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

(11) Publication number: 0 190 013
A2

(12) EUROPEAN PATENT APPLICATION

(21) Application number: 86300455.2

(51) Int. Cl.[4]: F 04 B 1/30

(22) Date of filing: 23.01.86

(30) Priority: 25.01.85 JP 10992/85

(43) Date of publication of application:
06.08.86 Bulletin 86/32

(84) Designated Contracting States:
DE FR GB IT SE

(71) Applicant: SANDEN CORPORATION
20 Kotobuki-cho
Isesaki-shi Gunma-ken(JP)

(72) Inventor: Terauchi, Kiyoshi
8-14 Heiwa-cho
Isesaki-shi Gunma, 372(JP)

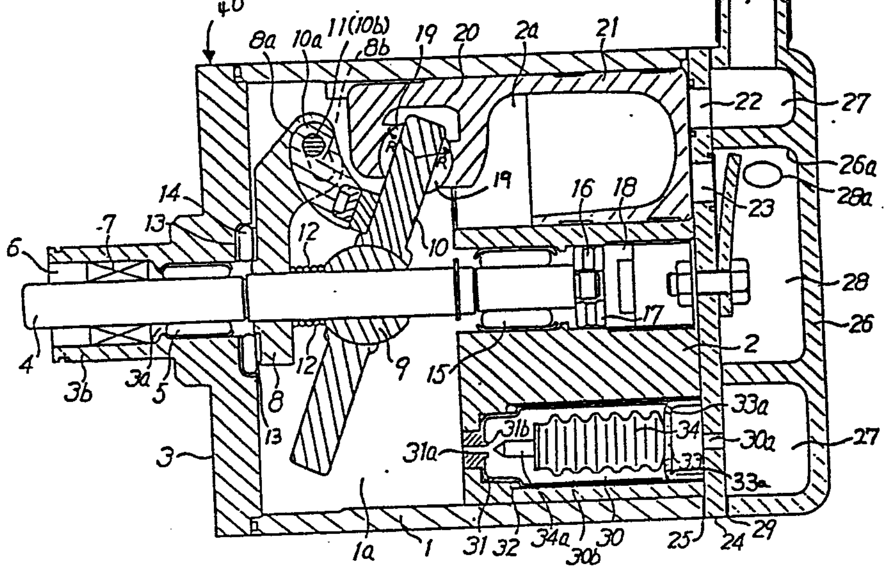
(74) Representative: Jackson, Peter Arthur et al,
GILL JENNINGS & EVERY 53-64 Chancery Lane
London WC2A 1HN(GB)

(54) Variable capacity compressor.

(57) A variable capacity compressor (40) has an inclined plate (10) which is rotated by a rotor (8) attached to a drive shaft (4). The rotary motion is converted by a pair of shoes (19) which make sliding contact with both faces of the plate (10) into reciprocating motion of a piston (21) so that fluid may be pumped from a suction chamber (27) into a discharge chamber (28). The shoes (19) have spherically curved surfaces which allow the plate (10) to pivot relatively to the piston (21) and which are arranged to be coincident with the surface of a notional common sphere of radius (R) so as to prevent pivoting of the plate (10) from causing a gap to appear between the plate (10) and the shoes (19). A bellows (34) is used to control the pressure in a chamber (1a) so that changes of pressure are used to cause the plate (10) to pivot about a pin (11) and thereby alter the stroke of the piston (21).

EP 0 190 013 A2

./...



27a
40
8a
10a
11(10b)
8b
19
20
2a
21
22
27
26a
28a
14
19
16
18
23
7
13
12
10
6
28
17
26
2
4
3b
3a
5
12
9
15
8
13
33a
3
31b
34
30a
27
31a
33
33a
1a
1
31
32
34a
30b
30
25
24
29

SANDEN CORPORATION. Ref: 50/2849/01

VARIABLE CAPACITY COMPRESSOR

The invention relates to an inclined plate type compressor suitable for use in a refrigerator, and more particularly to one having variable capacity.

In known compressors which compress fluid by converting the rotary motion of an inclined plate into the reciprocating, motion of pistons, the swept volumes are altered by changing the stroke length of the pistons, which are connected to the inclined plate, by varying the angle of inclination of the inclined plate.

