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

(11) EP 4 464 897 A1

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

(43) Date of publication:
20.11.2024 Bulletin 2024/47

(21) Application number: **24739378.8**

(22) Date of filing: **28.03.2024**

(51) International Patent Classification (IPC):
F04C 23/02 *(2006.01)* ***F04B 39/00*** *(2006.01)*
F04C 29/00 *(2006.01)*

(86) International application number:
PCT/JP2024/012928

(87) International publication number:
WO 2024/204663 (03.10.2024 Gazette 2024/40)

(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 ME MK MT NL NO PL PT RO RS SE SI SK SM TR
Designated Extension States:
BA
Designated Validation States:
GE KH MA MD TN

(30) Priority: **31.03.2023 JP 2023057770**

(71) Applicant: **DAIKIN INDUSTRIES, LTD.**
Osaka-shi, Osaka 530-0001 (JP)

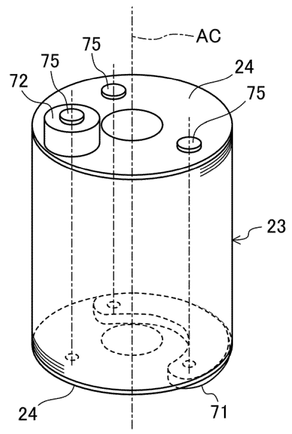
(72) Inventors:
- **WADA, Hikaru**
Osaka-shi
Osaka
5300001 (JP)
- **KATAYAMA, Tatsuya**
Osaka-shi
Osaka
5300001 (JP)
- **SOTOJIMA, Takazo**
Osaka-shi
Osaka
5300001 (JP)
- **KAWAE, Arisa**
Osaka-shi
Osaka
5300001 (JP)
- **UENO, Hiromichi**
Osaka-shi
Osaka
5300001 (JP)
- **KUMAZAKI, Tomoya**
Osaka-shi
Osaka
5300001 (JP)

(74) Representative: **Goddar, Heinz J.**
Boehmert & Boehmert
Anwaltspartnerschaft mbB
Pettenkoferstrasse 22
80336 München (DE)

(54) ROTARY COMPRESSOR AND REFRIGERATION DEVICE HAVING SAME

(57) A rotary compressor (10) includes a motor (21) disposed above a compression mechanism (30) and coupled to the compression mechanism through a drive shaft (25). The compression mechanism has a first cylinder (40) located on the upper side, a second cylinder (50) located on the lower side, a first eccentric portion (44) housed in the first cylinder, and a second eccentric portion (54) housed in the second cylinder. A first balance weight (71) is provided on a lower end portion of a rotor (23), and a second balance weight (72) is provided on an upper end portion of the rotor (23). A value of the product of the mass and eccentric distance of the second balance weight is smaller than a value of the product of the mass and eccentric distance of the first balance weight.

FIG.7



Description

TECHNICAL FIELD

[0001] The present disclosure relates to a rotary compressor and a refrigeration apparatus including the rotary compressor.

BACKGROUND ART

[0002] Two-cylinder rotary compressors have been known as a rotary compressor. The two-cylinder rotary compressors are vertical compressors having a motor, a drive shaft, and a compression mechanism housed in a casing, and the axial direction of which along the axis of the drive shaft is the vertical direction. The motor is disposed above the compression mechanism. The drive shaft couples a rotor that constitutes the motor and the compression mechanism to each other.

[0003] The compression mechanism has two cylinders arranged one above the other, and eccentric portions housed in the respective cylinders. The two eccentric portions are eccentric with respect to the axis of the drive shaft with a phase difference of 180 degrees in the circumferential direction. The inside of each cylinder is divided into a high-pressure chamber and a low-pressure chamber by a blade extending outward in the radial direction from a piston. Each eccentric portion eccentrically rotates in the cylinder along with the rotation of the drive shaft. By this operation, the compression mechanism sucks fluid into the low-pressure chamber, changes the low-pressure chamber to the high-pressure chamber, and compresses the fluid in the high-pressure chamber.

[0004] In the two-cylinder rotary compressor, the rotor that constitutes the motor is provided with a balancer for compensating the unbalance due to the eccentric rotational motion of the compression mechanism. The balancer includes two balance weights provided on both upper and lower surfaces of the rotor. One example of such a rotary compressor with the balancer is disclosed in Patent Document 1.

CITATION LIST

PATENT DOCUMENT

[0005] Patent Document 1: Japanese Unexamined Patent Publication No. 2009-180203

SUMMARY OF THE INVENTION

TECHNICAL PROBLEM

[0006] A general configuration of the rotary compressor with the balancer, such as one described above, is that it is balanced in a static condition. Specifically, the balance weight on the lower side is eccentric in the same direction as the eccentric portion on the upper side, and the balance weight on the upper side is eccentric in the same direction as the eccentric portion on the lower side. The eccentric portions and the balance weights are designed such that a value of the product of the mass and eccentricity of the eccentric portion on the upper side is equal to a value of the product of the mass and eccentricity of the eccentric portion on the lower side, and that a value of the product of the mass and eccentric distance of the balance weight on the upper side is equal to a value of the product of the mass and eccentric distance of the balance weight on the lower side.

[0007] However, the fluid is compressed in the cylinders when the rotary compressor is operated, and the pressure of the compressed fluid acts on each eccentric portion. The resulting compressive load is applied to the drive shaft, and the drive shaft bends slightly. The bent drive shaft results in a deviation of the rotor from a preset rotation center. If this happens, the centrifugal force of the rotor acts on the drive shaft in combination with the inertial force acting on the balance weight on the upper side, which increases bending of the drive shaft. As a result, a high-speed operation significantly destroys the static balance between the compression mechanism (two eccentric portions) and the balancer (two balance weights), and vibration increases.

[0008] It is an object of the present disclosure to reduce vibration in high-speed operation of a rotary compressor.

SOLUTION TO THE PROBLEM

[0009] A first aspect of the present disclosure is directed to a rotary compressor (10). The rotary compressor (10) of the first aspect includes: a compression mechanism (30) configured to suck and compress a fluid; a drive shaft (25) configured to drive the compression mechanism (30); a motor (21) having a rotor (23) coupled to the drive shaft (25); and a balancer (70) provided on the rotor (23). The compression mechanism (30) has a first cylinder (40) and a second cylinder (50) sequentially arranged next to each other from a side closer to the rotor (23) in an axial direction along an axis (AC) of the drive shaft (25), a first eccentric portion (44) housed in the first cylinder (40), and a second eccentric portion (54) housed in the second cylinder (50). The balancer (70) has a first balance weight (71) disposed at one end portion of the rotor (23) closer to the compression mechanism (30) in the axial direction, and a second balance weight (72) disposed at the other end portion of the rotor (23) in the axial direction. The first eccentric portion (44) and the first balance weight (71) are eccentric to the axis (AC) of the drive shaft (25) to one side, and the second eccentric portion (54) and the second balance weight (72) are eccentric to the axis (AC) of the drive shaft (25) to the other side. In the rotary compressor (10), along with rotation of the drive shaft (25), the first eccentric portion (44) eccentrically rotates in the first cylinder (40), and the second eccentric portion (54) eccentrically rotates in the second cylinder (50). A relationship of $m1 \times r1 > m2 \times r2$ is satisfied, where a mass of the first balance weight (71) is m1, an eccentric distance of a center of gravity (GC1) of the first balance weight (71) from the axis (AC) of the drive shaft (25) is r1, a mass of the second balance weight (72) is m2, and an eccentric distance of a center of gravity (GC2) of the second balance weight (72) from the axis (AC) of the drive shaft (25) is r2.

[0010] According to the first aspect, the value of the product of the mass and the eccentric distance of the second balance weight (72) ($m2 \times r2$) is smaller than the value of the product of the mass and the eccentric distance of the first balance weight (71) ($m1 \times r1$). This configuration reduces the inertial force of the second balance weight (72) acting in combination with the centrifugal force of the rotor (23) deviated from the predetermined rotation center when the drive shaft (25) is bent in the operation of the rotary compressor (10), thereby making it possible to reduce an increase in the bending of the drive shaft (25). The vibration in a high-speed operation of the rotary compressor (10) can thus be reduced.

[0011] A second aspect of the present disclosure is the rotary compressor (10) of the first aspect. In the second aspect, a maximum number of rotations of the drive shaft (25) is 120 rps or more.

[0012] According to the second aspect, the maximum number of rotations of the drive shaft (25) is 120 rps or more, which is a relatively great number of revolutions. The faster the drive shaft (25) rotates, the more the static balance between the first eccentric portion (44) and second eccentric portion (54) and the first balance weight (71) and second balance weight (72) tends to be destroyed significantly due to bending of the drive shaft (25), which tends to increase vibration of the rotary compressor (10). Thus, the technology according to the present disclosure is effective in the rotary compressor (10) that operates at the relatively great number of rotations.

[0013] A third aspect of the present disclosure is the rotary compressor (10) of the first or second aspect. In the third aspect, a relationship of $1.2 - 0.002 \times Nmax \leq (m2 \times r2 + m4 \times r4)/(m1 \times r1 + m3 \times r3) \leq 0.98$ is satisfied, where a mass of the first eccentric portion (44) is m3, an eccentric distance of a center of gravity (GC3) of the first eccentric portion (44) from the axis (AC) of the drive shaft (25) is r3, a mass of the second eccentric portion (54) is m4, an eccentric distance of a center of gravity (GC4) of the second eccentric portion (54) from the axis (AC) of the drive shaft (25) is r4, and a maximum number of rotations of the drive shaft (25) is Nmax [rps].

[0014] According to the third aspect, the static balance relation value obtained by dividing the sum of the value of the product of the mass and eccentric distance of the second balance weight (72) and the value of the product of the mass and eccentric distance of the second eccentric portion (54) ($m2 \times r2 + m4 \times r4$) by the sum of the value of the product of the mass and eccentric distance of the first balance weight (71) and the value of the product of the mass and eccentric distance of the first eccentric portion (44) ($m1 \times r1 + m3 \times r3$) is $1.2 - 0.002 \times Nmax$ (maximum number of rotations) or more and 0.98 or less. This can reduce vibration in a high-speed operation of the rotary compressor (10) in accordance with the maximum number of rotations of the drive shaft (25).

[0015] A fourth aspect of the present disclosure is the rotary compressor (10) of the third aspect. In the fourth aspect, a relationship of $0.88 \leq (m2 \times r2 + m4 \times r4)/(m1 \times r1 + m3 \times r3)$ is satisfied.

[0016] According to the fourth aspect, the static balance relation value is 0.88 or more. If the static balance relation value is smaller than 0.88, the vibration reduction effect in a high-speed operation of the rotary compressor (10) can only be expected only in a rotation number range relatively close to the maximum number of rotations of the drive shaft (25). On the other hand, if the static balance relation value is 0.88 or more, vibration of the rotary compressor (10) can be reduced in a relatively wide range of the number of rotations from the maximum number of rotations of the drive shaft (25).

[0017] A fifth aspect of the present disclosure is the rotary compressor of any one of the first to fourth aspects. In the fifth aspect, a relationship of $Vcc/\Phi^4 \geq 3 \times 10^{-4}$ is satisfied, where a sum of a volume of a first compression chamber (41s) formed between an inner peripheral surface of the first cylinder (40) and the first eccentric portion (44) and a volume of a second compression chamber (51s) formed between an inner peripheral surface of the second cylinder (50) and the second eccentric portion (54) is Vcc [cc], and a diameter of a main shaft portion (26) of the drive shaft (25) coupled to the rotor (23) is Φ [mm].

[0018] According to the fifth aspect, an index value indicating how easily the main shaft portion (26) is bent, which is obtained by dividing the sum Vcc of the volume of the first compression chamber (41s) and the volume the second compression chamber (51s) by the diameter Φ of the main shaft portion (26) of the drive shaft (25), is 3×10^{-4} or more. This configuration makes it possible to bend the main shaft portion (26) suitably such that the static balance between the first eccentric portion (44) and second eccentric portion (54) and the first balance weight (71) and second balance weight (72) can be maintained in the operation of the rotary compressor (10).

[0019] A sixth aspect of the present disclosure is the rotary compressor (10) of any one of the first to fifth aspects. In the sixth aspect, the drive shaft (25) has a main shaft portion (26) coupled to the rotor (23), and the main shaft portion (26) has a diameter of 16 mm or less. A relationship of $(m3 \times r3 + m4 \times r4)/2 \geq 600$ is satisfied, where a mass of the first eccentric portion (44) is m3 [g], an eccentric distance of a center of gravity (GC3) of the first eccentric portion (44) from the axis (AC) of the drive shaft (25) is r3 [mm], a mass of the second eccentric portion (54) is m4 [g], and an eccentric distance of a center of gravity (GC4) of the second eccentric portion (54) from the axis (AC) of the drive shaft (25) is r4 [mm],

[0020] According to the sixth aspect, the main shaft portion (26) of the drive shaft (25) has a relatively small diameter of 16 mm or less, and an average value of the values of the product of the mass and the eccentric distance of the first eccentric portion (44) and the product of the mass and the eccentric distance of the second eccentric portion (54) is relatively great, that is, 600 or more.

