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

(11) Publication number: 0 234 133 A1

(12) EUROPEAN PATENT APPLICATION

(21) Application number: 86400385.0

(51) Int. Cl.⁴: F04B 1/10

(22) Date of filing: 24.02.86

(43) Date of publication of application: 02.09.87 Bulletin 87/36

(84) Designated Contracting States: DE FR GB IT

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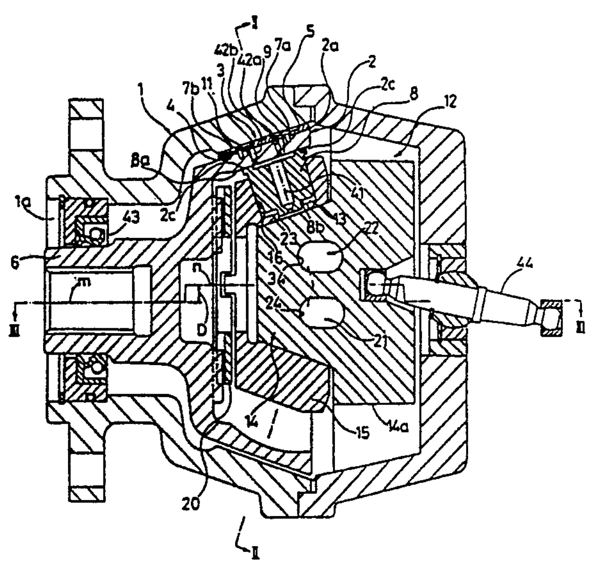
(54) Rotary fluid energy converter.

(57) A rotary fluid energy converter which is used either as a hydraulic pump or as a fluid motor has a housing (1), a torque ring (2), pistons (8), a cylinder barrel (15), and chambers (13). The ring (2) is closely held against the inner surface of the housing (1) via first static pressure bearings (3) which are circumferentially regularly spaced from one another. As the ring (2) is rotated relative to the housing (1), the volumes of the chambers (13) increase and decrease cyclically. Each of the first bearings (3) has two pressure pockets (7a, 7b ; 7a, 7b, 7c) axially neighboring each other. Fluid flows out of the chambers (13) and is distributed to the corresponding pressure pockets (7a, 7b, 7c) through restrictors - (40a, 40b) or through slide valve elements (50) for automatically selectively pressure fluid into said pressure pockets.

fig.1



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"ROTARY FLUID ENERGY CONVERTER"

The present invention relates to a rotary fluid energy converter that is used either as a hydraulic pump or as a hydraulic motor of the hydrostatic type.

A conventional rotary energy converter of this kind is employed as a hydrostatic hydraulic pump or motor and always uses mechanisms, such as cam mechanisms and linkages, for converting rotary power applied to its input shaft into rectilinear motion of a piston, plunger, or the like and for converting such rectilinear motion of the piston into rotary motion of its output shaft. Since its components are pressed against each other or a twisting force is applied to some components, the converter must employ bearings that make use of either the wedging action of oil films utilizing oiliness or viscosity of lubricating oil or the rolling action of balls, rollers, or the like. Therefore, an oil having an appropriate viscosity is required to be used as working fluid. If water or other fluid with a viscosity comparable to water is used as working fluid, it will be difficult to operate the convertor smoothly. This makes the operation life of the machine quite short. Thus, working fluids that can be used are limited to some kinds. If roller bearings are used, the operation life of the whole machine depends on the operation life of these bearings, making it difficult to enhance the durability. Moreover, roller bearings are relatively bulky. This renders it difficult to make the machine in smaller size or lightweight.

In recent years, as almost ideally efficient fluid energy converter operating on quite different principles from the prior art techniques described above was developed (see Japanese laid open patent specification n° 77179/1983). Specifically, this known converter comprises a housing having a tapering surface in its inner surface, a torque ring which is closely held against the tapering surface of the housing via first static pressure bearings disposed circumferentially regularly spaced from each other, said ring having flat surfaces corresponding to the first bearings on its inner surface, a plurality of pistons disposed on the inner side of the ring and having their front ends connected to the flat surfaces of the ring via second static pressure bearings, a cylinder barrel for slidably supporting the bottom ends of the pistons, a pintle being slidable perpendicularly to the axis of the housing and supporting the barrel, chambers formed between each piston and the barrel whose volume alternately increases and decreases during rotation of the housing relatively to the ring, a pair of fluid communication passages for communicating with the chambers whose volumes are increasing and decreasing, and fluid passages for introducing fluid from the chambers into the first and second bearings. Consequently, the static pressures of the fluid introduced into the first and second bearings develop a couple of forces about the axis of rotation on the torque ring.

In each first static pressure bearing having a single pressure pocket, the center of pressure of each bearing is maintained at a certain position and so the ring is submitted to another couple of forces being produced in a position being off the axis of rotation. This structure is now described by referring to Figs. 11 and 12, where the static pressure in each first static pressure bearing a produces a force F_a that acts along a line of action Ls. This line L_a passes across the center of pressure - (geometrical center) b of the bearing a, and every line of action L_a focus toward a point d on the axis m of both the housing c and the torque ring k. The static pressure in each second static pressure bearing e produces a force F_b that acts along a line of action L_b. Every such line of action L_b, (median lines g of the pistons f), focus toward a point i on the axis n of the pintle h. Therefore,knowing that the inner surface j of the housing c tapers, the torque ring k rotating relatively to the housing c and the axis n of the pintle h being displaced from the axis m of the housing c, the center g of the pintle f periodically moves away from the pressure center (geometrical center) b of the bearing a,while following an elliptic orbit p as shown in Fig. 12. In this case, the movement of the center g relative to the pressure center b along the axis X is needed to produce a couple of forces about the axis of rotation on the ring k. However, displacement along axis Y bends or twists the ring k. This can impair the features of this system, i.e., excellent durability and the ability to run smoothly and efficiently.

It is an object of the present invention to provide an energy converter equipped with a simple structure which effectively prevents a couple from occurring in a position being off the axis of rotation, which would otherwise be produced by the axial deviation of the pressure center of each first static pressure bearing from the center of each piston.

This is achieved by a converter characterized in that it further comprises : at least two axially adjacent pressure pockets formed in each of the first static means for automatically selectively distributing pressure fluid into said pressure pockets so that when the median axis of the piston is displaced relatively to the median axis of the corresponding pressure bearing the pressure pockets being remoter from said median axis of the piston have a pressure at least inferior as compared to the pressure pockets being closer to said axis.

In one embodiment of the invention, said means for automatically selectively distributing pressure fluid into the pressure pockets are constituted of restrictors through which the fluid flowing out of the chamber is distributed to the corresponding pressure pockets.

In another embodiment of the invention, said means for automatically selectively distributing pressure fluid into the pressure pockets are constituted of slide valve elements for selectively intermitting the supply of fluid into said remoter pressure pockets by making use of the axial relative movement between each piston and the torque ring.

In the structure constructed according to the abovementioned first embodiment, when the center of each piston axially moves away from the geometrical center of each first bearing, each fluid leakage gap of the bearings that is remoter from each piston becomes slightly larger than each fluid leakage gap of the bearings that is closer to each piston by the elastic deformation of the torque ring due to hydraulic pressure, so that the pressure inside each pressure pocket remoter from each piston becomes lower than the pressure inside each pressure pocket closer to each piston. As a result, the center of pressure of each first bearing comes closer to each corresponding piston than the geometrical center. Thus, the axial distance separating the center of pressure from the center of each piston is automatically reduced.

In the structure constructed according to the abovementioned second embodiment, when the center of each piston axially moves away from the geometrical center of each first bearing, the switching action of each slide valve element cuts off the supply of pressure fluid into some pressure pockets, so that the center of pressure of each first bearing comes closer to the piston than the geometrical center. As a result, the axial gap between the center of pressure and the center of each piston is automatically reduced.

Other objects and features of the invention will appear in the course of description that follows and in the corresponding drawings in which :

Fig. 1 is a front cross section view in elevation of an energy converter according to the invention.

Fig. 2 is a cross-sectional view taken along line II-II of Fig. 1.

Fig. 3 is a cross-sectional view taken along line III-III of Fig. 1.

Fig. 4 is a cross-sectional view taken along line IV-IV of Fig. 3.

Fig. 5 is an enlarged plan view of the pressure pockets in one of said first static pressure bearings of the converter shown in fig. 1.

Fig. 6 is a cross-sectional view taken along line VI-VI of Fig. 5.

Fig. 7 is a cross-sectional view taken along line VII-VII of Fig. 5.

Figs. 8-10 are fragmentary views for illustrating the operation of the converter shown in Fig. 1.

Fig. 11 is a partially cross-sectional view of a convention converter.

Fig. 12 is a fragmentary plan view of one of said first static pressure bearings of the converter shown in Fig. 11.

Fig. 13 is a front cross section view in elevation of another embodiment of the energy converter according to the invention.

Fig. 14 is a cross-sectional view taken along line II-II of Fig. 13.

