(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 1 148 239 A2

(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:
24.10.2001 Bulletin 2001/43

(51) Int Cl.^{7}: **F04B 27/10**

(21) Application number: **01109145.1**

(22) Date of filing: **12.04.2001**

(84) Designated Contracting States:
AT BE CH CY DE DK ES FI FR GB GR IE IT LI LU MC NL PT SE TR
Designated Extension States:
AL LT LV MK RO SI

(30) Priority: **17.04.2000 JP 2000114806**

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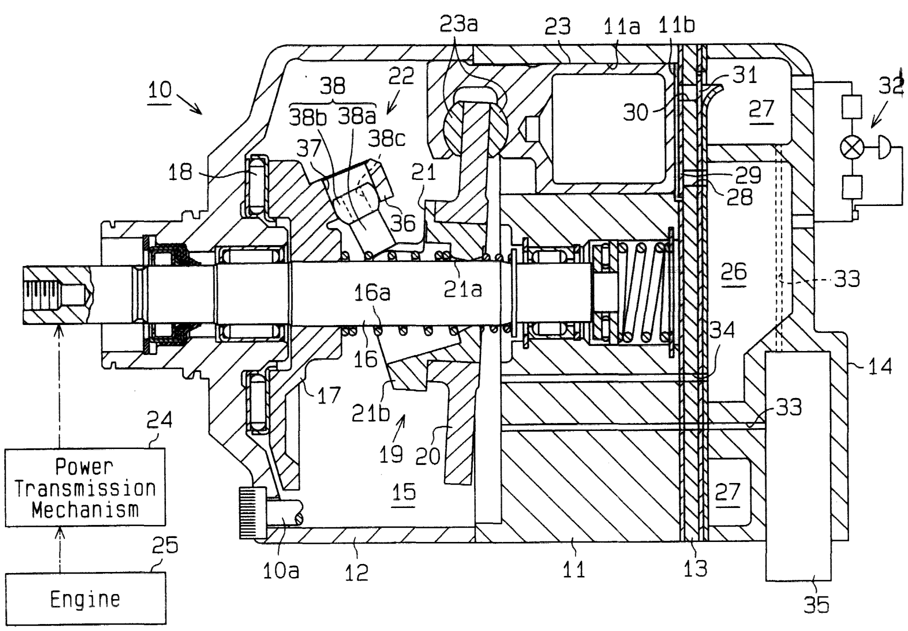
(54) **Hinge mechanism for variable displacement compressor**

(57) By changing the inclination of a cam plate (19), the stroke of a piston (23) is changed to change the discharge displacement a of compressor. A hinge mechanism (22) is positioned between a rotating support (17) and the cam plate (19). The hinge mechanism includes a guide pin (38), and the guide pin transfers the rotational motion of the rotating support (17) to the cam plate (19) and permits inclination of the cam plate. At least a part of the guide pin (38) is hollow.

Fig.1



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Description

[0001] The present invention relates to a hinge mechanism in a variable displacement compressor suitable for-vehicle air-conditioners and a guide pin used in such a hinge mechanism.

[0002] A typical variable displacement compressor includes a housing, which includes a cylinder block, and a drive shaft, which is supported by the housing. In the cylinder block, a plurality of bores are formed to surround the drive shaft, and a piston is located in each bore. A swash plate is driven by on the drive shaft through a special hinge mechanism such that the swash plate rotates integrally with the drive shaft and inclines with respect to the drive shaft. When the inclination of the swash plate changes, the swash plate slides along the surface of the drive shaft in the axial direction.

[0003] Each piston is connected to the outer periphery of the swash plate. Rotation of the drive shaft is converted to reciprocating movement of the pistons, and suction and compression are performed in each of the cylinder bores. By controlling the pressure in a crank chamber, in which the swash plate is located, the inclination angle of the swash plate is controlled, and the stroke of the pistons and the discharge displacement are changed accordingly.

[0004] Fig. 9 shows one example of a hinge mechanism between a swash plate and a drive shaft. The mechanism of Fig. 9 is from Japanese Unexamined Patent Publication No. Hei 11-93833. To a drive shaft 61 is fixed to a rotor 63 in a crank chamber 62, and a pair of support arms 64 projects from the rotor 63. Guide holes 65 are formed in the support arms 64, respectively.

[0005] In the crank chamber 62, a swash plate 66 is supported by the drive shaft 61. To limit the weight of the swash plate 66 and to prevent burning of a shoe 69, the swash plate 66 includes an aluminum based swash plate body 67 and an iron based swash plate guide 68. The swash plate body 67 is press fitted into the swash plate guide 68. The swash plate body 67 is connected to the pistons 70 via the shoes 69, which slide on the periphery of the swash plate body 67.

[0006] A pair of guide pins 71 extend from the swash plate 68. Each guide pins 71 includes a spherical portion 71a, which is received by one of the guide holes 65. The support arms 64, the guide holes 65 and the guide pins 71 form a hinge mechanism. When the pressure in the crank chamber 62 is changed, the inclination angle of the swash plate 66 is changed so that the stroke of the piston 70 is changed while the top dead center position of each piston 23 does not substantially change.

[0007] Essentially, two types of moments, that is, a moment due to centrifugal force and a moment generated based on mutual relationships between the internal pressures of the cylinder bores and the pressure (crank pressure Pc) in the crank chamber 62, act on the swash plate 66, and the inclination angle of the swash plate 66 is determined based on the balance of the moments. The mass of the guide pin 71, which form the hinge mechanism, influences the moment due to centrifugal force and acts to increase the inclination angle of the swash plate 66.

[0008] In consideration of the centrifugal force moment, an xyz coordinate system is used in Fig. 9. A vibration axis of the swash plate 66 is represented by z, and the axis of the drive shaft 61, which is perpendicular to the vibration axis z, is represented by y. An axis that is perpendicular to both the y and z axes is represented by x. The point of intersection of the axes is shown by an origin O. In such a right-angled coordinate system (x, y, z), the centrifugal force moment is obtained by multiplying the product of inertia Ixy of the swash plate 66 with respect to the xz plane and the yz plane by the square of the angular velocity with respect to the drive shaft 61 (refer to USP 5,573,379, which is incorporated herein by reference). Here the product of inertia Ixy is expressed by $Ixy = \int xy \, dm$. The dm represents mass of a minute component which forms the swash plate.