[0021] As a method for reducing vibration of the rotary compressor (10) caused by bending of the drive shaft (25) in a high-speed operation, it is conceivable to increase the diameter of the main shaft portion (26) and increase the rigidity of the drive shaft (25), or to reduce the mass and the eccentric distance of each of the first eccentric portion (44) and the second eccentric portion (54) and reduce the size of the rotor (23). However, the former method increases a mechanical loss such as a bearing loss, and the efficiency of the rotary compressor (10) decreases. The latter method limits the capacity of the rotary compressor (10).

[0022] On the other hand, the drive shaft (25) whose main shaft portion (26) has a diameter of 16 mm or less can reduce the mechanical loss such as the bearing loss and increase the efficiency of the rotary compressors (10). It is also possible to ensure a relatively great capacity of the rotary compressor (10) when the average value of the values of the product of the mass and the eccentric distance of the first eccentric portion (44) and the product of the mass and the eccentric distance of the second eccentric portion (54) is 600 or more. It is thus possible to achieve the rotary compressor (10) with high efficiency and great capacity, which is less likely to vibrate in a high-speed operation.

[0023] A seventh aspect of the present disclosure is directed to a refrigeration apparatus (1). The refrigeration apparatus (1) of the seventh aspect includes a refrigerant circuit (1a) configured to perform a refrigeration cycle. The refrigerant circuit (1a) has the rotary compressors (10) of any one of the first to sixth aspects.

[0024] According to the seventh aspect, the rotary compressor (10) according to the technology of the present disclosure is used in the refrigerant circuit (1a). Vibration in a high-speed operation is reduced in the rotary compressor (10). Use of the rotary compressor (10) in the refrigerant circuit (1a) contributes to execution of the refrigeration cycle with less vibration and noise.

BRIEF DESCRIPTION OF THE DRAWINGS

[0025]

FIG. 1 is a refrigerant circuit diagram illustrating, as an example, a configuration of a refrigeration apparatus of an embodiment.

FIG. 2 is a vertical cross-sectional view illustrating, as an example, a configuration of a rotary compressor of the embodiment.

FIG. 3 is a vertical cross-sectional view illustrating, as an example, a configuration of a compression mechanism of the embodiment.

FIG. 4 is a lateral cross-sectional view illustrating, as an example, configurations of a first cylinder and a first eccentric portion of the embodiment.

FIG. 5 is a lateral cross-sectional view illustrating, as an example, configurations of a second cylinder and a second eccentric portion of the embodiment.

FIG. 6 is a vertical cross-sectional view illustrating, as an example, a main portion (rotary system) of the rotary compressor of the embodiment.

FIG. 7 is a perspective view illustrating, as an example, configurations of a rotor and a balancer of the embodiment.

FIG. 8 is an exploded perspective view illustrating, as an example, the configurations of the rotor and the balancer of the embodiment.

FIG. 9 is a top view illustrating, as an example, the main portion (rotary system) of the rotary compressor of the embodiment.

FIG. 10 is a table showing parameters related to the static balance of the rotary system of the rotary compressor and a static balance relation value, for Comparative Example and Examples 1 to 16.

FIG. 11 is a graph showing results of simulation of the acceleration of vibration generated in the rotary compressors of Comparative Example and Examples 1 to 11.

FIG. 12 is a table showing results of simulation of the acceleration of vibration generated in the rotary compressors of Comparative Example and Examples 1 to 16, using the number of rotations on the high-speed side.

FIG. 13 is a lateral cross-sectional view illustrating, as an example, configurations of a first cylinder and a first eccentric portion of another embodiment.

DESCRIPTION OF EMBODIMENTS

[0026] An illustrative embodiment will be described below in detail with reference to the drawings. In the following embodiment, a case in which a rotary compressor according to the technique of the present disclosure is applied to a refrigeration apparatus will be described as an example. The drawings are used for conceptual description of the technique of the present disclosure. In the drawings, dimensions, ratios, or numbers may be exaggerated or simplified for easier understanding of the technique of the present disclosure.

[0027] In the following embodiment, a direction along the axis of a drive shaft of the rotary compressor will be referred to as an "axial direction," a direction perpendicular to the axial direction as a "radial direction," and a direction along the circumference of the drive shaft as a "circumferential direction." In addition, the expressions of "first," "second," ... are used to distinguish the terms to which these expressions are given, and do not limit the number and order of the terms.

<<Embodiment>>

[0028] As illustrated in FIG. 1, a rotary compressor (10) of this embodiment is provided in a refrigeration apparatus (1).

- Refrigeration Apparatus -

[0029] The refrigeration apparatus (1) includes a refrigerant circuit (1a). The refrigerant circuit (1a) is filled with a refrigerant. The refrigerant is an example of the fluid compressed by the rotary compressor (10). The refrigerant circuit (1a) includes the rotary compressor (10), a radiator (3), a decompression mechanism (4), and an evaporator (5). The decompression mechanism (4) is an expansion valve, for example. The refrigerant circuit (1a) performs a vapor compression refrigeration cycle.

[0030] In the refrigeration cycle, the gas refrigerant compressed by the rotary compressor (10) dissipates heat to the air in the radiator (3). At this time, the refrigerant is liquefied and changed into liquid refrigerant. The liquid refrigerant having dissipated heat is decompressed by the decompression mechanism (4). The decompressed liquid refrigerant is evaporated in the evaporator (5). At this time, the refrigerant is vaporized and changed into gas refrigerant. The evaporated gas refrigerant is sucked into the rotary compressor (10). The rotary compressor (10) compresses the sucked gas refrigerant.

[0031] The refrigeration apparatus (1) is an air conditioner, for example. The air conditioner may be a cooling and heating machine that switches between cooling and heating. In this case, the air conditioner has a switching mechanism that switches the direction of circulation of the refrigerant. The switching mechanism is a four-way switching valve, for example. The air conditioner may be a device for cooling only or a device for heating only.

[0032] The refrigeration apparatus (1) may be a water heater, a chiller unit, or a cooling apparatus that cools air in an internal space. The cooling apparatus is for cooling the air inside a water heater, a refrigerator, a freezer, or a container, for example.

- Rotary Compressor -

[0033] As illustrated in FIG. 2, the rotary compressor (10) is a two-cylinder rotary compressor. The maximum number of rotations of the rotary compressor (10) is 120 rps or more. The maximum number of rotations described herein is the number of rotations of a drive shaft (25) rotated by the operation of a motor (21), and refers to the maximum possible number of rotations in the operation range of the product. It is preferable to increase the maximum number of rotations of the rotary compressor (10) in order to increase the amount of circulation of the refrigerant in the refrigerant circuit (1a) and ensure the maximum amount of circulation of the refrigerant.

[0034] The rotary compressor (10) includes a casing (11), a drive mechanism (20), a compression mechanism (30), and a balancer (70). The drive mechanism (20), the compression mechanism (30), and the balancer (70) are housed in the casing (11).

<Casing>

[0035] The casing (11) is configured as a vertically-long cylindrical closed container with both ends closed. The casing (11) is placed in an upright position. The casing (11) has a barrel (12), a lower end plate (13), and an upper end plate (14). The barrel (12) is in the shape of a cylinder extending in the vertical direction. The lower end plate (13) is fixed to the lower end of the barrel (12) to close the lower end opening of the barrel (12). The upper end plate (14) is fixed to the upper end of the barrel (12) to close the upper end opening of the barrel (12).

[0036] A suction pipe (15) is attached to a lower portion of the barrel (12). The suction pipe (15) penetrates the barrel (12) and is connected to the compression mechanism (30). A discharge pipe (16) is attached to the upper end plate (14). The discharge pipe (16) penetrates the upper end plate (14) and is open to an upper space in the casing (11).

[0037] An oil reservoir (18) is provided at the bottom of the casing (11). The oil reservoir (18) is formed by inner walls of a lower portion of the barrel (12) and the lower end plate (13). The oil reservoir (18) stores oil. The oil lubricates sliding portions of the compression mechanism (30) and the drive shaft (25).

<Drive Mechanism>

[0038] The drive mechanism (20) includes the motor (21) and the drive shaft (25). The motor (21) is disposed above the compression mechanism (30). The motor (21) includes a stator (22) and a rotor (23). Each of the stator (22) and the rotor (23) has a cylindrical shape. The stator (22) is fixed to the barrel (12) of the casing (11). The rotor (23) is disposed in the hollow of the stator (22).

[0039] Circular plate-shaped end plates (24) having a hole are provided at both ends of the rotor (23) in the axial direction. The drive shaft (25) is inserted in the hollow of the rotor (23). The rotor (23) is fixed to the drive shaft (25). The drive shaft (25) rotates together with the rotor (23) when the motor (21) is energized. The drive shaft (25) is a shaft that drives the compression mechanism (30), and extends in the vertical direction in the casing (11).

[0040] The drive shaft (25) has a main shaft portion (26), a first eccentric shaft portion (27), and a second eccentric shaft portion (28). An upper portion of the main shaft portion (26) is coupled to the rotor (23). The first eccentric shaft portion (27) and the second eccentric shaft portion (28) are provided near the lower end of the main shaft portion (26). The first eccentric shaft portion (27) is disposed above the second eccentric shaft portion (28). The diameters of the first eccentric shaft portion (27) and the second eccentric shaft portion (28) are greater than the diameter of the main shaft portion (26).

[0041] The first eccentric shaft portion (27) and the second eccentric shaft portion (28) are eccentric from the axis (AC) of the drive shaft (25) (main shaft portion (26)) by a predetermined distance. The first eccentric shaft portion (27) and the second eccentric shaft portion (28) are eccentric with respect to the axis (AC) of the drive shaft (25) to the opposite sides. A portion of the main shaft portion (26) above the first eccentric shaft portion (27) is rotatably supported by a front head (31). A portion of the main shaft portion (26) below the second eccentric shaft portion (28) is rotatably supported by a rear head (33).

[0042] A first oil passage (25b) is formed inside the drive shaft (25). The first oil passage (25b) extends to the sliding portions of the compression mechanism (30) and the drive shaft (25). An oil supply pump (25a) is provided at the lower end of the drive shaft (25). The oil supply pump (25a) is immersed in the oil in the oil reservoir (18). The oil supply pump (25a) delivers the oil along with rotation of the drive shaft (25). The delivered oil is supplied to the sliding portions of the compression mechanism (30) and the drive shaft (25) through the first oil passage (25b).

<Compression Mechanism>

[0043] The compression mechanism (30) is a mechanism for sucking and compressing the refrigerant, and is disposed below the motor (21). The compression mechanism (30) includes the front head (31), a first cylinder (40), a middle plate (32), a second cylinder (50), and the rear head (33). The front head (31), the first cylinder (40), the middle plate (32), the second cylinder (50), and the rear head (33) are stacked in this order from top to bottom and fixed with a fastening bolt (35).

[0044] The front head (31) is fixed to the barrel (12) of the casing (11). The front head (31) is stacked on the top of the first cylinder (40). The front head (31) is disposed to cover a hollow (first cylinder bore (41)) of the first cylinder (40) from above. The main shaft portion (26) of the drive shaft (25) is inserted into a center portion of the front head (31). The front head (31) rotatably supports the drive shaft (25).

[0045] The middle plate (32) is sandwiched between the first cylinder (40) and the second cylinder (50). The middle plate (32) is disposed to cover the hollow (first cylinder bore (41)) of the first cylinder (40) from below. The middle plate (32) is disposed to cover a hollow (second cylinder bore (51)) of the second cylinder (50) from above.

[0046] The rear head (33) is stacked on the bottom of the second cylinder (50). The rear head (31) is disposed to cover the hollow (second cylinder bore (51)) of the second cylinder (50) from below. The main shaft portion (26) of the drive shaft (25) is inserted into a center portion of the rear head (31). The rear head (31) rotatably supports the drive shaft (25).

[0047] The first cylinder (40) is a substantially annular thick member. The first cylinder (40) has the first cylinder bore (41) at the center portion. The first cylinder bore (41) is a circular hole penetrating the first cylinder (40) in the thickness direction. The first cylinder bore (41) serves as a closed space defined by the front head (31) and the middle plate (32). The first cylinder (40) is fixed to the barrel (12) of the casing (11) with the centerline of the first cylinder bore (41) extending in the axial direction (vertical direction).

[0048] As illustrated in FIG. 4, the first cylinder (40) has a first bush hole (43a) and a first blade hole (43b). The first bush hole (43a) and the first blade hole (43b) penetrate the first cylinder (40) in the axial direction (thickness direction). The first bush hole (43a) and the first blade hole (43b) are each substantially in a circular shape. The first bush hole (43a) is open to the first cylinder bore (41). The first blade hole (43b) is located outside the first bush hole (43a) in the radial direction of the first cylinder (40), and communicates with the first bush hole (43a).

