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(54) **Variable displacement swash-plate type compressor**

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Compresseur à plateau en biais à compression variable

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- **PATENT ABSTRACTS OF JAPAN vol. 7, no. 291 (M-265)(1436) 27 December 1983, & JP-A-58 162 780 (TOYODA JIDO SHOKKI SEISAKUSHO) 27 September 1983**

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Description

The present invention relates to a variable displacement swash-plate type compressor according to the preamble of claim 1 and 7. Such a compression is suitable for use as a refrigerant compressor in automotive air conditioners.

Such a variable displacement swash-plate type compressor has been known from JP-A-58 162 780, wherein the angle of tilt of a swash plate is linearly changed so as to effect a linear control of displacement between 0% and 100%. This known swash-plate type compressor, however, suffers from the following problem: Namely, a decrease in the angle of tilt of the swash plate in this compressor causes not only a reduction in the piston stroke but also an increase in the dead volume on each rear side of each piston of the compressor. The increase in the dead volume in turn causes a problem that, due to expansion of the gas in the dead volume, the displacement of the compressor is significantly changed even with a slight change in the tilting angle of the swash plate.

In order to obviate this problem, Japanese Unexamined Patent Publication No. 60-175783 discloses a swash-plate type compressor which does not employ double-headed pistons but utilizes a swash plate arranged such that the angle and the position of the swash plate are changed by the control of a pressure acting on the rear side of each piston so that the dead volumes are not changed when the tilting angle is decreased. Thus, in this swash-plate type compressor, pistons are provided only on one side of the swash plate, so that the pressure of the gas discharged from the compressor greatly pulsates and the torque required for driving the compressor fluctuates undesirably. In addition, the capacity or displacement per size of the compressor is limited.

In order to obviate this problem, it is preferred that the compressor employs a double-headed piston type mechanism, i.e., pistons arranged on both sides of a swash plate. In this double-headed piston type, however, it is impossible to make use of back pressure acting on the rear sides of the pistons for the purpose of controlling the angle and position of the swash plate because working chambers are provided on both sides of the swash plate.

An object of the present invention is to provide a swash-plate type compressor wherein the displacement is linearly controlled with good response to a displacement control input and wherein the linear control of the displacement is possible even in an operating region in which the piston stroke has been slightly reduced from the maximum stroke, that is, in a condition in which the displacement is slightly reduced from the maximum displacement.

A further object of the present inventin is to provide a swash-plate type compressor wherein the displacement is linearly controlled with good response to a displacement control input and wherein the linear control of the displacement is conducted without fail even in an operating region in which the piston stroke has been largely reduced from the maximum stroke; that is, the object is to assure that the linear control of the displacement can be conducted down to the minimum displacement.

This object of the invention is achieved by the features in the characterizing part of claim 1 and 7.

According to an aspect of the present invention, there is provided a swash-plate type compressor having auxiliary spool urging means which assist a spool in making an axial movement when the amount of decrease in the piston stroke is comparatively small, i.e., when the displacement of the compressor has been reduced slightly from the maximum displacement. In an operating region where the displacement of the compressor has been slightly reduced from the maximum displacement, smooth movement of the spool tends to be impaired due to influence by a dead space formed in working chambers on one side of the pistons. Such a problem, however, can be overcome by the compressor of the aspect of the invention which incorporates the auxiliary spool urging means which assists the spool when the spool makes an axial movement so as to ensure a smooth movement of the spool when the compressor operates in an operating region where the displacement has been slightly reduced from the maximum displacement.

According to a further aspect of the present invention, there is provided a swash-plate type compressor having auxiliary tilting means adapted to urge the swash plate to the minimum tilt angle position where the piston stroke and, hence, the displacement of the compressor are minimized. In general, when the displacement of a piston has been decreased almost to the minimum level, a further tilting of the swash plate to the minimum tilting angle tends to be resisted by a force which is produced by pressure differential across the piston. In the swash-plate type compressor according to the further aspect of the present invention, however, this problem is overcome because the auxiliary tilting means operates to ensure that the swash plate can be tilted to the minimum tilting angle to enable the compressor to linearly and smoothly change its displacement to the minimum value.

Fig. 1 is an axial sectional view of an embodiment of the swash-plate type compressor in accordance with the present invention;

Fig. 2 is a sectional view of the compressor shown in Fig. 1 with the compressor shown in a different state of operation;

Fig. 3 is an axial sectional view of another embodiment of the swash-plate type compressor in accordance with the present invention;

Fig. 4 is an illustration of the thrust load applied to a spool in the compressor shwon in Fig. 3;

Fig. 5 is a sectional view of an essential portion of the compressor in accordance with the present invention;

Fig. 6 is a graph showing a relationship between the opening area of a relief port shown in Fig. 5 and the mean pressure in a pressure chamber;
Fig. 7 is a graph showing a relationship between the pressure difference across the spool and the displacement of the compressor;
Fig. 8 is a front elevational view of a further embodiment of the swash-plate type compressor of the present invention, illustrating an essential portion of this embodiment;
Fig. 9 is a sectional view of the embodiment shown in Fig. 8,
Fig. 10 is a sectional view of a still further embodiment of the swash-plate type compressor of the present invention, showing particularly an essential part thereof;
Fig. 11 is a sectional view of a still further embodiment of the swash-plate type compressor of the present invention, showing particularly an essential part thereof;
Fig. 12 is a graph showing a relationship between the compressor displacement ratio and thrust load acting on the spool;
Fig. 13 is an exploded perspective view of a part of the compressor shown in Fig. 2.
Fig. 14 is a graph showing a relationship between the load produced by back pressure acting on the spool and the amount of movement of the spool;
Fig. 15 is a graph showing a relationship between the amount of movement of the spool and the thrust load acting on the spool;
Fig. 16 is a graph illustrating a relationship between the piston stroke and the pressure in working chambers;
Fig. 17 is a graph showing a relationship between the reciprocatory motion of a piston and pressures in working chambers;
Fig. 18 is a graph showing a relationship between the stroke ratio of a spool and the thrust load acting on the spool;
Fig. 19 is a graph showing a relationship between piston stroke ratio and displacement ratio of the compressor;
Fig. 20 is an axial sectional view of a still further embodiment of the swash-plate type compressor of the present invention;
Fig. 21 is an axial sectional view of the compressor shown in Fig. 20 with the compressor shown in a different position of operation, and
Fig. 22 is a graphical illustration of the torque varied in the compressor shown in Fig. 20;

The compressor in Fig. 2 has an outer shell which is composed of the following parts assembled together by through bolts not shown: a front housing 4 made of an aluminum alloy; a front side plate 8; a suction valve 9; a front cylinder block 5; a rear cylinder block 6; a suction valve 12; a rear side plate 11 and a rear housing 13. Each of the cylinder blocks 5 and 6 is provided with five cylinder bores 64 formed therein in parallel with one another.

The compressor further has a shaft 1 which is rotatably supported on the front housing 4 and the front cylinder block 5 through bearings 2 and 3 and which is adapted to be driven by the power of an automotive engine which is not shown. During operaiton of the compressor, a thrust force is generated to act on the shaft 1 so as to urge the shaft to the left as viewed in Fig. 1. This thrust force is born by the front cylinder block 5 through a thrust bearing 15.

A rear shaft 40 is rotatably mounted in a spool 30 through a bearing 14. A thrust force which acts on the rear shaft 40 rightwards as viewed in Fig. 1 is born by the spool 30 through a thrust bearing 116. The spool 30 is axially slidably received in a cylindrical portion 65 of the rear cylinder block 6 and a cylindrical portion 135 of the rear housing 13.

A swash plate 10 is provided on the center thereof with a spherical surface portion 107 which receives a spherical portion 405 of the rear shaft 40 so that the swash plate 10 is rockably supported by the swash plate 10.

The shaft 1, the swash plate 10 and the rear shaft 40 are shown in an exploded perspective view in Fig.13. As will be seen from Fig. 13, the swash plate 10 is provided on its side adjacent to the shaft 1 with opposing walls defining a slit 105 which is adapted to receive a flat web portion 165 formed on the end of the shaft 1 adjacent to the swash plate 10. The flat web portion 165 makes a face-to-face contact with the opposing wall surfaces of the slit 105 so that a torque applied to the shaft 1 is transmitted to the swash plate 10.

Shoes 18 and 19 are slidably disposed on both sides of the swash plate 10. The cylinder bores 64 in the front cylinder block 5 and the rear cylinder block 5 slidably receive pistons 7. The shoes 18 and 19, which slidably engage with the swash plate 10 as stated above, rotatably engage with inner surfaces of the pistons 7. In consequence, an oscillatory rotational movement of the swash plate 10 is converted into reciprocatory motions of the pistons 7 through the shoes 18 and 19. The shoes 18 and 19 are so designed and sized that their outer surfaces constitute parts of a common sphere when they are assembled in the compressor.

The aforementioned flat web portion 165 of the shaft 1 is provided with an elongated slot 166, while the swash plate 10 is provided with pin-receiving holes formed in the opposing walls which define the slit 105. After the flat web portion 165 is placed in the slit 105, a pin 80 is inserted into the holes 106 and 108 through the elongated slot 166 so as to pivotally and movably connect the swash plate 10 to the shaft 1. A stopper ring 81 is provided on one end of the pin 80 to prevent the pin 80 from coming off these holes. The angle of tilt of the swash plate varies depending on a variable position of the pin 80 along the length of the elongated slot 166. A change

of the position of the pin 80 also causes a change in the position of the center (portion supported by the spherical portion 405 of the rear shaft 40) of the swash plate. Namely, the elongated slot 166 is so designed that, even when the stroke of the piston 7 is changed due to a change of the angle of tilt of the swash plate 10, the top dead center of the piston 7 in a working chamber 60 on the right side of each piston as viewed in Fig. 1 is not changed substantially, thus eliminating substantial increase in the dead volume in this working chamber 60. In contrast, in a working chamber 50 which is disposed on the left side of each piston 7 as viewed in Fig. 1, the top dead center of the piston is changed as a result of a change of the angle of tilt of the swash plate, thus causing a change in the dead volume.

The elongated slot 166 is so shaped, sized and positioned such that the position of the top dead center of the piston 7 in the working chamber 60 is not changed substantially even when the angle of tilt of the swash plate is changed. To meet this requirement, the elongated slot 166 must have an arcuate form in a strict sense. Particularly, however, such an arcuate form can be well approximated by a substantially linear elongated slot. In the described embodiment, the elongated slot 166 is disposed on the axis of the shaft 1 so as to prevent the shape and size of the flat web portion 165 from becoming excessively large-sized due to provision of the elongated slot 166. The reduction in the size of the flat web portion 165, which is realized by positioning the elongated slot 166 on the axis of the shaft 1, is advantageous particularly in the swash-plate compressor of the type in which the flat web portion 165 is disposed inwardly of pistons.

The compressor also has a shaft seal device 21 which prevents internal fluids such as a refrigerant gas or a lubricating oil from leaking along the surface of the shaft 1. The compressor further has discharge ports 24 which open to the working chambers 50 and 60, respectively, and communicate with discharge chambers 90 and 93, respectively. The discharge ports 24 are adapted to be opened and closed by discharge valves 22 which are fixed together with valve retainers 23 to the front side plate 8 and the rear side plate 11, respectively, by means of bolts which are not shown. The compressor further has suction valves 9 and 12 and a suction chamber 70 and 74 (Fig. 20).

An experiment conducted by the inventors, has revealed that it is often difficult to hold the spool 30 at a desired position.

Referring to Fig. 14, when the back pressure acting on the spool 30 is increased, the spool is moved in accordance with the rise in the back pressure as shown by a solid-line curve X-Y insofar as the back pressure is not higher than a predetermined level F_2. In Fig. 14, the axis of ordinate represents the amount of movement of the spool 30 which corresponds to the amount of change in the tilting angle of the swash plate 10 and also to the stroke of reciprocatory motion of the piston 7.

It has been confirmed that, when the back pressure acting on the spool 30 is increased beyond the level F_2, the travel of the spool 30 is not increased linearly but is increased to the maximum travel immediately when the pressure level F_2 is exceeded, as shown by a solid-line curve Y-Z. Thus, when the back pressure is above the predetermined level F_2, the spool 30 is fully moved to its travel and and is held at this position regardless of the level of the back pressure.

Conversely, when the back pressure acting on the spool 30 is decreased, the spool 30 is held at the travel and until the back pressure is decreased from F_3 to F_1 past the above-mentioned level F_2, as shown by a broken-line curve Z-K. When the back pressure comes down below the level F_1, the spool 30 is instantaneously moved by a predetermined distance as shown by broken-line curve K-L.

Thus, it has been impossible to delicately control and hold the position of the spool 30 particularly in the region near the travel end of the spool even though the back pressure of the spool 30 is controlled linearly and continuously.

The present inventors have made an intense study for clarifying the cause of this hysteresis in the behavior of the spool in relation to the back pressure and have reached a conclusion that this phenomenon is attributable to the fact that a relationship as shown in Fig. 15 exists between the stroke position of the spool 30 and an axial force exerted by the shaft 1 on the spool 30. In Fig. 15, a symbol O represents a state in which the travel of the spool 30 is minimum so that the tilting angle of the swash plate 10 and, hence, the stroke of the pistons 7 are minimum. As the travel of the spool 30 is increased from the first state shown by O, the stroke of the pistons 7 is increased correspondingly, so that the thrust force which is used for moving the spool 30 is also increased as shown by a solid-line curve O-P in Fig. 15. However, when the travel of the spool 30 is further increased, the force required for moving the spool 30 is not increased but decreased as shown by a solid-line curve P-Q in Fig. 15. The region of operation corresponding to the solid-line curve P-Q is the region where the stroke of reciprocatory motions of the pistons 7 is to be controlled up to the maximum stroke, i.e., the region in which the displacement of the compressor is slightly less than the maximum displacement of the compressor.

Thus, the curve representing the thrust force required for causing the movement of the spool 30 in relation to the travel of the spool between the minimum and maximum travel positions includes a peak or a maximum value F_2 of laod at the point P shown in Fig. 9. This maximum load F_2 corresponds to the travel position P_2 of the spool 30 which in turn corresponds to the point Y in Fig. 14. Thus, the spool 30 is fully moved at once to the maximum stroke position (point Q in Fig. 15 and point Z in Fig. 14) when the thrust force is increased beyond the predetermined value F_2. Once this state is reached, the spool 30 is held at this maximum travel position until the back pressure is decreased below the level of the thrust

force F_1 which is necessary for holding the spool 30 at this maximum stroke position.

When the back pressure acting on the spool 30 comes down below the level of the force F_1, the spool 30 is immediately moved from the position represented by the point Q in Fig. 15 to the position indicated by the point R in the same figure. The point R corresponds to the travel position P_1 of the spool 30, which in turn corresponds to the point L in Fig. 14.

The characteristic shown in Fig. 15 is attributable to the fact that, in the described swash-plate type compressor of the present invention, dead volume is generated only in the first working chamber 50 when the travel or amount of movement of the spool 30 is small. The operation will be described in more detail hereinunder with specific reference to Fig. 16.

