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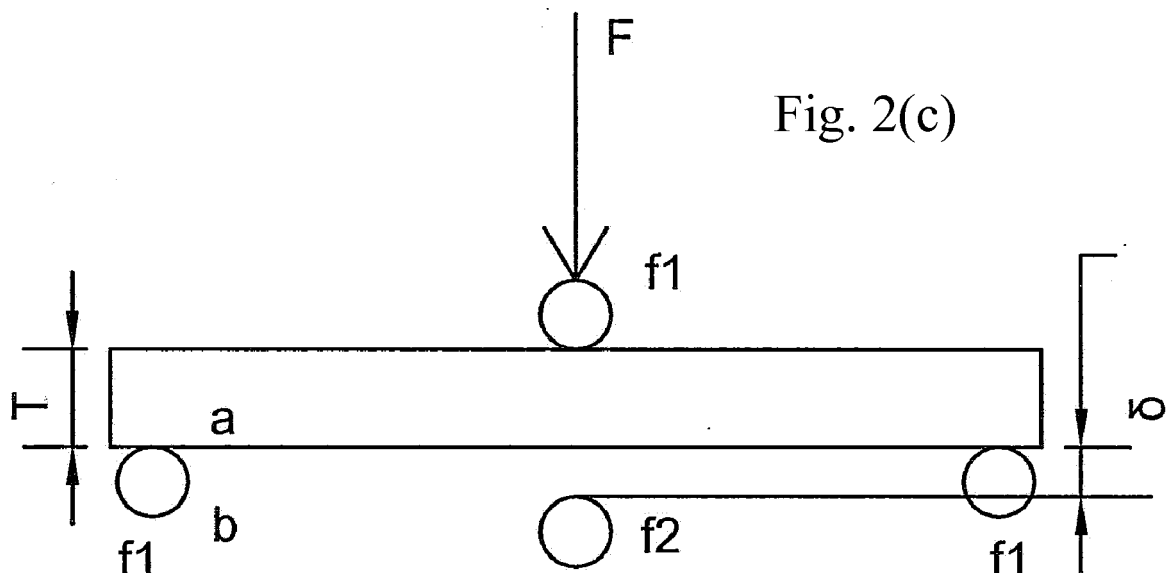
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(54) **A DYNAMIC LOAD ACCURACY ADJUSTABLE AUTOMATIC SCREWDRIVER**

(57) A dynamic load accuracy adjustable automatic screwdriver, more than one single or composite dynamic bearing (A,B,C,D,E,F,G,H) are mounted in the automatic screwdriver, the deformation (δ_1 , δ_2) under each load is preset, the load capacity in the full torque (including large torque, medium torque, small torque) range is calculated from this deformation (δ_1 , δ_2), and the

accurate error values under various torques and friction forces are obtained and removed. The preload torque of screw generates tensile stress, the gear clearance is minimized, the tension loss is reduced, and the overall stiffness is enhanced, so as to increase the strength of torque screwdriver or torque elbow gear, a large torque can be obtained in a small space.



Description

BACKGROUND OF INVENTION

1. Field of the Invention

[0001] The present invention relates generally to a plane bearing technology for automatic screwdriver structure, and more particularly to a dynamic load accuracy adjustable automatic screwdriver.

2. Description of Related Art

[0002] The error of automatic screwdrivers (including electric and pneumatic types) is universally resulted from subtracting the minimum tensile stress and compressive stress from the sum of total maximum tensile stress and compressive stress and maximum friction, after rotation, it is called error value of torque, known as torque accuracy.

[0003] For the torque accuracy of general electric and pneumatic screwdrivers, the torque error value is selected only from large, medium and small torques, the maximum torque is usually selected as the standard of error value, but the accuracy error value of medium and low torques is large, especially in low torque hardness, the error reaches its maximum.

[0004] However, general operators use medium or low torque as the standard of tools, seldom use the maximum torque as the maximum error value. As the maximum torque can shorten the life of tools, this design does not conform to the actual service condition. Sometimes the actual error value is larger than the reference error value in the catalog, the lock screws on aircraft, automobile or other articles are occasionally stripped or loose, resulting in accidents.

SUMMARY OF THE INVENTION

[0005] The primary objective of the present invention is to provide a dynamic load accuracy adjustable automatic screwdriver, more than one single or composite dynamic bearing is installed inside the automatic screwdriver, so that when the automatic screwdriver is in operation, the internal structure, such as clutch, uses the potential difference fulcrum for transfer dispersion under stress, so as to eliminate the torque error value resulted from friction force, and the optimal torque accuracy can be obtained in the full torque (i.e. different torque values) range.

[0006] Another objective of the present invention is to provide a dynamic load accuracy adjustable automatic screwdriver, the preload torque of screw or nut generates tensile stress to minimize the gear clearance, the tension loss decreases, and the overall stiffness is enhanced, so that the strength of elbow gear is increased, the large torque can be obtained in a small space, the corner dimension is reduced, the screws can be locked when lock-

ing small space, that cannot be done by general elbowed automatic screwdriver.

[0007] In order to attain said purposes, the technical proposal of this case is to distribute at least more than one dynamic bearing in the automatic screwdriver. The dynamic bearing comprises at least a flat washer in unlimited thickness and at least more than two high and low potential difference fulcrums composed of unlimited quantity of surrounding steel balls. The dynamic bearing is a bearing of composite design, more than one plane bearing. Each load capacity (e.g. total load tension, total load pressure) is calculated to know the deformation δ of material. The deformation of material is used to calculate the load capacity of full torque range (e.g. total load tension, total load pressure), so that the small torque has the friction error value of small torque, the medium torque has the friction error value of medium torque, and large torque has the friction error value of large torque. The torque error values of all torques are eliminated to make the torque value output more accurate.

[0008] The high and low potential difference fulcrum structures in said dynamic bearing can be formed of steel balls in different diameters.

[0009] The high and low potential difference fulcrum structures in said dynamic bearing can be implemented by making tracks with radians in different depths in the flat washer against the steel ball contact surface.

[0010] The high and low potential difference fulcrum structures in said dynamic bearing can be implemented by making tracks in different turning radii in the flat washer against the steel ball contact surface.

[0011] The high and low potential difference fulcrum structures in said dynamic bearing can be implemented by making tracks in different turning radii in the flat washer against the steel ball contact surface, and setting steel balls in different diameters.

[0012] The high and low potential difference fulcrum structures in said dynamic bearing can be implemented by mounting a support ring smaller than the steel ball diameter in the inner edge of surrounding steel balls.

[0013] The potential difference fulcrum of said dynamic bearing can be implemented by mounting a support base rotating ring in the outer edge of surrounding steel balls.

[0014] The preset deformation of said dynamic bearing includes radial deformation and axial deformation.

[0015] Said dynamic bearing has a prelocking structure to resist axial deformation, the axle through the dynamic bearing is combined with two bearings by screw locking of screws or nuts.

[0016] The present invention changes the constant friction force to variable friction force, so that different friction coefficients are used in different torque states, and the torque accuracy of automatic screwdriver is more accurate. The advantages include: 1) the torque error value decreases, the torque error value CMK increases by one to two units in front of the decimal point, the value is larger the better; 2) the effect of temperature on the automatic screwdriver is reduced, the friction force is re-

duced, so the effect of temperature and force on the automatic screwdriver is reduced, the automatic screwdriver can work at high speed in more cycles, and there will not be large error values; 3) the lubrication is good, the service life of automatic screwdriver is relatively long; 4) the error value decreases when the automatic screwdriver locks hard torque and soft torque; 5) as the friction force decreases, the automatic screwdriver can work in cold or hot operating environment.

BRIEF DESCRIPTION OF THE DRAWINGS

[0017]

FIG. 1 is the linear graph of friction force principle used in the present invention. 15

FIG. 2 (a) is the schematic diagram of bearing stress (small torque) principle used in the present invention. 20

FIG. 2 (b) is the schematic diagram (I) of general plane bearing stress.

FIG. 2 (c) is the schematic diagram (I) of dynamic bearing on the FIG. 2 (a) principle of the present invention. 25

FIG. 3 (a) is the schematic diagram of bearing stress (medium torque) principle used in the present invention. 30

FIG. 3 (b) is the schematic diagram (II) of general plane bearing stress.

FIG. 3 (c) is the schematic diagram (II) of dynamic bearing on the FIG. 3 (a) principle of the present invention. 35

FIG. 4 (a) is the schematic diagram of bearing stress (large torque) principle used in the present invention. 40

FIG. 4 (b) is the schematic diagram (III) of general plane bearing stress.

FIG. 4 (c) is the schematic diagram (III) of dynamic bearing on the FIG. 4 (a) principle of the present invention. 45

FIG. 5 is the schematic diagram of local structure according to the dynamic bearing in FIG. 4 (a) of the present invention. 50

FIG. 6 is the schematic diagram of difference between the steel balls for dynamic bearing of the present invention and general plane bearing. 55

FIG. 7 (a) is the schematic diagram (I) of steel ball setting for the dynamic bearing of the present inven-

tion.

FIG. 7 (b) is the schematic diagram (II) of steel ball setting for the dynamic bearing of the present invention.

FIG. 7 (c) is the schematic diagram (III) of steel ball setting for the dynamic bearing of the present invention.

FIG. 8 (a) is the schematic diagram of Hertz contact principle used in the present invention.

FIG. 8 (b) is the schematic diagram (I) of clutch stress according to FIG. 8 (a) principle.

FIG. 9 is the structural representation of the present invention applied to an upright automatic screwdriver.

FIG. 10 is the structural representation of the present invention applied to an elbowed automatic screwdriver.

FIG. 11 is the schematic diagram (II) of stress principle of the present invention applied to automatic screwdriver.

FIG. 12 is the schematic diagram (III) of stress principle of the present invention applied to automatic screwdriver.

FIG. 13 (a) is the schematic diagram (I) of dynamic bearing structure deformation of the present invention.

FIG. 13 (b) is the schematic diagram (II) of dynamic bearing structure deformation of the present invention.

FIG. 13 (c) is the schematic diagram (III) of dynamic bearing structure deformation of the present invention.

FIG. 13 (d) is the schematic diagram (IV) of dynamic bearing structure deformation of the present invention.

FIG. 13 (e) is the schematic diagram (V) of dynamic bearing structure deformation of the present invention.

FIG. 13 (f) is the schematic diagram (VI) of dynamic bearing structure deformation of the present invention.

FIG. 13 (g) is the schematic diagram (VII) of dynamic bearing structure deformation of the present inven-

tion.

FIG. 14 is the schematic diagram (VIII) of dynamic bearing structure deformation of the present invention.

FIG. 15 is the schematic diagram (IX) of dynamic bearing structure deformation of the present invention.

FIG. 16 is the schematic diagram (X) of dynamic bearing structure deformation of the present invention.

DETAILED DESCRIPTION OF THE INVENTION

[0018] The present invention provides a dynamic load accuracy adjustable automatic screwdriver, at least more than one dynamic bearing is mounted in the automatic screwdriver.

[0019] According to the principles followed by this design, referring to FIG. 1, the magnitude of friction force ($\mu_{\min}$ and $\mu_{\max}$) can influence the accuracy of torque. According to the figure, the total error value is closely related to the magnitude of friction force. In order to obtain the accurate torque, the friction error must be reduced. Therefore, it is very important to make variable friction force μ , this is the key point of the present invention. Because there are different tensile and compressive stresses under different torques, in the range of maximum clamping force $F_{v\max}$ and minimum clamping force $F_{v\min}$ (from spring force and force of clutch), the optimum friction coefficient can be obtained, and the optimum torque accuracy can be obtained (between minimum tolerable limit torque T_{ll} and maximum tolerable limit torque T_{ul}).

