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(54) **SCREW PUMP AND SCREW GEAR**

(57) A screw pump is provided with a pair of screw rotors serving as fluid transfer bodies. With respect to a rotation angle (x) around an axis of each of the screw rotors, a change of a lead angle (θ) from a winding start angle (0), which is the rotation angle (x) corresponding to a leading end of a spiral groove, to a winding end angle (E), which is the rotation angle (x) corresponding to a trailing end of the spiral groove, can be expressed by a

lead angle change function $\theta(x)$. The lead angle change function $\theta(x)$ is structured by a combination of a plurality of change functions $\theta_1(x)$ and $\theta_2(x)$ having different manners of changing. It is possible to arbitrarily set a manner in which the lead angle (θ) changes in accordance with a combination of a plurality of change functions $\theta_1(x)$ and $\theta_2(x)$. Therefore, it is possible to arbitrarily set a fluid compression characteristic of the pump in a relation with an axial length (L) of the screw rotor.

Fig. 2A

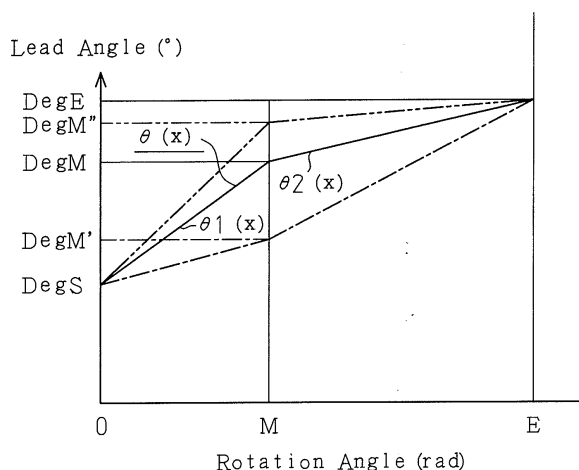
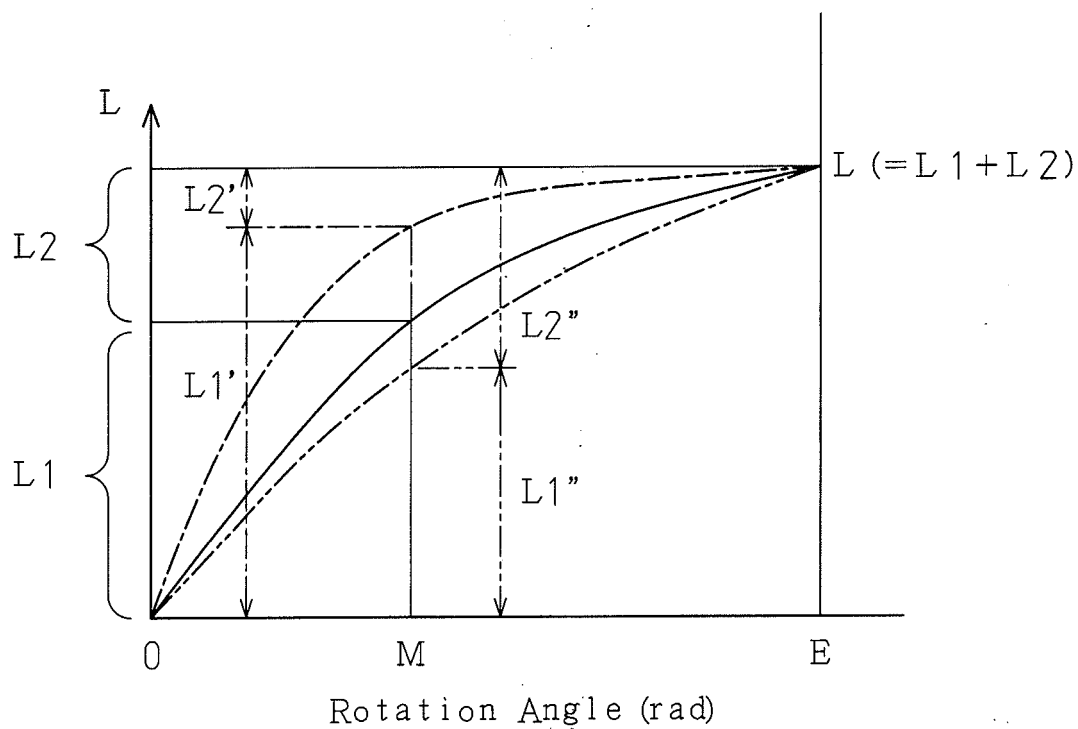


Fig. 2B



Description

TECHNICAL FIELD

5 **[0001]** The present invention relates to a screw pump, for example, used in a semiconductor manufacturing process and a screw gear suitable for the use in a screw pump.

BACKGROUND ART

10 **[0002]** In general, in a semiconductor manufacturing process, a screw pump is used as a vacuum pump for generating a vacuum environment. In other words, in the semiconductor manufacturing process, in order to apply various processes to a wafer under a vacuum environment, a clean vacuum environment is generated within a container by supplying an inert gas such as F₂ gas or the like, which is a fluid, into a container in which the wafer is accommodated, and the gas is sucked together with an impurity (O₂, CO₂ or the like) remaining within the container by a vacuum pump. As such a vacuum pump, there has been conventionally known a screw pump, for example, described in Patent Document 1.

15 **[0003]** The screw pump described in the Patent Document 1 is configured such that a pair of screw gears spirally mating with each other, that is, a pair of screw rotors serve as fluid transfer bodies (a gas transfer bodies). Each of the screw gears is coupled to a rotary shaft rotated by a drive source so as to be integrally rotated. A lead angle (a torsion angle) of each of the screw gears is continuously changed in accordance with a spiral groove (a helix) of the screw gear. Specifically, the lead angle is monotonically increased toward an end portion in an axial direction close to a high pressure (discharge) side from an end portion in an axial direction close to a low pressure (intake) side in the screw gear. In this case, the lead angle is defined as an angle of slope of the spiral groove with respect to an axis of the screw gear. In the case that both the screw gears are rotated in accordance with the rotation of the rotary shafts, the inert gas is sucked into a pump chamber from the outside, is transferred to the discharge side while being compressed by both the screw gears within the pump chamber, and is thereafter discharged to the outside from the interior of the pump chamber.

25 **[0004]** Fig. 4A is a graph showing a manner in which a lead angle θ changes in the screw gear in the Patent Document 1. Fig. 4A shows a change of the lead angle θ from a leading end (an intake side end portion) of the spiral groove (the helix) of the screw gear to a trailing end (a discharge side end portion) by setting a rotation angle x of the spiral gear around an axis to a horizontal axis. As shown in Fig. 4A, the change of the lead angle θ in the spiral groove from the intake side end portion to the discharge side end portion can be expressed as a function $\theta(x)$ of the rotation angle x of the screw gear around the axis. With respect to the horizontal axis of the graph in Fig. 4A, the rotation angle x corresponding to the intake side end portion of the spiral groove is defined as a winding start angle 0, and the rotation angle x corresponding to the discharge side end portion of the spiral groove is defined as a winding end angle E.

