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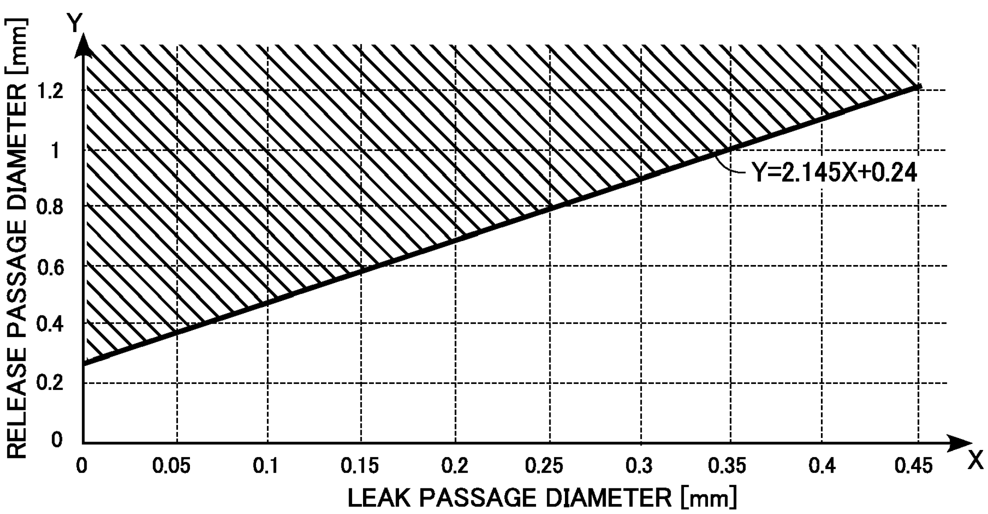
(54) **CONTROL VALVE FOR VARIABLE DISPLACEMENT COMPRESSOR**

(57) A control valve (1) according to one embodiment includes a valve element (24) that autonomously operates so that a differential pressure (Pd - Ps) between a discharge pressure (Pd) of a discharge chamber (114) and a suction pressure (Ps) of a suction chamber (110) is kept at a preset differential pressure according to a value of current supplied to the solenoid (3), and a bleed hole (96) that is provided in the body (5) separately from the valve hole (18), that has a leak passage (98) whose diameter smaller than that of the valve hole (18), and that permits refrigerant to be leaked from a discharge chamber communication port (10) to a crankcase communication port (12) even while a valve section is closed. The bleed hole (96) is formed such that Y > 2.145X + 0.24 is satisfied, where X represents the diameter of the leak passage (98) and Y represents a diameter of a release passage (119).

FIG.4



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Description

[0001] The present invention relates to a control valve for controlling the discharging capacity of a variable displacement compressor.

[0002] An automotive air conditioner is generally configured by arranging and placing a compressor, a condenser, an expander, an evaporator, and so forth in a refrigeration cycle. The compressor is, for example, a variable displacement compressor (hereinafter referred to simply as "compressor" also) capable of varying the refrigerant discharging capacity in order to maintain a constant level of cooling capacity irrespective of the engine speed. In this compressor, a piston for compression is linked to a wobble plate, which is mounted to a rotational shaft rotatingly driven by an engine. And the refrigerant discharging rate is regulated by changing the stroke of the piston through changes in the angle of the wobble plate. The angle of the wobble plate is changed continuously by changing the balance of pressure working on both faces of the piston as part of the discharged refrigerant is introduced into a hermetically-closed crankcase.

[0003] Specifically, the compressor is typically provided therein with a supply passage for supplying part of the discharged refrigerant into the crankcase and a release passage for releasing part of refrigerant in the crankcase to a suction chamber. The pressure Pc (hereinafter referred to as "crank pressure") within the crankcase is regulated by regulating the balance between the flow rate of refrigerant supplied into the crankcase through the supply passage and the flow rate of refrigerant released from the crankcase through the release passage, and the angle of the wobble plate is changed accordingly. The crank pressure Pc is controlled by regulating the opening degree of the release passage by a control valve for a variable displacement compressor (hereinafter referred to simply as "control valve" also), which is provided between the discharge chamber and the crankcase of the compressor. When the control valve is fully opened, the crank pressure Pc is increased and the wobble plate has become approximately perpendicular to the rotational shaft, the compressor can be switched to a minimum capacity operation mode where the compressor operates with the minimum capacity. In contrast, when the control valve is closed, the crank pressure Pc is decreased and the wobble plate is tilted greatly to the rotational shaft, the compressor can be switched to a maximum capacity operation mode where the compressor operates with the maximum capacity.

[0004] One example of such a control valve is a valve that controls the amount of refrigerant introduced into a crankcase according to a differential pressure (Pd - Ps) between a discharge pressure Pd and a suction pressure Ps of a compressor so as to control the crank pressure Pc (refer to Japanese Patent Application Publication No. 2001-132650, for example). Such a control valve has an internal passage constituting a release passage in a body thereof, and a valve hole in the internal passage. The flow rate of refrigerant introduced into the crankcase can be controlled by regulating the opening degree of a valve section by making a valve element, placed within the body, move toward and away from the valve hole. In this process, the valve element operates autonomously so that the differential pressure (Pd - Ps) is kept at a preset differential pressure set according to an amount of current supplied to a solenoid. Such a control valve can control the differential pressure (Pd - Ps) to be a desired value by adjusting the preset differential pressure, and can appropriately adjust the discharging capacity of refrigerant from the compressor. Since the capacity control is based on the magnitude of the discharge pressure Pd, this control valve also has an advantage of being highly responsive in changing the discharging capacity.

<u>Related Art List</u>

(1) Japanese Patent Application Publication No. 2001-132650

[0005] The body of such a control valve may have a bleed hole for oil circulation formed separately from the valve hole. Specifically, refrigerant flowing in a refrigeration cycle typically contains lubricating oil for preventing seizing at sliding portions in the compressor. The oil is preferably circulated constantly while the compressor is in operation. While the compressor is in the maximum capacity operation mode, however, the control valve is closed and the oil cannot be circulated through the valve hole. The bleed hole for at least ensuring minimum oil circulation may thus be provided separately from the valve hole.

[0006] With such a bleed hole, however, discharged refrigerant also continues to be introduced into the crankcase while the control valve is closed, which can cause an increase in the crank pressure Pc. In particular, in recent years, spurred by the global warming issue, it has been proposed that alternatives for chlorofluorocarbon (CFC), which have been conventionally used as refrigerant in the refrigeration cycle be replaced by carbon dioxide and the like. In the refrigeration cycle where carbon dioxide is used, the pressure of refrigerant is increased to a supercritical range exceeding the critical temperature thereof and therefore the discharge pressure of refrigerant gets very high. Thus, depending on the design of the bleed hole, the crank pressure Pc may also get higher than a required level while the control valve is closed and the compressor may not be able to maintain the maximum capacity operation. The inventors have recognized the aforementioned drawbacks, and as a result of thorough investigation seeking for a solution, have come to an idea that it is possible to both ensure the oil circulation and maintain a stably controlled state in the compressor by focusing on the relation between the release passage provided in the compressor and the bleed hole formed in the control valve.

[0007] A purpose of the present invention is to both achieve control stability in the maximum capacity operation and ensure oil circulation in a variable displacement compressor.

[0008] One embodiment of the present invention relates to a control valve for a variable displacement compressor. The control valve is applied to a variable displacement compressor for compressing refrigerant led into a suction chamber and discharging the compressed refrigerant from a discharge chamber, part of the discharged refrigerant being introduced into a crankcase via a supply passage while part of refrigerant in the crankcase being released to the suction chamber via a release passage, so that pressure in the crankcase is regulated in such a manner that a discharging capacity of the compressor is regulated, the control valve controlling the discharging capacity by regulating a flow rate of refrigerant flowing through the supply passage, the control valve including: a body having a discharge chamber communication port communicating with the discharge chamber, a crankcase communication port communicating with the crankcase, and a suction chamber communication port communicating with the suction chamber, an internal passage connecting the discharge chamber communication port and the control chamber communication port constituting the supply passage, the body having a valve hole provided in the internal passage; a valve element for adjusting an opening degree of a valve section by moving toward and away from the valve hole, the valve element receiving a differential pressure between a discharge pressure of the discharge chamber and a suction pressure of the suction chamber in a valve opening direction; and a solenoid, provided in the body, which generates a solenoidal force with which to drive the valve element in a valve closing direction according to an amount of current supplied thereto.

[0009] The valve element autonomously operates so that the differential pressure between the discharge pressure and the suction pressure is kept at a preset differential pressure according to a value of current supplied to the solenoid, and the control valve further has a bleed hole that permits refrigerant to be leaked from the discharge chamber communication port to the crankcase communication port even while the valve section is closed, the bleed hole being provided in the body separately from the valve hole and having a leak passage whose diameter is smaller than that of the valve hole. The bleed hole is formed such that the following expression (1) is satisfied, where X represents the diameter of the leak passage and Y represents a diameter of the release passage:

$$Y > 2.145X + 0.24 \quad \ldots (1).$$

[0010] By employing this embodiment, the maximum capacity operation of a compressor can be achieved and maintained even in a high-temperature and high-pressure environment and oil circulation for lubrication can be ensured, as will also be described in embodiments below.

FIG. 1 is a system chart showing a refrigeration cycle of an automotive air conditioner according to an embodiment;
FIG. 2 is a cross-sectional view showing a structure of a control valve according to an embodiment;
FIG. 3 is a partially enlarged cross-sectional view of the upper half of FIG. 2;
FIG. 4 is a graph showing an allowable range of combination of a release passage diameter and a leak passage diameter when carbon dioxide is used as refrigerant;
FIGS. 5A and 5B are graphs showing differential pressure characteristics depending on the combination of the release passage diameter and the leak passage diameter; and
FIG. 6 is a graph showing an allowable range of combination of the release passage diameter and the leak passage diameter when an alternative for chlorofluorocarbon (CFC) is used as refrigerant.

