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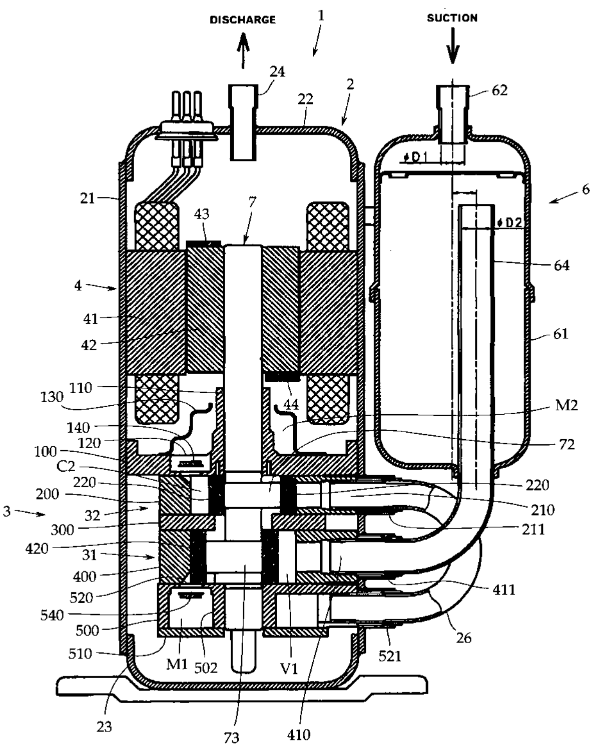
(54) Rotary compressor with accumulator and heat pump system

(57) There is provided a rotary compressor in which the pressure loss in a refrigerant suction pipe connecting the compressor and an accumulator to each other is reduced, and the compression efficiency is increased. The inside diameter D2 of a refrigerant suction pipe 64 on the outlet side of the accumulator 6 is made larger than the inside diameter D1 of a refrigerant return pipe 62 on the inlet side thereof (D2 > D1).

FIG. 1



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Description

TECHNICAL FIELD

[0001] The present invention relates to a rotary compressor used for the refrigerating cycle of a heat pump system for an air conditioner or the like. More particularly, it relates to a technique for reducing the pressure loss in a connection pipe connecting a compressor body and an accumulator to each other.

BACKGROUND ART

[0002] A rotary compressor used for a heat pump system for an air conditioner or the like is provided with an accumulator, which is used for separating a refrigerant returning from the refrigerating cycle of the system into gas and liquid, at the side of the compressor body to prevent liquid refrigerant from flowing into a compressing section in a transient state in terms of the reliability of the compressing section.

[0003] To the upper part of the accumulator, a refrigerant return pipe into which the refrigerant returning from the refrigerating cycle flows is connected. In the lower part of the accumulator, a refrigerant suction pipe, one end having an L shape of which is extended to an upper part in the accumulator and the other end of which is connected to a suction chamber of a low-stage compressing section from the side surface of the compressor, is provided.

[0004] For the accumulator provided for the conventional rotary compressor, there is no example in which the inside diameters of the refrigerant return pipe and the refrigerant suction pipe are considered in detail. That is to say, for example, as described in Japanese Patent Application Publication No. H05-195954, since the flow rates of refrigerant flowing in the refrigerant return pipe and the refrigerant suction pipe are the same at the time of steady continuous operation, the inside diameters of these pipes have been made approximately equal.

[0005] Conventionally, the refrigerant suction pipe connecting the rotary compressor and the accumulator to each other has presented a problem as described below. For the rotary compressor, since the rate of change of the suction volume during one turn is not constant, the flow velocity of refrigerant fluctuates in the refrigerant suction pipe.

[0006] In contrast, in a low-pressure refrigerant pipe and the refrigerant return pipe, which connect the accumulator and the refrigerating cycle to each other, since the accumulator has a volume 30 to 100 times that of the suction chamber of the compressing section, the fluctuations in flow velocity are reduced significantly. Therefore, even if the average flow velocities in the refrigerant return pipe and the refrigerant suction pipe are equal, since the pressure loss is substantially proportional to the square of flow velocity, the pressure loss increases with the increase in fluctuations in flow velocity, and resultantly the compression efficiency is decreased.

[0007] The present invention has been made to solve the above problems, and accordingly an object thereof is to provide a rotary compressor in which the pressure loss of a refrigerant suction pipe connecting the compressor and an accumulator to each other is reduced, thereby increasing the compression efficiency.

SUMMARY OF THE INVENTION

[0008] To achieve the above object, the present invention has some features described below. A rotary compressor having a compressor body including a motor and a rotary compressing section in a closed shell, and an accumulator at the side of the compressor body, in which a refrigerant return pipe connected to a refrigerating cycle is connected to the upper part of the accumulator, and a refrigerant suction pipe connected to the compressing section is connected to the lower part thereof, is characterized in that when the inside diameter of the refrigerant return pipe is taken as D1, and the inside diameter of the refrigerant suction pipe is taken as D2, D2 is larger than D1.

