

Description

The present invention relates to an adjustable slant plate type refrigerant compressor according to the first part of claim 1 and to a slant plate control mechanism for use in such a compressor according to the first part of claim 4. Such a compressor or control mechanism, resp., is particularly suitable for use in an automotive air conditioning system.

A wobble plate type refrigerant compressor with a variable displacement mechanism according to the first part of claim 1 is disclosed in US Patent 4,960,367 or EP-A-0 260 667. For purposes of explanation only, the left side of the Figure 1 will be referenced as the forward end or front and the right side of the Figure will be referenced as the rearward end.

Compressor 10 includes cylindrical housing assembly 20 including cylinder block 21, front end plate 23 at one end of cylinder block 21, crank chamber 22 formed between cylinder block 21 and front end plate 23, and rear end plate 24 attached to the other end of cylinder block 21. Front end plate 23 is mounted on cylinder block 21 forward of crank chamber 22 by a plurality of bolts 101. Rear end plate 24 is mounted on cylinder block 21 at its opposite end by a plurality of bolts 102. Valve plate 25 is located between rear end plate 24 and cylinder block 21. Opening 231 is centrally formed in front end plate 23 for supporting drive shaft 26 by bearing 30 disposed in the opening 231. The inner end portion of drive shaft 26 is rotatably supported by bearing 31 disposed within central bore 210 of cylinder block 21. Bore 210, which extends to a rearward end surface of cylinder block 21, contains valve control mechanism 19' as discussed below.

Cam rotor 40 is fixed on drive shaft 26 by pin member 261 and rotates with drive shaft 26. Thrust needle bearing 32 is disposed between the inner end surface of front end plate 23 and the adjacent axial end surface of cam rotor 40. Cam rotor 40 includes arm 41 having pin member 42 extending therefrom. Slant plate 50 is adjacent cam rotor 40 and includes opening 53 through which drive shaft 26 passes. Slant plate 50 includes arm 51 having slot 52. Cam rotor 40 and slant plate 50 are connected by pin member 42, which is inserted in slot 52 to create a hinged joint. Pin member 42 is slidable within slot 52 to allow adjustment of the angular position of slant plate 50 with respect to a plane perpendicular to the longitudinal axis of drive shaft 26.

Wobble plate 60 is rotatably mounted on slant plate 50 through bearings 61 and 62. Fork shaped slider 63 is attached to the outer peripheral end of wobble plate 60 and is slidably mounted on sliding rail 64 held between front end plate 23 and cyl-

inder block 21. Fork shaped slider 63 prevents rotation of wobble plate 60 so that wobble plate 60 nutates along rail 64 when cam rotor 40 rotates. Cylinder block 21 includes a plurality of peripherally located cylinder chambers 70 in which pistons 71 reciprocate. Each piston 71 is connected to wobble plate 60 by a corresponding connecting rod 72.

Rear end plate 24 includes peripherally located annular suction chamber 241 and centrally located discharge chamber 251. Valve plate 25 is located between cylinder block 21 and rear end plate 24 and includes a plurality of valved suction ports 242 linking suction chamber 241 with respective cylinders 70. Valve plate 25 also includes a plurality of valved discharge ports 252 linking discharge chambers 251 with respective cylinders 70. Suction ports 242 and discharge ports 252 are provided with suitable reed valves as described in U.S. Pat. No. 4,011,029 to Shimizu.

Suction chamber 241 includes inlet portion 241a which is connected to an evaporator of the external cooling circuit (not shown). Discharge chamber 251 is provided with outlet portion 251a connected to a condenser of the cooling circuit (not shown). Gaskets 27 and 28 are located between cylinder block 21 and the front surface of valve plate 25, and the rear surface of valve plate 25 and rear end plate 24 respectively, to seal the mating surfaces of cylinder block 21, valve plate 25 and rear end plate 24.

With reference to Figure 2, valve control mechanism 19' includes cup-shaped casing member 191 defining valve chamber 192 therewithin. O-ring 19a is disposed between an outer surface of casing member 191 and an inner surface of bore 210 to seal the mating surfaces of casing member 191 and cylinder block 21. A plurality of holes 19b are formed at the closed end (to the left in Figures 1 and 2) of casing member 191 to expose valve chamber 192 to the crank chamber pressure through gap 31a existing between bearing 31 and cylinder block 21. Valve device 193, which has a longitudinally expandable and contractable bellows 193a and valve element 193b attached at a rearward end of bellows 193a, is disposed in valve chamber 192. Bellows 193a longitudinally contracts and expands in response to the crank chamber pressure. Bellows 193a is made of an elastic material, for example, phosphor bronze and has an effective pressure receiving cross-sectional area which is designated below as area A_1 . Valve element 193b is generally hemispherical shaped and is attached at the rearward end of bellows 193a. Projection member 193c, which is attached at a forward end of bellows 193a, is secured to axial projection 19c formed at the center of the closed end of casing member 191. Bias spring 193d is

longitudinally and compressedly disposed within an inner hollow space of bellows 193a. The resultant force F of the restoring force of bellows 193a and bias spring 193d continuously urges valve element 193b rearwardly (to the right in Figures 1 and 2).

