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(54) Compressor

(57) A compressor comprises a compressor housing which comprises a suction port 29, a discharge port 30, a suction chamber 26, and a discharge chamber 27. The suction chamber and the discharge chamber are defined in the compressor housing and are connected to the suction port and the discharge port, respectively. Pistons 13 are reciprocally moved in the compressor by means of rotating a driving shaft 16 to compress the refrigerant introduced through the suction port. Compressed refrigerant is discharged through the discharge chamber. The compressor further comprises a flow path control mechanism 41 that controls an opening area of the suction port depending on a difference between the pressures in the discharge chamber and at the suction port. A pressure introduction control valve 47 leads the compressed refrigerant into the flow path control mechanism 41 through a bypass duct 48 depending on the driving condition of a vehicle to forcefully minimize the opening area of the suction port.

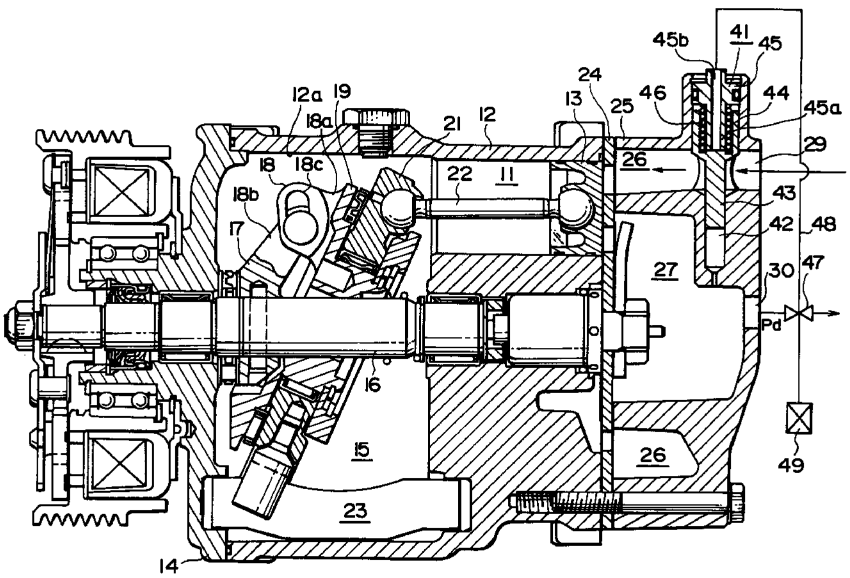

FIG. 3

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Description

Background of the Invention:

The present invention relates to a compressor and, more particularly, to a compressor suitable for a refrigerant compressor in a vehicle air conditioning system.

The compressor of this type is disclosed in, for example, United States Patent No. 4,905,477. This compressor is also referred to as a variable capacity compressor. It comprises a compressor housing. The compressor housing has a suction port, a discharge port, a suction chamber, and a discharge chamber. The suction chamber and the discharge chamber are defined in the compressor housing and are connected to the suction port and the discharge port, respectively. A plurality of pistons are reciprocally moved in the compressor by means of rotating a main shaft to compress the refrigerant introduced through the suction port. Compressed refrigerant is discharged through the discharge chamber. The compressor further comprises a flow path control mechanism that controls an opening area of the suction port depending on a difference between the pressures in the discharge chamber and at the suction port.

As described more in detail below, the opening area of the suction port is minimized at the beginning of the compressor operation with the compressor having a conventional flow path control mechanism. This limits an initial driving torque of the compressor to be minimal. However, the suction port has a maximum opening area when the compressor operates in the normal state. It may be difficult or even impossible to properly control the load on the compressor depending on the operational state of the vehicle when the compressor operates in the normal state with the suction port having the maximum opening area. In other words, the above-mentioned condition cannot provide proper control of driving load on the compressor depending on the state including, for example, acceleration of the vehicle or a trouble in the compressor.

Summary of the Invention:

Therefore, an object of the present invention is to provide a compressor of which driving load can be controlled depending on the necessity.

A compressor according to the present invention comprises a compressor housing, the compressor housing comprising a suction port, a discharge port, a suction chamber, and a discharge chamber, the suction chamber and the discharge chamber being defined in the compressor housing and connected to the suction port and the discharge port, respectively. A plurality of pistons are reciprocally moved in the compressor by means of rotating a main shaft to compress refrigerant introduced through the suction port. Compressed refrigerant is discharged through the discharge chamber.

According to an aspect of the present invention, the compressor further comprises first control section for controlling an opening area of the suction port depending on a difference in pressure between the discharge chamber and the suction port; and second control section for minimizing forcefully the opening area of the suction port.

Brief Description of the Drawing:

Fig. 1 is a cross sectional view of a conventional compressor with a suction port for refrigerant having a maximum opening area;

Fig. 2 is a cross sectional view of the compressor in Fig. 1 with the suction port for the refrigerant having a minimum opening area;

Fig. 3 is a cross sectional view of a compressor according to a first embodiment of the present invention with a suction port for the refrigerant having a maximum opening area;

Fig. 4 is a cross sectional view of the compressor in Fig. 3 with the suction port for the refrigerant having a minimum opening area;

Fig. 5 is a cross sectional view of a compressor according to a second embodiment of the present invention with a suction port for the refrigerant having the maximum opening area; and

Fig. 6 is a cross sectional view of the compressor in Fig. 5 with the suction port for the refrigerant having the minimum opening area.

Description of the Preferred Embodiment:

Referring to Fig. 1, a conventional variable capacity compressor is a swash plate variable capacity compressor mentioned before. The compressor comprises a cylinder block 12 having a plurality of cylinders 11 (only one of them is illustrated in the figure). The cylinders 11 are spaced with each other in a circumferential direction. Each cylinder 11 houses a piston 13 that is reciprocally mounted therein.

The cylinder block 12 has a hollow cylindrical section 12a. The hollow cylindrical section 12a is extended in an axial direction. A front end plate 14 is secured to an end of the hollow cylindrical section 12a. A space within the hollow cylindrical section 12a in the cylinder block 12 is used as a crank chamber 15. The crank chamber 15 houses a driving shaft (main shaft) 16. The driving shaft 16 is rotatably supported by the front end plate 14 and the cylinder block 12 via a bearing.

