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- **YOSHIDA, Hirofumi**
Osaka 540-6207 (JP)
- **OGATA, Takeshi**
Osaka 540-6207 (JP)
- **SHIOTANI, Yu**
Osaka 540-6207 (JP)
- **NAKAI, Hiroaki**
Osaka 540-6207 (JP)
- **KARINO, Tsuyoshi**
Osaka 540-6207 (JP)

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(71) Applicant: **Panasonic Intellectual Property Management Co., Ltd.**
Osaka-shi, Osaka 540-6207 (JP)

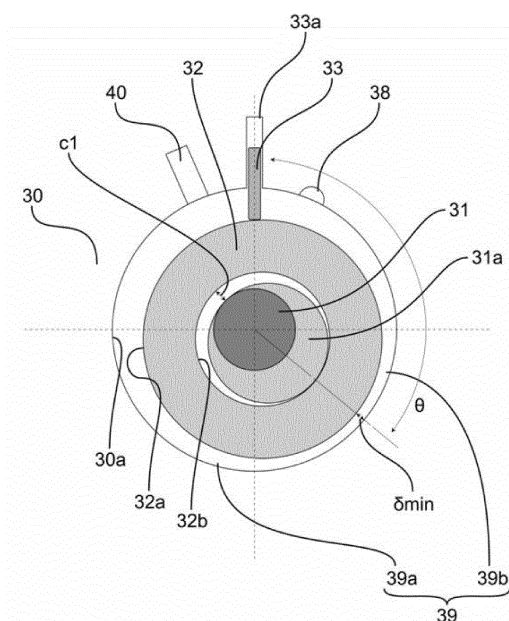
(74) Representative: **Eisenführ Speiser**
Patentanwälte Rechtsanwälte PartGmbB
Postfach 31 02 60
80102 München (DE)

(72) Inventors:
• **FUNAKOSHI, Daisuke**
Osaka 540-6207 (JP)

(54) **ROTARY COMPRESSOR**

(57) If a gap formed between an outer peripheral surface of a piston and an inner peripheral surface of a cylinder in a state where an eccentric portion is disposed at a position of a predetermined crank angle from a position of a vane and the piston is made to abut against a most eccentric position of the eccentric portion and an inner peripheral surface of an upper bearing is made to abut against a main shaft outer peripheral surface of the crankshaft when a rotary compressor is assembled is defined as δ , a minimum value $\delta_{\min}$ of the gap δ is set at a crank angle substantially opposite from a maximum load direction of the crankshaft during operation of the rotary compressor.

[Fig. 3]



Description

[TECHNICAL FIELD]

[0001] The present invention relates to a rotary compressor used for an air conditioner, a freezing machine, a blower, a water heater and the like.

[BACKGROUND TECHNIQUE]

[0002] Conventionally, a compressor is used in a freezing machine and an air conditioner. The compressor sucks gas refrigerant which is evaporated by an evaporator, compresses the gas refrigerant to pressure which is required for condensation, and discharge high temperature and high pressure refrigerant into a refrigerant circuit. A rotary compressor is known as one of such compressors.

[0003] Fig. 18 is a sectional view of essential portion of a conventional rotary compressor.

[0004] As shown in Fig. 18, in the rotary compressor, a motor (not shown) and a compression mechanism 3 are connected to each other through a crankshaft 31, and they are accommodated in a hermetic container 1. The compression mechanism 3 includes a compression chamber 39, a piston 32 and a vane (not shown). The compression chamber 39 is composed of a cylinder 30, and an upper bearing 34 and a lower bearing 35 which close both end surfaces of the cylinder 30. The piston 32 exists in the compression chamber 39, and is fitted over an eccentric portion 31a of the crankshaft 31 supported by the upper bearing 34 and the lower bearing 35. The vane abuts against a piston outer peripheral surface 32a of the piston 32, follows eccentric rotation of the piston 32 and reciprocates, and partitions an interior of the compression chamber 39 into a low pressure portion and a high pressure portion.

[0005] A suction port 40 opens in the cylinder 30, and gas is sucked through the suction port 40 toward the low pressure portion in the compression chamber 39. A discharge port 38 opens in the upper bearing 34, and gas is discharged from the high pressure portion through the discharge port 38. The low pressure portion is turned and formed into the high pressure portion in the compression chamber 39. The piston 32 is accommodated in the compression chamber 39 which is formed by the upper bearing 34, the lower bearing 35 and the cylinder 30. Upper and lower portions of the cylinder 30 are closed by the upper bearing 34 and the lower bearing 35. The discharge port 38 is formed as a hole penetrating the upper bearing 34. This hole is circular as viewed from above. The discharge port 38 is provided at its upper surface with a discharge valve 36 which opens when the discharge valve 36 receives pressure which is equal to or greater than predetermined pressure. A cup muffler 37 is provided above the upper bearing 34 for canceling noise of discharged gas.

[0006] In the rotary compressor having the above-de-

scribed configuration, on the side of the low pressure portion, if a sliding portion of an outer peripheral surface of the piston 32 passes through the suction port 40 and separates away from the suction port 40, the suction chamber gradually enlarges. Gas is sucked from the suction port 40 into the suction chamber. On the side of the high pressure portion, if the sliding portion of the outer peripheral surface of the piston 32 approaches the discharge port 38, the compression chamber 39 gradually shrinks. When pressure becomes equal to or greater than the predetermined pressure, the discharge valve 36 opens and gas in the compression chamber 39 is discharged from the discharge port 38 into the cup muffler 37.

[0007] In such a rotary compressor, there is concern that the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a strongly come into contact with each other, a problem of seizing or wearing occurs, input increases and efficiency of the compressor is lowered. Therefore, as shown in Fig. 16, an operation-time minimum gap W is provided between the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a. Magnitude of a leakage area S which is obtained by the operation-time minimum gap W and a height H of the compression chamber 39 exerts an influence on efficiency of the compressor.

[0008] Here, if the operation-time minimum gap W is set greater, an amount of compressed fluid which flows from the high pressure portion into the low pressure portion through the operation-time minimum gap W increases. Hence, compressed refrigerant gas leaks from the operation-time minimum gap W, a loss ("leakage loss", hereinafter) increases and thus, the efficiency of the compressor is deteriorated.

[0009] If the operation-time minimum gap W is set smaller on the other hand, although the leakage loss reduces, the piston outer peripheral surface and the cylinder inner peripheral surface strongly come into contact with each other. According to this, since a loss ("sliding loss", hereinafter) increases and thus, the efficiency of the compressor is deteriorated. Further, the piston outer peripheral surface and the cylinder inner peripheral surface strongly slide on each other, a problem of seizing or wearing occurs.

[0010] Therefore, the operation-time minimum gap W between the piston outer peripheral surface and the inner peripheral surface is set greater so that both the surfaces do not strongly come into contact with each other, the problem of seizing or wearing is eliminated and the sliding loss reduces.

[0011] Fig. 17 is a schematic diagram showing a shape of a cylinder having a non-circular (complex circular) cross section in the conventional rotary compressor described patent document 1.

[0012] For example, as shown in Fig. 17, a compression chamber has a non-circular cross section composed of a plurality of curvatures. According to this, even if an envelop locus of a piston outer peripheral surface be-

comes non-circular due to influence of an axial locus or the like, the operation-time minimum gap W while the piston rotates once can be maintained constant. As a result, the leakage loss and the sliding loss are reduced.

[0013] Further, in recent years, it is desired to enhance efficiency of an air conditioner and the like which circulates refrigerant by a compressor. Hence, it is important to further enhance the efficiency of the compressor.

[PRIOR ART DOCUMENT]

[PATENT DOCUMENT]

[0014] [Patent Document 1] Japanese Patent Application Laid-open No.2003-214369

[SUMMARY OF THE INVENTION]

[PROBLEM TO BE SOLVED BY THE INVENTION]

[0015] In the rotary compressor of the above-described conventional structure, however, the cross section shape of the cylinder inner peripheral surface is non-circular composed of the plurality of curvatures, precision on the order of several μm is required, and its machining operation is extremely difficult. Further, machining errors such as surface roughness and undulation of the cylinder inner peripheral surface exert large influence on the efficiency of the compressor, and this causes variation in performance.

[0016] Therefore, the present invention has been accomplished in view of the above circumstances, and it is an object of the invention to reduce a leakage loss from an operation-time minimum gap W from the ground up without deteriorating reliability, and to further enhance efficiency of a compressor without increasing a sliding loss.

[0017] It is another object of the invention to provide an efficient rotary compressor which can be machined easily without depending upon a cross section shape such as machining precision and surface roughness of a cylinder inner peripheral surface.

[MEANS FOR SOLVING THE PROBLEM]

[0018] A first aspect of the invention provides a rotary compressor comprising a motor and a compression mechanism both accommodated in a hermetic container, in which the compression mechanism connected to the motor through a crankshaft comprises a cylinder, an upper bearing and a lower bearing which close, from above and below, both end surfaces of the cylinder to form a compression chamber, a piston fitted over an eccentric portion of the crankshaft provided in the cylinder, a vane which follows eccentric rotation of the piston, which is provided in the cylinder, which reciprocates in a slot, and which partitions the compression chamber into a low pressure portion and a high pressure portion, a suction

port which is in communication with the low pressure portion, and a discharge port which is in communication with the high pressure portion, wherein if a gap formed between an outer peripheral surface of the piston and an inner peripheral surface of the cylinder in a state where the eccentric portion is disposed at a position of a pre-determined crank angle from a position of the vane and the piston is made to abut against a most eccentric position of the eccentric portion and an inner peripheral surface of the upper bearing is made to abut against a main shaft outer peripheral surface of the crankshaft when the rotary compressor is assembled is defined as δ , a minimum value δ_{min} of the gap δ is set at a crank angle substantially opposite from a maximum load direction of the crankshaft during operation of the rotary compressor.

[0019] In a second aspect of the invention, in the rotary compressor of the first aspect, when the rotary compressor is assembled, a first bearing gap is formed between the piston and the eccentric portion, a second bearing gap is formed between the upper bearing and the main shaft, in each of the crank angles, the crankshaft is moved by the first bearing gap in a load direction at a time of operation, the piston is moved by the second bearing gap in the load direction at the time of operation, when a minimum gap formed between an outer periphery of the piston and a phantom line of an inner periphery of the cylinder is defined as β , a direction of the minimum value δ_{min} is set such that a minimum gap β near a crank angle 45° and a minimum gap β near a crank angle 225° are substantially equal to each other.

[0020] In a third aspect of the invention, in the rotary compressor of the first or second aspect, the rotary compressor further includes one more compression chamber.

[0021] In a fourth aspect of the invention, in the rotary compressor of any one of the first to third aspects, the minimum value δ_{min} is about $5\ \mu\text{m}$ to $10\ \mu\text{m}$.

[EFFECT OF THE INVENTION]

[0022] Generally, when a compressor is operated, a crankshaft moves in a maximum load direction, and an operation-time minimum gap W increases at a crank angle opposite from the maximum load direction. According to the present invention, since a minimum gap δ_{min} is previously set at the crank angle opposite from the maximum load direction, the operation-time minimum gap W becomes small, leakage can be reduced, and it is possible to enhance the efficiency. Hence, it is possible to reduce the operation-time minimum gap W and a leakage loss without increasing a sliding loss, and efficiency of the compressor can further be enhanced.

