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(54) CENTRIFUGAL COMPRESSOR

ZENTRIFUGALVERDICHTER

COMPRESSEUR CENTRIFUGE

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EP 2 806 170 B1

Description

TECHNICAL FIELD

[0001] The present invention relates to a centrifugal compressor, and particularly to a centrifugal compressor with a large flow rate.

BACKGROUND ART

[0002] For improving the performances of products such as superchargers, gas turbines, and industrial compressors, it is a critical issue to increase the flow rate. Increasing the flow rate of a centrifugal compressor means to increase the discharge flow rate of a compressor with the same shell size, and further of an impeller with the same outer diameter.

[0003] One of the problems associated with the increase in flow rate is a decrease in efficiency. For this reason, a technique of increasing the flow rate while suppressing the decrease in efficiency is industrially very significant.

[0004] A conventional centrifugal compressor will be described by using FIG. 6. The centrifugal compressor mainly includes a suction inlet 1, an impeller 2, a hub 3, a rotary shaft 4, a diffuser 5, and a scroll 6.

[0005] The impeller 2 is connected to the rotary shaft 4 via the hub 3. The diffuser 5 is provided downstream of the impeller 2, has a flow passage extending in a direction away from the rotary shaft 4, and has an outlet directed in a radial direction in a meridian plane. Moreover, the scroll 6 is provided downstream of the diffuser 5 and communicates with the outlet of the diffuser 5.

[0006] The suction inlet 1 plays a roll of guiding a gas to the impeller 2. The centrifugal compressor is configured such that the gas guided to the impeller 2 is sucked into the centrifugal compressor by the impeller 2 being rotated by the rotary shaft 4. The velocity of the gas having passed through the impeller 2 is decreased, and the pressure of the gas is increased, in the diffuser 5. The gas having passed through the diffuser 5 flows into the scroll 6, and thereafter flows into a discharge port, which is not shown. In this way, the centrifugal compressor converts the kinetic energy of the gas to a pressure.

[0007] FIG. 7 is a cross-sectional view of the diffuser 5 and the impeller 2 of the conventional centrifugal compressor. The more the velocity vector of the gas (the arrow in the figure) is directed in a radial direction in the meridian plane, the less the energy loss is. However, a large flow rate causes the velocity distribution of the gas sacked through an impeller inlet 2a to be skewed to the hub 3 side at an impeller outlet 2b. As a result, the velocity vector is inclined toward an axial direction from the radial direction. In addition, if the gas flows further inside the diffuser 5 in this state, the velocity distribution is further skewed to become a cause of occurrence of shear stress, reducing the amount of static pressure recovery, and in turn leading to a decrease in efficiency of the entire compressor.

[0008] To solve the above-described problem, there is a method of bringing the velocity distribution of the gas into a more uniform distribution by providing the inside of the diffuser with a guide blade (see Japanese Patent No. 2569143) or a guide flow passage to the impeller inlet (see Japanese Patent No. 2703055).

PRIOR ART DOCUMENTS

PATENT DOCUMENTS

[0009] The centrifugal compressor according to EP 2 072 834 A1 has a diffusor structure which aims to prevent the airflow to separate from a hub side wall surface on a hub side wall surface downstream side within a diffusor passage. To this end, a hub side wall surface of the diffusor passage is provided with an inclined plane that approaches toward a shroud side in a position on a downstream side of a portion parallel with a normal line direction of a section surface of an impeller exit. The inclination angle is preferable set less than or equal to 20 degrees and preferably larger than 2 degrees and less than or equal to 10 degrees.

[0010] The compressor according to US 2002/0106278 A1 has a diffusor section which is constructed in such a way that the width dimension in the axial direction of the outlet opening is made larger than the width dimension of the inlet opening.

[0011] Another radial compressor as disclosed in US 2010/0178163 A1 has a diffusor constructed such that a low-pressure region in the area of the transition between the spiral housing and the tongue of the radial compressor is at least reduced.

[0012] In order to increase the efficiency of turbo-compressors, GB 152,689 A suggests to introduce air with a higher tension than that coming from the rotor into the space of the guiding device surrounding the outflow apertures of the rotor. This serves to oppose an increased resistance to the outflow of the air and to compress the air more completely.

[0013] The centrifugal compressor according to JP-H11 82389 A prevents the back flow in a diffuser by increasing the width of the channel toward the downstream side.

[0014] The centrifugal compressor according to JP 2008/175124 A also aims to restrain surge without bringing about a decline of the compressor unit efficiency by providing a shroud-sidewall surface of the diffusor path as a tapered surface.

[0015] EP 2 314 876 A2 represents the closest prior art. Further, EP 2 314 876 A2 discloses the provision of an angle formed by the diffusor inlet hub-side line with the radial direction in the meridian plane at a point closest to an outlet of impeller in the diffusor inlet hub-side line which ranges from 6° to 12°.