Cylinder member 194, which includes valve seat 194a, penetrates the center of valve plate assembly 200, which includes valve plate 25, gaskets 27, 28, suction reed valve 271 and discharge reed valve 281. Valve seat 194a is formed at a forward end of cylinder member 194 and is secured to an opened end of casing member 191. Nut 100 is screwed on cylinder member 194 from a rearward end of cylinder member 194 located in discharge chamber 251 to fix cylinder member 194 to valve plate assembly 200 with valve retainer 253. Conical-shaped opening 194b, which receives valve element 193b, is formed at valve seat 194a and is linked to cylinder 194c axially formed in cylinder member 194. Consequently, annular ridge 194d is formed at a location which is the boundary between conical-shaped opening 194b and cylinder 194c.

When bellows 193a expands to a certain longitudinal length, generally hemispherical-shaped valve element 193b is received by conical-shaped opening 194b to form a circular line contact 193e therebetween. Circular line contact 193e divides valve element 193b into front portion 193f and rear portion 193g, an exterior surface of which is responsive to pressure in suction chamber 241 conducted via later-mentioned radial hole 151, conduit 152 and hole 153. Rear portion 193g of valve element 193b has the effective pressure receiving cross-sectional area which is designated below as area A_2 , and which is approximately 50% of the effective pressure receiving cross-sectional area A_1 of bellows 193a.

Actuating rod 195, which is slidably disposed within cylinder 194c, slightly projects from the rearward end of cylinder 194c, and is linked to valve element 193b through bias spring 196, which smoothly transmits the force from actuating rod 195 to valve element 193b of valve device 193. Actuating rod 195 includes annular flange 195a which is integral with and radially extends from an outer surface of a front end portion of actuating rod 195. Annular flange 195a is located in conical shaped opening 194b, and prevents an excessive rearward movement of actuating rod 195 by contacting with annular ridge 194d. O-ring 197 is mounted about actuating rod 195 to seal the mating surfaces of cylinder 194c and actuating rod 195, thereby preventing the invasion of the refrigerant gas from discharge chamber 251 to conical shaped opening 194b via the gap created between cylinder 194c and rod 195. Cup-shaped member 103 having a threaded portion at its inner periph-

eral side wall is mounted on the rear end portion of cylinder member 194 to prevent O-ring 197 from falling off from the rear end of cylinder member 194.

Radial hole 151 is formed at valve seat 194a to link conical shaped opening 194b to conduit 152 formed in cylinder block 21. Conduit 152, which includes cavity 152a, is linked to suction chamber 241 through hole 153 formed at valve plate assembly 200. Passageway 150, which provides communication between crank chamber 22 and suction chamber 241, includes gap 31a, bore 210, holes 19b, valve chamber 192, conical shaped opening 194b, radial hole 151, conduit 152 and hole 153. As a result, the opening and closing of passageway 150 is controlled by the contraction and expansion of valve device 193 primarily in response to crank chamber pressure.

During operation of compressor 10, drive shaft 26 is rotated by the engine of the vehicle through an electromagnetic clutch 300. Cam rotor 40 is rotated with drive shaft 26, rotating slant plate 50 as well, which causes wobble plate 60 to nutate. Nutational motion of wobble plate 60 reciprocates pistons 71 in their respective cylinders 70. As pistons 71 are reciprocated, refrigerant gas which is introduced into suction chamber 241 through inlet portion 241a flows into each chamber 70 through suction ports 242 and then is compressed. The compressed refrigerant gas is discharged to discharge chamber 251 from each cylinder 70 through discharge ports 252, and therefrom into the cooling circuit through outlet 251a.

The capacity of compressor 10 is adjustable to maintain a constant pressure in suction chamber 241 in response to changes in the heat load on the evaporator or changes in the rotating speed of the compressor. Adjustment of the capacity of the compressor occurs by changing the angle of slant plate 50 which is dependent upon the crank chamber pressure. An increase in crank chamber pressure decreases the slant angle of slant plate 50 and wobble plate 60, decreasing the capacity of the compressor. A decrease in the crank chamber pressure increases the angle of slant plate 50 and wobble plate 60, increasing the capacity of the compressor.

As discussed in U.S. Patent No. 4,960,367, the effect of valve control mechanism 19 is to maintain a constant pressure at the outlet of the evaporator by controlling the capacity of the compressor in the following manner. Actuating rod 195 pushes valve element 193b in the direction to contract bellows 193a and bias spring 196. Actuating rod 195 moves in response to pressure in discharge chamber 251. Accordingly, increasing pressure in discharge chamber 251 further moves rod 195 toward bellows 193a, thereby increasing the contraction of bellows

193a. As a result, the control point for changing the displacement of the compressor is shifted to maintain a constant pressure at the evaporator outlet.

That is, valve control mechanism 19 makes use of the fact that the discharge pressure of the compressor is roughly directly proportional to the suction flow rate. Since actuating rod 195 moves in direct response to changes in discharge pressure, and applies a force directly to valve device 193, the control point at which valve device 193 operates is shifted in a direct and responsive manner by changes in discharge pressure.

Further operation of valve control mechanism 19 is described in detail below. In order to simplify the explanation of the operation of valve control mechanism 191, the above-mentioned effect of valve control mechanism 19 is neglected hereinafter.

With reference to Figures 3 and 4, and as particularly illustrated in Figure 4, in a situation where operation of the compressor is stopped, the suction chamber pressure P_s and the crank chamber pressure P_c are in a state of equilibration, i.e., $P_c = P_s$, which is greater than the operating point P_1' of valve device 193. This causes the contraction of bellows 193a so that valve element 193b permits communication between suction chamber 241 and valve chamber 192 through conical-shaped opening 194b, radial hole 151, conduit 152 and hole 153 to thereby establish communication between crank chamber 22 and suction chamber 241.