Fig. 15 is a cross-sectional view taken along line III-III of Fig. 13.

Fig. 16 is a cross-sectional view taken along line IV-IV of Fig. 15.

Fig. 17 is an enlarged plan view of the pressure pocket of one of the static pressure bearings of the converter shown in Fig. 13.

Fig. 18 is a cross-sectional view taken along line VI-VI of Fig. 17.

Fig. 19 is a view in perspective of the pressure pockets shown in Fig. 13.

Figs. 20-22 are fragmentary views for illustrating the operation of the converter shown in Fig. 13.

Fig. 23 is a partially cross-sectional view of a conventional converter.

Fig. 24 is a fragmentary plan view of one first static pressure bearing shown in fig. 23 ; and

Fig. 25 is a diagram for showing a further energy converter according to the invention.

Referring to Figs. 1-10, there is shown an energy converter according to the invention. This converter comprises a cylindrical housing 1 with a bottom portion, and a torque ring 2 is rotatably and closely held on the inner surface of the housing 1 via first static pressure bearings 3. The housing 1 is provided with an opening la at one of its ends. The inner surface of the housing has a surface 4 tapering toward the opening la, and the ring 2 is in contact with this tapering surface 4. The ring 2 is shaped like a cup and has a peripheral wall 2a presenting the same apical angle as the tapering surface 4. A rotating shaft 6 is formed integrally with the ring 2 and protrudes from one of its axial ends. The front end portion of the shaft 6 extends outwardly from the housing 1 through the opening la. The first bearings 3 rigidly comprise shoes 5 being rigidly fixed to the outer surface of the ring 2 at required positions, each shoe 5 being pressed on the tapering surface 4 of the housing 1. Each shoe 5 is provided with a pair of pressure pockets, 7a and 7b, axially neighboring one another. Hydraulic pressure is introduced into the pockets 7a and 7b. An odd number of bearings 3 are circumferentially regularly spaced apart from one another. The pockets 7a and 7b of each shoe 5 is surrounded by surrounding portions 5a, 5b, 5c which are shaped so that their cross section protrudes toward the tapering surface 4 as shown in Figs. 5-7. Each shoe 5 is in sliding contact with the tapering surface 4 by a small area. In addition, the surrounding portions 5a-5c are so shaped that they are not parallel to the axis of rotation X.

More specifically, only the surrounding portion 5a perpendicular to the axis of rotation X is straight in shape. The surrounding portion 5b is shaped like the letter "V". The surrounding portion 5c is shaped in a zigzag manner. The inner surface of the torque ring 2 has flat surfaces 2c at positions corresponding to the bearings 3.

The pistons 8 are disposed at positions corresponding to the inner flat surfaces 2c. The front ends 8a of the pistons 8 are pressed against their corresponding surfaces 2c via second static pressure bearings 9. The bearings 9 are made flat so that the front ends 8a of the pistons 8 come into close contact with their corresponding surfaces 2c. Each of the front end 8a has a pressure pocket 11 into which hydraulic pressure is introduced. The base end of each piston 8 is held by a piston retainer 12. A chamber 13 is formed between the retainer 12 and the piston 8 for the admission of fluid.

The piston retainer 12 consists of a pintle 14 having a sliding portion 14a cooperating with an annular cylinder barrel 15. The sliding portion 14a is supported on the housing 1. The pintle 14 can rotate about an axis n being parallel to the axis m of both the housing 1 and the torque ring 2. The barrel 15 is rotatably fitted over the outer periphery of the pintle 14. The barrel 15 is provided with cylinders 16 which are regularly circumferentially spaced apart from one another and are arranged radially. The axis of each cylinder 16 is substantially perpendicular to the outer surface of the pintle 14. The pistons 8 are fitted in the cylinders 16 so as to be slidable. The base surface 8b of each piston 8 and the inner surface of each cylinder 16 form the aforementioned chamber 13. The barrel 15 is connected to the torque ring 2 via an Oldham coupling 20 or similar, so that the barrel can rotate at the same angular velocity as the ring 2.

The pintle 14 takes the form of a truncated cone whose outer surface makes an apical angle substantially equal to the apical angle formed by the peripheral wall 2a of the ring 2. The pistons 8 are so held that they can move back and forth perpendicularly to the peripheral wall 2a of the ring 2. The sliding portion 14a of the pintle 14 is shaped as a longitudinally elongated block with a trapezoidal cross section. The sliding portion 14a is slidably fitted in a trapezoidal groove 19 formed in the housing 1. In other words, the pintle 14 is held in such a way that it can slide perpendicularly to the axis m. This makes it possible to set the distance D between the axis n of the pintle 14 and the axis m to any desired value, including zero.

As shown in Fig. 2, the inside of the housing 1 is divided into a first region A and a second region B by an imaginary line P that is drawn in the direction in which the pintle 14 slides. Those chambers 13 which are travelling through the first region A are placed in communication with a first fluid communication line 21. Those chambers 13 which are moving across the second region B communicate with a second fluid communication line 22.

The first fluid communication line 21 has fluid passages 23, a port 24 extending through the pintle 14, and a fluid inlet/outlet port 25 formed in the housing 1, corresponding to one end of port 24. The chambers 13 are in communication with the inside of the barrel 15 via the passages 23. One end of the port 24 extends to the outer periphery of the pintle 14 on the side of the first region A, while the other end extends to the inclined surface 14b of the sliding portion 14a of the pintle 14 that is on the side of the second region B. A pressure pocket 27 is formed between the outer periphery of the pintle 14 and the inner surface of the cylinder barrel 15, at one end of port 24, in order to form a third static pressure bearing 26. Another pressure pocket 29 is formed between the inclined surface 14b of the pintle 14 and the inner surface of the housing 1, at the other end of the port 24, to form a fourth static pressure bearing 28. The pocket 27 is elongated circumferentially, and acts to place all the chambers 13 present in the first region A in communication with the port 24 extending through the pintle. The pocket 29 is elongated in the direction in which pintle 14 slides. When pintle 14 is caused to slide, the pocket 29 prevents the port 24 from being disconnected from the fluid inlet/outlet port 25.

The second fluid communication line 22 has the fluid passages 23 already mentioned, a port 34 extending through the pintle, and a fluid inlet/outlet port 35 formed in the housing 1 at a position corresponding to one end of the port 34. The other end of port 34 extends to the outer surface of the pintle 14 on the side of the second region B, while said first end extends to the inclined surface 14c of the sliding portion 14a of the pintle 14 on the side of the first region A. At the other end of the port 34, a pressure pocket 37 is formed between the pintle 14 and the cylinder barrel 15 to form a third static pressure bearing 36. At said one end of the port 34, a further pressure pocket 39 is formed between the inclined surface 14c of the pintle 14 and the inner surface of the housing 1 to form a fourth static pressure bearing 38. The pockets 37 and 39 are similar in structure to pockets 27 and 29.

A pressure inlet passage 41 is formed along the axis of each piston 8. The fluid pressure within each chamber 13 corresponding to each piston 8 is introduced into the pressure pocket 11 in the corresponding second static pressure bearing 9 via the pressure inlet passage 41. The hydraulic pressure within the pocket 11 is introduced into the pressure pockets 7a, 7b in the corresponding first static pressure bearing 3 via fluid passages 42a, 42b formed in the ring 2. Restrictors 40a and 40b are disposed in the passages 42a and 42b, respectively.

The directions and area of the static pressure bearings 3 and 9 are so set that the force F_a acting on the ring 2 due to the static pressure of the fluid introduced into the first bearings 3 is identical in magnitude but opposite in sense to the force F_b acting on the torque ring due to the static pressure introduced into the second bearings 9. The area of the second bearings 9 is set to such a value that the force acting on the piston 8 due to the static pressure applied to the bearing 9 is cancelled by the force working on the piston 8 due to the static pressure of the fluid within the chambers 13. Moreover, the area of the third static pressure bearings 26 and 36 is set to such a value that the force acting on the barrel 15 due to the static pressure introduced into the bearings 26 and 36 is cancelled by the force acting on the barrel 15 due to the static pressure of the fluid within the chambers 13 being present in the corresponding regions A and B. The angle at which the surfaces 14b and 14c are inclined is set to such a value that the force acting on the pintle 14 due to the static pressure of the fluid introduced to the bearings 28 and 38 is cancelled by the force acting on the pintle 14 due to the static pressure of the fluid introduced to the third bearings 26 and 36 being present in the regions A and B in opposite relation to the inclined surfaces 14b and 14c on which the bearings 28 and 38 are respectively mounted. Referenced by numeral 43 are seal members. A control lever 44 is used to slide the pintle 14. Each shoe 5 is firmly fixed to the torque ring 2 with a fixing element 45.

