(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 2 789 861 A1

(12) **EUROPEAN PATENT APPLICATION**
published in accordance with Art. 153(4) EPC

(43) Date of publication:
15.10.2014 Bulletin 2014/42

(21) Application number: **12856231.1**

(22) Date of filing: **08.06.2012**

(51) Int Cl.:
F04D 29/44 *(2006.01)*

(86) International application number:
PCT/JP2012/064796

(87) International publication number:
WO 2013/084525 (13.06.2013 Gazette 2013/24)

(84) Designated Contracting States:
AL AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HR HU IE IS IT LI LT LU LV MC MK MT NL NO PL PT RO RS SE SI SK SM TR

(30) Priority: **08.12.2011 JP 2011268925**

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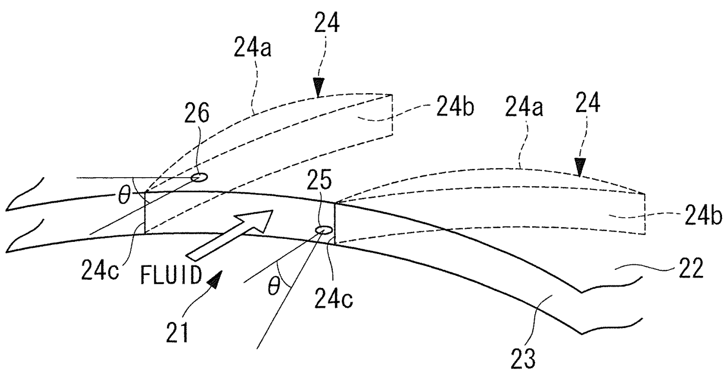
(54) CENTRIFUGAL FLUID MACHINE

(57) Provided is a centrifugal fluid machine in which flow separation due to the inconsistency in angles between a flow discharged from an impeller and a flow at vane inlet is suppressed and in which the performance of a bladed diffuser is enhanced. The present invention is a centrifugal compressor that is provided with a plurality of blades and an impeller having the plurality of blades; and a diffuser (21) that is provided on a downstream of the impeller and that has a plurality of vanes (24) between a shroud-side wall surface (22) and a hub-side wall surface (23) that face each other, and that is configured so that a fluid that is pressurized by means of rotation of the impeller flows out by passing through the diffuser (21), and the centrifugal compressor is provided with one or plurality of working fluid injection ports (25) that form openings at the hub-side wall surface (23) and that inject the working fluid in a main flow direction along pressure surfaces (24a) of the vanes (24), wherein the working fluid injection ports (25) are provided at radial positions closer to leading edges of the vanes (24) than to a throat radius (R2).

FIG. 1A



Printed by Jouve, 75001 PARIS (FR)

Description

{Technical Field}

[0001] The present invention relates to a centrifugal fluid machine equipped with a bladed diffuser, that is usualy used for a marine, a automotive, and a gas turbine engine for an aircraft, and so forth.

{Background Art}

[0002] A centrifugal fluid machine employed in a centrifugal compressor of a supercharger or the like is configured such that, by rotating an impeller in a casing, a fluid (gas) to be pressurized flows while passing through most of the flow channels in the impeller in the radial direction, so that the fluid is pressurized mainly by means of centrifugal force. To achieve a high pressure ratio, high efficiency, and a wide operational range are required for such a centrifugal fluid machine, a bladed diffuser, that is to say, provided with a plurality of vanes (diffuser blades), is provided downstream of the impeller, and it is essential to increase the performance of this bladed diffuser.

[0003] In the centrifugal compressor of Patent Literature 1, in order to prevent the flow separation from the vanes, communicating pathways that communicate between suction surfaces of the vanes and pressure surfaces thereof are formed.

[0004] The centrifugal fluid machine disclosed in Patent Literature 2 has a configuration in which radially-outer-side surfaces of vanes and radially-inner-side surfaces thereof communicate by means of a groove provided in at least one of the wall surfaces, namely, a shroud-side wall surface or a hub-side wall surface, or by means of communicating holes provided at tips of the vanes located close to a wall surface. With such a configuration, because the thickness of a boundary layer formed at the radially-inner-side surfaces of the vanes is reduced, the stall limit of a diffuser can be increased toward a lower flow ratio, which enables stable operation with a wider range.

{Citation List}

{Patent Literature}

[0005]

{PTL 1} Japanese Unexamined Patent Application, Publication No. Hei 10-331794
{PTL 2} Japanese Unexamined Patent Application, Publication No. 2005-155565

{Summary of Invention}

{Technical Problem}

[0006] Pressurized fluid discharged from an impeller flows into the vanes of a centrifugal fluid machine is distorted. Because of this, improvements that take account of the flow at the impeller exit are necessary to enhance the performance of the vanes.

