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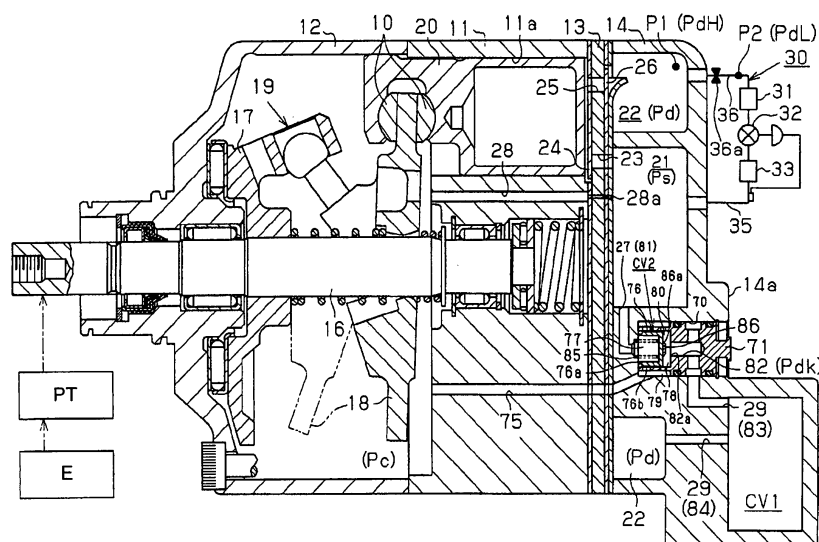
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(54) **Displacement control mechanism of a variable displacement type compressor**

(57) A displacement control mechanism controls displacement of a variable displacement type compressor that forms a refrigerant circulation circuit for an air conditioning apparatus. The displacement is decreased as a pressure in a crank chamber rises while being increased as the pressure in the crank chamber falls. The displacement control mechanism includes a bleed passage, a supply passage, a first control valve and a sec-

ond control valve. The second control valve includes a back pressure chamber, a valve chamber, a valve body, a spring and a second valve portion. The second valve portion is provided with the back surface of the valve body. The second valve portion closes an opening of an introduction passage in the back pressure chamber when the first valve portion maximizes the opening of the bleed passage.

FIG. 1



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Description

BACKGROUND OF THE INVENTION

[0001] The present invention relates to a displacement control mechanism for controlling the displacement of a variable displacement type compressor that forms a part of refrigerant circulation circuit of an air conditioning apparatus and the displacement of which is decreased as a pressure in a crank chamber of the compressor rises while being increased as the pressure in the crank chamber falls.

[0002] There is known a displacement control mechanism shown in FIG. 7, in which the pressure in a crank chamber 153 or crank pressure P_c is adjusted by technique, what is called, a supply control.

[0003] Namely, in a variable displacement type swash plate compressor (hereinafter the compressor), the crank chamber 153 communicates with a suction chamber 155 via a bleed passage 154. A discharge chamber 151 of the compressor communicates with the crank chamber 153 via a supply passage 152 in which a control valve 156 is arranged. The amount of refrigerant gas introduced into the crank chamber 153 via the supply passage 152 is controlled by adjusting the opening of the control valve 156, and the crank pressure P_c is determined in accordance with the relation between the amounts of refrigerant gas introduced into and bleeding from the crank chamber 153.

[0004] A fixed throttle 158 is arranged in the bleed passage 154 so that the refrigerant gas bleeds slowly from the crank chamber 153 to the suction chamber 155. Thus, even when the amount of the refrigerant gas supplied from the discharge chamber 151 to the crank chamber 153 via the supply passage 152 is small, the crank pressure P_c is steadily increased. Therefore, when the control valve 156 increases the opening of the supply passage 152, the crank pressure P_c is rapidly increased. Consequently, appropriate response in decreasing the compressor displacement is obtained.

[0005] Also, an amount of gas that blows from a cylinder bore 157 to the crank chamber 153 and that leaks to the suction chamber 155 via the bleed passage 154, and an amount of the refrigerant gas that moves from the discharge chamber 151 to the suction chamber 155 via the crank chamber 153 as mentioned above, so-called, a kind of internal leakage, are reduced as much as possible by the provision of the fixed throttle 158. Consequently, decrease in efficiency of the compressor caused by providing the displacement control mechanism is prevented.

[0006] However, the arrangement of the fixed throttle 158 on the bleed passage 154 makes decrease in a pressure in the crank chamber 153 slow. In other words, response in increasing the displacement of the compressor deteriorates. Especially, when the compressor is started, the crank pressure P_c tends to be excessively increased since the liquid refrigerant accumulated in the

crank chamber 153 evaporates and the fixed throttle 158 hampers smooth flow of the refrigerant gas from the crank chamber 153. Therefore, even when the control valve 156 closes the supply passage 152 so as to increase the displacement of the compressor in response to the requirement for cooling shortly after the compressor is started, it takes time before the displacement of the compressor is actually increased, and starting performance of an air conditioning apparatus deteriorates.

[0007] To solve such problems, it is proposed to provide a second control valve 161 for controlling the opening of the bleed passage 154 in addition to the control valve (first control valve) 156, as shown in FIG. 8. Please see Japanese Unexamined Patent Publication No. 2002-21721 (pages 7 to 10, and Figures 1, 4 and 5).

[0008] Specifically, in the proposed structure, a region K is provided in the supply passage 152 downstream of the position of the first control valve 156 (i.e. the position of the valve opening adjustment) and upstream of a fixed throttle 169, as shown in FIG. 8. The second control valve 161 is a spool type valve that includes a spool 162 and a back pressure chamber 166 into which the pressure in the region K is introduced. A valve chamber 167 of the second control valve 161 forms a part of the bleed passage 154 and communicates with the suction chamber 155. The valve chamber 167 also communicates with the crank chamber 153 via a valve hole 168 that forms the upstream portion of the bleed passage 154.

[0009] The spool 162 is movably fitted in a spool supporting recess 164 that is formed in a compressor housing. The spool 162 includes a valve portion 162a that is located in the valve chamber 167 and a back surface 162b that is located in the back pressure chamber 166. The spool 162 or the valve portion 162a is positioned by various forces applied thereto such as urging force of the pressure in the back pressure chamber 166 acting on the back surface 162b in the direction to close the valve, urging force of a spring 165 acting in the valve opening direction and force of the crank pressure P_c that is applied in the valve opening direction.

[0010] When the first control valve 156 closes the supply passage 152, a pressure P_{dK} in the back pressure chamber 166 of the second control valve 161 becomes substantially the same as the crank pressure P_c and, therefore, the spool 162 of the second control valve 161 is positioned by the spring 165 where the opening of the valve hole 168 is maximum. When the bleed passage 154 is widely opened by the second control valve 161, flowing of the refrigerant from the crank chamber 153 to the suction chamber 155 is prompted. Therefore, when the first control valve 156 closes the supply passage 152 so as to increase the displacement of the compressor shortly after the compressor is started, the displacement of the compressor is immediately increased, so that the starting performance of the air conditioning apparatus is improved.

[0011] A spring having a small urging force is utilized

as the urging spring 165. Thus, when the supply passage 152 is opened even slightly by the first control valve 156 and the pressure P_{dk} in the region K exceeds the crank pressure P_c , the spool 162 moves against the urging spring 165, and the valve portion 162a minimizes the opening of the valve hole 168 that is not zero. Therefore, when the valve hole 168 is thus set at the minimum opening that is not zero, the second control valve 161 functions similarly to the above-described fixed throttle 158 shown in FIG. 7, and the decrease in the efficiency of the compressor caused by providing the displacement control mechanism is prevented.

[0012] However, the first control valve 156 leaks the refrigerant gas by performance deterioration due to aged deterioration even in a state that the first control valve 156 closes the supply passage 152. Thus, the pressure P_{dk} in the back pressure chamber 166 of the second control valve 161 rises due to the refrigerant gas which leaks from the first control valve 156, and the second control valve 161 may inappropriately set the opening of the bleed passage 154 at the minimum opening. Therefore, the refrigerant gas is flowed slowly from the crank chamber 153 to the suction chamber 155 through the bleed passage 154, and the starting performance of the air conditioning apparatus is insufficient.

[0013] To solve such a problem, a spring having large urging force is adopted as the urging spring 165 so that the spool 162 or the valve portion 162a maintains the maximum opening of the valve hole 168 even if the pressure P_{dk} in the back pressure chamber 166 is raised somewhat.

[0014] However, when the spring having large urging force is adopted as the urging spring 165, the second control valve 161 cannot set the bleed passage 154 at the minimum opening unless the first control valve 156 widely opens the supply passage 152 and the pressure P_{dk} in the back pressure chamber 166 is greatly raised. Therefore, in a state that the first control valve 156 opens the supply passage 152, such period that the second control valve 161 sets the bleed passage 154 at an opening other than the minimum opening, in other words, such period that the second control valve 161 cannot function similarly to the fixed throttle 158 increases, and decrease in the efficiency of the compressor is caused.

SUMMARY OF THE INVENTION

[0015] The present invention is directed to a displacement control mechanism that prevents a second control valve from inappropriately operating even when performance of a first control valve deteriorates while preventing decrease in efficiency of a variable displacement type compressor.

[0016] According to the present invention, a displacement control mechanism controls displacement of a variable displacement type compressor that forms a refrigerant circulation circuit for an air conditioning apparatus.

