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

(11) EP 2 119 916 A1

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

(43) Date of publication:
18.11.2009 Bulletin 2009/47

(21) Application number: **07737530.1**

(22) Date of filing: **28.02.2007**

(51) Int Cl.:
F04C 28/08 *(2006.01)* ***F04C 18/32*** *(2006.01)*
F04C 23/00 *(2006.01)*

(86) International application number:
PCT/JP2007/053811

(87) International publication number:
WO 2008/105090 (04.09.2008 Gazette 2008/36)

(84) Designated Contracting States:
AT BE BG CH CY CZ DE DK EE ES FI FR GB GR HU IE IS IT LI LT LU LV MC NL PL PT RO SE SI SK TR

(71) Applicant: **Daikin Industries, Ltd.**
Osaka 530-8323 (JP)

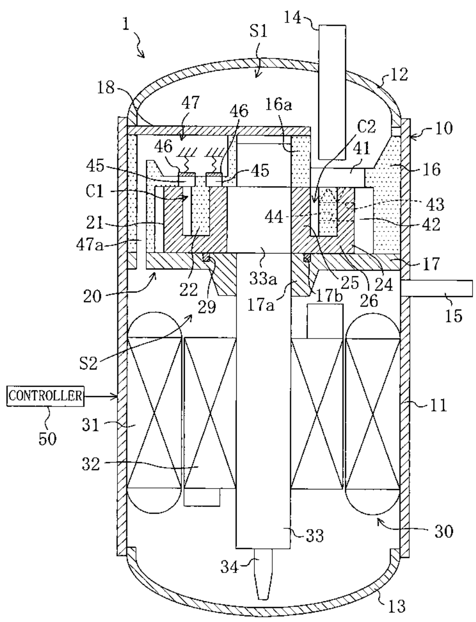
(72) Inventors:
- **SHIBAMOTO, Yoshitaka**
  Sakai-shi
  Osaka 591-8511 (JP)
- **SHIMIZU, Takashi**
  Sakai-shi
  Osaka 591-8511 (JP)
- **MASUDA, Masanori**
  Sakai-shi
  Osaka 591-8511 (JP)
- **SOTOJIMA, Takazou**
  Sakai-shi
  Osaka 591-8511 (JP)
- **FURUSHO, Kazuhiro**
  Sakai-shi
  Osaka 591-8511 (JP)
- **YAMAGIWA, Akio**
  Kusatsu-shi
  Shiga 525-8526 (JP)
- **YASUDA, Yoshiki**
  Kusatsu-shi
  Shiga 525-8526 (JP)

(74) Representative: **Goddar, Heinz J.**
Forrester & Boehmert
Pettenkoferstrasse 20-22
80336 München (DE)

(54) **ROTARY COMPRESSOR**

(57) An annular cylinder chamber (C1, C2) of a cylinder (21) is partitioned into an outer cylinder chamber (C1) and an inner cylinder chamber (C2) with an annular piston (22). The cylinder (21) is caused to eccentrically rotate by driving an electric motor (30), thereby changing the volumes of the outer cylinder chamber (C1) and the inner cylinder chamber (C2). The volume ratio Vr of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is set at about 0.7. In this state, the output torque of the electric motor (30) is changed in accordance with a variation in the load torque in one turn of rotation.

FIG. 1



EP 2 119 916 A1

Description

TECHNICAL FIELD

[0001] The present invention relates to rotary compressors, and particularly relates to measures against vibration caused by a variation in load torque.

BACKGROUND ART

[0002] As a conventional rotary compressor including two cylinder chambers, a compressor which compresses refrigerant by utilizing a variation in the volume of a cylinder chamber caused by eccentric rotation of an annular piston has been employed, as disclosed in, for example, Patent Document 1.

[0003] The compressor of Patent Document 1 includes: a cylinder having an annular cylinder chamber; and an annular piston placed in the cylinder chamber. The cylinder is composed of concentrically disposed outer and inner cylinders. Specifically, a cylinder chamber is formed between the outer cylinder and the inner cylinder, and this cylinder chamber is partitioned into an outer cylinder chamber and an inner cylinder chamber with an annular piston. The annular piston is configured to eccentrically rotate about the cylinder center by driving an electric motor, with the outer peripheral surface of the annular piston being in contact with the inner peripheral surface of the outer cylinder at substantially one point, and with the inner peripheral surface of the annular piston being in contact with the outer peripheral surface of the inner cylinder at substantially one point.

[0004] An outer blade is provided outside the annular piston. An inner blade is provided inside the annular piston on an extension of the outer blade. The outer blade is inserted in the outer cylinder, is radially biased toward the inside of the annular piston, and has its tip in pressure contact with the outer peripheral surface of the annular piston. The inner blade is inserted in the inner cylinder, is radially biased toward the outside of the annular piston, and has its tip in pressure contact with the inner peripheral surface of the annular piston. The outer and inner blades respectively partition the outer and inner cylinder chambers into high-pressure chambers and low-pressure chambers. In this compressor, eccentric rotation of the annular piston causes fluid to be sucked into the low-pressure chamber and to be compressed in the high-pressure chamber in each of the cylinder chambers.

Patent Document 1: Japanese Laid-Open Patent Publication No. 6-288358

DISCLOSURE OF INVENTION

Problems that the Invention is to Solve

[0005] However, disadvantageously, in the compressor of the Patent Document 1, the load torque of a drive shaft varies in one turn of rotation. This variation causes the rotation speeds of the drive shaft and a rotor of the electric motor for driving the drive shaft to vary, resulting in that vibration occurs in the tangential direction of a casing in which a stator of the electric motor is fixed. In a possible worst case, a pipe connected to the casing might be broken.

[0006] It is therefore an object of the present invention to suppress vibration caused by a load torque variation in one turn of rotation in a rotary compressor in which relative eccentric rotation is accomplished between a piston and a cylinder having two cylinder chambers.

Means of Solving the Problems

[0007] A first aspect of the present invention is directed to a rotary compressor including: a compression mechanism (20, 80) including a piston (22, 87a, 87b) and a cylinder (21, 81a, 81b) having two cylinder chambers (C1, C2, 82a, 82b); and an electric motor (30, 65) configured to change volumes of the cylinder chambers (C1, C2, 82a, 82b) by causing relative eccentric rotation between the cylinder (21, 81a, 81b) and the piston (22, 87a, 87b). The rotary compressor further includes a torque control means (50) configured to change an output torque of the electric motor (30, 65) in accordance with a variation in a load torque of the compression mechanism (20, 80) in one turn of rotation.

[0008] In this aspect, the relative eccentric rotation of the cylinder (21, 81a, 81b) and the piston (22, 87a, 87b) caused by driving the electric motor (30, 65) causes the volumes of the two cylinder chambers (C1, C2, 82a, 82b) to vary. In each of the cylinder chambers (C1, C2, 82a, 82b), fluid is sucked as the volume of the low-pressure chamber (i.e., the suction chamber) increases, whereas fluid in the high-pressure chamber (i.e., the compression chamber) is compressed as the volume of the high-pressure chamber decreases.

[0009] The load torque of the electric motor (30, 65) varies in accordance with the rotation angle in one turn of rotation of the compression mechanism (20, 80). Specifically, in each of the cylinder chambers (C1, C2, 82a, 82b), the load torque is highest substantially immediately before and after compressed fluid starts to be discharged. Accordingly, in this state, the rotation speed of the cylinder (21, 81a, 81b) or the piston (22, 87a, 87b) varies because the output torque of the electric motor (30, 65) is fixed. That is, when the load torque increases, the rotation speed decreases, whereas when the load torque decreases, the rotation speed increases. This variation in the rotation speed causes vibration of the compressor in the tangential direction of the casing.

[0010] On the other hand, in this aspect of the present invention, the torque control means (50) adjusts the output torque of the electric motor (30, 65) in accordance with a variation in the load torque in one turn of rotation. Specifically, the output torque of the electric motor (30, 65) decreases as the load torque decreases, and increases as the load torque increases. That is, torque control is performed such that the output torque of the electric motor (30, 65) is adjusted to a value commensurate with the load torque. This control makes the rotation speed of the cylinder (21, 81a, 81b) or the piston (22, 87a, 87b) substantially constant, thus suppressing vibration of the compressor.

[0011] In a second aspect of the present invention, in the rotary compressor of the first aspect, the cylinder (21) includes an annular cylinder chamber (C1, C2), and the piston (22) is an annular piston (22) housed in the annular cylinder chamber (C1, C2) and partitioning the annular cylinder chamber (C1, C2) into two cylinder chambers which are an outer cylinder chamber (C1) and an inner cylinder chamber (C2).

[0012] In this aspect, relative eccentric rotation of the cylinder (21) and the annular piston (22 (52)) caused by driving the electric motor (30) causes the volumes of the outer cylinder chamber (C1) and the inner cylinder chamber (C2) to vary. In each of the cylinder chambers (C1, C2), fluid is sucked as the volume of the low-pressure chamber (i.e., the suction chamber) increases, whereas fluid in the high-pressure chamber (i.e., the compression chamber) is compressed as the volume of the high-pressure chamber decreases.

[0013] In this case, the load torque of the electric motor (30) also varies in accordance with the rotation angle in one turn of rotation of the compression mechanism (20), thereby causing the rotation speed of the cylinder (21) or the annular piston (22) to vary. This variation causes vibration of the compressor in the tangential direction of the casing. On the other hand, in the present invention, the torque control means (50) adjusts the output torque of the electric motor (30) in accordance with a variation in the load torque in one turn of rotation. Accordingly, the rotation speed of the cylinder (21) or the annular piston (22) becomes substantially constant, thus suppressing vibration of the compressor.

[0014] In a third aspect of the present invention, in the rotary compressor of the second aspect, a volume ratio of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is in the range from 0.6 to 1.0.

[0015] In this aspect, since the volume ratio of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is set in the range from 0.6 to 1.0, the variation range in the load torque in one turn is small. Specifically, as shown in FIG. 5, as the volume ratio Vr of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) decreases, the variation range (i.e., the amount of variation) in the load torque increases. In particular, when the volume ratio Vr is about 0.6 or less, the variation range in the load torque becomes extremely large.

[0016] The torque control of the electric motor (30) changes the output torque of the electric motor (30) by adjusting, for example, an input current or an input voltage to the electric motor (30). For example, when the load torque is high, the input current is increased so as to increase the output torque of the electric motor (30). On the other hand, when the load torque is low, the input current is reduced so as to reduce the output torque of the electric motor (30). In general, the electric motor (30) exhibits a high operational efficiency when the electric motor (30) is driven with a substantially constant input current or voltage. Specifically, when the amount of variation (i.e., the degree of control) in, for example, the input current becomes large, the operational efficiency of the electric motor (30) greatly decreases.

[0017] On the other hand, in the present invention, the volume ratio Vr of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is limited within a given range as described above, thus reducing the amount of variation in the load torque in one turn of rotation. Accordingly, the amount of variation in the input current or the input voltage of the electric motor (30) is reduced in one turn of rotation, resulting in suppressing a decrease in the operational efficiency of the electric motor (30).

[0018] In a fourth aspect of the present invention, in the rotary compressor of the second or third aspect, the electric motor (30) is a brushless DC motor.

[0019] In this aspect, since a brushless direct-current (DC) motor is used as the electric motor (30), the operational efficiency of the electric motor (30) is higher than in the case of using an alternating-current (AC) motor. In particular, in torque control performed during low-speed rotation in which variation in the rotation speed is likely to be large, the DC motor can maintain a relatively high efficiently to a low speed, although the AC motor greatly decreases in efficiency, and thus substantially is not operable.

[0020] In a fifth aspect of the present invention, in the rotary compressor of the second or third aspect, the torque control means (50) is configured to change an output torque of the electric motor (30) by changing one of an input current, an input voltage, and an input current phase of the electric motor (30).

[0021] In this aspect, when the load torque decreases in one turn of rotation, the input current or the input voltage is reduced, thereby reducing the output torque of the electric motor (30). When the load torque increases, the input current or the input voltage is increased, thereby increasing the output torque of the electric motor (30). In this manner, the output torque of the electric motor (30) is adjusted to a value commensurate with the load torque. In addition, by adjusting (i.e., moving forward or backward) the input current phase, the output torque of the electric motor (30) increases or decreases to a value commensurate with the load torque. In particular, this adjustment of the input current phase can allow the output torque of the electric motor (30) to more closely follow a load torque which abruptly changes.

[0022] In a sixth aspect of the present invention, in the rotary compressor of the second or third aspect, the electric motor (30) is coupled to the cylinder (21) configured to rotate relative to the annular piston (22) which is stationary.

[0023] In this aspect, the cylinder (21) is movable, and the annular piston (22) is stationary. Specifically, the cylinder (21) eccentrically rotates relative to the annular piston (22), and the torque control means (50) suppresses a variation in the rotation speed of the cylinder (21). As a result, vibration caused by the variation in the rotation speed of the cylinder (21) can be suppressed.

[0024] In a seventh aspect of the present invention, the rotary compressor of the second or third aspect, the electric motor (30) is coupled to the annular piston (22) configured to rotate relative to the cylinder (21) which is stationary.

[0025] In this aspect, the cylinder (21) is stationary, and the annular piston (52) is movable. Specifically, the annular piston (22) eccentrically rotates relative to the cylinder (21), and the torque control means (50) suppresses a variation in the rotation speed of the annular piston (22). As a result, vibration caused by the variation in the rotation speed of the annular piston (22) can be suppressed.

[0026] In an eighth aspect of the present invention, the rotary compressor of the second aspect, the compression mechanism (20) is configured to perform two-stage compression on fluid, with one of the outer and inner cylinder chambers (C1) and (C2) used at a low-stage side and the other one of the outer and inner cylinder chambers (C1) and (C2) used at a high-stage side.

[0027] In this aspect, first, low-pressure fluid sucked into the outer cylinder chamber (C1) is compressed to be intermediate-pressure fluid. This intermediate-pressure fluid is sucked into the inner cylinder chamber (C2). This intermediate-pressure fluid in the inner cylinder chamber (C2) is further compressed to be high-pressure fluid. This series of processes is performed in one turn of rotation of the compression mechanism (20), thus changing the load torque of the electric motor (30) in accordance with the rotation angle. In this case, the torque control means (50) also makes the rotation speed of the cylinder (21) or the annular piston (22) substantially constant, resulting in suppressing vibration of the compressor.

[0028] In a ninth aspect of the present invention, the rotary compressor of the eighth aspect, a volume ratio of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is in the range from 0.6 to 0.8.

[0029] In this aspect, the volume ratio of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is set in the range from 0.6 to 0.8, and thus the variation range in the load torque in one turn of rotation is small. Specifically, as shown in FIG. 13, the variation range (i.e., the amount of variation) in the load torque is smaller in cases where the volume ratio Vr of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is 0.6 and 0.8, than in a case where the volume ratio Vr is 0.5 (or 1.0). In the case of the volume ratio Vr=1.0, the outer cylinder chamber (C1) and the inner cylinder chamber (C2) have the same volume, and thus this case corresponds to a one-cylinder compression mechanism performing so-called single-stage compression. In this aspect, the variation range in the load torque in one turn of rotation is smaller than in the one-cylinder compression mechanism.

[0030] In a tenth aspect of the present invention, in the rotary compressor of the first aspect, the cylinders (81 a, 81b) are respectively a low-stage first cylinder (81 a) and a high-stage second cylinder (81b) both including cylinder chambers (82a, 82b), the pistons (87a, 87b) are respectively a first rotary piston (87a) housed in the cylinder chamber (82a) of the first cylinder (81a) and a second rotary piston (87b) housed in the cylinder chamber (82b) of the second cylinder (81b), and the compression mechanism (80) is configured to perform two-stage compression on fluid in the cylinders (81a, 81b) by eccentric rotation of the rotary pistons (87a, 87b) caused by the electric motor (65).