[0020] According to said theory, the structure design of the dynamic bearing of the present invention is shown in FIG. 2 (c), including at least a flat washer a in unlimited thickness and at least more than two potential difference fulcrums b composed of unlimited quantity of surrounding steel balls. The fulcrum b with potential difference means any fulcrum has preset deformation in height against the other fulcrum.

[0021] Referring to FIGS. 2-4, an embodiment is shown to describe the structural difference between general plane bearing and dynamic bearing of the present invention according to the stress principle. FIG. 2 (a) is the schematic diagram of basic stress principle of a bearing under the static force F of torque spring (small torque). The infrastructure of the bearing comprises a flat washer a in thickness of t and a fulcrum b, where δ represents the deformation, the F represents the acting force of torque spring, Δ represents the fulcrum b, divided into fulcrum I f1 and fulcrum II f2. When the flat washer a and fulcrum b receive the static force F of torque spring (small torque), the acting force of the static force F on the flat washer a is slight, and the deformation is slight (because

the torque spring is not yet compressed). Therefore, the stress on fulcrum I f1 is also low, and the flat washer a is supported only by fulcrum I f1, not contacting the fulcrum II f2.

[0022] FIG. 2 (b) discloses the plane bearing in common automatic screwdrivers, its structure comprises a flat washer a and a fulcrum b; as the friction force generated in general application is relatively weak, the design does not need many fulcrums b (i.e. steel balls), there is only fulcrum I (i.e. steel ball I) f1 as support, the total friction force is relatively weak (the number of steel balls used is small, so the total friction force is weak). There will not be any problem in the torque accuracy under the static force F of torque spring (small torque), and the service life is not influenced.

[0023] FIG. 2 (c) discloses the dynamic bearing infrastructure of a preferred embodiment of the present invention, the design principle is the same as FIG. 2 (a), including a flat washer a and a fulcrum b, and the fulcrum b comprises steel ball I f1 and steel ball II f2 with potential difference setting. Therefore, when the dynamic bearing is used in the automatic screwdriver, there are still high accuracy torque and long service life under the static force F of torque spring (small torque). The steel ball (f1) above the flat washer a can be set according to actual demand, which can buffer the static force F of the torque spring.

[0024] FIG. 3 (a) discloses the schematic diagram of basic stress principle of a bearing under the dynamic force F of torque spring (medium torque). Based on this design, when the flat washer a and fulcrum b are under the dynamic force F (medium torque), a moderate deformation is generated to flat washer a, the stress on fulcrum I (f1) of fulcrum b is also moderate, and the flat washer a contacts the fulcrum II (f2) of fulcrum b due to deformation.

[0025] FIG. 3 (b) discloses a general plane bearing structure which comprises a flat washer a and a fulcrum b. When the bearing receives the dynamic force F of torque spring (medium torque), the fulcrum I (i.e. steel ball I)(f1) contact surface is enlarged compared with that under static force F (small torque), and the total friction force is increased, so there have been error values.

[0026] FIG. 3 (c) discloses the infrastructure of dynamic bearing of the present invention, the design principle is the same as FIG. 3 (a), the structure comprises a flat washer a and a fulcrum b with high and low potential difference settings. The fulcrum b comprises steel ball I f1 and steel ball II f2. The dynamic bearing is mounted in the automatic screwdriver to replace plane bearing, under the effect of dynamic force F of torque spring (medium torque), the stressed point can be diverted from steel ball I f1 to steel ball II f2, and the errors resulted from steel ball friction force can be eliminated, so as to obtain a high accuracy torque.

[0027] FIG. 4 (a) discloses the schematic diagram of basic stress principle of more than one bearing laminated structure under the high torque F of torque spring (large

torque). The structure comprises at least more than one layer of flat washers a, c, e and multiple fulcrums b, d, f laminated mutually. According to this schematic diagram of stress, when the multi-layer bearing receives the dynamic force F of torque spring (large torque), the laminated structure of at least two layers of flat washers a, c or more than two layers of flat washers a, c, e and more steel ball I f1~steel ball VI f6 as stratified fulcrums are required to withstand separately, so that the uppermost steel ball f1 will not break under direct stress. The final result of this design is that under whatever stress (large, medium and small torques), the friction force errors resulted from different forces in various stages can be eliminated, so as to implement accurate torque.

[0028] However, FIG. 4 (b) discloses the general plane bearing structure comprises a flat washer a and a fulcrum b. Under the dynamic force F of torque spring (large torque), the degree of deformation of flat washer a increases, the steel ball f1 contact surface is enlarged, the total friction force is increased, and the error value increases relatively.

[0029] FIG. 4 (c) discloses the infrastructure of dynamic bearing of the present invention, the design principle is the same as FIG. 4 (a). Therefore, the dynamic bearing of the present invention can use multilayer flat washers a, c, e and multilayer fulcrums b, d, f with high and low potential differences, including steel ball I f1~steel ball VI f6, so that when the dynamic bearing is used in the automatic screwdriver, under the dynamic force F of torque spring (large torque), the stress on various layers can be transferred by layers, and the friction force is eliminated, so the high accuracy torque can be obtained.

[0030] Said multi-layer dynamic bearing structure is not limited to the fixed laminated form shown in the figure, in actual implementation, multiple single or laminated dynamic bearings can be mounted in the automatic screwdriver to replace the original fixed bearing as required, so that multiple distributed single-layer or multi-layer dynamic bearings are formed in the automatic screwdriver, so that the dispersion can be diverted by the stress provided by all dynamic bearings, the friction force error is eliminated to obtain the required high torque accuracy.

[0031] Referring to FIGS. 4 and 5, according to the stress principle shown in FIG. 4 (a), under the dynamic F of torque spring (large torque), the fulcrum IV f4 in the laminated bearing structure generates shear force at the same time, so as shown in FIG. 4 (c), the fulcrum d is designed as multiple steel balls f4 under the flat washer c of dynamic bearing of the present invention, so as to disperse the Fmax (F1) into the stress type of two component forces (i.e. F2 and F3), the total down pressure can be reduced, then the counterforce is reduced, so as to obtain the optimal torque accuracy.

[0032] A conclusion is derived from said description: the magnitudes of total friction force and torque (acting force) are determined by the variation of the number of fulcrums (i.e. steel balls) of dynamic bearing and dissimilar thickness of at least a flat washer (t1~t3), the dynamic

bearing can obtain the optimum torque accuracy under large, medium and small torques.

[0033] Referring to FIG. 6, a preferred embodiment is shown to describe the structural difference between general plane bearing and the dynamic bearing of the present invention. According to the elementary theory shown in the figure, the flat washer a and fulcrum b (i.e. steel ball) in static state are described, under total friction torque of different loads (30%, 50%, 80%, 100%), the following result is obtained from equation $T=f\mu \times r$. When the friction force is fixed, but the steel ball radius is variable, $f\mu_1 > f\mu_2$, and $T_1 > T_2$, T1 loses major friction force, so the torque accuracy is poor; in a similar way, T2 loses minor friction force, so the torque accuracy is high.

[0034] The proportional relation shown in FIG. 6 shows the correlation between steel ball and friction force, and according to the conclusion: small torque only has small friction force, medium torque has medium friction force, and large torque has large friction force, so the friction force increases with the torque.

[0035] Referring to FIG. 7, the number of steel balls determines the magnitude of friction force. As shown in FIG. 7 (a), referring to said single-layer dynamic bearing structure composed of flat washer a and fulcrum b, under the static force F of torque spring (small torque), if three steel balls are used, the friction force is at its minimum, the total friction force = $3f\mu$; as shown in FIG. 7 (b), the single-layer dynamic bearing structure is taken as an example, under the dynamic force F of torque spring (medium torque), if five steel balls are used, the total friction force = $5f\mu$; as shown in FIG. 7 (c), the multi-layer dynamic bearing structure of the present invention is taken as an example, under high torque F (large torque), if ten steel balls are used, the friction force is maximized, the total friction force = $10f\mu$. Therefore, the larger the number of steel balls is, the more friction force is lost, and the worse is the torque accuracy.

[0036] Referring to FIG. 8, in actual dynamic state, the rotation of automatic screwdriver receives the thrust (FS) generated by torque spring. According to the Hertz contact theory in FIG. 8 (a), when two elastic bodies are in point contact, the formed contact area can be regarded as an ellipse. As shown in FIG. 8 (b), this theory is used to discuss the clutch (1) in general automatic screwdriver structure, the schematic of the thrust (FS) of torque spring is shown in the figure.

[0037] The present invention uses the elastic deformation of steel ball under load to design and describe the rolling, sliding and no-load frictions of bearing. As shown in FIGS. 9 and 10, for example, when the preset thrust (FS) is 750N, the 2.0mm steel ball has 8mm deformation, this δ deformation is preset in the dynamic bearing E of multi-layer dynamic bearing A~H in the automatic screwdriver, and the rolling and sliding are distinguished according to the steel ball radius and rotation radius, so that linear friction loss can be obtained when the thrust (FS) is 30%~100%. According to the compressive stress equation, $P_0 = 3N/2\pi ab$, the deformation δ can be ob-

tained, rolling friction force $FT=pN/r$, sliding friction force $FU=uN$. If the rolling friction force (T) is represented by moment: $T=pN$. When $a > b$, the steel ball rotates axially. When $FT > FU$, the steel ball rolls. When $a > b$, the steel ball rotates axially. When $FT < FU$, the steel ball slides.

[0038] FIGS. 9 and 10 disclose two types of automatic screwdriver. FIG. 9 discloses the local structure of an upright automatic screwdriver. FIG. 10 discloses the local structure of an elbowed automatic screwdriver. When the dynamic bearing of the present invention is implemented in said two types of automatic screwdriver, more than one layer of dynamic bearing can be mounted in the automatic screwdriver according to the actually preset torque value, so as to increase the torque accuracy. The figures only take two common automatic screwdriver structures as examples (not limited to them), as shown in FIGS. 9 and 10, the multi-layer dynamic bearings A~H are set in the automatic screwdriver; as mentioned above, the 2.0mm steel ball has δ mm deformation under 750N, and this δ deformation is preset in the dynamic bearing E, the thrust (FS) of the torque spring as 30%, 50%, 80% and 100% and the equation are used for description: (I) when the thrust (FS) is 30%, A layer rolls (pa), B layer rolls (pb), C layer rolls (pc), D layer rolls (pd), E layer is under no load in axial direction, F layer is under no load; total friction torque of load = $Na \times pa + Nb \times pb + Nc \times pc + Nd \times pd$. (II) when the thrust (FS) is 50%, A layer slides, B layer slides, C layer slides, D layer slides, E layer rolls (pe), F layer is under no load; total friction torque of load = $Na \times \mu_a \times Ra + Nb \times \mu_b \times Rb + Nc \times \mu_c \times Rc + Nd \times \mu_d \times Rd + Ne \times pe$. (III) when the thrust (FS) is 80%, A layer is stationary, B layer slides, C layer slides, D layer slides, E layer rolls (pe), F layer rolls (pf); total friction torque of load = $Na \times \mu_a \times Ra + Nb \times \mu_b \times Rb + Nc \times \mu_c \times Rc + Nd \times \mu_d \times Rd + Ne \times pe + Nf \times pf$. (IV) when the thrust (FS) is 100%, A layer is stationary, B layer slides, C layer slides, D layer slides, E layer rolls (pe), F layer slides; total friction torque of load = $Na \times \mu_a \times Ra + Nb \times \mu_b \times Rb + Nc \times \mu_c \times Rc + Nd \times \mu_d \times Rd + Ne \times pe + Nf \times \mu_f \times Rf$.