A rotor 17 is fixed to the driving shaft 16 at one end thereof. One surface of the rotor 17 is thrust supported by the inner wall surface of the front end plate 14 via a bearing. A swash plate 19 is attached to the rotor 17 via a hinge mechanism 18. The hinge mechanism 18 has a bracket 18a and a tab 18b. The bracket 18a extends from the swash plate 19 toward the rotor 17. The tab 18b is opposed to the bracket 18a and extends from the rotor 17 towards the swash plate 19. The bracket 18a has an elongate slot 18c formed therein. The tab 18b has a guide pin 18d to be engaged with the elongate slot 18c.

The swash plate 19 rotates integrally with the driving shaft 16 through the hinge mechanism 18. A sleeve 20 is slidably engaged with the drive shaft 16 so as to be slidable between the driving shaft 16 and the swash plate 19. With the above-mentioned structure, the swash plate 19 is supported such that it can be inclined at a variable angle relative to the driving shaft 16.

A wobble plate 21 is combined with the swash plate 19 through a bearing. A number of pistons 13 (only one of which is illustrated in the figure) are coupled to each other through rods 22 around the periphery of the wobble plate 21. A guide 23 is placed in the crank chamber 15. The guide 23 is supported by the front end plate 14 and the cylinder block 12 at the respective ends thereof. An end of the wobble plate 21 is engaged with the guide 23 such that it is slidable along the guide 23. The pistons 13 move reciprocally in the cylinder 11 when the driving shaft 16 rotates.

A cylinder head 25 is attached to the cylinder block 12 at the other end thereof through a valve body assembly 24. In the illustrative embodiment, the compressor housing is formed of the cylinder block 12, the front end plate 14, and the cylinder head 25. The cylinder head 25 has a number of suction chambers 26 (only one of which is illustrated in the figure) formed around the periphery thereof and a discharge chamber 27 formed at the center of the cylinder head 25. The cylinder head 25 has a suction port 29 and a discharge port 30, both of which are integrally provided with the cylinder head 25. The suction port 29 is for introducing refrigerant gas into a suction chamber 26. The discharge port 30 is for discharging the refrigerant gas out of the discharge chamber 27.

The valve body assembly 24 controls the flow of the refrigerant gas to ensure that the refrigerant gas flows from the suction chamber 26 to the discharge chamber 27 through the cylinders 11 when the pistons 13 move reciprocally.

A capacity control valve mechanism 31 is embedded in the cylinder block 12. The capacity control valve mechanism 31 controls opening/closing of a passage 32 communicating with the crank chamber 15 and the suction chamber 26. The capacity control valve mechanism 31 adjusts an angle of inclination of the swash plate 19 by using a difference in pressure P_E (=P_C - P_S) between the pressure P_C in the crank chamber 15 and the pressure P_S in the suction chamber 26.

A cylinder 33 is formed in the cylinder head 25. The cylinder 33 has one end opened to the discharge chamber 27 and the other end opened to the suction port 29. The cylinder 33 houses a piston 34 such that the piston 34 is movable reciprocally. A hollow cylindrical valve member 35 is fixedly secured to the piston 34 at one end thereof to change an opening area of the suction port 29 with the reciprocal movement of the piston 34. The valve member 35 houses a spring 36. The spring 36 is for biasing the valve member 35 to the direction closer to the central axis of the cylinder head 25. For this purpose, one end of the spring 36 engages with the valve member 35 and the other end of the spring 36 engages with a spring seat 37 placed on the periphery of the cylinder head 25. The position of the spring seat 37 can be determined in the direction of the reciprocal movement of the piston 34. A flow path control mechanism 39 for controlling the flow path in the suction port 35 is formed of the cylinder 33, the piston 34, the valve member 35, the spring 36, and the spring seat 37.

Referring also to Fig. 2, operation of the aforementioned compressor is described. The suction port 29 is at a low pressure side and the discharge chamber 27 is at a high pressure side. It is assumed that the compressor operation begins with the pressures at the low and the high pressure sides being balanced. The piston 34, which forms the flow path control mechanism 39, is depressed by the biasing force of the spring 36 because the suction port 29 has the same pressure as the discharge chamber 27. As a result, the refrigerant path at the low pressure side is opened minimally by the valve member 35.

Suction and discharge operation by the reciprocal movement of the piston 13 begins when the compressor is rotation driven by the driving shaft 16. At that time, only a limited volume of the refrigerant is drawn into the cylinder 11 because of the minimum opening area of the refrigerant flow path at the low pressure side. The pressure in the cylinder 11 thus drops suddenly. The pressure P_C in the crank chamber 15 becomes higher than the pressure in the suction chamber 26 accordingly. As a result, an inclination angle of the swash plate 19 becomes small relative to the driving shaft 16 due to the pressure difference P_E. The smaller the inclination angle of the swash plate 19 is, the closer the angle of the swash plate 19 is 90 degrees. An amplitude of the oscillation of the wobble plate 21 becomes small as the inclination angle of the swash plate 19 becomes small. Therefore, initial driving torque of the compressor is limited to a minimum level.

On the other hand, continued driving of the compressor increases the pressure in the discharge chamber 27 by the refrigerant gradually drawn into the cylinder 11 through the minimum opening area of the refrigerant flow path at the low pressure side. The increased pressure pushes the piston 34 of the flow path control mechanism 39 away from the central axis of the cylinder head 25. Generally, a discharge pressure from the compressor increases in proportion to the flow rate of the refrigerant, if the refrigerant flow path has a constant opening area. The piston 34 moves closer to the suction port 29 in the cylinder 33 when the pressure in the discharge chamber 27 exceeds the biasing force by the spring 36. The opening area of the refrigerant flow path from the suction port 29 to the suction chamber 26 increases as a result of the movement of the valve member 35 along with the above-mentioned movement of the piston 34. When the discharge pressure in the compressor reaches or exceeds a predetermined discharge pressure of, for example, P_0 = 13 kg/cm^{2}G, then the valve member 35 moves to the position where the suction port 29 is opened completely as shown in Fig. 2. As a result, the refrigerant flow path has the maximum opening area. Such a state is referred to as a normal operation state.

