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

(11) **EP 3 369 938 A1**

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

(43) Date of publication:
05.09.2018 Bulletin 2018/36

(21) Application number: **16884910.7**

(22) Date of filing: **14.01.2016**

(51) Int Cl.:
F04D 29/30 *(2006.01)* ***F04D 29/66*** *(2006.01)*

(86) International application number:
PCT/JP2016/050923

(87) International publication number:
WO 2017/122307 (20.07.2017 Gazette 2017/29)

(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
Designated Extension States:
BA ME
Designated Validation States:
MA MD

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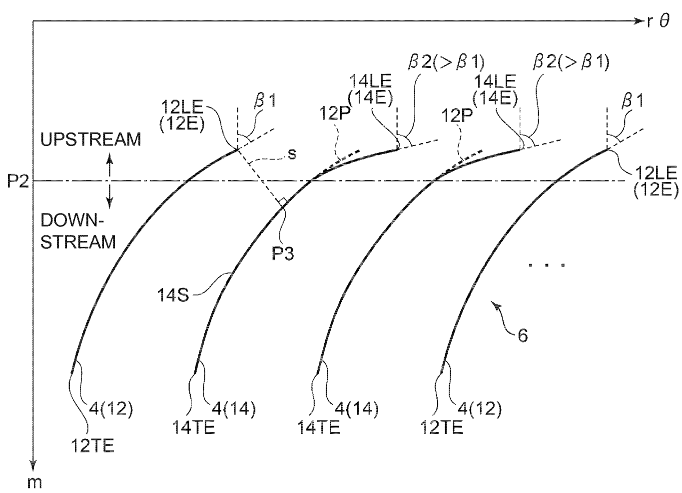
(54) COMPRESSOR IMPELLER AND METHOD FOR MANUFACTURING SAME

(57) A compressor impeller includes: a hub; and a blade group including a plurality of blades arranged along a circumferential direction on an outer peripheral surface of the hub, the blade group being configured such that hub-side ends of leading edges of the respective blades are aligned on the same circle. The plurality of blades include at least one first blade and at least one second blade having a different shape from the at least one first blade. When comparing a blade angle of a leading edge of the at least one first blade to a blade angle of a leading edge of the at least one second blade at the same position in a radial direction of the compressor impeller, the blade angle of the leading edge of the at least one first blade is different from the blade angle of the leading edge of the at least one second blade, at least in a partial range in the radial direction of the compressor impeller.

FIG. 4



EP 3 369 938 A1

Description

TECHNICAL FIELD

[0001] The present disclosure relates to a compressor impeller and a method for manufacturing the same.

BACKGROUND ART

[0002] Compressors such as a centrifugal compressor, an axial-flow compressor and an axial-flow compressor are configured to apply kinetic energy to a fluid through rotation of a compressor impeller, and convert the kinetic energy into pressure, thereby obtaining a high-pressure fluid.

[0003] Such a compressor is provided with various features to meet the need to improve the pressure ratio and the efficiency in a wide operational range.

[0004] Patent Document 1 discloses a centrifugal compressor for suppressing rotating stall. Rotating stall is an unstable phenomenon in which a stalling region generated on a blade propagates in the rotational direction from a blade to another blade at a speed lower than the tip speed of the impeller during operation in a low flow-rate range.

[0005] When rotating stall occurs, continuous reduction of the flow rate may lead to occurrence of surging of a flow accompanied by strong noise, where the compressor reaches its operation limit. Thus, to expand the operation range in which the compressor can operate stably, it is necessary to suppress rotating stall.

[0006] To suppress rotating stall, the compressor disclosed in Patent Document 1 includes a suppressing member for suppressing development of vortices of a fluid formed in the vicinity of a leading edge of a blade, disposed on the inner peripheral surface of the casing or the outer peripheral surface of the rotational shaft of the impeller at the upstream side of the blade leading edge of the impeller and configured to rotate relatively to the blade.

Citation List

Patent Literature

[0007] Patent Document 1: JP2014-118916A

SUMMARY

Problems to be Solved

[0008] To suppress rotating stall with the compressor disclosed in Patent Document 1, it is necessary to add a suppressing member configured to rotate relatively to the blade, on the inner peripheral surface of the casing or the outer peripheral surface of the impeller at the upstream side of the blade leading edge of the impeller. Thus, the number of necessary parts increases and the configuration of the compressor becomes complex.

[0009] The present invention was made in view of the above issue, and an object is to provide a compressor impeller and a method for manufacturing the same, whereby it is possible to suppress rotating stall with a simplified configuration.

Solution to the Problems

[0010]

(1) A compressor impeller according to at least one embodiment of the present invention includes: a hub; and a blade group including a plurality of blades arranged along a circumferential direction on an outer peripheral surface of the hub, the blade group being configured such that hub-side ends of leading edges of the respective blades are aligned on the same circle. The plurality of blades include at least one first blade and at least one second blade having a different shape from the at least one first blade, and, when comparing a blade angle of a leading edge of the at least one first blade to a blade angle of a leading edge of the at least one second blade at the same position in a radial direction of the compressor impeller, the blade angle of the leading edge of the at least one first blade is different from the blade angle of the leading edge of the at least one second blade, at least in a partial range in the radial direction of the compressor impeller.

With the compressor impeller described in the above (1), in the blade group configured such that the hub-side ends of the leading edges align on the same circle, when comparing the blade angle of the leading edge of the first blade to the blade angle of the leading edge of the second blade at the same position with respect to the radial direction of the compressor impeller, at least in a partial range in the radial direction of the compressor impeller, the blade angle of the leading edge of the first blade is different from the blade angle of the leading edge of the second blade. Accordingly, for the plurality of blades arranged so that the hub-side ends of the leading edges align on the same circle, it is possible to differentiate the stall characteristics between the first blade and the second blade. Thus, as compared to a case where the plurality of blades have a uniform shape, it is possible to impair uniform propagation and development of rotating stall. Accordingly, it is possible to improve the characteristics of the compressor at the low flow-rate side. Furthermore, it is unnecessary to provide an additional suppressing member disclosed in the prior art (JP2014-118916), and thus it is possible to simplify the configuration compared to the prior art.

(2) In some embodiments, in the above compressor impeller (1), the at least one first blade includes a plurality of first blades, the at least one second blade

includes a plurality of second blades, the number of the second blades in the blade group is smaller than the number of the first blades in the blade group, and the plurality of second blades include a pair of second blades between which the first blade is not disposed.

With the above compressor impeller (2), the second blades relatively fewer than the first blades and having different stalling characteristics from the first blades are aligned continuously in the circumferential direction of the compressor impeller, and thus it is possible to enhance the above effect to impair uniform propagation and development of rotating stall.

(3) In some embodiments, in the above compressor impeller (1) or (2), the number of the second blades in the blade group is smaller than the number of the first blades in the blade group, and, when comparing the blade angle of the leading edges of the first blades to the blade angle of the leading edges of the second blades at the same position in the radial direction of the compressor impeller, the blade angle of the leading edges of the second blades is greater than the blade angle of the leading edges of the first blades at least in a partial range in the radial direction of the compressor impeller.

With the above compressor impeller (3), the blade angle of the leading edges of the relatively large number of first blades is a relatively small blade angle taking into account the intake air amount of the high-flow rate side, while the blade angle of the leading edges of the relatively small number of second blades is a relatively large blade angle matched to the small flow rate side (less likely to cause stalling even at a low flow rate). Thus, it is possible to impair uniform propagation and development of rotating stall while suppressing a decrease in the intake air amount of the compressor impeller.

(4) In some embodiments, in the above compressor impeller (3), the blade angle of the tip-side ends of the leading edges of the second blades is greater than the blade angle of the tip-side ends of the leading edges of the first blades.

According to findings of the present inventors, rotating stall of the compressor impeller is likely to occur in a region on the tip side of a blade. Thus, as in the above (4), with the blade angle of the leading edges of the second blades on the tip side end being greater than the blade angle of the leading edges of the first blades on the tip side end as described above, it is possible to suppress rotating stall effectively.

(5) In some embodiments, in the above compressor impeller (4), the blade angle of the tip-side ends of the leading edges of the second blades is greater than the blade angle of the tip-side ends of the leading edges of the first blades by five degrees or more.

With the above compressor impeller (5), the effect in the above (4) can be exerted to a greater extent.

(6) In some embodiments, in the above compressor impeller (4) or (5), the blade angle of hub-side ends of the leading edges of the second blades is equal to the blade angle of hub-side ends of the leading edges of the first blades.

As described above, rotating stall of the compressor impeller is likely to occur in the tip-side region of a blade. Thus, even if the angle of the leading edge of the second blade at the hub-side end is greater than the blade angle of the leading edge of the first blade at the hub-side end, the effect to suppress rotating stall is relatively small. Furthermore, configuring the second blades to have a large blade angle matched to the small flow-rate side in a broad range in the radial direction of the compressor impeller may lead to a decrease in the intake air amount of the compressor impeller.

Thus, with the blade angle of the leading edges of the second blades on the tip side end being greater than the blade angle of the leading edges of the first blades on the tip side end, and the blade angle of the leading edges of the second blades on the hub-side end being equal to the blade angle of the leading edges of the first blades on the hub-side end as in the above (6), it is possible to impair uniform propagation and development of rotating stall while suppressing a decrease in the intake air amount of the compressor impeller.

(7) In some embodiments, in the above compressor impeller (6), when comparing the blade angle of the leading edges of the first blades to the blade angle of the leading edges of the second blades at the same position in the radial direction of the compressor impeller, the blade angle of the leading edges of the second blades is greater than the blade angle of the leading edges of the first blades in a range to the tip-side ends from a predetermined position of not less than 50% of a blade height of the second blades in the radial direction of the compressor impeller, and is equal to the blade angle of the leading edges of the first blades in a range to the predetermined position from the hub-side ends of the second blades in the radial direction of the compressor impeller.

With the above compressor impeller (7), the effect in the above (6) can be exerted to a greater extent.

(8) In some embodiments, in the compressor impeller according to any one of the above (1) to (7), the first blades and the second blades have different shapes only in an upstream region of a reference position in an axial direction of the compressor impeller, and have the same shape in a downstream region of the reference position in the axial direction of the compressor impeller.

The curve of a blade and the blade angle of the trailing edge of a blade have a great impact on the blade element performance, and thus the plurality of blades should have a uniform shape on the trailing edge side. Thus, in the above compressor impeller (8), the first blade and the second blade have different shapes only at the side of the leading edge where the shape is likely to contribute to improvement of rotating stall (the shape in the upstream region of the reference position in the axial direction of the compressor impeller), and have the same shape at the side of the trailing edge where the shape is less likely to contribute to improvement of rotating stall and more likely to have an impact on the blade element performance (the shape in the downstream region of the reference position in the axial direction of the compressor impeller). Accordingly, it is possible to suppress rotating stall while suppressing a decrease in the blade element performance, and thus it is possible to improve the performance of the compressor impeller effectively.

(9) In some embodiments, in the above compressor impeller (8), the reference position is a position upstream of an intersection between a suction surface of the second blade and a perpendicular line to the suction surface of the second blade from the tip-side end of the leading edge of the blade next to the suction surface of the second blade.

With the above compressor impeller (9), with the reference position described in the above (8) being positioned upstream of the intersection in the axial direction of the impeller (throat position of the second blade), it is possible to differentiate the blade angle of the first blade from the blade angle of the second blade without changing the throat width between the second blade and the blade next to the suction surface of the second blade as described in the above (1) to (8). Thus, it is possible to suppress rotating stall while suppressing a decrease in the intake air amount of the compressor impeller.