[0031] In this aspect, the compression mechanism (80) is a so-called two-cylinder rotary compression mechanism. In this compression mechanism (80), the rotary pistons (87a, 87b) eccentrically rotate by driving the electric motor (65). This eccentric rotation of the rotary pistons (87a, 87b) causes the volumes of the cylinder chambers (82a, 82b) to vary, thereby achieving two-compression of fluid in the cylinders (81a, 81b). Specifically, first, low-pressure fluid sucked into the cylinder chamber (82a) of the first cylinder (81a) is compressed to be intermediate-pressure fluid. This intermediate-pressure fluid is sucked into the cylinder chamber (82b) of the second cylinder chamber (82b). The intermediate-pressure fluid in the cylinder chamber (82b) is further compressed to be high-pressure fluid. This series of processes is performed in one turn of rotation of the compression mechanism (20), thus changing the load torque of the electric motor (30) in accordance with the rotation angle. In this case, the torque control means (50) also makes the rotation speed of the rotary pistons (87a, 87b) substantially constant, resulting in suppressing vibration of the compressor.

[0032] In an eleventh aspect of the present invention, in the rotary compressor of the tenth aspect, a volume ratio of the cylinder chamber (82b) of the second cylinder (81b) to the cylinder chamber (82a) of the first cylinder (8 1 a) is in the range from 0.6 to 0.8.

[0033] In this aspect, the volume ratio of the cylinder chamber (82b) of the second cylinder (81b) to the cylinder chamber (82a) of the first cylinder (81a) is set in the range from 0.6 to 0.8, and thus the variation range in the load torque in one turn of rotation is small. Specifically, as shown in FIG. 13, the variation range (i.e., the amount of variation) in the load torque is smaller in cases where the volume ratio Vr of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is 0.6 and 0.8, than in a case where the volume ratio Vr is 0.5 (or 1.0).

[0034] In a twelfth aspect of the present invention, in the rotary compressor of the tenth or eleventh aspect, the compression mechanism (80) is configured such that a rotational phase of the rotary piston (87a) of the first cylinder (81a) is shifted from a rotational phase of the rotary piston (87b) of the second cylinder (81 b) by 180°.

[0035] In this aspect, when the volume of the cylinder chamber (82a) decreases in the first cylinder (81a) with the rotation of the rotary piston (87a), fluid compressed to an intermediate pressure is discharged. At substantially the same time, the volume of the cylinder chamber (82b) increases in the second cylinder (81b) with the rotation of the rotary piston (87b), thereby sucking the intermediate-pressure fluid discharged from the first cylinder (81a). This sucked intermediate-pressure fluid is further compressed with a decrease in the volume of the cylinder chamber (82b) of the second cylinder (81b), and then is discharged.

EFFECTS OF THE INVENTION

[0036] Accordingly, in the present invention, the output torque of the electric motor (30, 65) is adjusted according to a variation in the load torque of the compression mechanism (20, 80) in one turn of rotation. Thus, a variation in the rotation speed of the cylinder (21, 81a, 81b) or the piston (22, 87a, 87b) can be reduced. As a result, vibration of the compressor caused by a variation in the rotation speed can be suppressed.

[0037] In the third aspect of the present invention, the volume ratio of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is limited within a given range (i.e., from 0.6 to 1.0), and thus the amount of variation in the load torque in one turn of rotation can be reduced. Accordingly, the amount of variation in the output torque of the electric motor (30) can be reduced, thus suppressing a decrease in the operational efficiency of the electric motor (30). As a result, energy saving in operating the compressor can be achieved.

[0038] In the fourth aspect of the present invention, a brushless direct-current motor is used as the electric motor (30). Thus, the efficiency of the electric motor (30) can be enhanced, as compared to the case of using an AC motor. As a result, further energy saving in operating the compressor can be achieved.

[0039] In the ninth or eleventh aspect of the present invention, the volume ratio of the high-stage cylinder chamber (C2, 82b) to the low-stage cylinder chamber (C1, 82a) is also limited within a given range (i.e., from 0.6 to 0.8) in a two-cylinder two-stage compression mechanism. Accordingly, the amount of variation in the load torque in one turn of rotation can be reduced, thus suppressing a decrease in the operational efficiency of the electric motor (30, 65).

BRIEF DESCRIPTION OF DRAWINGS

[0040]

[FIG. 1] FIG. 1 is a longitudinal cross-sectional view illustrating a compressor according to a first embodiment.

[FIG. 2] FIG. 2 is a transverse cross-sectional view illustrating a compression mechanism of the first embodiment.

[FIG. 3] FIG. 3 shows transverse cross sections showing operation of the compression mechanism of the first embodiment for every 90° of rotation.

[FIG. 4] FIG. 4 is a graph showing variations in compression torques in one turn of rotation.

[FIG. 5] FIG. 5 is a graph showing variations in compression torques with respect to the volume ratios Vr.

[FIG. 6] FIG. 6 is a graph showing the amounts of decrease in a torque variation ratio, a vibration ratio, and a motor efficiency with respect to volume ratios Vr.

[FIG. 7] FIG. 7 is a longitudinal cross-sectional view illustrating a compressor according to a second embodiment.

[FIG. 8] FIG. 8 is a transverse cross-sectional view illustrating a compression mechanism of the second embodiment.

[FIG. 9] FIG. 9 shows transverse cross-sections showing operation of the compression mechanism of the second embodiment for every 90° of rotation.

[FIG. 10] FIG. 10 is a longitudinal cross-sectional view illustrating a compressor according to a third embodiment.

[FIG. 11] FIG. 11 is a transverse cross-sectional view illustrating a compression mechanism of the third embodiment.

[FIG. 12] FIG. 12 is a graph showing relationships between operating pressure ratios and compression efficiencies based on volume ratios Vr.

[FIG. 13] FIG. 13 is a graph showing variations in compression torque with respect to volume ratios Vr.

[FIG. 14] FIG. 14 is a graph showing a torque variation ratio with respect to the volume ratio Vr.

[FIG. 15] FIG. 15 is a longitudinal cross-sectional view illustrating a compressor according to a fourth embodiment.

[FIG. 16] FIG. 16 is a transverse cross-sectional view illustrating a compression mechanism of the fourth embodiment.

Description of Characters

[0041]

1, 60	compressor
20	compression mechanism
21	cylinder
22	annular piston (piston)
30	electric motor
50	controller (torque control means)
C1	outer cylinder chamber (cylinder chamber)
C2	inner cylinder chamber (cylinder chamber)
65	electric motor
80	compression mechanism
81a	first cylinder (cylinder)
81b	second cylinder (cylinder)
82a	first cylinder chamber (cylinder chamber)

82b second cylinder chamber (cylinder chamber)
87a first rotary piston (piston)
87b second rotary piston (piston)

BEST MODE FOR CARRYING OUT THE INVENTION

[0042] Hereinafter, embodiments of the present invention will be specifically described with reference to the drawings. The following embodiments are merely preferred examples in nature, and are not intended to limit the scope, applications, and use of the invention.

«EMBODIMENT 1»

[0043] A first embodiment is directed to a rotary compressor as illustrated in FIG. 1. This compressor (1) is of a fully-enclosed type in which a compression mechanism (20) and an electric motor (30) for driving the compression mechanism (20) are housed in a casing (10), and are fully enclosed. The compressor (1) is used for compressing refrigerant sucked from an evaporator and for discharging the compressed refrigerant into a condenser in a refrigerant circuit of an air conditioner, for example.

[0044] The casing (10) includes a cylindrical body (11) and upper and lower heads (12) and (13) respectively fixed to the top and bottom of the body (11). A suction pipe (14) penetrates the upper head (12), and a discharge pipe (15) penetrates the body (11).

[0045] The compression mechanism (20) includes: upper and lower housings (16) and (17) fixed to the casing (10); and a cylinder (21). The cylinder (21) has an annular cylinder chamber (C1, C2), and is located between the upper housing (16) and the lower housing (17). The upper housing (16) includes an annular piston (22) which is located in the cylinder chamber (C1, C2) and is continuous to the upper housing (16). The cylinder (21) is configured to eccentrically rotate relative to the annular piston (22). Specifically, in this embodiment, the cylinder (21) and the annular piston (22) provide relative eccentric rotation in which the cylinder (21) is movable and the annular piston (22) is stationary.

[0046] The electric motor (30) is a brushless direct-current (DC) motor including a stator (31) and a rotor (32). The stator (31) is located below the compression mechanism (20), and is fixed to the body (11) of the casing (10). The rotor (32) is coupled to a drive shaft (33) which rotates together with the rotor (32). The drive shaft (33) longitudinally penetrates the compression mechanism (20), and has an eccentric part (33a) located in the cylinder chamber (C1, C2). This eccentric part (33a) has a diameter greater than the other portion, and is eccentric from the axis of the drive shaft (33) to a given extent.

[0047] An axially extending oil supply passageway (not shown) is provided in the drive shaft (33). An oil supply pump (34) is provided at the bottom of the drive shaft (33). This oil supply pump (34) pumps lubricating oil accumulated in the bottom of the casing (10), and supplies the oil to a sliding part of the compression mechanism (20) through the oil supply passageway of the drive shaft (33).

[0048] The cylinder (21) has an outer cylinder portion (24) and an inner cylinder portion (25). The outer cylinder portion (24) and the inner cylinder portion (25) are annular, and have the same axis. These outer and inner cylinder portions (24) and (25) are coupled to each other at an end by a head (26) to be continuous. The annular cylinder chamber (C1, C2) is formed between the inner peripheral surface of the outer cylinder portion (24) and the outer peripheral surface of the inner cylinder portion (25). The eccentric part (33a) of the drive shaft (33) is slidably fit into the inner cylinder portion (25). The cylinder (21) is made of cast steel or an aluminum alloy, for example.

[0049] Each of the upper housing (16) and the lower housing (17) has a bearing portion (16a, 17a) for rotatably supporting the drive shaft (33). In this manner, the compressor (1) of this embodiment has a penetration structure in which the drive shaft (33) longitudinally penetrates the cylinder chamber (C1, C2) and both axial ends of the eccentric part (33a) are held in the casing (10) with the bearing portions (16a, 17a) interposed therebetween.

[0050] The outer peripheral surface of the annular piston (22) has a diameter smaller than that of the inner peripheral surface of the outer cylinder portion (24), and the inner peripheral surface of the annular piston (22) has a diameter larger than that of the outer peripheral surface of the inner cylinder portion (25). The annular piston (22) is eccentrically placed in the annular cylinder chamber (C1, C2) to partition the cylinder chamber (C1, C2) into an outer cylinder chamber (C1) and an inner cylinder chamber (C2). Specifically, the outer cylinder chamber (C1) is formed between the inner peripheral surface of the outer cylinder portion (24) and the outer peripheral surface of the annular piston (22). The inner cylinder chamber (C2) is formed between the inner peripheral surface of the annular piston (22) and the outer peripheral surface of the inner cylinder portion (25). The head (26) of the cylinder (21) serves as a first block member for blocking one end of the cylinder chamber (C1, C2), and the upper housing (16) serves as a second block member for blocking the other end of the cylinder chamber (C1, C2).

[0051] The outer peripheral surface of the annular piston (22) is in contact with the inner peripheral surface of the outer cylinder portion (24) at substantially one point. At a position whose phase is 180° shifted from this contact point, the inner peripheral surface of the annular piston (22) is in contact with the outer peripheral surface of the inner cylinder portion (25) at substantially one point.

[0052] As illustrated in FIG. 2, the compression mechanism (20) has a blade (23) which partitions each of the outer cylinder chamber (C1) and the inner cylinder chamber (C2) into a high-pressure chamber (C1-Hp, C2-Hp) serving as a compression chamber and a low-pressure chamber (C1-Lp, C2-Lp) serving as a suction chamber. This blade (23) is in the shape of a rectangular plate which penetrates the annular piston (22) and extends in the direction along the diameter of the cylinder chamber (C1, C2). Both ends of the blade (23) are respectively fixed to the inner peripheral surface of the outer cylinder portion (24) and the outer peripheral surface of the inner cylinder portion (25).

[0053] The annular piston (22) has a C-shape obtained by partially cutting off the annular piston (22) so as to allow the blade (23) to penetrate therethrough. The cut-off portion of the annular piston (22) is provided with swing bushings (27). The swing bushings (27) are constituted by a discharge-side bushing (27A) and a suction-side bushing (27B). The discharge-side bushing (27A) and the suction-side bushing (27B) are located toward the high-pressure chamber (C1-Hp, C2-Hp) and the low-pressure chamber (C1-Lp, C2-Lp), respectively, with the blade (23) sandwiched therebetween.

[0054] The discharge-side bushing (27A) and the suction-side bushing (27B) are approximately semicircular in cross section, and have their plane surfaces face each other. That is, a blade groove (28) in which the blade (23) slides is formed between opposing surfaces of the bushings (27A, 27B). The swing bushings (27) are configured such that the blade (23) moves forward and backward in the blade groove (28), with the blade (23) and the cylinder (21) swinging in an integrated manner relative to the annular piston (22). The bushings (27A, 27B) are not necessarily separated from each other, and may be partially coupled to each other to be continuous.

[0055] In the compression mechanism (20), with the rotation of the drive shaft (33), the points of contact on the annular piston (22) with the outer and inner cylinder portions (24) and (25) sequentially move from the state shown in FIG. 3(A) to the state shown in FIG. 3(D). Specifically, the rotation of the drive shaft (33) causes the compression mechanism (20) to revolve about the drive shaft (33), without causing the outer cylinder portion (24) and the inner cylinder portion (25) to rotate.

[0056] The upper housing (16) has an inlet (41) in the shape of a slot below the suction pipe (14). This inlet (41) penetrates the upper housing (16) along the axis of the upper housing (16), and allows the low-pressure chamber (C1-Lp, C2-Lp) of the cylinder chamber (C1, C2) to communicate with the space (i.e., low-pressure space (S1)) above the upper housing (16). The outer cylinder portion (24) has a through hole (43) which allows suction space (42) to communicate with the low-pressure chamber (C1-Lp) of the outer cylinder chamber (C1). The annular piston (22) has a through hole (44) which allows the low-pressure chamber (C1-Lp) of the outer cylinder chamber (C1) to communicate with the low-pressure chamber (C2-Lp) of the inner cylinder chamber (C2).

[0057] The annular piston (22) and the outer cylinder portion (24) preferably have wedge shapes by chamfering top portions thereof corresponding to the inlet (41) as indicated by broken lines in FIG. 1. In this case, refrigerant can be efficiently sucked into the low-pressure chambers (C1-Lp, C2-Lp).

[0058] The housing (16) has two outlets (45). These outlets (45) penetrate the upper housing (16) along the axis of the upper housing (16). The bottoms of the outlets (45) are respectively open to the high-pressure chambers (C1-Hp, C2-Hp) of the cylinder chambers (C1, C2). On the other hand, the tops of the outlets (45) communicate with discharge space (47) through discharge valves (46) for opening and closing the outlets (45).

[0059] This discharge space (47) is formed between the upper housing (16) and a cover plate (18). The upper housing (16) and the lower housing (17) are provided with a discharge passageway (47a) which allows the discharge space (47) to communicate with space (i.e., high-pressure space (S2)) below the lower housing (17).

[0060] The lower housing (17) is provided with a seal ring (29). This seal ring (29) is placed in an annular groove (17b) of the lower housing (17), and is in pressure contact with the bottom of the head (26) of the cylinder (21). High-pressure lubricating oil is introduced between the cylinder (21) and the lower housing (17) at the inner side of the seal ring (29) in the radial direction of the seal ring (29). In this manner, the seal ring (29) constitutes a compliance mechanism in which an axial clearance between the bottom surface of the annular piston (22) and the head (26) of the cylinder (21) is reduced by utilizing the pressure of the lubricating oil.

[0061] In this embodiment, the volume Vout of the outer cylinder chamber (C1) is larger than the volume Vin of the inner cylinder chamber (C2). Specifically, the volume ratio Vr (Vin/Vout) of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is set at about 0.7. This volume ratio Vr is preferably set in the range from 0.6 to 1.0.

[0062] The compressor (21) includes a controller (50) which is a torque control means for controlling the output torque of the electric motor (30).

[0063] The controller (50) is configured to change the output torque of the electric motor (30) in accordance with a variation in the load torque of the compression mechanism (20) in one turn of rotation. This controller (50) receives the rotation angle of the rotor (32), and supplies, to the electric motor (30), current having a value which has been previously set in accordance with the rotation angle of the rotor (32). That is, the controller (50) changes the output torque of the electric motor (30) by controlling input current of the electric motor (30). The rotation angle of the rotor (32) is equal to the rotation angle of the drive shaft (33). As the rotation angle of the rotor (32), a value detected by a rotation sensor or a value calculated from the induced voltage or current of the electric motor (30) is used.

