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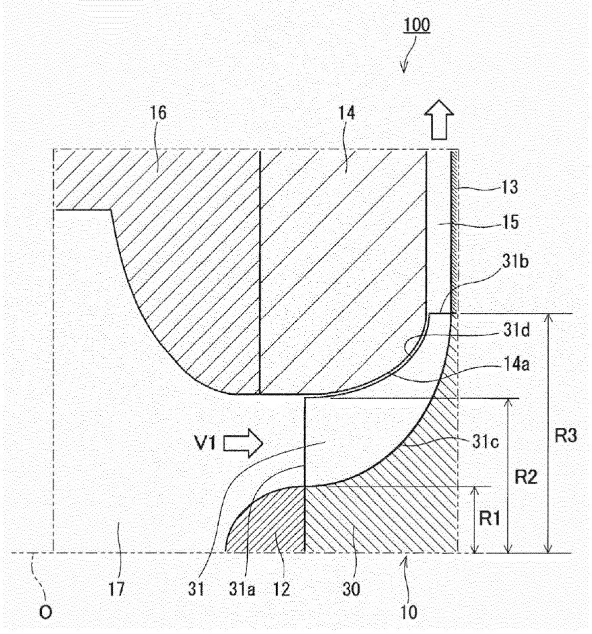
(54) TURBO COMPRESSOR

(57) An impeller of a turbo compressor has a hub and a plurality of blades. When a distance from a rotation axis to an intersection point between a front edge and a hub edge, a distance from the rotation axis to an intersection point between the front edge and a shroud edge, and a distance from the rotation axis to a rear edge are defined as R1, R2, and R3 respectively, the distances R1, R2, and R3 satisfy relations of $0.2 \leq R1/R3 \leq 0.3$ and $0.6 \leq R2/R3 \leq 0.8$. When a dimensionless distance from the front edge with respect to a code direction is defined as M, a blade angle βs is unchanged in a section where the dimensionless distance M is 0% or higher and 5% or lower, and a blade angle βh and the blade angle βs satisfy a relation of $\beta h \geq \beta s \times 2/3$ in a position where the dimensionless distance M is 5%.

FIG. 1



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Description

BACKGROUND

1. Technical Field

[0001] The present disclosure relates to a turbo compressor.

2. Description of the Related Art

[0002] A turbo compressor includes a component referred to as an impeller. The impeller is a rotary component used for sending air to or compressing a compressible fluid and accelerates the fluid mainly in a tangential direction of the rotation to add kinetic energy to the fluid. The impeller generally has a shape close to a truncated cone shape and rotates with an axis being a straight line connecting a center of a small-diameter surface and a center of a large-diameter surface. The impeller has a hub and a plurality of blades. The plurality of blades are radially disposed on a surface of the hub.

[0003] The fluid sucked into the turbo compressor collides with a front edge of a blade at a predetermined angle. This collision generates a velocity difference between a surface (suction surface) of the blade and a surface (pressure surface) of the blade, whereby a kinetic energy is added to the fluid. In a section between the front edge of the blade and a rear edge of the blade, with a turning radius of the impeller increased, a velocity component of the fluid mainly with respect to the tangential direction of the rotation is increased. In a position where the impeller has the maximum outer diameter, the maximum velocity component is obtained and the total amount of the kinetic energy added to the fluid is defined.

[0004] A flow of the fluid in a flow passage between the blades of the impeller is very complicated. In a complicated flow field, a vortex flow whose velocity is low and whose intensity is high (vortex flow having a high vorticity) is generated. The vortex flow inhibits kinetic energy from being effectively added to the fluid from the blades. Furthermore, friction of the fluid in the vortex flow generates losses. This causes lowering of the pressure ratio and adiabatic efficiency.

[0005] As one type of vortex flow, a vortex flow caused by a secondary flow arising from a relative pressure difference on the suction surface is known. The secondary flow indicates a flow having a velocity component perpendicular to the main flow. The secondary flow collides with a shroud wall, separates therefrom, and becomes a vortex flow to disturb the main flow.

[0006] Figs. 8A and 8B each are a graph illustrating a blade angle of an impeller of a turbo compressor disclosed in Japanese Patent No. 3693121. A horizontal axis of each graph indicates a dimensionless distance m of a length of a blade in a main flow direction (length in a code direction). A vertical axis of each graph indicates a blade angle. Fig. 8A illustrates a blade angle βH at a hub edge of the blade. Fig. 8B illustrates a blade angle βS at a shroud edge of the blade. Fig. 8C illustrates a value with respect to a dimensionless distance m (βH-βS).

[0007] Japanese Patent No. 3693121 discloses designing an impeller such that a relative pressure difference ΔCp is decreased as approaching an exit of a flow passage between blades. The relative pressure difference ΔCp is a pressure difference on the suction surface of a blade. When the value of (βH-βS) is increased, the relative pressure difference ΔCp is decreased.

[0008] Also see, for example, Colin Osborne et al., "AERODYNAMIC AND MECHANICAL DESIGN OF AN 8:1 PRESSURE RATIO CENTRIFUGAL COMPRESSOR", NASA CR-134782, April, 1975.

SUMMARY

[0009] Japanese Patent No. 3693121 merely focuses on the value of (βH-βS) near an exit of a flow passage of blades.

[0010] One non-limiting and exemplary embodiment provides techniques of suppressing a secondary flow and a vortex flow arising therefrom to improve efficiency of a turbo compressor.

[0011] In one general aspect, the techniques disclosed here feature a turbo compressor including a casing and an impeller disposed inside the casing, in which the impeller has a hub with an upper surface, a lower surface, and an outer peripheral surface and a plurality of blades radially disposed on the outer peripheral surface of the hub, each of the plurality of blades includes a front edge positioned at a side of the upper surface of the hub, a rear edge positioned at a side of the lower surface of the hub, a hub edge contacting the outer peripheral surface of the hub, and a shroud edge being a blade end positioned at an outer side in a span direction, when a distance from a rotation axis of the impeller to an intersection point between the front edge and the hub edge is defined as R1, a distance from the rotation axis of the impeller to an intersection point between the front edge and the shroud edge is defined as R2, and a distance from the rotation axis of the impeller to the rear edge is defined as R3, the distance R1, the distance R2, and the distance R3 satisfy relations of $0.2 \leq R1/R3 \leq 0.3$ and $0.6 \leq R2/R3 \leq 0.8$, when a dimensionless distance from the front edge with respect to a code direction is defined as M, a blade angle βs at the shroud edge is unchanged in a section where the dimensionless distance M is 0% or higher and 5% or lower, and a blade angle βh at the hub edge and the blade angle βs at the shroud edge satisfy a relation of $\beta h \geq \beta s \times 2/3$ in a position where the dimensionless distance M is 5%.

[0012] According to the techniques disclosed herein, on a suction surface near the front edge, a relative pressure difference in a span direction is decreased. This suppresses a secondary flow and a vortex flow arising therefrom to improve efficiency of a turbo compressor.

[0013] Additional benefits and advantages of the disclosed embodiments will become apparent from the specification and drawings. The benefits and/or advantages may be individually obtained by the various embodiments and features of the specification and drawings, which need not all be provided in order to obtain one or more of such benefits and/or advantages.

BRIEF DESCRIPTION OF THE DRAWINGS

[0014]

Fig. 1 is a meridional plane projection view of a turbo compressor according to one embodiment of the present disclosure;
Fig. 2 is a perspective view of an impeller of the turbo compressor illustrated in Fig. 1;
Fig. 3 is a perspective view of a blade of the impeller illustrated in Fig. 2;
Fig. 4 is a diagram illustrating an identification method of a blade angle;
Fig. 5 is a P-H diagram illustrating a state of a working fluid at an entrance to the impeller;
Fig. 6 is a graph illustrating blade angles βh and βs of the impeller illustrated in Fig. 2;
Fig. 7 is a configuration diagram of a refrigeration cycle apparatus using the turbo compressor illustrated in Fig. 1;
Fig. 8A is a graph illustrating a blade angle βH of an impeller of a turbo compressor disclosed in Japanese Patent No. 3693121;
Fig. 8B is a graph illustrating a blade angle βS of the impeller of the turbo compressor disclosed in Japanese Patent No. 3693121; and
Fig. 8C is a graph illustrating a value of (βH-βS) with respect to a dimensionless distance m.

DETAILED DESCRIPTION

(Underlying Knowledge Forming Basis of the Present Disclosure)

[0015] A substance having a saturated vapor pressure being negative at a normal temperature is used in a refrigeration cycle apparatus as a refrigerant in some cases. The substance whose saturated vapor pressure is negative at a normal temperature includes water. When an impeller of a turbo compressor of a refrigeration cycle apparatus using water is designed with the same specific velocity as an impeller of a turbo compressor of a refrigeration cycle apparatus using a fluorocarbon refrigerant (for example, R134a), a problem described below arises in the turbo compressor of the refrigeration cycle apparatus using water.

