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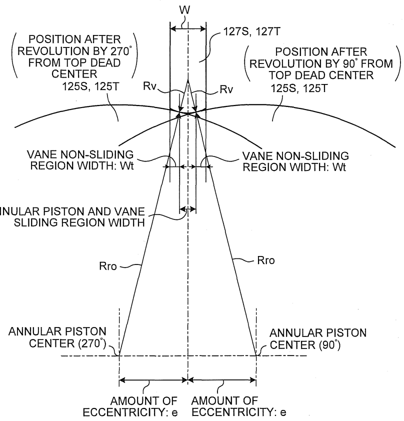
(54) **ROTARY COMPRESSOR**

(57) In a rotary compressor, when it is assumed that a vane width is W, the amount of eccentricity of an eccentric portion is e, a vane leading end curvature radius is R_v, an annular piston radius is R_{ro}, and a non-sliding region width on each of both side portions of a vane leading end is W_t, the vane width W and the vane leading end curvature radius R_v are set such that the non-sliding region width W_t on each of both the side portions of the vane leading end defined by the following equation (A) is a value satisfying an equation (B):

$$W_t = (W/2) - e \times R_v/(R_v + R_{ro}) \quad (A)$$

$$0.3\ \text{mm} \le W_t \le 0.6\ \text{mm} \quad (B).$$

FIG.4



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Description

[Technical Field]

[0001] The present invention relates to a rotary compressor that is used for air conditioning machines, freezer machines, and the like.

[Background Art]

[0002] As illustrated in FIG. 3, a vane leading end of a rotary compressor is formed into an arc surface having a vane leading end curvature radius R_v. Abutment of a ridge line portion formed by intersection of the arc surface and a vane side surface against the outer circumferential surface of a roller (annular piston) causes abnormal abrasion of the roller. As illustrated in FIG. 4, a non-sliding (non-contact) region width W_t of a vane is minimum at positions to which the roller revolves from a top dead center by 90° and 270° when a position at which the roller is located at the top dead center is assumed to be 0°. Conventionally, in order to prevent the abnormal abrasion of the roller due to the abutment of the ridge line portion of the vane against the outer circumferential surface of the roller, the vane leading end curvature radius R_v is decreased, a vane width W is increased (for example, W = 4 mm), and the non-sliding region width W_t of the vane is set to 0.8 mm to 1.0 mm.

[0003] In an abutment portion (roller and vane sliding region) between the vane leading end and the roller, under a condition where a pressure ratio between the low-pressure side and the high-pressure side of refrigerant gas is high, for example, at the time of heating when the outside air temperature is low, a gas temperature at the high-pressure side of the refrigerant gas is increased and a gas flow rate is decreased. For this reason, the temperature of the vane leading end becomes high and it is therefore difficult to form an oil film. In particular, with an R32 refrigerant having a lower gas density and a higher discharge temperature than those of an R410A refrigerant, the temperature of the sliding surfaces of the vane and the roller becomes much higher than that with the R410A refrigerant. This high temperature causes a problem in that the abnormal abrasion between the vane and the roller occurs, preventing reliability from being ensured.

[0004] As a rotary compressor for solving the above-mentioned problem, conventionally disclosed is a rotary compressor including a cylinder that has a suction port and a discharge port, a rotating axis that has a crank portion arranged on a cylinder axis line, a roller that is arranged between the crank portion and the cylinder and eccentrically rotates, and a vane that reciprocates in a groove provided in the cylinder and makes contact with the outer circumferential surface of the roller. In this rotary compressor, in a case where a position at which the vane starts reciprocating relative to the roller is set to a reference of the rotation angle of the roller, a contact surface of the vane with the roller has a curvature of equal to or smaller than a curvature of the roller when the roller is located at a rotation angle of close to 90° or close 270° or both of them (for example, refer to Patent Document 1).

[Citation List]

[Patent Citation]

[0005] Patent Document 1: Japanese Laid-open Patent Publication No. 07-229488

[Summary of Invention]

[Technical Problem]

[0006] The conventional technique disclosed in Patent Document 1 however has a problem in that production management is complicated because a plurality of surfaces having different curvatures form a vane leading end surface. Furthermore, a connecting portion between the surface of the vane leading end surface that has a curvature radius R and the surface thereof that has a curvature of equal to or smaller than the curvature of the roller needs to have a curvature radius smaller than the curvature radius R of the vane leading end surface. A hertz stress of the connection portion is therefore increased and abnormal abrasion can occur on the outer circumferential portion of the roller.

[0007] The present invention has been made in view of the above-mentioned circumstances and an object of the present invention is to provide a rotary compressor in which a vane leading end surface has a simple shape without being formed by a plurality of surfaces having different curvatures and the vane and a roller (annular piston) have high durability.

[Solution to Problem]

[0008] To solve the above mentioned problem and attain the purpose, the present invention is characterized in that a rotary compressor includes: a vertically placed and sealed compressor housing including a discharge portion of a refrigerant on an upper portion and a suction portion of the refrigerant on a lower side surface; a compression unit arranged on a lower portion of the compressor housing and including an annular cylinder, an end plate that has a bearing portion and a discharge valve portion and closes one end portion of the cylinder, an end plate or an intermediate partition plate that has a bearing portion and closes the other end portion of the cylinder, an annular piston that is fitted into an eccentric portion of a rotating axis supported on the bearing portion, revolves in the cylinder along a cylinder inner wall of the cylinder, and forms an operation chamber between the annular piston and the cylinder inner wall, and a vane that projects into the operation chamber from a vane groove provided in the cylinder, abuts against the annular piston, and divides the operation chamber into a suction chamber and a compression chamber; and a motor that is arranged on an upper portion of the compressor housing and drives the compression unit through the rotating axis, the rotary compressor sucking the refrigerant through the suction portion and discharging the refrigerant after passing through the compressor housing from the discharge portion, wherein when it is assumed that a vane width is W, the amount of eccentricity of the eccentric portion is e, a vane leading end curvature radius is R_v, an annular piston radius is R_{ro}, and a non-sliding region width on each of both side portions of a vane leading end is W_t, the vane width W and the vane leading end curvature radius R_v are set such that the non-sliding region width W_t on each of both the side portions of the vane leading end defined by the following equation (A) is a value satisfying an equation (B):

$$W_t = (W/2) - e \times R_v/(R_v + R_{ro}) \qquad (A)$$

$$0.3\ \text{mm} \leq W_t \leq 0.6\ \text{mm} \qquad (B).$$

[Advantageous Effects of Invention]

[0009] The present invention provides an advantageous effect in which a rotary compressor including a highly durable vane and roller (annular piston) is provided that does not need an increase in the width of the vane or a decrease in the amount of eccentricity of the roller.