[0064] Specifically, input current is increased at a rotation angle with a high load torque, whereas the input current is reduced at a rotation angle with a low load torque. Since the output torque of the electric motor (30) is proportional to the input current, increase/decrease in the input current causes the output torque of the electric motor (30) to increase/decrease accordingly. In this manner, the output torque of the electric motor (30) becomes commensurate with the load torque. Thus, a variation in the rotation speed of the drive shaft (33) in one turn can be suppressed, resulting in suppression of vibration.

[0065] In the present invention, instead of input current, an input voltage or an input current phase may be controlled in accordance with the rotation angle of the rotor (32) so as to change the output torque of the electric motor (30). For example, the input voltage is increased at a rotation angle with a high load torque, whereas the input voltage is reduced at a rotation angle with a low load torque. Accordingly, the output torque of the electric motor (30) increases/decreases in proportion to the input voltage, and is changed to a value commensurate with the load torque. A shift of the input voltage phase causes the output torque of the electric motor (30) to increase/decrease to a value commensurate with the load torque. In particular, adjusting the input current phase allows the output torque of the electric motor (30) to more closely follow the load torque which abruptly changes.

-Operation-

[0066] Now, it is described how the compressor (1) operates with reference to the drawings.

[0067] First, when the electric motor (30) is started, rotation of the rotor (32) is conveyed to the outer cylinder portion (24) and the inner cylinder portion (25) of the compression mechanism (20) via the drive shaft (33). Accordingly, the blade (23) reciprocates (i.e., moves forward and backward) between the swing bushings (27), and the blade (23) and the swing bushings (27) swing in an integrated manner with respect to the annular piston (22). The outer cylinder portion (24) and the inner cylinder portion (25) revolve about the annular piston (22) while swinging, thereby causing the compression mechanism (20) to perform given compression operation.

[0068] Specifically, in the outer cylinder chamber (C1), the volume of the low-pressure chamber (C1-Lp) is approximately smallest in the state shown in FIG. 3(D). While the drive shaft (33) rotates in the clockwise direction in the drawings to shift from the state of FIG. 3(D) to the states of FIGS. 3(A), 3(B), and 3(C) in this order, the volume of the low-pressure chamber (C1-Lp) increases. With this increase in the volume of the low-pressure chamber (C1-Lp), refrigerant passes through the suction pipe (14), the low-pressure space (S1), and the inlet (41) to be sucked into the low-pressure chamber (C1-Lp). At this time, the refrigerant is not only sucked directly into the low-pressure chamber (C1-Lp) from the inlet (41), but also partially enters the suction space (42) from the inlet (41), and then is sucked into the low-pressure chamber (C1-Lp) through the through hole (43).

[0069] When the drive shaft (33) returns to the state of FIG. 3(D) after one turn of rotation, the suction of refrigerant into the low-pressure chamber (C1-Lp) is completed. This low-pressure chamber (C1-Lp) is then changed to a high-pressure chamber (C1-Hp) in which the refrigerant is compressed, and a new low-pressure chamber (C1-Lp) is created with the blade (23) sandwiched between the high-pressure chamber (C1-Hp) and the low-pressure chamber (C1-Lp). Then, while the drive shaft (33) further rotates, suction of refrigerant is repeated in the low-pressure chamber (C1-Lp), whereas the volume of the high-pressure chamber (C1-Hp) decreases, thereby compressing the refrigerant in the high-pressure chamber (C1-Hp). The high-pressure refrigerant in the high-pressure chamber (C1-Hp) flows from the outlets (45) into the discharge space (47), and then flows into the high-pressure space (S2) through the discharge passageway (47a).

[0070] In the inner cylinder chamber (C2), the volume of the low-pressure chamber (C2-Lp) is approximately smallest in the state shown in FIG. 3(B). While the drive shaft (33) rotates in the clockwise direction in the drawings from the state of FIG. 3(B) to the states of FIGS. 3 (C), 3(D), and 3(A) in this order, the volume of the low-pressure chamber (C2-Lp) increases. With this increase in the volume of the low-pressure chamber (C2-Lp), refrigerant passes through the suction pipe (14), the low-pressure space (S1), and the inlet (41) to be sucked into the low-pressure chamber (C2-Lp). At this time, the refrigerant is not only sucked directly into the low-pressure chamber (C2-Lp) from the inlet (41), but also partially enters the suction space (42) from the inlet (41), and then passes through the through hole (43), the low-pressure chamber (C1-Lp) of the outer cylinder chamber, and the through hole (44) to be sucked into the low-pressure chamber (C2-Lp) of the inner cylinder chamber (C2).

[0071] When the drive shaft (33) returns to the state of FIG. 3(B) after one turn of rotation, the suction of refrigerant into the low-pressure chamber (C2-Lp) is completed. This low-pressure chamber (C2-Lp) is then changed to a high-pressure chamber (C2-Hp) in which the refrigerant is compressed, and a new low-pressure chamber (C2-Lp) is created with the blade (23) sandwiched between the high-pressure chamber (C2-Hp) and the low-pressure chamber (C2-Lp). Then, while the drive shaft (33) further rotates, suction of refrigerant is repeated in the low-pressure chamber (C2-Lp), whereas the volume of the high-pressure chamber (C2-Hp) decreases, thereby compressing refrigerant in the high-pressure chamber (C2-Hp). The high-pressure refrigerant in the high-pressure chamber (C2-Hp) flows from the outlets (45) into the discharge space (47), and then flows into the high-pressure space (S2) through the discharge passageway (47a).

[0072] In this manner, high-pressure refrigerant which has been compressed in the outer cylinder chamber (C1) and the inner cylinder chamber (C2) and has flown into the high-pressure space (S2) is discharged from the discharge pipe (15), is subjected to condensation, expansion, and evaporation processes in the refrigerant circuit, and then is sucked into the compressor (1) again.

[0073] Now, it is described how the torque of the electric motor (30) is controlled. In FIG. 3, it is assumed that

FIG. 3(A) corresponds to a rotation angle of 180°, FIG. 3(B) corresponds to a rotation angle of 270°, FIG. 3(C) corresponds to a rotation angle of 0° (360°), and FIG. 3 (D) corresponds to a rotation angle of 90°.

[0074] In the operation described above, the compression torque (i.e., the load torque) of the drive shaft (33) in one turn of rotation varies as indicated by the solid line in FIG. 4. Specifically, in the compressor (1) of this embodiment, the compression torque is highest around a rotation angle of 180°, and is lowest around rotation angles of 90° and 270°. On the other hand, as indicated by the broken line in FIG. 4, in a general one-cylinder rotary compressor, the compression torque is highest around a rotation angle of 180°, and is lowest around a rotation angle of 0° (360°). Comparison of the torque variation ranges (i.e., a difference between the maximum and minimum compression torques) in one turn of rotation shows that the torque variation range (about 1.1 Nm) of the compressor (1) of the present invention is greatly smaller than the torque variation range (about 2.3 Nm) of the one-cylinder rotary compressor. The torque variations shown in FIG. 4 are obtained when the operation pressure ratio (i.e., the condensation pressure/ the evaporation pressure) occurring in an air conditioner in an intermediate season is about 1.6.

[0075] Input current of the electric motor (30) is adjusted in accordance with the variation of the compression torque described above. Specifically, the input current value is largest when the compression torque is highest, and the input current value is smallest when the compression torque is lowest. In this manner, in one turn of rotation of the drive shaft (33), the input current of the electric motor (30) varies from the minimum value to the maximum value. However, the amount of variation in this input current (i.e., the degree of control) is smaller than that in the one-cylinder rotary compressor. That is, the one-cylinder rotary compressor exhibits a wide variation range of the compression torque in one turn of rotation, and thus the amount of variation in the input current is also large accordingly.

[0076] In general, as the amount of variation in the input current of an electric motor decreases, the efficiency in the electric motor increases (i.e., the amount of decrease in the efficiency decreases). This shows that even with the same torque control, the efficiency of the electric motor (30) less decreases in the compressor (1) of the present invention than in the one-cylinder rotary compressor. As a result, the compressor (1) can operate with energy saving as a whole.

[0077] Now, a relationship among the volume ratio Vr between the outer cylinder chamber (C1) and the inner cylinder chamber (C2), the torque variation ratio, and the vibration ratio is described.

[0078] First, FIG. 5 shows a relationship between the volume ratio Vr (Vin/Vout) between the outer cylinder chamber (C1) and the inner cylinder chamber (C2), and the torque variation range. In FIG. 5, torque variation ranges are shown for five patterns of volume ratios Vr (Vin/Vout): 50/50=1, 40/60=0.66, 25/75=0.33, 15/85=0.17, and 0/100=0. The pattern of the volume ratio Vr (Vin/Vout)=0/100 corresponds to a one-cylinder rotary compressor. The torque variations shown in FIG. 5 are obtained when the operating pressure ratio (i.e., the condensation pressure/the evaporation pressure) occurring in an air conditioner in rated operation is about 3.

[0079] Specifically, the case of the volume ratio Vr=0/100 exhibits the largest torque variation range, and the case of the volume ratio Vr=50/50 exhibits the smallest torque variation range. That is, as the volume ratio Vr approaches 1 (one), the torque variation range decreases. Accordingly, as the volume ratio Vr approaches 1 (one), vibration caused by a torque variation is more greatly suppressed.

[0080] It is also shown that the period (i.e., an interval between two valleys sandwiching one peak of a main torque variation) of a main torque variation becomes shorter as the volume ratio Vr approaches 1 (one). For example, the period ("c" in FIG. 5) of the main torque variation in the case of the volume ratio Vr=50/50 is shorter than the period ("b" in FIG. 5) of the main torque variation in the case of the volume ratio Vr=25/75. This period ("b" in FIG. 5) of the main torque variation is shorter than the period ("a" in FIG. 5) of the main torque variation in the case of the volume ratio Vr=0/100. As the period of the main torque variation increases, the motor vibrates more slowly, and thus the amplitude of this vibration increases. In general, the amplitude increases in proportion to the square of the period (=1/frequency).

[0081] FIG. 6 shows how the amounts of decrease in a torque variation ratio, a vibration ratio, and a motor efficiency (i.e., an electric-motor efficiency) are associated with the volume ratio Vr. In FIG. 6, the torque variation ratio and the vibration ratio are expressed as ratios of the torque variation range and vibration to the volume ratio Vr, with the torque variation range and the vibration in the case of the volume ratio Vr=0/100 being defined as "1". The amount of decrease in the motor efficiency is obtained when the rotation speed variation is suppressed to the lowest degree with torque control. In FIG. 6, the amount of decrease in the motor efficiency (i.e., the electric motor efficiency) is indicated by a solid line, the torque variation ratio is indicated by a broken line, and the vibration ratio is indicated by a dash-dotted line.

[0082] Specifically, as the volume ratio Vr approaches 1 (one), the torque variation ratio and the vibration ratio decrease. The amount of decrease in the motor efficiency is approximately 0%, i.e., is smallest, when the volume ratio Vr is 1 (one), and increases as the volume ratio Vr decreases. In addition, it is shown that the motor efficiency gradually decreases while the volume ratio Vr is in the range from 1.0 to 0.6, and steeply decreases when the volume ratio Vr is less than 0.6. In this manner, while the volume ratio Vr is in the range from 0.6 to 1.0, torque control can be performed with a small amount of decrease in the motor efficiency, and thus vibration can be suppressed as compared to the one-cylinder rotary compressor.

[0083] As described above, in the compressor (1) of this embodiment, torque control of the electric motor (30) enables suppression of a decrease in the efficiency of the electric motor (30) more greatly than torque control of a one-cylinder rotary compressor. In addition, despite performing torque control with a small amount of decrease in the efficiency of the electric motor (30), vibration can be further suppressed by setting the volume ratio Vr between the outer cylinder chamber (C1) and the inner cylinder chamber (C2) at about 0.7. As a result, vibration of the compressor (1) can be suppressed, and operation with energy saving can be achieved.

[0084] In this embodiment, a brushless direct-current (DC) motor having a higher efficiency than an alternating-current (AC) motor is employed as the electric motor (30), and thus high efficiency can be maintained even during low-speed operation necessary for an air conditioner incorporating the compressor (1) of the present invention in intermediate seasons, thus further saving energy.

[0085] In a conventional two-cylinder rotary compressor, two cylinders having a volume ratio Vr of 1 : 1 are disposed longitudinally, and thus a crank mechanism such as a rotary piston and an eccentric shaft is needed for each cylinder. On the other hand, in the compressor (1) of the present invention, one cylinder is partitioned into the outer cylinder chamber (C1) and the inner cylinder chamber (C2), and these cylinder chambers share the annular piston (22). As a result, only one crank mechanism is sufficient for the compressor (1), thus achieving cost reduction.

[0086] In addition, if the cylinder (21) is made of an aluminum alloy, centrifugal force during rotation is reduced. In this case, vibration in high-speed operation can be suppressed, and warping of the drive shaft (33) is also suppressed. Accordingly, highly efficient operation with small vibration can be achieved in a wide range.

«EMBODIMENT 2»

[0087] As illustrated in FIGS. 7 and 8, a second embodiment is obtained by modifying the configuration of the compression mechanism (20) of the first embodiment. Specifically, in this embodiment, an annular piston (52) is movable, a cylinder (21) is stationary, and the annular piston (52) eccentrically rotates relative to the cylinder (21).

[0088] A compression mechanism (20) of the second embodiment includes an upper housing (16) and a piston assembly (55). The upper housing (16) is continuous to the cylinder (21). The piston assembly (55) is configured to eccentrically rotate relative to the cylinder (21). In this embodiment, the lower housing (17) is omitted.

[0089] The cylinder (21) includes outer cylinder portion (24) and an inner cylinder portion (25). The outer cylinder portion (24) and the inner cylinder portion (25) are annular, and have the same axis. These outer and inner cylinder portions (24) and (25) are provided on the lower surface of a head (26) of the upper housing (16). An annular cylinder chamber (C1, C2) is formed between the inner peripheral surface of the outer cylinder portion (24) and the outer peripheral surface of the inner cylinder portion (25).

[0090] The piston assembly (55) includes: a head (51); the annular piston (52) positioned upright on, and continuous to, the upper surface of the head (51); and a cylindrical piston (53). The piston assembly (55) is made of cast steel or an aluminum alloy. The annular piston (52) has an inner diameter greater than the outer diameter of the cylindrical piston (53), and has the same axis as the cylindrical piston (53). The piston assembly (55) is configured such that the annular piston (52) is placed in the annular cylinder chamber (C1, C2) to partition the cylinder chamber (C1, C2) into an outer cylinder chamber (C1) and an inner cylinder chamber (C2). Specifically, the head (26) of the upper housing (16) serves as a first block member for blocking an end of the cylinder chamber (C1, C2). The head (51) of the piston assembly (55) serves as a second block member for blocking the other end of the cylinder chamber (C1, C2). The cylindrical piston (53) is located in the inner cylinder portion (25).

[0091] In this embodiment, the volume Vout of the outer cylinder chamber (C1) is also larger than the volume Vin of the inner cylinder chamber (C2), and the volume ratio Vr (Vin/Vout) of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is also set at about 0.7.

[0092] An eccentric part (33a) is formed at the top of a drive shaft (33) of an electric motor (30), and is coupled to the piston assembly (55). Specifically, this eccentric part (33a) of the drive shaft (33) is rotatably fitted in a fitting part (54) which has a cylindrical shape and is continuous to the lower surface of the piston assembly (55). In this manner, the piston assembly (55) is caused to eccentrically rotate relative to the cylinder (21) by the rotation of the drive shaft (33).

[0093] Next, it is described how this compressor (1) operates with reference to FIG. 9. Advantages of the cylinder chamber (C1, C2) in this operation are substantially the same as those in the first embodiment.