[0016] The density of a water vapor is lower than that of a fluorocarbon refrigerant vapor, and the Mach number (ratio of a flow velocity to a sound velocity) of a water vapor at an entrance to the impeller is thus larger than that of a fluorocarbon refrigerant. The Mach number of a water vapor is, for example, 1.3 times the Mach number of R134a. Because the Mach number is large, the width of a static pressure drop of a water vapor at the entrance to the impeller is also large. When the water vapor nearly in a saturated state advances into a flow passage between blades, the water vapor is condensed by a static pressure drop. A water drop may hit a blade to promote erosion and residence of condensed water may clog the flow passage. This may damage the reliability of the turbo compressor or lower the performance of the turbo compressor.

[0017] To prevent condensation of a refrigerant due to a static pressure drop, it is effective to lower the Mach number by increasing the area of the flow passage at the entrance to the impeller. However, increasing the area of the flow passage at the entrance causes an increase in the radius of the impeller and the length of the front edge of a blade. When the length of the front edge is increased, the relative Mach number between the refrigerant flowing near the tip (shroud edge) of the blade and the refrigerant flowing near the root (hub edge) of the blade is expanded. A load applied on the blade at the front edge is excessively increased and a relative pressure difference on the suction surface of the blade is increased. This enhances a secondary flow in a span direction. The secondary flow collides with a shroud wall, separates therefrom, and generates a vortex flow. When an adverse vortex flow is generated at the front edge of the blade, the vortex flow disturbs the main flow while advancing toward the downstream. A low-velocity region disturbing the main flow is extended to a diffuser, inhibiting the efficiency of static pressure recovery at the diffuser. This lowers the performance of the turbo compressor.

[0018] Japanese Patent No. 3693121 focuses on a difference between a blade angle βH and a blade angle βS, but merely focuses on the value of (βH-βS) near an exit of a flow passage of blades. Furthermore, Japanese Patent No. 3693121 only mentions suppressing a secondary flow of a general fluid such as air.

[0019] According to the above-described knowledge of the present inventors, on the suction surface near a front edge of a blade, decreasing a relative pressure difference in a span direction is effective in suppressing a secondary flow and a vortex flow arising therefrom and thus improving the performance of a turbo compressor. Increasing a load applied on the blade at the hub edge while decreasing a load applied on the blade at the shroud edge enables to effectively reduce the relative pressure difference.

[0020] A turbo compressor according to a first aspect of the present disclosure includes:

a casing; and
an impeller disposed inside the casing, in which
the impeller has a hub with an upper surface, a lower surface, and an outer peripheral surface and a plu-

rality of blades radially disposed on the outer peripheral surface of the hub,

each of the plurality of blades includes a front edge positioned at a side of the upper surface of the hub, a rear edge positioned at a side of the lower surface of the hub, a hub edge contacting the outer peripheral surface of the hub, and a shroud edge being a blade end positioned at an outer side in a span direction,

when a distance from a rotation axis of the impeller to an intersection point between the front edge and the hub edge is defined as R1, a distance from the rotation axis of the impeller to an intersection point between the front edge and the shroud edge is defined as R2, and a distance from the rotation axis of the impeller to the rear edge is defined as R3, the distance R1, the distance R2, and the distance R3 satisfy relations of $0.2 \leq R1/R3 \leq 0.3$ and $0.6 \leq R2/R3 \leq 0.8$,

when a dimensionless distance from the front edge with respect to a code direction is defined as M, a blade angle βs at the shroud edge is unchanged in a section where the dimensionless distance M is 0% or higher and 5% or lower, and

a blade angle βh at the hub edge and the blade angle βs at the shroud edge satisfy a relation of $\beta h \geq \beta s \times 2/3$ in a position where the dimensionless distance M is 5%.

[0021] According to the first aspect, the distance R1, the distance R2, and the distance R3 satisfy relations of $0.2 \leq R1/R3 \leq 0.3$ and $0.6 \leq R2/R3 \leq 0.8$, condensation of a working fluid is prevented. Furthermore, according to the first aspect, the blade angle βs at the shroud edge is unchanged in a section where the dimensionless distance M is 0% or higher and 5% or lower. In other words, in a section where the dimensionless distance M is 0% or higher and 5% or lower, the blade shape at the shroud edge is linear. With this configuration, a blade load applied on the shroud edge can be suppressed. Furthermore, when the blade angle βh at the hub edge in a section where the dimensionless distance M is 0% or higher and 5% or lower is adjusted so as to satisfy a relation of $\beta h \geq \beta s \times 2/3$ in a position where the dimensionless distance M is 5%, a load applied on the blade at the hub edge is increased, whereby the width of a static pressure drop on the suction surface at the hub edge is enlarged. This reduces a relative pressure difference in the span direction on the suction surface near the front edge. Consequently, generation of a vortex flow arising from a secondary flow is suppressed, so that the vortex flow disturbing the main flow is suppressed, whereby the performance of the turbo compressor is improved.

[0022] In a second aspect of the present disclosure, for example, the blade angle βs at the shroud edge of the turbo compressor according to the first aspect in a section where the dimensionless distance M exceeds 5% and is 10% or lower is smaller than 0.97 times the blade angle βs at the shroud edge in a position where the dimensionless distance M is 0%, and the blade angle βh and the blade angle βs satisfy a relation of $\beta h \geq \beta s \times 1/2$ in a position where the dimensionless distance M is 10%. According to the second aspect, even under a condition under which the impeller rotates with a higher velocity than a rated design point and a blade load applied on the entire blade is high, a blade load applied on the hub edge can be increased while a blade load applied on the shroud edge is decreased. This reduces a relative pressure difference in the span direction on the suction surface near the front edge, and generation of a secondary flow and a vortex flow arising therefrom is suppressed. This effect is remarkable under a high load condition, and the performance of the turbo compressor is improved.

[0023] In a third aspect of the present disclosure, for example, the impeller of the turbo compressor according to the first or the second aspect compresses a working fluid having a saturated vapor pressure being negative at a normal temperature. The techniques disclosed herein are effective especially for a turbo compressor for compressing a working fluid as described above.

[0024] A refrigeration cycle apparatus according to a fourth aspect of the present disclosure includes:

the turbo compressor according to the first or the second aspect, in which

a substance having a saturated vapor pressure being negative at a normal temperature is used as a refrigerant.

[0025] The techniques disclosed herein are effective especially for a refrigeration cycle apparatus using a substance having a saturated vapor pressure being negative at a normal temperature as a refrigerant.

[0026] In a fifth aspect of the present disclosure, for example, the substance for the refrigeration cycle apparatus according to the fourth aspect contains water. A refrigerant containing water causes less load to environments.

[0027] Embodiments according to the present disclosure will be described below with reference to the drawings. The present disclosure is not limited to the embodiments described below.

[0028] As illustrated in Fig. 1, a turbo compressor 100 in the present embodiment includes an impeller 10, a back plate 13, a shroud 14, and a casing 16. The impeller 10 has a hub 30 and a plurality of blades 31 and is disposed inside a casing 16. Between the back plate 13 and the shroud 14, a diffuser 15 is formed. A working fluid having passed through the impeller 10 flows into the diffuser 15. The shroud 14 has a shroud wall 14a surrounding the impeller 10. The casing 16 forms a suction space 17 for guiding the working fluid to be compressed to the impeller 10. The suction space 17 is an entrance to the impeller 10. To the impeller 10, a nose cone 12 is attached. The shroud 14 may be a part of the casing 16.

[0029] The turbo compressor 100 may be a centrifugal compressor. The techniques disclosed in the present disclosure are also applicable to a diagonal flow compressor.

[0030] The meridional plane projection view in Fig. 1 is a rotated projection view obtained by rotationally projecting a blade 31 and the shroud wall 14a on a meridional plane including a rotation axis O of the impeller 10. A shape represented on the meridional plane projection view is referred to as a "meridional shape" in the field of turbo compressors.

[0031] As illustrated in Fig. 2, the hub 30 of the impeller 10 has an upper surface 30a, a lower surface 30b, and an outer peripheral surface 30c. The hub 30 has a shape close to a truncated cone shape and the diameter thereof is widened from the upper surface 30a to the lower surface 30b smoothly. The plurality of blades 31 are radially disposed on the outer peripheral surface 30c of the hub 30.

[0032] Each of the blades 31 has a pressure surface 31p and a suction surface 31q. The pressure surface 31p is a surface positioned at the front side in a rotation direction D of the impeller 10. The suction surface 31q is a surface positioned at the rear side in the rotation direction D of the impeller 10.

[0033] The blade 31 further has a front edge 31a, a rear edge 31b, a hub edge 31 c, and a shroud edge 31d. The front edge 31a is a blade end positioned at the side of the upper surface 30a of the hub 30 in a code direction. The rear edge 31b is a blade end positioned at the side of the lower surface 30b of the hub 30 in the code direction. The hub edge 31c is a blade end contacting the outer peripheral surface 30c of the hub 30. The shroud edge 31d is a blade end positioned at the outer side in the span direction. In the span direction, the shroud edge 31d is positioned at the opposite side of the hub edge 31c.