[0009] According to this configuration, by making the inside diameter D2 of the refrigerant suction pipe on the outlet side of the accumulator larger than the inside diameter D 1 of the refrigerant return pipe on the inlet side thereof (D2 > D1), the fluctuations in flow velocity in the refrigerant suction pipe connecting the compressor body to the accumulator are restrained and therefore the pressure loss is reduced, by which the compression efficiency of compressor can be increased.

[0010] As a preferable mode, the rotary compressor is characterized in that the refrigerant suction pipe is arranged so as to be offset in the direction such as to be separated from the compressor body with respect to the center axis of the accumulator.

[0011] If the inside diameter of the refrigerant suction pipe is increased, the bending radius must be increased in terms of the compressive strength and working properties of pipe. Accordingly, by arranging the refrigerant suction pipe so as to be offset in the direction such as to be separated from the compressor body with respect to the center axis of the accumulator, even a pipe having a large pipe diameter can be attached to the accumulator unforcedly.

[0012] Also, it is preferable that the rotational speed of the compressing section be variable. According to this configuration, in the case where the compressor is operated at a rapid rotation by using an inverter system in which the rotational speed of the motor is variable, the flow velocity in the refrigerant suction pipe increases, and the pressure loss increases. Therefore, the operation effect further increases.

[0013] Further, the rotary compressor is characterized in that the rotary compressing section is provided with a low-stage compressing section and a high-stage compressing section, and a means for allowing the discharge side of the low-stage compressing section and the suction side of the high-stage compressing section to communicate with each other is provided, whereby a two-stage compressing section is formed.

[0014] According to this configuration, due to the use of the two-stage compressing section in which two compression chambers are connected to each other in series in such a manner that the compression phases thereof are shifted through 180 degrees, and are formed by the low-stage compressing section and the high-stage compressing section, the balance of compressive torques and the balance of centrifugal forces of the off-center parts are good, so that the compressor can be operated at a more rapid rotation, and therefore the flow velocity is increased. Therefore, the operation effect further increases.

[0015] The present invention also embraces a heat pump system including the above-described rotary compressor. The heat pump system having a refrigerating cycle including a compressor, a condenser, an expansion mechanism, and an evaporator and including a low-pressure refrigerant pipe for connecting an accumulator for the compressor to the evaporator is characterized in that the compressor described above is used as the compressor for the heat pump system, and taking the inside diameter of the low-pressure refrigerant pipe as D0, D2 is larger than D0.

BRIEF DESCRIPTION OF THE DRAWINGS

[0016]

FIG. 1 is a longitudinal sectional view of a rotary compressor in accordance with one embodiment of the present invention;
FIG. 2 is a transverse sectional view of a compressing section of the rotary compressor shown in FIG. 1;
FIG. 3 is a graph showing a change in suction flow velocity with respect to the rotation angle during one turn of the rotary compressor shown in FIG. 1;
FIG. 4 is a graph showing a change in suction pressure with respect to the rotation angle during one turn of the rotary compressor shown in FIG. 1;
FIG. 5 is a graph showing a change in suction pressure with respect to the suction volume during one turn of the rotary compressor shown in FIG. 1; and
FIG. 6 is a perspective view of the rotary compressor shown in FIG. 1 as viewed from the above.

DETAILED DESCRIPTION

[0017] A rotary compressor 1 includes a cylindrical closed vessel 2 arranged in the vertical direction, a motor 4 provided in an upper part in the closed vessel 2, and a compressing section 3 in a lower part therein.

[0018] The closed vessel 2 consists of a cylindrical main shell 21, a dome-shaped top shell 22 that closes the upper end part of the main shell 21, and a dome-shaped bottom shell 23 that closes the lower end part of the main shell 21. The top shell 22 and the bottom shell 23 are fixed to the main shell 21 by welding.

[0019] The top shell 22 is provided with a refrigerant discharge pipe 24 for discharging the refrigerant having been discharged into the closed vessel 2 from the compressing section 3 to the outside of the closed vessel 2.

[0020] A stator 41 of the motor 4 is shrinkage fitted to the main shell 21, and a rotor 42 of the motor 4 is shrinkage fitted onto a shaft 7 mechanically connecting the motor 4 to the compressing section 3. Above and below the rotor 42, an upper balancer 43 and a lower balancer 44 are attached, respectively, to balance the centrifugal forces of the whole of rotating parts.

[0021] The compressing section 3 is provided with a high-stage compressing section 32 in the upper part thereof and a low-stage compressing section 31 in the lower part thereof. The discharge side of the low-stage compressing section 31 and the suction side of the high-stage compressing section 32 are connected to each other by an intermediate connection pipe 26 on the outside of the closed vessel 2, by which what is called a two-stage compressing section is formed.

[0022] Next, the configuration of each of the compressing sections 31 and 32 is explained with reference to FIG. 2. FIG. 2 shows the transverse cross section of the low-stage compressing section 31 shown in FIG. 1. The configuration of the high-stage compressing section 32 is the same as that of the low-stage compressing section 31 except that the pistons are 180° out-of-phase.