[BRIEF DESCRIPTION OF THE DRAWINGS]

[0023]

Fig. 1 is a vertical sectional view of a rotary compres-

sor according to an embodiment of the present invention;

Fig. 2 is a sectional view of essential portions showing a relation between a piston of the rotary compressor and a gap of a crankshaft when the rotary compressor is assembled;

Fig. 3 is a plan view of essential portions showing a compression chamber of the rotary compressor when the rotary compressor is assembled;

Fig. 4 is a plan view of essential portions showing disposition of an upper bearing in Fig. 3;

Fig. 5 is a sectional view taken along line V-V in Fig. 4;

Fig. 6 is a plan view of essential portions showing the compression chamber of the rotary compressor when the rotary compressor is operated;

Fig. 7 is a sectional view showing gaps when the rotary compressor is operated;

Fig. 8 is a diagram showing magnitude and a direction of a load of a crankshaft in a one-piston rotary compressor;

Fig. 9 is a diagram showing a locus of a piston outer peripheral surface in the one-piston rotary compressor in which a minimum gap $\delta_{\min}$ is a general angle;

Fig. 10 is a diagram showing a locus of the piston outer peripheral surface in the one-piston rotary compressor when a minimum gap $\delta_{\min}$ direction is set such that a minimum gap β in the vicinity of 45° and a minimum gap β in the vicinity of 225° become equal to each other;

Fig. 11 is a diagram showing magnitude and a direction of a load of a crankshaft in a two-piston rotary compressor;

Fig. 12 is a diagram showing a locus of a piston outer peripheral surface in the two-piston rotary compressor in which a minimum gap $\delta_{\min}$ is a general angle;

Fig. 13 is a diagram showing a locus of the piston outer peripheral surface in the two-piston rotary compressor when a minimum gap $\delta_{\min}$ direction is set such that a minimum gap β in the vicinity of 45° and a minimum gap β in the vicinity of 225° become equal to each other;

Fig. 14 is a diagram showing a locus of the piston outer peripheral surface in the two-piston rotary compressor when the minimum gap $\delta_{\min}$ is a general angle and the minimum gap $\delta_{\min}$ is reduced to about 5 to 10 μm ;

Fig. 15 is a diagram showing a locus of the piston outer peripheral surface in the two-piston rotary compressor when the minimum gap $\delta_{\min}$ direction is set such that a minimum gap β in the vicinity of 45° and a minimum gap β in the vicinity of 225° become equal to each other and the minimum gap $\delta_{\min}$ is reduced to about 5 to 10 μm ;

Fig. 16 is a schematic diagram showing a leakage area S;

Fig. 17 is a schematic diagram showing a shape of a cylinder having a non-circular (complex circular) cross section in a conventional rotary compressor;

and

Fig. 18 is a sectional view of essential portions of a conventional rotary compressor.

5 [EXPLANATION OF SYMBOLS]

[0024]

1	hermetic container
2	motor
3	compression mechanism
4	upper shell
5	refrigerant discharge pipe
22	stator
24	rotor
26	air gap
28	notch
30	cylinder
30a	cylinder inner peripheral surface
31	crankshaft
31a	eccentric portion
31b	eccentric portion outer peripheral surface
31c	main shaft
32	piston
32a	piston outer peripheral surface
32b	piston inner peripheral surface
33	vane
34	upper bearing
34a	inner peripheral surface
35	lower bearing
36	discharge valve
37	cup muffler
38	discharge port
39	compression chamber
40	suction port

[MODE FOR CARRYING OUT THE INVENTION]

[0025] According to a rotary compressor of a first aspect of the present invention, if a gap formed between an outer peripheral surface of a piston and an inner peripheral surface of a cylinder in a state where an eccentric portion of a crankshaft is disposed at a position of a predetermined crank angle from a position of a vane and the piston is made to abut against a most eccentric position of the eccentric portion of the crankshaft and an inner peripheral surface of an upper bearing is made to abut against an outer peripheral surface of the crankshaft when the rotary compressor is assembled is defined as δ , a minimum value $\delta_{\min}$ of the gap δ is set at a crank angle substantially opposite from a maximum load direction of the crankshaft during operation of the rotary compressor. Generally, when the rotary compressor is operated, since the crankshaft moves in the maximum load direction, the operation-time minimum gap W becomes large at a crank angle opposite from the maximum load direction. According to the first aspect, since the minimum gap $\delta_{\min}$ is previously set at the crank angle op-

posite from the maximum load direction, the operation-time minimum gap W becomes small. Therefore, leakage can be reduced and efficiency can be enhanced.

[0026] In a second aspect of the invention, in the rotary compressor of the first aspect, when the rotary compressor is assembled, a first bearing gap is formed between the piston and the eccentric portion of the crankshaft, a second bearing gap is formed between the upper bearing and the main shaft of the crankshaft, in each of the crank angles, the crankshaft is moved by the first bearing gap in a load direction at a time of operation, the piston is moved by the second bearing gap in the load direction at the time of operation, when a minimum gap formed between an outer periphery of the piston and a phantom line of an inner periphery of the cylinder is defined as β , a direction of the minimum value $\delta_{\min}$ is set such that a minimum gap β near a crank angle 45° and a minimum gap β near a crank angle 225° are substantially equal to each other. According to the second aspect, the operation-time minimum gap W in the vicinity of the crank angle 45° and the operation-time minimum gap W in the vicinity of the crank angle 225° becomes substantially equal to each other, phantom lines in the load direction of the crankshaft become symmetric, the gaps are balanced and thus, a large sliding loss is generated. Therefore, leakage from the operation-time minimum gap W is reduced and efficiency can be enhanced while suppressing deterioration of reliability such as wearing and seizing.

[0027] In a third aspect of the invention, in the rotary compressor of the first or second aspect, the rotary compressor further includes one more compression chamber. According to the third aspect, in the case of a two-piston rotary, a load direction is substantially constant and a load becomes greater as compared with a one-piston rotary. Therefore, it is possible to reduce leakage from the operation-time minimum gap W and to enhance the efficiency while further suppressing deterioration of reliability such as wearing and seizing.

[0028] In a fourth aspect of the invention, in the rotary compressor of any one of the first to third aspects, the minimum value $\delta_{\min}$ is about $5\ \mu\text{m}$ to $10\ \mu\text{m}$. According to the fourth aspect, the phantom lines in the load direction of the crankshaft become symmetric, and the gaps are balanced. Hence, even if the minimum gap $\delta_{\min}$ is excessively reduced, a large sliding loss is not generated in the vicinity of the crank angle 45° and the crank angle 225° when the rotary compressor is operated. Therefore, it is possible to reduce leakage from the operation-time minimum gap W and to enhance the efficiency while suppressing deterioration of reliability such as wearing and seizing.

[0029] An embodiment of the present invention will be described with reference to the drawings. The invention is not limited to the embodiment.

[0030] Fig. 1 is a vertical sectional view of a rotary compressor according to an embodiment of the invention, and Fig. 6 is a plan view of essential portions showing a compression chamber of the rotary compressor when

the rotary compressor is operated.

[0031] In the drawings, according to the rotary compressor of the embodiment, a motor 2 and a compression mechanism 3 are accommodated in a hermetic container 1. The motor 2 and the compression mechanism 3 are connected to each other through a crankshaft 31. The motor 2 is composed of a stator 22 and a rotor 24. The compression mechanism 3 is composed of a cylinder 30, a piston 32, a vane 33, an upper bearing 34 and a lower bearing 35.

[0032] The compression chamber 39 is formed by the cylinder 30, and an upper bearing 34 and a lower bearing 35 which close both end surfaces of the cylinder 30. The piston 32 is accommodated in the compression chamber 39, and the piston 32 is fitted over an eccentric portion 31a of the crankshaft 31 which is supported by the upper bearing 34 and the lower bearing 35. The vane 33 reciprocates in a slot 33a provided in the cylinder 30 and always abuts against an outer peripheral surface 32a, thereby partitioning an interior of the compression chamber 39 into a low pressure portion 39a and a high pressure portion 39b. Two spaces are formed in the compression chamber 39 by the vane 33 and an operation-time minimum gap W . A space connected to a suction port 40 is the low pressure portion 39a, and a space connected to the discharge port 38 is the high pressure portion 39b. Here, the operation-time minimum gap W is an operation-time gap generated at a position where the piston 32 most approaches the cylinder 30.

[0033] The suction port 40 opens in the cylinder 30, and the suction port 40 sucks (supplies) refrigerant gas to the low pressure portion 39a in the compression chamber 39. The discharge port 38 opens in the upper bearing 34, and discharges gas from the high pressure portion 39b. The discharge port 38 is formed as a circular hole which penetrates the upper bearing 34. An upper surface of the discharge port 38 is provided with a discharge valve 36, and when the discharge valve 36 receives pressure which is equal to or greater than a predetermined value, the discharge valve 36 is opened. The discharge valve 36 is covered with a cup muffler 37.

[0034] As the operation-time minimum gap W separates away from the suction port 40, a volume of the low pressure portion 39a of the compression mechanism 3 gradually increases. By the increase in volume, refrigerant gas flows in from the suction port 40. The low pressure portion 39a moves while changing its volume by eccentric rotation of the piston 32, and if change in volume is turned from increase to reduction, the low pressure portion 39a becomes the high pressure portion 39b.

[0035] On the other hand, as the operation-time minimum gap W approaches the discharge port 38, the volume of the high pressure portion 39b gradually reduces, and pressure therein is increased by the reduction in volume. When the high pressure portion 39b is compressed to a predetermined pressure or more, the discharge valve 36 opens and high pressure refrigerant gas flows out from the discharge port 38.

[0036] Refrigerant gas is discharged into the hermetic container 1 by the cup muffler 37. The refrigerant gas passes through a notch 28 formed by the stator 22 and an inner periphery of the hermetic container 1 and through an air gap 26 of the motor 2, and the refrigerant gas is sent into an upper shell 4 of an upper portion of the motor 2. The refrigerant gas is discharged from the refrigerant discharge pipe 5 to outside of the hermetic container 1. Arrows in Fig. 1 show a flow of refrigerant.

[0037] There is a space 46 between an upper end surface of the eccentric portion 31a, the upper bearing 34 and an inner peripheral surface of the piston 32. There is a space 47 between a lower end surface of the eccentric portion 31a, the lower bearing 35 and the inner peripheral surface of the piston 32. Oil leaks into the spaces 46 and 47 from an oil hole 41 through oil-feeding holes 42 and 43. Pressure in each of the spaces 46 and 47 is almost always higher than pressure in the compression chamber 39.

[0038] A height of the cylinder 30 must be set slightly higher than a height of the piston 32 so that the piston 32 can slide in the cylinder 30. As a result, there is a gap between an end surface of the piston 32 and an end surface of the upper bearing 34, and between the end surface of the piston 32 and an end surface of the lower bearing 35. Hence, oil leaks from the spaces 46 and 47 into the compression chamber 39 through this gap.

[0039] Fig. 2 is a sectional view of essential portions showing a relation between the piston of the rotary compressor of the embodiment and a gap of the crankshaft when the rotary compressor is assembled, Fig. 3 is a plan view of essential portions showing the compression chamber of the rotary compressor when the rotary compressor is assembled, Fig. 4 is a plan view of essential portions showing disposition of the upper bearing in Fig. 3, and Fig. 5 is a sectional view taken along line V-V in Fig. 4.

[0040] In the rotary compressor of the invention, a gap between the piston inner peripheral surface 32b of the piston 32 and an eccentric portion outer peripheral surface 31b of the eccentric portion 31a of the crankshaft 31 is defined as a first bearing gap c1 as shown in Figs. 2 and 3. At this time, when the rotary compressor is assembled, the crankshaft 31 is disposed such that the eccentric portion 31a becomes equal to an angle θ from the vane 33 as shown in Fig. 3. The angle θ is an angle on a side substantially opposite from the maximum load direction of the crankshaft 31. Further, the crankshaft 31 is disposed such that a later-described minimum gap $\delta_{\min}$ is disposed such that the minimum gap $\delta_{\min}$ is closer to the discharge port 38 than a phantom line connecting the vane 33 and a center of the crankshaft 31 to each other. In a state where the eccentric portion 31a is disposed at the position of the angle θ , the piston 32 is brought into abutment against a most eccentric position of the eccentric portion 31a. As a result, a minimum gap $\delta_{\min}$ is formed between the piston outer peripheral sur-

face 32a and the cylinder inner peripheral surface 30a at the position of the angle θ . The first bearing gap c1 is formed between the piston inner peripheral surface 32b and the eccentric portion outer peripheral surface 31b at the position of the angle θ .

[0041] In a state where the disposition shown in Fig. 3 is maintained, the upper bearing 34 is disposed as shown in Fig. 4.

[0042] That is, by bringing the upper bearing 34 into abutment against a main shaft 31c (most non-eccentric position of the eccentric portion 31a) of the crankshaft 31 in a direction separated away from the vane 33 by the angle θ , a second bearing gap c2 is formed between an inner peripheral surface 34a of the upper bearing 34 and the main shaft 31c of the crankshaft 31.

[0043] By the above-described assembly, the minimum gap $\delta_{\min}$, the first bearing gap c1 and the second bearing gap c2 are disposed on a phantom line separated away from the vane 33 by the angle θ .

[0044] Fig. 5 shows a state where the minimum gap $\delta_{\min}$, the first bearing gap c1 and the second bearing gap c2 are disposed.

[0045] Generally, in a rotary compressor, there is concern that if the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a strongly come into contact with each other, a problem of seizing or wearing occurs.

[0046] Hence, the operation-time minimum gap W is provided between the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a as shown in Fig. 16. Magnitude of a leakage area S obtained by the operation-time minimum gap W and a height H of the compression chamber 39 exerts influence on efficiency of the compressor.