Fig. 16 shows a relationship between the stroke of the pistons 7 and the internal pressure of the working chamber 50, i.e., the relationship between the internal volume and the internal pressure of the working chamber 50. In Fig. 16, a solid-line curve A shows a state in which the piston has the maximum stroke, i.e., a state in which the compressor is set for operating with maximum displacement, while a curve B in one-dot-and-dash line shows a state in which the tilting angle of the swash plate has been slightly decreased from the maximum angle to reduce the stroke of the pistons 7. In the state shown by the curve B, therefore, a predetermined dead volume is formed between each piston 7 and the side plate 8. A broken-line curve C in Fig. 16 represents a state in which the tilting angle of the swash plate 10 has been further decreased to further increase the dead volume. A curve D shown by two-dot-and-dash line shows a state in which the tilting angle of the swash plate 10 has been minimized to minimize the stroke of each piston 7 and, hence, maximize the dead volume.

Referring first to the curve A showing the state in which the pistons 7 are each allowed to fully move to the maximum stroke end position, the internal volume of the working chamber 50 is decreased as the piston 7 moves from the fully retracted position a so that the internal pressure of the working chamber 50 is increased as shown by a curve a-b-c. When the pressure reaches a predetermined discharge pressure P_d, the discharge valve 24 is opened to relieve the compressed gas so that the internal pressure of the working chamber 50 does not increase any more. Thus, the pressure in the working chamber is maintained at the level of the predetermined discharge pressure P_d, as shown by a straight line c-d-e. The piston 7 then reaches the stroke end represented by e and then commences its backward stroke. In consequence, the suction port 25 is opened to cause the internal pressure of the working chamber 50 to be reduced immediately down to the level of the suction pressure P_s which is indicated by f in Fig. 16. The piston then returns to the fully retracted position shown by a. Thus, the pressure in the working chamber 50 changes following the curves a-c-e-f-a when the stroke of each piston is maximized.

When the tilting angle of the swash plate 10 has been slightly decreased, a slight dead volume is formed in the working chamber 50 so that a certain volume of compressed gas remains in the working chamber before the piston 7 commences its backward stroke. In consequence, when the piston moves backward, the compressed refrigerant gas ramaining in the working chamber 50 is allowed to expand, as shown by one-dot-and-dash line d-g, so that the pressure in the working chamber 50 is maintained at a level above the suction pressure P_s.

When the tilting angle of the swash plate 10 is further decreased, the stroke of the piston 7 is also decreased to allow a large dead volume to be formed in the working chamber 50. In this case, the pressure of the gas compressed in this working chamber 50 cannot reach the predetermined discharge pressure P_d so that the discharge valve 24 is never opened. This state is shown by the broken-line curve C in Fig. 16. The pressure in the working chamber is increased along the curve a-b-c and then decreased along the curve c-b-a.

When the tilting angle of the swash plate 10 is further decreased to further reduce the stroke of the piston 7, each piston 7 moves along the two-dot-and-dash line curve D in Fig. 16. In this case, the refrigerant gas is not sucked into nor discharged from the working chamber 50, so that the pressure in the working chamber 50 merely increased along the curve b-a and decreased along the curve b-a.

It will be seen, accordingly, that the pressure in the working chamber 50 is varied by the formation of a dead space in the working chamber 50.

Fig. 17 is a graph showing a relationship between the pressure in the working chamber 50 and the cycle of reciprocation of the piston 7. A solid-line curve A corresponds to the state shown by the solid-line curve A in Fig. 16. In this state, no dead space is formed on the end of each piston 7 so that the pressure in the working chamber 50 is lowered to the level of the suction pressure P_s without delay after the commencement of backward stroke of the piston 7. A one-dot-and-dash line curve B in Fig. 17 shows a state corresponding to the state represented by the one-dot-and-dash line curve B in Fig. 16. In this state, a certain dead volume is formed in the working chamber 50 so that there is a residual pressure in the working chamber 50 due to the presence of the dead volume. Namely, the pressure in the working chamber 50 is not lowered immediately to the level of the suction pressure after the piston 7 has commenced its backward stroke. More specifically, the pressure in the working chamber 50 is progressively decreased from the level of the discharge pressure P_d to the level of the suction pressure P_s. A broken-line curve C in Fig. 17 corresponds to the state shown by the broken-line curve C in Fig. 16. In this state, the dead volume is so large that the pressure in the working chamber 50 varies along a sine wave curve and the pressure in the working chamber 50

is not lowered below the level of the suction pressure P_s.

A curve D in two-dot-and-dash line in Fig. 17 corresponds to the state represented by the two-dot-and-dash line D in Fig. 16. In this state, the pressure in the working chamber 50 changes along a sine wave curve, but neither suction nor discharge of the refrigerant gas is conducted as in the case of the state shown by the curve C. In the state shown by the curve D, moreover, the pressure variation in the working chamber is decreased and the maximum pressure in the working chamber 50 is decreased.

The region between the points P and Q in Fig. 15 corresponds to the region between the solid-line curve A and the broken-line curve C in Fig. 16. AS will be understood from Fig. 17, in this region, the pressure in the working chamber 50 produces a force which acts to urge the piston 7 to the right as viewed in Fig. 1.

This rightward urging force acting on the piston 7, produced by the pressure in the first working chamber 50, serves to increase the tilting angle of the swash plate 50. Namely, the tilting angle of the swash plate 10 is increased due to the residual pressure in the working chamber 50 to increase the stroke of reciprocatory motion of the piston 7. The behavior of the piston 7 and the swash plate explained above is conducted in the region between the points P and Q in Fig. 15. In this region, the pressure remaining in the working chamber 50 is increased as the dead volume increases. In consequence, the thrust force required for urging the spool 30 to the left as viewed in Fig. 1 is increased as the dead volume is increased.

The foregoing description taken in conjunction with Figs. 15 to 17 is based on an assumption that the suction pressure P_s and the discharge pressure P_d are constant. When the compressor is used for the purpose of compressing a refrigerant gas in a refrigeration cycle, however, both the suction pressure P_s and the discharge pressure P_d vary in accordance with a varying condition of operation of the refrigeration cycle. For instance, when the refrigeration cycle operates under a comparatively light load, the suction pressure P_s and the discharge pressure P_d are, for example, 2.5 kg/cm^2 abs and 16 kg/cm^2 abs, repspectively. However, when the thermal load applied to the refrigeration cycle is increased, the suction pressure P_s and the discharge pressure P_d are increased to, for example, 4 kg/cm^2 abs and 26 kg/cm^2 abs, respectively. The change in the suction and discharge pressures P_s and P_d also causes a change in the compression ratio ε.

Fig. 18 shows the change in the thrust load required fro axially moving the spool 30 of the compressor in relation to a change in the discharge pressure P_d. As will be seen from Fig. 18, the thrust laod is increased as the discharge pressure becomes higher. It will also be seen that the thrust load varies largely particularly in the region immediately after the dead volume starts to be generated on the end of the piston 7. This is because the residual pressure generated due to the presence of the dead space produces a force which acts to force the spool 30 backward through the piston 7 and the swash plate 10. Namely, in the state where the discharge pressure is high, the residual pressure generated in the working chamber 50 due to the presence of the dead volume becomes higher, requiring a greater thrust load for axially moving the spool 30. As will be seen from Fig. 18, when the dead volume is increased beyond a predetermined value, the discharge pressure no longer affects the internal pressure of the working chamber 50. This means that, when the spool 30 has been moved to a predetermined position, the thrust load required for any further axial movement of the spool 30 is maintained constant regardless of any change in the discharge pressure. In consequence, the relationship between the travel of the spool and the displacement of the compressor varies in accordance with the change in the suction pressure P_s and the discharge pressure P_d as shown in Fig. 19. In Fig. 19, the solid-line curve shows the state in which the compressor operates steadily with a compression ratio of 5.0. The broken-line curve and the one-dot-and-dash line curve show, respectively, the states of operation of the compressor under a light load (compression ratio = 4.0) and under a heavy load (compression ratio = 6.0).

As will be understood from the foregoing description, the thrust load required for causing axial movement of the spool 30 from the position corresponding to the point P_2 (see Fig. 15) to the maximum stroke position is so influenced by the internal pressure of the working chamber 50 that the thrust load decreases as the spool 30 approaches the maximum stroke position. Thus, a non-linear relationship as shown in Fig. 15 is established between the travel of the spool 30 and the axial thrust force required for moving the spool 30. Under such a non-linear relationship, it is impossible to accurately control the displacement of the compressor solely by the control of the pressure in the control pressure chamber 200. In order to continuously control the displacement of the compressor, therefore, it is necessary to obtain such an operation characteristic as represented by a curve P-S in Fig. 15. To cope with this demand, the described embodiment of the swash-plate type compressor of the present invention employs an auxiliary loading means such as a biasing spring 900 (see Fig. 1) for biasing the spool 30 in the direction for reducing the displacement. Thus, the descending slope of the characteristic curve in the region between the points P and Q in Fig. 9 is changed by the auxiliary loading means into ascending slope as shown by broken-line curve P-S therein.

The biasing spring 900 is designed to be effective only when the travel of the spool 30 reaches a range between the point P_2, at which the thrust force shown in Fig. 15 is maximized, and the maximum stroke position MAX in Fig. 15. The spring constant of the biasing spring 900 is so selected as to be large enough to compensate for the decreasing tendency of the thrust load in the region P-Q in Fig. 15.

Fig. 1 shows a swash-plate type compressor in ac-

cordance with the present invention. It is assumed that the stroke of the spool 30 corresponding to the maximum tilting angle of the swash plate 10 is 0 mm, while the maximum stroke of the spool 30 corresponding to the minimum tilting angle of the swash plate 10 is 10 mm. When the spool 30 is positioned in the maximum stroke position, the stroke of the reciprocatory motion of the piston 7 is 20 mm. Assuming here that the maximum volume of the compressor is 180 cc and that the suction pressure P s is 3 kg/cm^2 abs while the discharge pressure P_d ranges between 12 kg/cm^2 abs and 18 kg/cm^2 abs, the change in the gradient of the characteristic curve in Fig. 18 from positive to negative gradient takes place when the travel of the spool 30 is 7 mm or greater. In the embodiment shown in Fig. 1, therefore, the biasing spring 900 is designed to apply a load when the stroke of the spool 30 has become 7 mm or greater. In such a case, the spring constant of the biasing spring 900 is, for example, 33 kg/mm.

The provision of the biasing spring 900 as the auxiliary loading means produces the following effect:

When the travel of the spool 30 is between 0 mm and 7 mm, the thrust load required for moving the spool 30 varies along a curve OP in Fig. 15, so that the travel of the spool 30 is increased substantially linearly in accordance with an increase in the back pressure acting on the spool 30. When the stroke of the spool 30 is increased beyond 7 mm, the biasing spring 900 becomes effective. In this state, the spool 30 cannot be moved reightwards as viewed in Fig. 1 unless a thrust load exceeding the load of the biasing spring 900 is applied to the rear side of the spool 30. Thus, the thrust force required for moving the spool 30 is increased along the curve P-S rather than being decreased along the curve P-Q. Fig. 2 shows the state in which the spool 30 has been moved in excess of 7 mm to commence a compression of the biasing spring 900.

From the foregoing description, it will be seen that the provision of the biasing spring 900 eliminates the reversed tendency of the travel of the spool 30 relative to the thrust load which is caused due to a dead volume in the working chamber 50.

In the embodiment shown in Figs. 1 and 2, the biasing spring 900 is a coiled spring disposed between one end of the shaft 1 and a retainer plate 901 provided on one end of the rear shaft 40. The shaft 1 is prevented by the thrust bearing 15 from being moved axially. The rear shaft 40 slidably fits on the shaft 1 and is connected to the spool 30 through a thrust bearing 116 so as to be able to move as a unit with the spool 30. Therefore, a leftward movement of the spool 30 as viewed in Fig. 1 is transmitted through the thrust bearing 116 to the rear shaft 40 and, hence, to the spherical portion 107. In consequence, the retainer plate 901 on the rear shaft 40 is also displaced and the end of the biasing spring adjacent to the shaft 1 is brought into contact with the shaft 1 when the travel of the spool 30 has reached 7 mm so that the biasing spring 900 begins to exert a pressing load on the spool 30 as shown in Fig. 2.

In the embodiment shown in Figs. 1 and 2, the shaft 1 extends through the swash plate 10 into the rear housing 6 and is supported at axially spaced points by a rear bearing 14 and a front bearing 3. Thus, the shaft 1 is stably supported for smooth rotation. It is also to be noted that the pin 80 is supported in the pin-receiving holes 106 and 108 in the walls defining the slit 105 through the intermediary of bearings 909. Therefore, the operation for changing the tilting angle of the swash plate 10 by an axial movement of the spool 30 encounters only a small friction resistance produced around the pin 80. In consequence, the movement of the spool 30 is smoothly converted into a change in the tilting angle of the swash plate 10 so that the compressor exhibits a highly continuous change in the displacement in accordance with the change in the tilting angle of the swash plate 10.

Although the biasing spring 90 is disposed on the rear end of the shaft 1 in the embodiment shown in Figs. 1 and 2, this is not exclusive and the biasing spring 900 may be disposed at any other portion provided that it becomes effective only when the spool 30 has been moved beyond a predetermined stroke.

Fig. 3 shows different examples of the arrangement of the biasing spring as the auxiliary loading means. A biasing spring 910 may be disposed between the spool 30 and the rear end plate 11. Alternatively, a biasing spring 911 may be disposed between the spool 30 and the rear housing 6. Further alternatively, a spring 912 may be disposed between the spherical support portion 107 and the portion 165 formed on the shaft 1.

In the described embodiment of the present invention, a return spring is used as the auxiliary loading means to compensate for the reduction in the thrust force required for displacing the spool which occurs when the travel of the spool has exceeded a certain value. It will be clear to those skilled in the art, however, the use of the return spring is not exclusive and other means such as pressure means may be used as the auxiliary loading means. It will also be understood that, even when a biasing spring is used as the auxiliary loading means, the spring may have a non-linear characteristic.

As will be understood from the foregoing description, since the swash-plate type compressor of the present invention is provided with the auxiliary loading means which becomes effective when the stroke of the spool has been increased beyond a predetermined value, the travel of the spool 30 can be smoothly and substantially linearly controlled up to the maximum travel by a continuous control of the pressure in the control pressure chamber.

According to experiments and studies made by the present inventors, however, it has been known that the control of the travel of the spool through the control of the pressure in the control pressure chamber 200 tends to become unstable particularly in the region where the stroke of the spool 30 is small. More specifically, in such a region of small stroke of the spool the spool cannot be

moved smoothly in response to a reduction in the pressure in the control pressure chamber 200.

This is attributable to the fact that, as will be seen from Fig. 18, the thrust load required for causing the leftward movement of the spool 30 as viewed in Fig. 1 takes a negative value when the amount of travel of the spool 30 is between 0 and 3 mm while the maximum stroke is 10 mm.