[0011] The invention will now be described by reference to the preferred embodiments. This does not intend to limit the scope of the present invention, but to exemplify the invention.

[0012] Embodiments of the present invention will now be described in detail with reference to the accompanying drawings. In the following description, for convenience of description, the positional relationship in each structure may be expressed as "vertical" or "up-down" with reference to how each structure is depicted in Figures. FIG. 1 is a system chart showing a refrigeration cycle of an automotive air conditioner according to an embodiment.

[0013] The air conditioner according to the present embodiment includes a so-called supercritical refrigeration cycle that uses carbon dioxide, which operates under a high pressure, as the refrigerant. This air conditioner includes a variable displacement compressor (hereinafter referred to simply as "compressor" also) 101, a gas cooler 102, an expander 103, an evaporator 104, and a receiver 105. Here, the compressor 101 compresses a gaseous refrigerant circulating through the refrigeration cycle. The gas cooler 102 functions as an external heat exchanger that cools a compressed high-pressure gaseous refrigerant. The expander 103 adiabatically expands the cooled refrigerant so as to reduce the pressure thereof. The evaporator 104 evaporates the expanded refrigerant and removes the evaporative latent heat so as to cool air inside a vehicle's compartment. The receiver 105 separates the evaporated refrigerant into gas refrigerant and liquid refrigerant and then returns the thus separated gaseous carbon dioxide to the compressor 101.

[0014] The compressor 101 has a not-shown rotational shaft, which is freely rotatably supported within crankcase 116, which functions as a "control chamber". A wobble plate is tiltably provided in this rotational shaft. And an end of the rotational shaft extends outside the crankcase 116 and is connected to an output shaft of an engine by way of a pulley. A plurality of cylinders 112 are arranged around the rotational shaft, and a piston, which performs a reciprocating motion by the rotational motion of the wobble plate, is provided in each cylinder 112. Each cylinder 112 is connected to a suction chamber 110 through a suction valve and is connected to a discharge chamber 114 through a discharge valve. The compressor 101 compresses the refrigerant, which has been led into the cylinders 112 through the suction chamber 110, and discharges the compressed refrigerant through the discharge chamber 114.

[0015] The angle of the wobble plate of the compressor 101 is kept in a position where, for example, the load of a spring biasing the wobble plate in the crankcase 116 and the load caused by the pressures working on both faces of the piston connected to the wobble plate are balanced. This angle of the wobble plate can be changed continuously as follows. That is, a crank pressure Pc is changed as part of the discharged refrigerant is introduced into the crankcase 116, and the balance of pressures working on the both faces of the piston is changed, thereby changing continuously the angle thereof. Changing the stroke of the piston by varying the angle of the wobble plate regulates the discharging capacity of refrigerant. The crank pressure Pc is controlled by a control valve 1, which is provided between the discharge chamber 114 and the crankcase 116 of the compressor 101.

[0016] In other words, part of the discharged refrigerant of the compressor 101 is led into the crankcase 116 by way of the control valve 1 and is used to control the capacity of the compressor 101. An internal passage of the control valve 1 constitutes a "supply passage" for introducing part of the discharged refrigerant from the discharge chamber 114 into the crankcase 116. The control valve 1 is configured as a solenoid-driven electromagnetic valve, and the electric conduction state and/or amount is controlled by a control unit 120. In the present embodiment, the control unit 120 outputs a pulse signal, which has been set to a predetermined duty ratio, to a drive circuit 122. Then the control unit 120 has the drive circuit 122 output a current pulse associated with the duty ratio. In this manner, the solenoid is driven. The control valve 1 regulates the flow rate of refrigerant delivered from the discharge chamber 114 to the crankcase 116 such that a differential pressure (Pd - Ps) between a discharge pressure Pd and a suction pressure Ps of the compressor 101 can be brought closer to a preset differential pressure, which is a control target value. Thereby, the discharging capacity of the compressor 101 varies. That is, the control valve 1 functions as a so-called (Pd - Ps) differential pressure regulating valve.

[0017] An orifice 119 is provided in a refrigerant passage 118 through which the crankcase 116 and the suction chamber 110 communicate with each other. The orifice 119 functions as a "release passage" through which part of the refrigerant inside the crankcase 116 is leaked to the suction chamber 110 side, so that the crank pressure Pc will not be excessively high. Thus, the pressure in the crankcase 116 is regulated by introducing part of the discharged refrigerant from the discharge chamber 114 into the crankcase 116 through the supply passage and releasing part of the refrigerant in the crankcase 116 into the suction chamber 110 through the release passage. As a result, the discharging capacity of the compressor 101 is regulated. A check valve 130 is provided in a refrigerant passage provided between the discharge chamber 114 and a refrigerant outlet in the compressor 101.

[0018] The control unit 120 includes a CPU for performing various arithmetic processings, a ROM for storing various control programs, a RAM used as a work area for data storage and program execution, an I/O interface, and so forth. The control unit 120 has a PWM output unit for outputting a pulse signal having a specified duty ratio. However, such a PWM output unit may be configured using a known art and therefore the detailed description thereof is omitted here. The control unit 120 determines the aforementioned preset differential pressure, based on predetermined external information detected by various sensors (e.g., the engine speed, the temperatures inside and outside the passenger compartment, and the air-blowout temperature of the evaporator 104). Also, the control unit 120 controls the electric conduction state of and/or amount to the control valve 1 in order to obtain a solenoidal force required to maintain the preset differential pressure. Suppose now that there is a request for cutting down on the acceleration for the purpose of reducing the load torque of the compressor 101 during a high load state (e.g., while a vehicle is accelerating or running uphill). Then, the control unit 120 turns off the solenoid or suppresses the electric conduction amount to a predetermined lower limit, and thereby switches the variable displacement compressor to a minimum capacity operation mode where the compressor operates with the minimum capacity.

[0019] The expander 103, which is configured as a so-called thermostatic-expansion valve, regulates a valve opening degree by feeding back the temperature of refrigerant at an outlet side of the evaporator 104 and then supplies a liquid refrigerant, which meets a thermal load, to the evaporator 104. The refrigerant, which has passed through the evaporator 104, is returned to the compressor 101 via the receiver 105 and is again compressed.

[0020] The check valve 130 maintains its opened state as long as the discharging capacity of the compressor 101 is large to a certain degree and a differential pressure (Pd - Pd1) between the discharge pressure Pd of the discharge chamber 114 and an outlet pressure Pd1 at the refrigerant outlet exceeds a valve opening differential pressure. This valve opening differential pressure is set by the load of a built-in spring of the check valve 130. If, in contrast thereto, the discharging capacity of the compressor 101 is small and the discharge pressure Pd does not sufficiently get high (e.g., during the minimum capacity operation), the check valve 130 will be closed due to the biasing force of the spring and thereby the back-flow of refrigerant from a gas cooler 102 side to the discharge chamber 114 will be prevented. Note that the check valve 130 is closed while the compressor 101 is operating with the minimum capacity. However, the refrigerant discharged from the discharge chamber 114 is returned to the suction chamber 110 via the control valve 1 and the crankcase 116. Thus, the internal circulation of refrigerant gas within the compressor 101 is assured.

[0021] FIG. 2 is a cross-sectional view showing a structure of the control valve 1 according to an embodiment.

[0022] The control valve 1 is constituted by integrally assembling a valve unit 2 and a solenoid 3. The valve unit 2 has a body 5 of stepped cylindrical shape. Though the body 5 is formed of brass in the present embodiment, it may be formed of an aluminum alloy. The body 5 has ports 10, 12, and 14 in this order from top down. Of these ports, the port 10 is provided in an upper end of the body 5, and the ports 12 and 14 are each provided on a lateral side thereof. The port 10 functions as a "discharge chamber communication port" that communicates with the discharge chamber 114. The port 12 functions as a "crankcase communication port" (corresponding to a "control chamber communication port") that communicates with the crankcase 116. The port 14 functions as a "suction chamber communication port" that communicates with the suction chamber 110. Though the "control chamber" in the present embodiment is formed by a crankcase, it may be a pressure chamber separately provided within or outside the crankcase, in a modification.

[0023] In the body 5, a valve seat forming member 16 of stepped cylindrical shape is provided in a passage that communicates between the port 10 and the port 12. The valve seat forming member 16 is formed by quenching a stainless steel (e.g., SUS420), and has a hardness higher than that of the body 5. The valve seat forming member 16 is coaxially inserted into an upper portion of the body 5 and is secured such that the upper portion of the body 5 is swaged inward. The valve seat forming member 16 has a through-hole along an axis line, and a lower half of the through-hole forms a valve hole 18. A valve chamber 20, which communicates with the port 12, is formed below the valve seat forming member 16 in the body 5. In the body 5, the port 10, the valve hole 18, and an internal passage connecting the valve chamber 20 and the port 12 constitute the "supply passage" for introducing part of the discharged refrigerant from the discharge chamber 114 into the crankcase 116.

[0024] The lower half of the valve seat forming member 16 is of tapered shape such that the outside diameter thereof is gradually reduced from an upper part to a lower part thereof, and extends into the valve chamber 20. A valve seat 22 is formed on a lower end surface of the valve seat forming member 16. A valve element 24 is provided in the valve chamber 20 in such a manner as to face the valve seat 22 from below. The opening degree of a valve section is regulated by moving the valve element 24 toward and away from the valve seat 22.