In one compressor operational situation indicated by time period "a" in Figure 4, which is a so-called cool down stage, the compressor operates as follows. In the beginning of operation of the compressor, the communication between crank chamber 22 and suction chamber 241 is maintained, thereby satisfying the equation $P_c = P_s$ as shown by the straight line "li" in Figure 3 until the suction chamber pressure P_s falls to the operating point P_1' of valve device 193. When the suction chamber pressure P_s falls to the operating point P_1' of valve device 193, valve element 193b contacts an inner surface of conical-shaped opening 194b due to expansion of bellows 193a. If the suction chamber pressure P_s drops below the operating point P_1' of valve device 193, valve element 193b frequently opens and closes conical-shaped opening 194b in accordance with the following equation:

$$F = (A_1 - A_2) P_c + A_2 \cdot P_s \quad (1)$$

wherein F is the resultant force of the restoring forces of bellows 193a and bias spring 193d, A_1 is the effective pressure receiving cross-sectional area of bellows 193a, A_2 is the effective pressure

receiving cross-sectional area of rear portion 193g of valve element 193b, P_s is the pressure in suction chamber 241, and P_c is the pressure in crank chamber 22. The above equation (1) can be converted into the following equation by solving for P_c :

$$P_c = A_2 \cdot P_s / (A_2 - A_1) + F / (A_1 - A_2) \quad (2)$$

Equation (2) shows that the crank chamber pressure P_c varies in accordance with the changes in the suction chamber pressure P_s . Furthermore, in this prior art, A_2 is $0.5A_1$ so that equation (2) can be further converted to the following equation by substituting $0.5A_1$ for A_2 .

$$P_c = -P_s + 2F/A_1 \quad (3)$$

Equation (3) is shown by the straight line "m'" in Figure 3. Therefore, suction chamber pressure P_s decreases in inverse proportion to the increase in the crank chamber pressure P_c with a proportion of one to one when the suction chamber pressure P_s is less than the operating point P_1' of valve device 193. At that time, the angular position of slant plate 50 is maintained at the maximum slant angle. However, as illustrated in Figure 4, once the suction chamber pressure P_s reaches one predetermined pressure P_5' at which the pressure difference between the crank and suction chambers 22 and 241 becomes ΔP_{max} , the angular position of slant plate 50 shifts to an angle which is smaller than its maximum slant angle. Therefore, the displacement of the compressor shifts to a value which is smaller than the maximum value.

Another compressor operational situation where the heat load on the evaporator gradually decreases is depicted by time period "b" in Figure 4. As long as the angular position of slant plate 50 is maintained at one angle, suction chamber pressure P_s gradually decreases while the crank chamber pressure P_c gradually increases so as to satisfy equation (3). However, once the suction chamber pressure P_s reaches predetermined pressure P_5' the angular position of slant plate 50 shifts from one angle to another angle which is smaller than the first angle. Therefore, the displacement of the compressor shifts from one value to another value which is smaller than the first value. When the displacement of the compressor shifts to the smaller value due to the change in the angular position of slant plate 50 to a smaller angle, the suction chamber pressure P_s quickly increases because the newly decreased displacement of the compressor insufficiently compensates the heat load on the evaporator. However, this quick increase in the suction chamber pressure P_s hits a peak before the suction chamber pressure P_s reaches predetermined pressure P_4' at which the pressure differ-

ence between the crank and suction chambers 22 and 241 becomes ΔP_{min} . Thereafter, as long as the angular position of slant plate 50 is maintained at another angle, the suction chamber pressure P_s gradually decreases while the crank chamber pressure P_c gradually increases so as to satisfy equation (3). The above-described operation is repeated while the heat load on the evaporator gradually decreases in accordance with time.

On the other hand, in yet another compressor operation situation where heat load on the evaporator gradually increases in accordance with time, which is indicated by the period "c" in Figure 4, as long as the angular position of slant plate 50 is maintained at one angle, the suction chamber pressure P_s gradually increases while the crank chamber pressure P_c gradually decreases so as to satisfy equation (3). However, once the suction chamber pressure P_s reaches predetermined pressure P_4' , the angular position of slant plate 50 shifts from one angle to another angle which is greater than the first angle. Therefore, the displacement of the compressor shifts from one value to another value which is greater than the first value. When the displacement of the compressor shifts to the greater value due to the change in the angular position of slant plate 50 to a greater angle, the suction chamber pressure P_s quickly decreases because the newly increased displacement of the compressor sufficiently compensates the heat load on the evaporator. However, this quick decrease in the suction chamber pressure P_s bottoms out before the suction chamber pressure P_s reaches predetermined pressure P_5' . Thereafter, as long as the angular position of slant plate 50 is maintained at one angle, the suction chamber pressure P_s gradually increases while the crank chamber pressure P_c gradually decreases so as to satisfy equation (3). The above-described operation is repeated while the heat load on the evaporator gradually increases in accordance with time.

