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(54) **Scroll compressor**

(57) Refrigerant gas undergoing compression is ex-
tracted from the compression chamber (1d) and intro-
duced into a closed frame space (15f) formed by dis-
posing two sealing materials (16c, 16d) respectively on
two cylindrical (or flat) surfaces formed by a compliant

frame (3), supporting the oscillating scroll (2), and a
guide frame (15) supporting the compliant frame (3).
The pressure in the frame space (15f) is 1.2 to 2 times
the suction pressure of the compressor.

Description

[0001] The present invention relates to a refrigerant compressor for use in refrigerating/air-conditioning equipment.

Background Art

[0002] Fig. 6 is a longitudinal sectional view showing the construction of a conventional scroll compressor disclosed in JP-A-2000-161254.

[0003] In Fig. 6, numeral 1 designates a fixed scroll having its outer circumferential portion fastened to a guide frame 15 by means of bolts (not shown). Plate-like scroll teeth 1b are formed on one surface (lower side in Fig. 6) of a base plate portion 1a. In addition, two Oldham's guide grooves 1c are formed substantially in a straight line in the outer circumferential portion. A claw 9c of an Oldham's ring 9 is reciprocally slidably engaged with each of the Oldham's guide grooves 1c. Further, from a side surface of the fixed scroll 1, a suction pipe 10a is press fitted through a closed vessel 10.

[0004] Numeral 2 designates an oscillating scroll, and plate-like scroll teeth 2b having substantially the same shape as the plate-like scroll teeth 1b of the fixed scroll 1 are provided on the upper surface of a base plate portion 2a. Thus, a compression chamber 1d is formed geometrically. A hollow cylindrical boss portion 2f is formed in the center portion of that surface of the base plate portion 2a which is opposite to the plate-like scroll teeth 2b. An oscillating bearing 2c is formed on the inner surface of the boss portion 2f. In addition, a thrust surface 2d which can slide in pressure contact with a thrust bearing 3a of a compliant frame 3, is formed on the same side surface as the boss portion 2f but on an outer side than the boss portion 2f. In the outer circumferential portion of the oscillating scroll base plate portion 2a, two Oldham's guide grooves 2e are formed substantially in a straight line to have a phase difference of 90 degrees with respect to the Oldham's guide grooves 1c of the fixed scroll 1. A claw 9a of the Oldham's ring 9 is reciprocally slidably engaged with each of the Oldham's guide grooves 2e. An extraction hole 2j is also provided in the base plate portion 2a so as to extend from the compression chamber 1d through the thrust surface 2d. An aperture portion 2k of the extraction hole 2j on the side of the thrust surface 2d is located so that the circular locus of the aperture portion 2k always stays inside the thrust bearing surface 3a of the compliant frame 3.

[0005] The compliant frame 3 has two upper and lower cylindrical surfaces 3d and 3e in its outer circumferential portion. The cylindrical surfaces 3d and 3e are supported in the radial direction of the scroll compressor by cylindrical surfaces 15a and 15b provided in the inner circumferential portion of the guide frame 15, respectively. A main bearing 3c and an auxiliary main bearing 3h for supporting a main shaft 4 in the radial direction of the scroll compressor are formed in the center por-

tions of the compliant frame 3. The main shaft 4 is driven to rotate by a motor 7. In addition, between the outside of the compliant frame 3 and the inside of the guide frame 15, a frame space 15f is defined by sealing materials 16a and 16b disposed on cylindrical surfaces 15c and 15d, respectively. The frame space 15f communicates with the compression chamber 1d through a communication passageway 3s and the extraction hole 2i which are interconnected via the surface of the thrust bearing 3a. Thus, the frame space 15f is filled with refrigerant gas which is supplied from the compression chamber 1d and which is on the way of compression.

[0006] A regulating valve receiving space 3p is also formed in the compliant frame 3. One end (lower end in Fig. 6) of the regulating valve receiving space 3p communicates with a boss portion outside space 2h. The boss portion outside space 2h is constituted by the inner circumference of the compliant frame 3 and the thrust surface 2d of the oscillating scroll 2. On the other hand, the other end (upper end in Fig. 6) of the regulating valve receiving space 3p is made open to a suction pressure atmosphere space 1g. An intermediate pressure regulating valve 3i is reciprocally movably received in the lower portion of the regulating valve receiving space 3p. On the other hand, received in the upper portion of the regulating valve receiving space 3p is an intermediate pressure regulating spring retainer 3t fixedly attached to the compliant frame 3. Between the intermediate pressure regulating valve 3i and the intermediate pressure regulating spring retainer 3t, an intermediate pressure regulating spring 3m is received in such a manner that the spring 3m is made shorter than its natural length.

[0007] An outer circumferential surface 15g of the guide frame 15 is fixedly attached to the closed vessel 10 by shrink-fitting, welding, or the like. However, a channel is ensured by a notch portion 15c provided in the outer circumferential portion of the guide frame 15. Thus, high pressure refrigerant gas discharged from a discharge port 1f of the fixed scroll 1 is directed through the channel to a discharge pipe 10b provided on the motor side.

[0008] Numeral 4 designates a main shaft and an oscillating shaft 4b is formed in the upper end portion of the main shaft 4. The oscillating shaft 4b is rotatably engaged with the oscillating bearing 2c of the oscillating scroll 2. A main shaft balancer 4e is shrink-fitted in the lower portion of the oscillating shaft 4b. Further, under the main shaft balancer 4e, a main shaft portion 4c is formed so as to be rotatably engaged with the main bearing 3c and the auxiliary main bearing 3h of the compliant frame 3. In addition, an auxiliary shaft portion 4d is formed in the lower portion of the main shaft 4 so as to be rotatably engaged with an auxiliary bearing 6a of a sub-frame 6. A rotor 8 is shrink-fitted between the auxiliary shaft portion 4d and the main shaft portion 4c.

[0009] An upper balancer 8a is fixed to the upper end surface of the rotor 8 and a lower balancer 8b is fixed to the lower end surface of the rotor 8. Static balance and

dynamic balance are ensured by the total of three balancers including the upper and lower balancers 8a and 8b in addition to the above-mentioned main shaft balancer 4e. Further, an oil pipe 4f is force fitted into the lower end of the main shaft 4. Thus, refrigerating machine oil 10e retained in the bottom portion of the closed vessel 10 is sucked up through the oil pipe 4f.

[0010] A glass terminal board 10f is provided at the side surface of the closed vessel 10. The motor 7 is connected with the glass terminal board 10f through lead wires.

[0011] Next, description will be made about the basic operation of the conventional scroll compressor.

[0012] A sucked refrigerant of low pressure enters the compression chamber 1d through the suction pipe 10a. The compression chamber 1d is defined by the plate-like scroll teeth of the fixed scroll 1 and the plate-like scroll teeth of the oscillating scroll 2. The oscillating scroll 2 driven by the motor 7 makes an eccentric turning motion while reducing the volume of the compression chamber 1d. On this compression stroke, the sucked refrigerant becomes high in pressure. Thus, the sucked refrigerant is discharged into the closed vessel 10 through the discharge port 1f of the fixed scroll 1.

[0013] on the other hand, the refrigerant gas of intermediate pressure on the way of compression on the above-mentioned compression stroke is directed from the extraction hole 2j of the oscillating scroll 2 to the frame space 15f through the communication passageway 3s of the compliant frame 3, so that the intermediate pressure atmosphere in this space is maintained.

[0014] The discharged gas of the high pressure fills the closed vessel 10 with the high pressure atmosphere. The discharged gas is eventually released from the discharge pipe 10b to the outside of the compressor.

[0015] The refrigerating machine oil 10e in the bottom portion of the closed vessel 10 is directed, by a differential pressure, to the oscillating bearing 2g through a hollow space 4g extending through the main shaft 4 in the axial direction and to the main bearing 3c through a side hole provided in the main shaft 4. The refrigerating machine oil 10e (which is generally formed into a two-phase flow of gas refrigerant and refrigerating machine oil because of the foaming of the refrigerant dissolved in the refrigerating machine oil) is made to have an intermediate pressure by the throttling action of the two bearings. The refrigerating machine oil 10e reaches the boss portion outside space 2h surrounded by the oscillating scroll 2 and the compliant frame 3. Then, the refrigerating machine oil 10e overcomes the force loaded by the intermediate pressure regulating spring 3m disposed in the regulating valve receiving space 3p. Thus, the refrigerating machine oil 10e pushes the intermediate pressure regulating valve 3i. Accordingly, the refrigerating machine oil 10e is introduced into the suction pressure atmosphere space 1g and sucked into the compression chamber 1d together with the low pressure refrigerant gas.

