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(54) **COMPRESSOR**

VERDICHTER

COMPRESSEUR

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EP 0 749 533 B1

Description

The present invention relates to a compressor.

When processing food, pharmaceutical and other sensitive material it is desirable to have a supply of compressed air or other working gas which is absolutely clean or "dry", that is to say completely free of oil or other bearing lubricating material.

In the past, there have been many attempts to produce oil-free compressors, but constructions such as dry screw compressors are expensive, inefficient, use large amounts of power and are cumbersome.

The overall market for air compressors comprises a number of performance bands with each performance band encompassing in combination a range of delivery pressures and a range of mass flows.

A delivery pressure of around 8.5 bara combined with a mass flow of 0.27kg per second is within one of the market bands for a dry air compressor. Delivery pressures can be met without difficulty at the present time, but the mass flow from a conventional turbo compressor of this sort is far greater than the mass flow which is required.

In addition, turbo compressors mounted on known oil lubricated, roller or ball journal bearings would be prohibitively inefficient at the high shaft rotational speeds (typically 50,000 to 100,000 rpm) required for the desired performance. Known turbo compressors operating in this band would therefore be extremely expensive, large and inefficient.

EP-A-0 150599 discloses a compressor comprising a rotatable shaft, bearing means provided for the shaft, drive means arranged to rotate the shaft, at least two impeller rotor stages mounted in spaced relationship on the shaft, at least a portion of the rotatable shaft between the spaced rotor stages being substantially hollow.

According to the present invention compressed air or working gas is bled from a relatively higher pressure rotor stage toward relatively lower pressure rotor stage along a path comprising the hollow portion of the rotatable shaft.

Desirably bleed passage means communicating with the hollow portion of the shaft is provided for this purpose. This is advantageous because the shaft is effectively cooled which results in further heat dissipation from the bearings.

The hollow portion of the shaft reduces the rotational moment of inertia of the combined impeller rotor stage and shaft assembly, thereby reducing the work needed to rotate the shaft and hence improving efficiency.

Preferably, thrust bearing means is provided, arranged to act directly on the impeller rotor stage such that when the shaft rotates, bearing contact is made between the thrust bearing and a bearing surface of the impeller rotor stage.

This enables heat generated at the thrust bearing means to be transferred via the impeller rotor stage directly to the working gas of the compressor, thereby cooling the bearing means and inhibiting overheating.

It is preferred that the compressor is provided with direct drive means arranged to rotate the shaft at high rotational speeds preferably in the range 50,000 to 100,000rpm. Preferably, the drive means therefore comprises an electric motor having a rotor mounted on the shaft.

Preferably the electric motor is positioned between the rotor stages. It is preferred that thrust bearing means is arranged to act directly on at least two impeller rotor stages to take up axial forces in opposed axial directions of the shaft.

Desirably, intercooler means is provided intermediate impeller rotor stages to enhance the efficiency of the compressor.

Desirably, the shaft comprises a composite shaft comprising the hollow rotor portion intermediately connecting spaced portions of the shaft, the spaced portions of the shaft desirably carrying respective impeller stages. Advantageously the hollow rotor portion of the shaft is of a magnetic or magnetisable material. By making the rotor portion of the shaft hollow, the mass (and hence the moment of inertia about the axis) of the composite shaft is kept to a minimum.

Advantageously securing means is provided for securing the hollow rotor portion and spaced portions of the shaft relative to one another. Preferably the securing means comprises a tie rod passing through the hollow rotor portion and the connected spaced portions of the shaft.

It is preferred that thrust bearing means are provided to act on impeller stages at both spaced portions of the shaft arranged such that axial thrust of the shaft in mutually opposed axial directions is taken up.

Desirably the thrust bearing means is arranged to act on the respective impeller stage rotor such that heat generated at the bearing is transferred to the impeller stage rotor. The thrust bearing means and the respective impeller are therefore preferably arranged to be in thermally communicative bearing contact when the compressor is operational.

This ensures that heat generated at the thrust bearing means is transferred to the respective impeller and subsequently to the working gas passing through the respective impeller stage of the compressor. The gas is then cooled as it passes into the following intercooler means.

It is preferred that the compressor further comprises journal bearing means arranged to support the shaft, preferably comprising at least one tilting pad journal bearing advantageously arranged to be self generating and air or gas lubricated and desirably having bearing pads provided with a ceramics bearing surface. The bearing pads may comprise homogenous pads of ceramics material.