[0007] To specifically describe the distorted flow, a pressurized fluid having a velocity distribution in the blade-height direction flows into the vanes of the diffuser. Because of this, an inconsistency occurs between the flow angle and the blade inlet angle at the vane inlet. Because such an inconsistency in the angles causes the flow of the fluid pressurized at the leading edges of the vanes to separate therefrom, there is a problem, which must be solved, that this separation causes a reduction in the efficiency of the bladed diffuser and stalling of the vanes.

[0008] The present invention has been conceived in light of the above-described circumstances, and an object thereof is to provide a centrifugal fluid machine in which flow separation due to the inconsistency in angles between a flow discharged from an impeller and a flow at the vane inlet is suppressed and in which the performance of a bladed diffuser is enhanced.

{Solution to Problem}

[0009] In order to solve the above-described problems, the present invention employs the following solutions.

[0010] A centrifugal fluid machine according to a first aspect of the present invention includes a plurality of blades and an impeller having the plurality of blades; and a diffuser that is provided on a downstream of the impeller and that has a plurality of vanes between a shroud-side wall surface and a hub-side wall surface that face each other, and is configured so that a fluid pressurized by means of rotation of the impeller flows out from the diffuser by passing therethrough. The centrifugal fluid machine is provided with one or a plurality of working fluid injection ports that form openings at the hub-side wall surface and that inject the working fluid in a fluid flow direction along pressure surfaces of the vanes, wherein the working fluid injection ports are provided at radial positions closer to leading edges of the vanes than to a throat radius (R2).

[0011] Because such a centrifugal fluid machine is provided with one or a plurality of working fluid injection ports that form openings at the hub-side wall surface and inject the working fluid in the main flow direction along the pressure surfaces of the vanes, and because these working fluid injection ports are provided at radial positions closer to the leading edges of the vanes than to the throat radius (R2), the working fluid that is injected toward the hub side of the pressure surfaces of the vanes joins a distorted flow that is discharged from the impeller and suppresses flow separation caused by the inconsistency in angles between a flow angle and a blade inlet angle.

[0012] Specifically, because a flow that tends to separate from the pressure surfaces at the leading-edge portions of the vanes is subjected to the influence of the working fluid that is injected from the working fluid injection ports in the main flow direction along the pressure surfaces, the flow is less likely to move away from the pressure surfaces. In this case, in order to suppress flow separation at the pressure surface side, it is desirable that the working fluid injection ports be formed as openings in regions on the pressure surface side that occupy 20 % or less of the throat width, and that the working fluid be injected therefrom.

[0013] A centrifugal fluid machine according to a second aspect of the present invention includes a plurality of blades; an impeller having the plurality of blades; and a diffuser that is provided on a downstream of the impeller and that has a plurality of vanes between a shroud-side wall surface and a hub-side wall surface that face each other, and is configured so that a fluid pressurized by means of rotation of the impeller flows out from the diffuser by passing therethrough. The centrifugal fluid machine is provided with one or a plurality of working fluid injection ports that form openings at the shroud-side wall surface and that inject the working fluid in a main flow direction along suction surfaces of the vanes, wherein the working fluid injection ports are provided at radial positions closer to leading edges of the vanes than to a throat radius (R2).

[0014] Because such a centrifugal fluid machine is provided with one or a plurality of working fluid injection ports that form openings at the shroud-side wall surface and that inject the working fluid in the fluid flow direction along the suction surfaces of the vanes, and because these working fluid injection ports are provided at radial positions closer to the leading edges of the vanes than to the throat radius (R2), the working fluid that is injected toward the shroud side of the suction surfaces of the vanes joins a distorted flow that is discharged from the impeller and suppresses flow separation caused by the inconsistency in angles between a flow angle and a blade inlet angle.

[0015] Specifically, because a flow that tends to separate from the suction surfaces at the leading-edge portions of the vanes is subjected to the influence of the working fluid that is injected from the working fluid injection ports in the main flow direction along the suction surfaces, the flow is less likely to move away from the suction surfaces. In this case, to suppress flow separation at the suction surfaces, it is desirable that the working fluid injection ports be formed as openings in regions on the suction surface side that occupy 20 % or less of the throat width, and that the working fluid be injected therefrom.

[0016] In the above-described centrifugal fluid machine, it is preferable that the radial positions at which the working fluid injection ports are provided have an inner circumferential limit at a 95 % minimum radius (Ro) with reference to a leading edge radius (R1) of the vanes, and an outer circumferential limit at the throat radius (R2); by doing so, the working fluid is injected into the region near the leading edges of the vanes, and thus, it is possible to efficiently suppress flow separation caused by the inconsistency in angles between a flow discharged from the impeller and a flow at vane inlet by using a minimum injection amount of the working fluid.

[0017] In the above-described centrifugal fluid machine, it is preferable that a working fluid injection angle (θ) of the working fluid injection ports from a wall surface be set so as to be 60° or less with respect to the fluid flow direction; by doing so, the injected working fluid efficiently forms a flow from the leading edges of the vanes toward the hub side of the pressure surfaces or the shroud side of the suction surfaces, thus enhancing the efficiency of flow separation suppression by the injected working fluid.