[Brief Description of Drawings]

[0010]

FIG. 1 is a longitudinal sectional view illustrating a rotary compressor to which the present invention is applied.
FIG. 2 is a transverse sectional view of first and second compression units when seen from the above.
FIG. 3 is a partial enlarged view of FIG. 2.
FIG. 4 is a partial enlarged view of FIG. 3.

[Embodiments for Carrying Out the Invention]

[0011] Hereinafter, an embodiment of a rotary compressor according to the present invention will be described in detail with reference to the accompanying drawings. Note that the embodiment does not limit the present invention. Embodiment

[0012] FIG. 1 is a longitudinal sectional view illustrating an embodiment of a rotary compressor according to the present invention. FIG. 2 is a transverse sectional view of first and second compression units in the embodiment when seen from the above.

[0013] As illustrated in FIG. 1, a rotary compressor 1 in the embodiment includes a compression unit 12 arranged on a lower portion of a cylindrical compressor housing 10 that is sealed and vertically placed, and a motor 11 that is arranged on an upper portion of the compressor housing 10 and drives the compression unit 12 through a rotating axis 15.

[0014] A stator 111 of the motor 11 is formed into a cylindrical shape and is shrink-fitted and fixed to the inner circumferential surface of the compressor housing 10. A rotor 112 of the motor 11 is arranged in the cylindrical stator 111 and is shrink-fitted and fixed to the rotating axis 15 mechanically connecting the motor 11 and the compression unit 12.

[0015] The compression unit 12 includes a first compression unit 12S, and a second compression unit 12T arranged in parallel with the first compression unit 12S and placed above the first compression unit 12S. As illustrated in FIG. 2, the first and the second compression units 12S and 12T include first and second annular cylinders 121S and 121T, respectively. The first and the second cylinders 121S and 121T include first and second suction holes 135S and 135T and first and second vane grooves 128S and 128T provided on first and second lateral expanding portions 122S and 122T in a radial manner, respectively.

[0016] As illustrated in FIG. 2, the first and the second cylinders 121S and 121T have respective first and second cylinder inner walls 123S and 123T having circular shapes that are formed concentrically with the rotating axis 15 of the motor 11. First and second annular pistons 125S and 125T having outer diameters smaller than the inner diameters of the cylinders are arranged at the inner side of the first and the second cylinder inner walls 123S and 123T, respectively. First and second operation chambers 130S and 130T for sucking, compressing, and discharging refrigerant gas are formed between the first and the second cylinder inner walls 123S and 123T and the first and the second annular pistons 125S and 125T, respectively.

[0017] The first and the second cylinders 121S and 121T include the first and the second vane grooves 128S and 128T formed from the first and the second cylinder inner walls 123S and 123T in the radial direction over entire height regions of the cylinders, respectively. First and second vanes 127S and 127T each formed into a flat plate shape are fitted into the first and the second vane grooves 128S and 128T in a slidable manner, respectively.

[0018] As illustrated in FIG. 2, first and second spring holes 124S and 124T are formed in deep portions of the first and the second vane grooves 128S and 128T so as to communicate with the first and the second vane grooves 128S and 128T from outer circumferential portions of the first and the second cylinders 121S and 121T, respectively. First and second vane springs (not illustrated) pressing the rear surfaces of the first and the second vanes 127S and 127T are inserted into the first and the second spring holes 124S and 124T, respectively.

[0019] When the rotary compressor 1 is activated, the first and the second vanes 127S and 127T project into the first and the second operation chambers 130S and 130T from the first and the second vane grooves 128S and 128T with repulsion of the first and the second vane springs, respectively. With the projection, the leading ends of the first and the second vanes 127S and 127T abut against the outer circumferential surfaces of the first and the second annular pistons 125S and 125T, respectively. Thus, the first and the second vanes 127S and 127T divide the first and the second operation chambers 130S and 130T into first and second suction chambers 131S and 131T and first and second compression chambers 133S and 133T, respectively.

[0020] Furthermore, the first and the second cylinders 121S and 121T include first and second pressure introduction paths 129S and 129T communicating the deep portions of the first and the second vane grooves 128S and 128T and an inner portion of the compressor housing 10 through openings R illustrated in FIG. 1, respectively. The first and the second pressure introduction paths 129S and 129T are used to introduce the compressed refrigerant gas in the compressor housing 10 and apply back pressure to the first and the second vanes 127S and 127T with pressure of the refrigerant gas, respectively.

[0021] The first and the second cylinders 121S and 121T have the first and the second suction holes 135S and 135T communicating the first and the second suction chambers 131S and 131T with the outside in order to suck the refrigerant into the first and the second suction chambers 131S and 131T from the outside, respectively.

[0022] As illustrated in FIG. 1, an intermediate partition plate 140 is arranged between the first cylinder 121S and the second cylinder 121T to partition and close the first operation chamber 130S (refer to FIG. 2) of the first cylinder 121S and the second operation chamber 130T (refer to FIG. 2) of the second cylinder 121T. The intermediate partition plate 140 closes an upper end portion of the first cylinder 121S and a lower end portion of the second cylinder 121T. A lower end plate 160S is arranged on a lower end portion of the first cylinder 121S and closes the first operation chamber 130S of the first cylinder 121S. An upper end plate 160T is arranged on an upper end portion of the second cylinder 121T and closes the second operation chamber 130T of the second cylinder 121T. The lower end plate 160S closes the lower end portion of the first cylinder 121S and the upper end plate 160T closes the upper end portion of the second cylinder 121T.

