



(12)

EUROPEAN PATENT APPLICATION

(43) Date of publication:
15.12.2004 Bulletin 2004/51

(51) Int Cl.⁷: **F04B 27/14**

(21) Application number: **04013660.8**

(22) Date of filing: **09.06.2004**

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

(30) Priority: **12.06.2003 JP 2003168392**

(71) Applicant: **Kabushiki Kaisha Toyota Jidoshokki**
Kariya-shi, Aichi-ken (JP)

(72) Inventors:
• Inoue, Yoshinori
Kariya-shi, Aichi-ken (JP)

- **Takahata, Junichi**
Kariya-shi, Aichi-ken (JP)
- **Kawamura, Hisato**
Kariya-shi, Aichi-ken (JP)
- **Mochizuki, Kenji**
Kariya-shi, Aichi-ken (JP)
- **Kawachi, Shigeki**
Kariya-shi, Aichi-ken (JP)

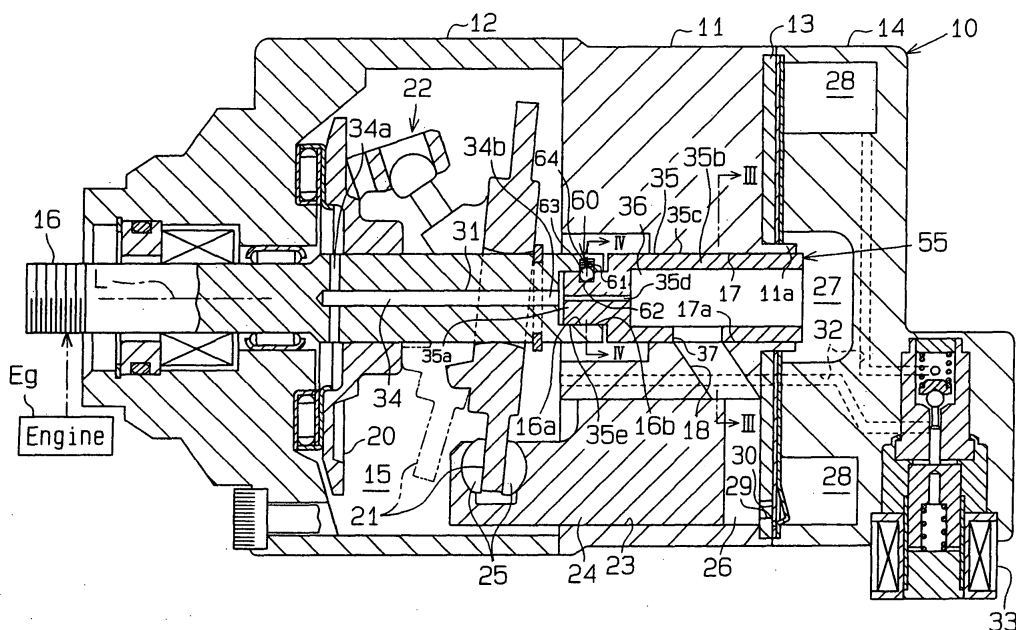
(74) Representative:
Leson, Thomas Johannes Alois, Dipl.-Ing.
Tiedtke-Bühling-Kinne & Partner GbR,
TBK-Patent,
Bavariaring 4
80336 München (DE)

(54) **Piston type compressor**

(57) A piston type compressor includes a suction valve mechanism. The suction valve mechanism includes a rotary valve coupled to a rotary shaft. A valve timing adjusting apparatus is capable of changing a rel-

ative rotational phase, which is a rotational phase of the rotary valve relative to the rotary shaft. Therefore, the compressor is capable of adjusting the valve timing of the rotary valve and has a reduced size in the axial direction.

Fig.1



Description

BACKGROUND OF THE INVENTION

[0001] The present invention relates to a piston type compressor in which rotation of a rotary shaft is converted into reciprocation of pistons, thereby drawing gas from a suction pressure zone to compression chambers through suction valve mechanism, and compressing the gas in the compression chambers.

[0002] Some piston type compressors have a rotary valve as a suction valve mechanism. A rotary valve is coupled to a rotary shaft and rotates in response to rotation of the rotary shaft, thereby selectively opening and closing a gas passage between compression chambers and a suction pressure zone. In a piston type compressor having a rotary valve as a suction valve mechanism, the timing at which a gas passage between a suction pressure zone and a compression chamber is closed (suction end timing) is determined by the position of a valve hole on the outer circumference of the rotary valve, which valve hole guides gas from the suction pressure zone to compression chambers.

[0003] However, the optimum suction end timing varies according to the rotation speed of the piston type compressor, that is, the rotation speed of the rotary shaft. Therefore, when a rotary valve is used as a suction valve mechanism, the actual suction end timing cannot be easily controlled to be in constant synchronization with the optimum suction end timing.

[0004] For example, when the speed of the rotary shaft is increased, the velocity of gas is also increased and the inertial force of the gas is increased, accordingly. Therefore, even if a piston is at or in the vicinity of the bottom dead center, gas is drawn into the compression chamber by the inertial force, which expectedly increases the compression efficiency. However, if the suction end timing of the rotary valve is optimized for a lower speed of the rotary shaft, the actual suction end timing is too advanced compared to the optimum suction end timing when the rotary shaft speed is high. In this case, the effect of the suction by inertial force cannot be expected.

[0005] On the other hand, when the rotary shaft speed is decreased, effective suction of gas by inertia is not expected. Therefore, if the rotary valve is open until the piston reaches the bottom dead center, gas can flow back to the suction pressure zone from the compression chamber. In this manner, if the suction end timing is optimized for a higher rotary shaft speed, backflow of gas from the compression chamber to the suction pressure zone reduces the compression efficiency when the rotary shaft speed is low.

[0006] The drawback that the actual valve timing of a rotary valve is not in synchronization with optimum valve timing presents not only with respect to the suction end timing, but also with respect to the timing at which a gas passage between the suction pressure zone and the

compression chamber is opened (suction start timing).

[0007] Conventionally, to eliminate the drawback that the suction end timing of a rotary valve is displaced from an optimum timing, a rotary valve having two or more axially separated valve holes corresponding to two or more suction end timings has been proposed (for example, Japanese Laid-Open Patent Publication No. 6-117363). According to the speed of the rotary shaft, one of the valve holes that corresponds to the optimum suction end timing at the time is selected.