[0009] Therefore, the larger the mass of the guide pin 71 is, the greater the influence on the moment during high velocity rotation is. Thus, to decrease the inclination angle of the swash plate 66 during high speed velocity, a high crank pressure Pc is necessary. As a result, hunting may occur in the control of the crank pressure, and wear of a sealing member that seals the drive shaft 61 is more likely to occur. Further, in a clutchless type compressor, the power consumption during minimum displacement operation is increased.

[0010] When the swash plate 67 is made of an aluminum-based metal and is fitted into an iron based guide 68, the distance from the plane of the swash plate body 67 to the spherical portion 71a of the guide pin 71 in the axial direction increases by about 20% compared to a compressor where the entire swash plate 66 is formed of an iron-based metal, to ensure the press-fit strength of the swash plate 66. As a result, the influence of the moment increases.

[0011] The present invention has been made in consideration of the above-described problems. An object of the present invention is to provide a hinge mechanism of a variable displacement compressor capable of reducing pressure in a crank chamber which is necessary for changing displacement at a high speed rotation, capable of suppressing the occurrence of hunting and capable of reducing the power dissipation in a clutchless type compressor, and a guide pin suitable for the hinge mechanism.

[0012] To achieve the foregoing and other objectives and in accordance with the purpose of the present invention, a variable displacement compressor is provided. The compressor includes a housing including a cylinder bore, a piston accommodated in the cylinder bore, a drive shaft supported by the housing, a rotating support integrally fixed to the drive shaft, a cam plate and a hinge mechanism. The cam plate is connected to the piston for converting rotational motion of the drive shaft to reciprocation of the piston. The cam plate inclines with respect to the drive shaft. The stroke of the piston changes to vary the discharge displacement of the compressor when the inclination of the cam plate changes. The hinge mechanism is positioned between the rotating support and the cam plate. The hinge mechanism includes a guide pin for transferring rotation of the rotating support to the cam plate and for permitting the inclination of the cam plate, wherein a part of the guide pin is hollow.

[0013] Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

[0014] The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a cross-sectional view of a first embodiment of the present invention;
Figs. 2(a), 2(b), and 2(c) are cross-sectional views of various guide pins;
Figs. 3(a) and 3(b) are schematic cross-sectional views illustrating swash plates;
Fig. 4 is a graph showing a relationship between an inclination angle of the swash plate and moment;
Fig. 5 is a partial cross-sectional view showing a hinge mechanism of a second embodiment;
Fig. 6 is a partial plan view of the hinge mechanism of Fig. 5;
Figs. 7(a) and 7(b) are cross-sectional views showing other embodiments of guide pins used in the compressor of the first embodiment, and Fig. 7(c) is a perspective view of another embodiment of a guide pin;
Fig. 8(a) is a cross-sectional view showing another embodiment of a guide pin used in a compressor of the second embodiment, and Fig. 8(b) is a perspective view showing a guide pin of still another embodiment; and
Fig. 9 is a partial cross-sectional view of a prior art variable displacement compressor.

[0015] A variable displacement compressor of a vehicle air-conditioner of a first embodiment of the invention will be described with reference to Figs. 1, 2, and 3.

[0016] As shown in Fig. 1, a compressor 10 includes a cylinder block 11, a front housing member 12, which is fixed to the front end of the cylinder block 11, and a rear housing member 14, which is fixed to the rear end of the cylinder block 11 through a valve plate 13. The housing members 12, 14, the valve plate 13 and the cylinder block 11 are secured to each other with a plurality of bolts 10a (only one shown). A crank chamber 15 is defined between the cylinder block 11 and the front housing member 12.

[0017] A drive shaft 16 is supported in the cylinder block 11 and the front housing member 12 with bearings. A lug plate 17 is fixed to the drive shaft 16 in the crank chamber 15. The lug plate 17 transmits thrust to an internal wall surface of the front housing member 12 through a thrust bearing 18.

[0018] A swash plate 19, or cam plate, includes an aluminum-based swash plate body 20 and an iron-based swash plate guide 21, and the swash plate body 20 is press fitted into the swash plate guide 21. Thus, burning between the swash plate 19 and an iron-based shoe 23a is inhibited. Also, the weight of the swash plate 19 is limited. The swash plate guide 21 is provided with a through hole 21a, and the drive shaft 16 passes through the through hole 21a. A hinge mechanism 22 is located between the lug plate 17 and the swash plate 19. Therefore, the swash plate 19 rotates in synchronization with the lug plate 17 and the drive shaft 16, and the swash plate 19 can incline with respect to the drive shaft 16 while sliding on the drive shaft 16 in the axial direction.

[0019] The swash plate 21 includes a counterweight 21b at a location that is opposite to the hinge mechanism 22 with respect to the drive shaft 16. Between the lug plate 17 and the swash plate guide 21 a spring 16a is fitted around the drive shaft 16. The spring 16a urges the swash plate 19 toward the cylinder block 11, that is, in the direction in which the inclination angle decreases.

[0020] A plurality of cylinder bores 11a (only one shown in Fig. 1) is provided in the cylinder block 11 such that the bores 11a are positioned at equal angular intervals. A single-headed piston 23 is fitted in each of the cylinder bores 11a. The front openings of the cylinder bores 11a are closed by the valve plate 13, and a compressor chamber 11b is defined in each cylinder bore 11a. The volume of the compressor 11b varies depending on the position of the corresponding piston 23. Each piston 23 is connected to the periphery of the swash plate 19 through a pair of the shoes 23a. Accordingly, the rotational motion of the swash plate 19, which is produced by the rotation of the drive shaft 16, is converted to reciprocating motion of the pistons 23.