[0034] As illustrated in Fig. 3, the span direction is a direction marked with an arrow A. The code direction is a direction marked with an arrow B. The span direction is orthogonal to the code direction.

[0035] The impeller 10 may include a plurality of splitter blades. Each of the plurality of splitter blades is a shorter blade than each of the blades 31, which is a full blade, and may be disposed between the blades 31.

[0036] As illustrated in Fig. 1, a distance from the rotation axis O of the impeller 10 to an intersection point between the front edge 31 a and the hub edge 31c is defined as R1. A distance from the rotation axis O of the impeller 10 to an intersection point between the front edge 31a and the shroud edge 31d is defined as R2. A distance from the rotation axis O of the impeller 10 to the rear edge 31b is defined as R3. The distance R1 is the radius of the hub 30 at the position of the front edge 31a. The distance R2 is the radius of the impeller 10 at the position of the front edge 31a. The distance R3 is the radius of the impeller 10 at the position of the rear edge 31b. The working fluid is guided to the impeller 10 through the suction space 17. A suction area is equal to an area (πx((R2)^2-(R1)^2)) of a plane formed in a donut shape that is defined by a locus of the front edge 31a.

[0037] Next, a relation among the distances R1, R2, and R3 that is required for preventing condensation of the working fluid when the working fluid is a substance having a saturated vapor pressure being negative at a normal temperature will be described.

[0038] The distance R3 is the maximum radius of the impeller 10. As an index representing the size of the impeller 10, a specific velocity Ns is used. When the specific velocity Ns and a peripheral velocity required for the impeller 10 are determined, the distance R3 can be obtained. The turbo compressor 100 has a specific velocity Ns of 0.6 to 0.8, for example. The peripheral velocity is the velocity (m/sec) of the rear edge 31b of a blade 31. The specific velocity Ns is defined by the following formula.

$$\mathrm{Ns} = \frac{N\sqrt{Q}}{\sqrt[3]{H^4}}$$

N: Number of rotations of axis [rpm]

Q: Volumetric flow rate of working fluid (entrance) [m^3/sec]

H: Heat drop (head) [m]

[0039] The distance R1 may be determined from a relation between an eigenvalue of the blade 31 and the number of rotations of the axis. The eigenvalue of the blade 31 is a value related to the strength of the hub 30, the length of the front edge 31 a of the blade 31, and the like. The distance R1 is substantially in a proportional relationship with the distance R3. The distances R1 and R3 satisfy a relation of 0.2≤R1/R3≤0.3, for example.

[0040] When a substance having a saturated vapor pressure being negative at a normal temperature is used as a working fluid, as described above, a problem due to condensation of the working fluid easily arises. In the present embodiment, the distances R2 and R3 satisfy a relation of 0.6≤R2/R3≤0.8. This prevents condensation of a working fluid, improving the reliability of the turbo compressor 100.

[0041] In a general turbo compressor, the distance R2 is adjusted such that a relative velocity between the front edge of a blade and a working fluid is minimized when the working fluid flows into a flow passage between blades. When the distance R2 is increased, the suction area is also increased, and an inflow velocity V1 of the working fluid is decreased. When the distance R2 is increased, a velocity VR2 of the intersection point between the front edge and the shroud edge is also increased. With this, for a synthetic velocity value of the inflow velocity V1 and the velocity VR2 (relative velocity between the front edges of the blades and the working fluid), a minimum value is present.

[0042] For example, in a two-step compression refrigeration cycle using water as a refrigerant, when the evaporation temperature is 6°C and the condensation temperature is 37°C, the saturated pressure in an evaporator is 0.94 kPa and the saturated pressure in a condenser is 6.28 kPa. The pressure ratio is 6.68 (=6.28 kPa/0.94 kPa). The compression ratio per step is approximately 2.58. The peripheral velocity of an impeller required for achieving this pressure ratio may be calculated by adiabatic efficiency and a slip coefficient. When the specific velocity Ns is 0.6 and the ratio (R1/R3) is 0.25, the value of R2/R3 with which the synthetic velocity value of the inflow velocity V1 and the velocity VR2 is minimized is approximately 0.54. This value is unchanged even when the working fluid is a fluorocarbon refrigerant (for example, R134a).

[0043] A mechanism of condensation of a working fluid will be described with reference to Fig. 5. In Fig. 5, the saturated pressure in an evaporator is P0 and the saturation temperature in the evaporator is T0. When it is assumed that the flow passage area in the evaporator is larger enough than that at the entrance to the impeller, the flow velocity of the working fluid in the evaporator can be approximated to zero. In this case, in the evaporator, the total pressure of the working fluid is equal to the static pressure (=P0). When the flow velocity of the working fluid at the entrance to the impeller is V1, the static pressure is decreased to P1 corresponding to the flow velocity. When the pressure of the working fluid is decreased, the state of the working fluid changes along the isentropic line if loss is ignored. Even in a case where the working fluid at the entrance to the impeller has an appropriate superheating degree sh, when the pressure is decreased from P0 to P1, the working fluid changes from a gas state to a gas-liquid two phase state, whereby condensation of the working fluid is generated.

[0044] In the present embodiment, a relation of $0.6 \leq R2/R3 \leq 0.8$ is satisfied. That is to say, the synthetic velocity value of the inflow velocity V1 and the velocity VR2 is larger than the minimum value. With this configuration, condensation of the working fluid may be prevented.

[0045] Next, a blade angle β of a blade 31 of the impeller 10 will be described.

[0046] Firstly, the relative pressure difference on the suction surface of the blade will be described. When it is assumed that the angular difference (angle of incidence) between the relative angle between the working fluid and the front edge 31a and the blade angle of the front edge 31a is unchanged in the span direction, the static pressure of the working fluid on the suction surface 31q is high near the hub edge 31c and low near the shroud edge 31d. The relative angle between the working fluid and the front edge 31a is determined by the inflow velocity V1 of the working fluid and the velocity of the front edge 31a in the circumferential direction.

[0047] In the position of the front edge 31a, the turning radius of the hub edge 31c is equal to the distance R1 and the turning radius of the shroud edge 31d is equal to the distance R2. Because the distance R2 is larger than the distance R1, the velocity of the shroud edge 31d in the circumference direction is higher than the velocity of the hub edge 31c in the circumference direction. The velocity in the circumference direction being high indicates that the relative velocity between the blade 31 and the working fluid is high. That is to say, near the front edge 31a, on the suction surface 31q of the blade 31, a pressure gradient (relative pressure difference) is present along the span direction.

[0048] In the present embodiment, to prevent condensation of the working fluid, the ratio of the distance R2 to the distance R3 (R2/R3) satisfies $0.6 \leq R2/R3 \leq 0.8$. This range is larger than the value of (R2/R3) in a general turbo compressor. With this, the relative pressure difference on the suction surface 31q in the span direction may become large. When the relative pressure difference on the suction surface 31q is increased, a secondary flow in the span direction is enhanced. The secondary flow in the span direction generates a vortex flow disturbing the main flow.

[0049] To suppress the secondary flow in the span direction, the impeller 10 in the present embodiment has a configuration described below. This configuration enables to suppress the secondary flow in the span direction while preventing condensation of the working fluid.

[0050] Fig. 6 illustrates blade angles βh and βs of the impeller 10 in the present embodiments. The vertical axis represents the blade angle β. The horizontal axis represents a dimensionless distance M (%) from the front edge 31a with respect to the code direction. The position of 0% corresponds to the front edge 31a of the blade 31 and the position of 100% corresponds to the rear edge 31b. For example, when the total length of the hub edge 31c (total length of the center line of the blade cross section of the hub edge 31c) is L, the position where the dimensionless distance M at the hub edge 31c is Y% corresponds to the position moved from the front edge 31a to the rear edge 31b along the hub edge 31c by a distance of (L*Y)/100. This also applies to the shroud edge 31d.

[0051] As illustrated in Fig. 6, in the present embodiment, the blade angle β has a negative value. The "blade angle βh" and the "blade angle βs" may be specified by the following method.

[0052] As illustrated in Fig. 4, in the position of a specified dimensionless distance M, a center line 311 of the blade cross section of the hub edge 31c is projected on a projection surface BP perpendicular to a normal line NL of the outer peripheral surface 30c of the hub 30. A reference flat surface H that is parallel to the rotation axis O and includes the normal line NL thereof is projected on the projection surface BP. In the obtained projection surface BP, the angle formed by the center line 311 of the blade cross section of the hub edge 31c and the reference flat surface H is the blade angle βh in the position of the specified dimensionless distance M.

[0053] Similarly, in the position of a specified dimensionless distance M, a center line 312 of the shroud edge 31 d is projected on a projection surface BP perpendicular to a normal line NL of a shroud surface. A reference flat surface H that is parallel to the rotation axis O and includes the normal line NL thereof is projected on the projection surface BP. In the obtained projection surface BP, the angle formed by the center line 312 of the shroud edge 31d and the reference flat surface H is the blade angle βs in the position of the specified dimensionless distance M. The "shroud surface" is a surface defined by a locus of the shroud edge 31d obtained when the impeller 10 is rotated.

