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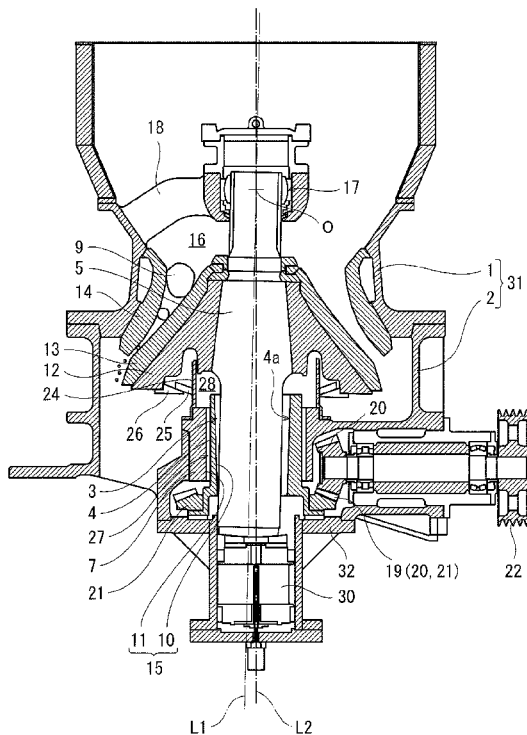
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(54) Title: GYRATORY CRUSHER

(54) 発明の名称: 旋動式破砕機



(57) Abstract: This gyratory crusher is provided with: a main shaft that is disposed rotatably inside a concavity and that rotates eccentrically with the central axis of the main shaft at an incline to the central axis of the concavity; an upper bearing that rotatably supports an upper end section of the main shaft; a lower bearing that rotatably supports a lower end section of the main shaft; and a hydraulic cylinder that causes the main shaft to move vertically through oil pressure. The lower bearing is provided with an eccentric sleeve that has a main shaft fitting-insertion hole into which the lower end section of the main shaft is inserted to fit rotatably, and an outer cylinder that has an eccentric sleeve fitting-insertion hole into which the eccentric sleeve is inserted to fit rotatably. At least one of an inner peripheral surface of the main shaft fitting-insertion hole, an outer peripheral surface of the eccentric sleeve, and an inner peripheral surface of the eccentric sleeve fitting-insertion hole has a tapered shape in at least a partial region thereof in the axial direction such that the distance from a surface that such tapered shape faces becomes greater as the tapered shape progresses upward.

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- 一 国際調査報告（条約第21条(3)）

(57) 要約：旋動式破碎機は、コーンケーブの内部に回転可能に配置され、その中心軸線がコーンケーブの中心軸線に対して傾斜して偏心旋回運動をする主軸と、主軸の上端部を回転自在に支持する上部軸受と、主軸の下端部を回転自在に支持する下部軸受と、主軸を油圧により上下動させる油圧シリンダと、を備え、下部軸受は、主軸の下端部が回転自在に嵌挿される主軸嵌挿穴を有する偏心スリーブと、偏心スリーブが回転自在に嵌挿される偏心スリーブ嵌挿穴を有する外筒と、を備え、主軸嵌挿穴の内周面、偏心スリーブの外周面、および偏心スリーブ嵌挿穴の内周面の少なくとも何れか1つが、軸線方向の少なくとも一部の領域において、上方に向かうほど対向する面との距離が広がるようなテーパ形状を有している。

DESCRIPTION**Title of Invention: GYRATION-TYPE CRUSHER****Technical Field**

[0001] The present invention relates to a gyration-type crusher that crushes rocks etc.

Background Art

[0002] Gyration-type crushers such as gyratory crushers and cone crushers have been conventionally used to crush large raw stones (rocks). In particular, there is a known hydraulic gyration-type crusher in which a main shaft equipped with a mantle is rotatably supported by upper and lower bearings and is moved upward and downward by hydraulic pressure (see Patent Literature 1, for example). Hereinafter, a hydraulic cone crusher will be taken as an example of the conventional gyration-type crushers, and the overview and crushing principle of the cone crusher will be described with reference to FIG. 22.

[0003] In the conventional gyration-type crusher shown in FIG. 22, a main shaft 5 is disposed in the center of an internal space defined by a frustum-shaped, tubular upper frame 1 and a lower frame 2 coupled to the upper frame 1. The central axis L1 of the main shaft 5 is tilted with respect to the central axis L2 of the upper frame 1. The upper and lower frames 1 and 2 may be collectively referred to as "frame 31".

[0004] The main shaft 5 has a cylindrical lower portion, which is rotatably supported by a lower bearing 15. The lower bearing 15 includes an eccentric sleeve 4 including a main shaft insertion hole 3 into which the lower end of the main shaft 5 is rotatably inserted. The eccentric sleeve 4 is equipped with an eccentric sleeve support 32 supporting the eccentric sleeve 4 from below in a manner permitting relative rotation of the eccentric sleeve 4. The eccentric sleeve support 32 is secured to the lower frame 2. The outer circumferential surface of the eccentric sleeve 4 is rotatably inserted into an eccentric sleeve insertion hole 27 of an outer barrel 7 mounted on the lower frame 2. The upper end of the main shaft 5 is rotatably supported by an upper bearing 17. The upper bearing 17 is supported by a spider 18 coupled to the upper frame 1. The spider 18 is a beam passing through the central portion of the upper frame 1 and connecting the central portion to the upper end of the upper frame 1.

[0005] In the gyration-type crusher shown in FIG. 22, a hydraulic cylinder 30 is disposed below the lower bearing 15 to move the main shaft 5 upward and downward by hydraulic pressure. A partition 24 having the shape of a hollow circular cylinder is disposed above the lower bearing 15, and a hydraulic pressure chamber 28 is defined inside the inner circumference of the partition 24. A lubricating oil is supplied from the hydraulic pressure chamber 28 to a

gap between the lower end of the main shaft 5 and the inner circumferential surface of the main shaft insertion hole 3 and a gap between the outer circumferential surface of the eccentric sleeve 4 and the inner circumferential surface of the eccentric sleeve insertion hole 27 to form oil films for the purpose of, for example, ensuring smooth sliding and preventing wearing down of the sliding surfaces. Thus, the eccentric sleeve 4 and outer barrel 7 of the lower bearing 15 function together as a radial sliding bearing. A dust seal 25 for preventing entry of dust into the hydraulic pressure chamber 28 is attached to the bottom surface of a mantle core 12 using a dust seal cover 26.

[0006] Hereinafter, the bearing portion constituted by the outer circumferential surface of the main shaft 5 inserted into the main shaft insertion hole 3 and the inner circumferential surface of the eccentric sleeve 4 (main shaft insertion hole 3) may be referred to as “main shaft bearing 10”, and the bearing portion constituted by the outer circumferential surface of the eccentric sleeve 4 and the inner circumferential surface of the outer barrel 7 (eccentric sleeve insertion hole 27) may be referred to as “eccentric sleeve bearing 11”. The main shaft bearing 10 and eccentric sleeve bearing 11 may be collectively referred to as “lower bearing 15” without distinction between the bearings 10 and 11.

[0007] The mantle core 12 having a frustum cone-shaped outer circumferential surface is securely attached to the outer surface of an upper portion of the main shaft 5 by shrink-fitting (the upper portion is below the upper bearing 17). A mantle 13 made of a wear-resistant material (e.g., high-manganese steel) and having a frustum cone-shaped outer circumferential surface is mounted on the outer circumferential surface of the mantle core 12.

[0008] A concave 14 made of a wear-resistant material (e.g., high-manganese steel) is mounted on the inner surface of the upper frame 1. The space defined by the concave 14 and mantle 13 and having a substantially wedge-like shape with a narrowed lower portion in vertical cross-section serves as a crushing chamber 16.

[0009] The central axis L1 of the main shaft 5 and the central axis L2 of the upper frame 1 intersect at an intersection point O located in an upper space of the crusher. The main shaft 5 is tilted with respect to the upper frame 1 in a plane including the central axis L1 of the main shaft 5 and the central axis L2 of the upper frame 1. The eccentric sleeve 4 has a central axis L4 substantially coinciding with the central axis L2 of the upper frame 1 (upper bearing 17), and is rotatable about the central axis L4.

[0010] In this configuration, drive power of an electric motor (not shown) disposed outside the frame 31 is transmitted to the eccentric sleeve 4 through power transmission mechanisms such as a pulley 22, a horizontal shaft, and bevel gears 19 (drive and driven bevel gears 20 and

21), and thus the eccentric sleeve 4 coupled to the driven bevel gear 21 is rotated about the central axis L2 of the upper frame 1. Consequently, in the crushing chamber 16, the main shaft 5 performs eccentric gyratory motion, namely precessional motion, with the intersection point O spatially fixed. It should be noted that this behavior is a geometrically ideal one. In an actual crusher, the intersection point O undergoes small displacements due to the bearing clearance of the upper bearing 17 or deformation of the casing in some situations such as during operation. The small displacements of the intersection point O could cause small variations in the geometric motion behavior of the main shaft 5.

[0011] In the crushing chamber 16, during the eccentric gyratory motion described above, the distance between any point on the inner surface of the concave 14 and the outer circumferential surface of the mantle 13 that faces the point changes with a period equal to that of the rotation of the main shaft 5. That is, when the main shaft 5 is gyrating in the crushing chamber 16 under the effect of the rotation of the eccentric sleeve 4, for example, the point which is located at the vertical bottom of the crushing chamber 16 and at which the distance between the outer surface of the mantle 13 and the inner surface of the concave 14 is minimum shifts along with the gyration of the main shaft 5 as shown in FIG. 2.

[0012] The rocks 9 as a crushing target (hereinafter referred to as “crushing target material”) are introduced into the crusher from above and fall into the crushing chamber 16. In the crushing chamber 16, the gap between the concave 14 and mantle 13 narrows downward, and the width of the gap changes periodically along with the gyration of the main shaft 5. Thus, falling in and compression of the crushing target material 9 are repeated as the crushing process proceeds. The crushing target material 9 crushed at a lower portion of the concave 14 into pieces smaller than the minimum width of the gap between the concave 14 and mantle 13 is discharged downward and collected as a crushed product.

[0013] According to the crushing principle of the gyration-type crusher, the crushing of the crushing target material 9 by the mantle 13 (the application of a crushing load W) induces a reaction force P1 acting on the main shaft 5 in the direction from the crushing point to the inside of the frame 31 (reaction force P described later with reference to FIG. 2) and a reaction force P2 acting on the frame 31 in the direction from the crushing point to the outside of the frame 31. The reaction force P1 acting on the main shaft 5 causes the lower end of the main shaft 5 to move toward the inner circumferential surface of the main shaft insertion hole 3 of the eccentric sleeve 4 (translational motion). Additionally, since the components such as the main shaft 5 and frame 31 are displaced or deformed by the two reaction forces, the central axis L1 of the main shaft 5 and the central axis L3 of the main shaft insertion hole 3 become non-parallel, and

the central axis L1 of the main shaft 5 is tilted with respect to the central axis L3 of the main shaft insertion hole 3 (rotational motion). This could bring the main shaft bearing 10 into what may be called a “partial contact state” where the minimum thickness of the oil film is reduced at the upper or lower end of the bearing 10. In case that the severity of such a partial contact state is increased, the outer circumferential surface of the lower end of the main shaft 5 and the inner circumferential surface of the main shaft insertion hole 3 of the eccentric sleeve 4 are shifted from a fluid lubrication state where there is a fluid film between the two surfaces into a mixed lubrication state where the two surfaces contact each other at a microscopic level or into a state where the solid surfaces slide in contact with each other. Consequently, the main shaft 5 and eccentric sleeve 4 could suffer so-called “seizure”.

[0014] Likewise, in the eccentric sleeve bearing 11, the reaction force P1 acting on the eccentric sleeve 4 via the main shaft 5 causes the eccentric sleeve 4 to move toward the inner circumferential surface of the outer barrel 7 that is opposite to the region where the reaction force P1 is at work. Since the components such as the eccentric sleeve 4 and frame 31 are displaced or deformed by the reaction force P1 acting on the main shaft 5 and the reaction force P2 acting on the frame 31, the central axis L4 of the eccentric sleeve 4 and the central axis L5 of the eccentric sleeve insertion hole 27 become non-parallel, and the central axis L4 of the eccentric sleeve 4 is tilted with respect to the central axis L5 of the eccentric sleeve insertion hole 27. This could bring the eccentric sleeve bearing 11 into what may be called a “partial contact state” where the minimum thickness of the oil film is reduced at the upper or lower end of the bearing 11. In case that the severity of such a partial contact state is increased, the outer circumferential surface of the eccentric sleeve 4 and the inner circumferential surface of the eccentric sleeve insertion hole 27 of the outer barrel 7 are shifted from a fluid lubrication state where there is a fluid film between the two surfaces into a mixed lubrication state where the two surfaces contact each other at a microscopic level or into a state where the solid surfaces slide in contact with each other. Consequently, the eccentric sleeve 4 and the outer barrel 7 could suffer so-called “seizure”.

[0015] Hereinafter, the partial contact that occurs at the upper end of the lower bearing 15 (main shaft bearing 10 or eccentric sleeve bearing 11) will be referred to as “upper partial contact”, and the partial contact that occurs at the lower end of the lower bearing 15 will be referred to as “lower partial contact”. The lower bearing 15 could suffer both the upper partial contact and the lower partial contact due to changes in factors related to the state of the crusher during crushing operation, such as the magnitudes of the reaction forces, the oil film

thickness (the size of the bearing clearance) of the lower bearing 15, the deformation of the main shaft 5, and the deformation of the eccentric sleeve 4.

[0016] Gyration-type crushers based on the crushing principle described above are inherently likely to experience partial contact of the bearings.

[0017] The lower bearing 15 in the partial contact state as described above is exposed to a large specific pressure locally acting on the end of the lower bearing 15 and could suffer failures such as wearing down and seizure under load conditions which would not cause any problems in normal use. This may require measures such as early replacement of the lower bearing 15.

[0018] Rocks, which are a typical crushing target of gyration-type crushers, vary widely in their properties such as strength and brittleness. In the case of crushing of the crushing target material 9 of a type which is hard to crush, the mantle 13 is subjected to a considerably large reaction force, and the lower bearing 15 wears down or is damaged within a short period of time. Thus, it is necessary, for example, to adjust or test crusher components such as the lower bearing 15, choose a suitable gyration-type crusher, or selectively use different gyration-type crushers, according to the type of the crushing target material 9. As such, gyration-type crushers are very complicated to handle and impose a heavy burden of cost and effort.

[0019] Further, in such a gyration-type crusher, the mantle 13 or concave 14 becomes thinner with passage of operation time due to gradual surface wear, and accordingly the distance between the outer circumferential surface of the mantle 13 and the inner surface of the concave 14 changes (increases). This renders it necessary to change (adjust) the position of the upper frame 1 or main shaft 5 as a function of the distance change. Consequently, even when the type of the crushing target material 9 remains the same, the crushing load or the crushing-induced reaction force changes, and the load conditions and other factors influencing the lower bearing 15 also change.

Citation List

Patent

Literature

[0020] PTL 1: Japanese Laid-Open Patent Application Publication No. H10-272374

Summary of Invention**Technical Problem**

[0021] It is an object of the present invention to overcome or ameliorate at least one of the disadvantages of the prior art, or to at least provide a useful alternative thereto. It is a further object of the present invention is to provide a hydraulic gyration-type crusher in which a main shaft is moved upward and downward by hydraulic pressure, the crusher having a simple configuration and having high robustness against changes in load conditions.

Solution to Problem

[0021a] Accordingly, in an aspect, the present invention provides a gyration-type crusher including: a main shaft rotatably disposed inside a concave to eccentrically gyrate in a manner in which a central axis of the main shaft is tilted with respect to a central axis of the concave; an upper bearing rotatably supporting an upper end of the main shaft; a mantle mounted on the main shaft; a lower bearing rotatably supporting a lower end of the main shaft; and a hydraulic cylinder disposed below the lower bearing to move the main shaft upward and downward by hydraulic pressure, wherein the lower bearing includes an eccentric sleeve including a main shaft insertion hole into which the lower end of the main shaft is rotatably inserted, and an outer barrel including an eccentric sleeve insertion hole into which the eccentric sleeve is rotatably inserted; and an inner circumferential surface of the main shaft insertion hole has a tapered shape in a region extending over at least a portion of the inner circumferential surface of the main shaft insertion hole in an axial direction of the main shaft insertion hole, the tapered shape being such that a distance between the inner circumferential surface of the main shaft insertion hole and an outer circumferential surface of the main shaft that faces the inner circumferential surface of the main shaft insertion hole increases upward, an inner circumferential surface of the eccentric sleeve insertion hole has a tapered shape in a region extending over at least a portion of the inner circumferential surface of the eccentric sleeve insertion hole in an axial direction of the eccentric sleeve insertion hole, the tapered shape being such that a distance between the inner circumferential surface of the eccentric sleeve insertion hole and an outer circumferential surface of the eccentric sleeve that faces the inner circumferential surface of the eccentric sleeve insertion hole increases upward, or the outer circumferential surface of the eccentric sleeve has a tapered shape in a region extending over at least a portion of the outer circumferential surface of the eccentric sleeve in an axial direction of the eccentric sleeve, the tapered shape being such

that a distance between the outer circumferential surface of the eccentric sleeve and the inner circumferential surface of the eccentric sleeve insertion hole that faces the outer circumferential surface of the eccentric sleeve increases upward.

[0022] Also disclosed herein is a gyration-type crusher that includes: a main shaft rotatably disposed inside a concave to eccentrically gyrate in a manner in which a central axis of the main shaft is tilted with respect to a central axis of the concave; an upper bearing rotatably supporting an upper end of the main shaft; a mantle mounted on the main shaft; a lower bearing rotatably supporting a lower end of the main shaft; and a hydraulic cylinder disposed below the lower bearing to move the main shaft upward and downward by hydraulic pressure, wherein: the lower bearing includes an eccentric sleeve including a main shaft insertion hole into which the lower end of the main shaft is rotatably inserted, and an outer barrel including an eccentric sleeve insertion hole into which the eccentric sleeve is rotatably inserted; and at least one of an inner circumferential surface of the main shaft insertion hole, an outer circumferential surface of the eccentric sleeve, or an inner circumferential surface of the eccentric sleeve insertion hole has a tapered shape in a region extending over at least a portion of the circumferential surface in an axial direction of the hole or eccentric sleeve, the tapered shape being such that a distance between the circumferential surface and another surface facing the circumferential surface increases upward.

[0023] In the above configuration, which includes the tapered shape as described above, the main shaft and the upper end of the inner circumferential surface of the lower bearing are prevented from being close to each other even in the situation where the crushing load is so large that the upper partial contact state would occur in conventional configurations. Thus, the upper partial contact state of the lower bearing can be avoided, and the reduction in minimum oil film thickness can be prevented. As such, a gyration-type crusher that prevents the occurrence of failures such as seizure in the lower bearing and that has high robustness against changes in load conditions can be constructed in a simple configuration.

[0024] The tapered shape may include a first tapered shape such that a diameter of the inner circumferential surface of the main shaft insertion hole increases upward.

[0025] The tapered shape may include a second tapered shape such that a diameter of the outer circumferential surface of the eccentric sleeve increases downward.

[0026] The tapered shape may include a third tapered shape such that a diameter of the inner circumferential surface of the eccentric sleeve insertion hole increases upward.

[0027] The first tapered shape may be formed in a region including an upper axial end of the inner circumferential surface of the main shaft insertion hole and having a length equal to or greater than $1/3$ of an axial length of the inner circumferential surface of the main shaft insertion

hole.

[0028] The third tapered shape may be formed in a region including an upper axial end of the inner circumferential surface of the eccentric sleeve insertion hole and having a length equal to or greater than $1/3$ of an axial length of the inner circumferential surface of the eccentric sleeve insertion hole.