[0023] A sub bearing portion 161S is formed on the lower end plate 160S and a sub axis portion 151 of the rotating axis 15 is supported on the sub bearing portion 161S in a rotatable manner. A main bearing portion 161T is formed on the upper end plate 160T and a main axis portion 153 of the rotating axis 15 is supported on the main bearing portion 161T in a rotatable manner.

[0024] The rotating axis 15 includes a first eccentric portion 152S and a second eccentric portion 152T that are made eccentric to each other while shifting phases thereof by 180°. The first eccentric portion 152S is fitted into the first annular piston 125S of the first compression unit 12S in a rotatable manner and the second eccentric portion 152T is fitted into the second annular piston 125T of the second compression unit 12T in a rotatable manner.

[0025] The rotation of the rotating axis 15 causes the first and the second annular pistons 125S and 125T to revolve in the clockwise direction in FIG. 2 in the first and the second cylinders 121S and 121T along the first and the second cylinder inner walls 123S and 123T, respectively. Following the revolution, the first and the second vanes 127S and 127T reciprocate. The volumes of the first and the second suction chambers 131S and 131T and the first and the second compression chambers 133S and 133T continuously change with the movements of the first and the second annular pistons 125S and 125T and the first and the second vanes 127S and 127T, respectively, and the compression unit 12 continuously sucks, compresses, and discharges the refrigerant gas.

[0026] As illustrated in FIG. 1, a lower muffler cover 170S is arranged at the lower side of the lower end plate 160S and a lower muffler chamber 180S is formed between the lower muffler cover 170S and the lower end plate 160S. The first compression unit 12S opens to the lower muffler chamber 180S. That is to say, a first discharge hole 190S (refer to FIG. 2) communicating the first compression chamber 133S of the first cylinder 121S and the lower muffler chamber 180S is provided in the lower end plate 160S in the vicinity of the first vane 127S. A reed valve-type first discharge valve 200S for preventing reverse flow of the compressed refrigerant gas is arranged in the first discharge hole 190S.

[0027] The lower muffler chamber 180S is one chamber formed in an annular form and is a part of a communication path communicating the discharge side of the first compression unit 12S into an upper muffler chamber 180T through refrigerant passages 136 (refer to FIG. 2) penetrating through the lower end plate 160S, the first cylinder 121S, the intermediate partition plate 140, the second cylinder 121T, and the upper end plate 160T. The lower muffler chamber 180S reduces pressure pulsation of the discharged refrigerant gas. A first discharge valve presser 201S for restricting the amount of deflected valve opening of the first discharge valve 200S is fixed together with the first discharge valve 200S with a rivet while being superimposed with the first discharge valve 200S. The first discharge hole 190S, the first discharge valve 200S, and the first discharge valve presser 201S configure a first discharge valve portion of the lower end plate 160S.

[0028] As illustrated in FIG. 1, an upper muffler cover 170T is arranged at the upper side of the upper end plate 160T and the upper muffler chamber 180T is formed between the upper muffler cover 170T and the upper end plate 160T. A second discharge hole 190T (refer to FIG. 2) communicating the second compression chamber 133T of the second cylinder 121T and the upper muffler chamber 180T is provided in the upper end plate 160T in the vicinity of the second vane 127T and a reed valve-type second discharge valve 200T for preventing reverse flow of the compressed refrigerant gas is arranged in the second discharge hole 190T. Furthermore, a second discharge valve presser 201T for restricting the amount of deflected valve opening of the second discharge valve 200T is fixed together with the second discharge valve 200T with a rivet while being superimposed with the second discharge valve 200T. The upper muffler chamber 180T reduces pressure pulsation of the discharged refrigerant gas. The second discharge hole 190T, the second discharge valve 200T, and the second discharge valve presser 201T configure a second discharge valve portion of the upper end plate 160T. Although not illustrated in the drawings, when the rotary compressor is of a single cylinder type, end plates close the upper and the lower end portions of the cylinder and no discharge valve portion may be provided on the end plate closing the lower end portion of the cylinder.

[0029] The first cylinder 121S, the lower end plate 160S, the lower muffler cover 170S, the second cylinder 121T, the upper end plate 160T, the upper muffler cover 170T, and the intermediate partition plate 140 are integrally fastened by a plurality of insertion bolts 175 and the like. In the compression unit 12 integrally fastened by the insertion bolts 175 and the like, the outer circumferential portion of the upper end plate 160T is firmly fixed to the compressor housing 10 by spot welding and the compression unit 12 is thereby fixed to the compressor housing 10.

[0030] First and second through-holes 101 and 102 are provided in the outer circumferential wall of the cylindrical compressor housing 10 in this order from the lower side so as to be separated from each other in the axis direction. First and second suction pipes 104 and 105 are inserted through the first and the second through-holes 101 and 102, respectively. An accumulator holder 252 and an accumulator band 253 hold an accumulator 25 formed by an independent cylindrical sealed container at the outside of the compressor housing 10.

[0031] A system connection pipe 255 that is connected to an evaporator of a refrigerant circuit is connected to the center of a top portion of the accumulator 25. First and second low-pressure communication pipes 31S and 31T one ends of which extend to an upper portion in the accumulator 25 and the other ends of which extend to the other ends of the first and the second suction pipes 104 and 105 are connected to bottom through-holes 257 provided in a bottom portion of the accumulator 25.

[0032] The first and the second low-pressure communication pipes 31S and 31T for introducing the low-pressure refrigerant in the refrigerant circuit to the first and the second compression units 12S and 12T through the accumulator 25 are connected to the first and the second suction holes 135S and 135T (refer to FIG. 2) of the first and the second cylinders 121S and 121T through the first and the second suction pipes 104 and 105 as suction portions, respectively. That is to say, the first and the second suction holes 135S and 135T are connected to the evaporator of the refrigerant circuit in parallel.

[0033] A discharge pipe 107 as a discharge portion that is connected to the refrigerant circuit and discharges high-pressure refrigerant gas to the condenser side of the refrigerant circuit is connected to a top portion of the compressor housing 10. That is to say, the first and the second discharge holes 190S and 190T are connected to the condenser of the refrigerant circuit.