It is preferred that the shaft is provided with hardened or ceramics surface portions against which the ceramics bearing surface of the respective tilting pads of

the journal bearing means is arranged to act.

Advantageously the bearing means comprises at least two journal bearings, each preferably being tilting pad journal bearings arranged to be air or gas lubricated and having bearing pads provided with respective ceramics bearing surfaces. Alternatively foil journal bearings may be used. Desirably, the journal bearings are provided to support spaced portions of the shaft advantageously adjacent opposed ends of the electric motor. It is preferred that at least one journal bearing is provided intermediately between a respective end of the motor and a respective impeller rotor stage.

The thrust bearing means preferably comprises a thrust bearing having tilting pads acting against the impeller rotor stage. Desirably the thrust bearing is of a self-generating air-or gas-lubricated type, having pads provided with ceramics bearing surfaces.

Advantageously, the impeller rotor stages are overhung at opposed ends of the shaft. It is preferred that each impeller rotor stage comprises a respective compressor impeller, with intercooler means being communicatively connected intermediate the impeller rotor stages.

Desirably, three impeller rotors are provided such that the compressor comprises three compression stages. It is preferred that respective intercooler means is provided intermediately between successive compressor stages. This improves the efficiency of the compressor. Advantageously, the flow of working gas into each respective impeller rotor is axial, and preferably in the direction of the electric motor.

It is accordingly preferred that at least two of the impeller stages are arranged in reverse formation relative to one another such that the respective flows into the respective impeller stages are in opposed directions, preferably towards one another. This has the advantage that the axial thrust load applied to the shaft by the respective impeller stages tend to cancel each other out, thereby reducing the axial thrust taken up by the thrust bearing means.

It is preferred that seal means, preferably comprising respective labyrinth seals, are provided for the shaft, arranged to inhibit access of the working gas from the impeller rotor stages to the motor and bearing means.

Advantageously, the electric motor comprises an electromagnetic or permanent magnet electric motor, preferably arranged to rotate the shaft at over 50,000 r.p.m. and more preferably at over 70,000 r.p.m. Desirably the electric motor is a direct current motor, preferably controlled by a variable frequency source.

The invention will now be further described by way of example only, and with reference to the accompanying drawings, in which:

Figure 1 is a schematic representation of a compressor;

Figure 2 is an enlarged detail of a part of the compressor of Figure 1;

Figure 3 is a schematic representation of a compressor according to the invention; and

Figure 4 is an enlarged detail of a part of the compressor of Figure 3.

Referring to the drawings, there is shown a compressor generally designated 1 which is generally as described in PCT specification WO94/05913. Although similar in general construction, and illustrative of preferred features, the compressor shown in Figures 1 and 2 does not include the specific structural features which result in the improved performance of a compressor according to the invention. The compressor 1 comprises an axial rotatable shaft 2 mounted in a housing 3, and having machined aluminium impeller rotors 4,5,6 mounted thereon.

Intake, first stage, rotor 4 is overhung at one end of the shaft, whereas second and third stage rotors 5 and 6 respectively are overhung at the cpposed end. Intermediately between impeller rotors 4 and 5 there is positioned a brushless D.C. motor having a rotor 7 comprising permanent magnets mounted on the shaft 2 and a stator 23 mounted in the housing. A solid state thyristor based inverter/controller (not shown) is used to generate a variable but high frequency current from a standard 415V/50Hz electrical supply. The high frequency current drives the motor (and therefore directly drives the shaft 2 without the need for intermediate gearing) at the required high operational speed which is typically of the order of 50,000 to 100,000 r.p.m. Because no gearing is required to couple shaft 2 to the drive, power losses are minimised.

The shaft 2 is supported in housing 3 on journal bearings 8,9 provided at either end of the electric motor, adjacent impeller rotors 4 and 5 respectively. A thrust bearing 10 is also mounted in the housing to act on thrust collar 11 provided on the shaft. Journal bearing 8,9 comprise tilting pad journal bearings which are self generating and air lubricated. The tilting pads 12 of each journal bearing 8,9 are supported on flexible pivots 24, and provided with ceramics bearing surfaces 13 which are arranged to act on immediately adjacent bearing surface portions of the shaft. The bearing surface portions of the shaft are coated with hardened deposit to increase wear resistance.

