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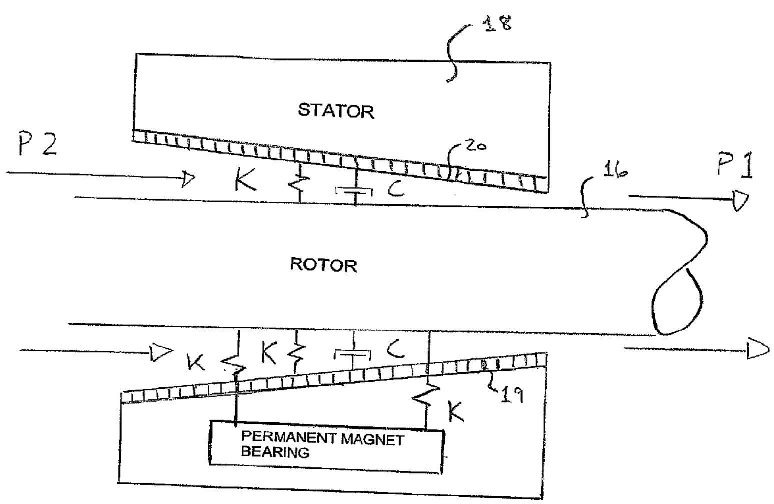
(54) Title: BEARING SYSTEM FOR ROTOR IN ROTATING MACHINES



P2 > P1

K: positive direct rigidity

C: positive direct damping

(57) Abstract: A bearing system for rotor in rotating machines, such as compressors, pumps, turbines, expanders, is distinguishing itself by points of bearing and sealing for the rotor (16) each in the form of a combined bearing and sealing (17) being formed by a stator (18) situated within a machine house (15) and surrounding the rotor (16). The stator (18) is formed with a bore (19), whereby an annular clearance is created between the stator and rotor, and the bore (19) is having sectional area gradually increasing in the direction of larger pressure (P2) within the rotating machine.

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BEARING SYSTEM FOR ROTOR IN ROTATING MACHINES

The present invention relates to a bearing system for the rotor in rotating machines, as disclosed in the preamble of claim 1.

In existing rotating machines, the rotor is supported both axially and radially. This may be done using bearings which are lubricated, magnetic, gas-dynamic etc. A common feature of all these bearings is that the length of the rotor shaft increases. Complex and costly support systems are also a necessity, except in the case of gas-dynamic bearings. Gas-dynamic bearings of the foil type do not require such support systems, but their bearing strength is for the present far less than that required for rotating machines with high output or pressure.

Other hydrodynamic and hydrostatic gas bearings have been proposed and to some extent tested but have not achieved significant popularity. Typical for hydrodynamic bearings, for example a foil bearing, is that their rotation generates a lift which gives bearing strength. In hydrostatic bearings, external pressurisation is carried out using specially formed recesses in the bearings. This bearing type requires separate seals.

This means that the existing technology provides solutions which are costly, complex, bulky and not particularly reliable.

The main object of the present invention is therefore to provide an improved bearing system for the rotor in rotating machines, with combined bearing and sealing of the rotor.

This object is achieved by the bearing system disclosed in claim 1. Preferred embodiments of the invention will be understood from the dependent claims and the following description of preferred embodiments.

Among the advantages of such a combined bearing system are that the design may be compact, allowing the rotor to be made shorter and more rigid for enhanced rotor-dynamic performance, or alternatively shorter and thinner for weight reduction, that the sealing aspect has much less importance than before, and that the costs are cut substantially as a result of the reliability or even the practicability of using the rotating machine in a subsea environment. The actual working medium in the machine may also be used during operation of the system so as to further reduce the complexity compared with known solutions.

By forming the bearing and seal combination according to the invention of an axial bearing in the form of a cylindrical disc on the rotor which rests against an associated portion of the stator, a gas film can be formed with rigidity and damping according to the same principle as in a radial bearing with desired dynamic rigidity and damping. Alternatively, the axial bearing can be formed according to the hydrostatic principle, which involves a flow restriction before and after the bearing surface, so as to obtain rigidity with accompanying damping. The axial bearing can also be formed using a combination of the two principles.