SUMMARY OF THE INVENTION

PROBLEMS TO BE SOLVED BY THE INVENTION

[0016] However, the above-described apparatuses require that new mechanisms should be provided in the diffuser, and have a possibility that the production costs and working hours are wasted.

[0017] In view of this, an object of the present invention is to solve the problems of the conventional techniques and to achieve higher efficiency in a centrifugal compressor with a large flow rate, not by providing a new mechanism, but by changing a shape of a diffuser.

MEANS FOR SOLVING THE PROBLEMS

[0018] This object is solved by a centrifugal compressor with the features of claim 1.

EFFECT OF THE INVENTION

[0019] According to the centrifugal compressor of the first invention, since θ-α>0°, skewing of the velocity distribution of the gas is eliminated, and accordingly a decrease in amount of static pressure recovery is suppressed. Therefore, a higher efficiency of the entire compressor can be achieved.

[0020] According to the centrifugal compressor of the invention, since 20°<θ<30°, the skewing of the velocity distribution of the gas can be further eliminated.

BRIEF DESCRIPTION OF THE DRAWINGS

[0021]

FIG. 1 is a cross-sectional view of a diffuser and an impeller of a centrifugal compressor according to Embodiment 1 of the present invention.

FIG. 2 is a graph showing a relationship between θ and an efficiency improvement rate of the centrifugal compressor according to Embodiment 1 of the present invention.

FIG. 3 is a cross-sectional view of a diffuser and an impeller of a centrifugal compressor according to Embodiment 2 of the present invention.

FIG. 4 is a schematic view showing differences between the centrifugal compressor according to Embodiment 1 of the present invention and the centrifugal compressor according to Embodiment 2 of the present invention.

FIG. 5 is a schematic diagram showing a relationship between 8 and α of the centrifugal compressor according to Embodiment 1 or 2 of the present invention.

FIG. 6 is a cross-sectional view of a conventional centrifugal compressor.

FIG. 7 is a cross-sectional view of a diffuser and an impeller of the conventional centrifugal compressor.

MODES FOR CARRYING OUT THE INVENTION

[0022] Hereinafter, a centrifugal compressor according to the present invention will be described referring to embodiments by use of the drawings.

Embodiment 1

[0023] An apparatus according to Embodiment 1 of the present invention will be described by use of FIG. 1. The apparatus mainly includes a suction inlet 1, an impeller 2, a hub 3, a rotary shaft 4, a diffuser 5, and a scroll 6, as in the case of the conventional centrifugal compressor. The impeller 2 is connected to the rotary shaft 4 via the hub 3. In addition, the diffuser 5 is provided downstream of the impeller 2, has a flow passage directed in a direction away from the rotary shaft 4, and has an outlet directed in a radial direction in a meridian plane. Moreover, the scroll 6 is provided downstream of the diffuser 5, and communicates with an outlet of the diffuser 5. Note that the rotary shaft 4 and the scroll 6 are not shown in FIG. 1, but are assumed to be the same as those of the conventional technical.

[0024] Furthermore, like the conventional technique, the suction inlet 1 plays a role of guiding a gas to the impeller 2. The centrifugal compressor is configured such that the gas guided to the impeller 2 is sucked into the centrifugal compressor by the impeller 2 being rotated by the rotary shaft 4. The velocity of the gas having passed through the impeller 2 is decreased, and the pressure of the gas is increased, in the diffuser 5. The gas having passed through the diffuser 5 flows into the scroll 6, and thereafter flows into a discharge port.

[0025] Here, a line on the hub 3 side in the inlet of the diffuser 5 (hereinafter, stated as a diffuser inlet hub-side line 5a) is inclined toward an axial direction from the radial direction in the meridian plane. At this time, an angle formed by the diffuser inlet hub-side line 5a with the radial direction at a point B closest to the impeller outlet 2b in the diffuser inlet hub-side line 5a is represented by θ.

[0026] Next, an angle formed by a tangent line 3b of a line on the hub 3 side in the impeller 2 (hereinafter, stated as an impeller hub-side line 3a) with the radial direction at a point A closest to an inlet of the diffuser 5 in the impeller hub-side line 3a is represented by α.

[0027] The conventional technique is set such that θ=α in order to smoothly connect the impeller hub-side line 3a and the diffuser inlet hub-side line 5a. On the other hand, the present apparatus is set such that θ-α>0° as shown in FIG. 5, and further θ is set such that 20°< θ <30°.

[0028] In addition, an angle formed by the impeller rear edge 2c with the axial direction is represented by β. Here, β has not necessary to be limited, but is set such that 0 °≤β≤35°, which is a value used in a general centrifugal compressor.