[0008] However, according to Japanese Laid-Open Patent Publication No. 6-117363, the rotary valve needs to be moved in the axial direction of the rotary shaft relative to the rotary shaft when selecting one of the valve holes that corresponds to the current optimum suction end timing. That is, a room that allows the rotary valve to be moved in the axial direction is required. This increases the piston type compression in the axial direction.

SUMMARY OF THE INVENTION

[0009] Accordingly, it is an objective of the present invention to provide a piston type compressor that is capable of adjusting the valve timing of a rotary valve and has a reduced size in the axial direction.

[0010] To achieve the above objective, the present invention provides a piston type compressor. The compressor includes a rotary shaft, a suction pressure zone, a compression chamber, a gas passage extending from the suction pressure zone to the compression chamber, and a piston defining the compression chamber. The piston reciprocates as the rotary shaft rotates, and as the piston reciprocates, gas is drawn into the compression chamber from the suction pressure zone through the gas passage, and the drawn gas is compressed in the compression chamber. A suction valve mechanism selectively opens and closes the gas passage. The suction valve mechanism includes a rotary valve coupled to the rotary shaft. The rotary valve rotates in response to rotation of the rotary shaft, thereby selectively opening and closing the gas passage. A valve timing adjusting apparatus is capable of changing a relative rotational phase, which is a rotational phase of the rotary valve relative to the rotary shaft.

[0011] Other aspects and advantages of the invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

[0012] 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 illustrating a swash plate type variable displacement compressor according to one embodiment of the present invention;

Fig. 2 is an enlarged partial cross-sectional view of the compressor shown in Fig. 1;

Fig. 3 is a cross-sectional view taken along line III-III in Fig. 2;

Fig. 4 is a cross-sectional view taken along line IV-IV in Fig. 2;

Fig. 5 is a cross-sectional view illustrating the compressor shown in Fig. 1 when the compressor is operating at a high speed; and

Fig. 6 is a cross-sectional view illustrating a modification according to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0013] In the drawings, like numerals are used for like elements throughout.

[0014] A preferred embodiment of the present invention will now be described.

[0015] First, a piston type compressor, which functions as a swash plate type variable displacement compressor for compressing refrigerant, will be described. The compressor is used in a vehicle air conditioner.

[0016] As shown in Fig. 1, the compressor 10 includes a cylinder block 11, a front housing member 12, a valve assembly 13, and a rear housing member 14. The front housing member 12 is secured to the front end of the cylinder block 11. The rear housing member 14 is secured to the rear end of the cylinder block 11 with the valve assembly 13 in between. The cylinder block 11, the front housing member 12, and the rear housing member 14 form a housing of the compressor 10. The left end of the compressor 10 in Fig. 1 is defined as the front of the compressor 10, and the right end is defined as the rear of the compressor 10.

[0017] The cylinder block 11 and the front housing member 12 define a crank chamber 15 in between. A rotary shaft 16 extends through the crank chamber 15 and is rotatably supported by the front housing member 12 and the cylinder block 11. The rotary shaft 16 is coupled to an external drive source, which is an engine Eg in this embodiment. The rotary shaft 16 is rotated by power supplied by the engine Eg. Therefore, the speed of the rotary shaft 16 is varied according to the speed of the engine Eg.

[0018] A lug plate 20 is coupled to the rotary shaft 16 and is located in the crank chamber 15. The lug plate 20 rotates integrally with the rotary shaft 16. A swash plate 21 is accommodated in the crank chamber 15. The swash plate 21 slides along and inclines with respect to the rotary shaft 16. A hinge mechanism 22 is arranged between the lug plate 20 and the swash plate 21. The lug plate 20 permits the swash plate 21 to rotate integrally with the rotary shaft 16 and to incline with respect

to the rotary shaft 16 while sliding along the rotation axis L of the rotary shaft 16.

[0019] As shown in Figs. 1 and 3, a plurality of cylinder bores 23, the number of which is five in this embodiment (only one is shown in Fig. 1) are formed through the cylinder block 11. The cylinder bores 23 surrounds the rear end of the rotary shaft 16 and are spaced from one another by a predetermined angular interval. A single headed piston 24 is accommodated in each cylinder bore 23. The piston 24 reciprocates inside the cylinder bore 23.

[0020] As shown in Fig. 1, the front and rear openings of each cylinder bore 23 are closed by the associated piston 24 and the valve assembly 13. A compression chamber 26 is defined in each cylinder bore 23. The volume of the compression chamber 26 changes according to the reciprocation of the corresponding piston 24. Each piston 24 is coupled to the peripheral portion of the swash plate 21 by a pair of shoes 25. The shoes 25 convert rotation of the swash plate 21, which rotates with the rotary shaft 16, into reciprocation of the pistons 24.

[0021] A suction chamber 27 and a discharge chamber 28 are defined in the rear housing member 14. The suction chamber 27 functions as a suction pressure zone. The suction chamber 27 is defined in a center portion of the rear housing member 14. The discharge chamber 28 is defined to surround the suction chamber 27. The suction chamber 27 is connected to an external pipe connected to a low pressure heat exchanger (not shown) of an external refrigerant circuit. The discharge chamber 28 is connected to an external pipe connected to a high pressure heat exchanger (not shown) of the external refrigerant circuit. The external refrigerant circuit and the compressor 10 form a refrigerant circuit (refrigerant cycle) of the vehicle air conditioner.

[0022] As each piston 24 moves from the top dead center to the bottom dead center, refrigerant gas in the suction chamber 27 is drawn into the corresponding compression chamber 26 through a suction valve mechanism 55 provided in the cylinder block 11. Refrigerant gas drawn into the compression chamber 26 is compressed to a predetermined pressure as the piston 24 is moved from the bottom dead center to the top dead center. Then, the gas is discharged to the discharge chamber 28 through one of discharge ports 29 while flexing one of discharge valve flaps 30, which discharge ports 29 and discharge valve flaps 30 are provided in the valve assembly 13.

[0023] A bleed passage 31, a supply passage 32, and a displacement control valve 33 are provided in the housing of the compressor 10. The bleed passage 31 connects the crank chamber 15 with the suction chamber 27. The bleed passage 31 includes an axial passage 34 formed along the axis L of the rotary shaft 16. An inlet 34a of the axial passage 34 is opened to the crank chamber 15 in the vicinity of the lug plate 20. An outlet 34b of the axial passage 34 is opened at the rear end face of the rotary shaft 16. The supply passage 32 con-

nects the crank chamber 15 with the discharge chamber 28. The supply passage 32 is regulated by the displacement control valve 33, which is a conventional electromagnetic valve.