[0039] The above is the implementation of multiple dynamic bearings in automatic screwdriver. The present invention uses composite bearing design, and uses more than one dynamic bearing for stress dispersion. There are different total friction forces of load under different loads, so there are different total friction torques of load under different torques (e.g. 30%, 50%, 80%, 100%), and the total friction torque is controlled in a numerical range smaller than normal condition for optimal and accurate locking torque. However, the general single plane bearing on the market has only one preset fixed total friction force value of load, so the screw torque under different requirements cannot be accurate. On the contrary, the dynamic bearing of the present invention is applicable to the locking operation required of different torques, and the error value is eliminated and more accurate torque value is obtained.

[0040] Referring to FIGS. 9-11, according to the elementary theory II of clutch stress in FIG. 11, the tensile stress and compressive stress generated before the clutch trip (B layer) under different torques (30%, 50%, 80%, 100%) are taken as examples; the FPS is the tensile stress generated by clutch trip, FS is the initial thrust of torque spring compressing clutch, FP is the connecting shaft prelocking force, Fsd is the binding force of torque spring pushing back clutch after clutch trip. The FPS tensile stress is generated when the clutch 1 trips, the tensile stress of the clutch 1 pulls off the connecting shaft 2, so the connecting shaft 1 is prelocked with FP (preload on connecting shaft), if $FPS > FP$, the dynamic bearing G layer acts; in the same way, the 2.0mm steel ball generates δ mm deformation under 750N, this δ deformation is preset in the dynamic bearing E layer, the tensile stress FPS under stress is compared with FIGS. 9 and 10 in the state of 30%, 50%, 80% and 100% respectively: (I) when FPS is 30%, A layer rolls (pa), B layer rolls (pb), C layer rolls (pc), D layer rolls (pd), E layer rolls (pe), F layer is under no load, G layer is under no load, H layer is under no load. (II) when FPS is 50%, A layer slides, B layer slides, C layer slides, D layer slides, E layer rolls (pe), F layer rolls (pf), G layer is under no load, H layer rolls (ph). (III) when FPS is 80%, A layer is stationary, B layer slides, C layer slides, D layer slides, E layer rolls (pe), F layer rolls (pf), G layer rolls (pg), H layer rolls (ph). (IV) when FPS is 100%, A layer is stationary, B layer slides, C layer slides, D layer slides, E layer rolls (pe), F layer slides, G layer rolls (pg), H layer rolls (ph).

[0041] At least one of said A~G layers of dynamic bearing can be trimmed in implementation, the steel balls of the selected layer of dynamic bearing are replaced by smaller steel balls, so that the steel balls of the layer of dynamic bearing turn from rolling state into sliding state, so as to reduce the total friction torque to export more accurate torque value.

[0042] Referring to FIGS. 9-10 and FIG. 12, according to the elementary theory in FIG. 12, when the clutch 1 trips (B layer) and connects connecting shaft 2, the Fsd thrust is generated, the thrust of the clutch 1 impacts the connecting shaft 2, so the connecting shaft 2 is prelocked with FP (i.e. preload on connecting shaft), once $FP > Fsd$, the dynamic bearing G layer takes effect.

[0043] Referring to FIG. 13, there are different numbers of fulcrums in order to obtain different loads, especially the different arrangements of steel balls, pivoted loop, fixed ring, support ring and support base of internal structure of automatic screwdriver (FIG. 9, FIG. 10), and the dynamic bearing variation is discussed according to $T=F \times R$ to improve the dynamic bearing structure, where δ represents the deformation of dynamic bearing contacting the next dynamic bearing under the F force; Δ represents the stressed fulcrum of plane bearing; R represents the radius of rotation; F represents the stress on object.

[0044] In the dynamic bearing of the present invention, the actual high and low potential difference fulcrums set-

tings have the following types, compared with the automatic screwdriver structure in FIGS. 9 and 10; as shown in FIG. 13 (a), the steel balls in the same rotation radius R are mounted on the flat washer of dynamic bearing, but the steel balls in different diameters are fulcrums, such as A and B layers in the automatic screwdriver. As shown in FIG. 13 (b), the fixed ring is mounted on the flat washer of dynamic bearing, the orbital arc difference results in height difference of steel balls, namely, the track in the contact surface of flat washer against steel ball has radian, such as A and B layers in the automatic screwdriver. As shown in FIG. 13 (c), the steel balls in different rotation radii R are mounted on the flat washer of dynamic bearing, but the steel balls are in the same diameter, namely, there are tracks in different rotation radii on the contact surface of flat washer against steel balls, such as A and B layers in the automatic screwdriver. As shown in FIG. 13 (d), there are steel balls in different rotation radii R on the flat washer of dynamic bearing, and the steel balls are in different diameters, such as A and B layers in the automatic screwdriver. As shown in FIG. 13 (e), the support ring adjusting rotating ring 4 for support is mounted on the flat washer of dynamic bearing, and there are steel balls in the same diameter, namely, a support ring adjusting rotating ring 4 in height slightly smaller than the steel ball diameter is mounted in the inner edge of steel balls, such as C and D layers in the automatic screwdriver. As shown in FIG. 13 (f), the support base adjusting rotating ring 5 is mounted in the outer edge of flat washer of dynamic bearing, and there are steel balls in the same diameter, a support base adjusting rotating ring 5 in height slightly smaller than the flat washer thickness is mounted in the outer edge of steel balls, such as C and D layers in the automatic screwdriver. As shown in FIG. 13 (g), the bearing fixing support base 6 is mounted between two layers of dynamic bearing, besides resisting radial deformation, this practice can resist axial deformation, such as E and F layers in the automatic screwdriver.

[0045] According to the (a)~(g) structure types, different materials and flat washer thicknesses (t) can result in different deformations δ , the deformation δ generated by different acting forces (e.g. small, medium and large torques), and the tensile and compressive stresses and deformations are calculated to generate the optimal dynamic bearing structure, the load capacity (e.g. total load tension, total load pressure) under full torque (small, medium and large torques) is calculated from the deformation δ of material, to replace the plane bearings in the automatic screwdriver freely, so as to increase the torque accuracy.

[0046] Referring to FIGS. 9 and 14, in addition, in order to avoid the screwdriver head trips off the clutch mechanism forming break-off effect which may cause torque error, the screw 3 locks the fixed support base formed of two bearings by screw thread at F_p to resist the final deformation δ , this combination results in axial balance. As shown in FIG. 14, where F_p is the prelocking force,

F_a is the axial force, F_r is the radial force, compared with the automatic screwdriver structure in FIG. 9, the dynamic bearing takes effect only if the load on G and H layers exceeds the F (object stress). When the load F (object stress) is removed, the generated tension has not effect, but there is sliding in radial direction, increasing the torque accuracy again.

[0047] Referring to FIGS. 10, 15 and 16, the structure of prelocking force is applied to elbowed automatic screwdriver, compared with the automatic screwdriver structure in FIG. 10, on G and H layers, the screw 3 (or nut) is combined with two dynamic bearings by screw thread locking, and the two bearings keep the support base structure of concentric pivot, so as to preset the axial deformation δ . This combination results in axial and radial equilibriums. As shown in FIG. 15, where F_p is the prelocking force, F_a is the axial force, F_r is the radial force, the dynamic bearing takes effect only if the load exceeds F (object stress), the generated tension has no effect, and the concentricity is better, the radial load has been absorbed by the dynamic bearing. The δ is resulted from axial and radial variations, this combination results in axial balance. In addition, as shown in FIG. 16, when the load exceeds F (object stress) on G and H layers, the second dynamic bearing takes effect.

Claims

1. A dynamic load accuracy adjustable automatic screwdriver wherein, more than one set of dynamic bearing are mounted in the automatic screwdriver; the dynamic bearing comprises at least a layer of flat washer in unlimited thickness, and at least more than two high and low potential difference fulcrums composed of unlimited quantity of surrounding steel balls; the dispersion of fulcrums under stress eliminates the torque error value resulted from friction force, and the optimal torque accuracy can be obtained in the full torque range.
2. The dynamic load accuracy adjustable automatic screwdriver as claimed in Claim 1, wherein the high and low potential difference fulcrums of dynamic bearing mean any fulcrum has preset deformation against another fulcrum height.
3. The dynamic load accuracy adjustable automatic screwdriver as claimed in Claim 1 or 2, wherein the high and low potential difference fulcrums of dynamic bearing can be implemented by steel balls in different diameters.
4. The dynamic load accuracy adjustable automatic screwdriver as claimed in one of the preceding claims, wherein the high and low potential difference fulcrums of dynamic bearing can be implemented by setting tracks with different radians in the contact

surface of flat washer against the steel balls.

5. The dynamic load accuracy adjustable automatic screwdriver as claimed in one of the preceding claims, wherein the high and low potential difference fulcrums can be implemented by setting tracks in different rotation radii in the contact surface of flat washer against the steel balls. 5

6. The dynamic load accuracy adjustable automatic screwdriver as claimed in one of the preceding claims, wherein the potential difference fulcrum of dynamic bearing can be implemented by setting tracks in different rotation radii in the contact surface of flat washer against the steel balls, and setting steel balls in different diameters. 10 15

7. The dynamic load accuracy adjustable automatic screwdriver as claimed in one of the preceding claims, wherein the potential difference fulcrum of dynamic bearing can be implemented by setting a support ring adjusting rotating ring smaller than the steel ball diameter in the inner edge of surrounding steel balls. 20 25

8. The dynamic load accuracy adjustable automatic screwdriver as claimed in one of the preceding claims, wherein the potential difference fulcrum of dynamic bearing can be implemented by setting a support base adjusting rotating ring in the outer edge of surrounding steel balls. 30

9. The dynamic load accuracy adjustable automatic screwdriver as claimed in one of claims 2 to 8, wherein the preset deformation includes the radial deformation and axial deformation of dynamic bearing. 35

10. The dynamic load accuracy adjustable automatic screwdriver as claimed in Claim 9, wherein the axial deformation includes a preventive prelocking structure, the shaft through the dynamic bearing is combined with two dynamic bearings by screw or nut locking. 40 45 50 55