The operation of the illustrated converter is now described. When it is used as a hydraulic motor, the high pressure fluid is supplied into the chambers 13 being present in the first region A through the first fluid communication line 21. Then, as shown, the axis n of pintle 14 is brought to a position which is at a given distance D from the axis m of the housing 1. Thus, as shown in Fig. 4, the line of action of the force F_a acting on the ring 2 due to the static pressure of the fluid introduced in the first bearings 3 rotates in the direction of axis X relatively to the line of force F_b acting on the ring 2 due to the static pressure of the fluid introduced in the corresponding second bearings 9 within the first region A. The forces F_a and F_b are identical in magnitude but opposite in direction to each other. Since they act parallelly to each other, they constitute couples. As can be seen from Fig. 4, said coupled induced by F_a and F_b developed at spaced locations on the ring 2 rotate the ring 2 in the same direction. Therefore, since the ring 2 receives the couples F_a, F_b directly from the fluid, the ring 2 rotates in the direction indicated by the arrow S.

It is now assumed for the illustrated embodiment that the magnitude of the couples induced by F_a and F_b is equal to F and that the distances of the lines of actions are 1_1, 1_2, 1_3. Then, the moment M acting on the ring 2 is given by

$$M = F.(1_1 + 1_2 + 1_3)$$

This moment M causes the ring 2 to rotate relatively to the housing 1. In this case, as the ring 2 rotates the volume of each chamber 13 present in the first region A gradually increases, while the volume of each chamber 13 present in the second region B gradually decreases. Accordingly, the high pressure fluid successively flows into the chambers 13 which are travelling across first region A, through first line 21. After doing work, the fluid flows out of said chambers 13 moving across the second region B and is discharged from the housing 1 through second line 22.

Under this condition, if the pintle 14 is slided into its neutral position where the axis n coincides with the axis m of the housing 1, then the distances 1_1, 1_2, 1_3 of the lines of action of the forces F_a and F_b are all reduced to zero. As a result, the moment acting on ring 2 disappears, making the output zero. If the pintle 14 is moved in the direction opposite to the direction shown across its neutral position, the distances 1_1, 1_2, 1_3 of the lines of action of the couples induced by F_a and F_b assume negative values and act reversely on ring 2.

When the converter is employed as a hydraulic pump, the ring 2 is rotated by an external force, for example, in the direction indicated by the arrow R. Then, couples of forces F_a, F_b are set up on ring 2 similarly to the foregoing. The input torque applied to the ring 2 is balanced by the couples F_a, F_b. The, fluid from outside the housing 1 is forced successively into the chambers 13 travelling across the second region B, through the second fluid communication line 22. The pressure fluid enters the chambers 13 moving across the first region A, and is then discharged from the housing 1 through the first line 21. In this case, if the pintle 14 slides to its neutral position, the amount of fluid discharged is made zero. This allows the ring 2 to run idle under an hydrostatically balanced condition. If the pintle 14 is moved in the direction opposite to the direction shown across the neutral position, then couples induced by F_a and F_b balanced by the input torque are produced in the second region. Then, the high pressure fluid is delivered out of the housing 1 via the second line 22. As the ring 2 is rotated relatively to the housing 1, the geometrical center b of each first bearing 3 and the center g of each piston 8 are shifted along the Y axis, whether the converter is used as a motor or a pump, as mentioned above. In this fluid energy converter, each first bearing 3 has a pair of pressure pockets 7a and 7b axially neighboring one another. The fluid flows out of the corresponding chambers 13 and is distributed to the pressure pockets 7a and 7b via the restrictors 40a and 40b. Hence, the configurations shown in Figs. 8-10 are obtained.

Referring to Fig. 8, when the geometrical center b of each first bearing 3 is not displaced with respect to the center g of each piston 8 in the direction of the Y axis, the pressures inside the pockets 7a and 7b are identical, and therefore the point of application q, or center of pressure, of the force F_a acting on the ring 2 due to the static pressure inside the bearings 3 is not displaced at all from the center g of each piston 8 in the direction of the Y axis.

Referring to Fig. 9, when the center g of each piston 8 is axially displaced from the geometrical center b of each first bearing 3, each fluid leakage gap 45b in the bearings 3 that is remoter from the pistons 8 becomes slightly larger than each fluid leakage gap 45a closer to the pistons 8 due to the elastic deformation of the peripheral wall 2a of the ring 2 due to the hydraulic pressure. Consequently, the pressure inside the pocket 7b that is remoter from the pistons 8 becomes lower than the pressure inside the pocket 7a that is closer to each piston 8. As a result, the center of pressure q of each first bearing 3 comes closer to each piston 8 than the geometrical center b, automatically reducing the axial distance between the center of pressure q and the center g of each piston 8.

The configuration shown in Fig. 10 is derived by rotating the above configuration through 180°. Specifically, when the center g of each piston 8 is displaced from the geometrical center b of each first bearing 3 in the axial direction opposite to the foregoing direction, each fluid leakage gap 45a of the bearings 3 that is remoter from each piston 8 becomes slightly larger than each fluid leakage gap 45b that is closer to each piston by the elastic deformation of the peripheral wall 2a of the ring 2 that is caused by the hydraulic pressure. Consequently, the pressure inside the pocket 7a remoter from each piston 8 becomes lower than the pressure inside the pocket 7b closer to each piston. As a result, the center of pressure q of each first bearing 3 comes closer to each piston 8 than the geometrical center b. Also, the axial gap between the center of pressure q and the center g of each piston 8 is automatically reduced.

The novel rotary fluid energy converter can be employed either as a hydraulic pump or as a hydraulic motor as mentioned above. In either case, only the hydrostatic pressure of the fluid introduced into the first bearings 3 and the second bearings 9 produces the couples induced by F_a and $F_b$on the ring 2. The couples induced by F_a and F_b are balanced by the input of output torque acting on the ring 2. Hence, the hydrostatic pressure of the fluid can be directly converted into rotary motion of the ring 2, only. Also it is possible to transform the rotary motion of the ring 2 into fluid pressure . Thus, a mechanism for mechanically converting rectilinear motion and rotary motion is entirely dispensed with. Moreover, as already described, the axial distance between the center of pressure of each first bearing and the center of each piston is minimized to thereby prevent undue bending or twisting force from acting on the ring.

Referring now to Figs. 13-22, there is shown an alternative embodiment of the energy converter according to the invention. This converter is similar to the converter already described in connection with Figs. 1-10, except for the structure of the shoes of the static pressure bearings. This converter has the first static pressure bearings 3 which attached shoes 5 to the outer periphery of the torque ring 2 at requisite positions, the shoes 5 being also attached to the tapering surface 4 of the housing 1. Each shoe 5 is provided with three pressure pockets 7a, 7b, 7c axially neighboring one another. Hydraulic pressure is introduced into these pockets 7a-7c. An odd number of bearings 3 are circumferentially regularly spaced from one another. The surrounding portions 5a, 5b, 5c, 5d, 5e that surround the pressure pockets 7a-7c are so shaped that their cross section protrudes toward the tapering surface 4, as shown in Figs. 17-19. this reduces the area with which each shoe 5 is in sliding contact with the tapering surface 4. Also, the surrounding portions 5a-5e are formed so as not be to parallel to the direction of rotation X. More specifically, only the surrounding portions 5 a and 5b which are perpendicular to the direction of rotation X are shaped into a rectilinear form. The surrounding portions 5c and 5e are shaped like the letter "V". The surrounding portion 5d is so shaped as to be oblique to the direction of rotation X. It is to be noted that Figs. 13-16 are basically the same as Figs. 1-4, and the components shown in those figures will not be described herein.

In this structure, the hydraulic pressure inside the chambers 13 corresponding to the pistons 8 is directed into the pressure pockets 11 in the corresponding second bearings 9 via the pressure inlet passage 41 formed along the axis of each piston 8. The hydraulic pressure inside the pockets 11 is routed into the pressure pockets 7a, 7b, 7c in the corresponding bearings 3 via the fluid passages 42a, 42b, 42c formed in the ring 2. These passages 42a-42c cooperate with the pressure pockets 11 to form slide valve elements 50.

Referring to Figs. 20-22, each valve element 50 acts to selectively cut off the supply of the fluid into the pockets 7a, 7b, 7c, making use of the relative movement between each piston 8 and the ring 2 in the direction of the Y axis. When the distance between the geometrical center b of each first bearing 3 and the center g of each piston 8 in the direction of the Y axis keeps within a given range, the pockets 11 are in communication with all the fluid passages 42a, 42b, 42c. When the distance increases beyond the range, the passage 42c or 42a remoter from the piston 8 interrupts communication with the pocket 11, as shown in Figs. 21 and 22. The restrictors 40a, 40b, 40c are formed in the passages 42a and 42b.