[0029] A lubricating oil may be supplied to a gap between an outer circumferential surface of the lower end of the main shaft and the inner circumferential surface of the main shaft insertion hole and to a gap between the outer circumferential surface of the eccentric sleeve and the inner circumferential surface of the eccentric sleeve insertion hole, and a cone angle or rate of taper of the tapered shape may be set to provide a robust zone in regard to changes in minimum oil film thickness of the lubricating oil with changes in crushing load.

[0030] In the above configuration, the setting of the robust zone can be accomplished relatively easily by adjusting the cone angle or rate of taper of the tapered shape.

[0031] When the tapered shape is viewed in cross-section along the central axis of the main shaft, an angle indicative of a ratio of amount of change in diameter Dt of the region of the tapered shape in the lower bearing to a length of the region in a direction of the central axis may be from 0.001 to 1° .

[0032] The lower bearing may include an eccentric sleeve support disposed below the eccentric sleeve to support the eccentric sleeve in a manner permitting relative rotation of the eccentric sleeve, the eccentric sleeve may include a first thrust bearing surface in a lower surface of the eccentric sleeve, the first thrust bearing surface being contactable with an upper surface of the eccentric sleeve support, the eccentric sleeve support may include a second thrust bearing surface in the upper surface of the eccentric sleeve support, the second thrust bearing surface being contactable with the first thrust bearing surface, and at least one of the first or second thrust bearing surface may have a fourth tapered shape such that a distance between the first and second thrust bearing surfaces facing each other increases outward in a radial direction of the eccentric sleeve.

[0033] In the above configuration, the fourth tapered shape for accommodating tilt of the eccentric sleeve is formed between the eccentric sleeve and the eccentric sleeve support. Thus, deformation of the eccentric sleeve or eccentric sleeve support due to tilt of the eccentric sleeve can be reduced.

[0034] A gyration-type crusher according to another aspect of the present invention includes: a main shaft rotatably disposed inside a concave to eccentrically gyrate in a manner in which a central axis of the main shaft is tilted with respect to a central axis of the concave; an

upper bearing rotatably supporting an upper end of the main shaft; a mantle mounted on the main shaft; a lower bearing rotatably supporting a lower end of the main shaft; and a hydraulic cylinder disposed below the lower bearing to move the main shaft upward and downward by hydraulic pressure, wherein: the lower bearing includes an eccentric sleeve including a main shaft insertion hole into which the lower end of the main shaft is rotatably inserted, an outer barrel including an eccentric sleeve insertion hole into which the eccentric sleeve is rotatably inserted, and an eccentric sleeve support disposed below the eccentric sleeve to support the eccentric sleeve in a manner permitting relative rotation of the eccentric sleeve; the eccentric sleeve includes a first thrust bearing surface in a lower surface of the eccentric sleeve, the first thrust bearing surface being contactable with an upper surface of the eccentric sleeve support; the eccentric sleeve support includes a second thrust bearing surface in the upper surface of the eccentric sleeve support, the second thrust bearing surface being contactable with the first thrust bearing surface; and at least one of the first or second thrust bearing surface has a tapered shape such that a distance between the first and second thrust bearing surfaces facing each other increases outward in a radial direction of the eccentric sleeve.

[0035] In the above configuration, which includes the tapered shape as described above, the main shaft and the upper end of the inner circumferential surface of the main shaft insertion hole of the eccentric sleeve are prevented from being close to each other even in the situation where the crushing load is so large that the upper partial contact state would occur in conventional configurations. Thus, the upper partial contact state of the lower bearing can be avoided, and the reduction in minimum oil film thickness can be prevented. As such, a gyration-type crusher that prevents the occurrence of failures such as seizure in the lower bearing and that has high robustness against changes in load conditions can be constructed in a simple configuration.

Advantageous Effects of Invention

[0036] The present invention can provide a hydraulic gyration-type crusher in which a main shaft is moved upward and downward by hydraulic pressure, the hydraulic gyration-type crusher having a simple configuration and having robustness against changes in load conditions.

Brief Description of Drawings

[0037] FIG. 1 is a cross-sectional view showing the overall configuration of a gyration-type crusher according to Embodiment 1 of the present invention.

FIG. 2 is a plan view for explaining the crushing principle of gyration-type crushers.

FIG. 3A shows the situation where a lower bearing of a gyration-type crusher is in a

lower partial contact state.

FIG. 3B shows the situation where the lower bearing of the gyration-type crusher is in a uniform contact state.

FIG. 3C shows the situation where the lower bearing of the gyration-type crusher is in an upper partial contact state.

FIG. 4A is an enlarged vertical cross-sectional view showing the lower bearing in the case where the shaft of FIG. 3C is a main shaft.

FIG. 4B is an enlarged vertical cross-sectional view showing the lower bearing in the case where the shaft of FIG. 3C is an eccentric sleeve.

FIG. 5 is a graph showing changes in minimum bearing oil film thickness with changes in crushing load for a lower bearing having specifications A.

FIG. 6 is a graph showing changes in shaft tilt angle with changes in crushing load for the lower bearing having the specifications A.

FIG. 7 is a graph showing changes in minimum bearing oil film thickness with changes in crushing load for a lower bearing having specifications B.

FIG. 8 is a graph showing changes in shaft tilt angle with changes in crushing load for the lower bearing having the specifications B.

FIG. 9 shows the oil film pressure distribution of the lower bearing in the upper partial contact state.

FIG. 10 shows the oil film pressure distribution of the lower bearing in the uniform contact state under crushing load conditions and specifications identical to those in FIG. 9.

FIG. 11 shows comparison between a lower bearing devoid of robust property and a lower bearing having the robust property.

FIG. 12 is a graph showing a schematic curve representing the robust property of the lower bearing having the specifications A.

FIG. 13 is a graph showing the robust property curve of FIG. 12 as approximated by quadratic functions.

FIG. 14 is a graph showing the robust property curve of FIG. 12 as approximated by cubic functions.

FIG. 15 is a graph showing a schematic curve representing the robust property of the lower bearing having the specifications B.

FIG. 16 is a graph showing the robust property curve of FIG. 15 as approximated by quadratic functions.

FIG. 17 is a graph showing the robust property curve of FIG. 15 as approximated by

cubic functions.

FIG. 18 is an enlarged cross-sectional view showing the lower bearing and its vicinity in the gyration-type crusher of FIG. 1.

FIG. 19A is an enlarged cross-sectional view showing a lower bearing and its vicinity in a gyration-type crusher according to Variant 1 of Embodiment 1.

FIG. 19B is an enlarged cross-sectional view showing a lower bearing and its vicinity in a gyration-type crusher according to Variant 2 of Embodiment 1.

FIG. 19C is an enlarged cross-sectional view showing a lower bearing and its vicinity in a gyration-type crusher according to Variant 3 of Embodiment 1.

FIG. 19D is an enlarged cross-sectional view showing a lower bearing and its vicinity in a gyration-type crusher according to Variant 4 of Embodiment 1.

FIG. 20A is an enlarged cross-sectional view showing the lower end of an eccentric sleeve and its vicinity in a gyration-type crusher according to Embodiment 2 of the present invention.

FIG. 20B is an enlarged cross-sectional view showing the lower end of an eccentric sleeve and its vicinity in a gyration-type crusher according to Variant 1 of Embodiment 2.

FIG. 20C is an enlarged cross-sectional view showing the lower end of an eccentric sleeve and its vicinity in a gyration-type crusher according to Variant 2 of Embodiment 2.

FIG. 21 is an enlarged cross-sectional view showing a lower bearing and its vicinity in a gyration-type crusher according to Embodiment 3 of the present invention.

FIG. 22 is a vertical cross-sectional view showing the overall configuration of an example of conventional gyration-type crushers.

Description of Embodiments

[0038] Hereinafter, embodiments of a gyration-type crusher according to one aspect of the present invention will be described with reference to the drawings.

[0039] The basic configuration of the gyration-type crusher of each embodiment is the same as that of the gyration-type crusher shown in FIG. 22. In the following description, the elements identical to those shown in FIG. 22 are denoted by the reference signs as used in FIG. 22 and may not be described repeatedly. The following description focuses on differences of the configuration of each embodiment from the configuration shown in FIG. 22. In each embodiment described below, a hydraulic cone crusher is taken as an example as in the description given above with reference to FIG. 22. It should be appreciated that the gyration-type crushers of the embodiments are not limited to hydraulic cone crushers and that the

present invention is applicable also to gyratory crushers and other forms of gyration-type crushers.

[0040] [Embodiment 1]

FIG. 1 is a vertical cross-sectional view showing the overall configuration of a gyration-type crusher according to Embodiment 1 of the present invention. The hydraulic cone crusher of the present embodiment includes a main shaft 5 disposed in the center of an internal space defined by a frustum-shaped, tubular upper frame 1 and a lower frame 2 coupled to the upper frame 1 (in other words, the main shaft 5 is disposed inside a concave 14). The central axis L1 of the main shaft 5 is tilted with respect to the central axis L2 of the crusher (the upper frame 1).

[0041] The upper end of the main shaft 5 is rotatably supported by an upper bearing 17. The lower end of the main shaft 5 is rotatably supported by a lower bearing 15. The lower bearing 15 includes an eccentric sleeve 4 including a main shaft insertion hole 3 into which the lower end of the main shaft 5 is rotatably inserted. The eccentric sleeve 4 is equipped with an eccentric sleeve support 32 supporting the eccentric sleeve 4 from below in a manner permitting relative rotation of the eccentric sleeve 4. The eccentric sleeve support 32 is secured to the lower frame 2. The outer circumferential surface of the main shaft 5 inserted into the main shaft insertion hole 3 and the inner circumferential surface of the eccentric sleeve 4 (main shaft insertion hole 3) constitute a radial sliding bearing (main shaft bearing 10) with a predetermined gap between the two surfaces. A lubricating oil is supplied to the predetermined gap, in which an oil film is formed.

[0042] The outer circumferential surface of the eccentric sleeve 4 is rotatably inserted into an eccentric sleeve insertion hole 27 of an outer barrel 7 mounted on the lower frame 2. The outer circumferential surface of the eccentric sleeve 4 and the inner circumferential surface of the outer barrel 7 (eccentric sleeve insertion hole 27) constitute a radial sliding bearing (eccentric sleeve bearing 11) with a predetermined gap between the two surfaces. A lubricating oil is supplied to the predetermined gap, in which an oil film is formed. Hereinafter, for convenience of explanation, the main shaft bearing 10 and eccentric sleeve bearing 11 may be collectively referred to as “lower bearing 15” without distinction between the bearings 10 and 11.

[0043] The main shaft 5 is rotatably supported by the upper and lower bearings 17 and 15 as described above, and is thus capable of performing eccentric gyratory motion with the central axis L1 tilted with respect to the central axis L2 of the concave 14. A hydraulic cylinder 30 is disposed below the lower bearing 15. Thus, the main shaft 5 is movable upward and downward by hydraulic pressure. The eccentric gyratory motion and up-down movement of the main shaft

5 allow a crushing target material 9 to be crushed in a crushing chamber 16 defined by a mantle 13 mounted on the main shaft 5 and the concave 14.

[0044] The following describes the details of the configuration of the lower bearing 15 of embodiments of the present invention. In the present embodiment, as shown in FIGS. 1 and 18, the inner circumferential surface 4a of the main shaft insertion hole 3 (eccentric sleeve 4) has a first tapered shape such that the diameter of the inner circumferential surface 4a increases upward.

[0045] The main shaft 5 or frame 31 is displaced or deformed due to changes in crushing load and hence in reaction force which are caused by changes in the type and properties (such as the constituents, size, and moisture content) of the crushing target material or by changes in operation conditions (such as the rotational speed and the amount of the crushing target material introduced). This leads to changes in the state of the lower bearing 15.

[0046] In particular, in the hydraulic gyration-type crusher in which the upper end of the main shaft 5 is supported by the upper bearing 17, the states into which the lower bearing 15 can be brought due to crushing load changes caused by changes in factors such as the type and properties of the crushing target material are broadly classified into three states shown in FIGS. 3A to 3C.

[0047] FIGS. 3A to 3C are diagrams for extracting and explaining specific motions or behaviors of the lower bearing 15 and schematically show three different relationships between the minimum oil film thickness T and the state of the lower bearing 15 which is affected by the magnitude of the bearing load F which changes as a function of the magnitude of the crushing load W . FIG. 3A shows the situation where the lower bearing 15 of the gyration-type crusher is in a lower partial contact state. FIG. 3B shows the situation where the lower bearing 15 of the gyration-type crusher is in a uniform contact state. FIG. 3C shows the situation where the lower bearing 15 of the gyration-type crusher is in an upper partial contact state. FIG. 4A is an enlarged vertical cross-sectional view showing the lower bearing 15 in the case where the shaft 41 of FIG. 3C is the main shaft 5. FIG. 4B is an enlarged vertical cross-sectional view showing the lower bearing 15 in the case where the shaft 41 of FIG. 3C is the eccentric sleeve 4.

For convenience of explanation, the lower bearing 15 that does not have a tapered shape is shown in FIGS. 3A to 3C and FIGS. 4A and 4B. The minimum oil film thicknesses in the lower partial contact state, uniform contact state, and upper partial contact state are denoted by T_1 , T_2 , and T_3 , respectively.

[0048] In FIGS. 3A to 3C, the central axis of the shaft 41 is denoted by La , and the central axis of the lower bearing 15 is denoted by Lb . In FIGS. 4A and 4B, the central axis of the main

shaft 5 is denoted by L1, the central axis of the upper bearing 17 is denoted by L2, the central axis of the main shaft insertion hole 3 is denoted by L3, the central axis of the eccentric sleeve 4 is denoted by L4, and the central axis of the eccentric sleeve insertion hole 27 is denoted by L5.

[0049] Hereinafter, each of the main shaft bearing 10 and the eccentric sleeve bearing 11 will be described individually.

[0050] In FIGS. 3A to 3C, assuming that the lower bearing 15 is the main shaft bearing 10, the shaft 41 corresponds to the main shaft 5, and the central axis La of the shaft 41 corresponds to the central axis L1 of the main shaft 5. The central axis Lb of the bearing 15 corresponds to the central axis L3 of the main shaft insertion hole 3 (see FIG. 4A). In this case, FIG. 3B shows the state where the central axis L1 of the main shaft 5 which corresponds to the axis La is substantially parallel to the central axis L3 of the main shaft insertion hole 3 which corresponds to the axis Lb and is close to the right inner surface of the main shaft insertion hole 3 in the plane of the figure. The state of FIG. 3B is one in which an oil film with a substantially uniform thickness is formed over the entire axial length of the bearing (uniform contact state). The bearing load F increases and decreases with increases and decreases in crushing load. The state of FIG. 3B can be considered a state where the shaft 41 is somewhat elastically deformed by a crushing load F_0 imposed on the portion of the shaft 41 that is located between the upper and lower bearings 17 and 15 (state where the central axis La of the shaft 41 is somewhat bent at the central portion of the shaft 41). Considering the state of FIG. 3B as a reference, the amount of displacement or deformation of components such as the main shaft 5 is small when the bearing load F is smaller than the bearing load F_0 which causes the lower bearing 15 to enter the uniform contact state. That is, the amount of bending of the central axis La of the shaft 41 is smaller than in the state of FIG. 3B. Thus, unlike in the state of FIG. 3B, the central axis L1 (La) of the lower end of the main shaft 5 is tilted counterclockwise in the plane of the figure with respect to the central axis L3 (Lb) of the main shaft insertion hole 3. This results in the lower partial contact state as shown in FIG. 3A, where the shaft 41 is close to the lower end of the lower bearing 15. Considering the state of FIG. 3B as a reference, the amount of displacement or deformation of components such as the main shaft 5 is large when the bearing load F is larger than the bearing load F_0 which causes the lower bearing 15 to enter the uniform contact state. That is, the amount of bending of the central axis La of the shaft 41 is larger than in the state of FIG. 3B. Thus, unlike in the state of FIG. 3B, the central axis L1 (La) of the lower end of the main shaft 5 is tilted clockwise in the plane of the figure with respect to the central axis L3 (Lb) of the main shaft insertion hole 3. This results in the upper partial contact state as shown in FIG. 3C, where the shaft 41 is close to the upper end of the lower bearing 15.

[0051] The main shaft 5 is pressed by the bearing load F toward the inner circumference of the frame 31 (rightward in the planes of FIGS. 3A to 3C). Thus, in general, the region where the oil film thickness is minimum is on the side of the main shaft bearing 10 that faces toward the inner circumference of the frame 31. As shown in FIGS. 3A to 3C, the oil film thickness in the main shaft bearing 10 is minimum at the lower end of the side opposite that exposed to the bearing load F in the lower partial contact state, over the entire axial length of the opposite side in the uniform contact state (in this state, the oil film thickness is substantially uniform), and at the upper end of the opposite side in the upper partial contact state. The minimum oil film thickness T is the largest in the lower partial contact state, the second largest in the uniform contact state, and the smallest in the upper partial contact state.

[0052] In FIGS. 3A to 3C, assuming that the lower bearing 15 is the eccentric sleeve bearing 11, the shaft 41 corresponds to the eccentric sleeve 4. FIG. 3B shows the state where the central axis $L4$ of the eccentric sleeve 4 is substantially parallel to the central axis $L5$ of the eccentric sleeve insertion hole 27 (see FIG. 4A) and is close to the right inner surface of the eccentric sleeve insertion hole 27 in the plane of the figure. The state of FIG. 3B is one in which an oil film with a substantially uniform thickness is formed over the entire axial length of the bearing (the uniform contact state). Considering the state of FIG. 3B as a reference, when the bearing load F is smaller, the central axis $L4$ of the eccentric sleeve 4 is tilted counterclockwise in the plane of the figure with respect to the central axis $L5$ of the eccentric sleeve insertion hole 27, as in the case of the main shaft bearing 10. This results in the lower partial contact state as shown in FIG. 3A, where the shaft 41 is close to the lower end of the lower bearing 15. Considering the state of FIG. 3B as a reference, when the bearing load F is larger, the central axis $L4$ of the eccentric sleeve 4 is tilted clockwise in the plane of the figure with respect to the central axis $L5$ of the eccentric sleeve insertion hole 27. This results in the upper partial contact state as shown in FIG. 3C, where the shaft 41 is close to the upper end of the lower bearing 15.

[0053] For the eccentric sleeve bearing 11, the minimum oil film thickness regions and the magnitudes of the minimum oil film thicknesses T in the lower partial contact state, uniform contact state, and upper partial contact state are the same as in the case where the shaft 41 is assumed to be the main shaft bearing 10.

[0054] In FIGS. 3A to 3C and FIG. 4, the size of the gap between the outer circumferential surface of the main shaft 5 and the inner circumferential surface of the eccentric sleeve 4 (main

shaft insertion hole 3) and the size of the gap between the outer circumferential surface of the eccentric sleeve 4 and the inner circumferential surface of the outer barrel 7 (eccentric sleeve

insertion hole 27) are exaggerated for ease of understanding.

[0055] The above three states of the lower bearing 15 which occur depending on the magnitude of the crushing load are summarized in Table 1 below.

[0056] [Table 1]

Crushing load	Small	Medium	Large
Bearing load F	Small	Medium	Large
Angle of tilt of La with respect to Lb	< 0	$\cong 0$	> 0
State of contact	Lower partial contact	Uniform contact	Upper partial contact
Minimum oil film thickness T	Large	Medium	Small

[0057] The primary reason why the lower bearing 15 of the gyration-type crusher of the present embodiment has the property as described above, i.e., the property of shifting toward the upper partial contact state with increasing crushing load and shifting toward the lower partial contact state with decreasing crushing load, is that the point of local contact of the shaft 41 with the lower bearing 15 shifts from the lower end to the upper end of the lower bearing 15 with changing crushing load. The shift of the contact point occurs due to the main shaft 5 being deformed (elastically deformed) with the lower and upper bearings 15 and 17 serving as support points under the effect of the bearing load F acting on the middle portion of the main shaft 5 that is located between the lower and upper bearings 15 and 17.