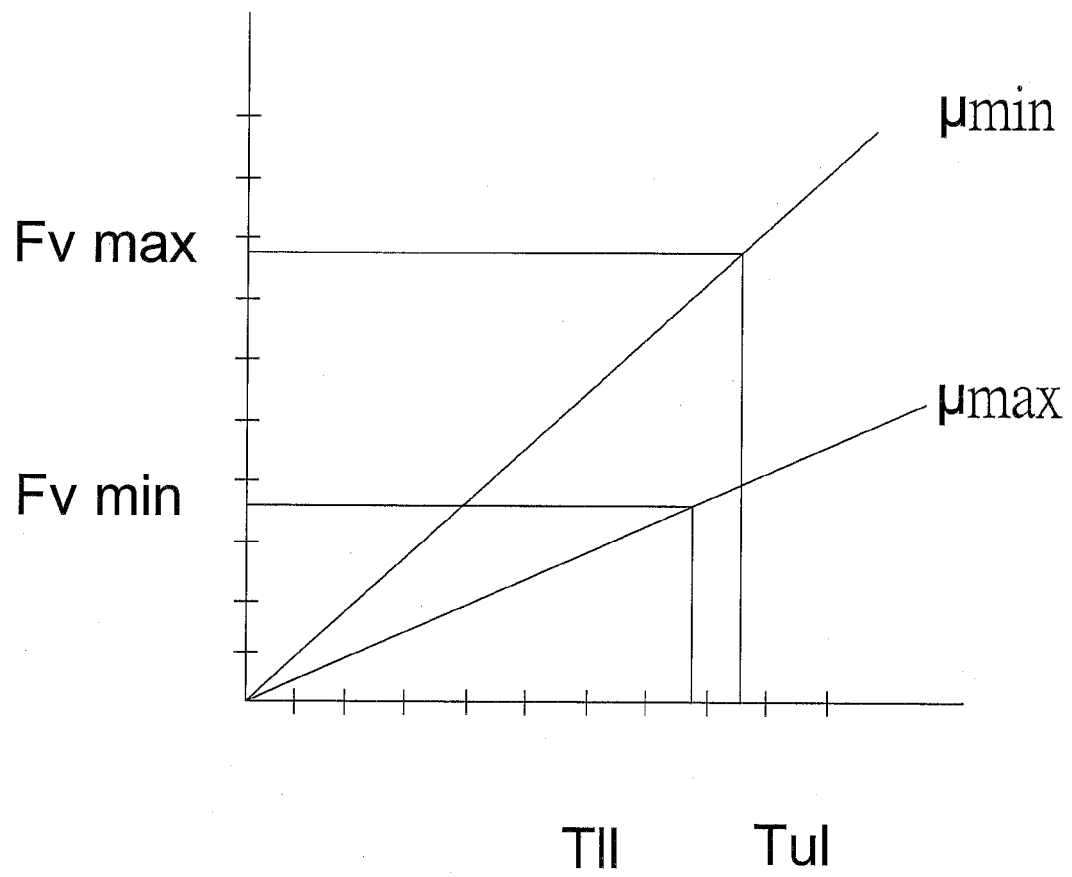
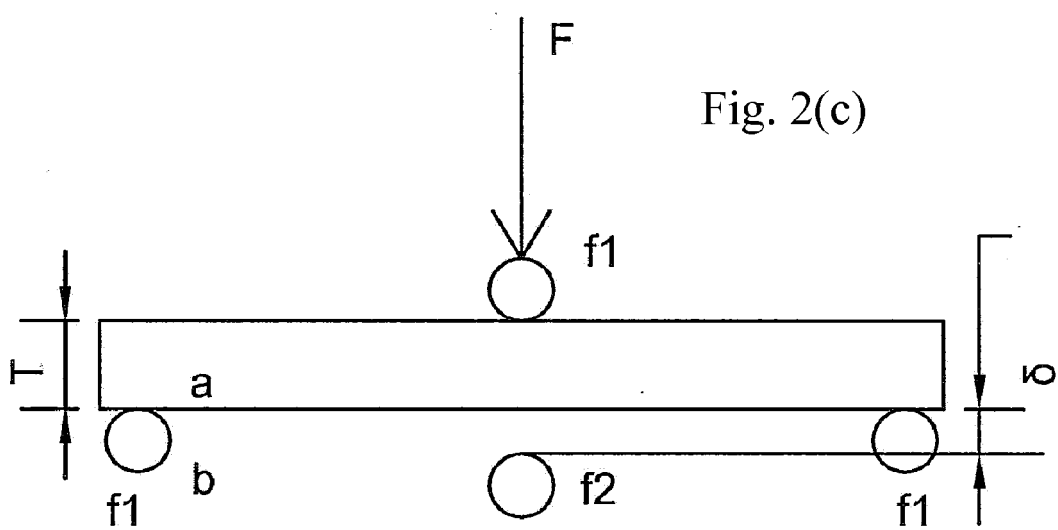
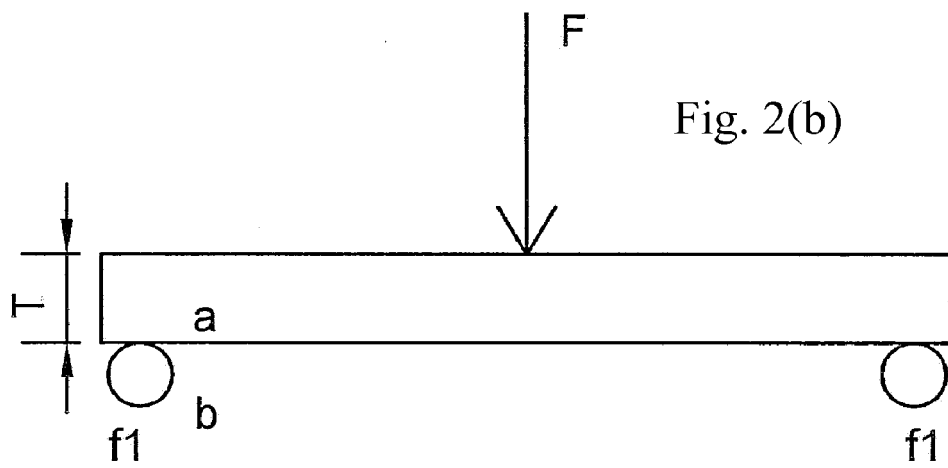
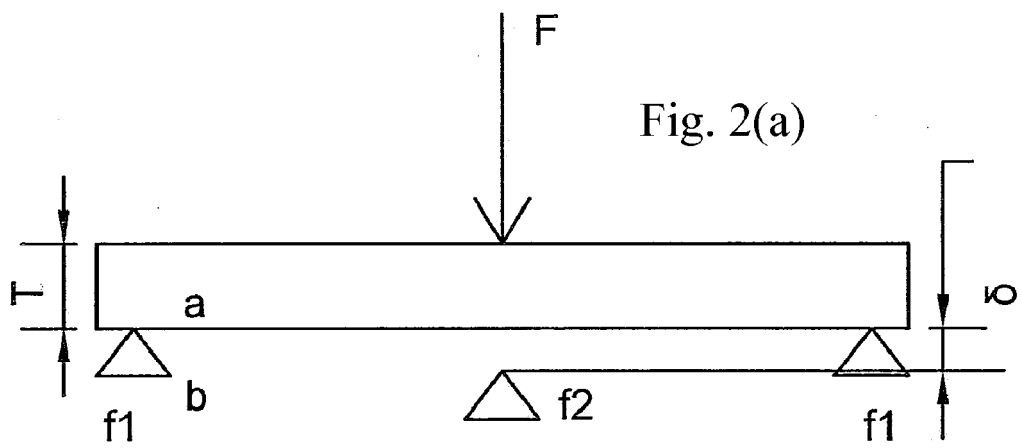
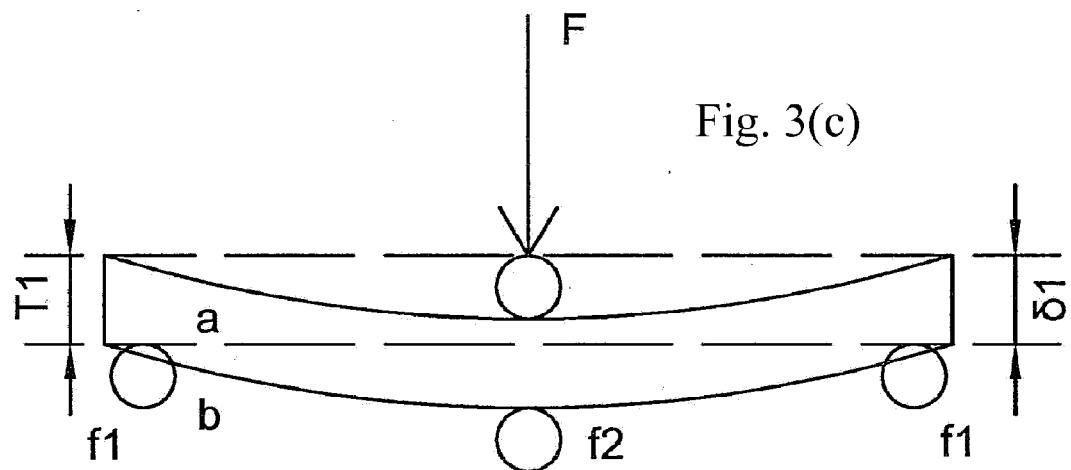
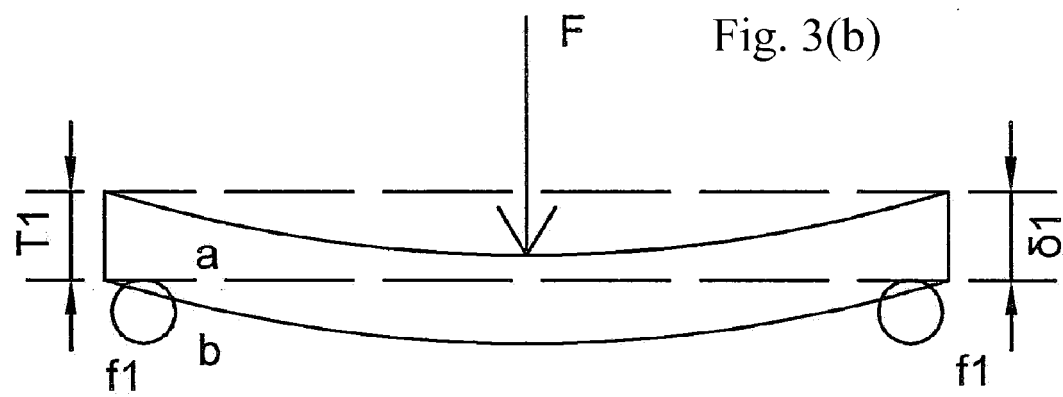
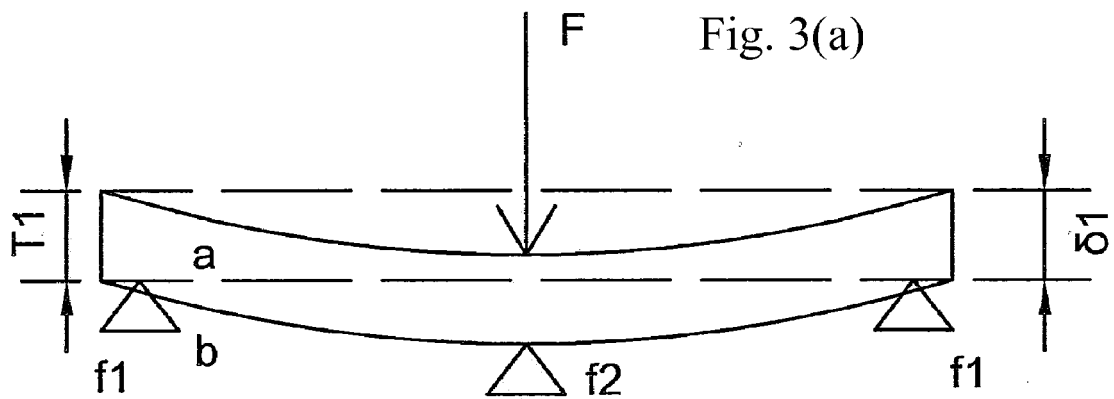
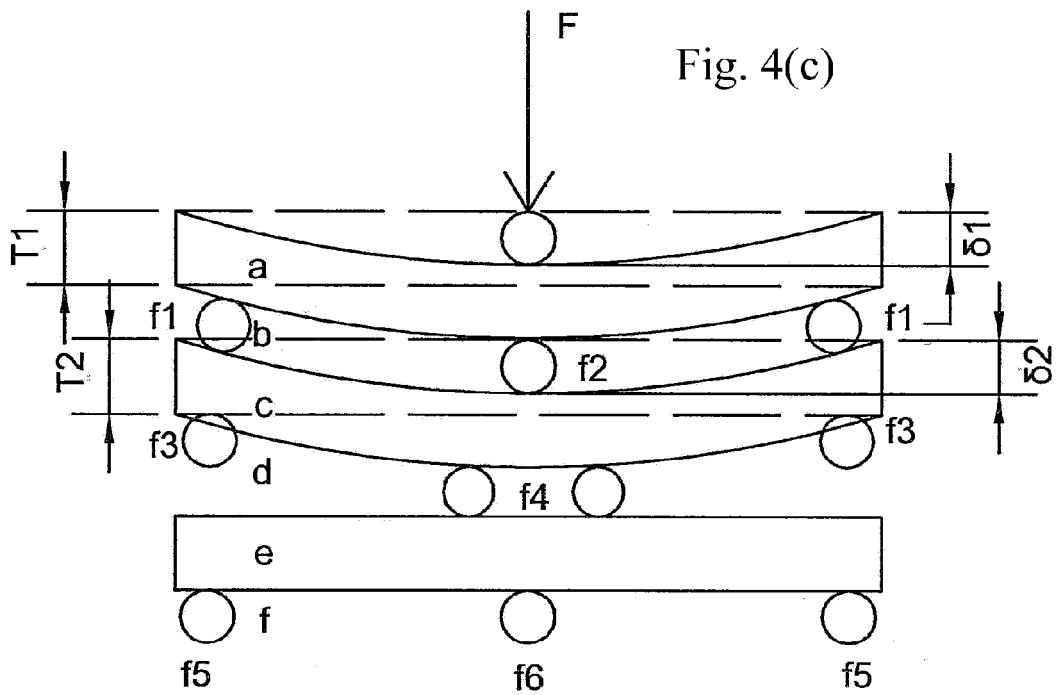
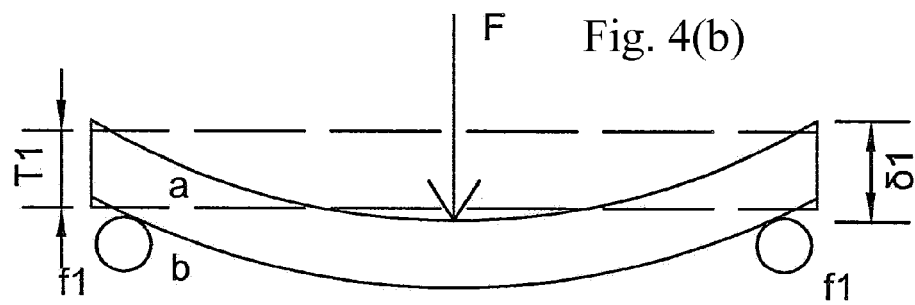
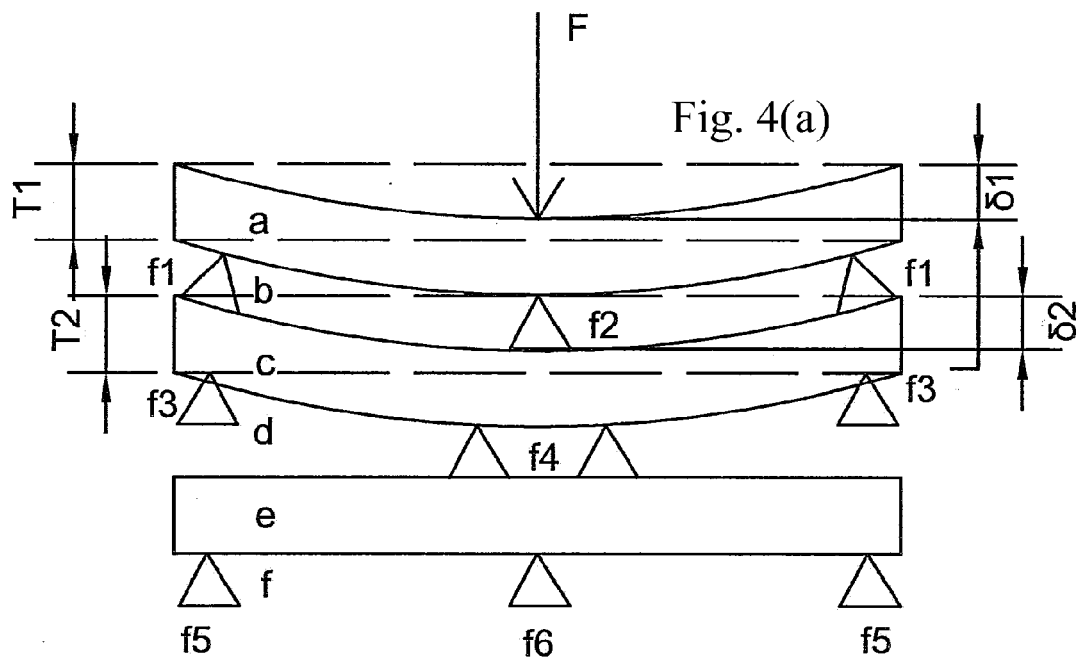