[0029] Note that a line of the shroud 7 is also inclined in conjunction with the inclination of θ to confirm with a diffuser width ratio of the conventional shape. The diffuser width ratio is b3/b2(see FIG. 1), and has a value set for each impeller. In general, the value of the diffuser width ratio is set such that b3/b2=0.6 to 1.0.

[0030] With the above-described structure, while the velocity vector of the gas at the time when the gas has flowed from the impeller outlet 2b to the diffuser 5 is not changed from that of the conventional one, the skewing of the velocity distribution can be suppressed.

[0031] FIG. 2 shows a result of simulation of the compressor efficiency of the present apparatus, conducted under conditions that α and β are certain constant values and only θ is a variable. In a graph of FIG. 2, the horizontal axis represents θ and the vertical axis represents a compressor efficiency improvement rate. The compressor efficiency improvement rate represents a difference, expressed in percentage, between the compressor efficiency of the present apparatus and the compressor efficiency of the conventional technique. As becoming higher in the graph, the compressor efficiency improvement rate indicates that the compressor efficiency of the present apparatus is higher. It can be understood from the graph that the compressor efficiency is improved when $20° < \theta < 30°$.

[0032] Accordingly, in the present apparatus, the skewing of the velocity distribution of the gas in the diffuser, which has conventionally occurred, is eliminated, and accordingly a decrease in the amount of static pressure recovery in the diffuser is suppressed. Therefore, a higher efficiency of the entire compressor can be achieved.

Embodiment 2

[0033] An apparatus according to Embodiment 2 of the present invention is one obtained by improving the apparatus according to Embodiment 1. FIG. 4 shows differences between the apparatus according to Embodiment 1 and the present apparatus. In the apparatus according to Embodiment 1, since the diffuser inlet hub-side line 5a is a straight line, directing the outlet of the diffuser 5 in the radial direction requires that the angle of the diffuser 5 has to be changed at a certain portion. As a result, as shown in FIG. 4, a stagnation region 11 where the flow of the gas stagnates is formed. Shear stress acts between the gas stagnating in the stagnation region 11 and the flowing gas, leading to a possibility of occurrence of an energy loss. The present apparatus reduces the stagnation region 11.

[0034] As in the case of the apparatus according to Embodiment 1, as shown in FIG. 3, the present apparatus mainly includes a suction inlet 1, an impeller 2, a hub 3, a rotary shaft 4, a diffuser 5, and a scroll 6. The impeller 2 is connected to the rotary shaft 4 via the hub 3. The diffuser 5 is provided downstream of the impeller 2, has a flow passage extending in a direction away from the rotary shaft 4, and has an outlet directed in a radial direction in a meridian plane. Moreover, the scroll 6 is provided downstream of the diffuser 5 and communicates with the outlet of the diffuser 5. Note that the rotary shaft 4 and the scroll 6 are not shown in FIG. 3, but are assumed to be the same as those of the conventional technique. Furthermore, the operation of the present apparatus is also the same as those of the apparatus according to Embodiment 1 and of the conventional technique, and is accordingly omitted.

[0035] Here, in the present apparatus, the diffuser inlet hub-side line 5b is made to be a concave curved line. An angle formed by a tangent line 5c of the diffuser inlet hub-side line 5b with a radial direction at a point B closest to an impeller outlet 2b in the diffuser inlet hub-side line 5b is represented by θ. The line of the shroud 7, α, and β are set such that $\theta - \alpha > 0°$ as shown in FIG. 5, and further θ is set such that $20° < \theta < 30°$ as in the case of the apparatus according to Embodiment 1. Note that the diffuser inlet hub-side line 5b may be a single arc, or may be a line obtained by smoothly combining a plurality of arcs or ovals, as long as it is a curved line.

[0036] With the above-described structure, as shown in FIG. 4, the present apparatus can reduce the stagnation region 11, which exists in the case of the apparatus according to Embodiment 1. Therefore, the present apparatus can reduce shear stress and makes it possible to achieve higher efficiency.

INDUSTRIAL APPLICABILITY

[0037] The present invention is favorable as a centrifugal compressor, and in particular a centrifugal compressor with a large flow rate.