[0023] Each of the compressing sections 31 and 32 has a cylinder 200, 400 and a cylindrical piston 220, 420 accommodated in a cylindrical cylinder bore 200a, 400a formed on the inside of the cylinder 200, 400. Between the internal wall of the cylinder bore 200a, 400a and the outer peripheral surface of the piston 220, 420, a working space for refrigerant is formed.

[0024] The cylinder 200, 400 is provided with a cylinder groove 200b, 400b directed from the cylinder bore 200a, 400a toward the outer periphery direction. In the cylinder groove 200b, 400b, a flat plate shaped vane 230, 430 is provided.

[0025] Between the vane 230, 430 and the internal wall of the closed vessel 2, a spring 240, 440 is provided. By the urging force of the spring 240, 440, the tip end of the vane 230, 430 is brought into sliding contact with the outer wall of the piston 220, 420, by which the working space is divided into a suction chamber V1, V2 and a compression chamber C1, C2.

[0026] Next, referring again to FIG. 1, the whole of the compressor 1 is explained. The compressor 1 has a main frame 100 above the high-stage side cylinder 200, an intermediate partition plate 300 between the high-stage side cylinder 200 and the low-stage side cylinder 400, and a sub-frame 500 below the low-stage side cylinder 400, and the upside and the downside of each of the two working spaces are closed by the main frame 100, the intermediate partition plate 300, and the sub-frame 500, whereby each of the two working spaces is formed into a closed space.

[0027] Above the main frame 100 and below the sub-frame 500, a high-stage side discharge muffler cover 130 and a low-stage side discharge muffler cover 510 are provided, respectively, and a high-stage side discharge muffler chamber M2 and a low-stage side discharge muffler chamber M1 are formed to reduce the pressure pulsation of discharged refrigerant.

[0028] The high-stage side discharge muffler cover 130, the main frame 100, the high-stage side cylinder 200, the intermediate partition plate 300, the low-stage side cylinder 400, the sub-frame 500, and the low-stage side discharge muffler cover 510 are fixed integrally with bolts (not shown), and further the outer peripheral part of the main frame 100 is fixed to the closed vessel 2 by spot welding.

[0029] The main frame 100 and the sub-frame 500 have bearing parts 110 and 502, respectively, so that the shaft 7 is fitted in the bearing parts 110 and 502 so as to be rotatably supported.

[0030] The shaft 7 has two crankshafts 72 and 73 that are off-centered in the 180 ° different direction. One crankshaft 72 engages with the piston 220 of the high-stage compressing section 32, and the other crankshaft 73 engages with the piston 420 of the low-stage compressing section 31.

[0031] Along with the rotation of the shaft 7, the pistons 220 and 420 turn while slidingly contacting with the inside walls of the respective cylinder bores 200a and 400a, and following this turning motion of the pistons 220 and 420, the vanes 230 and 430 reciprocate, by which the volumes of the suction chambers V1 and V2 and the compression chambers C1 and C2 are changed continuously. Thus, the compressing section 3 repeats the suction and compression of refrigerant.

[0032] The suction chamber V1 of the low-stage compressing section 31 is connected to a refrigerant suction pipe 64 via a low-stage side suction hole 410 provided in the cylinder 400. The compression chamber C1 of the low-stage compressing section 31 is connected to the intermediate connection pipe 26 via a low-stage side discharge hole 520 provided in the sub-frame 500 and the low-stage side discharge muffler chamber M1.

[0033] More specifically, the low-stage side discharge hole 520 is provided with a check valve 540. Also, the refrigerant suction pipe 64 is connected to the low-stage side suction hole 410 via a low-stage side suction connection pipe 411, and the intermediate connection pipe 26 is connected to the low-stage side discharge muffler chamber M1 via an intermediate discharge connection pipe 521.

[0034] The suction chamber V2 of the high-stage compressing section 32 is connected to the intermediate connection pipe 26 via a high-stage side suction hole 210 provided in the cylinder 200. The compression chamber C2 of the high-stage compressing section 32 is open to the interior of the closed vessel 2 via a high-stage side discharge hole 120 provided in the main frame 100 and the high-stage side discharge muffler chamber M2.

[0035] More specifically, the high-stage side discharge hole 120 is provided with a check valve 140. The intermediate connection pipe 26 is connected to the high-stage side suction hole 210 via an intermediate suction connection pipe 211.

[0036] At the side of the body of the compressor 1, an accumulator 6 consisting of an independent closed vessel 61 is provided. Above the accumulator 6, a refrigerant return pipe 62 is provided, the refrigerant return pipe 62 being connected to a heat pump system, not shown. Below the accumulator 6, there is provided the refrigerant suction pipe 64 one end having an L shape of which is extended to the upper part in the accumulator 6 and the other end of which is connected to the suction chamber V1 of the low-stage compressing section 31 from the side surface of the compressor 1.