Fig. 1







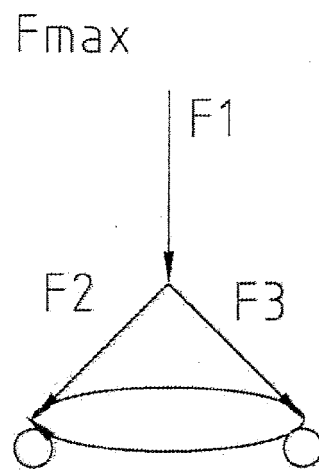


Fig. 5

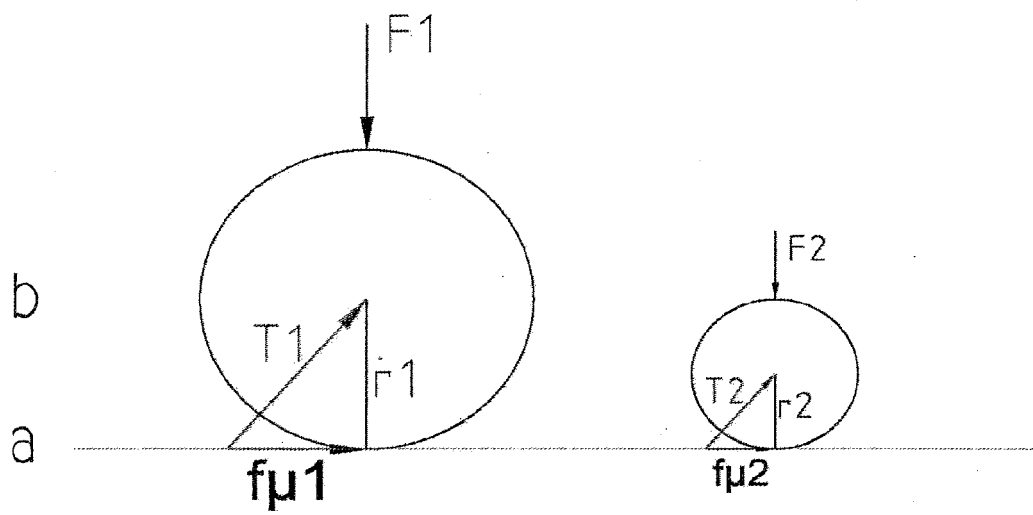


Fig. 6

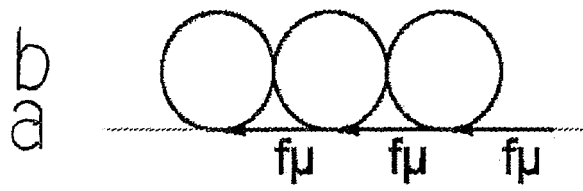


Fig. 7(a)

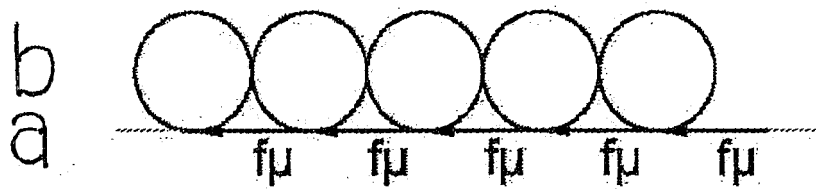


Fig. 7(b)

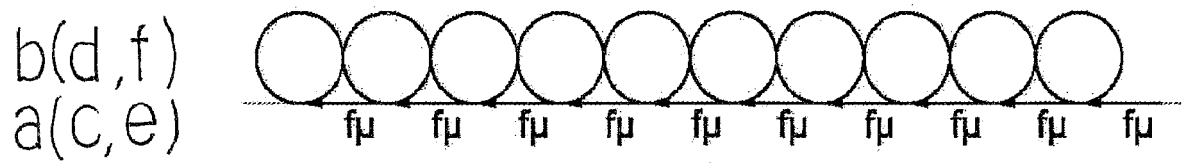


Fig. 7(c)

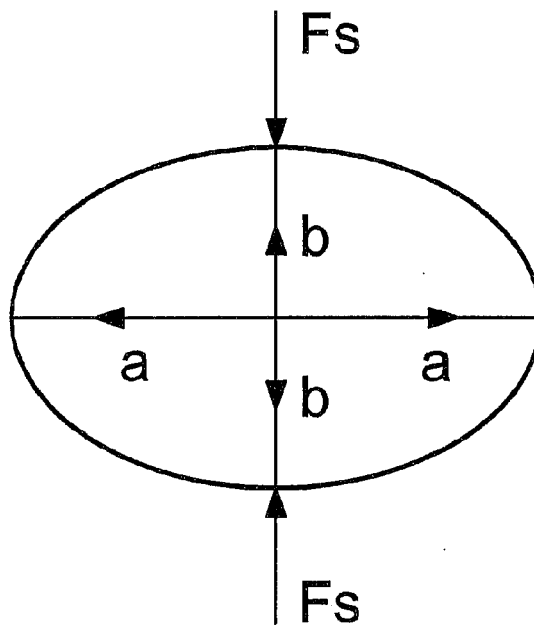


Fig. 8(a)