[0025] In the present embodiment as described above, a soft material is used for a material constituting the body 5 and thereby its high processability is kept. At the same time, a material or member constituting the valve seat 22 (the valve seat forming member 16) is formed of a material having a higher degree of hardness, so that the wear and deformation of the valve seat 22 are prevented or suppressed. This allows the seating characteristics of the valve element 24 to be satisfactorily maintained. In other words, since in the present embodiment the control valve 1 is applied to the supercritical refrigeration cycle that uses carbon dioxide as the refrigerant, the discharge pressure Pd of the compressor 101 becomes extremely high. This may possibly cause cavitation to occur when the high-pressure refrigerant passes through the valve section while the valve section is open, or have a foreign material contained in the refrigerant hit the valve seat 22 at high speed. As a result, the valve seat 22 is more likely to be worn away and a deformation (erosion) is more likely to progress in the valve seat 22 if the valve seat 22 is formed of a soft material similar to the body 5. This in turn may possibly degrade the sealing property of the valve section and cause a variation in a control set value (set value). In contrast to this, in the present embodiment, the material strength (the degree of hardness) of the valve seat 22 and its surrounding area is raised, so that the aforementioned adverse effects can be prevented or suppressed. In the present embodiment, the valve seat forming member 16 is formed of a material whose Vickers hardness is 500 or above (preferably 700 or above).

[0026] A partition wall 26 is so provided that an internal space of the body 5 is divided into an upper space and a lower space. The valve chamber 20 is formed on an upper side of the partition wall 26, and a working chamber 28 is formed on a lower side thereof. The valve chamber 20 communicates with the crankcase 116 through the port 12. The working chamber 28 communicates with the suction chamber 110 through the port 14. A guide portion 30, which extends in a direction of axis line, is provided in a center of the partition wall 26. A guiding passage 32 is so formed as to run through the guide portion 30 along the axis line, and an elongated actuating rod 34 is slidably inserted to the guiding passage 32 in the direction of axis line. The valve element 24 is provided coaxially on an upper end of the actuating rod 34. The valve element 24 and the actuating rod 34 are formed integrally with each other by performing a cutting work on a stainless steel.

[0027] The guide portion 30 protrudes as a small bump on an upper surface side of the partition wall 26 and protrudes as a large protrusion on a lower surface side thereof. The guide portion 30 is of tapered shape such that the outside diameter thereof is gradually reduced from an upper part to a lower part thereof, and the guide portion 30 extends into the working chamber 28. With this configuration and arrangement, a sufficient length of the guiding passage 32 is ensured and the actuating rod 34 is stably supported. The valve element 24 and the actuating rod 34 operate and move integrally together with each other, and the valve element 24 closes and opens the valve section by touching and leaving the valve seat 22, respectively, on the upper end surface of the valve element 24. The hardness of the valve seat forming member 16 is sufficiently high. Thus, the valve seat 22 is hardly deformed by repeated seating of the valve element 24 on the valve seat 22, thereby ensuring the durability of the valve section.

[0028] A retaining ring 36 (E-ring) is fitted to a lower part of the actuating rod 34, and a discoidal spring support 38 is provided such that the movement of the lower part of thereof in a downward direction is restricted. A spring 40, which biases the actuating rod 34 downward (in a valve closing direction) (functioning as a "first biasing member"), is set between the spring support 38 and the partition wall 26. The spring 40 is a tapered spring where the diameter thereof is reduced starting from the lower surface of the partition wall 26 toward the spring support 38 located therebelow. Having the guide portion 30 formed in a tapered shape as described above allows the tapered-shape spring 40 to be arranged as described above. A lower part of the body 5 is a small-diameter part 42 and constitutes a coupling portion with the solenoid 3.

[0029] A filter member 44, which suppresses foreign materials from entering the port 10, is provided in an upper end opening of the body 5. Since the foreign material, such as metallic powders, may possibly be contained in the refrigerant discharged from the compressor 101, the filter member 44 prevents or suppresses the foreign material from entering the interior of the control valve 1. The filter member 44 is configured such that two sheets of metal meshes are vertically superimposed on each other.

[0030] The solenoid 3 includes a cylindrical core 50, a bottomed cylindrical sleeve 52 inserted around the core 50, a plunger 54, which is contained in the sleeve 52 and which is disposed opposite to the core 50 in the direction of axis line, a cylindrical bobbin 56 inserted around the sleeve 52, an electromagnetic coil 58 wound around the bobbin 56, a cylindrical casing 60, which is so provided as to cover the electromagnetic coil 58 from outside, a connecting member 62 of stepped cylindrical shape, which is assembled, between the core 50 and the casing 60, in a position above the bobbin 56, and an end member 64, which is so provided as to seal off a lower end opening of the casing 60.

[0031] The sleeve 52 and the plunger 54 are each formed of electromagnetic soft iron (SUY) excellent in magnetic characteristics. The electromagnetic soft iron is a material with low impurity content, high magnetic flux density, high magnetic permeability, and a small magnetic coercive force. More specifically, the electromagnetic soft iron as used herein is, for example, SUY-1 where the processability is excellent (because it has an appropriate degree of hardness) and a small magnetic coercive force (60 to 80 A/m) is obtained. Thereby, a necessary solenoidal force can be ensured even though the current supplied to the solenoid 3 is relatively low. This allows the electromagnetic coil 58 and eventually the control valve 1 to be downsized.

[0032] The sleeve 52, which is formed of a non-magnetic material, houses the plunger 54 in a lower half thereof. A circular collar 66 is embedded in the end member 64. The collar 66 is set, between the sleeve 52 and the casing 60, in a position below the bobbin 56. The casing 60, the connecting member 62 and the collar 66, which are each formed of a magnetic material, form a yoke of the solenoid 3. The valve unit 2 and the solenoid 3 are secured such that the small-diameter part 42 (lower end part) of the body 5 is press-fitted to an upper end opening of the connecting member 62. It is to be noted here that, in the present embodiment, the body 5, the valve seat forming member 16, the connecting member 62, the casing 60 and the end member 64 form a body for the whole control valve 1.

[0033] An insertion hole 67 is so formed as to run through the core 50 in a center thereof in the direction of axis line. And a shaft 68 is inserted into the insertion hole 67 in such a manner as to penetrate along the insertion hole 67. The shaft 68 is formed coaxially with the actuating rod 34 and supports the actuating rod 34 from below. The diameter of the shaft 68 is larger than that of the actuating rod 34. The plunger 54 is assembled to a lower half of the shaft 68. In the present embodiment, the shaft 68 and the actuating rod 34 constitute a "transmitting rod" that transmits the solenoidal force to the valve element 24.

[0034] The plunger 54 is coaxially supported by the shaft 68 in an upper portion of the plunger 54. A retaining ring 70 (E-ring) is fitted to a predetermined position in an intermediate part of the shaft 68 in the direction of axis line, and the retaining ring 70 works to restrict the movement of the plunger 54 in an upward direction. A communicating groove 71 formed in parallel with the axis line is provided on a lateral surface of the plunger 54. The communicating groove 71 forms a communicating path through which the refrigerant is made to pass between the plunger 54 and sleeve 52.

[0035] A ring-shaped shaft support member 72 is press-fitted in an upper end of the core 50, and an upper end of the shaft 68 is slidably supported by the shaft support member 72 in the direction of axis line. An outer periphery of the shaft support member 72 is partially notched and thereby a communicating path is formed between the core 50 and the shaft support member 72. Through this communicating path, the suction pressure Ps of the working chamber 28 is led into the interior of the solenoid 3, too.

[0036] The diameter of a lower end of the sleeve 52 is slightly reduced, and a ring-shaped shaft support member 76 (functioning as a "supporting member") is press-fitted to a reduced diameter portion 74 of the sleeve 52. The shaft support member 76 slidably supports a lower end part of the shaft 68. In other words, the shaft 68 is two-point supported by both the shaft support member 72 in an upper side thereof and the shaft support member 76 in a lower side thereof, so that the plunger 54 can be stably operated in the direction of axis line. An outer periphery of the shaft support member 76 is partially notched and thereby a not-shown communicating path is formed between the sleeve 52 and the shaft support member 76. The suction pressure Ps introduced into the solenoid 3 fills the interior of the sleeve 52 through the communicating path between the core 50 and the shaft 68, the communicating path between the plunger 54 and the sleeve 52, and the not-shown communicating path between the shaft support member 76 and the sleeve 52.

[0037] A spring 78 (functioning as a "second spring") that biases the plunger 54 in an upward direction, namely in a valve closing direction, is set between the shaft support member 76 and the plunger 54. In other words, as the spring load, the valve element 24 receives the net force of a force exerted by the spring 40 in a valve opening direction and a force exerted by the spring 78 in a valve closing direction. However, the spring load of the spring 40 is larger than that of the spring 78. Thus, the overall spring load of the springs 40 and 78 works in a valve opening direction. The spring load thereof can be set by adjusting the press-fitting position of the shaft support member 76 in the sleeve 52. The press-fitting position thereof can be fine-adjusted such that a bottom center of the sleeve 52 is deformed in the direction of axis line by using a predetermined tool after the shaft support member 76 has been temporarily press-fitted to the sleeve 52.

[0038] A pair of connection terminals 80 connected to the electromagnetic coil 58 extend from the bobbin 56 and are led outside by passing through the end member 64. Note that only one of the pair of connection terminals 80 is shown in FIG. 2 for convenience of explanation. The end member 64 is mounted in such a manner as to seal the entire structure inside the solenoid 3 contained in the casing 60 from below. The ends of the connection terminals 80 are led out from the end member 64 and connected to a not-shown external power supply. The end member 64 also functions as a connector portion through which the connection terminal 80 is exposed.

[0039] The control valve 1 configured as above is secured into a not-shown mounting hole formed in the compressor 101 via a washer. A plurality of O-rings, which are set between the mounting holes and the control valve 1 and which achieve the sealing capability, are fitted on an outer peripheral surface of the control valve 1. Annular grooves are formed on peripheries of the body 5 above and below the port 12, respectively, and O-rings 82 and 84 are fitted on the annular grooves. An annular groove is also formed on a periphery of the connecting member 62 below the port 14, and an O-ring 86 is fitted on the annular groove. Furthermore, an O-ring 88 is fitted on a connection area where the casing 60 and the end member 64 are connected.