When the travel of the spool 30 is near the minimum value, the level of the residual pressure in the first working chamber 50 is low. On the other hand, in the second working chamber 60, the pressure changes between the levels of the suction pressure and the discharge pressure in accordance with reciprocal movements of the piston 7 because no dead volume is formed in this working chamber 60. Thus, the pressure in the second working chamber 60 can rise up to the level of the discharge pressure P_d even when the compressor is in the region near the minimum spool travel.

A detailed description will be made hereinunder as to the change in the thrust load required for axially displacing the spool 30 when the travel of the spool 30 is small. Fig. 4 shows the state of load applied to the spool 30 when the stroke of the spool 30 is small. In this figure, FPSi represents the sum of the pressing forces exerted to the swash plate 10 by pistons 7 in the compression phase, while FPN represents a force which acts on the pin 80. P_s and P_c represent, respectively, the pressure in the suction chamber 74 and the pressure in the control pressure chamber 200. The thrust loads produced by the pressures P_s and P_c are represented by FP_s and FP_c, respectively. A frictional force between the spherical support portion 107 and the swash plate 100 is represented by FB, while frictional forces between the shaft 1 and the rear shaft 40 and between the spool 30 and the inner surface of the housing 135 are represented by FS and FO, respectively.

As will be seen from Fig. 4, when the compressor is operating with a certain level of displacement with the spool 30 set at a certain travel position, the following condition has to be met if the spool 30 is to be moved toward the minimum stroke position, i.e., to the right as viewed in Fig. 4:

$$FPS_i + FP_s > FPN + FPC + FB + FS + FO$$

When the displacement of the compressor is small, there is no substantial pressure rise in the first working chamber 30 while the pressure in the second working chamber 60 can rise to the level of the discharge pressure. Thus, the value of the force FPS_i is too small to satisfy the condition shown by the above inequality, thus making it impossible to reduce the compressor displacement.

In order that the above-mentioned condition may be met, it is necessary that the value of the first term FPN of the right side of the inequality be reduced or, alternatively, the value of the first term FSP_i of the left side of the inequality be increased.

The force FPN, however, is produced by the pressure differential between the suction chamber 74 and the control pressure chamber 200 and, therefore, it is difficult to reduce this value from the view point of the mechanical construction. Namely, it is difficult to reduce the pressure P_c in the control pressure chamber 200 to a level below the suction pressure P_s. There also is a practical limit in the reduction of the mechanical frictional forces FB, FS and FO because there is a limit in reducing the friction coefficients of component parts.

The present inventors, therefore, have reached a conclusion that it will be a practical measure to increase the value of the force FPS_i for the purpose of enabling the spool 30 to the minimum stroke position without fail. Thus, the present invention also proposed an auxiliary for assuring a minimum spool travel. In an embodiment of the invention, this auxiliary means is constituted by a boosting passage means which provides a communication between the first working chamber 50 and the exterior. In an embodiment shown in Fig. 5, the boosting passage means comprises a boosting port 950 formed in the discharge valve 22. The boosting port is a minute port orifice which has a very small diameter of about 0.2 mm and is adapted for allowing a very small quantity of Q_0 of the refrigerant gas at the discahrge pressure to be returned into the first working chamber 50. It will be seen that the level of the pressure in the first working chamber 50 is elevated as a result of the returning the compressed refrigerant gas into this chamber 50.

Fig. 6 shows how the mean pressure P in the first working chamber 50 is varied in relation to a change in the area of opening of the boosting port 950. As will be seen from this figure, the pressure in the first working chamber 50 is increased as the opening area of the boosting port 950 is increased. In order to obtain a value of the mean pressure P which is large enough to enable the force FPS_i to meet the condition of the inequality mentioned before, it is sufficient to determine the opening area of the boosting port 950 to be greater than A_0.

The displacement C of the compressor was measured in relation to the pressure differential ΔP across the spool 30 in a compressor having the boosting port of an opening area greater than A_0, the result being shown by a solid-line curve H in Fig. 7. The pressure differential ΔP across the spool 30 is the difference between the suction pressure in the suction chambers 73 and 74 and the control pressure P_c in the control pressure chamber 200. From Fig. 7, it will be seen that the pressure differential ΔP across the spool 30 is reduced as a result of provision of the boosting port 950 to ensure that the spool 30 can be fully moved to the minimum travel position thereby enabling the compressor to reduce its displacement to the minimum value. For the purpose of comparison, a broken-line I in Fig. 7 shows the relationship between the pressure differential across the spool 30 and the displacement C observed in a compressor which is not provided with the boosting port 950.

The auxiliary biasing means can be in other forms

than the described boosting port 950, e.g., in the form of a pressure relief passage means. For instance, in an embodiment shown in Figs. 8 and 9 employs, the auxiliary biasing means is in the form of a boosting groove 951 which biases the end plate 8 and the discharge valve 22. This boosting groove 951 is advantageous over the boosting port 950 in that it is less liable to be clogged by foreign matters.

In an embodiment shown in Fig. 10, the auxiliary biasing means is constituted by through-hole 952 which opens at its one end in the discharge chamber 90 and at its other end in the portion of the first working chamber 50 adjacent to the bottom dead center of the piston 7 (shown by a broken line). It will be understood that this through-hole 952 effectively increases the pressure in the first working chamber 50 by introducing a high pressure from the discharge chamber 90. This through-hole 952 can have a comparatively large diameter because the open end 954 thereof can be covered by the side surface of the piston 7 during normal operation of the compressor.

The auxiliary biasing means in the described embodiments is intended to increase the mean pressure in the first working chamber 50 by introducing thereinto a high pressure available in the discharge chamber 90. This, however, is not exclusive and the auxiliary biasing means may be constituted by other types of means.

Fig. 11 shows an embodiment which employs another type of auxiliary biasing means. More specifically, in this embodiment, a dead space is formed also in the second working chamber 60. Namely, an apparent hinge point 1001 of the swash plate 10 is offset from the center of the shoes 18 and 19 towards the shaft 1 so that, when the tilting angle of the swash plate 10 has been decreased, the center of the shoes 18 and 19 is slightly displaced to the left as viewed in Fig. 11. The provision of this small dead volume reduces the mean pressure in the second working chamber 60.

Thus, in the embodiment shown in Fig. 11, a small dead volume is formed in the second working chamber 60 when the spool 30 is set at a small travel position. In consequence, a pressure change caused due to the dead volume produces a force which acts through the pistons 7 on the swash plate 10 thereby urging the swash plate in the direction to reduce the tilting angle. This is illustrated in Fig. 12 in which a solid-line curve J represents the characteristic obtained when the distance Δ between the apparent hinge point 1001 and the center of the shoes 18 and 19 is zero, i.e., when no dead space is formed in the second working chamber 60. Characteristics obtained when the distance Δ is 2.5 mm, 5 mm and 7.5 mm are represented, respectively, by a broken-line curve K, a one-dot-and-dash line L and a two-dot-and-dash line M. As will be understood from these lines K, L and M, the load F is increased as the dead volume is increased. The increase in the load F causes the total pressing force FPS_i exerted on the swash plate 10 by pistons 7, so that the aforementioned inequality is satisfied, thus ensuring that the displacement of the compressor can be controlled down to the minimum value.

Figs. 20 and 21 show a compressor provided with the auxiliary biasing means of the type shown in Fig. 11, in different states of operation. More specifically, Fig. 20 shows the state in which the displacement of the compressor has been reduced to the minimum. It will be apparent from this figure that the tilting angle θ of the swash plate 10 and, hence, the stroke S of the pistons 7 are minimum. Thus, the pressure in the first working chamber 50 in the state shown in Fig. 20 does not reach the level of the discahrge pressure, so that the discharge valve is kept closed. Fig. 21 shows the compressor in the state in which the displacement of the compressor has been maximized.

Fig. 22 shows the operation characteristic of this compressor in comparison with that of a known compressor of the type shown in Japanese Unexamined Patent Publication No. 60-175783 in which pistons are provided only on one side of the swash plate. Solid-line curves A and B in this figure represent the driving torques of the compressor in accordance with the present invention, while broken-line curves C and D represent the driving torques in the known compressor having pistons only on one side of the swash plate. The curves A and C show the levels of driving torques required when the respective compressors operate with their maximum displacements, while the curves B and C show the levels of driving torques required when the respective compressors operate with their minimum displacements. In Fig. 22, the axis of abscissa represents the angle of rotation of the shaft.

As will be understood from Fig. 22, the levels of the driving torques in both compressors are substantially the same when the displacements are minimum. This is because, in the minimum displacements only the second working chamber 60 are operative and the first working chambers 50 are inoperative even in the compressor of this embodiment.

However, when the displacement have been maximized, the level of the driving torque required by the compressor of this embodiment is much smaller than that required by the known compressor due to the fact that the amplitude of fluctuation in the required torque per rotation of the shaft is very small in the compressor of this embodiment because of the provision of the working chambers 50 and 60 on both sides of the pistons 7. In contrast, in the known compressor, the amplitude of fluctuation in the driving torque per rotation of the shaft is very large because pistons operate only on one side of the swash plate.

Another advantage provided by the compressor of the described embodiment is that the members for controlling the tilting movement of the swash plate 10, i.e., the pin 80, elongated slot 166 and the spherical support portion 405, do not directly bear the driving torque because the transmission of the driving torque from the

shaft to the swash plate relies upon the surface contact between the flat web portion 165 on the shaft 1 and the walls defining the slit 105 which receives this web portion.

In contrast, in the prior art compressors (known, for example, from Japanese Unexamined Patent Publication No. 58-162780), the torque for driving the swash plate and the thrust force exerted on the swash plate during compression are born by the pin which constitutes the center or fulcrum for the tilting motion of the swash plate. This arrangement undesirably limits the driving torque to be transmitted. The compressor of the described embodiment of the invention is free from this problem.

Furthermore, according to the invention, the control of the tilting motion of the swash plate 10 can be accomplished by the spool adapted to slide by a presssure differential across it, without necessitating any complicated and large-sized actuator such as a motor, thus enabling the swash-plate type compressor to have a reduced size and a compact construction.

Claims

1. A variable displacement swash-plate type compressor comprising:

   a cylinder block (5, 6) having cylinder bores (64) therein;
   a shaft (1) rotatably supported in said cylinder block;
   a swash plate (10) tiltably connected to said shaft (1) and adapted to be rotated together with said shaft;
   double-headed pistons (7) slidably received in said cylinder bores (64) and adapted to reciprocatorily stroke in said cylinder bores in accordance with an oscillatory motion of said swash plate (10),
   working chambers (50, 60) formed between both ends of each of said pistons and adjacent surfaces of said cylinder bores, characterized in that
   a support portion (40, 405, 107) disposed coaxially with said shaft (1) and adapted to support a central portion of said swash plate (10) rotatably and oscillatably;
   a spool (30) for driving said support portion axially of said shaft;
   control means (200) for driving said spool (30) axially of said shaft (1) between a maximum displacement position in which the angle of tilt of said spool is maximized and a minimum displacement position in which the angle of tilt of said swash plate is minimized; and
   auxiliary loading means (900) operative when said control means (200) has driven said spool (30) more than a predetermined distance from said minimum displacement position to apply to said spool a load which acts to suppress the movement of said spool towards said maximum displacement position more than said predetermined distance from said minimum displacement position;
   the movement of said spool (30) displacing said central portion of said swash plate (10) axially of said shaft (1) and changing the angle of tilt of said swash plate.

2. A variable displacement swash-plate type compressor according to Claim 1, wherein said auxiliary loading means comprise a spring means (900).

3. A variable displacement swash-plate type compressor according to Claim 2, wherein said spring means (900) is disposed between said spool (30) and one end of said shaft (1) in such a manner that the compression of said spring means is commenced when said spool has traveled in excess of said predetermined distance.

4. A variable displacement swash-plate type compressor according to Claim 2, wherein said spring means (911) is disposed between said spool (30) and said cylinder block (5, 6) in such a manner that the loading by said spring means is commenced when said spool has traveled in excess of said predetermined distance.

5. A variable displacement swash-plate type compressor according to Claim 2, further including an end plate (11) attached to an end of said cylinder block (5, 6) and covering said working chambers (50), and wherein said spring (910) means is disposed between said spool (30) and said end plate (11) in such a manner that said spring means becomes effective to produce a resilient load when said spool has traveled in excess of said predetermined distance.

6. A variable displacement swash-plate type compressor according to Claim 2, wherein said spring means (912) is disposed between said support portion (107) and a retainer portion (165) provided on said shaft (1) so as to oppose said support portion, said spring means (912) becoming effective to produce a resilient load when said support portion has traveled in excess of said predetermined distance.

7. A variable displacement swash-plate type compressor comprising:

   a cylinder block (5, 6) having cylinder bores (64) therein;
   a shaft (1) rotatably supported in said cylinder

block;
a swash plate (10) tiltably connected to said shaft (1) and adapted to be rotated together with said shaft;
double-headed pistons (7) slidably received in said cylinder bores (64) and adapted to reciprocatorily stroke in said cylinder bores in accordance with an oscillatory motion of said swash plate (10);
working chambers (50, 60) formed between both ends of each of said pistons and adjacent surfaces of said cylinder bores, characterized in that
a support portion (40, 405, 107) disposed coaxially with said shaft (1) and adapted to support a central portoin of said swash plate (10) rotatably and oscillatably;
a spool (30) for driving said support portion axially of said shaft;
control means (200) for driving said spool (30) axially of said shaft (1) to move said central portion of said swash plate axially of said shaft while changing the angle of tilt of said swash plate, thereby causing the strokes of reciprocatory movements of said pistons (7) in said cylinder bores (64) to vary in such a manner that the position of the stroke end of each piston in a first working chamber (50) on one end thereof is different from the position of the stroke end of said piston in a second working chamber (60) on the other end of said piston; and
auxiliary biasing means (450, 900) for applying a biasing force acting in a direction from said first working chamber (50) towards said second working chamber (60) so as to produce a torque which acts in the direction to reduce the angle of tilt of said swash plate (10).

8. A variable displacement swash-plate type compressor according to Claim 7, wherein said auxiliary biasing means comprises a spring (911, 912) which applies a load to said spool (30) in a direction to reduce the amount of movement of said spool towards said support portion (107).

9. A variable displacement swash-plate type compressor according to Claim 8, wherein said spring (900) is disposed to contact at its ends with said spool (30) and with said shaft (1).

10. A variable displacement swash-plate type compressor according to Claim 7, wherein said auxiliary biasing means comprises a boosting passage means (950) which introduces the discharge pressure of said compressor into said first working chamber (50).

11. A variable displacement swash-plate type compressor according to Claim 10, wherein said auxiliary biasing means comprises a communication means (952) which provides communication between said first working chamber (50) and a discharge chamber (90), said discharge chamber communicating with said second chamber (60) through a discharge port (24) and a discharge valve (22).