[0049] A pair of first bushes (48) is fitted in the first bush hole (43a). Each of the first bushes (48) is a semi-cylindrical member. The flat surfaces of the pair of first bushes (48) face each other with a space therebetween. The pair of first bushes (48) can swing about the centerline of the first bush hole (43a). The pair of first bushes (48) sandwiches a first blade (47), which will be described later, and restricts rotation of a first piston (45) on its own axis.

[0050] The first cylinder bore (41) houses the first piston (45). The first piston (45) has a first roller (46) and the first blade (47). The first roller (46) is a cylindrical member. The first eccentric shaft portion (27) of the drive shaft (25) is fitted in the first roller (46). The outer peripheral surface of the first roller (46) is in sliding contact with the inner peripheral surface of the first cylinder (40).

[0051] A space for compressing the refrigerant is formed between the outer peripheral surface of the first roller (46) and the inner peripheral surface of the first cylinder (40). This space is a first compression chamber (41s) formed by part of the first cylinder bore (41). The first roller (46) rotates integrally with the first eccentric shaft portion (27). The first roller (46) and the first eccentric shaft portion (27) form a first eccentric portion (44). As described above, the compression mechanism (30) includes the first eccentric portion (44) housed in the first cylinder (40). The first compression chamber (41s) is formed between the inner peripheral surface of the first cylinder (40) and the first eccentric portion (44).

[0052] The first blade (47) is provided on the outer peripheral surface of the first roller (46) and extends outward in the radial direction of the first roller (46). The first blade (47) is sandwiched between the pair of first bushes (48) so as to be movable back and forth. A tip end portion of the first blade (47) is housed in the first blade hole (43b). The space between the outer peripheral surface of the first roller (46) and the inner peripheral surface of the first cylinder (40) is divided into a first low-pressure chamber and a first high-pressure chamber by the first blade (47).

[0053] A first suction port (42) is formed in the first cylinder (40). The first suction port (42) penetrates the first cylinder (40) in the radial direction. One end of the first suction port (42) is open in the inner peripheral surface of the first cylinder (40), and communicates with the first low-pressure chamber at a position adjacent to the first bushes (48) (position on the right side of the first bushes (48) in FIG. 4). The other end of the first suction port (42) is open in the outer peripheral surface of the first cylinder (40). The suction pipe (15) is connected to the other end of the first suction port (42).

[0054] A first discharge port (49) is formed in the front head (31). The first discharge port (49) penetrates the front head (31) in the axial direction. One end of the first discharge port (49) is open in the lower surface of the front head (31), and communicates with the first high-pressure chamber at a position on the opposite side to the first suction port (42) with respect to the first bushes (48) (position on the left side of the first bushes (48) in FIG. 4). The other end of the first discharge port (49) is open in the upper surface of the front head (31).

[0055] A first discharge valve (60) is provided on the upper surface of the front head (31). The first discharge valve (60) opens and closes the first discharge port (49). The first discharge valve (60) is, for example, a reed valve. The first discharge valve (60) is in a closed state closing the first discharge port (49) while a gas pressure in the first high-pressure chamber is lower than a gas pressure in the casing (11) (pressure inside the dome). The first discharge valve (60) is in an open state opening the first discharge port (49) when the gas pressure in the first high-pressure chamber exceeds the pressure inside the dome.

[0056] The second cylinder (50) is a substantially annular thick member. The second cylinder (50) has the second cylinder bore (51) at the center portion. The second cylinder bore (51) is a circular hole penetrating the second cylinder (50) in the thickness direction. The second cylinder bore (51) serves as a closed space defined by the middle plate (32) and the rear head (33). The second cylinder (50) is provided with the centerline of the second cylinder bore (51) extending in the axial direction (vertical direction).

[0057] As illustrated in FIG. 5, the second cylinder (50) has a second bush hole (53a) and a second blade hole (53b). The second bush hole (53a) and the second blade hole (53b) penetrate the second cylinder (50) in the axial direction (thickness direction). The second bush hole (53a) and the second blade hole (53b) are each substantially in a circular shape. The second bush hole (53a) is open to the second cylinder bore (51). The second blade hole (53b) is located outside the second bush hole (53a) in the radial direction of the second cylinder (50), and communicates with the second bush hole (53a).

[0058] A pair of second bushes (58) is fitted in the second bush hole (53a). Each of the second bushes (58) is a semi-cylindrical member. The flat surfaces of the pair of second bushes (58) face each other with a space therebetween. The pair of second bushes (58) can swing about the centerline of the second bush hole (53a). The pair of second bushes (58) sandwiches a second blade (57), which will be described later, and restricts rotation of a second piston (55) on its own axis.

[0059] The second cylinder bore (51) houses the second piston (55). The second piston (55) has a second roller (56) and the second blade (57). The second roller (56) is a cylindrical member. The second eccentric shaft portion (28) of the drive shaft (25) is fitted in the second roller (56). The outer peripheral surface of the second roller (56) is in sliding contact with the inner peripheral surface of the second cylinder (50).

[0060] A space for compressing the refrigerant is formed between the outer peripheral surface of the second roller (56) and the inner peripheral surface of the second cylinder (50). This space is a second compression chamber (51s) formed by part of the second cylinder bore (51). The second roller (56) rotates integrally with the second eccentric shaft portion (28).

The second roller (56) and the second eccentric shaft portion (28) form a second eccentric portion (54). As described above, the compression mechanism (30) includes the second eccentric portion (54) housed in the second cylinder (50). The second compression chamber (51s) is formed between the inner peripheral surface of the second cylinder (50) and the second eccentric portion (54).

[0061] The second blade (57) is provided on the outer peripheral surface of the second roller (56) and extends outward in the radial direction of the second roller (56). The second blade (57) is sandwiched between the pair of second bushes (58) so as to movable back and forth. A tip end portion of the second blade (57) is housed in the second blade hole (53b). The space between the outer peripheral surface of the second roller (56) and the inner peripheral surface of the second cylinder (50) is divided into a second low-pressure chamber and a second high-pressure chamber by the second blade (57).

[0062] A second suction port (52) is formed in the second cylinder (50). The second suction port (52) penetrates the second cylinder (50) in the radial direction. One end of the second suction port (52) is open in the inner peripheral surface of the second cylinder (50), and communicates with the second low-pressure chamber at a position adjacent to the second bushes (58) (position on the right side of the second bushes (58) in FIG. 5). The other end of the second suction port (52) is open in the outer peripheral surface of the second cylinder (50). The suction pipe (15) is connected to the other end of the second suction port (52).

[0063] A second discharge port (59) is formed in the rear head (33). The second discharge port (59) penetrates the rear head (33) in the axial direction. One end of the second discharge port (59) is open in the upper surface of the rear head (33), and communicates with the second high-pressure chamber at a position on the opposite side to the second suction port (52) with respect to the second bushes (58) (position on the left side of the second bushes (58) in FIG. 5). The other end of the second discharge port (59) is open in the lower surface of the rear head (33).

[0064] A second discharge valve (61) is provided on the lower surface of the rear head (33). The second discharge valve (61) opens and closes the second discharge port (59). The second discharge valve (61) is, for example, a reed valve. The second discharge valve (61) is in a closed state closing the second discharge port (59) while a gas pressure in the second high-pressure chamber is lower than the pressure inside the dome. The second discharge valve (61) is in an open state opening the second discharge port (59) when the gas pressure in the second high-pressure chamber exceeds the pressure inside the dome.

[0065] In the compression mechanism (30), the first eccentric portion (44) eccentrically rotates in the first cylinder (40) along with the rotation of the drive shaft (25). As the volume of the first low-pressure chamber gradually increases with eccentric rotation of the first eccentric portion (44), the refrigerant flowing through the suction pipe (15) is sucked into the first low-pressure chamber through the first suction port (42). Further eccentric rotation of the first eccentric portion (44) causes isolation of the first low-pressure chamber from the first suction port (42), and the isolated space serves as the first high-pressure chamber.

[0066] The gas pressure in the first high-pressure chamber increases as the volume of the first high-pressure chamber gradually decreases along with further eccentric rotation of the first eccentric portion (44). When the gas pressure in the first high-pressure chamber exceeds the pressure inside the dome, the first discharge valve (60) is opened, and the refrigerant in the first high-pressure chamber flows out of the compression mechanism (30) through the first discharge port (49).

[0067] Along with rotation of the drive shaft (25), the first eccentric portion (44) eccentrically rotates, and the second eccentric portion (54) also eccentrically rotates in the second cylinder (50). As the volume of the second low-pressure chamber gradually increases with the eccentric rotation of the second eccentric portion (54), the refrigerant flowing through the suction pipe (15) is sucked into the second low-pressure chamber through the second suction port (52). Further eccentric rotation of the second eccentric portion (54) causes isolation of the second low-pressure chamber from the second suction port (52), and the isolated space serves as the second high-pressure chamber.

[0068] The gas pressure in the second high-pressure chamber increases as the volume of the second high-pressure chamber gradually decreases along with further eccentric rotation of the second eccentric portion (54). When the gas pressure in the second high-pressure chamber exceeds the pressure inside the dome, the second discharge valve (61) is opened, and the refrigerant in the second high-pressure chamber flows out of the compression mechanism (30) through the second discharge port (59).

[0069] The high-pressure refrigerant having flowed out of the compression mechanism (30) flows upward in the internal space of the casing (11), and passes through a core cut (not illustrated) or other portions of the motor (21). Then, the high-pressure refrigerant having flowed upward of the motor (21) is transferred to the refrigerant circuit (1a) through the discharge pipe (16).

[0070] As illustrated in FIG. 2, an oil supply siphon pipe (36) is connected to the rear head (33). The upper end of the oil supply siphon pipe (36) is connected to a second oil passage (37). The second oil passage (37) continuously penetrates the rear head (31), the second cylinder (50), the middle plate (32), and the first cylinder (40). The lower end of the oil supply siphon pipe (36) is immersed in the oil in the oil reservoir (18). The oil supply siphon pipe (36) sucks up the oil from the oil reservoir (18) and supplies the oil to the first blade hole (43b) and the second blade hole (53b) through the second oil passage (37).

<Accumulator>

[0071] An accumulator (80) is connected to the upstream side of the rotary compressor (10). The accumulator (80) temporarily stores the refrigerant that is to be sucked into the rotary compressor (10) and performs gas-liquid separation for a liquid refrigerant and oil contained in the gas refrigerant. The accumulator (80) includes a closed container (81), an inlet pipe (82), and an outlet pipe (83).

[0072] The closed container (81) is configured as a vertically long cylindrical member. The inlet pipe (82) is a pipe through which the refrigerant flows into the closed container (81). The inlet pipe (82) is connected to an upper portion of the closed container (81). The lower end of the inlet pipe (82) is open to the internal space of the closed container (81) at a position near the top of the closed container (81). The upper end of the inlet pipe (82) is connected to the refrigerant circuit (1a).

[0073] The outlet pipe (83) is a pipe through which the refrigerant flows out of the closed container (81). Two outlet pipes (83) are connected to a lower portion of the closed container (81). Each of the outlet pipes (83) has an upper end portion extending in the vertical direction in the closed container (81) and opening to the internal space of the closed container (81) at a position near the top of the closed container (81). Each of the outlet pipes (83) has a lower end portion extending downward from the lower end of the closed container (81) and bent to be connected to the suction pipe (15) of the rotary compressor (10).

<Balancer>

[0074] As also illustrated in FIGS. 6 to 8, the balancer (70) is for compensating the unbalance due to the eccentric rotational motion of the compression mechanism (30), and is provided on the rotor (23). The balancer (70) has a first balance weight (71) and a second balance weight (72).

[0075] The first balance weight (71) is an arc plate-shaped member. The first balance weight (71) is disposed on one end portion of the rotor (23) closer to the compression mechanism (30) in the axial direction, that is, a lower end portion of the rotor (23). The first balance weight (71) is attached to the lower surface of the end plate (24) on the lower side by placing the arc shape along the circumferential direction and fixing both end portions to the end plate (24) on the lower side with two rivets (75).

[0076] The second balance weight (72) is a cylindrical member. The second balance weight (72) is disposed on the other end portion of the rotor (23) opposite to the compression mechanism (30) in the axial direction, that is, an upper end portion of the rotor (23). The second balance weight (72) is attached to the upper surface of the end plate (24) on the upper side by placing the second balance weight (72) at a position opposed to the first balance weight (71) with respect to the axis (AC) of the drive shaft (25) in plan view and fixing a center portion to the end plate (24) on the upper side with one rivet (75).

[0077] As also illustrated in FIG. 9, the first balance weight (71) and the second balance weight (72) are eccentric to the axis (AC) of the drive shaft (25) to opposite sides, and are arranged with a phase difference of 180° in the circumferential direction. The first balance weight (71) is eccentric in the same direction as the first eccentric portion (44). The first balance weight (71) and the first eccentric portion (44) are in phase with each other. The second balance weight (72) is eccentric in the same direction as the second eccentric portion (54). The second balance weight (72) and the second eccentric portion (54) are in phase with each other.