It is an important feature of the design that frictional losses in the bearings are minimised to maximise the efficiency of the compressor. Typically, where fluid lubricated journal bearings (such as oil lubricated bearings) or ball or roller journal bearings are used in high speed rotating machinery frictional losses in the bearings amount tc between 5% and 10% of the driving power. The provision of tilting pad self generating air (or gas) bearings cuts frictional losses to approximately 0.5% of driving power. However due to the fact that the shaft rotation speed is extremely high (e.g. 80,000 r.p.m. for a compression from 1 bara to 8.5 bara at a mass flow of 0.27 kg/s for air) the temperature generated at the bear-

ings is extremely high, which can cause problems with bearing/shaft material expansion due to the necessarily small bearing shaft clearances required for the operation of air or gas lubricate tilting pad self generating journal bearings (typically 0.003" diametral clearance for journal bearings). This problem is overcome by utilising ceramics materials for the bearing surfaces of tilting pads 12; the provision of a hardened deposit surface covering for the bearing portions of the shaft 2 also assists in overcoming this problem.

Thrust bearing 10 is also provided with tilting pad thrust members 10a,10b provided with ceramics bearing surfaces. Pads 10a are arranged to take up normal thrust loading transferred from shaft 2 by thrust collar 11 during normal running of the compressor. Pads 10b act on the opposite side of collar 11 and act to take up reverse thrust loading during motor and shaft "run up" to normal operational speed.

To increase efficiency, an intercooler 15 is provided intermediately between first stage impeller 4 and second stage impeller 5. A second intercooler 16 is provided intermediately between second stage impeller 5 and final (third) stage impeller 6. It is an important feature of the compressor that the flow of working gas into the first stage impeller 4 is in an opposed direction to the flow of working gas into the second and third stage impellers 5,6. This has the effect of "balancing" the axial thrust acting on the shaft and reducing the usual axial thrust applied to thrust bearing 10. Bearing losses in thrust bearing 10 are thereby minimised.

In operation, the electric motor is run up to an operating speed of around 80,000 r.p.m. Working gas is then drawn axially into the first impeller stage 4 and forced out through duct 17 into intercooler 15. The working gas leaves intercooler 15 entering duct 18 and subsequently passing axially into second impeller stage 5. The working fluid leaves impeller 5 radially passing via duct 19 into second intercooler 16. Intercoolers 15 and 16 are substantially identical, except that intercooler 16 is arranged with its longitudinal dimension at 90° to the longitudinal dimension of intercooler 15 (i.e. the longitudinal dimension of intercooler 16 is out of the page in Figure 1).

Working gas leaves intercooler 16 via duct 20 and is directed to enter the third (and final) impeller stage 6 axially. The working gas leaves the final impeller stage 6 radially via outlet duct 21 (the outlet flow through duct 21 is out of the page in Figure 1).

Due to the combination of the high speed directly driven rotatable shaft, together with the minimisation of bearing losses and the split stage intercooled arrangement of the impeller rotors, an extremely efficient compressor is provided. The compressor enables a compact turbomachine to be used in applications previously served mainly by screw feed type compressors since, unusually for a turbo compressor high delivery pressures (8.5 bara typically) are achievable with relatively low mass flows (0.27 kg/s typically for air).

Referring to Figure 3, the embodiment of compressor 101 shown is generally similar to in terms of construction and operation to the arrangement shown in Figures 1 and 2, and like reference numerals have been used to identify like components of the compressors.

In the embodiment shown in Figure 3, the thrust collar 11 of the compressor embodiment shown in Figure 2 is dispensed with and a pair of spaced thrust bearings 210a,210b provided adjacent the first and second stage impellers 4,5 respectively to take up axial forces in respectively opposed directions acting on the shaft 2. It has been found that with the compressor shown in Figure 1, excessive heat is generated at the thrust bearing 10 which results in reduced efficiency in terms of compressor performance and operational life expectancy. By replacing the thrust collar 11 and bearing assembly 10 with thrust bearings 210a,210b acting directly on the rear substantially flat surfaces of impeller rotor stages 4,5 respectively (as shown in the embodiment of Figure 3), overheating problems are substantially ameliorated. Heat generated at the thrust bearings 210a and 210b is transferred directly to the respective impeller stage rotor 4,5 and subsequently to the working gas flowing through th respective impeller stage rotor. The working gas is then coolar by passing through respective intercoolers (not shown in Figure 3) which are provided intermediate each impeller stage rotor as for the apparatus shown in Figure 1. Heat is therefore effectively transferred away from the thrust bearings. The thrust bearings 210a,210b comprise bearing pads 110a,110b mounted in a respective annular support ring 37a,37b carried by housings 35a,35b. The pads may be homogenous ceramics material, or alternatively may be provided with a ceramics bearing surface.