12. A variable displacement swash-plate type compressor according to Claim 11, further including a side plate (8) provided on one end surface of said cylinder block (5), said side plate having said first working chamber and said discharge port (24) through which said second working chamber (60) communicates with said discharge chamber (90), said communicating means being provided in said side plate (8).

13. A variable displacement swash-plate type compressor according to Claim 11, wherein said communicating means comprises a relief port formed in said discharge valve (22).

14. A variable displacement swash-plate type compressor according to Claim 7, wherein said auxiliary biasing means comprises a dead volume generating means (1001) for changing the position of stroke end of said piston (7) in said second working chamber (60) and establishing a dead volume on the end of said piston in said second working chamber (60).

15. A variable displacement swash-plate type compressor according to Claim 14, further comprising:

a flat web portion (165) formed on said shaft (1);
a slit (105) provided on said swash plate (10) and receiving said web portion;
a slot (166) formed in said flat web portion (165), pin-receiving holes (106, 108) formed in portions of said swash plate (10) which define said slit (105); and
a pin (80) extending through said slot and received in said pin-receiving holes so as to tiltably connect said swash plate to said shaft;
said slot (166) being so designed that, when the angle of tilt of said swash plate (10) is decreased as a result of movement of said spool (30), a dead volume is formed in said second working chamber (60).

Patentansprüche

1. Taumelscheibenkompressor mit veränderlicher Verdrängung umfassend:

einen Zylinderblock (5, 6) mit darin befindlichen Zylinderbohrungen (64),
eine in dem Zylinderblock drehbar gelagerte

Welle (1),
eine Taumelscheibe (10), die mit der Welle (1) kippbar verbunden und geeignet ist, zusammen mit der Welle gedreht zu werden,
doppelseitige Kolben (7), die in den Zylinderbohrungen (64) aufgenommen und für einen hin- und hergehenden Hub in den Zylinderbohrungen entsprechend einer oszillierenden Bewegung der Taumelscheibe (10) geeignet sind,
Arbeitskammern (50, 60), die zwischen den beiden Enden jedes Zylinders und benachbarten Flächen der Zylinderbohrungen (64) ausgebildet sind, **gekennzeichnet durch**
einen Abstützbereich (40, 405, 107), der koaxial zu der Welle (1) angeordnet und zur drehbaren und oszillierenden Abstützung eines Zentralbereichs der Taumelscheibe (10) geeignet ist,
eine Nabe (30) zum axialen Antrieb des Abstützbereichs der Welle,
Regelungsmittel (200) zum axialen Antrieb der Nabe (30) hinsichtlich der Welle (1) zwischen einer Maximalverdrängungsstellung, in der der Kippwinkel der Taumelscheibe maximiert ist, und einer Mininalverdrängungsstellung, in der der Winkel der Taumelscheibe minimiert ist, und ein Hilfslastmittel (900), das in Betrieb genommen werden kann, wenn das Regelungsmittel (200) die Nabe (30) um mehr als eine vorbestimmte Strecke von der Minimalverdrängungsstellung aus angetrieben hat, um auf die Nabe eine Last zur Einwirkung zu bringen, die in Hinblick darauf wirkt, die Bewegung der Nabe in Richtung auf die Maximalverdrängungsstellung um mehr als die vorbestimmte Strecke von der Mininalverdrängungsstellung aus anzuhalten,
wobei die Bewegung der Nabe (30) den Zentralbereich der Taumelscheibe (10) axial zur Welle (1) verschiebt und den Kippwinkel der Taumelscheibe verändert.

2. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 1, wobei das Hilfslastmittel ein Federmittel (900) umfaßt.

3. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 2, wobei das Federmittel (900) zwischen der Nabe (30) und einem Ende der Welle (1) derart angeordnet ist, daß das Zusammendrücken des Federmittels beginnt, wenn sich die Nabe über die vorbestimmte Strecke hinaus bewegt hat.

4. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 2, wobei das Federmittel (911) zwischen der Nabe (30) und dem Zylinderblock (5, 6) derart angeordnet ist, daß das Belasten mittels des Federmittels beginnt, wenn sich die Nabe über die vorbestimmte Strecke hinaus bewegt hat.

5. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 2, weiter mit einer Stirnplatte (11), die an einem Ende des Zylinderblocks (5, 6) befestigt ist und die Arbeitskammern (50) abdeckt, und wobei das Federmittel (910) zwischen der Nabe (30) und der Stirnplatte (11) derart angeordnet ist, daß das Federmittel in Hinblick auf die Erzeugung einer nachgiebigen Last wirksam wird, wenn sich die Nabe über die vorbestimmte Strecke hinaus bewegt hat.

6. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 2, wobei das Federmittel (912) zwischen dem Abstützbereich (107) und einem Haltebereich (165) angeordnet ist, der an der Welle (1) vorgesehen ist, um so dem Abstützbereich gegenüberzuliegen, wobei das Federmittel (912) zur Erzeugung einer nachgiebigen Last wirksam wird, wenn sich der Abstützbereich über die vorbestimmte Strecke hinaus bewegt hat.

7. Taumelscheibenkompressor mit veränderlicher Verdrängung umfassend:

einen Zylinderblock (5, 6) mit darin befindlichen Zylinderbohrungen (64),
eine in dem Zylinderblock drehbar gelagerte Welle (1),
eine Taumelscheibe (10), die mit der Welle (1) kippbar verbunden und geeignet ist, zusammen mit der Welle gedreht zu werden,
doppelseitige Kolben (7), die in den Zylinderbohrungen (64) aufgenommen und für einen hin- und hergehenden Hub in den Zylinderbohrungen entsprechend einer oszillierenden Bewegung der Taumelscheibe (10) geeignet sind,
Arbeitskammern (50, 60), die zwischen den beiden Enden jedes Zylinders und benachbarten Flächen der Zylinderbohrungen (64) ausgebildet sind, **gekennzeichnet durch**
einen Abstützbereich (40, 405, 107), der koaxial zu der Welle (1) angeordnet und zur drehbaren und oszillierenden Abstützung eines Zentralbereichs der Taumelscheibe (10) geeignet ist,
eine Nabe (30) zum axialen Antrieb des Abstützbereichs der Welle,
ein Regelungsmittel (200) zum Antrieb der Nabe (30) axial zur Welle (1), um den zentralen Bereich der Taumelscheibe axial zur Welle zu bewegen, während der Kippwinkel der Taumelscheibe verändert wird, wodurch bewirkt wird, daß sich die Hübe der hin- und hergehenden Bewegungen der Kolben (7) in den Zylinderbohrungen (64) in solcher Weise verändern, daß

sich die Stellung des Hubendes jedes Kolbens in einer ersten Arbeitskammer (50) an einem Ende davon von der Stellung des anderen Hubendes des Kolbens in einer zweiten Arbeitskammer (60) am anderen Ende des Kolbens unterscheidet, und
ein Hilfsvorspannmittel (950, 900) um eine Vorspannkraft aufzubringen, die in einer Richtung von der ersten Arbeitskammer (50) aus zu der zweiten Arbeitskammer (60) hin wirkt, um so ein Moment zu erzeugen, das in der Richtung zur Verkleinerung des Kippwinkels der Taumelscheibe (10) wirkt.

8. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 7, wobei das Hilfsvorspannmittel eine Feder (911, 912) umfaßt, die eine Last auf die Nabe (30) in einer Richtung zur Verkleinerung der Größe der Bewegung der Nabe in Richtung auf den Abstützbereich (107) zur Einwirkung bringt.

9. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 8, wobei die Feder (900) so angeordnet ist, daß sie an ihren Enden mit der Nabe (30) und mit der Welle (1) in Berührung steht.

10. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 7, wobei das Hilfsvorspannmittel ein Vergrößerungsdurchtrittsmittel (950) umfaßt, das den Abgabedruck des Kompressors in die erste Arbeitskammer (50) einführt.

11. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 10, wobei das Hilfsvorspannmittel ein Verbindungsmittel (952) umfaßt, das eine Verbindung zwischen der ersten Arbeitskammer (50) und einer Abgabekammer (90) schafft, wobei die Abgabekammer mit der zweiten Kammer (60) über einen Abgabeanschluß (24) und ein Abgabeventil (22) in Verbindung steht.

12. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 11, weiter mit einer Seitenplatte (8), die an einer Stirnfläche des Zylinderblocks (5) angeordnet ist, wobei die Seitenplatte die erste Arbeitskammer und den Abgabeanschluß (24) aufweist, durch den hindurch die zweite Arbeitskammer (60) mit der Abgabekammer (90) in Verbindung steht, und wobei das Verbindungsmittel in der Seitenplatte (8) vorgesehen ist.

13. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 11, wobei das Verbindungsmittel einen Entlastungsanschluß umfaßt, der in dem Abgabeventil (22) ausgebildet ist.

14. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 7, wobei das Hilfsvorspannmittel ein ein Totvolumen erzeugendes Mittel (1001) zur Veränderung der Stellung des Hubendes des Kolbens (7) in der zweiten Arbeitskammer (60) und zur Ausbildung eines Totvolumens an dem Ende des Kolbens in der zweiten Arbeitskammer (60) umfaßt.

15. Taumelscheibenkompressor mit veränderlicher Verdrängung nach Anspruch 14, weiter umfassend

einen flachen stegartigen Bereich (165), der an der Welle (1) ausgebildet ist,
einen Schlitz (105), der an der Taumelscheibe (10) vorgesehen ist und den stegartigen Bereich aufnimmt,
einen Schlitz (166), der in dem flachen stegartigen Bereich (165) ausgebildet ist,
einen Stift aufnehmende Löcher (106, 108), die in Bereichen der Taumelscheibe (10) ausgebildet sind, die den Schlitz (105) bilden, und
einen Stift (80), der sich durch den Schlitz hindurch erstreckt und in den Stiftaufnahmelöchern aufgenommen ist, um die Taumelscheibe kippbar mit der Welle zu verbinden,
wobei der Schlitz (166) derart gestaltet ist, daß dann, wenn der Kippwinkel der Taumelscheibe (10) als Folge der Bewegung der Nabe (30) verkleinert wird, ein Totvolumen in der zweiten Arbeitskammer (60) gebildet wird.

Revendications

1. Compresseur à plateau en biais à compression variable comprenant :

un bloc-cylindres (5, 6) présentant des alésages de cylindre (64) ;
un arbre (1) supporté de façon à pouvoir tourner dans ledit bloc-cylindres ;
un plateau en biais (10) connecté de facon inclinable audit arbre (1) et adapté pour pouvoir être entraîné en rotation avec ledit arbre ;
des pistons double-tête (7) reçus de façon pouvoir coulisser dans lesdits alésages de cylindre (64) et adaptés pour avoir une course en va-et-vient dans lesdits alésages de cylindre en fonction d'un mouvement d'oscillation dudit plateau en biais (10) ;
des chambres de travail (50, 60) formées entre les deux extrémités de chacun desdits pistons et les surfaces adjacentes desdits alésages de cylindre ;
caractérisé par
une partie de support (40, 405, 107) située coaxialement audit arbre (1) et adaptée pour supporter une partie centrale dudit plateau en

biais (10) de façon à permettre sa rotation et son oscillation ;
une bobine (30) destinée à entraîner ladite partie de support axialement par rapport audit arbre ;
des moyens de commande (200) destinés à entraîner ladite bobine (30) axialement par rapport audit arbre (1) entre une position de déplacement maximal dans laquelle l'angle d'inclinaison du plateau en biais est maximal et une position de déplacement minimal dans laquelle l'angle d'inclinaison dudit plateau en biais est minimal ; et
des moyens de force auxiliaires (900) destinés à fonctionner quand lesdits moyens de commande (200) ont entraîné ladite bobine (30) sur une distance supérieure à une distance prédéterminée depuis ladite position de déplacement minimal, afin d'appliquer à ladite bobine une force destinée à empêcher le déplacement de ladite bobine vers ladite position de déplacement maximal sur une distance supérieure à ladite distance prédéterminée depuis ladite position de déplacement minimal ;
le déplacement de ladite bobine (30) déplaçant ladite partie centrale du plateau en biais (10) axialement par rapport audit arbre (1) et faisant varier l'angle d'inclinaison dudit plateau en biais.

2. Compresseur à plateau en biais à compression variable selon la revendication 1, dans lequel lesdits moyens de force auxiliaires comprennent des moyens de ressort (900).

3. Compresseur à plateau en biais à compression variable selon la revendication 2, dans lequel lesdits moyens de ressort (900) sont placés entre ladite bobine (30) et une extrémité dudit arbre (1), de façon à ce que la compression desdits moyens de ressort commence lorsque ladite bobine s'est déplacée davantage que ladite distance prédéterminée.

4. Compresseur à plateau en biais à compression variable selon la revendication 2, dans lequel lesdits moyens de ressort (911) sont placés entre ladite bobine (30) et ledit bloc-cylindres (5, 6), de façon à ce que la force exercée par lesdits moyens de ressort commence à s'appliquer lorsque ladite bobine s'est déplacée davantage que ladite distance prédéterminée.

5. Compresseur à plateau en biais à compression variable selon la revendication 2, comprenant en outre une plaque d'extrémité (11) fixée à une extrémité dudit bloc-cylindres (5, 6) et couvrant lesdites chambres de travail (50), et dans lequel lesdits moyens de ressort (910) sont placés entre ladite bobine (30) et ladite plaque d'extrémité (11), de façon à ce que lesdits moyens de ressort produisent une force élastique lorsque ladite bobine s'est déplacée davantage que ladite distance prédéterminée.

6. Compresseur à plateau en biais à compression variable selon la revendication 2, dans lequel lesdits moyens de ressort (912) sont placés entre ladite partie de support (107) et une partie de retenue (165) prévue sur ledit arbre (1) de façon à faire face à ladite partie de support, lesdits moyens de ressort (912) permettant de produire une force élastique lorsque ladite partie de support s'est déplacée davantage que ladite distance prédéterminée.

7. Compresseur à plateau en biais à compression variable comprenant :

un bloc-cylindres (5, 6) présentant des alésages de cylindre (64) ;
un arbre (1) supporté de façon à pouvoir tourner dans ledit bloc-cylindres ;
un plateau en biais (10) connecté de façon inclinable audit arbre (1) et adapté pour pouvoir être entraîné en rotation avec ledit arbre ;
des pistons double-tête (7) reçus de façon à pouvoir coulisser dans lesdits alésages de cylindre (64) et adaptés pour avoir une course en va-et-vient dans lesdits alésages de cylindre en fonction d'un mouvement d'oscillation dudit plateau en biais (10) ;
des chambres de travail (50, 60) formées entre les deux extrémités de chacun desdits pistons et les surfaces adjacentes desdits alésages de cylindre ;
caractérisé par
une partie de support (40, 405, 107) située coaxialement audit arbre (1) et adaptée pour supporter une partie centrale dudit plateau en biais (10) de façon à permettre sa rotation et son oscillation ;
une bobine (30) destinée à entraîner ladite partie de support axialement par rapport audit arbre ;
des moyens de commande (200) destinés à entraîner ladite bobine (30) axialement par rapport audit arbre (1) afin de déplacer ladite partie centrale dudit plateau en biais axialement par rapport audit arbre tout en faisant varier l'angle d'inclinaison dudit plateau en biais, de façon à faire varier la course des mouvements de va-et-vient desdits pistons (7) dans lesdits alésages de piston (64) de manière telle que la position de l'extrémité de course de chaque piston dans une première chambre de travail (50) à l'extrémité de celui-ci est différente de la position de l'extrémité de course dudit piston dans

une deuxième chambre de travail (60) à l'autre extrémité dudit piston ; et
des moyens de polarisation auxiliaires (950, 900) destinés à appliquer une force de polarisation agissant dans une direction allant de ladite première chambre de travail (50) à ladite deuxième chambre de travail (60), afin de produire un couple agissant dans cette direction pour réduire l'angle d'inclinaison dudit plateau en biais (10).