[0058] The amount of the elastic deformation or displacement of the main shaft 5 is highly dependent on the distance between the centers of the upper bearing 17 and the lower bearing 15 (main shaft bearing 10 or eccentric sleeve bearing 11) and the bending stiffness of the main shaft 5 which depends on parameters such as the diameter of the main shaft 5. Assuming that the crushing load is constant, for example, the amount of the deformation or displacement of the main shaft 5 increases with increasing distance between the centers of the upper and lower bearings 17 and 15. The amount of the deformation or displacement of the main shaft 5 decreases, for example, as the diameter of the lower end of the main shaft 5 inserted into the main shaft insertion hole 3 or the diameter of the bottom surface of the mantle 13 increases.

[0059] Thus, in general, gyration-type crushers are so structured that the lower bearing 15 is likely to be in the upper partial contact state. As such, seizure of the lower bearing 15 often occurs in the upper partial contact state. In particular, gyration-type crushers for use as primary or secondary crushers are so structured that the bearing center-to-center distance (the distance between the centers of the upper and lower bearings 17 and 15) is long relative to the diameter of the main shaft. Thus, in such gyration-type crushers, the lower bearing 15 is likely to be in a

severe upper partial contact state under increased crushing load.

[0060] The minimum oil film thickness decreases as the lower bearing 15 shifts from the uniform contact state to the upper partial contact state due to increasing crushing load (reaction force) and hence increasing amount of the displacement or deformation of the components such as the main shaft 5 and frame 31. Thus, in the upper partial contact state, the oil film pressure distribution in the lower bearing 15 has a peak at the upper end of the lower bearing 15 as shown in FIG. 9.

[0061] As the lower bearing 15 shifts from the lower partial contact state to the upper partial contact state, a support point (reaction point) at which the lower bearing 15 bears the crushing load-induced reaction force acting on the main shaft 5 (the middle portion of the main shaft 5) shifts from the lower end to the upper end of the lower bearing 15. This leads to a reduced distance between the point of the main shaft 5 on which the crushing load-induced reaction force acts and the support point of the lower bearing 15. Thus, in the upper partial contact state, the bearing load acting on the lower bearing 15 tends to be larger than in the lower partial contact state and uniform contact state even when the crushing load-induced reaction force acting on the main shaft 5 is the same as in the lower partial contact state and uniform contact state. As such, the bearing is exposed to harsh conditions.

[0062] As seen from the above, in the hydraulic gyration-type crusher including the main shaft 5 supported by the upper and lower bearings 17 and 15, it is desired to avoid the upper partial contact state. To this end, in the present embodiment, the inner circumferential surface of the lower bearing 15 has a tapered shape such that the diameter of the inner circumferential surface increases upward. Specifically, as shown in FIG. 1 and FIG. 18 which is an enlarged view showing the lower bearing 15 of FIG. 1 and its vicinity, the tapered shape includes the first tapered shape such that the diameter of the inner circumferential surface 4a of the main shaft insertion hole 3 of the eccentric sleeve 4 increases upward.

[0063] In the example of FIG. 3C, the tapered inner circumferential surface of the lower bearing 15 is shown by a dashed line 42. For the dashed line 42 in FIG. 3C and the inner circumferential surface 4a of the main shaft insertion hole 3 in FIGS. 1 and 18, the magnitude of the cone angle is exaggerated for clarity of illustration. In the configuration of FIGS. 1 and 18, which includes the tapered shape as described above, the shaft 41 and the upper end of the inner

circumferential surface 42 of the lower bearing 15 are prevented from being close to each other even in the situation where the crushing load is so large that the upper partial contact state would occur in conventional configurations. Referring back to FIGS. 3A to 3C, the minimum oil film thickness would be the smallest in the example of FIG. 3C in the absence of the tapered shape.

However, with the tapered shape, the minimum oil film thickness in the state of FIG. 3C can remain almost unchanged from that in the uniform contact state shown in FIG. 3B.

[0064] By virtue of the tapered shape (first tapered shape) as described above, the upper partial contact state of the lower bearing 15 can be avoided, and the reduction in the minimum oil film thickness T3 can be prevented. Thus, a gyration-type crusher that prevents the occurrence of failures such as seizure in the lower bearing 15 and that has high robustness against changes in load conditions can be constructed in a simple configuration.

[0065] The following describes the design parameters for the lower bearing 15 in the gyration-type crusher that can be in the states as shown in FIGS. 3A to 3C. In general, when the ratio of the bearing length to the shaft diameter (L/D) is in the range of about 0.5 to 2, the Sommerfeld number S is from about 0.0001 to 0.1, and the minimum oil film thickness is from about several micrometers to several hundreds of micrometers. The Sommerfeld number S is a dimensionless quantity used to evaluate the state of lubrication between a sliding bearing and a shaft which are lubricated with a fluid such as an oil (fluid lubrication), and is an evaluation indicator representing the oil film performance of a fluid lubrication bearing. The Sommerfeld number S is calculated by the following equation (1).

$$[0066] \quad S = (\eta n / P)(r / c)^2 \quad (1)$$

[0067] In this equation, η is the viscosity coefficient [Pa·s] of the lubricating oil, n is the shaft rotational speed [s⁻¹], P is the bearing specific pressure [Pa], and r is the shaft radius [m], and c is the bearing clearance [m] ($c = R - r$, R : bearing radius, r : shaft radius).

[0068] The cone angle of the above tapered shape is set to provide a robust zone described later in regard to changes in minimum oil film thickness of the lubricating oil with changes in crushing load. Specifically, for example, when the tapered shape is viewed in cross-section along the central axis L1 of the main shaft 5, the angle θ_t between the shaft 41 and the inner circumferential surface 42 of the lower bearing 15 (cone angle) is from 0.001 to 1°. That is, in the present embodiment, the cone angle θ_t , which is an angle indicative of the ratio of the amount of change in the diameter D_t of the region of the tapered shape in the lower bearing 15 to the length L_t of the region of the tapered shape in the direction of the central axis L_b is from 0.001 to 1°. In other words, the rate of taper, which is the ratio of the amount of change in the diameter D_t of the region of the tapered shape in the lower bearing 15 to the length L_t of the region of the tapered shape in the direction of the central axis, is from 2/100000 to 2/100. As shown in FIG. 18, the length L_t in the present embodiment is equal to the axial length of the eccentric sleeve 4. The diameter D_t of the region of the tapered shape is equal to the diameter

of the inner circumferential surface 4a of the main shaft insertion hole 3. The rate of taper is expressed as $\Delta Dt/Lt$. ΔDt represents the difference between the maximum diameter Dt_max and minimum diameter Dt_min ($\Delta Dt = Dt_max - Dt_min$) in the region of the tapered shape.

[0069] The following describes the relationship of minimum oil film thickness versus crushing load and the relationship of tilt angle versus crushing load in the gyration-type crusher which includes the tapered shape whose cone angle is set as described above. The relationships described below are those found by analysis for the case where the lower bearing 15 has specifications A (ratio L/D of bearing length to shaft diameter = about 1.4 and Sommerfeld number S = about 0.001) and the case where the lower bearing 15 has specifications B (L/D = about 0.8 and S = about 0.01).

[0070] FIG. 5 is a graph showing changes in minimum oil film thickness of the lower bearing 15 with changes in crushing load for the specifications A. In the case of FIG. 5, first, the amounts of deformation or displacement of the structural components such as the main shaft 5 and the frame 31 (upper and lower frames 1 and 2) in the specifications A were determined by structural analysis such as FEM (finite element method) or BEM (boundary element method). The determined values were used to find the oil film thickness of the lower bearing 15 in the specifications A by oil film analysis using the Reynolds equation based on fluid lubrication theory. The results were organized and plotted on the graph. FIG. 6 is a graph showing changes in tilt angle of the lower bearing 15 with changes in crushing load for the specifications A. FIG. 7 is a graph showing changes in minimum oil film thickness of the lower bearing 15 with changes in crushing load for the specifications B. The graph of FIG. 7 was created by finding the oil film thickness in the specifications B in the same manner as for the graph of FIG. 5. FIG. 8 is a graph showing changes in tilt angle of the lower bearing 15 with changes in crushing load for the specifications B.

[0071] The results of the structural analysis and oil film analysis are desirably evaluated for their validity by comparison with the bearing state (e.g., sliding marks) in a test machine or a practical machine. The oil film analysis is conducted using an analysis technique that takes into account the deformation or tilt of the shaft and bearing. Ideally, the structural analysis and oil film analysis are carried out using an analysis technique in which the structural analysis and oil film analysis are bidirectionally coupled. However, it is common practice to employ what may be called unidirectional coupling analysis in which the oil film analysis is conducted using the results of the structural analysis as described above.

[0072] For the validity evaluation of the analysis technique, it is effective, for example, to use a method in which the parameters determined by the analysis, such as the partial contact state

(specific pressure distribution) and the minimum oil film thickness, are examined in relation to sliding marks formed during operation of a practical machine.

[0073] In FIGS. 5 to 8, the crushing load on the abscissa is normalized by assuming a rated load of 100%.

[0074] The term “rated load”, as used for a gyration-type crusher operable with the rated output of the electric motor that drives the gyration-type crusher, refers to a crushing load that can be generated by the gyration-type crusher crushing the introduced raw material (e.g., rocks) with the rated output. The term “rated load”, as used for a gyration-type crusher that is not operable with the rated output of the electric motor because the crushing load that can be generated during crushing performed with the rated output of the electric motor is above the maximum load that can be continuously endured by the body or a component of the gyration-type crusher, refers to a crushing load generated with the maximum output at which the crushing can be continued safely (in this case, the maximum output is regarded as the “rated output”).

[0075] In general, cone crushers are designed based on the premise that they continue to be operated for continuous crushing. In contrast, gyratory crushers for use as primary crushers may not only be operated for continuous crushing but also be routinely used for single-particle crushing or intermittent crushing of large pieces of raw material (specific examples include stones). The above definition of the rated load applies also to gyration-type crushers such as gyratory crushers which are intermittently operated.

[0076] In FIGS. 5 and 7, the minimum oil film thickness on the ordinate is normalized by assuming that the minimum oil film thickness of the lower bearing 15 is 1 at a crushing load of 100%.

[0077] In FIGS. 6 and 8, the tilt angle on the ordinate is plotted on the supposition that the direction in which the shaft 41 is tilted clockwise in the plane of the figure with respect to the central axis L2 of the lower bearing 15 (the direction toward the upper partial contact state) is the positive direction. Additionally, in FIGS. 6 and 8, the tilt angle on the ordinate is normalized by assuming that the absolute value of the tilt angle is 1 at a crushing load of 50%. As for the positive and negative signs attached to the normalized tilt angle, the negative sign (−) indicates that the lower bearing 15 is in the lower partial contact state, and the positive sign (+) indicates that the lower bearing 15 is in the upper partial contact state.

[0078] In general, as shown in FIGS. 6 and 8, the tilt angle in the lower bearing 15 monotonically increases with increasing crushing load in such a fashion that the line representing the tilt angle is substantially straight or gently curved. In general, as shown in FIGS. 5 and 7,

the minimum oil film thickness in the lower bearing 15 decreases substantially monotonically on the whole with increasing crushing load. In the present embodiment, the rate of change (decrease) in minimum oil film thickness in the lower bearing 15 with increasing crushing load is lower when the crushing load is within a specific range than when the crushing load is outside the specific range. More specifically, for the lower bearing 15 of FIG. 5 which has the specifications A, the minimum oil film thickness decreases as the crushing load increases from 50%, and the rate of decrease in minimum oil film thickness continuously decreases with increasing crushing load. This trend continues until the rate of decrease in minimum oil film thickness with increasing crushing load sharply increases at a crushing load of about 105%. Likewise, for the lower bearing 15 of FIG. 7 which has the specifications B, the minimum oil film thickness decreases as the crushing load increases from 50%, and the rate of decrease in the minimum oil film thickness continuously decreases with increasing crushing load. This trend continues until the rate of decrease in minimum oil film thickness with increasing crushing load sharply increases at a crushing load of about 145%.

[0079] The above specific range, over which the rate of change (typically, decrease) in minimum oil film thickness with increasing crushing load is lower than over the other crushing load ranges and at the upper limit of which the rate of change in minimum oil film thickness sharply increases, is herein referred to as “robust zone”. For the lower bearing 15, the property of exhibiting such a robust zone is referred to as “robust property”. As shown in FIGS. 5 and 7, the change in minimum oil film thickness with increasing crushing load is gradual while the crushing load increases from a low value toward the upper crushing load limit in the robust zone. Thus, it is often the case that the lower crushing load limit in the robust zone cannot be unambiguously determined. In contrast, the upper crushing load limit in the robust zone can be determined based on the above characteristic phenomenon where the rate of decrease in minimum oil film thickness with increasing crushing load, which is initially low, sharply increases at a certain point. Specifically, for example, the upper crushing load limit in the robust zone is about 105% for the lower bearing 15 having the specifications A. For the specifications B, the upper crushing load limit in the robust zone is about 145%. The mathematical method of determining the upper crushing load limit in the robust zone will be described later.

[0080] In the case of the specifications A, as shown in FIG. 6, the tilt angle of the lower bearing 15 changes from negative to positive values at a crushing load of about 100%. Thus, for the specifications A, it can be thought that the lower bearing 15 is in the uniform contact state when the crushing load is about 105%. As such, the lower bearing 15 is in the lower partial

contact state when the crushing load is smaller than about 105% and in the upper partial contact state when the crushing load is larger than about 105%. Likewise, in the case of the specifications B, the tilt angle of the lower bearing 15 changes from negative to positive values at a crushing load of about 145% as shown in FIG. 8. Thus, for the specifications B, it can be thought that the lower bearing 15 is in the uniform contact state when the crushing load is about 145%. As such, the lower bearing 15 is in the lower partial contact state when the crushing load is smaller than about 145% and in the upper partial contact state when the crushing load is larger than about 145%.

[0081] For comparison, an analysis result of the oil film pressure distribution in the partial contact state is shown in FIG. 9. An analysis result of the oil film pressure distribution in the uniform contact state is also shown in FIG. 10. The specifications of the gyration-type crusher and the bearing load are the same for FIGS. 9 and 10. The tilt angle of the shaft 41 is 0.015 degrees in FIG. 9 and 0 degrees in FIG. 10, and the scale of the pressure distribution is the same for the two figures.

[0082] As shown in FIG. 10, the pressure distribution in the uniform contact state does not have any pronounced peak in the axial direction, and the pressure is generally low and uniform.

[0083] As the crushing load changes from the lower limit to the upper limit in the robust zone with increasing drive power of the motor rotationally driving the main shaft 5, the lower bearing 15, i.e., at least one of the main shaft bearing 10 or eccentric sleeve bearing 11, shifts from the lower partial contact state to the uniform contact state. Thus, the region where the oil film thickness of the lubricating oil is minimum shifts from the lower end of the bearing to the entire vertical length of the bearing. That is, along with the shift of the lower bearing 15 from the lower partial contact state to the uniform contact state, the oil film pressure distribution of the lower bearing 15 changes from a distribution where the oil film pressure is locally high around the lower end of the lower bearing 15 toward a distribution where the oil film pressure is even over the entire vertical length of the bearing.

[0084] As the crushing load further increases and exceeds the upper limit in the robust zone, the bearing 15 shifts to the upper partial contact state. Thus, the region where the oil film thickness is minimum shifts to the upper end of the lower bearing 15. Along with the shift from the uniform contact state to the upper partial contact state, the oil film pressure distribution changes from the distribution where the oil film pressure is even over the entire vertical length of the bearing to a distribution as shown in FIG. 9 where the oil film pressure increases sharply and locally around the upper end of the lower bearing 15.

[0085] In the partial contact state of FIG. 9, the minimum oil film thickness is reduced to

about 13% of the minimum oil film thickness in the uniform contact state of FIG. 10. This leads to the conclusion that the uniform contact state is advantageous for increasing the minimum oil film thickness under the same load conditions and the same specifications. In contrast, in the partial contact state, in particular the upper partial contact state, the bearing is exposed to harsh conditions because of the small minimum oil film thickness.

[0086] However, in the present embodiment, the lower bearing 15 has the robust zone over which the lower bearing 15 gradually shifts from a mild lower partial contact state to the uniform contact state with increasing crushing load, and the sensitivity of the minimum oil film thickness to changes in crushing load is lower in the robust zone than in the other crushing load ranges. Thus, the lower bearing 15 is characterized by being able to ensure a sufficient minimum oil film thickness.

[0087] Hereinafter, the features of a bearing having the robust property will be described in detail.

[0088] FIG. 11 shows comparison between a bearing devoid of the robust property and a bearing having the robust property. The graph 11A of FIG. 11 shows changes in minimum oil film thickness with changes in crushing load for the different bearings. The graph 11B of FIG. 11 shows changes in tilt angle with changes in crushing load for the different bearings. In FIG. 11, the crushing load is normalized by assuming a rated load of 100%, the minimum oil film thickness is normalized by assuming that the minimum oil film thickness is 1 when the crushing load is 100% of the rated load, and the tilt angle is normalized by assuming that the absolute value of the tilt angle is 1 when the crushing load is 20% of the rated load. In FIG. 11, the minimum oil film thickness and tilt angle are shown in a simplified fashion for ease of explanation and understanding. As for the robust zone in the bearing having the robust property, the upper crushing load limit in the robust zone is set to 120% to clarify the difference between the bearing having the robust property and the bearing devoid of the robust property.

[0089] For example, when the gyration-type crusher performs crushing operation with the crushing load set to the rated load, the magnitude of the crushing load during the operation fluctuates due to variations in the amount, shape, size, and nature of the raw material introduced into the crushing chamber 16. Thus, once the crushing load exceeds the rated load (e.g., by 5%), the tilt angle of the bearing increases and accordingly the bearing shifts to the partial contact state (upper partial contact) regardless of the presence or absence of the robust property (graph 11B of FIG. 11). As seen from the graph of the tilt angle, the bearing devoid of the robust property is in the upper partial contact state when the crushing load is 50% or more. The bearing having the robust property is in the lower partial contact state at a crushing load of 50%,

enters a complete uniform contact state at a crushing load of 120%, and shifts to the upper partial contact state once the crushing load exceeds 120%.

[0090] The minimum oil film thickness decreases substantially monotonically on the whole with increasing crushing load regardless of whether the bearing has the robust property. The bearing devoid of the robust property is already in the upper partial contact state at a crushing load of 50%. As the crushing load increases beyond 50%, the tilt angle increases and accordingly the bearing shifts to a severer upper partial contact state. Consequently, for the bearing devoid of the robust property, the minimum oil film thickness monotonically decreases without a significant change in the rate of decrease with increasing crushing load. For the bearing having the robust property, the minimum oil film thickness decreases as the crushing load increases from 50%; however, in the robust zone, the bearing exhibits the robust property, and the rate of change (typically, decrease) in minimum oil film thickness continuously decreases with increasing crushing load. Referring to FIG. 11, this trend continues until the rate of decrease in minimum oil film thickness with increasing crushing load sharply increases at a crushing load of about 105%. In the example of FIG. 11, in particular, the crushing load range from about 80 to 120% is the robust zone where the bearing exhibits the robust property. In the example of FIG. 11, considering changes in crushing load in relation to the partial contact state of the bearing, the crushing load range where the bearing shifts from a mild lower partial contact state to the uniform contact state is the robust zone where the robust property is exhibited in regard to the minimum oil film thickness.

[0091] In gyration-type crushers, as seen from the above, the robust zone where the robust property is exhibited in regard to changes in minimum oil film thickness with changes in crushing load can be created by adjusting the points at which shifts between different states of partial contact occur. It is seen that in the robust zone, the bearing can ensure the stability of the oil film performance against variations in crushing load.

[0092] As previously described, the robust zone is formed in the crushing load range where the bearing shifts from a mild lower partial contact state to the uniform contact state. Although in the example described with reference to FIG. 11, the bearing devoid of the robust zone is a bearing that is always in the upper partial contact state, a bearing that cannot be in a mild lower partial contact state but is always in a relatively severe lower partial contact state can also be considered devoid of the robust zone.