[0034] Lubricant oil is enclosed to approximately the height of the second cylinder 121T in the compressor housing 10. The lubricant oil is sucked from an oil supply pipe 16 mounted on a lower end portion of the rotating axis 15 with a pump blade (not illustrated) that is inserted into a lower portion of the rotating axis 15. The lubricant oil circulates in the compression unit 12, lubricates sliding components, and seals fine gaps in the compression unit 12.

[0035] The characteristic configuration of the rotary compressor 1 according to the embodiment will be described with reference to FIG. 3 and FIG. 4. FIG. 3 is a partial enlarged view of FIG. 2 and FIG. 4 is a partial enlarged view of FIG. 3. As illustrated in FIG. 3 and FIG. 4, when the first and the second vanes 127S and 127T are pressed against the first and the second annular pistons 125S and 125T, respectively, by vane pressing force P with back pressure, maximum contact stress σ_{max}, which is expressed by the following equations (1) and (2), is generated:

$$a = 2\sqrt{\frac{P}{\pi}\left(\frac{1-\nu_V{}^2}{E_V} + \frac{1-\nu_{ro}{}^2}{E_{ro}}\right)\Big/\left(\frac{1}{R_V} + \frac{1}{R_{ro}}\right)} \qquad (1)$$

$$\sigma_{max} = 2P/(\pi a) \qquad (2)$$

where σ_{max} is the maximum contact stress, a is a contact width, P is the vane pressing force, R_v is a vane leading end curvature radius, R_{ro} is an annular piston radius, E_v is a vane elastic modulus, E_{ro} is an annular piston elastic modulus, ν_v is a vane Poisson's ratio, and ν_{ro} is an annular piston Poisson's ratio.

[0036] A non-sliding region width W_t on each of both side portions of the leading ends of the first and the second vanes 127S and 127T is expressed by the following equation (A) (refer to dimensional relation of a similar triangle in FIG. 4):

$$W_t = (W/2) - e \times R_v/(R_v + R_{ro}) \qquad (A)$$

where W_t is the non-sliding region width on each of both the side portions of the leading end of the vane, W is the vane width, and e is the amount of eccentricity of the eccentric portion.

[0037] Under an oppressive operation condition of the rotary compressor 1 (such as an operation condition where a pressure ratio between the low-pressure side and the high-pressure side of the refrigerant gas is high, and a gas temperature at the high-pressure side is increased and a gas flow rate is decreased, for example, at the time of heating when the outside air temperature is low), an increase in the contact stress between the first and the second vanes 127S and 127T and the first and the second annular pistons 125S and 125T causes abnormal abrasion between the first and the second vanes 127S and 127T and the first and the second annular pistons 125S and 125T. For avoiding this, the maximum contact stress σ_{max}, which is expressed by the equation (2), needs to be decreased as low as possible.

[0038] In order to decrease the maximum contact stress σ_{max}, it is effective that the vane width W of the first and the second vanes 127S and 127T is decreased and the vane pressing force P with the back pressure of the refrigerant gas in the compressor housing 10 is decreased (refer to equation (2)). Furthermore, an increase in the vane leading end curvature radius R_v can increase the contact width a (the contact width a is the contact width in the circumferential direction with elastic deformation at each tangent point between the first and the second vanes 127S and 127T and the first and the second annular pistons 125S and 125T, and is observed only as a contact point in FIG. 4) between the first and the second vanes 127S and 127T and the first and the second annular pistons 125S and 125T, which is expressed by the equation (1), and can decrease the maximum contact stress σ_{max}, which is expressed by the equation (2).

[0039] When the vane leading end curvature radius R_v is excessively increased, the non-sliding region width W_t on each of both the side portions of the leading ends of the first and the second vanes 127S and 127T, which is expressed by the equation (A), is 0 and the vane ridge line portions illustrated in FIG. 3 hit the outer circumferential surfaces of the first and the second annular pistons 125S and 125T, resulting in an increase in the maximum contact stress σ_{max}. This causes the abnormal abrasion of the outer circumferential surfaces.

[0040] In order to prevent the non-sliding region width W_t from being 0 and prevent the vane ridge line portions from hitting the outer circumferential surfaces of the first and the second annular pistons 125S and 125T even with production tolerance of the first and the second vanes 127S and 127T, production tolerance of the first and the second vane grooves 128S and 128T, deflection of the first and the second vanes 127S and 127T, and the like, the vane width W and the vane leading end curvature radius R_v are set such that the non-sliding region width W_t on each of both the side portions of the leading ends of the first and the second vanes 127S and 127T that is defined by the equation (A) is a value satisfying the following equation (B) :

$$0.3\ \text{mm} \le W_t \le 0.6\ \text{mm} \qquad (B)$$

[0041] By setting the non-sliding region width W_t to a value satisfying the equation (B) (the non-sliding region width

W_t is smaller than a conventional value described in the background by equal to or more than 10%), the vane width W is smaller than the conventional width and the vane pressing force P with the back pressure is decreased by 20%, thereby decreasing the maximum contact stress σ_{max}.

[0042] By setting the non-sliding region width W_t on each of both the side portions of the leading ends of the first and the second vanes 127S and 127T that is defined by the equation (A) to a value satisfying the equation (B), the vane width W and the vane leading end curvature radius R_v appropriate for improving reliability of the rotary compressor 1 can be provided. The rotary compressor 1 can thereby be used under the oppressive operation condition where the discharge temperature of the refrigerant gas is high.

[0043] The rotary compressor according to the present invention is particularly effective when an R32 refrigerant having a lower gas density and a higher discharge temperature than those of an R410A refrigerant or a mixed refrigerant containing the R32 refrigerant of at least equal to or higher than 25% by weight is used.

