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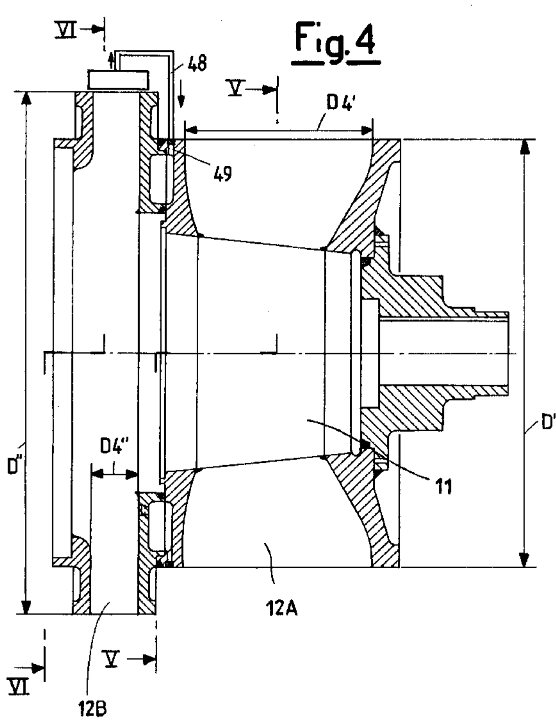
(54) **Liquid ring compressor**

(57) A liquid ring compressor (10), wherein the gas distribution is obtained by means of a central stator (11), which has a frusto conical shape, comprising at least a rotor (12A, 12B), provided with a plurality of radial blades (14) mounted on a shaft (13), which is concentric with the longitudinal axis (25) of the compressor (10); the configuration of the blade (14) is such that performances, substantially higher than the performances of the traditional compressors, can be obtained at relatively high working pressures; finally, in order to obtain higher working pressures, two rotors (12A, 12B) of the above described type are positioned one after the other and a hydraulic barrage is created between first and second stage, in order to oppose the natural gas flow from the second stage rotor (12B), at high pressure, to the first stage rotor (12A), at a lower pressure.

Fig.4



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Description

In general, the present invention refers to liquid ring pumps and, in particular, to a liquid ring compressor, of the single-stage or two-stage type, comprising at least a rotor provided with radial blades.

The manufacturing of liquid ring machines has been well known, said machines working both as vacuum pumps and as compressors; a traditional liquid ring machine comprises an inner fixed housing, wherein a rotor is mounted, provided with outwardly extending radial blades; the rotor is mounted on a rotation axis within a body concentric with the longitudinal axis of the machine inner housing, said body having two eccentricities which give said body an almost oval shape.

A certain amount of liquid is introduced inside said housing and is circulated so that, when the rotor rotates, the blades cause said liquid to move thus creating a ring which circulates in the oval body.

Since the housing is in an eccentric position as to the rotor, the inner surface of said liquid ring moves in the radial direction, alternatively, towards the outer, as to the rotor axis, and towards the inner part of the rotor, along a circular direction.

Consequently, the liquid, which partially fills the room between the radial blades adjacent to the rotor, alternatively expands (when it is pushed outwardly as to the rotor axis) and is compressed (when it moves towards the rotor inner part) along a closed circuit provided around the machine inner housing.

The gas to be pumped is injected in the interstices between the blades where the liquid expands and said expansion causes the gas to move towards the so called suction area of the machine.

The subsequent compression in the interstices between the rotor blades compresses the gas towards a so called compression area of the machine.

The interstices between the blades, wherein the compression occurs, are connected with an exhaust duct, through which the compressed gas comes out from the machine.

Further, there are known liquid ring machines, wherein the gas which leaves from a first stage is directed towards the suction area of a following stage in order to be compressed again; the gas coming out from the second stage can reach pressure values almost equal to the atmospheric pressure value.

A significant disadvantage, typical of the embodiment of all of the known machines, is the loss of energy and, therefore, the loss of performance of the machine itself due to the friction, which is caused in the fluid between the rotating liquid ring and the fixed housing.

Said disadvantage is partially overcome by using one or two rotors, positioned one after the other, which are mounted between at least one portion of the liquid ring and the inner surface of the housing, otherwise said inner surface would be directly in contact with said portion of the liquid ring.

In fact, usually the rotor rotates at a reduced angular speed compared to the rotation speed of the liquid ring and, therefore, the friction created by the fluid on the rotor blades is less than the friction existing between the liquid ring and the walls of the fixed housing.

However, the performance of said machines, particularly in the case wherein said machines are used as compressors, is always substantially low, especially in case of high working pressures and, at any rate, higher than the atmospheric pressure.