The embodiment of the invention shown in Figure 3 also differs from the arrangement shown in Figure 1 in that the shaft effectively comprises a hollow sectioned composite shaft comprising a first shaft portion 2a carrying impeller stage rotor 4), a second shaft portion 2b (carrying impeller stage rotor 5), and intermediate motor rotor section 7 extending between shaft portions 2a and 2b. Shaft portions 2a and 2b connect with opposed ends of the motor rotor section 7, the whole composite shaft being held together by means of axially extending tie rod 25. First and second shaft portions 2a, 2b are provided with respective hollow cylindrical cavities 31, 32 intersected by the axis of the shaft.

Tie rod 25 is provided along its length with sets of circumferentially spaced projections 40 which abut internal axial bores of shaft portions 2a, 2b and motor rotor 7. Circumferential spaces intermediate respective projections in each set 40 permit air communication along substantially the entire length of the interior of the composite shaft in the region adjacent tie rod 25. Compressed air or working gas is bled back from relatively higher pressure stage 5 (via bleed communication passage 42) and passes internally along the length of the composite shaft toward relatively lower pressure stage

4. Passage of the air or transport gas in the internal cavities 31, 32 and along the tie rod cause heat dissipation from the shaft portions 2a, 2b (and hence bearings 210a, 210b, 108, 109) and motor rotor 7.

Furthermore, because of the axial hollow cylindrical cavities provided within shaft portions 2a and 2b, the moment of inertia of the composite shaft about its rotational axes is reduced which increases the efficiency of the electromagnetic motor drive. Journal bearings 108, 109 are provided at opposed ends of the shaft and have bearings 112 which act on respective shaft portions 2a, 2b. Aided by the presence of cavities 31, 32 heat generated in the shaft from bearing contact with the journal bearings is transferred directly to impeller 5, 4 where it is transferred to the working gas of the compressor. At each end of the shaft the respective thrust bearing 210a, 210b and journal bearing 108, 109 are provided in a respective common unitary housing 35a, 35b. The compressor shown in Figures 3 and 4 operates in an almost identical manner to the compressor shown in Figure 1. Intercoolers (not shown) are provided intermediate each impeller rotor stage 4,5,6 and flow of working gas through the compressor is substantially as described in relation to the compressor shown in Figure 1.

Claims

1. A compressor (101) comprising a rotatable shaft (12), bearing means (8,9) provided for the shaft, drive means (7,23) arranged to rotate the shaft, at least two impeller rotor stages (4,5,6) mounted in spaced relationship on the shaft, at least a portion (31,32) of the rotatable shaft between the spaced rotor stages (4,5,6) being substantially hollow, characterised in that compressed air or working gas is bled from a relatively higher pressure rotor stage (5) toward relatively lower pressure rotor stage (4) along a path comprising the hollow portion (31,32) of the rotatable shaft.

2. A compressor (101) according to claim 1, characterised in that the shaft comprises a composite shaft comprising a rotor portion (7) intermediately connecting spaced portions (2a,2b) of the shaft (23), the spaced portions being provided with respective axial cavities (31,32) and carrying respective impeller rotor stages (4,5).

3. A compressor (101) according to claim 1 or claim 2, characterised in that the hollow portion (31,32) of the shaft is of a magnetic or magnetisable material.

4. A compressor (101) according to any preceding claim, characterised in that the drive means (7,23) comprises electro-magnetic drive means, preferably comprising an electric motor having a rotor (7) mounted on the shaft (2).

5. A compressor (101) according to any preceding claim, characterised in that thrust bearing means (210a,210b) is arranged to act directly on the impeller rotor stages (4,5) to take up forces acting in the axial direction of the shaft.

6. A compressor (101) according to any preceding claim, characterised in that intercooler means (15,16) is provided between impeller rotor stages (4,5).

7. A compressor (101) according to any preceding claim, characterised in that the compressor further comprises journal bearing means (108,109) arranged to support the shaft.

8. A compressor (101) according to claim 7, characterised in that the journal bearing (108,109) means comprises at least one tilting pad journal bearing.