[0078] As described above, the first balance weight (71), together with the first eccentric portion (44), is eccentric to the axis (AC) of the drive shaft (25) to one side, and the second balance weight (72), together with the second eccentric portion (54), is eccentric to the axis (AC) of the drive shaft (25) to the other side. The first balance weight (71) and the second balance weight (72) balance forces causing parallel displacement of the drive shaft (25) based on the inertial force acting on the first eccentric portion (44) and the second eccentric portion (54), thereby maintaining static balance (balance of the parallel displacement force).

<Configuration Related to Static Balance during High-Speed Operation>

[0079] In an operation of the rotary compressor (10), the drive shaft (25) is slightly bent due to a compressive load from the compression mechanism (30), and the rotor (23) deviates from a preset rotation center. The centrifugal force of the rotor (23) acts in combination with the inertial force acting on the second balance weight (72), which increases bending of the drive shaft (25). As a result, a high-speed operation significantly destroys the static balance between the compression mechanism (30) (the first eccentric portion (44) and the second eccentric portion (54)) and the balancer (70) (the first balance weight (71) and the second balance weight (72)), and vibration increases.

[0080] To address this, the rotary compressor (10) of this example is configured such that the centrifugal force acting on the second balance weight (72) becomes smaller than the centrifugal force acting on the first balance weight (71) to achieve static balance in a high-speed operation, in consideration of the fact that the centrifugal force of the rotor (23) acts in combination with the inertial force acting on the second balance weight (72) when the drive shaft (25) is bent.

[0081] Specifically, the mass and the eccentric distance of each of the first balance weight (71) and the second balance weight (72) satisfy a relationship expressed by Expression (1) below, where the mass of the first balance weight (71) is m1 [g], the eccentric distance of the center of gravity (GC1) of the first balance weight (71) from the axis (AC) of the drive shaft (25) is r1 [mm], the mass of the second balance weight (72) is m2 [g], and the eccentric distance of the center of gravity (GC2) of the second balance weight (72) from the axis (AC) of the drive shaft (25) is r2 [mm],

$$m1 \times r1 > m2 \times r2 \qquad (1)$$

[0082] A static balance relation value corresponding to the ratio between the total centrifugal force acting on the first eccentric portion (44) and the first balance weight (71) and the total centrifugal force acting on the second eccentric portion (54) and the second balance weight (72) is adjusted according to the maximum number of rotations of the rotary compressor (10) so as to achieve the static balance of the entire rotary system including the drive mechanism (20), the compression mechanism (30), and the balancer (70).

[0083] Specifically, the mass and the eccentric distance of each of the first eccentric portion (44), the second eccentric portion (54), the first balance weight (71), and the second balance weight (72) satisfy a relationship expressed by Expression (2) below, where the mass of the first eccentric portion (44) is m3 [g], the eccentric distance of the center of gravity (GC3) of the first eccentric portion (44) from the axis (AC) of the drive shaft (25) is r3 [mm], the mass of the second eccentric portion (54) is m4 [g], the eccentric distance of the center of gravity (GC4) of the second eccentric portion (54) from the axis (AC) of the drive shaft (25) is r4 [mm], and the maximum number of rotations of the compression mechanism (30) is Nmax [rps].

$$1.2 - 0.002 \times \mathrm{Nmax} \leq (m2 \times r2 + m4 \times r4)/(m1 \times r1 + m3 \times r3) \leq 0.98 \qquad (2)$$

[0084] The mass of the first eccentric portion (44) is calculated by adding the mass of the first eccentric shaft portion (27) and the mass of the first piston (45). The mass of the second eccentric portion (54) is calculated by adding the mass of the second eccentric shaft portion (28) and the mass of the second piston (55). It is preferable that the mass and the eccentric distance of each of the first eccentric portion (44), the second eccentric portion (54), the first balance weight (71), and the second balance weight (72) further satisfy a relationship expressed by Expression (3) below in order to ensure a relatively wide range of the number of rotations that can reduce vibration of the rotary compressor (10).

$$0.88 \leq (m2 \times r2 + m4 \times r4)/(m1 \times r1 + m3 \times r3) \qquad (3)$$

[0085] In the rotary compressor (10) of this example, the mass of the first eccentric portion (44) and the mass of the second eccentric portion (54) are designed to be equal to each other, and the eccentric distance of the first eccentric portion (44) and the eccentric distance of the second eccentric portion (54) are designed to be equal to each other. On the other hand, the mass of the first balance weight (71) is smaller than the mass of the second balance weight (72), and the eccentric distance of the first balance weight (71) is longer than the eccentric distance of the second balance weight (72).

[0086] In the rotary compressor (10) of this example, the static balance is maintained in the state in which the drive shaft (25) is bent. Thus, the diameter of the main shaft portion (26) of the drive shaft (25) is designed to be relatively small to allow bending. As illustrated in FIG. 3, the moment of inertia Is of the main shaft portion (26) is expressed by Expression (4) below, where the diameter of the main shaft portion (26) of the drive shaft (25) is Φ [mm]. This moment of inertia relates to the amount of bending of the main shaft portion (26). The amount of bending of the main shaft portion (26) is inversely proportional to the fourth power of the diameter Φ.

$$\mathrm{Is} = \pi \times \Phi^4/64 \qquad (4)$$

[0087] In the rotary compressor (10), the larger the sum of the volume of the first compression chamber (41s) and the volume of the second compression chamber (51s), the larger the sum of the mass of the first eccentric portion (44) and the mass of the second eccentric portion (54) tends to be. The larger the sum of the mass of the first eccentric portion (44) and the mass of the second eccentric portion (54), the greater the centrifugal force acting on the main shaft portion (26) of the drive shaft (25) from the compression mechanism (30) becomes. Thus, the amount of bending of the main shaft portion (26) increases.

[0088] From the above-described point, it is preferable to satisfy a relationship expressed by Expression (5) below, where the sum of the volume of the first compression chamber (41s) and the volume of the second compression chamber (51s) is Vcc [cc]. When the value of Vcc/Φ^4 is 3×10^{-4} or more, the main shaft portion (26) is relatively thin with respect to the total volume of the first compression chamber (41s) and the second compression chamber (51s), and therefore, the main shaft portion (26) is easily bent in an operation of the rotary compressor (10) due to the centrifugal force generated by operation of the compression mechanism (30). In the case where the main shaft portion (26) is relatively thin as described above, the problem of increased vibration during high speed operation becomes prominent in a conventional rotary compressor.

$$\mathrm{Vcc}/\Phi^4 \geq 3 \times 10^{-4} \qquad (5)$$

[0089] The volume V1 of the first compression chamber (41s) is expressed by Expression (6) below, where the inner diameter area of the first cylinder (40) is Sc1, the length of the first cylinder bore (41) is H1, the outer diameter area of the first roller (46) is Sr1, and the thickness of the first roller (46) is T1. Further, the volume V2 of the second compression chamber (51s) is expressed by Expression (7) below, where the inner diameter area of the second cylinder (50) is Sc2, the length of the second cylinder bore (51) is H2, the outer diameter area of the second roller (56) is Sr2, and the thickness of the second roller (56) is T2. The sum Vcc of the volume of the first compression chamber (41s) and the volume of the second compression chamber (51s) is expressed by Expression (8) below.

$$\mathrm{V1} = (\mathrm{Sc1} \times \mathrm{H1}) - (\mathrm{Sr1} \times \mathrm{T1}) \qquad (6)$$

$$\mathrm{V2} = (\mathrm{Sc2} \times \mathrm{H2}) - (\mathrm{Sr2} \times \mathrm{T2}) \qquad (7)$$

$$\mathrm{Vcc} = \mathrm{V1} + \mathrm{V2} \qquad (8)$$

[0090] For example, the diameter of the main shaft portion (26) is 16 mm or less. The sum of the volume of the first compression chamber (41s) and the volume of the second compression chamber (51s) is, for example, 20 cc or more. In addition, in order to ensure the capacity of the rotary compressor (10), the first eccentric portion (44) and the second eccentric portion (54) are designed such that the centrifugal force acting on the first eccentric portion (44) and the second eccentric portion (54) is relatively great. Specifically, the mass and the eccentric distance of each of the first eccentric portion (44) and the second eccentric portion (54) satisfy a relationship expressed by Expression (9) below.

$$(\mathrm{m3} \times \mathrm{r3} + \mathrm{m4} \times \mathrm{r4})/2 \geq 600 \qquad (9)$$

[0091] In view of the above-described point, a specific example is that each of the mass of the first eccentric portion (44) and the mass of the second eccentric portion (54) is 187.4 g. Each of the eccentric distance of the first eccentric portion (44) and the eccentric distance of the second eccentric portion (54) is 4.95 mm. The mass of the first balance weight (71) is 10.46 g. The mass of the second balance weight (72) is 5.4 g. The eccentric distance of the first balance weight (71) is 18.63 mm. The eccentric distance of the second balance weight (72) is 23.5 mm. The static balance relation value ((m2 × r2 + m4 × r4)/(m1 × r1 + m3 × r3)) in this example is 0.939.

- Evaluation of Vibration Reduction Effect -

[0092] The acceleration of vibration generated in a simulation of the operation of the rotary compressor (10) was obtained for a Comparative Example having a static balance relation value ((m2 × r2 + m4 × r4)/(m1 × r1 + m3 × r3)) of 1.0 and Examples 1 to 16 having different static balance relation values between 0.84 and 0.99 in increments of 0.01.

[0093] As shown in FIG. 10, the rotary compressors (10) of Comparative Example and Examples 1 to 16 have the same value of the product of the mass and eccentric distance of the first balance weight (71) (m1 × r1), the same value of the product of the mass and eccentric distance of the first eccentric portion (44) (m3 × r3), and the same value of the product of the mass and eccentric distance of the second eccentric portion (54) (m4 × r4), and are different from one another only in the value of the product of the mass and eccentric distance of the second balance weight (71) (m2 × r2).

[0094] In the rotary compressors (10) of Comparative Example and Examples 1 to 16, the value of the product of the mass and eccentric distance of the first balance weight (71) (m1 × r1) is 194.9 g mm, and the value of the product of the mass and eccentric distance of the first eccentric portion (44) (m3 × r3) and the value of the product of the mass and eccentric distance of the second eccentric portion (54) (m4 × r4) are the same as each other, i.e., 927.5 g·mm.

[0095] In the rotary compressor (10) of Comparative Example, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 195.1 g·mm, and the static balance relation value is 1.0.

[0096] In the rotary compressor (10) of Example 1, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 183.8 g·mm, and the static balance relation value is 0.99. In the rotary compressor

(10) of Example 2, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 172.5 g mm, and the static balance relation value is 0.98.

[0097] In the rotary compressor (10) of Example 3, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 161.2 g·mm, and the static balance relation value is 0.97. In the rotary compressor (10) of Example 4, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 149.9 g mm, and the static balance relation value is 0.96.

[0098] In the rotary compressor (10) of Example 5, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 138.7 g·mm, and the static balance relation value is 0.95. In the rotary compressor (10) of Example 6, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 127.6 g mm, and the static balance relation value is 0.94.

[0099] In the rotary compressor (10) of Example 7, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 116.3 g mm, and the static balance relation value is 0.93. In the rotary compressor (10) of Example 8, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 105.0 g mm, and the static balance relation value is 0.92.

[0100] In the rotary compressor (10) of Example 9, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 94.0 g mm, and the static balance relation value is 0.91. In the rotary compressor (10) of Example 10, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 82.7 g mm, and the static balance relation value is 0.90.

[0101] In the rotary compressor (10) of Example 11, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 71.4 g mm, and the static balance relation value is 0.89. In the rotary compressor (10) of Example 12, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 60.2 g mm, and the static balance relation value is 0.88.

[0102] In the rotary compressor (10) of Example 13, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 49.0 g mm, and the static balance relation value is 0.87. In the rotary compressor (10) of Example 14, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 37.8 g mm, and the static balance relation value is 0.86.

[0103] In the rotary compressor (10) of Example 15, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 26.5 g mm, and the static balance relation value is 0.85. In the rotary compressor (10) of Example 16, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is 15.3 g mm, and the static balance relation value is 0.84.

[0104] The simulation results for the rotary compressors (10) of Comparative Example and Examples 1 to 11 are shown in FIG. 11 in the form of a graph. In FIG. 11, the simulation results for the rotary compressors (10) of Examples 12 to 14 are omitted for the sake of convenience. As illustrated in FIG. 11, the vibration acceleration of all the rotary compressors (10) tends to increase once and then decrease to a local minimum value as the number of rotations increases, and tends to increase more along with a further increase in the number of rotations from the local minimum value.

[0105] The graph shows that the smaller the static balance relation value, the more the minimum value of the vibration acceleration tends to shift to a higher rotation side. That is, the number of rotations at the local minimum value of the vibration acceleration becomes greater in the order of the rotary compressor (10) of Comparative Example, the rotary compressor (10) of Example 1, the rotary compressor (10) of Example 2, the rotary compressor (10) of Example 3, the rotary compressor (10) of Example 4, the rotary compressor (10) of Example 5, the rotary compressor (10) of Example 6, the rotary compressor (10) of Example 7, the rotary compressor (10) of Example 8, the rotary compressor (10) of Example 9, the rotary compressor (10) of Example 10, and the rotary compressor (10) of Example 11.