The present invention will now be discussed in more detail with the aid of preferred illustrative embodiments shown in the drawings, in which:

Figs. 1A and 1B show schematically the difference between a traditional rotating machine in the form of a compressor with motor-powered rotor, and the corresponding machine formed in accordance with the invention;

Fig. 2 is a schematic sectional view of the basic structure of a component incorporating combined bearing and seal which constitutes a part of the present bearing system;

Fig. 3 shows schematically an embodiment with several of the components from Fig. 2 and a start-up/run-down accumulator;

Fig. 4 is a further schematic view of a second embodiment with the possibility of altering the clearance geometry of the bearing system or internal pressure differences in the rotating machine;

Fig. 5 shows schematically the same as Fig. 2, but in this case the component is made having two sets of radial hole patterns for increased damping in the bearing system by means of gas exchange; and

Fig. 6 shows schematically an embodiment of the invention with compressor and motor located in the same housing.

With reference to Fig. 1, the present invention will be explained in more detail in connection with rotating machines, as for example a compressor for use in subsea environments and which has a motor-powered rotor. However, this should not be understood as meaning that the invention relates solely to the illustrated compressor, as it is of course suitable other rotating machine types and environments of use. Furthermore, it should be noted that the figures only show details which are important for the understanding of the invention.

As can be seen from Fig. 1A, the traditional compressor has a motor-powered rotor, equipped with respectively a bearing system 13 and a sealing system 14 which are placed on a rotor shaft 12 at each end outside a compressor housing 11. As illustrated in Fig. 1B, these external bearing and sealing systems in the case of the present invention are replaced by at least two components 17, of which just one is shown, and which are located inside the compressor housing 15. The new component 17 also functions as a combination of a bearing and seal for the rotor 16, see Fig. 2. This means that the compressor can be equipped with a suitable motor, see Fig. 6, which is arranged inside the compressor housing. Thus, the need for external shaft seals no longer exists, with the result that the rotating machine as such has a far simpler design.

The principle of the combined bearing and seal 17 from Fig. 1B is illustrated in greater detail in Fig. 2. According to the general embodiment of the invention, an approximately cylindrical stator 18, i.e., the stationary part surrounding the rotor 16, is formed with a bore 19, whereby an annular clearance is formed between stator and rotor. The stator 18 thus constitutes a "bearing point" for the rotor 16. Furthermore, the pressure difference is used, i.e., the pressure drop across the clearance, to obtain the function as combined rotor bearing and seal. In Fig. 2 this is symbolised by means of P2 and P1, that is to say the outlet pressure and the inlet pressure of the compressor, respectively. The precondition for a successful result is, however, that the annular clearance has a geometric configuration that gives sufficient rigidity and damping in the relevant frequency ranges, as symbolised by K and C in Fig. 2. The rigidity can be provided by allowing the annular clearance to converge towards the lower pressure, so that the inlet clearance is greater than the outlet clearance. Positive direct rigidity is thus obtained in the bearing. Positive direct damping may be provided by means of the characteristics of the surface 20 of the stator facing the rotor, e.g., by means of a honeycomb structure or other type of roughness in the surface. The stator 18 is, for example, mounted in a T-shaped groove (not shown) with loose fit in the compressor housing.

Positive direct rigidity is a known concept in the field of rotor dynamics and entails the countering of radial motion of the rotor by the bearing, so that the same holds the rotor centred in the clearance for correct positioning in relation to the stator. Direct positive damping means that the rotor is "braked" or damped by the bearing.

With reference to Fig. 3, an embodiment is illustrated which is an example of the use of the present invention where the bearings/seals 3, 4, 5 are of the type referred to above in connection with Fig. 2. In this case, internal seals 1, 2 are also arranged between each impeller. The last-mentioned can be configured as a converging clearance or other suitable geometry so as to improve the rotordynamic properties if found expedient. The internal seals may alternatively only have a function as seals having conventional design.