[0094] Specifically, in the outer cylinder chamber (C1), the volume of a low-pressure chamber (C1-Lp) is approximately smallest in the state shown in FIG. 9(D). While the drive shaft (33) rotates to shift from this state to the states of FIGS. 9(A), 9(B), and 9(C) in this order, the volume of the low-pressure chamber (C1-Lp) increases, and refrigerant is sucked into the low-pressure chamber (C1-Lp). When the drive shaft (33) takes one turn, the low-pressure chamber (C1-Lp) changes to a high-pressure chamber (C1-Hp). Then, while the drive shaft (33) further rotates, the volume of the high-pressure chamber (C1-Hp) decreases, thereby compressing refrigerant.

[0095] On the other hand, in the inner cylinder chamber (C2), the volume of the low-pressure chamber (C2-Lp) is approximately smallest in the state shown in FIG. 9(B). While the drive shaft (33) rotates to shift from this state to the states of FIGS. 9(C), 9(D), and 9(A) in this order, the volume of the low-pressure chamber (C2-Lp) increases, and thus refrigerant is sucked into the low-pressure chamber (C2-Lp). When drive shaft (33) takes one turn, the low-pressure chamber (C2-Lp) changes to a high-pressure chamber (C2-Hp). Then, while the drive shaft (33) further rotates, the volume of the high-pressure chamber (C2-Hp) decreases, thereby compressing refrigerant.

[0096] As in the first embodiment, torque control of the electric motor (30) is performed by a controller (50) in this embodiment. Accordingly, as compared to a case where torque control is performed on a one-cylinder compressor, decrease in the efficiency of the electric motor (30) is greatly suppressed, thus achieving energy saving of the compressor (1).

[0097] As in the first embodiment, if the piston assembly (55) is made of an aluminum alloy, vibration and warping of the drive shaft (33) are suppressed during high-speed operation, thus performing highly efficient operation with small vibration in a wide range. Other configurations, operation, and advantages are similar to those in the first embodiment.

«EMBODIMENT 3»

[0098] As illustrated in FIGS. 10 and 11, a third embodiment is obtained by modifying the compression mechanism (20) of the first embodiment such that the compression mechanism (20) performs two-stage compression on refrigerant. Specifically, in a compression mechanism (20) of the third embodiment, an outer cylinder chamber (C1) serves as a low-stage compression chamber, and an inner cylinder chamber (C2) serves as a high-stage compression chamber.

[0099] A compressor (1) is used for, for example, a refrigerant circuit using carbon dioxide (CO_2) as refrigerant and operating in a two-stage compression one-stage expansion cycle. Although not shown, in this refrigerant circuit, the compressor (1), a heat dissipater (a gas cooler), a receiver, an intermediate cooler, an expansion valve, and an evaporator are sequentially connected to each other by refrigerant pipes. In this refrigerant circuit, high-pressure refrigerant discharged from the inner cylinder chamber (C2) of the compressor (1) sequentially flows in the heat dissipater, the receiver, the expansion valve, and the evaporator, and flows into the outer cylinder chamber (C1) of the compressor (1). On the other hand, intermediate-pressure refrigerant compressed in the outer cylinder chamber (C1) flows into the intermediate cooler, and part of liquid refrigerant from the receiver subjected to pressure reduction also flows into the intermediate cooler. In this intermediate cooler, the intermediate-pressure refrigerant from the outer cylinder chamber (C1) is cooled. This cooled intermediate-pressure refrigerant returns to the inner cylinder chamber (C2), and is compressed again. This circulation is repeated, thereby cooling the inside air in the evaporator, for example.

[0100] A body (11) of a casing (10) of the compressor (1) is provided with a suction pipe (14), an inflow pipe (1a), and an outflow pipe (1b). These pipes penetrate the body (11). The suction pipe (14) is connected to the evaporator, and the inflow pipe (1a) and the outflow pipe (1b) are connected to the intermediate cooler. A discharge pipe (15) is provided through an upper head (12) of the casing (10). This discharge pipe (15) is connected to the heat dissipater.

[0101] An upper housing (16) of the compression mechanism (20) is provided with a cover plate (18). In the casing (10), space above the cover plate (18) is defined as high-pressure space (4a), and space below a lower housing (17) is defined as intermediate-pressure space (4b). An end of the discharge pipe (15) is open to the high-pressure space (4a), and an end of the outflow pipe (1b) is open to the intermediate-pressure space (4b).

[0102] An intermediate-pressure chamber (4c) and a high-pressure chamber (4d) are formed between the upper housing (16) and the cover plate (18). The upper housing (16) has an intermediate-pressure passageway (4e). A pocket (4f) is formed on the outer periphery of an outer cylinder (24) between the upper housing (16) and the lower housing (17). The inflow pipe (1a) is connected to an end of the intermediate-pressure passageway (4e). The suction pipe (14) is connected to the pocket (4f) such that the pocket (4f) is in a low-pressure atmosphere at a suction pressure.

[0103] The outer cylinder (24) has a first inlet (41a) radially penetrating the outer cylinder (24). The first inlet (41a) is located at the right of a blade (23) in FIG. 11. That is, the first inlet (41a) establishes communication among the outer cylinder chamber (C1), the pocket (4f), and the suction pipe (14).

[0104] The other end of the intermediate-pressure passageway (4e) is configured as a second inlet (41b). This second inlet (41b) is located at the right of the blade (23), is open to the inner cylinder chamber (C2), and establishes communication between the inner cylinder chamber (C2) and the intermediate-pressure space (4b).

[0105] The upper housing (16) has a first discharge port (45a) and a second discharge port (45b). These discharge ports (45a, 45b) axially penetrate the upper housing (16). An end of the first discharge port (45a) is open to the high-pressure side of the outer cylinder chamber (C1), and the other end of the first discharge port (45a) is open to the intermediate-pressure chamber (4c). An end of the second discharge port (45b) is open to the high-pressure side of the inner cylinder chamber (C2), and the other end of the second discharge port (45b) is open to the high-pressure chamber (4d). Outer ends of the first discharge port (45a) and the second discharge port (45b) are provided with valves (46) which are reed valves.

[0106] The intermediate-pressure chamber (4c) and the intermediate-pressure space (4b) communicate with each other with a communication passageway (4g) formed in the upper housing (16) and the lower housing

(17). Although not shown, the high-pressure chamber (4d) communicates with the high-pressure space (4a) via a high-pressure passageway formed in the cover plate (18).

[0107] In this embodiment, the volume Vout of the outer cylinder chamber (C1) is also larger than the volume Vin of the inner cylinder chamber (C2), and the volume ratio Vr (Vin/Vout) of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is also set at about 0.7.

[0108] In this compressor (1), when the electric motor (30) is started, the outer cylinder (24) and the inner cylinder (25) revolve, while swinging relative to the annular piston (22), as in the first embodiment. Then, the compression mechanism (20) performs given compression operation.

[0109] When the volume of the outer cylinder chamber (C1) increases with rotation of a drive shaft (33), low-pressure refrigerant is sucked into the outer cylinder chamber (C1) from the suction pipe (14) through the pocket (4f) and the first inlet (41a). Then, the drive shaft (33) further rotates to cause the volume of the outer cylinder chamber (C1) to decrease, thereby compressing refrigerant. When the pressure of this outer cylinder chamber (C1) reaches a given intermediate pressure, and the differential pressure between the outer cylinder chamber (C1) and the intermediate-pressure chamber (4c) reaches a set value, the valves (46) are opened. Accordingly, intermediate-pressure refrigerant is discharged from the outer cylinder chamber (C1) into the intermediate-pressure chamber (4c), and flows from the outflow pipe (1b) through the intermediate-pressure space (4b).

[0110] On the other hand, when the volume of the inner cylinder chamber (C2) is caused to increase by the rotation of the drive shaft (33), intermediate-pressure refrigerant is sucked into the inner cylinder chamber (C2) from the inflow pipe (1a) through the intermediate-pressure passageway (4e) and the second inlet (41b). Then, the drive shaft (33) further rotates to cause the volume of the inner cylinder chamber (C2) to decrease, thereby compressing refrigerant. When the pressure of this inner cylinder chamber (C2) reaches a given high pressure, and the differential pressure between the inner cylinder chamber (C2) and the high-pressure chamber (4d) reaches a set value, the valves (46) are opened. Accordingly, high-pressure refrigerant is discharged from the inner cylinder chamber (C2) into the high-pressure chamber (4d), and flows from the discharge pipe (15) through the high-pressure space (4a). In this manner, in the compressor (1) of this embodiment, refrigerant compressed in the outer cylinder chamber (C1) is further compressed in the inner cylinder chamber (C2), thereby performing two-stage compression.

[0111] In general, an air conditioner (which is herein an inverter air conditioner) is most frequently operated in a range of low pressure ratios where the operating pressure ratio is in the range from about 1.6 to about 2.0. The operating pressure ratio herein is a ratio of a condensation pressure to an evaporation pressure in a refrigerant circuit.

[0112] As shown in FIG. 12, in a low operating pressure ratio range with a high operating frequency, as the volume ratio Vr of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) increases, the compression efficiency increases. For example, in the case of the volume ratio Vr=0.8, the compression efficiency is highest around an operating pressure ratio of 1.5, and in the case of the volume ratio Vr=0.6, the compression efficiency is highest around an operating pressure ratio of 1.9. On the other hand, in the case of the volume ratio Vr=0.5, the compression efficiency is highest around an operating pressure ratio of 2.5, and while the operating pressure ratio decreases to about 2.0 or less, the compression efficiency greatly decreases. That is, in the two-stage compression using the outer cylinder chamber (C1) and the inner cylinder chamber (C2), as the volume ratio Vr decreases, the compression ratio of the entire compression mechanism (20) increases. Then, under operating conditions with a low operating pressure ratio, overcompression losses are likely to occur, and thus the compression efficiency is also likely to decrease.

[0113] FIG. 13 shows a relationship between the volume ratio Vr (Vin/Vout) and the torque variation range: FIG. 14 shows a relationship between the volume ratio Vr (Vin/Vout) and the torque variation ratio. These graphs show that the torque variation range and the torque variation ratio in the cases of the volume ratio Vr=0.6 and 0.8 are smaller than those in the case of the volume ratio Vr=0.5 and 1. Accordingly, such small torque variations enable suppression of vibration. The torque variation ratio is the torque variation range expressed in terms of ratio for each volume ratio Vr, with the torque variation range in the case of the volume ratio Vr=1 set at "1". FIGS. 13 and 14 are obtained by measurement at an operating pressure ratio of about 2.

[0114] In the case of the volume ratio Vr=1, the outer cylinder chamber (C1) and the inner cylinder chamber (C2) have the same volume. In this case, refrigerant is not compressed in the outer cylinder chamber (C1), but is compressed only in the inner cylinder chamber (C2), i.e., is subjected to not two-stage compression but single-stage compression. That is, the case of the volume ratio Vr=1 substantially corresponds to the case where single-stage compression is performed by a conventional one-cylinder rotary compressor. Here, in the case of the volume ratio Vr=0.5, the torque variation range and the torque variation ratio are greater than those in the case of Vr=1. Specifically, large vibration requires suppression of the vibration by means of torque control, resulting in that the torque control might cause the operational efficiency to be lower than that in the conventional one-cylinder rotary compressor.

[0115] As described above, in the two-stage compression configuration of this embodiment, the volume ratio Vr between the inner and outer cylinder chambers (C1, C2) are set in the range from about 0.6 to about 0.8.

Accordingly, the compression efficiency can be enhanced and vibration can be suppressed, as compared to a conventional one-cylinder compressor.

«EMBODIMENT 4»

[0116] As illustrated in FIGS. 15 and 16, a fourth embodiment is obtained by modifying the two-stage compression configuration of the compression mechanism (20) of the third embodiment. Specifically, in the compression mechanism (20) of the third embodiment, two cylinder chambers (C1, C2) are formed in the same plane. On the other hand, in the fourth embodiment, a compression mechanism (80) is formed by longitudinally stacking two cylinder chambers (82a, 82b), and constitutes a so-called two-stage rotary compressor.

[0117] More specifically, a compressor (60) of the fourth embodiment is configured such that the compression mechanism (80) having a low-stage compression mechanism (80a) and a high-stage compression mechanism (80b) and an electric motor (65) are housed in a casing (61) which is a sealed vessel in an elongated cylindrical shape. In the casing (61), the electric motor (65) is located above the compression mechanism (80).

[0118] A suction pipe (62) penetrates the body of the casing (61), and a discharge pipe (63) penetrates the body at a position above the suction pipe (62). The discharge pipe (63) is bent at the inlet side thereof in the casing (61), then extends horizontally, and is open at the end.

[0119] The electric motor (65) includes a stator (66) and a rotor (67). The stator (66) is fixed to the inner peripheral surface of the casing (61). The rotor (67) is located at the inner side of the stator (66). A center portion of the rotor (67) is coupled to a main shaft portion (71) of a drive shaft (70) which longitudinally extends.

[0120] The drive shaft (70) constitutes a drive shaft. The drive shaft (70) has a first eccentric part (72) and a second eccentric part (73) which are located in this order from the bottom. Each of the first eccentric part (72) and the second eccentric part (73) has a diameter greater than that of the main shaft portion (71), and is eccentric to the axis of the main shaft portion (71). The first eccentric part (72) and the second eccentric part (73) are respectively eccentric to the axis of the main shaft portion (71) in opposite directions. The height of the first eccentric part (72) is greater than that of the second eccentric part (73).

[0121] The compression mechanism (80) is configured such that a rear head (84), a first cylinder (81a), a middle plate (86), a second cylinder (81b), and a front head (83) are stacked in this order. The first cylinder (81 a) houses a first rotary piston (87a). The second cylinder (81b) houses a second rotary piston (87b).

[0122] The first cylinder (81a), the first rotary piston (87a), the rear head (84), and the middle plate (86) constitute the low-stage compression mechanism (80a). The second cylinder (81b), the second rotary piston (87b), the front head (83), and the middle plate (86) constitute the high-stage compression mechanism (80b). Each of the low-stage compression mechanism (80a) and the high-stage compression mechanism (80b) is configured by a rotary fluid machine of a swinging piston type, which is a type of a positive-displacement fluid machine.

[0123] As illustrated in FIG. 16, the first rotary piston (87a) of the low-stage compression mechanism (80a) has an annular shape. The first eccentric part (72) is rotatably fitted in this first rotary piston (87a) of the low-stage compression mechanism (80a). The second rotary piston (87b) of the high-stage compression mechanism (80b) also has an annular shape. The second eccentric part (73) is rotatably fitted in this second rotary piston (87b) of the high-stage compression mechanism (80b).

[0124] The inner peripheral surfaces of the rotary pistons (87a, 87b) are respectively in slidable contact with the outer peripheral surfaces of the eccentric parts (72, 73), and the outer peripheral surfaces of the rotary pistons (87a, 87b) are respectively in slidable contact with the inner peripheral surfaces of the cylinders (81a, 81b). A cylinder chamber (82a, 82b) is formed between the outer peripheral surface of the rotary piston (87a, 87b) and the inner peripheral surface of the cylinder (81a, 81b). A blade (74) in the shape of a flat plate projects from a side surface of each of the rotary pistons (87a, 87b). The blade (74) is supported by the cylinder (81a, 81b) with a swing bushing (75) interposed therebetween. The swing bushing (75) is provided between a discharge port (89a, 89b) and a suction port (88a, 88b) which will be described later. The blade (74) partitions the cylinder chamber (82a, 82b) into a high-pressure side and a low-pressure side.

[0125] In this manner, the compression mechanism (80) is configured such that rotation of the eccentric part (72, 73) causes the rotary piston (87a, 87b) to revolve and swing in the cylinder chamber (82a, 82b). The rotational phases of the rotary pistons (87a, 87b) shift from each other by 180°.

[0126] The first cylinder (81a) of the low-stage compression mechanism (80a) and the second cylinder (81 b) of the high-stage compression mechanism (80b) have the same inner diameter. The first rotary piston (87a) of the low-stage compression mechanism (80a) and the second rotary piston (87b) of the high-stage compression mechanism (80b) have the same outer diameter. The height of the first cylinder (81 a) of the low-stage compression mechanism (80a) is greater than that of the second cylinder (81b) of the high-stage compression mechanism (80b).