(10) A method for manufacturing the compressor impeller according to any one of the above (1) to (9) includes: a first blade forming step of forming a plurality of first blades having the same shape; and a second blade forming step of forming at least one second blade by performing a bending process on a leading-edge side portion of a part of the first blades formed in the first blade forming step.

[0011] According to the method for manufacturing a compressor impeller described in the above (10), it is possible to form the second blade merely by performing a bending process on the leading edge portion of only a part of the first blades after forming a plurality of first blades having the same shape, and thereby it is possible to easily manufacture the centrifugal compressor impeller described in any one of the above (1) to (9).

Advantageous Effects

[0012] According to at least one embodiment of the present invention, provided is a compressor impeller and a method for manufacturing the same, whereby it is possible to suppress rotating stall with a simplified configuration.

BRIEF DESCRIPTION OF DRAWINGS

[0013]

FIG. 1 is an axial-directional view of a compressor impeller 100 (100A) according to an embodiment.

FIG. 2 is a meridional cross-sectional view of a part of a compressor impeller 100 (100A) according to an embodiment, taken along the axial direction.

FIG. 3 is a schematic diagram for describing the shape of the first blade 12 and the second blade 14.

FIG. 4 is a blade-row expanded view schematically showing the positional relationship of the plurality of blades 4 on the tip side. In FIG. 4, the line indicating each blade 4 is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4.

FIG. 5 is a blade-row expanded view schematically showing the positional relationship of the plurality of blades 4 on the hub side. In FIG. 5, the line indicating each blade 4 is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4.

FIG. 6 is a diagram schematically showing a rotating stall state in a comparative embodiment. In FIG. 6, the line indicating each blade 4 is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4.

FIG. 7 is a diagram schematically showing a rotating stall state in an embodiment. In FIG. 7, the line indicating each blade 4 is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4.

FIG. 8 is a view showing comparison of surge lines in an embodiment and a comparative embodiment.

FIG. 9 is a schematic diagram for describing another example of the shape of the first blade 12 and the second blade 14. In FIG. 4, the line indicating each blade 4 is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4.

FIG. 10 is an axial-directional view of a compressor impeller 100 (100B) according to an embodiment.

FIG. 11 is a meridional cross-sectional view of a part of a compressor impeller 100 (100B) according to an embodiment, taken along the axial direction.

FIG. 12 is a blade-row expanded view schematically showing an example of the positional relationship of a plurality of full blades 4f and a plurality of splitter blades 4s on the tip side. In FIG. 12, the line indicating each blade 4 (4f, 4s) is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4.

FIG. 13 is a blade-row expanded view schematically showing an example of the positional relationship of a plurality of full blades 4f and a plurality of splitter blades 4s on the tip side. In FIG. 13, the line indicating each blade 4 (4f, 4s) is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4 (4f, 4s).

FIG. 14 is a blade-row expanded view schematically showing an example of the positional relationship of a plurality of full blades 4f and a plurality of splitter blades 4s on the tip side. In FIG. 14, the line indicating each blade 4 (4f, 4s) is the camber line connecting the middle points of the suction surface and the pressure surface of the blade 4 (4f, 4s).

FIG. 15 is a partial meridional cross-sectional view of a compressor impeller 100 according to an embodiment, taken along the axial direction.

DETAILED DESCRIPTION

[0014] Embodiments of the present invention will now be described in detail with reference to the accompanying drawings. It is intended, however, that unless particularly identified, dimensions, materials, shapes, relative positions and the like of components described in the embodiments shall be interpreted as illustrative only and not intended to limit the scope of the present invention.

[0015] For instance, an expression of relative or absolute arrangement such as "in a direction", "along a direction", "parallel", "orthogonal", "centered", "concentric" and "coaxial" shall not be construed as indicating only the arrangement in a strict literal sense, but also includes a state where the arrangement is relatively displaced by a tolerance, or by an angle or a distance whereby it is possible to achieve the same function.

[0016] For instance, an expression of an equal state such as "same" "equal" and "uniform" shall not be construed as indicating only the state in which the feature is strictly equal, but also includes a state in which there is a tolerance or a difference that can still achieve the same function.

[0017] Further, for instance, an expression of a shape such as a rectangular shape or a cylindrical shape shall not be construed as only the geometrically strict shape, but also includes a shape with unevenness or chamfered corners within the range in which the same effect can be achieved.

[0018] On the other hand, an expression such as "comprise", "include", "have", "contain" and "constitute" are not intended to be exclusive of other components.

[0019] FIG. 1 is an axial-directional view of a compressor impeller 100 (100A) according to an embodiment. FIG. 2 is a meridional cross-sectional view of a part of a compressor impeller 100 (100A) shown in FIG. 1, taken along the axial direction.

[0020] As shown in FIGs. 1 and 2, the compressor impeller 100 includes a hub 2 and a blade group 6 including a plurality of blades 4 arranged at intervals in the circumferential direction on the outer peripheral surface 2a of the hub 2. In the blade group 6, the blades 4 are aligned such that the hub-side ends 4A of the leading edges of the respective blade 4 are on the same circle C1 centered at the rotational axis O of the compressor impeller. The blade group 6 is configured such that the hub-side ends 4A of the plurality of blades 4 are at the same position in the axial direction of the compressor impeller 100. The plurality of blades 4 includes at least one first blade 12, and at least one second blade 14 having a different shape from the first blade 12.

[0021] FIG. 3 is a schematic diagram for describing the shape of the first blade 12 and the second blade 14. FIG. 4 is a blade-row expanded view schematically showing the positional relationship of the plurality of blades 4 on the tip side. FIG. 5 is a blade-row expanded view schematically showing the positional relationship of the plurality of blades 4 on the hub side.

[0022] In FIGs. 4 and 5, the horizontal axis represents the position 'rθ' in the circumferential direction of the compressor impeller 100, and the vertical axis represents the distance from the leading edge 12LE in the meridional direction 'm'. Herein, as shown in FIG. 15, the "meridional direction 'm'" refers to the direction along a line connecting points at which the blade height ratio is the same, from the leading edge 12LE to the trailing edge 12TE of the blade 4. Herein, "blade height ratio" is defined as follows. First, as shown in FIG. 15, 'mh' refers to the meridional length from the position of the leading edge 12LE to the position of the trailing edge TE at the hub-side end of the blade 4, and 'mt' refers to the meridional length from the position of the leading edge LE to the position of the trailing edge TE at the tip-side end of the blade 4. Furthermore, provided that the position P and the position Q are such positions that the ratio of the meridional length from the position of the leading edge 12LE on the hub-side end of the blade 4 to the position P divided by the meridional length 'mh' is equal to the ratio of the meridional length from the position of the leading edge LE on the tip-side end of the blade 4 to the position Q divided by the meridional length 'mh' (e.g. the position P and the position Q where both of the ratios are 20%), the length of the segment connecting the position P and the position Q is defined as the blade height 'h' at a meridional position (%). Further, the value y/h obtained by dividing the distance y from the outer peripheral surface 2a of the hub 2 in the blade height direction along the segment by the blade height 'h' is defined as the blade height ratio.

[0023] In FIGs. 3 and 4, when comparing the blade angle $\beta 1$ of the leading edge LE of the first blade 12 to the blade angle $\beta 2$ of the leading edge 14LE of the second blade 14 at the same position 'r' with respect to the radial direction of the compressor impeller 100, at least in a partial range w1 in the radial direction of the compressor impeller 100, the blade angle $\beta 1$ of the leading edge LE of the first blade 12 is different from the blade angle $\beta 2$ of the leading edge 14LE of the second blade 14. Herein, "blade angle β" refers to the angle β (see FIG. 4, for instance) formed between the meridional direction 'm' and the camber line at a blade height y/h, and is defined, using the position 'm' in the meridional direction and the position 'rθ' in the circumferential direction, by the following equation 1: $\beta = tan^{-1}\frac{d(r\theta)}{dm}$.

[0024] With the above configuration, for the plurality of blades 4 (see FIG. 1) arranged so that the hub-side ends 4A of the leading edges align on the same circle, it is possible to differentiate the stall characteristics between the first blade 12 and the second blade 14. Thus, as compared to a case where the plurality of blades 4 have a uniform shape, it is possible to impair uniform propagation and development of rotating stall. Accordingly, it is possible to improve the characteristics of the compressor at the low flow-rate side. Furthermore, it is unnecessary to provide an additional suppressing member disclosed on the prior art (JP2014-118916), and thus it is possible to simplify the configuration compared to the prior art.

[0025] In an embodiment, as shown in FIG. 1 for instance, the at least one first blade 12 includes a plurality of first blades 12, and the at least one second blade 14 includes a plurality of second blades 14. The number of second blades 14 in the blade group 6 is smaller than the number of the first blades 12 in the blade group 6. Further, as shown in FIGs. 1 and 4, the plurality of second blades 14 includes a pair of second blades 14 between which the first blade 12 is not disposed in the circumferential direction of the compressor impeller 100. In the exemplary embodiment shown in FIG. 1, the blade group 6 includes six blades 4, and the six blades 4 includes four first blades 12 and two second blades 14. None of the first blades 12 is disposed between the two second blades 14.

[0026] With the above configuration, the second blades 14 relatively fewer than the first blades 12 and having different stalling characteristics from the first blades 12 are aligned continuously in the circumferential direction of the compressor impeller 100, and thus it is possible to enhance the above effect to impair uniform propagation and development of rotating stall.

[0027] In an embodiment, as shown in FIGs. 3 and 4, when comparing the blade angle β1 of the leading edge LE of the first blade 12 to the blade angle β2 of the leading edge 14LE of the second blade 14 at the same position 'r' with respect to the radial direction of the compressor impeller 100, at least in the partial range w1 in the radial direction of the compressor impeller 100, the blade angle β2 of the leading edge 14LE of the second blade 14 is greater than the blade angle β1 of the leading edge 12LE of the first blade 12.

[0028] With the above configuration, as compared to a comparative embodiment in which the plurality of blades 4 have a uniform shape, that is, a case in which the plurality of blades 4 includes only the first blades 12 (see FIG. 6), the leading edges LE of the second blades 14 have the relatively large blade angle β2 matched to the small flow-rate side (less likely to stall even at a small flow rate), and thereby stalling is less likely to occur in the region A on the suction surface side of the second blade 14, which makes it possible to impair propagation and development of rotating stall effectively. Accordingly, as shown in FIG. 7, as compared to the above comparative embodiment, it is possible to shift the surge line to the small flow-rate side and expand the operational range at the small flow-rate side. Furthermore, the leading edges 12LE of the relatively large number of first blades 12 have the relatively small blade angle β1 taking account of the intake air amount at the high flow-rate side in the range 21, and thereby it is possible to suppress a decrease in the intake air amount of the compressor impeller 100. Thus, it is possible to suppress a decrease in the intake air amount of the compressor impeller 100 while impairing uniform propagation and development of rotating stall.

[0029] In an embodiment, in FIGs. 3 and 4, the blade angle β2 of the tip side ends 14E of the leading edges 14LE of the second blades 14 is greater than the blade angle β1 of the tip-side ends 12E of the leading edges 12LE of the first blades 12. Preferably, the blade angle β2 of the tip side ends 14E of the leading edges 14LE of the second blades 14 is greater than the blade angle β1 of the tip-side ends 12E of the leading edges 12LE of the first blades 12 by 5 degrees or more.