[0040] FIG. 3 is a partially enlarged cross-sectional view of the upper half of FIG. 2.

[0041] The filter member 44 is configured such that the two sheets of metal meshes 46 and 48 are superimposed on each other in the thickness direction. In other words, the mesh 46 is so arranged as to face the outer side of the body 5, and the mesh 48 is so arranged as to face the inner side thereof. The reason why the meshes 46 and 48 are made from metal is that the filter member 44 is placed on a high pressure side of the supercritical refrigeration cycle; thus, if resin-made meshes are used, the pressure resistance strength thereof will be insufficient.

[0042] The meshes 46 and 48 are both formed in a circular sheet shape, but differ in mesh size and in rigidity (stiffness properties) from each other. In other words, the mesh 46 has a finer mesh size than the mesh 48; the mesh 46 has a smaller porosity than the mesh 48. On the other hand, the mesh 48 has a larger wire diameter than the mesh 46 and has a higher rigidity than the mesh 46. This is to achieve a high filtering function by the mesh 46 and ensure (reinforce) the strength of the entire filter member 44 by the mesh 48.

[0043] The filter member 44 is placed in such a manner as to be inserted in the upper end opening of the body 5, and then the filter member 44 is secured such that an upper end of the body 5 is swaged inward. In this manner, the filter member 44 is of a simple structure such that the two sheets of meshes are directly placed on each other and are fixed directly to the body 5, so that the component cost and the manufacturing cost can be suppressed. Since, as shown in FIG. 3, the filter member 44 is placed inside the body 5, any deformation and/or damage caused through contact with an external structural object or the like can be prevented or suppressed.

[0044] The diameter of a through-hole 90, which is formed in a center of the valve seat forming member 16, is reduced in a lower half thereof. This reduced diameter portion of the through-hole 90 forms the valve hole 18. In other words, the upper half of the through-hole 90 is a large-diameter part 92, whereas the lower half thereof is a small-diameter part 94. And the small-diameter part 94 forms the valve hole 18. A connection area in between the large-diameter part 92 and the small-diameter part 94 is a tapered surface where the inside diameter thereof is gradually reduced downward. The diameter of the through-hole 90 is reduced in stages from an upstream side to a downstream side.

[0045] A bleed hole 96 in parallel with the through-hole 90 is formed in radially outward direction of the through-hole 90 in the valve seat forming member 16. The bleed hole 96 is used to ensure the circulation of oil in the compressor 101 by delivering a minimum required amount of refrigerant to the crankcase 116 even when the valve section is closed. The refrigerant contains a lubricating oil in order to ensure a stabilized operation of the compressor 101, and the bleed hole 96 is to ensure the oil circulation inside and outside the crankcase 116.

[0046] The bleed hole 96 is formed such that a leak passage 98 located in an upper part thereof and a communication passage 99 located in a lower part thereof are connected together. The inside diameter of the leak passage 98 is of a size to a degree that the refrigerant is made to leak therethrough, and the inside diameter thereof is fairly smaller than that of the valve hole 18. The inside diameter of the communication passage 99 is smaller than that of the large-diameter part 92 of the through-hole 90 and larger than that of the small-diameter part 94 thereof. In a modification, the inside diameter of the communication passage 99 may be greater than or equal to that of the large-diameter part 92 of the through-hole 90 or may be less than or equal to that of the small-diameter part 94 thereof.

[0047] A connection area of the leak passage 98 and the communication passage 99 is a tapered surface where the inside diameter thereof is gradually enlarged downward. The diameter of the bleed hole 96 is enlarged in stages from an upstream side to a downstream side. An annular raised portion 150 is formed on a top surface of the valve seat forming member 16 in such a manner as to surround the through-hole 90, and the raised portion 150 is of a stepped shape such that a radially inward portion and a radially outward portion of the valve seat forming member 16 are lower than the raised portion 150. The width of the raised portion 150 is sufficiently small and is less than or equal to that of the valve hole 18 in the present embodiment. The leak passage 98 is opened upward in a position of the raised portion 150.

[0048] As described above, the bleed hole 96 is formed such that an inlet of refrigerant has a small diameter and the inlet thereof is opened on the top surface of a stepped shape. Thus, the entry of foreign material through the bleed hole 96 is prevented or suppressed. In other words, if a foreign material, whose size is smaller than the mesh size (mesh width) of the filter member 44, enters the port 10, it is highly improbable that the foreign material will enter through the bleed hole 96. This is because the width of the raised portion 150 is sufficiently small and the size of the inlet of the bleed hole 96 is smaller than the width of the raised portion 150. If the foreign material hits the raised portion 150, it is highly probable that the foreign material is dropped to a lower position inside or outside the raised portion 150. In particular, even though the refrigerant flows through the bleed hole 96 when the valve section is closed, the foreign material contained in the refrigerant is unlikely to be led into the bleed hole 96. If the foreign material enters the port 10 when the valve section is open, most of such foreign material will pass through the valve hole 18 and be discharged from the port 12.

[0049] Also, in the valve chamber 20, the guide portion 30 protrudes in a central part of the upper surface of the partition wall 26 and thereby an annular groove 152 is formed on the periphery of this protrusion (the guide portion 30). The outside diameter of the valve element 24 is slightly larger than that of the actuating rod 34 located immediately beneath the valve element 24. Thus, if the foreign material enters the valve chamber 20 through the valve hole 18, it is highly improbable that the foreign material will enter a sliding portion of the actuating rod 34 relative to the guiding passage 32. In the event that the foreign material passes through the valve hole 18, most of such the foreign material will be discharged through the port 12 or stay on in the annular groove 152 even though it should remain in the valve chamber 20. Thus, the remaining foreign material is less likely to enter a spacing or gap between the actuating rod 34 and the guiding passage 32. In other words, the annular groove 152 can function to trap the foreign material therein. Hence, this structure prevents the valve element 24 from being locked as a result of the entanglement of foreign material in the sliding portion of the actuating rod 34 relative to the guiding passage 32.

[0050] In the present embodiment, the pressure sensitivity of the valve element 24 is optimally set such that a seal section diameter **a** (the inside diameter of the valve hole 18) in the valve section of the valve element 24 is slightly (e.g., by a very small amount) larger than a diameter **b** of the sliding portion of the actuating rod 34 (**a>b**). In other words, such the setting as this increases the extent of contribution of the crank pressure Pc in a valve closing direction at the time the valve section is opened, thereby making it slightly difficult for the valve section to be opened. Thereby, the differential pressure (Pd - Ps) slowly rises and the effect of the crank pressure Pc is raised as compared with the case where **a=b.** As a result, the actuation responsiveness of the wobble plate (cam plate) of the compressor 101 is lowered so as to prevent or suppress the control hunting occurring when the valve section is opened. It is to be noted here that, for example, the technique disclosed in Japanese Patent Application Publication No. 2006-57506 can be used to adjust the pressure sensitivity.

[0051] In the present embodiment, as described earlier, the guide portion 30 protrudes as a larger protrusion on a working chamber 28 side than a valve chamber 20 side. Thereby, a lower end of the actuating rod 34 can protrude from a lower end position of the body 5 (i.e., a lower end opening of the small-diameter part 42). This enables the retaining ring 36 to be easily mounted to the actuating rod 34. In other words, in order for the retaining ring 36 to be fitted to the actuating rod 34, the actuating rod 34 must first be inserted from the valve chamber 20 side. This is because the outside diameter of the valve element 24 is larger than the size of the guiding passage 32. On the other hand, in order for the retaining ring 36 to be fitted to the actuating rod 34, a fitting part formed in the actuating rod 34 needs to be exposed from an opening end of the body 5 or at least the fitting part needs to be positioned near the opening end thereof in consideration of the workability. For this reason, if the guide portion 30 extends (protrudes) uniformly both above and below the partition wall 26, the actuating rod 34 needs to be unnecessarily made longer, which is not preferable at all. In the light of this, in the present embodiment, the guide portion 30 is configured such that the guide portion 30 is positioned in a lower part of the body 5. This configuration and arrangement ensure a more stabilized guiding function of the guide portion 30 and maintain an excellent workability when the retaining ring 36 is to be mounted. Since the actuating rod 34 will not be unnecessarily long, the body 5 and eventually the control valve 1 are made smaller-sized.

[0052] Furthermore, in the present embodiment as described above, the guide portion 30 and the spring 40 are each taper-shaped such that the outside diameter thereof becomes gradually smaller downward. Thus, a lower half of the spring 40 is contained in an upper end opening of the core 50, and the outside diameter of the small-diameter part 42 is made as small as possible. Thereby, the outside diameter of the connecting member 62 is made smaller, and an O-ring whose outside diameter is smaller can be selected as the O-ring 86. As a result, when the control valve 1 is to be mounted through the mounting holes of the compressor 101, the effect of the refrigerant pressure acting in a direction opposite to a mounting direction is reduced. That is, an area below the O-ring 86 has an atmospheric air pressure; if the size of the O-ring 86 is large, a fixing structure having a high pressure withstanding property need be implemented in order to prevent the control valve 1 from fall off. In this regard, the O-ring 86 can be made small in the present embodiment and therefore it suffices that the control valve 1 has a simple fixing structure such as a washer.

[0053] In the above-described configuration, the diameter of the actuating rod 34 is slightly smaller than the inside diameter of the valve hole 18 but is of a size approximately identical thereto. Thus, the effect of the crank pressure Pc acting on the valve element 24 in the valve chamber 20 is almost canceled out. As a result, the differential pressure (Pd - Ps) between the discharge pressure Pd and the suction pressure Ps practically acts on the valve element 24 for a pressure-receiving area having approximately the same size as that of the valve hole 18. The valve element 24 operates and moves such that the differential pressure (Pd - Ps) is kept at a preset differential pressure set by a control current supplied to the solenoid 3.