30 **[0005]** As shown in the graph in Fig. 4A, the lead angle θ is monotonically increased from a winding start lead angle DegS (for example, 50 degrees) serving as the lead angle corresponding to the winding start angle 0, to a winding end lead angle DegE (for example, 80 degrees) serving as the lead angle corresponding to the winding end angle E. Accordingly, in the Patent Document 1, as shown in Fig. 4B, an entire length L in the axial direction of the screw gear is univocally defined by a monotone increasing function $\theta(x)$ using the winding start lead angle DegS and the winding end lead angle DegE.

35 **[0006]** In other words, the monotone increasing function $\theta(x)$ indicating the change of the lead angle θ of the screw gear can be expressed by the following expression (11), and a constant k in the expression (11) can be expressed by the following expression (12). In this case, reference symbol r denotes a radius of a pitch circle of the screw gear.

$$45 \quad \theta(x) = \text{DegS} + k \cdot x \quad (11)$$

$$50 \quad k = (\text{DegE} - \text{DegS}) / (2\pi r \cdot E) \quad (12)$$

[0007] In accordance with the expressions (11) and (12), the entire length L of the screw gear can be univocally defined by the following expression (13).

$$55 \quad L = 1/k \cdot \log(\sin(\text{DegS} + k \cdot 2\pi r \cdot E) / \sin(\text{DegS})) \quad (13)$$

[0008] The expression (13) mentioned above indicates the fact that the entire length L of the screw gear is determined by the winding start lead angle DegS and the winding end lead angle DegE in the screw gear.

[0009] Further, in the screw pump of the Patent Document 1 mentioned above, a volumetric capacity of a plurality of gas actuation chambers defined within the pump chamber by the screw gear becomes gradually small toward the discharge side from the intake side, and the gas is compressed as it is transferred toward the discharge side actuation chamber. To change the manner in which the volumetric capacity of the actuation chamber is changed from the intake side to the discharge side, in other words, to change the manner in which a gas compression characteristic of the screw pump, the winding start lead angle DegS and the winding end lead angle DegE affecting the entire length L of the screw gear are changed. On the other hand, since the screw gear is accommodated within the pump chamber in the vacuum pump, it is necessary that the entire length L of the screw gear be set to a value by which the screw gear can be accommodated within the pump chamber. However, in the case of changing the winding start lead angle DegS and the winding end lead angle DegE for changing the gas compression characteristic of the screw pump, there can be generated a case that the entire length L of the screw gear comes to a value by which the screw gear cannot be accommodated within the pump chamber. Accordingly, the gas compression characteristic of the screw pump in the Patent Document 1 cannot be freely adjusted.

Patent Document 1: Japanese Laid-Open Patent Publication No. 9-32766

DISCLOSURE OF THE INVENTION

[0010] An objective of the present invention is to provide a screw pump and a screw gear which have a high degree of flexibility in a fluid compression characteristic.

[0011] In order to achieve the objective mentioned above, in accordance with the present invention, there is provided a screw gear having a portion in which a lead angle is continuously changed from a leading end to a trailing end of a spiral groove. In the case of expressing, as a lead angle change function, a change of the lead angle from a winding start angle serving as a rotation angle corresponding to the leading end of the spiral groove to a winding end angle serving as a rotation angle corresponding to a trailing end of the spiral groove with respect to a rotation angle of the screw gear around an axis, the lead angle change function is constituted by a combination of a plurality of change functions having different manners of changing.

[0012] Further, in accordance with the present invention, there is provided a screw pump apparatus comprising a pair of screw gears mating with each other and a pump chamber accommodating the screw gears. The screw gears rotate while mating with each other, whereby a fluid sucked into the pump chamber is transferred in an axial direction of the screw gears while being compressed within the pump chamber. Each of the screw gears is constituted by the screw gear structured as mentioned above, and an actuation chamber for compressing the fluid is defined between adjacent thread ridge portions in the axial direction of each respective screw gear.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013]

Fig. 1 is a cross-sectional plan view of a screw vacuum pump in accordance with an embodiment of the present invention;

Fig. 2A is a graph showing a change of a lead angle of a screw rotor;

Fig. 2B is a graph explaining an axial length of the screw rotor;

Fig. 3A is a graph showing a change of a lead angle of a screw rotor;

Fig. 3B is a graph explaining an axial length of the screw rotor;

Fig. 4A is a graph showing a change of a lead angle of a prior art screw rotor; and

Fig. 4B is a graph explaining an axial length of the prior art screw rotor.

BEST MODE FOR CARRYING OUT THE INVENTION

[0014] A description will be given of one embodiment according to the present invention with reference to Figs. 1 to 3B.

[0015] As shown in Fig. 1, a screw vacuum pump 11 in accordance with the present embodiment is provided with a cylindrical rotor housing member 12, a lid-shaped front housing member 13 jointed to a front end (a left end in Fig. 1) of the rotor housing member 12, and a plate-shaped rear housing member 14 jointed to a rear end (a right end in Fig. 1) of the rotor housing member 12. A stepped mounting hole 14a is formed in the rear housing member 14, and a bearing body 15 is fixed to the rear housing member 14 by bolts in a state of being fitted to the mounting hole 14a. A pair of screw rotors (screw gears) 16 serving as fluid transfer bodies are accommodated within the rotor housing member 12. A pump chamber 17 is defined between outer peripheral surfaces of the screw rotors 16 and an inner peripheral surface

of the rotor housing member 12. The specific structure of the screw rotors 16 will be described later.

[0016] A pair of support holes 18 are formed in the bearing body 15 to extend therethrough, and a rotary shaft 19 is inserted in and supported by each of the support holes 18. An end portion (a left end in Fig. 1) of each of the rotary shafts 19 protrudes into the pump chamber 17 from the corresponding support hole 18, and one of both the screw rotors 16 is fixed to the end portion of each of the rotary shafts 19 by a bolt. In other words, each of the screw rotors 16 is coupled to the corresponding rotary shaft 19 so as to integrally rotate with the rotary shaft 19.

[0017] A gear housing member 20 formed as a cylinder having a closed end is fixed to a rear end of the rear housing member 14. End portions (right ends in Fig. 1) 19a of both the rotary shafts 19 respectively protrude into the gear housing member 20, and gears 21 are fastened and attached to the protruding end portions 19a in a mating state. An electric motor 22 serving as a drive source is attached to an outer surface of the gear housing member 20. The end portion 19a of one rotary shaft (the rotary shaft in a lower side in Fig. 1) 19 of both the rotary shafts 19 is coupled to an output shaft 22a of the electric motor 22 extending into the gear housing member 20 via a shaft coupling 23.

[0018] An suction port 24 allowing a fluid, specifically an inert gas such as F_2 gas or the like to be introduced is formed substantially in a center portion of the front housing member 13 in such a manner as to be communicated with the pump chamber 17. A discharge port (not shown) allowing the inert gas to be discharged is formed in a peripheral wall of the rotor housing member 12 near an end portion of the rotor housing member 12 positioned in an opposite side to the suction port 24 in such a manner as to be communicated with the pump chamber 17. The discharge port is positioned in a lower portion in a substantially center of a width direction (a vertical direction in Fig. 1) of the rotor housing member 12. If the electric motor 22 is driven, the screw rotors 16 are rotated inversely in accordance with the rotation of the rotary shafts 19. Accordingly, the inert gas sucked into the pump chamber 17 via the suction port 24 is transferred toward the discharge port within the pump chamber 17 in an axial direction of the screw rotors 16 while being compressed, and is thereafter discharged to an outer portion from the discharge port.