[0021] The drive shaft 16 is connected to an engine 25 through a power transmission mechanism 24. The power transmission mechanism 24 may be a clutch mechanism (for example, an electromagnetic clutch), which connects and disconnects power transmission and is externally controlled. Alternatively, the power transmission mechanism 24 may be a clutchless (for example, the combination of a belt and a pulley). The present embodiment has a clutchless type power transmission mechanism 24.

[0022] A suction chamber 26 and a substantially annular discharge chamber 27, which surrounds the suction chamber 26, are defined in the rear housing member 14. In the valve plate 13, a suction port 28, a suction valve 29, which opens and closes the suction port 28, a discharge port 30, and a discharge valve 31, which opens and closes the discharge port 30, are formed in correspondence with each cylinder bore 11a. The suction chamber 26 and the discharge chamber 27 are connected to each other by an external refrigerant circuit 32.

[0023] The cylinder block 11, the valve plate 13 and the rear housing member 14 are provided with an air-supply passage 33, which connects the crank chamber 15 and the discharge chamber 27, and a bleed passage 34, which connects the crank chamber 15 and the suction chamber 26. A control valve 35 is installed on the way of the supply passage 33. The control valve 35 is similar to the control valve disclosed in Japanese Unexamined Patent Publication No. Hei 9-268973, and includes a bellows, which moves in response to changes in the suction pressure, a solenoid, which produces electromagnetic force, and a valve mechanism, which controls the degree of opening of the supply passage 33 according to the displacement of the bellows and the electromagnetic force of the solenoid. The parts of the control valve 35 are not shown.

[0024] When the pressure in the suction chamber 26 is a predetermined value or less, the bellows are displaced so that the supply passage 33 is opened, and when the pressure in the suction chamber 26 is a predetermined value or more, the supply passage 33 is held in a closed state. The discharge displacement of the compressor is adjusted by controlling the pressure (crank pressure) Pc in the crank chamber with the control valve 35. That is, when the pressure in the suction chamber 26 is relatively low, the degree of opening of the control valve 35 is increased so that the crank pressure Pc is increased. Accordingly, the inclination angle (angle formed by a plane perpendicular to the drive shaft 16 and the swash plate 19) of the swash plate 19 is decreased so that the stroke of each piston 23 is decreased and the discharge displacement is reduced. On the other hand, when the pressure in the suction chamber 26 is relatively high, the degree of opening of the control valve 35 is decreased so that the crank pressure Pc is lowered. Accordingly, the inclination angle of the swash plate 19 is increased so that the stroke of each piston 23 is increased and the discharge displacement is increased.

[0025] Next, the hinge mechanism 22 will be described in further detail. The hinge mechanism 22 has two supporting arms 36 (only one is shown), which extend from a rear surface of the lug plate 17, a guide hole 37 formed in each of the supporting arms 36, and two guide pins 38, which are fixed to the swash plate 19. Each guide hole 37 is cylindrical. The guide pins 38 are parallel, and an imaginary plane that includes the axis of the drive shaft 16 lies between the pins 38. One pin 38 corresponds to each supporting arm 36. The guide pins 38 are identical in shape and size, and are symmetrical with respect to the previously mentioned imaginary plane.

[0026] Each of the guide pins 38 includes a shaft portion 38a attached to the swash plate 19 and a spherical portion 38b formed at the distal end of the shaft portion 38a. The spherical portions 38b engage the guide holes 37. Each spherical portion 38b has a larger outer diameter than the shaft portion 38a, and the distal end of each spherical portion 38b is truncated along a plane. At least a portion of each spherical portion 38b is hollow. Each guide pin 38 is forged by use of, for example, a header or a former.

[0027] A hollow chamber 38c in each guide pin 38 is opened at the distal end of the spherical portion 38b. The shape of the hollow chamber 38c can be appropriately selected. For example, any one of many shapes such as a first shape, in which the hollow chamber 38c extends to approximately the center of the spherical portion 38b as shown in Fig. 2(a), a second shape, in which the hollow chamber 38c extends to the shaft portion 38a as shown in Fig. 2(b), and a third shape, in which the hollow chamber 38c extends to the vicinity of the location where the shaft portion 38a joins with the swash plate guide 21 as shown in Fig. 2(c), can be selected. The masses of the guide pins 38 shown in Figs. 2(a), 2(b) and 2(c) decreases as indicated by the following inequality: Fig. 2(a) > Fig. 2(b) > Fig. 2(c).

[0028] The shape of the swash plate 19 is like that of the swash plate disclosed in Japanese Unexamined Patent Publication, No. Hei 7-293429 (USP 5,573,379). Figs. 3(a) and 3(b) employ an xyz coordinate system. Additionally, a vibration axis of the swash plate 19 is represented by S. The axis of the drive shaft 16, is represented by y. A z-axis is perpendicular to the plane of the sheet of Fig. 3(a) and is parallel to the vibration axis S. An x-axis is perpendicular to both the y and z axes. A point of intersection the x, y and z axes is defined as an origin point ○. When the inclination angle of the swash plate 19 starts rotation at an angular portion of zero degrees as shown in Fig. 4, the product of inertia Ixy of the swash plate 19 with respect to the xy plane and the yz plane generates moment M in a direction such that the displacement is increased (the direction in which the inclination angle of the swash plate 19 is increased).

[0029] The operation of the compressor is as follows.

[0030] With rotation of the drive shaft 16, the swash plate 19 rotates, and the rotation of the swash plate 19 is converted to reciprocating motion of each piston 23 through the shoes 23a. As a result, in the compressor chamber 11b, suction, compression and discharge of refrigerant are sequentially repeated. Refrigerant supplied from the external refrigerant circuit 32 to the suction chamber 26 is drawn into the compressor 11b through the suction port 28. After the refrigerant is compressed, it is discharged to the discharge chamber 27 through the discharge port 30. The refrigerant discharged to the discharge chamber 27 enters the external refrigerant circuit 32 via the discharge hole.