The inclined plate comprises a cam rotor and a swash plate. The angle of the cam rotor, which is connected to a rotary shaft, can be varied and, since the swash plate moves with the motion of the cam rotor, the angle of inclination of the swash plate varies with the change in angle of the cam rotor. The pistons are connected to the swash plate via connecting rods and their stroke length varies with the variations in the angle of inclination of the swash plate. In this manner, the total swept volume of the compressor is altered.

However, the construction of the swash plate and cam rotor supports and the bearings needed to permit variations in the cam rotor angle is complicated, and a rotation prevention mechanism has to be provided in order to prevent rotation of the swash plate.

Furthermore, since the rotation prevention mechanism in a variable capacity compressor must function even when the angle of inclination of the plate is being varied, it cannot use the bevel gear construction used in fixed capacity compressors. Instead, it has comprised a slide rod which projects radially from the outer edge of the swash plate and engages a guide groove formed axially in the crank case. Rotation of the swash plate is prevented by sliding motion of the slide rod in the guide groove. However, durability is a problem because of the rubbing that occurs. Also, since the sliding friction changes in response to the range of movement of the swash plate, the angular velocity, which should be transfered to the swash plate, is not uniform during variations of the angle of inclination and hence vibration is produced.

In accordance with the present invention, a variable capacity compressor comprises a housing having a chamber; a drive shaft supported rotatably in the housing; an inclined plate coupled to the drive shaft; a plurality of pistons arranged to be driven reciprocably by rotation of the inclined plate, each piston being connected to the inclined plate by a connecting means which abuts slidably against both faces of the inclined plate; a hinge mechanism for coupling the inclined plate to the drive shaft and for permitting the angle of inclination of the inclined plate to be varied within a predetermined range; and a means for controlling the pressure in the chamber, the arrangement of the hinge mechanism being such that variations in the chamber pressure force the mechanism to move and vary the angle of inclination of the inclined plate, and wherein each connecting means comprises a pair of sliding shoes, each shoe being mounted in a respective holding portion of the respective piston and having a first surface in sliding contact with the respective face of the inclined plate and a second, spherically curved, surface in sliding abutment with a complementary surface on the respective holding portion, the second surfaces of the two shoes being coincident with the surface of a common notional sphere.

The invention will now be described by way of example with reference to the accompanying drawings, in which;-

Figure 1 is a sectional side view of a variable capacity compressor in accordance with the invention showing the inclined plate at its largest angle of inclination;

Figure 2 is a sectional side view similar to that of Figure 1, but showing the inclined plate at its smallest angle of inclination;

Figure 3(a) is a sectional side view of a conventional mechanism for coupling a piston and an inclined plate.

Figure 3(b) is a sectional side view of the conventional coupling mechanism of Figure 3(a) illustrating the situation in which a gap appears between the inclined plate and a pair of sliding shoes when the angle of inclination of the inclined plate is reduced; and,

Figure 4 is a sectional side view of a coupling mechanism in accordance with the invention.

Figure 1 shows an example of a variable capacity compressor 40 which has a casing 1 covered at one end by a front housing 3 and at the other end by a valve plate 24 and a cylinder head 26. A cylinder block 2 is mounted inside the other end of the casing 1 and has a cylinder bore 2a. A hole 3a is formed in the centre of the front housing 3 to allow a drive shaft 4 to pass through. A radial bearing 5 is disposed in hole 3a to support rotatably the drive shaft 4.

A sleeve 3b extends from the outer wall of the front housing 3 and surrounds the drive shaft 4. A mechanical seal 7 is positioned in a seal chamber 6 formed between the outer surface of the drive shaft 4 and the inner wall of the sleeve 3b. A chamber 1a is formed between the inner wall of the front housing 3 and an end surface of the cylinder block 2. A rotor 8 is mounted on the drive shaft 4 in the chamber 1a, and has an ear-shaped portion 8a which is formed in the ear-shape by an end of the rotor 8 being bent towards the drive shaft 4. An elongate hole 8b is formed in the ear-shaped portion 8a.