[0024] The opening degree of the control valve 33 is adjusted to control the balance between the flow rate of highly pressurized gas supplied to the crank chamber 15 through the supply passage 32 and the flow rate of gas conducted out of the crank chamber 15 through the bleed passage 31. The pressure in the crank chamber 15 is thus adjusted. As the pressure in the crank chamber 15 varies, the difference between the pressure in the crank chamber 15 and the pressure in the compression chambers 26 with the pistons 24 in between is changed. This changes the inclination angle of the swash plate 21. Accordingly, the stroke of each piston 24, or the compressor displacement, is controlled.

[0025] For example, when the pressure in the crank chamber 15 is lowered as the opening degree of the displacement control valve 33 is reduced, the inclination angle of the swash plate 21 is increased. This lengthens the stroke of each piston 24 and the compressor displacement is increased, accordingly. In contrast, when the pressure in the crank chamber 15 is increased as the opening degree of the displacement control valve 33 is increased, the inclination angle of the swash plate 21 is decreased. This shortens the stroke of each piston 24 and the compressor displacement is decreased, accordingly.

[0026] The suction valve mechanism 55 will now be described.

[0027] As shown in Figs. 2 and 3, a cylindrical accommodation hole 17 is formed in the housing of the compressor 10. The accommodation hole 17 is located in a center portion of the cylinder block 11 that is surrounded by the cylinder bores 23. A boss portion 11a is formed at the rear end of the cylinder block 11. The boss portion 11a projects through the valve assembly 13 from a portion about the opening of the accommodation hole 17. Part of the boss portion 11a is located in the center portion of the rear housing member 14. Accordingly, the accommodation hole 17 and the suction chamber 27 are continuously arranged along the direction of the axis L. A plurality of introducing passages 18, the number of which is five in this embodiment (only one is shown in Fig. 2), are formed in the cylinder block 11. The introducing passages 18 extend radially from the axis L. Each compression chamber 26 is connected to the accommodation hole 17 by one of the introducing passages 18.

[0028] A rotary valve 35 is rotatably accommodated in the accommodation hole 17. The rotary valve 35 substantially has a hollow cylindrical shape with a bottom. The bottom is located at the front end of the rotary valve 35. A front portion of the rotary valve 35 has a small diameter (small diameter portion 35a). A valve receiving hole 16a is formed at the rear end of the rotary shaft 16, which faces the accommodation hole 17. The small di-

ameter portion 35a of the rotary valve 35 is fitted in the valve receiving hole 16a of the rotary shaft 16. Accordingly, the rotary valve 35 and the rotary shaft 16 are aligned along the common axis L. The rotary valve 35 and the rotary shaft 16 are permitted to be displaced relative to each other in the rotation direction of the rotary shaft 16 about the axis L.

[0029] A spherical body, which is a steel ball 63, is located at an engaging sections of the rotary shaft 16 and the rotary valve 35. The rotary valve 35 is coupled to the rotary shaft 16 with the ball 63 in between, which permits the rotary valve 35 to rotate in response to rotation of the rotary shaft 16, that is, to reciprocation of the pistons 24. The rotary valve 35 has a large diameter portion 35b. A circumferential surface 35c of the large diameter portion 35b and an inner circumferential surface 17a of the accommodation hole 17 form sliding bearing surfaces that rotatably support the rear end portion of the rotary shaft 16.

[0030] The inner circumferential surface 17a of the accommodation hole 17 and the circumferential surface 35c of the large diameter portion 35b of the rotary valve 35 closely and slidably contact each other. A communication hole 35d is formed through a front end portion of the rotary valve 35. The communication hole 35d extends in a front-rear direction. An inside space of the rotary valve 35, which is an introduction chamber 36, is connected to the axial passage 34 in the rotary shaft 16 (the outlet 34b) by the communication hole 35d. The introduction chamber 36 communicates with the suction chamber 27. The communication hole 35d and the introduction chamber 36 form part of the bleed passage 31.

[0031] A suction guide hole 37 is formed in the circumferential wall of the rotary valve 35. The suction guide hole 37 extends in a predetermined circumferential section and functions as a valve hole that always communicates with the introduction chamber 36. When each piston 24 is in a suction stroke, the suction guide hole 37 in the rotary valve 35 communicates with the associated introducing passage 18 in the cylinder block 11. Therefore, refrigerant gas in the suction chamber 27 is drawn into each compression chamber 26 via the introduction chamber 36 in the rotary valve 35, the suction guide hole 37, and the associated introducing passage 18 formed in the cylinder block 11 in order. The introduction chamber 36, the suction guide hole 37, and the associated introducing passage 18 form a gas passage, which extends from the suction pressure zone to the associated compression chamber 26.

[0032] At the end of the suction stroke of each piston 24, the suction guide hole 37 is completely displaced from the associated introducing passage 18. Accordingly, suction of refrigerant gas from the introduction chamber 36 to the compression chamber 26 is stopped. When the piston 24 shifts to compression-discharge stroke, the outer circumferential surface 35c of the large diameter portion 35b of the rotary valve 35 maintains the as-

sociated introducing passage 18 disconnected from the introduction chamber 36. This prevents compression of refrigerant gas and discharge of compressed gas to the discharge chamber 28 from being hindered.

[0033] A valve timing adjusting apparatus 60 will now be described. The valve timing adjusting apparatus 60 changes the relative rotational phase of the rotary valve 35 to the rotary shaft 16, that is, the valve timing of the rotary valve 35 (in this embodiment, suction end timing).

[0034] As shown in Figs. 2, 4, and 5, an accommodating recess 61 is formed in the inner circumference 16b of the valve receiving hole 16a of the rotary shaft 16. The accommodating recess 61 is cylindrical and extends radially outward from an opening in the inner circumference 16b of the valve receiving hole 16a. The ball 63 is accommodated in the accommodating recess 61 to be movable in the extending direction of the recess 61, that is, along a radial direction of the rotary shaft 16. Therefore, the ball 63 is located at an eccentric position relative to the rotary shaft 16. An urging member, which is an urging spring 64, is accommodated in the accommodating recess 61. The urging spring 64 is a coil spring. The urging spring 64 urges the ball 63 radially inward toward the axis L.