[0093] In the bearing having the robust property, the sensitivity of the minimum oil film thickness to changes in crushing load (the rate of change in minimum oil thickness with changing crushing load) is the lowest at the upper crushing load limit in the robust zone or at a

crushing load slightly smaller than the upper crushing load limit. In general, the minimum oil film thickness monotonically decreases with increasing crushing load. However, in some cases, the rate of change in minimum oil film thickness is zero at a crushing load slightly smaller than the upper crushing load limit in the robust zone. In such a case, the minimum oil film thickness may slightly increase as the crushing load increases from the value at which the rate of change in minimum oil film thickness is zero to the upper crushing load limit in the robust zone, and may begin to decrease with increasing crushing load once the crushing load exceeds the upper limit. However, this behavior is imperceptible and can occur only under limited conditions. As such, the minimum oil film thickness can generally be considered to monotonically decrease with increasing crushing load.

[0094] Sliding marks in a bearing having the robust property differ from those in a bearing devoid of the robust property because between these bearings there is the above-described difference in minimum oil film thickness due to the presence or absence of the robust property. The following describes the difference in sliding marks between the two bearings.

[0095] In a common gyration-type crusher, if the shaft, bearing, and lubricating oil are in good condition, sliding motion involving slight contact does not cause an immediate seizure, because of the material properties of the bearing or the effect of agents such as the extreme-pressure additive in the lubricating oil. In many cases, slight contact experienced by the bearing leads to natural formation of desired crowning around the ends of the bearing or reduction in roughness of the bearing surface. Thus, the quality of the bearing is improved compared to that in the crusher as newly produced, and the bearing becomes able to work well in the event of a severe partial contact state or a small oil film thickness. This process is commonly called “breaking-in” or “running-in”. During this process, some sliding marks are formed on the surface of the shaft or bearing. Even in the case where the bearing oil film formed is in good condition, contamination of the lubricating oil by foreign matter whose size or amount is non-negligible relative to the oil film thickness could cause formation of sliding marks such as linear marks and abrasion marks or formation of foreign matter biting marks.

[0096] As previously stated, the lower bearing 15 in the present embodiment exhibits the robust property in regard to the minimum oil film thickness while shifting from a mild lower partial contact state to the uniform contact state.

[0097] In the robust zone, smooth sliding marks extending over a relatively wide area are formed, rather than localized coarse sliding marks. In the case where the formation of the sliding marks in the robust zone is due to minute foreign matter contained in the lubricating oil, the foreign matter acts like an abrasive to form the sliding marks (abrasion marks) over a

relatively wide area.

[0098] When the lower bearing 15 is in a mild lower partial contact state, the oil film thickness is minimum at the lower end of the lower bearing 15 and gradually changes (typically, decreases) upward. Thus, sliding marks are likely to be formed over a region from the lower end of the lower bearing 15 to about 1/5 to about 1/3 of the axial length of the lower bearing 15. When the lower bearing 15 is in the uniform contact state, continuous sliding marks are likely to be formed over a region between the region from the lower end of the lower bearing 15 to about 1/5 to 1/3 of the axial length of the lower bearing 15 and a region from the upper end of the lower bearing 15 to about 1/5 to 1/3 of the axial length of the lower bearing 15. Once the lower bearing 15 shifts from the uniform contact state to the upper partial contact state, the lower bearing 15 loses the robust property due to the exceeding of the upper crushing load limit in the robust zone. When the bearing is in the upper partial contact state, the oil film thickness is minimum at the upper end of the lower bearing 15 and increases downward. Thus, sliding marks are likely to be formed over the region from the upper end of the lower bearing 15 to about 1/5 to 1/3 of the axial length of the lower bearing 15.

[0099] Thus, in the lower bearing 15 of the present embodiment which has the robust zone, the sliding marks in the uniform contact state and the sliding marks in the upper partial contact state are formed as the lower bearing 15 shifts from the uniform contact state to the upper partial contact state with increasing crushing load from the upper crushing load limit in the robust zone.

[0100] For the above reasons, in the lower bearing 15 having the robust property, failures including seizure due to causes such as defect of the oil film are not likely to occur despite variations in crushing load, and smooth sliding marks are likely to be formed over a relatively wide area in the axial direction of the lower bearing 15. The minimum oil film thickness in the lower bearing 15 having the robust property varies as the lower bearing 15 shifts between the different states of partial contact in response to the crushing load; specifically, the minimum oil film thickness T2 in the uniform contact state and the minimum oil film thickness T1 in the lower partial contact state are larger than the minimum oil film thickness T3 in the upper partial contact state. Thus, in the case of the lower bearing 15 having the robust property, the oil film is in better condition and sliding marks are less likely to be formed in a mild lower partial contact state or the uniform contact state than in the upper partial contact state. Therefore, the lower bearing 15 having the robust property can be free of sliding marks in some cases, and even in the case where sliding marks are formed, the sliding marks are relatively insignificant.

[0101] Outside the robust zone, the bearing is in the lower partial contact state where the tilt is relatively large or in the upper partial contact. In the lower partial contact state occurring

outside the robust zone, sliding marks are formed locally around the lower end of the lower bearing 15. In the upper partial contact state occurring outside the robust zone, sliding marks are formed in a region from the upper end of the lower bearing 15 to about $1/5$ to $1/3$ of the axial length of the lower bearing 15. In the lower bearing 15 which is in the upper partial contact state occurring outside the robust zone, a further increase in crushing load causes an increase in the severity of the upper partial contact state and a sharp decrease in the minimum oil film thickness, thus increasing the likelihood of local formation of coarse sliding marks. In some cases, not only are sliding marks formed, but also failures can occur, including seizure due to causes such as defect of the oil film.

[0102] A bearing devoid of the robust property, as defined in terms of the minimum oil film thickness versus crushing load relationship in the lower bearing 15 having the robust property, can be considered a bearing that is always in the upper partial contact state or in the lower partial contact state where the tilt is relatively large. In other words, such a bearing devoid of the robust property can be considered a bearing that cannot be in a mild lower partial contact state or the uniform contact state. Thus, in such a bearing, sliding marks are formed which have the characteristics of any one of the different types of sliding marks described above as those formed outside the robust zone, and any sliding marks characteristic of the bearing having the robust zone are not formed.

[0103] In general, sliding marks are not formed when the oil film thickness is sufficiently large or when the lubricating oil is not contaminated by foreign matter whose size or amount is non-negligible relative to the oil film thickness. In such cases, the presence or absence of the robust zone in the bearing cannot be determined based on sliding marks. However, the robust zone can be determined to be present if sliding marks are observed which are described above as arising from the presence of the robust zone.

[0104] In the graph 11A of FIG. 11, where the minimum oil film thickness is normalized, the normalized minimum oil film thickness at the rated load is the same for the bearing having the robust property and the bearing devoid of the robust property. However, given the rate of change in minimum oil film thickness with changing crushing load in the bearing devoid of the robust property, the actual minimum oil film thickness at the rated load is larger for the bearing devoid of the robust property than for the bearing having the robust property.

[0105] The following describes how to determine the upper crushing load limit in the robust zone in the lower bearing 15 having the robust property.

[0106] In the lower bearing 15 having the robust property, as described above with reference to FIGS. 5 and 7, the rate of change in minimum oil film thickness with changing

crushing load clearly changes before and after exceeding of the upper crushing load limit in the robust zone. In FIGS. 5 and 7, the dots (●) indicate values determined through oil film analysis, and the solid line is one drawn by connecting the dots. For the bearing having the robust property, the upper crushing load limit in the robust zone can easily be determined by obtaining the values of the minimum oil film thickness at many values of the crushing load and analyzing the obtained values as in FIGS. 5 and 7. The upper crushing load limit in the robust zone can be determined to be about 105% in FIG. 5 and about 145% in FIG. 7.

[0107] With the use of a mathematical technique, the robust property can be approximated by two fitted curves. Specifically, the robust property can be approximated, for example, by fitted curves of quadratic or cubic functions. The upper crushing load limit in the robust zone can be identified from the point of intersection between the two fitted curves. FIG. 12 is a graph showing a schematic curve representing the robust property of the bearing having the specifications A. The graph of FIG. 12 is the same as that of FIG. 5. FIG. 13 is a graph showing the robust property curve of FIG. 12 as approximated by quadratic functions. FIG. 14 is a graph showing the robust property curve of FIG. 12 as approximated by cubic functions. In the example of FIG. 13, the upper crushing load limit in the robust zone, as determined by finding the point of intersection between the fitted curves, is 104.7%. In the example of FIG. 14, the upper crushing load limit in the robust zone, as determined by finding the point of intersection between the fitted curves, is 105.1%. Likewise, FIG. 15 is a graph showing a schematic curve representing the robust property of the bearing having the specifications B. The graph of FIG. 15 is the same as that of FIG. 7. FIG. 16 is a graph showing the robust property curve of FIG. 15 as approximated by quadratic functions. FIG. 17 is a graph showing the robust property curve of FIG. 15 as approximated by cubic functions. In the example of FIG. 16, the upper crushing load limit in the robust zone, as determined by finding the point of intersection between the fitted curves, is 144.1%. In the example of FIG. 17, the upper crushing load limit in the robust zone, as determined by finding the point of intersection between the fitted curves, is 145.4%.

[0108] The above examples are those where the robust property is relatively explicit. In the case where the difference between the rate of change in minimum oil film thickness with changing crushing load in the robust zone and the rate of change in minimum oil film thickness with changing crushing load outside the robust zone is small, the boundaries of the robust zone are ambiguous. Even in such a case, the curve representing the robust property can be approximated by two fitted curves of quadratic or cubic functions. The bearing is considered to have the robust property in any case where the upper crushing load limit in the robust zone can

be determined from the point of intersection between two fitted curves.

[0109] For a bearing in which the upper crushing load limit in the robust zone is in a significantly high or low load range, the oil film thickness could fall below the allowable oil film thickness before the upper crushing load limit is reached. In such a case, the bearing is not regarded as having the robust property even if the amount of change in minimum oil film thickness with changing crushing load is significantly small in a specific crushing load range.

[0110] The magnitude of the crushing load in the robust zone and the width of the robust zone vary depending on the levels of the stiffness of components such as the frame 31, main shaft 5, and bearing support components (the outer barrel 7 and eccentric sleeve support 32) or on the balance among the stiffness levels. Thus, the stiffness of the components, as well as the crushing load, is a key parameter in designing the robust zone.

[0111] In addition to the parameters mentioned above, the amount of wear of the upper bearing 17 is also important in designing the robust zone. In the hydraulic gyration-type crusher where the main shaft 5 needs to be supported by the upper and lower bearings 17 and 15, the bearing metal used in the upper bearing 17 wears down over time. With the wearing down of the upper bearing 17, the lower bearing 15 becomes more likely to be in the upper partial contact state, and the robust zone varies from that as originally designed or that in the crusher as newly produced. Specifically, for example, as the upper bearing 17 wears down, the robust zone shifts to a lower load range from the load range where the robust zone in the newly-produced crusher free of the wearing down was located. In each of the examples of FIGS. 5 and 7, the curve representing the robust property shifts leftward in the plane of the figure with wearing down of the upper bearing 17.

[0112] In the gyration-type crusher, the raw material may be retained in the crushing chamber 16 defined by the mantle 13 and concave 14 for some reasons. In this case, if it is attempted to continue the operation after disruption of the crushing or discharge step, the rotational motion of the crusher is disturbed by the raw material retained in the crushing chamber 16. This could result in a phenomenon where the load imposed on the gyration-type crusher far exceeds the rated load instantaneously.

[0113] Upon the occurrence of such a phenomenon, a shaft torque larger than that at the rated output of the motor is produced because of the torque characteristics of the motor. Specifically, for example, when the motor is a three-phase induction motor, the maximum torque produced generally corresponds to 160% or more of that produced under the rated load conditions. In this situation, the bearing could be subjected to a bearing load associated with the shaft torque. This could cause a crushing load of 160% or more. It is common practice to

equip the gyration-type crusher with a kind of safety device in order to prevent mechanical damage to the body of the gyration-type crusher. In this case, the upper crushing load limit is preferably 200% or less of the rated load of the gyration-type crusher. The occurrence of a crushing load extremely larger than the rated load leads to overload being imposed on the motor. Thus, the upper crushing load limit is more preferably 160% or less.

[0114] In the case where the above phenomenon occurs, the robust zone may be deliberately set in a range of crushing loads larger than the rated load. This can ensure the reliability against unexpected events. In such a case, a usual crushing load (crushing load usually employed in light of the type or nature of the raw material) or the rated load can be smaller than the lower crushing load limit in the robust zone. As previously stated, the lower bearing 15 tends to be in the lower partial contact state when the crushing load is small. However, as shown in FIGS. 5 and 7, a sufficient minimum oil film thickness $T1$ is likely to be ensured because of the smallness of the crushing load W (bearing load F).

[0115] In plants for crushing relatively soft raw materials, gyration-type crushers are often operated at a crushing load smaller than the rated load, in particular, for example, a crushing load of about 50%. For such operation, the reliability of the bearing during the operation can be enhanced by setting the robust zone in a low crushing load range.

[0116] A gyration-type crusher employing the lower bearing 15 having the features described above can continue to be used while preventing the occurrence of the partial contact states even if the crushing load changes for reasons such as changes in the type of the crushing target material and changes in the operation conditions (including changes in crushing load due to wearing down of the mantle 13 or concave 14). Thus, there is no need to adjust or test the components such as the lower bearing 15 according to changes in crushing load. It is also unnecessary to choose a suitable gyration-type crusher or selectively use different gyration-type crushers according to changes in crushing load. As such, the effort and cost required to address changes in crushing load can be reduced, and the operating rate can be increased.

[0117] The crushing load is substantially proportional to the motor power (shaft torque). In actual operation of gyration-type crushers, the motor power is easier to directly measure and manage than the crushing load. Thus, it is more convenient to use the relationship between the motor power and the minimum oil film thickness in data organization and understanding than to use the relationship between the crushing load and the minimum oil film thickness. In particular, in the above-described results where the crushing load is normalized based on the rated load (rated value), the crushing load can be regarded as (is directly interchangeable with) the motor power.

[0118] The gyration-type crusher of the above embodiment can have a simple configuration and at the same time have high robustness against changes in load conditions.

[0119] In the gyration-type crusher of the above embodiment, an extreme upper partial contact state or lower partial contact of the lower bearing 15 can be avoided.

[0120] In this technical field, a certain degree of upper partial contact or lower partial contact is acceptable. During the operation period of the gyration-type crusher, the occurrence of both a certain degree of upper partial contact and a certain degree of lower partial contact can even be beneficial to avoid an extreme upper partial contact state or lower partial contact state.

[0121] In this context, the formation of partial contact-induced sliding marks in both upper and lower portions of the inner circumferential surface of the lower bearing 15 can be considered to indicate that an ideal operation state is established as in the above embodiment.

[0122] In the robust zone in the above embodiment, the upper crushing load limit is preferably about 70% or more, about 80% or more, or about 100% or more of the rated value of the motor power.

[0123] In the robust zone in the above embodiment, the upper crushing load limit is preferably about 200% or less, about 160% or less, or about 110% or less of the rated value of the motor power.

[0124] The present invention is particularly beneficial for large-sized gyration-type crushers. To be specific, the present invention is advantageous especially in a gyration-type crusher having an inlet size of 200 mm or more. The inlet size refers to the distance between the inner surface of the concave 14 and the upper end of the mantle 13. The inlet size defines the maximum size of raw materials that can be supplied to the gyration-type crusher.

[0125] In the above embodiment, the setting of the robust zone described above can be accomplished by adjusting the cone angle or rate of taper of the tapered shape of the inner circumferential surface 42 of the lower bearing 15. The cone angle θ_t of the tapered shape is preferably from 0.001 to 1°. In other words, the rate of taper is preferably from 2/100000 to 2/100.

[0126] In the above embodiment, the inner circumferential surface 4a of the main shaft insertion hole 3 has the first tapered shape as the tapered shape for avoiding the upper partial contact state, the first tapered shape being such that the diameter of the inner circumferential surface 4a increases upward. However, the present invention is not limited to this tapered shape.

[0127] For example, as shown in FIG. 19A, the tapered shape may include a second tapered shape such that the diameter of the outer circumferential surface 4b of the eccentric sleeve 4

increases downward. Alternatively, as shown in FIG. 19B, the tapered shape may include a third tapered shape such that the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 of the outer barrel 7 increases upward. In these cases, the cone angle θ_t for providing the robust zone is in the same range as that in the case previously described. In the example of FIG. 19A, the length L_t of the region of the tapered shape of the lower bearing 15 in the central axis direction is equal to the axial length of the outer barrel 7. The diameter D_t of the region of the tapered shape is equal to the diameter of the outer circumferential surface 4b of the eccentric sleeve 4. In the example of FIG. 19B, the length L_t of the region of the tapered shape of the lower bearing 15 in the central axis direction is equal to the axial length of the outer barrel 7. The diameter D_t of the region of the tapered shape is equal to the diameter of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27.

[0128] In the cases of the variants shown in FIGS. 19A and 19B, the shaft 41 of FIG. 3C corresponds to the eccentric sleeve 4. In the variants shown in FIGS. 19A and 19B, as in FIG. 3C, the relative relationship between the shaft 41 and the lower bearing 15 is such as to form a shape that allows the avoidance of the upper partial contact state. Thus, also with the second or third tapered shape, the upper partial contact state of the lower bearing 15 can be avoided, and the reduction in minimum oil film thickness T_3 can be prevented. As such, a gyration-type crusher that prevents the occurrence of failures such as seizure in the lower bearing 15 and that has high robustness against changes in load conditions can be constructed in a simple configuration.

[0129] In the variant shown in FIG. 19B, the third tapered shape is formed over the entirety of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 in the axial direction. However, the third tapered shape may be formed over a portion of the inner circumferential surface 7a in the axial direction. For example, as shown in FIG. 19C, the third tapered shape is formed in a region 7a1 extending between the upper axial end of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 and the axial center of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27. Thus, a region 7a2 extending between the axial center of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 and the lower axial end of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 is the region where there is no tapered shape (where the inner circumferential surface extends parallel to the axial direction). In the example of FIG. 19C, the length L_t of the region of the tapered shape of the lower bearing 15 in the central axis direction is equal to 1/2 of the axial length of the outer barrel 7.

[0130] Thus, the third tapered shape is formed in a region including the upper axial end of

the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 and having a length equal to $1/2$ of the axial length of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27. In the case where a tapered shape (third tapered shape) is formed over the entirety of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27, an increased cone angle could increase the degree of partial contact occurring under low loads, leading to the oil film thickness being insufficient in spite of the low loads. When the third tapered shape is formed over a portion of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 in the axial direction, the reduction in the minimum oil film thickness T3 can be prevented both under low loads and under high loads.

[0131] The formation of the third tapered shape over a portion of the inner circumferential surface 7a in the axial direction is not limited to that in the above example. The third tapered shape may be formed in a region including the upper axial end of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 and having a length equal to or greater than $1/3$ of the axial length of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27. For example, the third tapered shape is formed in the region 7a1 having a length equal to $1/3$ of the axial length of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27, and the lower end of this region is spaced downward from the upper end by a distance equal to $1/3$ of the axial length of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27. Alternatively, for example, the third tapered shape is formed in the region 7a1 having a length equal to $2/3$ of the axial length of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27, and the lower end of this region is spaced downward from the upper end by a distance equal to $2/3$ of the axial length of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27. The axial length of the region 7a1 of the third tapered shape is more preferably $1/2$ or more of the axial length of the inner circumferential surface 7a of the eccentric sleeve insertion hole 27.