[0019] Next, a description will be given of the screw rotors 16.

[0020] As shown in Fig. 1, each of the screw rotors 16 is formed as a single thread screw gear, and has a spiral groove, that is, a thread ridge 16a and a thread groove 16b on an outer peripheral surface thereof. The screw rotors 16 extend in parallel to each other within the pump chamber 17 such that the thread ridge 16a on one of the screw rotors 16 and the thread groove 16b in the other are mated with each other. Actuation chambers 25 for the inert gas are formed between the adjacent thread ridges 16a in the axial direction of the respective screw rotors 16 within the pump chamber 17. These actuation chambers 25 transfer the inert gas toward the discharge port from the suction port 24, that is, toward a high pressure side from a low pressure side, while compressing the inert gas.

[0021] Each of the screw rotors 16 has a lead angle (also called as a torsion angle) θ continuously changing in accordance with a spiral groove (a helix) of the screw rotor 16. In this case, the lead angle θ is defined as an angle of slope of the spiral groove (the thread ridge 16a and the thread groove 16b) with respect to an axis of the screw rotor 16. The screw rotor 16 is formed such that a lead P1 in a portion closest to the suction port 24 becomes maximum and a lead P4 in a portion closest to the discharge port becomes minimum, so that a volumetric capacity of the actuation chamber 25 is gradually reduced from the suction port 24 (the intake side) toward the discharge port (the discharge side). Specifically, the lead angle θ is changed such that the lead gradually becomes smaller from a maximum lead P1 to a smaller lead P2 in a first range (an intake side range) from the portion closest to the suction port 24 of the screw rotor 16 to an intermediate position m in the axial direction. The lead angle θ is changed in a different manner from that in the first range, such that the lead gradually becomes shorter from a lead P3 to a smaller lead P4 in a second region (a discharged side range) from the intermediate position m of the screw rotor 16 to the portion closest to the discharge port. In the present embodiment, since the screw rotor 16 is formed as the shape of the single thread screw gear, a distance in the axial direction at a time of making one turn of the axis of the screw rotor 16 along the lead of the screw rotor 16, that is, the spiral groove (the helix) is equal to the pitch of the thread ridge 16a.

[0022] Fig. 2A is a graph showing a manner in which the lead angle θ of the screw rotor 16 changes in the present embodiment. Fig. 2A, in which the horizontal axis represents the rotation angle x around the axis of the screw rotor 16, shows the change of the lead angle θ from a leading end (an intake side end portion) to a trailing end (a discharge side end portion) of the spiral groove (the helix) of the screw rotor 16. As shown in Fig. 2A, the change of the lead angle θ in the spiral groove from the intake side end portion to the discharge side end portion can be expressed as a function $\theta(x)$ of the rotation angle x around the axis of the screw rotor 16. Hereinafter, the function $\theta(x)$ is called as the lead angle change function $\theta(x)$. In this case, with respect to the horizontal axis of the graph in Fig. 2A, the rotation angle x corresponding to the intake side end portion of the spiral groove is defined as a winding start angle 0, the rotation angle x corresponding to the intermediate position m is defined as a switch angle M, and the rotation angle x corresponding to the discharge side end portion of the spiral groove is defined as a winding end angle E. In other words, in the case of tracking back the spiral groove from the intake side end portion to the discharge side end portion while turning around the axis of the screw rotor 16, the rotation angle x corresponding to the intake side end portion of the spiral groove is defined as the winding start angle 0, the rotation angle x at a time of reaching the intermediate position m is defined as the switch angle M, and the rotation angle x at a time of reaching the discharge side end portion of the spiral groove is

defined as the winding end angle E.

[0023] As shown in Fig. 2A, during a period of the rotation angle x from the winding start angle 0 to the winding end angle E, the lead angle change function $\theta(x)$ is constituted by a combination of a plurality of (two in Fig. 2A) change functions $\theta_1(x)$ and $\theta_2(x)$ having different manners of changing. In other words, during a period of the rotation angle x from the winding start angle 0 to the winding end angle E, the change of the lead angle θ is expressed by the combination of a plurality of change functions $\theta_1(x)$ and $\theta_2(x)$ having different manners of changing.

[0024] The change function $\theta_1(x)$ corresponds to a first change function (an intake side change function) corresponding to an angle range from the winding start angle 0 to the switch angle M, and expresses the change of the lead angle θ in the first range (the intake side range). The change function $\theta_2(x)$ corresponds to a second change function (the intake side change function) corresponding to an angle range from the switch angle M to the winding end angle E, and expresses the change of the lead angle θ in the second range (the discharge side range). The second change function $\theta_2(x)$ expresses the change of the lead angle θ by a slow change degree in comparison with the first change function $\theta_1(x)$. Both of the first change function $\theta_1(x)$ and the second change function $\theta_2(x)$ are constituted by a monotone increasing function which gradually increases the lead angle θ in accordance with change of the rotation angle x from the winding start angle 0 toward the winding end angle E.

[0025] With respect to a vertical axis of the graph in Fig. 2A, "DegS" corresponds to a lead angle in the intake side end portion of the spiral groove corresponding to the winding start angle 0, that is, a winding start lead angle, "DegM" corresponds to a lead angle in the intermediate position m corresponding to the switch angle M, that is, a switch lead angle, and "DegE" corresponds to a lead angle in the discharge side end portion of the spiral groove corresponding to the winding end angle E, that is, a winding end lead angle. For example, it is assumed that the winding start lead angle DegS is set to 50 degrees, the switch lead angle DegM is set to 70 degrees, and the winding end lead angle DegE is set to 80 degrees. In this case, the lead angle is monotonically increased by 20 degrees in a comparatively steep manner from the winding start angle 0 to the switch angle M. On the other hand, the lead angle is monotonically increased by 10 degrees in a comparatively slow manner from the switch angle M to the winding end angle E.

[0026] A total of lead obtained by turning around the axis of each screw rotor 16 from the winding start angle 0 to the winding end angle E can be determined as an entire length L in the axial direction of the screw rotor 16 on the basis of the lead angle change function $\theta(x)$ obtained by combining the first change function $\theta_1(x)$ and the second change function $\theta_2(x)$. In other words, as shown in Fig. 2B, an axial length in the first range of the screw rotor 16, that is, a first axial length (an intake side axial length) L1 is determined on the basis of the first change function $\theta_1(x)$ corresponding to the angle range from the winding start angle 0 to the switch angle M. Further, an axial length in the second range of the screw rotor 16, that is, a second axial length (a discharge side axial length) L2 is determined on the basis of the second change function $\theta_2(x)$ corresponding to the angle range from the switch angle M to the winding end angle E. Further, the total of both the axial lengths L1 and L2 is determined as the entire length L in the axial direction of the screw rotor 16.

[0027] The first change function $\theta_1(x)$, the second change function $\theta_2(x)$ and the entire length L (= L1 + L2) in the axial direction of the screw rotor 16 obtained on the basis of the change functions $\theta_1(x)$ and $\theta_2(x)$ can be expressed by the following expressions.