In this case, it would be desirable to increase the pressure of the carrying liquid or at least of a portion of said liquid; however, in order to do so, it is necessary to have an auxiliary pump (for instance a centrifugal pump) with a consequent increase in the production and operation costs of the machine and a decrease in reliability.

A purpose of the present invention is, therefore, to disclose a liquid ring compressor which overcomes the above mentioned disadvantages, and to provide a liquid ring compressor which allows to obtain a high performance, compared with the known art, at working pressures substantially higher than the atmospheric pressure.

Another purpose of the present invention is to disclose a high performance liquid ring compressor which allows to obtain working pressure values as high as ten times the value of the atmospheric pressure.

A further purpose of the invention is to provide a liquid ring compressor, wherein the technical characteristics are higher than those of the traditional liquid ring compressors.

Another purpose of the invention is to provide a liquid ring compressor which is less expensive than the traditional compressors, which does not require the use of complex or particularly expensive technologies and which allows to substantially reduce the energy losses and, therefore, the efficiency losses of the compressor, as to the known art.

These and other purposes are achieved by a liquid ring compressor as claimed in claim 1, which is taken as reference.

Advantageously, the configuration of the blades of a rotor, which is used in a compressor according to the invention, has a tapering at an angle of about 30 degrees.

Said tapering, used in rotors of liquid ring machines having a gas distribution through a central stator with a frusto conical shape, allows to obtain higher performances than in the known art at working pressures of about 300-400 KPa.

In order to obtain higher pressures of about 700-1100 KPa, two rotors are positioned one after the other, thus obtaining a two-stage compressor; the rotor of the first stage has a smaller diameter and a bigger thickness than the rotor of the second stage, so as to obtain similar compression ratios.

In said embodiments, there is also a hydraulic barrage between the first and the second stage to oppose the natural gas flow from the second stage rotor, at high pressure, to the first stage rotor, at a lower pressure; this is accomplished by transferring a portion of the carrying liquid from a second stage housing to an area which is intermediate between the first stage rotor and the second stage rotor.

It is therefore possible to obtain high working pressures and outstanding performances of the compressor.

Further purposes and advantages of the present invention will be more apparent from the following description and from the accompanying drawings, given by way of non-limiting example, wherein:

Figure 1 is a longitudinal section of a two-stage liquid ring compressor, according to a known embodiments;
Figure 2 shows an enlarged detail of a longitudinal section of a single-stage liquid ring compressor, according to the present invention;
Figure 3 is a sectional view taken along line III-III of Figure 2;
Figure 4 shows an enlarged detail of a longitudinal section of a two-stage liquid ring compressor, according to the present invention;
Figure 5 is a sectional view taken along line V-V of Figure 4;
Figure 6 is a sectional view taken along line VI-VI of Figure 4.

With reference to Figure 1, the numeral 10 indicates, generally, a liquid ring compressor which comprises a drive section, indicated by numeral 15, of a shaft 13, around which the rotors 12A, 12B, belonging to a first and to a second stage of the compressor 10, respectively, rotate.

20 indicates a process phase, during which the gas distribution is obtained by means of a central stator, with a frusto conical shape, indicated by numeral 11; the gas to be compressed flows from the duct 42, through a first opening 43 to the central stator 11 (along the direction of arrows G in Figure 1).

In a first exemplifying and non-limiting embodiment of the present invention, the compressor 10 comprises one single stage and one single housing wherein the rotor rotates.

Alternatively, a second exemplifying and non-limiting embodiment of the present invention comprises a two-stage compressor, wherein a rotor 12A, positioned in a first stage, is mounted on the shaft 13 and rotates inside the walls of a first housing 30A; said rotor co-operates, together with a service liquid ring (not shown) of the first stage, in causing the gas to flow from one side to the other of the compressor 10 and simultaneously in compressing said gas.

The service liquid is introduced from the outside through the opening 24 (along the direction of arrow L) and, then, it flows inside the housing 30A of the first stage rotor 12A, through a second opening 26.

The compressed gas mixed with the liquid comes out from the first stage rotor 12A through a third opening 28 and flows towards the inlet of the second stage, through a fourth opening 29; the second stage rotor 12B, which rotates inside the walls of a second housing 30B, helps, together with the service liquid in the second stage, to compress again the gas which, finally, comes out from a fifth opening 32 in the exhaust duct 33 (along the direction of arrow G1).

The service liquid is discharged outside through a plurality of ducts, indicated by numeral 33, along the directions of arrows L1.