[0035] As shown in Figs. 4 and 5, a guide projection 65 is formed on the inner circumference 16b of the valve receiving hole 16a in an area that is rearward of the opening of the accommodating recess 61 with respect to the rotational direction of the rotary shaft 16. In other words, the projection 65 has a phase delayed relative to the phase of the recess 61 and is located rearward of the recess 61 in the clockwise direction. A surface 65a of the guide projection 65 at the accommodating recess 61 is formed continuously with a part (an area 61a at the trailing side in the rotational direction) of the inner wall of the accommodating recess 61, and guides the movement of the ball 63. The surface 65a of the guide projection 65 corresponding to the accommodating recess 61 and the area 61a of the inner surface of the accommodating recess 61 function as power transmitting surfaces 61a, 65a that transmit rotational force of the rotary shaft 16 to the rotary valve 35 with the ball 63.

[0036] In the rotary valve 35, a groove 62 is formed in a part of the outer circumference 35e of the small diameter portion 35a. The groove 62 extends along the circumferential direction and receives the guide projection 65 of the rotary shaft 16. A flat power receiving surface 62a and a flat rear surface 62b are formed in the bottom of the groove 62. The power receiving surface 62a is located at an advancing side in the rotational direction and faces toward a direction opposite the rotational direction. The rear surface 62b is located at the trailing side in the rotational direction and faces the rotational direction. The power receiving surface 62a and the rear surface 62b are inclined relative to each other so that the joint of the surfaces 62a, 62b is dented toward the axis L.

[0037] The ball 63 is located between the power trans-

mitting surfaces 61a, 65a of the rotary shaft 16 and the power receiving surface 62a of the rotary valve 35. Rotational force that is transmitted from the power transmitting surfaces 61a, 65a of the rotary shaft 16 to the ball 63 is then transmitted to the rotary valve 35 through the power receiving surface 62a. Accordingly, the rotary valve 35 is rotated. The power transmission from the rotary shaft 16 to the rotary valve 35 with the ball 63 is performed by tightly holding the ball 63 between the power transmitting surfaces 61a, 65a and the power receiving surface 62a. The distance between the power transmitting surfaces 61a, 65a and the power receiving surface 62a, which tightly hold the ball 63, or perform power transmission (torque transmission) with the ball 63, is varied according to the position of the ball 63 with respect to the radial direction of the rotary shaft 16.

[0038] For example, as shown in Fig. 4, when the ball 63 is moved radially inward from a certain position, the distance between the power transmitting surfaces 61a, 65a and the power receiving surface 62a is increased while the surfaces 61a, 65a and 62a hold the ball 63. To increase the distance between the power transmitting surfaces 61a, 65a and the power receiving surface 62a, the rotary valve 35 needs to be shifted relative to the rotary shaft 16 in the rotational direction of the rotary shaft 16. Shifting the rotary valve 35 relative to the rotary shaft 16 in the rotational direction, that is, advancing the relative rotational phase of the rotary valve 35 to the rotary shaft 16, advances the suction end timing of the rotary valve 35.

[0039] The rear surface 62b of the groove 62 of the rotary valve 35 and a surface of the guide projection 65 of the rotary shaft 16 that faces the rear surface 62b function as most advanced phase defining surfaces 62b, 65b. When the defining surfaces 62b, 65b contact each other, the relative rotational phase of the rotary valve 35 is most advanced, that is, the suction end timing of the rotary valve 35 is most advanced. That is, the guide projection 65 and the groove 62 form an engaging mechanism that defines the most advanced phase relative to the rotary shaft 16.

[0040] In contrast, as shown in Fig. 5, when the ball 63 is moved radially outward from a certain position, the distance between the power transmitting surfaces 61a, 65a and the power receiving surface 62a is decreased while the surfaces 61a, 65a and 62a hold the ball 63. To decrease the distance between the power transmitting surfaces 61a, 65a and the power receiving surface 62a, the rotary valve 35 needs to be shifted relative to the rotary shaft 16 in a direction opposite to the rotational direction of the rotary shaft 16. Shifting the rotary valve 35 relative to the rotary shaft 16 in a direction opposite to the rotational direction, that is, retarding the relative rotational phase of the rotary valve 35 to the rotary shaft 16, retards the suction end timing of the rotary valve 35.

[0041] The position of the ball 63 in the radial direction of the rotary shaft 16 is determined by the equilibrium of centrifugal force acting on the ball 63, an urging force,

or the reaction force from the power receiving surface 62a that acts on the ball 63 based on torque transmitted from the rotary shaft 16 to the rotary valve 35, and the radially inward urging force of the spring 64. The urging force based on the transmitted torque and the urging force of the spring 64 are fixed parameters, which are determined during the machine design. Only the centrifugal force is variable parameter that varies according to the rotation speed of the rotary shaft 16. Therefore, the position of the ball 63 is determined by the rotation speed of the rotary shaft 16.

[0042] For example, when the rotation speed of the rotary shaft 16 is lowered, the centrifugal force acting on the ball 63 is decreased, which causes the ball 63 to be moved radially inward by the spring 64. Accordingly, the rotary valve 35 is rotated relative to the rotary shaft 16 in the rotation direction, or the relative rotational phase of the rotary valve 35 is advanced. This advances the suction end timing of the rotary valve 35, and thus prevents backflow of gas from the compression chambers 26 to the introduction chamber 36, which tends to occur when the pistons 24 are at or in the vicinity of the bottom dead center. Accordingly, the compression efficiency is prevented from being lowered by backflow of gas.

[0043] In contrast, when the rotation speed of the rotary shaft 16 is increased, the centrifugal force acting on the ball 63 is increased, which causes the ball 63 to be moved radially outward against the spring 64. Accordingly, the rotary valve 35 is rotated relative to the rotary shaft 16 in a direction opposite to the rotation direction, or the relative rotational phase of the rotary valve 35 is retarded. This retards the suction end timing of the rotary valve 35. Therefore, even if the pistons 24 are at or in the vicinity of the bottom dead center, suction by inertial force of the gas is effectively used to increase the compression efficiency.

[0044] In this embodiment, the ball 63 functions as a power transmitting member forming the valve timing adjusting apparatus 60. The urging spring 64 and power receiving surface 62a, which apply force to the ball 63 to determine the position of the ball 63, function as a position determining device that forms a part of the valve timing adjusting apparatus 60.

[0045] The above embodiment provides the following advantages.