[0047] For example, if the operation-time minimum gap W is set large, an amount of compressed fluid which flows out from the high pressure portion to the low pressure portion through the operation-time minimum gap W is increased. Therefore, since the compressed refrigerant gas leaks from the operation-time minimum gap W and a leakage loss increases and thus, efficiency of the compressor is deteriorated.

[0048] If the operation-time minimum gap W is set small on the other hand, although the leakage loss reduces, the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a strongly come into contact with each other. According to this, a sliding loss increases and thus, efficiency of the compressor is deteriorated. Further, the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a strongly slide on each other, a problem of seizing or wearing occurs.

[0049] An operation state of the compression mechanism assembled as described above will be described using Figs. 6 and 7.

[0050] First, a relation between the minimum gap $\delta_{\min}$ and the operation-time minimum gap W when the compression mechanism is operated will be described using

Fig. 6.

[0051] As described above, when the compression mechanism is assembled, the minimum gap $\delta_{\min}$ is formed between the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a.

[0052] When the compression mechanism is operated, a pressure difference X is added to the piston 32 as shown by an arrow in Fig. 6. Since the low pressure portion 39a and the high pressure portion 39b are formed in the compression chamber 39, the pressure difference X is applied from the high pressure portion 39b toward the low pressure portion 39a. The piston 32 is pushed toward the low pressure portion 39a by the pressure difference X and the piston 32 is displaced. Hence, when the compression mechanism is operated, the operation-time minimum gap W is not formed at a position of the minimum gap $\delta_{\min}$ which is set when the compression mechanism is assembled, a position of an angle $(\theta + \alpha)$ becomes the operation-time minimum gap W where the piston outer peripheral surface 32a and the cylinder inner peripheral surface 30a most approach each other. The operation-time minimum gap W becomes a gap which is narrower than the minimum gap $\delta_{\min}$ (α is a minute angle which is varied depending upon an operation state).

[0053] Next, a relation between the operation-time minimum gap W, the first bearing gap c1 and the second bearing gap c2 when the compression mechanism is operated will be described using Fig. 7.

[0054] As shown in Fig. 7, when the compression mechanism is operated, the eccentric portion 31a of the crankshaft 31 located inside of the piston 32 and the crankshaft 31 located inside of the upper bearing 34 move to a center by oil film pressure. Therefore, the minimum gap $\delta_{\min}$ which is set when the compression mechanism is assembled becomes narrow by 1/2 of the first bearing gap c1 and by 1/2 of the second bearing gap c2 when the compression mechanism is operated. According to this, the operation-time minimum gap W which is theoretically close to zero is formed, and the compression mechanism is operated with a gap size of only oil film size in practice.

[0055] Generally, when the compression mechanism is operated, since the crankshaft 31 moves in the maximum load direction, the operation-time minimum gap W becomes large at a crank angle on the opposite side from the maximum load direction. According to this embodiment, since the minimum gap $\delta_{\min}$ is previously set at the crank angle on the opposite side from the maximum load direction, it is possible to keep the operation-time minimum gap W small at the crank angle on the opposite side from the maximum load direction, and leakage is reduced. Further, the operation-time minimum gap W does not become small also at other crank angles, input is not increased and efficiency can be enhanced.

[0056] Here, Fig. 8 shows magnitude and a direction of a load at each of crank angles which is applied to the crankshaft 31 of a one-piston rotary compressor during one rotation (a direction of the vane is a plus side of y

axis, and a direction of suction is a minus side of x axis and a plus side of y axis). As shown in the drawing, a load becomes the maximum in the vicinity of a crank angle 225° .

[0057] Figs. 9 and 10 show, at each of crank angles, a relation between a locus of the piston outer peripheral surface 32a and a position of the cylinder inner peripheral surface 30a when the crankshaft 31 moves by the second bearing gap c2 in the load direction at the time of operation and the piston 32 moves by the first bearing gap c1 in the load direction at the time of operation assuming that the cylinder 30 does not exist (at each of crank angles, a minimum gap formed between the piston outer peripheral surface 32a and a phantom line of the cylinder inner peripheral surface 30a is defined as β . If a gap when the piston outer peripheral surface 32a spreads outward more than the cylinder inner peripheral surface 30a is defined as substantially zero (oil film holding), the minimum gap β becomes substantially equal to the operation-time minimum gap W). In Fig. 9, a direction of the minimum gap $\delta_{\min}$ is set to a general direction. In Fig. 10, a direction of the minimum gap $\delta_{\min}$ is set such that the minimum gap β in the vicinity of a crank angle 45° and the minimum gap β in the vicinity of a crank angle 225° becomes substantially equal to each other. If Figs. 9 and 10 are compared with each other, a portion of the piston outer peripheral surface 32a which spreads outward more than the cylinder inner peripheral surface 30a is held by an oil film, operation is actually carried out along the cylinder inner peripheral surface 30a. However, a length of a sliding portion in Fig. 10 is apparently shorter, and increase in a sliding loss can be suppressed as small as possible. Hence, the minimum gap β can be uniformed in a wide range of a crank angle, the leakage loss can be reduced and the efficiency can be enhanced.

[0058] Fig. 11 shows magnitude and a direction of a load in each of the crank angles which is applied to the crankshaft 31 of a two-piston rotary compressor (not shown) during one rotation. As shown in Fig. 11, the load is the maximum in the vicinity of a crank angle 225° .

[0059] Figs. 12 and 13 show, at each of crank angles, positional relations between a locus of the piston outer peripheral surface 32a and a phantom line of the cylinder inner peripheral surface 30a when the crankshaft 31 moves by the second bearing gap c2 in the load direction at the time of operation and the piston 32 moves by the first bearing gap c1 in the load direction at the time of operation assuming that the cylinder 30 does not exist (only cylinder 30 on one side is shown). In Fig. 12, a direction of the minimum gap $\delta_{\min}$ is set to a general direction. In Fig. 13, a direction of the minimum gap $\delta_{\min}$ is set such that the minimum gap β in the vicinity of a crank angle 45° and the minimum gap β in the vicinity of a crank angle 225° become substantially equal to each other. If Figs. 12 and 13 are compared with each other, a portion of the piston outer peripheral surface 32a which spreads outward more than the cylinder inner peripheral surface 30a is held by an oil film, operation is actually

carried out along the cylinder inner peripheral surface 30a. However, a length of the sliding portion in Fig. 13 is apparently shorter, and increase in a sliding loss can be suppressed as small as possible. Hence, the minimum gap β can be uniformed in a wide range of the crank angle, the leakage loss can be reduced and the efficiency can be enhanced. If this is compared with the one-piston rotary, a direction of a bearing load is substantially constant, the minimum gap β in the vicinity of a crank angle 45° and the minimum gap β in the vicinity of a crank angle 225° can be uniformed while keeping excellent balance and thus, efficiency can further be enhanced.

[0060] Fig. 14 shows a state where a direction of the minimum gap $\delta_{\min}$ is set to a general direction and the minimum gap $\delta_{\min}$ is extremely reduced as small as 5 to 10 μm . Fig. 15 shows a state where a direction of the minimum gap $\delta_{\min}$ is set such that the minimum gap β in the vicinity of a crank angle 45° and the minimum gap β in the vicinity of a crank angle 225° become substantially equal to each other and the minimum gap $\delta_{\min}$ is extremely reduced as small as 5 to 10 μm . If Figs. 14 and 15 are compared with each other, a length of the sliding portion in Fig. 14 is largely increased, but the minimum gap β is more uniform in Fig. 15 over its entire circumference. In Fig. 14, although the minimum gap $\delta_{\min}$ is made small, the minimum gap β is not made small and thus, volume efficiency is not enhanced and only input increases. In Fig. 15, input does not increase so much and volume efficiency is largely enhanced. Generally, it is considered that if the minimum gap $\delta_{\min}$ is made small, volume efficiency is enhanced, but its limit value is about 10 μm . If the minimum gap $\delta_{\min}$ is set to a direction opposite from the maximum load direction of the crankshaft 31 as in this embodiment, efficiency can further be enhanced even if the minimum gap $\delta_{\min}$ is set to 10 μm or less (compare Fig. 13 and 15).

[INDUSTRIAL APPLICABILITY]

[0061] As described above, according to the rotary compressor of the present invention, it is possible to suppress deterioration of reliability such as wearing and seizing, to reduce both leakage loss and sliding loss, and to enhance the efficiency of the compressor. According to this, the invention can also be applied to a compressor for an air conditioner using HFC-based refrigerant and HCFC-based refrigerant, and to an air conditioner and a heat pump water heater using carbon dioxide which is natural refrigerant.

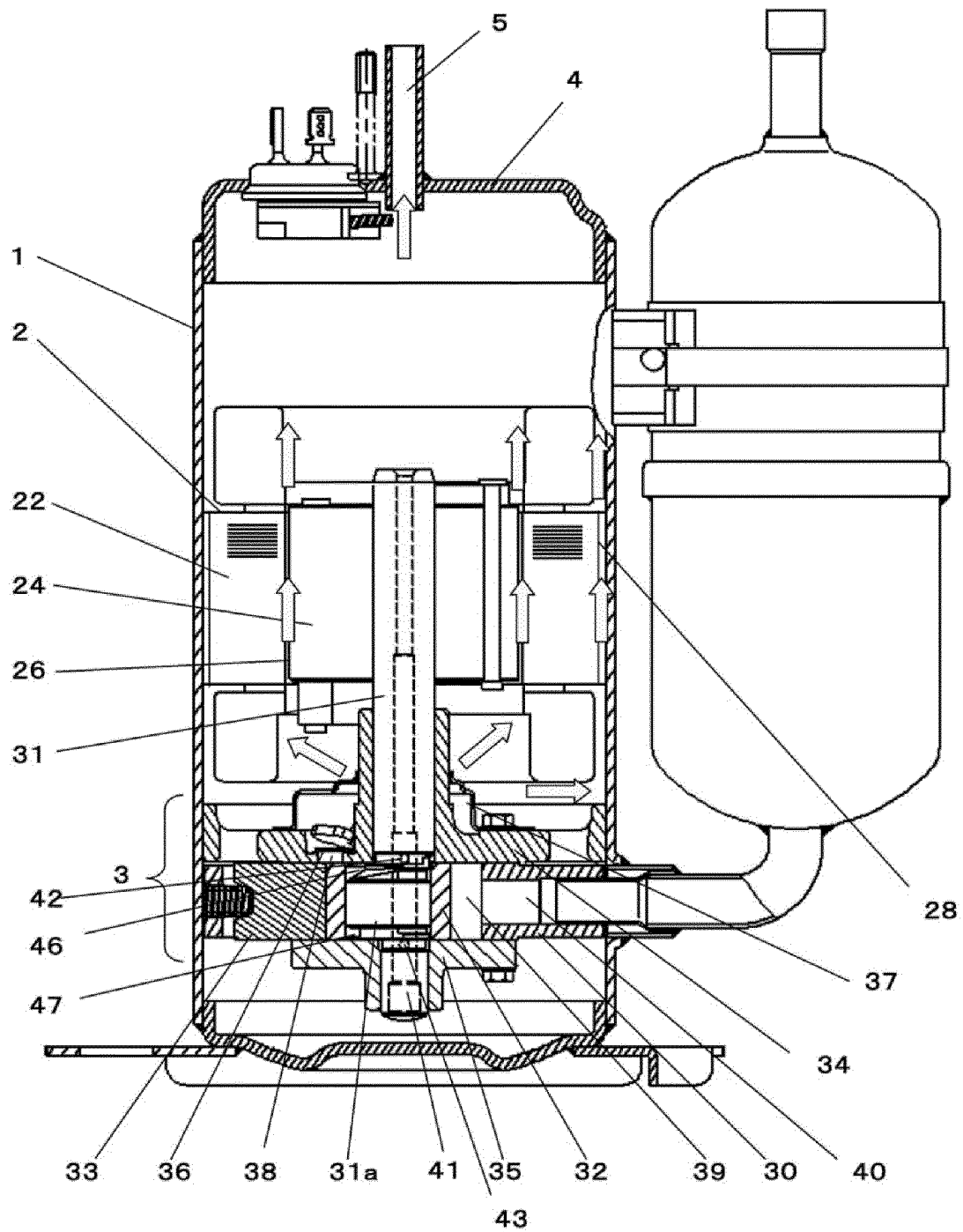
Claims

1. A rotary compressor comprising a motor and a compression mechanism both accommodated in a hermetic container, in which the compression mechanism connected to the motor through a crankshaft comprises