Each one of the rotors 12A, 12B of the first and second stage is concentric with the rotation axis 25 of the shaft 13 and with the compressor body; the body, however, has not a circular shape but, since it is provided with two eccentricities, it has an almost oval shape.

Further, as is clearly indicated in Figures 3, 5 and 6, each rotor 12A, 12B is provided with a plurality of radial blades, indicated by numeral 14, which define interstices and which cause the rotation of the service liquid, which fills at least partially the housing 30A, thus forming a ring with a substantially constant thickness, with an eccentric shape.

In the interstices between consecutive blades 14, the service liquid is pushed along the radial direction towards the periphery during a first turn portion, while during a second turn portion the liquid is pushed towards the centre; in this way, the service liquid works as a hydraulic piston, which continuously sucks the gas during the first turn portion and which compresses the gas during the following turn portion.

The blades 14 are radial and curved towards the periphery of the rotor 12A, 12B and are one another equally spaced in order to allow the passage of the same amount of gas.

The configuration and the arrangement of the rotors 12A, 12B allow to reduce the viscous friction forces, which are in effect at the interface between the fluid and the inner walls of the gas distribution cone of the compressor 10, and therefore to increase the machine efficiency.

In this specific case, it has been demonstrated that in practice the tapering of each blade 14 of the rotors 12A, 12B at an angle of about 30 degrees, indicated by α in Figure 2, allows the achievement of machine efficiencies higher than the efficiencies of the traditional solutions, at working pressures around 300-400 KPa; said tapering can be introduced or, better, is suitable for liquid ring compressors which have the gas distribution by means of a central stator (as is the case of the compressor of the present invention).

Further, said efficiencies are achieved by using the following design parameters for the rotors 12A, 12B: the outer diameter, indicated by the letter D in Figure 3, is comprised within the 0.16-0.67 m (meter) range of values, while the inner diameter, indicated by D1, is comprised within the 0.08-0.335 m range of values; further, the length indicated by D2 in Figure 2 is comprised within the 0.05-0.2 m range of values, the length indicated by D3 in Figures 2 and 3 is comprised within the 0.005-0.015 m range of values, while the inner width of the rotor 12A, 12B, indicated by D4 in Figure 2, is comprised within the 0.08-0.3 m range of values.

Finally, the radiuses of curvature indicated by R1 and R2 in Figure 2 are comprised within the 0.1-0.3 m and 0.05-0.15 m ranges of values, respectively.

In order to obtain working pressures higher than 300-400 Kpa and therefore to achieve values of about 700-1100 KPa, two rotors 12A, 12B, having the same above described technical characteristics, are positioned one after the other in a typical embodiment of the two-stage compressor 10, said embodiment is shown, in particular, in Figures 4, 5 and 6.

Said rotors 12A, 12B have two different diameters and two different width, so that the first stage rotor 12A is characterised by having a smaller diameter D' and a larger width D4' than the corresponding dimensions, indicated by D" and D4", of the second stage rotor 12B. Said differences are necessary to obtain similar gas compression ratios in both rotors 12A, 12B.

In the case of the two-stage compressor 10, it is further provided a hydraulic barrage between the first and the second stage, said barrage is formed by a tube, indicated by numeral 48 in Figure 4, suitable to circulate back the service liquid, which is carried from the second stage rotor 12B and is sent back inside the first stage housing 30A.

The hydraulic barrage 48 can be positioned either inside or outside the machine and said barrage is utilised in such a way that the system can oppose the natural gas flow from the second stage rotor 12B at high pressure to the first stage rotor 12A at a lower pressure.

In particular, the operation is executed by sucking a portion of the service liquid from the second stage housing 30B (i.e. from the area at the highest working pressure) and by discharging said liquid in the intermediate area, indicated by 49 in Figure 4, between the first stage rotor 12A and the second stage rotor 12B; this allows to achieve high working pressures (700-1100 KPa) with outstanding machine efficiencies.

The characteristics and the advantages of the high efficiency liquid ring compressor of the present invention have been clearly disclosed by the preceding description.

In particular, said characteristics and advantages are as follows:

- higher performances (as regards working pressure and efficiency) in comparison with the performances of similar known compressors;
- reduced costs in comparison with the known art. It is clear that several changes can be done to the high efficiency liquid ring compressor of the present invention without leaving from the innovative principles of the invention, it is also clear that, in the embodiments of the invention, the materials, the shapes and the dimensions of the illustrated details can be different and these same details can be substituted by technically equivalent elements.