As described above, as the ring 2 is rotated relatively to the housing 1, the geometrical center b of each first bearing 3 and the center g of each piston 8 are moved in the direction of the Y axis, whether the converter is used as a motor or as a pump. In this fluid energy converter, each first bearing 3 is provided with the pressure pockets 7a, 7b, 7c axially neighboring one another. Each slide valve element 50 is provided to selectively interrupt the supply of the fluid into the pockets 7a-7c, making use of the relative movement between each piston 8 and the ring 2 in the direction of the Y axis. Consequently, actions as shown in Figs. 20-22 are obtained. Specifically, when the geometrical center b of each first bearing 3 is not displaced from the center g of each piston 8 in the direction of the Y axis as shown in fig. 20, all the fluid communication passages 42a, 42b, 42c are in communication with the pressure pockets 11, so that the pressures inside the pockets 7a, 7b, 7c are equal. Consequently, the point of application q, or center of pressure, of the force F_a acting on the ring 2 due to the static pressure in the first bearings 3 is not displaced at all from the center g of each piston 8 in the direction of the Y axis. When the center g of the piston 8 is displaced only slightly in the direction of the Y axis but displaced considerably to the vicinities of points t and u shown in fig. 24 in the direction of the X axis, the fluid passages 42a and 42c are disconnected from the pockets 11, leaving only the fluid passages 42b in communication with the pockets 11. The result is that the center g of each piston 8 is axially displaced only slightly from the point of application q, or the center of pressure, of the force F_a acting on the ring 2.

When the center g of each piston 8 is displaced from the geometrical center b of each first bearing 3 in the direction of the Y axis as shown in Fig. 21, the passage 42c remoter from the piston 8 is not in communication with the pocket 11. This permits pressure fluid to be supplied only into two pressure pockets 7a and 7b in the bearings 3 which are closer to the piston 8. As a result, the center of pressure q of the first bearing 3 comes closer to the piston 8 than the geometrical center b. In this way, the axial distance between the center of pressure q and the center g of each piston 8 is automatically reduced. When this configuration is rotated through 180°, the configuration shown in Fig. 22 is derived.

Referring to Fig. 22, when the center g of each piston 8 is axially displaced from the geometrical center b of each first bearing 3 in the direction opposite to the foregoing direction, the passage 42a remoter from the piston 8 is disconnected from the pocket 11. Hence, pressure fluid is supplied only into the two pressure pockets 7b and 7c in the bearings 3 which are closer to the piston 8. As a result, the center of pressure q of each first bearing 3 comes closer to each piston 8 than the geometrical center b. Also in this case, the gap between the center of pressure q and the center g of the piston is automatically reduced. Immediately after the fluid passage 42c or 42a is isolated, i.e. when the geometrical center b of the bearing 3 is not yet greatly displaced from the center g of the piston 8, there arise the possibility that the center of pressure q becomes remoter from the geometrical center b than the center g of the piston 8. If the pressure center q moves past the center g of the piston 8 and across the geometrical center in the direction of the Y axis, the fluid leakage gap 45c or 45a to which the pressure center q has come closer becomes slightly larger due to the elastic deformation of the peripheral wall 2a of the ring 2. Then, the pressure inside the pressure pocket 7a or 7b to which the pressure center has come closer decreases slightly. Therefore, the position of the pressure center is brought closer to the center g of the piston 8. In Figs. 20-22, F_a, F_b, and F_c schematically indicate forces acting on the ring 2 because of the pressures inside the pockets 7a, 7b, 7c, respectively.

Since the converter is designed as described above, the distance between the pressure center of each first static pressure bearing and the center of each piston along the Y axis is reduced to a minimum in the same manner as in the converter already described in conjunction with Figs. 1-10.

This can prevent undue bending or twisting force from acting on the torque ring. Therefore, it is easy to design the structure in such a way that its components are not severely pressed against each other or twisting force does not act on them. Further, it is possible to quite dispense with bearings utilizing the wedging action of oil films relying on the oiliness or viscosity of lubricating oil, or with bearings utilizing the rolling action of balls, rolls, or the like. Moreover, adequate static pressure bearings may be provided enabling using water or other fluid exhibiting a viscosity comparable to that of water without difficulty. Also, when static pressure bearings are used instead of roller bearings, the machine is not affected by the operation life of roller bearings. This makes it possible to increase the operation life of the machine. In addition, it helps make the machine in smaller size and lighweight.

When the eccentric position of the pintle relative to the axis of the housing is adjusted as in the illustrated embodiment, the converter can be advantageously used as a hydraulic pump or motor of the variable displacement type. Of course, the invention is not limited to this scheme.In addition, as the eccentric position of the pintle is adjustable, the adjusting means is not limited to the foregoing means. For instance, the pintle may be reciprocated by a hydraulic actuator.

Furthermore, the cross-sectional shape of the surrounding portions that surround the pressure pockets in the first static pressure bearings is not limited to the shape described above. Where the cross section protrudes as described already, however, shapes of wedge-shaped cross section are formed between the surrounding portions and the tapering surface. When the converter operates, fluid enters the wedge-shaped spaces, producing hydrodynamic pressure. This allows the housing and the torque ring to be rotated relatively to each other more smoothly. Since the surrounding portions are so shaped that no portion is parallel to the direction of rotation, the hydrodynamic pressure is generated on every portion of the surrounding portions. Therefore, when the converter runs at high speeds, especially excellent bearing action can be obtained. Obviously, it is possible to fabricate the torque ring 2 and the shoes 5 integral as shown in Fig. 25. When the ring 2 and the shoes 5 are integrally molded, angle θ_1 which is half of the angle that the protruding portion of each surrounding portion makes is made larger than the complementary angle θ_3 of the cone angle θ_2 at the tapering portion of the outer periphery of the torque ring. Then,moulds for the outer periphery of the ring can be removed axially, enhancing the productivity. In other words, by making the gradient of the protruding portion of the cross section of the surrounding portion not larger than the gradient of the cone formed by the inner surface of the housing, mould release is facilitated. Additionally, the number of the pistons is not limited to the number in the illustrated embodiment. The working fluid is not limited to liquids. For example, it can be a gas such as air.

Since the novel rotary fluid energy converter is constructed as described thus far, it can act either as a pump or as a motor without using a mechanism for mechanically converting rectilinear or rotary motion into another form of motion. Further, it includes a simple structure which does not use valve element or the like at all but which can effectively prevent couple from occurring on the torque ring on a position off the axis of rotation, which would otherwise be caused by the presence of axial distance between the pressure center of each first static pressure bearing and the center of each piston.

Claims

1. A rotary fluid energy converter comprising :
a housing (1) having its inner surface with a tapering portion (4) ;
a torque ring (2) closely held against said tapering surface (4) of the housing (1) via first static pressure bearings (3) that are circumferentially regularly spaced apart from one another, said ring (2) having flat inner surfaces (2c) cooperating with said first bearings (3) ;
pistons (8) disposed on the inner side of the torque ring (2) and having their front ends (8a) attached to the flat inner surfaces (2c) of the ring (2) via second static pressure bearings (9) ;
a cylinder barrel (15) for slidably holding the base ends of said pistons (8) ;
a pintle (14) which is slidable in a direction perpendicular to the axis of the housing (1) and which rotatably holds the cylinder barrel (15) ;
chambers (13) formed between each piston (8) and the cylinder barrel (15), the volumes of the chambers (13) cyclically increasing or decreasing as the torque ring (2) rotates relatively to the housing (1);
two fluid communication lines (21, 22) with which the chambers (13) alternately communicate as their volume is increasing and decreasing, respectively ;
fluid passages (23) for directing fluid from the chambers (13) to the first and second static pressure bearings (3, 9) whereby the static pressure of the fluid introduced in the first static pressure bearings (3) and the static pressure of the fluid introduced in the second static pressure bearings (9) produce a couple of forces (F_a, F_b) about the axis of rotation of the torque ring (2) ;
characterized in that it further comprises :

at least two axially adjacent pressure pockets (7a, 7b ; 7a, 7b, 7c) formed in each of the first static pressure bearings (3) ; and

means (40a, 40b ; 50) for automatically selectively distributing pressure fluid into said pressure pockets (7a, 7b) so that when the median axis of the piston (8) is displaced relatively to the median axis of the corresponding pressure bearing (3) the pressure pockets being remoter from said median axis of the piston (8) have a pressure at least inferior as compared to the pressure pockets being closer to said axis.

2. A rotary fluid energy converter according to claim 1, characterized in that said means for automatically selectively distributing pressure fluid into the pressure pockets (7a, 7b) are constituted of restrictors (40a, 40b) through which the fluid flowing out of the chambers (13) is distributed to the corresponding pressure pockets (7a, 7b).