The opening area of the suction port 29 changes depending on the pressure difference between the low and the high pressure sides at the beginning of the operation of the compressor, so that a torque shock against the compressor can be avoided. This means a so-called soft start is achieved.

In the above-mentioned compressor, the opening area of the suction port 29 is minimized at the beginning of the compressor operation. This limits an initial driving torque of the compressor to be minimal. However, the suction port 29 has the maximum opening area when the compressor operates in the normal state. The capacity control is achieved by the control valve mechanism 31. It may be difficult or even impossible to properly control the load on the compressor depending on the operational state of the vehicle when the compressor operates in the normal state with the suction port 29 having the maximum opening area. In other words, the above-mentioned condition cannot provide a proper control of the driving load on the compressor depending on the state including, for example, acceleration of the vehicle or a trouble in the compressor. The vehicle with such compressor may not often comfortable to ride in or the compressor may be damaged or broken down.

Referring to Fig. 3, a compressor according to a first embodiment of the present invention is described. In Fig. 3, similar components and parts to those illustrated in Fig. 1 are depicted by the same reference numerals as in Fig. 1. The compressor comprises the cylinder block 12 having a plurality of cylinders 11 (only one of them is illustrated in the figure). The cylinders 11 are spaced with each other in a circumferential direction. Each cylinder 11 houses the piston 13 that is reciprocally mounted therein.

The cylinder block 12 has the hollow cylindrical section 12a. The hollow cylindrical section 12a is extended in the axial direction. The front end plate 14 is secured to an end of the hollow cylindrical section 12a. The space within the hollow cylindrical section 12a in the cylinder block 12 is used as the crank chamber 15. The crank chamber 15 houses the driving shaft 16. The driving shaft 16 is rotatably supported by the front end plate 14 and the cylinder block 12 via the bearing.

The rotor 17 is fixed to the driving shaft 16 at one end thereof. One surface of the rotor 17 is thrust supported by the inner wall surface of the front end plate 14 via the bearing. The swash plate 19 is attached to the rotor 17 via the hinge mechanism 18. The hinge mechanism 18 has the bracket 18a and the tab 18b. The bracket 18a extends from the swash plate 19 toward the rotor 17. The tab 18b is opposed to the bracket 18a and extends from the rotor 17 towards the swash plate 19. The bracket 18a has the elongate slot 18c formed therein. The tab 18b has the guide pin 18d to be engaged with the elongate slot 18c.

The swash plate 19 rotates integrally with the driving shaft 16 through the hinge mechanism 18. In this embodiment, an inner peripheral surface of the swash plate 19 contacts with the driving shaft 16. The swash plate 19 is slidable along the driving shaft 16. The swash plate 19 is thus supported such that it can be inclined at a variable inclination angle relative to the driving shaft 16.

The wobble plate 21 is assembled with the swash plate 19 through the bearing. The above-mentioned pistons 13 are coupled to each other through the rods 22 around the periphery of the wobble plate 21. The guide 23 is placed in the crank chamber 15. The guide 23 is supported by the front end plate 14 and the cylinder block 12 at the respective ends thereof. An end of the wobble plate 21 is engaged with the guide 23 such that it is slidable along the guide 23. The pistons 13 move reciprocally in the cylinders 11 when the driving shaft 16 rotates.

The cylinder head 25 is attached to the cylinder block 12 at the other end thereof through the valve body assembly 24. The cylinder head 25 has a number of suction chambers 26 formed around the periphery thereof and the discharge chamber 27 formed at the center of the cylinder head 25. The cylinder head 25 has the suction port 29 and the discharge port 30, both of which are integrally provided with the cylinder head 25. The suction port 29 is for introducing the refrigerant gas into the suction chamber 26. The discharge port 30 is for discharging the refrigerant gas out of the discharge chamber 27.

While not being illustrated, the capacity control valve mechanism is embedded in the cylinder block 12 as mentioned in conjunction with Fig. 1. The capacity control valve mechanism controls opening/closing of the passage communicating with the crank chamber 15 and the suction chamber 26. The capacity control valve mechanism adjusts the angle of inclination of the swash plate 19 by using the difference in pressure P_E (= P_C - P_S) between the pressure P_C in the crank chamber 15 and the pressure P_S in the suction chamber 26. This means that the stroke of the piston 13 is adjusted.

The compressor has a flow path control mechanism that is different from the flow path control mechanism 39 described in Fig. 1. The flow path control mechanism of this embodiment is depicted by the reference numeral 41. The flow path control mechanism 41 has a cylinder (high pressure chamber) 42. The cylinder 42 is formed with the cylinder head 25. The cylinder 42 is opened to the discharge chamber 27 at one end thereof. The other end of the cylinder 42 opens to the suction port 29. The cylinder 42 houses a piston 43 such that the piston 43 is movable reciprocally. A hollow cylindrical valve member 44 is fixedly secured to the piston 43 at one end thereof to change the opening area of the suction port 29 with the reciprocal movement of the piston 43. The piston 43 and the valve member 44 are generally called a spool valve.

The cylinder head 25 has a sealing member 45 opposing to the valve member 44. The sealing member 45 has a tip portion 45a that is to be inserted into a hollow portion in the valve member 44. The sealing member 45 has an injection passage 45b formed therein along the center of the sealing member 45. The injection passage 45b is opened at both ends thereof. As described later, a high pressure gas is introduced into the injection passage 45b. The valve member 44 houses a spring 46 for biasing the piston 43 towards the discharge chamber 27. One end of the spring 46 engages with the piston 43 and the other end of the spring 46 engages with the sealing member 45.

As apparent from the figures, the discharge port 30 is coupled to the injection passage 45 through a pressure introduction control valve 47 by a bypass duct 48. The pressure introduction control valve 47 is controlled by a control signal supplied by a controller 49. The pressure introduction control valve 47 may be, for example, a solenoid valve or a three-way switching valve.

Next, operation of the compressor is described with reference to Figs. 3 and 4. The suction port 26 is at the low pressure side and the discharge chamber 27 is at the high pressure side. It is assumed that the compressor operation begins with the pressures at the low and the high pressure sides being balanced. This corresponds to the initial state where the compressor operation is started. The piston 43 of the flow path control mechanism 41 is depressed towards the discharge chamber 27 by the biasing force of the spring 46 because the suction port 29 has the same pressure as the discharge chamber 27. As a result, the refrigerant path at the low pressure side is opened minimally by the valve member 44, though not illustrated in the figures.