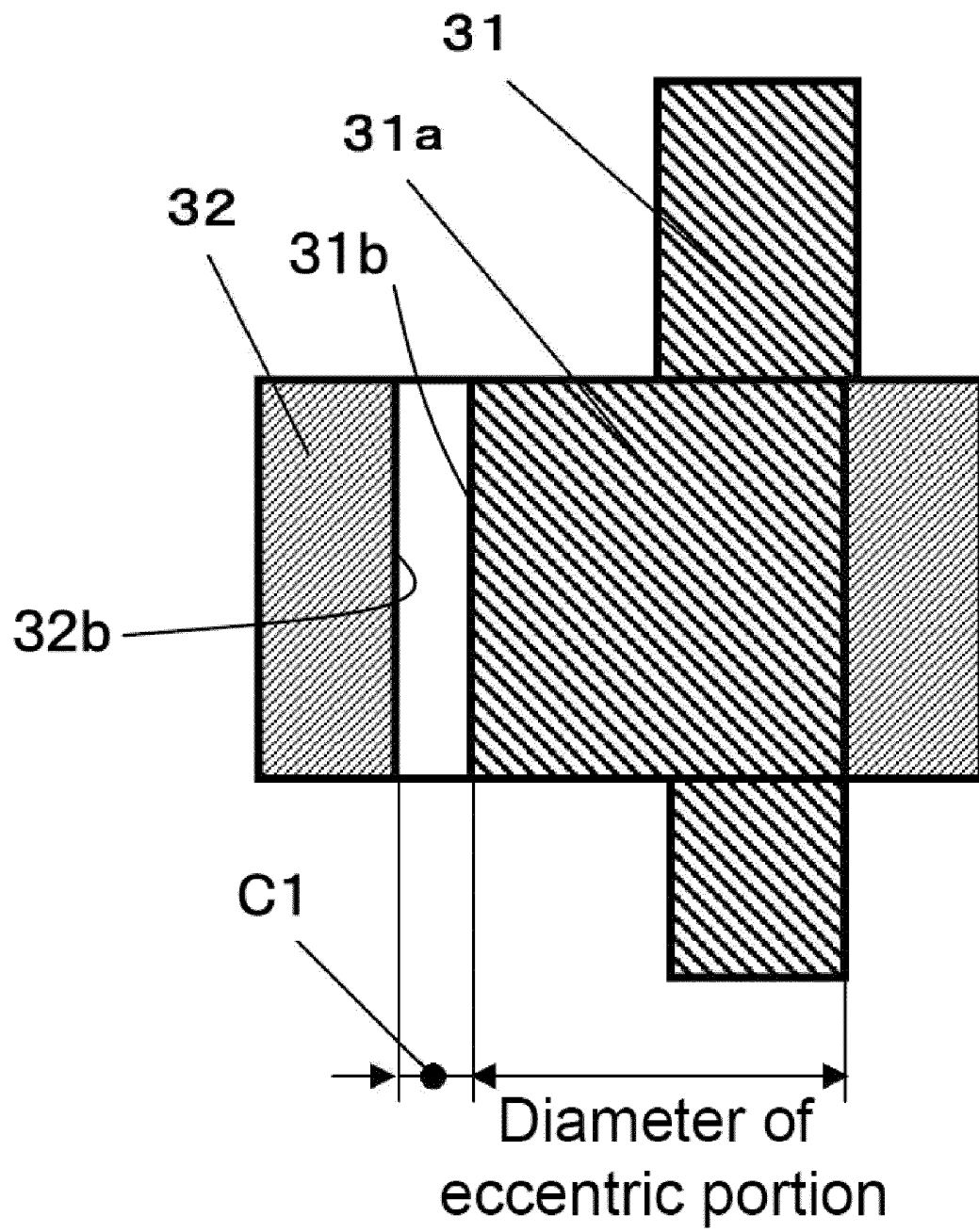
a cylinder,
an upper bearing and a lower bearing which close, from above and below, both end surfaces of the cylinder to form a compression chamber,
a piston fitted over an eccentric portion of the crankshaft provided in the cylinder,
a vane which follows eccentric rotation of the piston, which is provided in the cylinder, which reciprocates in a slot, and which partitions the compression chamber into a low pressure portion and a high pressure portion,
a suction port which is in communication with the low pressure portion, and
a discharge port which is in communication with the high pressure portion, wherein
if a gap formed between an outer peripheral surface of the piston and an inner peripheral surface of the cylinder in a state where the eccentric portion is disposed at a position of a predetermined crank angle from a position of the vane and the piston is made to abut against a most eccentric position of the eccentric portion and an inner peripheral surface of the upper bearing is made to abut against a main shaft outer peripheral surface of the crankshaft when the rotary compressor is assembled is defined as δ ,
a minimum value $\delta_{\min}$ of the gap δ is set at a crank angle on a side substantially opposite from a maximum load direction of the crankshaft during operation of the rotary compressor.

2. The rotary compressor according to claim 1, wherein when the rotary compressor is assembled,
a first bearing gap is formed between the piston and the eccentric portion,
a second bearing gap is formed between the upper bearing and the main shaft,
in each of the crank angles,
the crankshaft is moved by the first bearing gap in a load direction at a time of operation,
the piston is moved by the second bearing gap in the load direction at the time of operation, when a minimum gap formed between an outer periphery of the piston and a phantom line of an inner periphery of the cylinder is defined as β ,
a direction of the minimum value $\delta_{\min}$ is set such that a minimum gap β near a crank angle 45° and a minimum gap β near a crank angle 225° are substantially equal to each other.
3. The rotary compressor according to claim 1 or 2, further comprising one more compression chamber.
4. The rotary compressor according to any one of claims 1 to 3, wherein the minimum value $\delta_{\min}$ is about 5 μm to 10 μm .

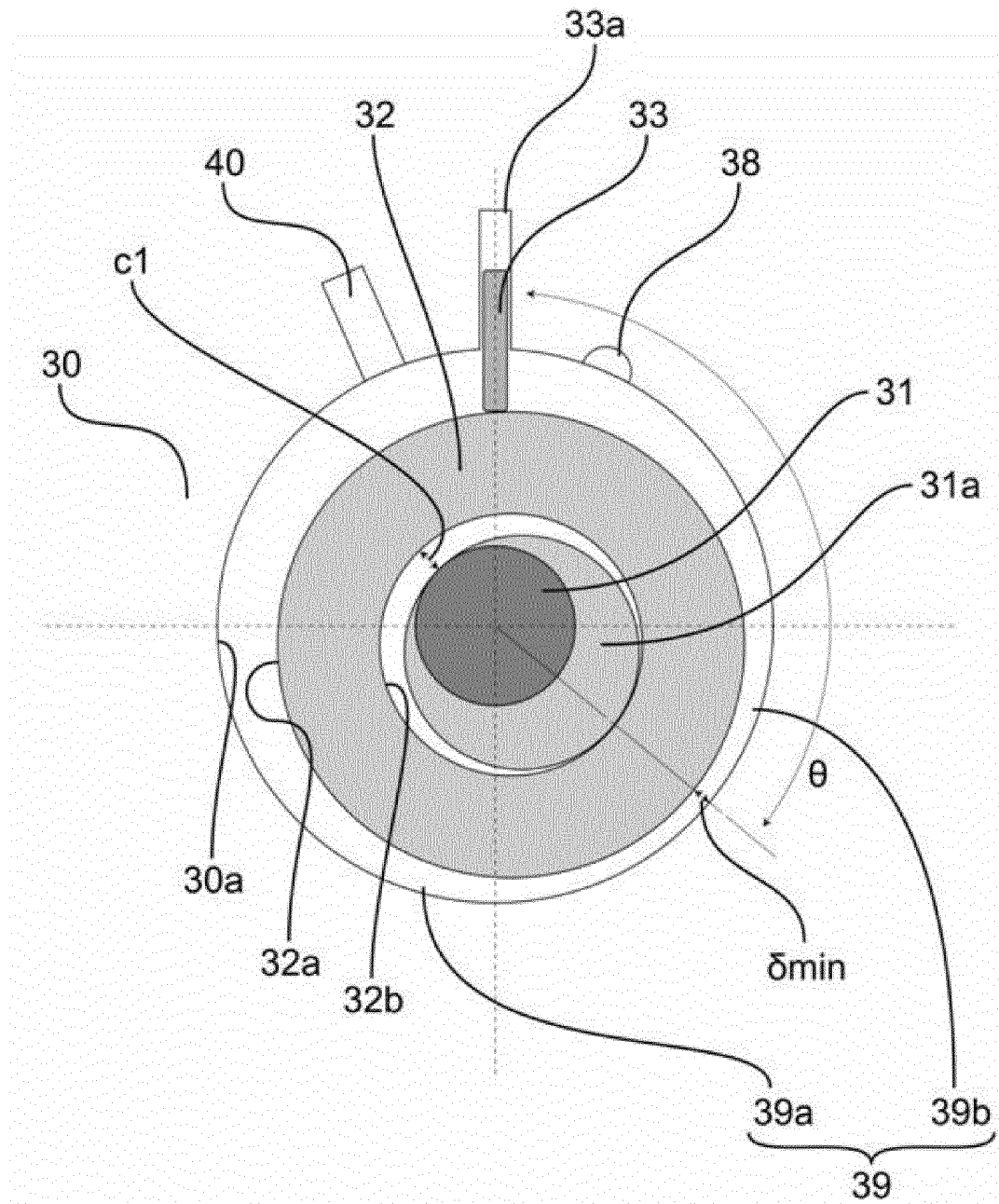
[Fig. 1]



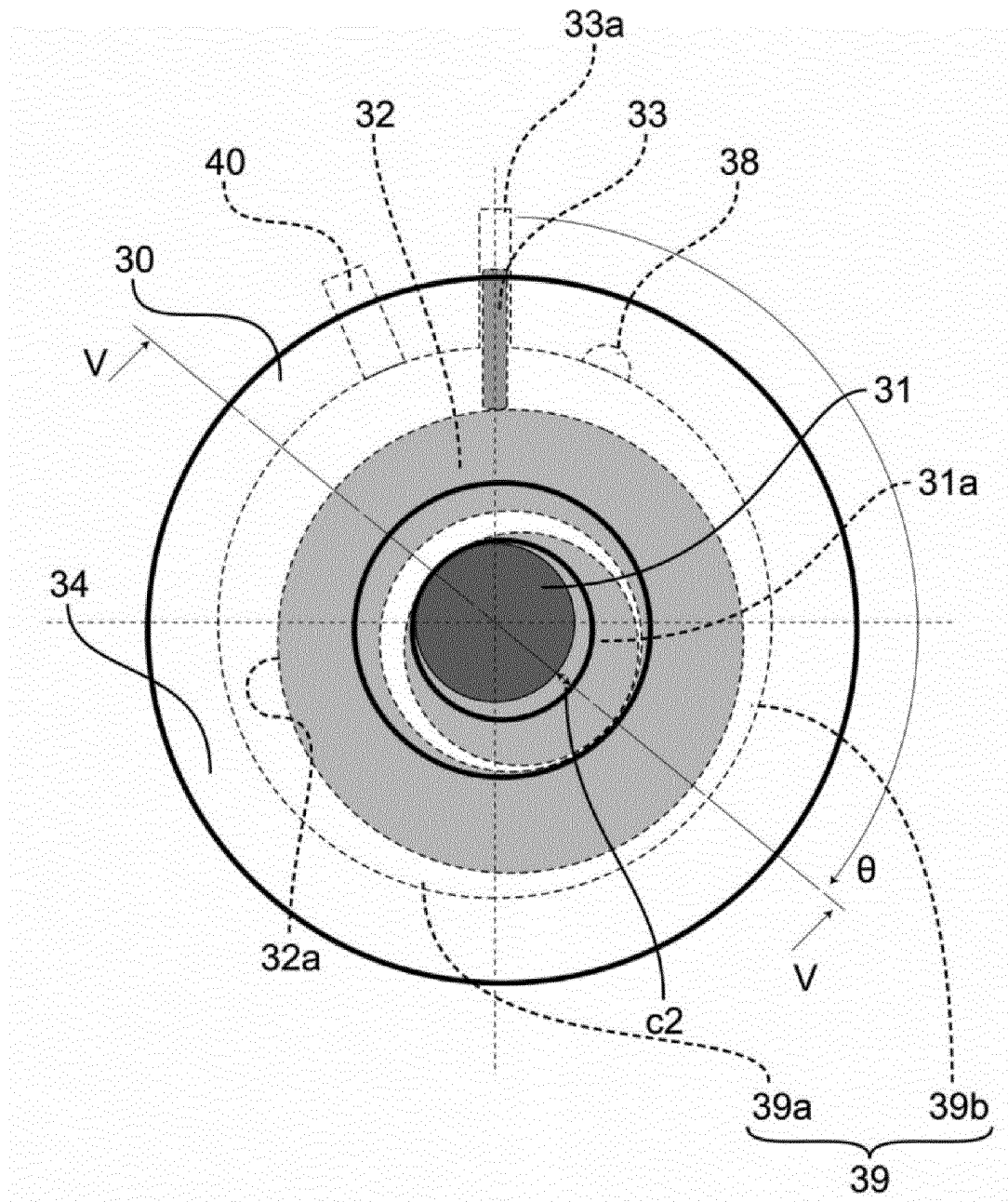
[Fig. 2]