[0037] The refrigerant suction pipe 64 is arranged so as to be offset to the opposite side of the compressor body with respect to the center axis of the closed vessel 61 of the accumulator 6. The inside diameter D2 of the refrigerant suction pipe 64 is larger than the inside diameter D1 of the refrigerant return pipe 62 ($D2 > D1$, preferably $D2 \geqq 1.2 \times D1$). The configuration in which the refrigerant suction pipe 64 is offset to the opposite side of the compressor body is not limited to the configuration in which the refrigerant suction pipe 64 is offset on the imaginary line connecting the center of compressor body 1 to the center of the accumulator 6. As shown in FIG. 6, the refrigerant suction pipe 64 may be offset to any position farther from the compressor body 1 than the center of the accumulator 6.

[0038] Next, the flow of refrigerant in the above-described configuration is explained with reference to FIGS. 1 and 2. The refrigerant flows from the heat pump system side into the accumulator 6 passing through the refrigerant return pipe 62. In the accumulator 6, the refrigerant is separated so that a liquid refrigerant lies in the lower part of the accumulator 6 and a gas refrigerant lies in the upper part thereof.

[0039] When the low-stage side piston 420 is turned to increase the volume of the low-stage side suction chamber V1, the gas refrigerant in the accumulator 6 is sucked into the low-stage side suction chamber V1 of the compressor body 1 through the refrigerant suction pipe 64.

[0040] After one turn of the piston 420, the low-stage side suction chamber V1 comes to a position isolated from the low-stage side suction hole 410, and is turned to the low-stage side compression chamber C1 as it is, by which the refrigerant is compressed.

[0041] When the pressure of the compressed refrigerant reaches the pressure in the low-stage side discharge muffler chamber M1 on the outside of the check valve 540 provided in the low-stage side discharge hole 520, that is, an intermediate pressure, the check valve 540 is opened, by which the compressed refrigerant is discharged into the low-stage side discharge muffler chamber M1.

[0042] After the pressure pulsation of refrigerant, which may cause noise, has been reduced in the low-stage side discharge muffler chamber M1, the refrigerant is guided into the suction chamber V2 of the high-stage compressing section 32 through the intermediate connection pipe 26.

[0043] The refrigerant guided into the suction chamber V2 of the high-stage compressing section 32 is sucked, compressed, and discharged in the high-stage compressing section 32 on the same principle as that of the low-stage compressing section 31. After the pressure pulsation of refrigerant has been reduced in the high-stage side discharge muffler chamber M2, the refrigerant is discharged into the closed vessel 2.

[0044] The refrigerant is further guided to a portion above the motor 4 after passing through a core notch (not shown) in the stator 41 of the motor 4 and a gap between a core and a coil, and is discharged to the system side through the discharge pipe 24.

[0045] In the above-described refrigerant flow, since the rate of change of the volume of the low-stage side suction chamber V1 during one turn is not constant, the flow velocity of refrigerant fluctuates in the refrigerant suction pipe 64. In contrast, in the refrigerant return pipe 62, since the accumulator 6 has a volume 30 to 100 times that of the low-stage side suction chamber V1, the fluctuations in flow velocity are reduced significantly.

[0046] Therefore, by making the inside diameter D2 of the refrigerant suction pipe 64 in which the flow velocity fluctuates greatly larger than the inside diameter D1 of the refrigerant return pipe 62 ($D2 > D1$), the flow velocity in the refrigerant suction pipe 64 is decreased, so that the pressure loss in the suction process decreases, thereby improving the compression efficiency of compressor.

[0047] The above-described effect is explained with reference to FIGS. 3, 4 and 5. FIG. 3 is a graph showing the suction flow velocity in the refrigerant suction pipe 64 with respect to the rotation angle of the piston. The rotation angle at the time when the piston comes to a position closest to the vane groove in the cylinder is set at 0 degree. Also, the suction flow velocity in the case (1) where the inside diameter D2 of the refrigerant suction pipe 64 is made equal to the inside diameter D 1 of the refrigerant return pipe 62 and the flow velocity does not change, that is, the average flow velocity in terms of time is set at 1. In contrast to case (1), the actual flow velocity (case (3) and case (4)) changes during one turn.

[0048] FIG. 4 is a graph showing the suction pressure at the time when the pressure loss is assumed to be proportional to the square of the suction flow velocity, corresponding to FIG. 3 showing the suction flow velocity. The suction pressure in the case where the pressure loss is completely absent is set at 1.0 MPa. The suction pressure is changed during one turn by the change in suction flow velocity as shown in the graph. In contrast to the case (3) where the inside diameter D2 of the refrigerant suction pipe 64 is equal to the inside diameter D1 of the refrigerant return pipe 62, in the case (4) where the inside diameter D2 of the refrigerant suction pipe 64 is 1.2 times the inside diameter D1 of the refrigerant return pipe 62, the change width of pressure is reduced to about one-half.

[0049] FIG. 5 is a graph in which the rotational angle on the abscissa of FIG. 4 is changed to the volume of suction chamber. By choosing the volume as the abscissa, the loss in the suction process, that is, the increase in power consumption of compressor is represented as the area of a portion surrounded by the straight line of suction pressure 1.0 MPa and a curve.