Suction and discharge operation by the reciprocal movement of the piston 13 begins when the compressor is rotation driven by the driving shaft 16. At that time, only the limited volume of refrigerant is drawn into the cylinder 11 because of the minimum opening area of the refrigerant flow path at the low pressure side. The pressure in the cylinder 11 thus drops suddenly. The pressure P_C in the crank chamber 15 becomes higher than the pressure in the suction chamber 26 accordingly. As a result, the inclination angle of the swash plate 19 becomes small relative to the driving shaft 16 due to the pressure difference P_E. In other words, the angle of the swash plate 19 becomes closer to 90 degrees. The amplitude of the oscillation of the wobble plate 21 becomes small as the inclination angle of the swash plate 19 becomes small. Therefore, the initial driving torque of the compressor is limited to the minimum level.

On the other hand, continued driving of the compressor increases the pressure in the discharge chamber 27 by the refrigerant gradually drawn into the cylinder 11 through the minimum opening area of the refrigerant flow path at the low pressure side. The increased pressure pushes the piston 43 of the flow path control mechanism 41 away towards the suction port 29. Generally, the discharge pressure from the compressor increases in proportion to the flow rate of the refrigerant, if the refrigerant flow path has a constant opening area. The piston 43 moves closer to the suction port 29 in the cylinder 32 when the pressure in the discharge chamber 27 exceeds the biasing force by the spring 46. The opening area of the refrigerant flow path from the suction port 29 to the suction chamber 26 increases as a result of the movement of the valve member 44 along with the above-mentioned movement of the piston 43. When the pressure in the discharge chamber 27 reaches or exceeds a predetermined discharge pressure of, for example, P_0 = 13 kg/cm^{2}G, then the valve member 44 moves to the position where the suction port 29 is opened completely as shown in Fig. 3. As a result, the refrigerant flow path has the maximum opening area. Such a state is referred to as the normal operation state.

The controller 49 monitors driving condition of the vehicle. For example, the controller 49 controls, in response to an acceleration instruction signal, the pressure introduction control valve 47 to introduce high pressure gas (having a pressure P_d) from the discharge port 30 to the injection passage 45b through the bypass duct 48.

By the operation of the pressure introduction control valve 47, the high pressure gas is introduced into the injection passage 45b. As a result, the pressure in the cylinder 42 is balanced with that in the injection passage 45b. In this event, the piston 43 moves towards the discharge chamber 27 because of the biasing force of the spring 46. The cylinder 42 may be called as a high pressure chamber or a first chamber. The injection passage 45b is a hollow portion in the valve member 44 and may be called as a second chamber. The valve member 44 minimizes the opening area of the refrigerant flow path at the low pressure side. As described above, only the limited volume of refrigerant is drawn into the cylinder 11 because of the minimum opening area of the refrigerant flow path at the low pressure side. The pressure in the cylinder 11 thus drops suddenly. The pressure P_C in the crank chamber 15 becomes higher than the pressure in the suction chamber 26 accordingly. As a result, the inclination angle of the swash plate 19 becomes small relative to the driving shaft 16 due to the pressure difference P_E. The amplitude of the oscillation of the wobble plate 21 becomes small as the inclination angle of the swash plate 19 becomes small.

Accordingly, the operational load on the compressor can be reduced depending on the driving condition or the like of the vehicle. In other words, the operational load on the compressor can be reduced depending on the necessity.

While the present invention has thus been described in conjunction with the case where the pressure introduction control valve 47 is controlled depending on the driving condition of the vehicle, the pressure introduction control valve 47 may be controlled in response to, for example, a detection of a trouble in the compressor.

Referring to Figs. 5 and 6, a compressor according to a second embodiment of the present invention is described. This compressor is also a variable capacity compressor. Similar components and parts to those described in conjunction with Fig. 3 are depicted by the same reference numerals as in Fig. 3. Description of such components and parts is omitted.

The compressor comprises the swash plate 19 that is coupled to the rotor 17 through the hinge mechanism 18. A number of pistons 13 (only one of which is illustrated in the figure) are coupled to each other through rods 22 at the edge of the swash plate 19. The following description is made for only one piston.

In this embodiment, a pair of shoes 19a is provided on the edge of the periphery of the swash plate 19 such that the shoes 19 holds the swash plate 19 therebetween. The pair of shoes 19a forms a substantial sphere. A grasping section 22a is formed at one end of the rod 22. The grasping section 22a holds the shoes 19a as illustrated in the figure. The grasping section 22a is slidable on the spherical surface formed by the shoes 19a.

As illustrated, a spring 16b is placed between the swash plate 19 and a ring 16a attached to the main shaft 16 along the main shaft 16. The spring 16b biases the swash plate 19 towards the left side in the figure.

Operation of this variable capacity compressor is similar to that of the variable capacity compressor described in Figs. 3 and 4. Therefore, detailed description thereof will be omitted. Briefly, the opening area of the refrigerant flow path from the suction port 29 to the suction chamber 26 becomes maximum as shown in Fig. 5 after the beginning of the compressor operation and continued driving for a certain period of time. This is referred to as the normal operation state.

On the other hand, operation of the pressure introduction control valve 47 leads the opening area of the refrigerant flow path at the low pressure side to be minimized as shown in Fig. 6. This limits the volume of the refrigerant to be drawn into the cylinder 11. The pressure in the cylinder 11 thus drops suddenly. As a result, the inclination angle of the swash plate 19 becomes small relative to the driving shaft 16 and the stroke of the piston 13 also becomes small.

The present invention has thus been described in conjunction with the case where it is applied to the variable capacity compressor. However, the present invention may equally be applied to a fixed capacity compressor. The amount of the gas drawn into the compressor is reduced when the above-mentioned flow path control mechanism 41 is used in the fixed capacity compressor. As a result, the compressor load is also reduced and the operational load thereon becomes small.

In the above-mentioned embodiments, the pressure introduction control valve 47 is placed outside the compressor housing. However, the pressure introduction control valve 47 and the bypass duct 48 may be placed within the cylinder head 25.