[Explanation of Reference]

[0044]

1 ROTARY COMPRESSOR
10 COMPRESSOR HOUSING
11 MOTOR
12 COMPRESSION UNIT
15 ROTATING SHAFT
25 ACCUMULATOR
31S FIRST LOW-PRESSURE COMMUNICATION PIPE
31T SECOND LOW-PRESSURE COMMUNICATION PIPE
101 FIRST THROUGH-HOLE
102 SECOND THROUGH-HOLE
104 FIRST SUCTION PIPE
105 SECOND SUCTION PIPE
107 DISCHARGE PIPE (DISCHARGE PORTION)
111 STATOR
112 ROTOR
12S FIRST COMPRESSION UNIT
12T SECOND COMPRESSION UNIT
121S FIRST CYLINDER (CYLINDER)
121T SECOND CYLINDER (CYLINDER)
122S FIRST LATERAL EXPANDING PORTION
122T SECOND LATERAL EXPANDING PORTION
123S FIRST CYLINDER INNER WALL (CYLINDER INNER WALL)
123T SECOND CYLINDER INNER WALL (CYLINDER INNER WALL)
124S FIRST SPRING HOLE
124T SECOND SPRING HOLE
125S FIRST ANNULAR PISTON (ANNULAR PISTON)
125T SECOND ANNULAR PISTON (ANNULAR PISTON)
127S FIRST VANE (VANE)
127T SECOND VANE (VANE)
128S FIRST VANE GROOVE (VANE GROOVE)
128T SECOND VANE GROOVE (VANE GROOVE)
129S FIRST PRESSURE INTRODUCTION PATH
129T SECOND PRESSURE INTRODUCTION PATH
130S FIRST OPERATION CHAMBER (OPERATION CHAMBER)
130T SECOND OPERATION CHAMBER (OPERATION CHAMBER)
131S FIRST SUCTION CHAMBER (SUCTION CHAMBER)
131T SECOND SUCTION CHAMBER (SUCTION CHAMBER)
133S FIRST COMPRESSION CHAMBER (COMPRESSION CHAMBER)
133T SECOND COMPRESSION CHAMBER (COMPRESSION CHAMBER)
135S FIRST SUCTION HOLE (SUCTION HOLE)
135T SECOND SUCTION HOLE (SUCTION HOLE)
136 REFRIGERANT PASSAGE

140 INTERMEDIATE PARTITION PLATE
151 SUB SHAFT PORTION
152S FIRST ECCENTRIC PORTION (ECCENTRIC PORTION)
152T SECOND ECCENTRIC PORTION (ECCENTRIC PORTION)
153 MAIN SHAFT PORTION
160S LOWER END PLATE (END PLATE)
160T UPPER END PLATE (END PLATE)
161S SUB BEARING PORTION (BEARING PORTION)
161T MAIN BEARING PORTION (BEARING PORTION)
170S LOWER MUFFLER COVER
170T UPPER MUFFLER COVER
175 INSERTION BOLT
180S LOWER MUFFLER CHAMBER
180T UPPER MUFFLER CHAMBER
190S FIRST DISCHARGE HOLE (DISCHARGE VALVE PORTION)
190T SECOND DISCHARGE HOLE (DISCHARGE VALVE PORTION)
200S FIRST DISCHARGE VALVE (DISCHARGE VALVE PORTION)
200T SECOND DISCHARGE VALVE (DISCHARGE VALVE PORTION)
201S FIRST DISCHARGE VALVE PRESSER (DISCHARGE VALVE PORTION)
201T SECOND DISCHARGE VALVE PRESSER (DISCHARGE VALVE PORTION)
252 ACCUMULATOR HOLDER
253 ACCUMULATOR BAND
255 SYSTEM CONNECTION PIPE
257 BOTTOM THROUGH-HOLE
R OPENING

Claims

1. A rotary compressor comprising:

a vertically placed and sealed compressor housing including a discharge portion of a refrigerant on an upper portion and a suction portion of the refrigerant on a lower side surface;

a compression unit arranged on a lower portion of the compressor housing and including an annular cylinder, an end plate that has a bearing portion and a discharge valve portion and closes one end portion of the cylinder, an end plate or an intermediate partition plate that has a bearing portion and closes the other end portion of the cylinder, an annular piston that is fitted into an eccentric portion of a rotating axis supported on the bearing portion, revolves in the cylinder along a cylinder inner wall of the cylinder, and forms an operation chamber between the annular piston and the cylinder inner wall, and a vane that projects into the operation chamber from a vane groove provided in the cylinder, abuts against the annular piston, and divides the operation chamber into a suction chamber and a compression chamber; and

a motor that is arranged on an upper portion of the compressor housing and drives the compression unit through the rotating axis,

the rotary compressor sucking the refrigerant through the suction portion and discharging the refrigerant after passing through the compressor housing from the discharge portion, wherein