[0030] According to findings of the present inventors, rotating stall of a compressor impeller is likely to occur in a region on the tip side of the leading edge of a blade. Thus, with the blade angle β2 of the tip side ends 14E of the leading edges 14LE of the second blades 14 being greater than the blade angle β1 of the tip side ends 12E of the leading edges 12LE of the first blades 12 as described above, it is possible to suppress rotating stall effectively.

[0031] In an embodiment, in FIGs. 3 and 5, the blade angle β2 of the hub side ends 14A of the leading edges 14LE of the second blades 14 is equal to the blade angle β1 of the hub side ends 12A of the leading edges 12LE of the first blades 12.

[0032] As described above, rotating stall of a compressor impeller is likely to occur in the tip-side region of a blade. Thus, even if the blade angle β2 of the leading edge LE of the second blade 14 at the hub-side end 14A is greater than the blade angle β1 of the leading edge 12LE of each blade 12 at the hub-side end 12A, the effect to suppress rotating stall is relatively small. Furthermore, configuring the second blades 14 to have a large blade angle β2 matched to the small flow-rate side in a broad range in the radial direction of the compressor impeller 100 may lead to a decrease in the intake air amount of the compressor impeller 100.

[0033] Thus, with the blade angle β2 of the leading edges 14LE of the second blades 14 on the tip side end 14E being greater than the blade angle β1 of the tip side ends 12E of the leading edges 12LE of the first blades 12, and the blade angle of the leading edges 14LE of the second blades 14 on the hub-side end 14E being equal to the blade angle β1 of the hub-side ends 12A of the leading edges 12LE of the first blades 12, it is possible to impair uniform propagation and development of rotating stall while suppressing a decrease in the intake air amount of the compressor impeller 100.

[0034] In an embodiment, as shown in FIGs. 3 to 5, when comparing the blade angle β1 of the leading edge LE of the first blade 12 to the blade angle β2 of the leading edge 14LE of the second blade 14 at the same position 'r' with respect to the radial direction of the compressor impeller 100, the blade angle β2 of the leading edge 14LE of the second blade 14 is greater than the blade angle β1 of the leading edge LE of the first blade 12 in the range w1 from a predetermined position P1 of not less than 50% of the blade height 'h' of the second blade 14 in the radial direction of the compressor impeller 100 (e.g. a predetermined position of 70% to 80% of the blade height 'h' of the second blade 14) to the tip side end 14E, and is equal to the blade angle β1 of the leading edge LE of the first blade 12 in the range w2 from the hub side end 14A of the second blade 14 in the radial direction of the compressor impeller 100 to the predetermined position P1.

[0035] As described above, rotating stall of a compressor impeller is likely to occur in the tip-side region of the leading edge of a blade. Thus, even if the angle β2 of the leading edge LE of the second blade 14 at the hub-side end is greater than the blade angle β1 of the leading edge 12LE of the first blade at the hub-side end, the effect to suppress rotating stall is relatively small. Furthermore, configuring the second blades 14 so that the leading edges 14 have a large blade angle β2 matched to the small flow-rate side in a broad range in the radial direction of the compressor impeller 100 may lead to a decrease in the intake air amount of the compressor impeller 100.

[0036] Thus, with the blade angle β2 of the leading edge 14LE of the second blade 14 being greater than the blade angle β1 of the leading edge LE of the first blade 12 in the range w1 from the predetermined position P1 of not less than 50% of the blade height h of the second blade 14 in the radial direction of the compressor impeller 100 to the tip side end 14E, and being equal to the blade angle β1 of the leading edge LE of the first blade 12 in the range w2 from the hub side end 14A of the second blade 14 in the radial direction of the compressor impeller 100 to the predetermined position PI, it is possible to impair uniform propagation and development of the rotating stall while suppressing a decrease in the intake air amount of the compressor impeller 100.

[0037] In an embodiment, as shown in FIGs. 3 and 4, the first blade 12 and the second blade 14 have different shapes only in the upstream region of the reference position P2 in the axial direction of the compressor impeller 100, and have the same shape in the downstream region of the reference position P2 in the axial direction of the compressor impeller 100.

[0038] The curve of a blade and the blade angle of the trailing edge of a blade have a great impact on the blade element performance, and thus the plurality of blades 4 should have a uniform shape on the trailing edge side. That is, preferably, the shape of the blade 12 at the side of the trailing edge 12TE and the shape of the blade 14 at the side of the trailing edge 14TE are the same. Thus, the first blade 12 and the second blade 14 have different shapes only at the side of the leading edge where the shape is likely to contribute to improvement of rotating stall (the shape in the upstream region of the reference position P2 in the axial direction of the compressor impeller 100), and have the same shape at the side of the trailing edge where the shape is less likely to contribute to improvement of rotating stall and more likely to have an impact on the blade element performance (the shape in the downstream region of the reference position P2 in the axial direction of the compressor impeller 100). Accordingly, it is possible to suppress rotating stall while suppressing a decrease in the blade element performance, and thus it is possible to improve the performance of the compressor impeller 100 effectively.

[0039] In an embodiment, as shown in FIGs. 3 and 4 for instance, the above reference position P2 is upstream of an intersection P3 (throat position of the second blade) between the suction surface 14S of the second blade 2 and a perpendicular line to the suction surface 14S of the second blade 14 from the tip-side end of the leading edge of the blade 4 next to the suction surface 14S of the second blade 4 (the tip side end 12E of the leading edge 12LE of the first blade 12 in the embodiment shown in the drawings).

[0040] With the reference position P2 being positioned upstream of the intersection P3 in the axial direction of the impeller 100, it is possible to differentiate the blade angle β1 of the first blade 12 from the blade angle β2 of the second blade 14 while suppressing a change in the throat width S between the second blade 14 and the blade 4 next to the suction surface 14S of the second blade 14. Thus, it is possible to suppress rotating stall while suppressing a decrease in the intake air amount of the compressor impeller 100.

[0041] While the above described compressor impeller 100 can be manufactured by machining, casting, or the like, the manufacturing method may include a first blade forming step of forming the plurality of first blades 12 having the same shape, and a second blade forming step of forming at least one second blade 14 by performing a bending process only on a portion 12P (see FIG. 3) on the tip side and on the leading edge side of a part of the first blades 12 formed in the first blade forming step so as to curve smoothly toward the pressure surface side in an arc shape.

[0042] Accordingly, as compared to a case in which the first blade 12 and the second blade 14 are formed in separate steps, it is possible to form the second blades 14 by merely performing a bending process on the first blades 12 formed via the first blade forming step, and thus it is possible to manufacture the compressor impeller 100 easily.

[0043] Embodiments of the present invention were described in detail above, but the present invention is not limited thereto, and various amendments and modifications may be implemented.

[0044] For instance, in FIG. 4, the first blade 12 and the second blade 12 have different shapes only in the upstream region of the reference position P2 in the axial direction of the compressor impeller 100, and have the same shape in the downstream region of the reference position P2 in the axial direction of the compressor impeller 100.

[0045] However, the present invention is not limited to this embodiment. As shown in FIG. 9 for instance, the second blade 14 may have a different shape from the first blade 12 in the entire range of the second blade 14 in the axial direction of the compressor impeller 100. Also in this configuration, it is sufficient if, when comparing the blade angle β1 of the leading edge 12LE of the first blade 12 to the blade angle β2 of the leading edge 14LE of the second blade 14 at the same position with respect to the radial direction of the compressor impeller 100, at least in a partial range w1 in the radial direction of the compressor impeller 100, the blade angle β1 of the leading edge LE of the first blade 12 is different from the blade angle β2 of the leading edge 14LE of the second blade 14. Thus, in terms of suppression of a decrease in the intake air amount of the compressor impeller 100, it is possible to suppress uniform propagation and development of rotating stall.

[0046] Nevertheless, in the embodiment shown in FIG. 4, it is possible to differentiate the blade angle β1 of the first blade 12 from the blade angle β2 of the second blade 14 while suppressing a change in the throat width S between the second blade 14 and the blade 4 next to the suction surface 14S of the second blade 14. Thus, the embodiment shown in FIG. 4 is more preferable than the embodiment shown in FIG. 9.

[0047] Furthermore, in FIG. 1, the compressor impeller 100 includes a single blade group 6 (which includes a plurality of blades 4 arranged at intervals in the circumferential direction on the outer peripheral surface 2a of the hub 2, and in which the blades 4 are aligned such that the hub-side ends 4A of the leading edges of the respective blades 4 are on the same circle C1 centered at the rotational axis O of the compressor impeller).

[0048] However, the present invention is not limited to this embodiment. As shown in FIGs. 10 and 11 for instance, the compressor impeller 100 may include a plurality of blade groups. In the exemplary embodiment shown in FIG. 10, the compressor impeller 100 (100B) includes two blade groups: a full blade group 6f and a splitter blade group 6s.

[0049] The full blade group 6f includes a plurality of full blades 4f disposed at intervals in the circumferential direction on the outer peripheral surface 2a of the hub 2. The hub-side ends 4Af of the leading edges of the respective full blades 4f are aligned on the same circle Cf centered at the rotational axis O of the compressor impeller.

[0050] The splitter blade group 6s includes a plurality of splitter blades 4s disposed at intervals in the circumferential direction on the outer peripheral surface 2a of the hub 2. The splitter blades 4s have a shorter blade length than the full blades 4f, and each of the plurality of splitter blades 4s is disposed between two adjacent full blades 4f. The hub-side ends 4As of the leading edges of the plurality of splitter blades 4s are aligned on the same circle Cs centered at the rotational axis O of the compressor impeller 100. Herein, the hub-side ends 4As of the leading edges of the plurality of splitter blades 4s are disposed downstream of the hub-side ends 4Af of the leading edges of the plurality of full blades 4f. That is, the circle Cs has a greater radius than the circle Cf, and is positioned downstream of the circle Cf with respect to the intake direction of the compressor impeller 100.

[0051] In the embodiment shown in FIG. 10, the invention according to the blade group 6 of the compressor impeller 100 (100A) described with reference to FIGs. 1 to 9 may be applied only to the full blade group 6f as shown in FIG. 12, or only to the splitter blade group 6s as shown in FIG. 13, or to both of the full blade group 6f and the splitter blade group 6s as shown in FIG. 14.

[0052] In the embodiment shown in FIG. 12, the plurality of full blades 4f constituting the full blade group 6f includes at least one first blade 12f, and at least one second blade 14f having a different shape from the first blade 12f. Furthermore, when comparing the blade angle β1f of the leading edge 12LEf of the first blade 12f to the blade angle β2f of the leading edge 14LEf of the second blade 14f at the same position with respect to the radial direction of the compressor impeller 100, at least in a partial range (see the range w1 in FIG. 3) in the radial direction of the compressor impeller 100, the blade angle β1f of the leading edge 21LEf of the first blade 12f is different from the blade angle β2f of the leading edge 14LEf of the second blade 14f.

[0053] Also with the above embodiment, for the plurality of full blades 4f arranged so that the hub-side ends 4Af of the leading edges align on the same circle, it is possible to differentiate the stall characteristics between the first blade 12f and the second blade 14f. Thus, as compared to a case where the plurality of full blades 4f have a uniform shape, it is possible to impair uniform propagation and development of rotating stall. Accordingly, it is possible to improve the performance of the compressor at the low flow-rate side. Furthermore, it is unnecessary to provide an additional suppressing member disclosed in the prior art (JP2014-118916), and thus it is possible to simplify the configuration compared to the prior art.