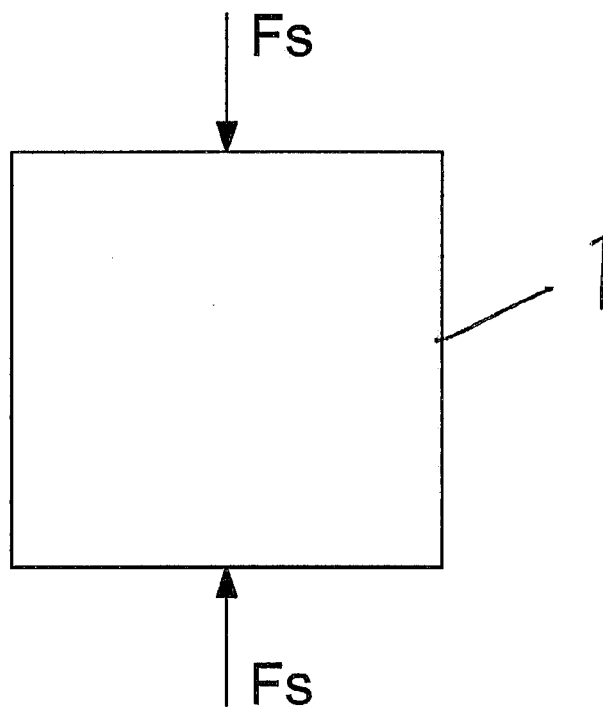


Fig. 8(b)

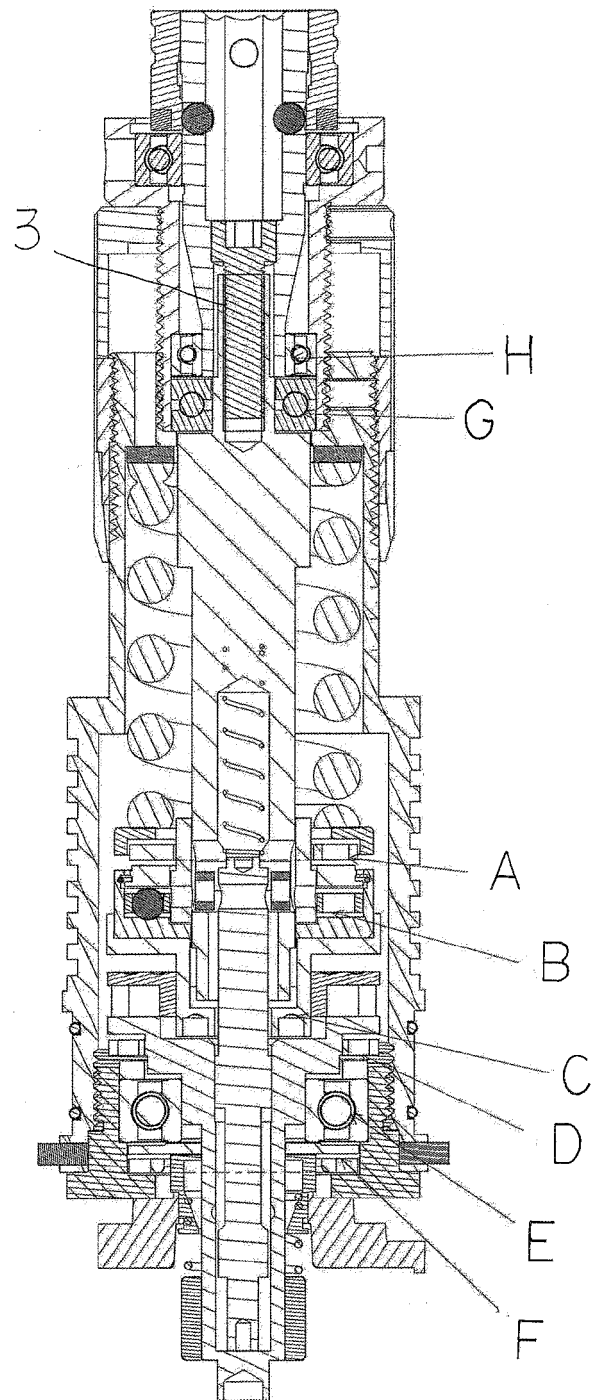


Fig. 9

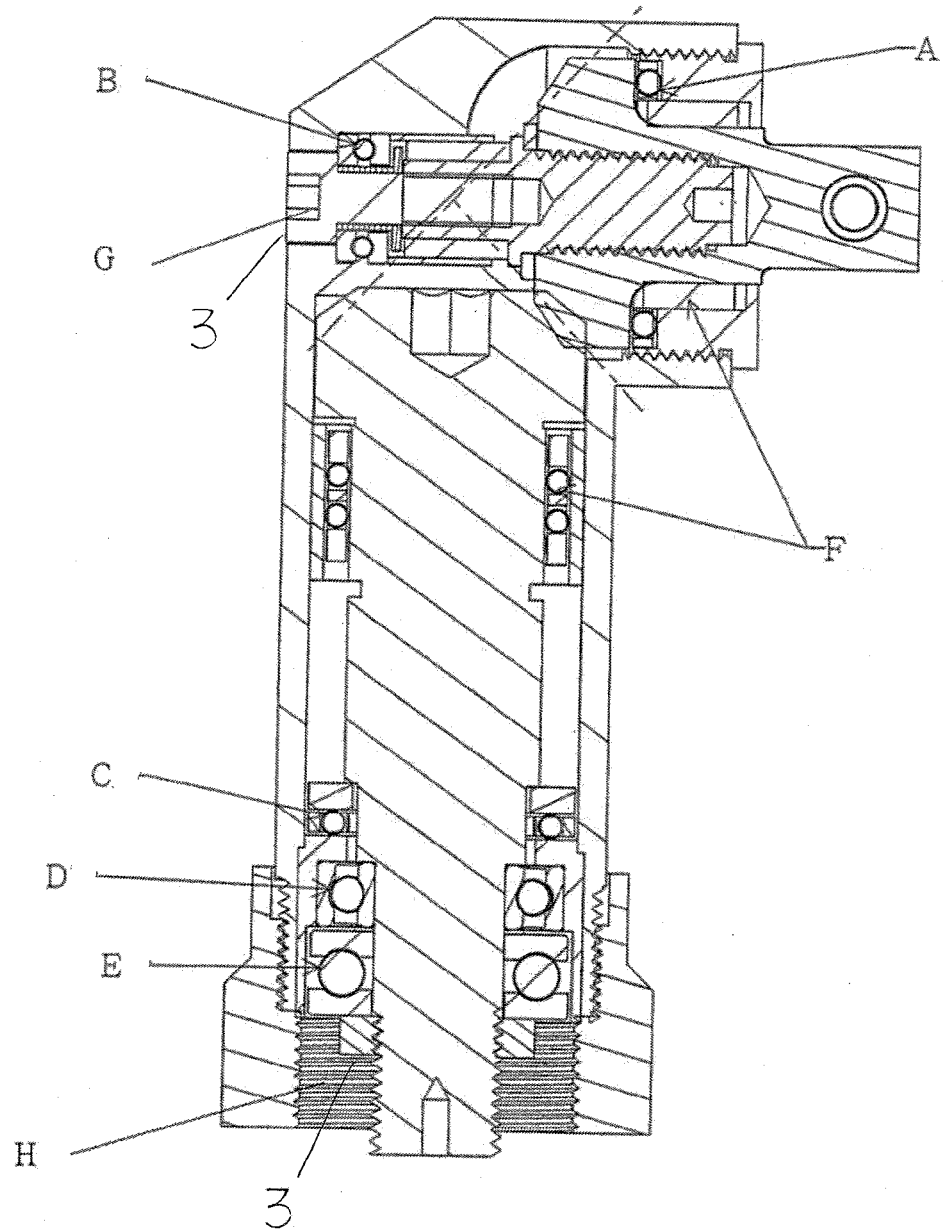


Fig. 10

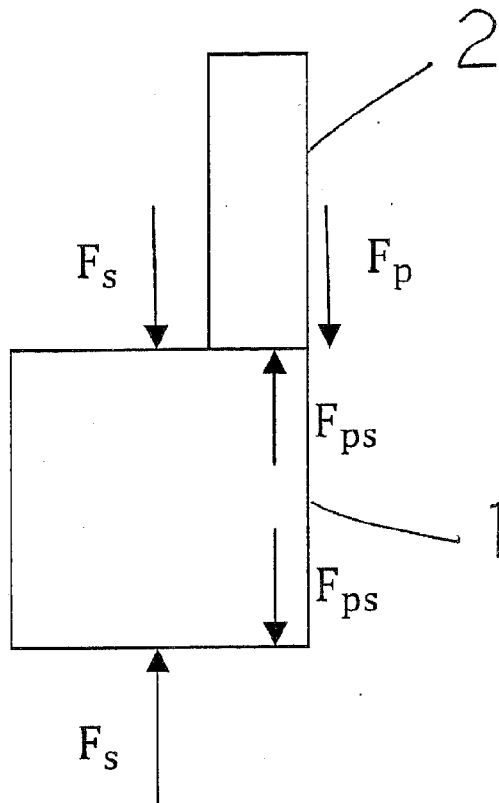


Fig. 11

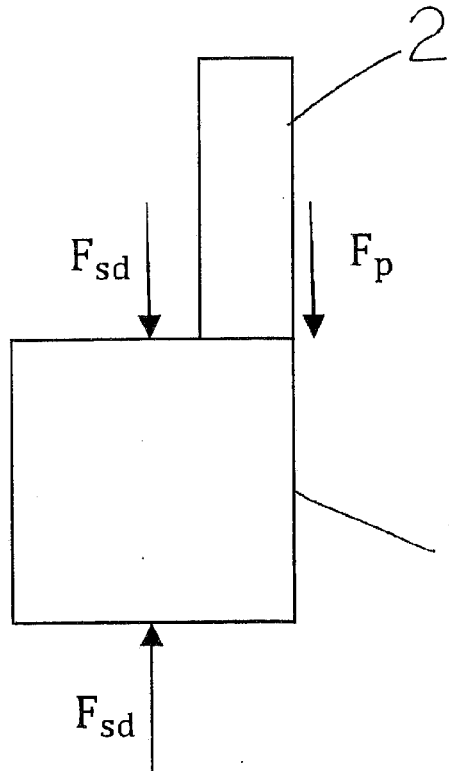


Fig. 12

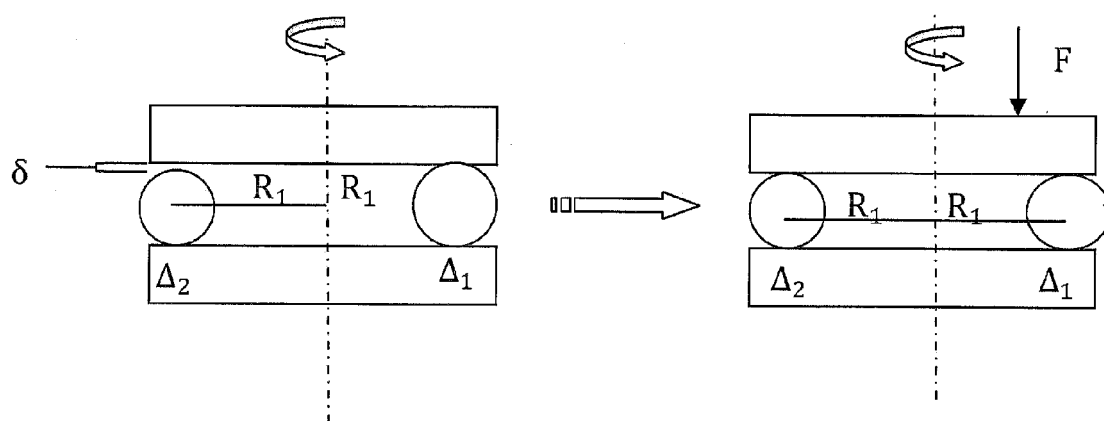


Fig. 13(a)

