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(54) **A centrifugal compressor having inlet guide vanes**

Zentrifugalverdichter mit Einlassleitschaufeln

Compresseur centrifuge avec des aubes de guidage pour l'entrée

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- **PATENT ABSTRACTS OF JAPAN vol. 005, no. 193 (M-101), 9 December 1981 (1981-12-09) & JP 56 115897 A (MITSUBISHI HEAVY IND LTD), 11 September 1981 (1981-09-11)**
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Description

BACKGROUND OF THE INVENTION

[0001] The present invention relates to a centrifugal compressor, and more particular, to a centrifugal compressor, which uses inlet guide vanes to perform flow rate control.

[0002] JP-A-56-115897 describes an example of a conventional centrifugal compressor having inlet guide vanes. The centrifugal compressor described in JP-A-56-115897 comprises a detector for detecting a discharge air pressure, a controller for finding a required angle of inclination of inlet guide vanes, and an actuator driven by a signal of the controller, in order that running at high efficiency is achieved by changing an angle of inclination of inlet guide vanes in response to pressure change in discharge air pressure to change a surge line of the compressor.

[0003] JP-A-57-65898 describes another example of a conventional centrifugal compressor having inlet guide vanes. The centrifugal compressor described in JP-A-57-65898 comprises means for detecting the rotational speed of the compressor and means for detecting an air temperature, and the inlet guide vanes are driven on the basis of signals from these means to impart pre-whirl to an air flowing into the compressor to change the flow characteristics of the compressor.

[0004] Further, JP-A-11-62894 describes a further example of a conventional centrifugal compressor. The centrifugal compressor described in JP-A-11-62894 comprises one or more free rotors provided between inlet guide vanes and a centrifugal impeller, and the free rotors store therein flow energy to make running of the compressor further stable.

[0005] With the centrifugal compressors described in JP-A-56-115897 and JP-A-57-65898, flow rate control by means of inlet guide vanes enables making the centrifugal compressors high in performance but no adequate consideration is given to the case where suction gas pressure of the centrifugal compressors are increased. That is, with compressors used for chemical plants or the like, in which a suction-side pressure becomes several times to ten times or more as high as atmospheric pressure, starting cannot be in some cases done unless a pressure difference between upstream and downstream sides of inlet guide vanes is large. In such case, it is feared that a pressure difference between upstream and downstream sides of inlet guide vanes is increased and a load applied on vanes of the inlet guide vanes is increased to give damage to the inlet guide vanes. However, the above-described patent publications take no account of such increase in load.

[0006] With the centrifugal compressor described in the JP-A-11-62894, the free rotors store therein kinetic energy to be able to prevent surging. However, the publication discloses nothing about a fear of generation of a situation, in which a large pressure difference is generated between upstream and downstream sides of inlet guide vanes to possibly give damage to the inlet guide vanes, at start-up, at which a suction pressure of a compressor becomes high, and does not describe cancellation of such disadvantages.

[0007] DE-A-1 013 033 discloses a centrifugal compressor comprising an outer cylinder having an inner diameter being largest at the area where inlet guide vanes are arranged. An inner cylinder is not arranged in the suction flow passage.

[0008] DE-A 821 879 describes a centrifugal compressor comprising an outer cylinder having an inner diameter decreasing in the direction of the inlet of the impeller. An inner cylinder is arranged coaxially within the outer cylinder. Guide vanes are attached at the end of a pin between the end face inner cylinder, the pin being slidably supported in the inner cylinder.

[0009] The problem underlying the invention is to provide a centrifugal compressor avoiding damage to the inlet vanes even when the suction pressure and thus a load on the inlet guide vanes is increased.

[0010] This problem is solved by a centrifugal compressor comprising the features of claim 1. Preferred embodiments are claimed in claim 2 to 3.

[0011] Other objects, features and advantages of the invention will become apparent from the following description of the embodiments of the invention taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWING

[0012]

Fig. 1 is a longitudinal, cross sectional view showing an embodiment of a centrifugal compressor according to the invention;

Fig. 2 is a view viewed from an arrow A in Fig. 1;

Fig. 3 is a longitudinal, cross sectional view showing a centrifugal compressor not constructed according to the invention; and

Figs. 4 and 5 are sectional views viewed from an arrow B in Fig. 3.