A spherical surface bush 9 is disposed on the drive shaft 4 at a distance along from the rotor 8 so that the spherical surface bush 9 can slide along the drive shaft 4. A disk-shaped inclined plate 10 is pivotably mounted on the spherical surface bush 9 and is attached to an ear-shaped portion 10a located adjacent to the ear-shaped portion 8a and having a hole 10b in alignment with the elongate hole 8b, a pin 11 being inserted through both the elongate hole 8b and the hole 10b with the pin 11 being able to slide within the elongate hole 8b. A hinge mechanism which couples the inclined plate 10 to the drive shaft 4 thus comprises the following components: the elongate hole 8b in the ear-shaped portion 8a of the rotor 8, the hole 10b in the portion 10a and the pin 11. A spring 12 is mounted over the drive shaft 4 and biases apart the rotor 8 and the bush 9. A pair of thrust races 13 are formed on the inner wall of the front housing 3 and the opposing surface of the rotor 8 with a thrust needle bearing 14 placed therebetween.

The other end of the drive shaft 4 is supported rotatably by a radial needle bearing 15 mounted in the cylinder block 2 and is restrained axially by a thrust needle bearing 16 and a plate 17 held in position by an adjusting screw 18.

Sliding shoes 19 each having a flat surface and a spherically curved surface are disposed on each side of the inclined plate 10 with the flat surfaces in contact with the plate 10 so that it can slide between them. As shown in Figure 1, the pair of shoes 19 are positioned so that their spherically curved surfaces form part of the surface of a common sphere having a radius of curvature of R. The centre of the sphere defined by the curved surfaces is located in a plane halfway between the two faces of the inclined plate 10, i.e. at the middle of the thickness of the inclined plate 10. A piston rod 20 has a forked structure on one of its ends which is used to hold together the sliding shoes 19 and the inclined plate 10. A piston 21 is disposed on, or as in this case forms an integral extension of, the other end of the piston rod 20, and is mounted slidably in the cylinder bore 2a. A number of such piston rods/pistons and pairs of sliding shoes are located in the compressor.

A suction hole 22 and a discharge hole 23 are formed in the valve plate 24. One side of the valve plate 24 abuts the cylinder block 2 through a gasket 25; and the other side abuts the cylinder head 26 through a gasket 29. A partition 26a divides the cylinder head 26 into a suction chamber 27 and a discharge chamber 28. The casing 1 is closed by the valve plate 24 and the cylinder head 26 which are attached to the top of the cylinder block 2. A suction port 27a allows flow into the suction chamber 27, and a discharge port 28a allows flow out of the discharge chamber 28.

A duct 30 formed mainly in the cylinder block 2 connects the suction chamber 27 with the chamber 1a and comprises a hole 30a which penetrates the valve plate 24 and the gaskets 25 and 29 and a chamber 30b formed in the cylinder block 2 itself and within which a bellows 34 is positioned.

A coupling 31, which has a hole 31a and a valve seat 31b, is located in the chamber 30b at the end adjacent to the chamber 1a. An O-ring 32 is inserted between the coupling 31 and the cylinder block 2 to prevent leakage of fluid gas. A pedestal 33, which has holes 33a on two sides of its top, is fixed in the chamber 30b and the bellows 34, which has a needle 34a on its top end, is mounted on the surface of the pedestal 33 and contains gas at a certain pressure. When the top end of the needle 34a is inserted into the hole 31a, a seal is formed between the needle 34a and the valve seat 31b of the coupling 31.

The operation of the above-described compressor when used in a refrigeration circuit will now be explained with reference to Figures 1 and 2.

When rotary motion from a drive source rotates the drive shaft 4, the inclined plate plate 10 is also rotated since it is connected to the shaft 4 through the rotor 8 and the hinge portion. However, the rotary motion is not transmitted to the piston rod 20 because the sliding shoes 19 allow the inclined plate 10 to slide between their flat surfaces. Thus, the rotary motion of the inclined plate 10 is converted only into reciprocating motion of the piston 21 (i.e. motion towards the left and right in these figures). This reciprocating motion causes the fluid to be sucked through the suction hole 22, compressed in the cylinder bore 2a and discharged into the discharge chamber 28 through the discharge hole 23.

The swept volume of the compressor is changed in the following manner. When the load on the refrigeration system rises above its "substantially off-load" capacity (compressor as in Figure 2), the suction pressure of the refrigerant increases, thereby increasing the pressure in the suction chamber 27. Accordingly, the pressure in the chamber 30b, which is connected to the suction chamber 27, also increases.