Accordingly, during a capacity control stage of operation, which includes time periods "b" and "c" shown in Figure 4, the suction chamber pressure P_s varies in a range $\Delta P_s' = P_4' - P_5'$ while the crank chamber pressure P_c varies in a range $\Delta P_c' = P_2' - P_3'$. Furthermore, the range of variation $\Delta P_s'$ in the suction chamber pressure is equal to the range of variation $\Delta P_c'$ in the crank chamber pressure because the suction chamber pressure P_s decreases in inverse proportion to the increase in the crank chamber pressure P_c at a proportion of one to one. Therefore, the range of variation $\Delta P_s'$ in the suction chamber pressure during the capacity control stage is not negligible. Accordingly, when the prior art compressor is used in an automotive air conditioning system, the temperature of cooled air which leaves the evaporator varies over

a range which is not negligible so that the air conditioning in a passenger compartment of an automobile is not effectively and efficiently controlled.

Accordingly, it is an object of this invention to provide a slant plate type refrigerant compressor having a capacity control mechanism which can sufficiently reduce the range of variation in the suction chamber pressure during a capacity control stage of operation.

In order to obtain the above object the present invention provides a slant plate type refrigerant compressor as defined in claim 1 and a slant plate control mechanism for use in such a refrigerant compressor as defined in claim 4.

Figure 1 is a vertical longitudinal sectional view of a conventional wobble plate type refrigerant compressor with a variable displacement mechanism.

Figure 2 is an enlarged sectional view of a valve control mechanism shown in Figure 1.

Figure 3 is a graph showing the relationship between the pressures in a crank chamber and a suction chamber of the wobble plate type refrigerant compressor shown in Figure 1.

Figure 4 is a graph showing the relationship between the elapsed time and the pressures in the crank chamber and the suction chamber of the wobble plate type refrigerant compressor shown in Figure 1.

Figure 5 is an enlarged sectional view of a valve control mechanism provided in a wobble plate type refrigerant compressor with a variable displacement mechanism in accordance with one embodiment of the present invention.

Figure 6 is a graph showing the relationship between the pressures in a crank chamber and a suction chamber of the wobble plate type refrigerant compressor shown in Figure 5.

Figure 7 is a graph showing the relationship between the elapsed time and the pressures in the crank chamber and the suction chamber of the wobble plate type refrigerant compressor shown in Figure 5.

Figure 5 illustrates a construction of valve control mechanism 19 provided in a wobble plate type refrigerant compressor with a variable displacement mechanism in accordance with one embodiment of the present invention. In the drawing, the same numerals are used to denote the same elements shown in Figures 1 and 2. Furthermore, for purposes of explanation only, the left side of the Figure will be referred to as the forward end or front and the right side of the Figure will be referred to as the rearward end.

With reference to Figure 5, valve control mechanism 19 includes valve device 293 having a longitudinally expandable and contractable bellows

193a and valve element 293b attached at a rearward end of bellows 193a. Bellows 193a longitudinally contracts and expands in response to crank chamber pressure. Bellows 193a is made of an elastic material, for example, phosphor bronze and has an effective pressure receiving cross-sectional area which is designated below as area A_1 . Valve element 293b has a generally truncated cone shape and is attached at the rearward end of bellows 193a. Projection member 193c, which is attached at a forward end of bellows 193a, is secured to axial projection 19c formed at the center of the closed end of casing member 191. Bias spring 193d is longitudinally and compressedly disposed within an inner hollow space of bellows 193a. The resultant force F of the restoring forces of bellows 193a and bias spring 193d continuously urges valve element 293b rearwardly (to the right in Figure 5).

When bellows 193a expands to a certain longitudinal length, generally truncated cone-shaped valve element 293b is received by conical-shaped opening 194b to form a circular line contact 293e therebetween. Circular line contact 293e divides valve element 293b into front portion 293f and rear portion 293g, an exterior surface of which is responsive to pressure in suction chamber 241 conducted via radial hole 151, conduit 152 and hole 153. Rear portion 293g of valve element 293b has an effective pressure receiving cross-sectional area which is designated below as area A_2 , and which is approximately 80% of the effective pressure receiving cross-sectional area A_1 of bellows 193a.

With reference to Figures 6 and 7, and as particularly illustrated in Figure 7, in a situation where operation of the compressor is stopped, the suction chamber pressure P_s and the crank chamber pressure P_c are in a state of equilibration, i.e., $P_c = P_s$, which is greater than the operating point P_1 of valve device 293. This causes the contraction of bellows 193a so that valve element 293b permits communication between suction chamber 241 and valve chamber 192 through conical-shaped opening 194b, radial hole 151, conduit 152 and hole 153 to thereby establish communication between crank chamber 22 and suction chamber 241.

In one compressor operational situation indicated by time period "a" in Figure 7, which is a so-called cool down stage, the compressor operates as follows. In the beginning of operation of the compressor, the communication between crank chamber 22 and suction chamber 241 is maintained, thereby satisfying the equation $P_c = P_s$ as shown by the straight line "li" in Figure 6 until the suction chamber pressure P_s falls to the operating point P_1 of valve device 293. When the suction chamber pressure P_s falls to the operating point P_1 of valve device 293, valve element 293b contacts

an inner surface of conical-shaped opening 194b due to expansion of bellows 193a. If the suction chamber pressure P_s drops below the operating point P_1 of valve device 293, valve element 293b frequently opens and closes conical-shaped opening 194b in accordance with the following equation:

$$F = (A_1 - A_2)P_c + A_2 \cdot P_s \quad (1)$$

wherein F is the resultant force of the restoring forces of bellows 193a and bias spring 193d, A_1 is the effective pressure receiving cross-sectional area of bellows 193a, A_2 is the effective pressure receiving cross-sectional area of rear portion 293g of valve element 293b, P_s is the pressure in suction chamber 241, and P_c is the pressure in crank chamber 22. The above equation (1) can be converted into the following equation by solving for P_c :

$$P_c = A_2 \cdot P_s / (A_2 - A_1) + F / (A_1 - A_2) \quad (2)$$

Equation (2) shows that the crank chamber pressure P_c varies in accordance with the changes in the suction chamber pressure P_s . Furthermore, in this valve control mechanism, A_2 is $0.8A_1$ so that equation (2) can be further converted to the following equation by substituting $0.8A_1$ for A_2 .