It is clear, furthermore, that the bearings/seals require a pressure difference in order to cause the required rigidity and damping. This is a fact that must be taken into account during start-up and shut-down. As shown in Fig. 3, the difference in pressure can be obtained by means of an accumulator 6 which is put in communication with the respective bearings/seals 3, 4, 5 in any suitable way. The accumulator is filled with gas pressurised to the outlet pressure P2 for injection into the bearings/seals 3, 4, 5. An alternative is the mounting of special start-up and run-down bearings 7 which also additionally are shown in Fig. 3, for example, bearings of the same type as used in magnetic bearings, and which withstand contact for a brief period during start-up or run-down.

The concept according to the invention can be implemented in a hermetic compressor, where the motor is placed in a pressurised gas atmosphere, or in a conventional, externally powered rotating machine. In the last-mentioned case, a designated shaft or axle seal 8 must then be used to seal against the atmosphere. Such a seal may however be made substantially smaller than normal, as the shaft diameter will only be dimensioned for transfer of necessary torque. The advantage of the reduced shaft diameter is that the area of use of high-pressure compressors is extended as a consequence of the fact that smaller seals withstand greater pressure. Expressed briefly, technical limitations on allowed seal pressure depend on seal diameter. Since when using the present invention there is no need for a separate support bearing outside the seal, the diameter can be made substantially smaller than in a conventional rotating machine.

Moreover, as shown in Fig. 4, it is possible in addition to envisage an adjustment of the geometry of the combined bearings/seals by using gas pressure differences in the rotating machine and lead the necessary bores out to a control means, for example, a control valve 17 which alters the pressure in a cavity between the stator and the compressor housing, or in the stator, in such manner that the pressure forces change the geometry of the bearing and seal clearance. With such a variant, it is possible to obtain a necessary degree of freedom in order to cater for different operating conditions by changing rigidity and damping in the combined bearings and seals.

As shown in Fig. 5, the damping can be further increased with the aid of the alternative configuration of the surface of the stator 18 facing the rotor 16. In this case, the bore 19 in the stator 18 has a surface structure 21 consisting of an external radial hole pattern and a corresponding internal hole pattern, but so positioned relative to each other that a gas exchange can take place in the direction of the greater pressure P2.

The motor 22 and the compressor 23 can, as shown schematically in Fig. 6, also be located in the same housing 24. This means that external sealing is advantageously not required.

Also, it should be noted that a passive permanent magnetic bearing, see Fig. 2, can be used to support the rotor. This will reduce the load on the fluid film and increase the overall rigidity of the bearing, which is essential at start-up or shut-down. In such a case, the passive permanent magnetic bearing is arranged integrated in the bearing and seal combination or separately next to the combination.

Patent claims

1.

A bearing system for the rotor in rotating machines, such as compressors, pumps, turbines and expanders, wherein the rotor is provided with at least two bearings and associated seals, characterised in that each bearing and sealing point for the rotor (16) is in the form of a bearing and seal combination (17) that is formed of a stator (18) located within a rotating machine housing (15) and surrounding the rotor (16), that the stator (18) is formed with a bore (19), whereby an annular clearance is formed between the stator and the rotor, and that the bore (19) has a gradually increasing sectional area in the direction of higher pressure (P2) within the rotating machine.

2.

A bearing system according to claim 1, characterised in that the bearing and seal combination (17) is an axial bearing formed as a cylindrical disc on the rotor (16) which bears against an associated portion of the stator (18), thereby enabling a gas film to be formed with rigidity and damping according to the same principle as in a radial bearing with desired dynamic rigidity and damping.

3.

A bearing system according to claim 1, characterised in that the axial bearing is formed according to the hydrostatic principle which entails a flow restriction before and after its bearing surface so as to thereby obtain rigidity with accompanying damping.

4.