[0016] As described above, the intermediate pressure Pm1 (MPa) of the boss portion outside space 2h is substantially defined on the basis of the spring force of the intermediate pressure regulating spring 3m and the intermediate pressure exposure area of the intermediate pressure regulating valve 3i. Thus, the intermediate pressure Pm1 is controlled by a predetermined value α as follows:

$$P_{m1} = P_s + \alpha \quad (1)$$

wherein Ps represents the suction pressure or low pressure (MPa).

[0017] Here, the difference between the closed vessel pressure Pd (MPa) (i.e. the discharge pressure) and the boss portion outside space pressure Pm1 is an oil feed differential pressure ΔP required for feeding the refrigerating machine oil 10e to the main bearing 3c and the oscillating bearing 2g. It is necessary to always ensure a positive value for the oil feed differential pressure ΔP .

$$\Delta P = P_d - P_{m1} > 0 \quad (2)$$

[0018] On the compression stroke, the refrigerating machine oil 10e is released from the discharge port 1f into the closed vessel 10 together with the high pressure refrigerant gas. Here, the refrigerating machine oil 10e is separated from the refrigerant gas, and returned to the bottom portion of the closed vessel again.

[0019] The compression chamber 1d for the refrigerant gas always or intermittently communicates with the frame space 15f through the extraction hole 2j provided in the base plate portion 2a of the oscillating scroll 2 and the communication passageway 3s provided in the compliant frame 3. Since the frame space 15f is a space closed by the two sealing materials 16a and 16b, the pressure in the frame space 15f breathes and changes in response to the change in pressure of the compression chamber 1d. The pressure in the frame space 15f is roughly equal to the integrated average value of the pressure changes in the compression chamber 1d with which the extraction hole 2j communicates.

[0020] As described above, the intermediate pressure Pm2 (MPa) of the frame space 15f is controlled by a predetermined magnification value β determined by the position of the compression chamber 1d with which the extraction hole 2j communicates, as follows.

$$P_{m2} = P_s \times \beta \quad (3)$$

wherein Ps represents the suction pressure or low pressure (MPa).

[0021] Here, Fpm1 represents the force tending to cause the compliant frame 3 and the oscillating scroll 2 to separate from each other due to the intermediate

pressure Pm1 in the boss portion outside space 2h. In addition, Fgth represents the thrust gas force tending to cause the fixed scroll 1 and the oscillating scroll 2 to separate from each other in the axis direction due to the compression operation. Thus, the sum of the two forces Fpm1 and Fgth acts on the compliant frame 3 as a force for moving the compliant frame 3 in the opposite direction to the compression chamber 1d.

[0022] On the other hand, Fpm2 represents the force tending to cause the compliant frame 3 and the guide frame 15 to separate from each other due to the intermediate pressure Pm2 of the frame space 15f to which the refrigerant gas on the way of compression has been directed. In addition, Fpd2 represents the differential pressure which acts on the lower portion exposed to the high pressure atmosphere. Thus, the sum of the two forces Fpm2 and Fpd2 acts on the compliant frame 3 as a force to move the compliant frame 3 toward the compression chamber.

[0023] During the steady-state operation, the force to move the compliant frame 3 toward the compression chamber is set to exceed the force to move the compliant frame 3 in the opposite direction to the compression chamber. Thus, the compliant frame 3 is guided by the engaging upper and lower cylindrical surfaces 3d and 3e so as to move toward the compression chamber. The oscillating scroll 2 moves in the same direction as the compliant frame 3 while sliding on the compliant frame 3 in close contact therewith and also causing its plate-like scroll teeth 2b to slide in contact with the fixed scroll 1.

[0024] On the other hand, the above-mentioned thrust gas force Fgth increases during the starting, fluid compression, or the like. Thus, the oscillating scroll 2 strongly presses down the compliant frame 3 through the thrust bearing 3a. As a result, there is produced a comparatively large clearance between the tooth tips and the tooth bottoms of the oscillating scroll 2 and the fixed scroll 1. Thus, the pressure in the compression chamber is prevented from abnormally increasing. This action is called "relief action", and the amount of the produced clearance is called "relief amount".

[0025] The relief amount is controlled by a distance of travel by which the compliant frame 3 and the guide frame 15 collide with each other.

[0026] A part or the whole of upsetting moment generated in the oscillating scroll 2 is transmitted to the compliant frame 3 through the thrust bearing 3a. However, a bearing load applied by the main bearing 3c, and a resultant of two reactions thereof, that is, a couple produced by a resultant of counterforces applied by the two upper and lower cylindrical engaging surfaces 3d and 3e of the compliant frame 3 and the guide frame 15 act on the compliant frame 3 so as to cancel the above-mentioned upsetting moment. Thus, excellent steady-state operation follow-up action and relief action stability are ensured.

[0027] Next, detailed description will be made about

the relationship of axial forces acting on the conventional scroll compressor.

[0028] Fig. 7 illustrates the relationship of axial forces acting on the oscillating scroll 2 and the compliant frame 3 in the conventional scroll compressor.

[0029] Fgth represents the counterforce generated by compressing the refrigerant gas, and Ftip represents the tooth tip contact force generated by making the fixed scroll 1 and the oscillating scroll 2 slide in contact with each other at the tooth tips. Thus, the counterforce Fgth and the tooth tip contact force Ftip act on the oscillating scroll 2 in the downward direction in Fig. 7. On the other hand, Fpm1 represents the force tending to cause the oscillating scroll 2 and the compliant frame 3 to separate from each other by the pressure Pm1 in the boss portion outside space 2h. In addition, Fpd1 represents the force acting on the inside of the boss portion of the oscillating scroll exposed to the high pressure atmosphere due to the differential pressure. Further, Fth represents the thrust contact force generated by the thrust surface sliding in contact with the compliant frame 3. Thus, the forces Fpm1, Fpd1 and Fth act on the oscillating scroll 2 as upward forces in Fig. 7. Here:

$$F_{pm1} = S_{pm1} \times (P_{m1} - P_s) \quad (4)$$

$$F_{pd1} = S_{pd1} \times (P_d - P_s) \quad (5)$$

wherein:

Spm1 represents an acting area (m²) of the intermediate pressure Pm1 in the boss portion outside space

Spd1 represents an acting area (m²) of the discharge pressure Pd in the boss portion inside space
Pd represents the discharge pressure (MPa)

Ps represents the suction pressure (MPa).

[0030] Accordingly, the force acting on the oscillating scroll 2 is expressed by:

$$F_{gth} + F_{tip} = F_{th} + F_{pm1} + F_{pd1} \quad (6)$$

[0031] On the other hand, the force Fpm2 and the thrust contact force Fth act on the compliant frame 3 as downward forces in Fig. 7. The force Fpm2 is a force tending to cause the oscillating scroll 2 and the compliant frame 3 to separate from each other due to the intermediate pressure Pm1 of the boss portion outside space 15h. The thrust contact force Fth is generated when the compliant frame 3 slides in contact with the oscillating scroll 2. On the other hand, force Fpm2 and force Fpd2 act on the compliant frame 3 in the upward direction in Fig. 7. The force Fpm2 is a force tending to cause the compliant frame 3 and the guide frame 15 to

separate from each other due to the intermediate pressure P_{m2} of the frame space 15f. The force F_{pd2} is generated by the differential pressure acting on the lower end portion of the compliant frame exposed to the high pressure atmosphere.

$$F_{pm2} = S_{pm2} \times (P_{m2} - P_s) \quad (7)$$

$$F_{pd2} = S_{pd2} \times (P_d - P_s) \quad (8)$$

wherein:

S_{pm2} represents an active area (m^2) of the intermediate pressure P_{m2} in the frame space

S_{pd2} represents the area (m^2) in which the compliant frame is exposed to the discharge pressure atmosphere at its lower end

P_d represents the discharge pressure (MPa)

P_s represents the suction pressure (MPa).