[0132] In the above embodiment, as shown in FIG. 18, the first tapered shape is formed over the entirety of the inner circumferential surface 4a of the main shaft insertion hole 3 in the axial direction. However, as in FIG. 19C, the first tapered shape may be formed over a portion of the inner circumferential surface 4a in the axial direction. For example, as shown in FIG. 19D, the first tapered shape is formed in a region 4a1 extending between the upper axial end of the inner circumferential surface 4a of the main shaft insertion hole 3 and the axial center of the inner circumferential surface 4a of the main shaft insertion hole 3. Thus, a region 4a2 extending between the axial center of the inner circumferential surface 4a of the main shaft insertion hole 3 and the lower axial end of the inner circumferential surface 4a of the main shaft insertion hole 3

is the region where there is no tapered shape (where the inner circumferential surface extends parallel to the axial direction). In the example of FIG. 19D, the length L_t of the region of the tapered shape of the lower bearing 15 in the central axis direction is equal to $1/2$ of the axial length of the eccentric sleeve 4.

[0133] Thus, the first tapered shape is formed in a region including the upper axial end of the inner circumferential surface 4a of the main shaft insertion hole 3 and having a length equal to $1/2$ of the axial length of the inner circumferential surface 4a of the main shaft insertion hole 3. In the case where a tapered shape (first tapered shape) is formed over the entirety of the inner circumferential surface 4a of the main shaft insertion hole 3, an increased cone angle could increase the degree of partial contact occurring under low loads, leading to the oil film thickness being insufficient in spite of the low loads. When the first tapered shape is formed over a portion of the inner circumferential surface 4a of the main shaft insertion hole 3 in the axial direction, the reduction in the minimum oil film thickness T_3 can be prevented both under low loads and under high loads.

[0134] The formation of the first tapered shape over a portion of the inner circumferential surface 4a in the axial direction is not limited to that in the above example. The first tapered shape may be formed in a region including the upper axial end of the inner circumferential surface 4a of the main shaft insertion hole 3 and having a length equal to or greater than $1/3$ of the axial length of the inner circumferential surface 4a of the main shaft insertion hole 3. The axial length of the region 4a1 of the first tapered shape is more preferably $1/2$ or more of the axial length of the inner circumferential surface 4a of the main shaft insertion hole 3.

[0135] Two or more of the first to third tapered shapes may be used in combination. That is, it is sufficient that at least one of the inner circumferential surface 4a of the main shaft insertion hole 3, the outer circumferential surface 4b of the eccentric sleeve 4, or the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 have a tapered shape such that the distance between the circumferential surface and another surface facing the circumferential surface increases upward. In any case, as in the case of FIG. 1, the upper partial contact state of the lower bearing 15 can be avoided, and the reduction in minimum oil film thickness T_3 (FIG. 3C) can be prevented. Thus, a gyration-type crusher that prevents the occurrence of failures such as seizure in the lower bearing 15 and that has high robustness against changes in load conditions can be constructed in a simple configuration.

[0136] In the configuration combining two or more of the first to third tapered shapes, the cone angle for providing the robust zone is in the same range as that in the configuration previously described. The cone angle refers to the sum of the cone angles of the combined

tapered shapes.

[0137] [Embodiment 2]

In Embodiment 1 described above, the gyration-type crusher has the configuration where at least one of the inner circumferential surface 4a of the main shaft insertion hole 3, the outer circumferential surface 4b of the eccentric sleeve 4, or the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 has a tapered shape such that the distance between the circumferential surface and another surface facing the circumferential surface increases upward. Alternatively, the gyration-type crusher may have, as the configuration for avoiding the upper partial contact state, a configuration in which a tapered shape is formed in a thrust bearing 33 located between the eccentric sleeve 4 and the eccentric sleeve support 32.

[0138] FIG. 20A is an enlarged cross-sectional view showing the lower end of an eccentric sleeve and its vicinity in a gyration-type crusher according to Embodiment 2 of the present invention. In the example of FIG. 20A, the eccentric sleeve 4 includes a first thrust bearing surface 23a in its lower surface, the first thrust bearing surface 23a being contactable with the upper surface of the eccentric sleeve support 32. The eccentric sleeve support 32 includes a second thrust bearing surface 23b in its upper surface, the second thrust bearing surface 23b being contactable with the first thrust bearing surface 23a. The first and second thrust bearing surfaces 23a and 23b constitute a thrust bearing 23 for the eccentric sleeve support 32 of the lower bearing 15 (lower frame 2). The first thrust bearing surface 23a has a fourth tapered shape such that the distance between the first and second thrust bearing surfaces 23a and 23b facing each other increases outward in the radial direction of the eccentric sleeve 4.

[0139] In the above configuration, which includes the tapered shape as described above, the main shaft 5 and the upper end of the inner circumferential surface of the main shaft insertion hole 3 of the eccentric sleeve 4 are prevented from being close to each other even in the situation where the crushing load is so large that the upper partial contact state would occur in conventional configurations. Thus, the upper partial contact state of the lower bearing 15 can be avoided, and the reduction in minimum oil film thickness can be prevented. As such, a gyration-type crusher that prevents the occurrence of failures such as seizure in the lower bearing 15 and that has high robustness against changes in load conditions can be constructed in a simple configuration.

[0140] As shown in FIG. 20B, the second thrust bearing surface 23b may have the fourth tapered shape instead of the first thrust bearing surface 23a. Alternatively, as shown in FIG. 20C, both the first and second thrust bearing surfaces 23a and 23b may have the fourth tapered shape.

[0141] [Embodiment 3]

The gyration-type crusher may have both the tapered shape in Embodiment 1 (at least one of the first to third tapered shapes) and the fourth tapered shape in Embodiment 2 (the tapered shape formed in at least one of the first or second thrust bearing surface 23a or 23b).

[0142] FIG. 21 is an enlarged cross-sectional view showing a lower bearing and its vicinity in a gyration-type crusher according to Embodiment 3 of the present invention. In the example of FIG. 21, the lower bearing 15 includes the second tapered shape formed in the outer circumferential surface 4b of the eccentric sleeve 4, the third tapered shape formed in the inner circumferential surface 7a of the eccentric sleeve insertion hole 27 of the outer barrel 7, and the fourth tapered shapes formed respectively in the first and second thrust bearing surfaces 23a and 23b of the thrust bearing 23.

[0143] By virtue of the second and third tapered shapes formed in the lower bearing 15, the above configuration can accommodate a larger tilt of the eccentric sleeve 4 than conventional configurations (configurations devoid of the second and third tapered shapes). This can lead to an increase in the stress applied to the eccentric sleeve 4 and eccentric sleeve support 32 due to the tilt of the eccentric sleeve 4. The fourth tapered shapes (the first and second thrust bearing surfaces 23a and 23b) for accommodating tilt of the eccentric sleeve 4 are formed between the eccentric sleeve 4 and eccentric sleeve support 32. This reduces the increase in the stress applied to the eccentric sleeve 4 and eccentric sleeve support 32. As such, the deformation of the eccentric sleeve 4 or eccentric sleeve support 32 due to tilt of the eccentric sleeve 4 can be reduced.

[0144] The combination of the first to fourth tapered shapes is not limited to that in the above example. Various other combinations may be employed, such as a configuration including the first and third tapered shapes and a configuration including all of the first to fourth tapered shapes.

Reference Signs List

- [0145]
- 3 main shaft insertion hole
 - 4 eccentric sleeve
 - 4a inner circumferential surface having first tapered shape
 - 4b outer circumferential surface having second tapered shape
 - 5 main shaft
 - 7 outer barrel
 - 7a inner circumferential surface having third tapered shape

- 13 mantle
- 14 concave
- 15 lower bearing
- 17 upper bearing
- 23a first thrust bearing surface
- 23b second thrust bearing surface
- 27 eccentric sleeve insertion hole
- 30 hydraulic cylinder
- 32 eccentric sleeve support

CLAIMS

1. A gyration-type crusher including:
 - a main shaft rotatably disposed inside a concave to eccentrically gyrate in a manner in which a central axis of the main shaft is tilted with respect to a central axis of the concave;
 - an upper bearing rotatably supporting an upper end of the main shaft;
 - a mantle mounted on the main shaft;
 - a lower bearing rotatably supporting a lower end of the main shaft; and
 - a hydraulic cylinder disposed below the lower bearing to move the main shaft upward and downward by hydraulic pressure, wherein:
 - the lower bearing includes
 - an eccentric sleeve including a main shaft insertion hole into which the lower end of the main shaft is rotatably inserted, and
 - an outer barrel including an eccentric sleeve insertion hole into which the eccentric sleeve is rotatably inserted; and
 - an inner circumferential surface of the main shaft insertion hole has a tapered shape in a region extending over at least a portion of the inner circumferential surface of the main shaft insertion hole in an axial direction of the main shaft insertion hole, the tapered shape being such that a distance between the inner circumferential surface of the main shaft insertion hole and an outer circumferential surface of the main shaft that faces the inner circumferential surface of the main shaft insertion hole increases upward,
 - an inner circumferential surface of the eccentric sleeve insertion hole has a tapered shape in a region extending over at least a portion of the inner circumferential surface of the eccentric sleeve insertion hole in an axial direction of the eccentric sleeve insertion hole, the tapered shape being such that a distance between the inner circumferential surface of the eccentric sleeve insertion hole and an outer circumferential surface of the eccentric sleeve that faces the inner circumferential surface of the eccentric sleeve insertion hole increases upward, or
 - the outer circumferential surface of the eccentric sleeve has a tapered shape in a region extending over at least a portion of the outer circumferential surface of the eccentric sleeve in an axial direction of the eccentric sleeve, the tapered shape being such that a distance between the outer circumferential surface of the eccentric sleeve and the inner circumferential surface of the eccentric sleeve insertion hole that faces the outer circumferential surface of the eccentric sleeve increases upward.

2. The gyration-type crusher according to claim 1, wherein the tapered shape includes a first tapered shape such that a diameter of the inner circumferential surface of the main shaft insertion hole increases upward.
3. The gyration-type crusher according to claim 1 or 2, wherein the tapered shape includes a second tapered shape such that a diameter of the outer circumferential surface of the eccentric sleeve increases downward.
4. The gyration-type crusher according to any one of claims 1 to 3, wherein the tapered shape includes a third tapered shape such that a diameter of the inner circumferential surface of the eccentric sleeve insertion hole increases upward.
5. The gyration-type crusher according to claim 2, wherein the first tapered shape is formed in a region including an upper axial end of the inner circumferential surface of the main shaft insertion hole and having a length equal to or greater than $1/3$ of an axial length of the inner circumferential surface of the main shaft insertion hole.
6. The gyration-type crusher according to claim 4, wherein the third tapered shape is formed in a region including an upper axial end of the inner circumferential surface of the eccentric sleeve insertion hole and having a length equal to or greater than $1/3$ of an axial length of the inner circumferential surface of the eccentric sleeve insertion hole.
7. The gyration-type crusher according to any one of claims 1 to 6, wherein
a lubricating oil is supplied to a gap between an outer circumferential surface of the lower end of the main shaft and the inner circumferential surface of the main shaft insertion hole and to a gap between the outer circumferential surface of the eccentric sleeve and the inner circumferential surface of the eccentric sleeve insertion hole, and
a cone angle or rate of taper of the tapered shape is set to provide a robust zone in regard to changes in minimum oil film thickness of the lubricating oil with changes in crushing load.
8. The gyration-type crusher according to any one of claims 1 to 7, wherein when the tapered shape is viewed in cross-section along the central axis of the main shaft, an angle indicative of a ratio of amount of change in diameter Dt of the region of the tapered shape in the lower bearing to a length of the region in a direction of the central axis is from 0.001 to 1° .

9. The gyration-type crusher according to claim 3 or 4, wherein
- the lower bearing includes an eccentric sleeve support disposed below the eccentric sleeve to support the eccentric sleeve in a manner permitting relative rotation of the eccentric sleeve,
 - the eccentric sleeve includes a first thrust bearing surface in a lower surface of the eccentric sleeve, the first thrust bearing surface being contactable with an upper surface of the eccentric sleeve support,
 - the eccentric sleeve support includes a second thrust bearing surface in the upper surface of the eccentric sleeve support, the second thrust bearing surface being contactable with the first thrust bearing surface, and
 - at least one of the first or second thrust bearing surface has a fourth tapered shape such that a distance between the first and second thrust bearing surfaces facing each other increases outward in a radial direction of the eccentric sleeve.

10. A gyration-type crusher including:
- a main shaft rotatably disposed inside a concave to eccentrically gyrate in a manner in which a central axis of the main shaft is tilted with respect to a central axis of the concave;
 - an upper bearing rotatably supporting an upper end of the main shaft;
 - a mantle mounted on the main shaft;
 - a lower bearing rotatably supporting a lower end of the main shaft; and
 - a hydraulic cylinder disposed below the lower bearing to move the main shaft upward and downward by hydraulic pressure, wherein:
 - the lower bearing includes
 - an eccentric sleeve including a main shaft insertion hole into which the lower end of the main shaft is rotatably inserted,
 - an outer barrel including an eccentric sleeve insertion hole into which the eccentric sleeve is rotatably inserted, and
 - an eccentric sleeve support disposed below the eccentric sleeve to support the eccentric sleeve in a manner permitting relative rotation of the eccentric sleeve;
 - the eccentric sleeve includes a first thrust bearing surface in a lower surface of the eccentric sleeve, the first thrust bearing surface being contactable with an upper surface of the eccentric sleeve support;
 - the eccentric sleeve support includes a second thrust bearing surface in the upper surface of the eccentric sleeve support, the second thrust bearing surface being contactable with the first thrust bearing surface; and

at least one of the first or second thrust bearing surface has a tapered shape such that a distance between the first and second thrust bearing surfaces facing each other increases outward in a radial direction of the eccentric sleeve.