[0028] First, the first change function $\theta_1(x)$ corresponding to the angle range ($0 < x < M$) from the winding start angle 0 to the switch angle M can be expressed by the following expression (1), and a constant k1 in the expression (1) can be expressed by the following expression (2). In this case, r in the expression (2) corresponds to the radius of the pitch circle of the screw rotor 16.

$$\theta_1(x) = \text{DegS} + k_1 \cdot x \quad (1)$$

$$k_1 = (\text{DegM} - \text{DegS}) / (2\pi r \cdot M) \quad (2)$$

[0029] It is assumed that the winding start lead angle DegS is changed to a large value without changing the switch angle M and the switch lead angle DegM, with respect to the first change function $\theta_1(x)$ shown by a solid line in Fig. 2A. In this case, a change degree of the lead angle θ from the winding start angle 0 to the switch angle M becomes gentler than that in the first change function $\theta_1(x)$ shown by the solid line. In other words, the change degree of the volumetric capacity of the actuation chamber 25 from the intake side to the discharge side determining a gas compression characteristic of the pump 11 becomes gentler than that of the case of the first change function $\theta_1(x)$ shown by the solid line in the first range of the screw rotor 16. In contrast, it is assumed that the winding start lead angle DegS is changed to a small value, with respect to the first change function $\theta_1(x)$ shown by the solid line in Fig. 2A. In this case, the change degree of the lead angle θ from the winding start angle 0 to the switch angle M becomes steeper than that in the first

change function $\theta_1(x)$ shown by the solid line. In other words, the change degree of the volumetric capacity of the actuation chamber 25 from the intake side to the discharge side becomes steeper than that of the case of the first change function $\theta_1(x)$ shown by the solid line in the first range of the screw rotor 16.

[0030] On the other hand, the second change function $\theta_2(x)$ corresponding to the angle range ($M < x < E$) from the switch angle M to the winding end angle E can be expressed by the following expression (3), and a constant k_2 in the expression (3) can be expressed by the following expression (4).

$$\theta_2(x) = \text{DegM} + k_2 \cdot (x - M) \quad (3)$$

$$k_2 = (\text{DegE} - \text{DegM}) / (2\pi r \cdot E) \quad (4)$$

[0031] It is assumed that the winding end lead angle DegE is changed to a large value without changing the switch lead angle DegM, with respect to the second change function $\theta_2(x)$ shown by the solid line in Fig. 2A. In this case, a change degree of the lead angle θ from the switch angle M to the winding end angle E becomes steeper than that in the second change function $\theta_2(x)$ shown by the solid line. In other words, the change degree of the volumetric capacity of the actuation chamber 25 from the intake side to the discharge side becomes steeper than that of the case of the second change function $\theta_2(x)$ shown by the solid line in the second range of the screw rotor 16. In contrast, it is assumed that the winding end lead angle DegE is changed to a small value, with respect to the second change function $\theta_2(x)$ shown by the solid line in Fig. 2A. In this case, the change degree of the lead angle θ from the switch angle M to the winding end angle E becomes gentler than that in the second change function $\theta_2(x)$ shown by the solid line. In other words, the change degree of the volumetric capacity of the actuation chamber 25 from the intake side to the discharge side becomes gentler than that of the case of the second change function $\theta_2(x)$ shown by the solid line in the second range of the screw rotor 16.

[0032] Next, a description will be given of the entire length $L (= L_1 + L_2)$ in the axial direction of the screw rotor 16 introduced from the lead angle change functions $\theta_1(x)$ and $\theta_2(x)$ mentioned above.

[0033] The first axial length L_1 in the first range corresponding to the angle range ($0 < x < M$) from the winding start angle 0 to the switch angle M can be expressed by the following expression (5).

$$L_1 = 1/k_1 \cdot \log(\sin(\text{DegS} + k_1 \cdot 2\pi r \cdot M) / \sin(\text{DegS})) \quad (5)$$

[0034] Further, the second axial length L_2 in the second range corresponding to the angle range ($M < x < E$) from the switch angle M to the winding end angle E can be expressed by the following expression (6).

$$L_2 = 1/k_2 \cdot \log(\sin(\text{DegM} + k_2 \cdot 2\pi r \cdot E) / \sin(\text{DegM})) \quad (6)$$

[0035] Accordingly, it is possible to determine the entire length $L (= L_1 + L_2)$ in the axial direction of the screw rotor 16 on the basis of the expressions (5) and (6) mentioned above.

[0036] Next, a description will be given of an operation of the pump 11 structured as mentioned above.

[0037] When rotated by the electric motor 22, the screw rotors 16 mating with each other are rotated together with the rotary shafts 19, and the inert gas is sucked into the pump chamber 17 from the outside via the suction port 24. The inert gas sucked into the pump chamber 17 is transferred toward the discharge port while being compressed within each of the actuation chambers 25 in accordance with the rotation of both the screw rotors 16, and is discharged to the outside from the interior of the pump chamber 17 via the discharge port. Accordingly, in the case that the pump 11 is actuated in a state in which the suction port 24 is connected to a working room or a working container executing various processes with respect to the wafer (not shown) in the semiconductor manufacturing process, a clean vacuum environment is generated within the working room and the working container.

[0038] On the other hand, the screw rotor 16 executes a compression operation in accordance with the following manner. In other words, the inert gas sucked into the pump chamber 17 from the suction port 24 is rapidly compressed at a time of being transferred in the actuation chamber 25 in the first range of the screw rotor 16, because the volumetric capacity change degree of the actuation chamber 25 is comparatively steep. Thereafter, the inert gas is slowly com-

pressed at a time of being transferred in the actuation chamber 25 in the second range of the screw rotor 16, because the volumetric capacity change degree of the actuation chamber 25 is comparatively slow. Accordingly, it is possible to avoid the matter that the rapid pressure increase is generated near the discharge port, and it is possible to suppress the local temperature increase near the discharge port.

[0039] The entire length L in the axial direction of the screw rotor 16 can be defined on the basis of the expressions (1) to (6). In the assumption mentioned above, in the case of changing the manner in which the volumetric capacity of the actuation chamber 25 changes from the intake side to the discharge side determining the gas compression characteristic of the pump 11 without changing the entire length L in the axial direction, the switch lead angle DegM is changed, for example, as shown in Fig. 2A. In this case, in the example in Fig. 2A, the winding start angle θ , the switch angle M and the winding end angle E are not changed, and the winding start lead angle DegS and the winding end lead angle DegE are not changed. In other words, in the case that the switch lead angle DegM is changed, for example, to a value DegM' smaller than a value in the lead angle change function $\theta(x)$ shown by the solid line in Fig. 2A, the first change function $\theta_1(x)$ expresses the gentler change degree of the lead angle θ , as shown by a one-dot chain line in Fig. 2A, and the second change function $\theta_2(x)$ expresses the steeper change degree of the lead angle θ . In this case, as shown by a one-dot chain line in Fig. 2B, the entire length $L (= L_1' + L_2')$ in the axial direction of the screw rotor 16 becomes equal to the entire length $L (= L_1 + L_2)$ in the axial direction before changing the switch lead angle DegM . Further, in the case that the switch lead angle DegM is changed, for example, to a value DegM'' larger than the value in the lead angle change function $\theta(x)$ shown by the solid line in Fig. 2A, the first change function $\theta_1(x)$ expresses the steeper change degree of the lead angle θ , as shown by a two-dot chain line in Fig. 2A, and the second change function $\theta_2(x)$ expresses the gentler change degree of the lead angle. In this case, as shown by a two-dot chain line in Fig. 2B, the entire length $L (= L_1'' + L_2'')$ in the axial direction of the screw rotor 16 becomes equal to the entire length $L (= L_1 + L_2)$ in the axial direction before changing the switch lead angle DegM . As mentioned above, if the lead angle change function $\theta(x)$ is constructed by the combination of a plurality of change functions $\theta_1(x)$ and $\theta_2(x)$ having different manner of changing, the compression characteristic of the pump 11 can be changed by changing the manner in which the lead angle θ changes from the winding start angle θ to the winding end angle E , even in the case that there is any circumstance by which the entire length L in the axial direction of the screw rotor 16 cannot be changed.