[0054] In the embodiment shown in FIG. 13, the plurality of splitter blades 4 constituting the splitter blade group 6s includes at least one first blade 12s, and at least one second blade 14s having a different shape from the first blade 12s. Furthermore, when comparing the blade angle β1s of the leading edge 12LEs of the first blade 12s to the blade angle β2s of the leading edge 14LEs of the second blade 14s at the same position with respect to the radial direction of the compressor impeller 100, at least in a partial range (see the range w1 in FIG. 3) in the radial direction of the compressor impeller 100, the blade angle β1s of the leading edge 21LEs of the first blade 12s is different from the blade angle β2s of the leading edge 14LEs of the second blade 14s.

[0055] Also with the above embodiment, for the plurality of splitter blades 4s arranged so that the hub-side ends 4As of the leading edges align on the same circle, it is possible to differentiate the stall characteristics between the first blade 12s and the second blade 14s. Thus, as compared to a case where the plurality of splitter blades 4s have a uniform shape, it is possible to impair uniform propagation and development of rotating stall. Accordingly, it is possible to improve the characteristics of the compressor at the low flow-rate side. Furthermore, it is unnecessary to provide an additional suppressing member disclosed on the prior art, and thus it is simplify the configuration compared to the prior art.

[0056] In the embodiment shown in FIG. 14, the plurality of full blades 4f constituting the full blade group 6f includes at least one first blade 12f, and at least one second blade 14f having a different shape from the first blade 12f. Furthermore, when comparing the blade angle β1f of the leading edge 12LE of the first blade 12f to the blade angle β2f of the leading edge 14LEf of the second blade 14f at the same position with respect to the radial direction of the compressor impeller 100, at least in a partial range (see the range w1 in FIG. 3) in the radial direction of the compressor impeller 100, the blade angle β1f of the leading edge 21LEf of the first blade 12f is different from the blade angle β2f of the leading edge 14LEf of the second blade 14f. Furthermore, the plurality of splitter blades 4s constituting the splitter blade group 6s includes at least one first blade 12s, and at least one second blade 14s having a different shape from the first blade 12s. Furthermore, when comparing the blade angle β1s of the leading edge 12LEs of the first blade 12s to the blade angle β2s of the leading edge 14LEs of the second blade 14s at the same position with respect to the radial direction of the compressor impeller 100, at least in a partial range (see the range w1 in FIG. 3) in the radial direction of the compressor impeller 100, the blade angle β1s of the leading edge 21LEs of the first blade 12s is different from the blade angle β2s of the leading edge 14LEs of the second blade 14s.

[0057] Also with the above embodiment, for the plurality of full blades 4f arranged so that the hub-side ends 4Af of the leading edges align on the same circle, it is possible to differentiate the stall characteristics between the first blade 12f and the second blade 14f. Thus, as compared to a case where the plurality of full blades 4f have a uniform shape, it is possible to impair uniform propagation and development of rotating stall. Furthermore, for the plurality of splitter blades 4s arranged so that the hub-side ends 4As of the leading edges align on the same circle, it is possible to differentiate the stall characteristics between the first blade 12s and the second blade 14s. Thus, as compared to a case where the plurality of splitter blades 4s have a uniform shape, it is possible to impair uniform propagation and development of rotating stall. Accordingly, it is possible to improve the performance of the compressor at the low flow-rate side. Furthermore, it is unnecessary to provide an additional suppressing member disclosed in the prior art, and thus it is to simplify the configuration compared to the prior art.

[0058] While a centrifugal compressor is described as an example in the above embodiment, the present invention is not limited to a centrifugal compressor and may be applied to an axial-flow compressor or a mixed-flow compressor, for instance.

Description of Reference Numerals

[0059]

2 Hub
2a Outer peripheral surface
4 Blade
4A Hub-side end
4f Full blade
4s Splitter blade
6 Blade group
6f Full blade group
6s Splitter blade group
12 Second blade
12LE Leading edge
12E Tip-side end of leading edge
12A Hub-side end of leading edge
12P Portion
14 Second blade
14LE Leading edge
14E Tip-side end of leading edge
14A Hub-side end of leading edge
14S Suction surface
100 Impeller
C1 Circle
C2 Circle
L Perpendicular line
O Rotational axis
P1 Predetermined position
P2 Reference position
P3 Intersection
w1, w2 Range
r, z Position

Claims

1. A compressor impeller, comprising:

a hub; and
a blade group including a plurality of blades arranged along a circumferential direction on an outer peripheral surface of the hub, the blade group being configured such that hub-side ends of leading edges of the respective blades are

aligned on the same circle,
wherein the plurality of blades include at least one first blade and at least one second blade having a different shape from the at least one first blade, and
wherein, when comparing a blade angle of a leading edge of the at least one first blade to a blade angle of a leading edge of the at least one second blade at the same position in a radial direction of the compressor impeller, the blade angle of the leading edge of the at least one first blade is different from the blade angle of the leading edge of the at least one second blade, at least in a partial range in the radial direction of the compressor impeller.

2. The compressor impeller according to claim 1,
wherein the at least one first blade includes a plurality of first blades,
wherein the at least one second blade includes a plurality of second blades,
wherein the number of the second blades in the blade group is smaller than the number of the first blades in the blade group, and
wherein the plurality of second blades include a pair of second blades between which the first blade is not disposed.

3. The compressor impeller according to claim 1 or 2,
wherein the number of the second blades in the blade group is smaller than the number of the first blades in the blade group, and
wherein, when comparing the blade angle of the leading edges of the first blades to the blade angle of the leading edges of the second blades at the same position in the radial direction of the compressor impeller, the blade angle of the leading edges of the second blades is greater than the blade angle of the leading edges of the first blades at least in a partial range in the radial direction of the compressor impeller.

4. The compressor impeller according to claim 3,
wherein the blade angle of tip-side ends of the leading edges of the second blades is greater than the blade angle of tip-side ends of the leading edges of the first blades.

5. The compressor impeller according to claim 4,
wherein the blade angle of the tip-side ends of the leading edges of the second blades is greater than the blade angle of the tip-side ends of the leading edges of the first blades by five degrees or more.

6. The compressor impeller according to claim 4 or 5,
wherein the blade angle of hub-side ends of the leading edges of the second blades is equal to the blade angle of hub-side ends of the leading edges of the first blades.

7. The compressor impeller according to claim 6,
wherein, when comparing the blade angle of the leading edges of the first blades to the blade angle of the leading edges of the second blades at the same position in the radial direction of the compressor impeller,
the blade angle of the leading edges of the second blades is greater than the blade angle of the leading edges of the first blades in a range to the tip-side ends from a predetermined position of not less than 50% of a blade height of the second blades in the radial direction of the compressor impeller, and is equal to the blade angle of the leading edges of the first blades in a range to the predetermined position from the hub-side ends of the second blades in the radial direction of the compressor impeller.

8. The compressor impeller according to any one of claims 4 to 7,
wherein the first blades and the second blades have different shapes only in an upstream region of a reference position in an axial direction of the compressor impeller, and have the same shape in a downstream region of the reference position in the axial direction of the compressor impeller.

9. The compressor impeller according to claim 8,
wherein the reference position is a position upstream of an intersection between a suction surface of the second blade and a perpendicular line to the suction surface of the second blade from the tip-side end of the leading edge of the blade next to the suction surface of the second blade.

10. A method for manufacturing the compressor impeller according to any one of claims 1 to 9, comprising:

a first blade forming step of forming a plurality of first blades having the same shape; and
a second blade forming step of forming at least one second blade by performing a bending process on a leading-edge side portion of a part of the first blades formed in the first blade forming step.