Claims

1. High efficiency liquid ring compressor (10), wherein the gas distribution is obtained by means of a central stator (11), which has a frusto conical shape, comprising at least a rotor (12A, 12B), provided with a plurality of radial blades (14) and said rotor is concentric with a longitudinal rotation shaft (13) and with the body of the compressor (10), said blades (14) are used to cause the rotation of predefined amount of service liquid, which fills at least partially one or more inner housings (30A, 30B) of the compressor (10), thus forming a ring with a substantially constant thickness and with an eccentric shape, said blades (14) being curved towards the outward portion of said rotor (12A, 12B) and being one another equally spaced in order to allow the passage of the same amount of gas, characterised in that each one of said blades (14) has at least a tapered portion at an angle substantially equal to 30 degrees.

2. Compressor (10) as claimed in claim 1, characterised in that the outer diameter value (D) of the rotor (12A, 12B) is comprised within the 0.16-0.67 m range of values (including the end values), while the inner diameter (D1) is comprised within the 0.08-0.335 m range of values (including the end values).

3. Compressor (10) as claimed in claim 2, characterised in that the inner width (D4) of said rotor (12A, 12B) has a value which is comprised within the 0.08-0.3 m range (including the end values).

4. Compressor (10) as claimed in claim 3, characterised in that each radius of curvature (R1, R2) of the blades (14) has a value which is comprised within the 0.05-0.3 m range (including the end values).

5. Compressor (10) as claimed in claim 1, characterised in that the working pressure is higher than or equal to 300 KPa.

6. Compressor (10) as claimed in claim 5, characterised in that the working pressure is comprised within the 300-700 KPa range of values (including the end values).

7. Compressor (10) as claimed in claim 4, characterised by providing two side by side rotors (12A, 12B) which define two stages, said rotors (12A, 12B) hav-

ing, respectively, two different diameters (D4', D4") and two different widths (D', D"), wherein a first rotor (12A) belonging to the first stage has a smaller diameter (D') and a larger width (D4') in comparison with the corresponding values (D", D4") of a second rotor (12B) belonging to the second stage, so as to obtain similar compression ratios in the rotors (12A, 12B).

8. Compressor (10) as claimed in claim 7, characterised by providing a hydraulic barrage between the first and the second stage, said barrage being obtained by using at least a duct (48) which sucks, at least partially, said predefined amount of service liquid from an area of the compressor (10) at maximum working pressure provided in the second stage, said duct then discharging said liquid in an intermediate area (49) comprised between said first rotor (12A) and said second rotor (12B).

9. Compressor (10) as claimed in claim 8, characterised in that the working pressure is higher or equal to 700 KPa.

10. Compressor (10) as claimed in claim 9, characterised in that the working pressure is comprised within the 700-1200 KPa range of values (including the end values).