The gas in the bellows 34 is at a pressure which is a little higher than the suction pressure produced when the load is within the capabilities of the "substantially off-load" system. Therefore, the rise in suction pressure causes the bellows 34 to contract and the free end to move towards the right as shown in Figure 1. The needle 34a moves away from valve seat 31b and this opens the hole 31a. The suction chamber 27 is now connected to the chamber 1a and, since the gas which leaks from the cylinder chambers 2a into the chamber 1a during the compression strokes is now able to return to the suction chamber 27, the pressure in the chamber 1a reduces to a value almost equal to the pressure in the suction chamber 27.

The pistons, which are equiangularly spaced around the plate 10, experience reactive forces from the gas being compressed during the compression strokes and this produces a moment M_1 tending to cause clockwise rotation of the portion 10a (as viewed in Figures 1 and 2) about the pin 11. An anticlockwise moment M_2 is produced by the compression of the coil spring 12; and an anticlockwise moment M_3 is produced by any pressure difference between the chamber 1a and the suction chamber 27 acting across the pistons. Thus, when the aperture 31a is open and no pressure difference exists between the chamber 1a and the suction chamber 27, only the moment M_2 is opposing the moment M_1. The spring rate of the coil spring 12 is chosen to result in M_1 being greater than M_2, and therefore the inclined plate 10 moves towards the rotor 8 and is rotated about the pin 11 of the hinge coupling mechanism with pin 11 being pushed to the radially outer end of the elongate hole 8b (i.e. the compressor becomes as shown in Figure 1). In this way, the slant angle of the inclined plate 10 is set to its maximum, thereby maximizing the stroke length of the pistons 21 and causing the compressor to bring the refrigeration system up to its "on-load" capacity.

When the load on the system decreases, the compressor runs at high speed and the swept volume of the compressor is now too large. The pressure in the suction chamber 27 decreases and the bellows 34 move to close the hole 31a with its needle 34a, i.e. the bellows is as shown in Figure 2. The pressure in the chamber 1a gradually rises as gas leaks from the cylinder chambers into the chamber 1a through the gaps between the pistons 21 and the cylinder block 2 during the compression strokes. As the pressure rises, a pressure difference between the chamber 1a and the suction chamber 27 appears, thereby generating an increasing moment M_3. This moment M_3 is opposed to the moment M_1 so that, at some point, the combined magnitude of moments M_2 and M_3 exceeds the moment M_1. When this occurs, a net moment in the anticlockwise direction about pin 11 appears and rotates the inclined plate 10 so that the slant angle of the inclined plate 10 decreases. The slant angle continues to decrease until the pin 11 contacts the radially inner end of the elongate hole 8b (i.e. the compressor returns to being as shown in Figure 2). This decrease in slant angle reduces the stroke of the pistons 21 in the cylinders 2a and hence the swept volume of the compressor. Since it is undesirable to stop completely movement of the pistons, as this would result in the flow of refrigerant gas and lubricating oil also stopping, some movement of the pistons should be maintained to continue lubrication of the compressor. For this reason, the length of the elongate hole 8b is chosen such that the minimum swept volume is between 20 and 30 percent of the maximum swept volume.

Figure 3(a) shows a conventional coupling mechanism for a piston and an inclined plate. The sliding shoes 19' are half spheres having a radius B and are located on respective sides of an inclined plate 10' of thickness t, the distance A between the furthest extremities of the shoes being defined by the following formula (1).

$$A = 2B + \frac{t}{\cos \alpha} \qquad (1)$$

Where α is the angle between the axis of the shaft 4' and the bisection plane D' of the inclined plate 10', i.e. α is the angle of inclination of the inclined plate 10'.

As has been described previously, the angle of inclination of the inclined plate is varied and, therefore, the distance A defined in the formula (1) varies in accordance with the variations of the angle α. The distance A between the furthest extremities of the shoes is equal to the distance C between the holding portions 35' of the piston rod 20' when the inclined plate is as shown in Figure 3(a). However, the distance C is fixed and this results in a gap being produced between the sliding shoes 19' and the inclined plate 10' when the angle is reduced (see Figure 3(b)), thereby disturbing the smooth motion of the piston 21'.