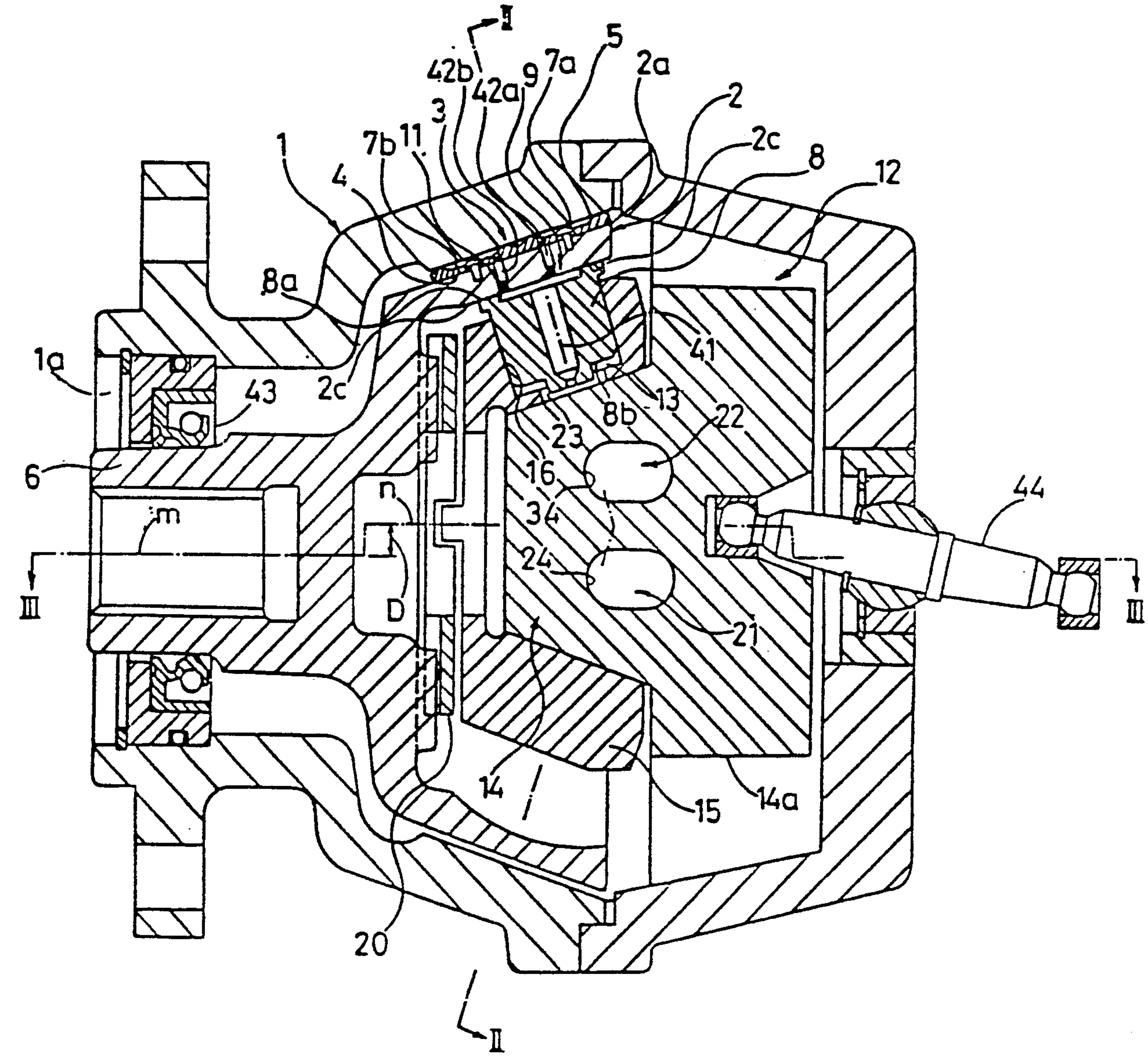
3. A rotary fluid energy converter according to claim 1, characterized in that said means for automatically selectively distributing pressure fluid into the pressure pockets (7a, 7b) are constituted of slide valve elements (50) for selectively intermitting the supply of fluid into said remoter pressure pockets (7a, 7b) by making use of the axial relative movement between each piston (8) and the torque ring (2).

4. A rotary fluid energy converter according to any claims 1 to 3, characterized in that said pressure pockets (7a, 7b ; 7a, 7b, 7c) are delimited by continuous wall portions being in broken line, and extending substantially not parallely to the circumferential direction X.

5. A rotary fluid energy converter according to any claims 1 to 4, characterized in that said pressure pockets (7a, 7b ; 7a, 7b, 7c) are delimited by wall portions having their cross section narrowing in direction of said tapering surface (4) of said torque ring (2), so as to reduce their contact area with said surface (4).

6. A rotary fluid energy converter according to claim 2, characterized in that said torque ring (2) exhibits a sufficient elasticity for enabling its peripheral wall 2a be deformed at the level of said first bearings (3) due to the hydraulic pressure and delimit unequal leakage gaps (45a, 45b) on each side of said bearings.

fig.1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2a
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2c
7b
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1
12
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8a
41
1a
43
2c
8b
13
22
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6
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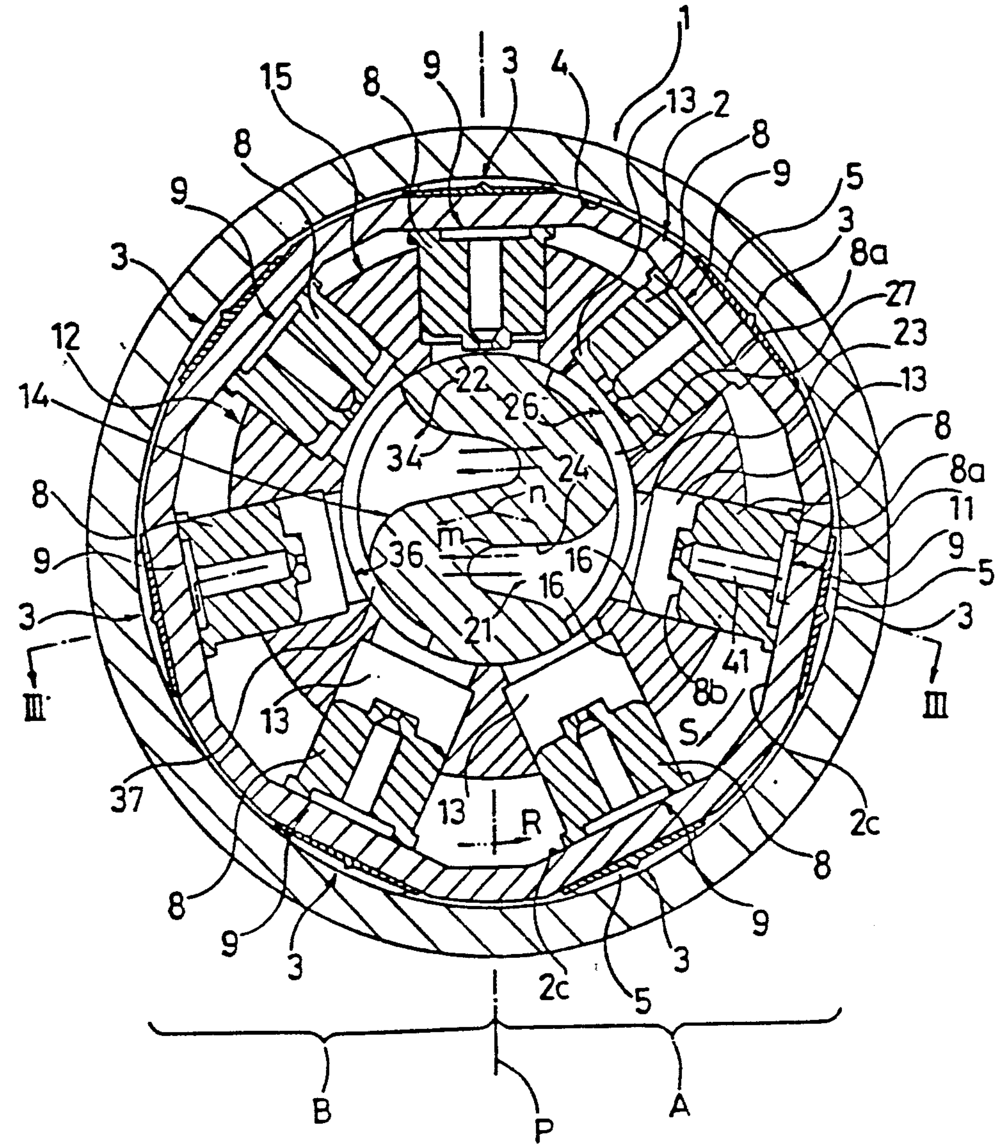
34
44
m
III
24
III
D
21
14
15
14a
20
II