DETAILED DESCRIPTION OF THE INVENTION

[0013] Several embodiments of a centrifugal compressor according to the invention will be described below with reference to the drawings. Figs. 1 and 2 are views showing an embodiment of a centrifugal compressor, Fig. 1 is a longitudinal, cross sectional view, and Fig. 2 is a view viewed from an arrow A. A main shaft 100a of the centrifugal compressor is connected to a shaft of a driving machine (not shown) directly or through a speed-increasing gear. A centrifugal impeller 11a is mounted on a tip end of the main shaft 100a. A diffuser 101, which is defined by a side of a casing 42 and a side of an outer cylinder described in detail later is provided downstream of the centrifugal impeller 11a. The casing 42 receives therein bearings holding the main shaft 100a and shaft seal means. The diffuser 101 may be a ribbed diffuser, on which blades having a slight height in a widthwise direction of a flow passage as shown in the drawing are arranged at intervals in a circumferential direction, or a vaned diffuser, or a vaneless diffuser. A spiral-shaped scroll 102 is disposed downstream of the diffuser 101.

[0014] The scroll 102 is defined by a part of an outer cylinder 14a and the casing 42. An inner peripheral surface of the outer cylinder 14a forms a cylindrical-shaped suction flow passage on a suction side of the impeller 11a. An inner cylinder 13a supported by stays 15a is arranged centrally of the suction flow passage. A tip end of the inner cylinder 13a assumes a streamline shape to reduce a flow resistance. Outer peripheral sides of the stays 15a are fixed to the outer cylinder 14a. The stays 15a are disposed in a plurality of positions spaced substantially equally in the circumferential direction.

[0015] Inlet guide vanes 12a are arranged in the suction flow passage between the stays 15a and the impeller 11a and at a position to correspond to an intermediate portion of the inner cylinder 13a. The inlet guide vanes 12a are eleven in number in the embodiment and spaced equally in the circumferential direction. Rotating shafts 31 are provided on root sides of the inlet guide vanes 12a. The rotating shafts 31 are rotatably supported by bearings 16a, which are held on the outer cylinder 14a. An extended rotating shaft 17a, which extends outside of the casing 42 and is rotatably supported by a bearing 16c held on the casing 42, is connected to the rotating shaft 31 for one inlet guide vane among the plurality of inlet guide vanes 12a.

[0016] A radially extending arm 19a is attached to an end of the extended rotating shaft 17a extending outside of the casing 42. Meanwhile, arms 20a extending perpendicularly to the rotating shafts 31 are attached to intermediate portions of the rotating shafts 31 in the axial direction. Ends of the arms 20a are connected to a ring 18a through other arms 32. When a pneumatic power device or the like (not shown) is used to rotatingly drive the arm 19a disposed outside the casing 42, the extended rotating shaft 17a and the rotating shaft 31 connected with the extended rotating shaft 17a are rotated, and the arm 20a attached to the rotating shaft 31 connected with the extended rotating shaft 17a turns the ring 18a around a rotational axis of the impeller 11a. As the ring 18a turns around the rotational axis of the impeller, the arms 32 on the respective inlet guide vanes, which are connected with the ring 18a, are moved. And the respective rotating shafts 31 attached to the arms 32 are all together turned the same angle in the same direction as that, in which the extended rotating shaft 17a turns. Thereby, all the inlet guide vanes 12a are subjected to a change of the same magnitude in their angles.

[0017] The inlet guide vanes 12a configured in this manner are used to permit a working gas to flow into the impeller 11a at a predetermined flow angle. The working gas compressed by the impeller flows into the scroll 102 through the diffuser 101. At the time of steady state running, in which no flow rate control is effected, the inlet guide vanes 12a are fully opened. At this time, the inlet guide vanes 12a are oriented in a flow direction. At startup of the compressor, the inlet guide vanes 12a are turned to cause the working gas to have a whirl component. At this time, the suction flow passage is narrowed.