[0018] In the above-described centrifugal fluid machine, it is preferable that a fluid in which a portion thereof is returned from the downstream of the diffuser or a fluid that is introduced from an external pressure source be used as the working fluid.

{Advantageous Effects of Invention}

[0019] The present invention described above affords a notable advantage in that a centrifugal fluid machine is provided, in which flow separation due to the inconsistency in angles between the flow discharged from an impeller and the flow at vane inlet is suppressed and in which the performance of a bladed diffuser is enhanced.

[0020] In particular, because the working fluid injected toward the hub side of the pressure surfaces of the vanes is effective for enhancing the performance, and because the working fluid injected toward the shroud side of the suction surfaces of the vanes is effective for increasing the operation range, by injecting the working fluid toward both the hub side of the pressure surfaces and the shroud side of the suction surfaces, it is possible to realize a centrifugal fluid machine having a high pressure ratio, high efficiency, and a wide operation range.

{Brief Description of Drawings}

[0021]

{Fig. 1A} Fig. 1A is a diagram showing an embodiment of a centrifugal fluid machine according to the present invention and is a perspective view showing an example configuration of a relevant portion of a bladed diffuser.

{Fig. 1B} Fig. 1B is a diagram showing the embodiment of the centrifugal fluid machine according to the present invention and is a cross-sectional view of the bladed diffuser in Fig. 1A viewed from a shroud side.

{Fig. 1C} Fig. 1C is a diagram showing the embodiment of the centrifugal fluid machine according to the present invention and is a cross-sectional view taken along A-A in Fig. 1B.

{Fig. 2} Fig. 2 is a longitudinal cross-sectional view of an impeller and a bladed diffuser, showing an outline of the centrifugal fluid machine.

{Fig. 3} Fig. 3 is a diagram of the centrifugal fluid machine in Fig. 2 viewed from the upstream of the impeller.

{Description of Embodiment}

[0022] An embodiment of a centrifugal fluid machine according to the present invention will be described below (a centrifugal compressor will be described below as an example) based on the drawings.

[0023] As shown in Figs. 2 and 3, a centrifugal compressor 10 is configured having an impeller 11 and a diffuser 21 as main components and discharges a fluid introduced thereinto after pressurizing the fluid. The impeller 11 has a plurality of blades 12 and a hub 13 disposed at base portions R of these blades 12, and the individual blades 12 are disposed at a surface of the hub 13 so that leading edges LE thereof are positioned at a small-diameter-side end portion 13a of the hub 13 and so that trailing edges TE thereof are positioned at a large-diameter-side end portion 13b of the hub 13.

[0024] In the drawings, reference sign 14 indicates a shroud that is disposed so as to cover tips of the blades 12.

[0025] The diffuser 21 is provided on the downstream of the impeller 11 described above, has a plurality of vanes (diffuser blades) 24 between a shroud-side wall surface 22 and a hub-side wall surface 23 that face each other, and has a function for converting the kinetic energy possessed by the fluid (gas), which is pressurized by passing through the impeller 11, into pressure energy. Specifically, at the diffuser 21, the dynamic pressure of a flow is converted to an increase in the static pressure by decelerating the flow speed of the fluid.

[0026] The flow whose static pressure is increased in this way is guided to an outlet of the centrifugal compressor 10 by a spiral volute 31.

[0027] In other words, the centrifugal compressor 10 is provided with the impeller 11, which has the plurality of blades 12 and the hub 13 disposed at the base portions R of the plurality of blades 12, and the diffuser 21, which is provided on the downstream of the impeller 11 and which has the plurality of vanes 24 between the shroud-side wall surface 22 and the hub-side wall surface 23 that face each other, and is configured so that the fluid pressurized by means of rotation of the impeller 11 flows out from the diffuser 21 by passing therethrough.

[0028] Fig. 1A is a perspective view in which a portion of the diffuser 21 is enlarged, and is a diagram viewed from the trailing edge TE side of the blades 12, which is the exit side for the fluid that has passed through the impeller 11, that is to say, as viewed from the inlet side of the diffuser 21.

[0029] In a first aspect of this embodiment shown in Figs. 1A to 1C, the centrifugal compressor 10 described above is provided with one or a plurality of working fluid injection ports 25 that form openings at the hub-side wall surface 23 and that inject the working fluid (arrow Fa in the figures) in the fluid flow direction along the pressure surfaces 24a of the vanes 24. Also, these working fluid injection ports 25 are provided at radial positions closer to the leading edges 24c of the vanes 24 than to the throat radius (R2).