[0127] The middle plate (86) has an annular intermediate passageway (90). A discharge port (89a) of the low-stage compression mechanism (80a) is formed in the middle plate (86). This discharge port (89a) allows the high-pressure side of the first cylinder chamber (82a) of the low-stage compression mechanism (80a) to communicate with the intermediate passageway (90). On the other hand, a discharge port (89b) of the high-stage compression mechanism (80b) is formed in the front head (83). This discharge port (89b) allows the high-pressure side of the second cylinder chamber (82b) of the high-stage compression mechanism (80b) to communicate with the space in the casing (61). These discharge ports (89a, 89b) have discharge valves (not shown) for opening and closing the respective outlets of the discharge ports (89a, 89b).

[0128] The first cylinder (81a) of the low-stage compression mechanism (80a) has a suction port (88a). This suction port (88a) radially penetrates the first cylinder (81a). The terminal end of the suction port (88a) is open to the first cylinder chamber (82a). This suction port (88a) is connected to the suction pipe (62). The second cylinder (81b) of the high-stage compression mechanism (80b) has a suction port (88b) extending from the middle plate (86). The starting end of the suction port (88b) is open to the intermediate passageway (90), and the terminal end of the suction port (88b) is open to the second cylinder chamber (82b).

[0129] An oil sump for storing lubricating oil is formed at the bottom of the casing (61). A centrifugal oil supply pump (92) immersed in the oil sump is provided at the bottom of the drive shaft (70). This oil supply pump (92) longitudinally extends in the drive shaft (70), and is connected to an oil supply passageway (91) which communicates with the low-stage compression mechanism (80a) and the high-stage compression mechanism (80b). The oil supply pump (92) is configured to supply lubricating oil in the oil sump to a sliding portion of the low-stage compression mechanism (80a) and to a sliding portion of the high-stage compression mechanism (80b) through the oil supply passageway (91).

[0130] In this embodiment, the volume V1 of the first cylinder chamber (82a) is also larger than the volume V2 of the second cylinder chamber (82b), and the volume ratio Vr (V2/V1) of the second cylinder chamber (82b) to the first cylinder chamber (82a) is also set at about 0.7.

[0131] In this compressor (60), when the electric motor (65) is started, the rotary pistons (87a, 87b) revolve, while swinging in the cylinder chambers (81a, 81b). Then, the compression mechanism (80) performs given compression operation.

[0132] This compression operation is described with reference to FIG. 16. In FIG. 16, the rotary piston (87a, 87b) swings, while rotating in the clockwise direction. The position in which the rotary piston (87a, 87b) is in contact with the top dead center is defmed as a rotation angle of 0°, and the position in which the rotary piston (87a, 87b) is in contact with the bottom dead center is defined as a rotation angle of 180°. First, when the drive shaft (70) slightly rotates from the state of a rotation angle of 0° so that the contact point between the first rotary piston (87a) and the first cylinder (81a) passes by the opening of the suction port (88a), refrigerant starts to flow from the suction port (88a) to the first cylinder chamber (82a). The refrigerant continues to flow into the first cylinder chamber (82a) until the rotation angle of the drive shaft (70) reaches 360°.

[0133] Subsequently, in a state where the flow of refrigerant into the first cylinder chamber (82a) is finished (i.e., the rotation angle of the drive shaft (70) is 360°), when the drive shaft (70) slightly rotates so that the contact point between the first rotary piston (87a) and the first cylinder (81a) passes by the opening of the suction port (88a), confinement of refrigerant in the first cylinder chamber (82a) is completed. Then, from this state, the drive shaft (70) further rotates, thereby starting compression of refrigerant. When the pressure of refrigerant in the first cylinder chamber (82a) exceeds the pressure of refrigerant in the intermediate passageway (90), the discharge valve is opened, thereby discharging intermediate-pressure refrigerant from the discharge port (89a) into the intermediate passageway (90). The discharge of refrigerant continues until the rotation angle of the drive shaft (70) reaches 360°.

[0134] On the other hand, in the high-stage compression mechanism (80b), the rotation of the drive shaft (70) causes intermediate-pressure refrigerant in the intermediate passageway (90) to flow from the suction port (88b) into the second cylinder chamber (82b). Specifically, when the contact point between the second rotary piston (87b) and the second cylinder (81b) passes by the opening of the suction port (88b), refrigerant starts to flow from the intermediate passageway (90) into the second cylinder chamber (82b). The flow of intermediate-pressure refrigerant continues until the rotation angle of the drive shaft (70) reaches 360°.

[0135] Thereafter, when the contact point between the second rotary piston (87b) and the second cylinder (81b) passes by the opening of the suction port (88b) so that confinement of refrigerant in the second cylinder chamber (82b) is completed, compression of refrigerant starts. Then, when the pressure of refrigerant in the second cylinder chamber (82b) exceeds the pressure of refrigerant in the space in the casing (61), the discharge valve is opened, thereby discharging the high-pressure refrigerant from the discharge port (89b) into the space in the casing (61). The discharge of refrigerant continues until the rotation angle of the drive shaft (70) reaches 360°. The refrigerant discharged into the space in the casing (61) is discharged from the discharge pipe (63) to the refrigerant circuit. In this manner, in the compressor (60) of this embodiment, refrigerant compressed in the low-stage first cylinder chamber (82a) is further compressed in the high-stage second cylinder chamber (82b), thereby performing two-stage compression.

[0136] As in the third embodiment, the relationship between the volume ratio Vr (V2n/V1) and the torque variation range and the relationship between the volume ratio Vr (V2n/V1) and the torque variation ratio also apply to the relationships shown in FIGS. 13 and 14. That is, the torque variation range and the torque variation ratio in the case of the volume ratio Vr=0.6 and 0.8 are smaller than those in the case of the volume ratio Vr= 0.5 and 1. Accordingly, such small torque variations enable suppression of vibration. Thus, in the two-stage compression configuration in which the low-stage and high-stage cylinder chambers (82a, 82b) are longitudinally stacked, as long as the volume ratio Vr between these cylinder chambers (82a, 82b) are set in the range from about 0.6 to about 0.8, the compression efficiency can be enhanced and vibration can be suppressed, as compared to a conventional one-cylinder compressor.

INDUSTRIAL APPLICABILITY

[0137] As described above, the present invention is useful for rotary compressors each including two cylinder chambers whose volume varies with eccentric rotation of a piston.

Claims

1. A rotary compressor, comprising:

    a compression mechanism (20, 80) including a piston (22, 87a, 87b) and a cylinder (21, 81a, 81b) having two cylinder chambers (C1, C2, 82a, 82b);
    an electric motor (30, 65) configured to change volumes of the cylinder chambers (C1, C2, 82a, 82b) by causing relative eccentric rotation between the cylinder (21, 81a, 81b) and the piston (22, 87a, 87b); and
    a torque control means (50) configured to change an output torque of the electric motor (30, 65) in accordance with a variation in a load torque of the compression mechanism (20, 80) in one turn of rotation.

2. The rotary compressor of claim 1, wherein the cylinder (21) includes an annular cylinder chamber (C1, C2), and
the piston (22) is an annular piston (22) housed in the annular cylinder chamber (C1, C2) and partitioning the annular cylinder chamber (C1, C2) into two cylinder chambers which are an outer cylinder chamber (C1) and an inner cylinder chamber (C2).

3. The rotary compressor of claim 2, wherein a volume ratio of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is in the range from 0.6 to 1.0.

4. The rotary compressor of claim 2 or 3, wherein the electric motor (30) is a brushless DC motor.

5. The rotary compressor of claim 2 or 3, wherein the torque control means (50) is configured to change an output torque of the electric motor (30) by changing one of an input current, an input voltage, and an input current phase of the electric motor (30).

6. The rotary compressor of claim 2 or 3, wherein the electric motor (30) is coupled to the cylinder (21) configured to rotate relative to the annular piston (22) which is stationary.

7. The rotary compressor of claim 2 or 3, wherein the electric motor (30) is coupled to the annular piston (22) configured to rotate relative to the cylinder (21) which is stationary.

8. The rotary compressor of claim 2, wherein the compression mechanism (20) is configured to perform two-stage compression on fluid, with one of the outer and inner cylinder chambers (C1) and (C2) used at a low-stage side and the other one of the outer and inner cylinder chambers (C1) and (C2) used at a high-stage side.

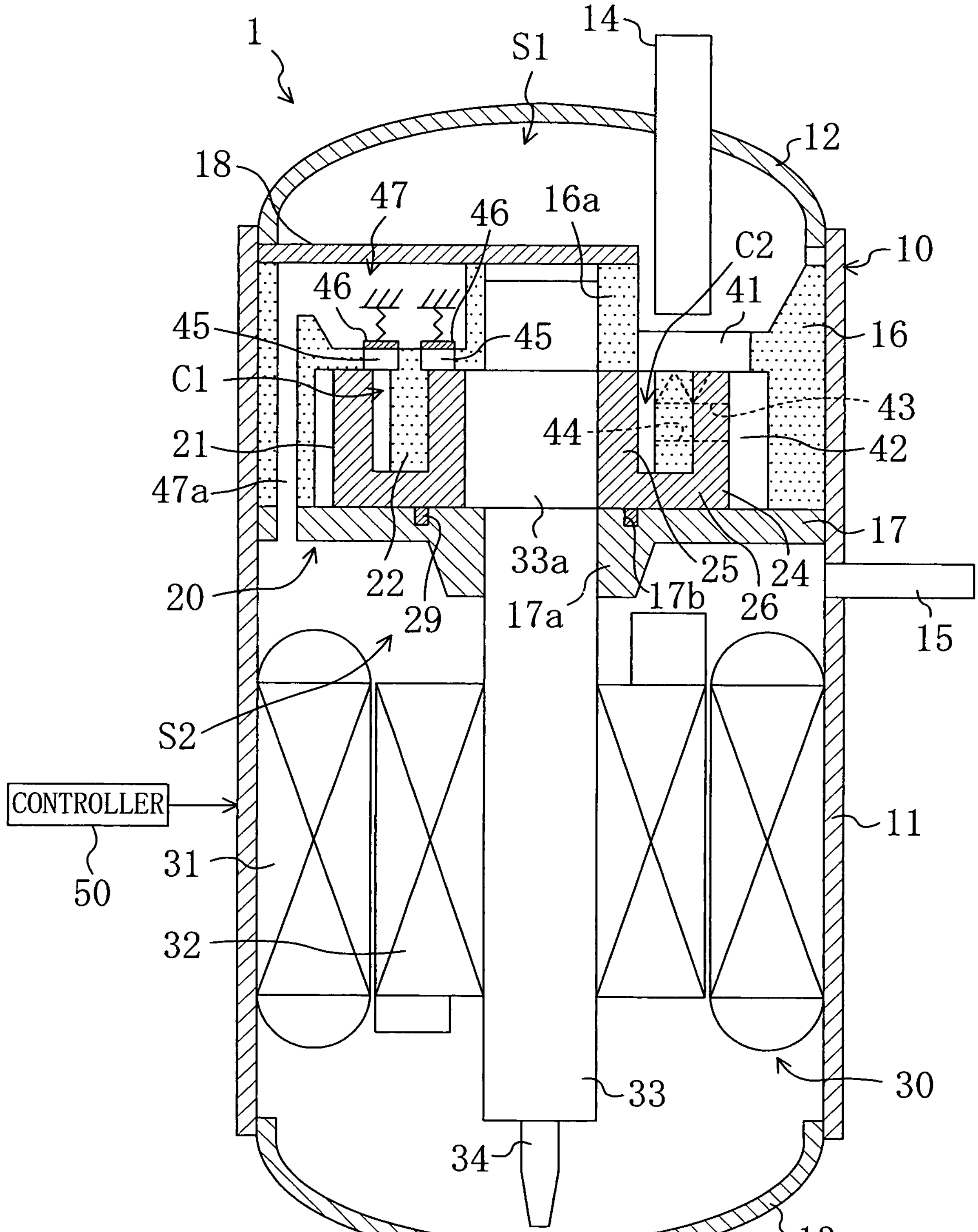
9. The rotary compressor of claim 8, wherein a volume ratio of the inner cylinder chamber (C2) to the outer cylinder chamber (C1) is in the range from 0.6 to 0.8.

10. The rotary compressor of claim 1, wherein the cylinders (81a, 81b) are respectively a low-stage first cylinder (81a) and a high-stage second cylinder (81b) both including cylinder chambers (82a, 82b), the pistons (87a, 87b) are respectively a first rotary piston (87a) housed in the cylinder chamber (82a) of the first cylinder (81a) and a second rotary piston (87b) housed in the cylinder chamber (82b) of the second cylinder (81b), and
the compression mechanism (80) is configured to perform two-stage compression on fluid in the cylinders (81a, 81b) by eccentric rotation of the rotary pistons (87a, 87b) caused by the electric motor (65).

11. The rotary compressor of claim 10, wherein a volume ratio of the cylinder chamber (82b) of the second cylinder (81b) to the cylinder chamber (82a) of the first cylinder (81a) is in the range from 0.6 to 0.8.

12. The rotary compressor of claim 10 or 11, wherein the compression mechanism (80) is configured such that a rotational phase of the rotary piston (87a) of the first cylinder (81a) is shifted from a rotational phase of the rotary piston (87b) of the second cylinder (81b) by 180°.

FIG. 1



1
S1
14
12
18
47
46
16a
C2
10
46
41
45
45
16
C1
43
44
21
42
47a
17
33a
25
24
20
22
17a
17b
26
15
29
S2
CONTROLLER
11
50
31
32
33
30
34
13