EXPLANATION OF REFERENCE NUMERALS

[0038]

1	suction inlet
2	impeller
2a	impeller inlet
2b	impeller outlet
2c	impeller rear edge
3	hub
3a	impeller hub-side line
3b	tangent line
4	rotary shaft
5	diffuser
5a	diffuser inlet hub-side line (in the apparatus according
to	Embodiment 1 of the present invention)
5b	diffuser inlet hub-side line (in the apparatus according
to	Embodiment 2 of the present invention)
5c	tangent line
6	scroll
7	shroud
11	stagnation region

Claims

1. A centrifugal compressor comprising:

an impeller (2) connected to a rotary shaft via a hub (3); and
a diffuser (5) provided downstream of the impeller (2), the diffuser (5) having a flow passage which extends in a direction away from the rotary shaft and an outlet which is directed in a radial direction in a meridian plane, wherein
the centrifugal compressor satisfies θ-α>0°
where

θ is an angle formed by a diffuser inlet hub-side line (5a) with the radial direction in the meridian plane at a point closest to an outlet (2b) of the impeller (2) in the diffuser inlet hub-side line (5a), the diffuser inlet hub-side line (5a) being a line on the hub (3) side in an inlet of the diffuser (5), and
α is an angle formed by a tangent line (3b) of an impeller hub-side line (3a) with the radial direction in the meridian plane at a point closest to the inlet of the diffuser (5) in the impeller hub-side line (3a), the impeller hub-side line (3a) being a line on the hub (3) side in the impeller, and **characterized in that** the angle θ is such that 20°<θ<30° when the diffuser inlet hub-side line (5a) is a straight line, β is such that 0°≤β≤35° where β is an angle formed by a rear edge (2c) of the impeller (2) with an axial direction.

Patentansprüche

1. Zentrifugalverdichter, umfassend:

ein Laufrad (2), das über eine Nabe (3) mit einer Drehwelle verbunden ist; und
einen Diffusor (5), der dem Laufrad (2) nachgelagert bereitgestellt ist, wobei der Diffusor (5) einen Strömungsdurchgang, der sich in eine Richtung weg von der Drehwelle erstreckt, und einen Auslass aufweist, der in eine radiale Richtung in einer meridianen Ebene gerichtet ist, wobei
der Zentrifugalverdichter θ-α>0° erfüllt
wo

θ ein Winkel ist, der durch eine Diffusoreinlass-Nabenseitenlinie (5a) mit der radialen Richtung in der meridianen Ebene an einem Punkt gebildet wird, der einem Auslass (2b) des Laufrads (2) in der Diffusoreinlass-Nabenseitenlinie (5a) am nächsten ist, wobei die Diffusoreinlass-Nabenseitenlinie (5a) eine Linie auf der Seite der Nabe (3) in einem Einlass des Diffusors (5) ist, und
α ein Winkel ist, der durch eine Berührungslinie (3b) einer Laufrad-Nabenseitenlinie (3a) mit der radialen Richtung in der meridianen Ebene an einem Punkt gebildet wird, der dem Einlass des Diffusors (5) in der Laufrad-Nabenseitenlinie (3a) am nächsten ist, wobei die Laufrad-Nabenseitenlinie (3a) eine Linie auf der Seite der Nabe (3) im Laufrad ist, und **dadurch gekennzeichnet, dass** der Winkel θ so ist, dass 20°<θ<30°, wenn die Diffusoreinlass-Nabenseitenlinie (5a) eine gerade Linie ist, β so ist, dass 0°≤β≤35°, wo β ein Winkel ist, der durch eine Hinterkante (2c) des Laufrads (2) mit einer axialen Richtung gebildet wird.

Revendications

1. Compresseur centrifuge comprenant :

une roue à aubes (2) raccordée à un arbre rotatif via un moyeu (3) ; et
un diffuseur (5) prévu en aval de la roue à aubes (2), le diffuseur (5) présentant un passage de flux qui s'étend dans une direction loin de l'arbre rotatif et une sortie qui est dirigée dans une direction radiale dans un plan méridien, dans lequel
le compresseur centrifuge satisfait à θ-α>0°
où

θ est un angle formé par une ligne côté moyeu d'entrée de diffuseur (5a) avec la direction radiale dans le plan méridien sur un point le plus près d'une sortie (2b) de la roue à aubes (2) dans la ligne côté moyeu d'entrée de diffuseur (5a), la ligne côté moyeu d'entrée de diffuseur (5a) étant une ligne sur le côté de moyeu (3) dans une entrée du diffuseur (5), et
α est un angle formé par une ligne tangente (3b) d'une ligne côté moyeu de roue à aubes (3a) avec la direction radiale dans le plan méridien sur un point le plus près de l'entrée du diffuseur (5) dans la ligne côté moyeu de roue à aubes (3a), la ligne côté moyeu de roue à aubes (3a) étant une ligne sur le côté de moyeu (3) dans la roue à aubes, et **caractérisé en ce que** l'angle θ est tel que 20°<θ<30° lorsque la ligne côté moyeu d'entrée de diffuseur (5a) est une ligne droite, β est tel que 0°≤β≤35° où β est un angle formé par une arête arrière (2c) de la roue à aubes (2) avec une direction axiale.