[0032] Accordingly, the force acting on the compliant frame 3 is expressed by:

$$F_{pm1} + F_{th} = F_{pm2} + F_{pd2} \quad (9)$$

[0033] By the simultaneous equations (6) and (9), the tooth tip contact force F_{tip} and the thrust contact force F_{th} can be obtained.

$$F_{tip} = F_{pd1} + F_{pd2} + F_{pm2} - F_{gth} \quad (10)$$

$$F_{th} = F_{pm2} + F_{pd2} - F_{pm1} \quad (11)$$

[0034] The expression (10) shows that the tooth tip contact force F_{tip} increases as the force F_{pm2} (the force tending to cause the compliant frame 3 and the guide frame 15 to separate from each other due to the pressure P_{m2} of the frame space 15f) is set to be larger. In other words, the tooth tip contact force F_{tip} increases as the intermediate pressure P_{m2} of the frame space 15f is set to be higher (the value β is set to be larger).

[0035] On the other hand, the expression (11) shows that the thrust contact force F_{th} decreases as the force F_{pm1} (the force tending to cause the compliant frame 3 and the oscillating scroll 2 to separate from each other due to the pressure P_{m1} of the boss portion outside space 2h) is set to be larger. In other words, the thrust contact force F_{th} decreases as the intermediate pressure P_{m1} of the boss portion outside space 2h is set to be higher (the value α is set to be larger). That is, it is so constructed that the thrust sliding loss can be reduced so as to be useful in saving the electrical power supplied to the compressor.

[0036] As described above, the tooth tip contact force F_{tip} or the thrust contact force F_{th} can be adjusted desirably by adjusting the pressure P_{m1} in the boss portion outside space or the pressure P_{m2} in the frame space. However, positive values must be always ensured for the two forces in order that the compressor performs out a normal compressing operation.

$$F_{tip} > 0 \quad (12)$$

$$F_{th} > 0 \quad (13)$$

[0037] Referring now to Fig. 8, the sealing materials will be described hereinafter. The sealing materials are provided on the cylindrical engaging surfaces of the guide frame 15 and the compliant frame 3 so as to form the frame space 15f.

[0038] Since the refrigerant gas on the way of compression is extracted and introduced into the frame space 15f, the pressure levels during the normal operation are generally expressed by:

$$P_s < P_{m2} < P_d \quad (14)$$

[0039] Accordingly, the sealing materials usually constituted by a U-ring for preventing discharge pressure gas from entering the frame space 15f and another U-ring for preventing gas from leaking from the frame space 15f to the suction pressure atmosphere are provided in the direction shown in Fig. 8. Teflon or the like is often used as the material of the U-rings.

[0040] In the conventional scroll compressor, as described previously, when the intermediate pressure P_{m1} of the boss portion outside space 2h is set to be high, the thrust contact force F_{th} shown by the expression (11), that is, the thrust sliding loss can be reduced so that the electrical power supplied to the compressor can be saved. However, if the pressure P_{m1} is set to be too high, the thrust contact force F_{th} takes a negative value ($F_{th} < 0$). Accordingly, the oscillating scroll 2 and the compliant frame 3 are separated from each other so that a normal compressing operation cannot be carried out. In addition, the oscillating scroll 2 fluctuates in the clearance of the axial relief amount so that the oscillating bearing functions as one-sided bearing. Thus, there is a problem that abnormal wear, damage, or the like, is caused.

[0041] Likewise, if the pressure P_{m1} is set to be too large, the expression (2) $\Delta P = P_d - P_{m1}$ takes a negative value ($\Delta P = P_d - P_{m1} < 0$). Accordingly, the differential pressure for feeding oil to the oscillating bearing 2c and the main bearing 3c cannot be ensured, so that there is a problem that the bearings are damaged or the like.

[0042] In the conventional scroll compressor, if the intermediate pressure P_{m2} of the frame space 15f is set

to be too low, no force is generated to move the compliant frame 3 toward the compression chamber. As a result, the value of the tooth tip contact force F_{tip} becomes negative. Thus, during the steady-state operation, the fixed scroll 1 and the oscillating scroll 2 are separated from each other so that a normal compressing operation cannot be effected. In addition, there is such a problem that the oscillating scroll 2 fluctuates in the clearance of the axial relief amount so that the bearings are damaged. On the contrary, if the intermediate pressure P_{m2} is set to be too high, the tooth tip contact force F_{tip} becomes so large that the sliding loss increases. Thus, the electrical power supplied to the compressor increases. In addition, there is such a problem that the tooth tips are worn abnormally, and as the worst case, the tooth tips seize.

[0043] The present invention has been made in order to solve such problems, and it is an object of the present invention to provide a scroll compressor of the type described which has high performance and high reliability in that the value β in the expression (3) is set in a proper range with the result that the compliant frame 3 is moved toward the compression chamber positively so that the fixed scroll and the oscillating scroll are brought into close contact with each other by a proper pressing force in the axial direction of the compressor and thus the tooth tip contact force F_{tip} is maintained so proper that a normal compressing operation is ensured, that the bearings, for example, are prevented from being damaged, that the sliding loss is prevented from increasing, and that the tooth tips are prevented from being abnormally worn or from seizing.

[0044] In addition, the conventional scroll compressor uses U-rings made of Teflon or the like as the sealing materials. Accordingly, the material itself is comparatively expensive.

[0045] In addition, in the case where the closed vessel is in balanced pressure, as before the compressor starts up, the pressure increases as follows. In the frame space 15f where the refrigerant gas of the intermediate pressure is extracted on the way of compression carried out in the compression chamber 1d immediately after the compressor starts up, the pressure increases comparatively rapidly, while, in the closed vessel, the volume is much larger than the volume of the frame space 15f so that the pressure increases more slowly than the frame space 15f.

[0046] In such a case, for a certain period of time, the pressure levels of the pressure P_{m2} of the frame space 15f and the closed vessel pressure (that is, discharge pressure) P_d come into the condition shown by the following expression.

$$P_{m2} > P_d \quad (15)$$

[0047] On the assumption of steady-state operation, the sealing materials are formed so as to prevent discharge pressure gas from entering the frame space 15f.

However, the sealing materials cannot prevent the flow reverse to that of the discharge pressure gas. In the condition shown by the expression (15), the refrigerant gas in the frame space 15f leaks out into the closed space so that the pressure P_{m2} in the frame space does not increase. Thus, the force required to move the compliant frame 3 toward the compression chamber becomes insufficient. In other words, it takes a long time to start a normal compressing operation. In addition, during this period, the compliant frame 3 and the oscillating scroll 2 moving in the axial direction of the compressor in contact with the compliant frame 3 fluctuate in the clearance of the axial relief amount. Thus, there is a problem that damage, seizing, or the like, is caused to the bearings by the occurrence of one-sided bearing of the bearings.

[0048] According to the present invention, O-rings may be used instead of Teflon so that the material cost can be reduced.

[0049] Even during the starting of the compressor, the pressure P_{m2} of the frame space 15f is quickly increased without leaking the refrigerant gas of intermediate pressure supplied from the compression chamber 1d to the frame space 15f. Thus, the force required to move both the compliant frame 3 and the oscillating scroll 2 toward the compression chamber is generated positively so that a normal compressing operation can be started quickly.

[0050] It is therefore another object of the present invention to provide a scroll compressor of the type described which is low in cost, superior in starting performance, free from damage of bearings, and high in reliability.

[0051] In addition, if conventional O-rings typically made of CR (chloroprene rubber) are used as the sealing materials in the case where an HFC refrigerant (R407C, R410A, etc.) is used as a working fluid, the O-rings are swollen and deteriorated due to compatibility with the refrigerant. Thus, there is a problem that the sealing materials lose their sealing properties.

[0052] In order to solve such a problem, it is therefore another object of present invention to provide a highly reliable scroll compressor of the type described in which O-rings made of HNBR (in which hydrogen atoms are bonded with a part of acrylonitrile-butadiene rubber molecules) are used for the HFC refrigerant so that the O-rings do not deteriorate and do not lose their sealing properties.