9. A compressor (101) according to any preceding claim, characterised in that impeller (4,5,6) rotor stages are overhung at opposed ends of the shaft (2).

10. A compressor (101) according to any preceding claim, characterised in that impeller rotor stages are arranged in reverse formation (4,6 and 4,5) relative to one another such that the respective flows into the respective impeller stages are in opposed directions.

Patentansprüche

1. Kompressor (101) umfassend eine drehbare Welle (12), für die Welle vorgesehene Lagermittel (8, 9), Antriebsmittel (7, 23) zur Drehung der Welle, wenigstens zwei Flügelradstufen (4, 5, 6), die im Abstand voneinander auf der Welle angeordnet sind, wenigstens ein im wesentlichen hohles Teil (31, 32) der drehbaren Welle zwischen den im Abstand voneinander angeordneten Flügelradstufen (4, 5, 6), dadurch gekennzeichnet, daß komprimierte(s) Luft oder Arbeitsgas von einer Flugelradstufe (5) relativ höheren Drucks zu einer Flügelradstufe (4) relativ niedrigeren Drucks strömt, und zwar entlang eines Weges, der den hohlen Teil (31, 32) der drehbaren Welle umfaßt.

2. Kompressor (101) nach Anspruch 1, dadurch gekennzeichnet, daß die welle eine zusammengesetzte Welle umfaßt, die einen Rotorteil (7) enthält, der die im Abstand voneinander angeordneten Teile (2a, 2b) der Welle (23) untereinander verbindet, wobei die einen Abstand voneinander aufweisenden Teile mit entsprechenden axialen Hohlräumen (31, 32) versehen sind und entsprechende Flügelrad-

stufen (4, 5) tragen.

3. Kompressor (101) nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß der hohle Teil (31, 32) der Welle aus einem magnetischen oder magnetisierbaren Material besteht.

4. Kompressor (101) nach einem der vorhergehenden Ansprüche, dadurch gekennzeichnet, daß die Antriebsmittel (7, 23) elektromagnetische Antriebsmittel umfassen, vorzugsweise einen Elektromotor mit einem Rotor (7), der auf der Welle (2) montiert ist.

5. Kompresor (101) nach einem der vorhergehenden Ansprüche, dadurch gekennzeichnet, daß Druck-Lagermittel (210a, 210b) so angeordnet sind, daß sie direkt auf die Flügelradstufen (4, 5) einwirken, um Kräfte aufzunehmen, die in Axialrichtung der Welle wirken.

6. Kompressor (101) nach einem der vorhergehenden Ansprüche, dadurch gekennzeichnet, daß Zwischenkühlermittel (15, 16) zwischen den Flügelradstufen (4, 5) vorgesehen sind.

7. Kompressor (101) nach einem der vorhergehenden Ansprüche, dadurch gekennzeichnet, daß der Kompressor ferner Achslagermittel (108, 109) zur Lagerung der Welle umfaßt.

8. Kompressor (101) nach Ansruch 7, dadurch gekennzeichnet, daß die Achslagermittel (108, 109) wenigstens ein Klotz-Achslager umfassen.

9. Kompressor (101) nach einem der vorhergehenden Ansrüche, dadurch gekennzeichnet, daß Flügelradstufen (4, 5, 6) an entgegengesetzten Enden der Welle (2) fliegend angeordnet sind.

10. Kompressor (101) nach einem der vorhergehenden Ansprüche, dadurch gekennzeichnet, daß Flügelradstufen in entgegengesetzter Formation (4, 6 und 4, 5) relativ zueinander angeordnet sind, so daß die entsprechenden Strömungen in die entsprechenden Flügelradstufen in entgegengesetzten Richtungen fließen.

Revendications

1. Compresseur (101) comprenant un arbre rotatif (2), des moyens de palier (8,9) prévus pour l'arbre, un moyen de commande (7, 23) disposé pour faire tourner l'arbre, au moins deux étages de roues de rotor (4, 5, 6) montés de façon espacée sur l'arbre, au moins une portion (31, 32) de l'arbre rotatif entre les étages de rotor espacés (4, 5, 6) étant essentiellement creuse, caractérisé en ce que de l'air comprimé ou du gaz actif est purgé àepuis un étage de rotor (5) à pression relativement élevée vers un étage de rotor (4) à pression relativement basse le long d'un trajet comprenant la portion creuse (31, 32) de l'arbre rotatif.