The displacement is decreased as a pressure in a crank chamber rises while being increased as the pressure in the crank chamber falls. The refrigerant circulation circuit has a suction pressure region and a discharge pressure region. The displacement control mechanism includes a bleed passage, a supply passage, a first control valve and a second control valve. The bleed passage interconnects the crank chamber with the suction pressure region. The supply passage interconnects the crank chamber with the discharge pressure region. The first control valve is located on the supply passage for adjusting an opening of the supply passage at a position of valve opening adjustment. The second control valve includes a back pressure chamber, a valve chamber, a valve body, a spring and a second valve portion. A pressure on a downstream side of the position of valve opening adjustment of the first control valve in the supply passage is introduced to the back pressure chamber through an introduction passage. The valve chamber forms a part of the bleed passage. The valve body has a first valve portion located in the valve chamber and a back surface located in the back pressure chamber. The first valve portion decreases an opening of the bleed passage as a pressure in the back pressure chamber which is applied to the back surface rises. The spring urges the valve body so that the first valve portion increases the opening of the bleed passage. The second valve portion is provided with the back surface of the valve body. The second valve portion closes an opening of the introduction passage in the back pressure chamber when the first valve portion maximizes the opening of the bleed passage.

[0017] Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

[0018] The features of the present invention that are believed to be novel are set forth with particularity in the appended claims. The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

FIG. 1 is a longitudinal sectional view illustrating a variable displacement type swash plate compressor;

FIG. 2 is a longitudinal sectional view illustrating a first control valve;

FIG. 3 is a partially enlarged view illustrating a second control valve and its vicinity of FIG. 1;

FIG. 4 is a longitudinal sectional view illustrating operation of the second control valve;

FIG. 5 is an enlarged longitudinal sectional view illustrating another second control valve and its vicinity;

FIG. 6 is an enlarged longitudinal sectional view illustrating yet another second control valve and its vicinity;

FIG. 7 is a schematic view illustrating a prior art displacement control mechanism; and

FIG. 8 is a longitudinal sectional view illustrating a prior art second control valve and its vicinity.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0019] The following will describe a preferred embodiment of the present invention. In the preferred embodiment, the present invention is applied to a variable displacement type swash plate compressor (hereinafter the compressor) that is used in a vehicle air conditioning apparatus for compressing refrigerant gas.

[0020] Referring to FIG. 1, the compressor includes a cylinder block 11, a front housing 12, a valve plate assembly 13 and a rear housing 14. In FIG. 1, the left side and the right side respectively correspond to the front side and the rear side of the compressor. The front housing 12 is fixedly joined to the front end of the cylinder block 11, and the rear housing 14 is fixedly joined to the rear end of the cylinder block 11 via the valve plate assembly 13. The cylinder block 11, the front housing 12 and the rear housing 14 cooperate to form a compressor housing.

[0021] A crank chamber 15 is defined by the cylinder block 11 and the front housing 12. A drive shaft 16 is rotatably supported in the crank chamber 15. A lug plate 17 is fixed to the drive shaft 16 so as to be integrally rotated with the drive shaft.

[0022] The front end of the drive shaft 16 is operatively connected to a vehicle engine E as an external drive source via a power transmission mechanism PT. The power transmission mechanism PT may be a clutch mechanism (e.g. an electromagnetic clutch) that selectively transmits and blocks driving power according to electric control from an external device, or a continuous transmission type clutchless mechanism (e.g. the combination of a belt and a pulley) that dispenses with the above clutch mechanism. In the present preferred embodiment, the clutchless type power transmission mechanism PT is utilized.

[0023] A swash plate 18 as a cam plate is accommodated in the crank chamber 15. The swash plate 18 is slidably and inclinably supported by the drive shaft 16. A hinge mechanism 19 is interposed between the lug

plate 17 and the swash plate 18. Thus, a hinge connection between the lug plate 17 and the swash plate 18 via the hinge mechanism 19 and the support of the swash plate 18 by the drive shaft 16 allow the swash plate 18 to rotate synchronously with the lug plate 17 and the drive shaft 16 as well as to incline with respect to an axis of the drive shaft 16 in accordance with the sliding movement of the swash plate 18 in the axial direction of the drive shaft 16.

[0024] A plurality of cylinder bores 11a is formed in the cylinder block 11 extending axially through the cylinder block 11 and is arranged around the drive shaft 16. In FIG. 1, only one cylinder bore is shown. A single-head piston 20 is accommodated in each of the cylinder bores 11a for reciprocation therein. The front and rear openings of the cylinder bores 11a are respectively closed by the pistons 20 and the valve plate assembly 13. Compression chambers are defined in the cylinder bores 11a, and the volumes of the compression chambers are varied in accordance with the reciprocating movement of the pistons 20. Each of the pistons 20 is engaged with the periphery of the swash plate 18 via a pair of shoes 10, so that the rotation of the swash plate 18 with the drive shaft 16 is converted into linear reciprocating movement of the pistons 20.

[0025] A suction chamber 21 and a discharge chamber 22 are defined between the valve plate assembly 13 and the rear housing 14. The suction chamber 21 is located in the middle region of the rear housing 14 and is surrounded by the discharge chamber 22. A suction port 23 and a suction valve 24 are formed in the valve plate assembly 13 for each of the cylinder bores 11a. The suction valve 24 is adapted to open and close the suction port 23. A discharge port 25 and a discharge valve 26 are also formed in the valve plate assembly 13 for each of the cylinder bores 11a. The suction chamber 21 communicates with each of the cylinder bores 11a via the corresponding suction port 23, and each of the cylinder bores 11a communicates with the discharge chamber 22 via the corresponding discharge port 25.

[0026] As each of the pistons 20 moves from the top dead center toward the bottom dead center, the refrigerant gas is drawn into the corresponding cylinder bore 11a via the associated suction port 23 pushing away the associated suction valve 24. As the pistons 20 move from the bottom dead center toward the top dead center, the refrigerant gas introduced into the cylinder bore 11a is compressed to a predetermined pressure and is discharged into the discharge chamber 22 via the associated discharge port 25 pushing away the discharge valve 26.

[0027] An inclination angle of the swash plate 18, which is defined as an angle made between the swash plate 18 and a plane perpendicular to the axis of the drive shaft 16 is varied in accordance with the pressure in the crank chamber 15 (or a crank pressure P_c) between the minimum inclination angle as indicated by a solid line in FIG. 1 and the maximum inclination angle as in-

licated by a two-dot chain line in FIG. 1.

[0028] A displacement control mechanism for controlling the crank pressure P_c which has bearing on control of the inclination angle of the swash plate 18 includes a first bleed passage 27, a second bleed passage 28, a supply passage 29, a first control valve CV1 and a second control valve CV2.

[0029] The first and second bleed passages 27 and 28 interconnect the crank chamber 15 with the suction chamber 21 as a suction pressure (P_s) region. The second bleed passage 28 has a fixed throttle 28a and extends through the cylinder block 11 and the valve plate assembly 13. The supply passage 29 interconnects the discharge chamber 22 as a discharge pressure (P_d) region with the crank chamber 15. The first control valve CV1 is arranged in the supply passage 29 for adjusting the opening of the supply passage 29. It is noted that the first bleed passage 27 and the supply passage 29 are partially shared therebetween as will be later described.

[0030] The first control valve CV1 adjusts the opening of the supply passage 29 while the second control valve CV2 adjusts the opening of the supply passage 29 and the first bleed passage 27. By so doing, the balance between [an] the amount of high-pressure discharge gas introduced from the discharge chamber 22 into the crank chamber 5 via the supply passage 29 and [an] the amount of the refrigerant gas flowing from the crank chamber 5 into the suction chamber 21 via the first and second bleed passages 27 and 28 is controlled, and the crank pressure P_c is determined, accordingly. Pressure difference between the crank pressure P_c and the internal pressure in the cylinder bores 11 a via the pistons 20 is changed in accordance with the variation of the crank pressure P_c , and the inclination angle of the swash plate 12 is varied, accordingly. Consequently, the stroke of pistons 20, that is, the displacement of the compressor is adjusted.

[0031] For example, when the first control valve CV1 reduces the opening of the supply passage 29 and the crank pressure P_c is decreased, the inclination angle of the swash plate 18 is increased, and the displacement of the compressor is increased. On the other hand, when the first control valve CV1 increases the opening of the supply passage 29 and the crank pressure P_c is increased, the inclination angle of the swash plate 18 is decreased, and the displacement of the compressor is decreased. It is note that the minimum displacement of the compressor is set at zero or about zero

[0032] A refrigerant circulation circuit (or a refrigeration cycle) of the vehicle air conditioning apparatus includes the above-described compressor and an external refrigerant circuit 30. The external refrigerant circuit 30 includes a condenser 31, an expansion valve 32 and an evaporator 33. A circulation pipe 35 for the refrigerant is provided on the downstream side of the external refrigerant circuit 30, interconnecting the outlet of the evaporator 33 with the suction chamber 21 of the com-

pressor. A circulation pipe 36 for the refrigerant is provided on the upstream side of the external refrigerant circuit 30, interconnecting the discharge chamber 22 of the compressor with the inlet of the condenser 31. The compressor draws and compresses therein the refrigerant gas which is introduced from the downstream side of the external refrigerant circuit 30 into the suction chamber 21, and then discharges the compressed refrigerant gas to the discharge chamber 22 which interconnects with the upstream side of the external refrigerant circuit 30.