[0031] The degree of opening of the control valve 35 is adjusted according to the cooling load. When the cooling load is high and the pressure in the suction chamber

26 is high, the opening degree of the control valve 35 is reduced, and the pressure (crank pressure Pc) in the crank chamber 15 is decreased as a result, which increases the inclination angle of the swash plate 19. The stroke of each piston 23 is increased accordingly, and the compressor 10 operates with a large displacement. On the other hand, when the cooling load is low and the pressure in the suction chamber 26 is low, the opening degree of the control valve 35 is increased and the pressure (crank pressure Pc) in the crank chamber 15 is increased so that the inclination angle of the swash plate 19 is decreased. As a result, the stroke of the piston 23 is decreased, and the compressor 10 operates with a small displacement.

[0032] The moment that is generated based on the relationship between the internal pressure of the cylinder bores and the crank pressure Pc, the moment due to centrifugal force, and the force of the spring 16a act on the swash plate 19, and the inclination angle of the swash plate 19 is determined based on the equilibrium of these components.

[0033] The moment that is generated by rotation of the swash plate 19 (the moment due to centrifugal force) is the product of inertia Ixy of the swash plate 19 with respect to the xz plane and the yz plane in the right-angled coordinate system (x, y, z) multiplied by the square of the angular velocity ω of the drive shaft 16.

[0034] As shown in Fig. 3(a), y axis coincides with the axis of the drive shaft 16, the z axis is parallel to the vibration shaft S, and the x axis is perpendicular to the y axis and the z axis. When the upward direction of the x axis is positive and the front direction of the y axis is also positive, the product of inertia Ixy of the swash plate is expressed by $Ixy = \int xy\, dm$. Here, dm is the minute mass of the swash plate 19 including the guide pin. Therefore, even if the outer diameter, number and arrangement of the pistons 23, the outer diameter of the swash plate 19, the outer shape of the guide pin 38, and the rotational speed (angular velocity ω) are constant, the moment due to the rotation varies depending on the distance L between the center of the spherical portion 38b of the guide pin 38 and the xz plane.

[0035] When the swash plate 19 is formed by press fitting the iron based metallic swash plate guide 21 into the aluminum based metallic swash plate body 20, to produce a large press-fit area, the distance L is increased by about 20%, as shown in Fig. 3(a), as compared with the swash plate 19 shown in Fig. 3(b), which is an entirely iron-based swash plate. As a result, even if the shapes of the guide pins 38 are the same, the product of inertia Ixy of the swash plate 19 of Fig. 3(a) is significantly increased compared with the swash plate 19 of Fig. 3(b).

[0036] The moment M that is generated by rotation of the swash plate 19 in the vicinity of the minimum inclination angle acts to increase the inclination angle of the swash plate 19. Thus, when the product of inertia Ixy is large, the influence is greater at a high rotation speeds. Therefore, to reduce the inclination angle of the swash plate 19, high crank pressure Pc is required. Even if the mass of the guide pins 38 is the same as, the product of inertia Ixy of the guide pins 38 becomes large when the mass of the portion spaced apart from the xy plane, the distal end portion of the guide pin 38 is larger.

[0037] On the contrary, in this embodiment, since the hollow chambers 38c are formed so that the masses of the front spherical portions 38b of the guide pins 38 are decreased, even if the mass of the guide pins 38 are the same, the product of inertia Ixy is smaller than that of a swash plate lacking the hollow chambers 38c. Further, the moment M based on the rotation of the swash plate 19 is smaller than the moment Mo of a conventional swash plate, as shown in Fig. 4. Thus, the crank pressure Pc necessary for changing the inclination angle of the swash plate 19 is reduced. When the crank pressure Pc necessary for changing the inclination angle is high, the inclination angle is likely to shift by a slight variation in the compression load and hunting is likely to occur even if adjusted to a predetermined inclination angle. However, when the crank pressure Pc necessary for changing the inclination angle is reduced, hunting is less likely to occur. This can be understood from the fact that the rate of change in the moment M with respect to the inclination angle of the swash plate 19 is smaller than that of the moment Mo of the swash plate provided with conventional guide pins of Fig. 4. Reducing the mass of the guide pins 38 permits the mass of the counterweight 21b to be reduced, and contributes to reducing the rate of change of the rotating moment M of the swash plate 19.

[0038] This embodiment has the following effects.

(1) At least a part of the guide pin 38 is hollow. Thus, the product of inertia of a portion of the guide pin 38 that influences the rotational moment rotation of the drive shaft 16 and the swash plate 19 is decreased. As a result, the crank pressure Pc that is necessary for changing the displacement of a compressor can be reduced at a high rotation speeds, and hunting can be inhibitted. Further, in a clutchless type compressor, even if the vehicle air-conditioner is off, the power of engine is transferred to the compressor. However, at the time the inclination angle of the swash plate 19 approaches zero degree so that power dissipation can be decreased.
Further, when a check valve is provided downstream of the discharge port of the compressor, the valve opening pressure can be reduced and performance is consequently improved.

(2) The spherical portion 38b of the guide pin 38 and the guide hole 37 of the supporting arm 36 form the hinge mechanism 22. Therefore, sliding of the swash plate 19 on the drive shaft 16 is smoothly guided by forming the guide hole 37 in a simple cylindrical shape or the like.

(3) When the guide pins 38 are hollowed out, the mass that has the greatest influence on the product of inertia Ixy is removed. Since the distal end of the pin 38 has a greater effect than the proximal end, the distal end is hollowed out.

(4) The hollow chamber 38c is opened at the distal end. Thus, by changing the depth of the hollow chamber 38c, the product of inertia Ixy can be easily changed. Further, machining of the hollow chamber 38c is relatively simple.

(5) The distal ends of the spherical portions 38b of the guide pins 38 are truncated. Accordingly, the product of inertia Ixy is reduced, as compared with a conventional guide pin having a full spherical end.

(6) The guide pin is formed by forging. Therefore, the guide pin is stronger than a pin in which the hollow chamber 38c of the guide pin 38 is formed by a cutting operation. In addition, if the guide pin is manufactured by a header or a former, productivity is higher.