This contrasts with the sliding shoes 19 shown in Figure 4, which are in accordance with the present invention. They each have a spherically curved surface in addition to a flat surface which abuts against the inclined plate 10. The extent of each spherically curved surface is such that when the shoes 19 are located on either side of the inclined plate 10, these surfaces are coincident with the surface of an imaginary common sphere of radius R centered at a point on the bisection plane D of the inclined plate 10. This ensures that the distance between the extremities of the shoes is not dependent upon the angle of inclination of the inclined plane. Thus, no gap is ever produced between the shoes 19 and the inclined plate 10, and this results in smooth operation of the piston 21 irrespective of its swept volume.

CLAIMS

1. A variable capacity compressor (40) comprising a housing (1,2,3,24,26) having a chamber (1a); a drive shaft (4) supported rotatably in the housing (1,2,3,24,26); an inclined plate (10) coupled to the drive shaft (4); a plurality of pistons (21) arranged to be driven reciprocably by rotation of the inclined plate (10), each piston (21) being connected to the inclined plate (10) by a connecting means (19) which abuts slidably against both faces of the inclined plate (10); a hinge mechanism (8,8a,8b,10a,10b,11) for coupling the inclined plate (10) to the drive shaft (4) and for permitting the angle of inclination of the inclined plate (10) to be varied within a predetermined range; and a means (34) for controlling the pressure in the chamber (1a), the arrangement of the hinge mechanism (8,8a,8b,10a,10b,11) being such that variations in the chamber pressure force the mechanism to move and vary the angle of inclination of the inclined plate (10), and wherein each connecting means (19) comprises a pair of sliding shoes (19), each shoe (19) being mounted in a respective holding portion (35) of the respective piston (21) and having a first surface in sliding contact with the respective face of the inclined plate (10) and a second, spherically curved, surface in sliding abutment with a complementary surface on the respective holding portion (35), the second surfaces of the two shoes (19) being coincident with the surface of a common notional sphere.

2. A compressor (40) according to claim 1, wherein the controlling means (34) is a bellows (34).

3. A compressor (40) according to claim 1 or claim 2, wherein the notional sphere has its centre located

halfway between the two faces of the inclined plate (10).

Fig 1.