[0018] Hereupon, the inlet guide vanes 12a used in controlling a flow rate of the centrifugal compressor have an advantage that normally efficiency is favorable at other points than an operating point and torque at startup can be decreased comparatively. The inlet guide vanes 12a cause whirling of a gas sucked into the centrifugal compressor in addition to flow rate control. When the sucked gas has a whirling component, work amount of the impeller 11a varies. More specifically, head Δh given to the working gas by the impeller is

$$\Delta h = (1/g) \cdot (u_2 v_{u2} - u_1 v_{u1})$$

where u_2 designates a peripheral speed at an outlet of the impeller 11a, u_1 designates a peripheral speed at an inlet of the impeller 11a, v_{u2} designates a circumferential component of an absolute velocity of the working gas at the outlet of the impeller, v_{u1} designates a circumferential component of an absolute velocity of the working gas at the inlet of the impeller, and g designates the gravitational acceleration.

[0019] Here, when the inlet guide vanes 12a are not provided, a direction of an absolute velocity of the working gas at the inlet of the impeller 11a is radial. As a result, $v_{u1} = 0$ results. When the inlet guide vanes 12a impart whirling to a gas flow, $v_{u1} \neq 0$ results to enable increasing or decreasing the work amount of the impeller 11a.

[0020] Further, the use of the inlet guide vanes 12a makes it possible to decrease torque at startup. When a centrifugal compressor is driven by an induction motor capable of running only at a fixed speed, there is a case which must reduce torque at startup of the centrifugal compressor by virtue of restriction on current value and voltage value. For example, with compressors used for chemical plants or the like, there is a case in which inlet gas temperature, gas pressure and gas density are higher at startup than in a steady state. In this case, it is necessary to decrease torque at startup. Since with the use of inlet guide vanes, whirling is imparted to a working gas and a cross sectional area around an inlet guide vane portion is reduced, a mass flow rate decreases to reduce a load on an impeller. Thereby, a compressor can be started up.

[0021] The provision of the inlet guide vanes 12a makes it possible to start up a compressor even when temperature and pressure of a sucked gas are high, but pressure at the inlet of the impeller becomes smaller than the suction pressure. Thereby, a pressure difference is generated between upstream and downstream sides of the inlet guide vanes 12a and a load corresponding to the pressure difference is imposed on the inlet guide vanes 12a. With air compressors for atmospheric pressure suction, a pressure difference between upstream and downstream sides of inlet guide vanes is 1 atmospheric pressure at maximum, and so a load on the inlet guide vanes is comparatively small. With centrifugal compressors, in which a suction pressure is several times to ten times or more as high as atmospheric pressure, however, an inlet guide vane portion is narrowed for the purpose of starting up the compressor, with the result that a pressure difference between upstream and downstream sides of inlet guide vanes becomes large, so that a load on the inlet guide vanes is greatly increased.

[0022] Hereupon, according to the embodiment, instead of having the outer cylinder 14a and the inner cylinder 13a assuming a cylindrical shape of a fixed radius, both the outer cylinder 14a and the inner cylinder 13a are made larger in diameter around the inlet guide vanes 12a than at other areas. Along with this, both the outer cylinder 14a and the inner cylinder 13a are gently decreased in outer diameter in an area running from the large-diameter area to the inlet of the impeller. As compared with the case where the outer cylinder 14a and the inner cylinder 13a are fixed in radius in the flow direction, a cross sectional area of the flow passage is increased in the embodiment, in which the outer cylinder 14a increases a cross sectional area of the flow passage and the inner cylinder 13a decreases a cross sectional area of the flow passage.

[0023] In addition, when a cross sectional shape of the flow passage is changed with the above technique, an increase in diameter of the outer cylinder 14a can be made smaller than that of the inner cylinder 13a. For example, consideration is given to the case where an inner diameter d and an outer diameter D of the flow passage is changed without changing a cross sectional area S of the flow passage. When a lower suffix 1 and a lower suffix 2 designate states before and after change, $S = \pi(D_1^2 - d_1^2)/4 = \pi(D_2^2 - d_2^2)/4$ holds, so that a difference Δ between inner and outer diameters lead to $\Delta_2 < \Delta_1$ when $D_2 > D_1$. Accordingly, it is possible to shorten a radial length L of the inlet guide vanes 12a obtained by subtracting the inner diameter of the inner cylinder 13a from the inner diameter of the outer cylinder 14a. Since the radial length L of the inlet guide vanes 12a is shortened, strength of the inlet guide vanes 12a can be set in that range, which can resist a pressure difference between upstream and downstream sides of the inlet guide vanes 12a, required at startup of the compressor. The reason for this is as follows.