FIG. 1

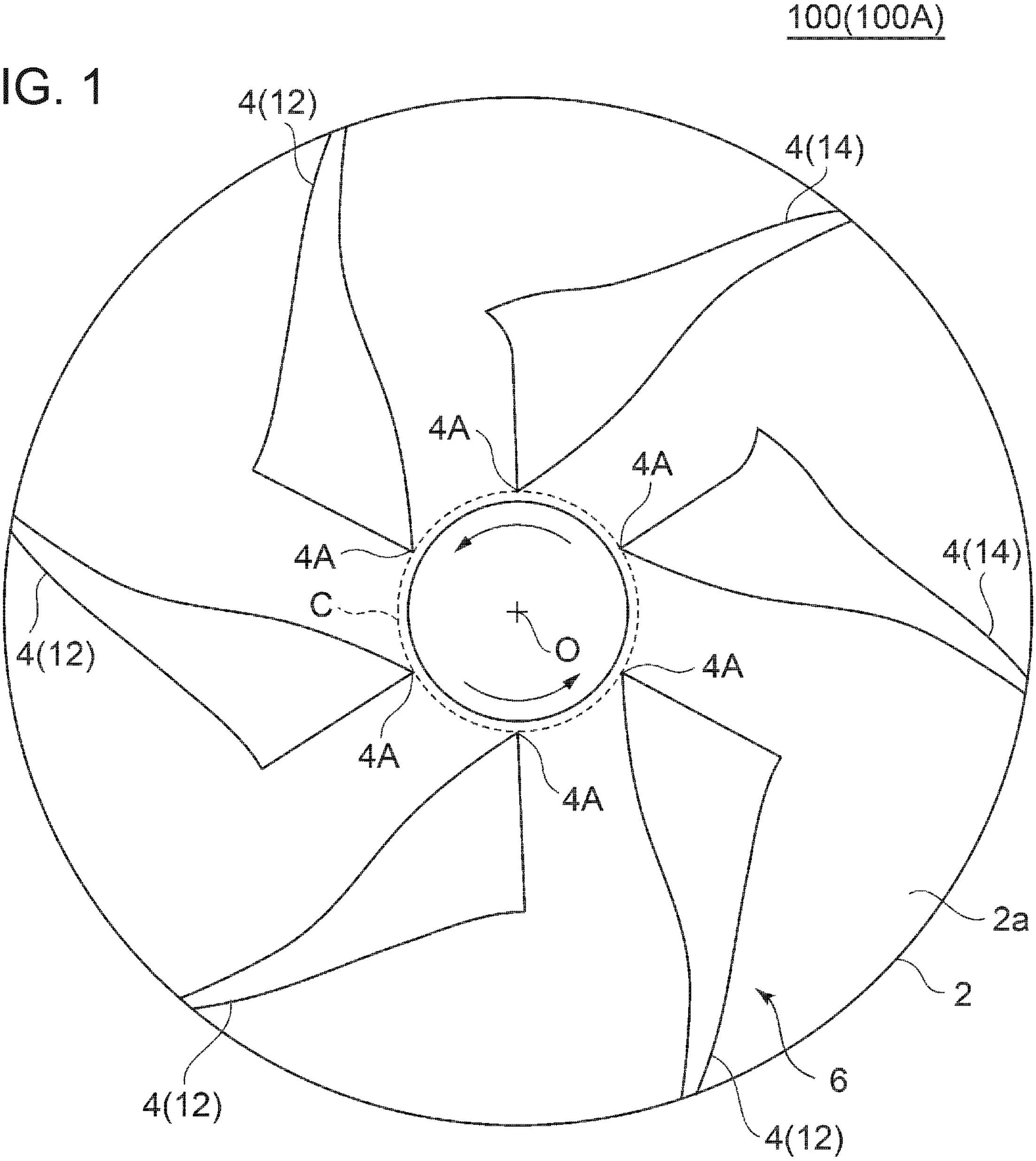


FIG. 2

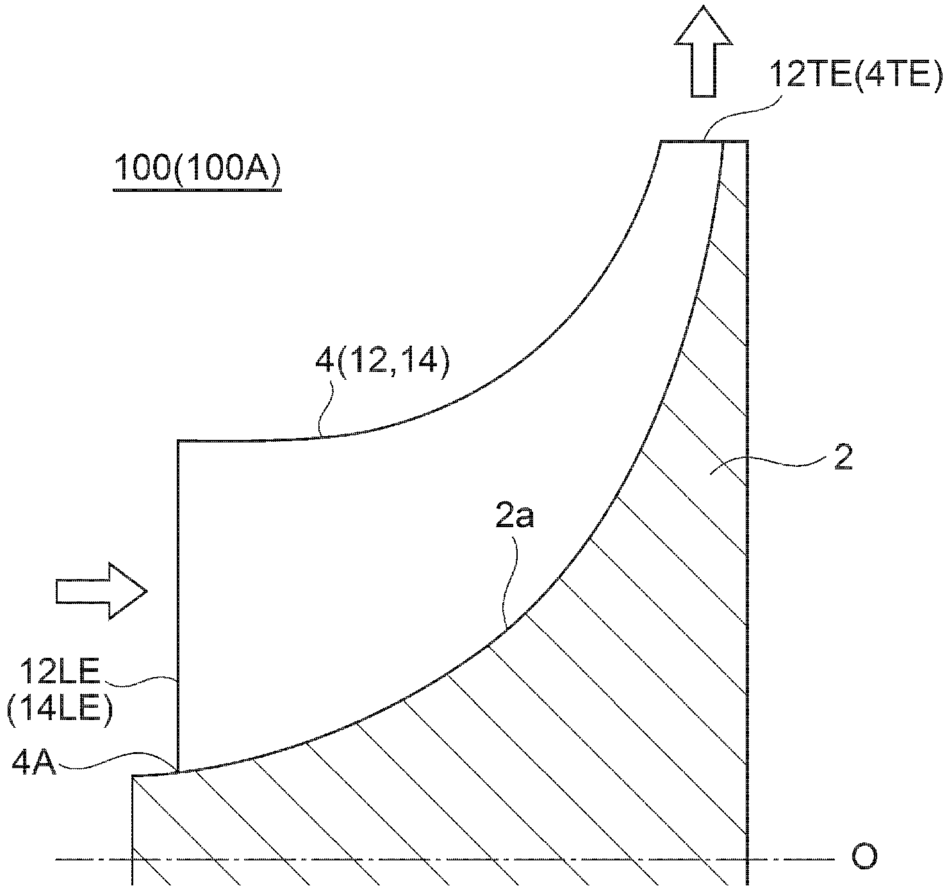


FIG. 3

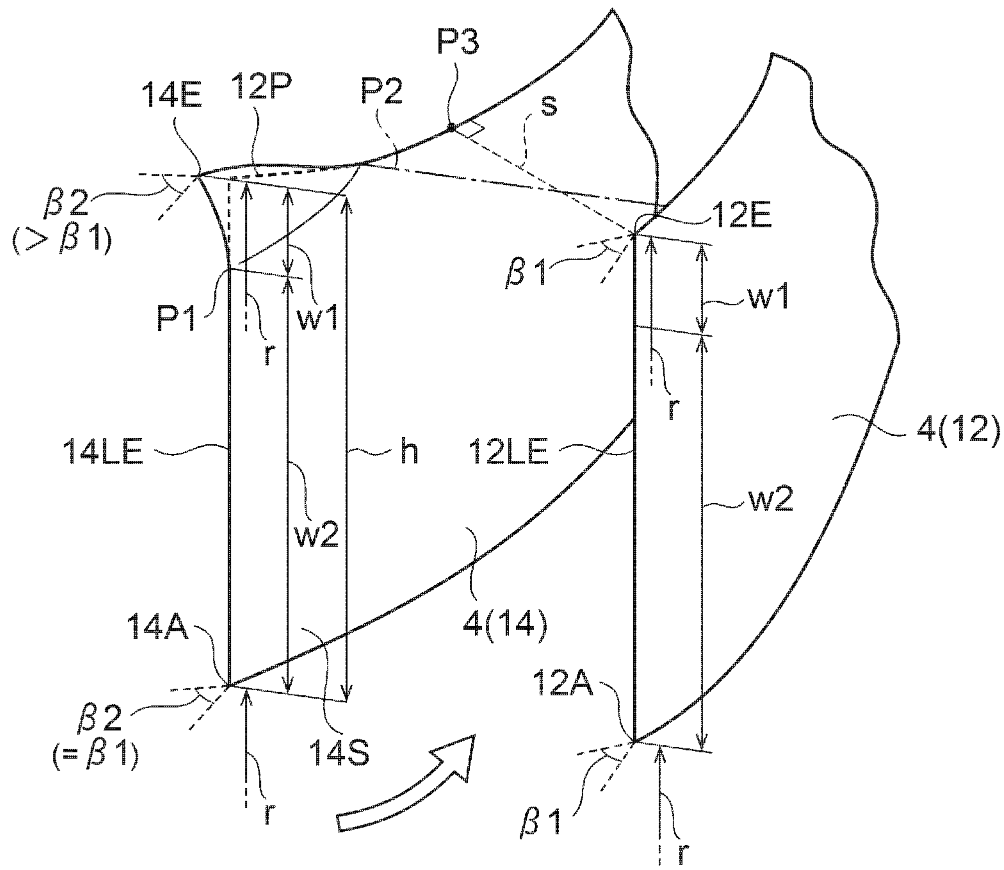

P3
14E
12P
P2
s
β2
(>β1)
12E
β1
w1
P1
r
w1
r
14LE
h
12LE
4(12)
w2
w2
14A
4(14)
12A
β2
(=β1)
14S
r
β1
r

FIG. 4

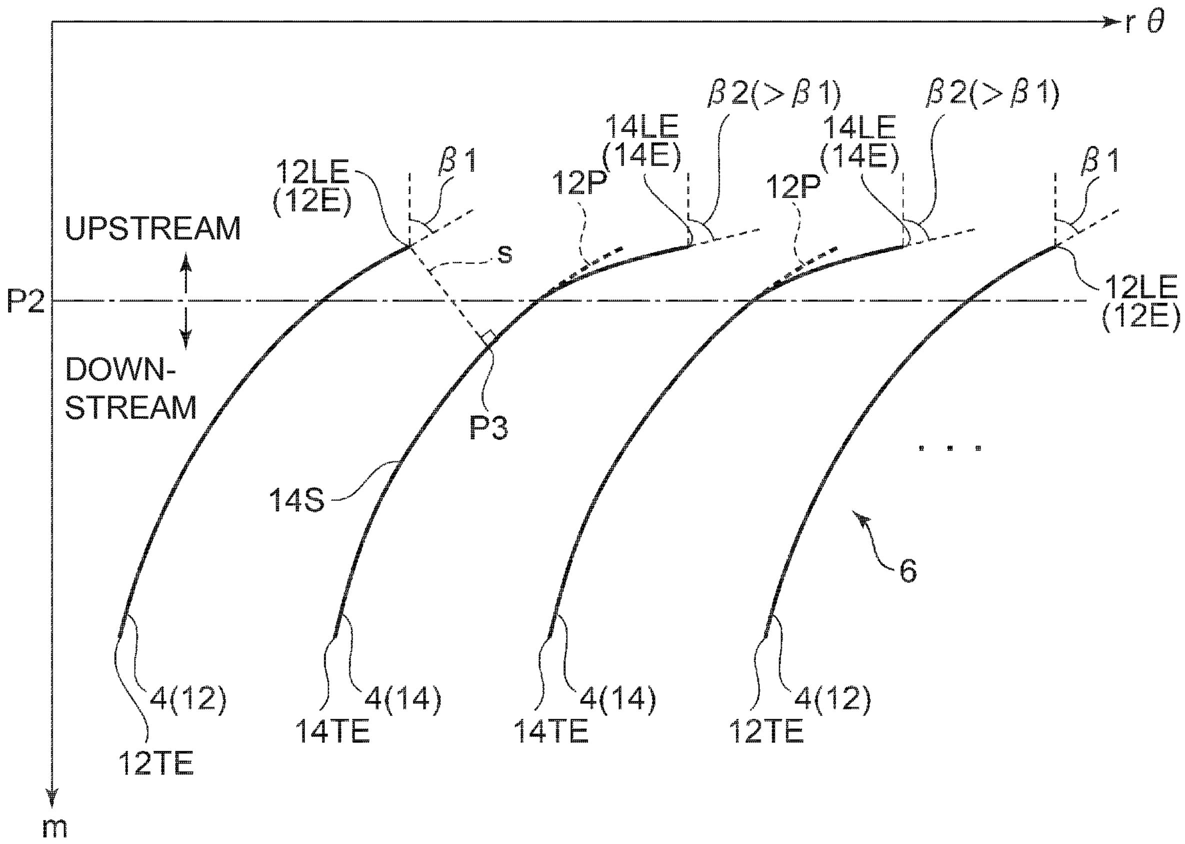

r θ
β1
β2(>β1)
14LE
(14E)
β2(>β1)
14LE
(14E)
β1
12LE
(12E)
12P
12P
UPSTREAM
s
P2
12LE
(12E)
DOWN-
STREAM
P3
14S
6
4(12)
12TE
4(14)
14TE
4(14)
14TE
4(12)
12TE
m

FIG. 5

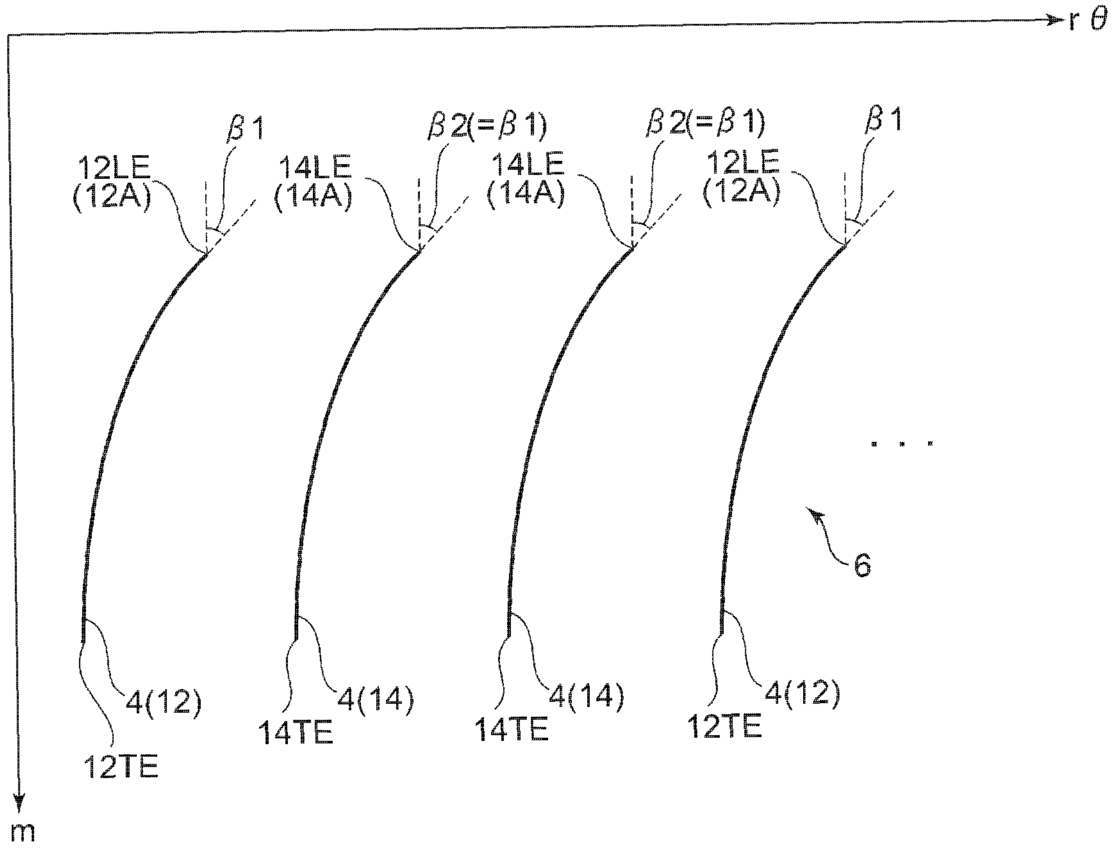

r θ
m
β1
12LE
(12A)
β2(=β1)
14LE
(14A)
β2(=β1)
14LE
(14A)
β1
12LE
(12A)
6
4(12)
12TE
4(14)
14TE
4(14)
14TE
4(12)
12TE