$$P_c = -4P_s + 5F/A_1 \quad (4)$$

Equation (4) is shown by the straight line "mi" in Figure 6. Therefore, the suction chamber pressure P_s decreases in inverse proportion to the increase in the crank chamber pressure P_c with a proportion of one to four when the suction chamber pressure P_s is less than the operating point P_1 of valve device 293. At that time, the angular position of slant plate 50 is maintained at the maximum slant angle. However, as illustrated in Figure 7, once the suction chamber pressure P_s reaches third predetermined pressure P_5 at which the pressure difference between the crank and suction chambers 22 and 241 becomes ΔP_{max} , the angular position of slant plate 50 shifts to an angle which is smaller than the maximum slant angle. Therefore, the displacement of the compressor shifts to a value which is smaller than its maximum value.

Another compressor operational situation where the heat load on the evaporator gradually decreases is depicted by time period "b" in Figure 7. As long as far as the angular position of slant plate 50 is maintained at one angle, the suction chamber pressure P_s gradually decreases while the crank chamber pressure P_c quickly increases so as to satisfy equation (4). However, once the suction chamber pressure P_s reaches predetermined pressure P_5 , the angular position of slant plate 50 shifts from one angle to another angle which is smaller

than the first angle. Therefore, the displacement of the compressor shifts from one value to another value which is smaller than the first value. When the displacement of the compressor shifts to the smaller value due to the change in the angular position of slant plate 50 to a smaller angle, the suction chamber pressure P_s quickly increases because the newly decreased displacement of the compressor insufficiently compensates the heat load on the evaporator. However, this quick increase in the suction chamber pressure P_s hits a peak before the suction chamber pressure P_s reaches predetermined pressure P_4 at which the pressure difference between the crank and suction chambers 22 and 241 becomes ΔP_{min} . Thereafter, as long as the angular position of slant plate 50 is maintained at another angle, the suction chamber pressure P_s gradually decreases while the crank chamber pressure P_c quickly increases so as to satisfy equation (4). The above-described operation is repeated while the heat load on the evaporator gradually decreases in accordance with time.

On the other hand, in yet another compressor operation situation where heat load on the evaporator gradually increases in accordance with time, which is indicated by the period "c" in Figure 7, as long as the angular position of slant plate 50 is maintained at one angle, the suction chamber pressure P_s gradually increases while the crank chamber pressure P_c quickly decreases so as to satisfy equation (4). However, once the suction chamber pressure P_s reaches predetermined pressure P_4 , the angular position of slant plate 50 shifts from one angle to another angle which is greater than the first angle. Therefore, the displacement of the compressor shifts from one value to another value which is greater than the first value. When the displacement of the compressor shifts to the greater value due to the change in the angular position of slant plate 50 to a greater angle, the suction chamber pressure P_s quickly decreases because the newly increased displacement of the compressor sufficiently compensates the heat load on the evaporator. However, this quick decrease in the suction chamber pressure P_s bottoms out before the suction chamber pressure P_s reaches predetermined pressure P_5 . Thereafter, as long as the angular position of slant plate 50 is maintained at one angle, the suction chamber pressure P_s gradually increases while the crank chamber pressure P_c quickly decreases so as to satisfy equation (4). The above-described operation is repeated while the heat load on the evaporator gradually increases in accordance with time.

Accordingly, during a capacity control stage of operation, which includes time periods "b" and "c" shown in Figure 7, in the compressor of the preferred embodiment, the suction chamber pressure

P_s varies in a range $\Delta P_s = P_4 - P_5$ while the crank chamber pressure P_c varies in a range $\Delta P_c = P_2 - P_3$. Furthermore, the range of variation ΔP_s in the suction chamber pressure is one-fourth the range of variation ΔP_c in the crank chamber pressure because the suction chamber pressure P_s decreases in inverse proportion to the increase in the crank chamber pressure P_c with a proportion of one to four. For example, experimental data comparing conventional compressors and the compressor of the present invention shows that the range of variation in the suction chamber pressure during the capacity control stage decreases from 0.26 to 0.1 kgf/cm²G. Therefore, in the compressor of the present invention, the range of variation in the suction chamber pressure during the capacity control stage can be effectively decreased by a significant amount as compared with conventional compressors. Accordingly, when the present invention compressor is used in an automotive air conditioning system, the temperature of cooled air which leaves the evaporator varies over a range which is negligible so that the air conditioning in a passenger compartment of an automobile can be effectively and efficiently controlled.