Fig.1

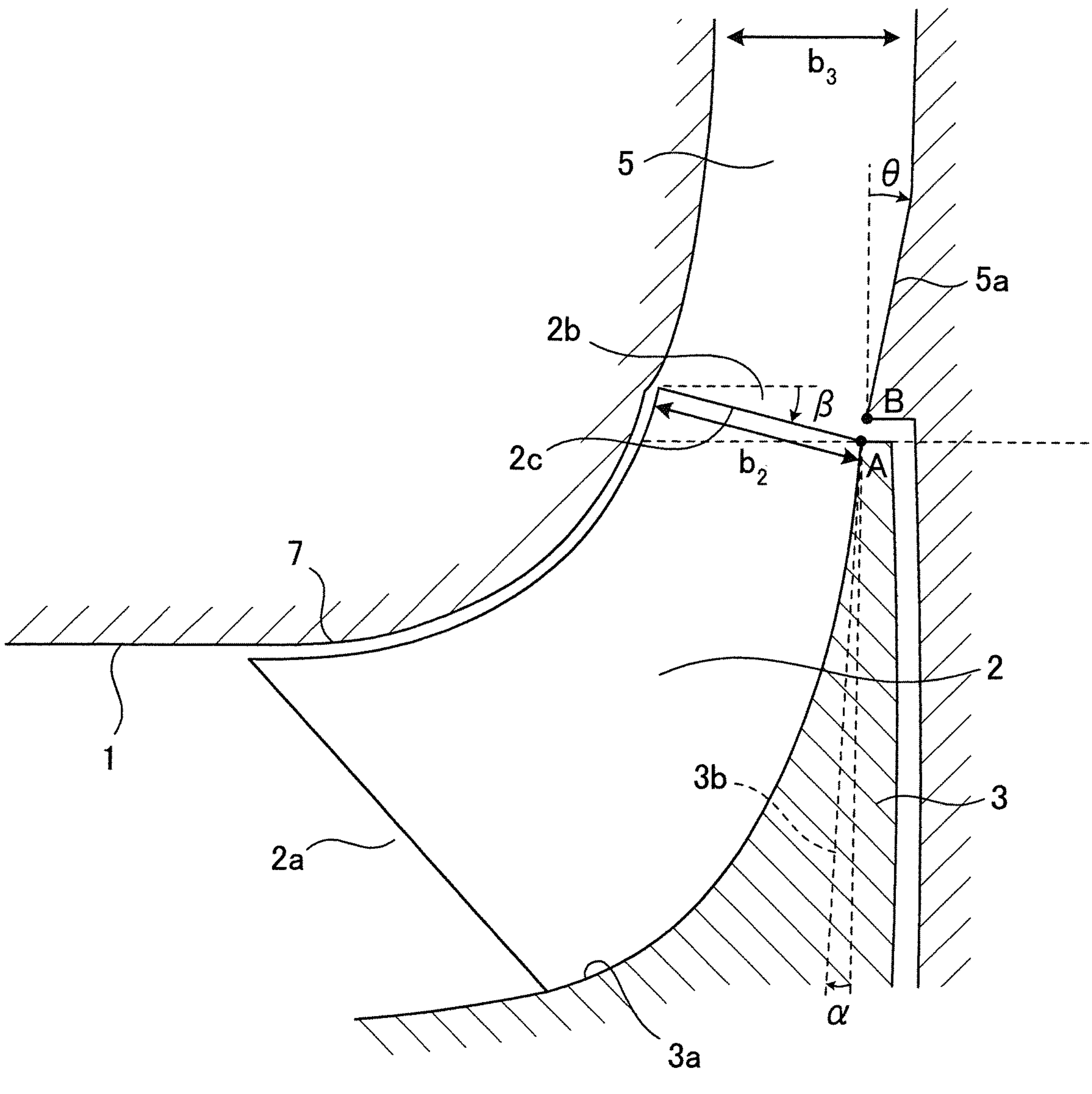


Fig.2

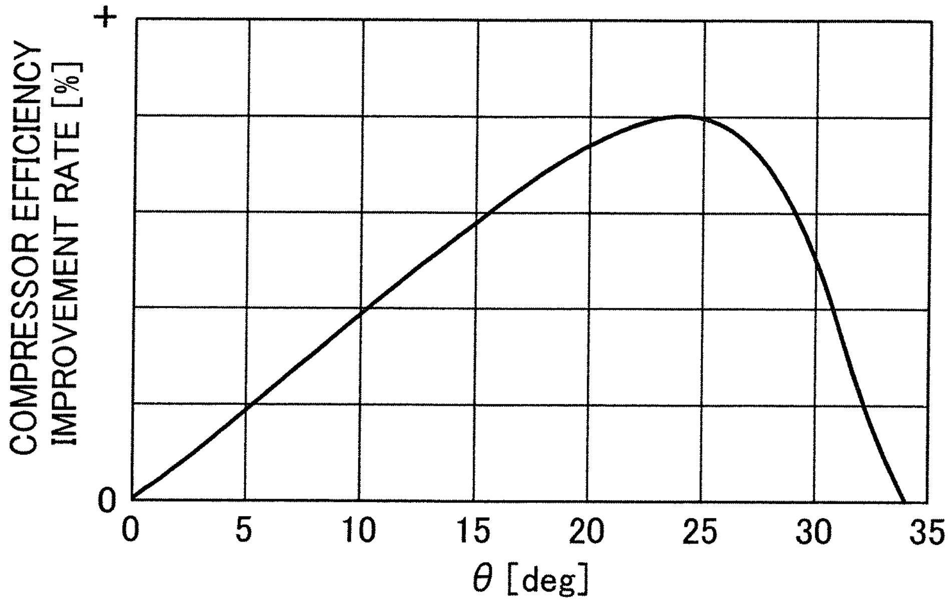

+
0
COMPRESSOR EFFICIENCY IMPROVEMENT RATE [%]
0
5
10
15
20
25
30
35
θ [deg]