As described above, according to the present invention, the operational load on the compressor can be reduced depending on, for example, the driving condition of the vehicle. In other words, the operational load on the compressor can be reduced depending on the necessity.

Claims

1. A compressor comprising a compressor housing comprising a suction port (29), a discharge port (30), a suction chamber (26), and a discharge chamber (27), said suction chamber and said discharge chamber being defined in said compressor housing and connected to said suction port and said discharge port, respectively, and a plurality of pistons reciprocally moved in said compressor by means of rotating a main shaft (16) to compress refrigerant introduced through said suction port, compressed refrigerant being discharged through said discharge chamber, which is characterized in that said compressor further comprises:

   first control means (41) for controlling an opening area of said suction port depending on a difference in pressure between said discharge chamber and said suction port; and
   second control means (49) for minimizing forcefully the opening area of said suction port.

2. A compressor as claimed in claim 1, wherein said first control means comprises:

   a cylinder formed in a direction across said suction port, said cylinder being communicated with said discharge chamber;
   a valve member (44) slidably placed in said cylinder to change the opening area of said suction port; and
   biasing means (46) for biasing said valve member to such a direction that minimizes the opening area of said suction port,
   said cylinder being divided into a fist chamber (42) and a second chamber (45b) by said valve member, said first chamber being communi-

cated with said discharge chamber being separated from said first chamber, and wherein said second control means moves said valve member to such a direction that minimizes the opening area of said suction port by means of selectively drawing the compressed refrigerant into said second chamber.

3. A compressor as claimed in claim 1 or 2, wherein said second control means comprises:

a bypass duct (48) for introducing the compressed refrigerant into said second chamber; and
a valve mechanism (47) to control opening and closing of said bypass duct to lead the compressed refrigerant into said second chamber through said bypass duct.

4. A compressor as claimed in one of claims 1 to 3, wherein said compressor is mounted on a vehicle and said second control means minimizes forcefully the opening area of said suction port depending on a driving condition of said vehicle.