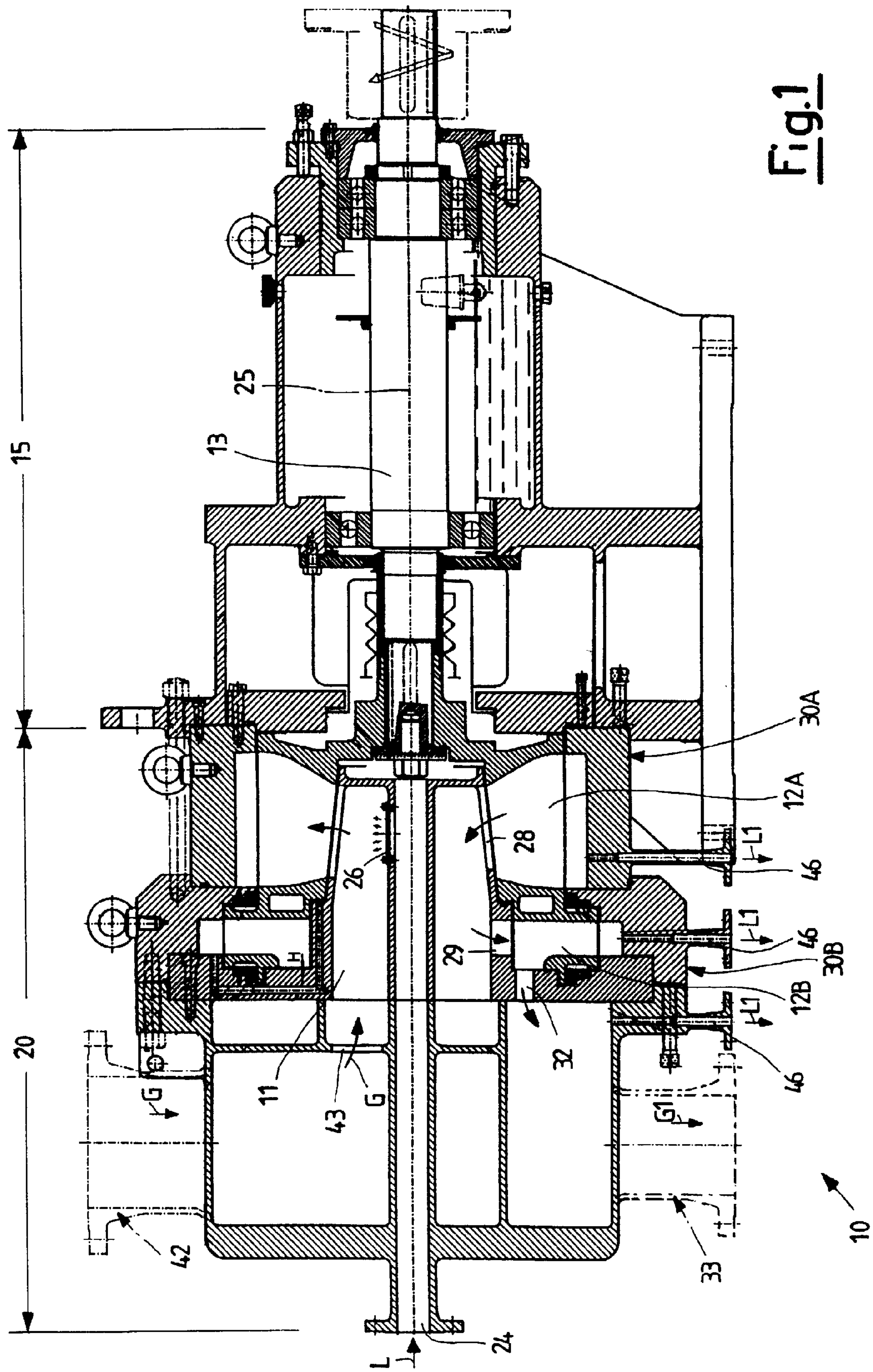

15
20
25
13
30A
12A
28
26
L1
46
46
30B
12B
29
32
11
43
G
G1
42
33
10
24
L

Fig.1