[0024] Those portions of the inlet guide vanes 12a, in which a maximum stress is generated, are root portions of the inlet guide vanes 12a. A maximum bending stress in these portions is proportional to the radial length L of the inlet guide vanes 12a to the third power and inversely proportional to a thickness of the inlet guide vanes 12a to the third power. When the inlet guide vanes 12a are increased in thickness, a flow sucked into the compressor is made turbulent, so that a nonuniform flow enters into the impeller 11a. Hereupon, instead of increasing the inlet guide vanes 12a in thickness, the radial length L of the inlet guide vanes 12a is shortened. When the radial length L of the inlet guide vanes 12a is shortened, there is also produced an effect that the inlet guide vanes 12a rise in natural frequency since the natural frequency of the inlet guide vanes 12a is inversely proportional to the radial length L to the second power.

[0025] Another example of a centrifugal compressor will be described with reference to a longitudinal, cross sectional view shown in Fig. 3. This centrifugal compressor is suitable in the case where a pressure difference between the front and the back of inlet guide vanes is larger than that in the embodiment shown in Fig. 1. That is, the present embodiment is applied in the case where with only a change in a ratio of inner and outer diameters of a flow passage in an inlet guide vane portion, it is feared that a bending stress acting on inlet guide vanes is increased to make the inlet guide vanes incapable of resisting the pressure difference.

[0026] In this case instead of changing a shape of a suction flow passage of an impeller, inlet guide vanes are divided radially into a plurality of sections. Also, portions communicated to a downstream side from an upstream side are formed in portions of a cross section perpendicular to an axis of the suction flow passage and in the neighborhoods of cut-off points of the inlet guide vanes by the divided inlet guide vanes. Thereby, a necessary pressure difference can be generated between the front and the back of the inlet guide vanes at startup, and strength of the inlet guide vanes is made to be able to resist the pressure difference.

[0027] The divided guide vanes 12c are arranged on an inner diameter side of the inlet guide vanes 12b. When the inlet guide vanes 12b and the divided guide vanes 12c are registered with each other, a projection in a direction perpendicular to an axis is made sector-shaped. At the time of steady state running, both the inlet guide vanes 12b and the divided guide vanes 12c are positioned to extend along a flow direction Fin as shown in Fig. 4. Meanwhile, the inlet guide vanes 12b are turned at startup to an angle, at which the suction flow passage is closed. However, the divided guide vanes 12c are turned to a different angle from that angle, to which the inlet guide vanes 12b are turned.

[0028] Here, the divided guide vanes 12c are mounted on an inner cylinder 13b. Rotating shafts 26 extending in a direction perpendicular to a rotating shaft 100a of an impeller 11a are arranged around a piston 23 within the inner cylinder 13b. Pinions 27 are mounted on the rotating shafts 26. Center positions of the rotating shafts 26 in a flow direction correspond to center positions of the inlet guide vanes 12b in the flow direction. Racks 28 adapted to mesh with the pinions 27 are mounted on the piston 23. An end of the piston 23 toward the impeller 11a is fitted into a sleeve 25 arranged in the inner cylinder 13b. An end of the piston 23 on a suction side is restrained to the inner cylinder by a spring 24. A hole 21 providing communication between the suction flow passage and an interior of the inner cylinder 13b is formed in a wall of the inner cylinder 13b at a position beyond the sleeve 25 toward the impeller 11a. Likewise, Another hole 22 providing communication between the suction flow passage and the interior of the inner cylinder is formed in the wall of the inner cylinder 13b at a position, which corresponds to the spring 24. These holes 21, 22 are formed on upstream and downstream sides in the flow direction with respect to the divided guide vanes 12c. Pivot bearings are formed centrally on inner peripheral surfaces of the inlet guide vanes 12b, and pivots adapted to be fitted into the pivot bearings are formed centrally on outer peripheral surfaces of the divided guide vanes 12c.