A bearing system according to claim 2 or 3, characterised in that the axial bearing is formed as a combination of the radial bearing with gas film and the hydrostatic principle with flow restriction before and after the bearing surface.

5.

A bearing system according to any one of the preceding claims, characterised in that the bore (19) is formed having an uneven surface structure.

6.

A bearing system according to any one of the preceding claims, c h a r a c - t e r i s e d i n t h a t t the bore (19) is formed having a honeycomb structure or a pattern of holes (20).

7.

A bearing system according to any one of the preceding claims, c h a r a c - t e r i s e d i n t h a t the surface structure (21) of the bore (19) has an outer zone consisting of an external radial pattern of holes and an internal pattern of channels, but so positioned relative to each other as to allow gas exchange to take place in the direction of the higher pressure (P2).

8.

A bearing system according to any one of the preceding claims, c h a r a c - t e r i s e d i n t h a t at the start-up or the run-down of the rotating machine the higher pressure (P2) is provided by means of an accumulator (6) which contains gas at such a pressure, and which is in communication with each individual bearing and seal combination (17).

9.

A bearing system according to any one of the preceding claims, c h a r a c - t e r i s e d i n t h a t the system comprises at least two support bearings (7) arranged in connection with the respective bearing and seal combination (17), and which are of a type suitable for withstanding contact for a brief period during start-up or run-down.

10.

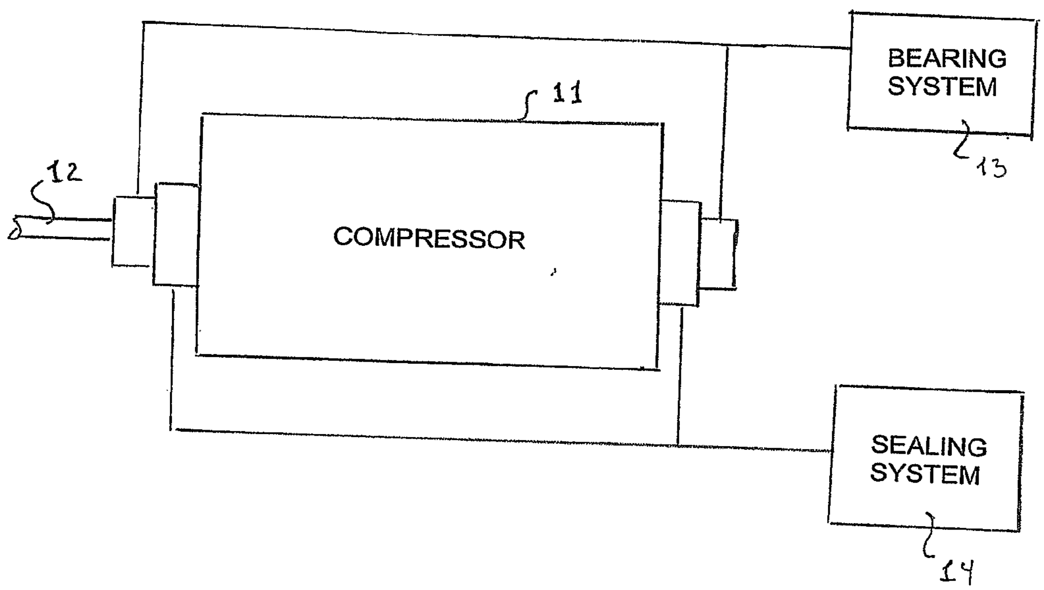
A bearing system according to any one of the preceding claims, c h a r a c - t e r i s e d i n t h a t the system comprises a control means (7) such as a control valve, so as to adjust the geometry of the respective bearing and seal combination (17) by means of applied pressure forces.

11.