[0054] A basic operation of the control valve for the variable displacement compressor is now explained.

[0055] Now refer back to FIG. 2. In the control valve 1, when the solenoid 3 is turned off, the valve element 24 gets separated away from the valve seat 22 by the net force of the springs 40 and 78 in a valve opening direction with the result that the valve section is remained at a fully opened state. At this time, a high-pressure refrigerant having the discharge pressure Pd introduced into the port 10 from the discharge chamber 114 of the compressor 101 passes through the fully-opened valve section and then flows into the crankcase 116 through the port 12. As a result, the crank pressure Pc is raised and the compressor 101 carries out a minimum capacity operation where the discharging capacity is the minimum.

[0056] When, on the other hand, at the startup of the automotive air conditioner or when the cooling load is the maximum, the value of current supplied to the solenoid 3 is the maximum and the plunger 54 is attracted by a maximum suction force of the core 50. At this time, the actuating rod 34 (including the valve element 24), the shaft 68 and the plunger 54 operate and move integrally altogether in a valve closing direction, and the valve element 24 is seated on the valve seat 22. The crank pressure Pc drops by this valve closing movement and therefore the compressor 101 carries out a maximum capacity operation where the discharging capacity is the maximum.

[0057] When the value of current supplied to the solenoid 3 is set to a predetermined value while the capacity is being controlled, the actuating rod 34 (including the valve element 24), the shaft 68 and the plunger 54 operate and move integrally altogether. At this time, the valve element 24 stops at a valve-lift position. This valve-lift position is a position where five loads/forces are all balanced thereamong. Here, the five loads/forces are the spring load of the spring 40 that biases the actuating rod 34 in a valve opening direction, the spring load of the spring 78 that biases the plunger 54 in a valve opening direction, the load of the solenoid 3 that biases the plunger 54 in a valve closing direction, the force by the discharge pressured Pd that the valve element 24 receives in a valve opening direction, and the force by the suction pressure Ps that the valve element 24 receives in a valve closing direction.

[0058] If, in this balanced state, the rotating speed of the compressor 101 rises simultaneously with an increased engine speed and thereby the discharging capacity increases, the differential pressure (Pd - Ps) will increase and then the force in a valve opening direction will exert on the valve element 24. As a result, the valve element 24 further uplifts its position and thereby the flow rate of refrigerant flowing from the discharge chamber 114 to the crankcase 116 increases. This, in turn, causes the crank pressure Pc to rise and then the compressor 101 operates in a direction such that the discharging capacity is reduced. Then the compressor 101 is controlled such that the differential pressure (Pd - Ps) becomes the preset differential pressure. If the engine speed drops, the compressor 101 operates in a manner reverse to the aforementioned operation and then the compressor 101 is controlled such that the differential pressure (Pd - Ps) becomes the preset differential pressure.

[0059] FIG. 4 is a graph showing an allowable range of combination of a release passage diameter and a leak passage diameter when carbon dioxide is used as refrigerant. In FIG. 4, the horizontal axis indicates the diameter (mm) of the leak passage and the vertical axis indicates the diameter (mm) of the release passage. A hatched part in FIG. 4 indicates the allowable range of combination of the passage diameters determined on the basis of experimental results.

[0060] Verification conducted by the inventors has shown that combination of the release passage diameter and the leak passage diameter set within the allowable range in FIG. 4 can ensure switching to and maintenance of the maximum capacity operation while ensuring oil circulation in the compressor 101. The "allowable range" used herein refers to a range in which a differential pressure (Pc - Ps) between the crank pressure Pc and the suction pressure Ps can be set at not higher than 0.5 MPa while the valve section is closed, that is, a range in which the wobble plate is tilted greatly so that the maximum capacity operation can be carried out.

[0061] It is assumed herein that the preset differential pressure (Pd - Ps) is 10 MPaG and that the suction pressure Ps is controlled so as not to become lower than 3.5 MPa. Note that the lower limit of 3.5 MPa of the suction pressure Ps is a pressure at which the temperature of carbon dioxide is 0°C. Thus, as a result of setting the suction pressure Ps at the lower limit or higher, it is possible to prevent the temperature of carbon dioxide from becoming lower than 0°C and to prevent freezing from being caused in the evaporator 104 and the like. Note that the leak passage diameter for sufficiently ensuring oil circulation is set within the allowable range as appropriate.

[0062] As shown in FIG. 4, the allowable range can be met by forming the bleed hole 96 such that the following expression (1) is satisfied, where X represents the diameter of the leak passage 98 and Y represents the diameter of the release passage (the orifice 119):

$$Y > 2.145X + 0.24 \quad \dots (1)$$

where X > 0.

[0063] In general, the manufacturer of compressors is often different from the manufacturer of control valves. In such a case, the manufacturer of the control valve 1 may acquire information on the diameter of the release passage (the orifice 119) from the manufacturer of the compressor 101 and set the diameter of the leak passage 98 so that the expression (1) is satisfied.

[0064] FIGS. 5A and 5B are graphs showing differential pressure characteristics depending on the combination of the release passage diameter and the leak passage diameter. FIGS. 5A and 5B show the relation between the differential pressure (Pd - Ps) to be controlled and the differential pressure (Pc - Ps) obtained therefrom at each value I (I_A to I_D) of current supplied to the solenoid 3. In FIG. 5A, solid lines indicate a case (case 1) where the leak passage diameter Φ1 and the release passage diameter Φ2 are both large, and broken lines indicate a case (case 2) where the leak passage diameter Φ1 and the release passage diameter Φ2 are both small. In FIG. 5B, one-dot chain lines indicate a case (case 3) where the leak passage diameter Φ1 is large while the release passage diameter Φ2 is small, and two-dot chain lines indicate a case (case 4) where the leak passage diameter Φ1 is small while the release passage diameter Φ2 is large. In each of FIGS. 5A and 5B, the horizontal axis indicates the differential pressure (Pd - Ps) and the vertical axis indicates the differential pressure (Pc - Ps). The magnitudes of the supplied current values I satisfy the following relation: $I_A < I_B < I_C < I_D$.

[0065] FIGS. 5A and 5B show the following. First, as a whole, as the supplied current value I is larger, points (hereinafter referred to as "valve opening points") p_A to p_D where the differential pressure (Pc - Ps) starts to change sharply relative to the change in the differential pressure (Pd - Ps) are greater. This means that the valve opens less easily and the maximum capacity operation is maintained more easily as the supplied current value I is larger. Note that an increase in the differential pressure (Pc - Ps) before the differential pressure (Pd - Ps) reaches a valve opening point indicates that refrigerant is also introduced into the crankcase 116, even while the valve is closed, via the leak passage 98.

[0066] In addition, FIG. 5A shows that the change in the differential pressure (Pc - Ps) relative to the change in the differential pressure (Pd - Ps) is relatively larger before the valve opening point and relatively smaller after the valve opening point in case 1 than that in case 2. This indicates that an effect of accelerated pressure increase is produced owing to the large leak passage diameter Φ1 before the valve opening point while an effect of accelerated pressure increase is produced owing to the small release passage diameter Φ2 after the valve opening point.

[0067] Furthermore, FIG. 5B shows that the change in the differential pressure (Pc - Ps) relative to the change in the differential pressure (Pd - Ps) is large in the valve closed state in case 3. This indicates that if the leak passage diameter Φ1 is excessively larger than the release passage diameter Φ2, the effect of accelerated pressure increase becomes great, the differential pressure (Pc - Ps) may exceed 0.5 MPa before the valve opens, and the maximum capacity operation may not be maintained. In contrast, FIG. 5B shows that the change in the differential pressure (Pc - Ps) relative to the change in differential pressure (Pd - Ps) in the valve closed state is small in case 4. This indicates that if the leak passage diameter Φ1 is excessively smaller than the release passage diameter Φ2, the effect of decelerated pressure increase becomes great, maintenance of the maximum capacity operation in the valve closed state can be ensured, but sufficient oil circulation may not be achieved.

[0068] In view of the above, the combination of the leak passage diameter Φ1 and the release passage diameter Φ2 may be set so as to meet the allowable range shown in FIG. 4 with the differential control characteristics shown in FIGS. 5A and 5B taken into consideration.

[0069] The description of the present invention given above is based upon illustrative embodiments. These embodiments are intended to be illustrative only and it will be obvious to those skilled in the art that various modifications could be further developed within the technical idea underlying the present invention.

[0070] FIG. 6 is a graph showing an allowable range of combination of the release passage diameter and the leak passage diameter when an alternative (HFC-134a) for chlorofluorocarbon (CFC) is used as refrigerant. In FIG. 6, the horizontal axis indicates the diameter (mm) of the leak passage and the vertical axis indicates the diameter (mm) of the release passage. A hatched part in FIG. 6 indicates the allowable range of combination of the passage diameters determined on the basis of experimental results.

[0071] Verification conducted by the inventors has shown that, when HFC-134a is used as refrigerant, combination of the release passage diameter and the leak passage diameter set within the allowable range in FIG. 6 can ensure switching to and maintenance of the maximum capacity operation while ensuring oil circulation in the compressor 101. The "allowable range" used herein refers to a range in which the differential pressure (Pc - Ps) between the crank pressure Pc and the suction pressure Ps can be set at not higher than 0.05 MPa while the valve section is closed, that is, a range in which the maximum capacity operation can be carried out when HFC-134a is used.

[0072] It is assumed herein that the preset differential pressure (Pd - Ps) is 1.5 MPaG and that the suction pressure Ps is controlled so as not to become lower than 0.2 MPa. Note that the lower limit of 0.2 MPa of the suction pressure Ps is a pressure at which the temperature of HFC-134a is 0°C. Note that the leak passage diameter for sufficiently ensuring oil circulation is set within the allowable range as appropriate.