[0054] The rotation axis O of the impeller 10 may be present on the reference flat surface H including the normal line NL.

[0055] It is to be noted that the definition of the blade angle disclosed in Japanese Patent No. 3693121 is different from the definition of the blade angle described herein. The blade angle β described herein corresponds to an angle ($\beta=\beta 1-90°$) obtained by subtracting 90° from the blade angle $\beta 1$ disclosed in Japanese Patent No. 3693121.

[0056] As illustrated in Fig. 6, in the impeller 10 in the present embodiment, in a section where the dimensionless distance M is 0% or higher and 5% or lower, the blade angle βs at the shroud edge 31d is unchanged. In other words, in a section where the dimensionless distance M is 0% or higher and 5% or lower, the blade shape at the shroud edge 31d is linear. The "blade angle βs being unchanged" indicates that a change in the blade angle βs is within ± 1 % with respect to the blade angle βs in a position where the dimensionless distance M is 0%.

[0057] As illustrated in Fig. 6, in a section where the dimensionless distance M is 0% or higher and 5% or lower, the blade angle βh at the hub edge 31c is rapidly increased. In a position where the dimensionless distance M is 5%, the blade angle βh at the hub edge 31c and the blade angle βs at the shroud edge 31d satisfy a relation of $\beta h \geq \beta s \times 2/3$. That is to say, in a position where the dimensionless distance M is 5%, an adequate difference is secured between the blade angle βh and the blade angle βs. An upper limit value of the blade angle βh at the hub edge 31c in a position where the dimensionless distance M is 5% is 0 degrees, for example.

[0058] When the blade angle βs at the shroud edge 31d is unchanged, a velocity difference between the pressure surface 31p and the suction surface 31q is less likely generated. This enables to suppress a blade load applied on the shroud edge 31d. When the blade angle βh at the hub edge 31c in a section where the dimensionless distance M is 0% or higher and 5% or lower is adjusted so as to satisfy a relation of $\beta h \geq \beta s \times 2/3$ in a position where the dimensionless distance M is 5%, a blade load applied on the hub edge 31c is increased, whereby the width of a static pressure drop on the suction surface 31q at the hub edge 31c is enlarged. This reduces a relative pressure difference in the span direction on the suction surface 31q near the front edge 31a. Consequently, generation of a vortex flow arising from a secondary flow is suppressed, so that the vortex flow disturbing the main flow is suppressed, whereby the performance of the turbo compressor is improved. It can be thought that a disadvantage (relative pressure difference in the span direction) which may be generated by the ratio (R2/R3) satisfying $0.6 \leq R2/R3 \leq 0.8$ is offset by the adjustment of the blade angles βh and βs.

[0059] As illustrated in Fig. 6, in a section where the dimensionless distance M exceeds 5% and is 10% or lower, the blade angle βs gradually changes. The blade angle βs in a section where the dimensionless distance M exceeds 5% and is 10% or lower is smaller than 0.97 times the blade angle βs in a position where the dimensionless distance M is 0%. In a position where the dimensionless distance M is 10%, the blade angles βh and βs satisfy a relation of $\beta h \geq \beta s \times 1/2$. With this configuration, even under a condition under which the impeller 10 rotates with a higher velocity than a rated design point and a blade load applied on the entire blade 31 is high, a blade load applied on the hub edge 31c can be increased while a blade load applied on the shroud edge 31d is decreased. This reduces a relative pressure difference in the span direction on the suction surface 31q near the front edge 31a, and generation of a secondary flow and a vortex flow arising therefrom is suppressed. This effect is remarkable under a high load condition, and the performance of the turbo compressor 100 is improved.

[0060] The impeller 10 of the turbo compressor 100 in the present embodiment is what is called an open-type impeller. However, the techniques disclosed herein can be applied to a closed-type impeller in which the shroud wall 14a contacts the shroud edge 31d of the impeller 10.

(Embodiment of Refrigeration Cycle Apparatus)

[0061] As illustrated in Fig. 7, a refrigeration cycle apparatus 200 in the present embodiment includes a main circuit 6, a heat absorbing circuit 7, and a heat dissipation circuit 8 for circulating a refrigerant. In the main circuit 6, a liquid refrigerant is charged at a normal temperature. More specifically, as a refrigerant, a substance having a saturated vapor pressure being negative at a normal temperature (Japanese Industrial Standard: 20°C $\pm$15°C/JIS Z8703) is used. This type of refrigerant includes a refrigerant whose main component is water or alcohol. When the refrigeration cycle apparatus 200 is operated, the inside of the main circuit 6 is in a negative pressure state in which the pressure is lower than the atmospheric pressure. The "main component" herein means a component whose mass ratio is the largest.

[0062] The main circuit 6 includes an evaporator 66, a first compressor 61, an intercooler 62, a second compressor 63, a condenser 64, and an expansion valve 65. These devices are connected by a flow passage (piping) in the above-described order.

[0063] The heat absorbing circuit 7 is a circuit for using a refrigerant liquid cooled in the evaporator 66 and includes necessary devices such as a pump 70 and an indoor heat exchanger 71. A part of the heat absorbing circuit 7 is positioned inside the evaporator 66. In the inside of the evaporator 66, a part of the heat absorbing circuit 7 may be positioned in a position upper than the liquid level of the refrigerant liquid, or may be positioned in a position lower than the liquid level of the refrigerant liquid. In the heat absorbing circuit 7, a heating medium such as water or brine is charged.

[0064] The refrigerant liquid accumulated in the evaporator 66 contacts a member (piping) composing the heat absorbing circuit 7. With this, heat exchange is performed between the refrigerant liquid and the heating medium inside the heat absorbing circuit 7 to evaporate the refrigerant liquid. The heating medium inside the heat absorbing circuit 7 is cooled by an evaporation latent heat of the refrigerant liquid. For example, when the refrigeration cycle apparatus 200 is an air conditioner apparatus that performs indoor cooling, the heating medium in the heat absorbing circuit 7 cools the indoor air. The indoor heat exchanger 71 is a fin-tube heat exchanger, for example.

[0065] The heat dissipation circuit 8 is a circuit used for taking heat from the refrigerant inside the condenser 64 and includes necessary devices such as a pump 80 and an outdoor heat exchanger 81. A part of the heat dissipation circuit 8 is positioned inside the condenser 64. More specifically, in the inside of the condenser 64, a part of the heat dissipation circuit 8 is positioned in a position upper than the liquid level of the refrigerant liquid. In the heat dissipation circuit 8, a heating medium such as water or brine is charged. When the refrigeration cycle apparatus 200 is an air conditioner apparatus that performs indoor cooling, the heating medium in the heat dissipation circuit 8 cools the refrigerant in the condenser 64.

[0066] A high-temperature refrigerant vapor discharged from the second compressor 63 contacts a member (piping) composing the heat dissipation circuit 8 inside the condenser 64. With this, heat exchange is performed between the refrigerant vapor and the heating medium inside the heat dissipation circuit 8 to condense the refrigerant vapor. The heating medium inside the heat dissipation circuit 8 is heated by the condensation latent heat of the refrigerant vapor. The heating medium heated by the refrigerant vapor is cooled by the outdoor air or cooling water in the outdoor heat exchanger 81.

[0067] The evaporator 66 is formed of a container having heat insulating property and pressure resistance, for example. The evaporator 66 evaporates the refrigerant liquid in the inside thereof while accumulating the refrigerant liquid. The refrigerant liquid inside the evaporator 66 absorbs heat brought from the outside of the evaporator 66 to boil. That is to say, the refrigerant liquid heated by absorbing heat from the heat absorbing circuit 7 is boiled and evaporated in the evaporator 66. In the present embodiment, the refrigerant liquid accumulated in the evaporator 66 indirectly contacts the heating medium circulating in the heat absorbing circuit 7. That is to say, a part of the refrigerant liquid accumulated in the evaporator 66 is heated by the heating medium in the heat absorbing circuit 7 and used for heating the refrigerant liquid in a saturated state.

[0068] The first compressor 61 and the second compressor 63 compress the refrigerant vapor in two steps. As the first compressor 61, the turbo compressor 100 according to the present embodiment may be used. The second compressor 63 may be a displacement-type compressor independent from the first compressor 61 or may be a turbo compressor coupled to the first compressor 61 with a shaft 11. As the second compressor 63, the turbo compressor 100 according to the present embodiment may be used. An electric motor 67 for rotating the shaft 11 is disposed between the first compressor 61 and the second compressor 63. Along the longitudinal direction of the shaft 11, the first compressor 61, the second compressor 63, and the electric motor 67 may be disposed in this order, or the electric motor 67, the first compressor 61, and the second compressor 63 may be disposed in this order. Because the first compressor 61 and the second compressor 63 are coupled to each other with the shaft 11, the number of components of the first compressor 61 and the second compressor 63 is reduced.