[0030] As the above-described working fluid injection ports 25, one or a plurality of holes are provided along the pressure surfaces 24a within an area from a radius R1, which corresponds to the leading edges 24c of the vanes 24, that is to say, a 95 % radial position Ro (95 % R1) determined with reference to the radius R1 of a circle connecting the positions of the leading edges 24c, to a radial position R2 where the throats 27 of the vanes 24 and the blade suction surfaces 24b intersect. In other words, the radial positions (in a region in the radial direction) at which the working fluid injection ports 25 are provided are in an area whose inner circumferential limit is set at the 95 % minimum radius Ro determined with reference to the leading edge radius R1 of the vanes 24 and whose outer circumferential limit is set at the throat radius R2.

[0031] The holes that serve as the working fluid injection ports 25 are set so that, for example, as shown in Fig. 1C, a working fluid injection angle (θ) is 60° or less (θ < 60°) from the hub-side wall surface 23 with respect to the fluid flow direction. Although the working fluid injection ports 25 generally have a circular cross-section, they are not limited thereto; naturally, the cross-sectional shape thereof may be elliptical, rectangular, and so forth, and the working fluid injection ports 25 may be thin, long slits formed along the pressure surfaces 24a.

[0032] For example, as shown in Fig. 2, a fluid in which a portion thereof is returned from the downstream of the diffuser 21 by passing through the extracted-fluid flow channel 28 or a fluid that is introduced from an external pressure source 40 may be used as the working fluid to be injected from the working fluid injection ports 25.

[0033] With the centrifugal compressor 10 having such working fluid injection ports 25, because the openings are formed at the hub-side wall surface 23 and the working fluid Fa is injected toward the pressure surfaces 24a of the vanes 24, the working fluid injected on the hub side of the pressure surfaces of the vanes 24 joins a distorted flow of the pressurized fluid that is discharged from the impeller 11 and suppresses flow separation caused by the inconsistency in the angles between the flow angle and the blade inlet angle.

[0034] Specifically, at the leading-edge portions of the vanes 24, because a flow of the fluid that tends to separate from the pressure surfaces 24a in the vicinity of the leading edges 24c is influenced by the working fluid Fa that is injected from the working fluid injection ports 25 in the main flow direction along the pressure surfaces

24a, the flow is less likely to move away from the pressure surfaces 24a. In other words, in the flow of the fluid that tends to separate from the pressure surfaces 24a in the vicinity of the leading-edge portions, a flow in the separating direction is suppressed by the flow of the working fluid Fa injected from the working fluid injection ports 25 in the main flow direction along the pressure surfaces 24a; therefore, the flow is less likely to move away from the pressure surfaces 24a.

[0035] By injecting the working fluid from the injection ports 25 toward the hub side of the pressure surfaces of the vanes 24 in this way, flow separation due to the inconsistency between the flow discharged from the impeller 11 and the blade inlet angle of the vanes 24 is suppressed, and, as a result, the performance of the diffuser 21 and the centrifugal compressor 10 is enhanced.

[0036] In this case, in order to efficiently suppress the flow separation of the fluid from the pressure surfaces 24a, it is desirable that, in the width direction of the throats 27, the working fluid injection ports 25 be provided so as to form openings in regions on the pressure surface 24a side that occupy 10% or less of the throat width. Specifically, it is desirable that, in the width direction of the throats 27, the working fluid be injected from positions close to the pressure surfaces 24a where flow separation is to be suppressed.

[0037] A injected flow ratio of the working fluid required for such separation suppression is about 2 to 5 % of the flow ratio of the fluid to be compressed; at 5 % or greater, there is almost no change in the separation prevention effect, and, at 2 % or less, it is not possible to achieve a satisfactory separation prevention effect.

[0038] Regarding the radial positions at which the working fluid injecttion ports 25 are provided, the inner circumferential limit is set at the minimum radius (Ro) in order to realize the separation prevention effect by the working fluid from the injection ports, and the outer circumferential limit is set at the throat radius (R2) because a further increase achieves almost no change in the separation prevention effect.

[0039] Next, with a second aspect of this embodiment shown in Figs. 1A to 1C, the centrifugal compressor 10 described above is provided with one or a plurality of working fluid injection ports 26 that form openings at the shroud-side wall surface 22 and that inject the working fluid (arrow Fa in the figures) in the fluid flow direction along the suction surfaces 24b of the vanes 24.

[0040] Also, as with the working fluid injection ports 25 described above, these working fluid injection ports 26 are provided at radial positions closer to the leading edges 24c of the vanes 24 than to the throat radius (R2). In other words, the working fluid injection ports 26 are positioned at the shroud-side wall surfaces 22, differing from the first aspect described above; however, with regard to the basic items such as the installation positions (regions) of the working fluid injection ports 26, the working fluid, and so forth, the hub-side wall surface 23 can be taken as the shroud-side wall surface 22, and the pressure surfaces 24a can be taken as the suction surfaces 24b.