[0050] As shown in this graph, in contrast to the case (3) where the inside diameter D2 of the refrigerant suction pipe 64 is equal to the inside diameter D1 of the refrigerant return pipe 62, in the case (4) where the inside diameter D2 of the refrigerant suction pipe 64 is 1.2 times the inside diameter D1 of the refrigerant return pipe 62, the area representing the loss is reduced to about one-half. This area corresponds to the area in the case (1) where the inside diameter D2 of the refrigerant suction pipe 64 is made equal to the inside diameter D1 of the refrigerant return pipe 62 and the flow velocity does not change.

[0051] From the above description, it can be seen that by making the inside diameter D2 of the refrigerant suction pipe 64 larger than the inside diameter D1 of the refrigerant return pipe 62, the power consumption can be reduced, and that by making D2 equal to or larger than 1.2 times D1, the power consumption can be reduced to a level at which the flow velocity does not fluctuate in the refrigerant suction pipe 64.

[0052] Also, since the refrigerant suction pipe 64 is fixed so as to be offset to the opposite side of the compressor body 1 with respect to the closed vessel 61 of the accumulator 6, even in the case where the inside diameter of the refrigerant suction pipe 64 is larger than in the conventional example, the accumulator 6 can be arranged closer to the compressor body 1, so that a compact shape can be provided in mounting in the system.

[0053] In this embodiment, as the rotary compressor 1, the compressor provided with the two-stage compression type compressing section 3 having the low-stage compressing section 31 and the high-stage compressing section 32 has been shown typically as a preferred mode. However, the rotary compressor 1 may be a two-stage compression rotary compressor configured so that a gas injection cycle is used as the refrigerating cycle, and an injection refrigerant is allowed to flow into an intermediate compressing section between the low-stage compressing section 31 and the high-stage compressing section 32.

[0054] Also, the present invention may be applied to a single-stage compression rotary compressor having one compression chamber. Also, the compressing mechanism of the compressing section 3 is not limited to the compressing mechanism shown in this embodiment if the compressor is configured so that the change in volumes of the suction chamber V1, V2 and the compression chamber C1, C2 caused by the turning motion of the piston 220, 420 imparted by the crankshaft 72, 73 is utilized.

Claims

1. A rotary compressor comprising a compressor body including a motor and a rotary compressing section in a closed shell, and an accumulator at the side of the compressor body, in which a refrigerant return pipe connected to a refrigerating cycle is connected to the upper part of the accumulator, and a refrigerant suction pipe connected to the compressing section is connected to the lower part of the accumulator, wherein
   when the inside diameter of the refrigerant return pipe is taken as D1, and the inside diameter of the refrigerant suction pipe is taken as D2, D2 is larger than D1.

2. The rotary compressor according to claim 1, wherein the refrigerant suction pipe is arranged so as to be offset in the direction such as to be separated from the compressor body with respect to the center axis of the accumulator.

3. The rotary compressor according to claim 1 or 2, wherein the rotational speed of the compressing section is variable.

4. The rotary compressor according to claim 1 or 2, wherein the rotary compressing section is provided with a low-stage compressing section and a high-stage compressing section, and a means for allowing the discharge side of the low-stage compressing section and the suction side of the high-stage compressing section to communicate with each other is provided, whereby a two-stage compressing section is formed.

5. A heat pump system having a refrigerating cycle including a compressor, a condenser, an expansion mechanism, and an evaporator and comprising a low-pressure refrigerant pipe for connecting an accumulator for the compressor to the evaporator, wherein
   the compressor described in any one of claims 1 to 4 is used as the compressor, and taking the inside diameter of the low-pressure refrigerant pipe as D0, D2 is larger than D0.