fig. 2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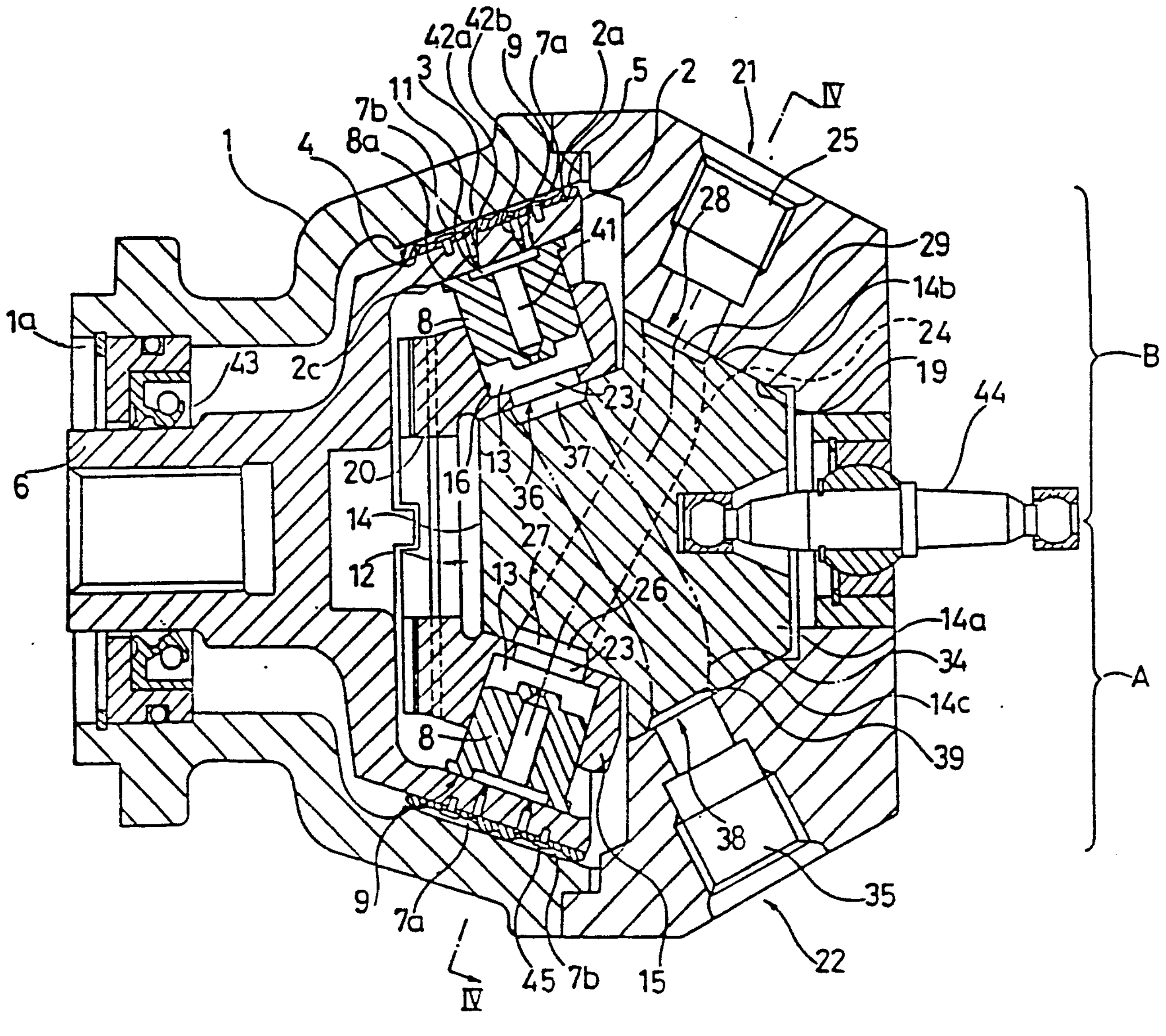
26
27
34
36
37
41
2c
m
n
S
R
III
A
B
P

fig.3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14b
8
24
1a
43
2c
19
B
23
44
20
16
13
37
6
36
14
27
12
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A
14c
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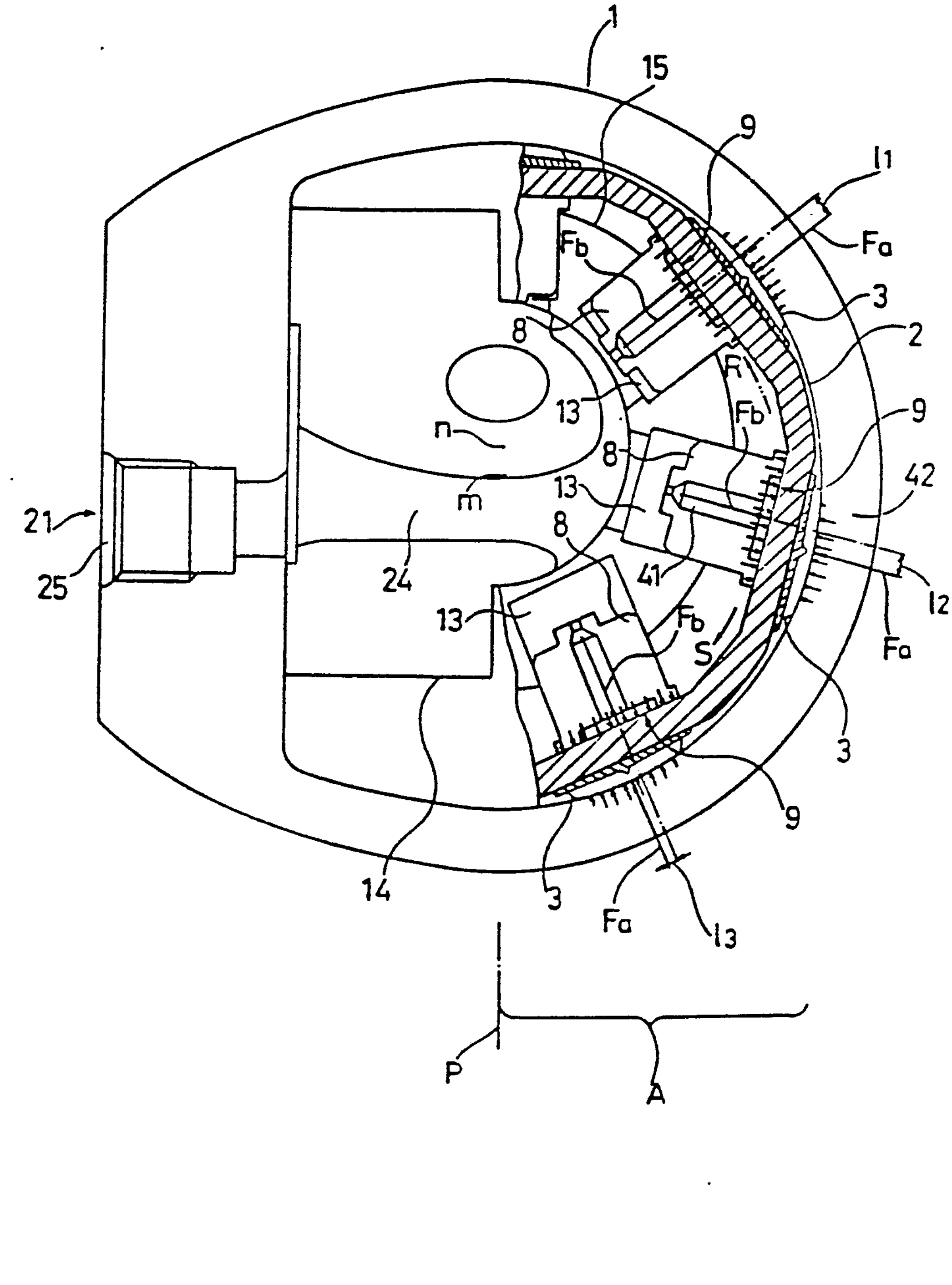
35
9
7a
45
7b
15
22
IV

fig.4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m
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l_2
13
Fb
S
Fa
3
9
14
3
Fa
l_3
P
A