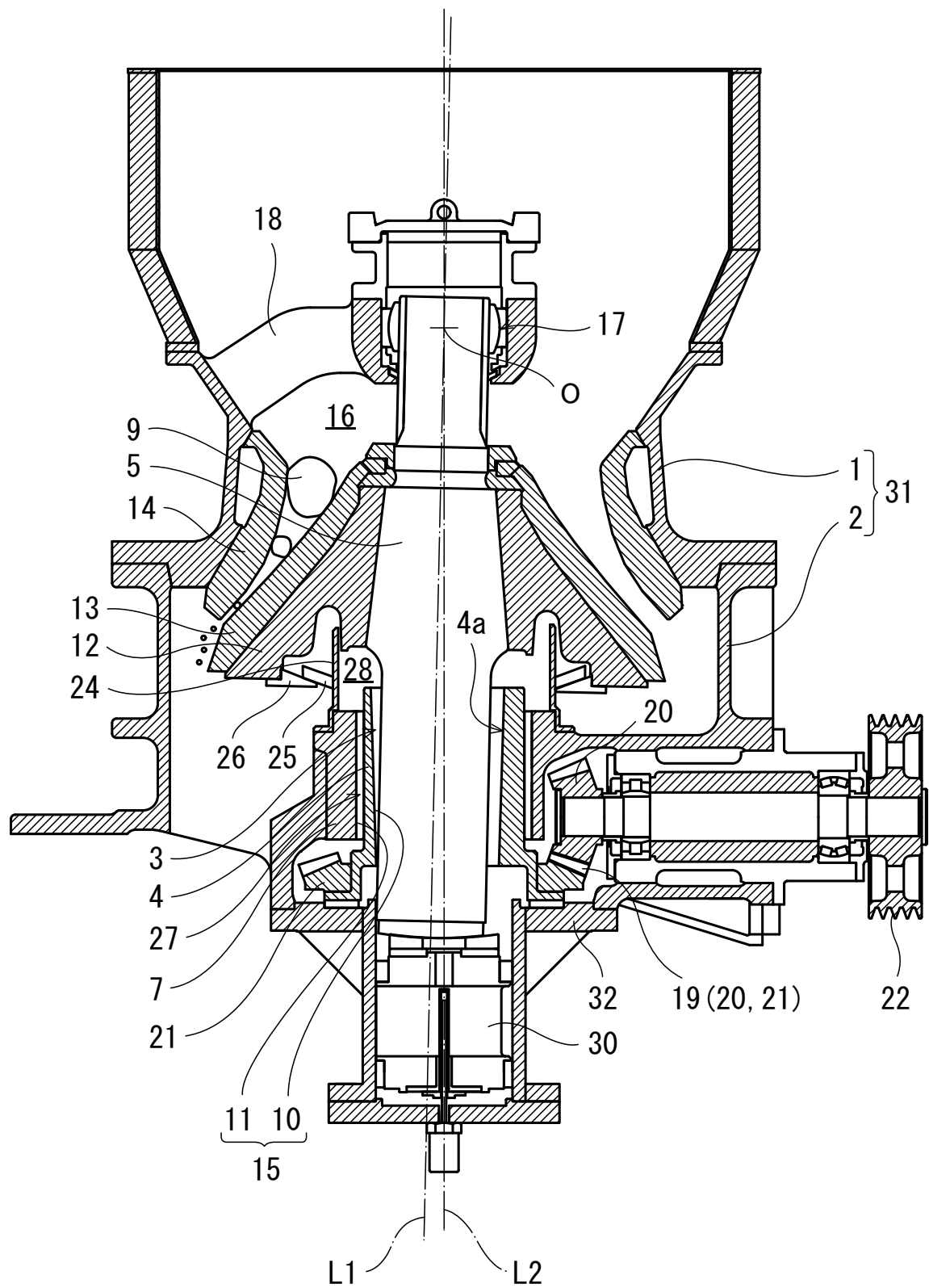


FIG. 1

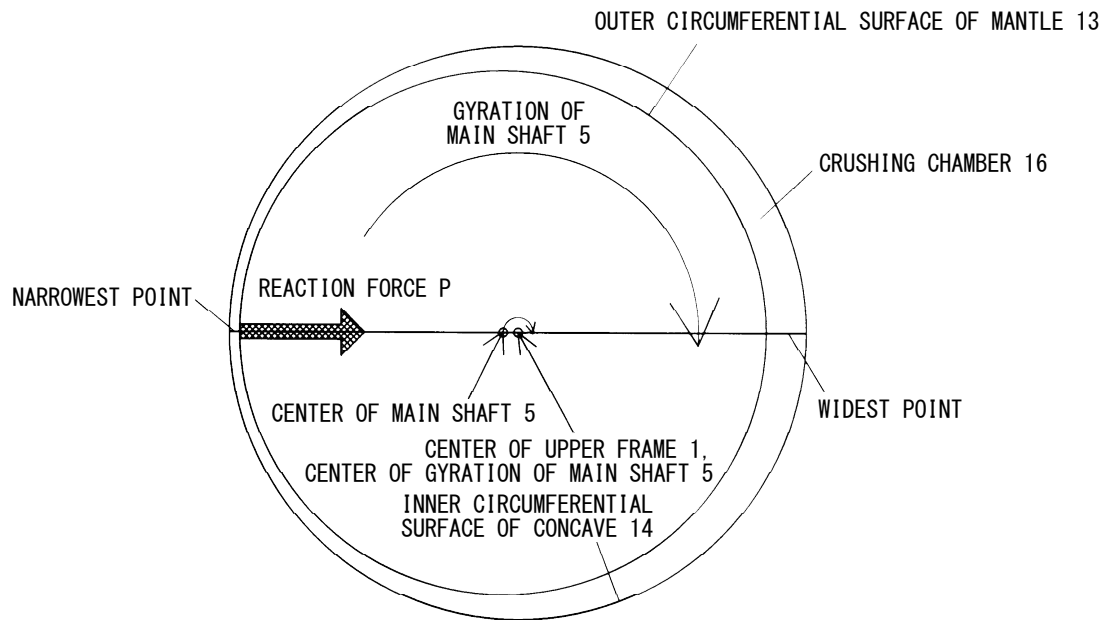


FIG. 2

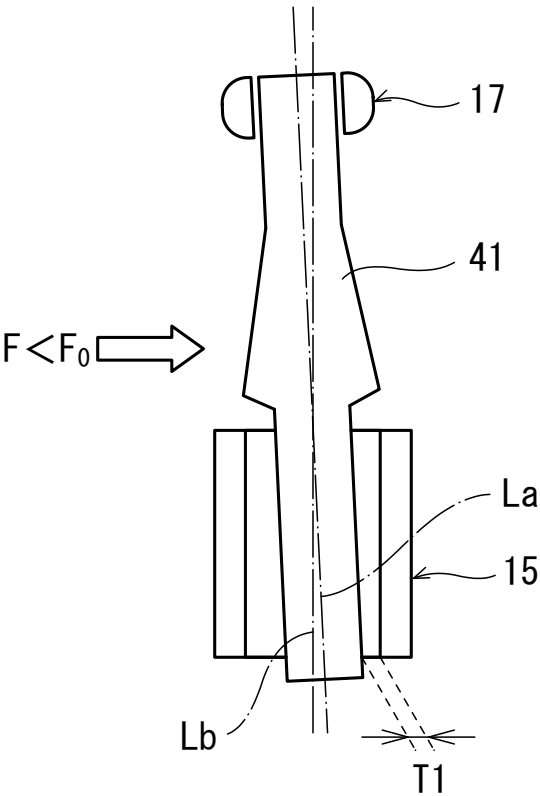


FIG. 3A

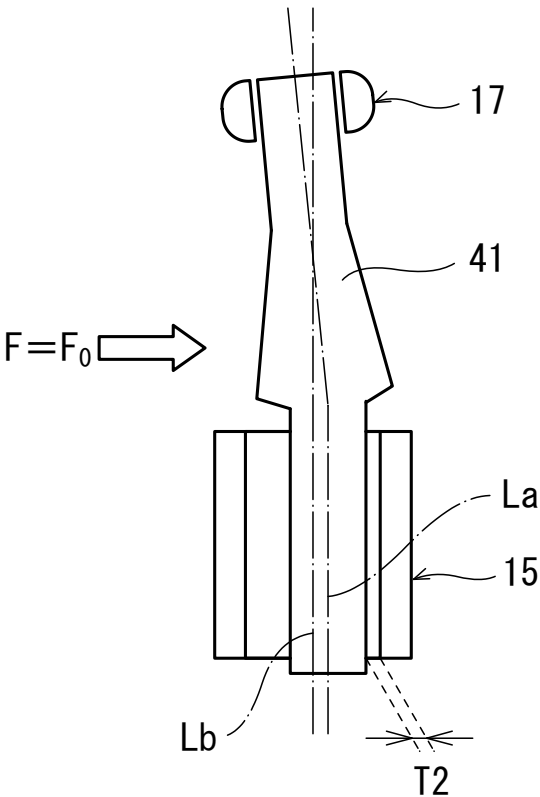


FIG. 3B

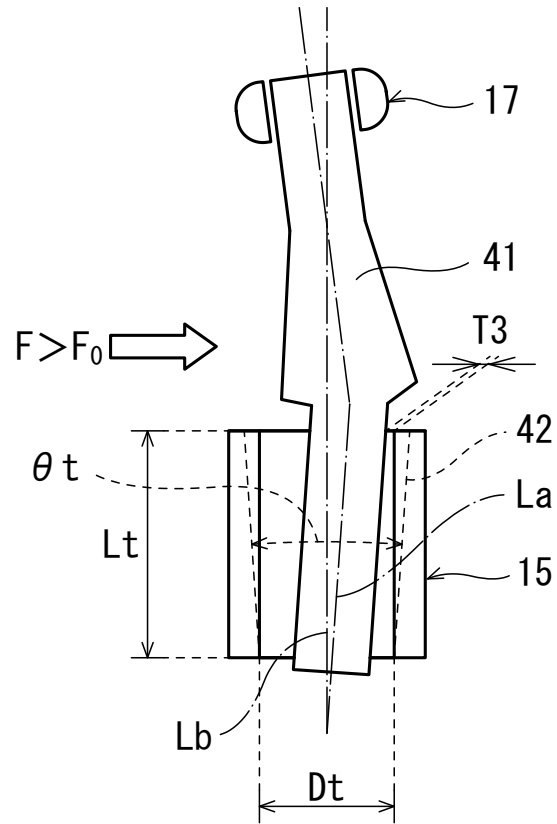


FIG. 3C

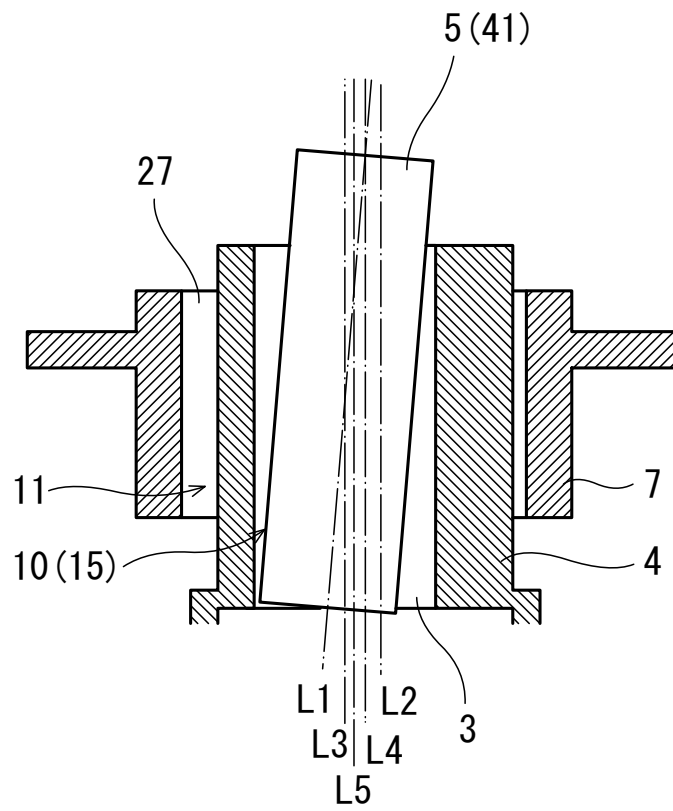


FIG. 4A

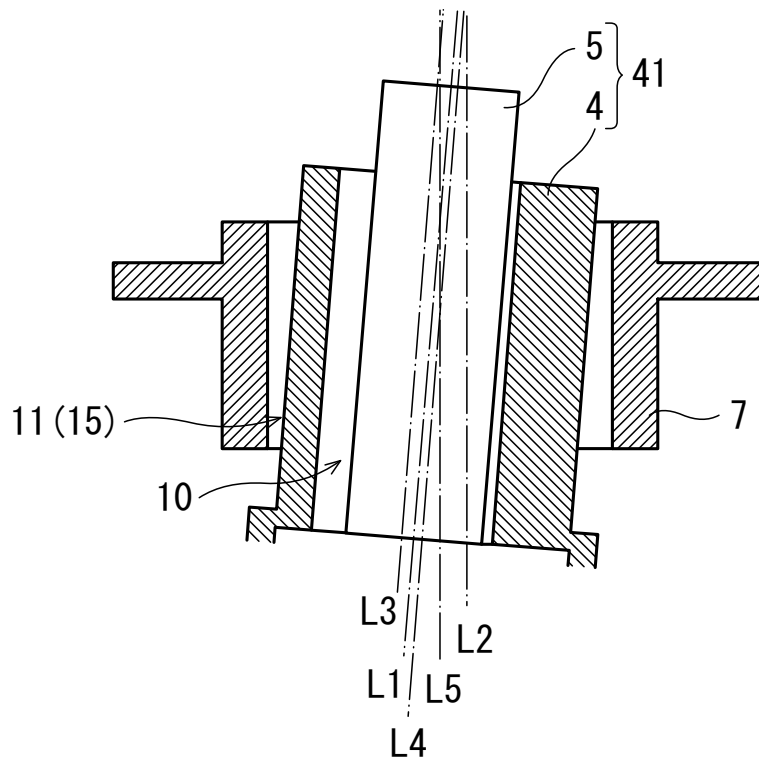


FIG. 4B

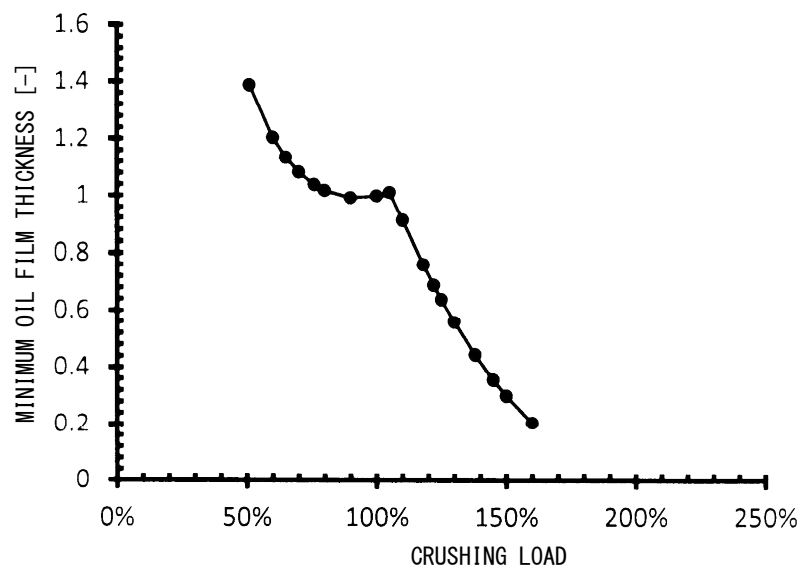


FIG. 5

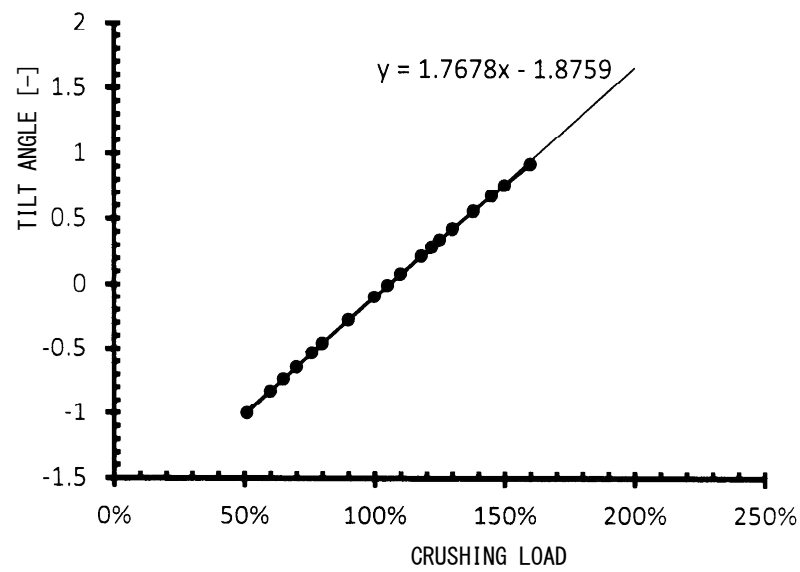


FIG. 6

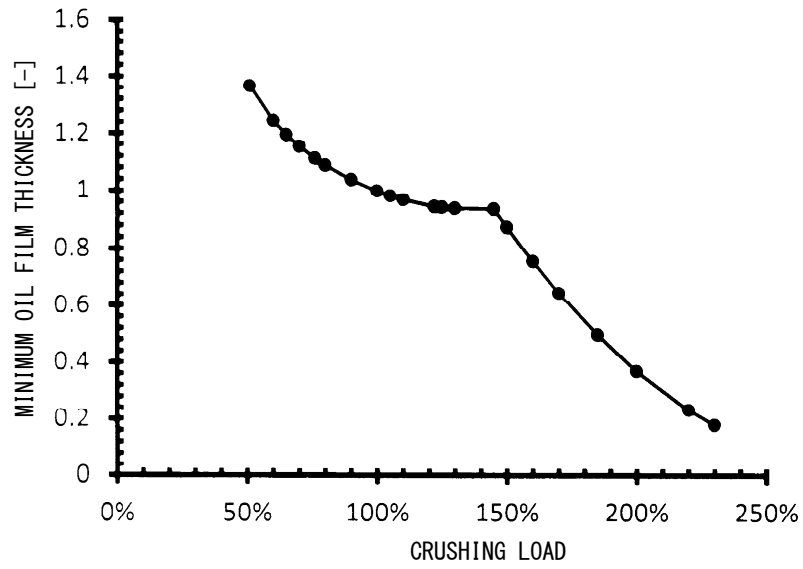


FIG. 7

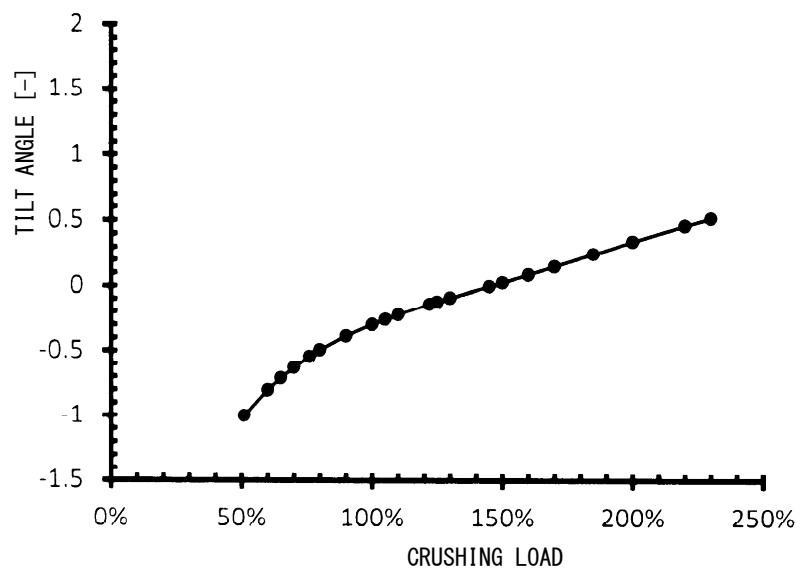


FIG. 8

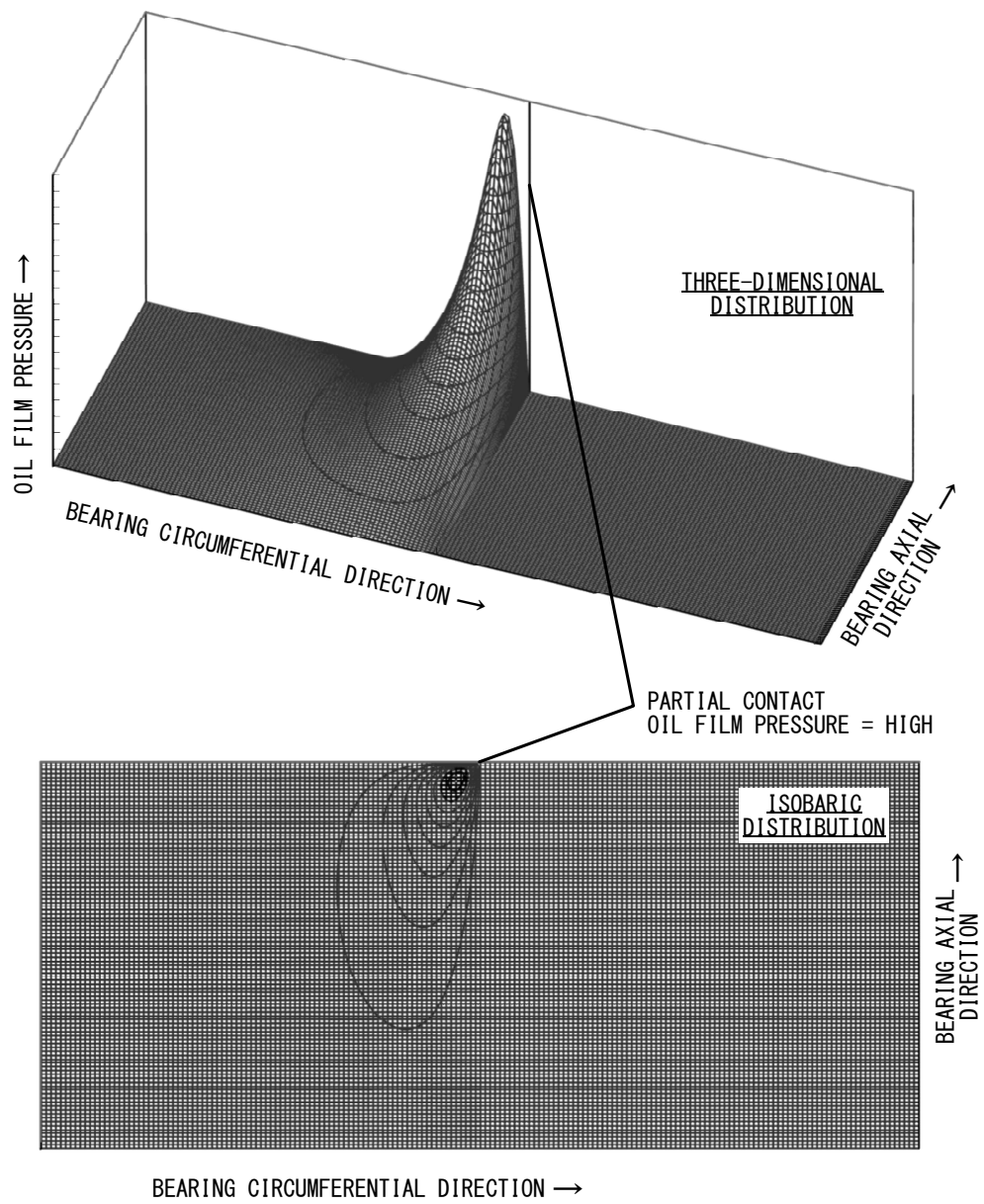


FIG. 9

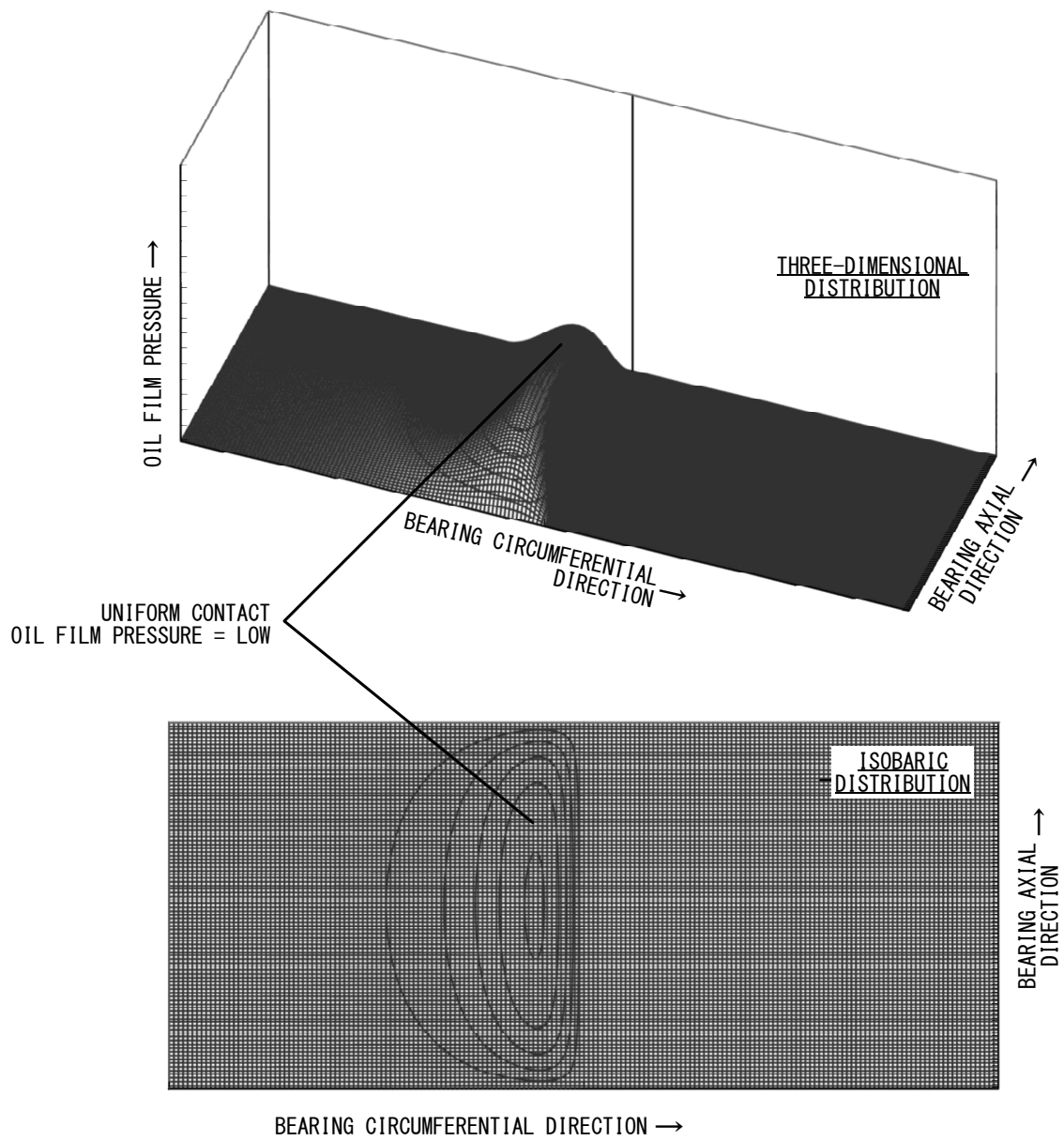
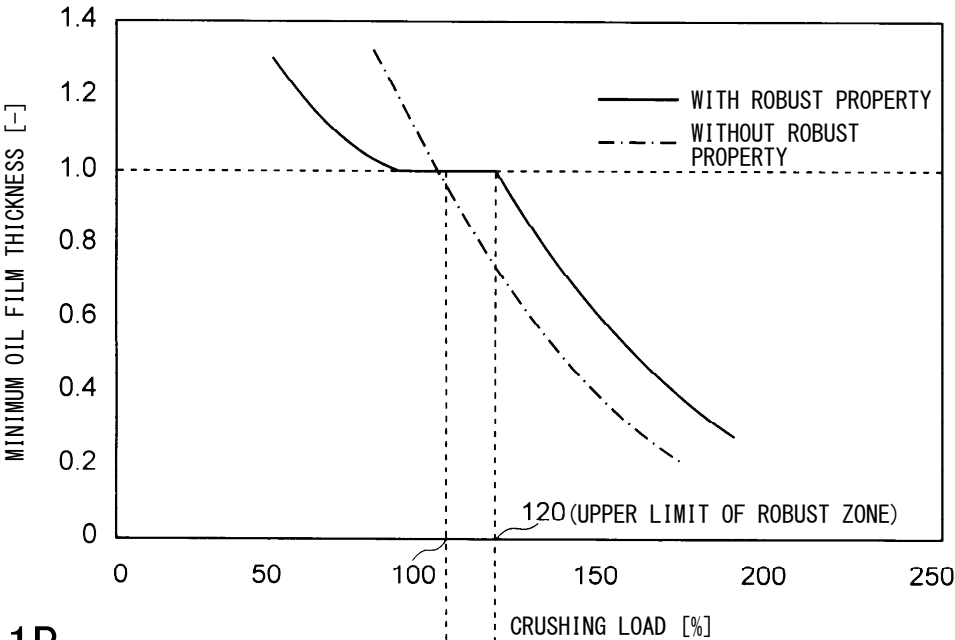