FIG. 1

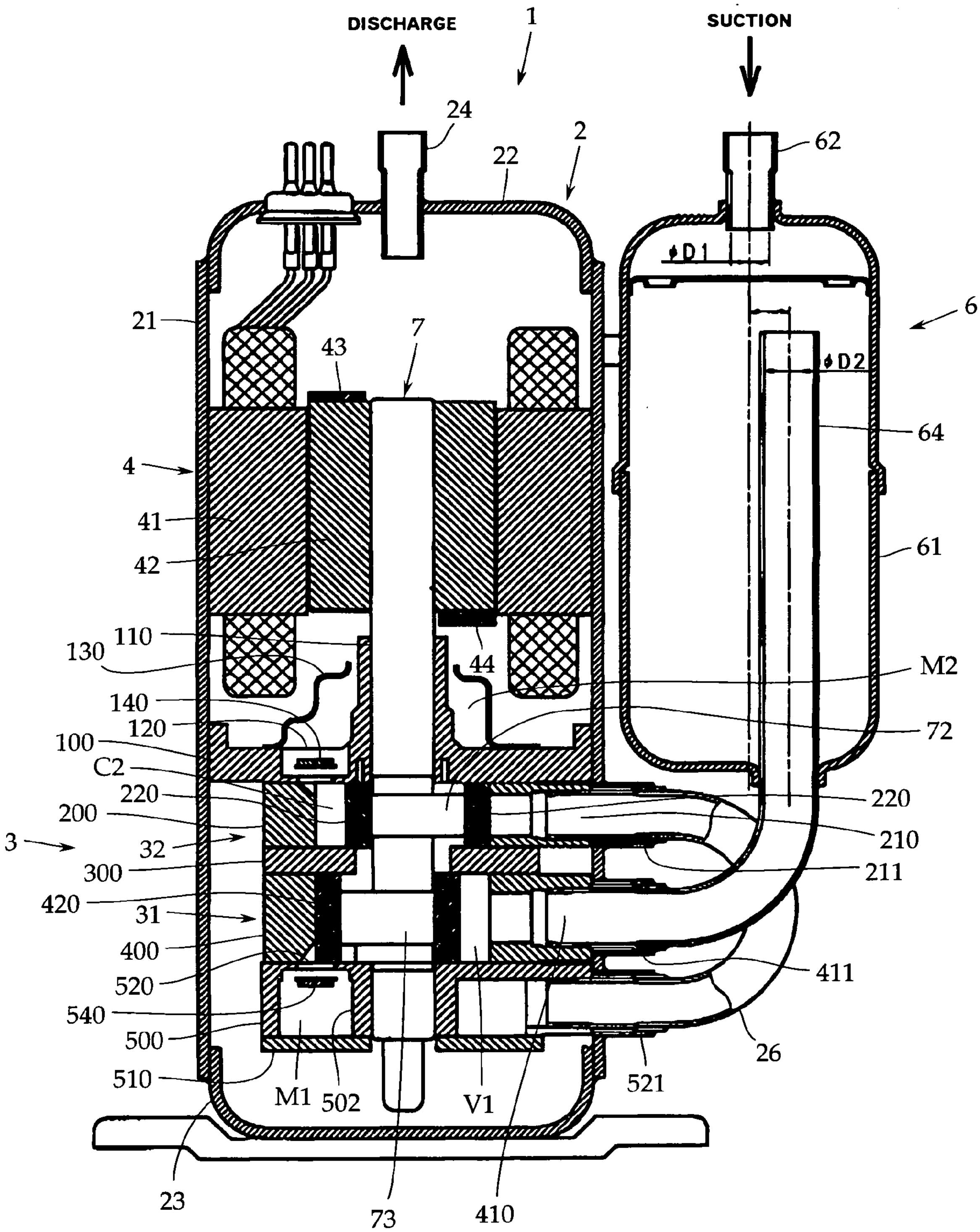

DISCHARGE
SUCTION
1
2
4
6
7
21
22
23
24
26
3
31
32
41
42
43
44
61
62
64
72
73
ϕD1
ϕD2
100
110
120
130
140
C2
200
210
211
220
300
400
410
411
420
500
502
510
520
521
540
M1
M2
V1

FIG. 2

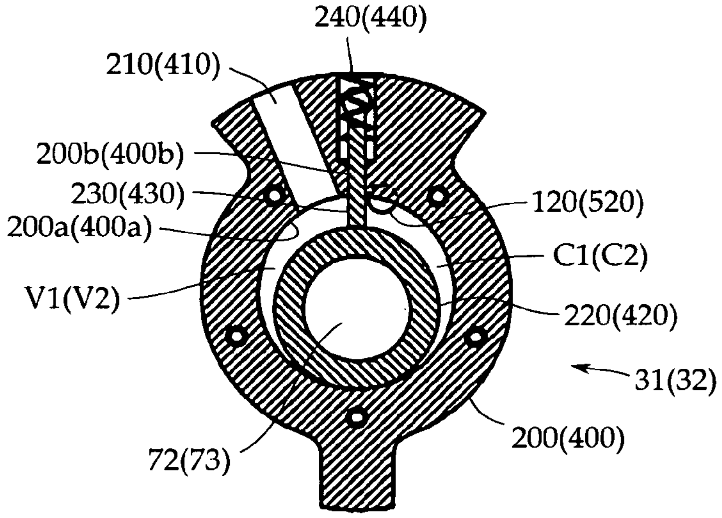

240(440)
210(410)
200b(400b)
230(430)
200a(400a)
120(520)
C1(C2)
V1(V2)
220(420)
31(32)
72(73)
200(400)