fig.5

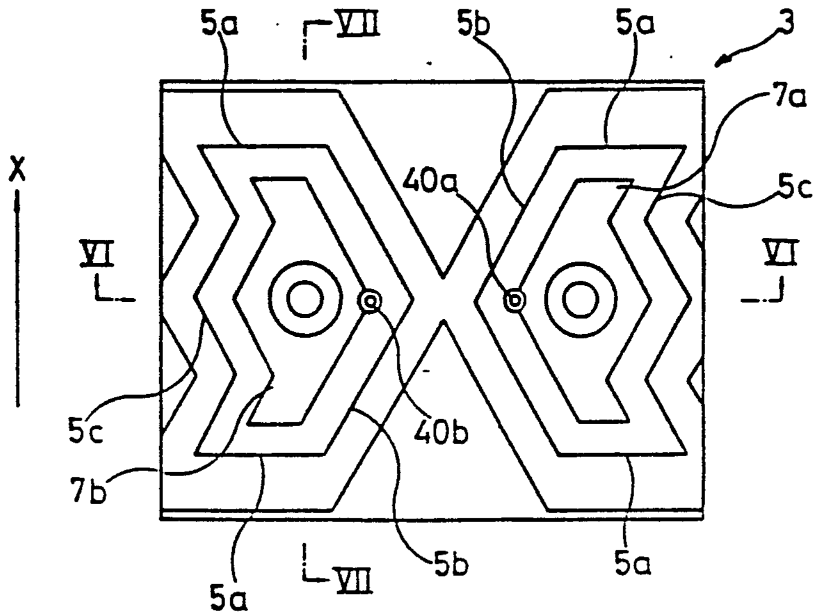


fig.6

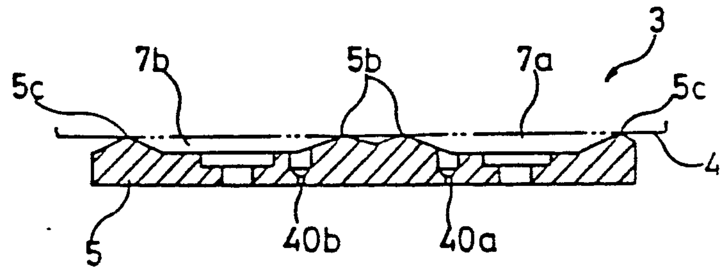


fig.7

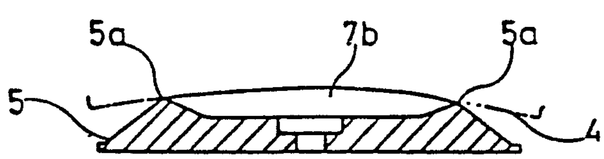


fig.8

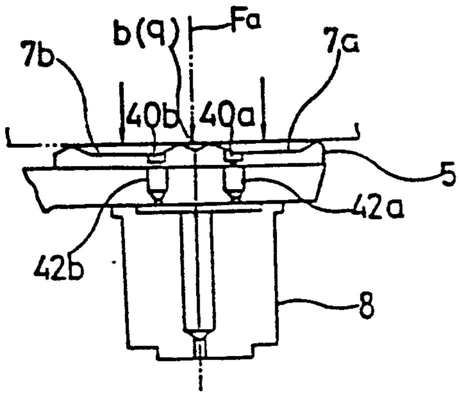


fig.9

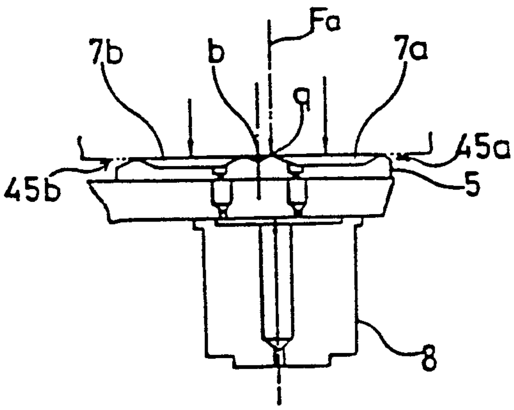


fig.10

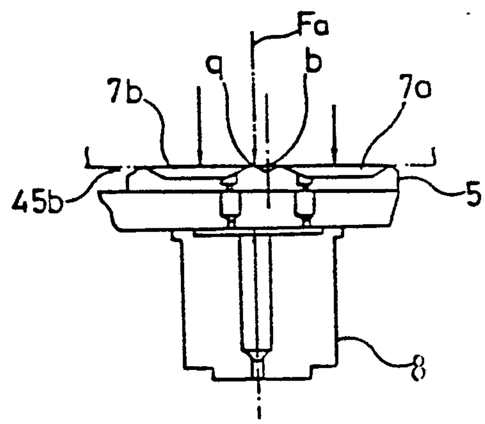


fig.11

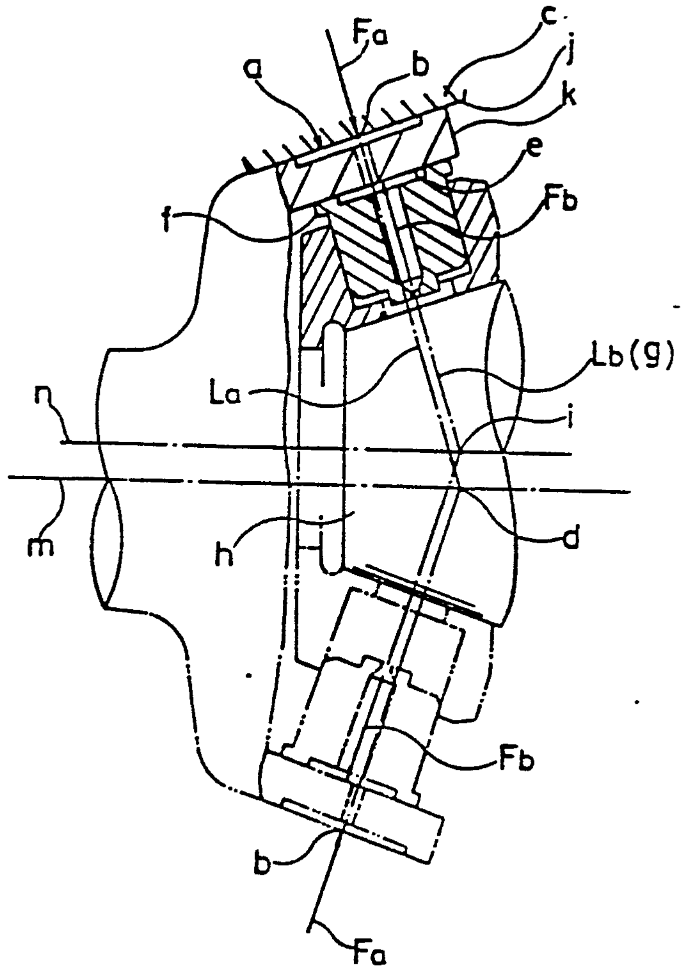


fig.12

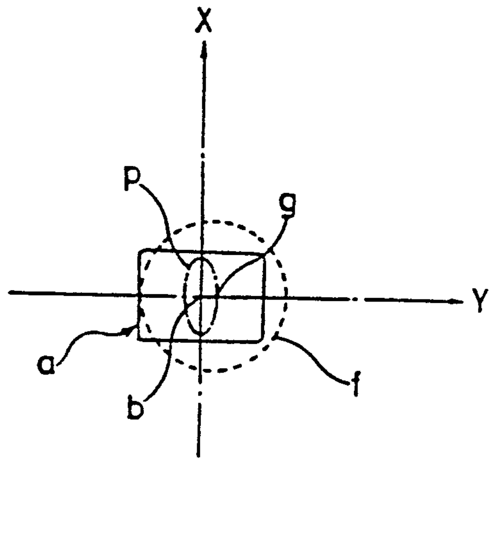
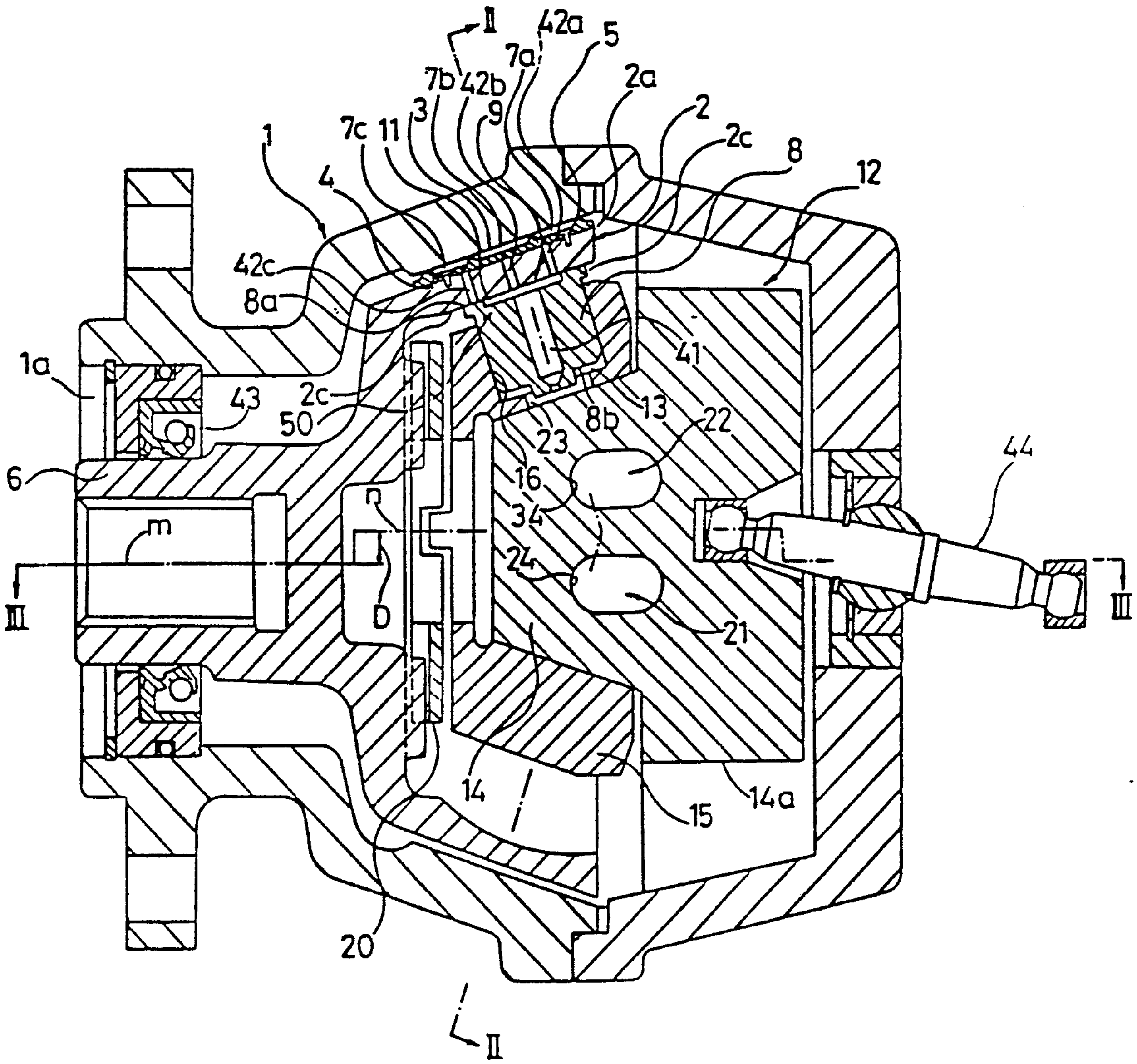