[0033] As shown in FIG. 2, the first control valve CV1 includes a valve portion in the upper half thereof as seen on the drawing of FIG. 2 and a solenoid portion 60 in the lower half. The valve portion adjusts the opening (a degree of throttle) of the supply passage 29 that interconnects the discharge chamber 22 with the crank chamber 15. The solenoid portion 60 is an actuator for controlling the operation of a valve rod 40 arranged in the control valve CV1 in response to a control signal from an external device. The valve rod 40 is a rod-like member [and] which includes a partition portion 41 at the top of the rod, a connection portion 42, a valve body portion 43 at the middle and a guide rod portion 44 at the base.

[0034] A valve housing 45 for the first control valve CV1 includes a valve body housing 45a [in] forming its upper part and an actuator housing 45b [in] forming its lower part. A valve accommodating chamber 46, a communication passage 47 and a pressure sensing chamber 48 are defined in the valve body housing 45a. The valve rod 40 is arranged in the valve accommodating chamber 46 and the communication passage 47 for axial movement, that is, movement in the vertical direction [of] as seen in FIG. 2. The partition portion 41 of the valve rod 40 is inserted [into] through the communication passage 47 thereby to shut off the communication between the pressure sensing chamber 48 [from] and the communication passage 47.

[0035] Ports 51 and 52 are formed through the peripheral wall of the valve body housing 45a. The port 51 communicates with the valve accommodating chamber 46, and the port 52 communicates with the communication passage 47, respectively. The valve accommodating chamber 46 communicates with the discharge chamber 22 of the compressor via the port 51 and the upstream part of the supply passage 29, or a passage 84. The communication passage 47 communicates with the crank chamber 15 of the compressor via the port 52, the downstream part of the supply passage 29 or a passage 83, the second control valve CV2 and a passage 75. The supply passage 29 includes the passage 84, the port 51, the valve accommodating chamber 46, the communication passage 47, the port 52, the passage 83, the second control valve CV2 and the passage 75.

[0036] The valve body portion 43 of the valve rod 40 is located in the valve accommodating chamber 46. A valve seat 53 is formed at the stepped portion located between the valve accommodating chamber 46 and the

communication passage 47, and the communication passage 47 functions as a valve hole. When the valve rod 40 moves upward from the position of FIG. 2, where the communication passage 47 (or the supply passage 29) is opened, to a position where the valve body portion 43 contacts the valve seat 53, the communication passage 47 (the supply passage 29) is closed.

[0037] A bellows 50 is accommodated in the pressure sensing chamber 48. The upper end of the bellows 50 is fixed to the valve housing 45. The top of the partition portion 41 of the valve rod 40 is fitted into the lower end of the bellows 50. The pressure sensing chamber 48 is divided into two chambers by the bellows 50, namely a first pressure chamber 54 formed inside the bellows and a second pressure chamber 55 formed outside the bellows 50.

[0038] As shown in FIG. 1, a throttle 36a is formed on the circulation pipe 36 between the discharge chamber 22 and the external refrigerant circuit 30. Referring back to FIG. 2, the first pressure chamber 54 communicates via a first pressure introducing passage 37 with the discharge chamber 22 at a first pressure monitoring point P1 that is located on the upstream side of the throttle 36a. The second pressure chamber 55 communicates via a second pressure introducing passage 38 with the circulation pipe 36 at a second pressure monitoring point P2 that is located on the downstream side of the throttle 36a. Thus, a monitored pressure PdH at the first pressure monitoring point P1 is introduced into the first pressure chamber 54, and a monitored pressure PdL at the second pressure monitoring point P2 is introduced into the second pressure chamber 55.

[0039] The lower end of the bellows 50 vertically moves in accordance with the pressure difference (PdH - PdL) between the pressures on opposite sides of the throttle 36a. Thus, the position of the valve rod 40 (or the valve body portion 43) is determined by varying the pressure difference. The pressure difference (PdH - PdL) between the pressures on opposite sides of the throttle 36a varies depending on the refrigerant flow rate in the refrigerant circulation circuit. For example, when the refrigerant flow rate is increased, the pressure difference (PdH - PdL) is increased. On the other hand, when the refrigerant flow rate is decreased, the pressure difference (PdH - PdL) is decreased. The bellows 50 operates the valve body portion 43 such that the displacement of the compressor is changed so as to cancel the variation of the pressure difference (PdH - PdL).

[0040] The solenoid portion 60 of the first control valve CV1 has in the middle of the actuator housing 45b an accommodating cylinder 61 that has a cylindrical shape with a bottom. A fixed core 62 of a column shape is fittingly fixed to the upper opening of the accommodating cylinder 61. Thus, a solenoid chamber 63 is defined in the lower portion of the accommodating cylinder 61.

[0041] A movable core 64 is axially movably accommodated in the solenoid chamber 63. A guide hole 65 extends through the center of the fixed core 62 in the

axial direction of the valve rod 40. The guide rod portion 44 of the valve rod 40 is arranged in the guide hole 65 so as to move in the axial direction of the valve rod 40. The guide rod portion 44 is fittingly fixed to the movable core 64 of the solenoid chamber 63. Thus, the movable core 64 and the valve rod 40 vertically move together.

[0042] A helical spring 66 is accommodated between the fixed core 62 and the movable core 64 in the solenoid chamber 63 for urging the valve rod 40 in such direction that causes the valve body portion 43 to move away from the valve seat 53.

[0043] A coil 67 is wound around the outer periphery of the accommodating cylinder 61 over a range covering the fixed core 62 and the movable core 64. Driving signal is transmitted from a driving circuit 68a to the coil 67, based on the command from a control device 68 in accordance with air conditioning load. With such driving signal transmitted to the coil 67, electromagnetic force (or electromagnetic attraction) is generated between the fixed core 62 and the movable core 64, the magnitude of which electromagnetic force is determined by amount of electric power supplied to the coil 67. The electromagnetic force is transmitted to the valve rod 40 (or the valve body portion 43) through the movable core 64. Controlling energization of the coil 67 is performed by adjusting the voltage applied to the coil 67, and duty cycle control is utilized in the present preferred embodiment.

[0044] The solenoid portion 60 of the first control valve CV1 varies the electromagnetic force for application to the valve body portion 43 in accordance with the amount of the electric power supplied from an external device. In the first control valve CV1, therefore, control target (or set pressure difference) for the pressure difference (PdH - PdL) between the pressures on opposite sides of the throttle 36a, that is, a standard for positioning the valve body portion 43 by the bellows 50 is changed by varying the electromagnetic force for application to the valve body portion 43. In other words, the first control valve CV1 is formed to internally autonomously position the valve rod 40 (or the valve body portion 43) in accordance with the variation of the pressure difference (PdH - PdL) between the first and second pressure monitoring points P1 and P2 such that the set pressure difference determined by the amount of the electric power supplied to the coil 67 is maintained.

[0045] The set pressure difference of the first control valve CV1 is varied by adjusting the amount of the electric power supplied to the coil 67 from the external device. For example, when the duty ratio that is commanded from the control device 68 to the driving circuit 68a is increased, electromagnetic urging force of the solenoid portion 60 is increased, and the set pressure difference of the first control valve CV1 is increased, accordingly. With the set pressure difference of the first control valve CV1 thus increased, the displacement of the compressor is increased. On the other hand, when the duty ratio that is commanded from the control device 68 to the driving circuit 68a is decreased, electromag-

netic urging force of the solenoid portion 60 is decreased, and the set pressure difference of the first control valve CV1 is decreased. When the set pressure difference of the first control valve CV1 is decreased, the displacement of the compressor is decreased.

[0046] It is noted that the compressor of the present preferred embodiment is what is called a clutchless type compressor, and the drive shaft 16 is continuously rotated while the engine E is driven. When the air conditioning is not needed, however, supplying the electric power to the coil 67 is stopped by switching off the air conditioning apparatus, that is, the duty ratio is zero, and the swash plate is set at the minimum inclination angle. Thus, the displacement of the compressor is set at the minimum displacement, namely, zero or about zero by only one meaning. Therefore, even when the drive shaft 16 is rotated, supplying the refrigerant from the compressor to the external refrigerant circuit 30 is substantially stopped, and the refrigeration cycle is stopped.

[0047] As shown in FIGS. 1, 3 and 4, an accommodation hole 70 is formed in a rear end surface of the rear housing 4 for accommodating therein the second control valve CV2. A valve housing 71 is fittingly fixed to the accommodation hole 70. The valve housing 71 includes a cylindrical portion 72 whose outside diameter is smaller than that of the accommodation hole 70 and a fitting portion 73 that continues from the cylindrical portion 72 on the opening side of accommodation hole 70 and is fittingly fixed to the accommodation hole 70. The valve housing 71 is pushed into the accommodation hole 70 such that the distal end of the cylindrical portion 72 contacts an inner bottom surface 70a of the accommodation hole 70.

[0048] The cylindrical portion 72, the end surface 73a of the fitting portion 73 that faces the inside of the cylindrical portion 72, and the inner bottom surface 70a of the accommodation hole 70 define an accommodation chamber 74 in the cylindrical portion 72. A communication space 79 is formed between the outer peripheral surface of the cylindrical portion 72 and the inner peripheral surface of the accommodation hole 70. The communication space 79 communicates with the crank chamber 15 via a passage 75 arranged on the side of the crank chamber 15.