FIG. 10

11A



11B

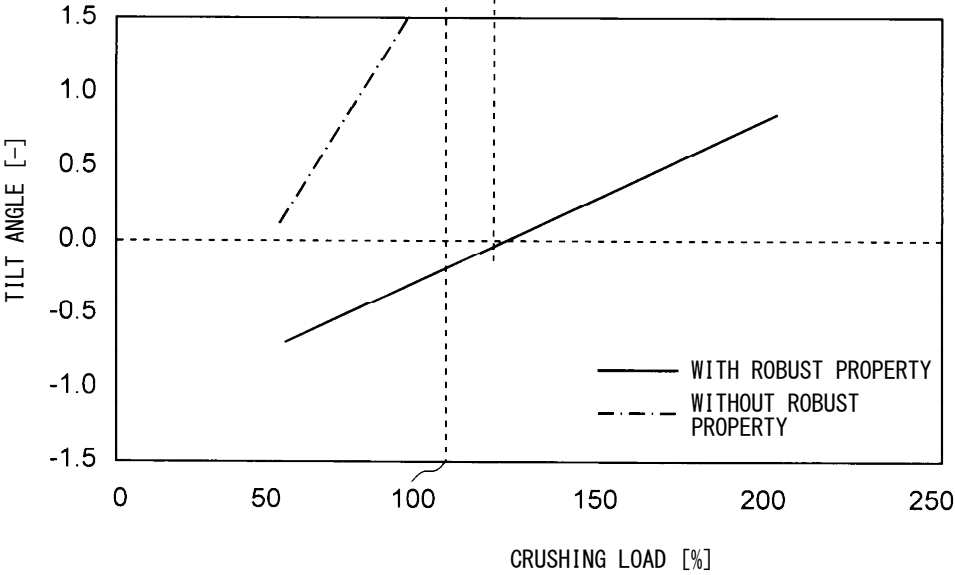


FIG. 11

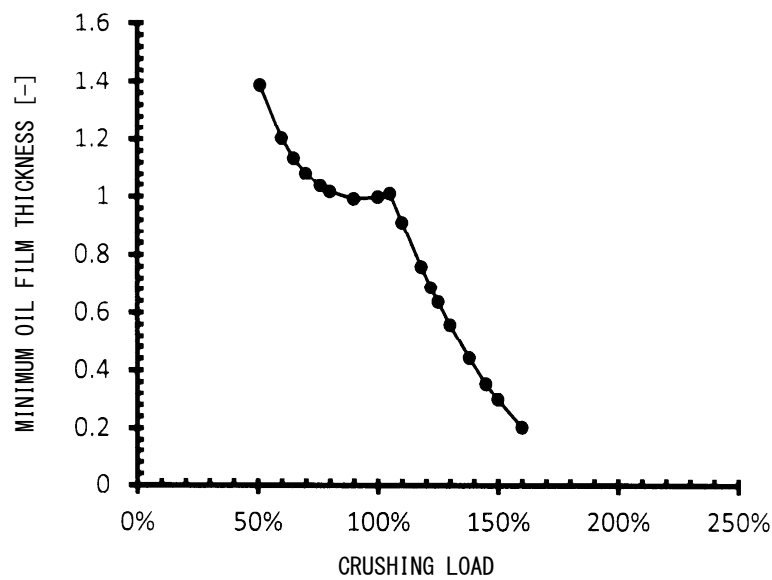


FIG. 12

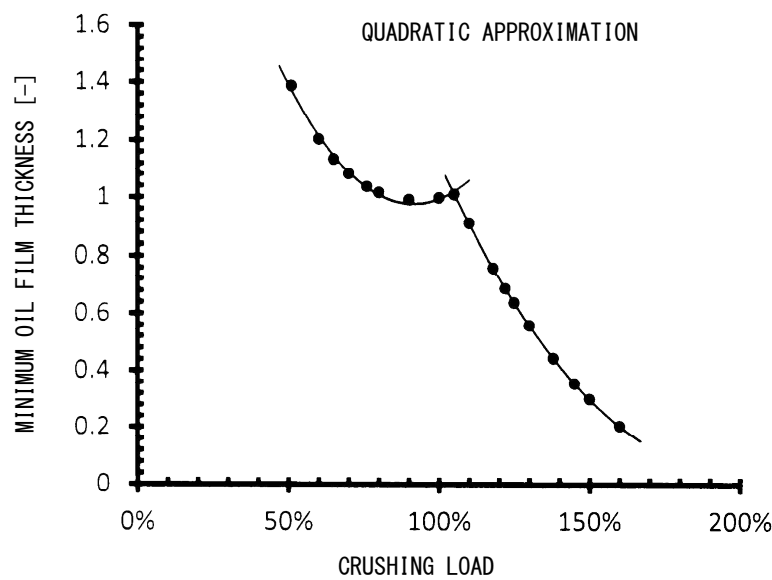


FIG. 13

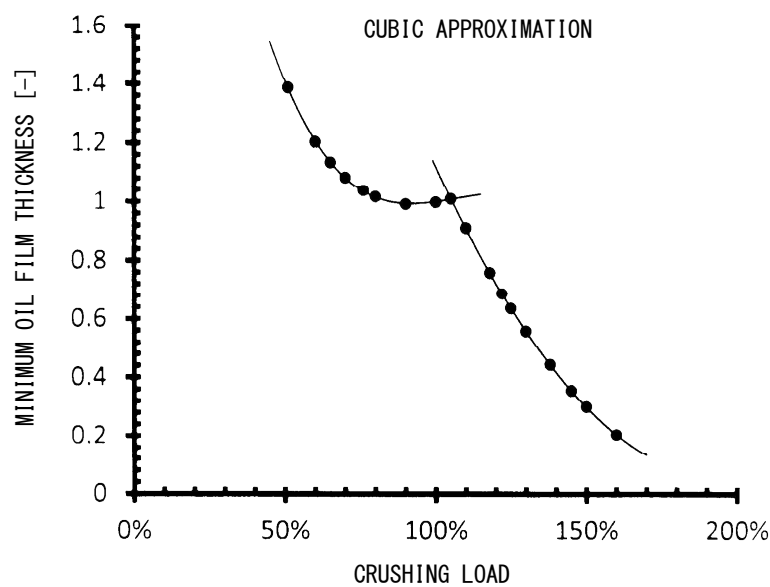


FIG. 14

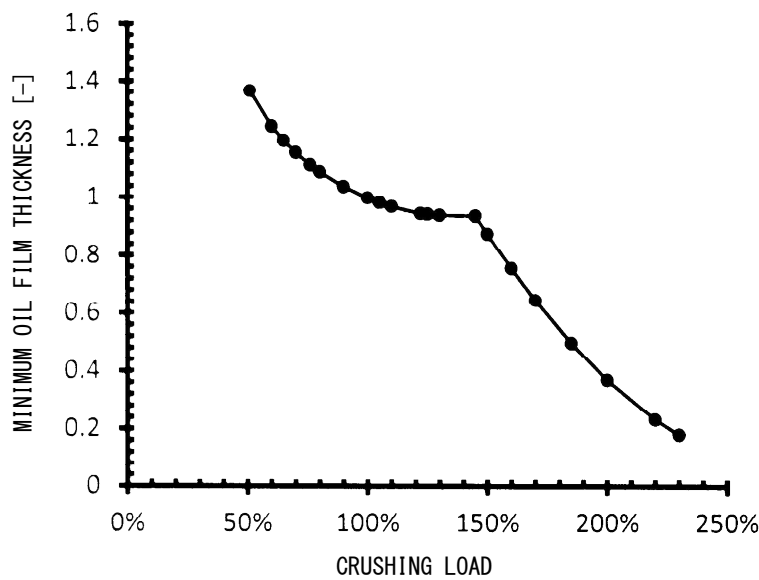


FIG. 15

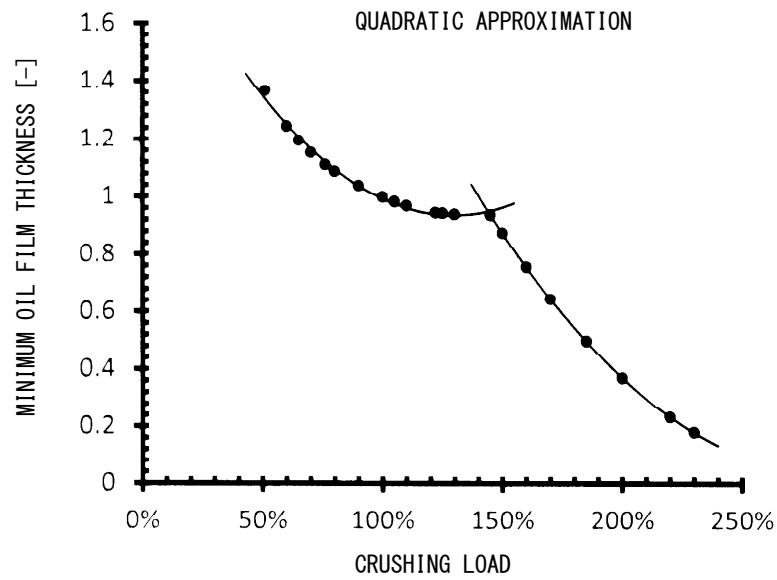


FIG. 16

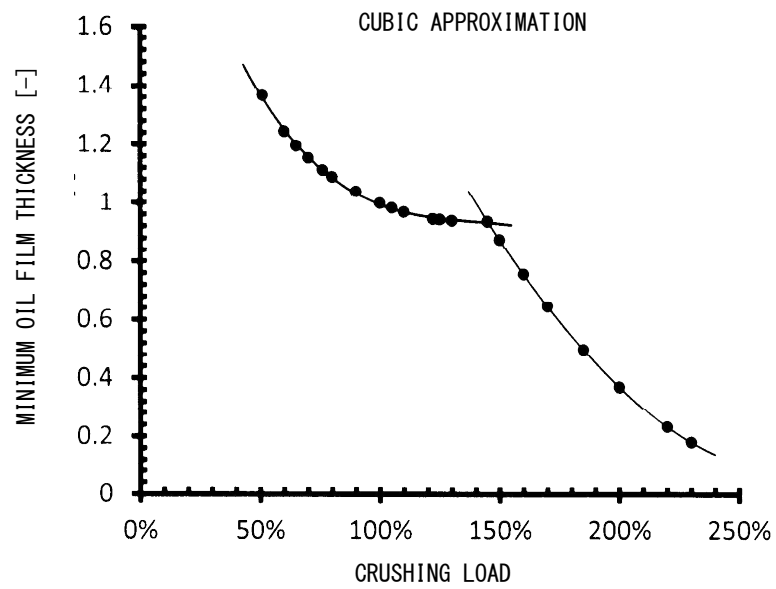


FIG. 17

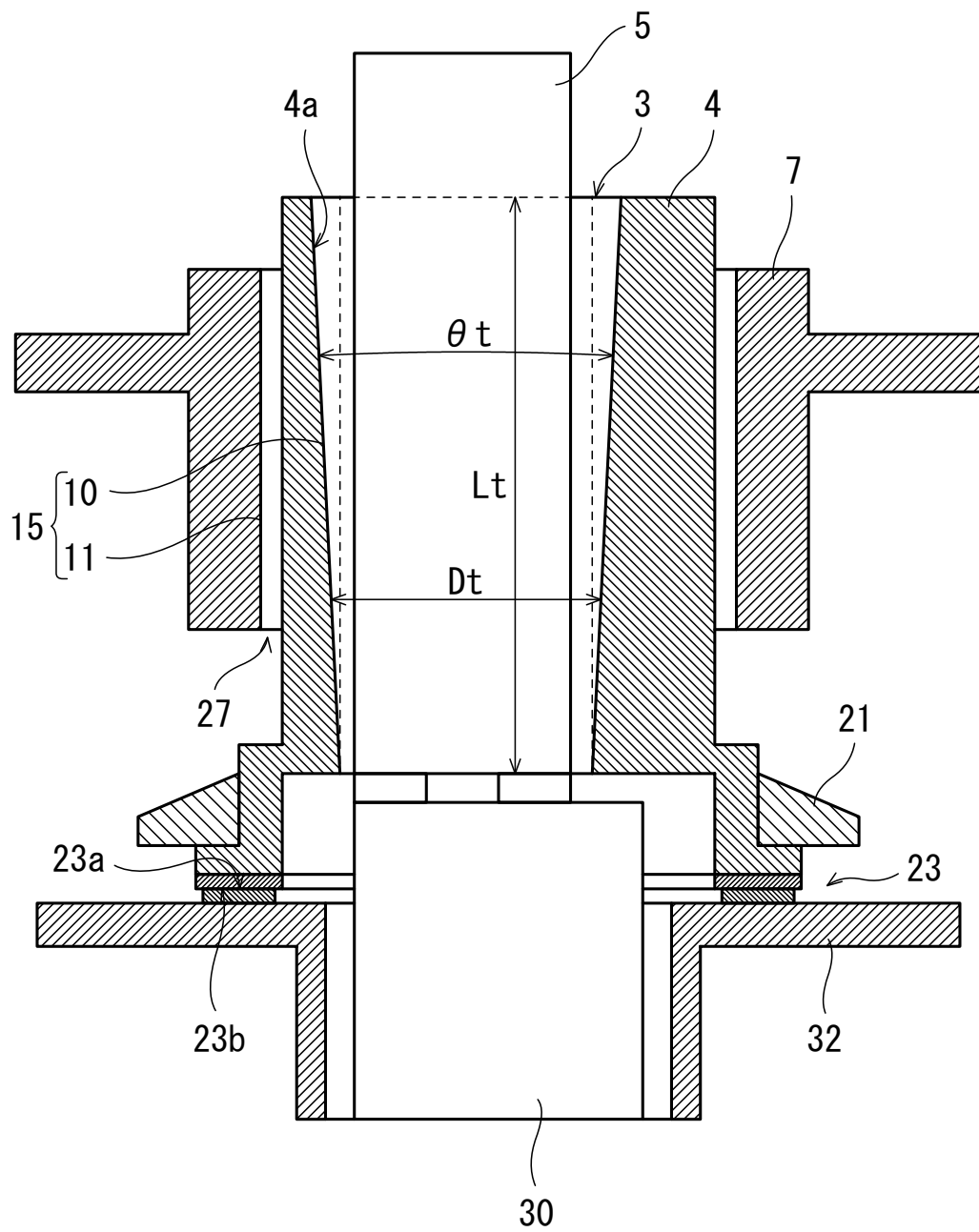