[0069] The intercooler 62 cools the refrigerant vapor discharged from the first compressor 61 before the refrigerant vapor is suctioned by the second compressor 63. The intercooler 62 may be a direct-contact-type heat exchanger or may be an indirect-contact-type heat exchanger.

[0070] The condenser 64 is formed of a container having heat insulating property and pressure resistance, for example. The condenser 64 condenses the refrigerant vapor and accumulates the refrigerant liquid generated by condensing the refrigerant vapor. In the present embodiment, the refrigerant vapor in a superheated state indirectly contacts the heating medium cooled by dissipation of heat into the outside environment to be condensed. That is to say, the refrigerant vapor is cooled by the heating medium in the heat dissipation circuit 8 to be condensed.

[0071] The expansion valve 65 is an example of a pressure reducing mechanism that reduces the condensed refrigerant liquid. The expansion valve 65 may be omitted.

[0072] In the present embodiment, the evaporator 66 and the condenser 64 are indirect-contact-type heat exchangers (for example, a shell tube heat exchanger). However, the evaporator 66 and the condenser 64 may be direct-contact-type heat exchangers. That is to say, the refrigerant liquid may be heated or cooled by being circulated in the heat absorbing circuit 7 and the heat dissipation circuit 8. Furthermore, at least one of the heat absorbing circuit 7 and the heat dissipation circuit 8 may be omitted.

[0073] When the turbo compressor 100 according to the present embodiment is used as the first compressor 61, even if the superheating degree of the refrigerant is relatively small, condensation of the refrigerant is prevented. The temperature of the refrigerant at the entrance (entrance to the impeller 10) of the first compressor 61 may be equal to or lower than the temperature obtained by adding 5°C to the saturation temperature in the evaporator 66. Because the superheating degree of the refrigerant is relatively low, the theoretical power in the compression process can be reduced, whereby the power consumption of the first compressor 61 is reduced.

[0074] The techniques disclosed herein are suitable for a refrigeration cycle apparatus represented by a chiller and a turbo refrigerator. The refrigeration cycle apparatus is used in an air conditioner for business or home, for example.

Claims

1. A turbo compressor comprising:

a casing; and
an impeller disposed inside the casing, wherein the impeller has a hub with an upper surface, a lower surface, and an outer peripheral surface and a plurality of blades radially disposed on the outer peripheral surface of the hub,
each of the plurality of blades includes a front edge positioned at a side of the upper surface of the hub, a rear edge positioned at a side of the lower surface of the hub, a hub edge contacting the outer peripheral surface of the hub, and a shroud edge being a blade end positioned at an outer side in a span direction,
when a distance from a rotation axis of the impeller to an intersection point between the front edge and the hub edge is defined as R1, a distance from the rotation axis of the impeller to an intersection point between the front edge and the shroud edge is defined as R2, and a distance from the rotation axis of the impeller to the rear edge is defined as R3, the distance R1, the distance R2, and the distance R3 satisfy relations of $0.2 \leq R1/R3 \leq 0.3$ and $0.6 \leq R2/R3 \leq 0.8$,
when a dimensionless distance from the front edge with respect to a code direction is defined as M, a blade angle βs at the shroud edge is unchanged in a section where the dimensionless distance M is 0% or higher and 5% or lower, and
a blade angle βh at the hub edge and the blade angle βs at the shroud edge satisfy a relation of $\beta h \geq \beta s \times 2/3$ in a position where the dimensionless distance M is 5%.

2. The turbo compressor according to Claim 1, wherein the blade angle βs at the shroud edge in a section where the dimensionless distance M exceeds 5% and is 10% or lower is smaller than 0.97 times the blade angle βs at the shroud edge in a position where the dimensionless distance M is 0%, and
the blade angle βh and the blade angle βs satisfy a relation of $\beta h \geq \beta s \times 1/2$ in a position where the dimensionless distance M is 10%.

3. The turbo compressor according to Claim 1, wherein the impeller compresses a working fluid having a saturated vapor pressure being negative at a normal temperature.

4. A refrigeration cycle apparatus comprising:

the turbo compressor according to Claim 1, wherein
a substance having a saturated vapor pressure being negative at a normal temperature is used as a refrigerant.