[0040] On the other hand, in the case that the entire length L in the axial direction of the screw rotor 16 is changed without changing the first change function $\theta_1(x)$ and the second change function $\theta_2(x)$, the switch angle M is changed, for example, as shown in Fig. 3A. In this case, in the example, shown in Fig. 3A, the winding start angle θ and the winding end angle E are not changed, and the winding start lead angle DegS is not changed. In other words, in the case that the switch angle M is changed to a small value M' , for example, as shown by a one-dot chain line in Fig. 3A, the switch lead angle DegM and the winding end lead angle DegE become smaller in a state in which the change degrees of the lead angle θ respectively expressed by the first and second change functions $\theta_1(x)$ and $\theta_2(x)$ are not changed. As a result, in this case, as shown by a one-dot chain line in Fig. 3B, the entire length L in the axial direction of the screw rotor 16 is changed to a larger value L' . Further, in the case that the switch angle M is changed to a larger value M'' , for example, as shown by a two-dot chain line in Fig. 3A, the switch lead angle DegM and the winding end lead angle DegE become larger in a state in which the change degrees of the lead angle θ respectively expressed by the first and second change functions $\theta_1(x)$ and $\theta_2(x)$ are not changed. As a result, in this case, the entire length L in the axial direction of the screw rotor 16 is changed to a smaller value L'' , as shown by a two-dot chain line in Fig. 3B. As mentioned above, it is possible to arbitrarily change the entire length L in the axial direction of the screw rotor 16 by changing the switch angle M without changing a plurality of change functions $\theta_1(x)$ and $\theta_2(x)$ constituting the lead angle change function $\theta(x)$.

[0041] The embodiment mentioned above has the following advantages.

(1) In the present embodiment, the change of the lead angle θ in the screw rotors 16 is expressed by the lead angle change function $\theta(x)$ obtained by combining a plurality of change functions $\theta_1(x)$ and $\theta_2(x)$ having different manners of changing, from the winding start angle θ to the winding end angle E . Accordingly, it is possible to arbitrarily set the manner in which the lead angle θ changes in accordance with the manner of combining a plurality of change functions $\theta_1(x)$ and $\theta_2(x)$. Accordingly, it is possible to arbitrarily set the compression characteristic (the manner in which the volumetric capacity of the actuation chamber 25 changes) introduced by the manner in which the lead angle θ changes on the basis of a plurality of combined change functions $\theta_1(x)$ and $\theta_2(x)$, in a relation with the entire length L in the axial direction of the screw rotors 16, and it is possible to set such that a compression efficiency becomes optimum in correspondence to the kind of the inert gas (the fluid) to be compressed.

(2) The change degree of the lead angle θ is gentler in the second range in each screw rotor 16 than in the first range. In other words, the volumetric capacity change degree of the actuation chamber 25 determining the compression characteristic of the pump 11 becomes gentler in the second range in each screw rotor 16 than in the first range. Therefore, the volumetric capacity change degree of the actuation chamber 25 becomes gentle near the discharge port of the pump 11 at the time of the operation of the pump. Accordingly, it is possible to reliably avoid

the steep pressure increase near the discharge port and the local temperature increase caused by the steep pressure increase.

(3) Each of the first and second change functions $\theta_1(x)$ and $\theta_2(x)$ constituting the lead angle change function $\theta(x)$ is constituted by the monotone changing function of gradually increasing the lead angle θ in accordance with change of the rotation angle x from the winding start angle 0 to the winding end angle E. Accordingly, the lead in the screw rotor 16 is decreased from the winding start angle 0 toward the winding end angle E. Therefore, in the case that a pair of the screw rotors 16 are rotated within the pump chamber 17 while being mated with each other, the rotation load of the screw rotors 16 becomes small, and it is possible to achieve an improved compression operation of the pump 11.

(4) There is a case that it is required to change the compression characteristic of the pump 11 (the manner in which the volumetric capacity of the actuation chamber 25 changes) in correspondence to the kind of the inert gas to be compressed in the pump 11. In the case mentioned above, in the present embodiment, the switch lead angle DegM is changed at the switch angle M corresponding to the intermediate position m in the axial direction of the screw rotor 16. As a result, it is possible to easily change the compression characteristic of the pump 11 without changing the entire length L in the axial direction of the screw rotor 16 accommodated within the pump chamber 17 having a constraint in space, and it is possible to compress and transfer the various inert gases at an optimum compression efficiency.

(5) There is a case that it is required to change the entire length L in the axial direction of the screw rotor 16 without changing the compression characteristic of the pump 11 (the manner in which the volumetric capacity of the actuation chamber 25 changes) in the case of changing the volumetric capacity of the pump chamber 17 or the like. In the case mentioned above, in the present embodiment, the rotation angle x coming to a boundary where two change functions $\theta_1(x)$ and $\theta_2(x)$ are switched, that is, the switch angle M is changed. In this case, the switch lead angle DegM is also changed in accordance with the change of the switch angle M. As a result, it is possible to easily change the entire length L in the axial direction of the screw rotor 16 without changing the compression characteristic of the pump.

[0042] In this case, the embodiment mentioned above may be changed as follows.

[0043] The fluid transferred while being compressed within the pump chamber 17 in accordance with the rotation of the screw rotors 16 may be constituted by a gas other than the inert gas (F_2 gas or the like), for example, a cooling medium gas, or may be constituted by a liquid such as a working fluid or the like. Further, the screw pump in accordance with the present invention may be applied to other pumps than the vacuum pump.

[0044] A plurality of change functions $\theta_1(x)$ and $\theta_2(x)$ combined for constructing the lead angle change function $\theta(x)$ are not limited to the monotone increasing function, but may be constituted by a quadratic function, an nth degree function, an exponential function or the like.

[0045] The number of the change functions $\theta_1(x)$ and $\theta_2(x)$ combined for constructing the lead angle change function $\theta(x)$ is not limited to two but may be set to three or more as far as it is a plural number.