FIG. 3

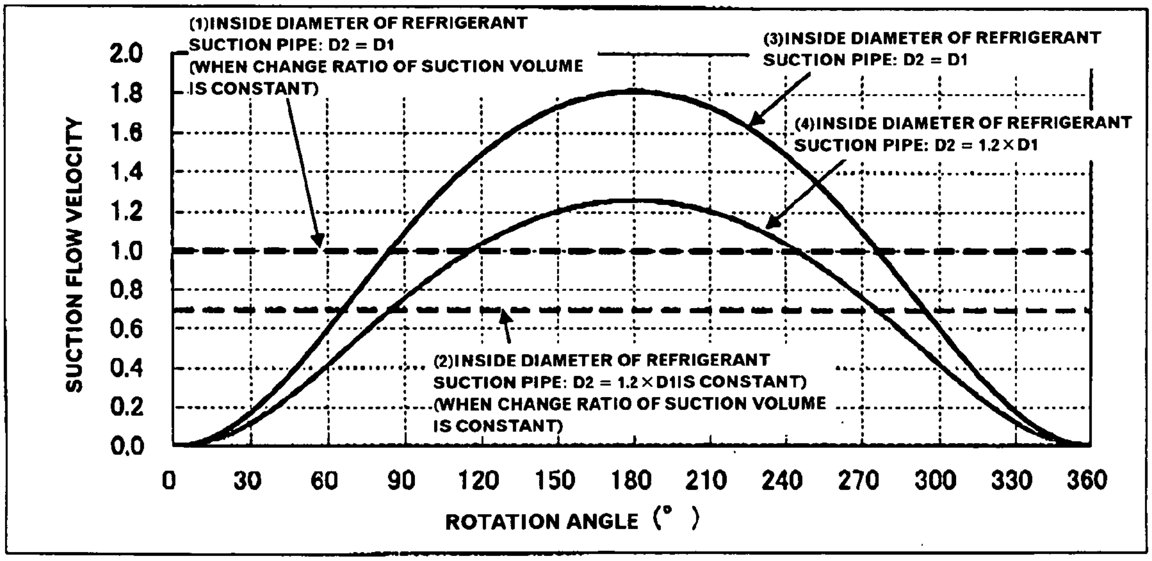

(1)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = D1
(WHEN CHANGE RATIO OF SUCTION VOLUME
IS CONSTANT)
(3)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = D1
(4)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = 1.2×D1
(2)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = 1.2×D1IS CONSTANT)
(WHEN CHANGE RATIO OF SUCTION VOLUME
IS CONSTANT)
SUCTION FLOW VELOCITY
2.0
1.8
1.6
1,4
1.2
1.0
0,8
0.6
0.4
0.2
0,0
0
30
60
90
120
150
180
210
240
270
300
330
360
ROTATION ANGLE (°)

FIG. 4

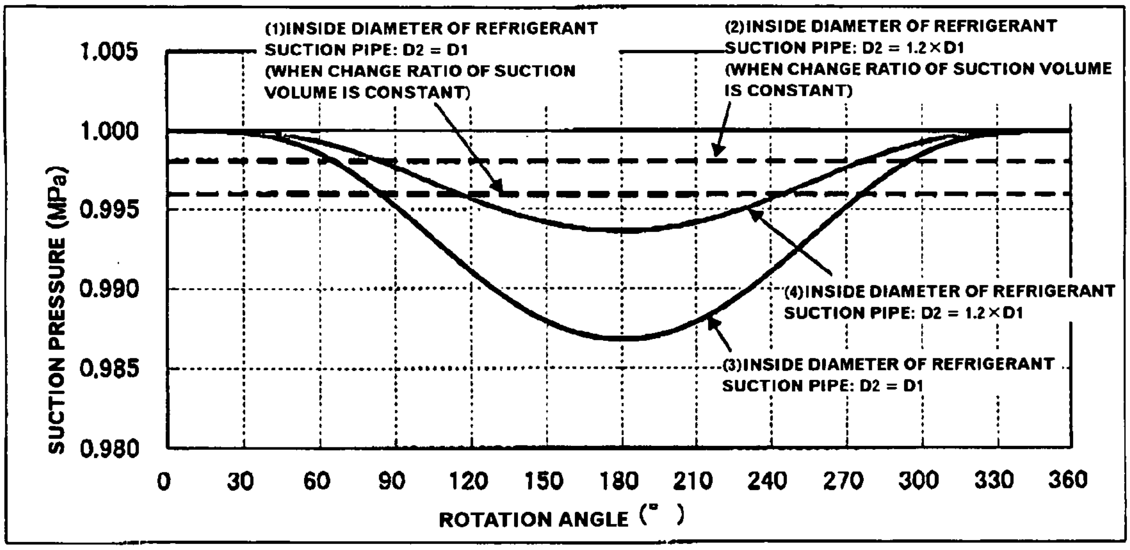

(1)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = D1
(WHEN CHANGE RATIO OF SUCTION
VOLUME IS CONSTANT)
(2)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = 1.2×D1
(WHEN CHANGE RATIO OF SUCTION VOLUME
IS CONSTANT)
(4)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = 1.2×D1
(3)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = D1
1.005
1.000
0.995
0.990
0.985
0.980
SUCTION PRESSURE (MPa)
0
30
60
90
120
150
180
210
240
270
300
330
360
ROTATION ANGLE (°)