2. Compresseur (101) selon la revendication 1, caractérisé en ce que l'arbre comprend un arbre composite comprenant une portion de rotor (7) connectant de façon intermédiaire des portions espacées (2a, 2b) de l'arbre 123), les portions espacées étant munies de cavités axiales respectives (31, 32) et portant des étages de roues de rotor respectifs (4, 5).

3. Compresseur (101) selon la revendication 1 ou la revendication 2, caractérisé en ce que la portion creuse (31, 32) de l'arbre est en un matériau magnétique ou susceptible d'être magnétisé.

4. Compresseur (101) selon l'une quelconque des revendications précédentes, caractérisé en ce que le moyen de commande (7, 23) comprend des moyens de commande électromagnétiques, comprenant de préférence un moteur électrique ayant un rotor (7) monté sur l'arbre (2).

5. Compresseur (101) selon l'une quelconque des revendications précédentes, caractérisé en ce que des moyens de palier de butée (210a, 210b) sont arrangés pour agir directement sur les étages de roues de rotor (4, 5) pour reprendre les forces agissant dans la direction axiale de l'arbre.

6. Compresseur (101) selon l'une quelconque des revendications précédentes, caractérisé en ce qu'un moyen de refroidisseur intermédiaire (15, 16) est prévu entre les étages de roues de rotor (4, 5).

7. Compresseur (101) selon l'une quelconque des revendications précédentes, caractérisé en ce que le compresseur comprend en outre un moyen de palier intermédiaire (108, 109) disposé pour supporter l'arbre.

8. Compresseur (101) selon la revendication 7, caractérisé en ce que le moyen de palier intermédiaire (108, 109) comprend au moins un palier intermédiaire à patins oscillants.

9. Compresseur (101) selon l'une quelconque des revendications précédentes, caractérisé en ce que les étages de roues de rotor (4, 5, 6) font saillie aux extrémités opposées de l'arbre (2).

10. Compresseur (101) selon l'une quelconque des revendications précédentes, caractérisé en ce que les étages de roue de rotor sont disposés selon une configuration inversée (4, 6 et 4, 5) l'un par rapport

à l'autre, de sorte que les flux respectifs à l'intérieur des étages de roues respectifs sont dans des directions opposées.