(1) The suction end timing of the rotary valve 35 is adjusted by rotating the rotary valve 35 relative to the rotary shaft 16, thereby changing the relative rotational phase of the rotary valve 35 to the rotary shaft 16. For example, Japanese Laid-Open Patent Publication No. 6-117363 discloses a technique to move a rotary valve in the axial direction relative to a rotary shaft, thereby adjusting the suction end timing. Compared to the technique of the Japanese Laid-Open Patent Publication 6-117363, the present invention is capable of reducing the size of

the compressor 10 in the direction of the axis L.

(2) The relative rotational phase of the rotary valve 35 to the rotary shaft 16 is changed according to the rotation speed of the rotary shaft 16. Therefore, the suction end timing of the rotary valve 35 is optimized for the rotational speed of the rotary shaft 16.

(3) The rotational phase of the rotary valve 35 is changed to delay the suction end timing when the rotation speed of the rotary shaft 16 is increased. In contrast, the rotational phase of the rotary valve 35 is changed to advance the suction end timing when the rotation speed of the rotary shaft 16 is decreased. Accordingly, the compression efficiency of the compressor 10 is increased as described above.

(4) A spherical body (steel ball 63) is used as a power transmitting member. Since a spherical body need not be set in a specific orientation, the ball 63 is easily assembled in the compressor 10. For example, the ball 63 would neither be inclined between the rotary shaft 16 and the rotary valve 35 nor be immovable.

(5) The position of the ball 63, that is the suction end timing of the rotary valve 35, is automatically changed based on change in the centrifugal force acting on the ball 63. Therefore, compared to a case where the position of the ball 63 is determined, for example, by an actuator, the device for determining the position of the ball 63 is simplified.

[0046] The invention may be embodied in the following forms.

[0047] In the above embodiment, the settings of the components are determined while placing importance on the function of the rotary valve 35 to change the suction end timing. Alternatively, the settings of the components may be determined such that the rotary valve 35 is capable of adjusting the suction start timing. That is, the present invention is not limited to the suction end timing adjusting apparatus as in the above embodiment, but may be embodied in a suction start timing adjusting apparatus.

[0048] For example, the rotary valve 35 may have a bypass groove for bypassing residual gas in a cylinder bore 23 in a state immediately after a compression stroke to another cylinder bore in a state immediately before the end of a suction stroke. The bypass groove is thus capable of increasing the volumetric efficiency of the compressor. In this configuration, the period of bypassing gas is shortened as the rotation speed of the rotary shaft 16 increases. Therefore, the gas pressure in the cylinder bore 23 in a state immediately after a compression stroke is not sufficiently lowered. As a result, when a suction stroke of the cylinder bore 23 is

started, backflow of gas occurs if the internal pressure of the introduction chamber 36 in the rotary valve 35 is low, which can add to the compressor noise.

[0049] Therefore, the configuration in which the suction start timing is delayed when the rotary speed of the rotary shaft 16 increases is suitable for a compressor with a rotary valve that has a bypass groove, since the configuration reduces the compressor noise.

[0050] In the above embodiment, the position of the ball 63 is automatically determined by the equilibrium of centrifugal force, which varies according to the rotation speed of the rotary shaft 16, and the radially inward urging force applied by the urging spring 64. This configuration may be changed. Specifically, an actuator may be used to determine the position of the ball 63, or the relative rotational phase of the rotary valve 35. In this case, the valve timing of the rotary valve 35 is adjusted by the actuator.

[0051] For example, as shown in Fig. 6, an actuator, which is an electromagnetic attractive force applying device 70, may be provided in a position in the cylinder block 11 surrounding the joint of the rotary shaft 16 and the rotary valve 35. The attractive force applying device 70 is capable of applying electromagnetic attractive force (radially outward urging force) to the steel ball 63. In this case, the urging spring 64 is configured to have a strong spring force that does not permit the ball 63 to be displaced during a high speed rotation of the rotary shaft 16. Therefore, the position of the ball 63 in the radial direction is adjusted according to the electromagnetic attractive force applied to the ball 63 by the attractive force applying device 70.

[0052] The attractive force applying device 70 receives a driving signal from the driving circuit 72 based on a command from the control computer 71. The computer 71 and the driving circuit 72 form a control device. The attractive force applying device 70 applies to the steel ball 63 an electromagnetic attractive force the magnitude of which corresponds to the driving signal from the driving circuit 72. The control computer 71 adjusts the driving signal supplied to the attractive force applying device 70 by the driving circuit 72 based on detected information from a rotation speed sensor 73 that detects the rotation speed of the rotary shaft 16.

[0053] When the rotation speed of the rotary shaft 16 increases, the control computer 71 increases the electromagnetic attractive force generated by the attractive force applying device 70, thereby displacing the ball 63 radially outward. Accordingly, the suction end timing of the rotary valve 35 is delayed. When the rotation speed of the rotary shaft 16 decreases, the control computer 71 decreases the electromagnetic attractive force generated by the attractive force applying device 70, thereby displacing the ball 63 radially inward. Accordingly, the suction end timing of the rotary valve 35 is advanced.

[0054] For example, compared to the above embodiment of Fig. 2, in which the position of the ball 63 is automatically determined by centrifugal force (an internal

control), the embodiment of Fig. 6 is capable of accurately determining the position of the ball 63. In other words, the actual suction end timing is brought close to an optimal suction end timing.

[0055] In the embodiment of Fig. 6, the position determining device forming the valve timing adjusting apparatus 60 includes: the urging spring 64, the power receiving surface 62a, and the attractive force applying device 70, which apply force to the ball 63 for determining the position of the ball 63; the control computer 71 and the driving circuit 72, which control the attractive force applying device 70; and the rotation speed sensor 73, which provides the control computer 71 with the rotation speed information of the rotary shaft 16.

[0056] In the embodiment of Fig. 6, the valve timing of the rotary valve 35 is adjusted according to the rotation speed of the rotary shaft 16. However, the valve timing of the rotary valve 35 may be adjusted, for example, according to the displacement of the compressor 10.

The control computer 71 is capable of obtaining the displacement of the compressor 10 based on information related to, for example, current supplied to the displacement control valve 33. That is, the capability to control the valve timing of the rotary valve 35 permits the valve timing to be adjusted based on information other than the information related to the rotation speed of the rotary shaft 16.

[0057] In the above embodiments, the ball 63, which is a spherical body, is used as the power transmitting member. However, the power transmitting member may be a cylindrical body, a body formed by combining a semispherical body and a cylindrical body, or a triangle pole body, as long as the power transmitting member is capable of transmitting power from the rotary shaft 16 to the rotary valve 35.