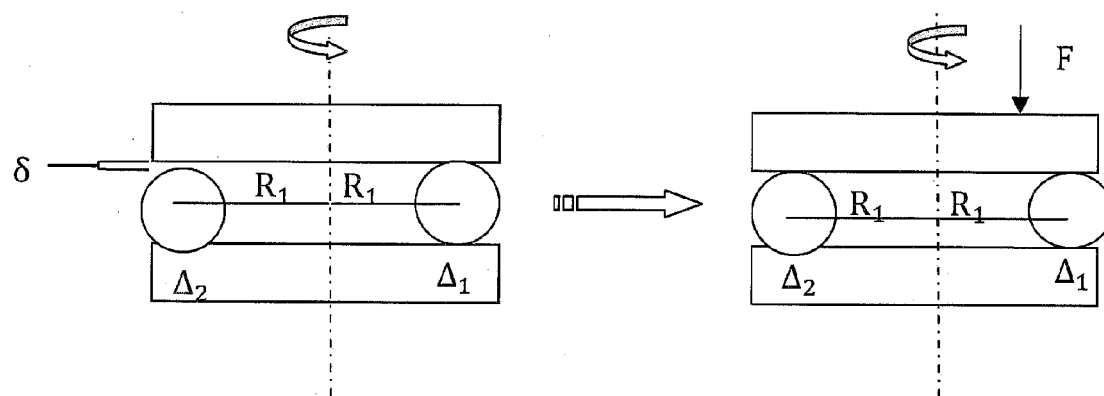


Fig. 13(b)

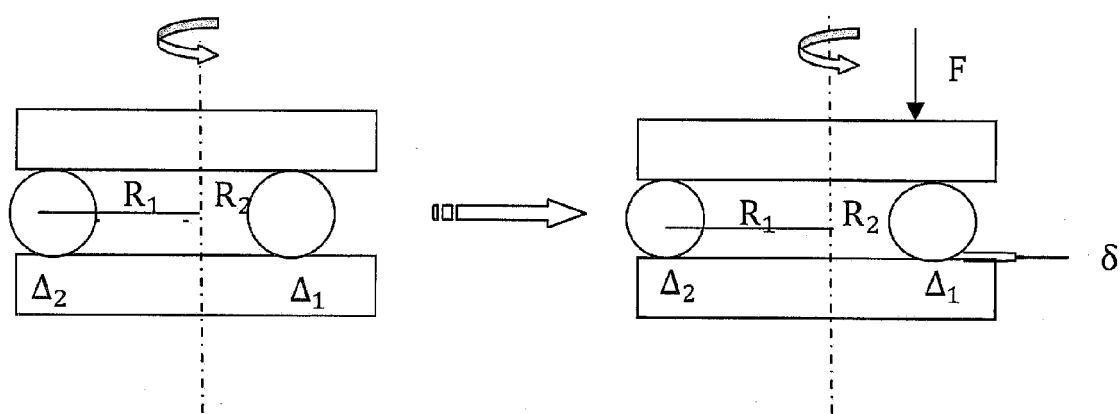


Fig. 13(c)

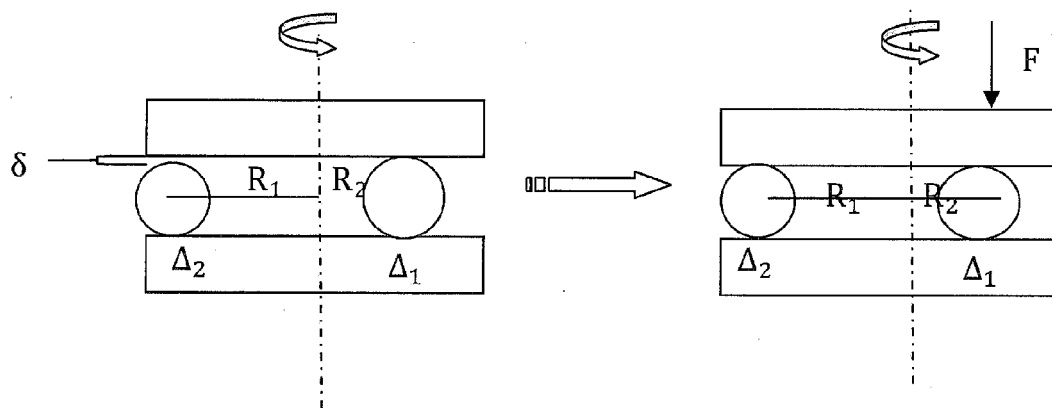


Fig. 13(d)

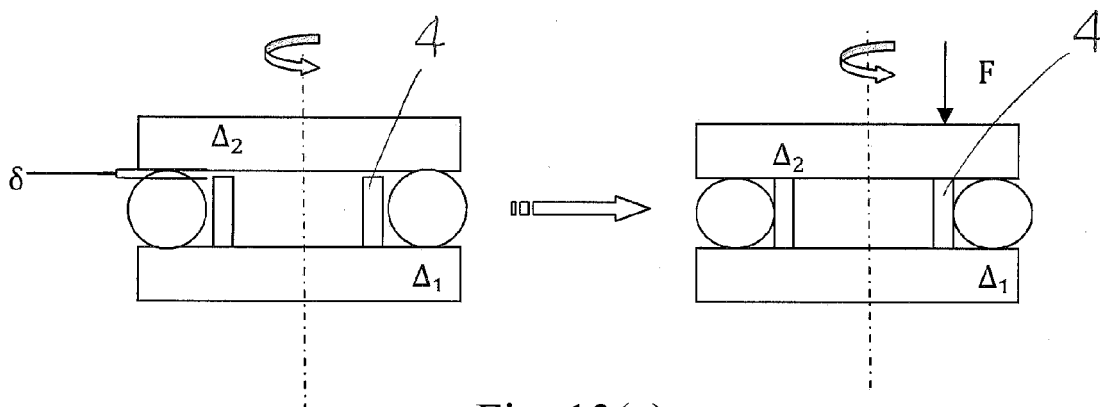


Fig. 13(e)

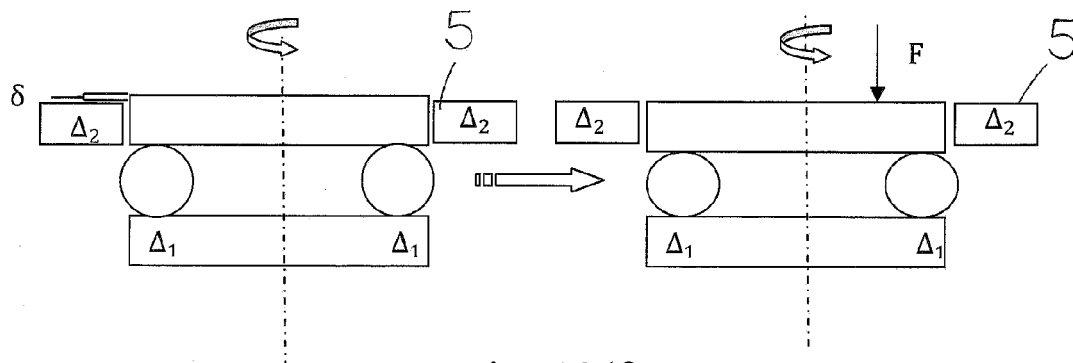


Fig. 13(f)

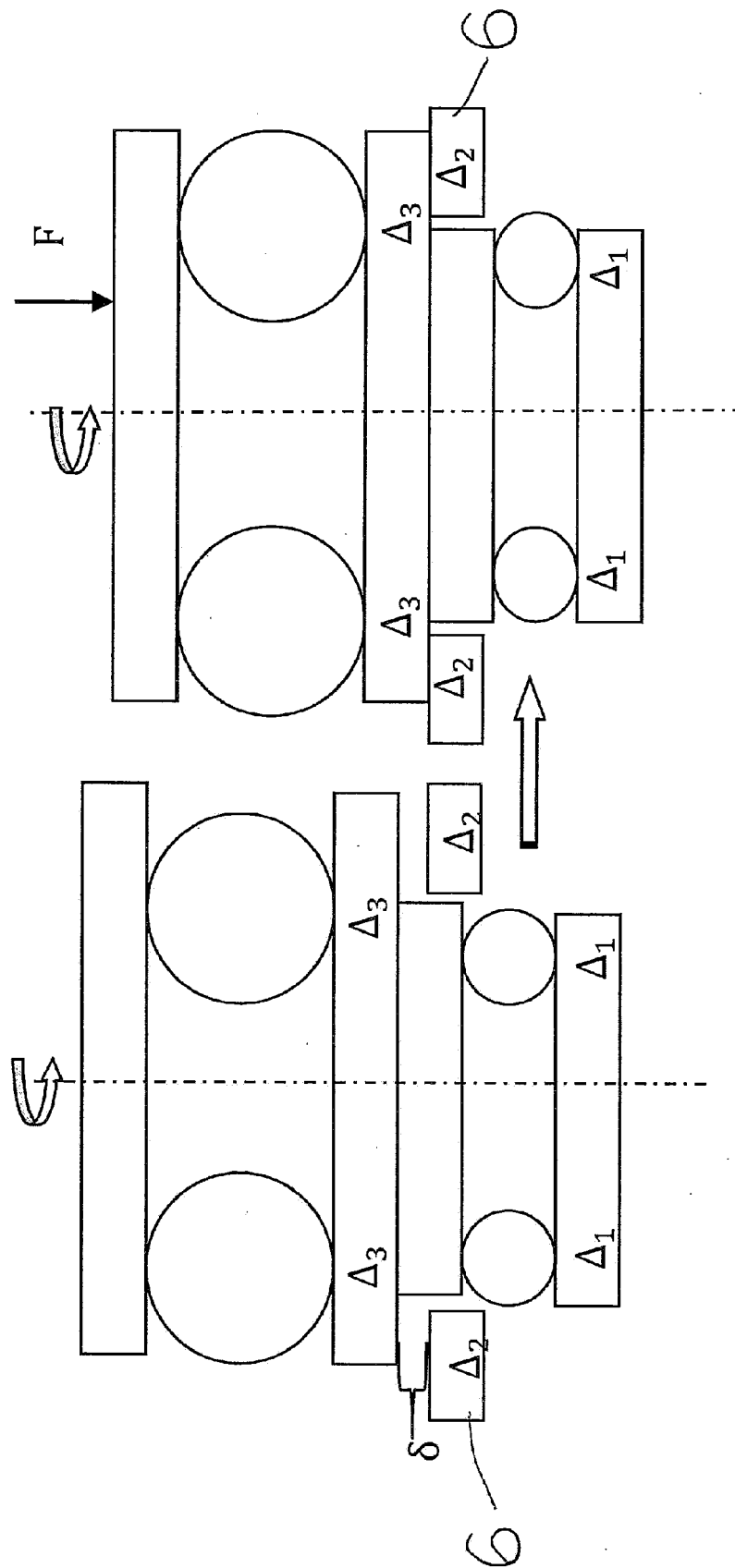


Fig. 13(g)