FIGURE 1

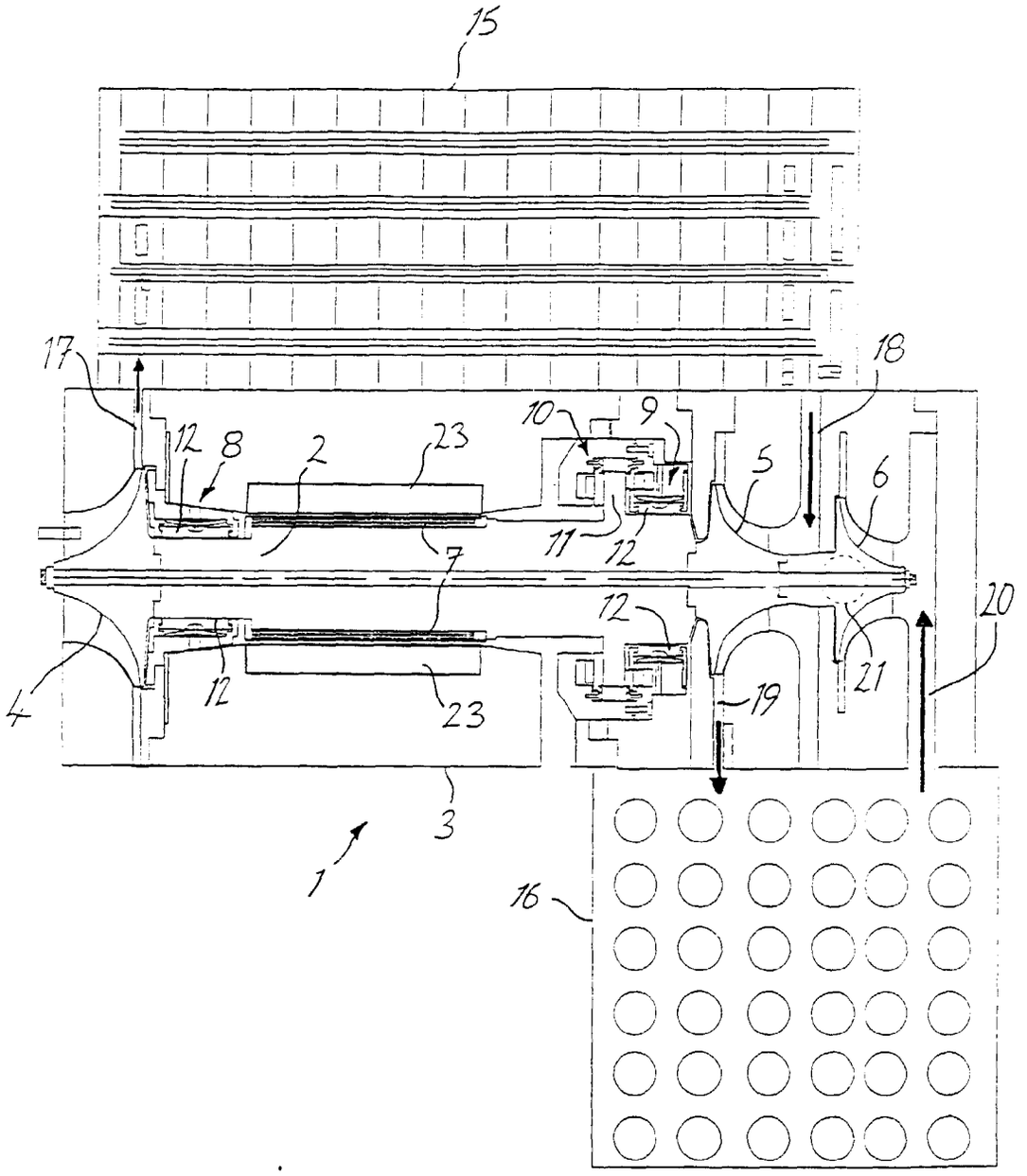


FIGURE 2

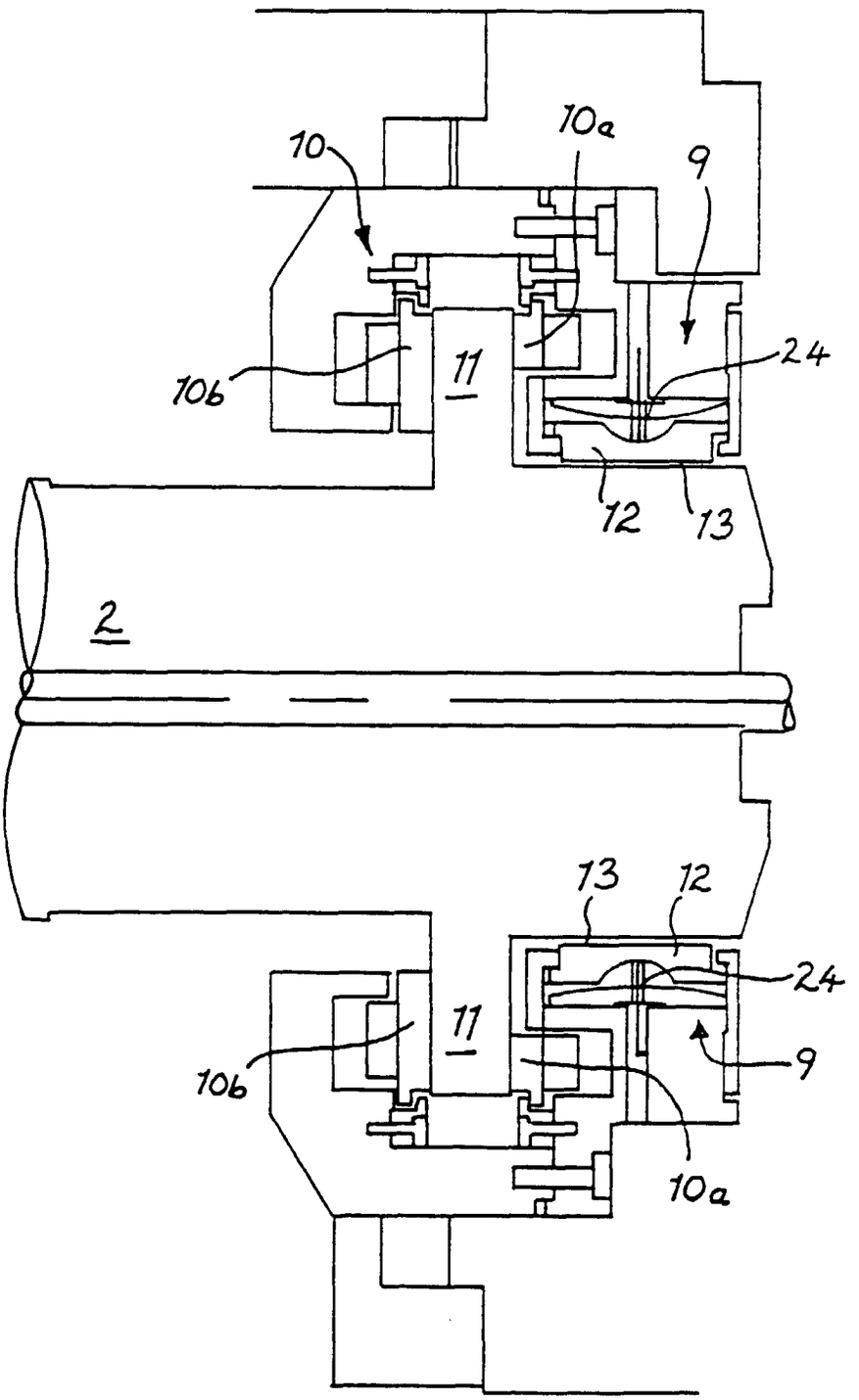