Fig.3

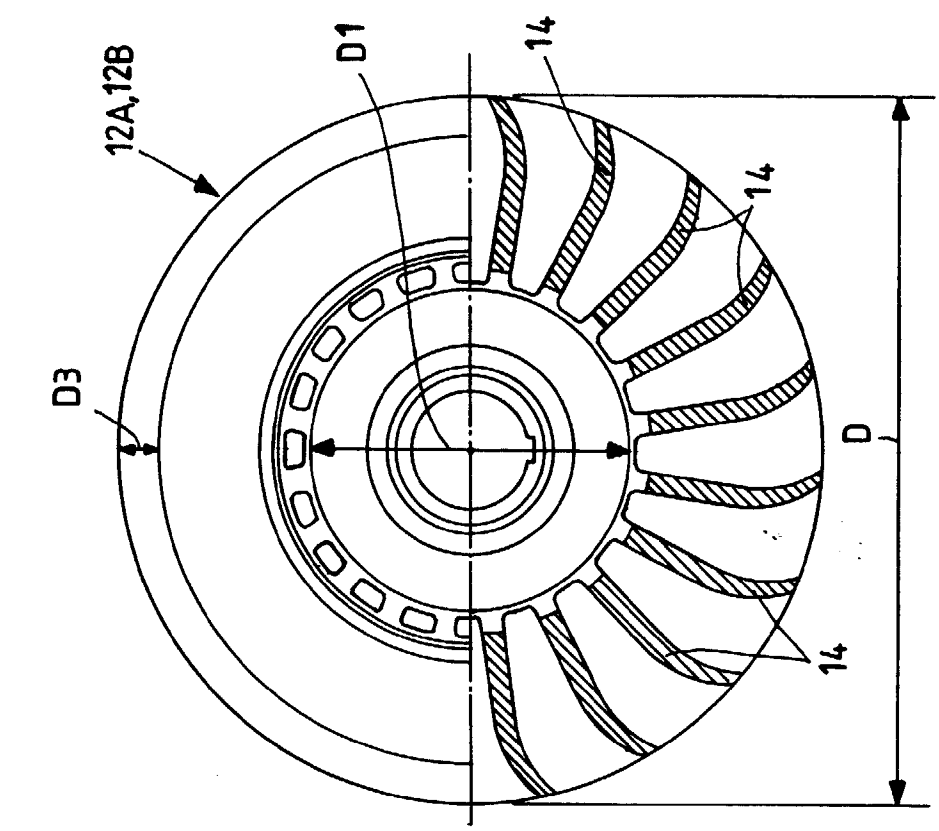


Fig.2

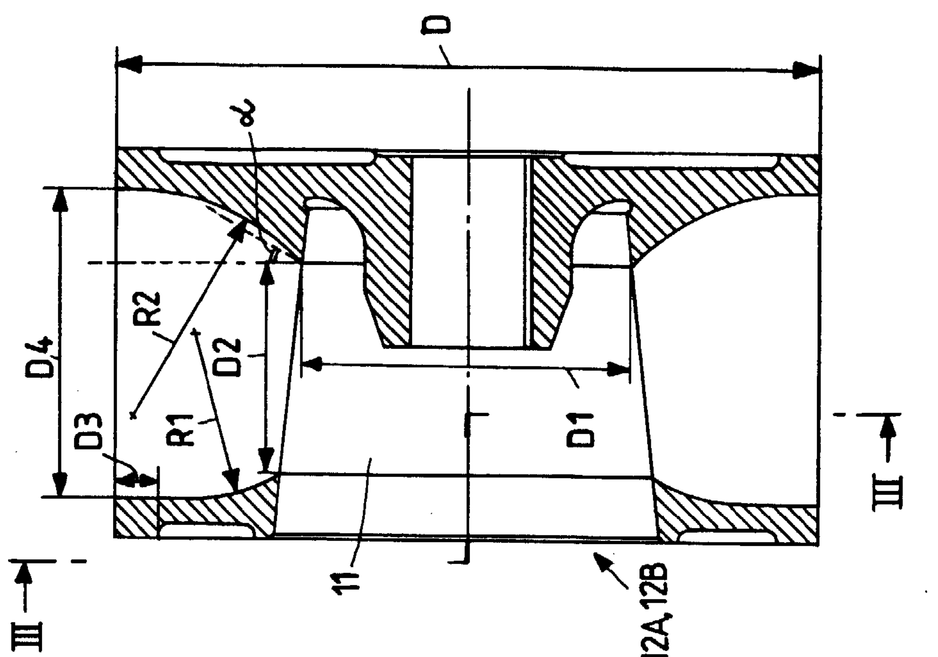


Fig.5