[0053] US-A-6 135 739 discloses a scroll compressor in accordance with the preamble of claim 1.

Disclosure of Invention

[0054] According to the present invention, there is provided a scroll compressor as set forth in claim 1.

[0055] In a scroll compressor which is provided in a closed vessel and which comprises: a fixed scroll and an oscillating scroll respectively having plate-like scroll

teeth engaging with each other so as to form a compression chamber therebetween; a compliant frame for supporting the oscillating scroll in an axial direction of the scroll compressor while supporting a main shaft in a radial direction of the scroll compressor for driving the oscillating scroll, the compliant frame being displaceable in the axial direction; and a guide frame for supporting the compliant frame in the radial direction, the oscillating scroll being made movable in the axial direction due to movement of the compliant frame in the axial direction relative to the guide frame, refrigerant gas undergoing compression is extracted from the compression chamber and introduced into a closed frame space formed by disposing two sealing materials on cylindrical surfaces or flat surfaces formed by the compliant frame and the guide frame, and pressure P_{m2} (MPa) in the frame space is set to fall in a range of not less than 1.2 times and not more than 2 times the suction pressure P_s (MPa) of the compressor.

[0056] Thus, the highly reliable high-efficiency scroll compressor is obtained which makes the fixed scroll and the oscillating scroll slide in contact with each other by a proper pressing force in the whole running pressure range of the compressor so that the fixed scroll and the oscillating scroll are prevented from separating from each other and any increase in sliding loss or seizing caused by excessive pressing is prevented.

[0057] In addition, if each sealing material is formed into an O-ring, the cost of the sealing material can be reduced. Further, even during starting of the compressor, the compliant frame and the oscillating scroll move toward the compression chamber quickly without leaking the pressure of the frame space into the closed vessel. Thus, a normal compressing operation can be started. Accordingly, the scroll compressor is obtained which is low in cost and high in reliability.

[0058] In addition, in the case of an HFC refrigerant (R407C, R410A, etc.) used as a working fluid, the sealing material may be made of HNBR (in which hydrogen atoms are bonded with a part of acrylonitrile-butadiene rubber molecules) and formed into an O-ring. As a result, it is possible to obtain sealing properties which ensure the reduced danger of swelling or deteriorating of the O-ring. Thus, the highly reliable scroll compressor is obtained.

Brief Description of Drawings

[0059]

Fig. 1 is a longitudinal sectional view of an embodiment of the present invention;

Fig. 2 is a graph showing the running temperature range of a compressor;

Fig. 3 is a graph showing the correlation between value α and rated performance ratio in the case where a refrigerant is R407C;

Fig. 4 is a graph showing the correlation between

the value α and the rated performance ratio in the case where a refrigerant is R410A;

Fig. 5 is a graph showing the correlation between value β and the rated performance ratio;

Fig. 6 is a longitudinal sectional view of a conventional scroll compressor;

Fig. 7 is an explanatory view of axial forces acting on the respective parts;

Fig. 8 is an enlarged sectional view showing the sealing materials and the adjoining parts; and

Fig. 9 is a table of low compression ratio running pressures in respective refrigerants according to the present invention.

15 Best Mode for Carrying Out the Invention

[0060] Fig. 1 is a longitudinal sectional view showing a scroll compressor in which the names and functions of respective parts are similar to their counterparts in the above-mentioned conventional apparatus. Therefore, those parts are designated by the same reference numerals and the description thereof will be omitted.

[0061] Two sealing materials for forming a frame space 15f are O-rings 16c and 16d. The O-rings 16c and 16d are disposed on cylindrical surfaces 15d and 15d formed by the inner circumference of a guide frame 15 and the outer circumference of a compliant frame 3. The O-rings are made of HNBR. There is no danger of the O-rings being swollen and deteriorated even when an HFC refrigerant is used. A suitable material may be selected for the O-rings in accordance with the kind of the refrigerant filling the compressor, the atmosphere temperature, and so on.

[0062] During the starting of the compressor, pressure P_{m2} in the frame space 15f for extracting and introducing refrigerant gas on the way of compression in a compression chamber 1d increases more rapidly than the pressure P_d (i.e., discharge pressure) in a closed vessel. However, pressure leakage from the frame space 15f into the closed vessel can be prevented by the O-rings forming the frame space 15f. Accordingly, the pressure P_{m2} in the frame space increases so quickly that a force tending to move the compliant frame 3 toward the compression chamber 1d is imparted to the compliant frame 3. Thus, a normal compressing operation can be started quickly.

[0063] A boss portion outside space 2h is disposed midway in an oil feed passageway of a refrigerating machine oil 10e in the closed vessel. The refrigerating machine oil 10e passes through the differential pressure oil feed passageway as follows. The refrigerating machine oil 10e in the high pressure bottom portion of the closed vessel passes through a main shaft hollow portion 4g and reaches the boss portion outside space via a main bearing 3c and an oscillating bearing 2c. Then, the refrigerating machine oil 10e is introduced into a low pressure space 1g via an intermediate pressure regulating valve receiving space 3p provided in the compliant

frame 3. It is preset so that by virtue of the throttling action of the main bearing 3c and the oscillating bearing 2c and adjustment of the spring constant of an intermediate pressure regulating spring 3m which is provided in the regulating valve receiving space, the pressure Pm1 of the boss portion outside space 2h becomes such that the value α shown in the expression (1) becomes about 0.3. As a result, thrust contact force Fth is reduced to reduce the thrust sliding loss in the whole running pressure range of the compressor. At the same time, a normal compressing operation can be ensured without separating the oscillating scroll 2 and the compliant frame 3 from each other. In addition, a positive value is ensured for oil feed differential pressure ΔP of the refrigerant machine oil. Thus, there is no danger of the oil feeding to the oscillating bearing 2c and the main bearing 3c being interrupted.

[0064] The frame space 15f is filled with intermediate pressure refrigerant gas supplied continuously or intermittently through an extraction hole 2j and a communication passageway 3s. The pressure Pm2 in the frame space is set in accordance with the position of the compressor chamber 1d with which the extraction hole 2j communicates, so that the value β shown in the expression (3) becomes about 1.6. As a result, in the entire running pressure range of the compressor, tooth tip contact force Ftip does not assume a negative value. Thus, the oscillating scroll 2 and the fixed scroll 1 are prevented from separating from each other so that a normal compressing operation can be ensured. In addition, there is no danger of the sliding loss being increased by excessive pressing of the tooth tips.

[0065] Incidentally, the intermediate pressure acting area or the high pressure acting area of the boss portion outside space or the frame space is defined on the basis of the value α or the value β described above. The optimum values α and β change in accordance with the adjustment of these areas. Generally, an intermediate pressure acting area Spm1 of the boss portion outside space 2h is determined by geometric shapes of an Oldham's ring, a thrust bearing, and so on. Accordingly, the intermediate pressure acting area Spm1 has a limited degree of freedom in setting. On the other hand, an intermediate pressure acting area Spm2 of the frame space 15f has a relatively large degree of freedom in adjustment. It is therefore preferable that the intermediate pressure active area Spm2 is set to be as large as possible so that the value β is set to be small. That is, it is preferable that the intermediate pressure Pm2 of the frame space is set to be low. Thus, stable tooth tip contact force Ftip can be obtained in the wide running pressure range of the compressor. In addition, the compliant frame 3 and the oscillating scroll 2 can be moved toward the compression chamber by low intermediate pressure Pm2. Thus, calculative and experimental results have been obtained showing improvements in the starting characteristics of the compressor, or the like.

[0066] Now, description will be made about the setting

of the value α determining the pressure Pm1 of the boss portion outside space 2h.

[0067] By setting the value α to be large, the thrust contact force Fth or the thrust sliding loss can be reduced as described previously in connection with the conventional apparatus. However, if the value α is set to be too large, that is, if the pressure Pm1 of the boss portion outside space 2h is set to be too high, the thrust contact force Fth assumes a negative value. Thus, there is a problem that the oscillating scroll 2 and the compliant frame 3 are separated from each other, or the differential pressure ΔP for feeding oil to the oscillating bearing 2c and the main bearing 3c cannot be ensured.