FIG. 5

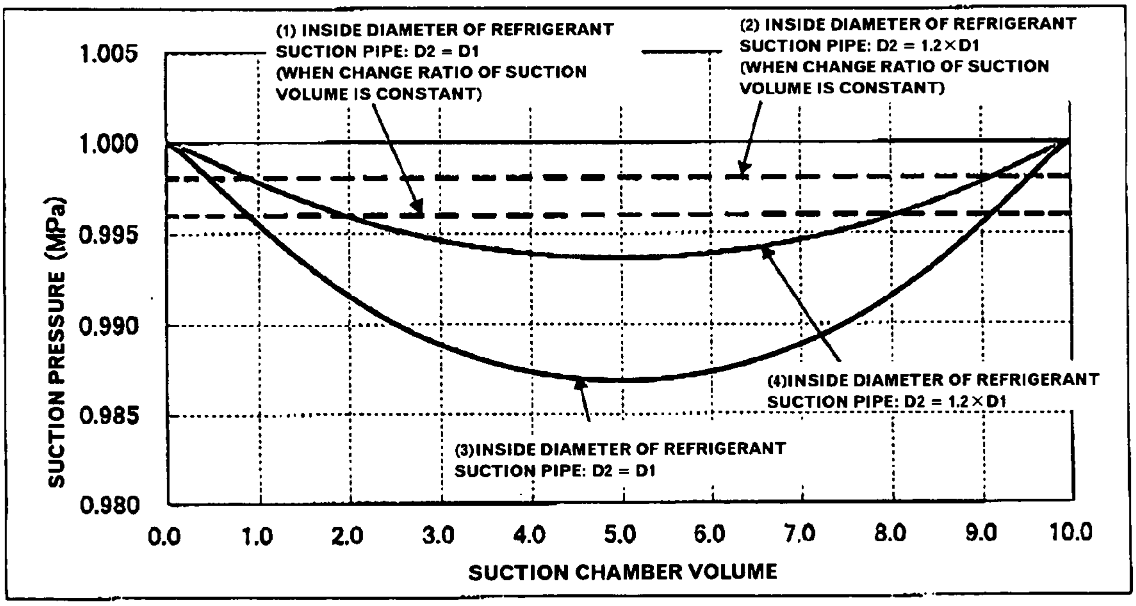
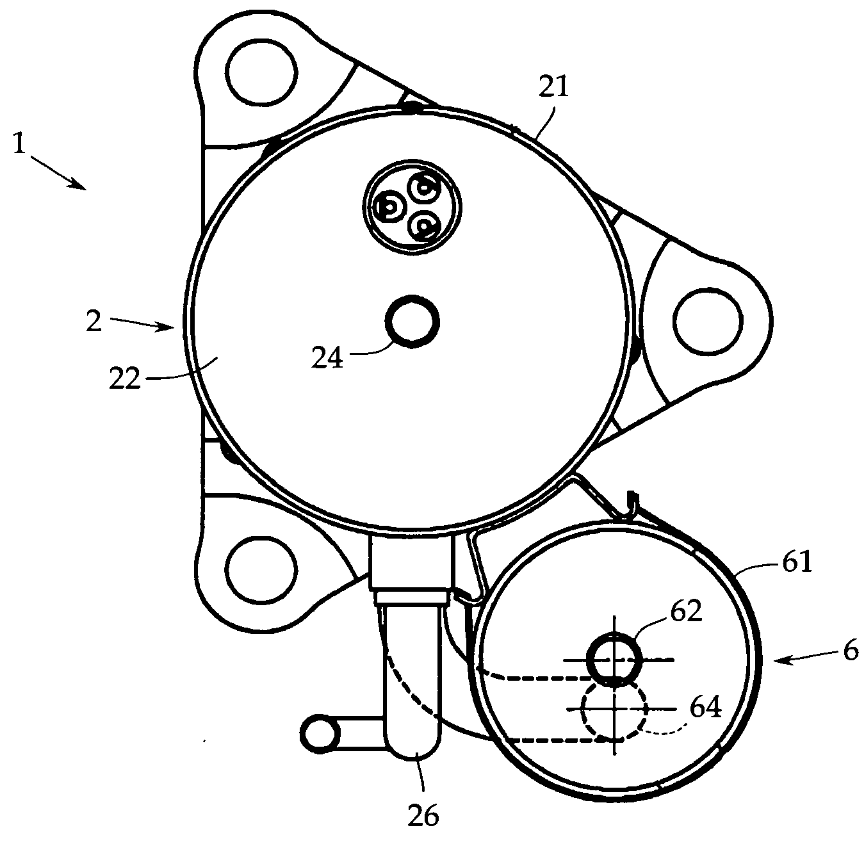

(1) INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = D1
(WHEN CHANGE RATIO OF SUCTION
VOLUME IS CONSTANT)
(2) INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = 1.2×D1
(WHEN CHANGE RATIO OF SUCTION
VOLUME IS CONSTANT)
(4)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = 1.2×D1
(3)INSIDE DIAMETER OF REFRIGERANT
SUCTION PIPE: D2 = D1
1.005
1.000
0.995
0.990
0.985
0.980
SUCTION PRESSURE (MPa)
0.0
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
10.0
SUCTION CHAMBER VOLUME

FIG. 6



European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 08 25 0948

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
X	KR 2007 0009958 A (HITACHI HOME & LIFE SOLUTIONS [JP]) 19 January 2007 (2007-01-19) * figures 1-3,6 *	1-5	INV. F01C21/10 F04C18/356 F04C23/00 F25B43/00 F04C28/08
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