[0106] Further, FIG. 12 shows the numerical values [m/s^2] of the vibration accelerations of the simulation results for the rotary compressors (10) of Comparative Example and Examples 1 to 17 at the number of rotations of 120 rps, 130 rps, 140 rps, 150 rps, 160 rps, 170 rps, and 180 rps. As illustrated in FIG. 12, the rotary compressors (10) of Examples 1 to 17 show lower vibration accelerations than the rotary compressor (10) of Comparative Example, which confirms the vibration reduction effect in a high-speed operation. In FIG. 12, the numerical values indicating the vibration reduction effect are underlined. This simulation shows that the vibration reduction effect is obtained only on the side of the higher number of rotations as the static balance relation value is smaller, and the vibration reduction effect is obtained in a wide range of the number of rotations on the side of greater static balance relation values (i.e., side closer to 1.0).

- Features of First Embodiment -

[0107] In the rotary compressor (10) of the first embodiment, the value of the product of the mass and eccentric distance of the second balance weight (72) (m2 × r2) is smaller than the value of the product of the mass and eccentric distance of the first balance weight (71) (m1 × r1). This configuration reduces the inertial force of the second balance weight (72) acting in combination with the centrifugal force of the rotor (23) deviated from the predetermined rotation center when the drive shaft (25) is bent in the operation of the rotary compressor (10), thereby making it possible to reduce an increase in the bending of the drive shaft (25). The vibration in a high-speed operation of the rotary compressor (10) can thus be reduced.

[0108] The rotary compressor (10) of the first embodiment has the maximum number of rotations of 120 rps or more, which is a relatively great number of rotations. The faster the rotary compressor (10) rotates, the more the static balance between the first eccentric portion (44) and second eccentric portion (54) and the first balance weight (71) and second balance weight (72) tends to be destroyed significantly due to bending of the drive shaft (25), which tends to increase vibration of the rotary compressor (10) in operation. Thus, the technology according to the present disclosure is effective in the rotary compressor (10) that operates at the relatively great number of rotations.

[0109] In the rotary compressor (10) of the first embodiment, the static balance relation value obtained by dividing the sum of the value of the product of the mass and eccentric distance of the second balance weight (72) and the value of the product of the mass and eccentric distance of the second eccentric portion (54) ($m2 \times r2 + m4 \times r4$) by the sum of the value of the product of the mass and eccentric distance of the first balance weight (71) and the value of the product of the mass and eccentric distance of the first eccentric portion (44) ($m1 \times r1 + m3 \times r3$) is $1.2 - 0.002 \times$ Nmax (maximum number of rotations) or more and 0.98 or less. This can reduce vibration in a high-speed operation in accordance with the maximum number of rotations of the rotary compressor (10).

[0110] In the rotary compressor (10) of the first embodiment, the static balance relation value is 0.88 or more. If the static balance relation value is smaller than 0.88, the vibration reduction effect in a high-speed operation of the rotary compressor (10) can only be expected only in a rotation number range relatively close to the maximum number of rotations of the compression mechanism (30). On the other hand, if the static balance relation value is 0.88 or more, vibration of the rotary compressor (10) can be reduced in a relatively wide range of the number of rotations from the maximum number of rotations of the compression mechanism (30).

[0111] In the rotary compressor (10) of the first embodiment, an index value indicating how easily the main shaft portion (26) is bent, which is obtained by dividing the sum Vcc of the volume of the first compression chamber (41s) and the volume the second compression chamber (51s) by the diameter Φ of the main shaft portion (26) of the drive shaft (25), is 3×10^{-4} or more. This configuration makes it possible to bend the main shaft portion (26) suitably such that the static balance between the first eccentric portion (44) and second eccentric portion (54) and the first balance weight (71) and second balance weight (72) can be maintained in the operation of the rotary compressor (10).

[0112] In the rotary compressor (10) of the first embodiment, the main shaft portion (26) of the drive shaft (25) has a relatively small diameter of 16 mm or less, and an average value of the values of the product of the mass and the eccentric distance of the first eccentric portion (44) and the product of the mass and the eccentric distance of the second eccentric portion (54) is relatively great, that is, 600 or more.

[0113] As a method for reducing vibration of the rotary compressor (10) caused by bending of the drive shaft (25) in a high-speed operation, it is conceivable to increase the diameter of the main shaft portion (26) and increase the rigidity of the drive shaft (25), or to reduce the mass and the eccentric distance of each of the first eccentric portion (44) and the second eccentric portion (54) and reduce the size of the rotor. However, the former method increases a mechanical loss such as a bearing loss, and the efficiency of the rotary compressor (10) decreases. The latter method limits the capacity of the rotary compressor (10).

[0114] On the other hand, the drive shaft (25) whose main shaft portion (26) has a diameter of 16 mm or less can reduce the mechanical loss such as the bearing loss and increase the efficiency of the rotary compressors (10). It is also possible to ensure a relatively great capacity of the rotary compressor (10) when the average value of the values of the product of the mass and the eccentric distance of the first eccentric portion (44) and the product of the mass and the eccentric distance of the second eccentric portion (54) is 600 or more. It is thus possible to achieve the rotary compressor (10) with high efficiency and great capacity, which is less likely to vibrate in a high-speed operation.

[0115] In the refrigeration apparatus (1) of the first embodiment, the rotary compressor (10) is used in the refrigerant circuit (1a). Vibration in a high-speed operation is reduced in the rotary compressor (10). Use of the rotary compressor (10) in the refrigerant circuit (1a) contributes to execution of the refrigeration cycle with less vibration and noise.

<<Other Embodiments>>

[0116] As illustrated in FIG. 13, the compression mechanism (30) of the rotary compressor (10) of the above-described embodiment may be of a rolling piston type in which the first blade (47) of the first piston (45) is formed separately from the first roller (46). In such a compression mechanism (30), the flat plate-shaped first blade (47) is fitted in a first blade groove (90) extending in the radial direction of the first cylinder (40) so as to be movable move back and forth, and the first bushes (48) are omitted. The first blade (47) is pressed against the outer peripheral surface of the first roller (46) by a spring (91). A tip end portion of the first blade (47) is in sliding contact with the outer peripheral surface of the first roller (46). The same may also apply to the second piston (55).

[0117] In the rotary compressor (10) of the above-described embodiment, the mass of the first balance weight (71) and the mass of the second balance weight (72) may be the same as each other, or the eccentric distance of the first balance weight (71) and the eccentric distance of the second balance weight (72) may be the same as each other. The first balance weight (71) and the second balance weight (72) may have the same shape, which may be any shape. In short, it is only necessary to have the value of the product of the mass and the eccentric distance of the second balance weight (72) (m2 × r2) smaller than the value of the product of the mass and the eccentric distance of the first balance weight (71) (m1 × r1).

[0118] In the rotary compressor (10) of the above-described embodiment, the mass of the first eccentric portion (44) and the mass of the second eccentric portion (54) may be different from each other, and the eccentric distance of the first eccentric portion (44) and the eccentric distance of the second eccentric portion (54) may be different from each other. In short, it is only necessary that the static balance of the entire rotary system including the drive mechanism (20), the compression mechanism (30), and the balancer (70) is maintained in the operation of the rotary compressor (10).

[0119] While the embodiments and variations thereof have been described above, it will be understood that various changes in form and details may be made without departing from the spirit and scope of the claims. The foregoing embodiments and variations thereof may be combined and replaced with each other without deteriorating the intended functions of the present disclosure.

INDUSTRIAL APPLICABILITY

[0120] As described above, the present disclosure is useful for the rotary compressor and the refrigeration apparatus including the rotary compressor.

DESCRIPTION OF REFERENCE CHARACTERS

[0121]

AC	Axis
GC1	Center of Gravity of First Balance Weight
GC2	Center of Gravity of Second Balance Weight
GC3	Center of Gravity of First Eccentric Portion
GC4	Center of Gravity of Second Eccentric Portion
1	Refrigeration Apparatus
1a	Refrigerant Circuit
10	Rotary Compressor
21	Motor
23	Rotor
25	Drive Shaft
30	Compression Mechanism
40	First Cylinder
41s	First Compression Chamber
44	First Eccentric Portion
50	Second Cylinder
51s	Second Compression Chamber
54	First Eccentric Portion
70	Balancer
71	First Balance Weight
72	Second Balance Weight

Claims

1. A rotary compressor comprising:

a compression mechanism (30) configured to suck and compress a fluid; a drive shaft (25) configured to drive the compression mechanism (30); a motor (21) having a rotor (23) coupled to the drive shaft (25); and a balancer (70) provided on the rotor (23), wherein

the compression mechanism (30) has a first cylinder (40) and a second cylinder (50) sequentially arranged next to each other from a side closer to the rotor (23) in an axial direction along an axis (AC) of the drive shaft (25), a first eccentric portion (44) housed in the first cylinder (40), and a second eccentric portion (54) housed in the second cylinder (50),

the balancer (70) has a first balance weight (71) disposed at one end portion of the rotor (23) closer to the compression mechanism (30) in the axial direction, and a second balance weight (72) disposed at the other end

portion of the rotor (23) in the axial direction,
the first eccentric portion (44) and the first balance weight (71) are eccentric to the axis (AC) of the drive shaft (25) to one side, and the second eccentric portion (54) and the second balance weight (72) are eccentric to the axis (AC) of the drive shaft (25) to the other side,
along with rotation of the drive shaft (25), the first eccentric portion (44) eccentrically rotates in the first cylinder (40), and the second eccentric portion (54) eccentrically rotates in the second cylinder (50), and
a relationship of

$$m1 \times r1 > m2 \times r2$$

is satisfied, where a mass of the first balance weight (71) is m1, an eccentric distance of a center of gravity (GC1) of the first balance weight (71) from the axis (AC) of the drive shaft (25) is r1, a mass of the second balance weight (72) is m2, and an eccentric distance of a center of gravity (GC2) of the second balance weight (72) from the axis (AC) of the drive shaft (25) is r2.

2. The rotary compressor of claim 1, wherein a maximum number of rotations of the drive shaft (25) is 120 rps or more.

3. The rotary compressor of claim 1 or 2, wherein

a relationship of

$$1.2 - 0.002 \times Nmax \leq (m2 \times r2 + m4 \times r4)/(m1 \times r1 + m3 \times r3) \leq 0.98$$

is satisfied, where a mass of the first eccentric portion (44) is m3, an eccentric distance of a center of gravity (GC3) of the first eccentric portion (44) from the axis (AC) of the drive shaft (25) is r3, a mass of the second eccentric portion (54) is m4, an eccentric distance of a center of gravity (GC4) of the second eccentric portion (54) from the axis (AC) of the drive shaft (25) is r4, and a maximum number of rotations of the drive shaft (25) is Nmax [rps].

4. The rotary compressor of claim 3, wherein

a relationship of

$$0.88 \leq (m2 \times r2 + m4 \times r4)/(m1 \times r1 + m3 \times r3)$$

is satisfied.

5. The rotary compressor of any one of claims 1 to 4, wherein

a relationship of

$$Vcc/\Phi^4 \geq 3 \times 10^{-4}$$

is satisfied, where a sum of a volume of a first compression chamber (41s) formed between an inner peripheral surface of the first cylinder (40) and the first eccentric portion (44) and a volume of a second compression chamber (51s) formed between an inner peripheral surface of the second cylinder (50) and the second eccentric portion (54) is Vcc [cc], and a diameter of a main shaft portion (26) of the drive shaft (25) coupled to the rotor (23) is Φ [mm].

6. The rotary compressor of any one of claims 1 to 5, wherein

the drive shaft (25) has a main shaft portion (26) coupled to the rotor (23), the main shaft portion (26) having a diameter of 16 mm or less, and
a relationship of

$$(m3 \times r3 + m4 \times r4)/2 \geq 600$$

is satisfied, where a mass of the first eccentric portion (44) is m3 [g], an eccentric distance of a center of gravity

(GC3) of the first eccentric portion (44) from the axis (AC) of the drive shaft (25) is r3 [mm], a mass of the second eccentric portion (54) is m4 [g], and an eccentric distance of a center of gravity (GC4) of the second eccentric portion (54) from the axis (AC) of the drive shaft (25) is r4 [mm].

7. A refrigeration apparatus comprising: a refrigerant circuit (1a) configured to perform a refrigeration cycle, wherein the refrigerant circuit (1a) includes the rotary compressor (10) of any one of claims 1 to 6.