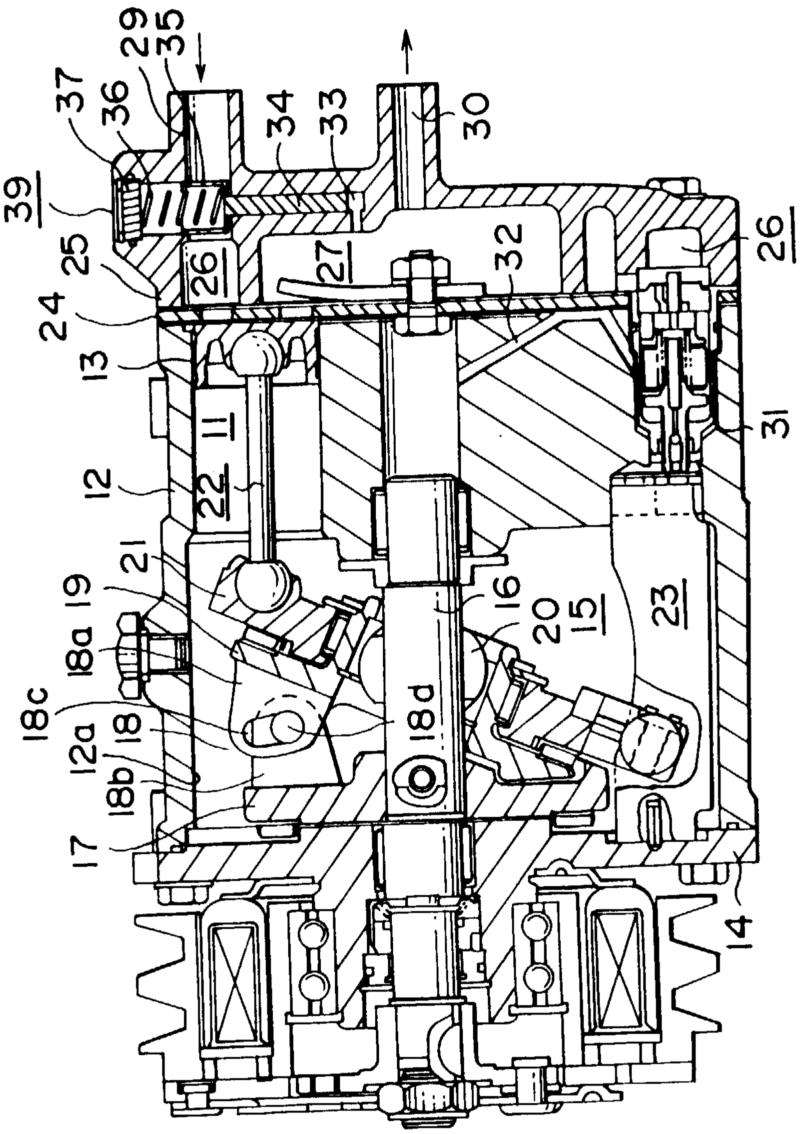

FIG. 1 PRIOR ART

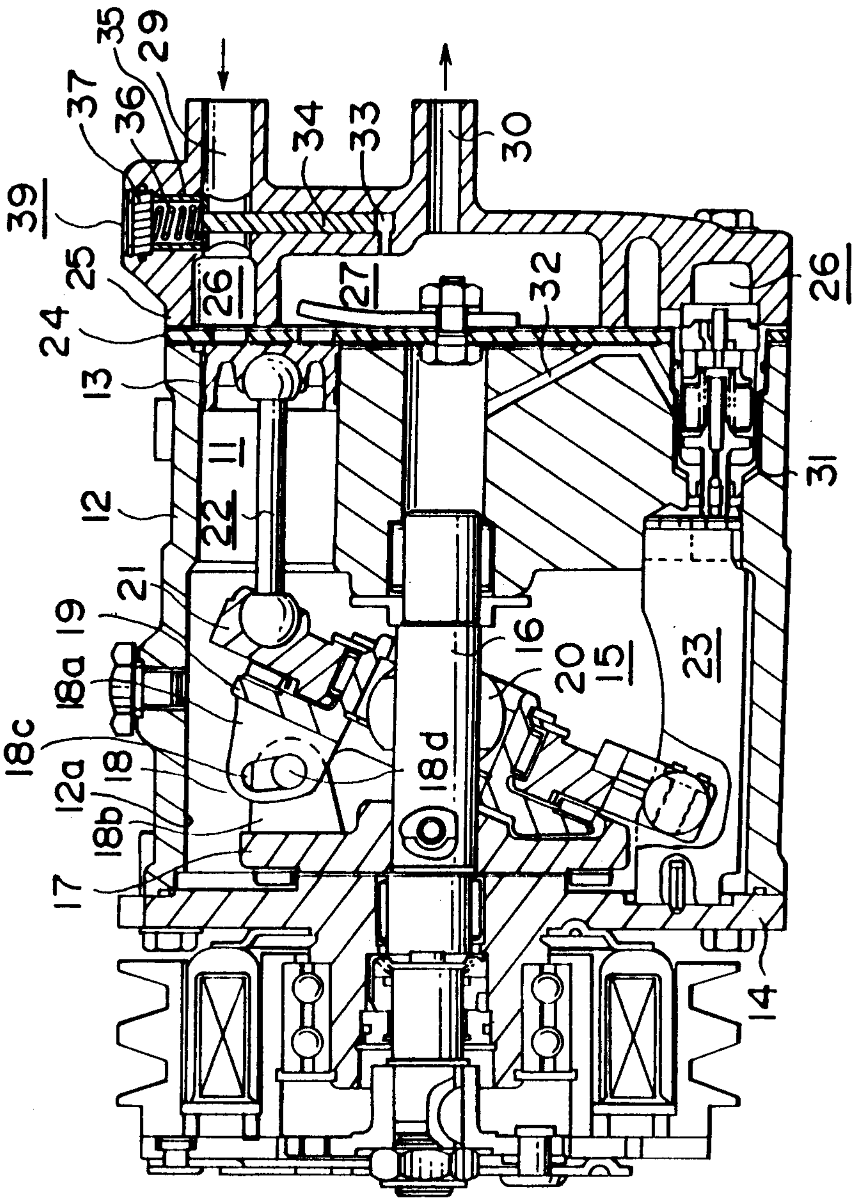

FIG. 2 PRIOR ART

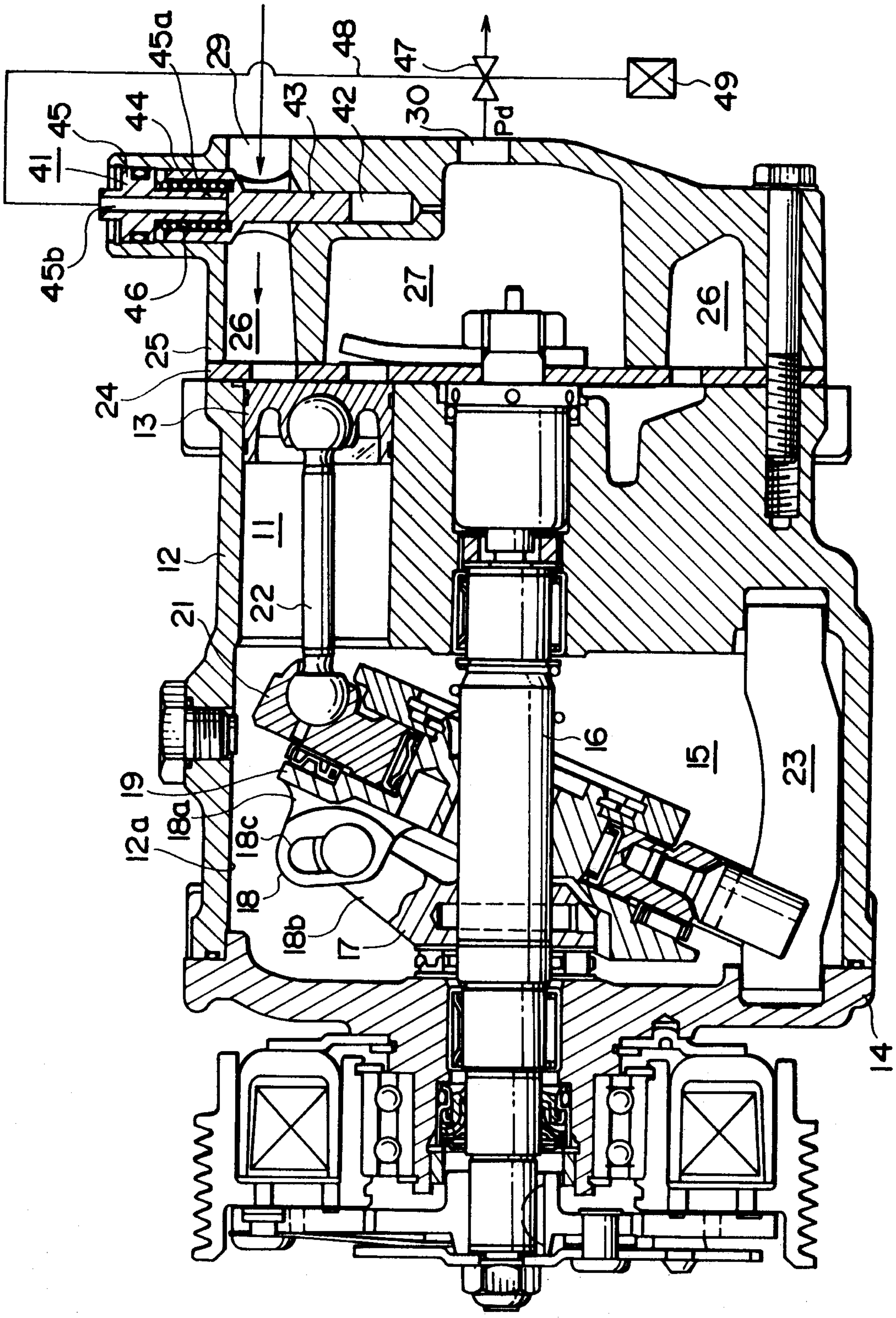
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FIG. 3