[0068] Fig. 2 shows an ordinary running temperature range quarantined by the compressor. Oil feeding has to be ensured in such a wide range. In Fig. 2, the condition that makes it difficult for the compressor to feed oil appears to be at a running point (low compression ratio) at which the difference between condensation temperature CT and evaporation temperature ET is minimum, that is, the difference between discharge pressure Pd and suction pressure Ps is minimum. In Fig. 2, this running point is a right lower point in the running temperature range, and the ratio CT/ET is 30/10°C. The difference min(Pd-Ps) between the discharge pressure Pd and the suction pressure Ps at this point varies in accordance with the refrigerant to be used. The points are summarized in Fig. 9.

[0069] A differential pressure head for feeding oil to the oscillating bearing 2 and the main bearing 3c becomes the differential pressure ΔP between the closed vessel pressure (i.e., discharge pressure) Pd and the boss portion outside space pressure Pm1 as shown in the expression (2). For example, in the case where the refrigerant used is R407C, if the value α reaches 0.6 or larger, the pressure Pm1 and the differential pressure ΔP take the following values respectively at the running point (Pd/Ps=1.27/0.71 MPa) shown in Fig. 9.

$$Pm1 = Ps + \alpha = 0.71 + 0.6 = 1.31 \text{ (MPa)}$$

$$\Delta P = Pd - Pm1 = 1.27 - 1.31 = -0.04 \text{ (MPa)} < 0$$

The value of the differential pressure ΔP shows that oil cannot be fed under such running pressure conditions. Specifically, in the case where R407C is used as a working refrigerant, the value α has to be set not larger than the value of the high/low pressure difference min(Pd-Ps), that is, not larger than 0.56 at the low compression ratio running pressure (Pd/Ps=1.27/0.71 MPa).

[0070] Likewise, the value α has to be set smaller than 0.51 ($\alpha < 0.51$) when R22 is used as a working refrigerant, and the value α has to be set smaller than 0.8 ($\alpha < 0.8$) when R410A is used as a working refrigerant. Otherwise, an area where no oil is fed appears in the running pressure range of the compressor. Therefore,

the value α has to be set not larger than any one of the above-mentioned values.

[0071] Also in the case where the refrigerant to be used in the compressor or the running pressure range of the compressor differs from the above-mentioned refrigerant or running pressure range, it is necessary to set the value α to be not larger than the differential pressure value $\min(P_d - P_s)$ where a difference between high and low pressures is minimum in the running pressure range of the compressor.

[0072] Fig. 3 shows the rated performance ratio η when the value α is varied with R407C as a working refrigerant. The rated performance ratio is expressed by the performance ratio on the assumption that the performance MAX value is 100%. In the area where the value α is small, the effect of relieving the thrust contact force F_{th} cannot be obtained sufficiently, and there is a tendency that the thrust sliding loss increases so that the performance deteriorates gradually. If the value α is increased gradually, the effect of relieving the thrust sliding loss is exhibited so that the performance is improved. The performance reaches a peak (100%) when the value α is about 0.3. If the value α is increased further, the thrust sliding loss becomes smaller. However, the thrust contact force F_{th} becomes somewhat insufficient. Thus, upsetting moment generated in the oscillating scroll cannot be maintained so that, though a very small, a clearance begins to appear between the tooth tips. Thus, there is a tendency that the performance lowers again due to the deterioration of the volumetric efficiency or the increase of the internal leakage loss. If the value α exceeds 0.7, the thrust contact force F_{th} becomes completely insufficient. Thus, the compliant frame 3 and the oscillating scroll 2 are separated from each other so that the performance deteriorates rapidly. In Fig. 3, the value α which was required for ensuring the performance to be 95% or more with respect to the performance MAX value was in a range of from 0 to 0.5.

[0073] Next, description will be made about merits of this embodiment in the case where a high pressure working refrigerant is used.

[0074] A high pressure working refrigerant (e.g., R401A or R32) is higher in working/running pressure than any other refrigerant (e.g., R22 or R407C). Therefore, the radial load on the oscillating bearing 2c or the main bearing 3c and the load on the thrust bearing 3a increase.

[0075] Generally, in the case of high pressure working refrigerant, a stroke volume V_{st} of the compressor becomes small due to the thermal physical properties of the refrigerant itself. In order to relieve the stress generated in the scroll teeth due to the high pressure refrigerant, generally, the stroke volume V_{st} in the scroll compressor is adjusted by reducing the height of the scroll teeth or increasing the tooth thickness. In such a method, the radial load on the oscillating bearing 2c or the main bearing 3c can be reduced to the conventional level. However, in this method, the load on the thrust bear-

ing cannot be reduced. Thus, the thrust sliding loss increases and therefore a decline in the performance of the compressor is caused.

[0076] In order to solve this problem, the scroll compressor according to the present invention is constructed so that the thrust bearing load can be reduced when the pressure P_{m1} of the boss portion outside space 2h is set to be high (the value α is set to be large). In addition, as shown in Fig. 9, the value α , in the case of R410A, has an upper limit of about 0.8 to ensure the oil feed differential pressure. This upper limit is higher than that of any other refrigerant (R22 or R407C). Thus, the degree of freedom with which the value α can be set to be large is high so that the effect of reducing the thrust bearing load is also large. In other words, the scroll compressor shown in this embodiment exhibits advantages more as the pressure of working refrigerant is high.

[0077] Fig. 4 shows the correlation between the value α and the rated performance ratio when R410A i.e., a high pressure working refrigerant is used. In Fig. 4, the correlation in the above-mentioned case of R407C is also shown.

[0078] In the area where the value α is small, the thrust bearing load is large. In addition, the effect of canceling the load according to this embodiment is not exhibited sufficiently. Thus, the performance ratio in the case of R410A takes a smaller value than that in the case of R407C. If the value α is increased gradually, the effect of canceling the thrust bearing load according to this embodiment appears. Then, the performance reaches a peak point at a higher level of the value α than that in the case of R407C. In this embodiment, the performance reached a peak point at $\alpha=0.5$. As described above, the high pressure working refrigerant (R410A) has a larger thrust bearing load than R407C or R22. Accordingly, better performance can be obtained by setting the intermediate pressure P_{m1} of the boss portion outside space 2h to be higher, that is, by setting the value α to be larger. If the value α is increased further, the thrust contact force F_{th} becomes so insufficient that the performance lowers again by the same reason as described in connection with Fig. 3.

[0079] In Fig. 4, the value α which was required for keeping the performance ratio to be 95% or more was approximately in a range of $0.2 < \alpha < 0.7$.

[0080] Accordingly, as shown in Fig. 9, it is necessary that the upper limit of the value α is set to be the differential pressure value $\min(P_d - P_s)$ at which the difference between high and lower pressures is minimum in the running pressure range of the compressor. The optimum value α is therefore not larger than the value $\min(P_d - P_s)$. The optimum value α must be determined experimentally by measuring the performance or the like within a range where the thrust contact force F_{th} is neither too small nor too large.

[0081] Although the value α changed somewhat in accordance with the intermediate pressure acting area S_{pm1} , the optimum value α obtained experimentally in

this embodiment was a value substantially in the vicinity of half the value $\min(P_d - P_s)$ shown in Fig. 9, i.e., $\alpha \approx \{\min(P_d - P_s)\}/2$.

[0082] Next, description will be made about determination of the value β in the expression (3) for setting to a proper value the force F_{pm2} in the expression (11), which tends to cause the guide frame 15 and the compliant frame 3 to be separated from each other.

[0083] If the value β is set to be too small, it becomes difficult to ensure a positive value for the tooth tip contact force F_{tip} at certain running pressures. Thus, a normal compressing operation cannot be warranted. On the other hand, if the value β is set to be too large, the tooth tip contact force F_{tip} in the expression (10) becomes larger than required. Thus, the resulting increase in sliding loss causes an inconvenience such as deterioration of the performance of the compressor, seizing of the tooth tips, or the like.

[0084] Fig. 5 shows the rated performance ratio when the value β is varied in the scroll compressor of this embodiment. The rated performance ratio is expressed in terms of the performance ratio by taking the performance MAX value as 100% in the same manner as described above.