Fig. 3

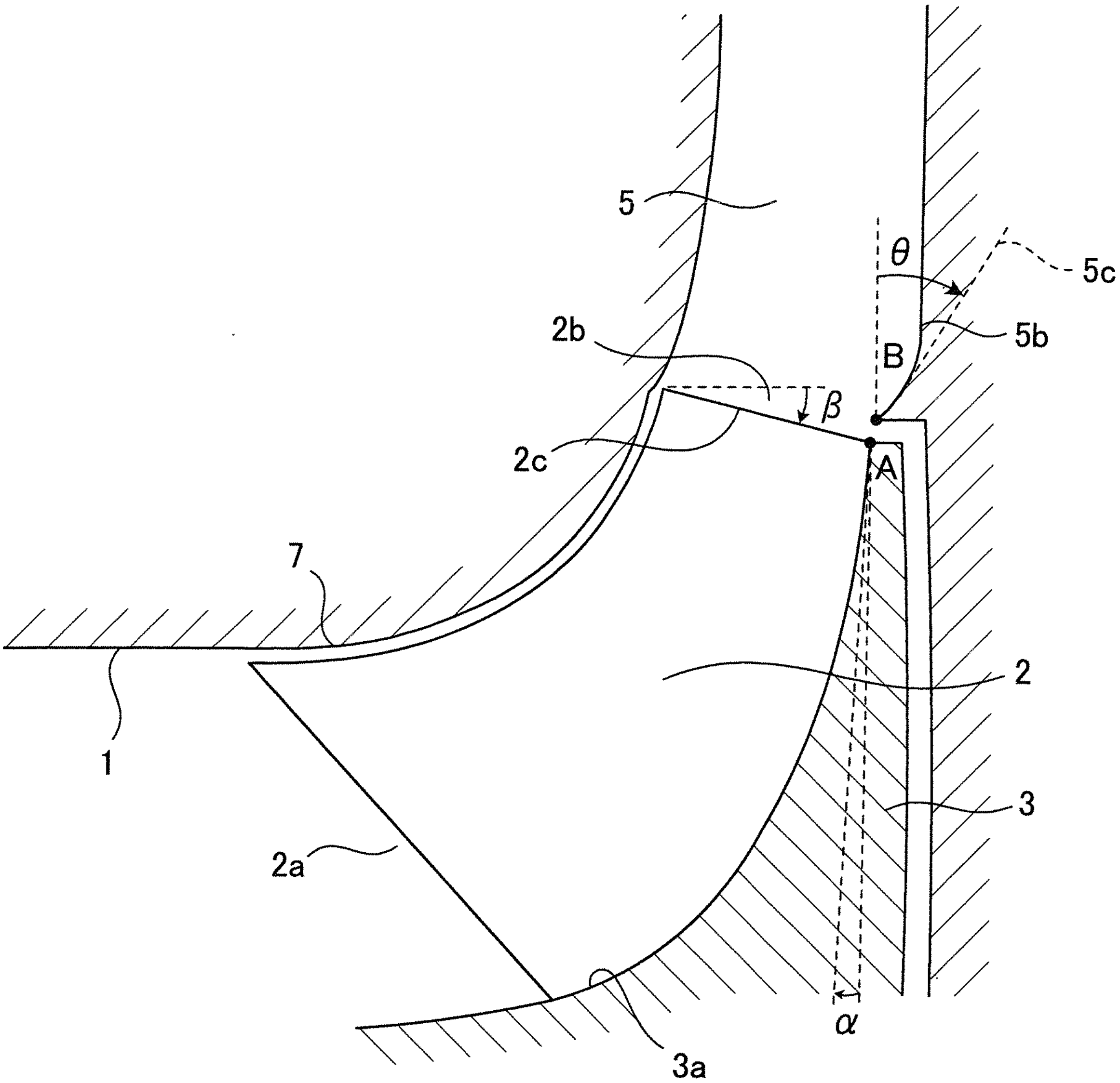


Fig.4

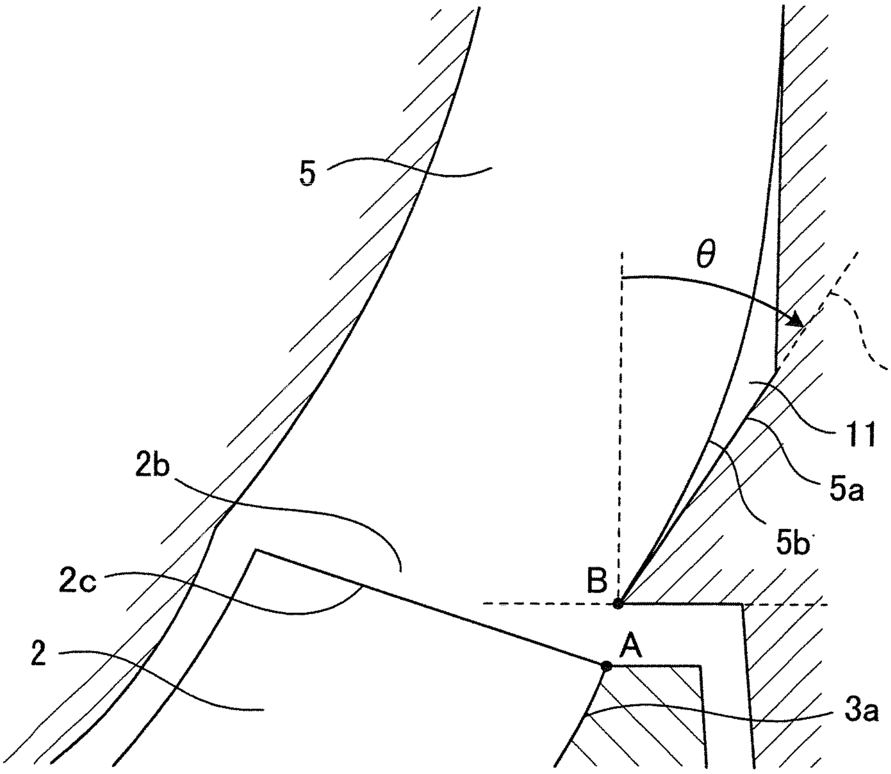


Fig.5