FIG. 18

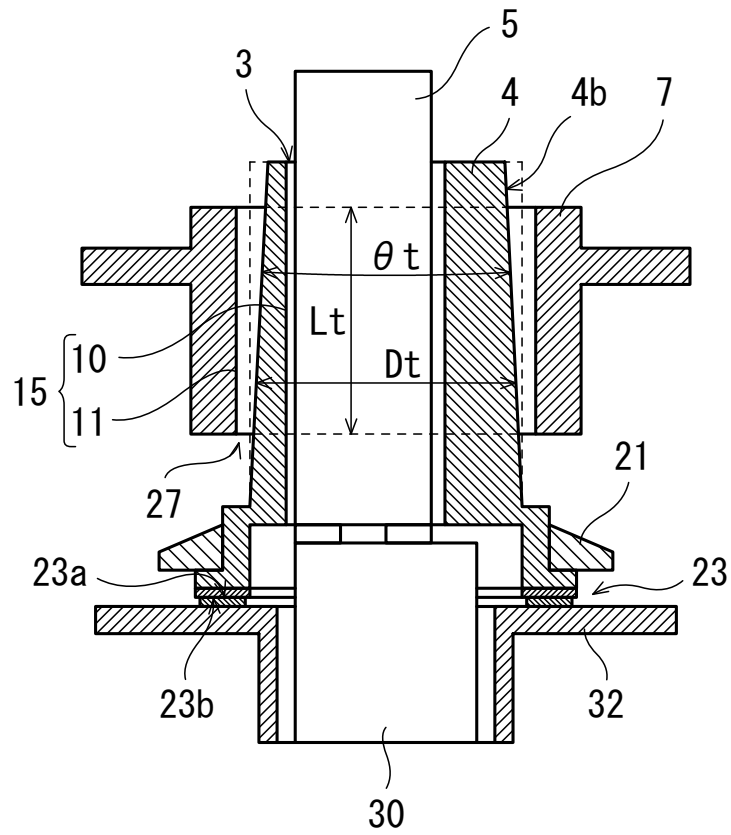


FIG. 19A

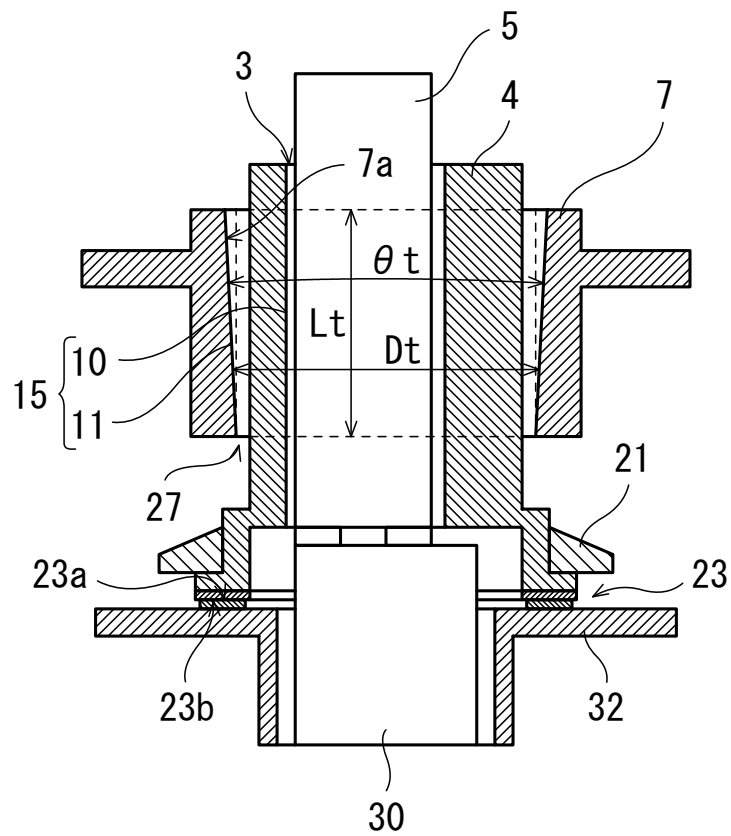


FIG. 19B

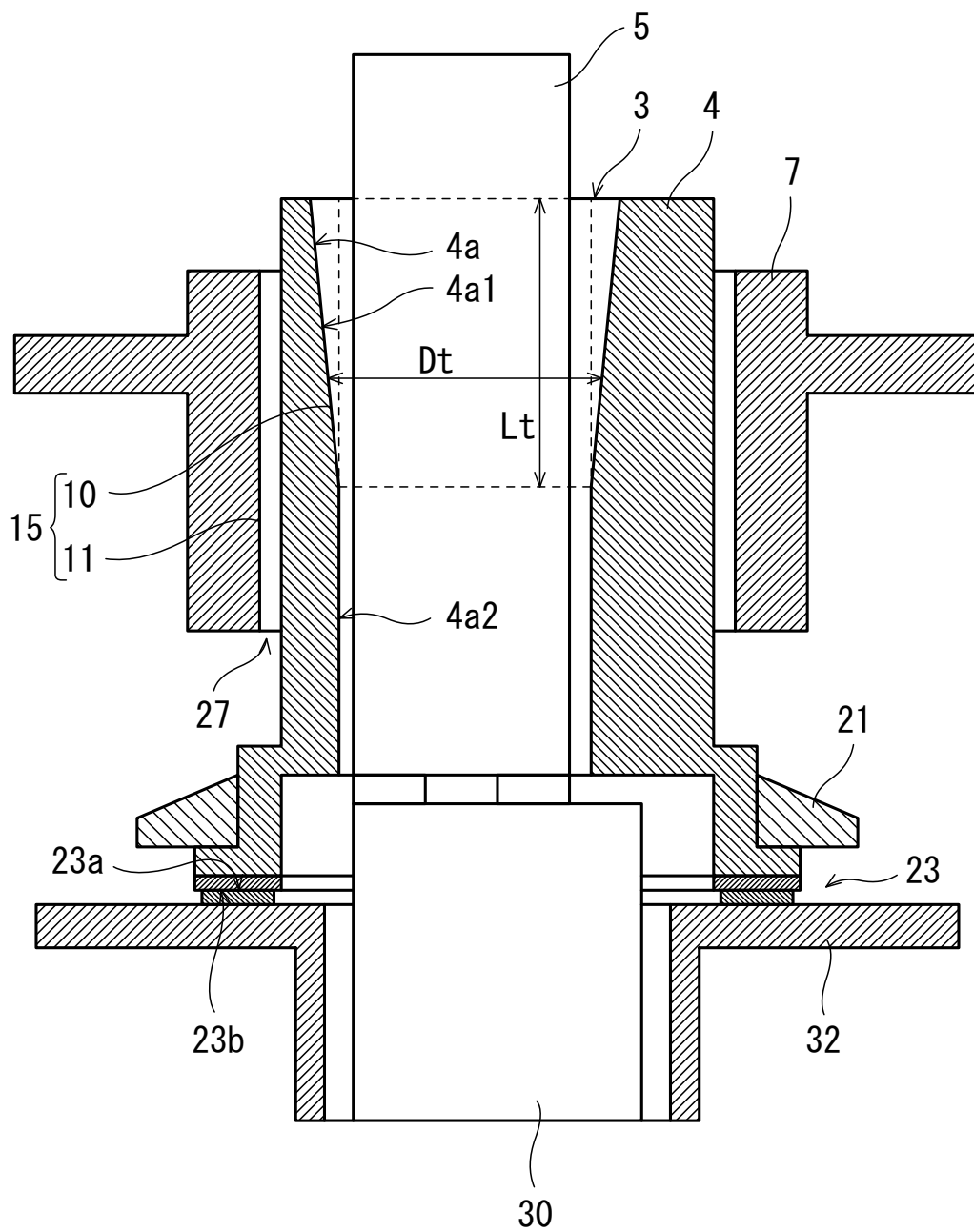


FIG. 19D

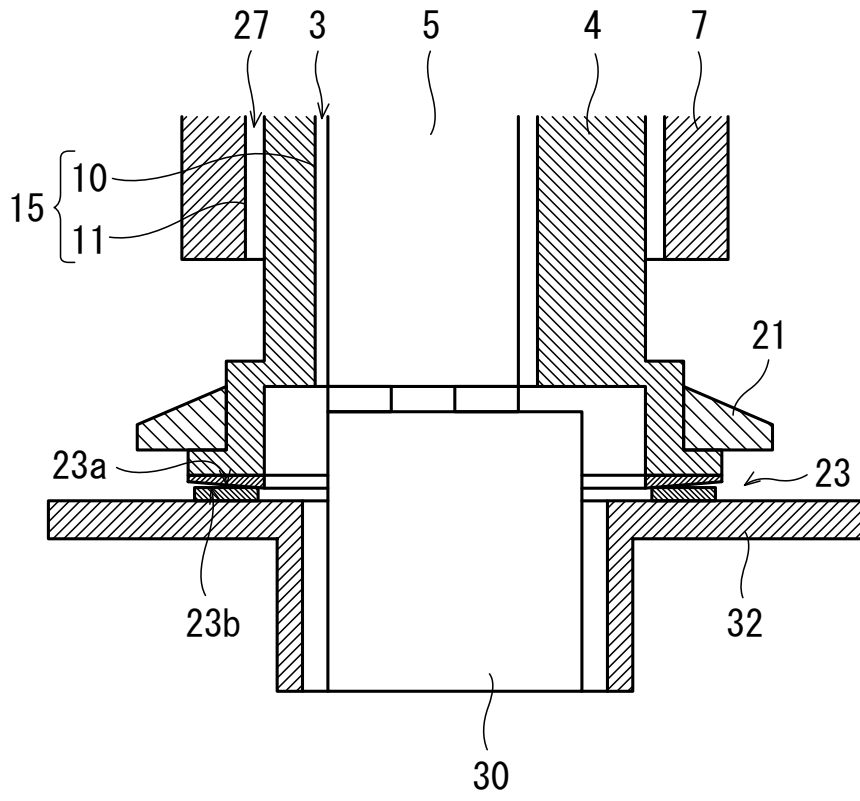


FIG. 20A

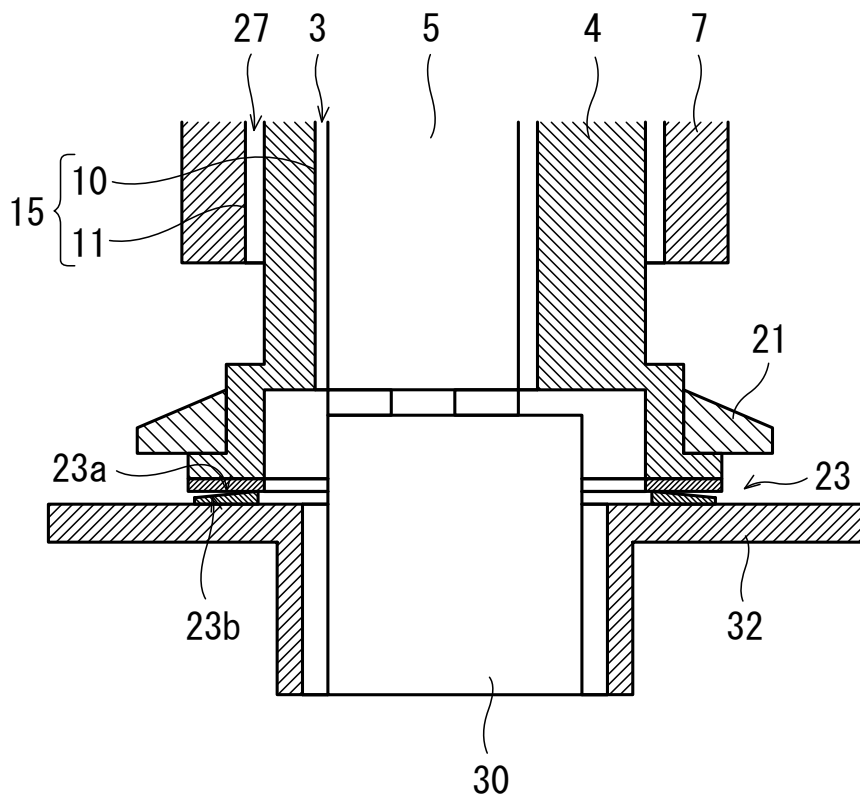


FIG. 20B

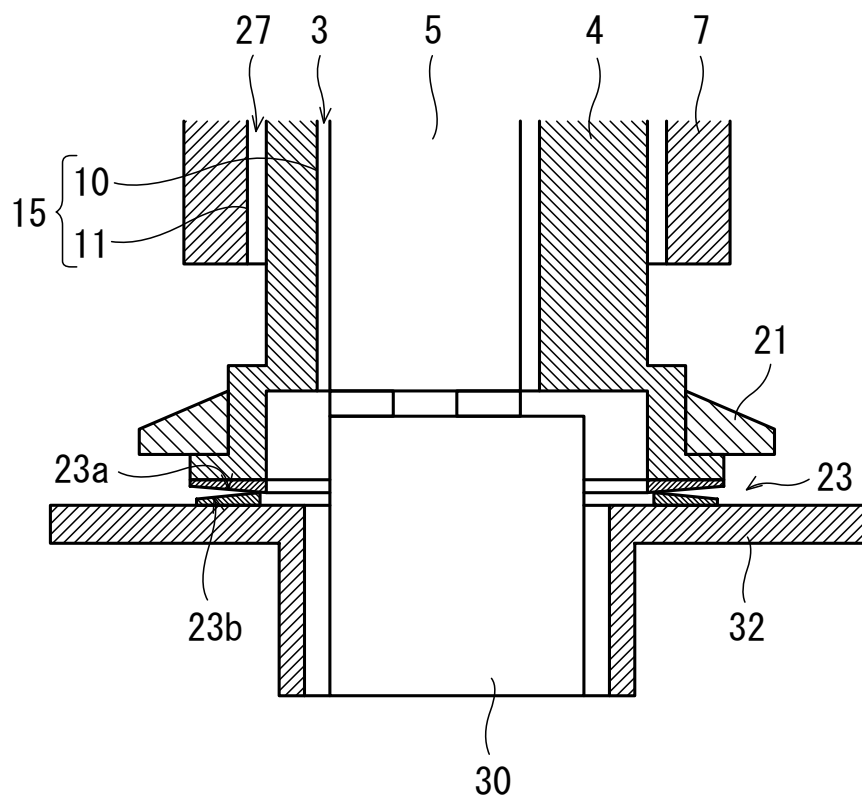


FIG. 20C

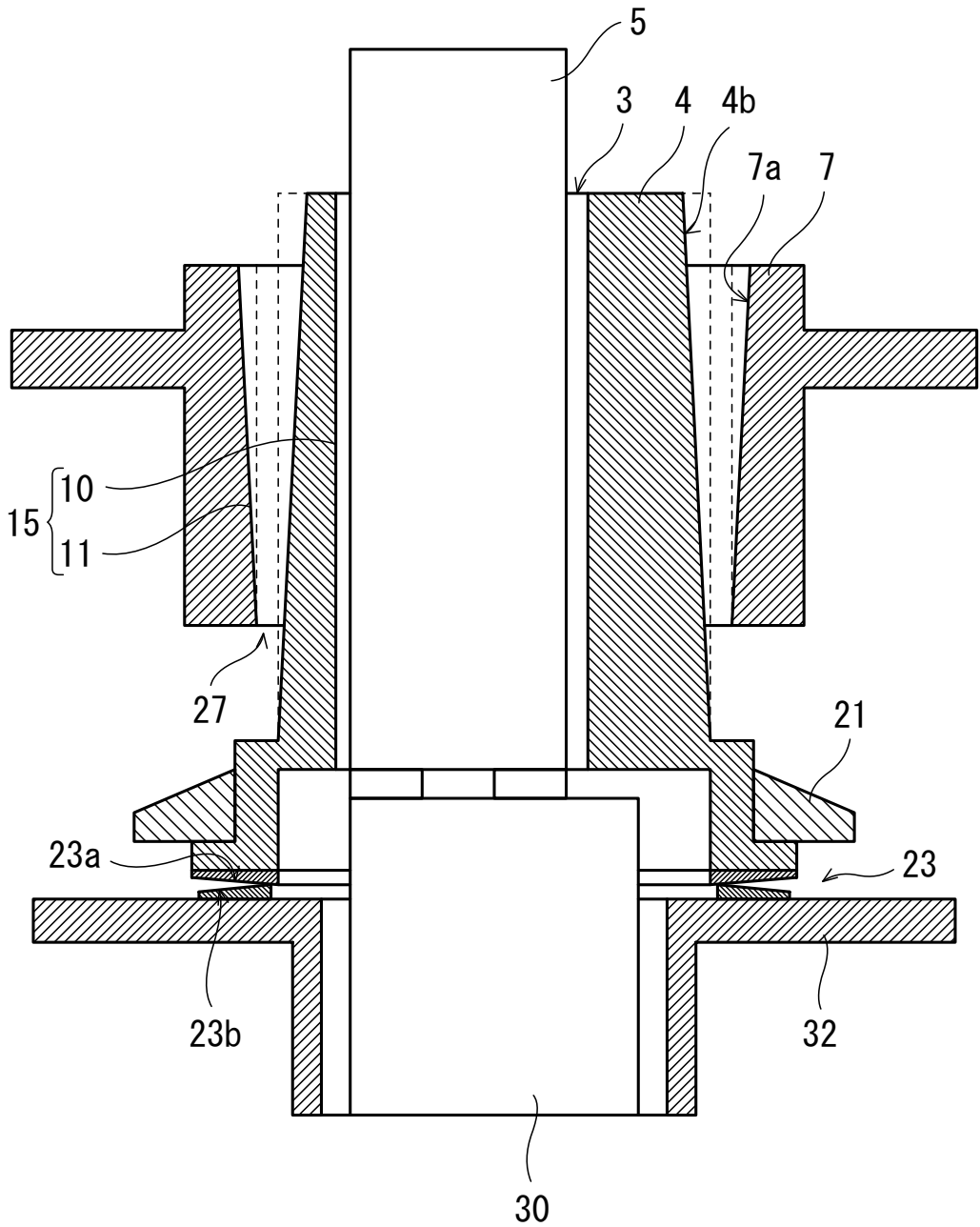


FIG. 21

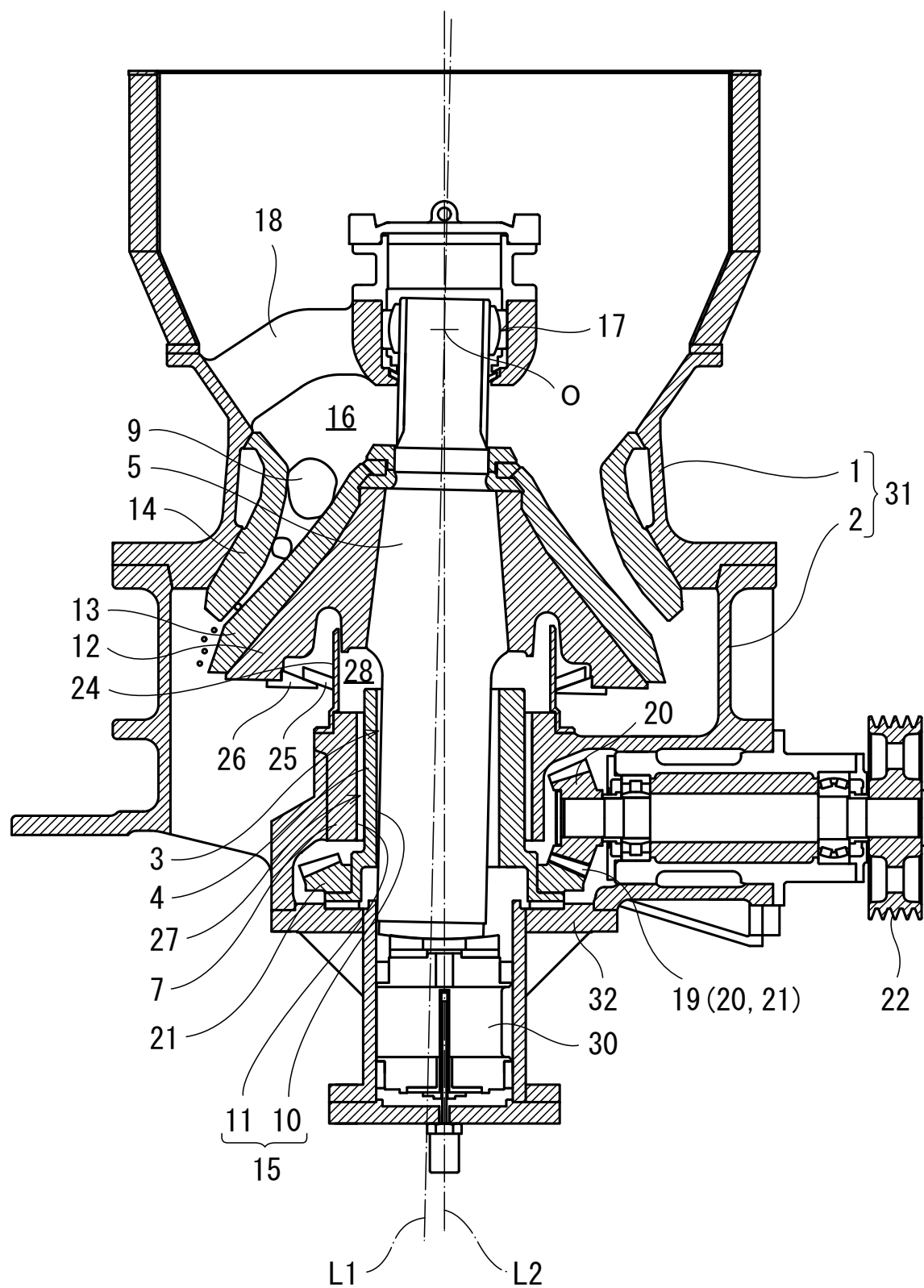


FIG. 22