[0046] The change functions $\theta_1(x)$ and $\theta_2(x)$ combined for constructing the lead angle change function $\theta(x)$ may be structured such that the first change function $\theta_1(x)$ expresses the change of the lead angle θ by the gentler change degree than that of the second change function $\theta_2(x)$, as is different from that shown by the solid line in Fig. 2A.

[0047] In the case that a plurality of functions combined for constructing the lead angle change function $\theta(x)$ are constituted, for example, by the combination of two functions, the structure may be made such that one function is constituted by a change function indicating a state in which the lead angle θ is continuously changed, and the other function is constituted by a function indicating a state in which the lead angle θ is not continuously changed. In other words, it is preferable that the screw rotor 16 has at least a part of the portion where the lead angle θ is continuously changed from the leading end (the intake side end portion) of the spiral groove (the helix) to the trailing end (the discharge side end portion).

Claims

1. A screw gear having a portion in which a lead angle is continuously changed from a leading end to a trailing end of a spiral groove, the screw gear being **characterized in that** in the case of expressing, as a lead angle change function, a change of said lead angle from a winding start angle serving as a rotation angle corresponding to the leading end of said spiral groove to a winding end angle serving as a rotation angle corresponding to a trailing end of said spiral groove with respect to a rotation angle of said screw gear around an axis, said lead angle change

function is constituted by a combination of a plurality of change functions having different manners of changing.

2. The screw gear according to claim 1, **characterized in that** a predetermined rotation angle between said winding start angle and said winding end angle is set as a switch angle, said lead angle change function includes a first change function corresponding to an angle range from said winding start angle to said switch angle, and a second change function corresponding to an angle range from said switch angle to said winding end angle, and said second change function expresses the change of the lead angle by a gentler change degree than said first change function.
3. The screw gear according to claim 1 or 2, **characterized in that** each of said change functions constituting said lead angle change function is constituted by a monotone changing function gradually increasing the lead angle in accordance with change of said rotation angle from said winding start angle to said winding end angle.
4. The screw gear according to claim 1, **characterized in that** the following expressions are satisfied:

$$\theta_1(x) = \text{DegS} + k_1 \cdot x \quad (\text{in this case } 0 < x < M)$$

$$k_1 = (\text{DegM} - \text{DegS}) / (2\pi r \cdot M)$$

$$\theta_2(x) = \text{DegM} + k_2 \cdot (x - M) \quad (\text{in this case } M < x < E)$$

$$k_2 = (\text{DegE} - \text{DegM}) / (2\pi r \cdot E)$$

$$L_1 = 1/k_1 \cdot \log(\sin(\text{DegS} + k_1 \cdot 2\pi r \cdot M) / \sin(\text{DegS}))$$

$$L_2 = 1/k_2 \cdot \log(\sin(\text{DegM} + k_2 \cdot 2\pi r \cdot E) / \sin(\text{DegM}))$$

$$L = L_1 + L_2$$

in which x : rotation angle, 0: winding start angle, E: winding end angle, M: switch angle corresponding to a rotation angle x set between the winding start angle 0 and the winding end angle E, DegS: winding start lead angle corresponding to a lead angle at the winding start angle 0, DegE: winding end lead angle corresponding to a lead angle at the winding end angle E, DegM: switch lead angle corresponding to a lead angle at the switch angle M, $\theta_1(x)$: change function indicating a change of the lead angle in an angle range ($0 < x < M$) from the winding start angle 0 to the switch angle M, $\theta_2(x)$: change function indicating a change of the lead angle in an angle range ($M < x < E$) from the switch angle M to the winding end angle E, k_1 and k_2 : constants, r: radius of a pitch circle of the screw gear, L: entire length in an axial direction of the screw gear, L_1 : length in an axial direction of the screw gear corresponding to the angle range $0 < x < M$, L_2 : length in an axial direction of the screw gear corresponding to the angle range $M < x < E$.

5. A screw pump apparatus comprising a pair of screw gears mating with each other and a pump chamber accommodating the screw gears, wherein the screw gears rotate while mating with each other, whereby a fluid sucked into the pump chamber is transferred in an axial direction of the screw gears while being compressed within said pump chamber, the apparatus being **characterized in that** each of said screw gears is constituted by the screw gear according to any one of claims 1 to 4, and an actuation chamber for compressing the fluid is defined between adjacent thread ridge portions in the axial direction of each screw gear.