[0029] Pressure of a gas in the suction flow passage upstream of the inlet guide vanes 12b is applied on the piston 23 through the hole 22. Pressure of the gas in the suction flow passage downstream of the inlet guide vanes 12b is applied on the piston 23 through the hole 21. Sealing is provided between the piston 23 and the sleeve 25 to prevent the working gas from going and coming. When a pressure difference between gas pressures on upstream and downstream sides of the inlet guide vanes 12b is small, the force of the spring 24 acting on the piston 23 acts on the racks 28 to move the racks 28 toward the impeller 11a. When the racks 28 are moved, the pinions 27 meshing therewith are rotated to push the divided guide vanes 12c against the inlet guide vanes 12b. That is, at the time of steady state running of the compressor, the inlet guide vanes 12b and the divided guide vanes 12c are put in an assembled state, and the inlet guide vanes 12b and the divided guide vanes 12c are turned in the same direction.

[0030] At startup of the compressor, the inlet guide vanes 12b are turned as shown in Fig. 5 to shut off the suction flow passage. Since the suction flow passage is shut off, a pressure difference is generated between upstream and downstream sides of the inlet guide vanes 12b. The working gas flowing into the holes 22 and 21 produces a force corresponding to the pressure difference to move the piston 23 against the force of the spring 24. The racks 28 mounted on the piston 23 are moved to turn the pinions 27, thus separating the divided guide vanes 12c from the inlet guide vanes 12b. As a result, the suction flow passage is shifted from the shut-off state to be put in a state, in which slight openings are present. Since the openings are formed in the suction flow passage, a pressure difference between pressures of the working gas on upstream and downstream sides of the inlet guide vanes 12b decreases and a bending stress acting on the inlet guide vanes 12b is reduced. In the present embodiment, it is necessary to appropriately set the spring force of the spring 24 connected to the piston 23. Setting is made so that the spring force of the spring 24 moves the piston when a pressure difference between the upstream side and the downstream side of the inlet guide vanes 12b exceeds that pressure difference, at which the compressor can be started up.

[0031] Since a bending stress caused by pressure of the working gas acting on the inlet guide vanes is reduced, a centrifugal compressor can be improved in reliability even when a suction pressure is high. Also, it is possible to avoid surging of a centrifugal compressor.

Claims

1. A centrifugal compressor comprising:

a centrifugal impeller (11a) mounted on a shaft (100a) which is connected to a driving machine;
an outer cylinder (14a) forming a cylindrical-shaped suction flow passage on a suction side of the impeller (11a);
an inner cylinder (13a) arranged radially centrally in the suction flow passage; and
a plurality of inlet guide vanes (12a) arranged in circumferential direction between the inner cylinder (13a) and the outer cylinder (14a) at a position corresponding to an intermediate portion of the inner cylinder (13a), wherein the inner diameter of the outer cylinder (14a) and the outer diameter of the inner cylinder (13a) are larger around said plurality of inlet guide vanes (12a) than in other areas and decrease to the inlet of the centrifugal impeller (11a), and wherein said plurality of inlet guide vanes (12a) has rotating shafts (31) which are provided on root sides thereof.

2. A centrifugal compressor according to claim 1, **characterized in that** the cross sectional area of the suction flow passage perpendicular to the axis of the suction flow passage at the area where the inlet guide vanes (12a) are arranged is larger than that of the inlet of the impeller (11a).

3. A centrifugal compressor according to claim 1 or 2, **characterized in that** the pressure of a working gas sucked into the centrifugal compressor is at least 1 MPa.

Patentansprüche

1. Zentrifugalverdichter

   - mit einem Zentrifugallaufrad (11 a) das auf einer Welle (1 00a) montiert ist, die mit einer Antriebsmaschine verbunden ist,
   - mit einem Außenzylinder (14a), der einen zylinderförmigen Ansaugströmungskanal auf einer Ansaugseite des Laufrades (11a) bildet,
   - mit einem Innenzylinder (13a), der radial zentral in dem Ansaugströmungskanal angeordnet ist, und
   - mit einer Vielzahl von Einlassleitschaufeln (12a), die in Umfangsrichtung zwischen dem Innenzylinder (13a) und dem Außenzylinder (14a) an einer Position angeordnet sind, die einem Zwischenabschnitt des Innenzylinders (13a) entspricht,
   - wobei der Innendurchmesser des Außenzylinders (14a) und der Außendurchmesser des Innenzylinders (13a) um die Vielzahl von Einlassleitschaufeln (12a) herum größer als in anderen Bereichen sind und zum Einlass des Zentrifugallaufrads (11a) hin abnehmen, und
   - wobei die Vielzahl von Einlassleitschaufeln (12a) Drehachsen (31) aufweist, die an ihren Fußseiten vorgesehen sind.