5. The refrigeration cycle apparatus according to Claim 4, wherein
the substance contains water.

FIG. 1

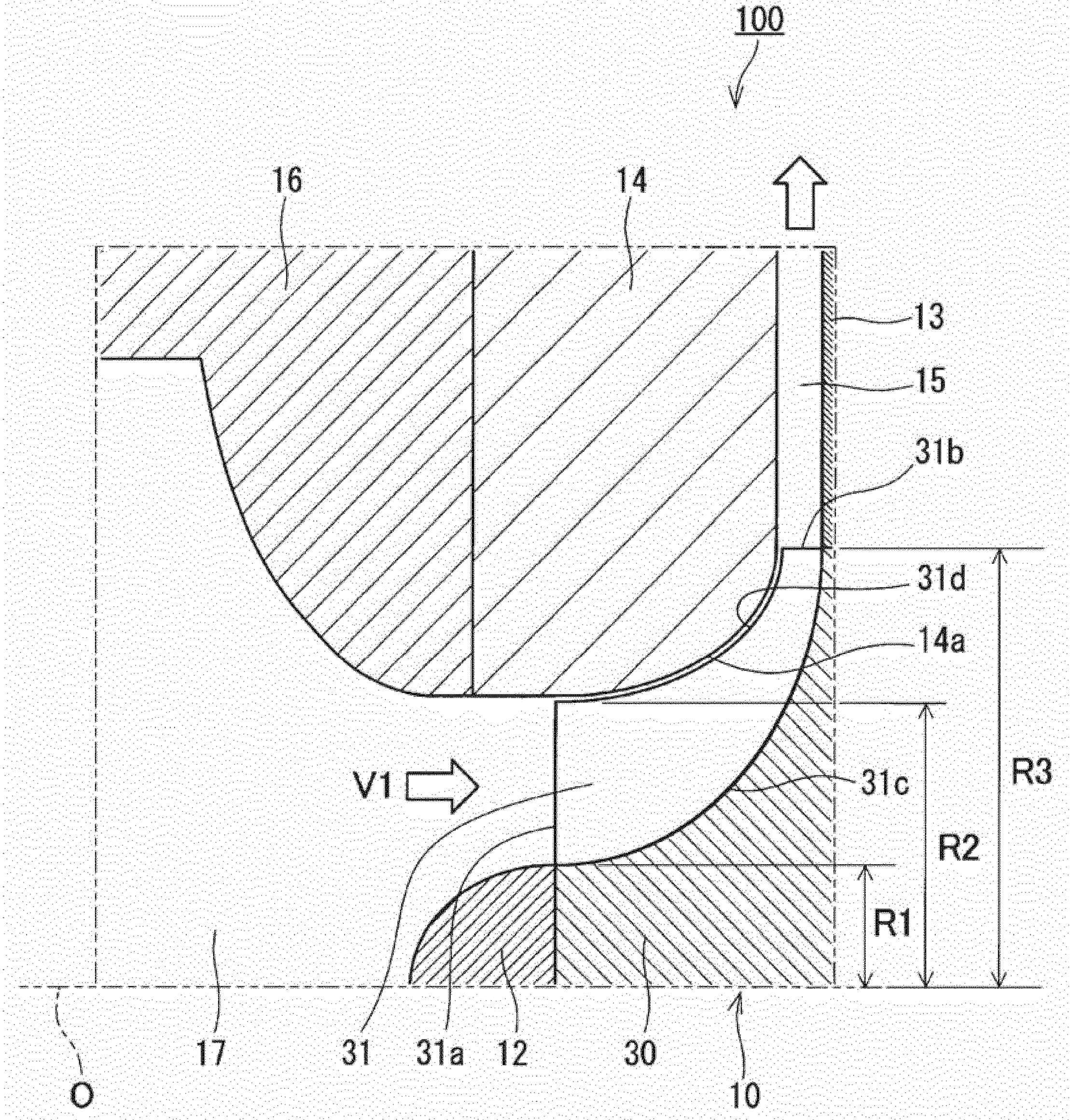


FIG. 2

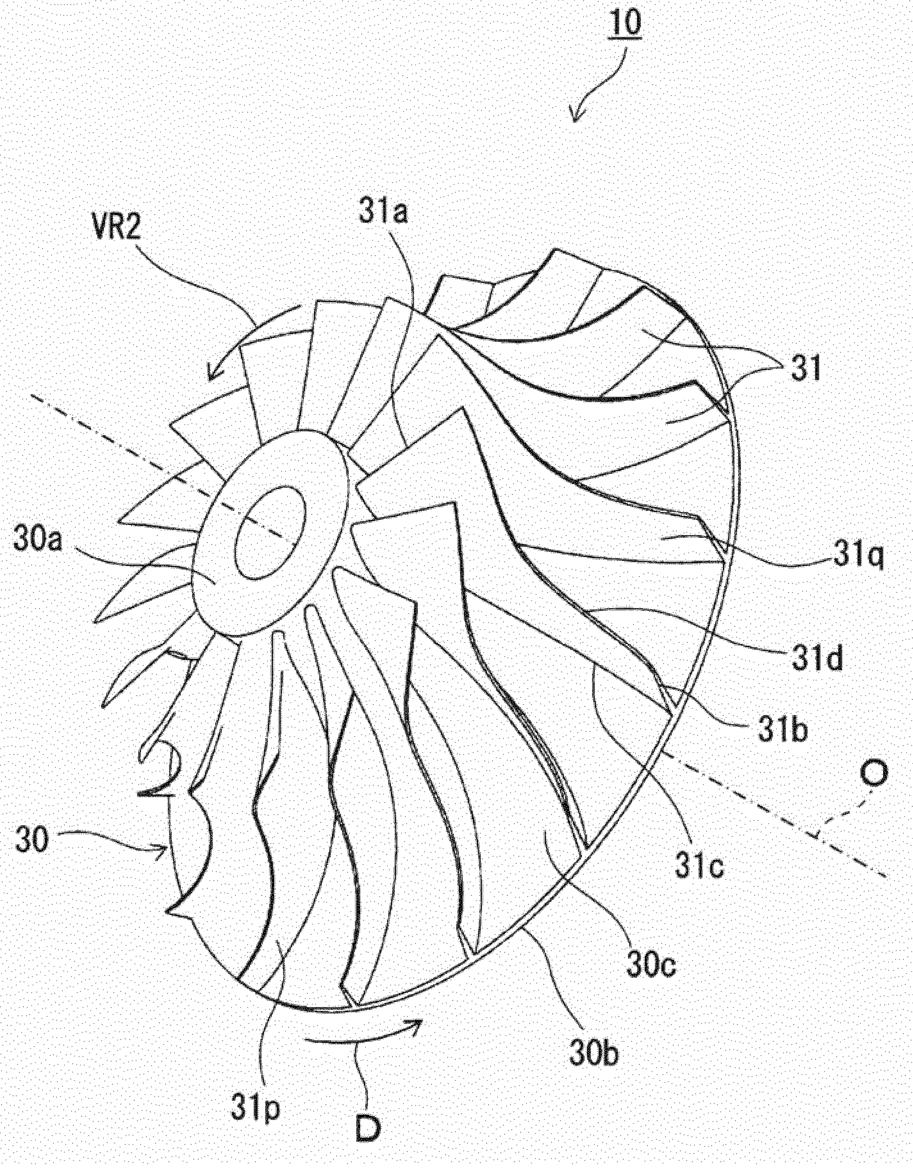


FIG. 3

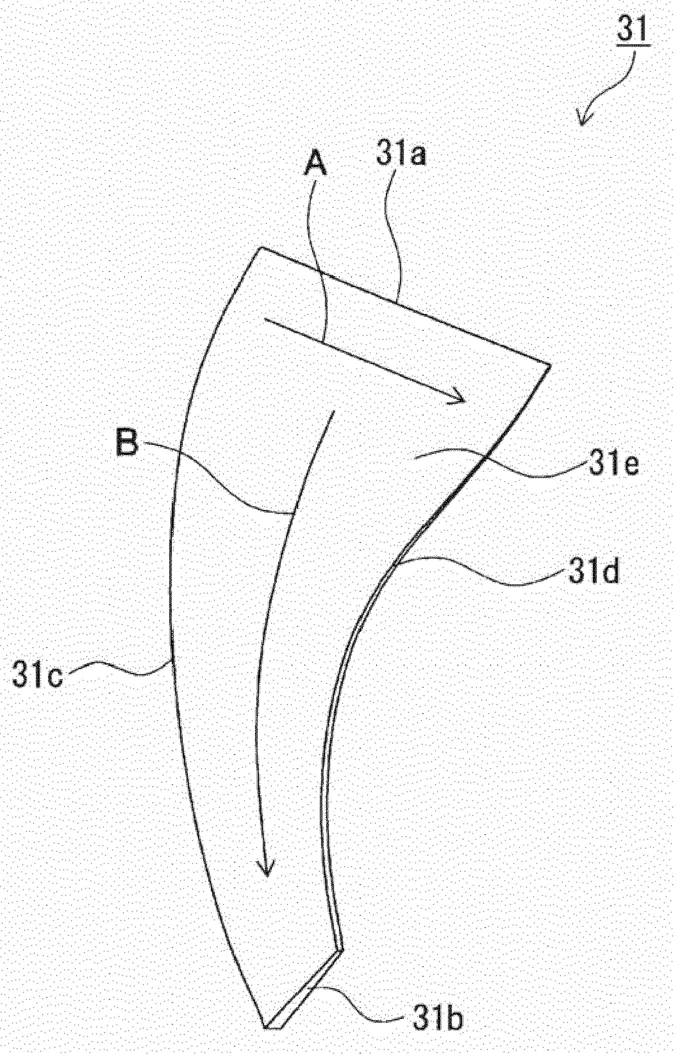


FIG. 4

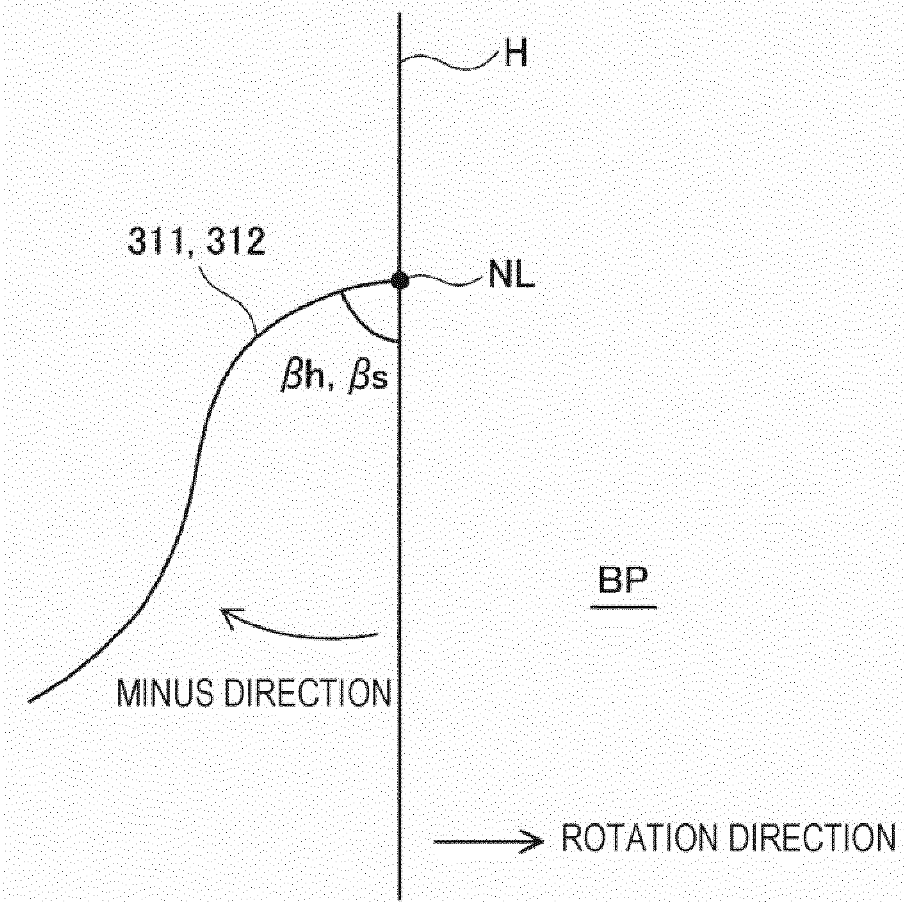
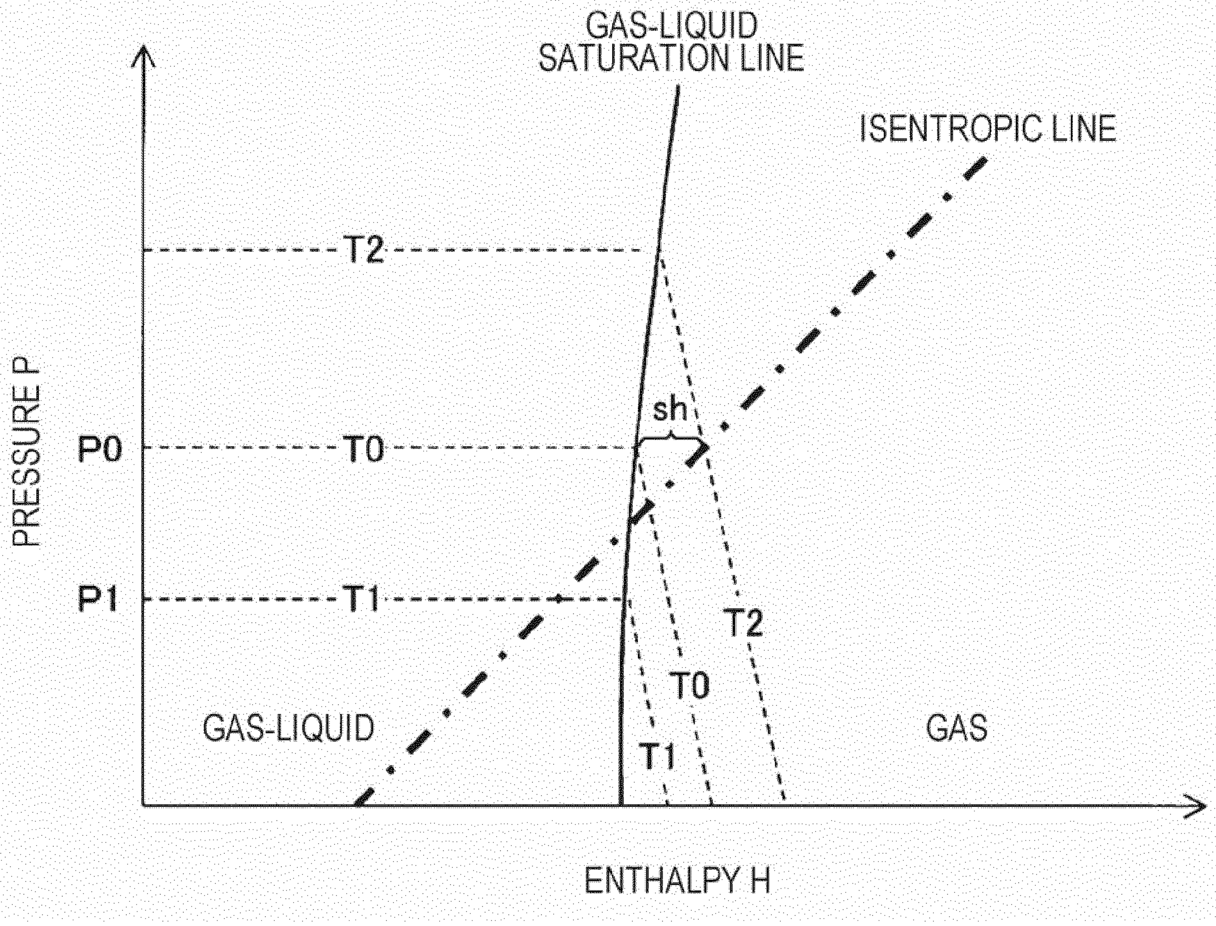