FIG. 2

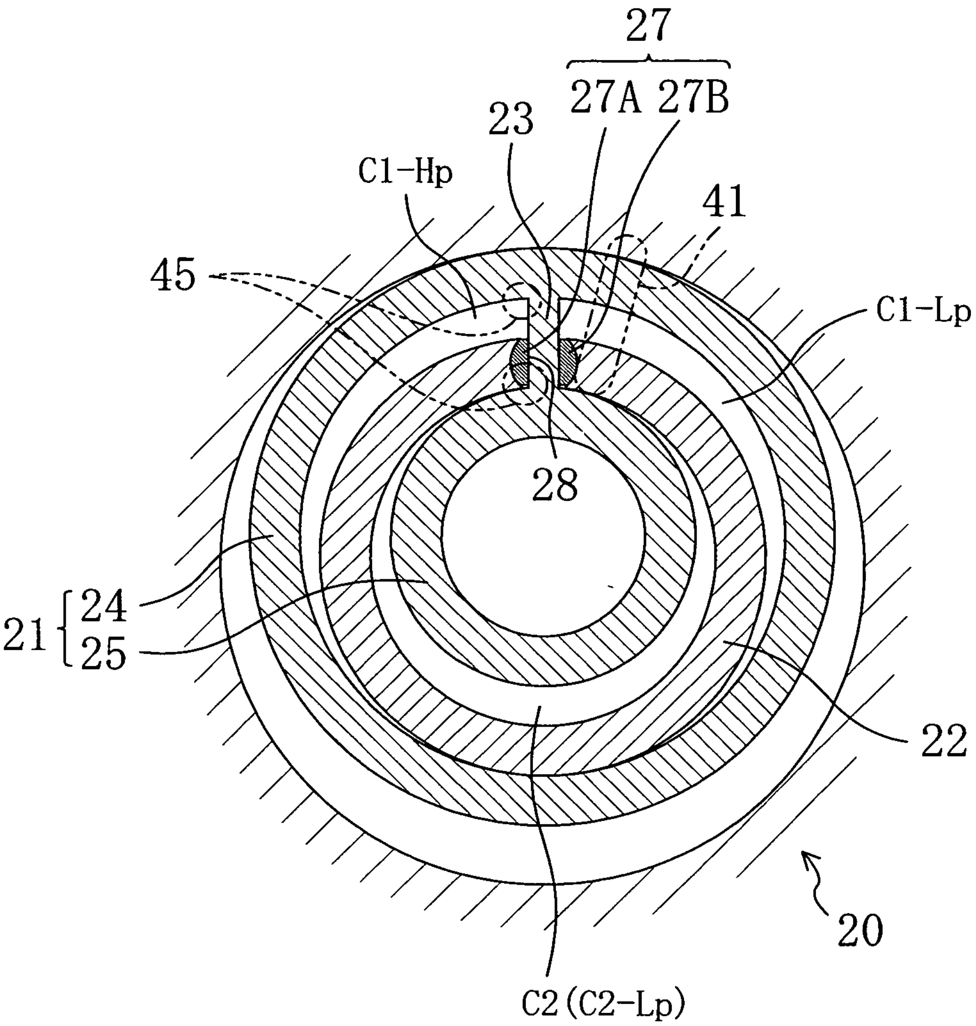


FIG. 3

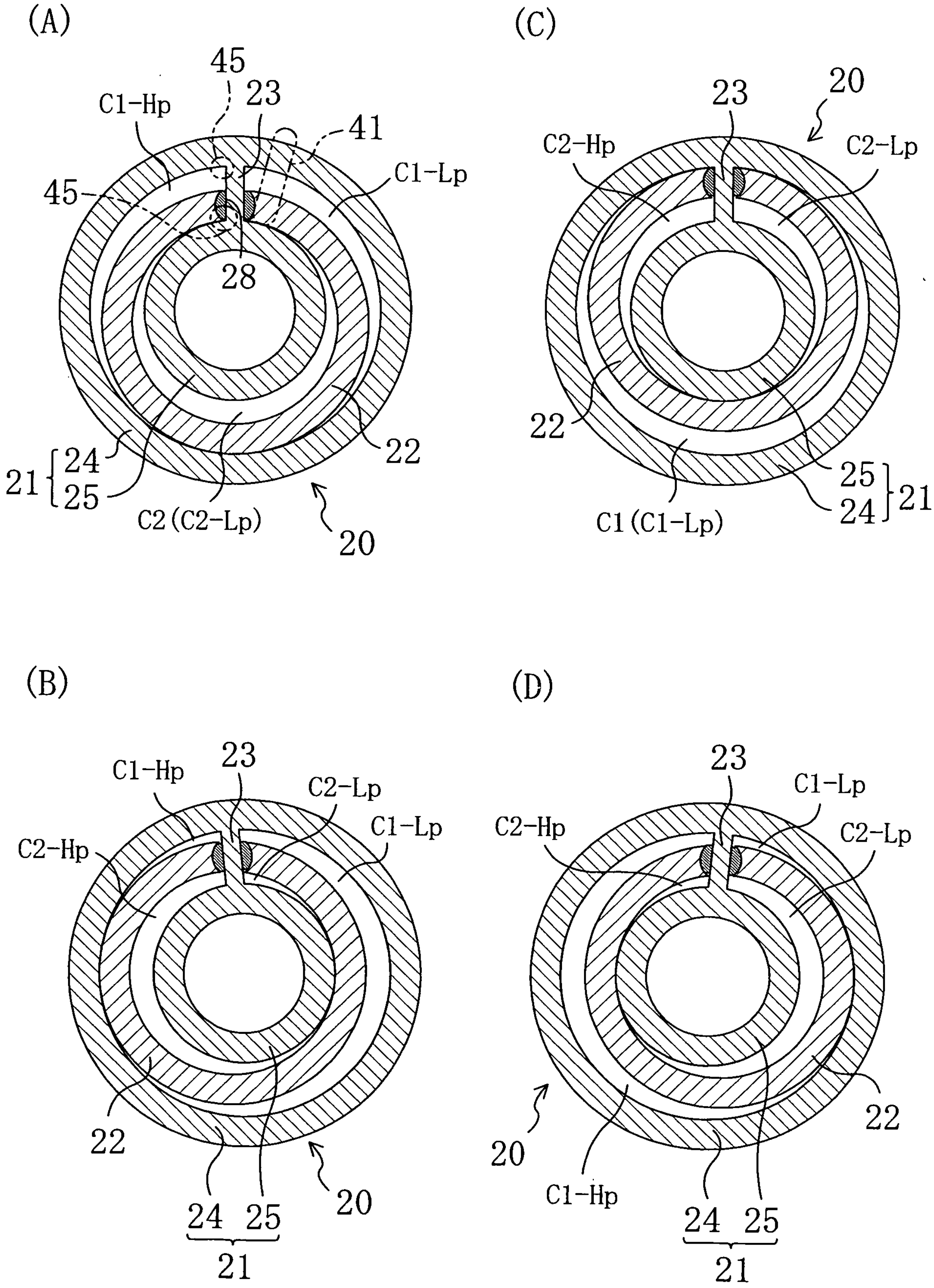

(A)
45
23
C1-Hp
41
C1-Lp
45
28
21
24
25
C2(C2-Lp)
22
20
(C)
23
20
C2-Hp
C2-Lp
22
25
24
21
C1(C1-Lp)
(B)
C1-Hp
23
C2-Lp
C1-Lp
C2-Hp
22
24
25
21
20
(D)
23
C1-Lp
C2-Hp
C2-Lp
20
22
C1-Hp
24
25
21

FIG. 4

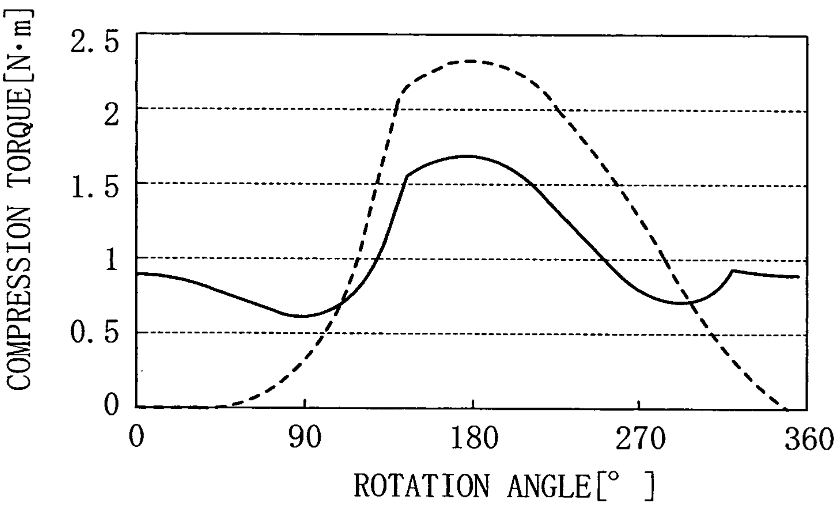

COMPRESSION TORQUE[N·m]
2.5
2
1.5
1
0.5
0
0
90
180
270
360
ROTATION ANGLE[°]

FIG. 5

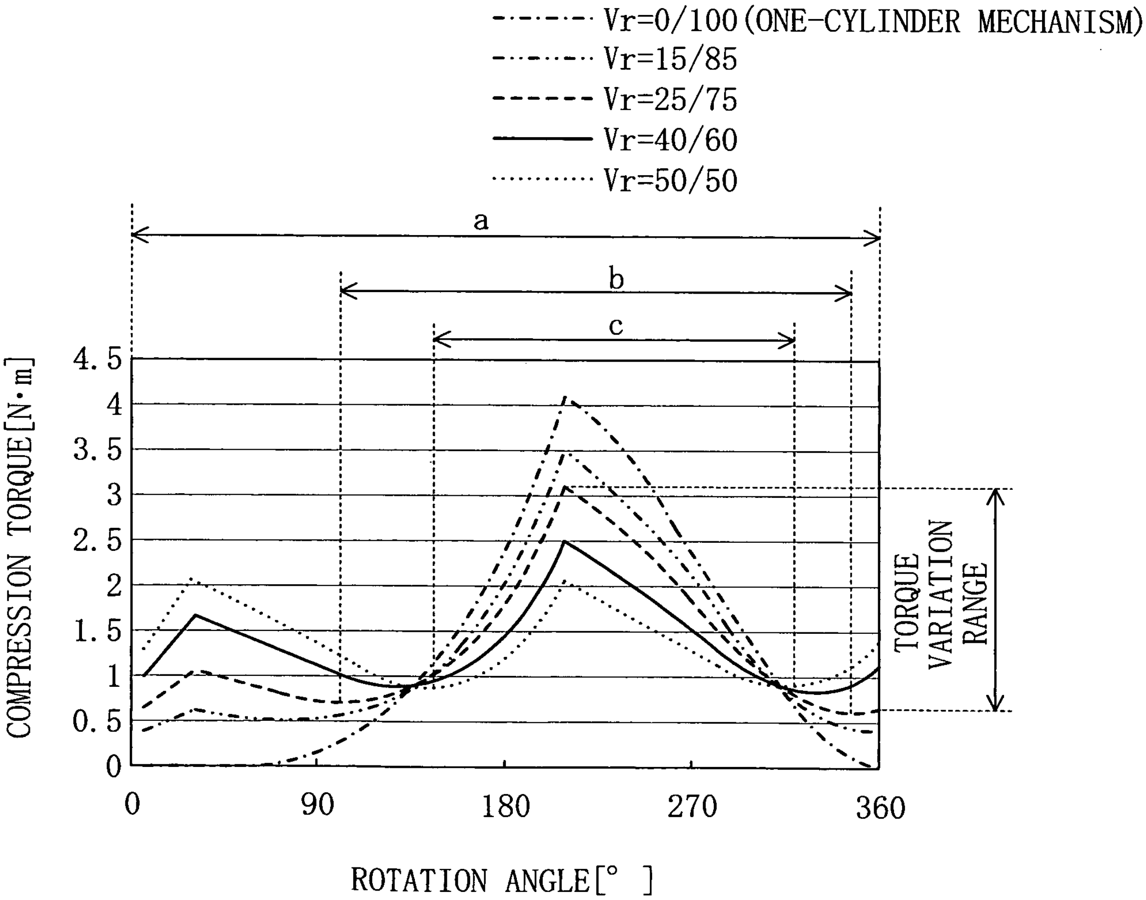

Vr=0/100(ONE-CYLINDER MECHANISM)
Vr=15/85
Vr=25/75
Vr=40/60
Vr=50/50
a
b
c
COMPRESSION TORQUE[N·m]
4.5
4
3.5
3
2.5
2
1.5
1
0.5
0
0
90
180
270
360
TORQUE VARIATION RANGE
ROTATION ANGLE[°]

FIG. 6

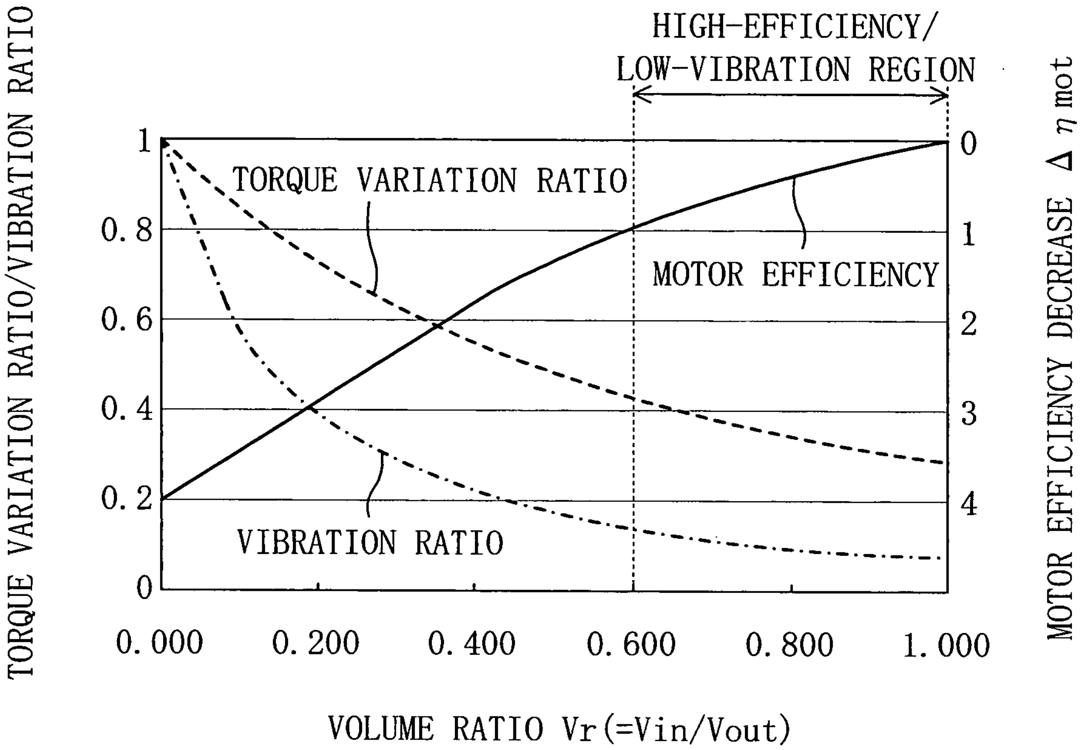

HIGH-EFFICIENCY/
LOW-VIBRATION REGION
TORQUE VARIATION RATIO
MOTOR EFFICIENCY
VIBRATION RATIO
TORQUE VARIATION RATIO/VIBRATION RATIO
MOTOR EFFICIENCY DECREASE Δ η mot
1
0.8
0.6
0.4
0.2
0
0
1
2
3
4
0.000
0.200
0.400
0.600
0.800
1.000
VOLUME RATIO Vr(=Vin/Vout)

FIG. 7

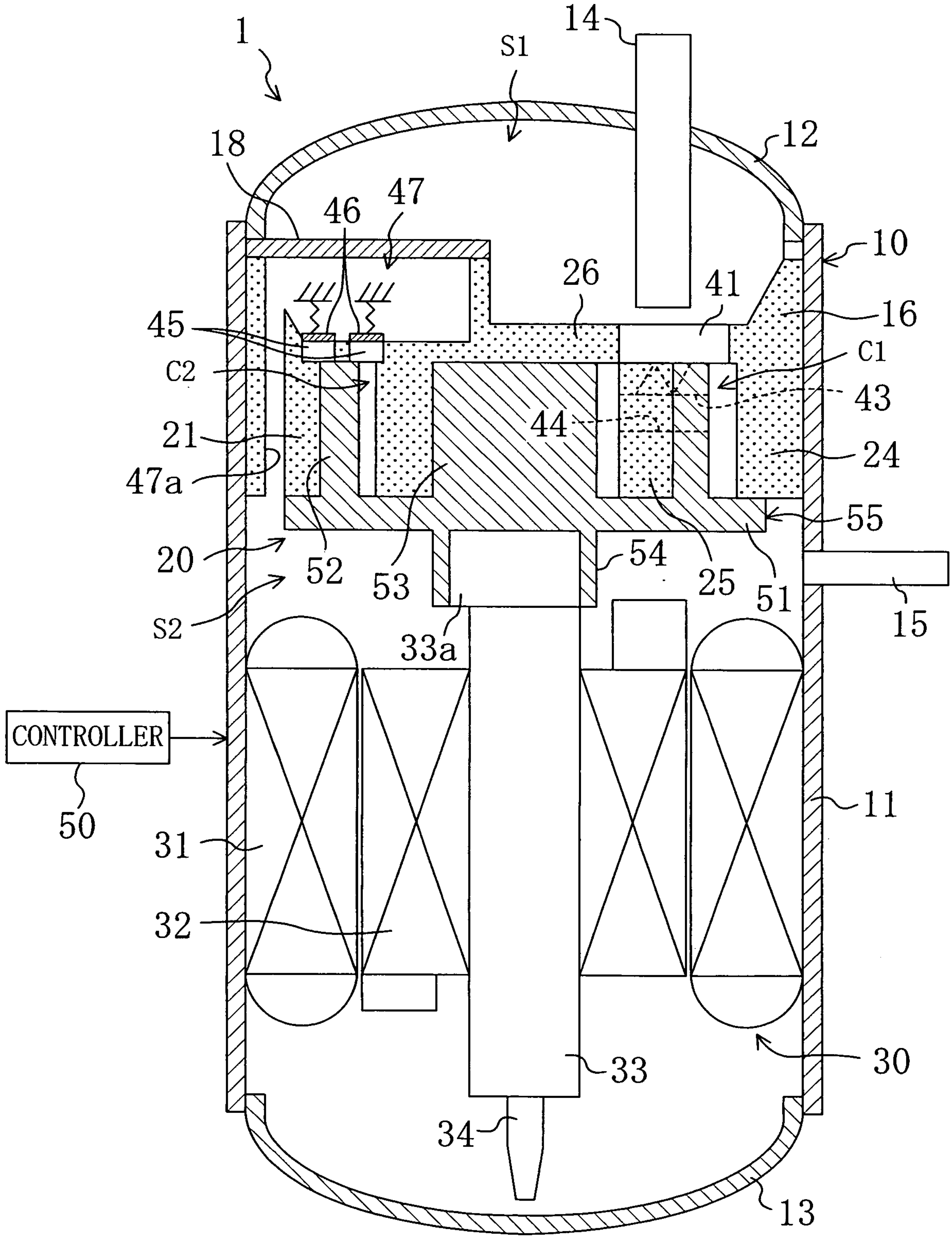

1
S1
14
12
18
46
47
10
26
41
16
45
C1
C2
43
44
21
24
47a
55
20
54
52
53
25
51
S2
15
33a
CONTROLLER
50
11
31
32
33
30
34
13