Claims

1. An adjustable slant plate type refrigerant compressor comprising:
 - a compressor housing (20) provided with a plurality of cylinders (70), a suction chamber (241), a discharge chamber (251) and an enclosed crank chamber (22);
 - a piston (71) slidably fitted within each of said cylinders;
 - a drive mechanism including a rotor (40);
 - an adjustable slant plate (50) having an inclined surface adjustably connected to said rotor and having an adjustable slant angle, the slant angle changing in response to a change in pressure in said crank chamber to thereby change the capacity of said compressor;
 - coupling means (60, 72) for operationally coupling said slant plate to said pistons such that rotation of said rotor and slant plate reciprocates said pistons in said cylinders;
 - a passageway (150) in said compressor housing linking said suction chamber with said crank chamber; and
 - a valve control means (19') for controlling the opening and closing of said passageway, said valve control means including a bellows (193a) having a first effective pressure receiving cross-sectional area responsive to crank chamber pressure and a valve element (293b) attached at one end of said bellows to open and close said passageway, said passageway



than the first angle. Therefore, the displacement of the compressor shifts from one value to another value which is smaller than the first value. When the displacement of the compressor shifts to the smaller value due to the change in the angular position of slant plate 50 to a smaller angle, the suction chamber pressure P_s quickly increases because the newly decreased displacement of the compressor insufficiently compensates the heat load on the evaporator. However, this quick increase in the suction chamber pressure P_s hits a peak before the suction chamber pressure P_s reaches predetermined pressure P_4 at which the pressure difference between the crank and suction chambers 22 and 241 becomes ΔP_{min} . Thereafter, as long as the angular position of slant plate 50 is maintained at another angle, the suction chamber pressure P_s gradually decreases while the crank chamber pressure P_c quickly increases so as to satisfy equation (4). The above-described operation is repeated while the heat load on the evaporator gradually decreases in accordance with time.

On the other hand, in yet another compressor operation situation where heat load on the evaporator gradually increases in accordance with time, which is indicated by the period "c" in Figure 7, as long as the angular position of slant plate 50 is maintained at one angle, the suction chamber pressure P_s gradually increases while the crank chamber pressure P_c quickly decreases so as to satisfy equation (4). However, once the suction chamber pressure P_s reaches predetermined pressure P_4 , the angular position of slant plate 50 shifts from one angle to another angle which is greater than the first angle. Therefore, the displacement of the compressor shifts from one value to another value which is greater than the first value. When the displacement of the compressor shifts to the greater value due to the change in the angular position of slant plate 50 to a greater angle, the suction chamber pressure P_s quickly decreases because the newly increased displacement of the compressor sufficiently compensates the heat load on the evaporator. However, this quick decrease in the suction chamber pressure P_s bottoms out before the suction chamber pressure P_s reaches predetermined pressure P_5 . Thereafter, as long as the angular position of slant plate 50 is maintained at one angle, the suction chamber pressure P_s gradually increases while the crank chamber pressure P_c quickly decreases so as to satisfy equation (4). The above-described operation is repeated while the heat load on the evaporator gradually increases in accordance with time.

Accordingly, during a capacity control stage of operation, which includes time periods "b" and "c" shown in Figure 7, in the compressor of the preferred embodiment, the suction chamber pressure

P_s varies in a range $\Delta P_s = P_4 - P_5$ while the crank chamber pressure P_c varies in a range $\Delta P_c = P_2 - P_3$. Furthermore, the range of variation ΔP_s in the suction chamber pressure is one-fourth the range of variation ΔP_c in the crank chamber pressure because the suction chamber pressure P_s decreases in inverse proportion to the increase in the crank chamber pressure P_c with a proportion of one to four. For example, experimental data comparing conventional compressors and the compressor of the present invention shows that the range of variation in the suction chamber pressure during the capacity control stage decreases from 0.26 to 0.1 kgf/cm²G. Therefore, in the compressor of the present invention, the range of variation in the suction chamber pressure during the capacity control stage can be effectively decreased by a significant amount as compared with conventional compressors. Accordingly, when the present invention compressor is used in an automotive air conditioning system, the temperature of cooled air which leaves the evaporator varies over a range which is negligible so that the air conditioning in a passenger compartment of an automobile can be effectively and efficiently controlled.

Claims

1. An adjustable slant plate type refrigerant compressor comprising:
 - a compressor housing (20) provided with a plurality of cylinders (70), a suction chamber (241), a discharge chamber (251) and an enclosed crank chamber (22);
 - a piston (71) slidably fitted within each of said cylinders;
 - a drive mechanism including a rotor (40);
 - an adjustable slant plate (50) having an inclined surface adjustably connected to said rotor and having an adjustable slant angle, the slant angle changing in response to a change in pressure in said crank chamber to thereby change the capacity of said compressor;
 - coupling means (60, 72) for operationally coupling said slant plate to said pistons such that rotation of said rotor and slant plate reciprocates said pistons in said cylinders;
 - a passageway (150) in said compressor housing linking said suction chamber with said crank chamber; and
 - a valve control means (19') for controlling the opening and closing of said passageway, said valve control means including a bellows (193a) having a first effective pressure receiving cross-sectional area responsive to crank chamber pressure and a valve element (293b) attached at one end of said bellows to open and close said passageway, said passageway