(7) The swash plate 19 is made of an aluminum based metallic swash plate body 20 and an iron based metallic swash plate guide 21. Therefore, the entire swash plate 19 is lighter than an iron-based swash plate 19.

(8) The swash plate 19 is directly supported on the drive shaft 16 by the wall of the through hole 21a. Thus, since it is not necessary to provide a sleeve on the drive shaft 16 and a pivot shaft that connects the swash plate to the sleeve, the number of parts is low.

A second embodiment of the present invention will be described with reference to Figs. 5 and 6. In the second embodiment, the hinge mechanism 22 is different from that of the first embodiment. Otherwise, the second embodiment is basically the same as the first embodiment. Therefore, parts that are the same are denoted by the same reference numerals, and only the differences will be explained.

As shown in Fig. 5, a sleeve 39 is fitted on the drive shaft 16 and is permitted to slide on the drive shaft 16. A swash plate guide 21 is pivotally supported by a pair of supporting shafts 40 (only one shown) to the sleeve 39. The supporting shafts 40 extend perpendicular to the drive shaft 16.

The hinge mechanism 22 includes two swing arms 41 that extend from the swash plate guide 21 toward the lug plate 17. A supporting arm 42 extends from on the lug plate 17, and a guide pin 43 connects the swing arms 41 to the supporting arm 42. The swing arms 41 surrounds the supporting arm 42 as shown in Fig. 6. A guide hole 44 is formed in the supporting arm 42. Each of the swing arms 41 has a mounting hole 45, the axes of which are parallel to the supporting shaft 40. The guide pin 43 is press fitted into the mounting holes 45 and fitted into the guide hole 44.

The guide hole 44 is elongated so that, even if the inclination angle of the swash plate is changed, the top dead center position, of the pistons 23 do not substantially change. That is, the guide hole 44 extends so that the closer the guide hole 44 is to the swash plate 19, the further it is from the drive shaft 16. The guide pin 43 is a hollow cylinder.

In the hinge mechanism 22 of this embodiment, as the guide pin 43 moves along the guide hole 44, the swash plate 19 is rotated integrally with the lug plate 17 and the sliding and inclining movements of the swash plate 19 on the drive shaft 16 are guided.

Therefore, this embodiment has the following effects in addition to the effects (1) to (7) described in the first embodiment.

(9) The guide pin 43, which is part of the hinge mechanism 22, moves along the guide hole 44, and the sliding motion and inclination of the swash plate 19 are guided. Therefore, the guide pin 43 can have a simple linear shape, which further simplifies manufacturing.

(10) The guide hole 44 is formed in the supporting arm 42, and mounting holes 45 are formed in the swing arms 41. Therefore, the structure of the swash plate 19 is simple as compared to a swash plate where the guide hole 44 is formed in the swing arm 41.

[0039] The second embodiment is not limited to the structure described above, and may be constructed as follows for example.

[0040] In a construction in which the spherical guide pin 38 is used, as in the first embodiment, the hollow chamber 38c of the guide pin 38 may be formed in the shaft 38a of the guide pin 38, as shown in Figs. 7(a) and 7(b). Alternatively, a hollow chamber 38c may be formed in the guide pin 38 and a slit 38d may be formed in the shaft portion 38a, as shown in Fig, 7(c). In this case, the outer diameter tolerance of the shaft portion 38a may be increased.

[0041] In the second embodiment, the guide pin 43 may be formed such that a partition is formed in the hollow portion 43a as shown in Fig. 8(a). Alternatively, the ends of the pin 43 may be formed by solid bodies, as shown in Fig. 8(b), instead of the simple pipe shape of Fig. 5. The force on the guide pin 43 acts strongly on the ends of the guide pin 43. However, when the ends of the guide pin 43 are solid, the strength of the guide pin 43 is improved.

[0042] In the first embodiment, the swash plate 19 may be pivotally connected to a sleeve 39 that is fitted on the drive shaft 16, as in the second embodiment, through the supporting shafts 40. Alternatively, in the second embodiment, the swash plate 19 may be supported on the drive shaft 16 as in the first embodiment. In addition, a spherical sleeve may be fitted on the drive shaft 16, and the swash plate 19 may be pivotally supported on the outer surface of the spherical sleeve.

[0043] The hinge mechanisms of Fig. 1 and 6 have two joints. That is, two pins 38 couple with two holes 37 in Fig. 1, and two arms 41 form joints in Fig. 6. Alternatively, each hinge may have just one joint. However, two sets are preferable from the viewpoints of rotational balance and stability in the driving power transmission.

[0044] The swash plate 19 may be made of only one kind of metal, such as an iron based metal or the like. In this case, a press-fit margin is not needed, and the distance between the guide pin 38 and the xz plane can be decreased. Accordingly, the product of inertia Ixy can be further decreased by forming the guide pin 38 in a hollow shape. When the material of the swash plate 19 is the same as that of the shoe 23a, a surface treatment (for example, aluminum spray coating) for the prevention of burning is applied to the sliding face of the shoe 23a.

[0045] In the hinge mechanism 22 of the second embodiment, the guide hole 44 may be formed in the swing arm 41, and the mounting holes 45 may be formed in the supporting arm 42.

[0046] The guide pin 38 may be manufactured by cutting or casting.

[0047] The present invention may be applied to a wobble type variable displacement compressor.

[0048] It should be apparent to those skilled in the art that the present invention may be embodied in many other specific forms without departing from the spirit or scope of the invention. Particularly, it should be understood that the invention may be embodied in the following forms.

[0049] Therefore, the present examples and embodiments are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope and equivalence of the appended claims.

[0050] By changing the inclination of a cam plate (19), the stroke of a piston (23) is changed to change the discharge displacement a of compressor. A hinge mechanism (22) is positioned between a rotating support (17) and the cam plate (19). The hinge mechanism includes a guide pin (38), and the guide pin transfers the rotational motion of the rotating support (17) to the cam plate (19) and permits inclination of the cam plate. At least a part of the guide pin (38) is hollow.