8. Compresseur à plateau en biais à compression variable selon la revendication 7, dans lequel lesdits moyens de polarisation auxiliaires comprennent un ressort (911, 912) qui applique une force à ladite bobine (30) dans une direction pour réduire l'importance du déplacement de ladite bobine vers ladite partie de support (107).

9. Compresseur à plateau en biais à compression variable selon la revendication 8, dans lequel ledit ressort (900) est placé de façon à ce que ses extrémités soient en contact avec ladite bobine (30) et avec ledit arbre (1).

10. Compresseur à plateau en biais à compression variable selon la revendication 7, dans lequel lesdits moyens de polarisation auxiliaires comprennent des moyens de passage d'élévation (950) qui introduisent la pression de décharge dudit compresseur à l'intérieur de ladite première chambre de travail (50).

11. Compresseur à plateau en biais à compression variable selon la revendication 10, dans lequel lesdits moyens de polarisation auxiliaires comprennent des moyens de communication (952) qui assurent la communication entre ladite première chambre de travail (50) et une chambre de décharge (90), ladite chambre de décharge communicant avec ladite seconde chambre (60) par l'intermédiaire d'un orifice de décharge (24) et d'une soupape de décharge (22).

12. Compresseur à plateau en biais à compression variable selon la revendication 11, comprenant en outre une plaque latérale (8) prévue sur une surface d'extrémité dudit bloc-cylindres (5), ladite plaque latérale comprenant ladite première chambre de travail et ledit orifice de décharge (24) par l'intermédiaire duquel ladite deuxième chambre de travail (60) communique avec ladite chambre de décharge (90), lesdits moyens de communication étant prévus dans ladite plaque latérale (8).

13. Compresseur à plateau en biais à compression variable selon la revendication 11, dans lequel lesdits moyens de communication comprennent un orifice de décharge formé dans ladite soupape de décharge (22).

14. Compresseur à plateau en biais à compression variable selon la revendication 7, dans lequel lesdits moyens de polarisation auxiliaires comprennent des moyens de créer un volume mort (1001) destinés à faire varier la position de l'extrémité de course dudit piston (7) dans ladite deuxième chambre de travail (60) et à établir un volume mort à l'extrémité dudit piston dans ladite deuxième chambre de travail (60).

15. Compresseur à plateau en biais à compression variable selon la revendication 14, comprenant en outre :

une partie de bande plate (165) formée sur ledit arbre (1) ;
une fente (105) prévue sur ledit plateau en biais (10) et recevant ladite partie de bande ;
une rainure (166) formée dans ladite partie de bande plate (165) ;
des trous destinés à recevoir des broches (106, 108) formés dans des parties dudit plateau en biais (10) qui définissent ladite fente (105) ; et
une broche (80) s'étendant à travers ladite rainure et reçue dans lesdits trous destinés à recevoir des broches afin de relier de façon inclinable ledit plateau en biais audit arbre ;
ladite rainure (166) étant conçue de manière à ce que, lorsque l'angle d'inclinaison dudit plateau en biais (10) est réduit en conséquence du déplacement de ladite bobine (30), un volume mort est formé dans ladite deuxième chambre de travail (60).

FIG. 1

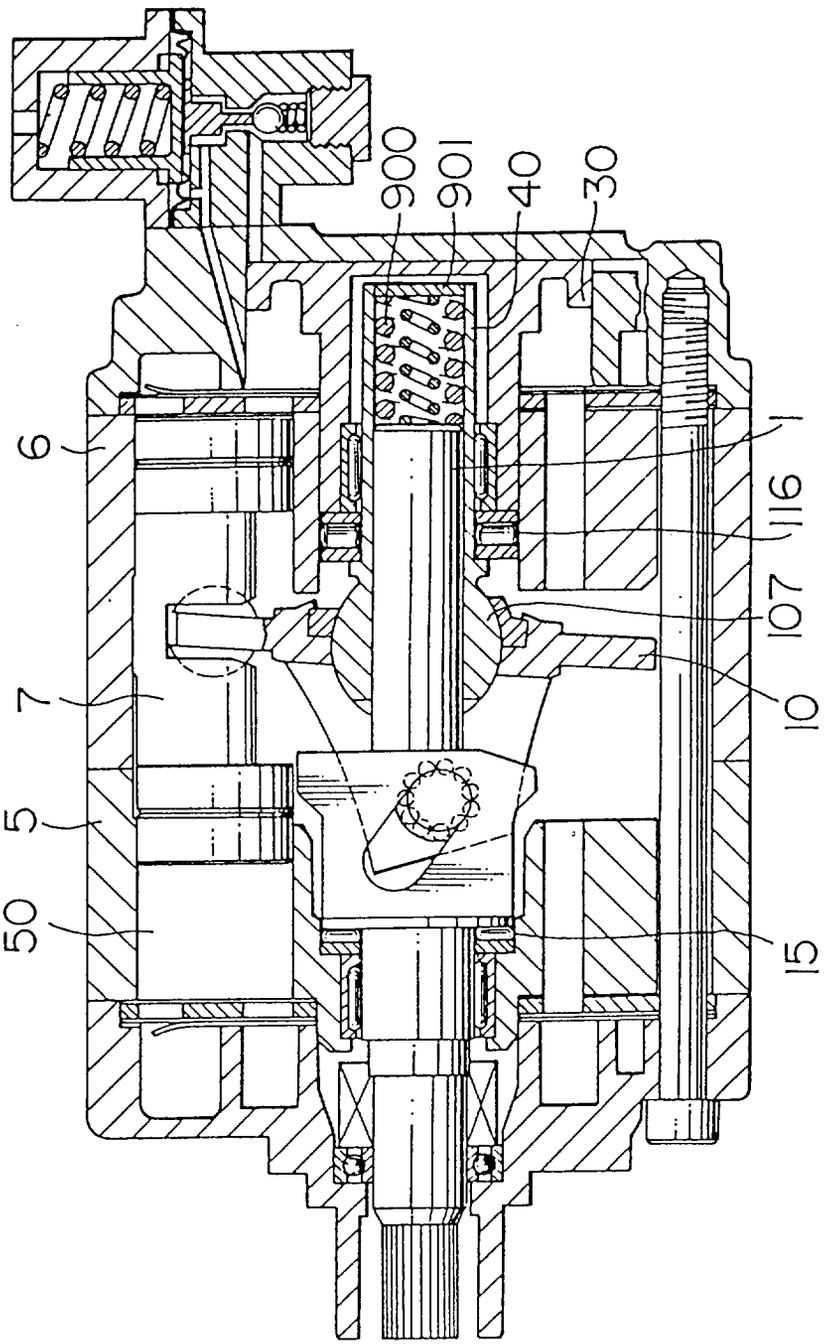


FIG. 2

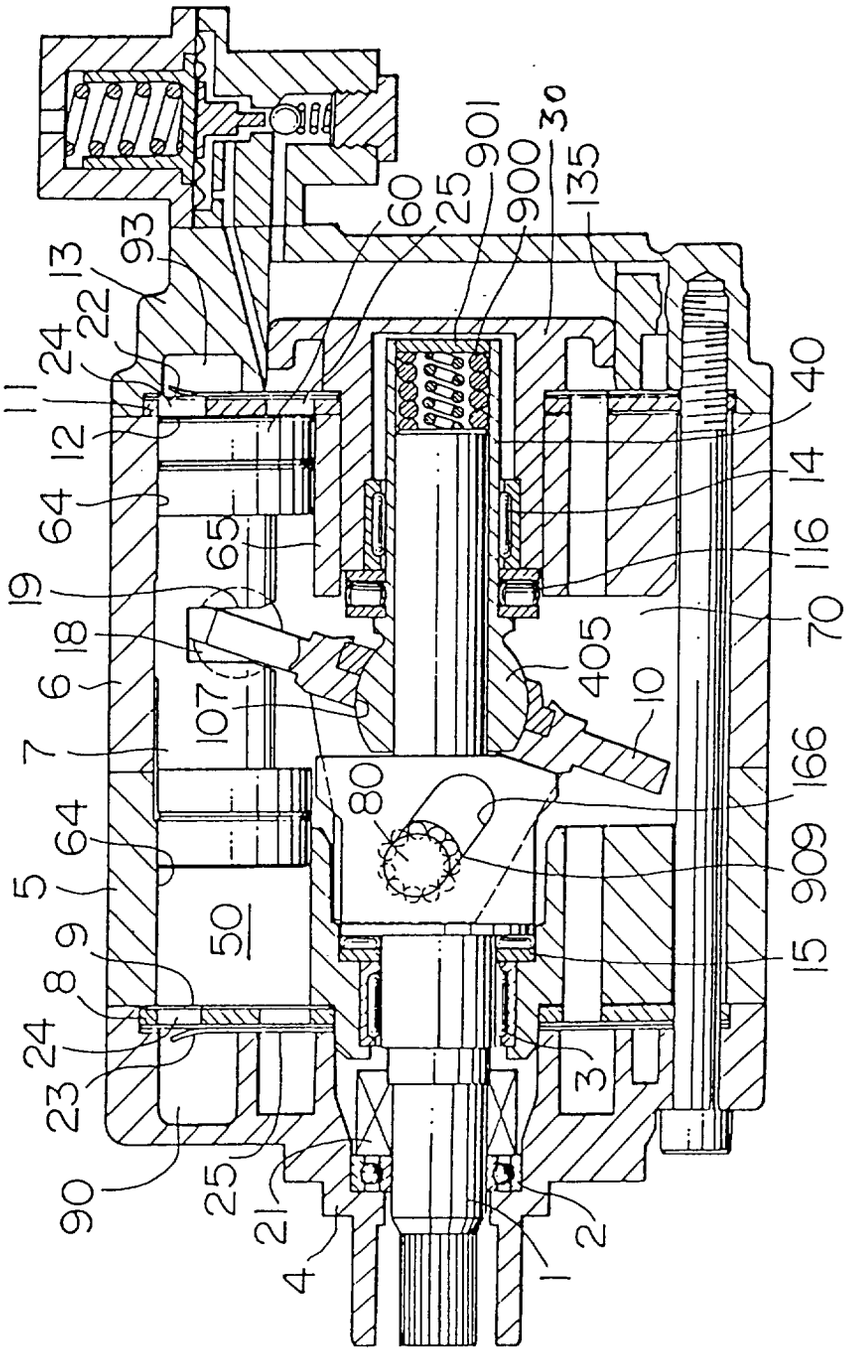

60
25
901
900
30
135
93
13
22
24
11
12
64
65
19
18
6
107
7
64
5
50
9
8
24
23
90
25
21
4
1
2
3
80
405
10
166
909
15
40
14
116
70