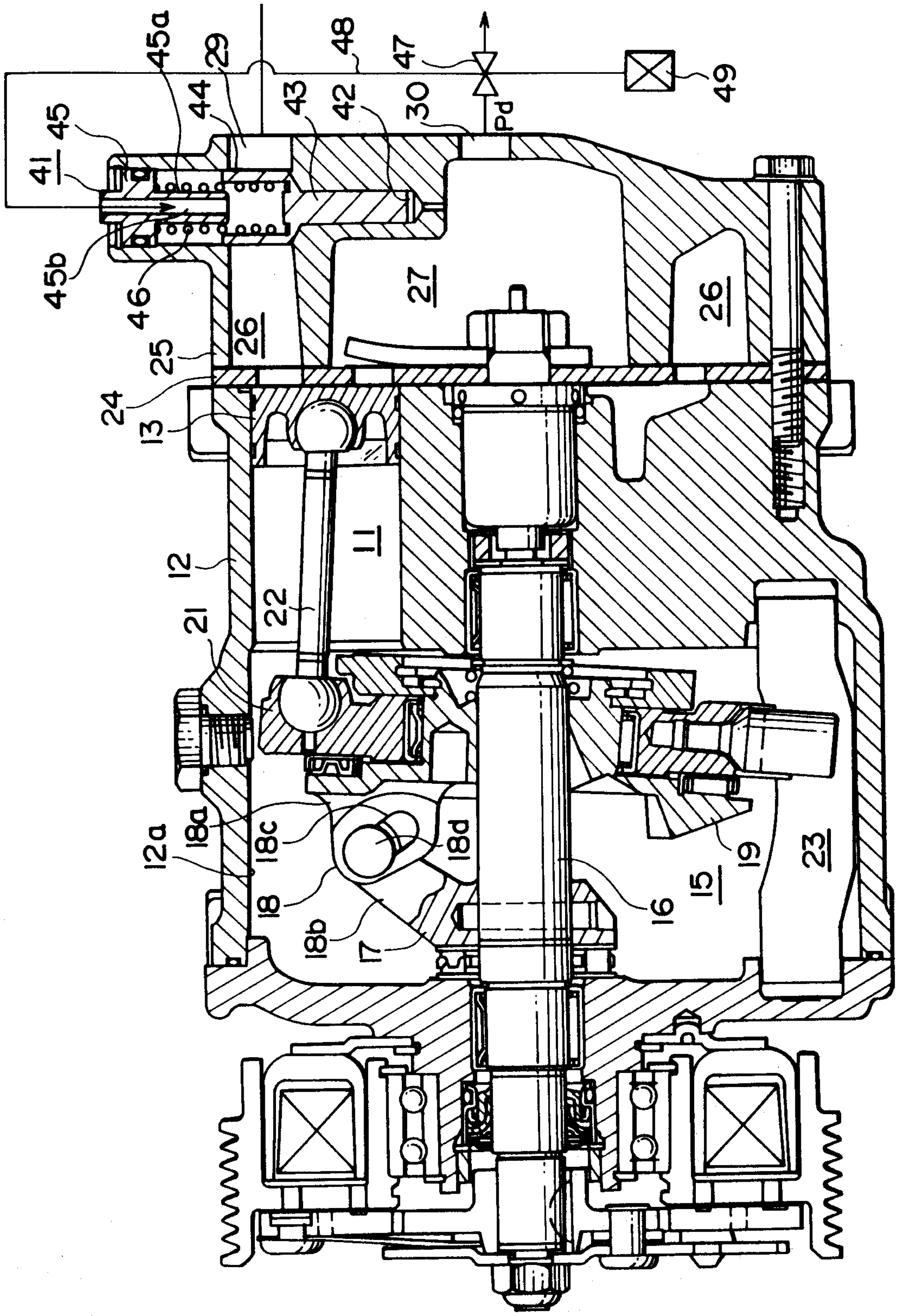
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FIG. 4

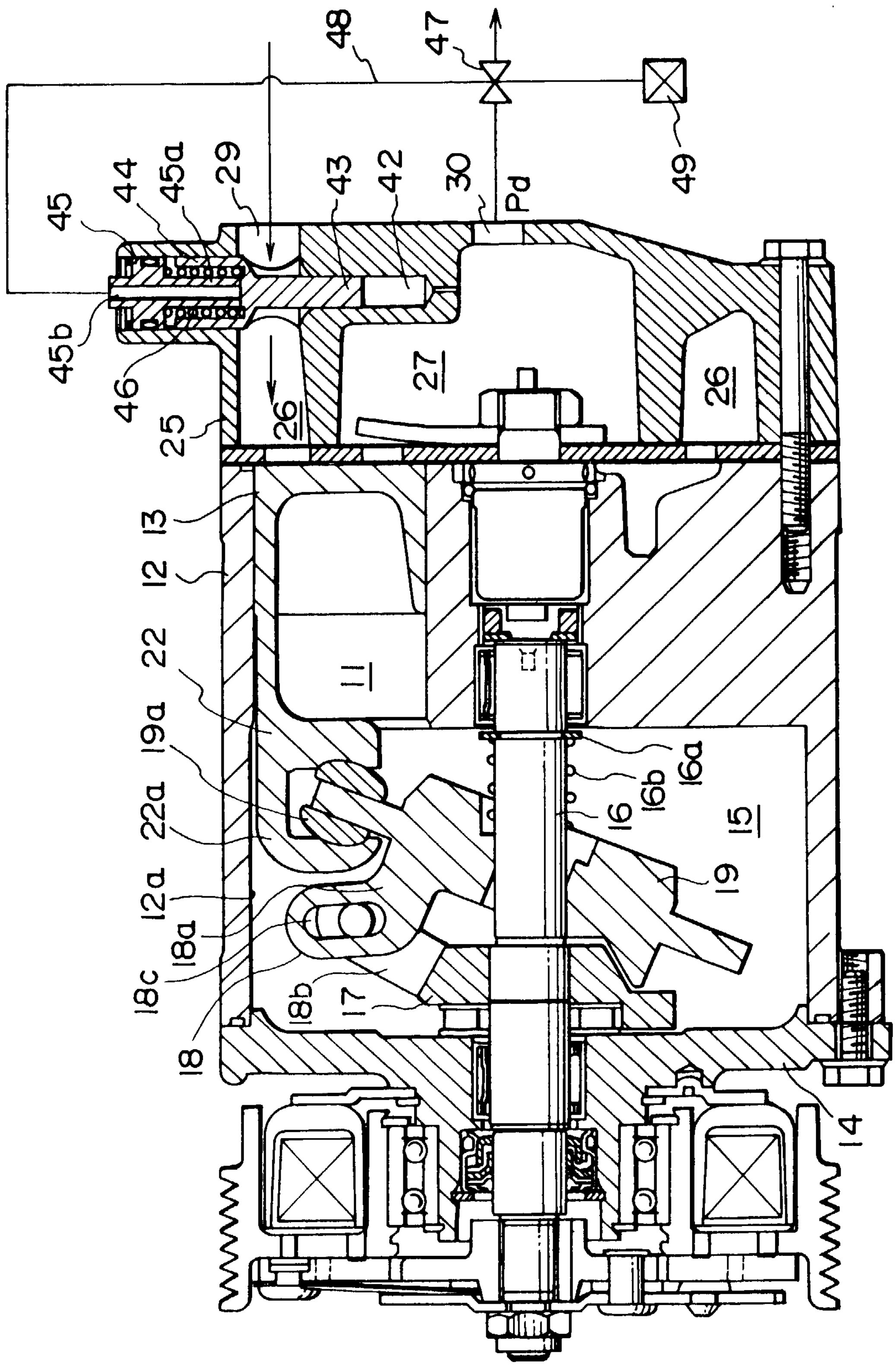
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FIG. 5