Claims

1. A variable displacement compressor comprising a housing including a cylinder bore(11a); a piston (23) accommodated in the cylinder bore; a drive shaft (16) supported by the housing; a rotating support (17) integrally fixed to the drive shaft; a cam plate (19) connected to the piston for converting rotational motion of the drive shaft to reciprocation of the piston, wherein the cam plate (19) inclines with respect to the drive shaft, and wherein the stroke of the piston (23) changes to vary the discharge displacement of the compressor when the inclination of the cam plate changes; and a hinge mechanism (22) positioned between the rotating support (17) and the cam plate (19), **characterized by that** the hinge mechanism (22) includes a guide pin (38) for transferring rotation of the rotating support (17) to the cam plate (19) and for permitting the inclination of the cam plate, wherein a part of the guide pin (22) is hollow.

2. The compressor according to claim 1, **characterized by that** the hinge mechanism includes a supporting arm (42) extending from the rotating support toward the cam plate and a guide portion (37) provided in the support arm, the guide pin (38) comprising:

   a shaft portion (38a), which is fixed to the cam plate, and a spherical portion (38b), which has a larger diameter than the shaft portion and is provided on the shaft portion, wherein the spherical portion (38b) engages the guide portion (37), and at least a part of the spherical portion (38b) is hollow.

3. The compressor according to claim 2, **characterized by that** the guide pin (38) has a hollow chamber that is open on the outer periphery of the spherical portion, wherein the hollow chamber extends substantially to the center of the spherical portion.

4. The compressor according to claim 2, **characterized by that** the guide pin (38) has a hollow chamber that is open on the outer periphery of the spherical portion, wherein the hollow chamber extends to the shaft portion.

5. The compressor according to claim 2, **characterized by that** a part of the shaft portion (38a) is embedded in the cam plate, and a part of the shaft portion (38a) is exposed from the cam plate (19), and wherein the guide pin has a hollow chamber that is open on the outer surface of the spherical portion, wherein the hollow chamber extends within the entire exposed portion of the shaft portion.

6. The compressor according to claim 1, **characterized by that** the hinge mechanism comprises:

   a swing arm (41) extending from the cam plate (19) toward the rotating support (17);
   a supporting arm (42) on the rotating support (17);

a guide hole (44) formed in one of the swing arm and the supporting arm; and
a mounting hole (45) formed in the other of the swing arm and the supporting arm, wherein the guide pin is located within the guide hole and the mounting hole.

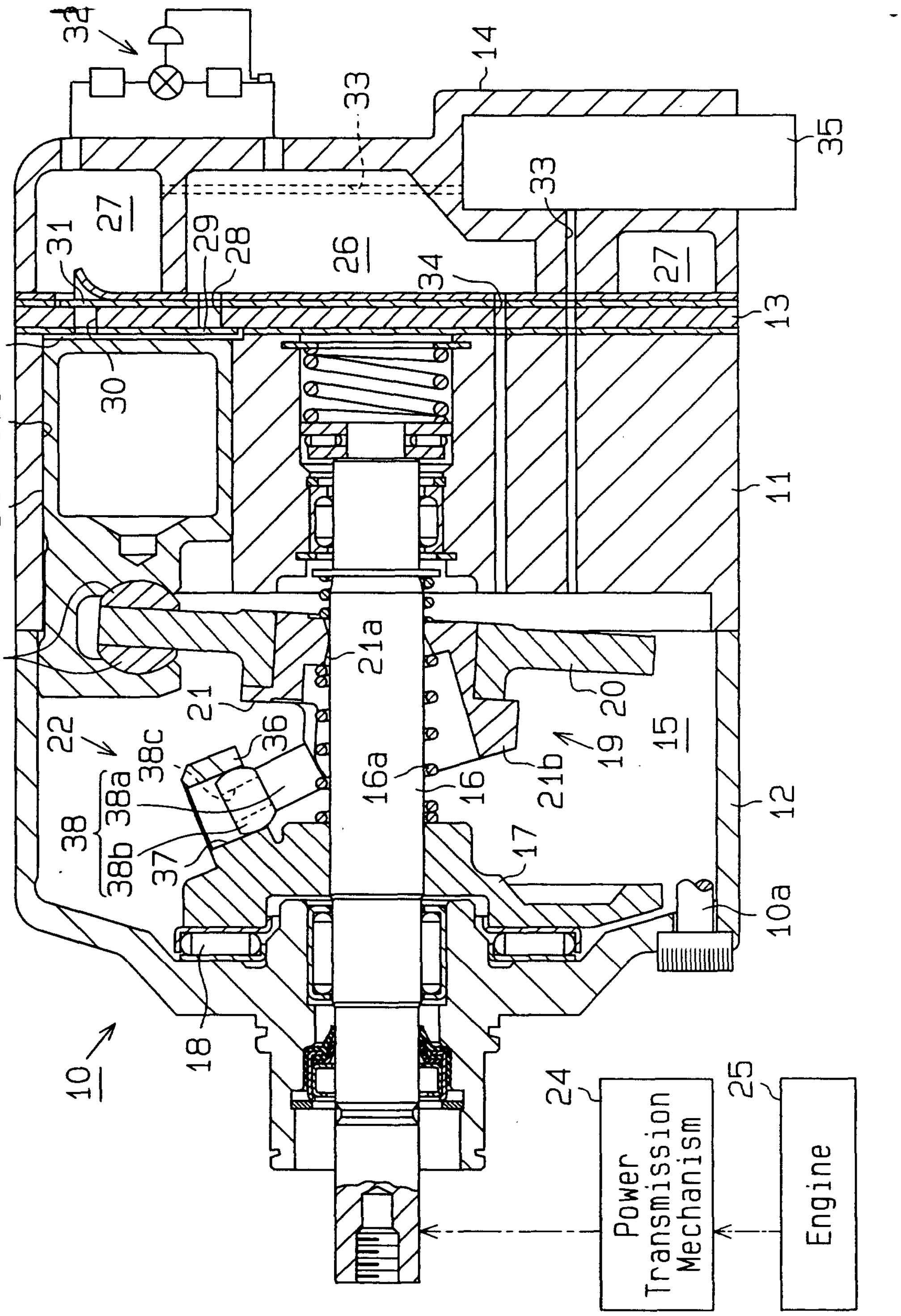
7. The compressor according to claim 6, **characterized by that** the guide hole (44) is formed in the supporting arm (42) and the mounting hole (45) is formed in the swing arm (41).