FIG. 3

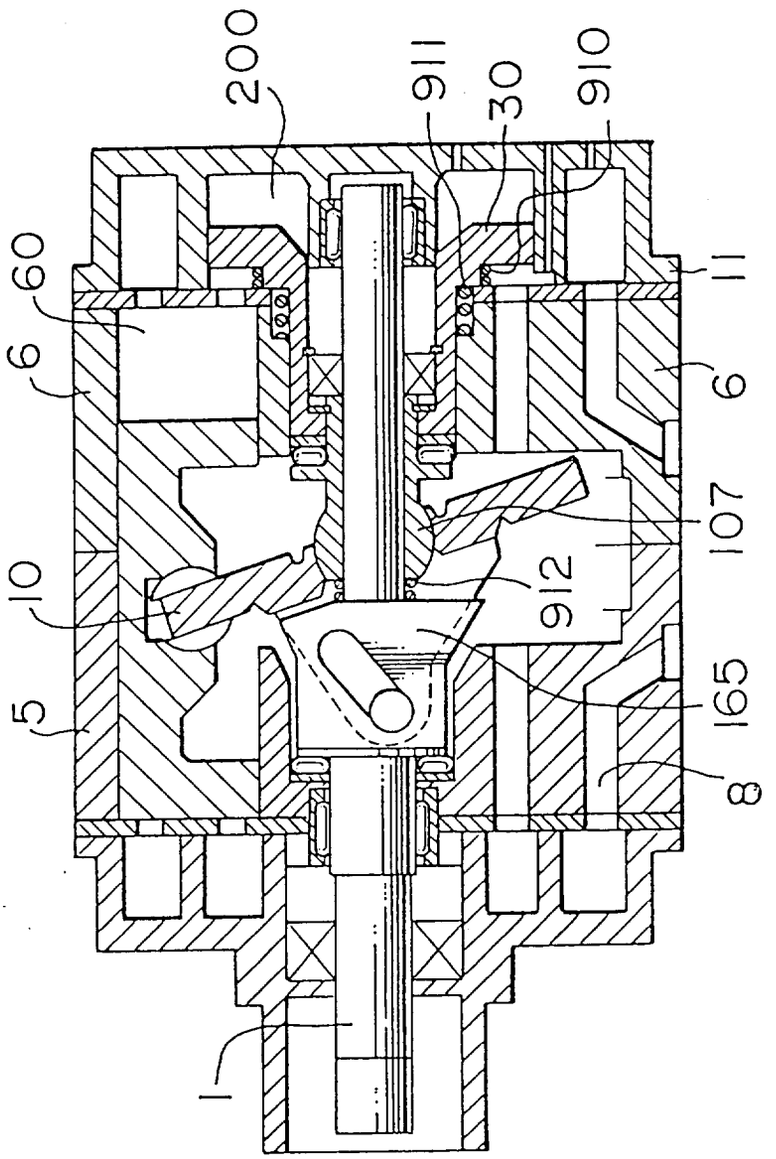


FIG. 4

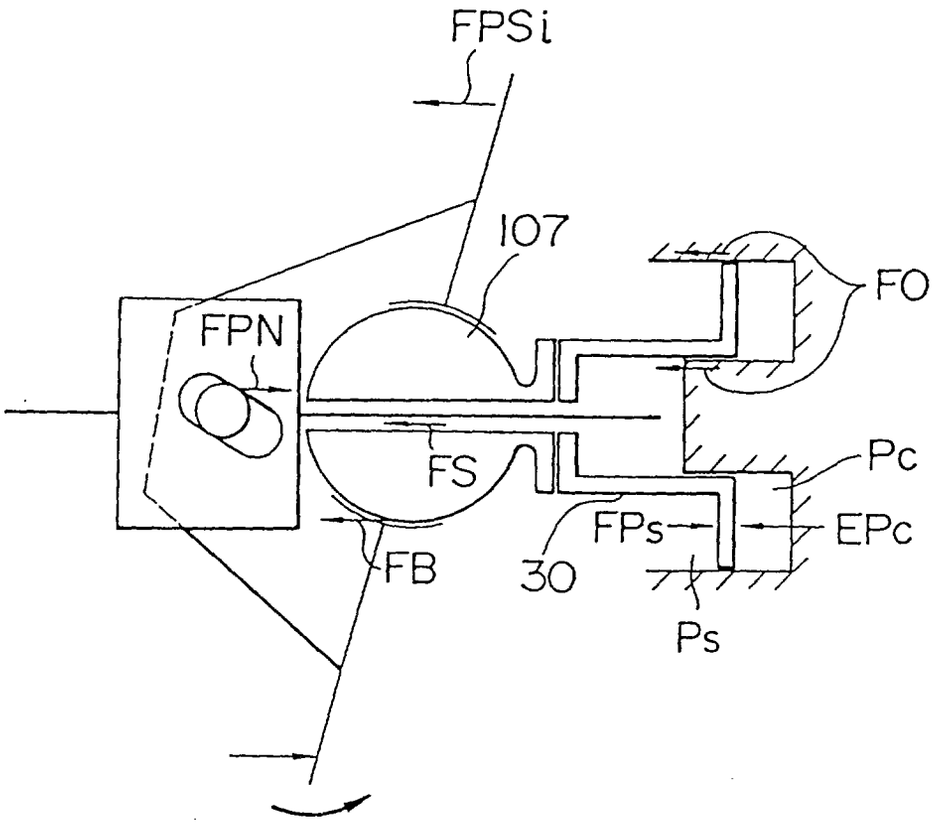


FIG. 5

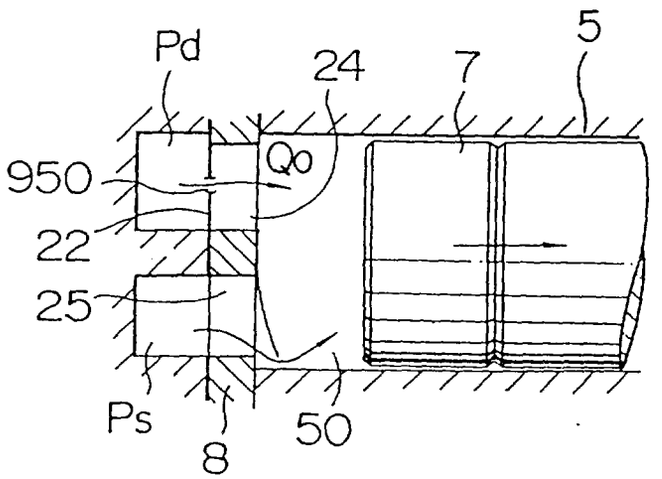


FIG. 6

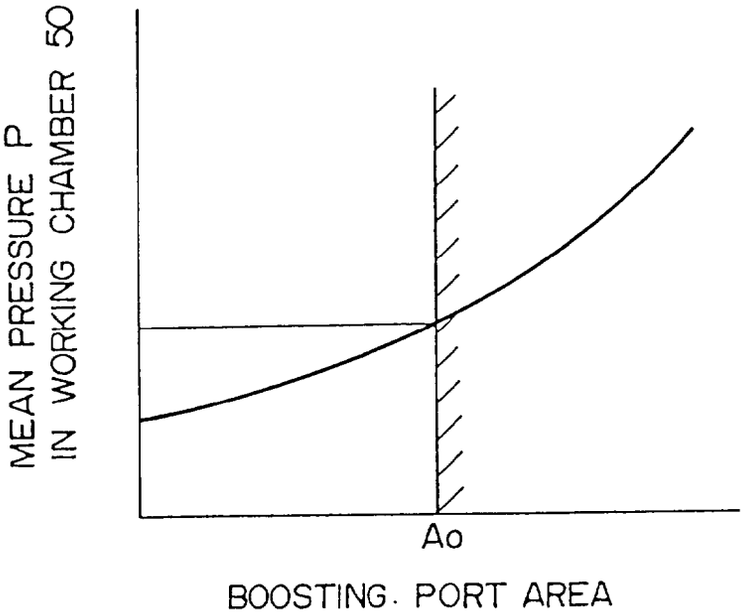

MEAN PRESSURE P
IN WORKING CHAMBER 50
Ao
BOOSTING PORT AREA

FIG. 7

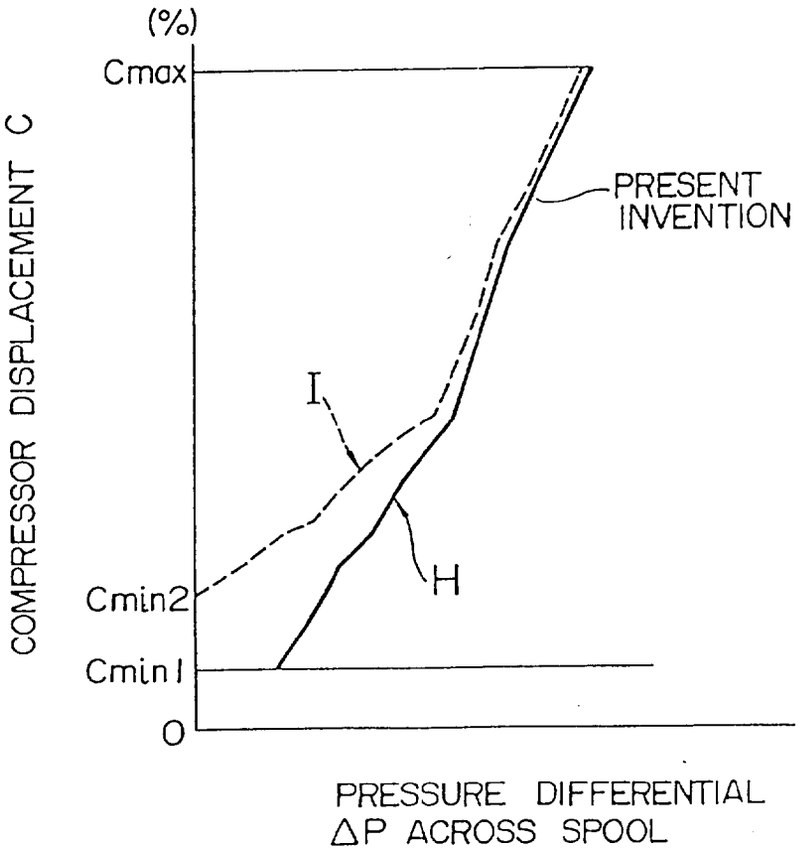

(%)
Cmax
PRESENT
INVENTION
I
H
Cmin2
Cmin1
0
COMPRESSOR DISPLACEMENT C
PRESSURE DIFFERENTIAL
ΔP ACROSS SPOOL

FIG. 8

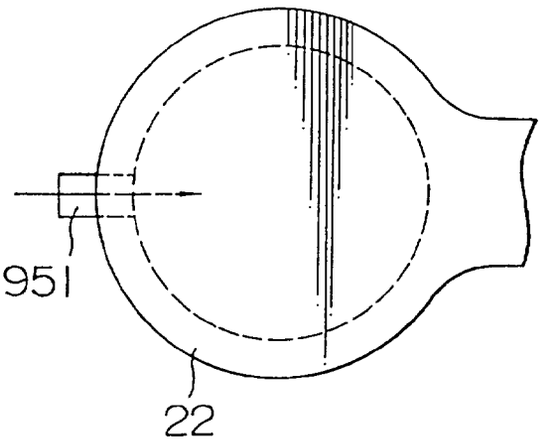


FIG. 9

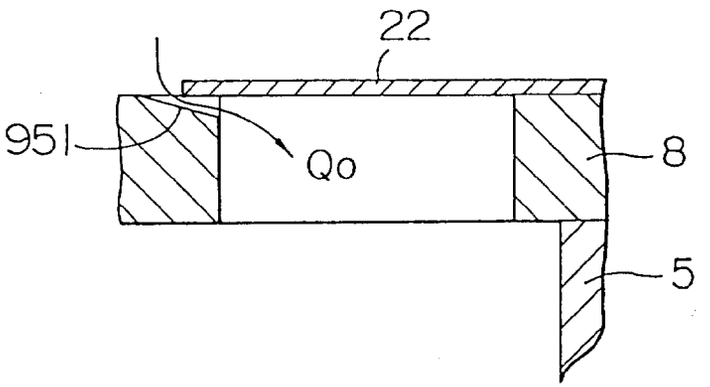


FIG. 10

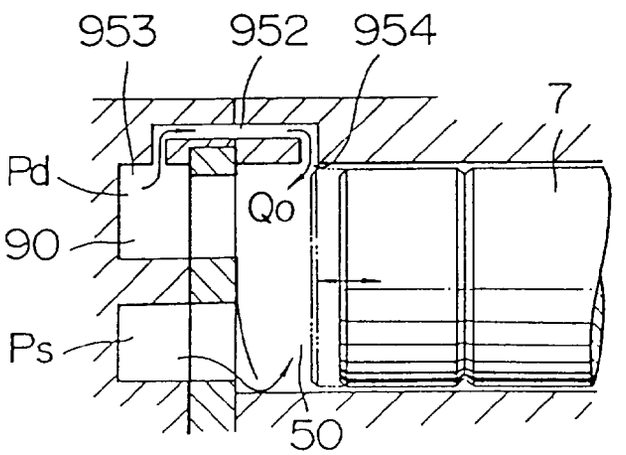


FIG. 11

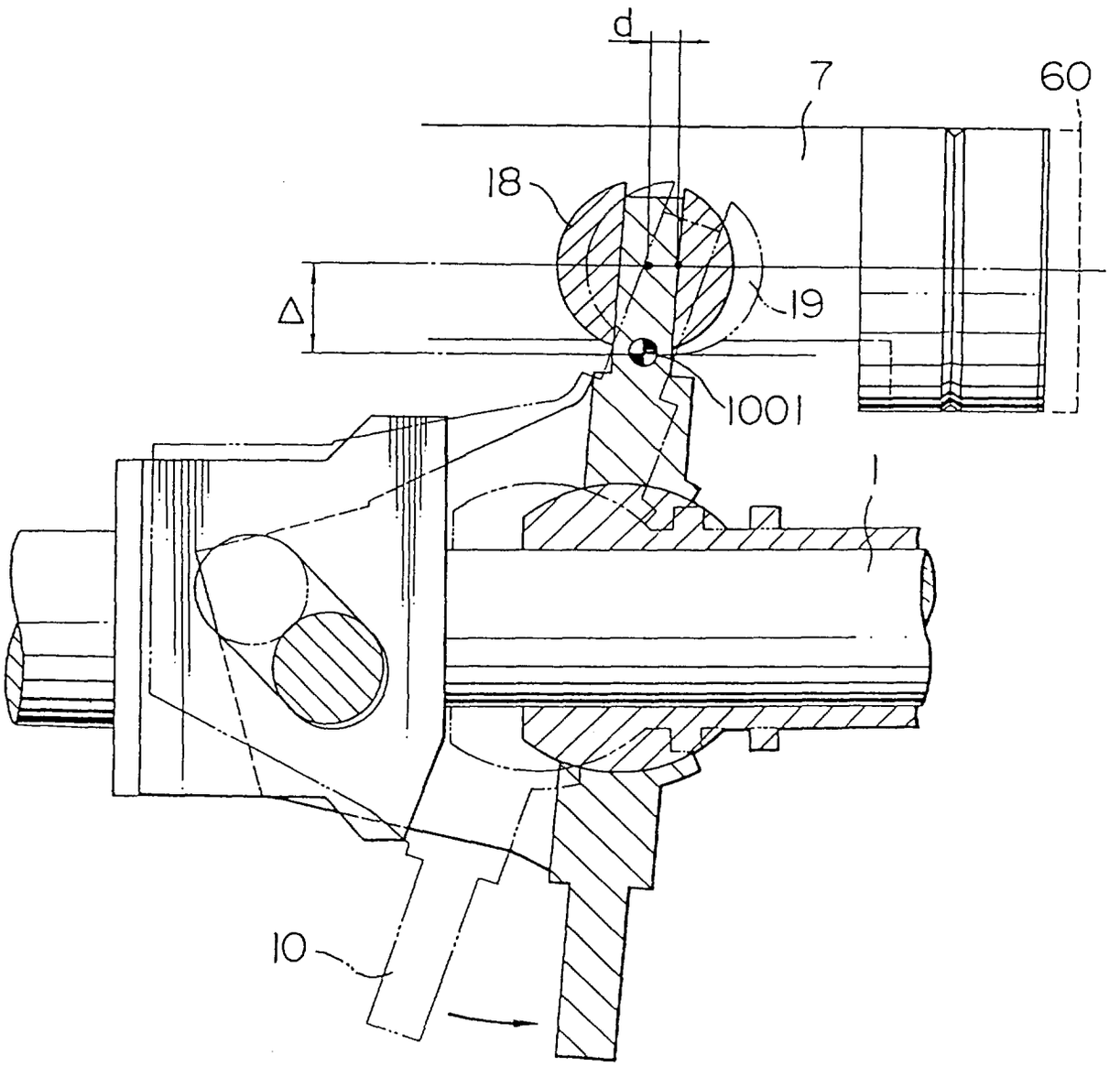


FIG. 12

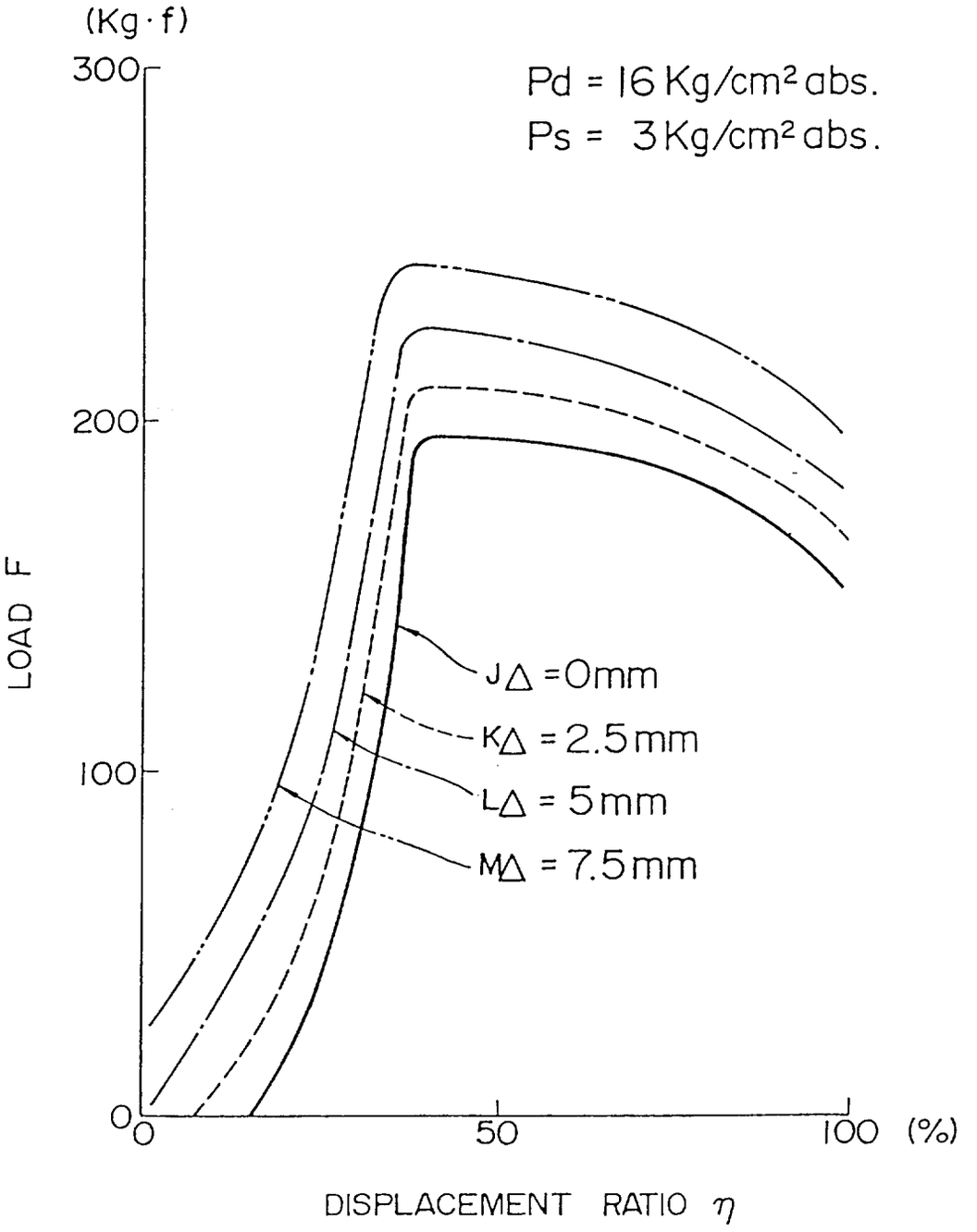

(Kg·f)
300
200
100
0
LOAD F
Pd = 16 Kg/cm² abs.
Ps = 3 Kg/cm² abs.
JΔ = 0mm
KΔ = 2.5mm
LΔ = 5mm
MΔ = 7.5mm
0
50
100 (%)
DISPLACEMENT RATIO η