FIG. 6

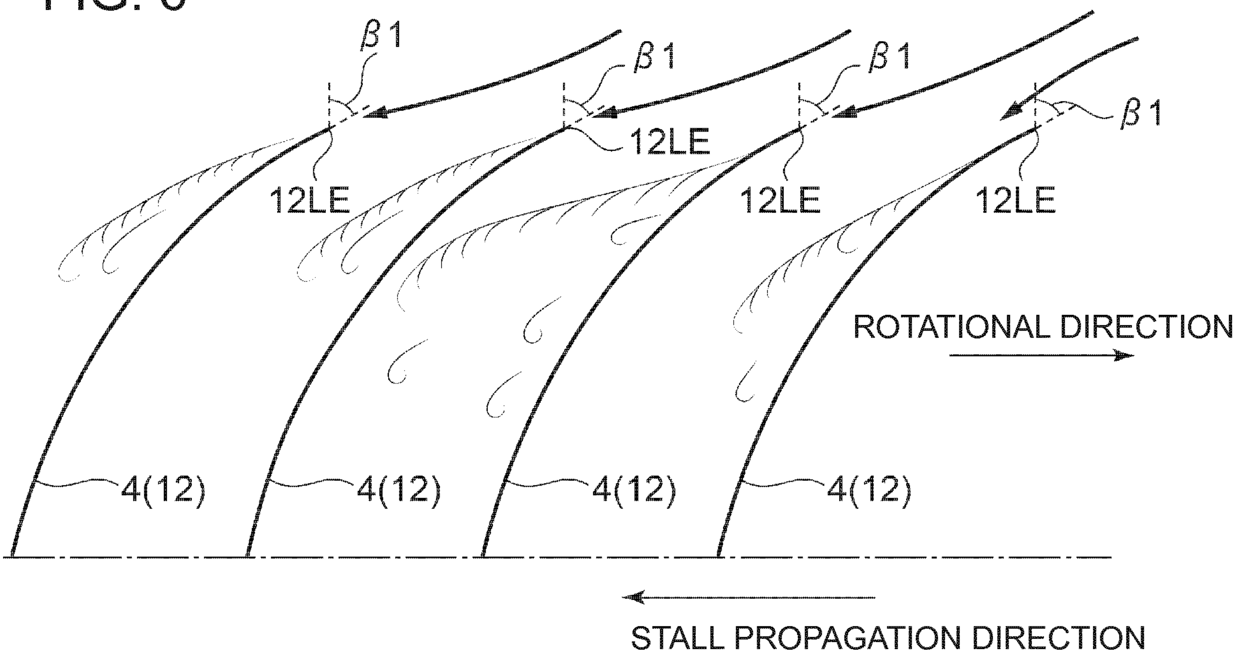

β1
β1
β1
β1
12LE
12LE
12LE
12LE
ROTATIONAL DIRECTION
4(12)
4(12)
4(12)
4(12)
STALL PROPAGATION DIRECTION

FIG. 7

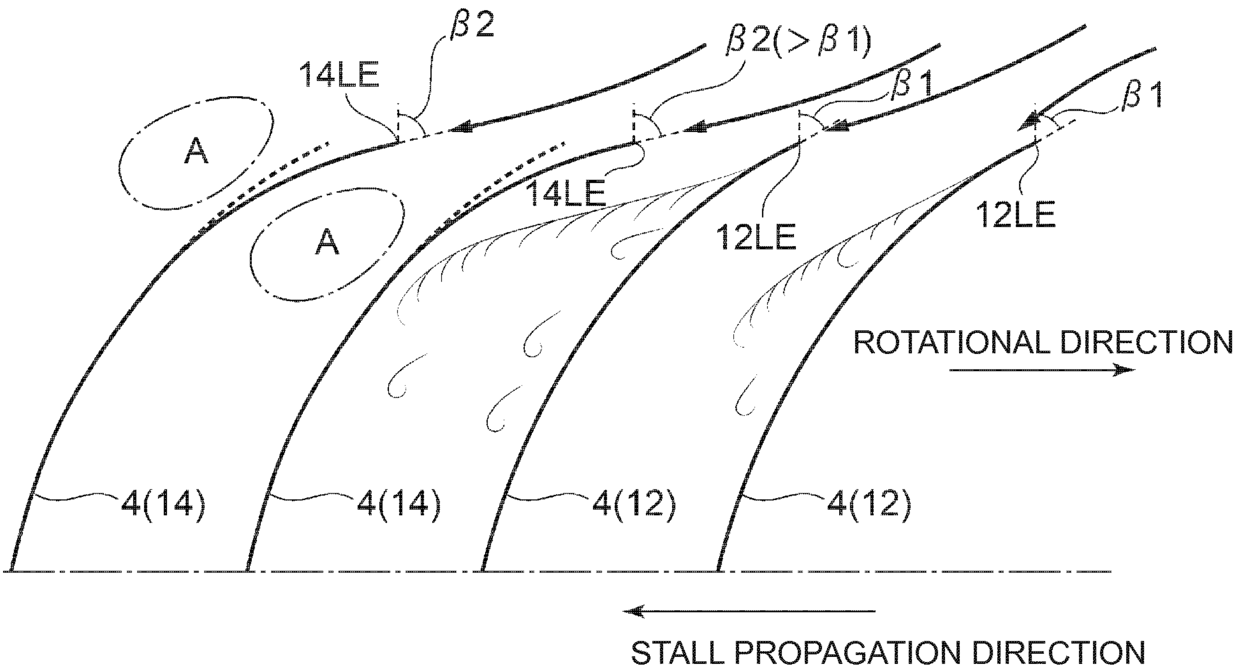

β2
14LE
β2(>β1)
β1
β1
A
A
14LE
12LE
12LE
ROTATIONAL DIRECTION
4(14)
4(14)
4(12)
4(12)
STALL PROPAGATION DIRECTION

FIG. 8

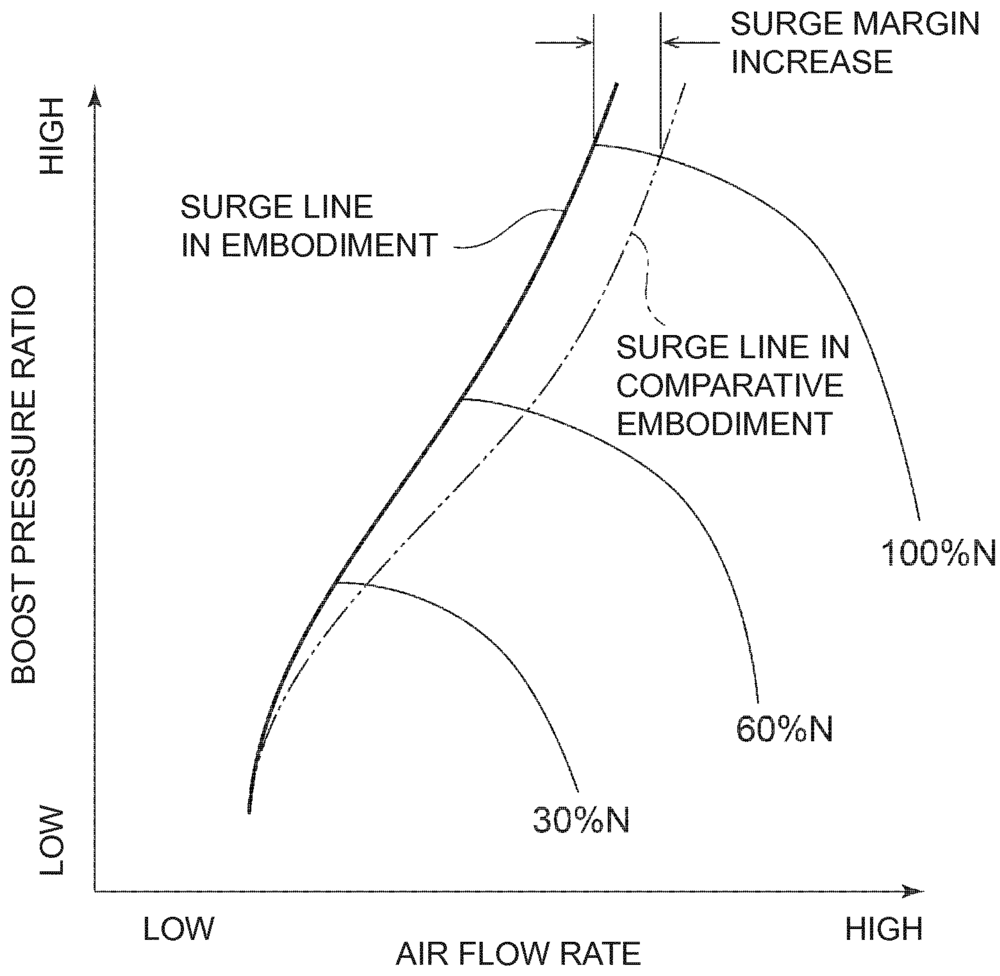

SURGE MARGIN
INCREASE
SURGE LINE
IN EMBODIMENT
SURGE LINE IN
COMPARATIVE
EMBODIMENT
HIGH
BOOST PRESSURE RATIO
LOW
100%N
60%N
30%N
LOW
AIR FLOW RATE
HIGH