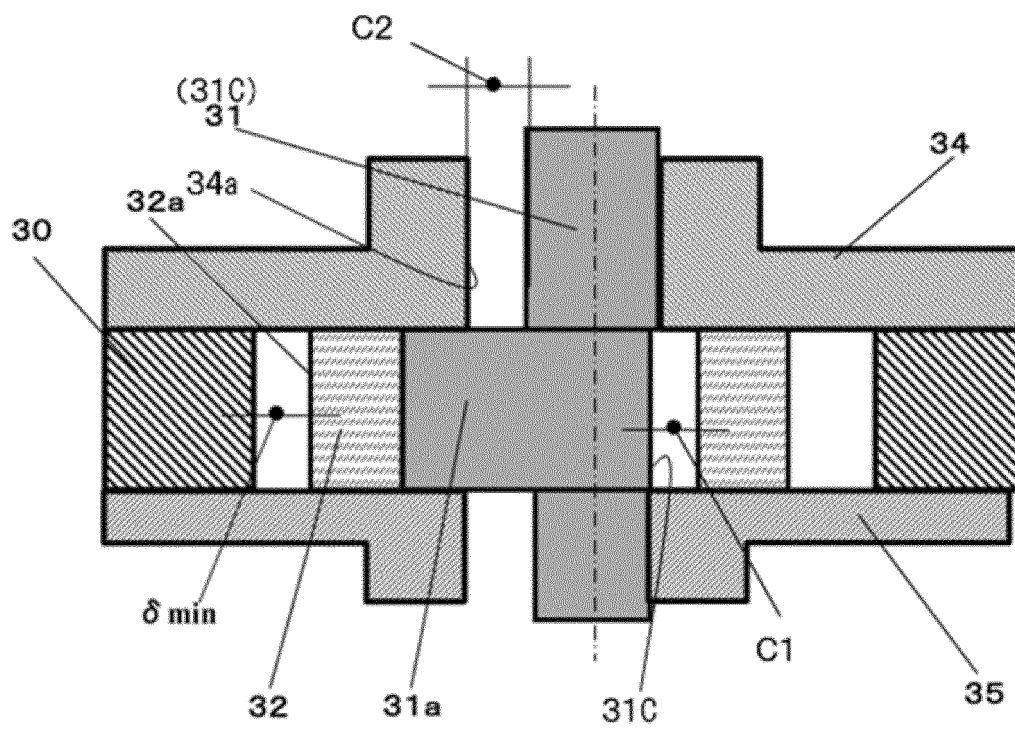
[Fig. 3]



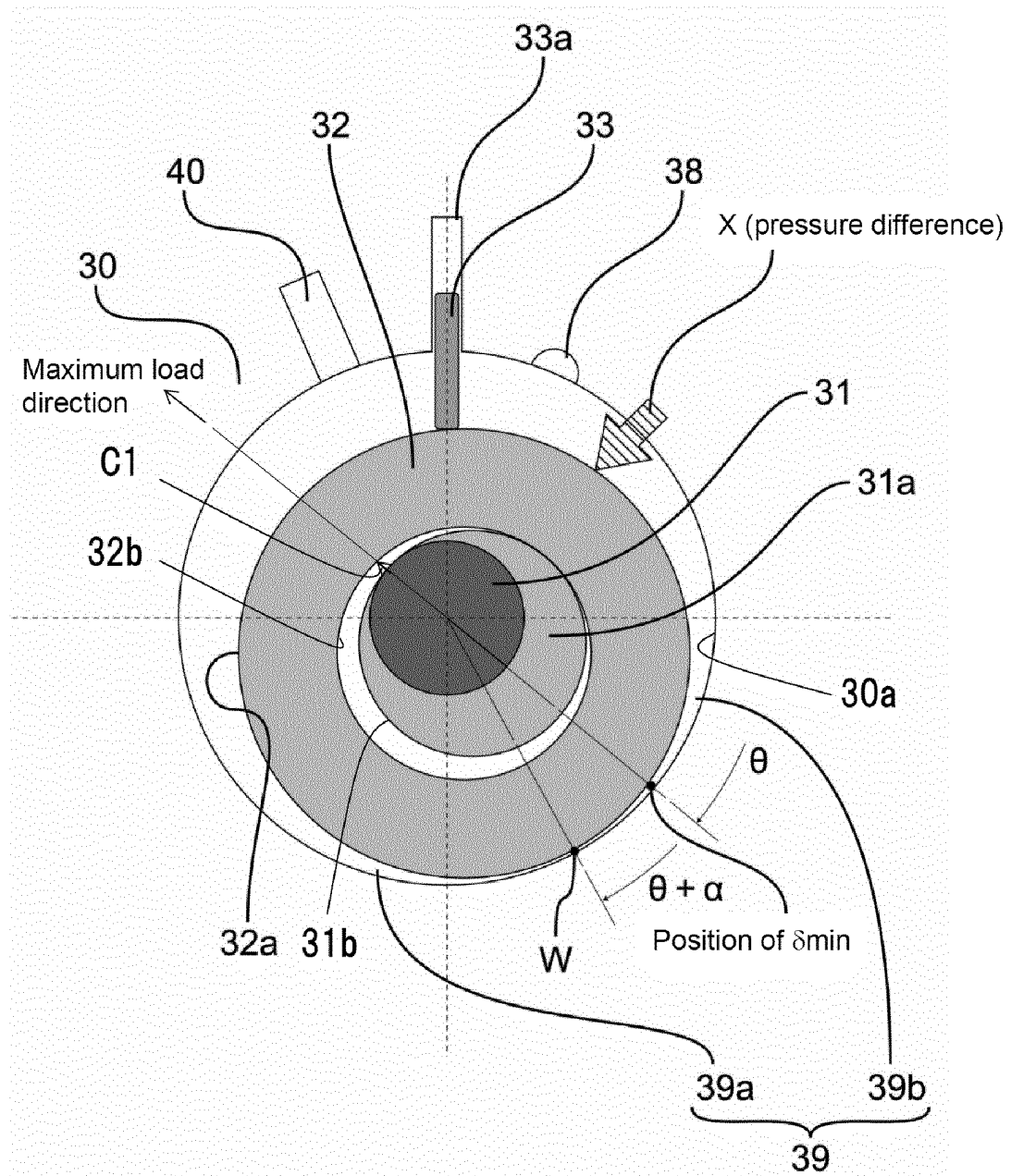
[Fig. 4]



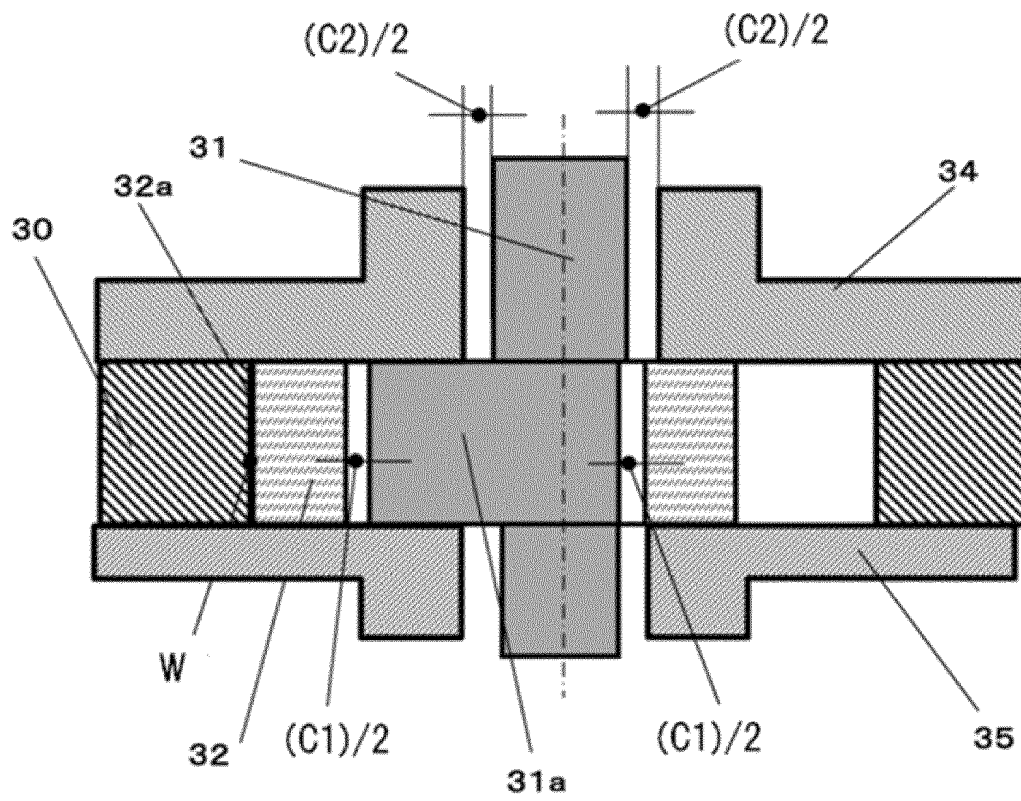
[Fig. 5]



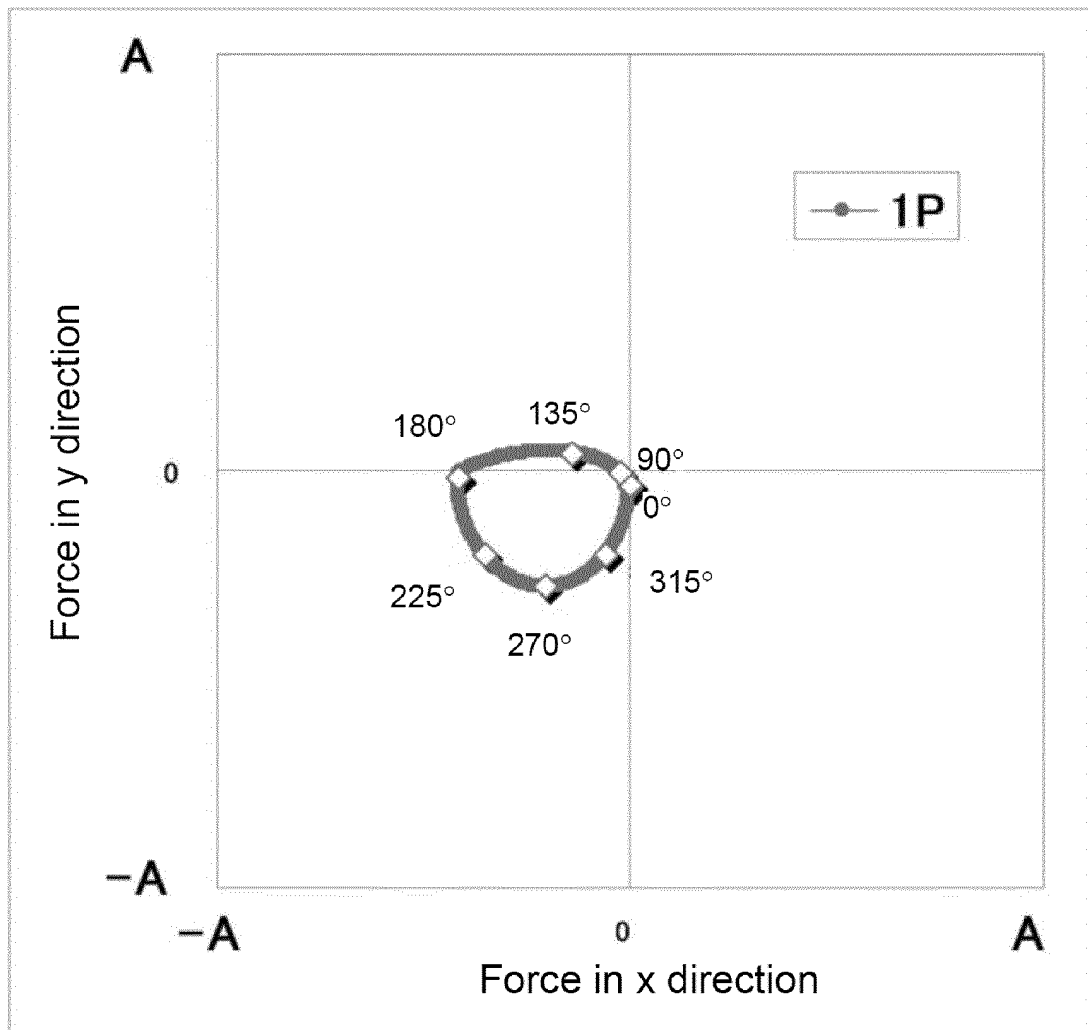
[Fig. 6]