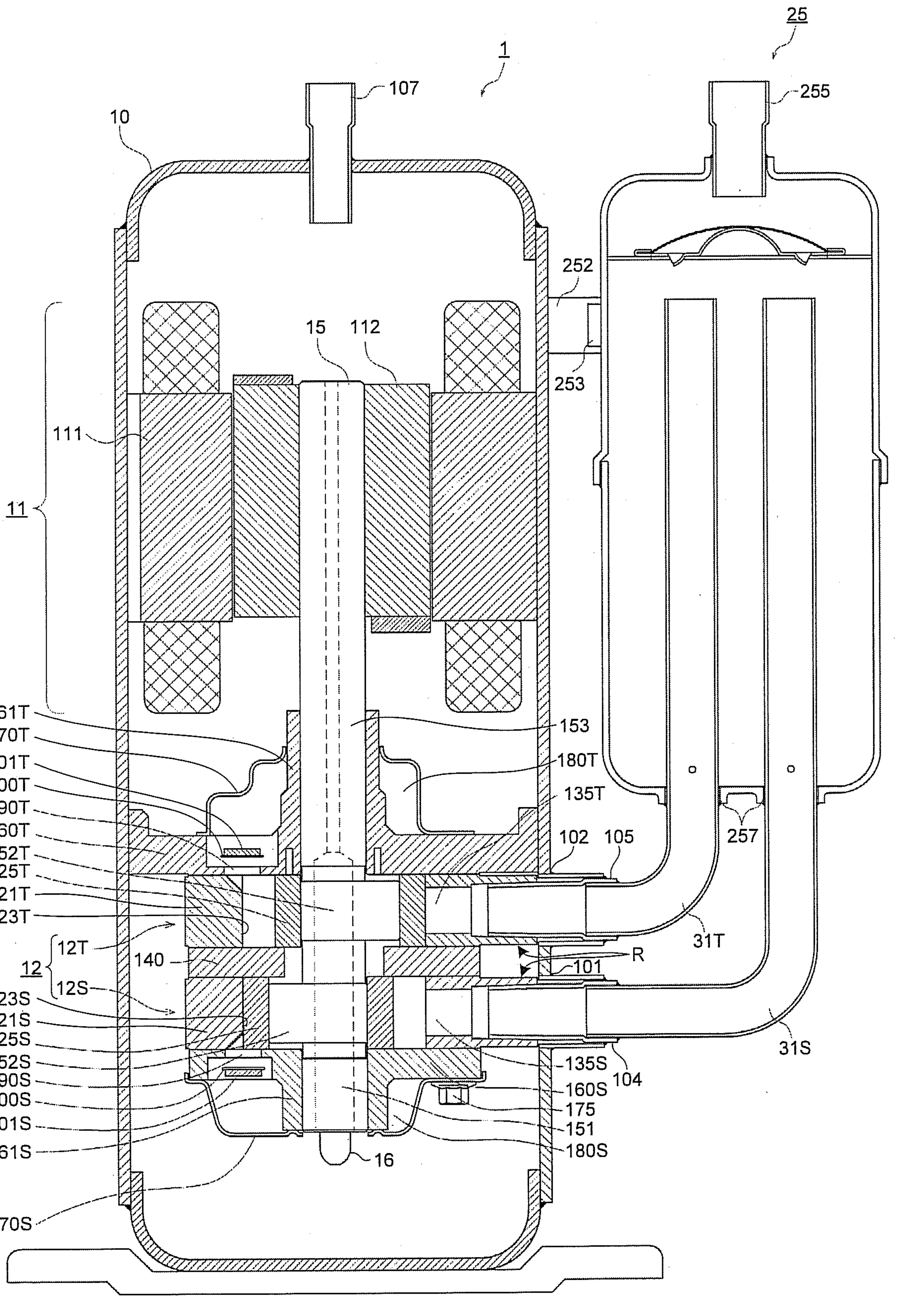
when it is assumed that a vane width is W, the amount of eccentricity of the eccentric portion is e, a vane leading end curvature radius is R_v, an annular piston radius is R_{ro}, and a non-sliding region width on each of both side portions of a vane leading end is W_t, the vane width W and the vane leading end curvature radius R_v are set such that the non-sliding region width W_t on each of both the side portions of the vane leading end defined by the following equation (A) is a value satisfying an equation (B):

$$W_t = (W/2) - e \times R_v/(R_v + R_{ro}) \qquad (A)$$

$$0.3\ \text{mm} \leq W_t \leq 0.6\ \text{mm} \qquad (B).$$

2. The rotary compressor according to claim 1, wherein the refrigerant is an R32 refrigerant or a mixed refrigerant containing the R32 refrigerant of at least equal to or higher than 25% by weight.

FIG.1



25
1
107
255
10
252
253
15
112
111
11
161T
170T
201T
200T
190T
160T
152T
125T
121T
123T
153
180T
135T
102
105
257
12T
12
12S
140
R
101
31T
123S
121S
125S
152S
190S
200S
201S
161S
170S
135S
104
160S
175
151
180S
16
31S