FIGURE 3

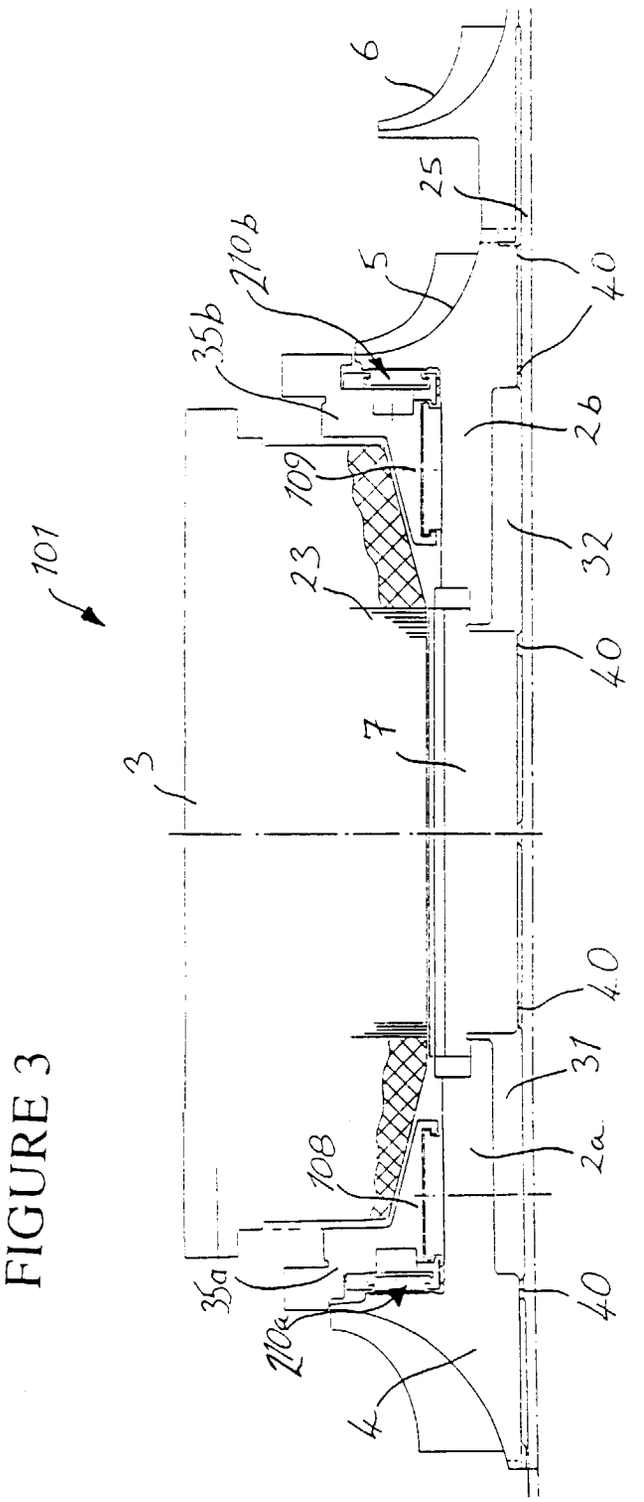


FIGURE 4