[0049] In the accommodation chamber 74, a spool 76 that serves as a valve body is movably accommodated in the direction in which the cylindrical portion 72 extends. The spool 76 is slidable between the position at which the spool 76 contacts the inner bottom surface 70a of the accommodation hole 70 and the position at which the spool 76 contacts the end surface 73a of the fitting portion 73, and has a cylindrical shape with a bottom on the side of the end surface 73a of the fitting portion 73.

[0050] The spool 76 divides the accommodation chamber 74 into front and rear spaces, which are blocked by the contact between the outer peripheral surface of the spool 76 and the inner peripheral surface of

the accommodation chamber 74. The blocked front and rear spaces are respectively defined as a valve chamber 77 on the side of the inner bottom surface 70a of the accommodation hole 70 and a back pressure chamber 78 on the side of the end surface 73a of the fitting portion 73. In the spool 76, the end surface on the opening side of the spool 76 arranged in the valve chamber 77 is defined as an end valve portion 76a and the outer bottom surface of the spool 76 arranged in the back pressure chamber 78 is defined as a back surface 80. The spool 76 contacts the inner bottom surface 70a of the accommodation hole 70 with the end valve portion 76a.

[0051] The cylindrical portion 72 of the valve housing 71 forms a first gap-hole 72a and a second gap-hole 72b therethrough. The first gap-hole 72a communicates with the inside and the outside of the cylindrical portion 72. The second gap-hole 72b is located nearer the fitting portion 73 than the first gap-hole 72a, and communicates with the inside and the outside of the cylindrical portion 72.

[0052] The first gap-hole 72a communicates with the valve chamber 77 and the communication space 79 in a state that the spool 76 is in contact with the end surface 73a of the fitting portion 73 as shown in FIG. 4. The first gap-hole 72a is blocked by a region on the side of the valve chamber 77 in the outer peripheral surface of the spool 76, that is, a first peripheral valve portion 76b in a state that the spool 76 is in contact with the inner bottom surface 70a of the accommodation hole 70. Thus, the communication between the valve chamber 77 and the communication space 79 is blocked as shown in FIG. 3.

[0053] The second gap-hole 72b communicates with the back pressure chamber 78 and the communication space 79 in a state that the spool 76 is in contact with the inner bottom surface 70a of the accommodation hole 70 as shown in FIG. 3. The second gap-hole 72b is blocked by a region on the side of the back pressure chamber 78 in the outer peripheral surface of the spool 76, that is, a second peripheral valve portion 76c in a state that the spool 76 is in contact with the end surface 73a of the fitting portion 73. Thus, the communication between the back pressure chamber 78 and the communication space 79 is blocked as shown in FIG. 4.

[0054] The valve chamber 77 communicates with the suction chamber 21 via a passage 81 formed in the rear housing 14. The passage 81 is opened more inwardly than an annular region or a sealed region in which the end valve portion 76a of the spool 76 contacts the inner bottom surface 70a of the accommodation hole 70.

[0055] Therefore, the communication of the inside and the outside of the valve chamber 77 relative to the sealed region of the end valve portion 76a are blocked in a state that the spool 76 is in contact with the inner bottom surface 70a of the accommodation hole 70. In addition, the first gap-hole 72a is blocked by the first peripheral valve portion 76b. Thus, the communication between the passage 81 and the communication space 79 (or the passage 75) is blocked as shown in FIG. 3. The

communication of the inside and the outside of the valve chamber 77 relative to the sealed region of the end valve portion 76a are opened in a state that the spool 76 is in contact with the end surface 73a of the fitting portion 73. In addition, the first gap-hole 72a is opened by the first peripheral valve portion 76b of the spool 76. Thus, the communication between the passage 81 and the communication space 79 (or the passage 75) is opened as shown in FIG. 4.

[0056] In the present preferred embodiment, the passage 81, the valve chamber 77, the first gap-hole 72a, the communication space 79, and the passage 75 which is shared with the supply passage 29 form the first bleed passage 27. Therefore, in the spool 76, the end valve portion 76a and the first peripheral valve portion 76b which open and close the communication between the passage 81 and the communication space 79 are regarded as a first valve portion for adjusting the opening of the first bleed passage 27.

[0057] The back pressure chamber 78 communicates with the port 52 of the first control valve CV1 via a passage 82 formed in the fitting portion 73 of the valve housing 71 and a passage 83 that forms the supply passage 29. The passage 82 is opened at an opening 82a formed at the center of the end surface 73a of the fitting portion 73 in the back pressure chamber 78. Therefore, the refrigerant gas flowed from the discharge chamber 22 is introduced into the back pressure chamber 78 via a passage 84, the first control valve CV1 which is in an opening state, the passages 83 and 82. That is, a pressure Pdk on the downstream side of the position of the valve opening adjustment of the first control valve CV1, or the valve seat portion 53, in the supply passage 29 is applied to the back pressure chamber 78 via the passage 82 that serves as an introduction passage.

[0058] The refrigerant gas introduced from the discharge chamber 22 to the back pressure chamber 78 is flowed into the crank chamber 15 via the second gap-hole 72b, the communication space 79 and the passage 75. That is, in the second control valve CV2, the passage 82, the back pressure chamber 78, the second gap-hole 72b and the communication space 79 form the supply passage 29.

[0059] The spool 76 is urged toward the inner bottom surface 70a of the accommodation hole 70, that is, in such direction that the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion decrease the opening of the first bleed passage 27 by the force of the pressure Pdk in the back pressure chamber 78 applied to the back surface 80. On the other hand, the spool 76 is urged toward the end surface 73a of the fitting portion 73, that is, in such direction that the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion increase the opening of the first bleed passage 27 by the force of the suction pressure Ps which is applied to the end valve portion 76a and the valve chamber 77.

[0060] A helical spring 85 is arranged in the spool 76

of the valve chamber 77. The spring 85 has a movable end and a fixed end on the opposite sides thereof. The movable end of the spring 85 is in contact with the spool 76 while the fixed end of the spring 85 is held and accommodated in an accommodating groove 70b formed in the inner bottom surface 70a of the accommodation hole 70. The spring 85 urges the spool 76 in such direction that the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion increase the opening of the first bleed passage 27.

[0061] That is, the spool 76 is positioned by the balance between the urging force in the valve closing direction of the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion caused by the force of the pressure Pdk in the back pressure chamber 78, the urging force in the valve opening direction of the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion caused by the force of the pressure Ps in the valve chamber 77, and the urging force in the valve opening direction of the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion caused by the force of the urging force 85.

[0062] Meanwhile, in the present embodiment, the back surface 80 of the spool 76 forms thereon a second valve portion 86 for opening and closing the opening 82a of the passage 82 in the back pressure chamber 78 in accordance with the position of the spool 76. The second valve portion 86 protrudes from the center of the back surface 80 of the spool 76 so as to face the opening 82a of the passage 82. The second valve portion 86 is shaped into a circular shape in a transverse section, and is tapered so that the distal end of the second valve portion 86 becomes a minor diameter. The taper shape of the second valve portion 86 is such shaped that the diameter of the proximal end thereof becomes larger than that of the opening 82a of the passage 82 and the diameter of the distal end thereof becomes smaller than that of the opening 82a. The second valve portion 86 is made of resilient material such as synthetic rubber or synthetic resin.

[0063] As shown in FIG. 4, the movement of the spool 76 toward the fitting portion 73 is regulated by the contact of the second valve portion 86 with the end surface 73a of the fitting portion 73. In such a state that the movement of the spool 76 is regulated by the contact with the end surface 73a of the fitting portion 73, that is, in a state that the first bleed passage 27 is fully opened by the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion, the distal end of the second valve portion 86 enters the inside of the passage 82 via the opening 82a while a taper surface 86a of the second valve portion 86 contacts at an annular region on the rim of the opening 82a of the passage 82. Thus, the communication between the back pressure chamber 78 and the passage 82 is blocked. In addition, in such a state, the second gap-hole 72b is blocked by the second peripheral valve portion 76c of

the spool 76. Thus, the communication between the passage 83 and the communication space 79 (or the passage 75) is blocked.

[0064] In contrast, as shown in FIG. 3, in a state that the movement of the spool 76 is regulated by the contact with the inner bottom surface 70a of the accommodation hole 70, that is, in a state that the first bleed passage 27 is fully closed by the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion, the second valve portion 86 is distanced from the end surface 73a of the fitting portion 73 and the opening 82a of the passage 82 is opened. In addition, in such a state, the second gap-hole 72b is opened by the second peripheral valve portion 76c of the spool 76. Thus, the passage 83 and the communication space 79 (or the passage 75) are interconnected with each other.