FIG.2

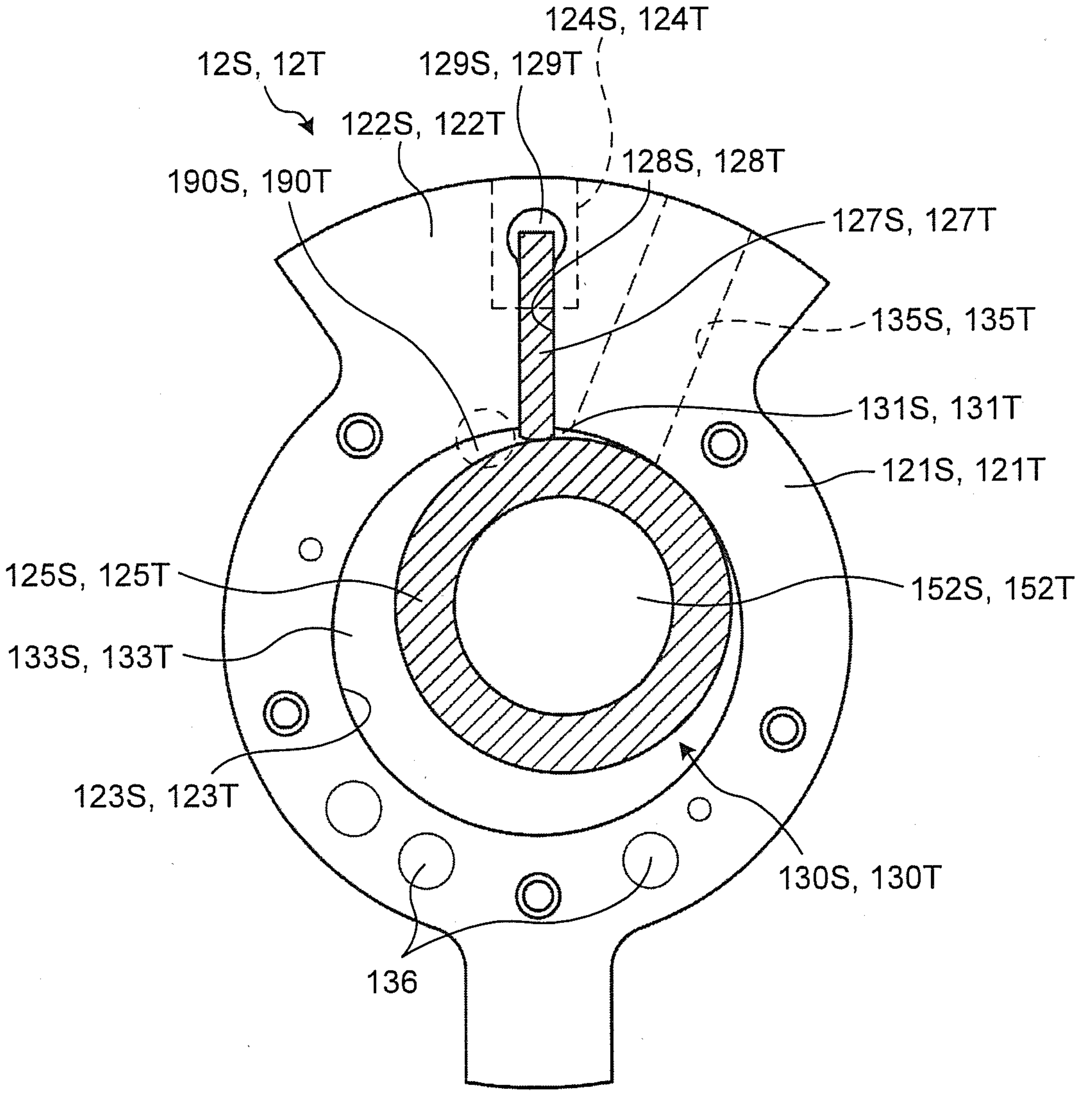

124S, 124T
12S, 12T
129S, 129T
122S, 122T
128S, 128T
190S, 190T
127S, 127T
135S, 135T
131S, 131T
121S, 121T
125S, 125T
152S, 152T
133S, 133T
123S, 123T
130S, 130T
136

FIG.3

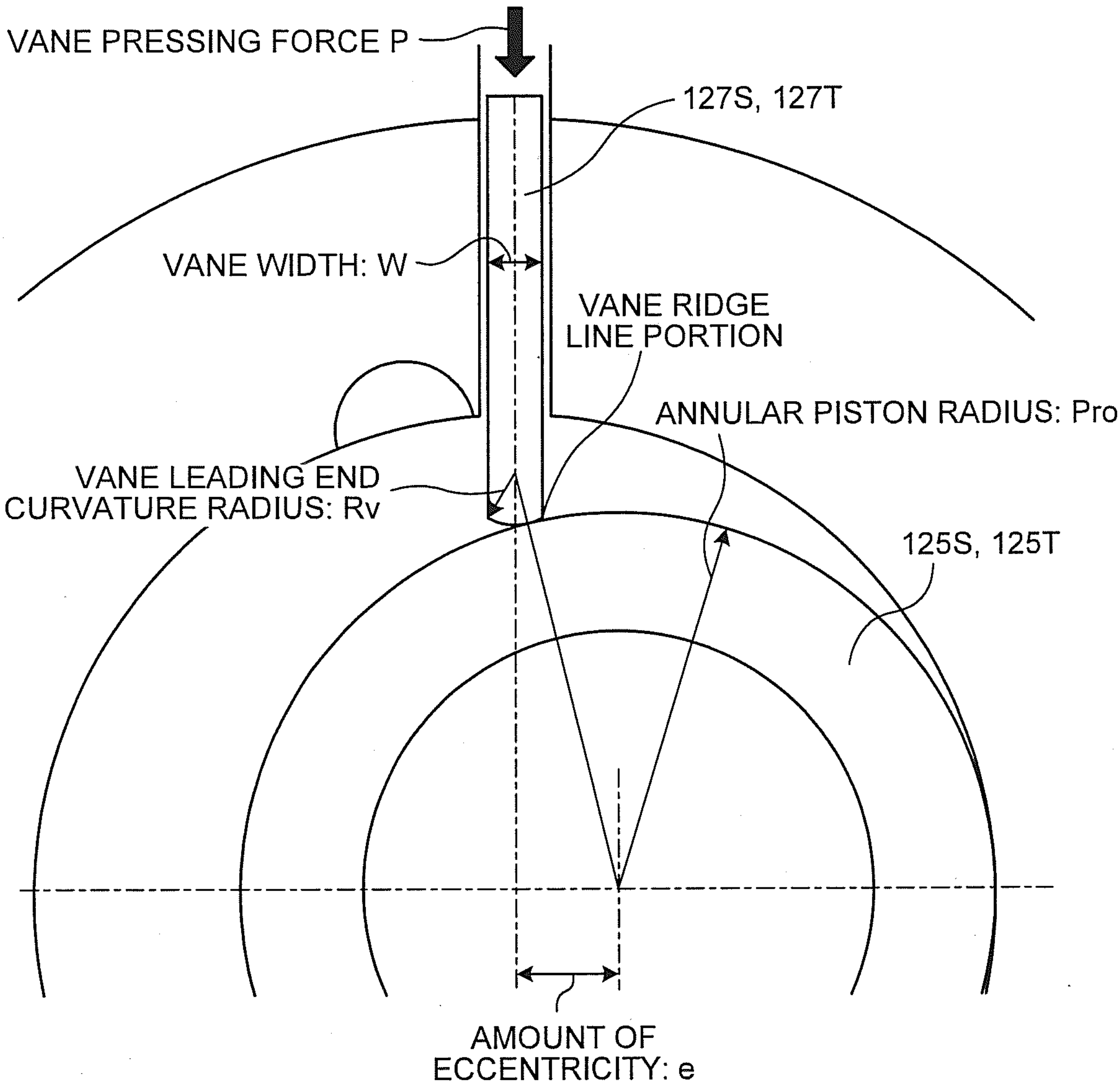
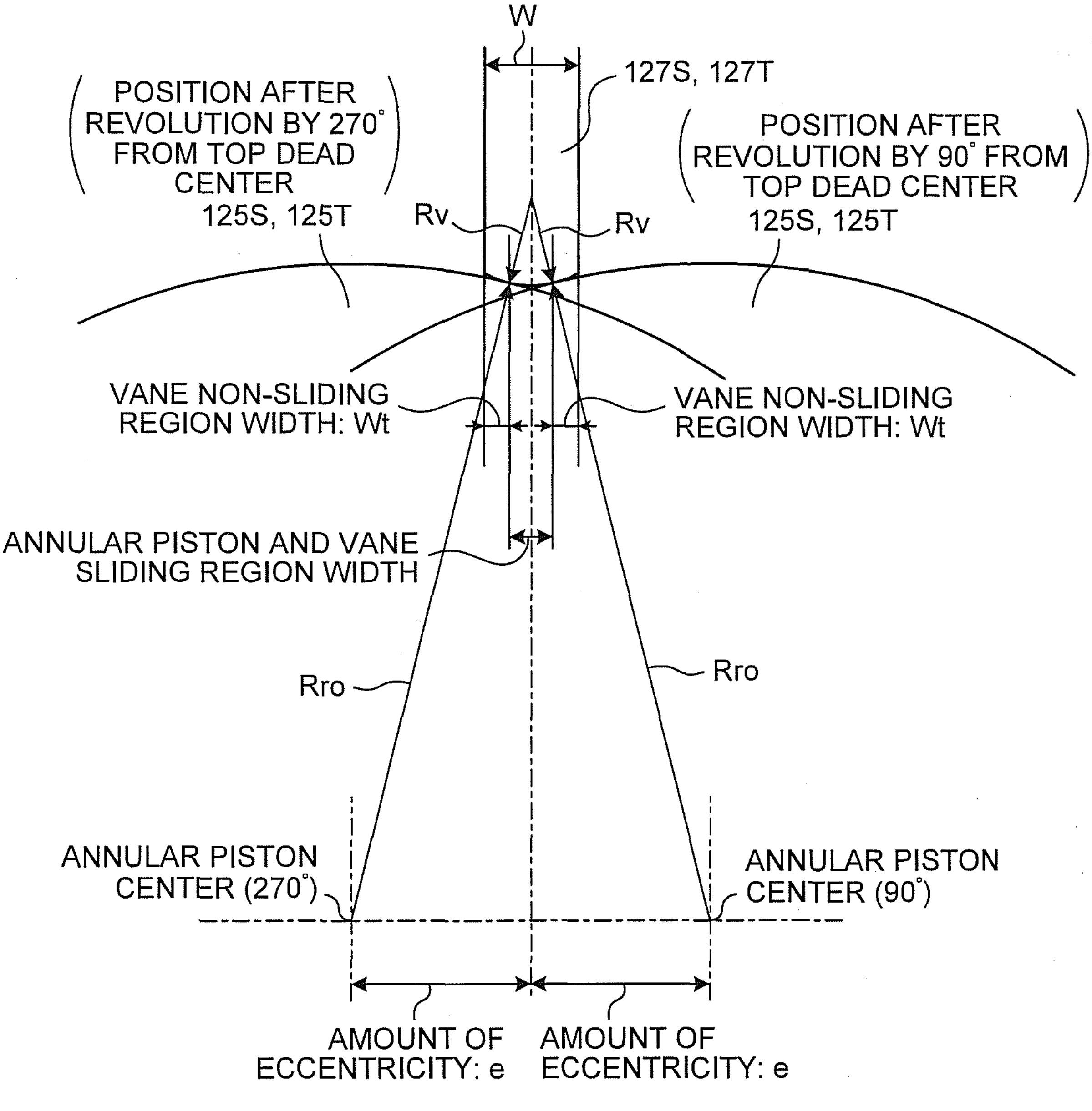