[0058] In the above embodiments, the urging spring 64, which is a coil spring, is used as the urging member. However, the urging member may be, for example, a leaf spring, or a rubber body, as long as the urging member is capable of urging the power transmitting member.

[0059] The present invention may be applied to a wobble plate type variable displacement compressor.

[0060] The present invention may be applied to a double-headed piston type compressor.

[0061] The present invention may be applied to a wave cam type compressor.

[0062] 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.

[0063] A piston type compressor includes a suction valve mechanism. The suction valve mechanism includes a rotary valve coupled to a rotary shaft. A valve timing adjusting apparatus is capable of changing a relative rotational phase, which is a rotational phase of the rotary valve relative to the rotary shaft. Therefore, the compressor is capable of adjusting the valve timing of

the rotary valve and has a reduced size in the axial direction.

Claims

1. A piston type compressor, comprising:

a rotary shaft;
 a suction pressure zone;
 a compression chamber;
 a gas passage extending from the suction pressure zone to the compression chamber;
 a piston defining the compression chamber, wherein the piston reciprocates as the rotary shaft rotates, and as the piston reciprocates, gas is drawn into the compression chamber from the suction pressure zone through the gas passage, and the drawn gas is compressed in the compression chamber; and
 a suction valve mechanism that selectively opens and closes the gas passage, the suction valve mechanism including a rotary valve coupled to the rotary shaft, wherein the rotary valve rotates in response to rotation of the rotary shaft, thereby selectively opening and closing the gas passage,
 the compressor being **characterized by**:

a valve timing adjusting apparatus capable of changing a relative rotational phase, which is a rotational phase of the rotary valve relative to the rotary shaft.

2. The compressor according to claim 1, **characterized in that** the adjusting apparatus determines the relative rotation phase according to a rotation speed of the rotary shaft.

3. The compressor according to claim 1 or 2, **characterized in that** the adjusting apparatus rotates according to the rotation speed of the rotary shaft, and determines the relative rotation phase according to a centrifugal force acting on the adjusting apparatus.

4. The compressor according to claim 2 or 3, **characterized in that**, when the rotation speed of the rotary shaft is increased, the adjusting apparatus changes the relative rotational phase to delay timing at which suction of gas from the suction pressure zone to the compression chamber is ended, and wherein, when the rotation speed of the rotary shaft is decreased, the adjusting apparatus changes the relative rotational phase to advance the suction end timing.

5. The compressor according to any one of claims 2

to 4, **characterized in that**, when the rotation speed of the rotary shaft is increased, the adjusting apparatus changes the relative rotational phase to delay timing at which suction of gas from the suction pressure zone to the compression chamber is started, and wherein, when the rotation speed of the rotary shaft is decreased, the adjusting apparatus changes the relative rotational phase to advance the suction start timing.

6. The compressor according to any one of claims 2 to 5, **characterized in that** the rotary valve is arranged coaxial with the rotary shaft and engaged with the rotary shaft such that the rotary valve rotates relative to the rotary shaft, wherein the adjusting apparatus has a power transmitting member for transmitting power from the rotary shaft to the rotary valve, wherein the power transmitting member is displaceably located between the rotary shaft and the rotary valve, wherein the relative rotational phase is changed according to the position of the power transmitting member, and wherein the adjusting apparatus has a position determining device that determines the position of the power transmitting member according to the rotation speed of the rotary shaft.

7. The compressor according to claim 6, **characterized in that** the power transmitting member is a spherical body.

8. The compressor according to claim 6 or 7, **characterized in that** the power transmitting member is located at an eccentric position relative to the rotary shaft and is displaceable in a radial direction of the rotary shaft, wherein the position determining device has an urging member for urging the power transmitting member radially inward with respect to the rotary shaft, and wherein the position of the power transmitting member is determined according to a centrifugal force that acts on the power transmitting member as the rotary shaft rotates and an urging force applied to the power transmitting member by the urging member.

9. The compressor according to any one of claims 6 to 8, **characterized in that** the rotary shaft has a power transmitting surface that transmits power to the power transmitting member, the rotary valve has a power receiving surface that receives power from the power transmitting member, and wherein the distance with respect to the rotation direction of the rotary shaft between the power transmitting surface and the power receiving surface is reduced toward the axis of the rotary shaft.

10. The compressor according to any one of claims 6 to 9, **characterized in that** one of the rotary shaft

and the rotary valve has an engaging projection that extends in the axial direction of the rotary shaft, and the other has an engaging recess to which the engaging projection is fitted, and wherein the power transmitting member and the position determining device are located between the engaging projection and the engaging recess. 5

11. The compressor according to any one of claims 6 to 10, **characterized in that** the rotary shaft and the rotary valve have an engaging mechanism that defines a relative rotational phase that is most advance relative to the rotary shaft. 10

12. The compressor according to any one of claims 6 to 11, **characterized in that** the position determining device includes: 15

a rotation speed sensor for detecting the rotation speed of the rotary shaft; 20

an urging force applying device for applying an urging force to the power transmitting member, wherein the urging force applying device changes the urging force based on a driving signal from the outside, thereby determining the position of the power transmitting member; 25

and
a control device that adjusts the driving signal supplied to the urging force applying device based on detected information from the rotation speed sensor. 30

13. The compressor according to claim 12, **characterized in that** the power transmitting member is made of metal, the urging force applying device includes an electromagnet, and wherein the electromagnet generates an electromagnetic attractive force for urging the power transmitting member. 35

14. The compressor according to claim 1, **characterized in that** the rotary valve is arranged coaxial with the rotary shaft and engaged with the rotary shaft such that the rotary valve rotates relative to the rotary shaft, wherein the adjusting apparatus has a power transmitting member for transmitting power from the rotary shaft to the rotary valve, wherein the power transmitting member is displaceably located between the rotary shaft and the rotary valve, wherein the relative rotational phase is changed according to the position of the power transmitting member, and wherein the adjusting apparatus has a position determining device that determines the position of the power transmitting member. 40 45 50

15. The compressor according to claim 14, **characterized in that** the position determining device includes: 55

an urging force applying device for applying an urging force to the power transmitting member, wherein the urging force applying device changes the urging force based on a driving signal from the outside, thereby determining the position of the power transmitting member; and

a control device that adjusts the driving signal supplied to the urging force applying device.