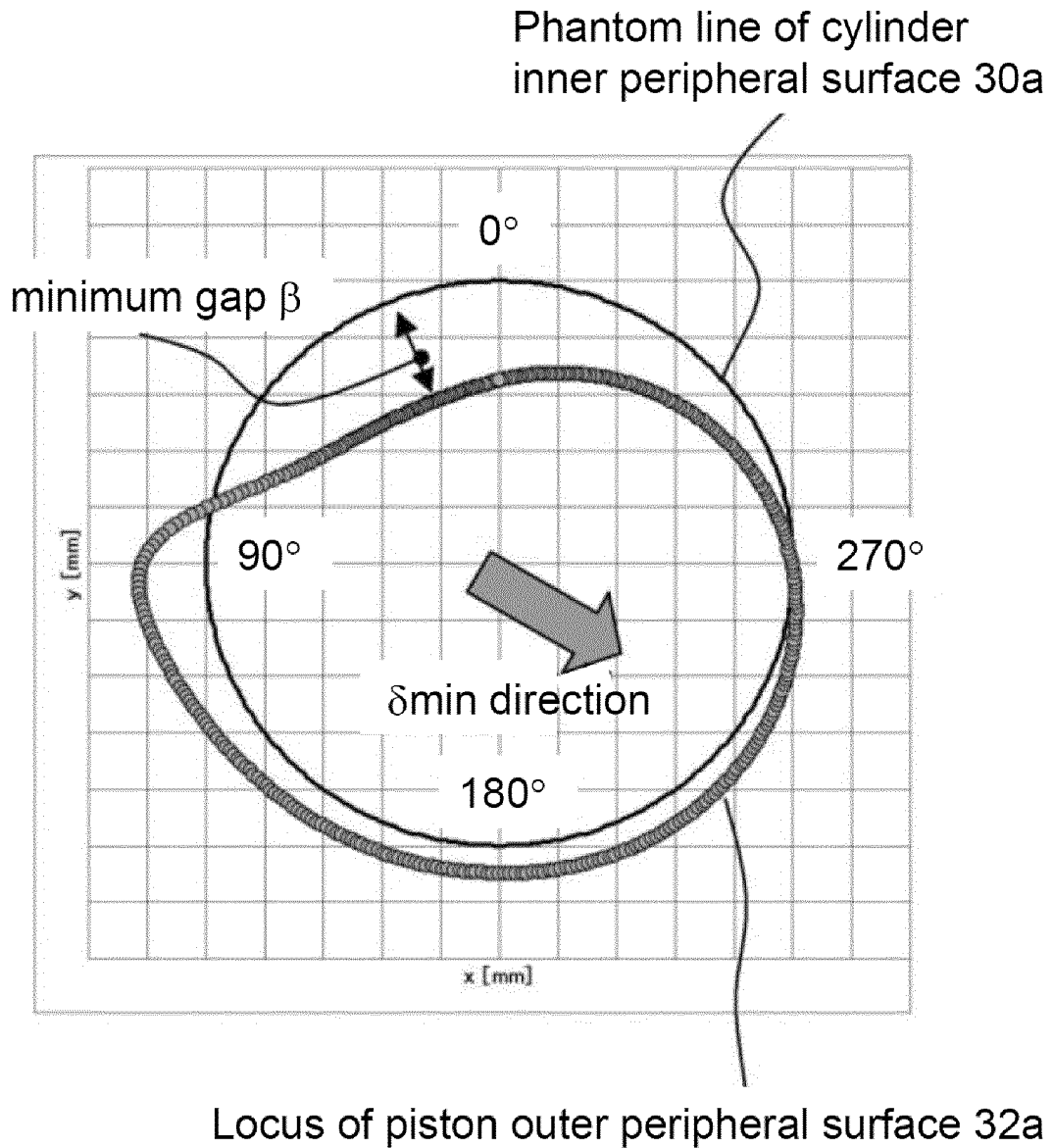
[Fig. 7]



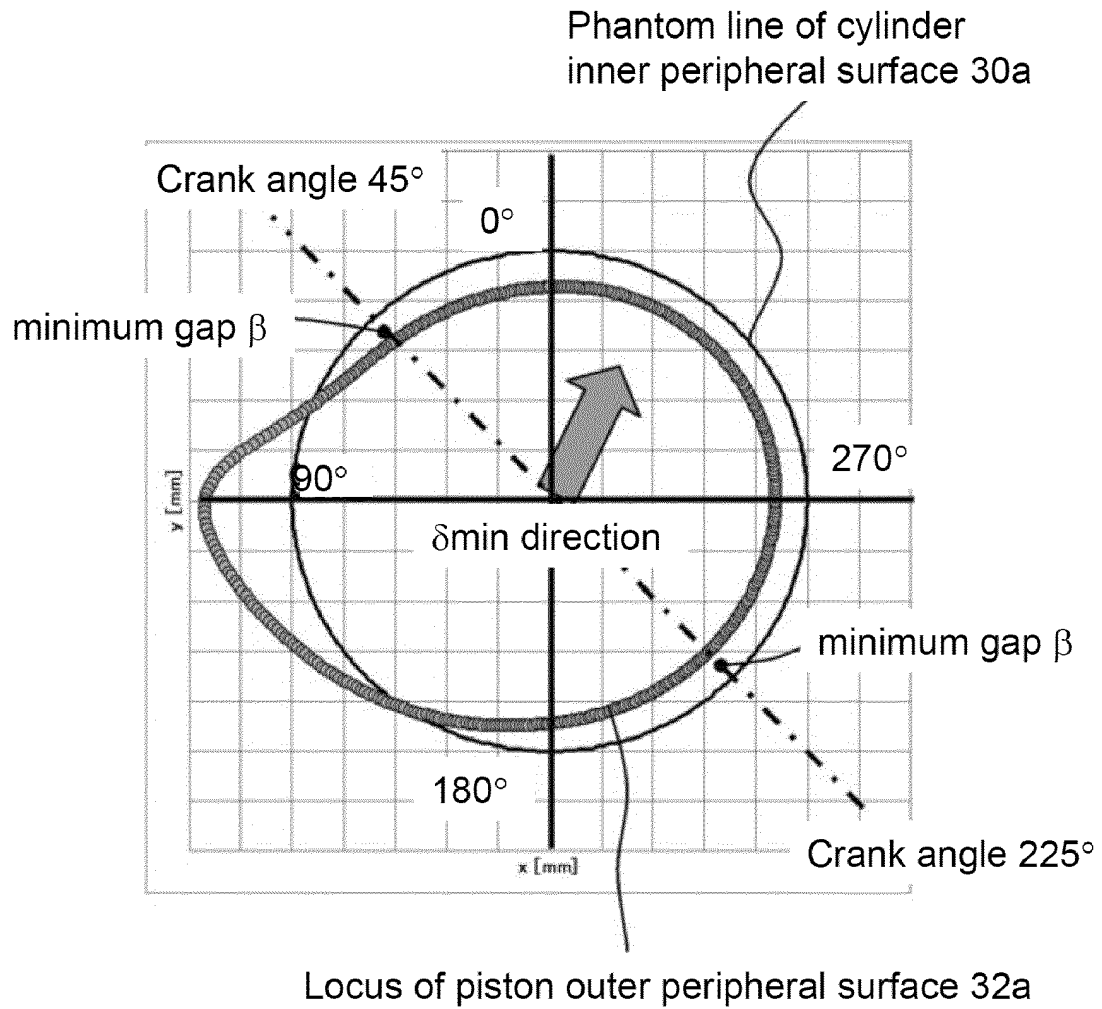
[Fig. 8]



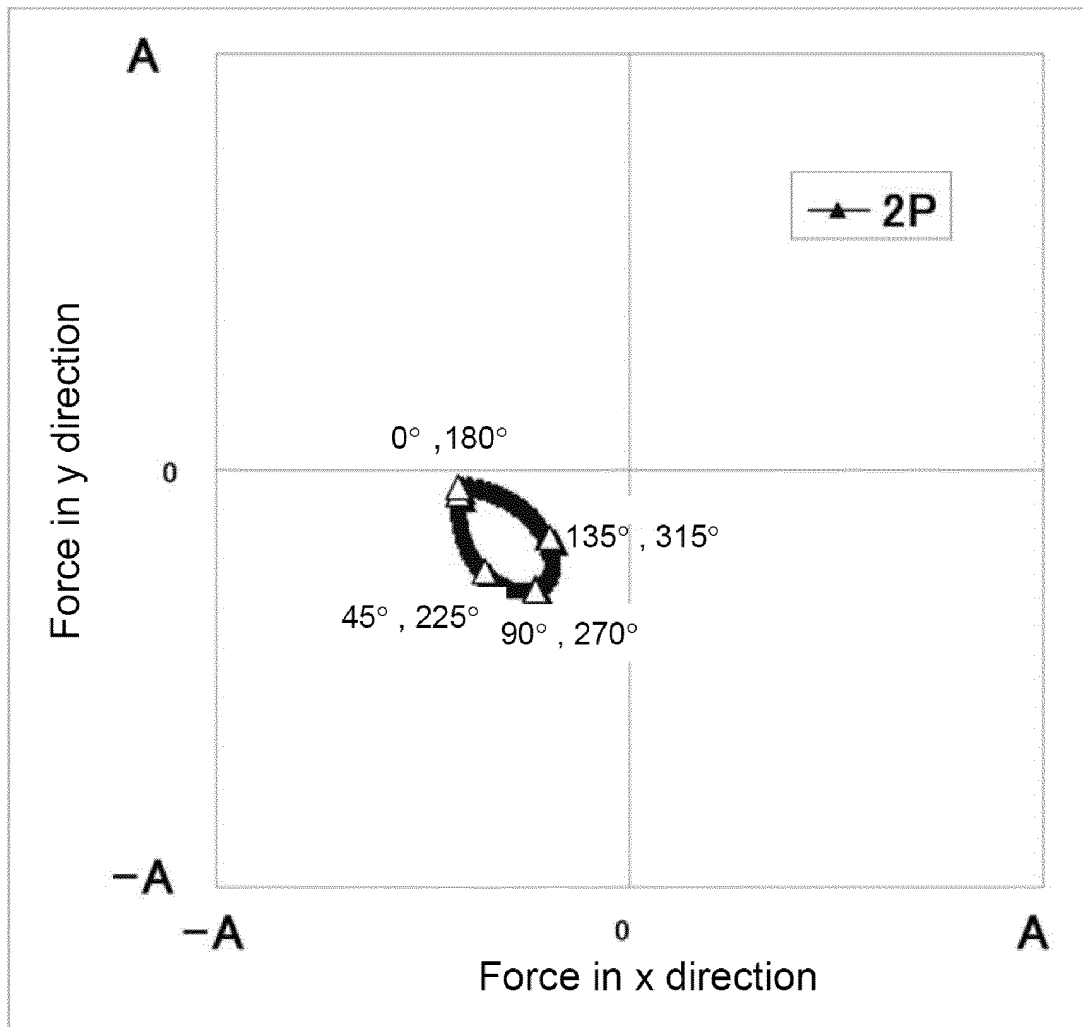
[Fig. 9]