FIG.1

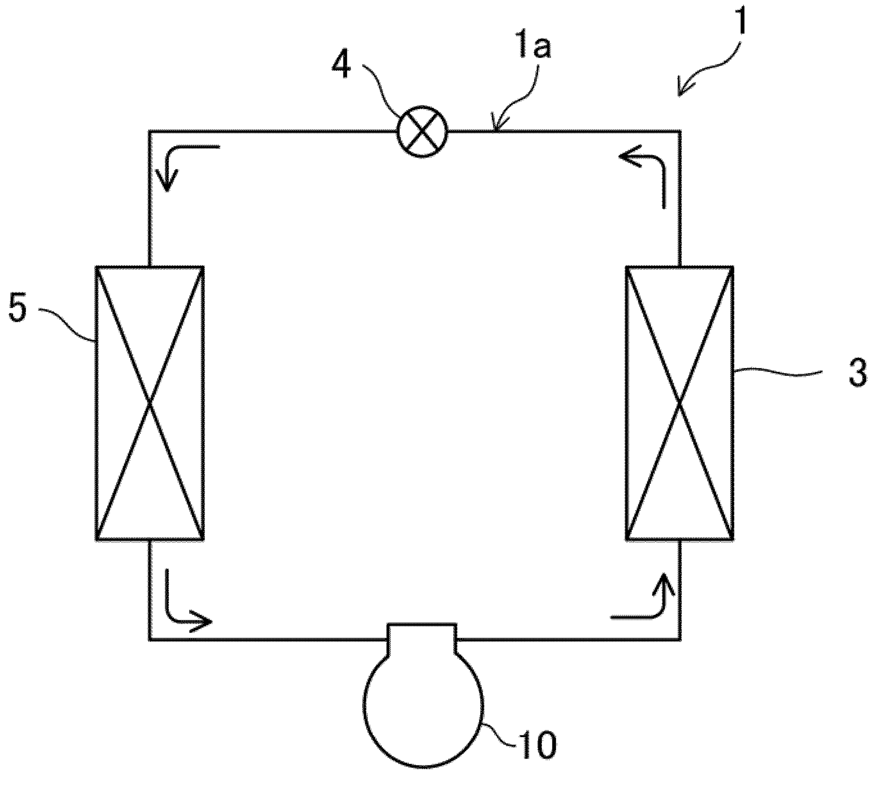


FIG.2

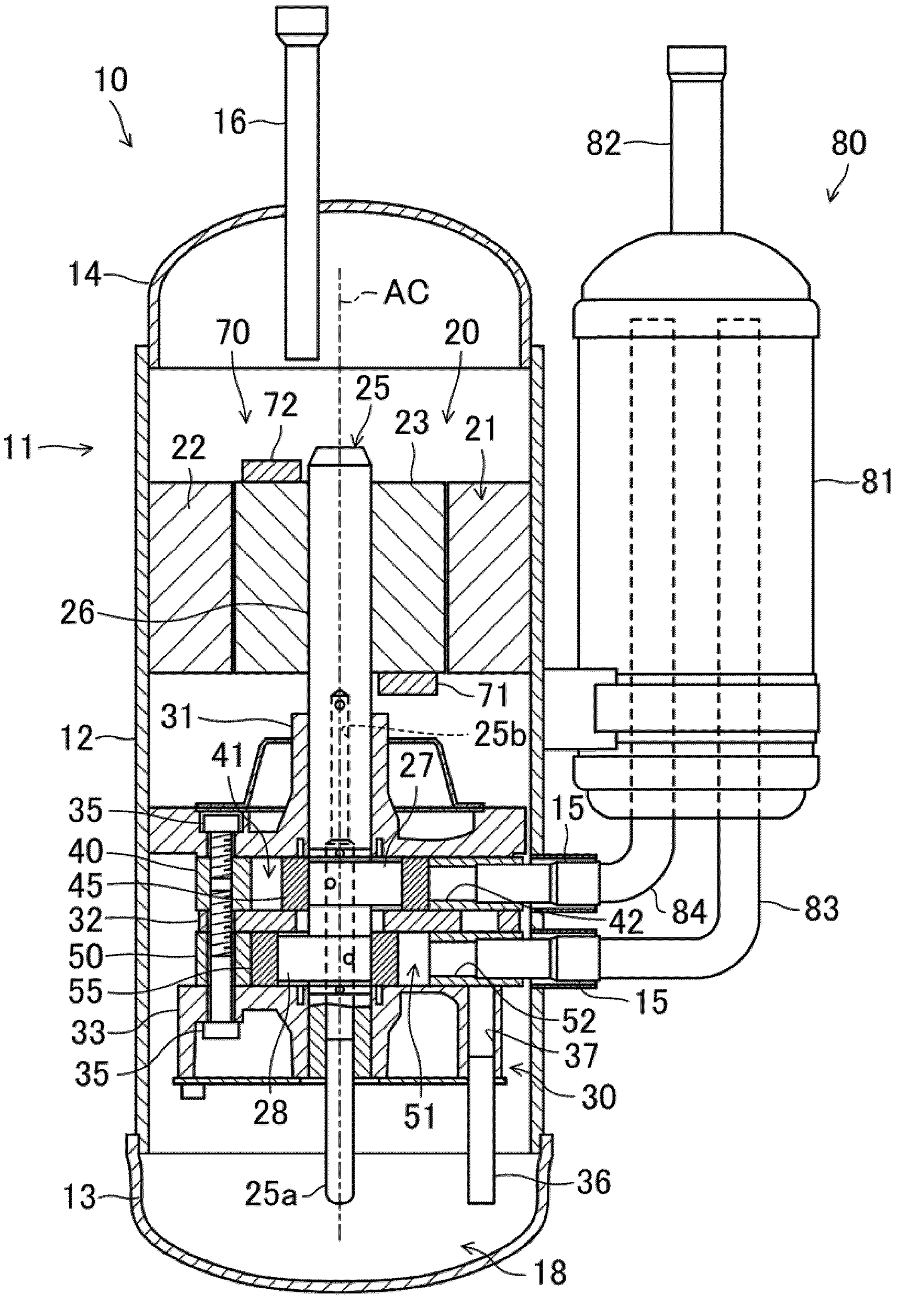

10
16
82
80
14
AC
70
20
25
72
23
21
11
22
81
26
71
31
25b
12
41
27
35
15
40
45
84
83
42
32
50
55
15
52
33
37
35
30
28
51
13
25a
36
18

FIG.3

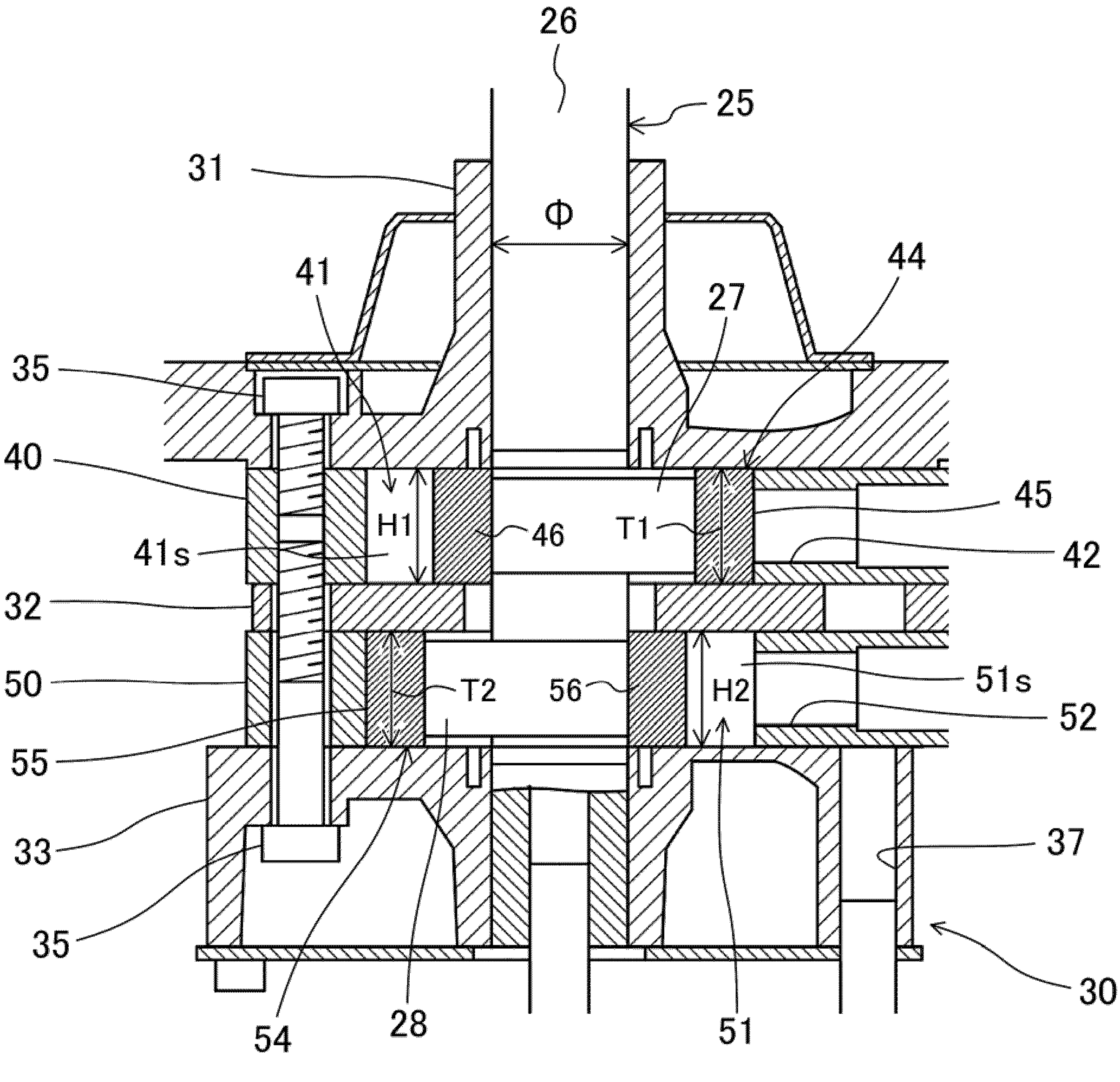

26
25
31
Φ
41
44
27
35
40
H1
46
T1
45
41s
42
32
50
T2
56
H2
51s
52
55
33
37
35
30
54
28
51