[0041] Because such a centrifugal compressor 10 is provided with one or a plurality of working fluid injection ports 26 that form openings at the shroud-side wall surface 22 and that inject the working fluid in the fluid flow direction along the suction surfaces 24b of the vanes 24, and because these working fluid injection ports 26 are provided at the radial positions closer to the leading edges 24c of the vanes 24 than to the throat radius (R2), the working fluid injected toward the shroud side of the suction surfaces of the vanes 24 joins the distorted flow discharged from the impeller 11 and suppresses flow separation caused by the inconsistency in the angles between the flow angle and the blade inlet angle. As a result, it is possible to increase the operational range of the diffuser 21 and the centrifugal compressor 10, specifically, the operational flow ratio range from a low flow ratio (lower-limit value), determined by a flow ratio value at which surging occurs, to a high flow ratio (upper-limit value), determined by a flow ratio value at which choking occurs.

[0042] Specifically, because a flow that tends to separate from the suction surfaces 24b at the leading-edge 24c of the vanes 24 is affected by the working fluid that is injected from the injection ports 26 in the main flow direction along the suction surfaces 24b, the flow is less likely to move away from the suction surfaces 24b. In this case, in order to suppress flow separation at the suction surfaces 24b, it is desirable that, as with the working fluid injection ports 25, the working fluid injection ports 26 be formed as openings in regions on the suction surface 24b side that occupy 10% or less of the throat width, and that the working fluid be injected therefrom.

[0043] The ratio of the injection flow to the main flow ratio to be compressed, the reason for setting the inner circumferential limit at the minimum radius (Ro), the reason for setting the outer circumferential limit at the throat radius (R2), the reason for setting the working fluid injection angle (θ) equal to 60° or less, the supply source of the working fluid, and so forth are the same as those for the above-described working fluid injection ports 25 at the hub-side wall surface 23.

[0044] Finally, with a third aspect of this embodiment shown in Figs. 1A to 1C, the above-described centrifugal compressor 10 is provided with one or a plurality of working fluid injection ports 25 that form openings at the hub-side wall surface 23 and that injects the working fluid in the main flow direction along the pressure surfaces 24a of the vanes 24 and one or a plurality of working fluid injection ports 26 that form openings at the shroud-side wall surface 22 and that injects the working fluid in the main flow direction along the suction surfaces 24b of the vanes 24. In other words, this configuration includes both the working fluid injection ports 25 and 26 of the first aspect and the second aspect, described above.

[0045] Because it is possible to achieve the operational advantages of the first aspect and the second aspect described above by employing such a configuration, it is possible to enhance the performance of the diffuser 21 and the centrifugal compressor 10 and to increase the operational range thereof.

[0046] As described above, with this embodiment described above, it is possible to provide a centrifugal fluid machine, such as the centrifugal compressor 10 or the like, in which flow separation due to the inconsistency in angles between the flow discharged from the impeller 11 and the flow at the vane inlet is suppressed and in which the performance of the diffuser 21 provided with the vanes 24 is enhanced.

[0047] In particular, because the working fluid injected from the working fluid injection ports 25 toward the hub side of the pressure surfaces of the vanes 24 is effective for enhancing the performance, and because the working fluid injected from the working fluid injection ports 26 toward the shroud side of the suction surfaces of the vanes 24 is effective for increasing the operational range, by injecting the working fluid toward both the hub side of the pressure surfaces and the shroud side of the suction surfaces, it is possible to realize a centrifugal fluid machine having a high pressure ratio, high efficiency, and a wide operational range.

[0048] The present invention is not limited to the embodiment described above, and, for example, the centrifugal fluid machine is not limited to a centrifugal fluid machine (turbocharger for marine use, an automobile supercharger, an aircraft gas turbine, and so forth) and also encompasses a centrifugal pump and a centrifugal blower, thus allowing appropriate alterations within a range that does not depart from the scope of the present invention.

{Reference Signs List}

[0049]

10 centrifugal compressor (centrifugal fluid machine)
11 impeller
12 blade
13 hub
14 shroud
21 diffuser
22 shroud-side wall surface
23 hub-side wall surface
24 vane (diffuser blade)
24a pressure surface
24b suction surface
24c leading edge
25, 26 working fluid injection port
27 throat
28 extracted-fluid flow channel
40 pressure source

Claims

1. A centrifugal fluid machine that is provided with a plurality of blades and an impeller having the plurality of blades; and
a diffuser that is provided on a downstream of the impeller and that has a plurality of vanes between a shroud-side wall surface and a hub-side wall surface that face each other,
and that is configured so that a fluid pressurized by means of rotation of the impeller flows out from the diffuser by passing therethrough,
the centrifugal fluid machine comprising:

one or a plurality of working fluid injection ports that form openings at the hub-side wall surface and that inject the working fluid in a main flow direction along pressure surfaces of the vanes,
wherein the working fluid injection ports are provided at radial positions closer to leading edges of the vanes than to a throat radius (R2).