Fig.1

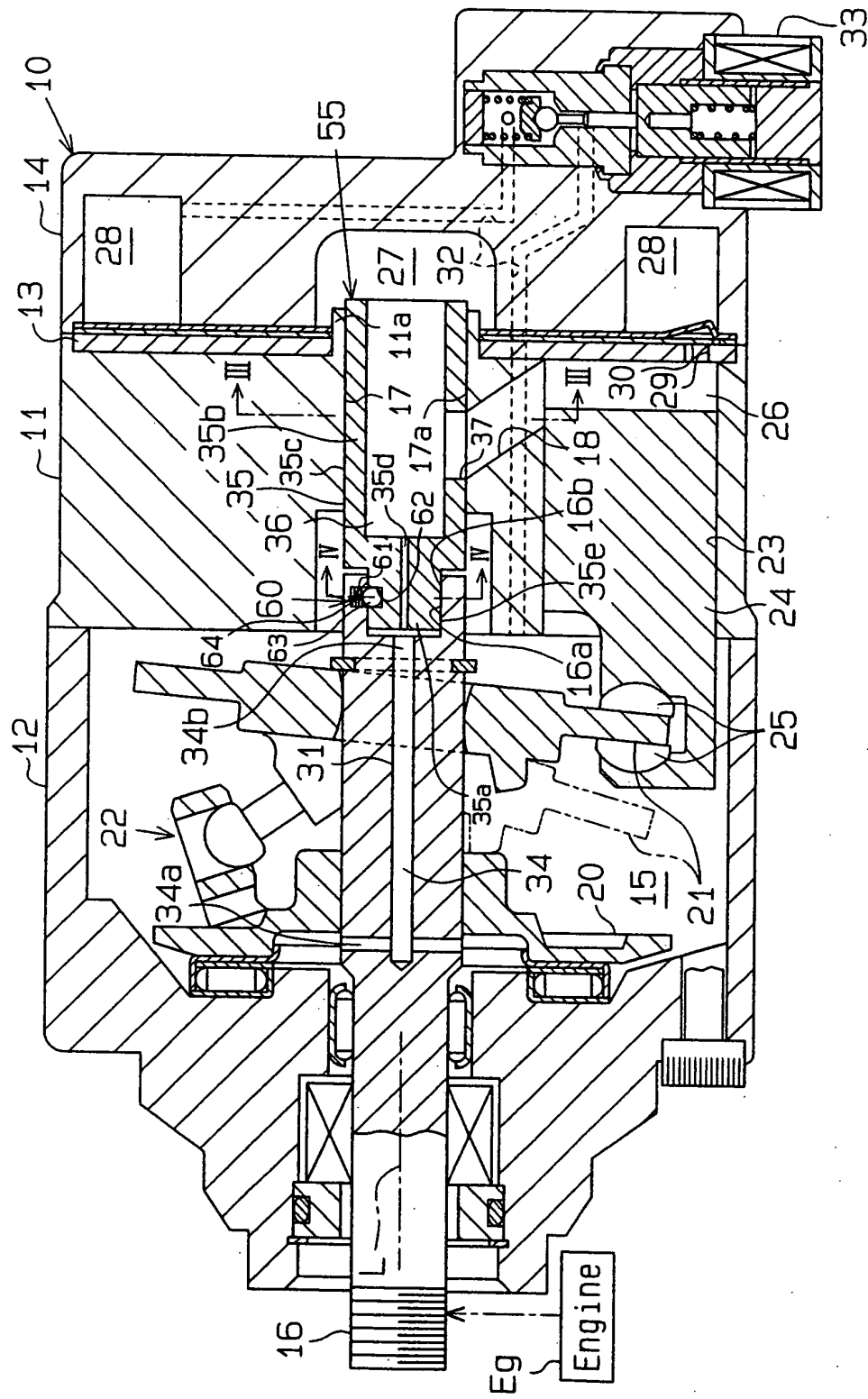


Fig. 2

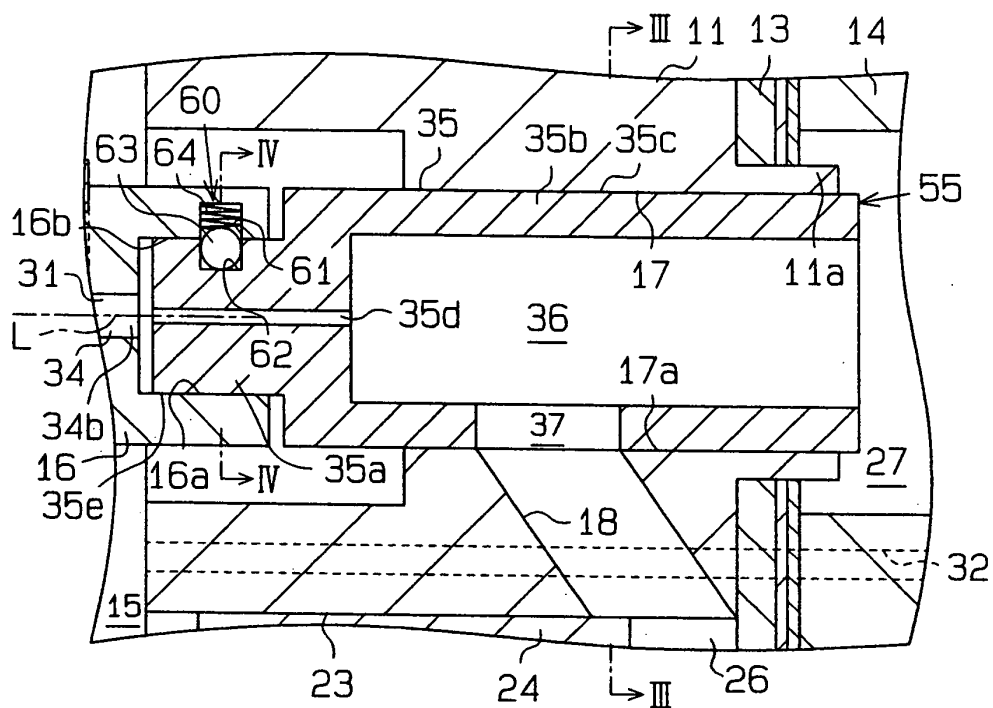


Fig. 3

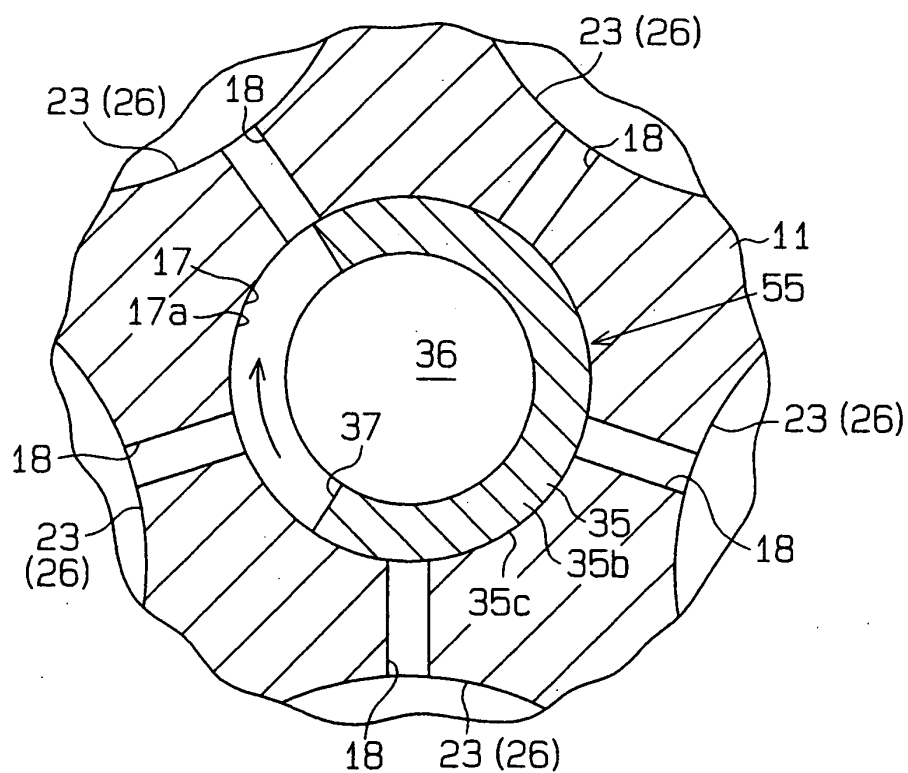