FIG.4

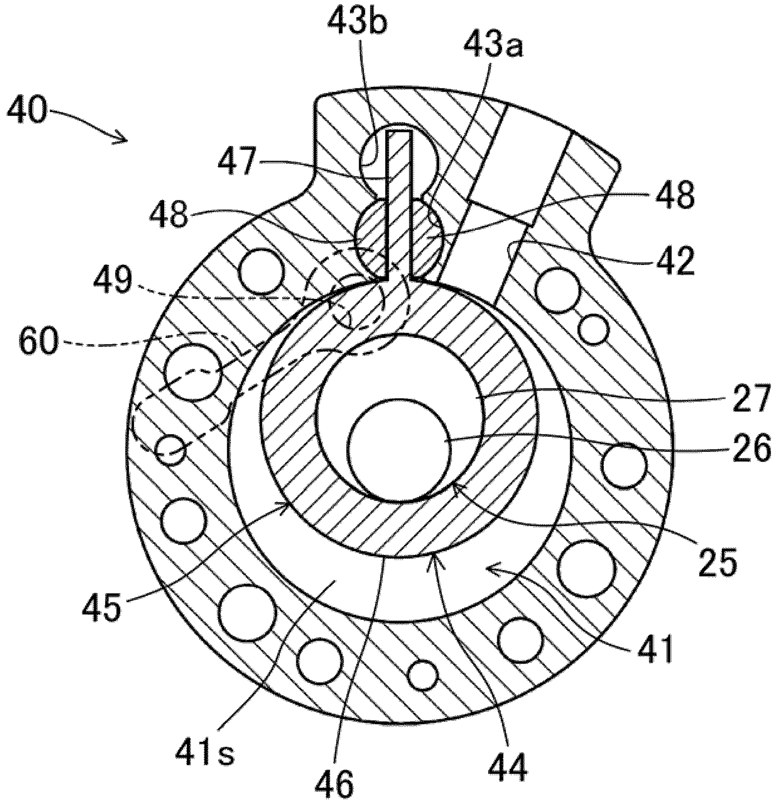

43b
43a
40
47
48
48
42
49
60
27
26
25
45
41
41s
46
44

FIG.5

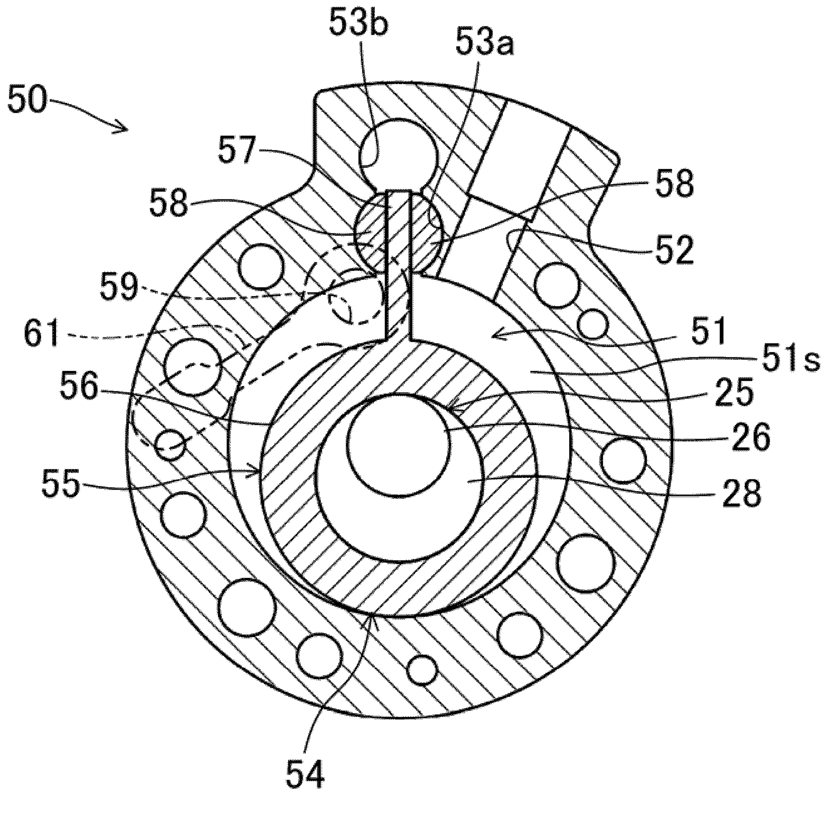

53b
53a
50
57
58
58
52
59
51
61
51s
56
25
26
55
28
54

FIG.6

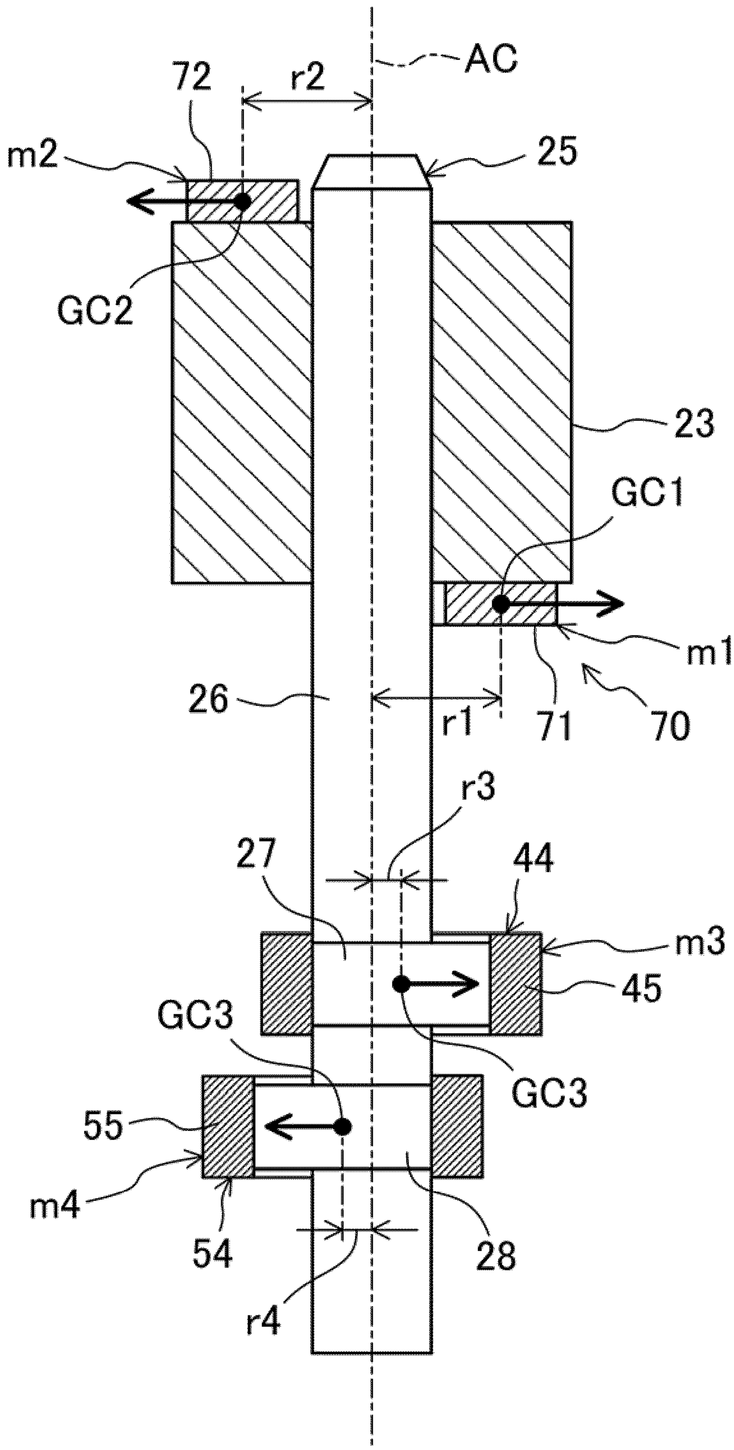

AC
72
r2
m2
25
GC2
23
GC1
m1
26
r1
71
70
r3
27
44
m3
45
GC3
GC3
55
m4
54
28
r4

FIG.7

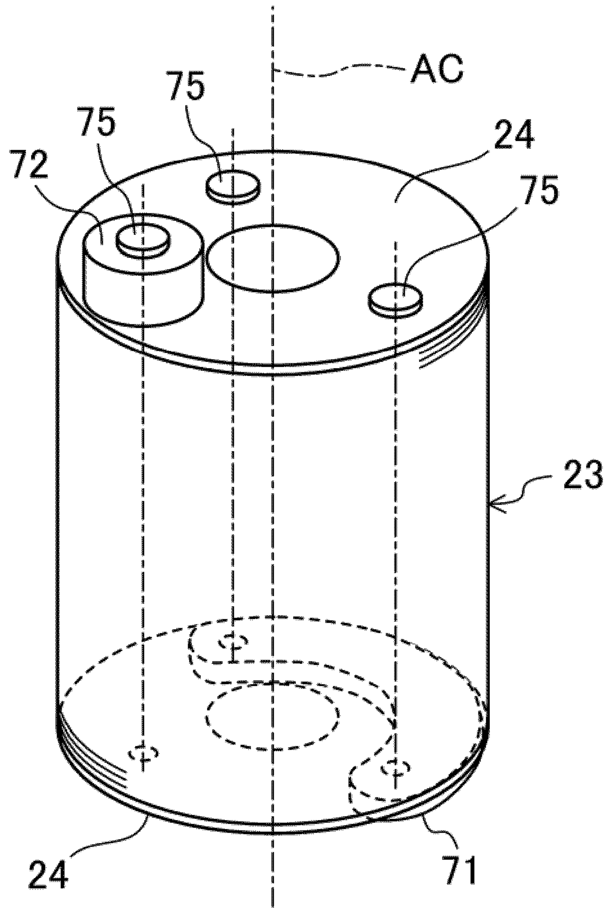


FIG.8

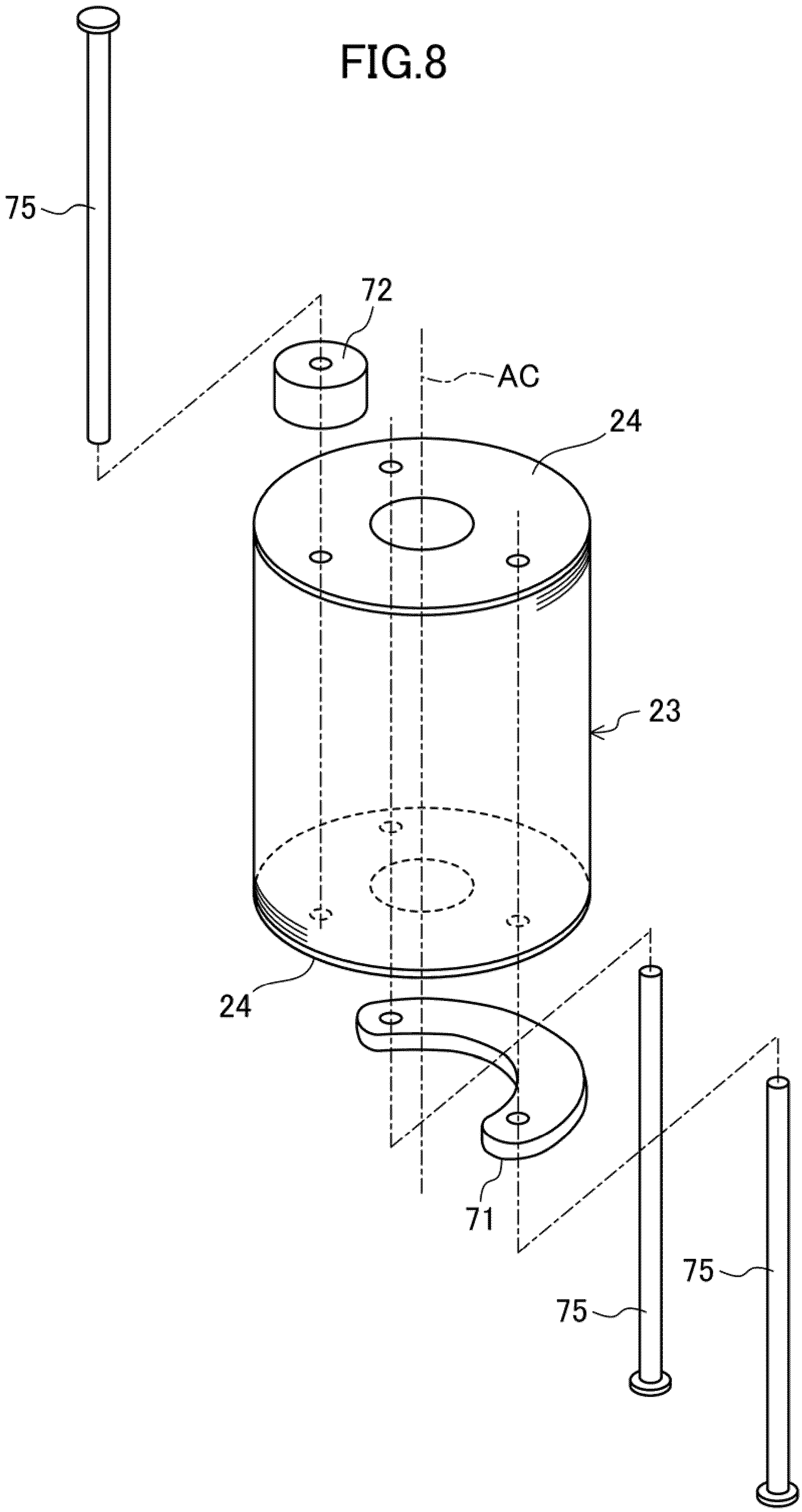


FIG.9

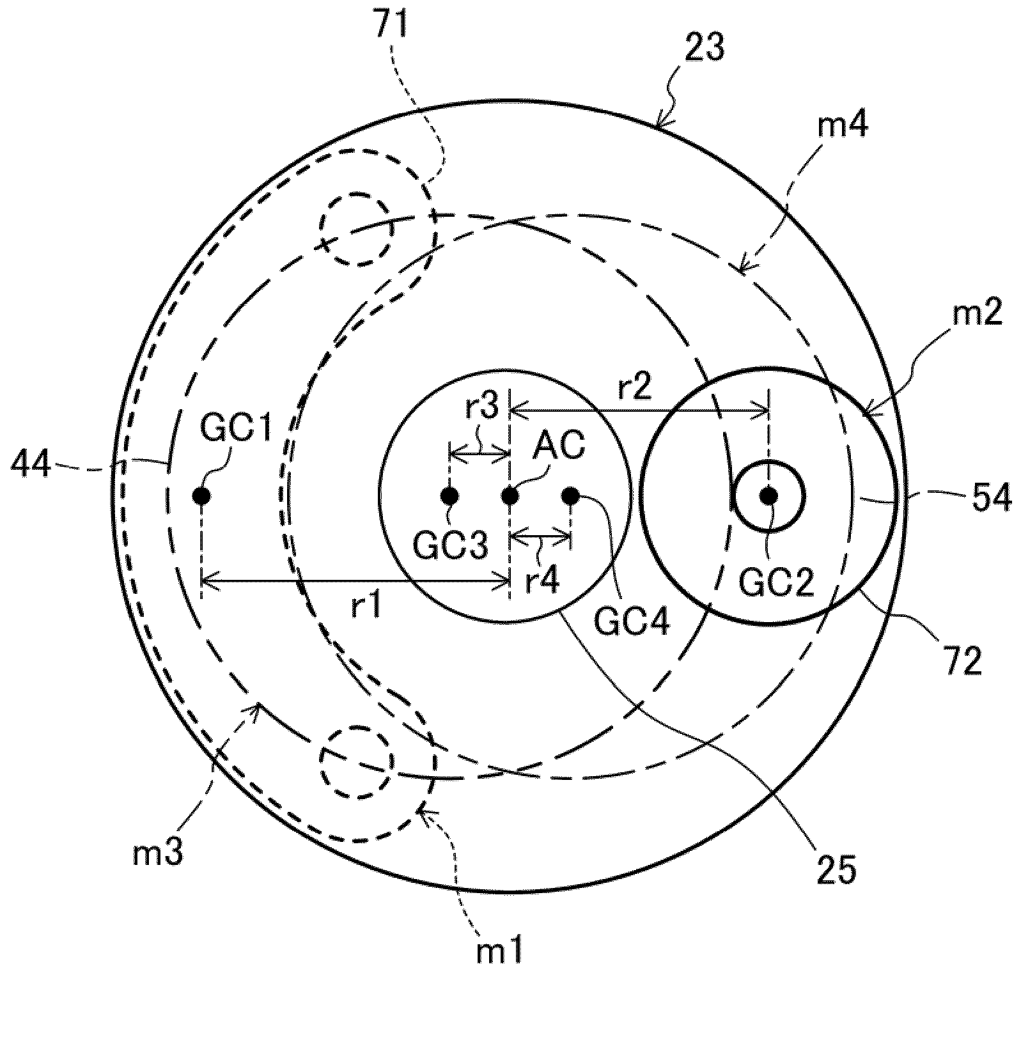


FIG.10

	FIRST BALANCE WEIGHT m1×r1[g·mm]	SECOND BALANCE WEIGHT m2×r2[g·mm]	FIRST ECCENTRIC PORTION M3×r3[g·mm]	SECOND ECCENTRIC PORTION M4×r4[g·mm]	STATIC BALANCE RELATION VALUE
COMPARATIVE EXAMPLE	194.9	195.1	927.5	927.5	1.0
EXAMPLE 1	194.9	183.8	927.5	927.5	0.99
EXAMPLE 2	194.9	172.5	927.5	927.5	0.98
EXAMPLE 3	194.9	161.2	927.5	927.5	0.97
EXAMPLE 4	194.9	149.9	927.5	927.5	0.96
EXAMPLE 5	194.9	138.7	927.5	927.5	0.95
EXAMPLE 6	194.9	127.6	927.5	927.5	0.94
EXAMPLE 7	194.9	116.3	927.5	927.5	0.93
EXAMPLE 8	194.9	105.0	927.5	927.5	0.92
EXAMPLE 9	194.9	94.0	927.5	927.5	0.91
EXAMPLE 10	194.9	82.7	927.5	927.5	0.90
EXAMPLE 11	194.9	71.4	927.5	927.5	0.89
EXAMPLE 12	194.9	60.2	927.5	927.5	0.88
EXAMPLE 13	194.9	49.0	927.5	927.5	0.87
EXAMPLE 14	194.9	37.8	927.5	927.5	0.86
EXAMPLE 15	194.9	26.5	927.5	927.5	0.85
EXAMPLE 16	194.9	15.3	927.5	927.5	0.84

FIG.11

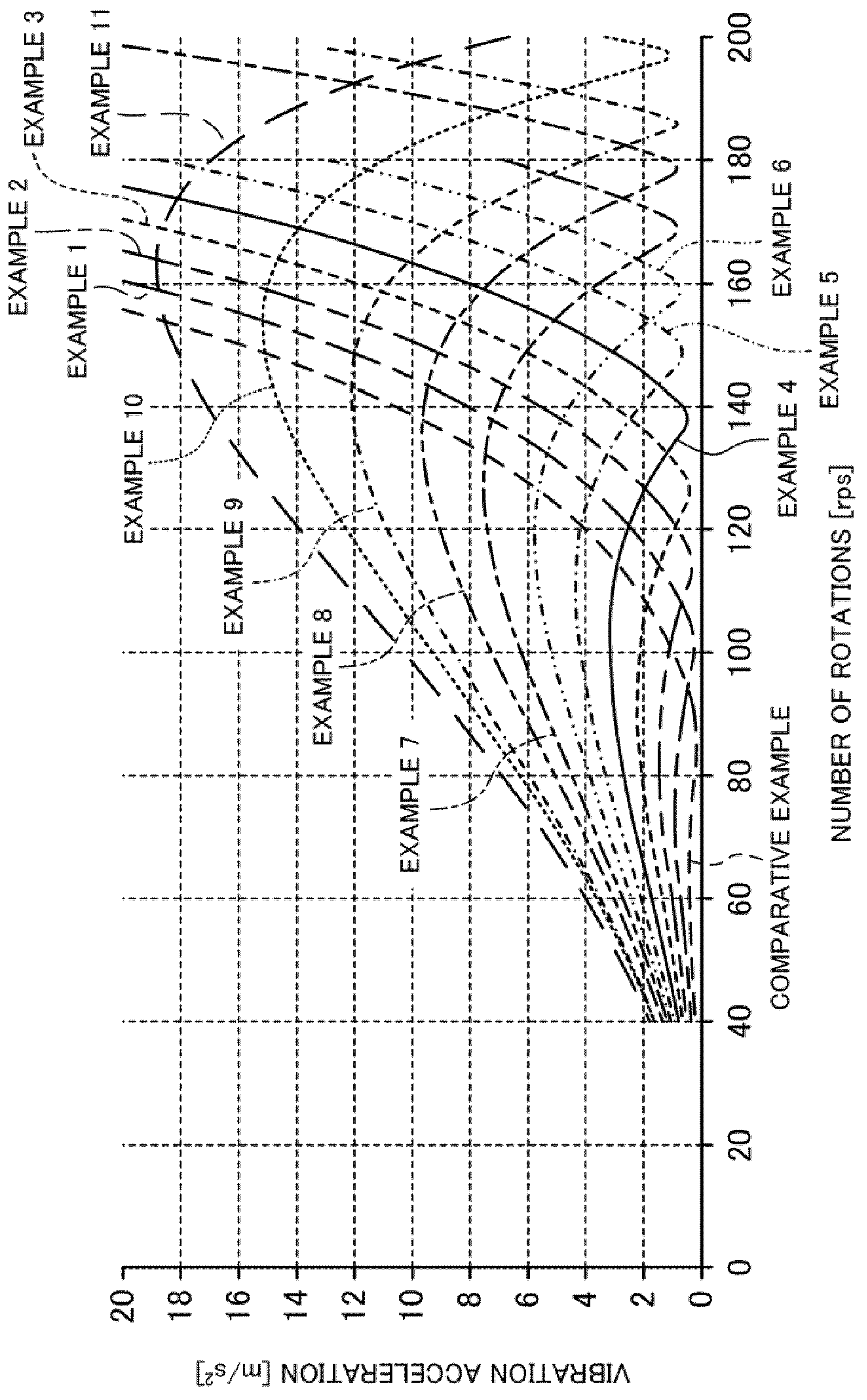
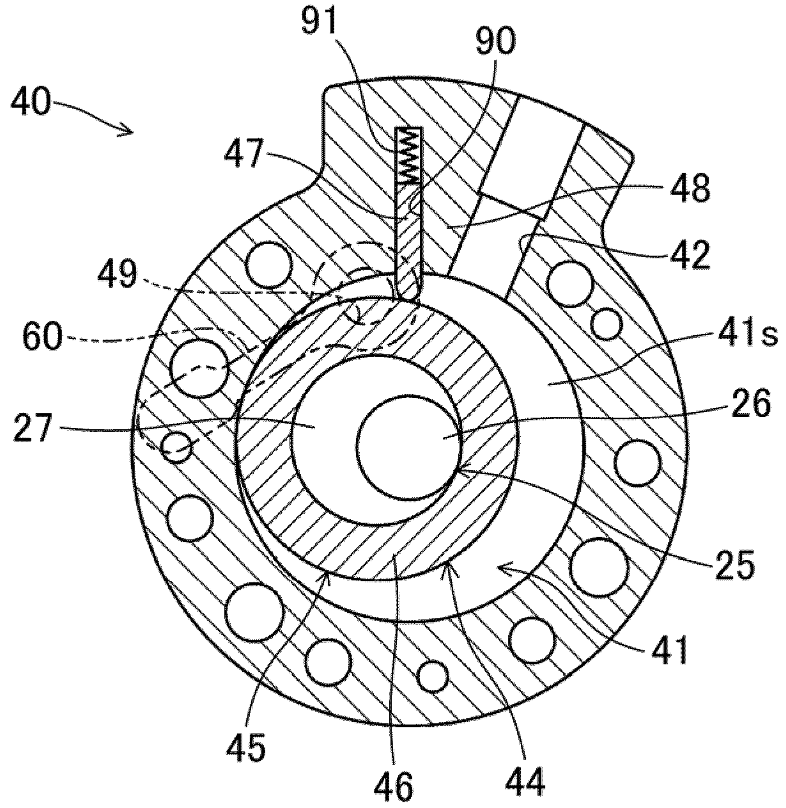

VIBRATION ACCELERATION [m/s²]
0
2
4
6
8
10
12
14
16
18
20
NUMBER OF ROTATIONS [rps]
0
20
40
60
80
100
120
140
160
180
200
COMPARATIVE EXAMPLE
EXAMPLE 1
EXAMPLE 2
EXAMPLE 3
EXAMPLE 4
EXAMPLE 5
EXAMPLE 6
EXAMPLE 7
EXAMPLE 8
EXAMPLE 9
EXAMPLE 10
EXAMPLE 11

FIG.12

		COMPARATIVE EXAMPLE	EXAMPLE															
			1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
NUMBER OF ROTATIONS [rps]	180	48.9	<u>42.8</u>	<u>36.7</u>	<u>30.7</u>	<u>24.8</u>	<u>18.8</u>	<u>12.8</u>	<u>6.9</u>	<u>1.2</u>	<u>5.1</u>	<u>11.0</u>	<u>17.0</u>	<u>22.9</u>	<u>28.8</u>	<u>34.7</u>	<u>40.6</u>	<u>46.5</u>
	170	34.1	<u>29.2</u>	<u>24.4</u>	<u>19.6</u>	<u>14.8</u>	<u>10.0</u>	<u>5.3</u>	<u>0.9</u>	<u>4.4</u>	<u>9.0</u>	<u>13.8</u>	<u>18.5</u>	<u>23.3</u>	<u>28.0</u>	<u>32.7</u>	37.5	42.2
	160	23.5	<u>19.6</u>	<u>15.7</u>	<u>11.9</u>	<u>8.1</u>	<u>4.2</u>	<u>0.7</u>	<u>3.6</u>	<u>7.4</u>	<u>11.1</u>	<u>15.0</u>	<u>18.8</u>	<u>22.6</u>	26.4	30.3	34.1	37.9