2. A centrifugal fluid machine that is provided with a plurality of blades; an impeller having the plurality of blades; and
a diffuser that is provided on a downstream of the impeller and that has a plurality of vanes between a shroud-side wall surface and a hub-side wall surface that face each other,
and that is configured so that a fluid pressurized by means of rotation of the impeller flows out from the diffuser by passing therethrough,
the centrifugal fluid machine comprising:

one or a plurality of working fluid injection ports that form openings at the shroud-side wall surface and that inject the working fluid in a main flow direction along suction surfaces of the vanes,
wherein the working fluid injection ports are provided at radial positions closer to leading edges of the vanes than to a throat radius (R2).

3. A centrifugal fluid machine according to Claim 1 or 2, wherein the radial positions at which the working fluid injection ports are provided have an inner circumferential limit at a 95 % minimum radius (Ro) with reference to a leading edge radius (R1) of the vanes, and an outer circumferential limit at the throat radius (R2).

4. A centrifugal fluid machine according to any one of Claims 1 to 3, wherein a working fluid injection angle (θ) of the working fluid injection ports from a wall surface is set so as to be 60° or less with respect to the main flow direction.

5. A centrifugal fluid machine according to any one of Claims 1 to 4, wherein a fluid in which a portion thereof is returned from the downstream of the diffuser or a fluid that is introduced from an external pressure source is used as the working fluid.

FIG. 1A

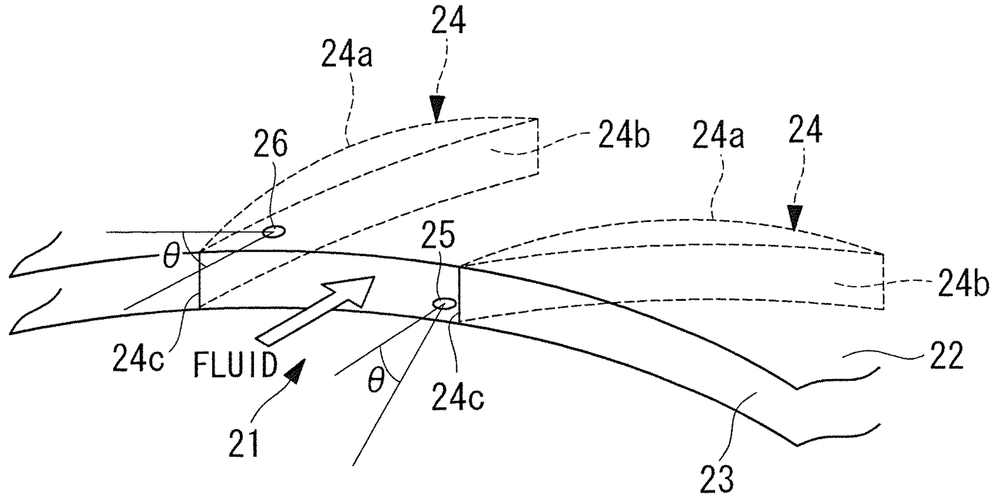
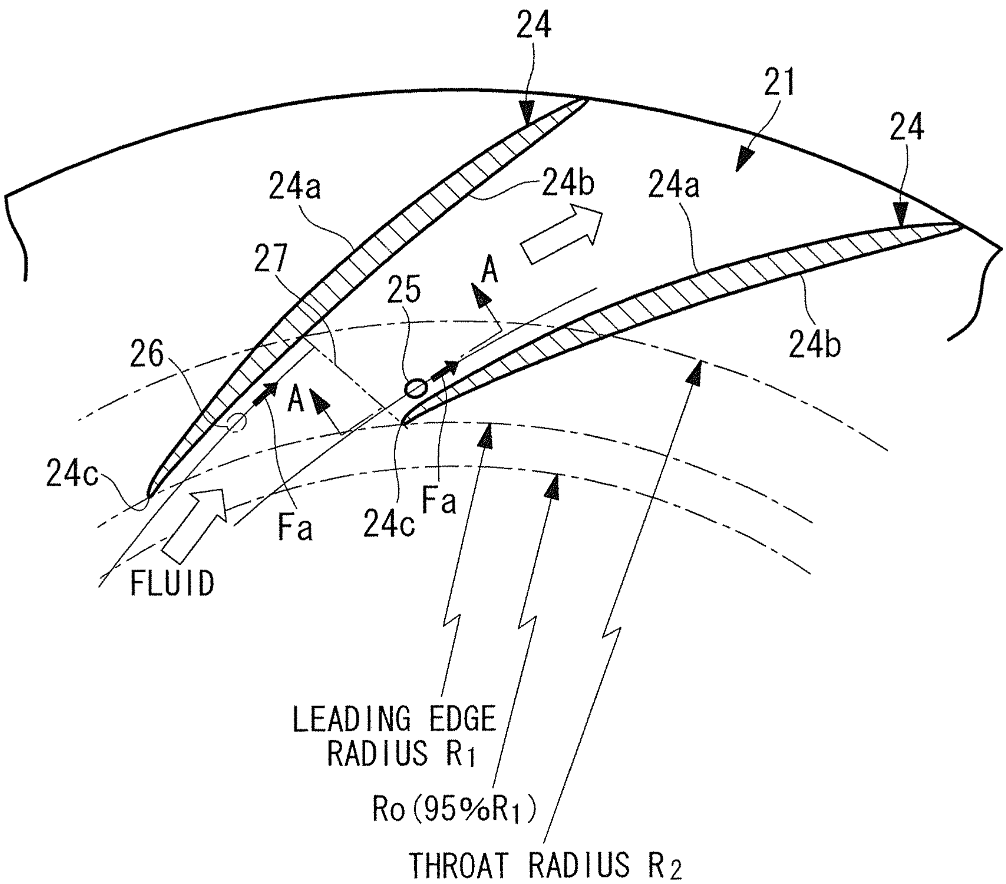