[0073] As shown in FIG. 6, the allowable range can be met by forming the bleed hole 96 such that the following expression (2) is satisfied, where X represents the diameter of the leak passage 98 and Y represents the diameter of the release passage (the orifice 119):

$$Y > 2.3X + 0.27 \ldots (1)$$

where $X > 0$.

[0074] Note that the range of the expression (2) is included in the range of the expression (1).

[0075] In the above-described embodiments, an example where the release passage is formed in a housing of the compressor 101 is presented. In a modification, the release passage may be formed in the control valve 1. For example, the body 5 may be provided with a second crankcase communication port, through which the crank pressure Pc can be introduced from the crankcase 116, separately from the port 12, and may be provided with a communication passage through which the second crankcase communication port and the port 14 communicate with each other. In addition, the orifice 119 may be provided in the communication passage.

[0076] In the above-described embodiments, an example is shown where the entire valve seat forming member 16 is formed of a material having a high degree of hardness. Instead, only the valve seat 22 and a surrounding part thereof may be formed of a material having a high degree of hardness. For example, the valve seat forming member may be formed of a material having the same or equivalent softness as the body 5, a fitting hole may be formed in a center of end part thereof and then a valve seat member having a high degree of hardness may be press-fitted to the fitting hole. An end surface of this valve seat member may function as the valve seat 22.

[0077] In the above-described embodiments, an exemplary structure is shown where the actuating rod 34 and the shaft 68 are manufactured as separate units and then they are coupled together such that both of them are coaxially abutted against each other in the direction of axis line, thereby constituting the thus coupled one as a transmitting rod for transmitting the solenoidal force to the valve element 24. In a modification, the actuating rod 34 and the shaft 68 may be integrally formed as a single element.

[0078] In the above-described embodiments, a structure is shown where, in valve seat forming member 16, the inlet of refrigerant in the bleed hole 96 has a small diameter and the inlet thereof is opened on the top surface of the stepped shape (see FIG. 3). This structure markedly achieves the function of suppressing the entry of foreign material in the refrigeration cycle where the refrigerant pressure gets high as in the above-described embodiments. In other words, the higher the discharge pressure Pd is, the more it is likely that the foreign material enters the port 10 by passing through the filter member 44. In spite of this problem, this function of suppressing the entry of foreign material reduces at least the possibility that the foreign material will reach up to the valve chamber 20 through the bleed hole 96. This eventually leads to maintaining the excellent control characteristic of the control valve 1 in a high pressure environment.

[0079] In the above-described embodiments, an example is shown where the raised portion 150 is formed annularly on the top surface of the valve seat forming member 16. It goes without saying that a shape other than this may be employed. For example, the raised portion may be formed only around the inlet of refrigerant of the bleed hole 96. Although, in the above described-embodiments, an example is shown where only a single bleed hole 96 is formed, a plurality of bleed holes 96 may be formed in a plurality of positions. In such a case, too, the inlet of refrigerant of each bleed hole 96 may preferably be provided on the top surface of the raised portion (stepped shape).

[0080] The present invention is not limited to the above-described embodiments and modifications only, and those components may be further modified to arrive at various other embodiments without departing from the scope of the invention. Also, various other embodiments may be further formed by combining, as appropriate, a plurality of structural components disclosed in the above-described embodiments and modification. Also, one or some of all of the components exemplified in the above-described embodiments and modifications may be left unused or removed.

Claims

1. A control valve (1), for a variable displacement compressor (101), applied to the compressor (101) for compressing refrigerant led into a suction chamber (110) and discharging the compressed refrigerant from a discharge chamber (114), part of the discharged refrigerant being introduced into a control chamber (116) via a supply passage while part of refrigerant in the control chamber (116) being released to the suction chamber (110) via a release passage (119), so that pressure in the control chamber (116) is regulated in such a manner that a discharging capacity of the compressor (101) is regulated, the control valve (1) con-trolling the discharging capacity by regulating a flow rate of refrigerant flowing through the supply passage, the control valve (1) comprising:

   a body (5) having a discharge chamber communication port (10) communicating with the discharge chamber (114), a control chamber communication port (12) communicating with the control chamber (116), and a suction chamber communication port (14) communicating with the suction chamber (110), an internal passage connecting the discharge chamber communication port (10) and the control chamber communication port (12) constituting the supply passage, the body (5) having a valve hole (18) provided in the internal passage;
   a valve element (24) for adjusting an opening degree of a valve section by moving toward and away from the valve hole (18), the valve element (24) receiving a differential pressure (Pd - Ps) between a discharge pressure (Pd) of the discharge chamber (114) and a suction pressure (Ps) of the suction chamber (110) in a valve opening direction; and
   a solenoid (3), provided in the body (5), which generates a solenoidal force with which to drive the valve element (24) in a valve closing direction according to an amount of current supplied thereto,
   wherein the valve element (24) autonomously operates so that the differential pressure (Pd - Ps) between the discharge pressure (Pd) and the suction pressure (Ps) is kept at a preset differential pressure according to a value of current supplied to the solenoid (3),
   wherein the control valve (1) further has a bleed hole (96) is provided in the body (5) separately from the valve hole (18), has a leak passage (98) whose diameter is smaller than that of the valve hole (18), and permits refrigerant to be leaked from the discharge chamber communication port (10) to the control chamber communication port (12) even while the valve section is closed, and
   wherein the bleed hole (96) is formed such that the following expression (1) is satisfied, where X represents the diameter of the leak passage (98) and Y represents a diameter of the release passage (119):

$$Y > 2.145X + 0.24 \quad \ldots (1).$$

2. A control valve (1), for a variable displacement compressor, according to claim 1, wherein the body (5) has a valve chamber (20) formed between the valve hole (18) and the control chamber communication port (12), wherein the valve element (24) opens and closes the valve section by moving toward and away from the valve hole (18) from a valve chamber (20) side,
   wherein the bleed hole (96) is provided radially outward of the valve hole (18) in the body (5), and is formed such that the leak passage (98) and a communication passage (99), whose diameter is larger than that of the leak passage (98), are connected together in a direction of axis line, and
   wherein the communication passage (99) is open toward the valve chamber (20).

3. A control valve (1), for a variable displacement compressor, according to claim 2, wherein the body (5) has a working chamber (28) through which the suction chamber communication port (14) and an inside of the solenoid (3) communicate with each other, a partition wall (26) that separates the valve chamber (20) from the working chamber (28), and a guiding passage (32) formed in the partition wall (26),
   the control valve (1) further comprising a transmitting rod (34, 68) for transmitting the solenoidal force to the valve element (24), the transmitting rod (34, 68) being slidably supported in the guiding passage (32), one end side of the transmitting rod (34, 68) being integrated with the valve element (24) in the valve chamber (20), the other side thereof being coupled to the solenoid (3).

4. A control valve (1), for a variable displacement compressor, according to any one of claim 1 to claim 3, wherein the control valve (1) is applied to a refrigeration cycle in which a refrigeration operation is performed in a supercritical range exceeding a critical temperature of the refrigerant.