Fig.4

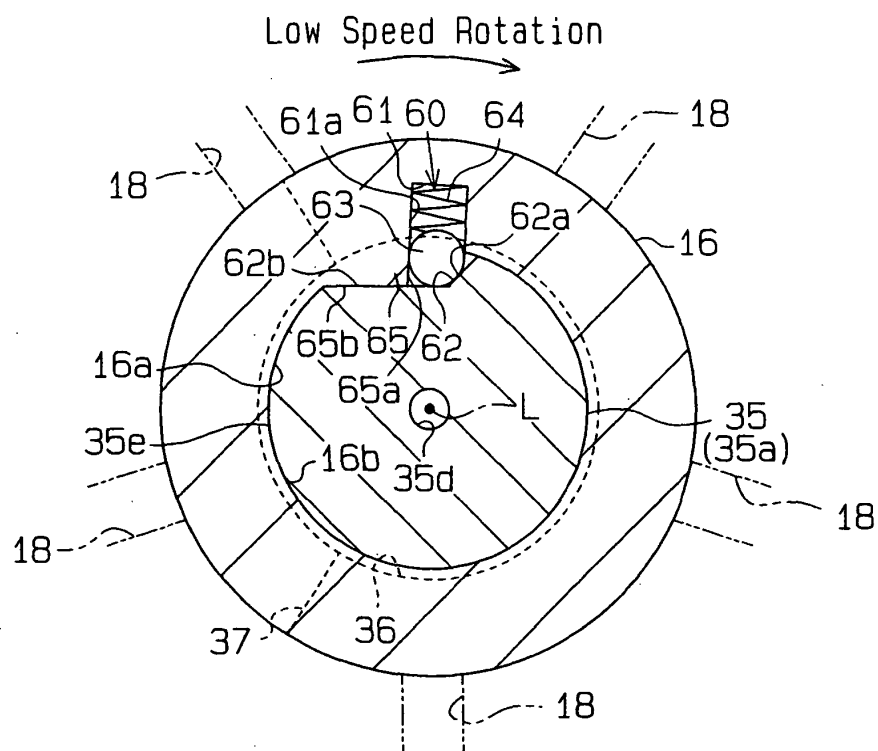


Fig.5

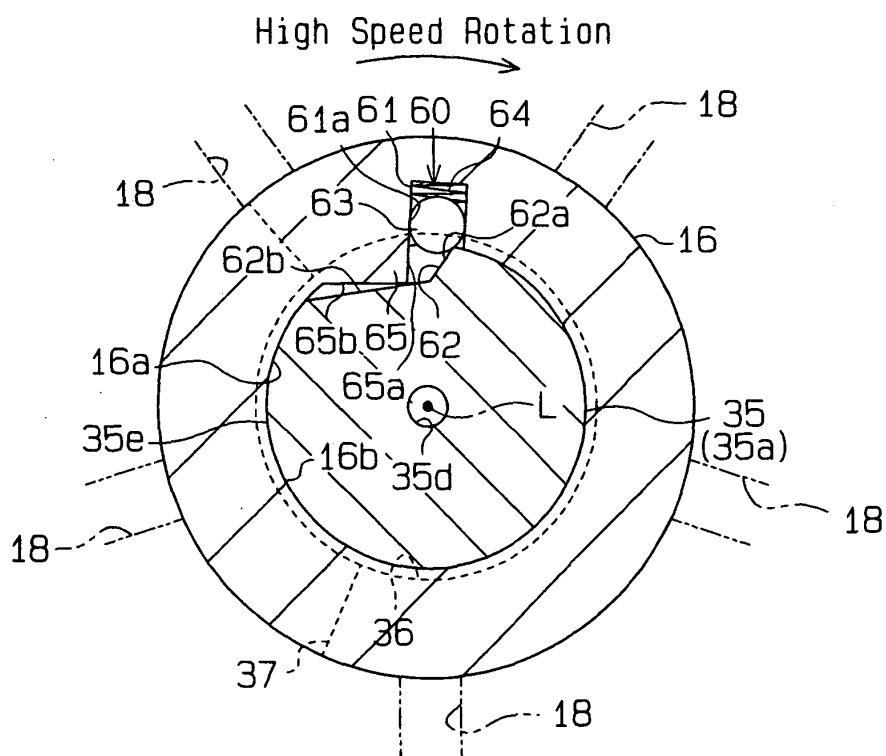


Fig. 6