[0065] The operating characteristics of the control valve CV2 will be now described. As shown in FIG. 3, in a state that the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion of the spool 76 of the second control valve CV2 have decreased the opening of the first bleed passage 27 from the fully opening state of the first bleed passage 27, the second valve portion 86 of the spool 76 opens the division of the back pressure chamber 78 and the passage 82 and the pressure Pdk in the passage 82 is applied to the back pressure chamber 78. Therefore, in the second control valve CV2, if the cross sectional area of the back pressure chamber 78 that is perpendicular to the axial direction of the spool 76 is represented as "SA", and the urging force of the spring 85 is represented as "f", condition expression (1) for increasing the opening of the first bleed passage 27 in the second control valve CV2 is expressed as follows:

$$(Pdk - Ps) \cdot SA < f \quad (1)$$

[0066] As shown in FIG 4, in the second control valve CV2, in a state that the first bleed passage 27 is fully opened by the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion of the spool 76, the second valve portion 86 of the spool 76 blocks the communication between the back pressure chamber 78 and the passage 82. Thus, the pressure Pdk in the passage 82 is not applied to the back pressure chamber 78. Therefore, the pressure Pdk in the passage 82 is applied only to the second valve portion 86 of the back surface 80 of the spool 76. If the cross sectional area at the opening 82a of the passage 82 that is perpendicular to the axial direction of the passage 82 is represented as "SB" (< "SA"), condition expression for decreasing the opening of the first bleed passage 27 in the second control valve CV2 in a state that the first bleed passage 27 is fully opened is expressed as follows:

$$(Pdk - Ps) \cdot SB > f \quad (2)$$

[0067] When time has passed for more than a predetermined time after the vehicle engine E was stopped, the pressure in the refrigerant circulation circuit is equalized at a relatively small value, and thus the pressure Pdk and the suction pressure Ps equalized to each other. Since the condition expression (1) is effective and the condition expression (2) is not effective, as shown in FIG. 4, the spool 76 moves by the spring 85 and the second valve portion 86 blocks the supply passage 29. At the same time, the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion fully opens the first bleed passage 27.

[0068] In a conventional compressor for a vehicle air-conditioning apparatus, any liquid refrigerant existing on the low pressure side of the external refrigerant circuit 30 with the vehicle engine E kept at a stop for a long time flows into the crank chamber 15 via the suction chamber 21 due to the fluid communication between the crank chamber 15 and the suction chamber 21 via the first and second bleed passages 27 and 28. Especially, when the temperature in the engine room where the compressor is located is lower than that in the vehicle interior, a large amount of the liquid refrigerant flows into the crank chamber 15 via the suction chamber 21 and is accumulated in the crank chamber 15.

[0069] Therefore, when the vehicle engine E is started and the compressor is also started thereby through the clutchless type power transmission mechanism PT, the liquid refrigerant evaporates under the influence of heat generated by the vehicle engine E and also of the stirring effect of the swash plate 18, with the result that the crank pressure Pc tends to be increased regardless the opening of the first control valve CV1.

[0070] For example, when the vehicle engine E is started while the vehicle interior is hot, the control device 68 is operated in response to the demand from an occupant to command maximum duty ratio to the drive circuit 68a, and the set pressure difference of the first control valve CV1 is set at the maximum value, accordingly, for performing cooling as required from the occupant. For this purpose, the first control valve CV1 closes the supply passage 29, and no high pressure refrigerant gas is supplied from the discharge chamber 22 to the back pressure chamber 78 of the second control valve CV2 and the crank chamber 15. Therefore, even if evaporation of the liquid refrigerant occurs in the crank chamber 15, the state wherein the pressure difference between the crank pressure Pc and the suction pressure Ps does not exceed the urging force f, that is, the state wherein the condition expression (2) is not effective, continues.

[0071] Consequently, the spool 76 of the second control valve CV2 is maintained by the urging force f of the spring 85 in such a state that the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion fully opens the first bleed passage 27, and the

liquid refrigerant in the crank chamber 15, as well as the refrigerant gas evaporated from part of the liquid refrigerant, are immediately flowed into the suction chamber 21 via the fully-opened first bleed passage 27. Thus, the crank pressure P_c is maintained at a low value since the first control valve CV1 closes the supply passage 29, and the compressor increases the inclination angle of the swash plate 18 thereby to increase the displacement of the compressor to its maximum.

[0072] If the first control valve CV1 still closes the supply passage 29 even after the liquid refrigerant is flowed out of the crank chamber 15, the first bleed passage 27 is fully opened by the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion of the second control valve CV2 as described above. Thus, even if the amount of blow-by gas from the cylinder bores 11a to the crank chamber 15 is increased from the amount initially designed, the blow-by gas is immediately flowed into the suction chamber 21 via the first and second bleed passages 27 and 28. Therefore, the crank pressure P_c is maintained at substantially the same level as the suction pressure P_s , and the maximum inclination angle of the swash plate 18, that is, the maximum displacement operation (100% displacement operation) of the compressor is maintained.

[0073] When the vehicle interior is cooled to a certain extent due to the above maximum displacement operation of the compressor, the control device 68 reduces the duty ratio that is commanded to the driving circuit 68a from the maximum. Accordingly, the first control valve CV1 opens the supply passage 29 so that the pressure P_{dk} in the passage 82 exceeds the suction pressure P_s in the valve chamber 77. Thus, the condition expression (2) is satisfied, so that the spool 76 moves against the urging force f of the spring 85 in the direction to reduce the valve opening of the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion from the fully-opened state as shown in FIG. 3.

[0074] In the second control valve CV2, in a state that the end valve portion 76a and the first peripheral valve portion 76b of the first valve portion of the spool 76 decreases the opening of the first bleed passage 27 from the fully-opened state, the second valve portion 86 of the spool 76 opens the division of the back pressure chamber 78 and the passage 82. Therefore, condition expression (3) for decreasing the opening of the first bleed passage 27 in the second control valve CV2 in a state that the first bleed passage 27 is opened but is not fully opened is expressed as follows:

$$(P_{dk} - P_s) \cdot SA > f \quad (3)$$

The condition expression (3) is effective due to the relation " $SA > SB$ " as long as the urging force f of the spring 85 is fixed, even if the pressure difference " $P_{dk} - P_s$ " between the pressure P_{dk} in the passage 82 and the pres-

sure P_s in the valve chamber 77 is smaller than the minimum value that satisfies the condition expression (2). Therefore, the spool 76 which has been distanced from the fully-opened state of the first bleed passage 27 by the formation of the condition expression (2) is moved in the direction to reduce the opening of the first bleed passage 27 without stopping on the way by the formation of the condition expression (3). Since the urging force of the spring 85 is relatively small, the spool 76 which has been distanced from the fully-opened state of the first bleed passage 27 is immediately moved to the closed state of the first bleed passage 27.

[0075] Thus, the crank pressure P_c is immediately raised by opening the supply passage 29 of the first control valve CV1 and closing the first bleed passage 27 of the second control valve CV2. Consequently, the compressor decreases the inclination angle of the swash plate 18 thereby to decrease the displacement of the compressor.

[0076] An amount of the compressed refrigerant gas that leaks from the discharge chamber 22 to the crank chamber 15 further to the suction chamber 21 is reduced to the amount of compressed refrigerant gas which leaks only through the second bleed passage 28 by closing the first bleed passage 27 in the second control valve CV2, so that a decrease in the efficiency of the compressor is prevented. Furthermore, although the refrigerant circulation circuit in the present preferred embodiment is formed such that the refrigerant circulation stops by operating the compressor at the minimum displacement (so called an off operation of the clutchless compressor), the off operation of the compressor is ensured by closing the first bleed passage 27 in the second control valve CV2.

[0077] The present embodiment provides the following advantageous effects.

(1) If the performance of the first control valve CV1 deteriorates due to its aged deterioration, the first control valve CV1 leaks the refrigerant gas even when the first control valve CV1 is operated on the maximum duty ratio. When the first control valve CV1 leaks the refrigerant gas, the pressure P_{dk} in the passage 82 is raised and the spool is urged in the direction to reduce the opening of the first bleed passage 27 in accordance with the pressure P_{dk} .

However, the spool 76 of the second control valve CV2 provides with the second valve portion 86 which blocks the opening 82a of the passage 82 in the back pressure chamber 78 in a state that the second control valve CV2 fully opens the first bleed passage 27. Therefore, in the back surface 80 of the spool 76, referring to the condition expression (2), the pressure P_{dk} in the passage 82 is applied only to the second valve portion 86, but is not applied to the back surface 80 other than the second valve portion 86. Thus, in the first control valve CV1 in a state that the supply passage 29 is blocked,

even if leakage of the refrigerant gas which is caused by the performance deterioration of the first control valve CV1 generates, the fully-opened state of the first bleed passage 27 is maintained even by the spring having small urging force f , such that mechanical error of the second control valve CV2 is prevented. Consequently, the maximum inclination angle of the swash plate 18, that is, the maximum displacement operation (100% displacement operation) of the compressor is maintained.

If the spring 85 whose urging force is relatively small is adopted, the second control valve CV2 can set the first bleed passage 27 at the minimum opening without increasing the pressure P_{dk} in the back pressure chamber 78 by widely opening the supply passage 29 in the first control valve CV1. Therefore, in a state that the first control valve CV1 opens the supply passage 29, the period in which the second control valve CV2 sets the first bleed passage 27 at the opening other than the closed state is not increased. Thus, decrease in the efficiency of the compressor is prevented.

(2) The second valve portion 86 of the second control valve CV2 is shaped in a protruding shape and a taper shape so as to enter the passage 82. Therefore, the opening 82a of the passage 82 is closed by the second valve portion 86.