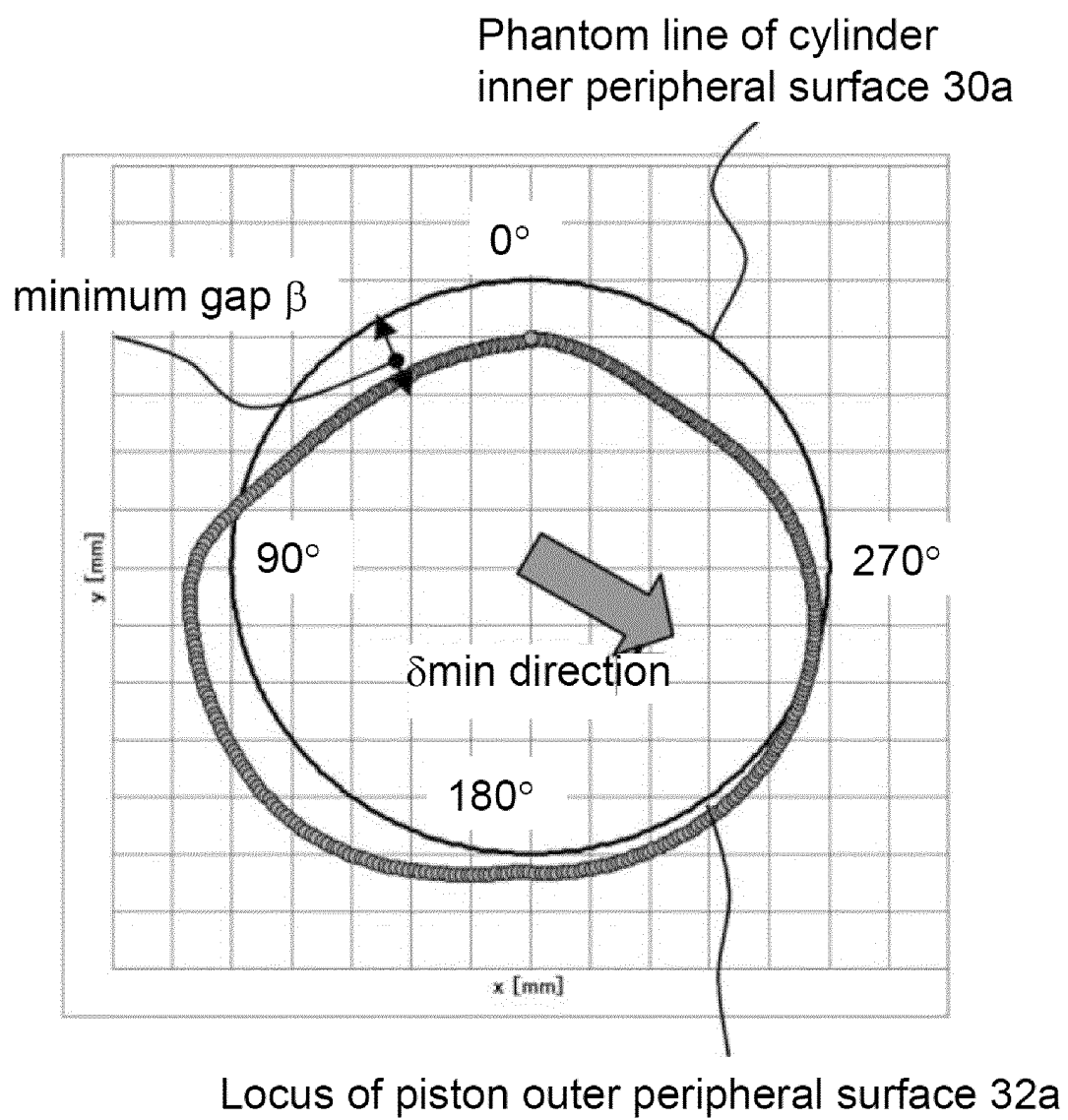
[Fig. 10]



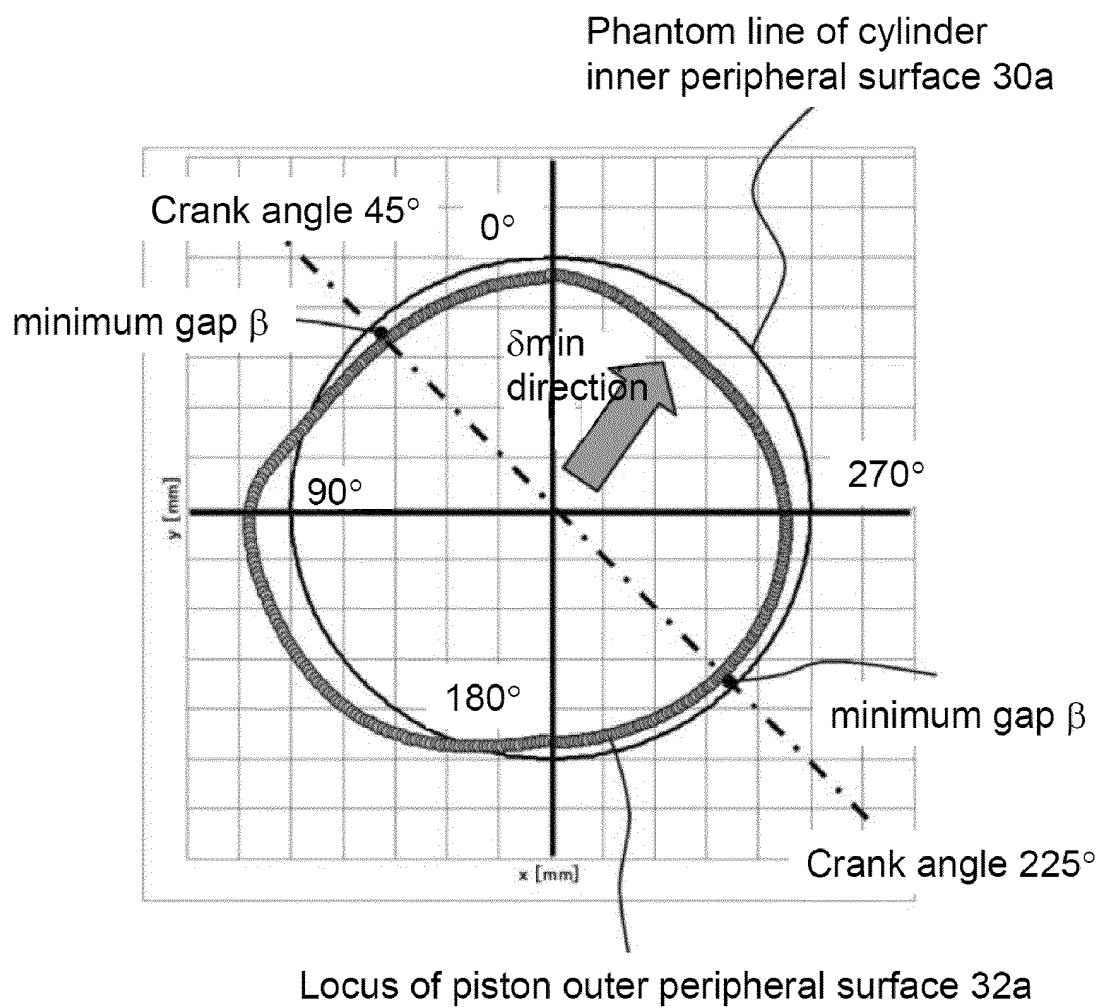
[Fig. 11]



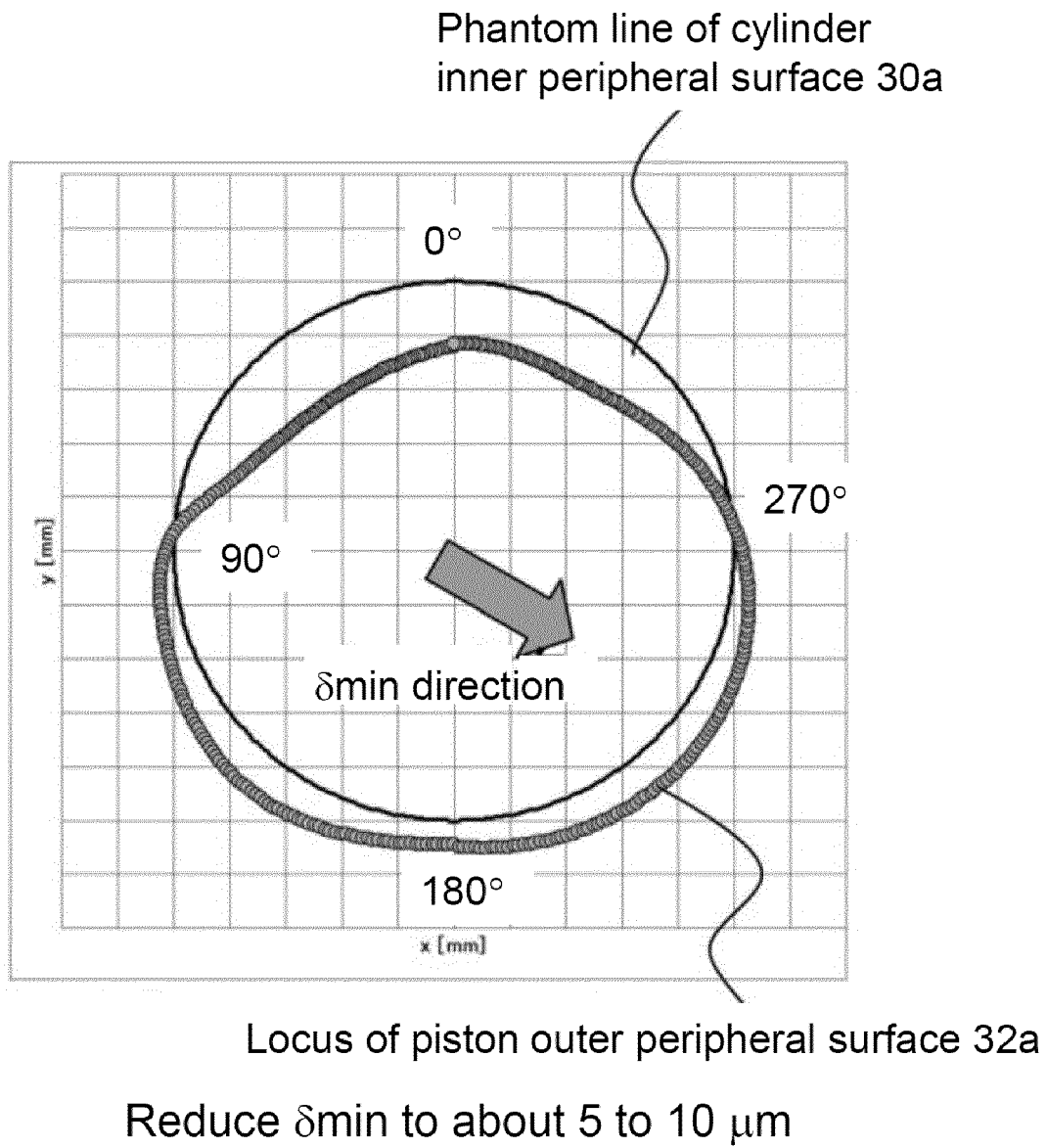
[Fig. 12]