- including a valve seat (194b) formed therein for receiving said valve element, said valve element including a boundary line (293e) which is defined at an exterior surface of said valve element when said valve element is received in said valve seat, said boundary line dividing said valve element into first (293g) and second (293f) portions; said first portion having an exterior surface responsive to pressure in said suction chamber when said valve element is received in said valve seat, characterized in that said first portion of said valve element has a second effective pressure receiving cross-sectional area which is approximately equal to or greater than 80 % of said first effective pressure receiving cross-sectional area.
2. The compressor according to claim 1, characterized by a spring member (193d) within said bellows for urging said valve element towards said seat.
 3. The compressor according to claims 1 or 2, characterized in that said valve element is frusto-conical and engages said seat along a circular line (293e).
 4. A slant plate control mechanism controlling the angular position of an adjustable slant plate (50) in a slant plate refrigerant compressor in response to crank chamber pressure, said compressor including a compressor housing (20) defining a crank chamber (22) and a suction chamber (241), said slant plate control mechanism comprising:
 - a passageway (151) connecting said crank and suction chambers;
 - a valve seat (194b) encircling said passageway;
 - a valve element (293b) engageable with said valve seat to close said passageway, a boundary (293e) between said valve element and said passageway defining a second effective pressure area on said valve element when said valve element engages said valve seat; and
 - a bellows (193a) connected with said valve element for moving said valve element into engagement with said valve seat, the cross-sectional area of said bellows defining a first effective pressure area, characterized in that said second effective pressure area on said valve element is approximately 80% or more of the first effective pressure area on said bellows.
 5. The slant plate control mechanism according to claim 4, characterized by a spring member

(193d) within said bellows for urging said valve element towards said seat.

6. The slant plate control mechanism of claim 4 or 5, characterized in that said valve element is frusto-conical and engages said seat along a circular line (293e).

Patentansprüche

1. Einstellbarer Schiefscheibenkühlkompressor mit:
 - einem Kompressorgehäuse (20), das mit einer Mehrzahl von Zylindern (70), einer Ansaugkammer (241), einer Auslaßkammer (251) und einer eingeschlossenen Kurbelkammer (22) versehen ist;
 - einem Kolben (71), der in jeden der Zylinder gleitend verschiebbar eingepaßt ist;
 - einem Antriebsmechanismus mit einem Rotor (40);
 - einer einstellbaren Schiefscheibe (50) mit einer geneigten Oberfläche, die einstellbar mit dem Rotor verbunden ist und einen einstellbaren Neigungswinkel aufweist, wobei sich der Neigungswinkel als Reaktion auf eine Änderung im Druck in der Kurbelkammer zum dadurch Ändern der Kapazität des Kompressors ändert;
 - Verbindungsmittel (60, 72) zum betriebsmäßigen Verbinden der Schiefscheibe mit den Kolben derart, daß die Rotation des Rotors und der Schiefscheibe die Kolben in den Zylindern hin- und herbewegt;
 - einem Durchgang (150) in dem Kompressorgehäuse, der die Ansaugkammer mit der Kurbelkammer verbindet; und
 - Ventilsteuermittel (19') zum Steuern des Öffnens und Schließens des Durchganges, wobei das Ventilsteuermittel einen Balgen 193a mit einer ersten effektiven Querschnittsfläche zum Druckaufnehmen, der auf den Kurbelkammerdruck reagiert, und ein Ventilelement (293b), das an einem Ende des Balgens angebracht ist, zum Öffnen und Schließen des Durchganges aufweist, der Durchgang einen Ventilsitz (194b) aufweist, der darin zum Aufnehmen des Ventilelementes gebildet ist, das Ventilelement eine Grenzlinie aufweist, die auf einer äußeren Oberfläche des Ventilelementes definiert ist, wenn das Ventilelement in dem Ventilsitz aufgenommen ist, die Grenzlinie das Ventilelement in einen ersten (293g) und einen zweiten (293f) Abschnitt unterteilt, der erste Abschnitt eine äußere Oberfläche aufweist, die auf den Druck in der Ansaugkammer reagiert, wenn das Ventilelement in dem Ventilsitz aufgenommen ist;
- dadurch gekennzeichnet, daß der erste Ab-



- schnitt des Ventilelementes eine zweite effektive Querschnittsfläche zum Druckaufnehmen aufweist, die ungefähr gleich oder größer als 80% der ersten effektiven Querschnittsfläche zum Druckaufnehmen ist.
2. Kompressor nach Anspruch 1, gekennzeichnet durch ein Federteil (193d) innerhalb des Balgens zum Drücken des Ventilelementes zu dem Sitz.
 3. Kompressor nach Anspruch 1 oder 2, dadurch gekennzeichnet, daß das Ventilelement kegelförmig ist und den Sitz entlang einer kreisförmigen Linie (293e) angreift.
 4. Schiefscheibensteuermechanismus, der die Winkelposition einer einstellbaren Schiefscheibe (50) in einem Schiefscheibenkühlkompressor als Reaktion auf Kurbelkammerdruck steuert, wobei der Kompressor ein Kompressorgehäuse (20) aufweist, das eine Kurbelkammer (22) und eine Ansaugkammer (241) definiert, der Schiefscheibensteuermechanismus aufweist:
 - einen Durchgang (151), der die Kurbel- und Ansaugkammer verbindet;
 - einen Ventilsitz (194b), der den Durchgang umschließt;
 - ein Ventilelement (293b), das in Eingriff mit dem Ventilsitz zum Schließen des Durchganges bringbar ist, wobei eine Grenze (293e) zwischen dem Ventilelement und dem Durchgang eine zweite effektive Druckfläche auf dem Ventilelement definiert, wenn das Ventilelement in Eingriff mit dem Ventilsitz steht; und
 - einen Balgen (193a), der mit dem Ventilelement zum Bewegen des Ventilelementes in Eingriff mit dem Ventilsitz verbunden ist, wobei die Querschnittsfläche des Balgens eine erste effektive Druckfläche definiert, dadurch gekennzeichnet, daß die zweite effektive Druckfläche auf dem Ventilelement ungefähr 80% oder mehr der ersten effektiven Druckfläche auf dem Balgen ist.
 5. Schiefscheibensteuermechanismus nach Anspruch 4, gekennzeichnet durch ein Federteil (193d) innerhalb des Balgens zum Drücken des Ventilelementes zu dem Sitz.
 6. Schiefscheibensteuermechanismus nach Anspruch 4 oder 5, dadurch gekennzeichnet, daß das Ventilelement kegelförmig ist und mit dem Sitz entlang einer kreisförmigen Linie (293e) in Eingriff steht.