(3) The back pressure chamber 78 of the second control valve CV2 and the passage 82 for introducing the refrigerant gas from the discharge chamber 22 to the back pressure chamber 78 form a part of the supply passage 29. That is, in a state that the first control valve CV1 closes the supply passage 29, the second valve portion 86 of the second control valve CV2 closes the supply passage 29 on the downstream side of the first control valve CV1. Therefore, in this state, even if the refrigerant gas leaks by the performance deterioration of the first control valve CV1, the leaked refrigerant gas is not supplied into the crank chamber 15. Thus, the maximum inclination angle of the swash plate 18, that is, the maximum displacement operation of the compressor is maintained.

(4) The second control valve CV2 opens and closes the supply passage 29 at a plurality of places. In the present preferred embodiment, the second control valve CV2 opens and closes the supply passage 29 at two places of the second peripheral valve portion 76c and the second valve portion 86. Therefore, the second control valve CV2 surely closes the supply passage 29 thereby further effectively preventing the refrigerant gas that leaks from the first control valve CV1 from being supplied into the crank chamber 15.

(5) The second valve portion 86 is made of resilient material. Therefore, the opening 82a of the passage 82 is surely closed by the second valve portion 86 in response to the resilient deformation of the second valve portion 86.

(6) The first valve portion of the second control valve CV2 opens and closes the first bleed passage 27 at a plurality of places. In the present preferred embodiment, the first valve portion of the second control valve CV2 opens and closes the first bleed passage 27 at two places of the end valve portion 76a and the first peripheral valve portion 76b. Therefore, the second control valve CV2 surely closes the first bleed passage 27 thereby further effectively preventing the decrease in the efficiency of the compressor.

[0078] The present invention is not limited to the above-described embodiment, but is modified as follows.

[0079] In the above-preferred embodiment, the second control valve CV2 is arranged on the supply passage 29. In an alternative embodiment to the above embodiment, as shown in FIG. 5, the second gap-hole 72b of the second control valve CV2 is eliminated and the passage 83 directly communicates with the crank chamber 15. In addition, a branch passage 90 is branched off the passage 83 and communicates with the passage 82 of the second control valve CV2. In this case, the passage 75 is exclusive for the first bleed passage 27.

[0080] In an alternative embodiment to the above embodiment, the aspect of FIG. 5 is partially modified. The minimum opening of the first valve portion 76a and 76b is set to a value that is not zero by grooving the first valve portion 76a and 76b of the second control valve CV2 such that the first bleed passage 27 is continuously opened. The second bleed passage 28 may be eliminated. In this case, the passages of the displacement control mechanism are simply formed.

[0081] In an alternative embodiment to the above embodiment, the aspect of FIG. 5 is partially modified. As shown in FIG. 6, positions at which the passage 75 and the passage 81 communicate with the second control valve CV2 are replaced by each other. In addition, a fixed throttle 83a is formed on the passage 83. In this case, if the first bleed passage 27 is continuously opened by setting the minimum opening of the first valve portion 76a and 76b at a value that is not zero, and further if the second bleed passage 28 is eliminated, the second valve portion 86 of the preferred embodiment of the present invention can be applied to the structure similar to the prior art control valve which is shown in FIG. 8.

[0082] In the above-described embodiments, the second valve portion 86 of the second control valve CV2 is shaped in a protruding shape on the back surface 80 of the spool 76. In alternative embodiments to the above

embodiments, the second valve portion 86 is eliminated from the above-described embodiments. Instead, the back surface 80 may be regarded as a flat second valve portion by adhering resilient coat such as rubber coat and resin coat on the back surface 80 of the spool 76. In a technique other than adhering resilient coat on the back surface 80 of the spool 76 such that the back surface 80 of the spool 76 serves as a second valve portion, it is proposed that the back surface 80 and the end surface 73a of the fitting portion 73 are polished in high accuracy.

[0083] In the above-described embodiments, the second control valve CV2 opens and closes the supply passage 29 at a plurality of places (at two places of the second peripheral valve portion 76c and the second valve portion 86). In alternative embodiments to the above embodiments, the second control valve CV2 opens and closes the supply passage 29 at a singular place, or at the second valve portion 86.

[0084] In the above-described embodiments, the first valve portion 76a and 76b of the second control valve CV2 opens and closes the first bleed passage 27 at a plurality of places (at two places of the end valve portion 76a and the first peripheral valve portion 76b). In alternative embodiments to the above embodiments, the first valve portion of the second control valve CV2 opens and closes the first bleed passage 27 at a singular place such as the end valve portion 76a or the first peripheral valve portion 76b.

[0085] In the above-described embodiments, the spool 76 (a tubular body) is adopted as a valve body of the second control valve CV2. In alternative embodiments to the above embodiments, a spherical body may be adopted as the valve body. In this case, a hemispherical part of the spherical body on the side of the valve chamber 77 forms the first valve portion while the rest hemispherical part of the spherical body on the side of the back pressure chamber 78 forms the back surface and the second valve portion.

[0086] In the above-described embodiments, the spring 85 is a coil spring. In the present invention, however, the spring is not limited to the coil spring. Other type of springs such as plate spring and torsion bar may be adopted.

[0087] In the above-described embodiments, the first control valve CV1 varies the displacement of the compressor such that the pressure difference (PdH-PdL) between the pressures on opposite sides of the throttle 36a is maintained at a predetermined target value (set pressure difference). Also, in the first control valve CV1, the set pressure difference is varied by external electric control. In alternative embodiments to the above embodiments, the first control valve CV1 is operated such that the pressure in the suction pressure region is maintained at a predetermined target value (set suction pressure) while the set suction pressure is varied by external electric control. In this case, the first control valve CV1 is so-called a control valve of variable set suction pres-

sure type.

[0088] In the above-described embodiments, pressure sensing mechanism such as the pressure sensing chamber 48 and the bellows 50 may be eliminated from the first control valve CV1, and the first control valve CV1 may be varied to a simple electromagnetic valve.

[0089] In the above-described embodiments, the solenoid 60 may be eliminated from the first control valve CV1, and the first control valve CV1 may be varied to a simple pressure sensing valve which does not provide with external control function.

[0090] The present invention may be applied to a displacement control device for a variable displacement type compressor of a wobble type.

[0091] Therefore, the present examples and embodiments are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein but may be modified within the scope of the appended claims.

A displacement control mechanism controls displacement of a variable displacement type compressor that forms a refrigerant circulation circuit for an air conditioning apparatus. The displacement is decreased as a pressure in a crank chamber rises while being increased as the pressure in the crank chamber falls. The displacement control mechanism includes a bleed passage, a supply passage, a first control valve and a second control valve. The second control valve includes a back pressure chamber, a valve chamber, a valve body, a spring and a second valve portion. The second valve portion is provided with the back surface of the valve body. The second valve portion closes an opening of an introduction passage in the back pressure chamber when the first valve portion maximizes the opening of the bleed passage.

Claims

1. A displacement control mechanism for controlling displacement of a variable displacement type compressor that forms a refrigerant circulation circuit for an air conditioning apparatus, the displacement being decreased as a pressure in a crank chamber rises while being increased as the pressure in the crank chamber falls, the refrigerant circulation circuit having a suction pressure region and a discharge pressure region, the displacement control mechanism comprising a bleed passage interconnecting the crank chamber with the suction pressure region, a supply passage interconnecting the crank chamber with the discharge pressure region, a first control valve located on the supply passage for adjusting an opening of the supply passage at a position of valve opening adjustment, and a second control valve including a back pressure chamber to which a pressure on a downstream side of the position of valve opening adjustment of the first control

valve in the supply passage is introduced through an introduction passage, a valve chamber forming a part of the bleed passage, a valve body having a first valve portion located in the valve chamber and a back surface located in the back pressure chamber, the first valve portion decreasing an opening of the bleed passage as a pressure in the back pressure chamber which is applied to the back surface rises, and a spring for urging the valve body so that the first valve portion increases the opening of the bleed passage, **characterized in that** a second valve portion is provided with the back surface of the valve body, and **in that** the second valve portion closes an opening of the introduction passage in the back pressure chamber when the first valve portion maximizes the opening of the bleed passage.

2. The displacement control mechanism according to claim 1, wherein the second valve portion is so shaped as to protrude and taper from the back surface toward the opening of the introduction passage, the second valve portion closing the opening of the introduction passage by entering the introduction passage.
3. The displacement control mechanism according to claim 1 or 2, wherein the introduction passage and the back pressure chamber form a part of the supply passage.
4. The displacement control mechanism according to claim 3, wherein the second control valve opens and closes the supply passage at a plurality of places.
5. The displacement control mechanism according to any one of claims 1 through 3, wherein the second valve portion is made of resilient material.
6. The displacement control mechanism according to any one of claims 1 through 5, wherein the first valve portion of the second control valve opens and closes the bleed passage at a plurality of places.
7. The displacement control mechanism according to any one of claims 1 through 6, wherein the variable displacement type compressor is of a clutchless type, the variable displacement type compressor of clutchless type is such a variable displacement type compressor that continues to rotate a drive shaft in a state that a minimum inclination angle of a cam plate is maintained such that a refrigerant is not substantially flowed into an external refrigerant circuit while a refrigeration cycle stops.