FIG.1

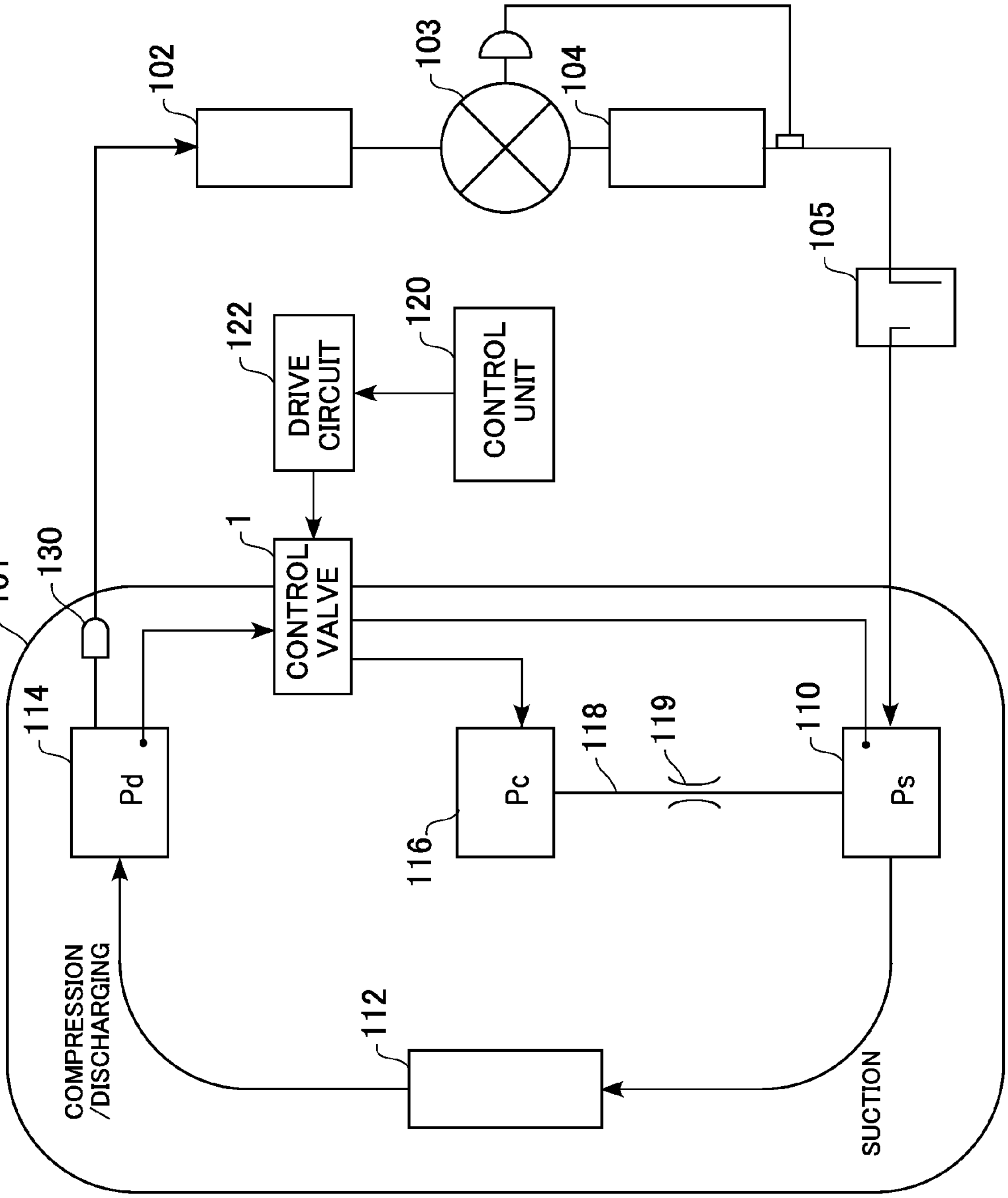
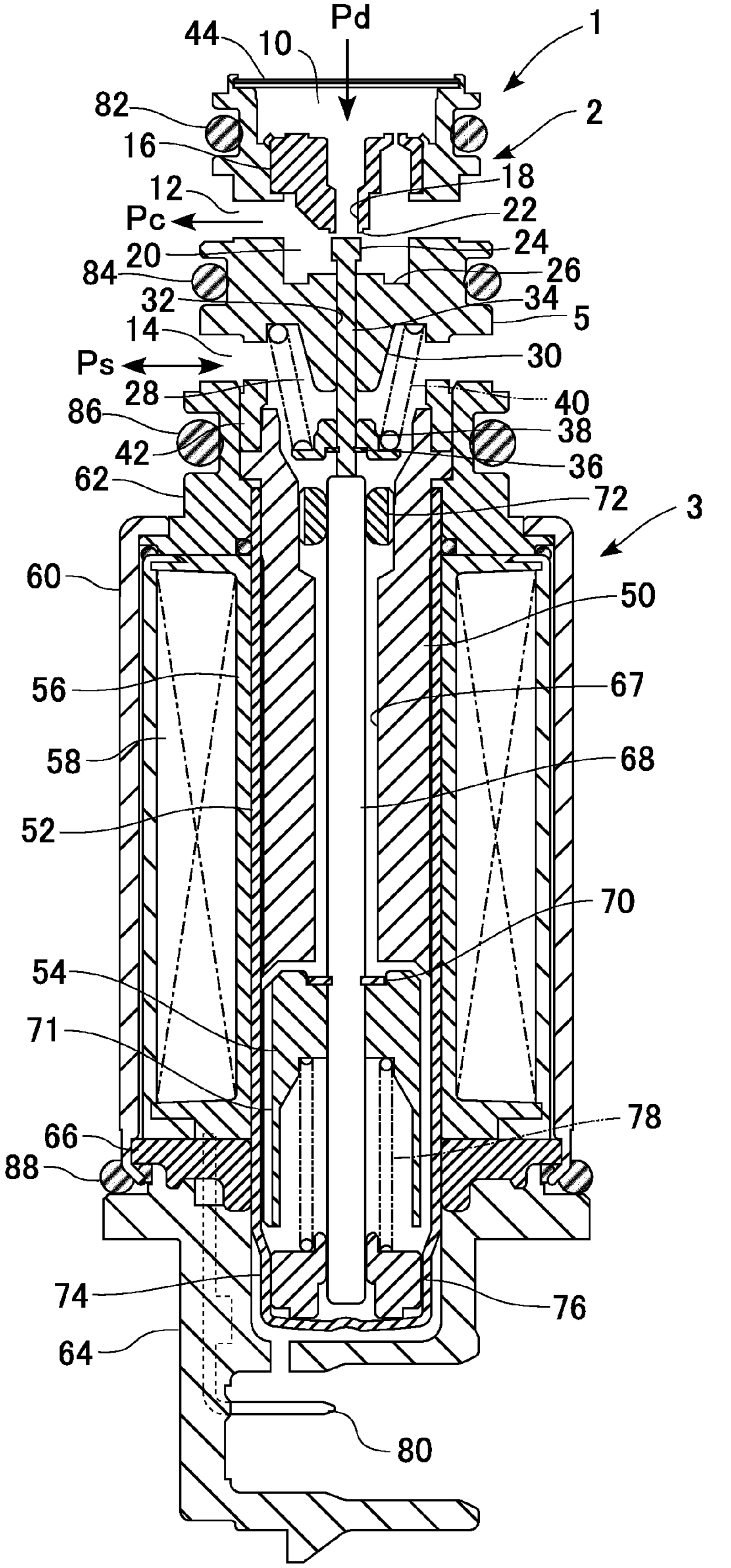

101
130
102
103
104
105
122
DRIVE CIRCUIT
120
CONTROL UNIT
1
CONTROL VALVE
114
Pd
116
Pc
118
119
110
Ps
112
COMPRESSION /DISCHARGING
SUCTION

FIG.2


Pd
1
2
3
5
10
12
14
16
18
20
22
24
26
28
30
32
34
36
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42
44
50
52
54
56
58
60
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78
80
82
84
86
88
Pc
Ps

FIG.3

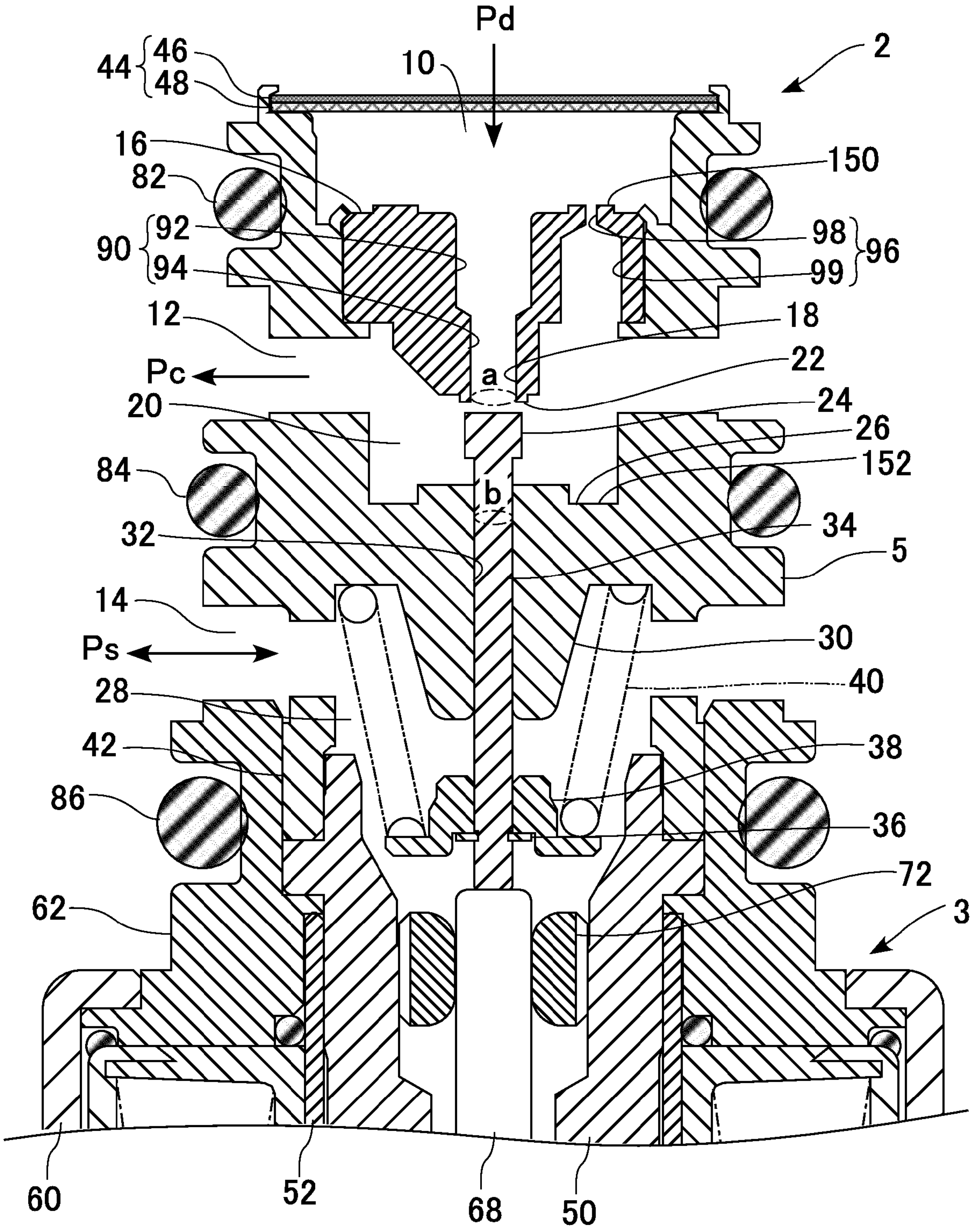

Pd
2
44
46
48
10
16
150
82
90
92
94
98
96
99
12
18
Pc
a
22
20
24
26
84
152
b
32
34
5
14
30
Ps
28
40
42
38
86
36
72
62
3
60
52
68
50