FIG. 9

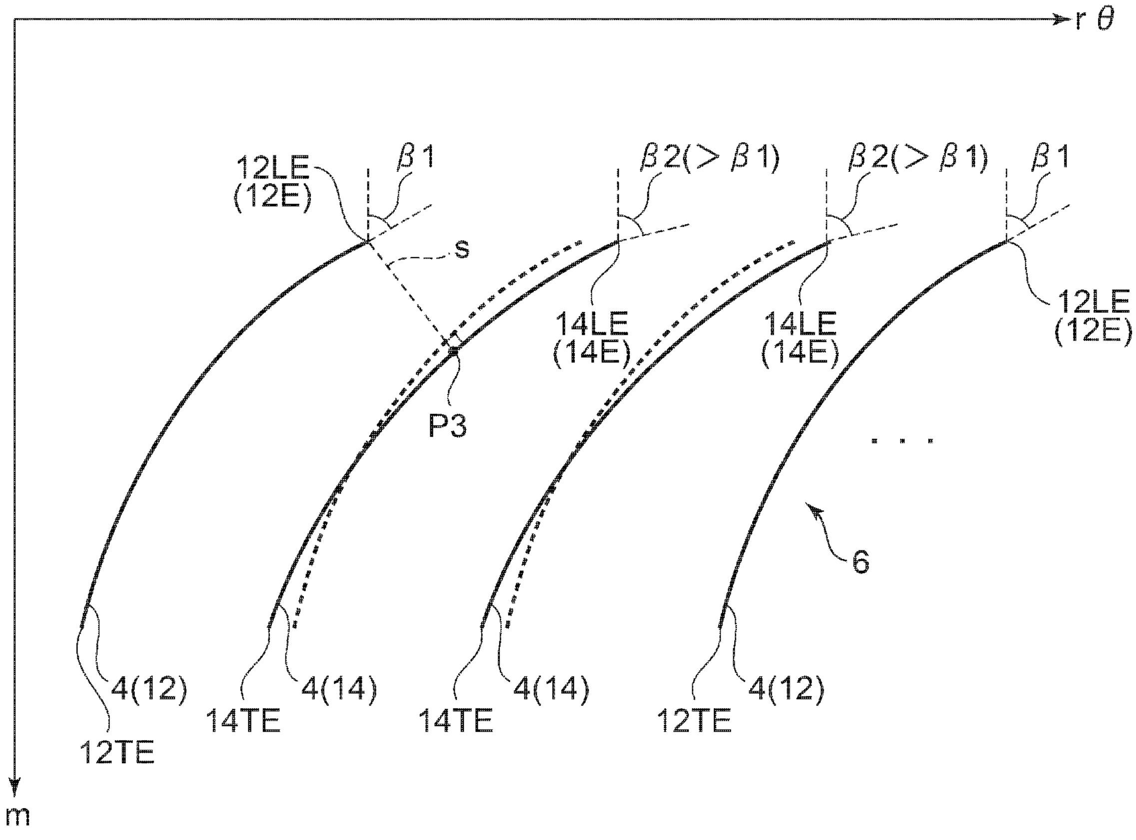

r θ
m
12LE
(12E)
β1
β2(>β1)
β2(>β1)
β1
s
P3
14LE
(14E)
14LE
(14E)
12LE
(12E)
6
4(12)
12TE
14TE
4(14)
14TE
4(14)
12TE
4(12)

FIG. 10

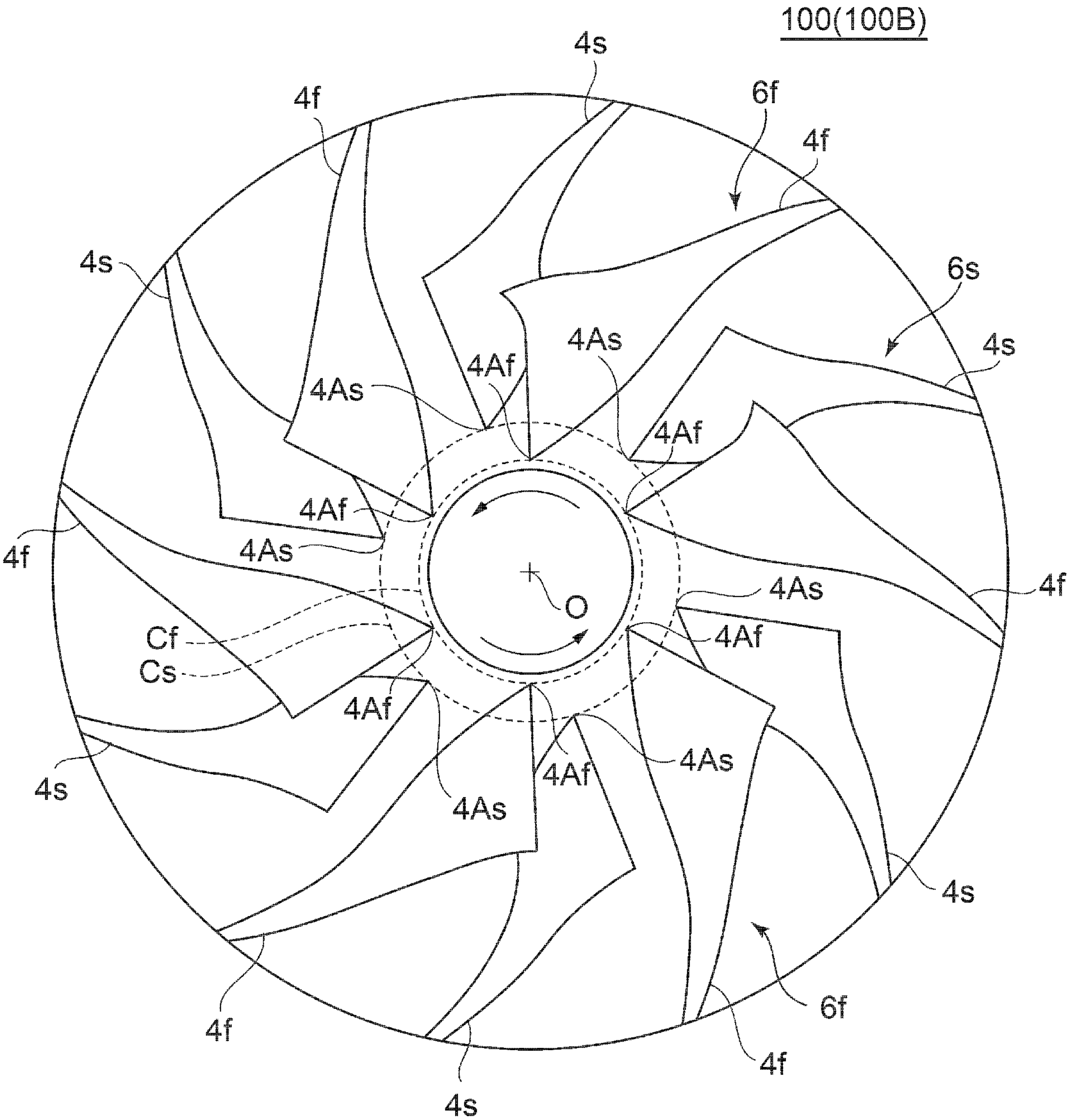

100(100B)
4s
4f
6f
4f
4s
6s
4s
4As
4Af
4As
4Af
4Af
4As
4f
O
4As
4f
4Af
Cf
Cs
4Af
4As
4Af
4s
4As
4s
6f
4f
4f
4s

FIG. 11

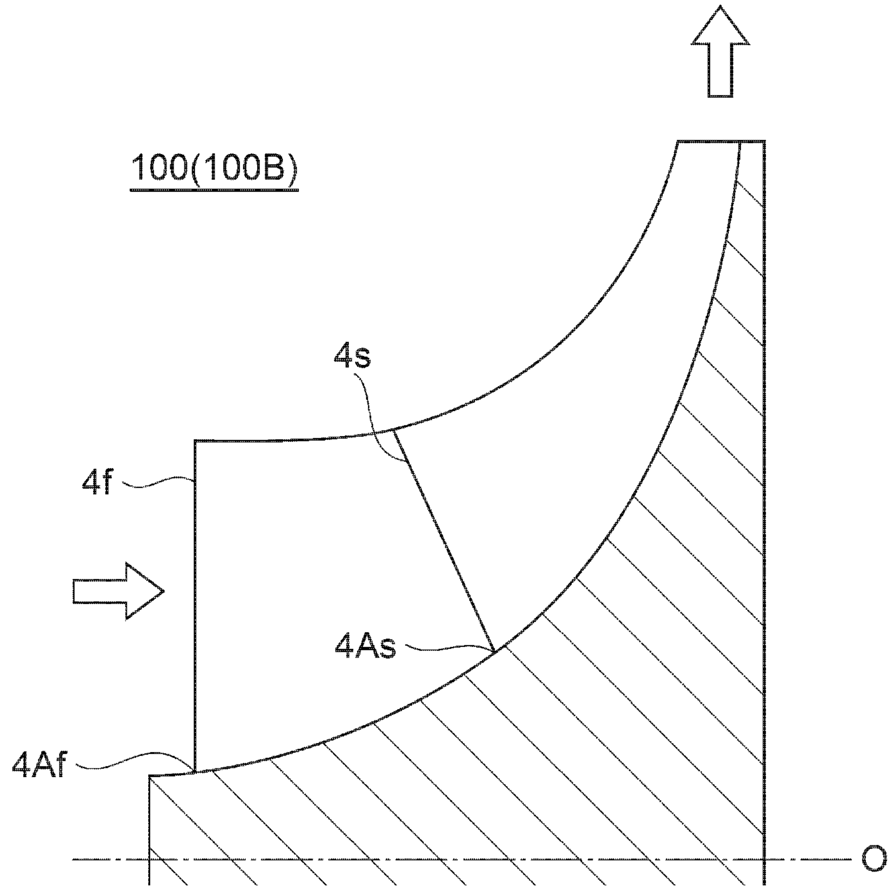
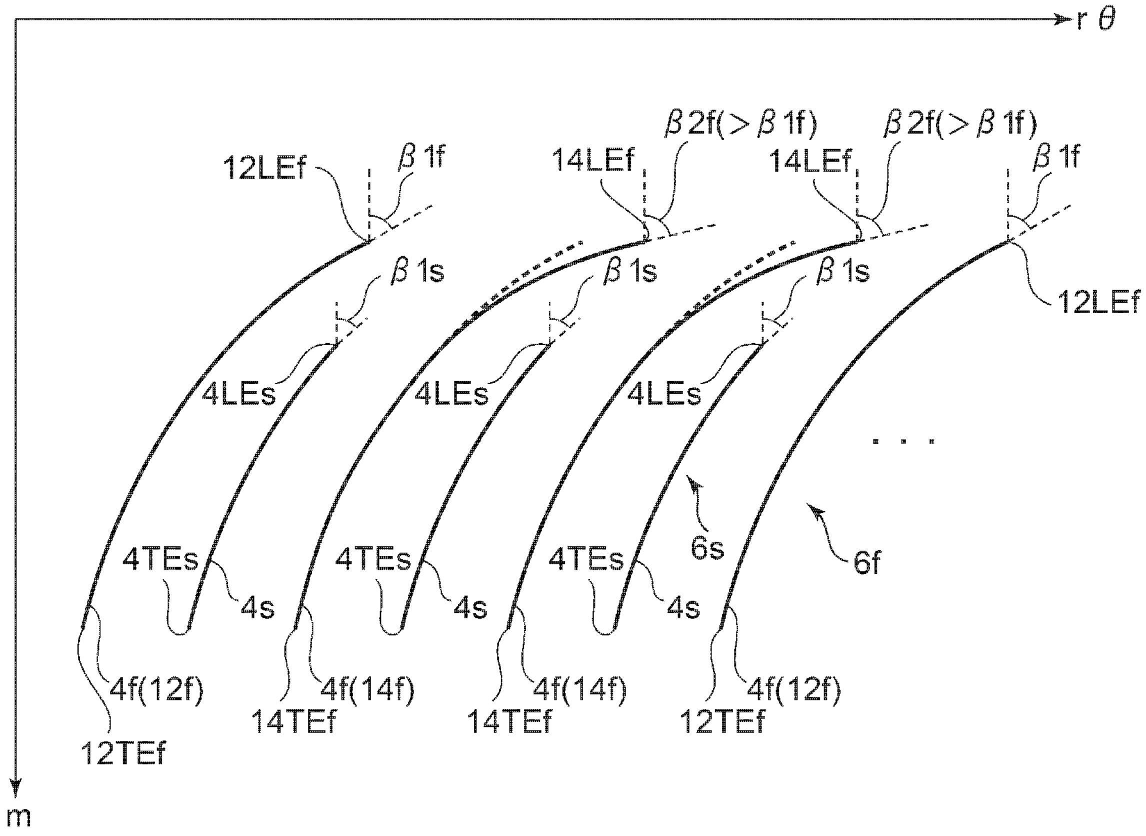