FIG.13

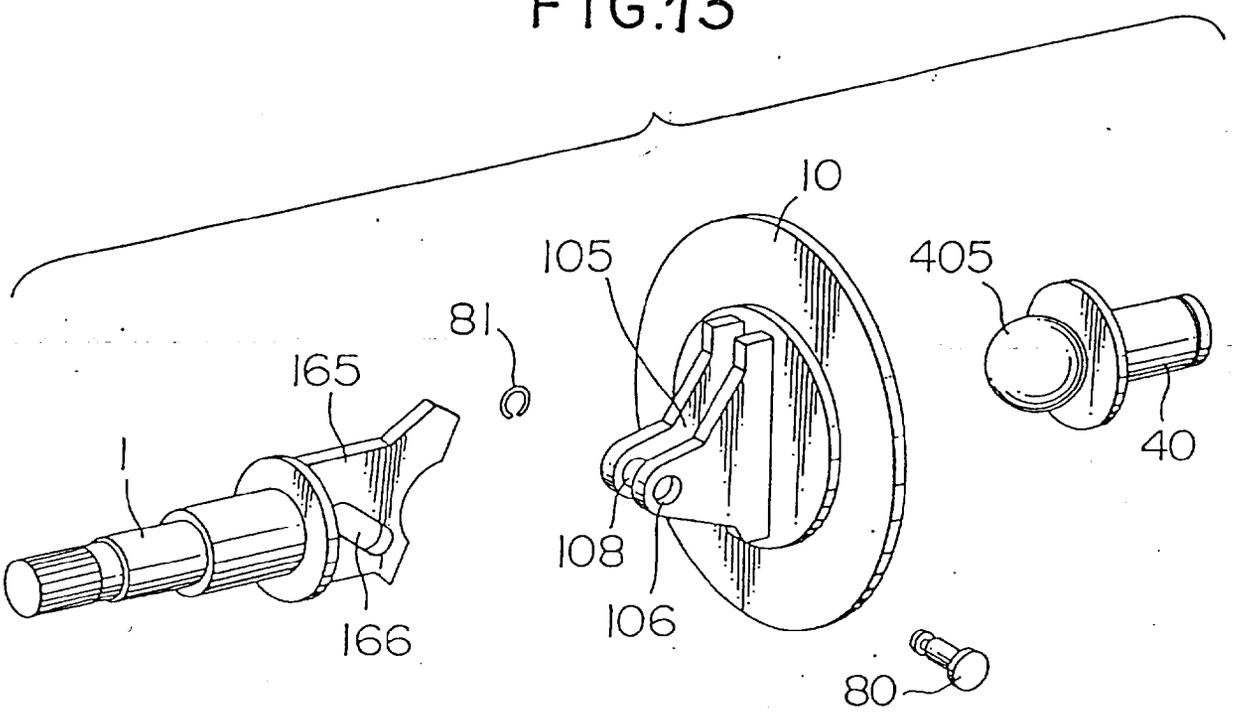


FIG. 14

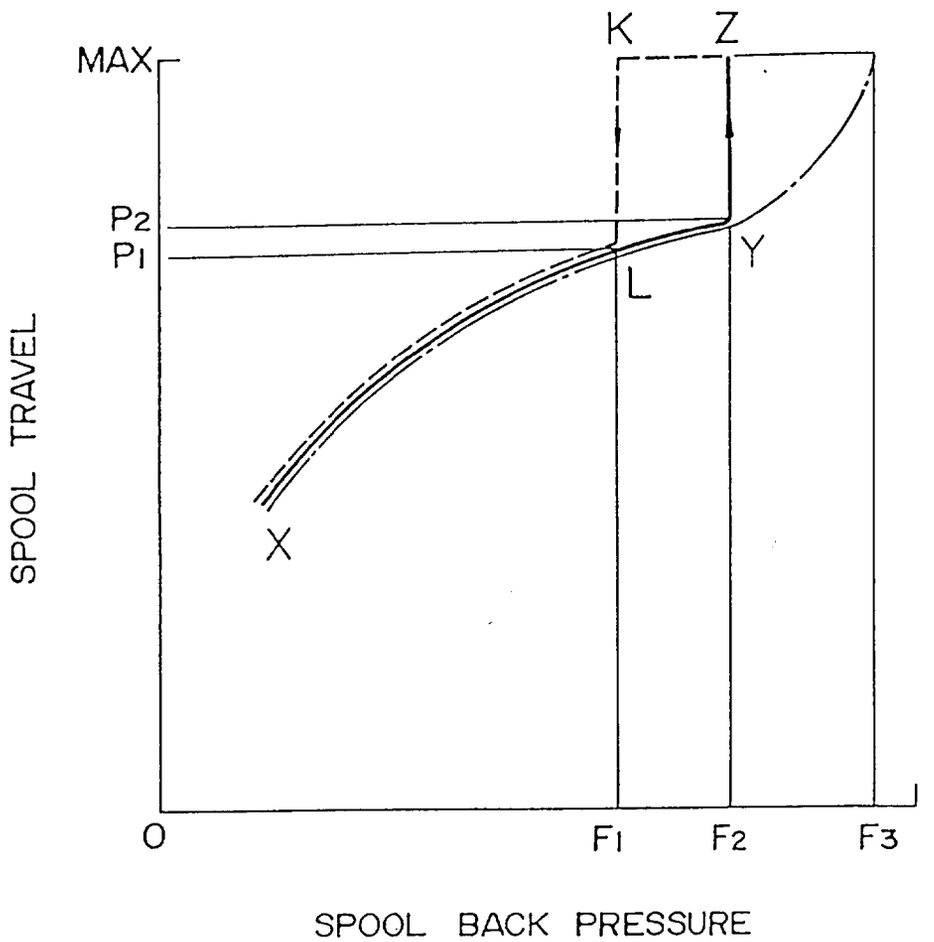


FIG. 15

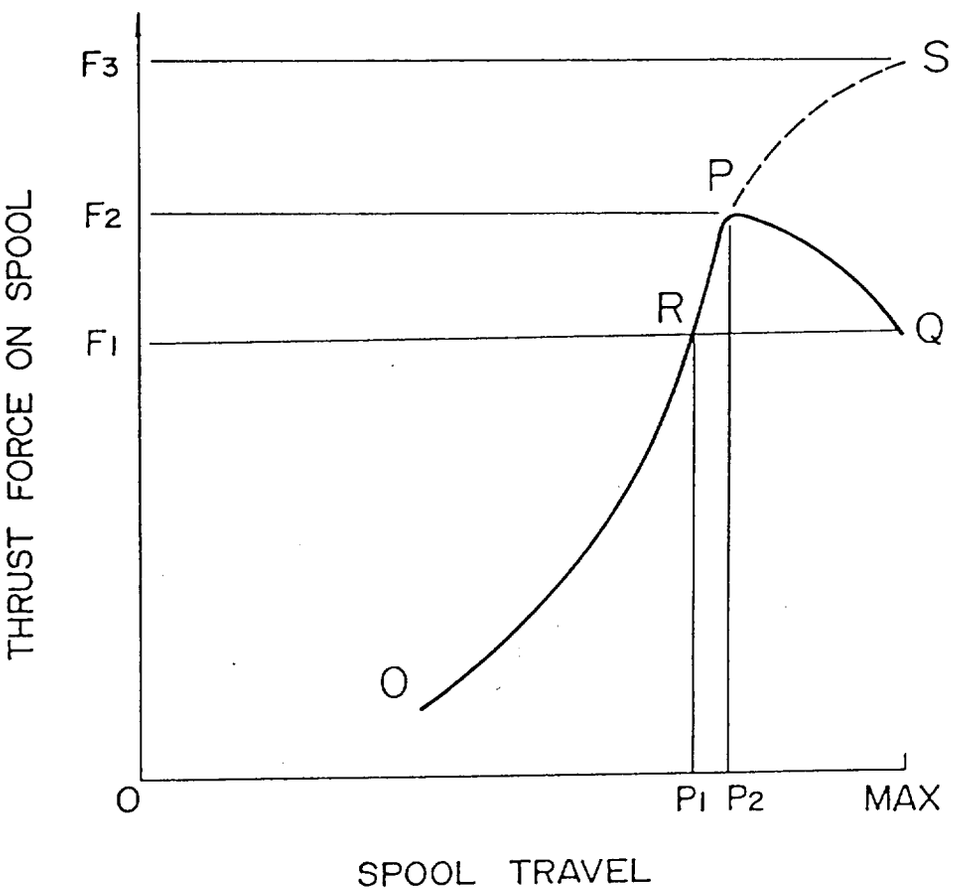


FIG. 16

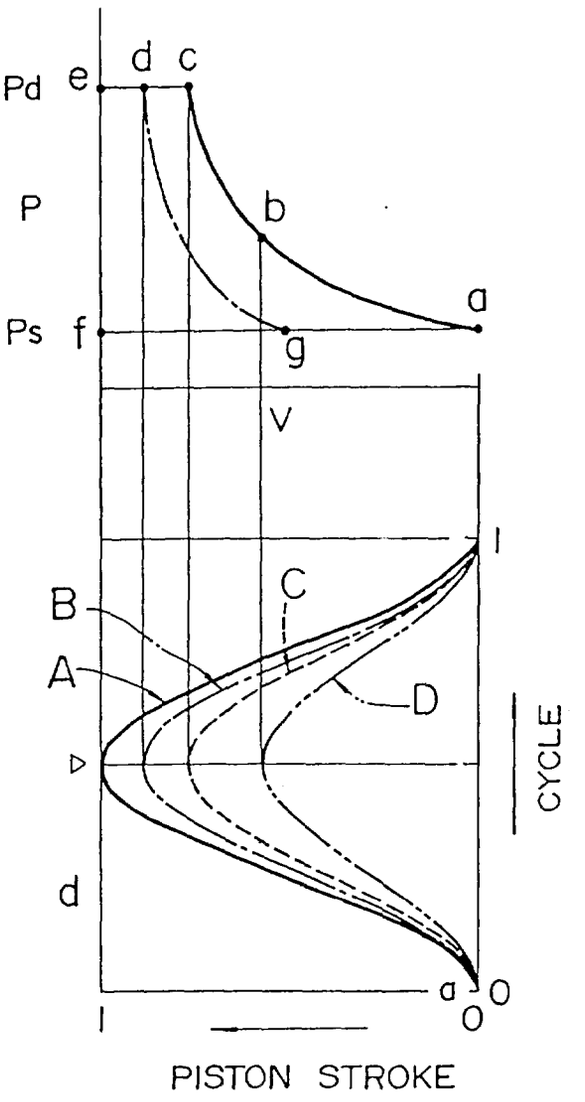


FIG. 17

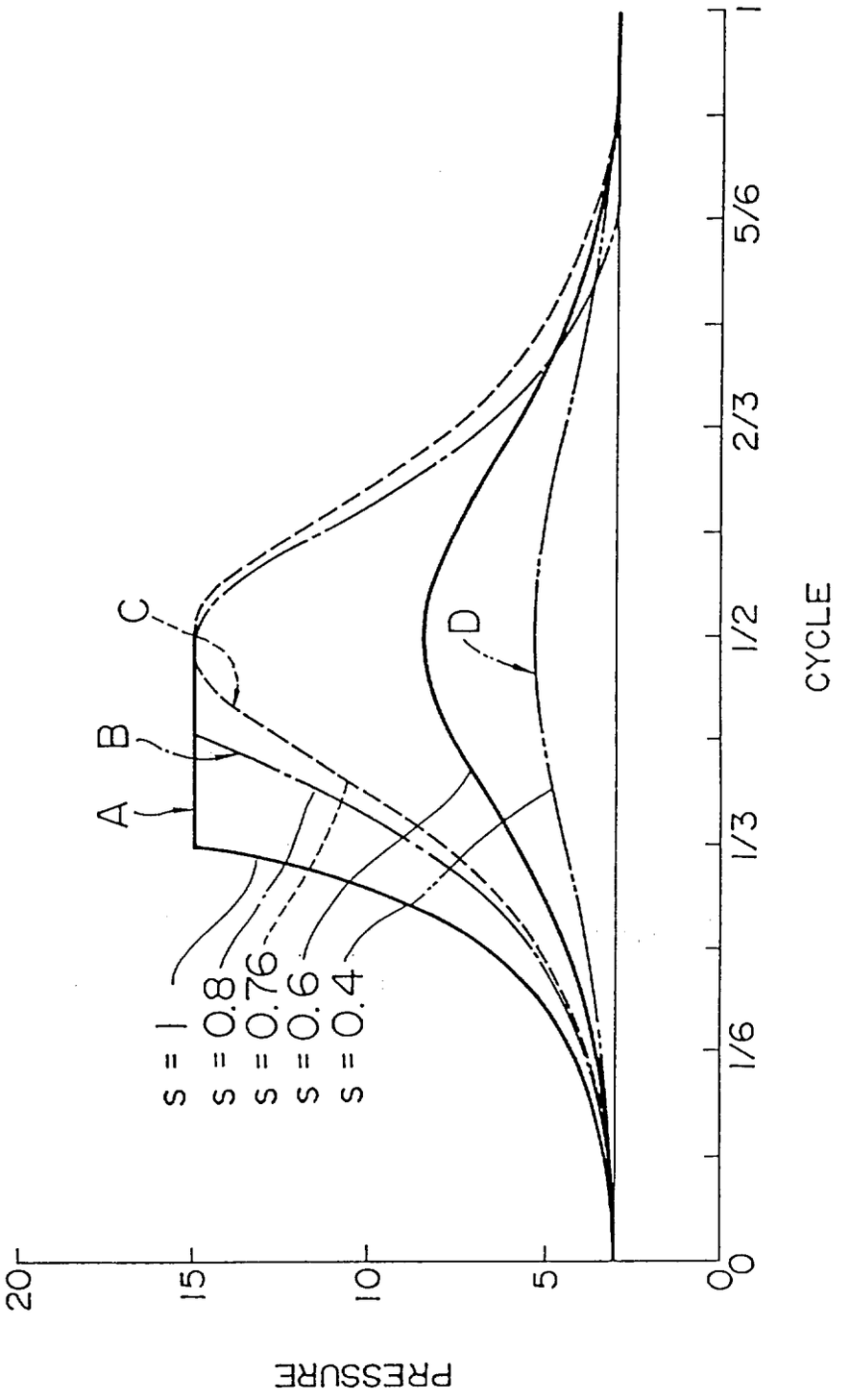

A
B
C
D
s = 1
s = 0.8
s = 0.76
s = 0.6
s = 0.4
20
15
10
5
0
PRESSURE
0
1/6
1/3
1/2
2/3
5/6
1
CYCLE

FIG. 18

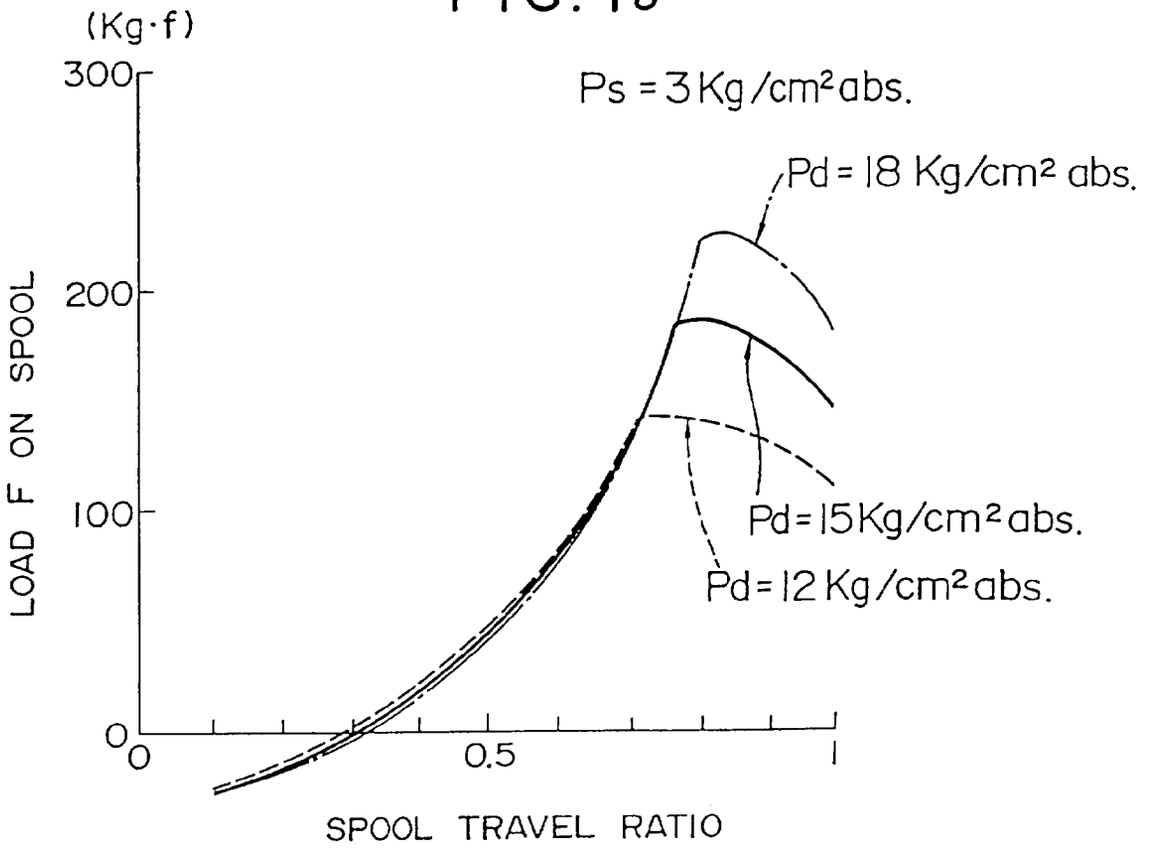