FIG. 1

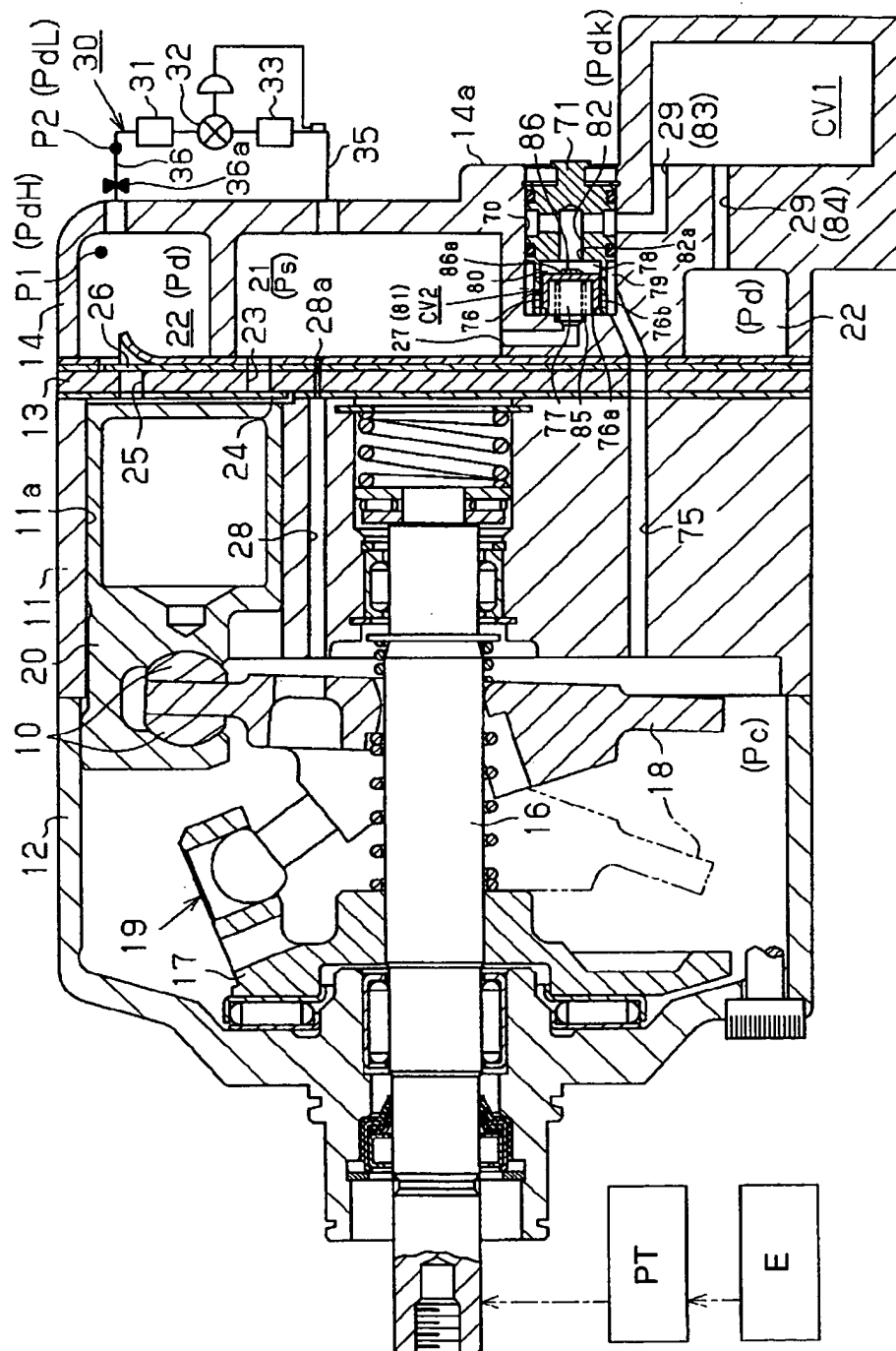


FIG. 2

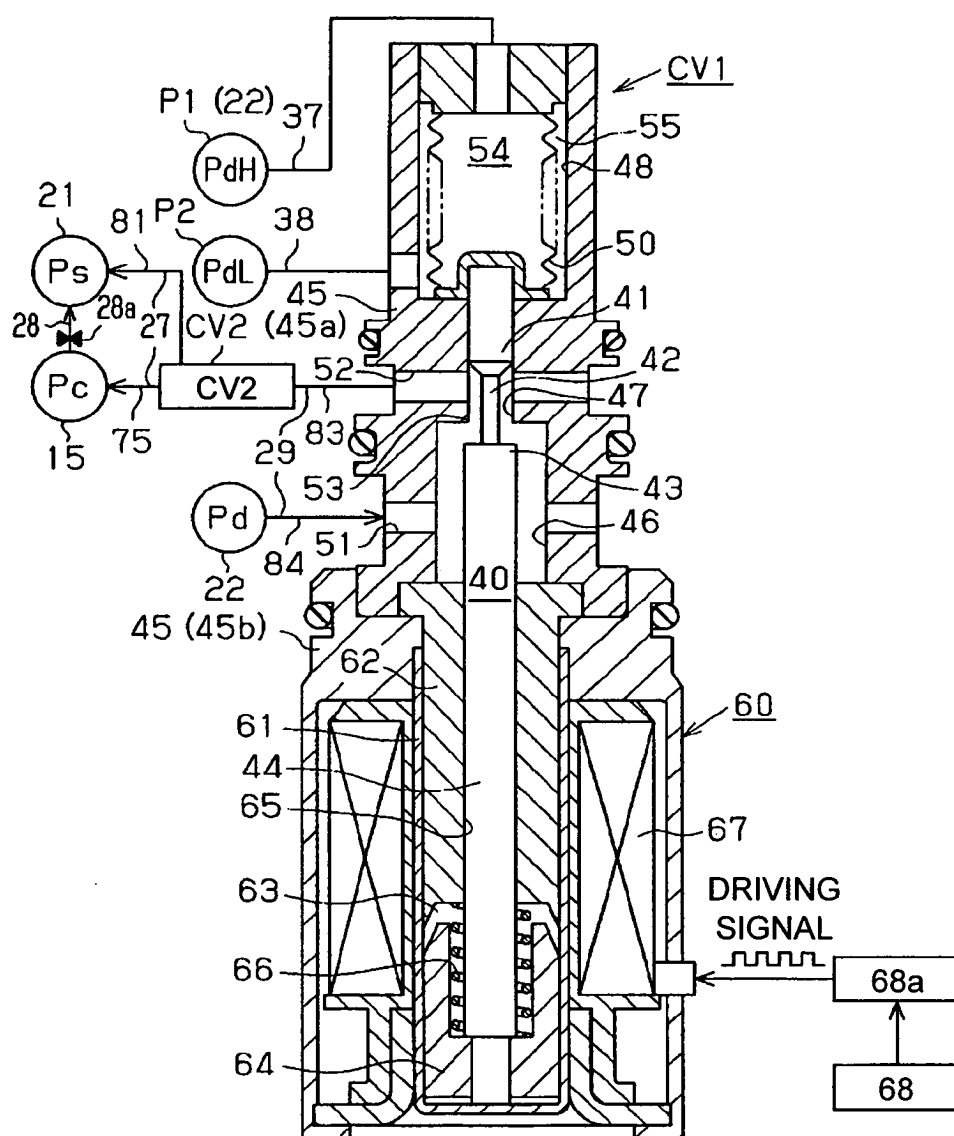


FIG. 3

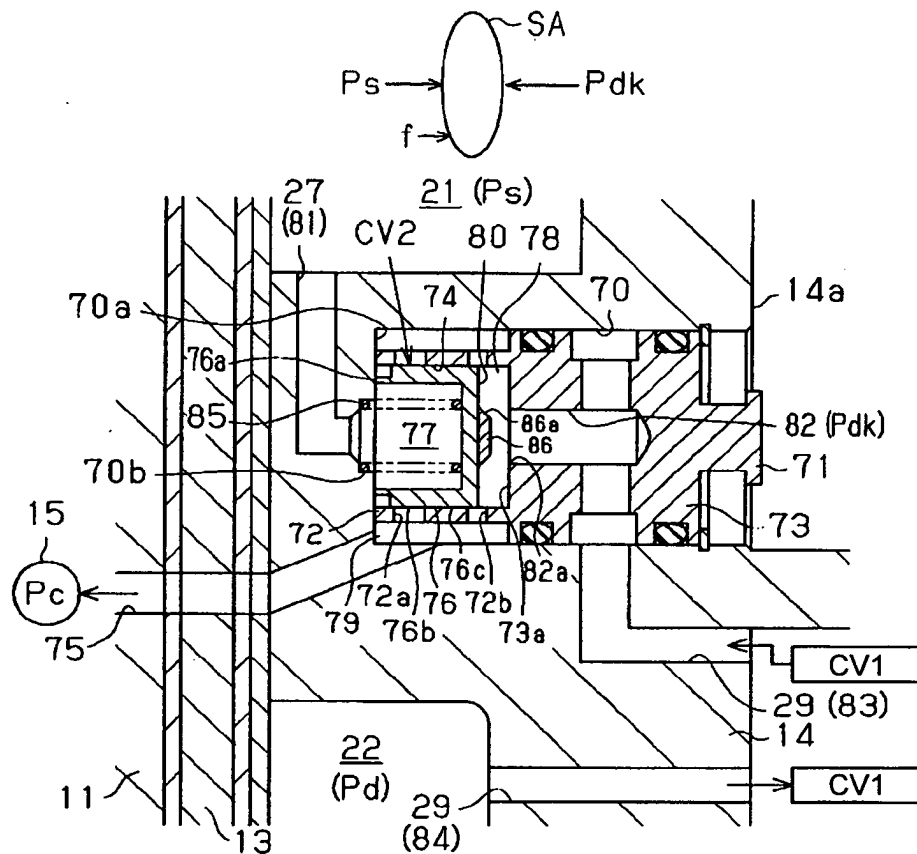


FIG. 4

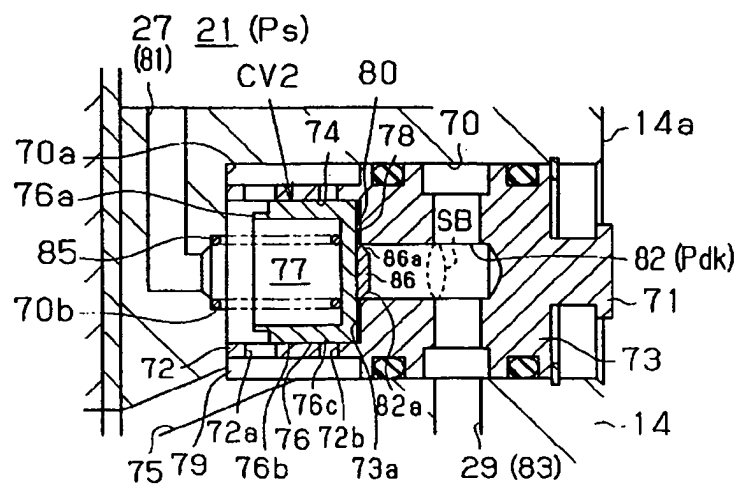


FIG. 5

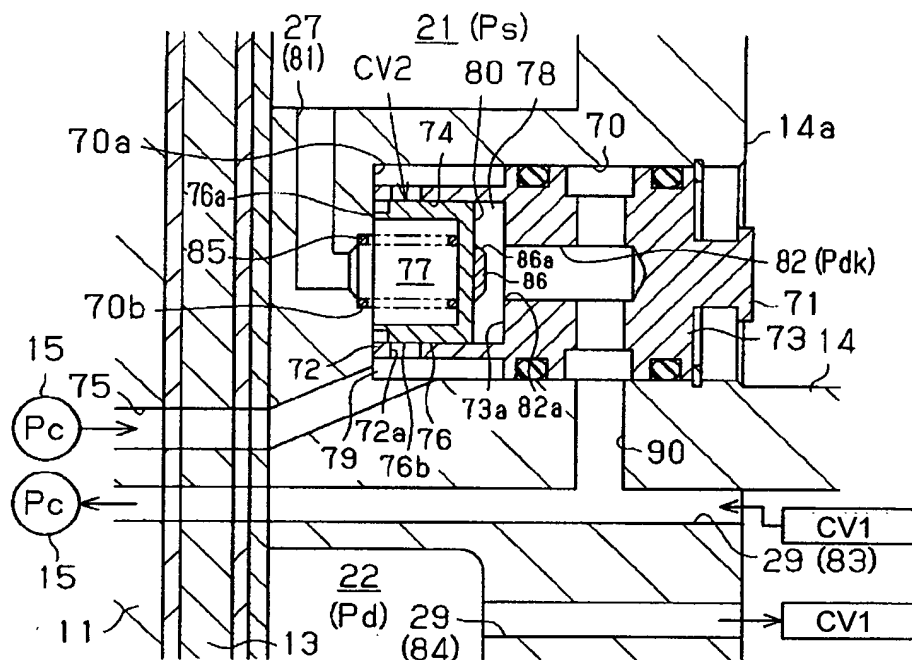


FIG. 6

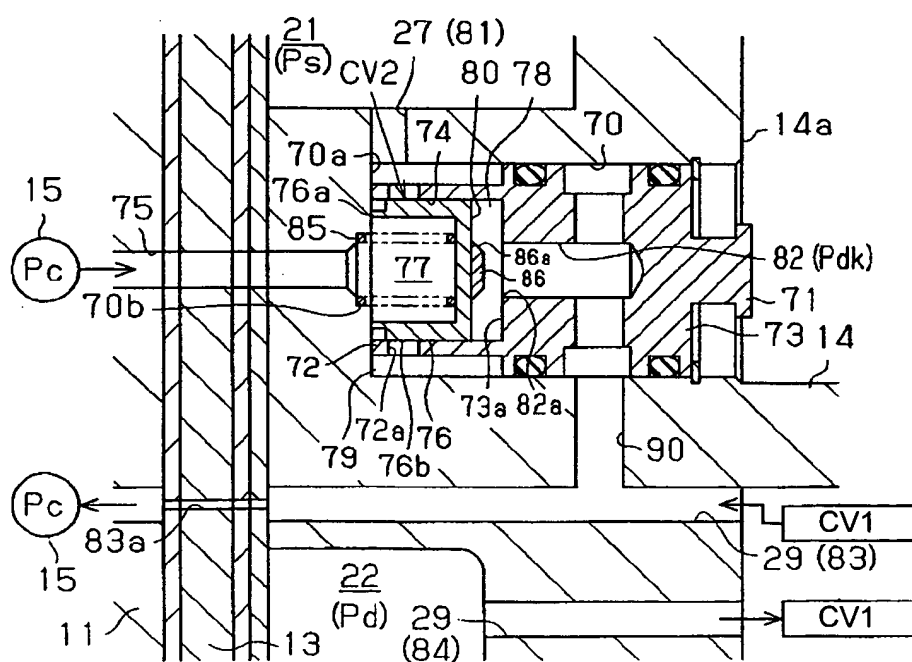


FIG. 7

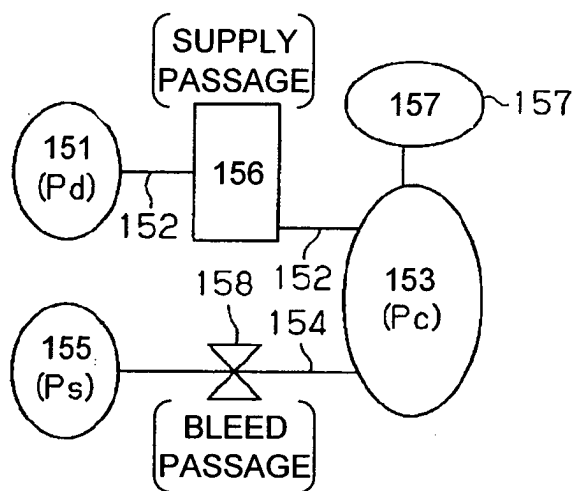
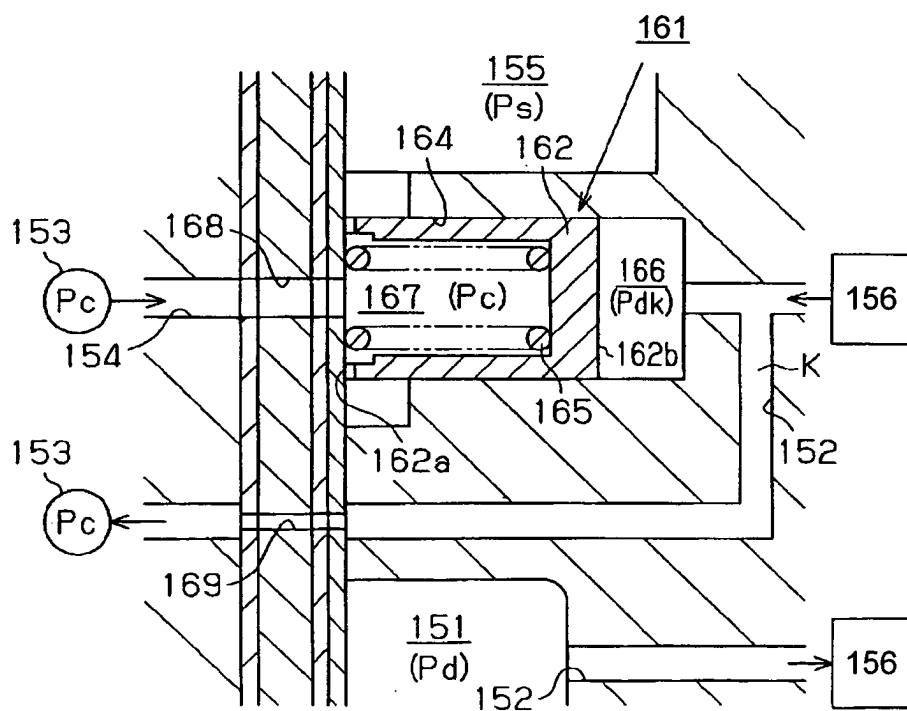


FIG. 8





European Patent
Office

EUROPEAN SEARCH REPORT

Application Number
EP 04 01 4234

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.7)
X	EP 1 172 559 A (TOYOTA JIDOSHOKKI KK) 16 January 2002 (2002-01-16) * paragraph [0001] - paragraph [0005] * * paragraph [0042] - paragraph [0045] * * paragraph [0073] * * figures 1,4,6 *	1,3,7	F04B27/18
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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 20 October 2004	Examiner Gnüchtel, F
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document			

EPO FORM 1503 03.82 (P04C01)

**ANNEX TO THE EUROPEAN SEARCH REPORT
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EP 04 01 4234

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
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