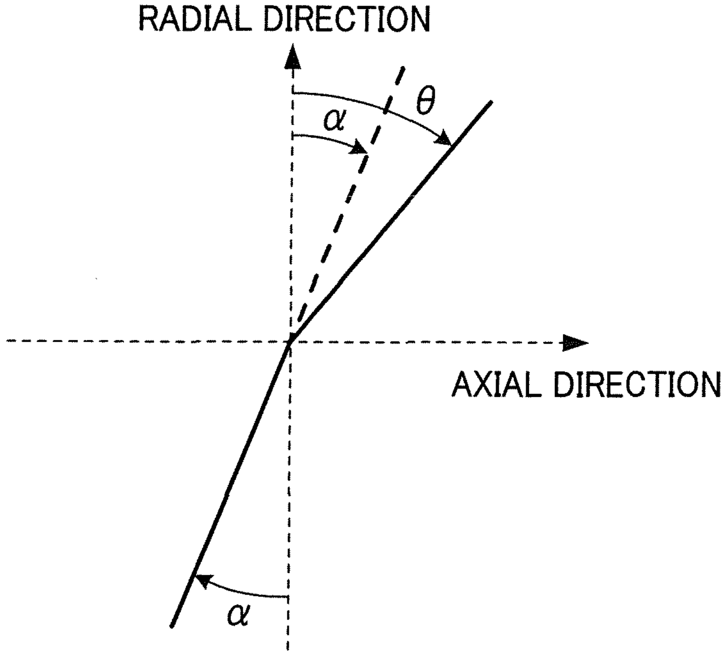


Fig.6

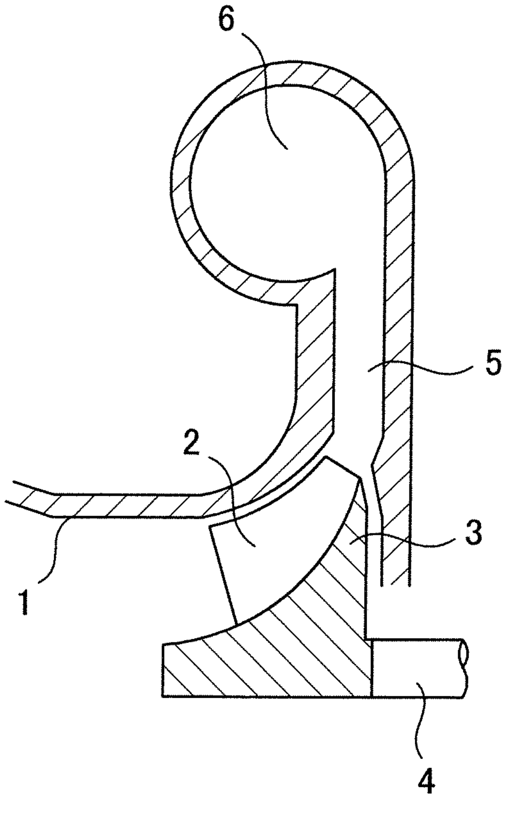
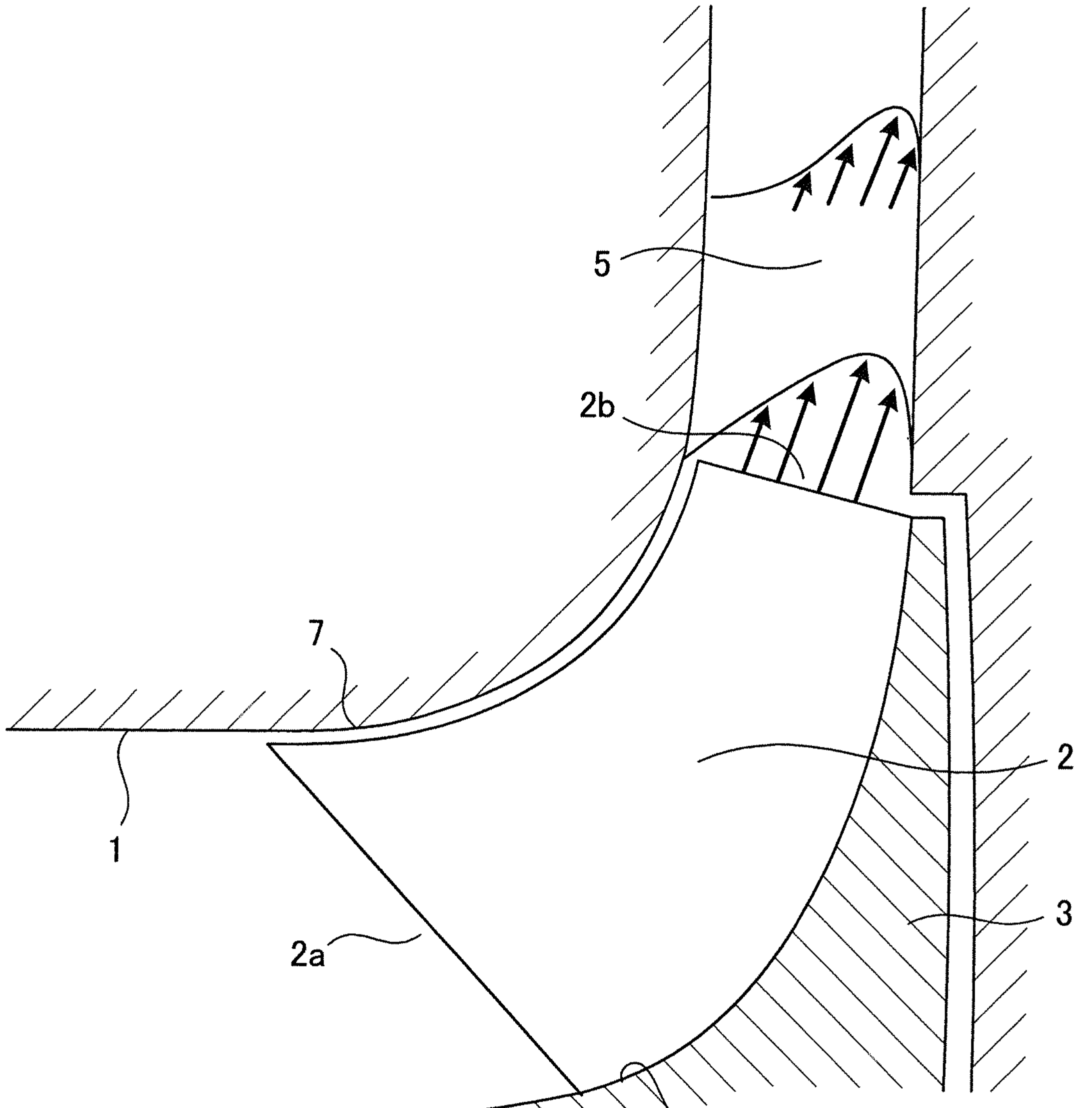


Fig. 7



REFERENCES CITED IN THE DESCRIPTION

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