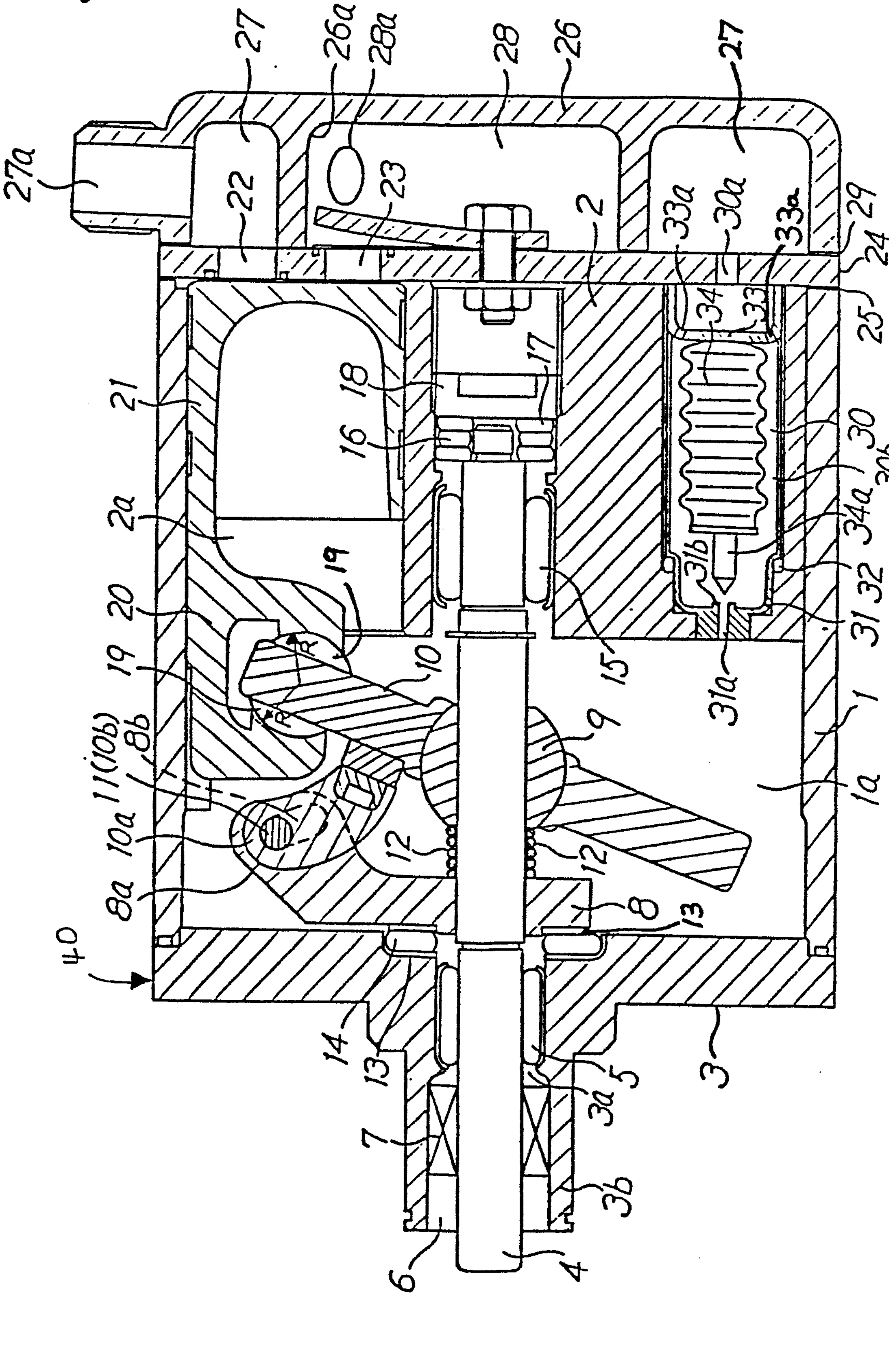

27a
27
22
26a
28a
23
28
26
2
27
33a
30a
33a
29
24
25
34
33
21
18
17
16
30
30b
34a
2a
31b
32
31
20
19
15
31a
10
19
9
1a
1
11 (10b)
8b
10a
12
12
8a
8
13
40
14
13
5
3
3a
7
3b
6
4

Fig 2.