8. The compressor according to claim 7, **characterized by that** the swing arm is a first swing arm and the mounting hole is a first mounting hole, and the compressor further comprises a second swing arm and a corresponding second mounting hole, the first and second swing arms being arranged on opposite sides of the supporting arm (42), wherein the first mounting hole is coaxial with the second mounting hole, and end portions of the pin, which extend between the swing arms and the supporting arm, are solid.

9. The compressor according to claim 1, **characterized by that** the guide pin (38) is manufactured by forging.

10. The compressor according to claim 2, **characterized by that** the distal end portion of the spherical portion (38b) is truncated.

Fig.1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11
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11b
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13
14
15
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21b
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23a
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34
35
36
37
38
38a
38b
38c
10a
Power Transmission Mechanism
Engine

Fig.2(a)

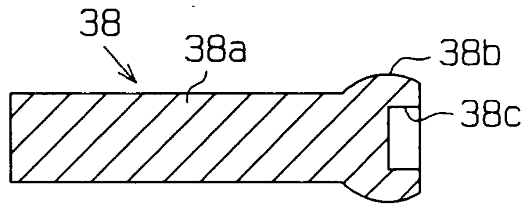

Fig.2(b)

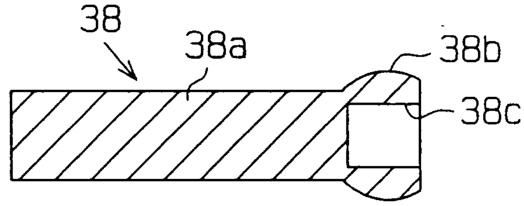

Fig.2(c)

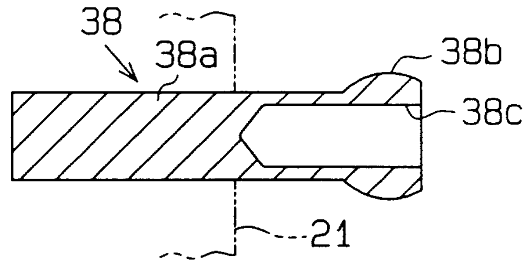

Fig.3(a)

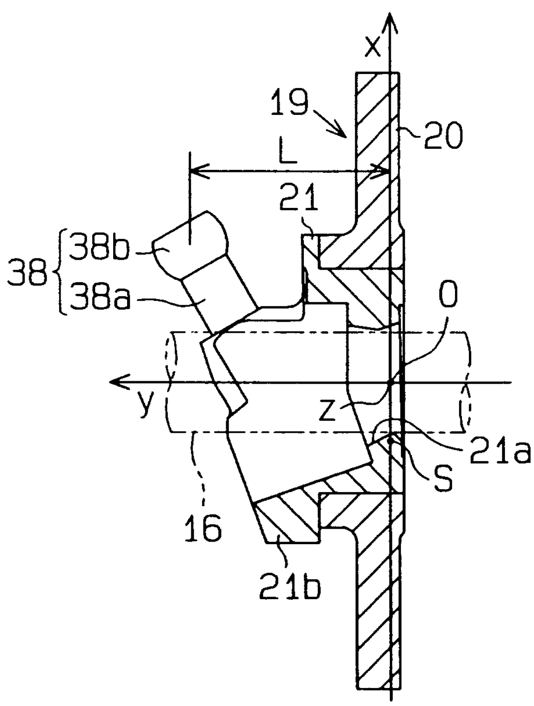

Fig.3(b)

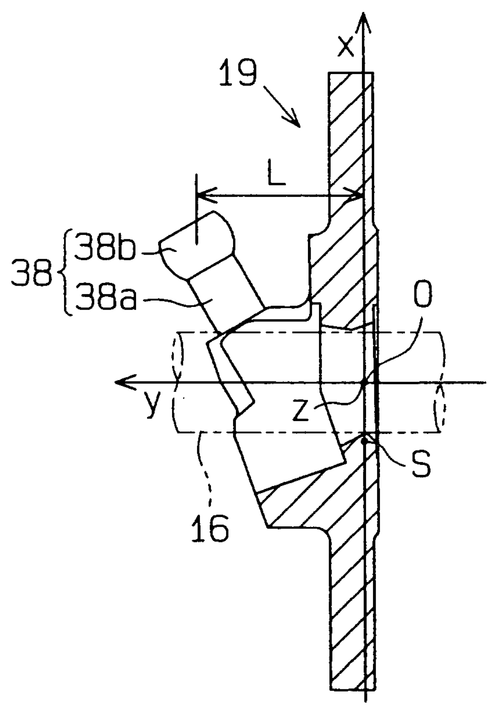

Fig.4

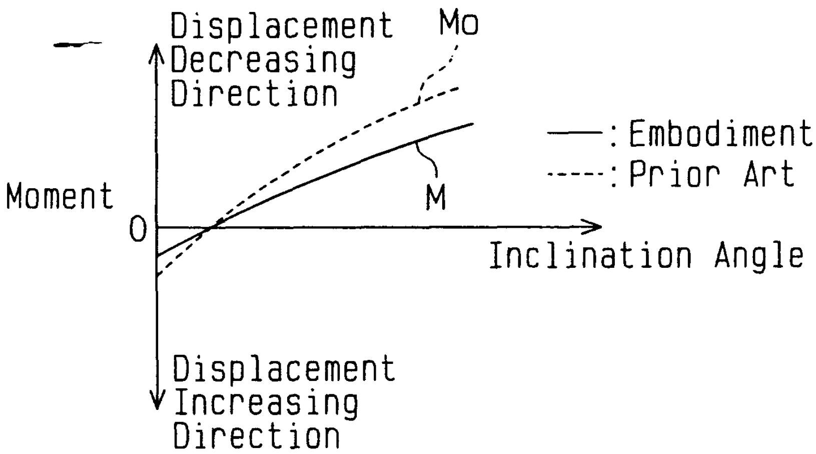
Displacement
Decreasing
Direction
Mo
—: Embodiment
-----: Prior Art
Moment
M
0
Inclination Angle
Displacement
Increasing
Direction