2. Zentrifugalverdichter nach Anspruch 1, **dadurch gekennzeichnet, dass** die Querschnittsfläche des Ansaugströmungskanals senkrecht zur Achse des Ansaugströmungskanals in dem Bereich, in welchem die Einlassleitschaufeln (12a) angeordnet sind, größer ist als die des Einlasses des Laufrads (11a).

3. Zentrifugalverdichter nach Anspruch 1 oder 2, **dadurch gekennzeichnet, dass** der Druck des in den Zentrifugalverdichter gesaugten Arbeitsgases wenigstens 1 MPa beträgt.

Revendications

1. Compresseur centrifuge comprenant :

   une roue centrifuge (11a) montée sur un arbre (100a) qui est relié à une machine d'entraînement ;
   un cylindre externe (14a) formant un passage d'écoulement d'aspiration en forme de cylindre sur un côté d'aspiration de la roue (11a) ;
   un cylindre interne (13a) agencé de manière radiale au centre du passage d'écoulement d'aspiration ; et
   une pluralité d'aubes directrices d'admission (12a) agencées dans le sens de la circonférence entre le cylindre interne (13a) et le cylindre externe (14a) dans une position correspondant à une partie intermédiaire du cylindre interne (13a), dans lequel
   le diamètre interne du cylindre externe (14a) et le diamètre externe du cylindre interne (13a) sont plus importants autour de ladite pluralité d'aubes directrices d'admission (12a) que dans d'autres zones et diminuent vers l'admission de la roue centrifuge (11a), et dans lequel
   ladite pluralité d'aubes directrices d'admission (12a) ont des arbres tournants (31) qui sont réalisés sur les côtés d'emplantures de ces dernières.

2. Compresseur centrifuge selon la revendication 1, **caractérisé en ce que** la zone de section transversale du passage d'écoulement d'aspiration perpendiculaire à l'axe du passage d'écoulement d'aspiration au niveau de la zone dans laquelle les aubes directrices d'admission (12a) sont agencées est supérieure à celle de l'admission de la roue (11a).

3. Compresseur centrifuge selon la revendication 1 ou 2, **caractérisé en ce que** la pression d'un gaz de travail aspiré dans le compresseur centrifuge est d'au moins 1 MPa.