24
24a
26
24b
24a
24
25
θ
24b
24c
FLUID
θ
24c
22
21
23

FIG. 1B


24
21
24
24a
24b
24a
27
25
A
26
A
24b
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Fa
24c
Fa
FLUID
LEADING EDGE RADIUS R_1
$Ro(95\%R_1)$
THROAT RADIUS R_2

FIG. 1C

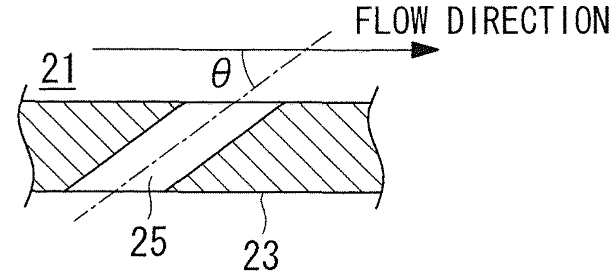

FLOW DIRECTION
21
θ
25
23

FIG. 2

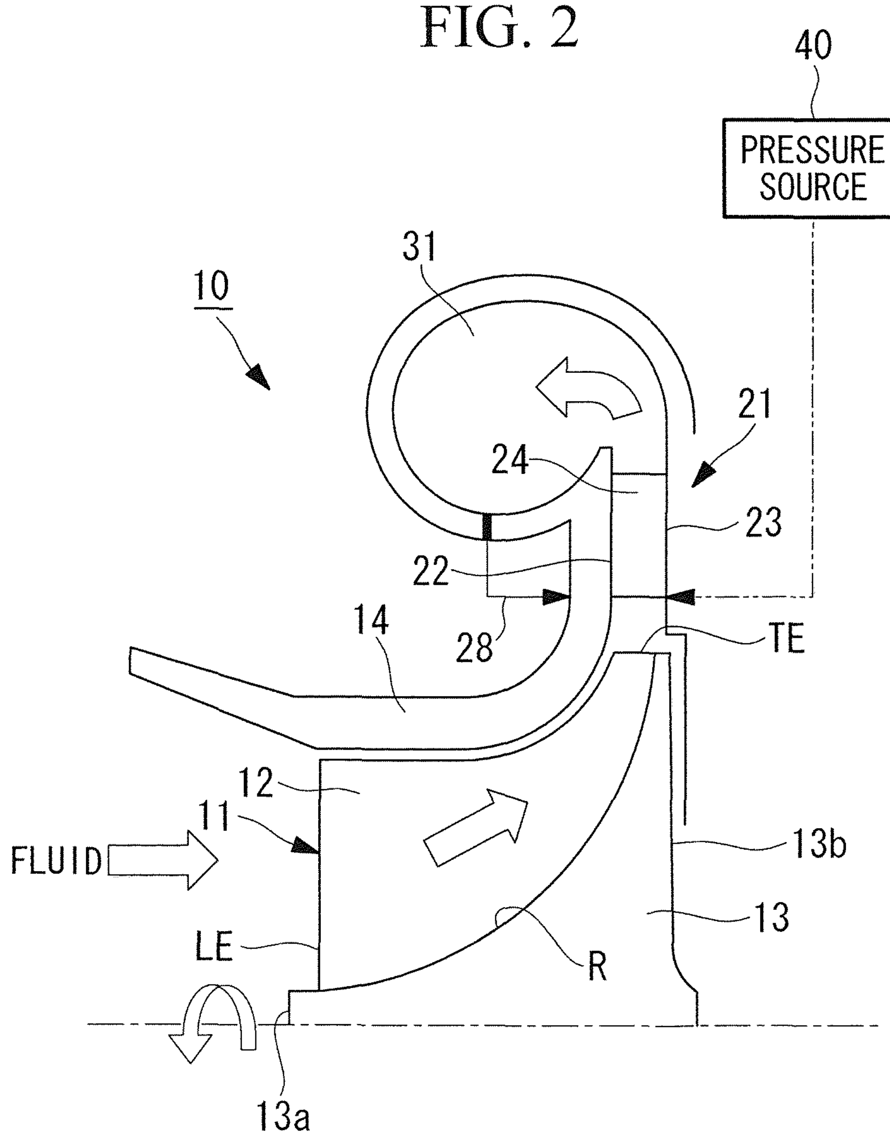
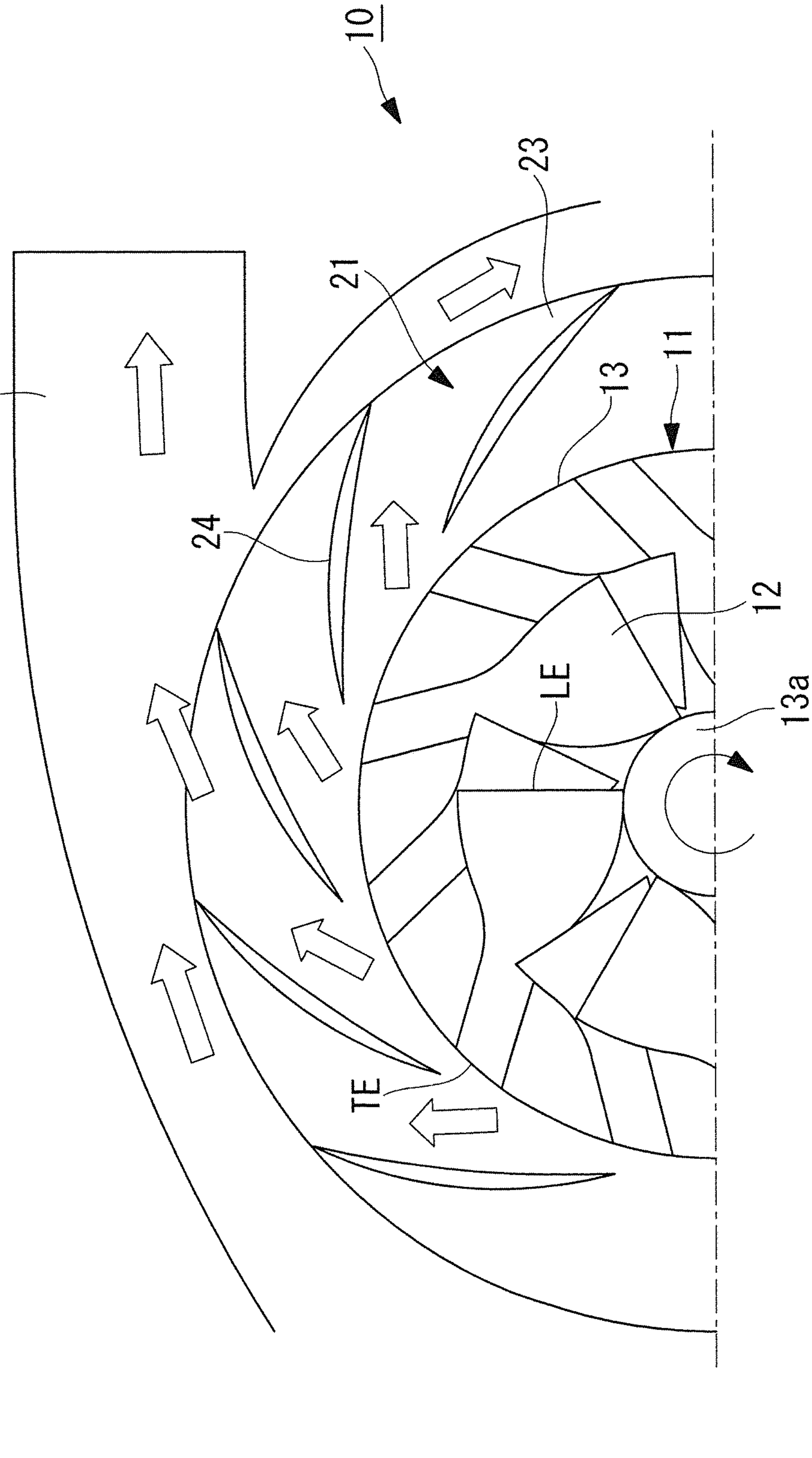

40
PRESSURE SOURCE
31
10
21
24
23
22
28
14
TE
12
11
FLUID
13b
13
LE
R
13a

FIG. 3



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2012/064796

A. CLASSIFICATION OF SUBJECT MATTER
F04D29/44(2006.01)i

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)
F04D29/44

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

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2012
Kokai Jitsuyo Shinan Koho	1971-2012	Toroku Jitsuyo Shinan Koho	1994-2012

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 2010-255609 A (Toyota Motor Corp.), 11 November 2010 (11.11.2010), entire text; all drawings (Family: none)	1-5
A	JP 2006-29200 A (Toshiba Corp.), 02 February 2006 (02.02.2006), entire text; all drawings (Family: none)	1-5

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:
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Date of the actual completion of the international search	Date of mailing of the international search report
04 September, 2012 (04.09.12)	11 September, 2012 (11.09.12)
Name and mailing address of the ISA/ Japanese Patent Office	Authorized officer
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