Fig.5

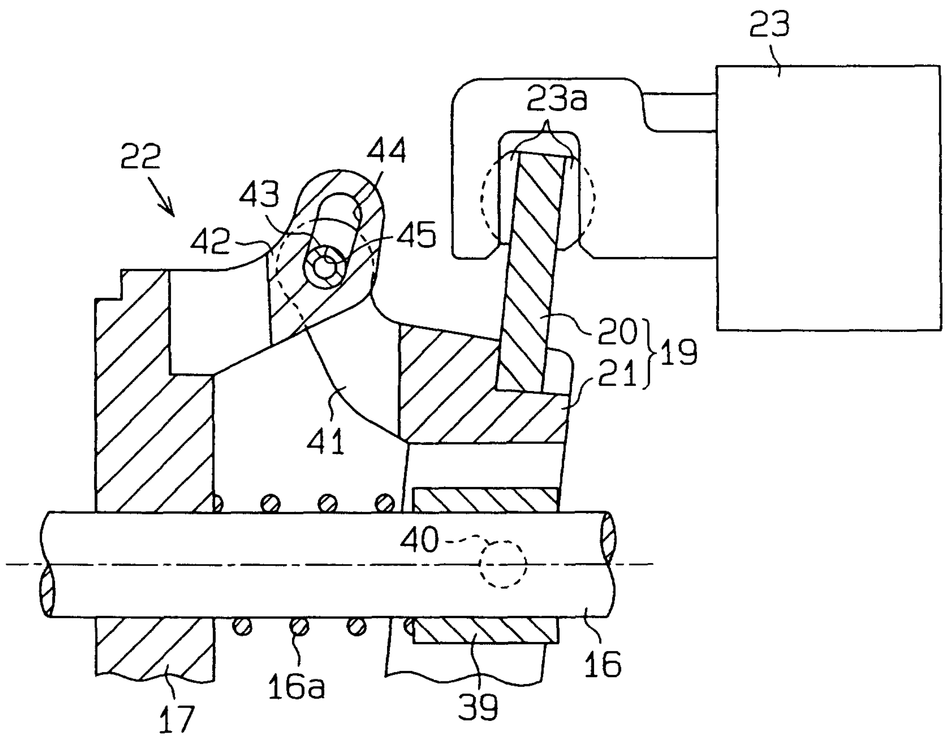
23
23a
22
44
43
42
45
20
19
21
41
40
16
17
16a
39

Fig.6

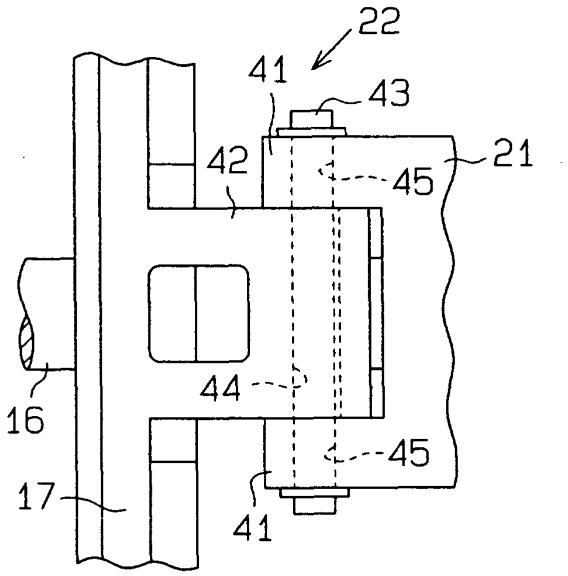

Fig.7(a)

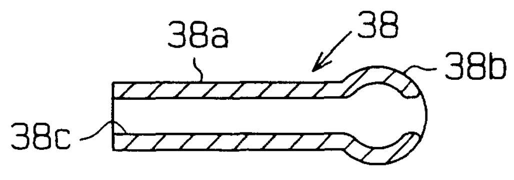

Fig.7(b)

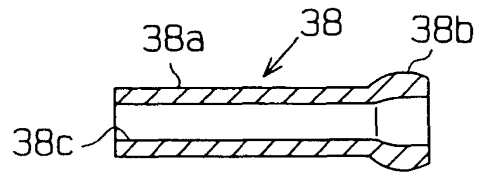

Fig.7(c)

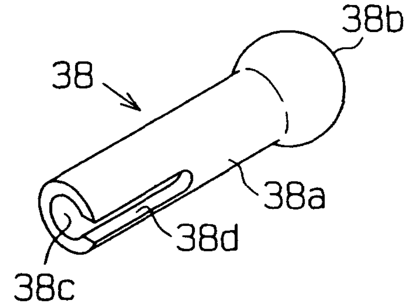

Fig.8(a)

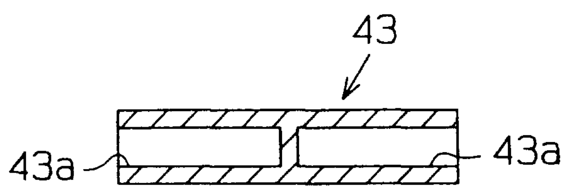

Fig.8(b)

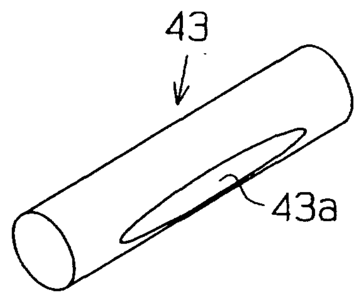

Fig.9