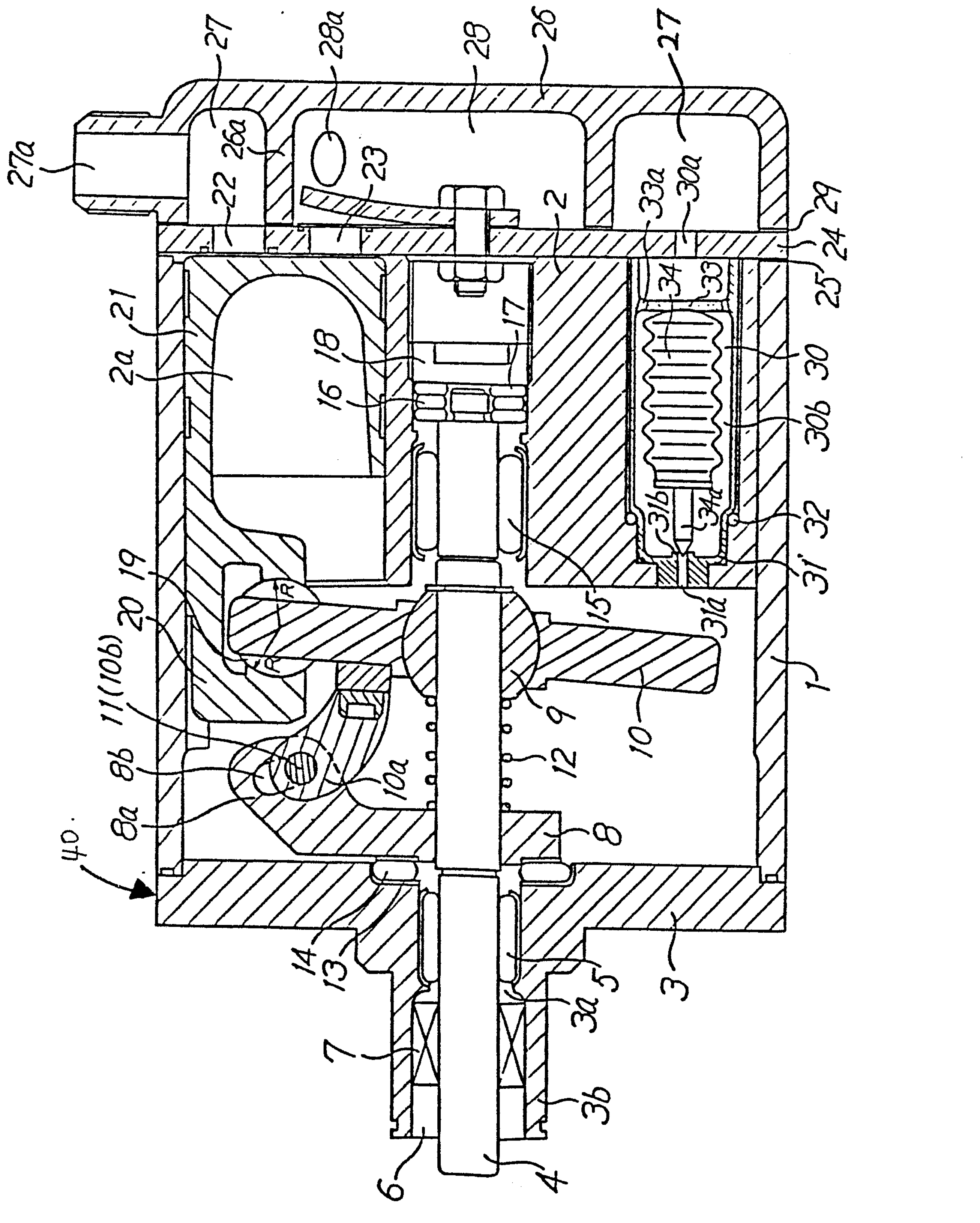

27
28a
28
26
27
27a
26a
23
22
23
2
33a
30a
29
24
25
33
34
21
2a
18
17
16
30
30b
31b
34a
32
19
20
15
31
31a
11(10b)
9
10
8b
12
10a
8a
8
40
14
13
5
3a
3
7
3b
6
4

Fig 3(a).

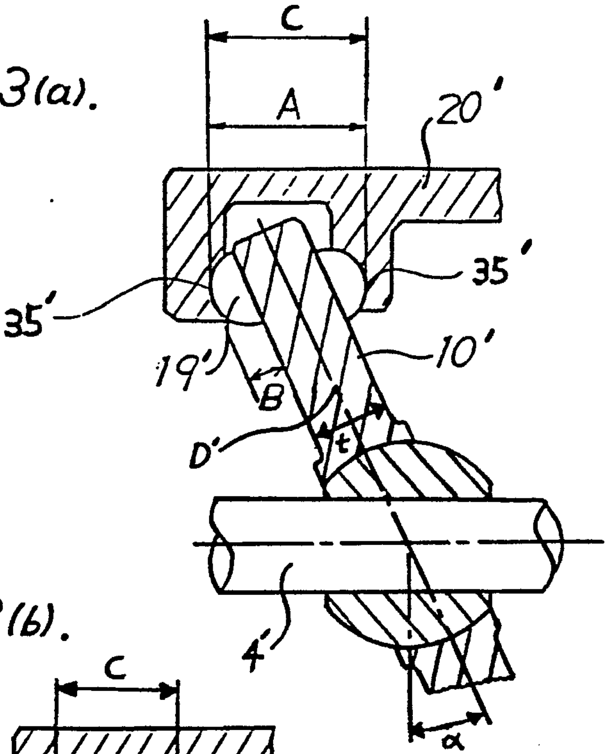


Fig 3(b).

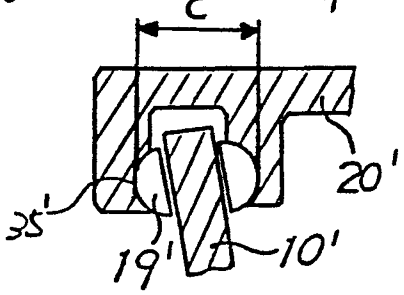


Fig 4.