FIG. 12


r θ
m
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β 1f
14LEf
β 2f(> β 1f)
14LEf
β 2f(> β 1f)
β 1f
12LEf
β 1s
β 1s
β 1s
4LEs
4LEs
4LEs
6s
6f
4TEs
4s
4TEs
4s
4TEs
4s
4f(12f)
12TEf
4f(14f)
14TEf
4f(14f)
14TEf
4f(12f)
12TEf

FIG. 13

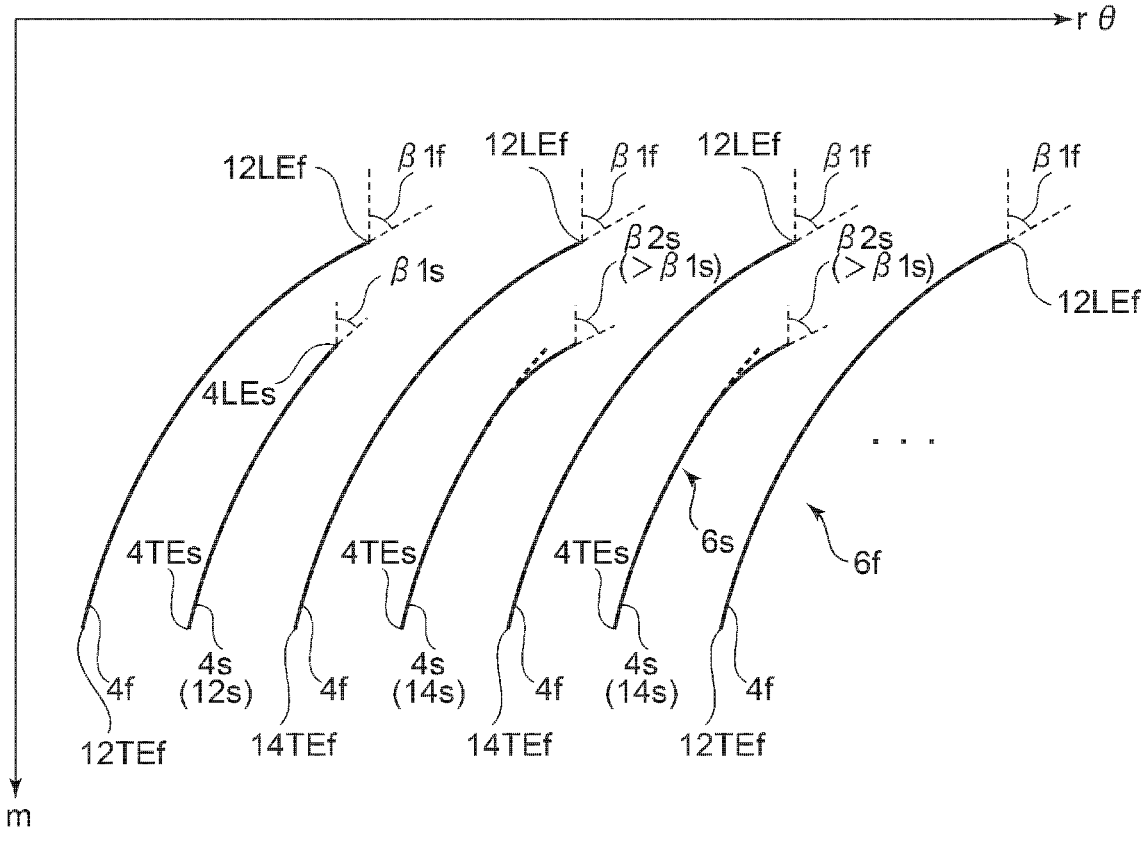
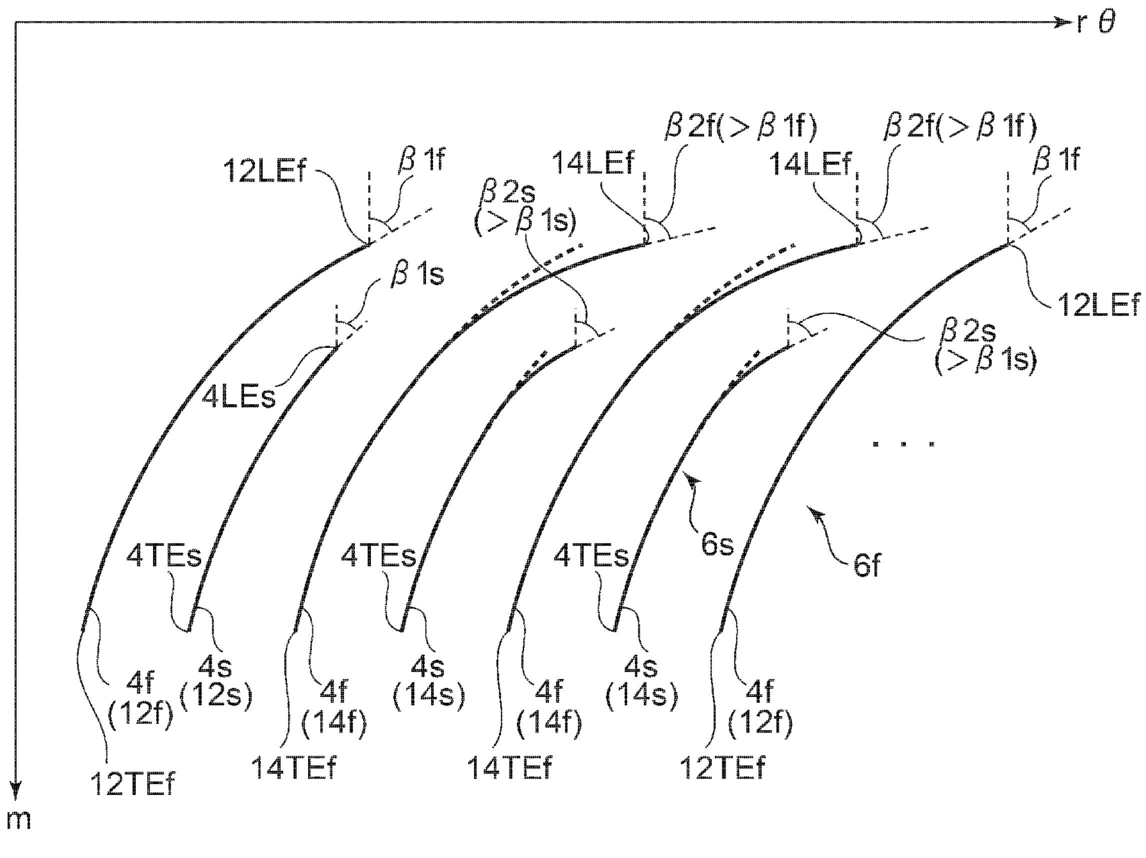
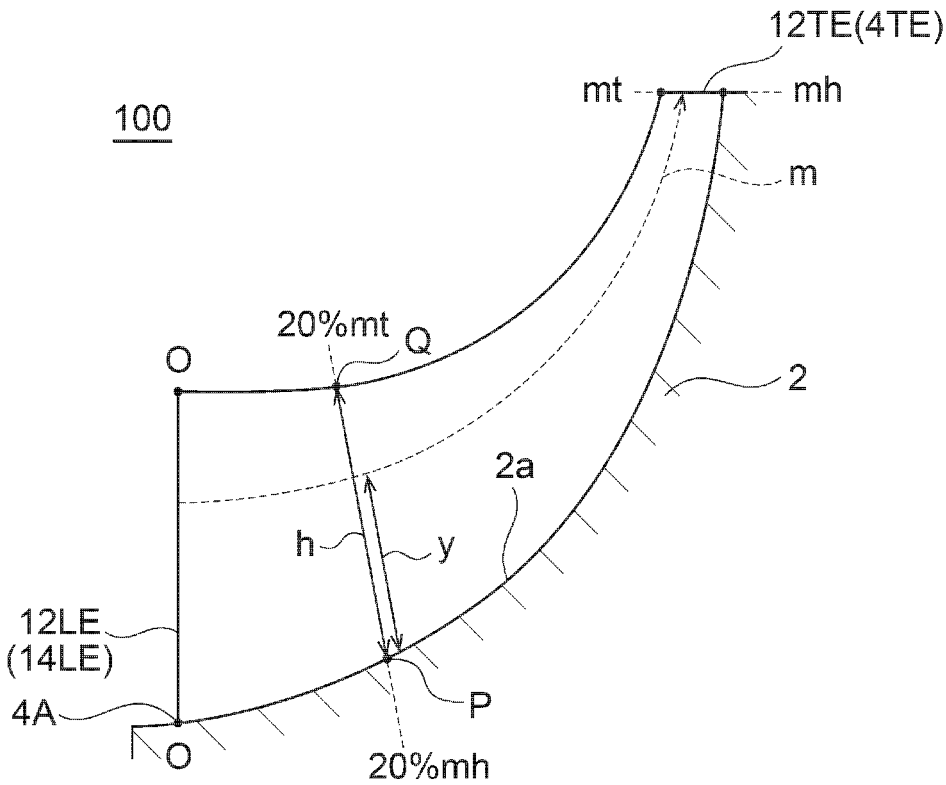

r θ
m
12LEf
β 1f
β 1s
4LEs
β 2s
(> β 1s)
4TEs
4s
(12s)
(14s)
4f
12TEf
14TEf
6s
6f
. . .

FIG. 14



r θ
m
12LEf
β1f
14LEf
β2f(>β1f)
14LEf
β2f(>β1f)
β1f
β2s
(>β1s)
β1s
12LEf
4LEs
β2s
(>β1s)
6s
6f
4TEs
4TEs
4TEs
4f
(12f)
4s
(12s)
4f
(14f)
4s
(14s)
4f
(14f)
4s
(14s)
4f
(12f)
12TEf
14TEf
14TEf
12TEf

FIG. 15



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2016/050923

A. CLASSIFICATION OF SUBJECT MATTER
F04D29/30(2006.01)i, *F04D29/66*(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)
F04D29/30, F04D29/66

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	1996-2016
Kokai Jitsuyo Shinan Koho	1971-2016	Toroku Jitsuyo Shinan Koho	1994-2016

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 2014-109214 A (Mitsubishi Heavy Industries, Ltd.), 12 June 2014 (12.06.2014), paragraph [0024] (Family: none)	1-10
A	JP 2001-214894 A (Seiko Epson Corp.), 10 August 2001 (10.08.2001), paragraphs [0072] to [0074] & US 2002/0182053 A1 paragraphs [0129] to [0132]	1-10
A	JP 2003-214390 A (NIDEC Corp.), 30 July 2003 (30.07.2003), paragraph [0017] (Family: none)	1-10

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

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
18 March 2016 (18.03.16)	05 April 2016 (05.04.16)

Name and mailing address of the ISA/	Authorized officer
Japan Patent Office 3-4-3, Kasumigaseki, Chiyoda-ku, Tokyo 100-8915, Japan	Telephone No.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2016/050923

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	Microfilm of the specification and drawings annexed to the request of Japanese Utility Model Application No. 100845/1988(Laid-open No. 022700/1990) (Nissan Motor Co., Ltd.), 15 February 1990 (15.02.1990), page 5, lines 13 to 20 (Family: none)	1-10
A	JP 2001-248595 A (Ishigaki Co., Ltd.), 14 September 2001 (14.09.2001), paragraph [0011] (Family: none)	1-10
A	JP 58-128243 A (Nippon Light Metal Co., Ltd.), 30 July 1983 (30.07.1983), column 9, lines 10 to 12 (Family: none)	10

Form PCT/ISA/210 (continuation of second sheet) (January 2015)

REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 2014118916 A **[0007] [0010] [0024] [0053]**