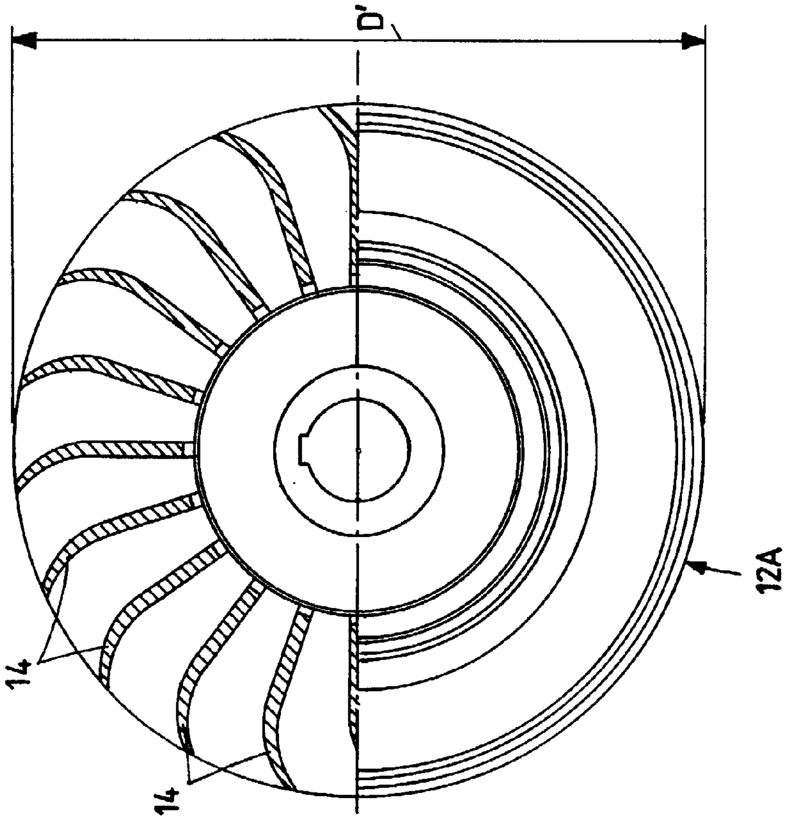


Fig.4

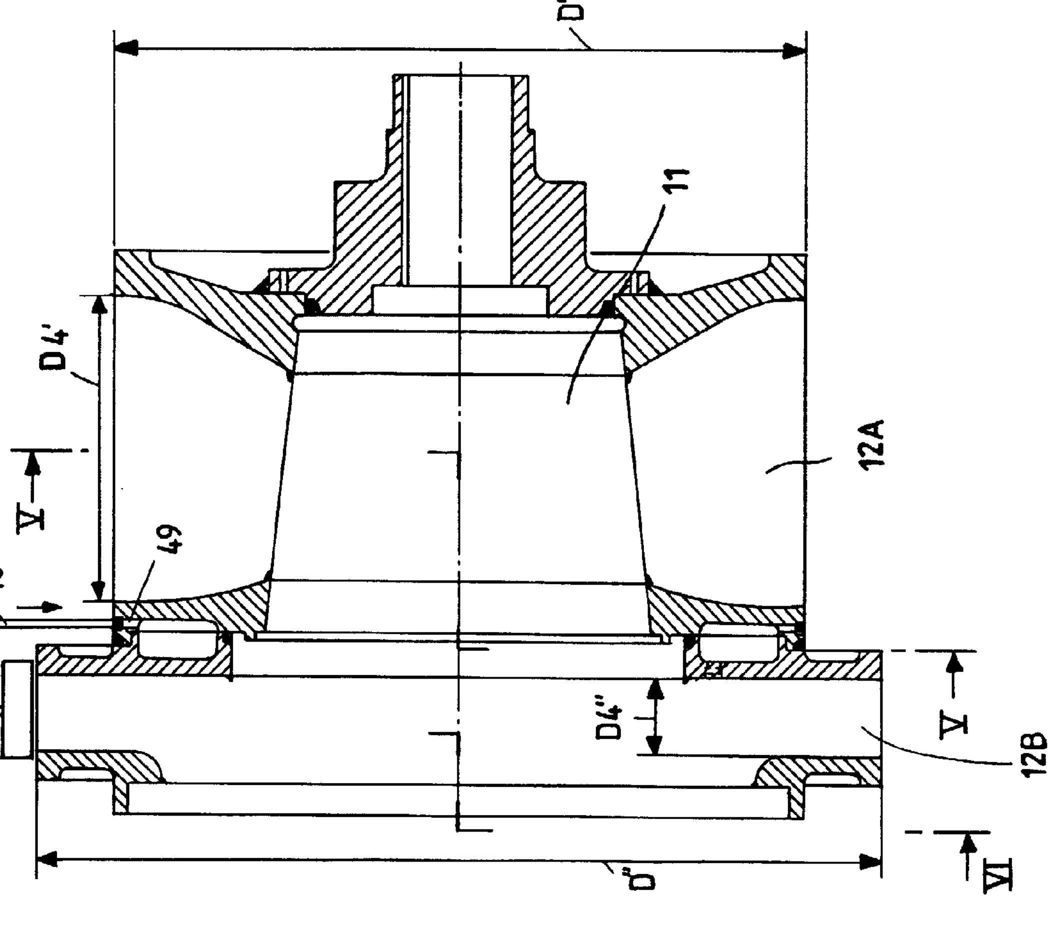
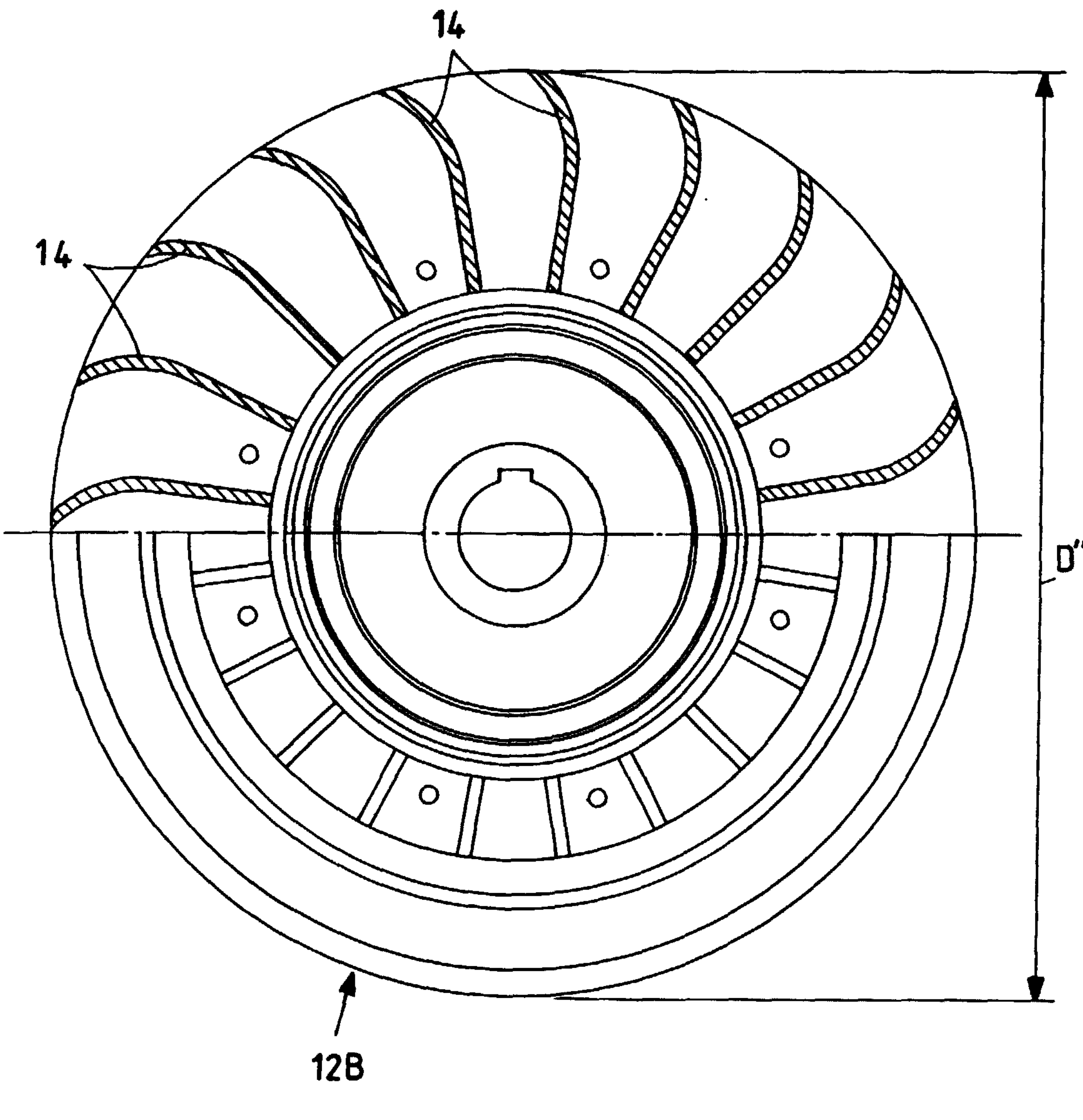