VANE PRESSING FORCE P
127S, 127T
VANE WIDTH: W
VANE RIDGE
LINE PORTION
ANNULAR PISTON RADIUS: Pro
VANE LEADING END
CURVATURE RADIUS: Rv
125S, 125T
AMOUNT OF
ECCENTRICITY: e

FIG.4


W
127S, 127T
POSITION AFTER REVOLUTION BY 270° FROM TOP DEAD CENTER
125S, 125T
POSITION AFTER REVOLUTION BY 90° FROM TOP DEAD CENTER
125S, 125T
Rv
Rv
VANE NON-SLIDING REGION WIDTH: Wt
VANE NON-SLIDING REGION WIDTH: Wt
ANNULAR PISTON AND VANE SLIDING REGION WIDTH
Rro
Rro
ANNULAR PISTON CENTER (270°)
ANNULAR PISTON CENTER (90°)
AMOUNT OF ECCENTRICITY: e
AMOUNT OF ECCENTRICITY: e

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2015/052976

A. CLASSIFICATION OF SUBJECT MATTER
F04C18/356(2006.01)i, *F04C23/00*(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)
F04C18/356, F04C23/00

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	1996-2015
Kokai Jitsuyo Shinan Koho	1971-2015	Toroku Jitsuyo Shinan Koho	1994-2015

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.
X Y	JP 2007-92575 A (Mitsubishi Electric Corp.), 12 April 2007 (12.04.2007), paragraphs [0018] to [0025]; fig. 1 to 5 (Family: none)	1 1-2
Y	JP 8-14175 A (Daikin Industries, Ltd.), 16 January 1996 (16.01.1996), paragraphs [0096] to [0102]; fig. 17, 19 (Family: none)	1-2
A	JP 2001-263280 A (Sanyo Electric Co., Ltd.), 26 September 2001 (26.09.2001), paragraphs [0026] to [0049]; tables 1 to 4 & US 2001/0043879 A1 & EP 1134418 A2	1-2

☒ 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
07 April 2015 (07.04.15)	21 April 2015 (21.04.15)

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/JP2015/052976

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 7-229488 A (Hitachi, Ltd.), 29 August 1995 (29.08.1995), entire text; all drawings & US 5494423 A	1-2
A	JP 7-293463 A (Matsushita Refrigeration Co.), 07 November 1995 (07.11.1995), entire text; all drawings (Family: none)	1-2
A	JP 2002-242867 A (Sanyo Electric Co., Ltd.), 28 August 2002 (28.08.2002), paragraphs [0023] to [0036]; table 1 & US 2002/0150493 A1 & EP 1233186 A2	1-2

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 7229488 A **[0005]**