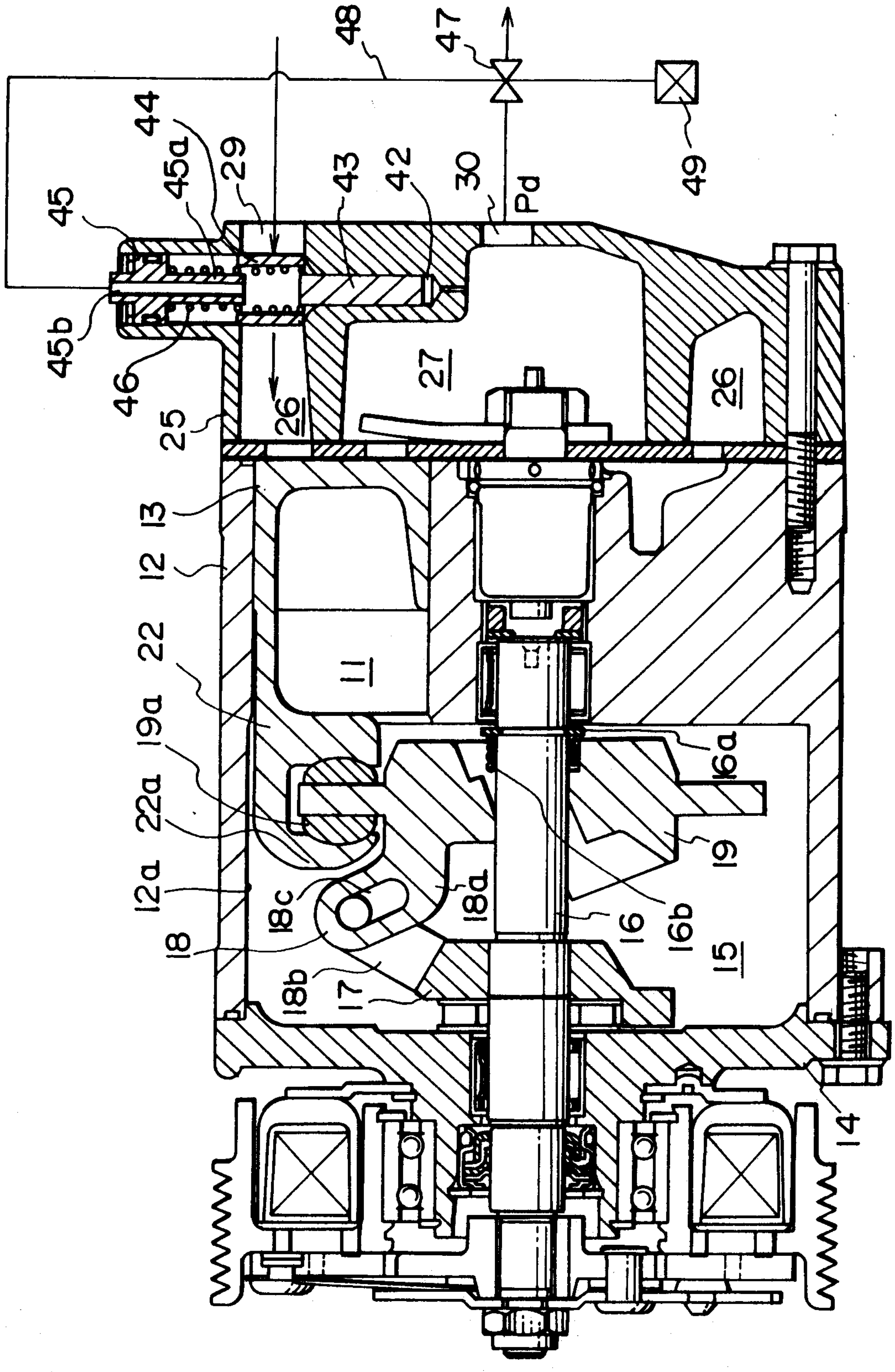
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FIG. 6

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 98 10 2109

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.6)
Y,D A	US 4 905 477 A (KAZUHIKO TAKAI) 6 March 1990 * column 3, line 65 - column 4, line 59; figures 2-4 *	1 2	F04B49/22
Y	US 5 173 032 A (TAGUCHI TATSUHISA ET AL) 22 December 1992 * column 4, line 64 - column 5, line 47; figures 1,2,7 *	1	
A	EP 0 711 918 A (TOYODA AUTOMATIC LOOM WORKS) 15 May 1996 * column 18, line 52 - column 20, line 16; figures 6,7 *	1,3	
A	EP 0 489 164 A (TOYODA AUTOMATIC LOOM WORKS) 10 June 1992 * column 11, line 7 - line 47; figures 1,2 *	1,4	

TECHNICAL FIELDS SEARCHED (Int.Cl.6)
F04B

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	8 May 1998	Bertrand, G

CATEGORY OF CITED DOCUMENTS

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