FIG. 8

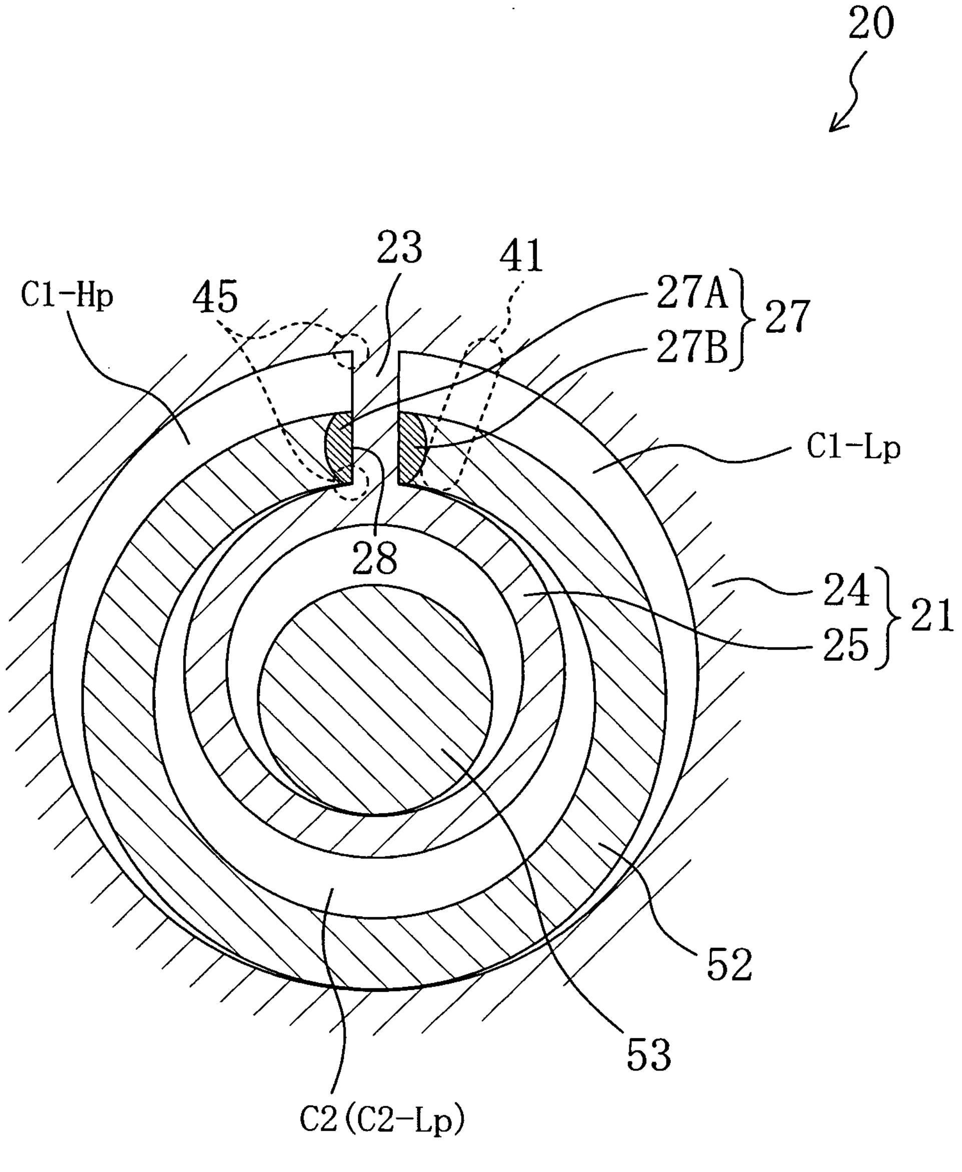


FIG. 9

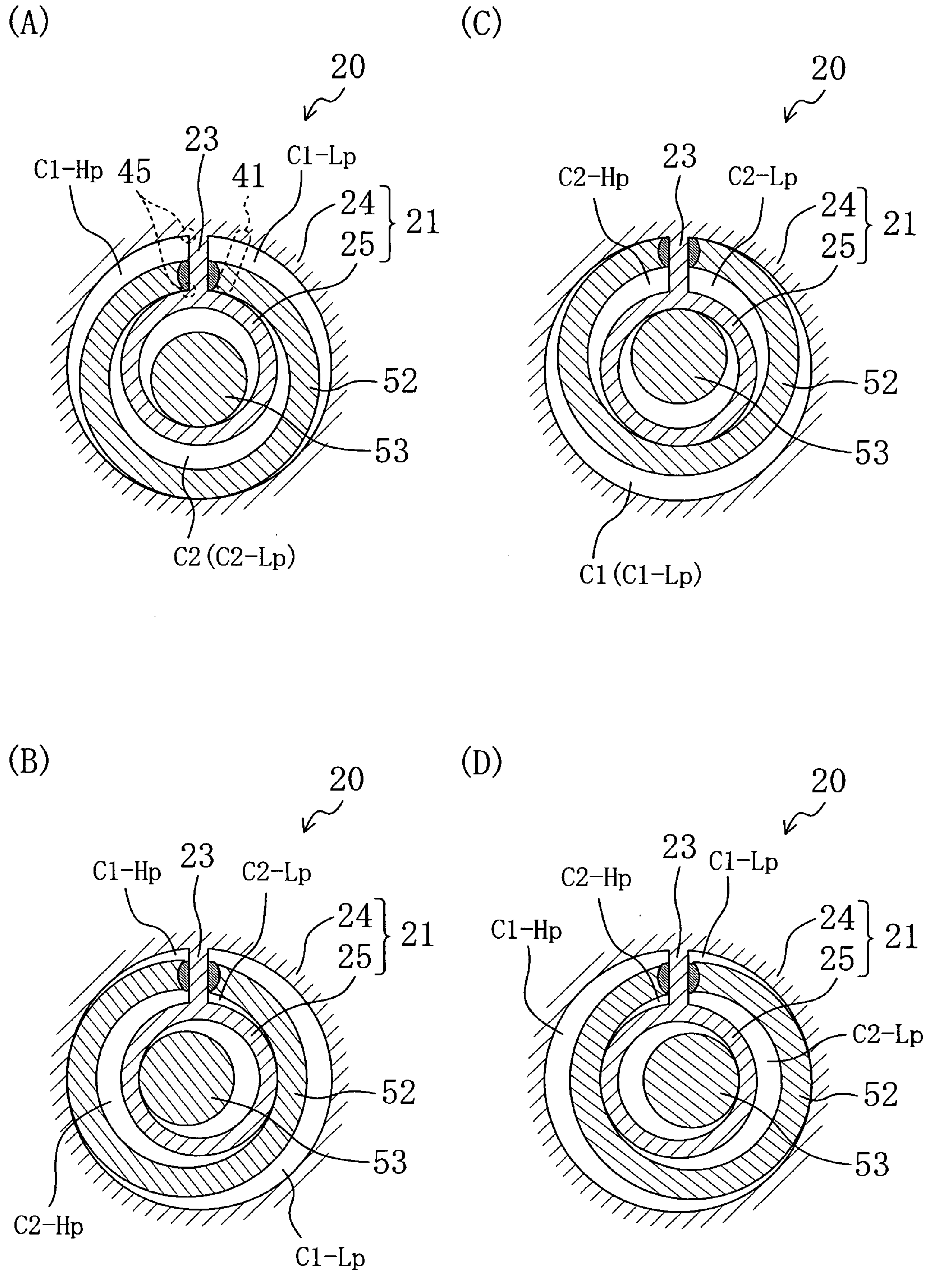

(A)
20
23
C1-Hp
45
41
C1-Lp
24
25
21
52
53
C2 (C2-Lp)
(C)
20
23
C2-Hp
C2-Lp
24
25
21
52
53
C1 (C1-Lp)
(B)
20
23
C1-Hp
C2-Lp
24
25
21
52
53
C2-Hp
C1-Lp
(D)
20
23
C2-Hp
C1-Lp
C1-Hp
24
25
21
C2-Lp
52
53

FIG. 10

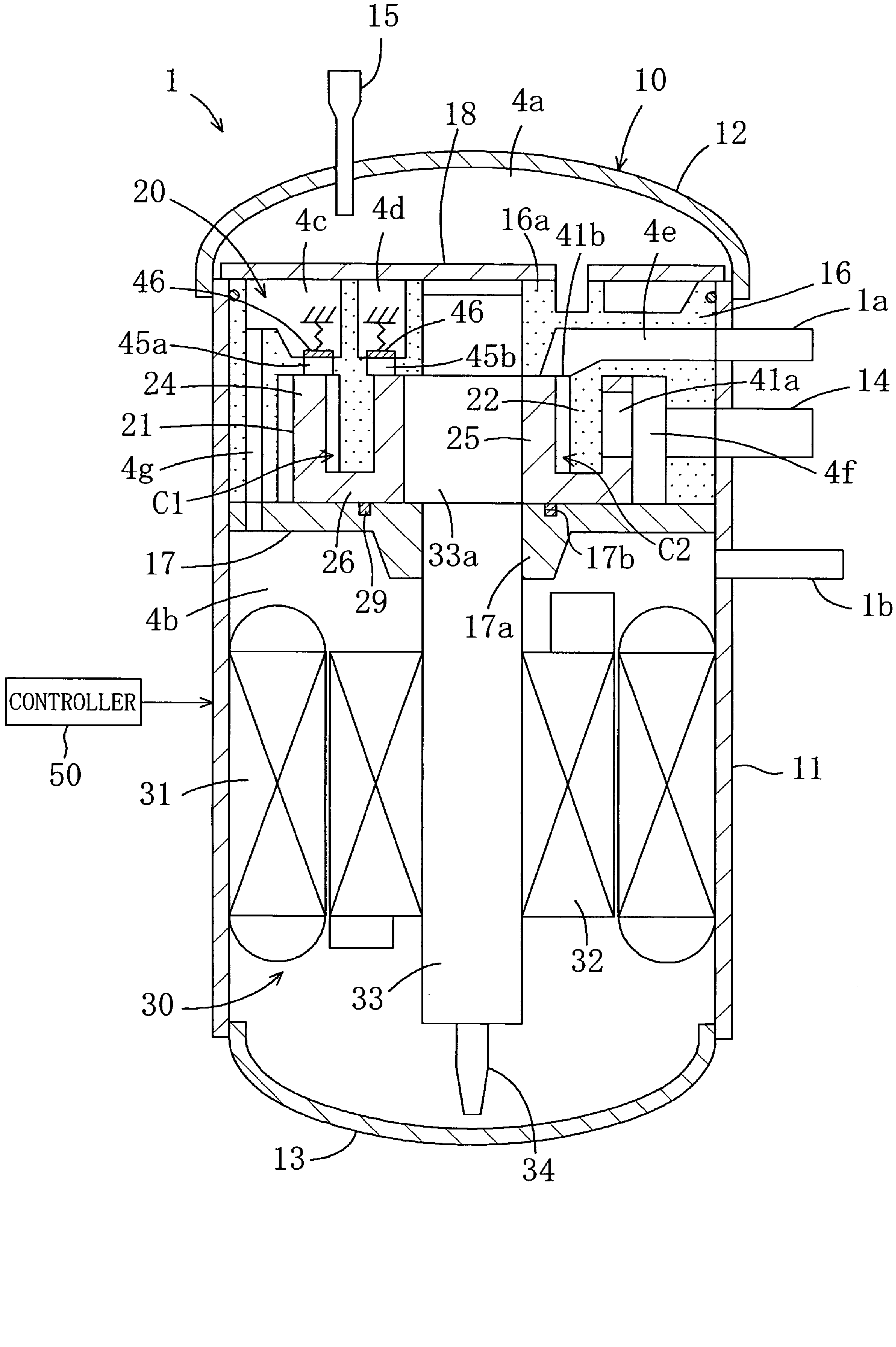

15
1
10
18
4a
12
20
4c
4d
16a
41b
4e
16
46
1a
46
45a
45b
24
22
41a
14
21
25
4g
4f
C1
C2
26
33a
17b
17
4b
29
17a
1b
CONTROLLER
50
31
11
32
30
33
13
34

FIG. 11

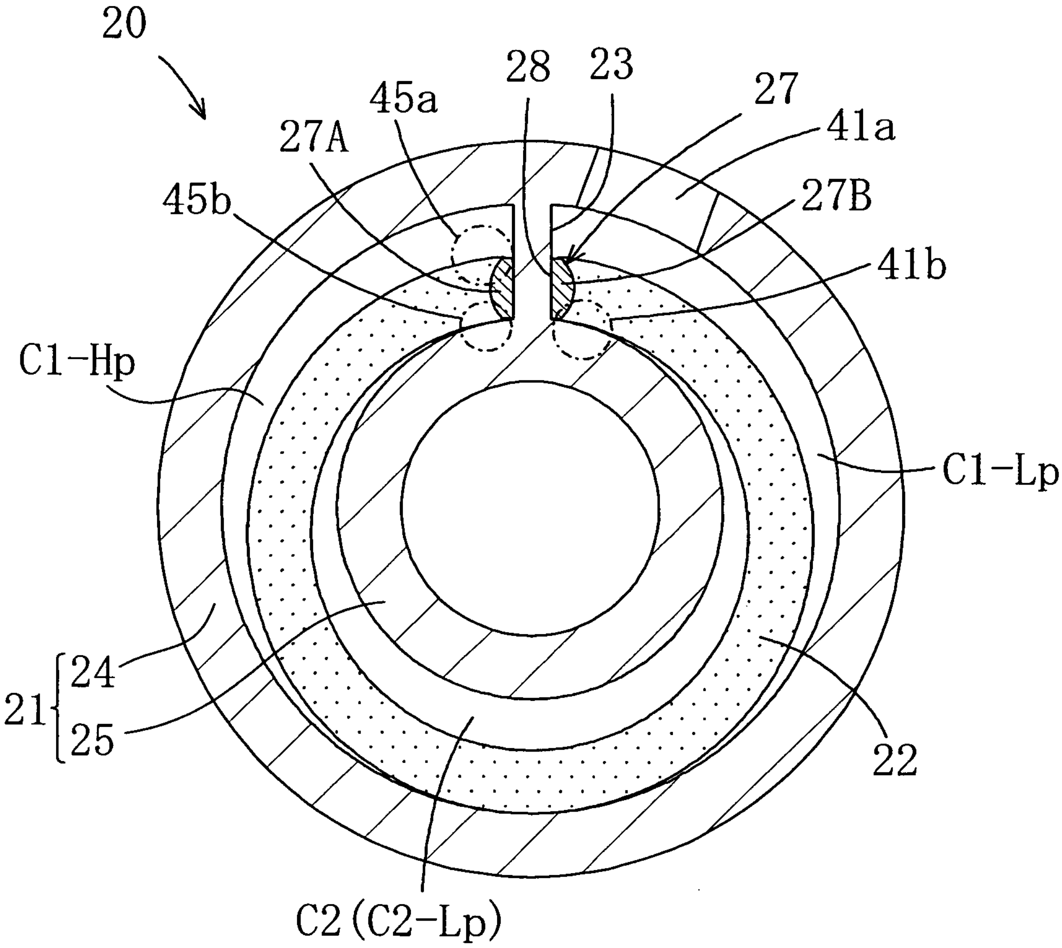


FIG. 12

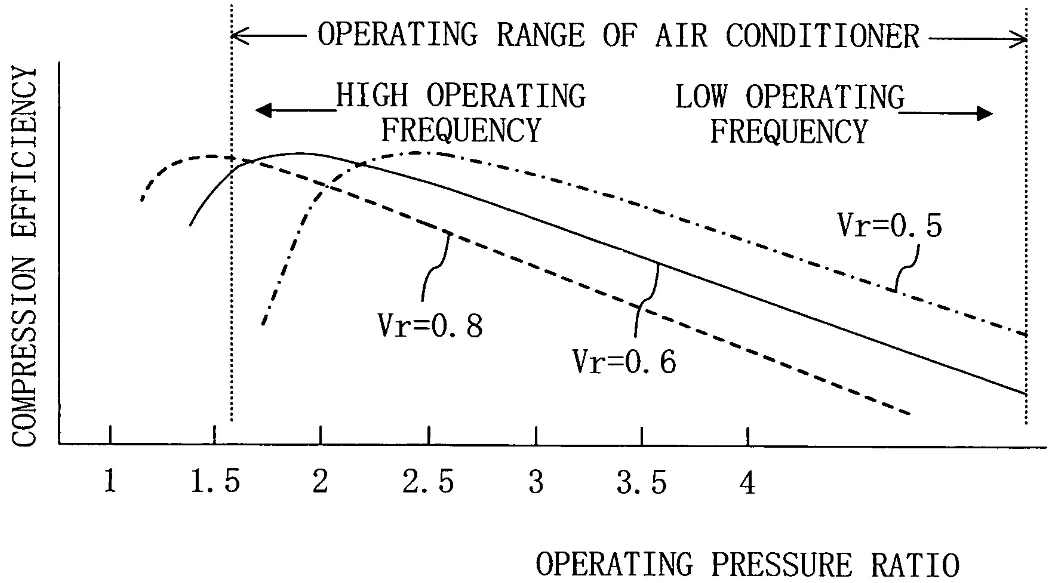

OPERATING RANGE OF AIR CONDITIONER
HIGH OPERATING FREQUENCY
LOW OPERATING FREQUENCY
COMPRESSION EFFICIENCY
Vr=0. 5
Vr=0. 8
Vr=0. 6
1
1. 5
2
2. 5
3
3. 5
4
OPERATING PRESSURE RATIO