[0085] In the area where the value β is small, the tooth tip contact force F_{tip} is entirely insufficient. Thus, the compliant frame 3 and the oscillating scroll 2 cannot move toward the compression chamber so that a normal compressing operation cannot be carried out. Accordingly, the performance is considerably low. If the value β is increased gradually, the tooth tip contact force F_{tip} assumes a positive value, but the upsetting moment generated in the oscillating scroll 2 cannot be maintained so that a very small clearance appears at the tooth tips. Thus, the performance cannot be said sufficient as yet in view of the deterioration of the volumetric efficiency or the increase of the internal leakage loss. However, such a leakage phenomenon is reduced gradually in the vicinity of $\beta=1.2$ so that the tooth tip contact force F_{tip} becomes sufficient. Thus, the performance improves and reaches a peak (100%) at about $\beta=1.6$. Thereafter the tooth tip sliding loss increases due to the increase of the tooth tip contact force F_{tip} and the performance tends to deteriorate again.

[0086] In Fig. 5, the value β which was required for ensuring the performance ratio to be 95% or more was in a range of $1.2 < \beta < 2.0$.

Claims

1. A scroll compressor provided in a closed vessel, comprising:

a fixed scroll (1) and an oscillating scroll (2) respectively having plate-like scroll teeth (1b, 2b) which engage with each other so as to form a compression chamber (1d) therebetween;

a compliant frame (3) for supporting said oscillating scroll (2) in an axial direction of said scroll compressor while supporting a main shaft (4) in a radial direction of said scroll compressor for driving said oscillating scroll (2), said compliant frame (3) being displaceable in said axial direction; and

a guide frame (15) for supporting said compliant frame (3) in said radial direction; said oscillating scroll (2) being made movable in said axial direction due to movement of said compliant frame (3) in said axial direction relative to said guide frame (15);

characterised in that:

refrigerant gas undergoing compression is extracted from said compression chamber (1d) and introduced into a closed frame space (15f) formed by disposing two sealing materials (16c, 16d) respectively on two cylindrical surfaces or flat surfaces formed by said compliant frame (3) and said guide frame (15); and the pressure P_{m2} in said frame space (15f) is set to fall in a range of not less than 1.2 times and not more than 2 times the suction pressure P_s of said compressor.

2. A scroll compressor according to Claim 1, wherein said sealing materials (16c, 16d) are O-rings.
3. A scroll compressor according to Claim 1, wherein O-rings made of HNBR in which hydrogen atoms are bonded with a part of acrylonitrile-butadiene rubber molecules are used for said sealing materials (16c, 16d) when an HFC refrigerant is used as a working fluid.