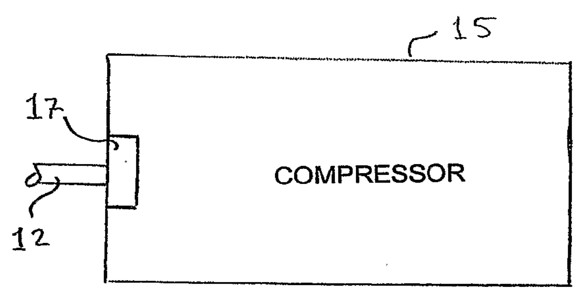
A bearing system according to any one of the preceding claims, c h a r a c - t e r i s e d i n t h a t the motor (22) and the compressor (23) are located in the same housing (24).

12.

A bearing system according to any one of the preceding claims, c h a r a c - t e r i s e d i n t h a t a passive permanent magnetic bearing for support of the rotor (16) at start-up or shut-down is arranged integrated in the bearing and seal combination (17) or separately adjacent thereto.

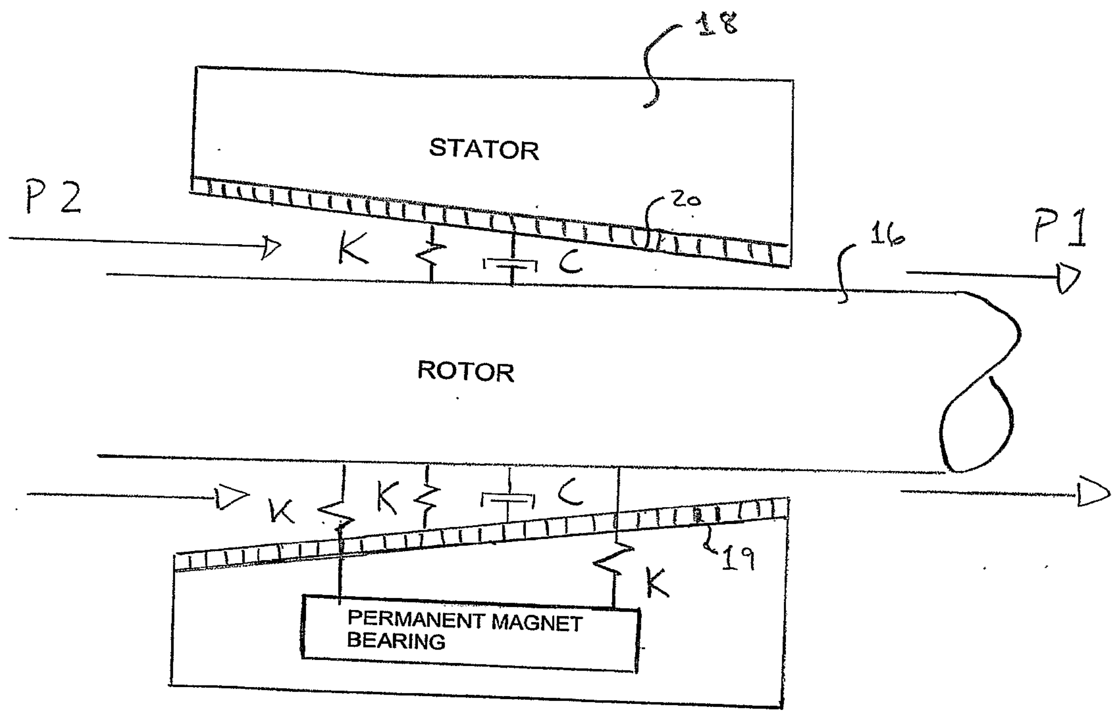


1A: Traditional