FIG. 1

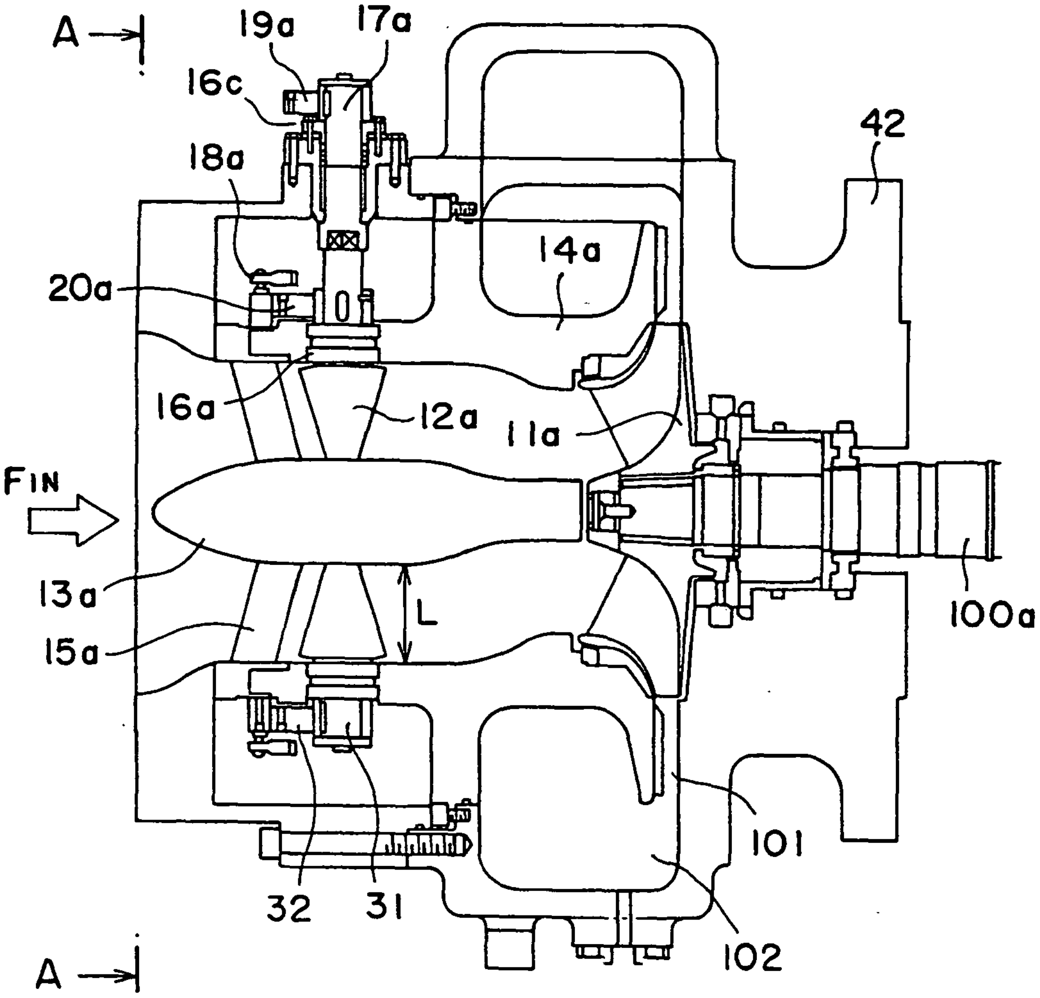

A
19a
17a
16c
42
18a
14a
20a
16a
12a
11a
FIN
13a
L
100a
15a
101
32
31
102
A

FIG. 2

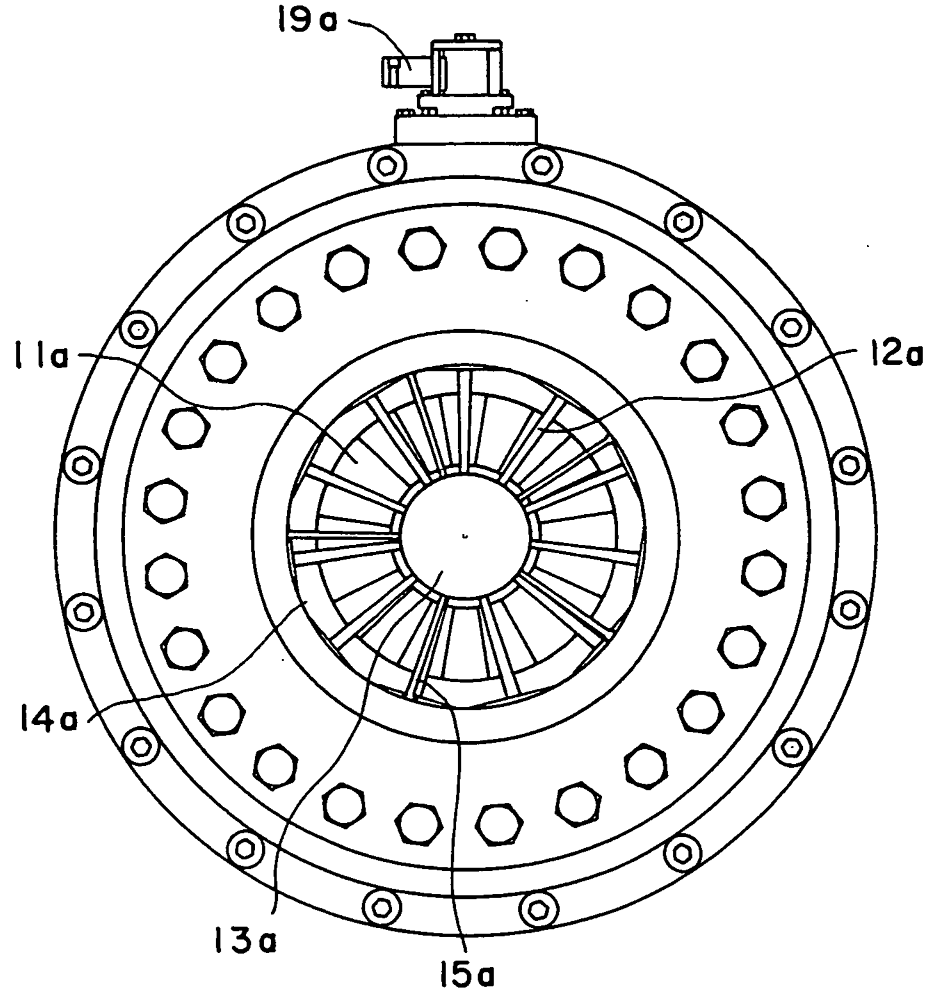


FIG. 3

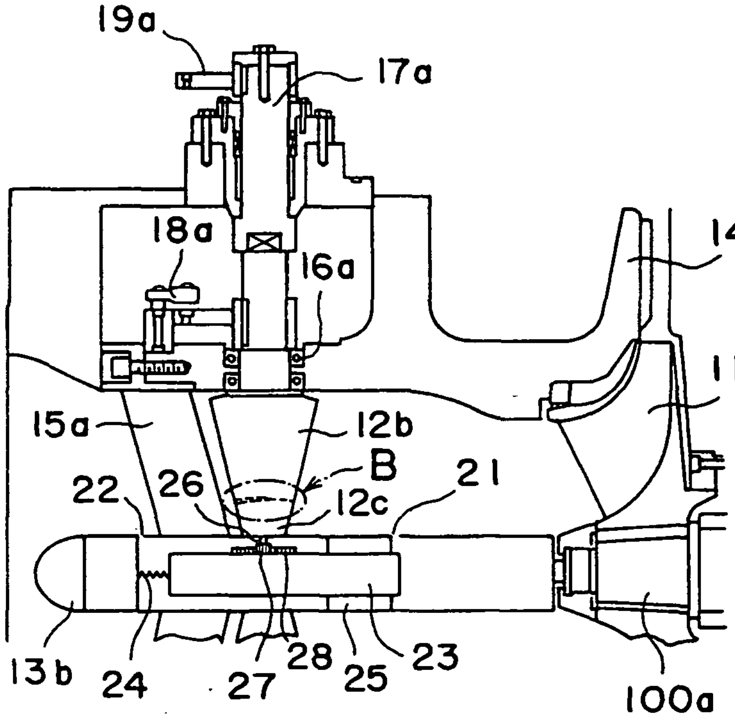

19a
17a
18a
16a
14b