Fig. 1

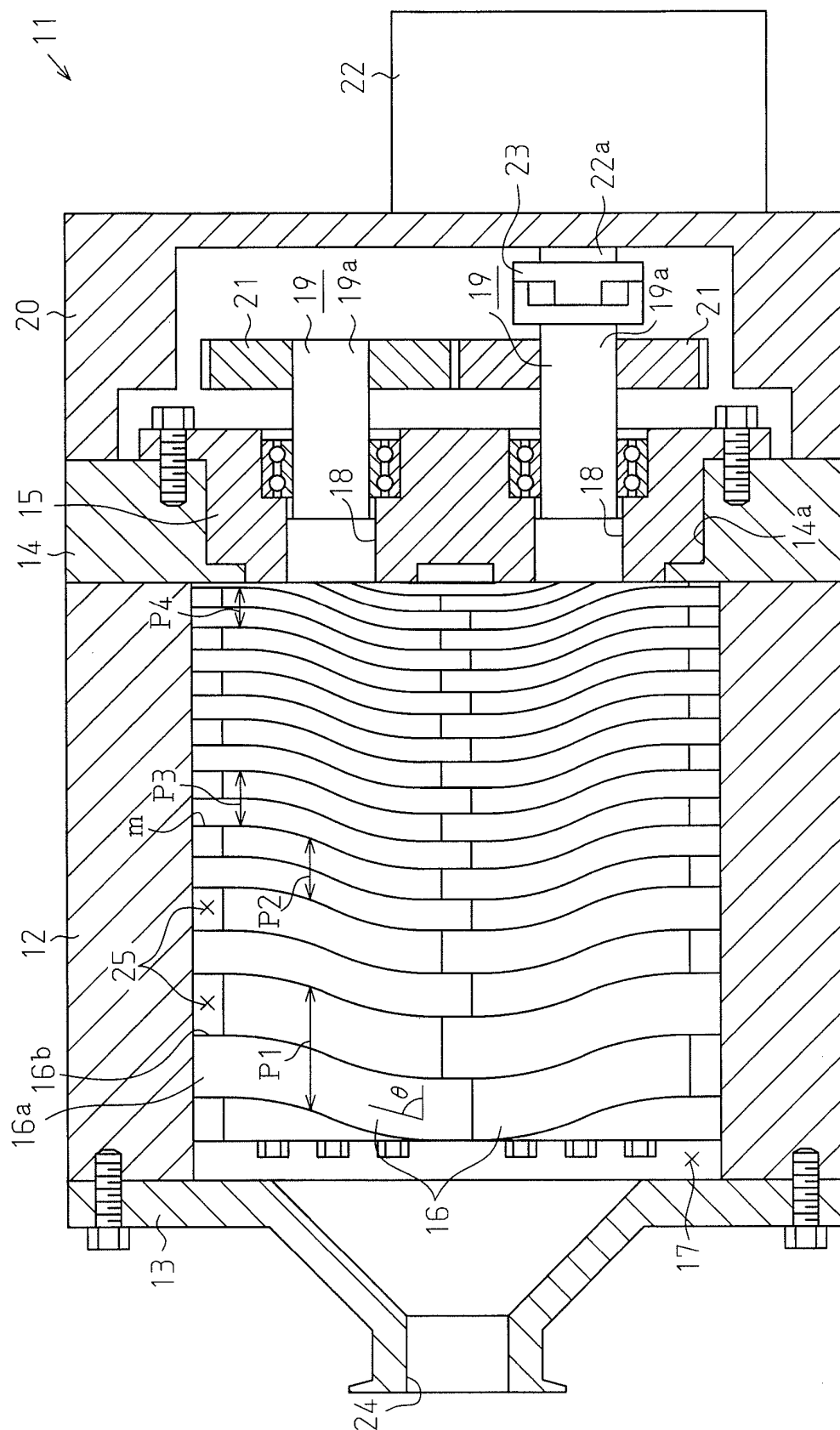


Fig. 2A

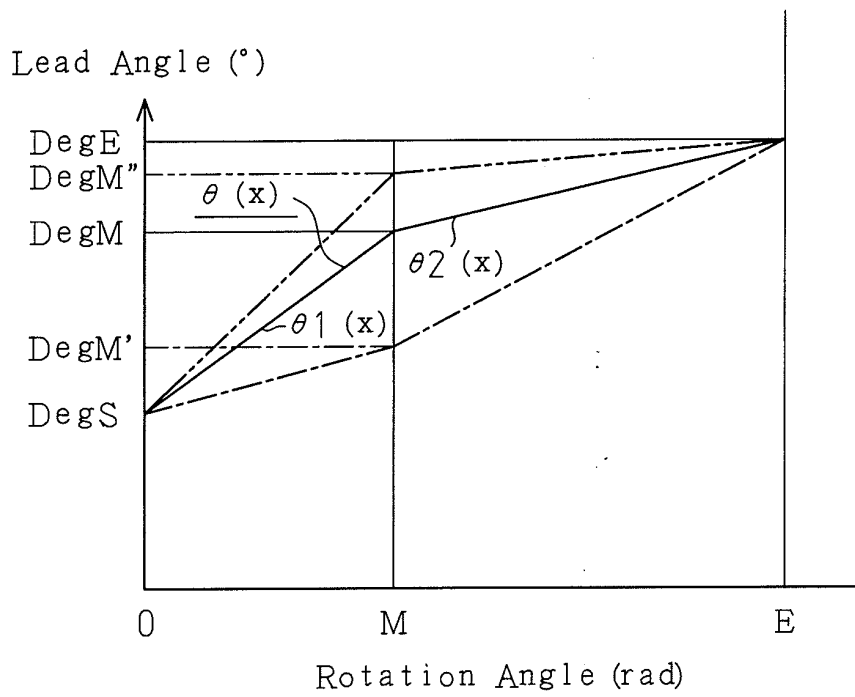


Fig. 2B

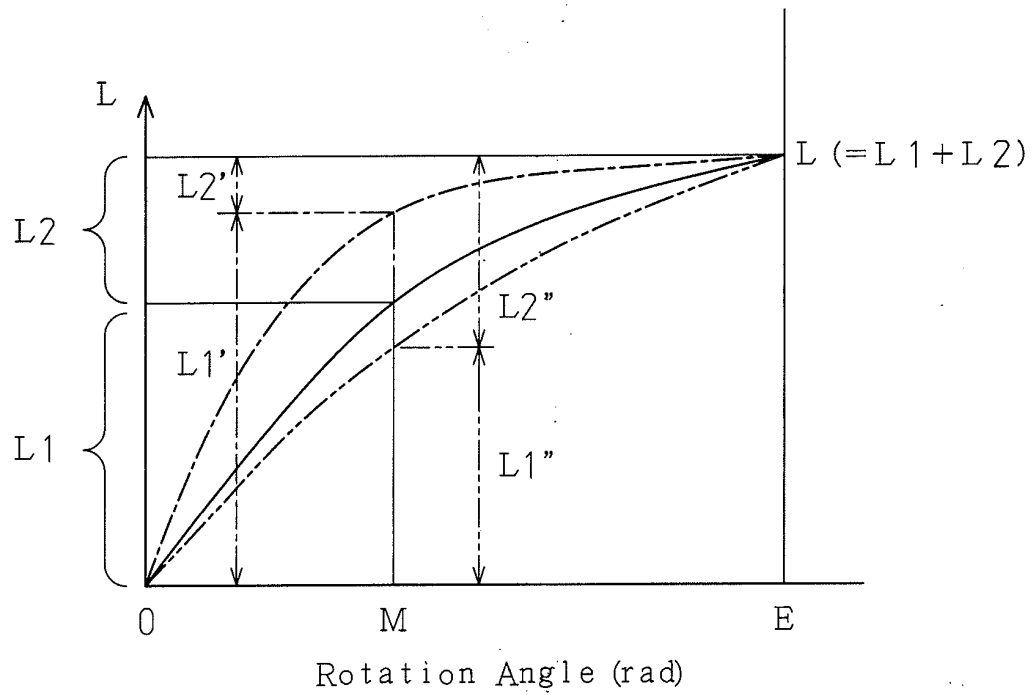


Fig. 3A

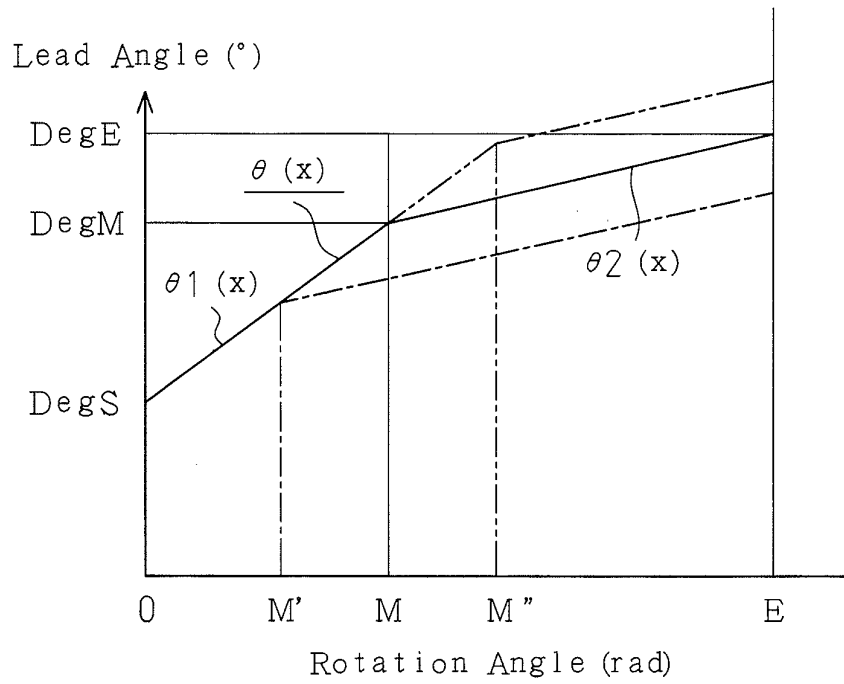


Fig. 3B

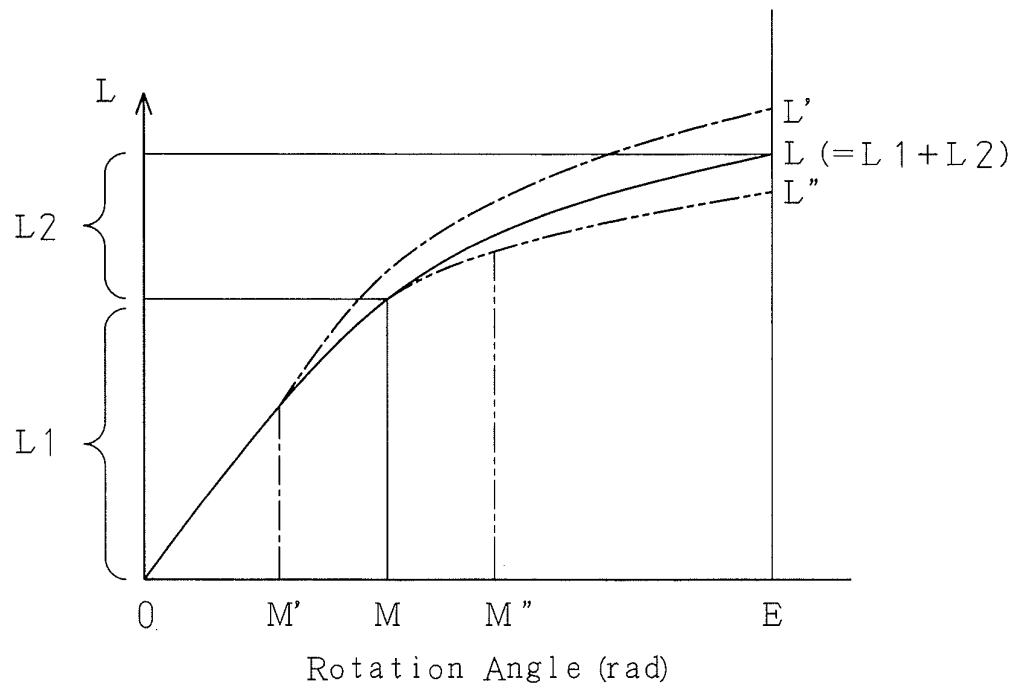


Fig. 4A

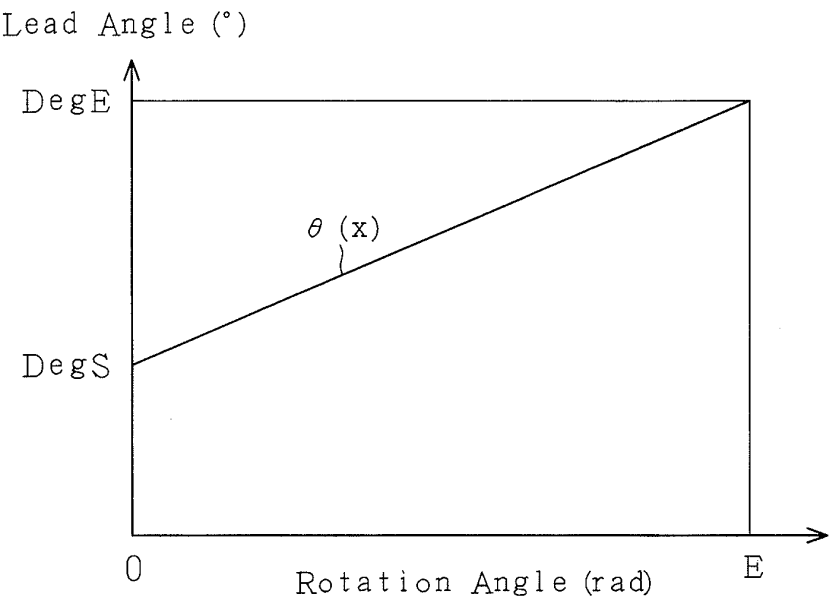
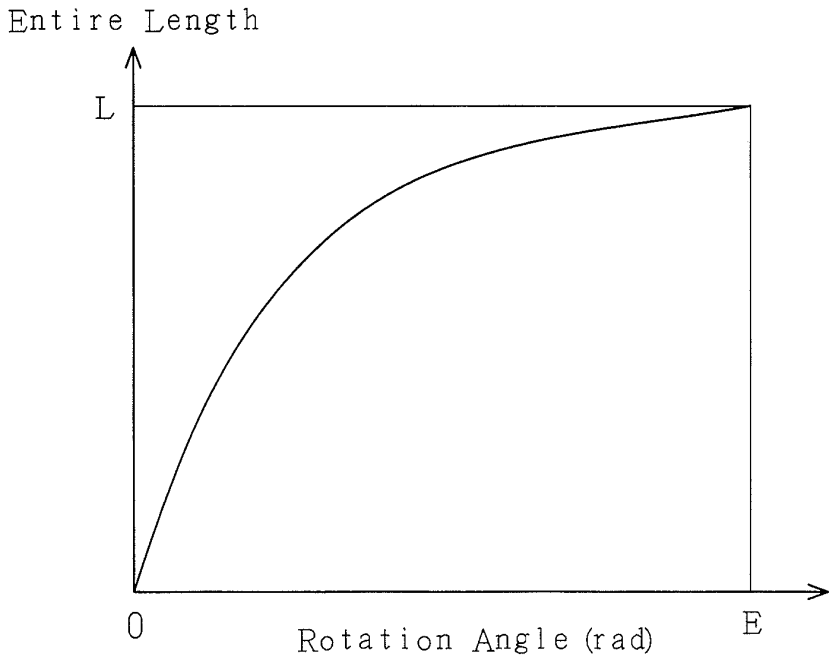


Fig. 4B



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP2005/010853

A. CLASSIFICATION OF SUBJECT MATTER

Int.Cl.⁷ F04C18/16, 25/02, 29/00

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

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int.Cl.⁷ F04C18/16, 25/02, 29/00

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

Jitsuyo Shinan Koho 1922-1996 Jitsuyo Shinan Toroku Koho 1996-2005

Kokai Jitsuyo Shinan Koho 1971-2005 Toroku Jitsuyo Shinan Koho 1994-2005

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

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X A	JP 11-270485 A (Teijin Seiki Co., Ltd.), 05 October, 1999 (05.10.99), Full text; all drawings (Family: none)	1, 5 2-4
A	JP 9-32766 A (Diavac Ltd.), 04 February, 1997 (04.02.97), Full text; all drawings (Family: none)	1-5
A	JP 2000-136787 A (Teijin Seiki Co., Ltd.), 16 May, 2000 (16.05.00), Full text; all drawings (Family: none)	1-5

☐ Further documents are listed in the continuation of Box C.☐ See patent family annex.

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"&" document member of the same patent family

Date of the actual completion of the international search
26 September, 2005 (26.09.05)Date of mailing of the international search report
11 October, 2005 (11.10.05)Name and mailing address of the ISA/
Japanese Patent Office

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REFERENCES CITED IN THE DESCRIPTION

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Patent documents cited in the description

- JP 9032766 A [0009]