1B: Proposed technology

Fig. 1

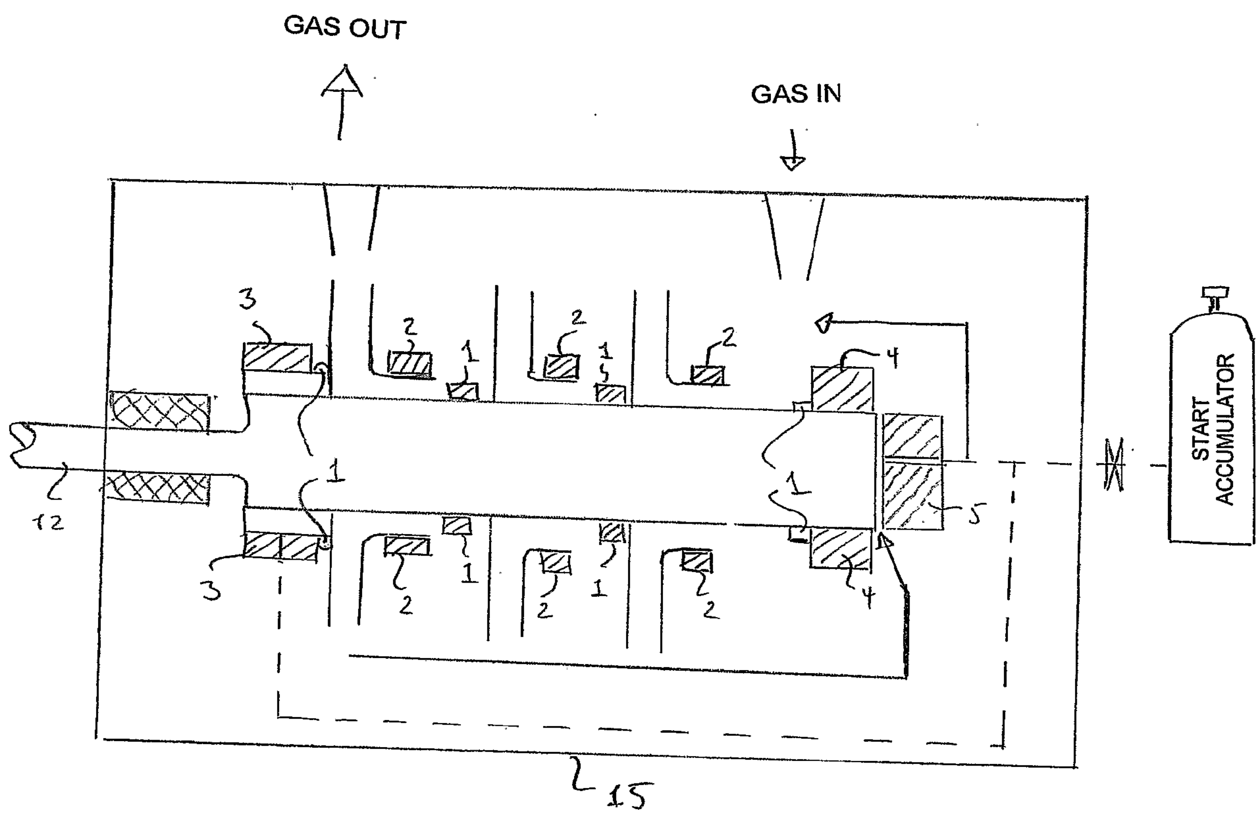


P2 > P1

K: positive direct rigidity

C: positive direct damping

Fig. 2



: BEARING/SEALING

: AXLE SEALING AGAINST ATMOSPHERE

Fig. 3

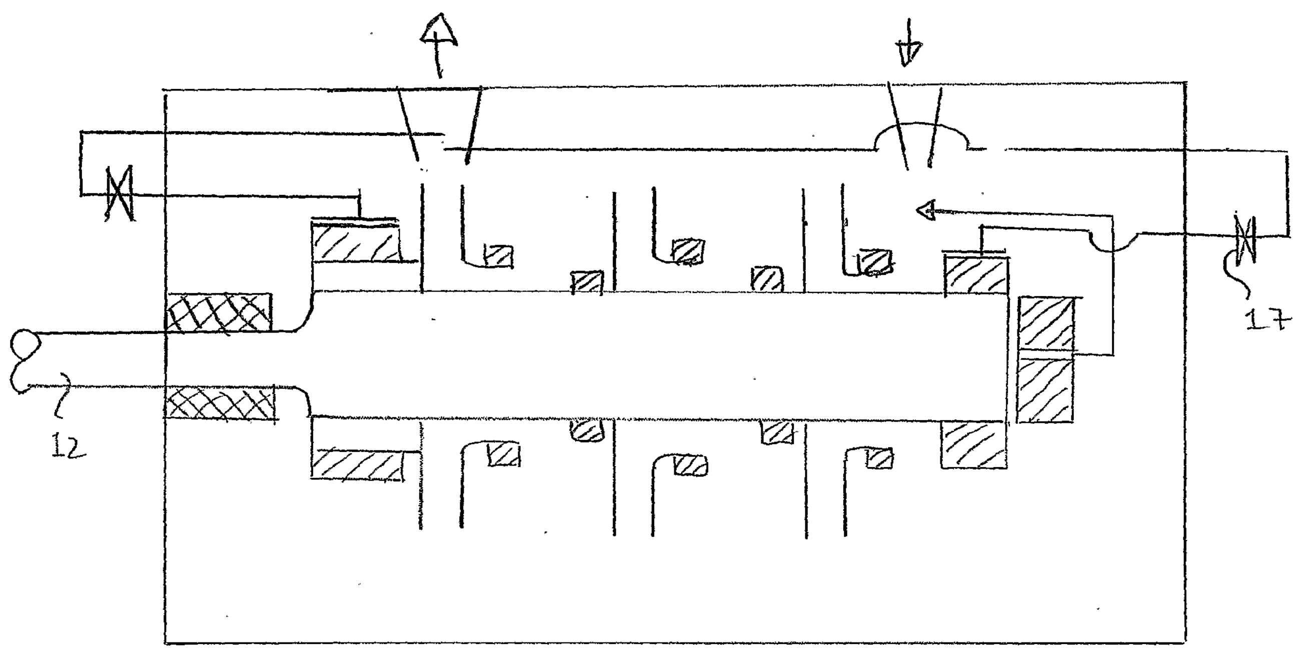


Fig. 4

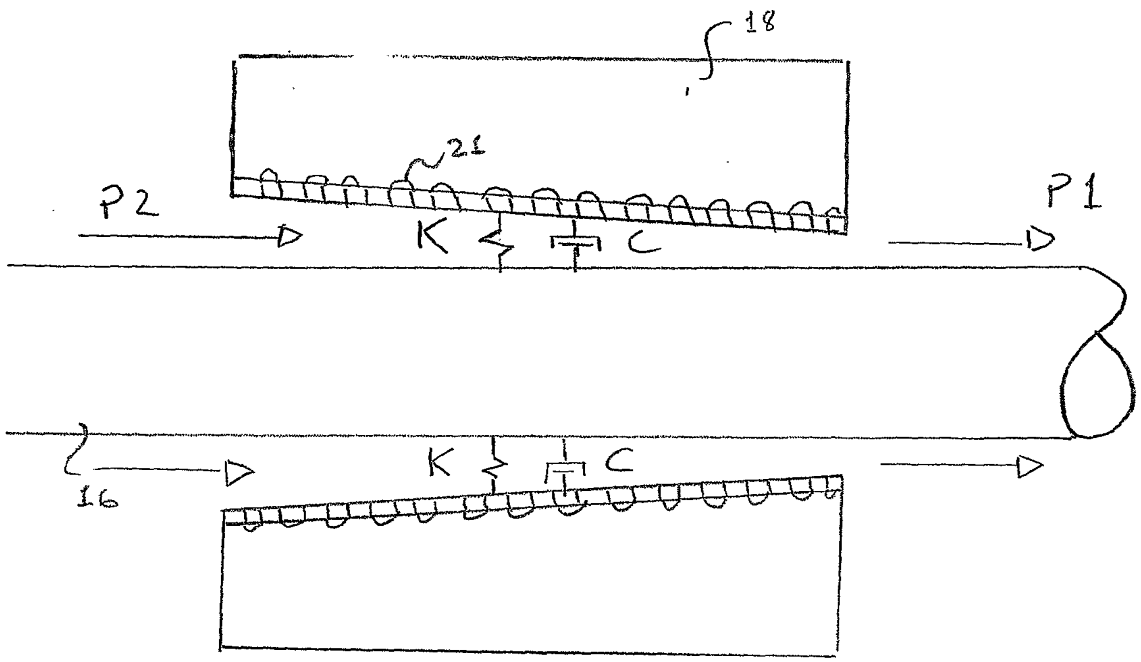


Fig. 5

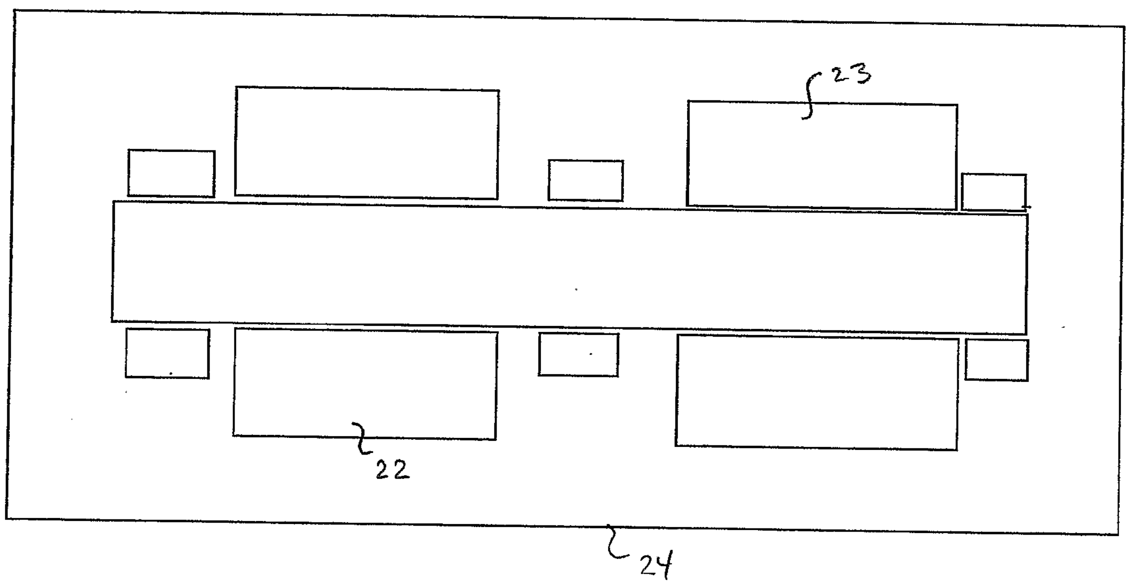


Fig. 6

INTERNATIONAL SEARCH REPORT

International application No.
PCT/NO2007/000279

A. CLASSIFICATION OF SUBJECT MATTER

IPC: see extra sheet
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)

IPC: F16C

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

SE,DK,FI,NO classes as above

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

EPO-INTERNAL, WPI DATA, PAJ

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5310265 A (STANGELAND ET AL), 10 May 1994 (10.05.1994), column 2, line 27 - line 51; column 3, line 15 - line 42, figures 4-6, abstract	1-12
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☐ Further documents are listed in the continuation of Box C. ☒ See patent family annex.

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Date of the actual completion of the international search	Date of mailing of the international search report
19 November 2007	20 -11- 2007
Name and mailing address of the ISA/ Swedish Patent Office Box 5055, S-102 42 STOCKHOLM Facsimile No. +46 8 666 02 86	Authorized officer Jan-Axel Ylivainio / JA A Telephone No. +46 8 782 25 00

International patent classification (IPC)

F16C 32/06 (2006.01)

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Paper copies can be ordered at a cost of 50 SEK per copy from PRV InterPat (telephone number 08-782 28 85).

Cited literature, if any, will be enclosed in paper form.

US	5310265	A	10/05/1994	NONE