(Kg·f)
300
200
100
0
LOAD F ON SPOOL
Ps = 3 Kg/cm² abs.
Pd = 18 Kg/cm² abs.
Pd = 15 Kg/cm² abs.
Pd = 12 Kg/cm² abs.
0
0.5
1
SPOOL TRAVEL RATIO

FIG. 19

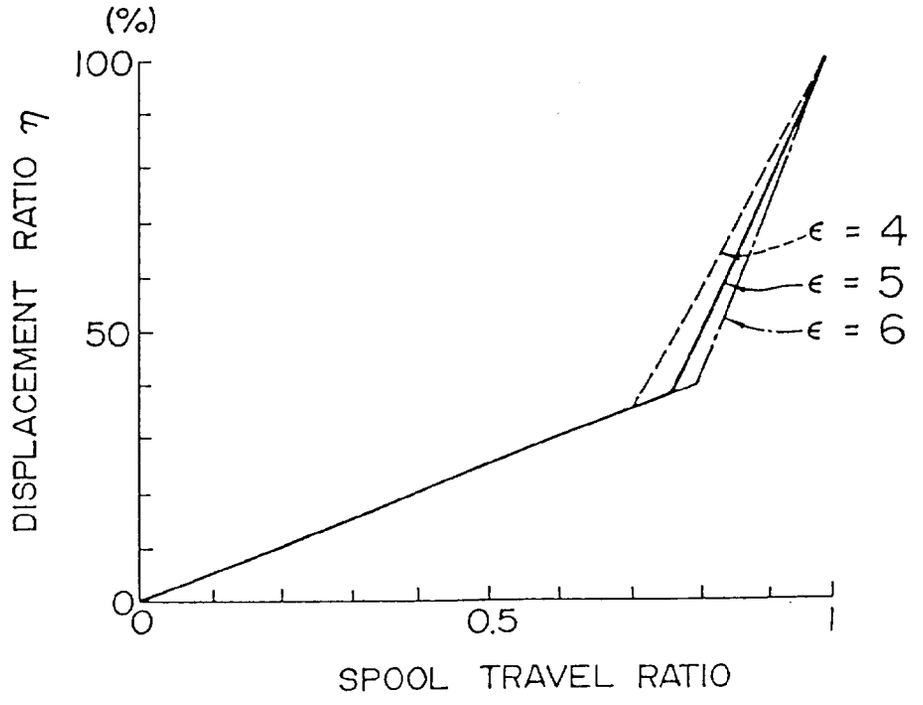

(%)
100
50
0
DISPLACEMENT RATIO η
ε = 4
ε = 5
ε = 6
0
0.5
1
SPOOL TRAVEL RATIO

FIG. 20

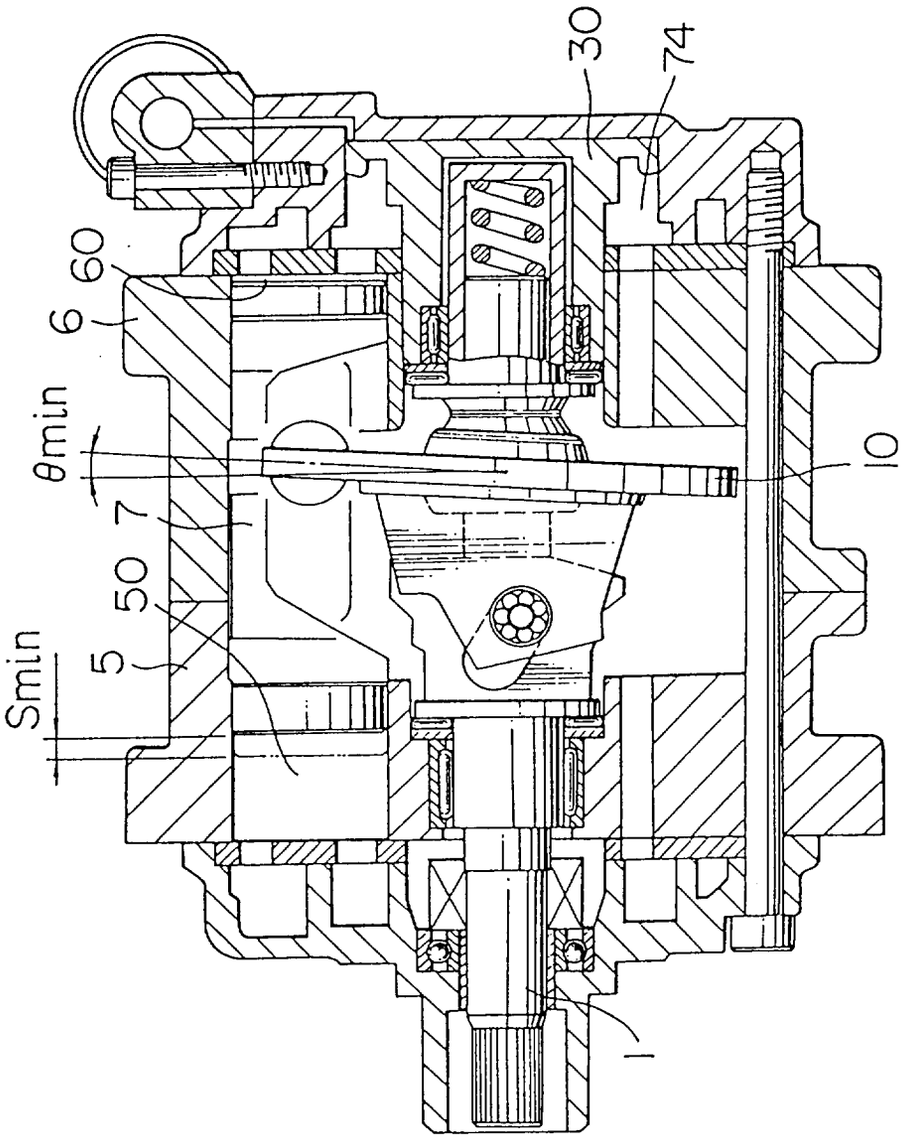


FIG. 21

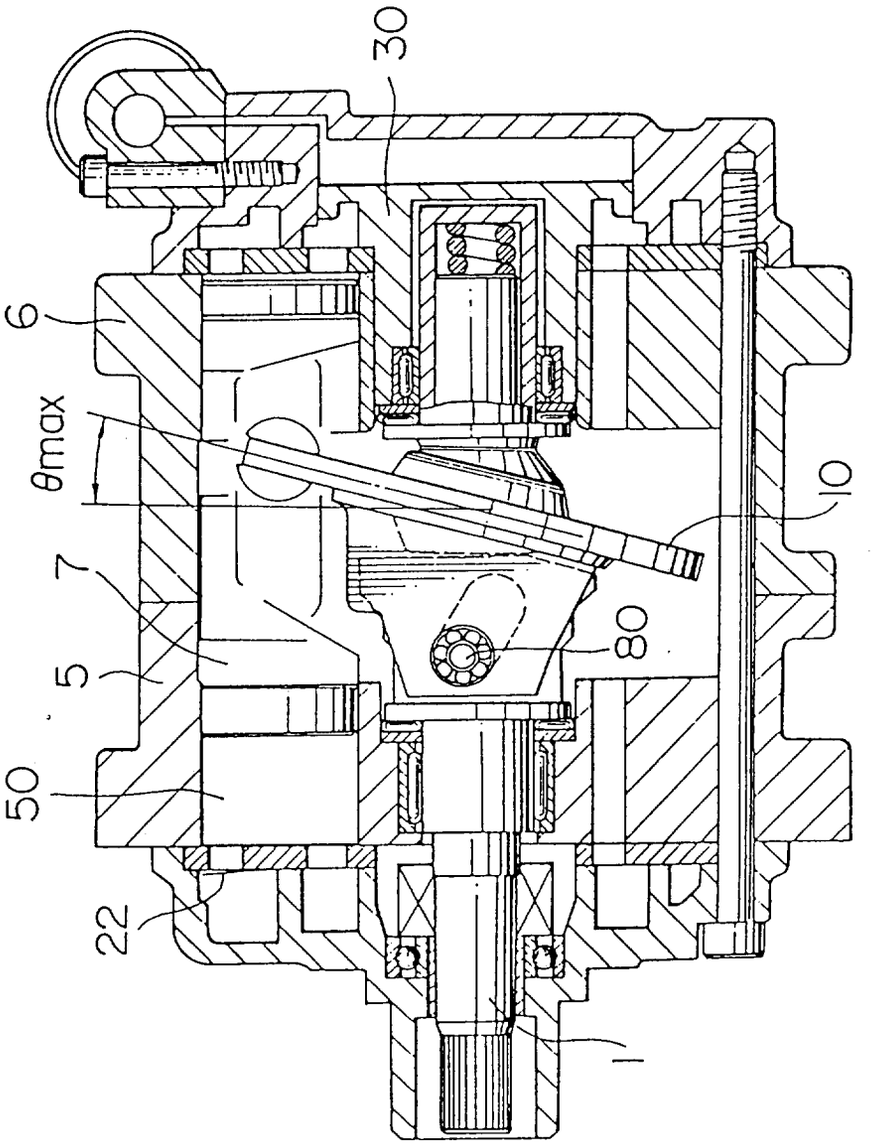


FIG. 22