FIG. 5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SATURATION LINE
ISENTROPIC LINE
PRESSURE P
P0
P1
T2
T0
T1
sh
T2
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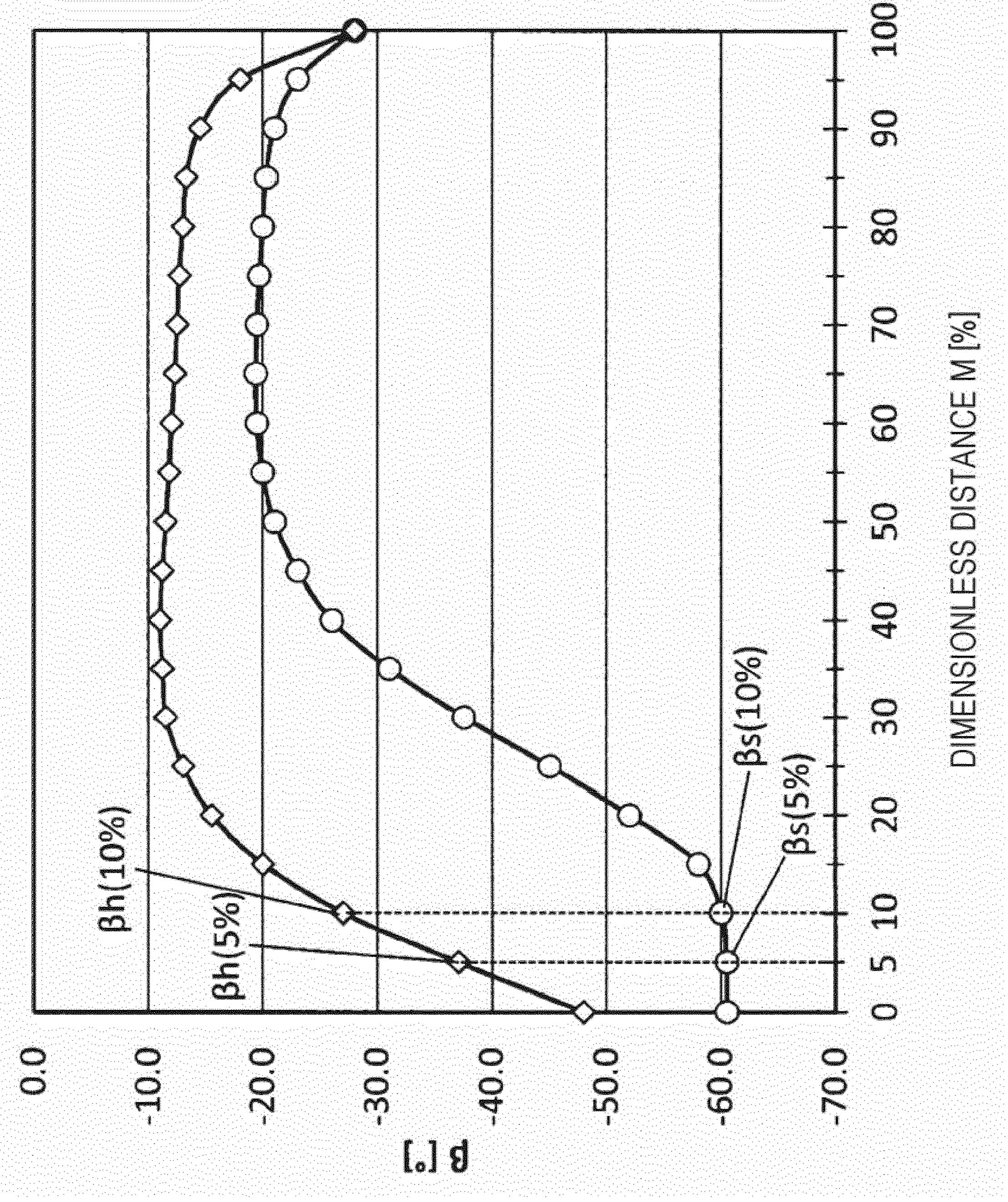
T1
GAS-LIQUID
GAS
ENTHALPY H

FIG. 6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βh(5%)
βs(10%)
βs(5%)
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-30.0
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-50.0
-60.0
-70.0
β [°]
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20
30
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50
60
70
80
90
100
DIMENSIONLESS DISTANCE M [%]