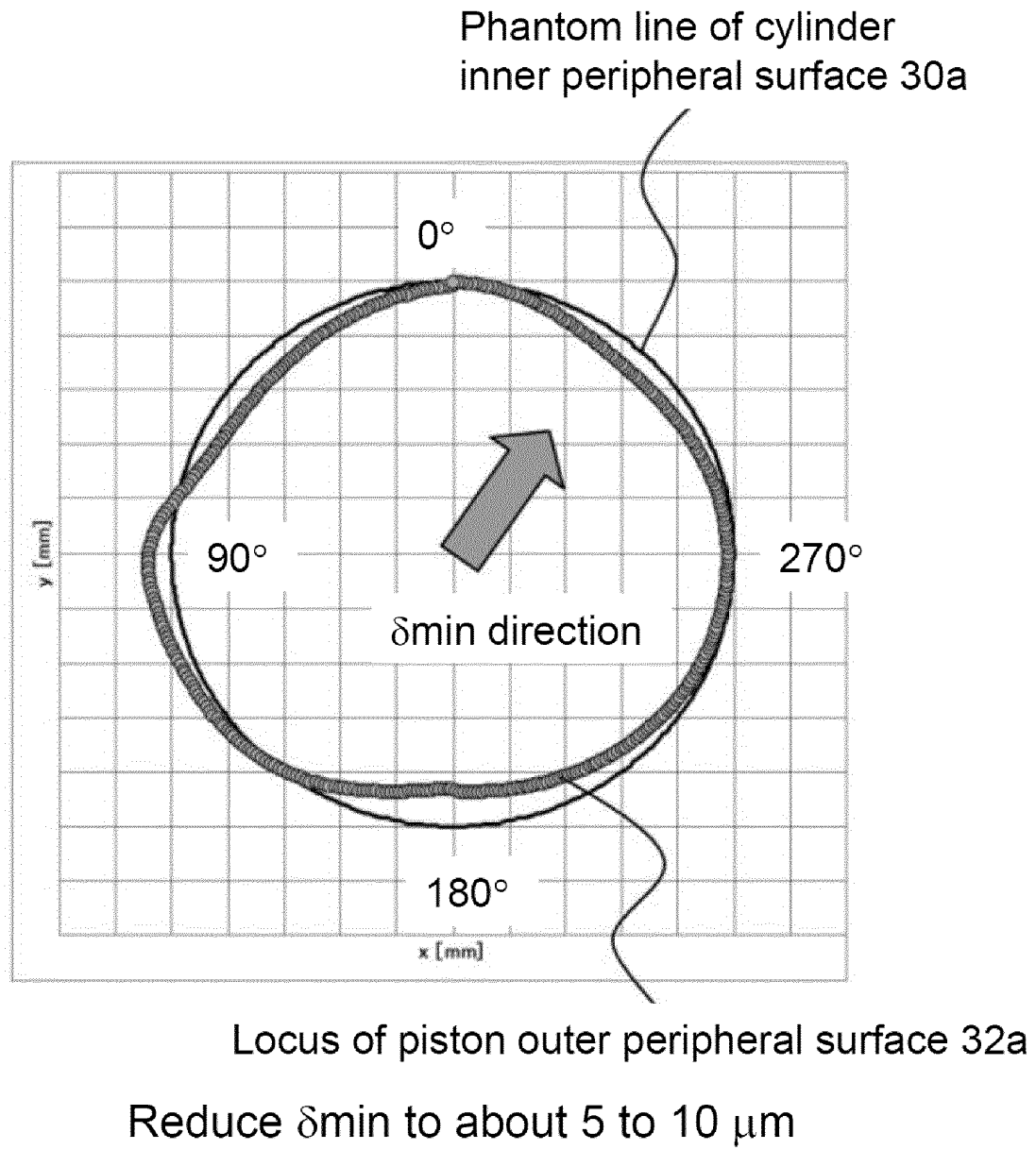
[Fig. 13]



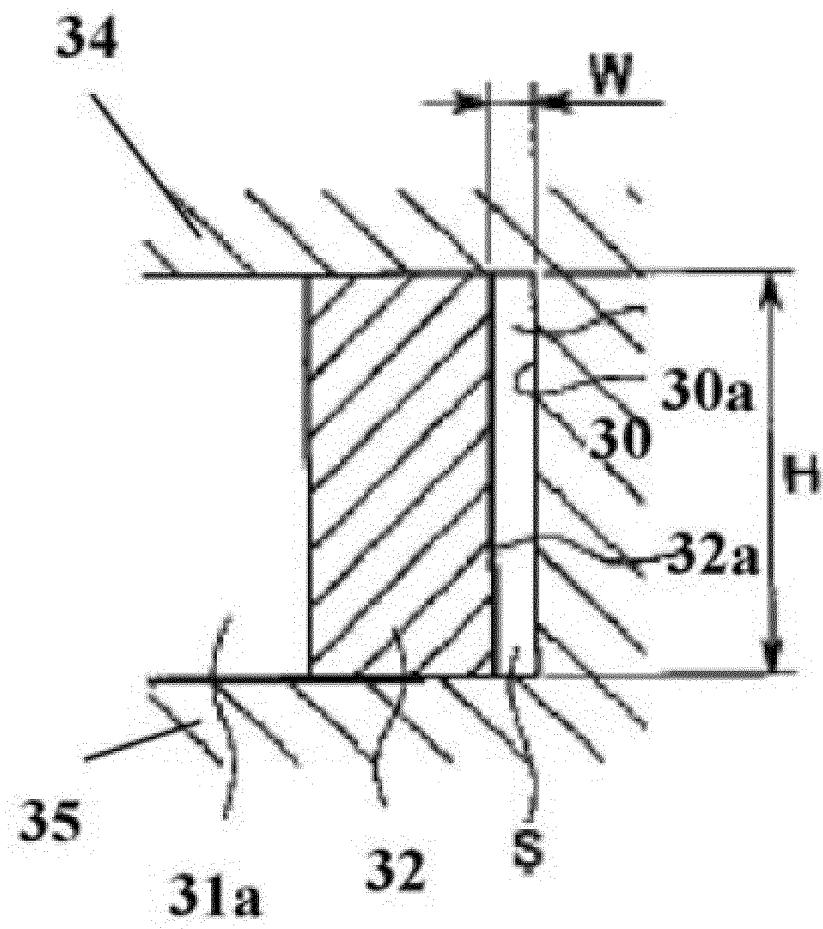
[Fig. 14]



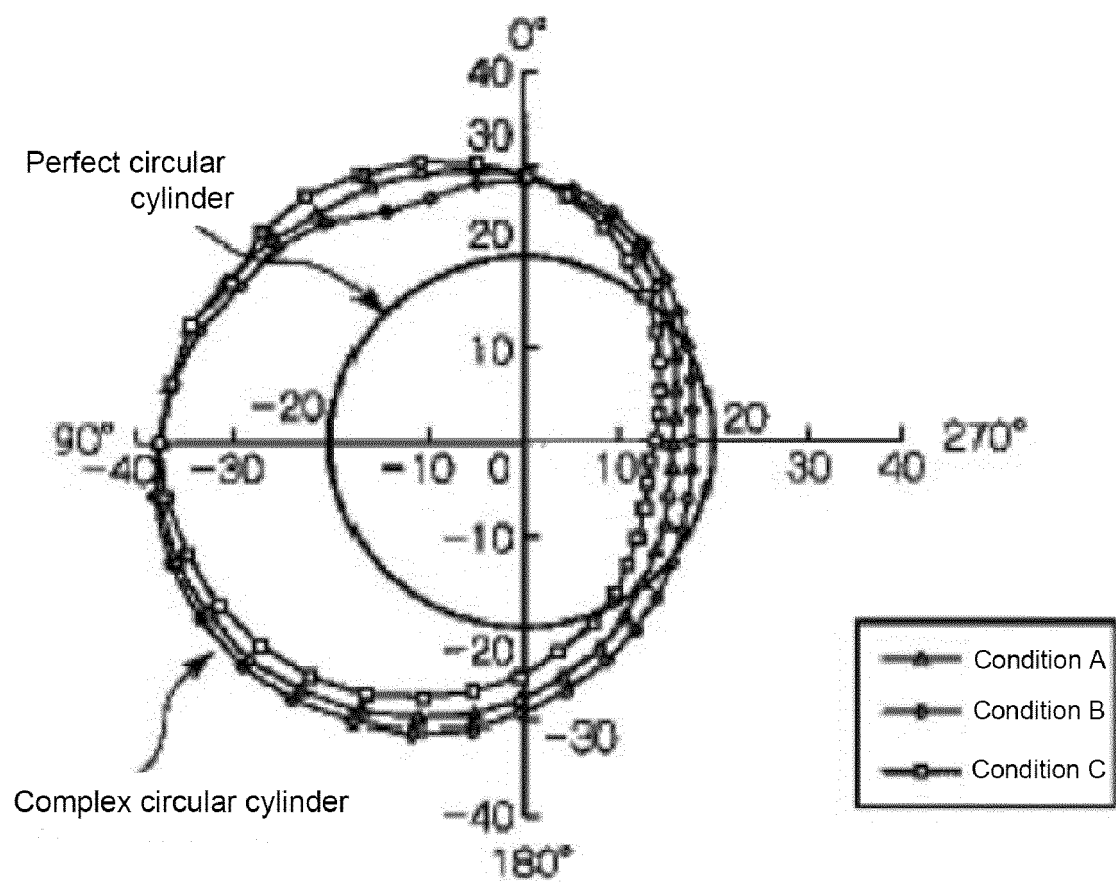
[Fig. 15]



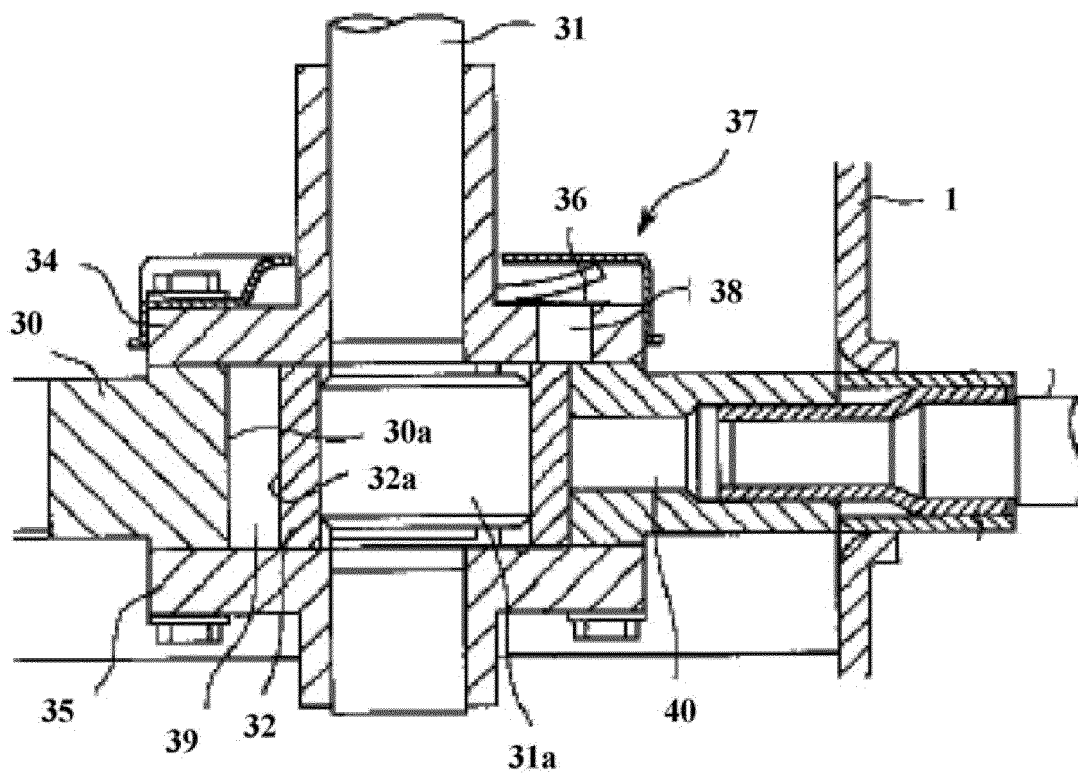
[Fig. 16]



[Fig. 17]



[Fig. 18]



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2013/003446

A. CLASSIFICATION OF SUBJECT MATTER

F04C18/356(2006.01) i, F04C29/00(2006.01) i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F04C18/356, F04C29/00

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho 1922-1996 Jitsuyo Shinan Toroku Koho 1996-2013

Kokai Jitsuyo Shinan Koho 1971-2013 Toroku Jitsuyo Shinan Koho 1994-2013

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X Y	JP 2003-214369 A (Mitsubishi Heavy Industries, Ltd.), 30 July 2003 (30.07.2003), entire text; all drawings (Family: none)	1 2-4
Y	JP 61-142389 A (Daikin Industries, Ltd.), 30 June 1986 (30.06.1986), entire text; all drawings (Family: none)	2-4
Y	JP 2006-152950 A (Sanyo Electric Co., Ltd.), 15 June 2006 (15.06.2006), entire text; all drawings (Family: none)	3-4

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 ☐ See patent family annex.

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Date of the actual completion of the international search
01 August, 2013 (01.08.13)Date of mailing of the international search report
13 August, 2013 (13.08.13)Name and mailing address of the ISA/
Japanese Patent Office

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INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2013/003446

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	JP 2005-240564 A (Mitsubishi Electric Corp.), 08 September 2005 (08.09.2005), entire text; all drawings (Family: none)	1-4
A	JP 2002-98075 A (Matsushita Electric Industrial Co., Ltd.), 05 April 2002 (05.04.2002), entire text; all drawings (Family: none)	1-4
A	JP 55-14278 B2 (Tokyo Shibaura Electric Co., Ltd.), 15 April 1980 (15.04.1980), entire text; all drawings (Family: none)	1-4

Form PCT/ISA/210 (continuation of second sheet) (July 2009)

REFERENCES CITED IN THE DESCRIPTION

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