11a
15a
12b
B
22
26
21
12c
13b
24
27
28
25
23
100a

FIG. 4

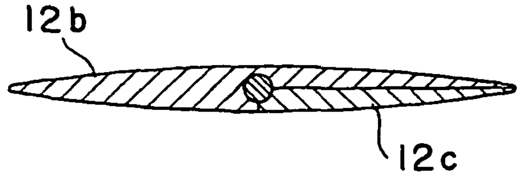
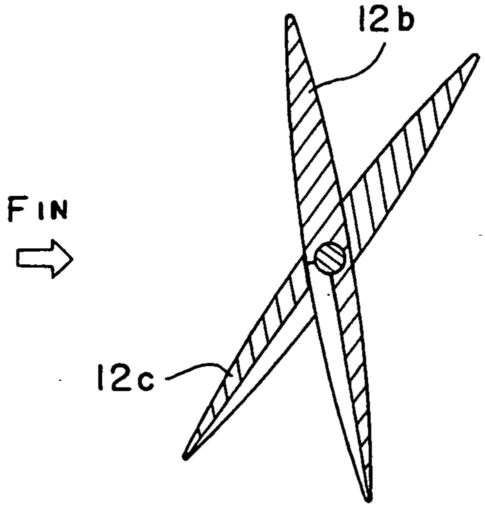

12b
12c

FIG. 5



REFERENCES CITED IN THE DESCRIPTION

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