FIG. 7

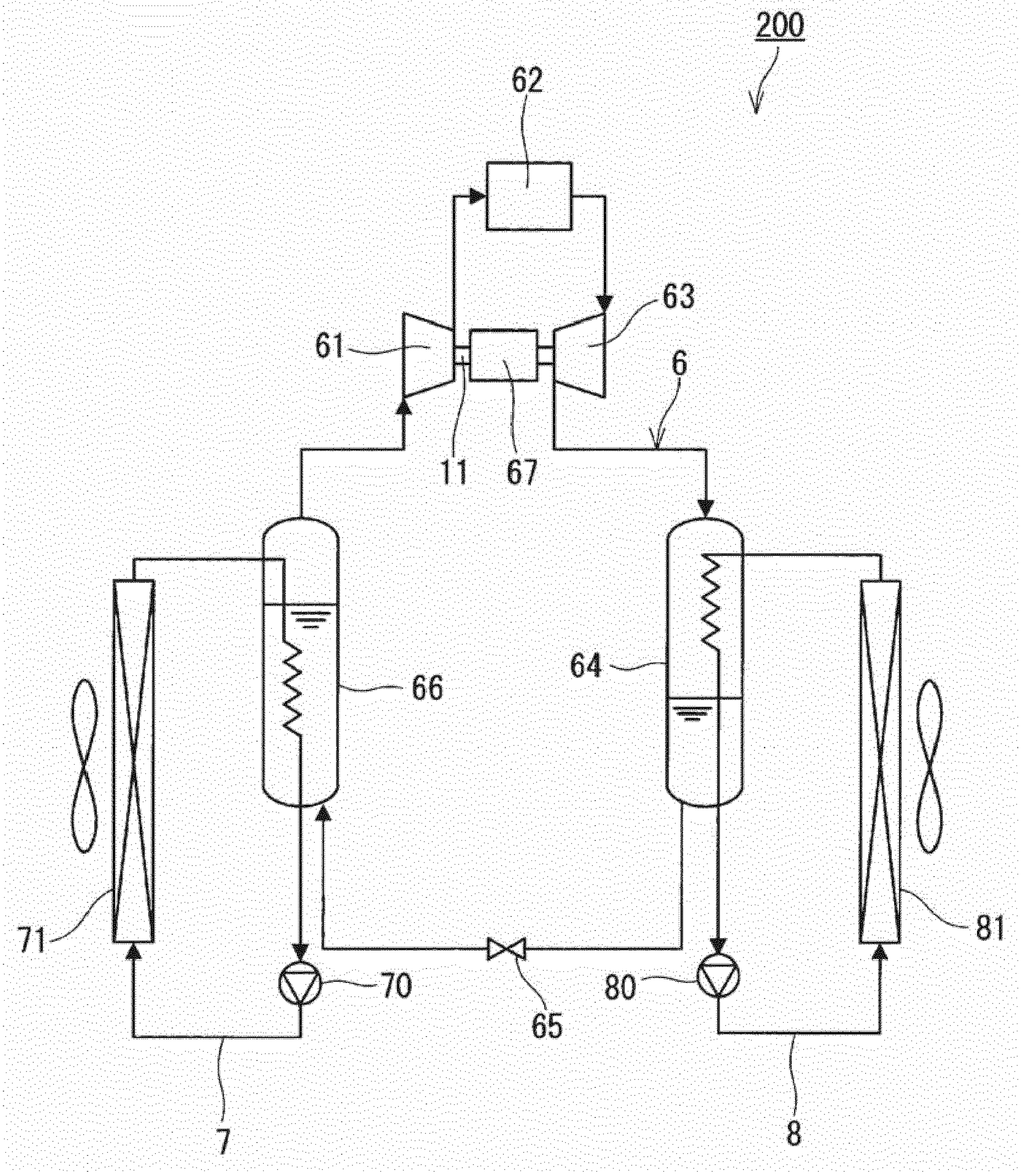
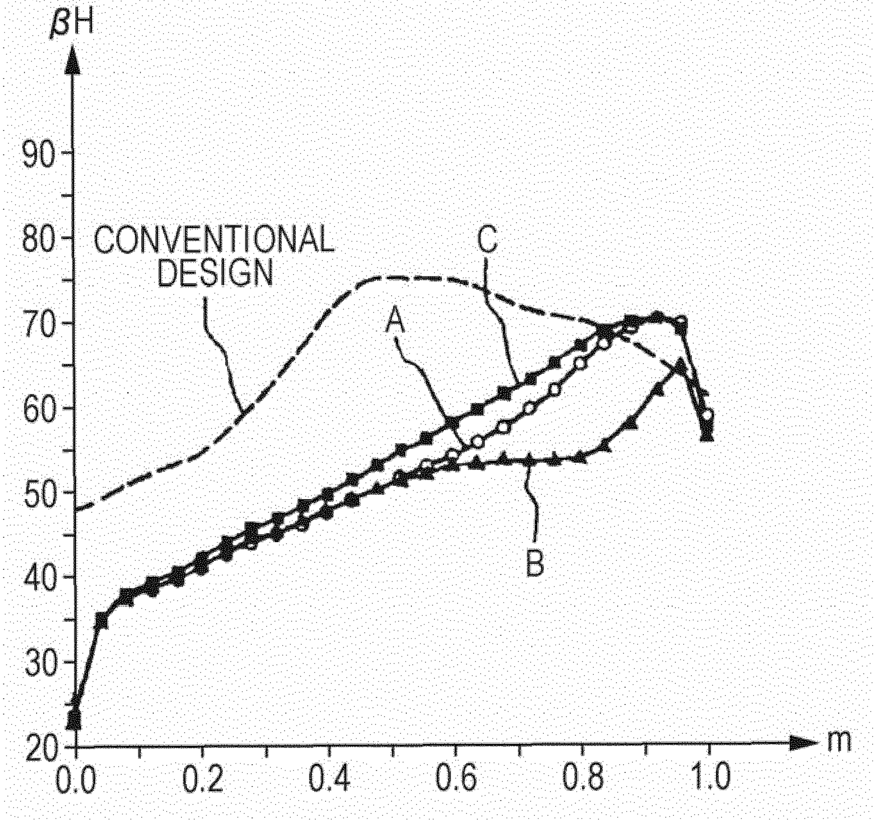


FIG. 8A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m
CONVENTIONAL DESIGN
C
A
B

FIG. 8B

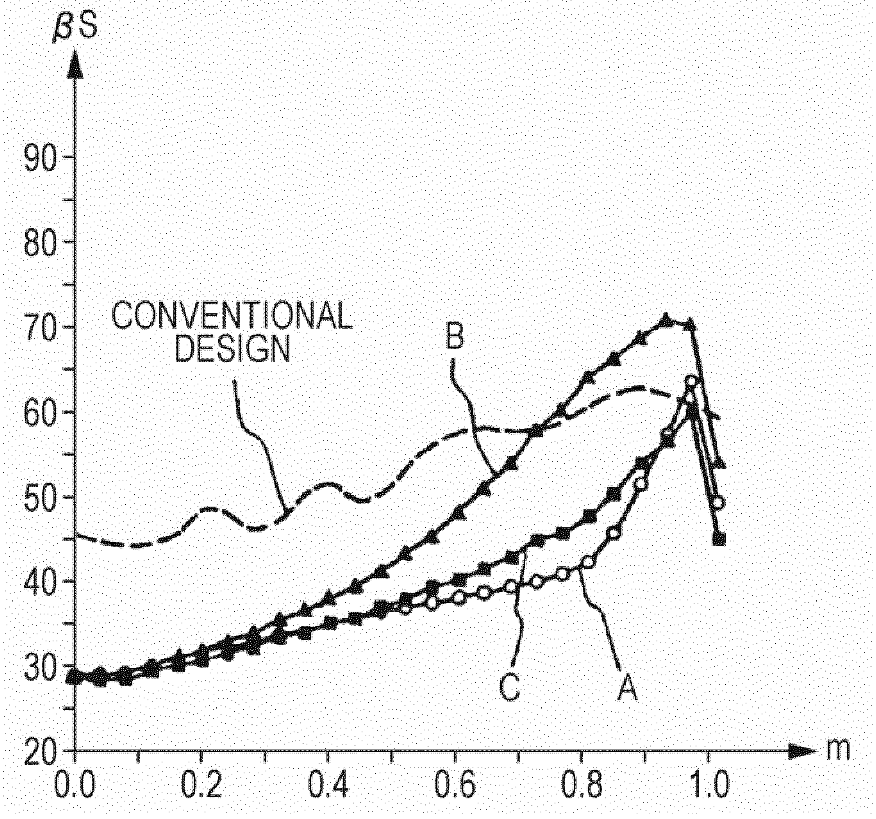
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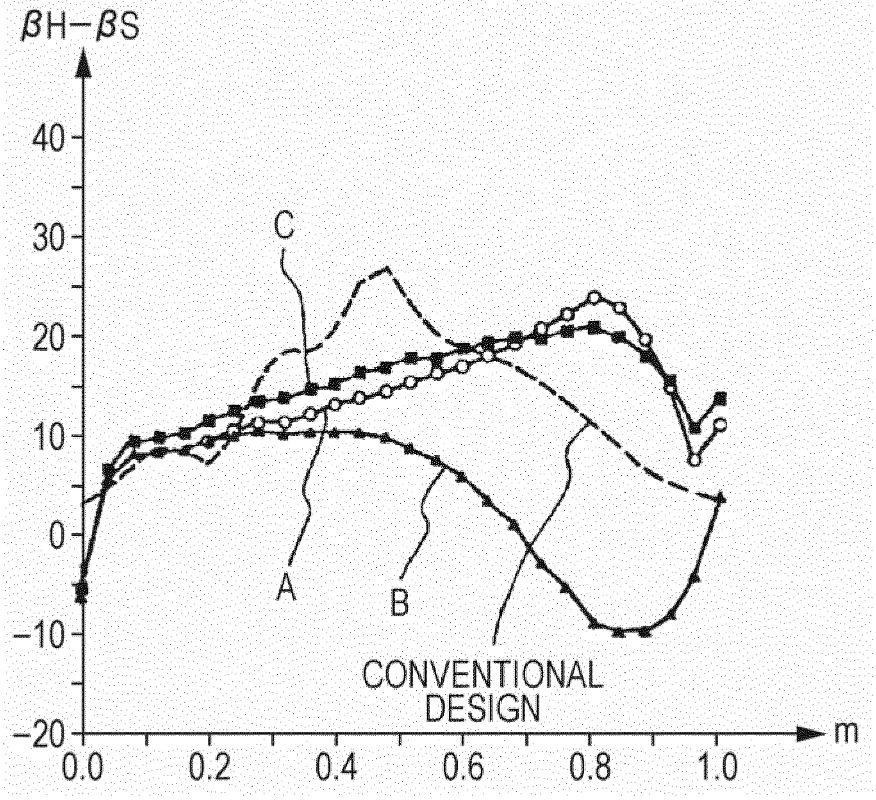
FIG. 8C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m
C
A
B
CONVENTIONAL
DESIGN

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EUROPEAN SEARCH REPORT

Application Number
EP 18 15 7880

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F04D

The present search report has been drawn up for all claims

1

Place of search	Date of completion of the search	Examiner
The Hague	13 July 2018	Hermens, Sjoerd

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