	150	16.0	<u>12.8</u>	<u>9.7</u>	<u>6.6</u>	<u>3.5</u>	<u>0.7</u>	<u>2.8</u>	<u>5.9</u>	<u>9.0</u>	<u>12.0</u>	<u>15.1</u>	18.2	21.3	24.4	27.5	30.6	33.6
	140	10.6	<u>8.0</u>	<u>5.5</u>	<u>3.0</u>	<u>0.7</u>	<u>2.1</u>	<u>4.6</u>	<u>7.1</u>	<u>9.6</u>	12.1	14.6	17.1	19.6	22.1	24.6	27.1	29.6
	130	6.7	<u>4.7</u>	<u>2.6</u>	<u>0.7</u>	<u>1.5</u>	<u>3.5</u>	<u>5.5</u>	7.5	9.5	11.5	13.6	15.6	17.6	19.6	21.6	23.7	25.7
	120	4.0	<u>2.4</u>	<u>0.8</u>	<u>1.0</u>	<u>2.6</u>	4.2	5.8	7.4	9.0	10.6	12.3	13.9	15.5	17.1	18.8	20.4	22.0

FIG.13



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2024/012928

A. CLASSIFICATION OF SUBJECT MATTER

F04C 23/02(2006.01)i; ***F04B 39/00***(2006.01)i; ***F04C 29/00***(2006.01)i
FI: F04C23/02 J; F04C29/00 T; F04B39/00 106D

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)

F04C23/02; F04B39/00; F04C29/00

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

Published examined utility model applications of Japan 1922-1996
Published unexamined utility model applications of Japan 1971-2024
Registered utility model specifications of Japan 1996-2024
Published registered utility model applications of Japan 1994-2024

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	JP 2013-204564 A (DAIKIN INDUSTRIES, LTD.) 07 October 2013 (2013-10-07) paragraphs [0039]-[0047], fig. 1, 3	1-7

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

* Special categories of cited documents:
"A" document defining the general state of the art which is not considered to be of particular relevance
"D" document cited by the applicant in the international application
"E" earlier application or patent but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed
"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
05 June 2024	**18 June 2024**

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

Form PCT/ISA/210 (second sheet) (July 2022)

INTERNATIONAL SEARCH REPORT
Information on patent family members

International application No.
PCT/JP2024/012928

Patent document cited in search report	Publication date (day/month/year)	Patent family member(s)	Publication date (day/month/year)
JP 2013-204564 A	07 October 2013	(Family: none)	

Form PCT/ISA/210 (patent family annex) (July 2022)

REFERENCES CITED IN THE DESCRIPTION

Patent documents cited in the description

- JP 2009180203 A **[0005]**