FIG. 1

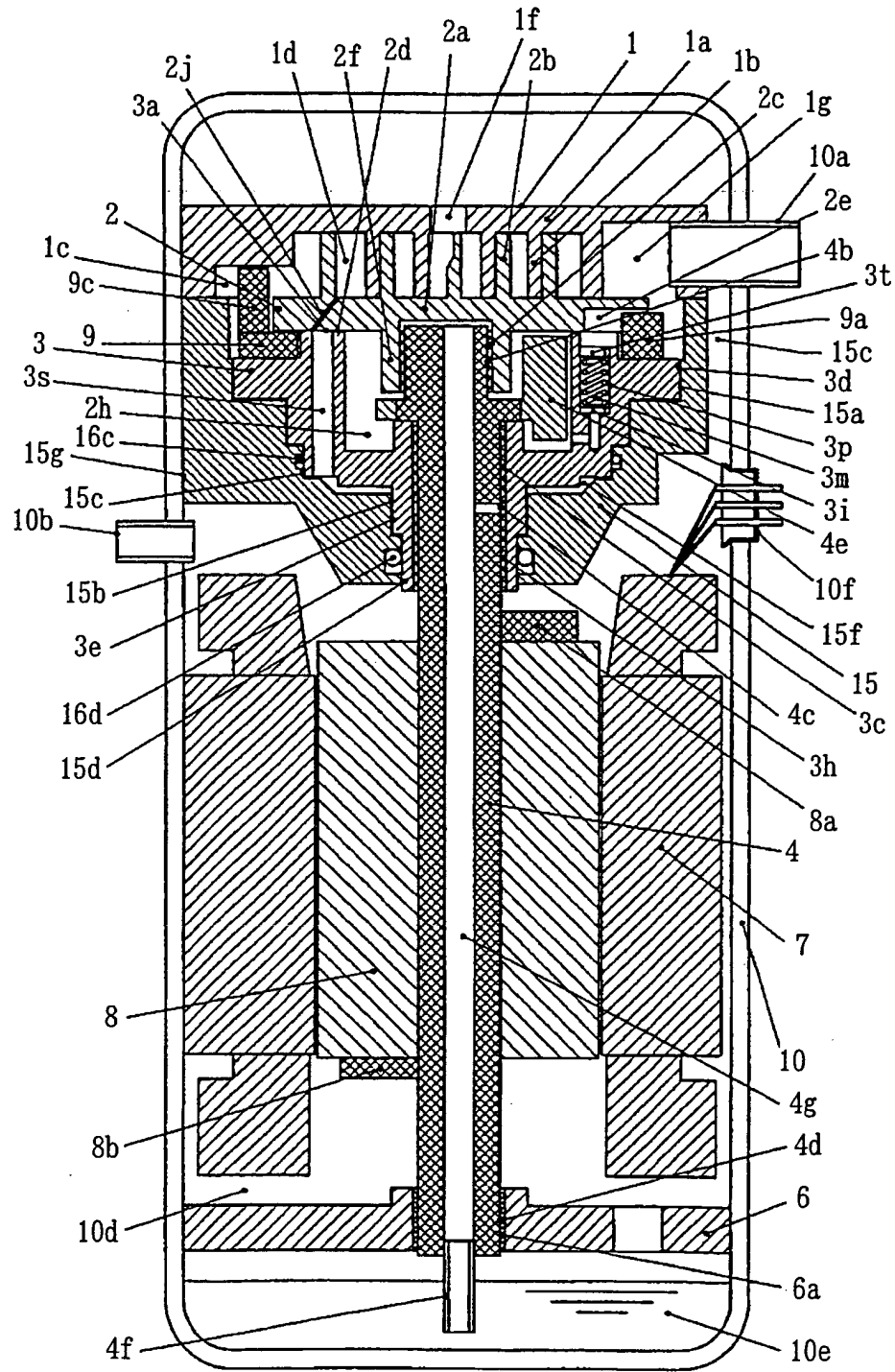


FIG. 2

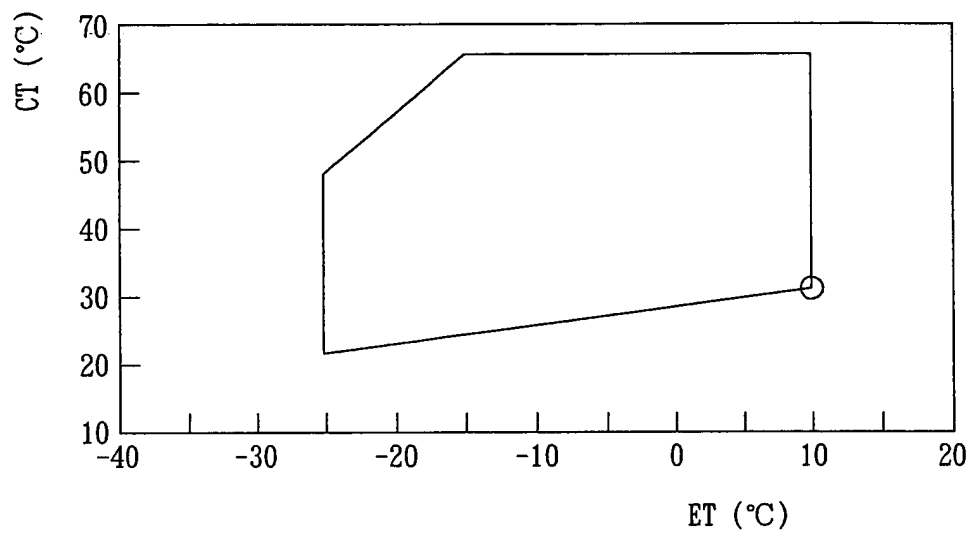


FIG. 3

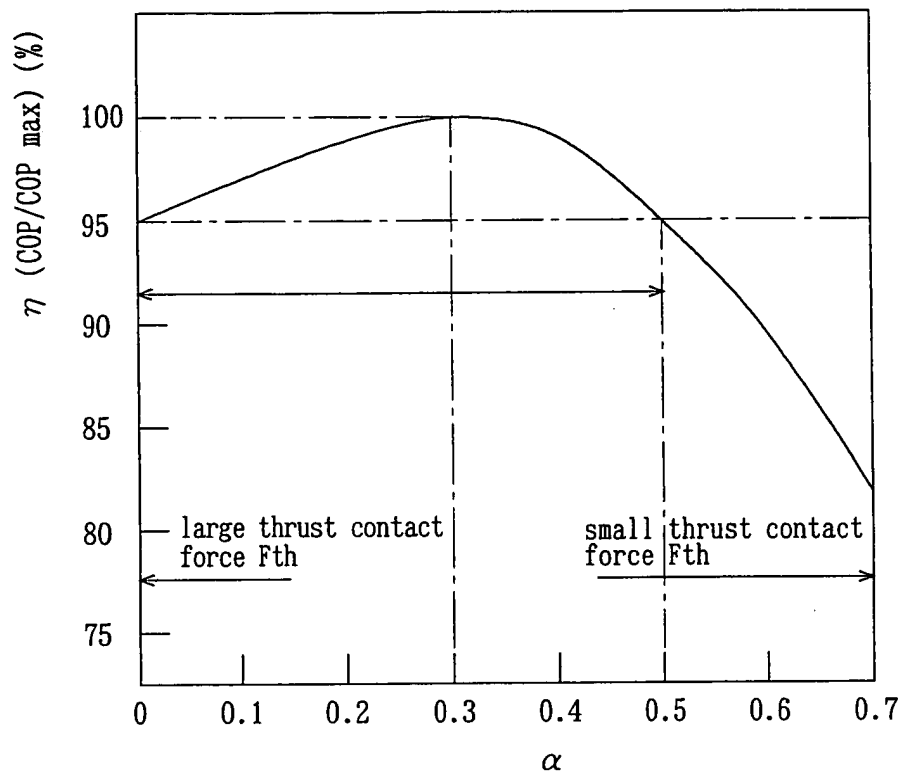


FIG. 4

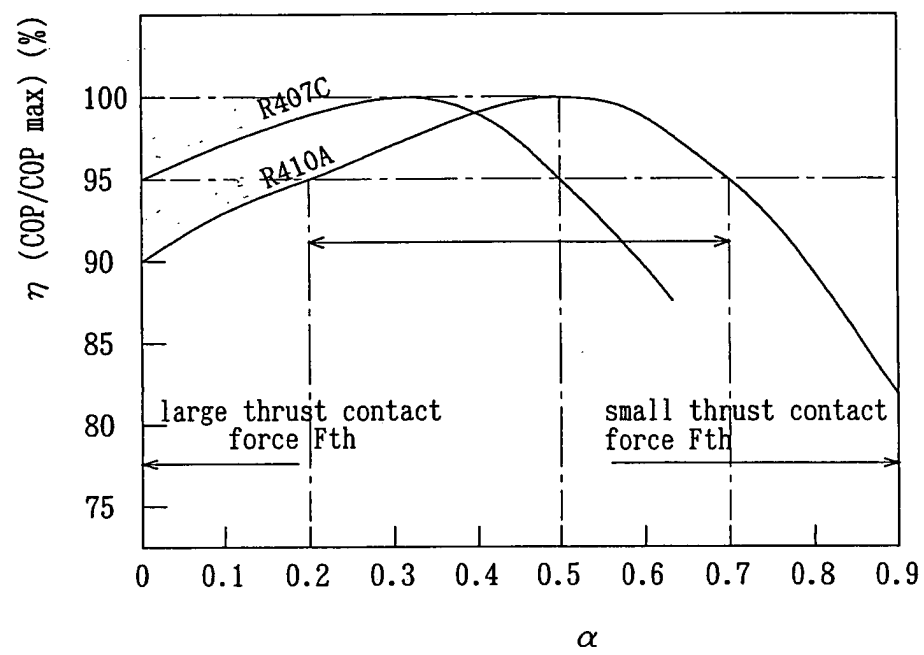


FIG. 5

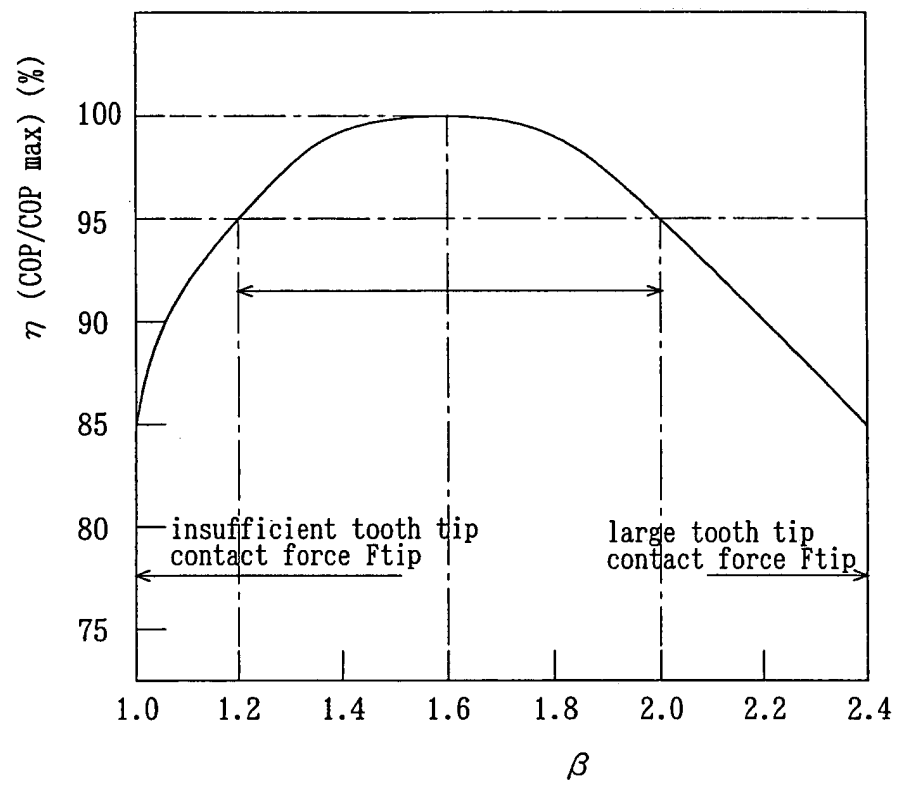


FIG. 6

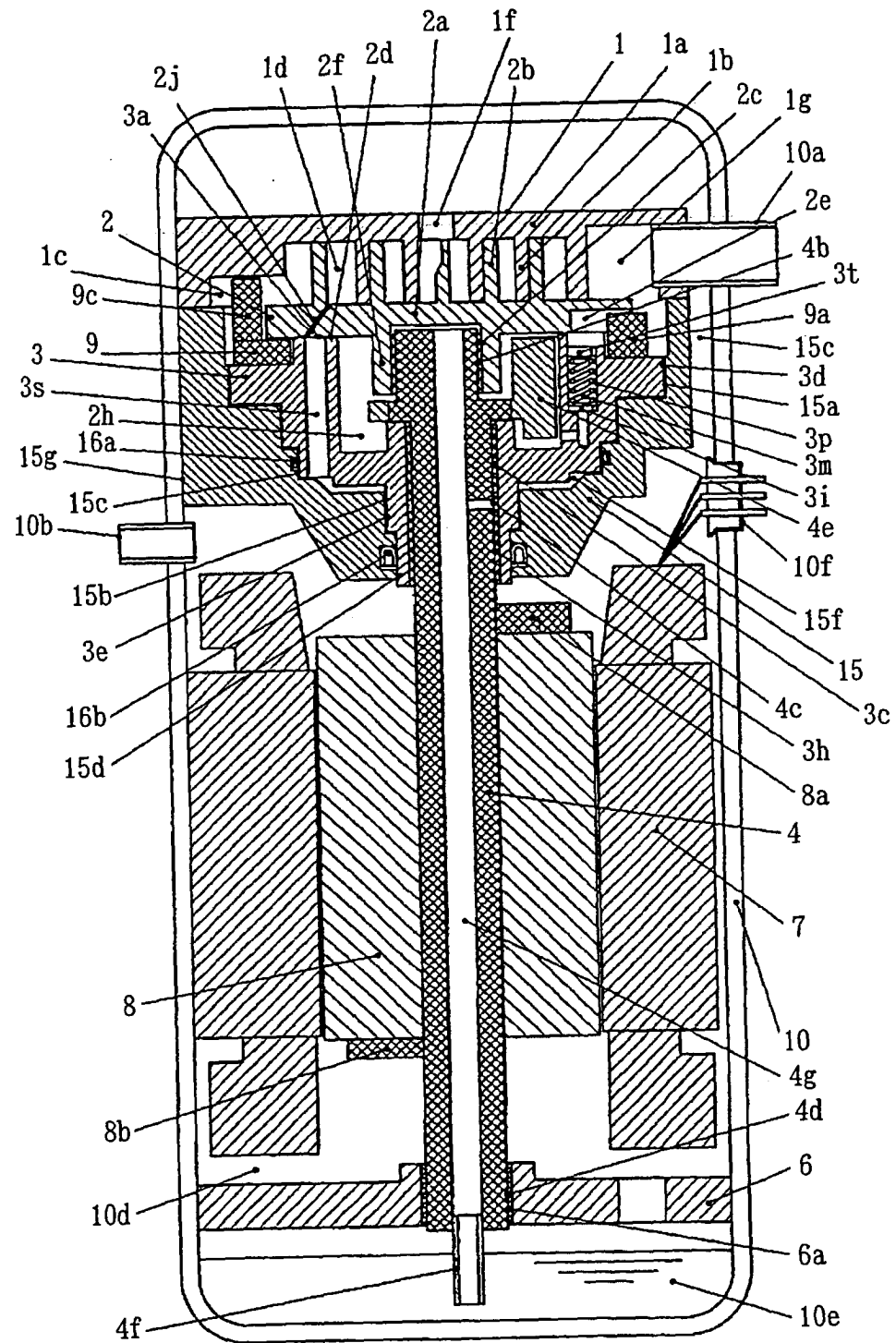


FIG. 7

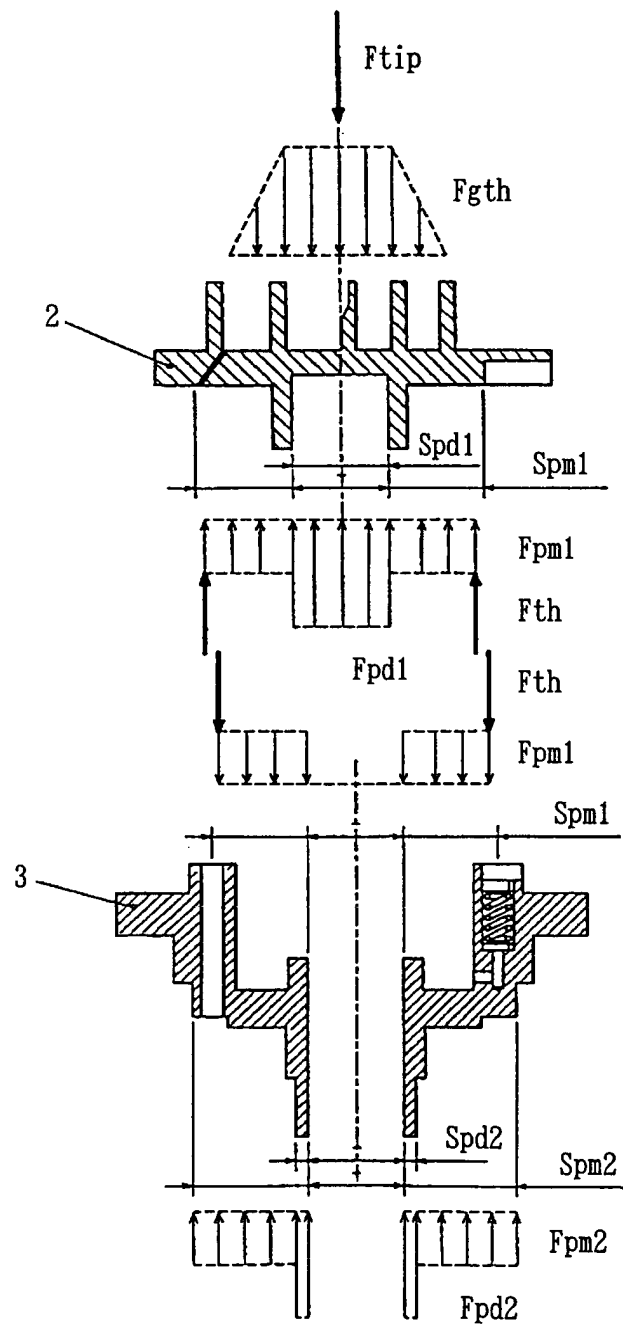


FIG. 8

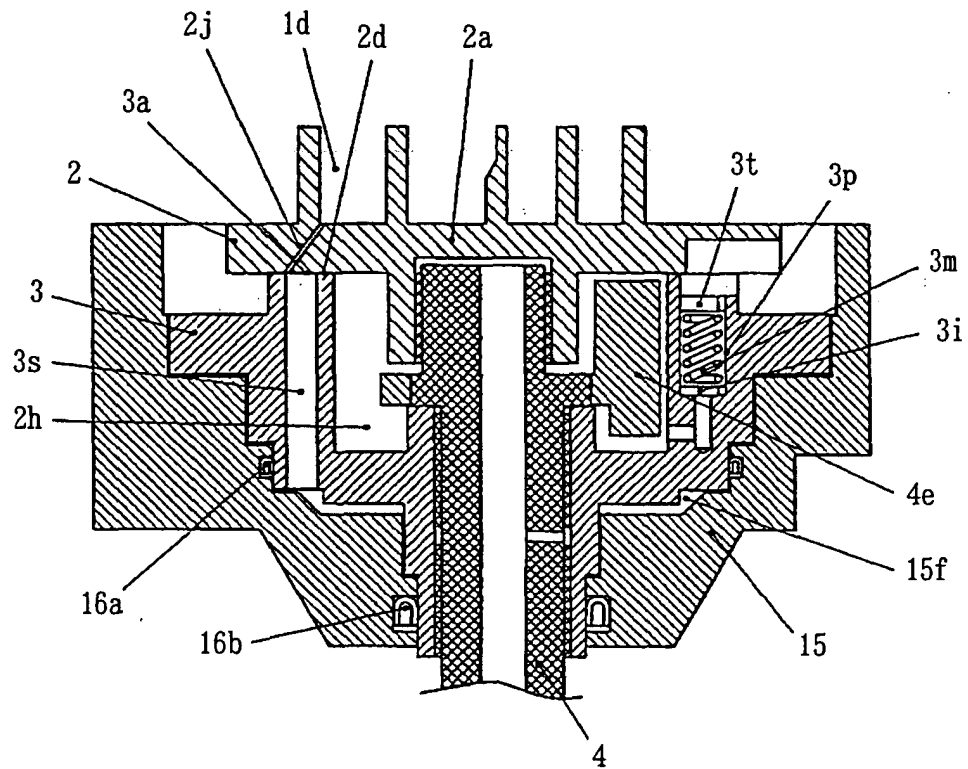


FIG. 9

	Pd(MPa)	Ps(MPa)	min(Pd-Ps)(MPa)
R22	1.19	0.68	0.51
R407C	1.27	0.71	0.56
R410A	1.88	1.08	0.80

(CT/ET = 30/10 °C)



European Patent
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EUROPEAN SEARCH REPORT

Application Number
EP 05 01 2629

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The present search report has been drawn up for all claims			
Place of search Munich		Date of completion of the search 19 July 2005	Examiner Descoubes, P
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