FIG. 13

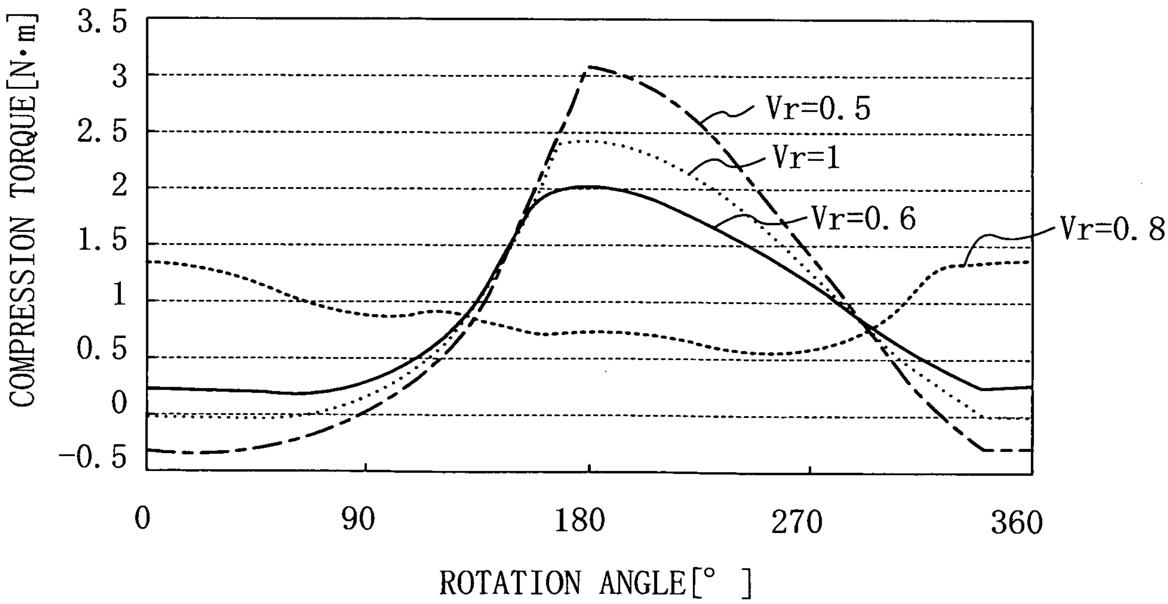

COMPRESSION TORQUE[N·m]
3. 5
3
2. 5
2
1. 5
1
0. 5
0
-0. 5
Vr=0. 5
Vr=1
Vr=0. 6
Vr=0. 8
0
90
180
270
360
ROTATION ANGLE[°]

FIG. 14

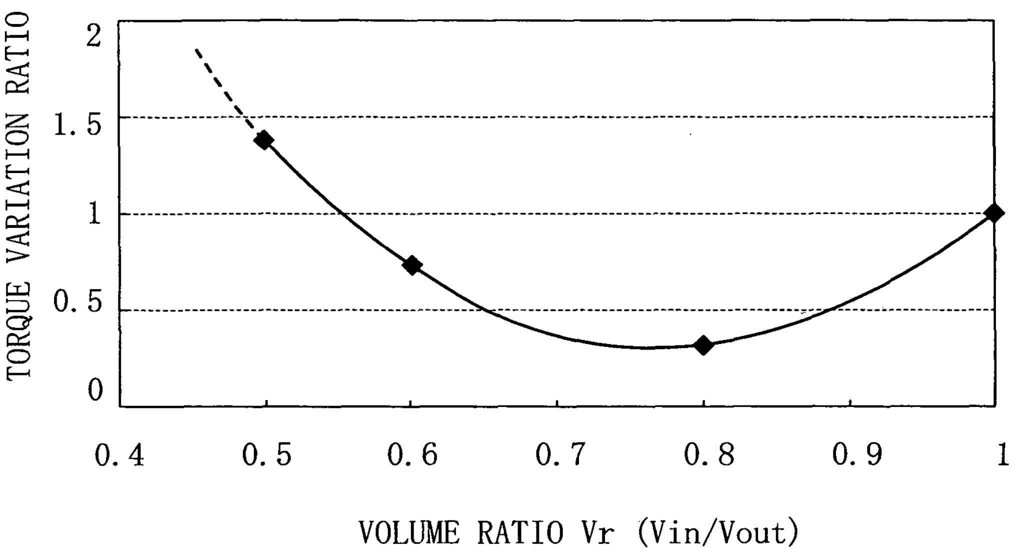

TORQUE VARIATION RATIO
2
1.5
1
0.5
0
0.4
0.5
0.6
0.7
0.8
0.9
1
VOLUME RATIO Vr (Vin/Vout)

FIG. 15

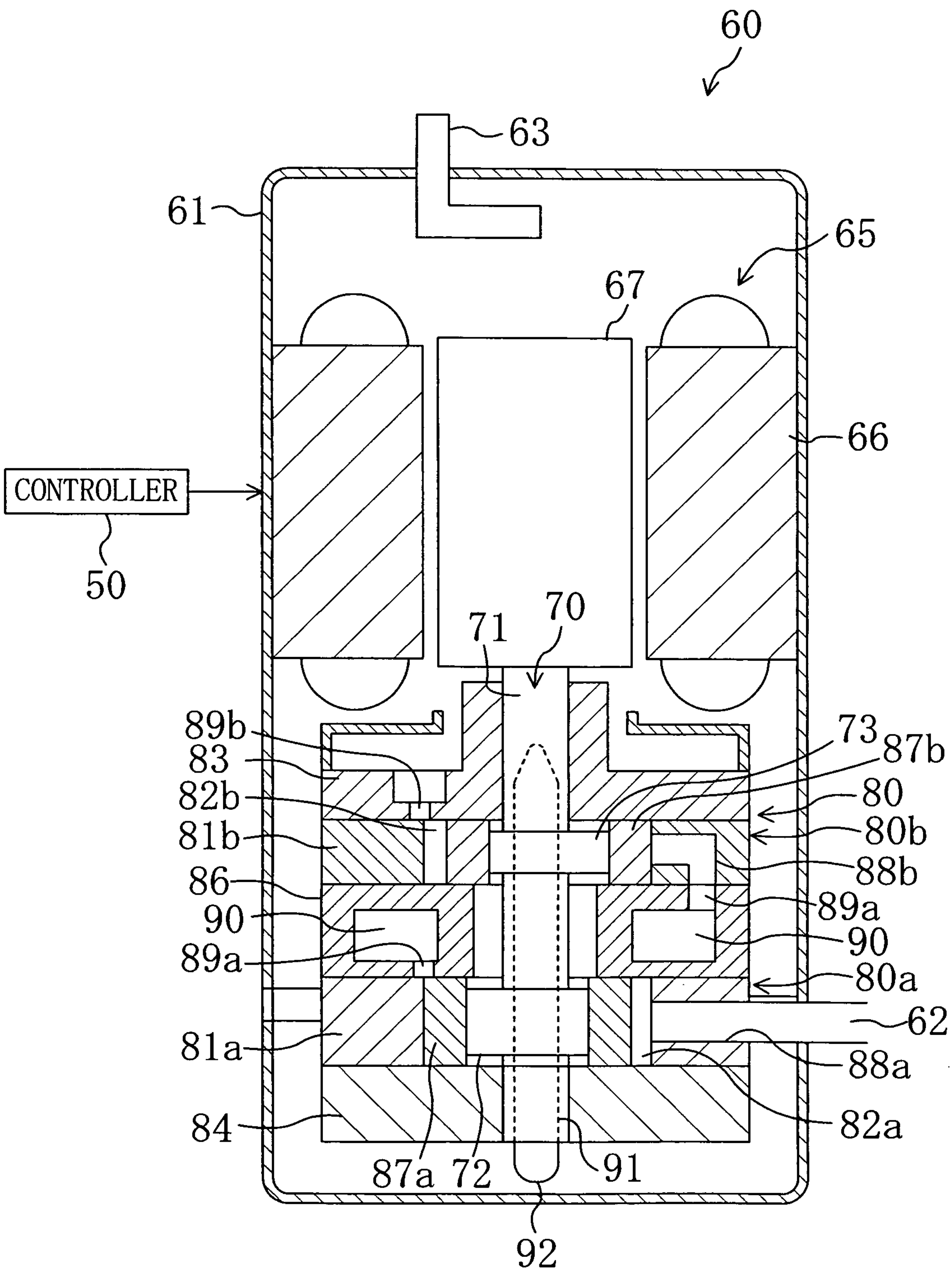
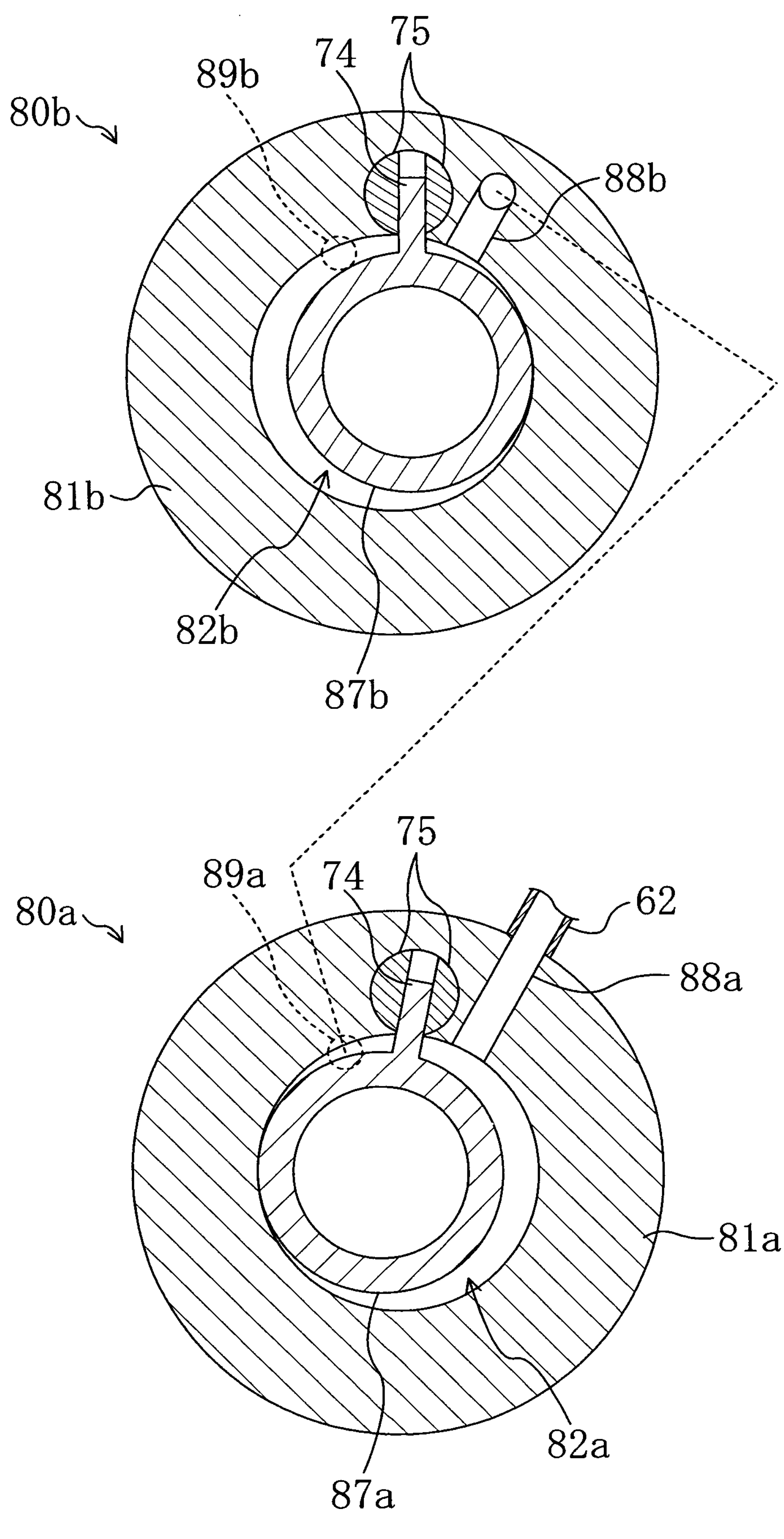

60
63
61
65
67
66
CONTROLLER
50
71
70
89b
83
82b
81b
86
90
89a
73
87b
80
80b
88b
89a
90
80a
62
81a
88a
84
82a
87a
72
91
92

FIG. 16



INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP2007/053811

A. CLASSIFICATION OF SUBJECT MATTER
F04C28/08(2006.01)i, *F04C18/32*(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)
F04C28/08, F04C18/32, 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	1996-2007
Kokai Jitsuyo Shinan Koho	1971-2007	Toroku Jitsuyo Shinan Koho	1994-2007

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.
Y	JP 63-147986 A (Daikin Industries, Ltd.), 20 June, 1988 (20.06.88), Page 3, lower right column, line 16 to page 4, upper right column, line 14 (Family: none)	1-12
Y	JP 2005-320927 A (Daikin Industries, Ltd.), 17 November, 2005 (17.11.05), Par. Nos. [0043] to [0048], [0102]; Figs. 1 to 2 & WO 2005/108793 A1	1-9
Y	JP 2006-207559 A (Toshiba Carrier Corp.), 10 August, 2006 (10.08.06), Par. Nos. [0009] to [0013]; Figs. 1 to 2 (Family: none)	1,10-12

☐ 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
"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 19 March, 2007 (19.03.07)	Date of mailing of the international search report 03 April, 2007 (03.04.07)
Name and mailing address of the ISA/ Japanese Patent Office Facsimile No.	Authorized officer Telephone No.

Form PCT/ISA/210 (second sheet) (April 2005)

REFERENCES CITED IN THE DESCRIPTION

This list of references cited by the applicant is for the reader's convenience only. It does not form part of the European patent document. Even though great care has been taken in compiling the references, errors or omissions cannot be excluded and the EPO disclaims all liability in this regard.

Patent documents cited in the description

- JP 6288358 A **[0004]**