Fig.6



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PARTIAL EUROPEAN SEARCH REPORT

which under Rule 45 of the European Patent Convention shall be considered, for the purposes of subsequent proceedings, as the European search report

Application Number
EP 98 20 2222

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
Y	EP 0 474 476 A (A. AHLSTROM CO.) 11 March 1992 * column 4, line 35 - line 46; figure 2 * ---	1-6	F04C19/00
Y	WO 96 22467 A (ENERGY CONVERTERS LTD.) 25 July 1996 * page 15, paragraph 2 - paragraph 4 * * page 17, paragraph 2 * * page 20, paragraph 2 * ---	1-6	
X	US 3 712 764 A (SHEARWOOD) 23 January 1973 * column 2, line 17 - line 23 * ---	1	
A	EP 0 599 545 A (ASSAF) 1 June 1994 * column 1, line 7 - line 17 * * column 3, line 16 - line 23 * ---	1-3,5,6	
A	GB 1 185 754 A (ABBIONETA) 25 March 1970 * the whole document * ---	1-6	
A	GB 935 881 A (RAVERA) * the whole document * --- -/--	1-6	TECHNICAL FIELDS SEARCHED (Int.Cl.6) F04C

INCOMPLETE SEARCH

The Search Division considers that the present application, or one or more of its claims, does/do not comply with the EPC to such an extent that a meaningful search into the state of the art cannot be carried out, or can only be carried out partially, for these claims.

Claims searched completely :

Claims searched incompletely :
1-10

Claims not searched :
-

Reason for the limitation of the search:

The angle "a" is not defined clearly neither in the description nor in figure 2.

Place of search	Date of completion of the search	Examiner
THE HAGUE	22 September 1998	Dimitroulas, P

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
A	FR 1 439 894 A (NEYRPIC) 5 August 1966 * the whole document *	1	
A	US 3 351 272 A (JENNINGS) 7 November 1967 * column 3, line 42 - line 45 * * column 4, line 17 - line 22; figures 1,4,10 *	7,8	
			TECHNICAL FIELDS SEARCHED (Int.Cl.6)