Revendications

1. Compresseur de réfrigérant du type à plaque inclinée réglable comprenant :
 - un boîtier de compresseur (20) qui comporte une pluralité de cylindres (70), une chambre d'aspiration (241), une chambre de refoulement (251) et une chambre à manivelle (22) fermée,
 - un piston (71) monté en coulissement à l'intérieur de chacun desdits cylindres ;
 - un mécanisme d'entraînement comprenant un rotor (40) ;
 - une plaque inclinée réglable (50) ayant une surface inclinée reliée de manière réglable audit rotor et ayant un angle d'inclinaison réglable, l'angle d'inclinaison variant en réponse à un changement de pression dans ladite chambre à manivelle afin de modifier ainsi la capacité dudit compresseur ;
 - des moyens d'accouplement (60, 72) pour accoupler de manière fonctionnelle ladite plaque inclinée auxdits pistons de manière que la rotation dudit rotor et de la plaque inclinée impose auxdits pistons un mouvement alternatif à l'intérieur desdits cylindres ;
 - un passage (150) dans ledit boîtier de compresseur reliant ladite chambre d'aspiration avec ladite chambre à manivelle ; et
 - un moyen (19) de commande de soupape pour commander l'ouverture et la fermeture dudit passage, ledit moyen de commande de soupape comprenant un soufflet (193a) ayant une première surface utile de section transversale soumise à la pression répondant à la pression dans la chambre à manivelle et un élément de soupape (293b) relié à une extrémité dudit soufflet pour ouvrir et fermer ledit passage, ledit passage contenant un siège de soupape (194b) formé à l'intérieur pour recevoir ledit élément de soupape, ledit élément de soupape comprenant une ligne de limite (293e) qui est définie sur une surface extérieure dudit élément de soupape lorsque ledit élément de soupape est reçu à l'intérieur dudit siège de soupape, ladite ligne limite partageant ledit élément de soupape en une première (293g) et une seconde (293f) parties, ladite première partie ayant une surface extérieure sensible à la pression dans ladite chambre d'aspiration quand ledit élément de soupape est reçu dans ledit siège de soupape,
 - caractérisé en ce que ladite première partie dudit élément de soupape comporte une seconde surface utile de section transversale soumise à la pression qui est sensiblement égale ou supérieure à 80 % de ladite première surface utile de section transversale soumise à



la pression.

2. Compresseur selon la revendication 1, caractérisé par un élément de ressort (193d) placé dans ledit soufflet pour solliciter ledit élément de soupape vers ledit siège. 5
3. Compresseur selon la revendication 1 ou 2, caractérisé en ce que ledit élément de soupape est tronconique et vient en prise avec ledit siège le long d'une ligne circulaire (293e). 10
4. Mécanisme de commande de plaque inclinée commandant la position angulaire d'une plaque inclinée réglable (50) dans un compresseur de réfrigérant à plaque inclinée en réponse à la pression dans la chambre à manivelle, ledit compresseur comprenant un boîtier de compresseur (20) définissant une chambre (22) à manivelle et une chambre d'aspiration (241), ledit mécanisme de commande de plaque inclinée comprenant : 15
un passage (151) reliant ladite chambre à manivelle et ladite chambre d'aspiration ;
un siège de soupape (194b) entourant ledit passage ; 20
un élément de soupape (293b) susceptible de venir en prise avec ledit siège de soupape pour fermer ledit passage, une limite (293e) entre ledit élément de soupape et ledit passage définissant une seconde surface utile sensible à la pression sur ledit élément de soupape lorsque ledit élément de soupape vient en prise avec ledit siège de soupape ; et 25
un soufflet (193a) relié audit élément de soupape pour déplacer ledit élément de soupape en l'amenant en prise avec ledit siège de soupape, la surface de la section transversale dudit soufflet définissant une première surface sensible à la pression, 30
caractérisé en ce que ladite seconde surface utile sensible à la pression sur ledit élément de soupape est sensiblement égale à 80 % ou plus de la première surface utile sensible à la pression dudit soufflet. 35
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5. Mécanisme de commande de plaque inclinée selon la revendication 4, caractérisé par un élément de ressort (193d) placé à l'intérieur dudit soufflet pour solliciter ledit élément de soupape vers ledit siège. 50
6. Mécanisme de commande de plaque inclinée selon la revendication 4 ou 5, caractérisé en ce que ledit élément de soupape est tronconique et vient en prise avec ledit siège le long d'une ligne circulaire (293e). 55