FIG.4

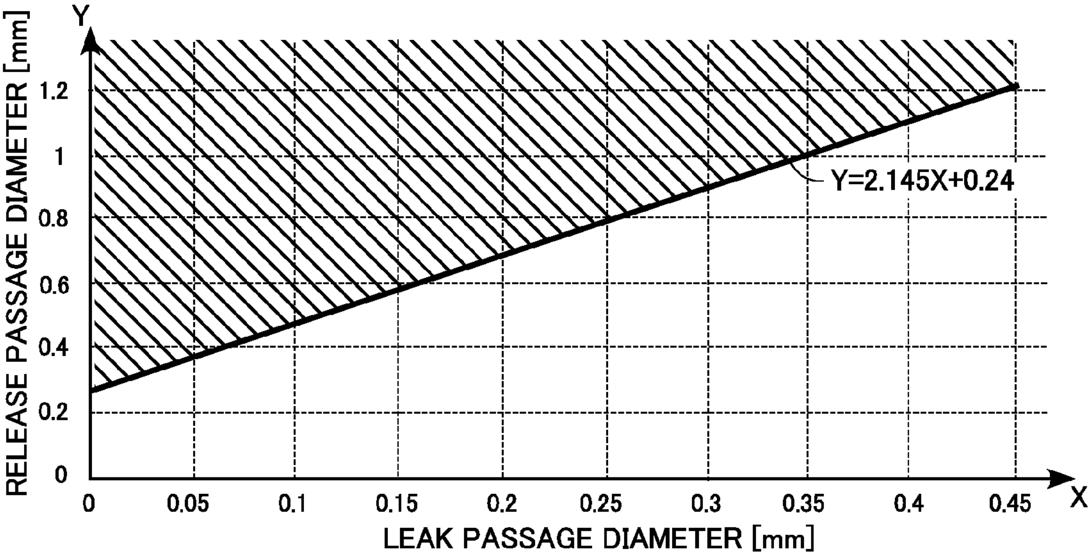

Y
RELEASE PASSAGE DIAMETER [mm]
1.2
1
0.8
0.6
0.4
0.2
0
Y=2.145X+0.24
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
0.45
X
LEAK PASSAGE DIAMETER [mm]

FIG.5A

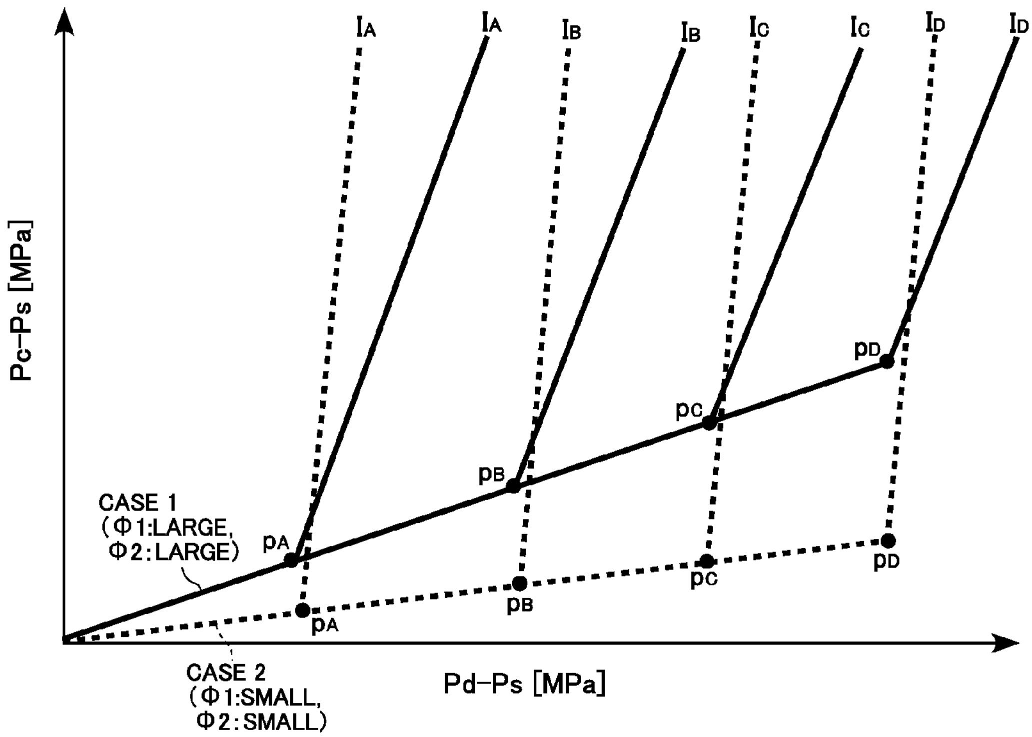

IA
IA
IB
IB
IC
IC
ID
ID
Pc-Ps [MPa]
pD
pC
pB
CASE 1
(Φ1:LARGE,
Φ2:LARGE)
pA
pC
pD
pB
pA
CASE 2
(Φ1:SMALL,
Φ2:SMALL)
Pd-Ps [MPa]

FIG.5B

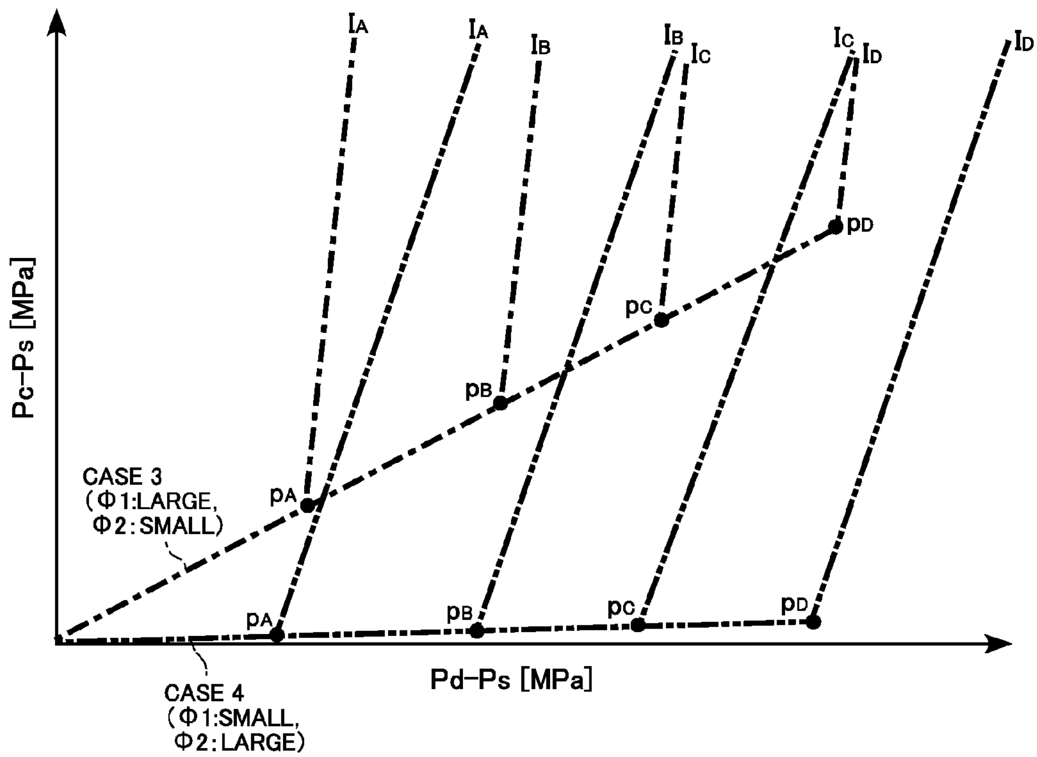
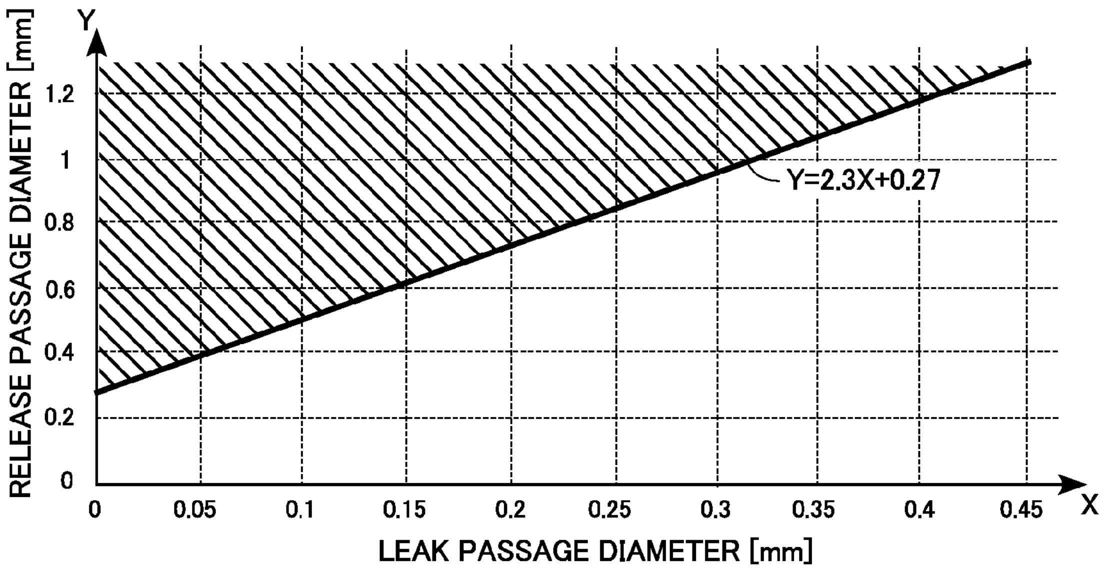

IA
IA
IB
IB
IC
IC
ID
ID
Pc-Ps [MPa]
pD
pC
pB
CASE 3
(Φ1:LARGE,
Φ2:SMALL)
pA
pA
pB
pC
pD
CASE 4
(Φ1:SMALL,
Φ2:LARGE)
Pd-Ps [MPa]

FIG.6


Y
RELEASE PASSAGE DIAMETER [mm]
1.2
1
0.8
0.6
0.4
0.2
0
Y=2.3X+0.27
0
0.05
0.1
0.15
0.2
0.25
0.3
0.35
0.4
0.45
X
LEAK PASSAGE DIAMETER [mm]

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EUROPEAN SEARCH REPORT

Application Number
EP 15 17 3253

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Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (IPC)
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