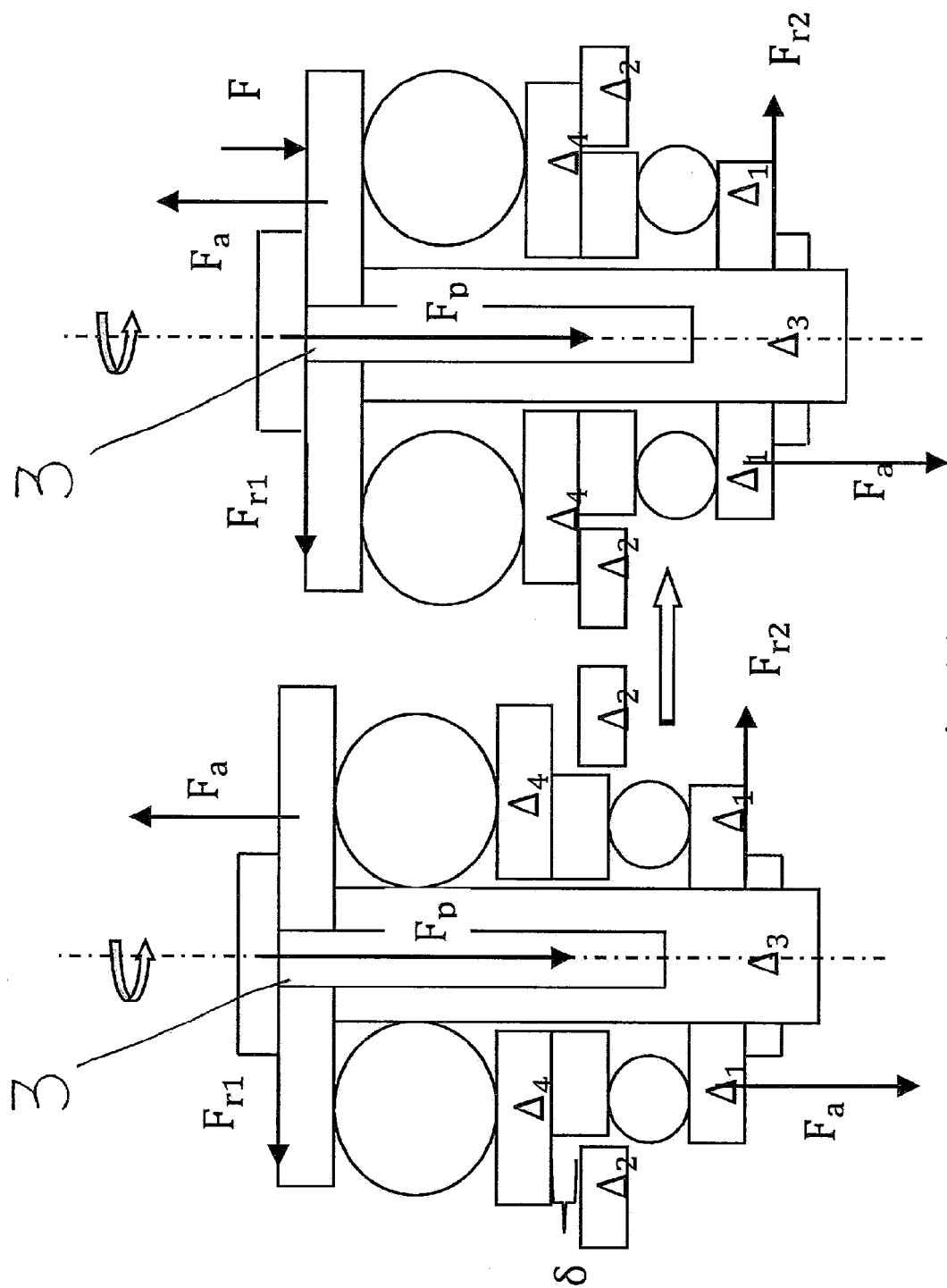


Fig. 14

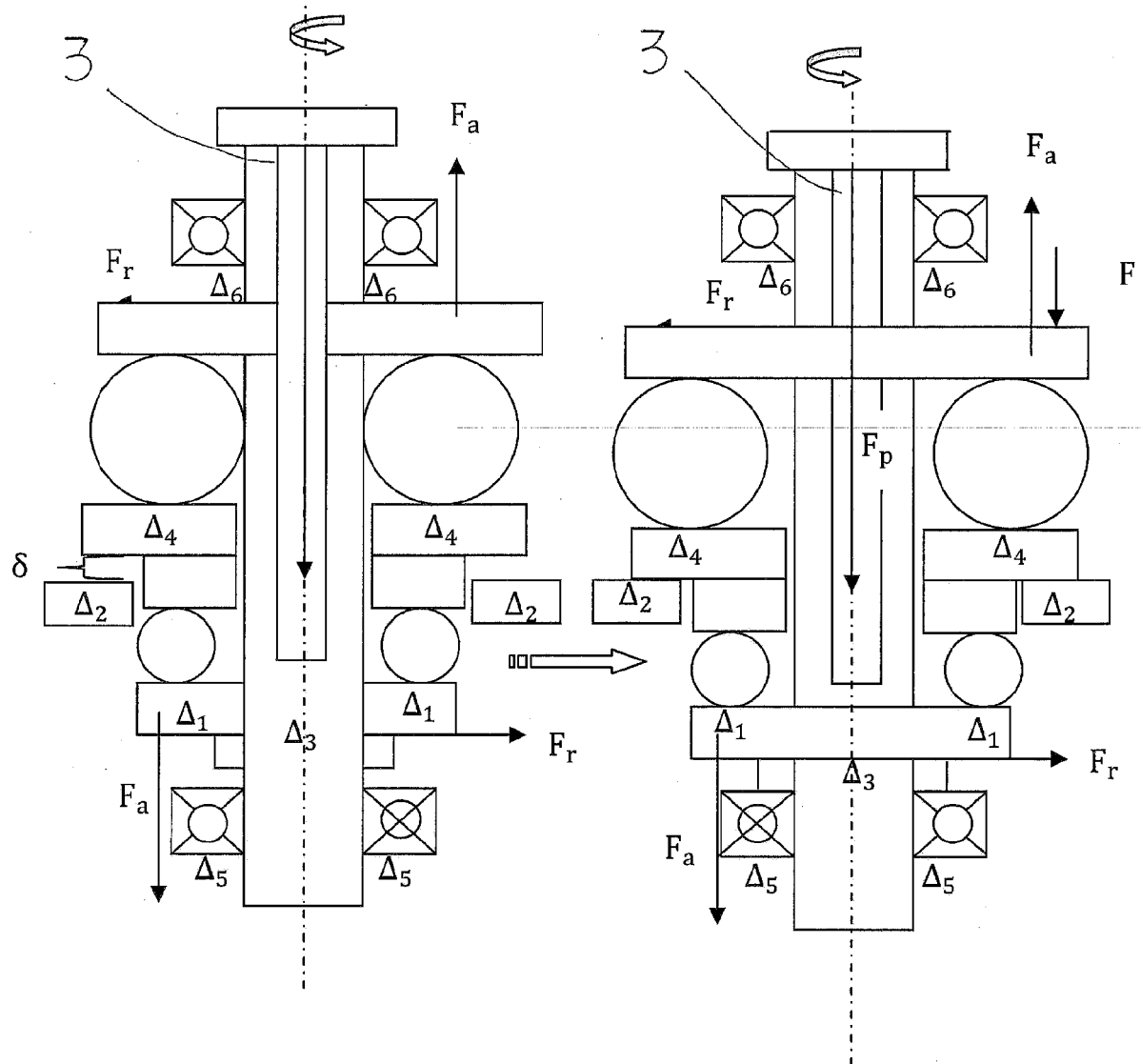


Fig. 15

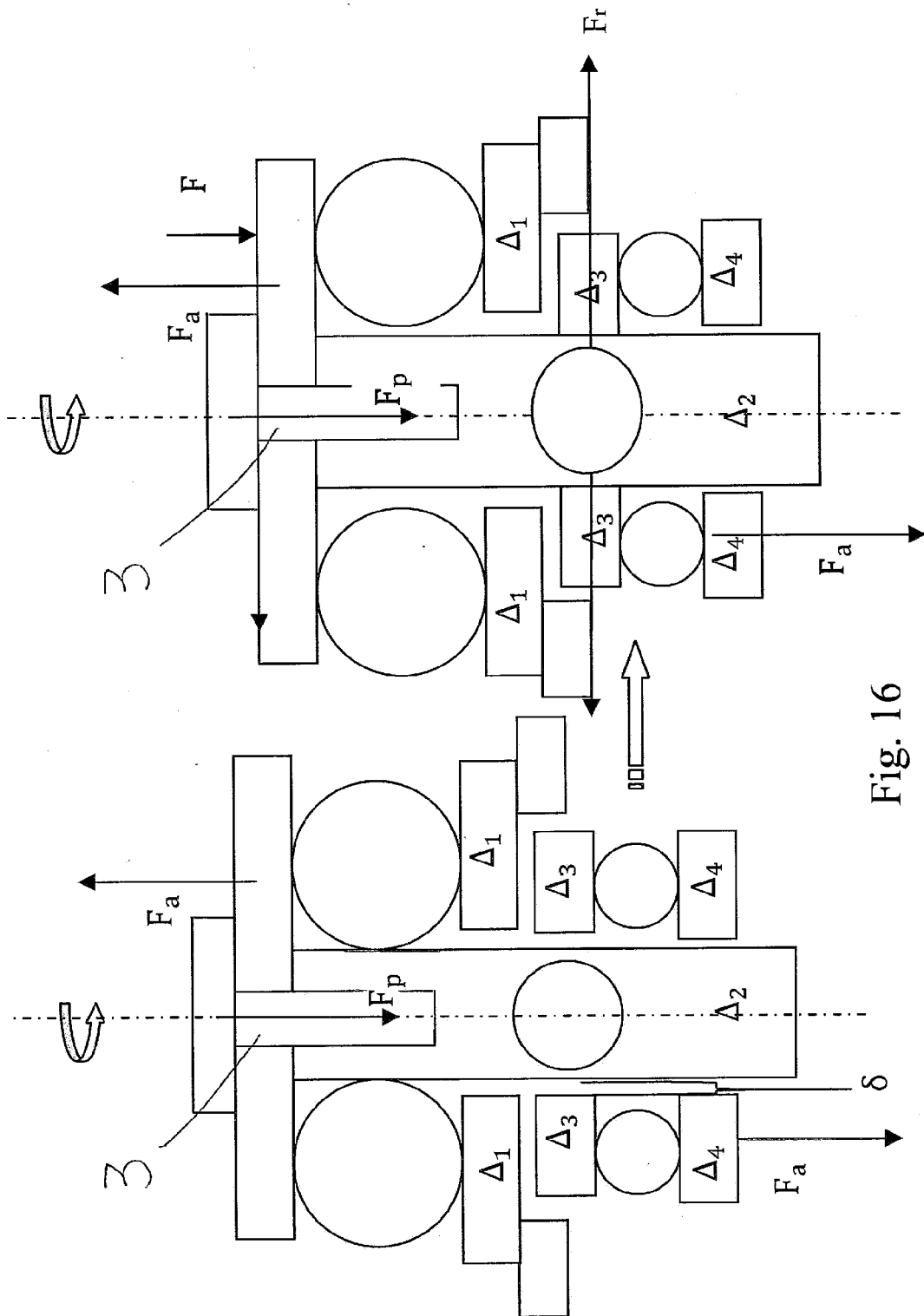


Fig. 16



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Application Number
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Place of search The Hague		Date of completion of the search 19 January 2016	Examiner Hartnack, Kai
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