fig.13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1a
2c
43
50
13
23
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m
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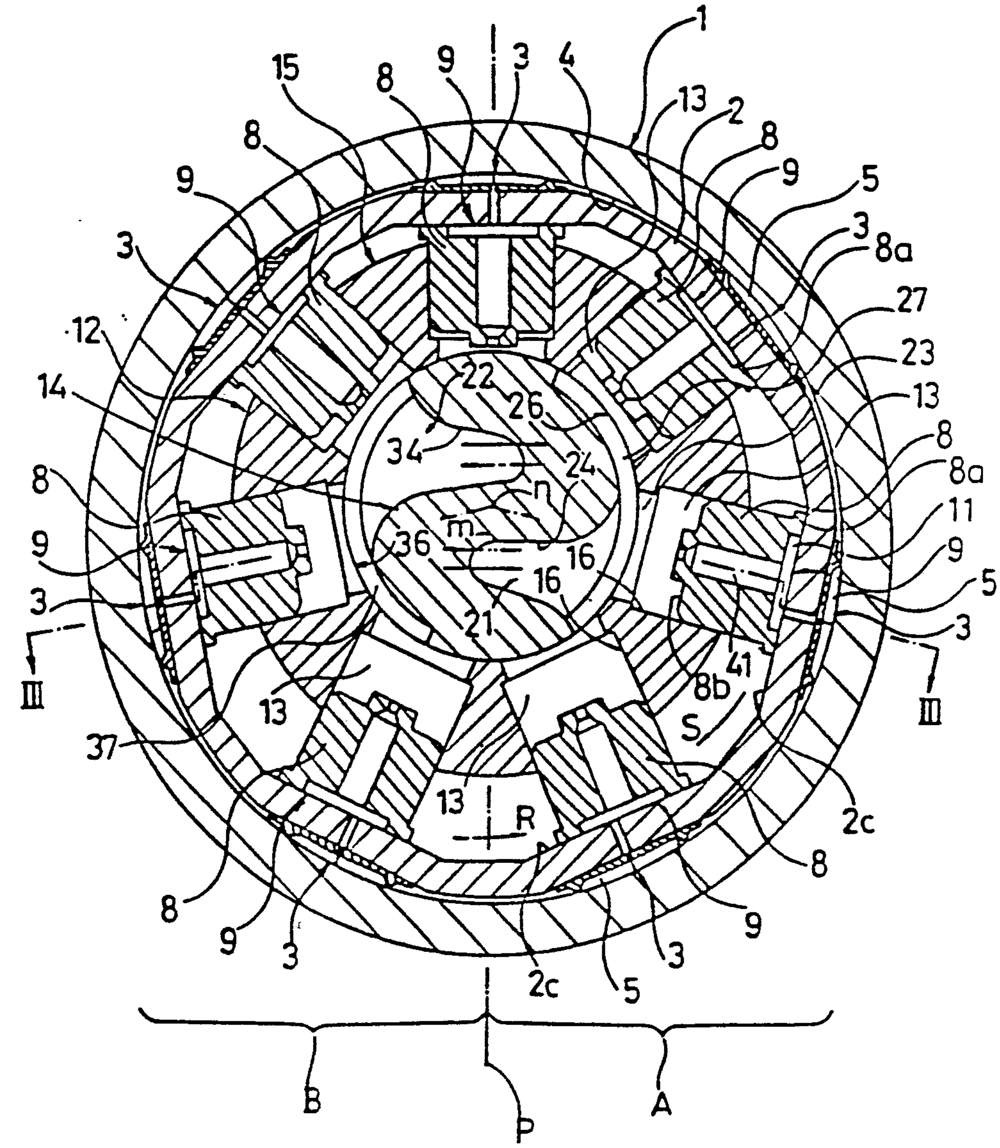
III
D
24
21
14
15
14a
20
II

fig.14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m
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21
8b
41
S
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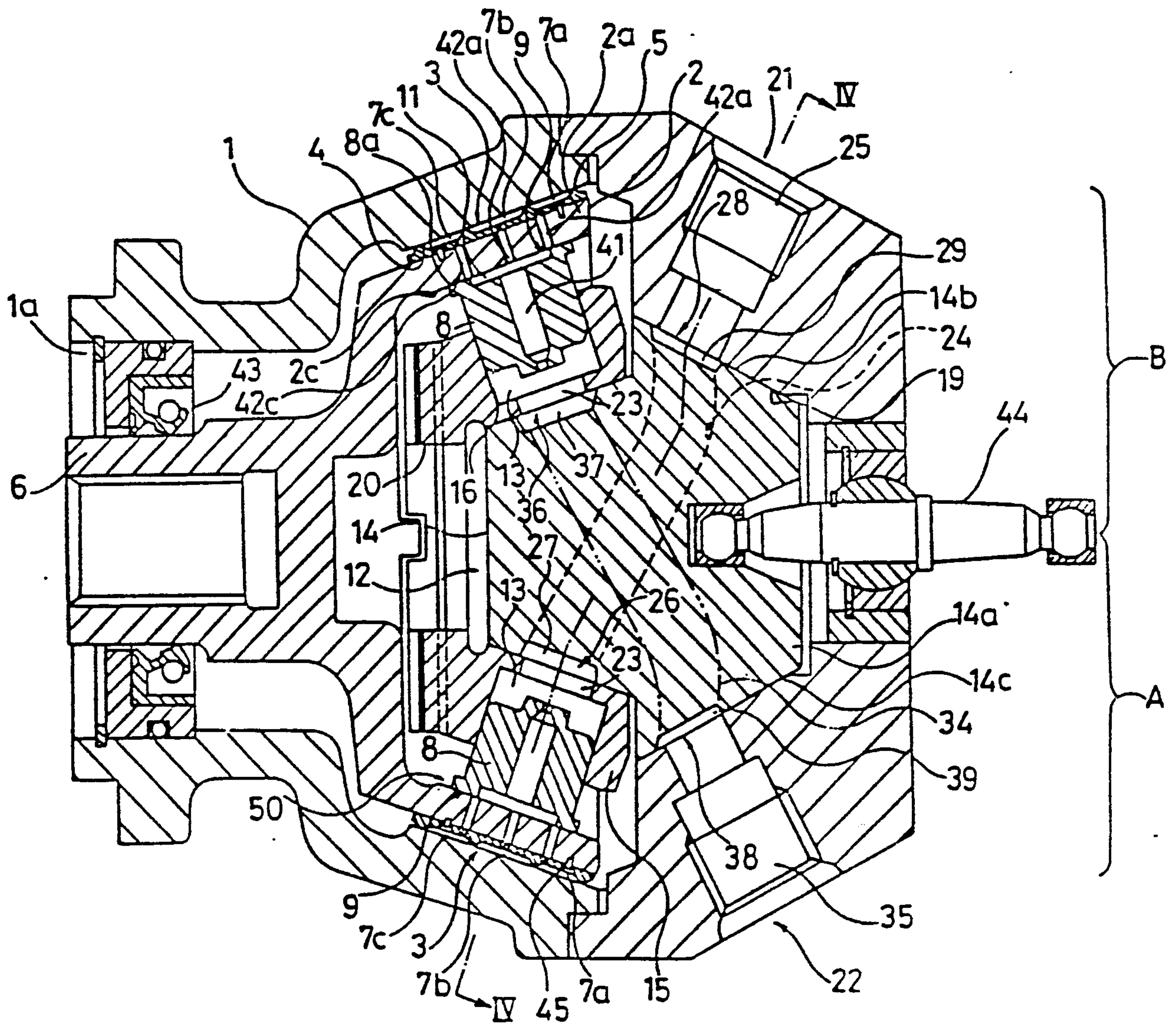
5
B
P
A

fig.15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14b
14c
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16
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22
23
24
25
26
27
28
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34
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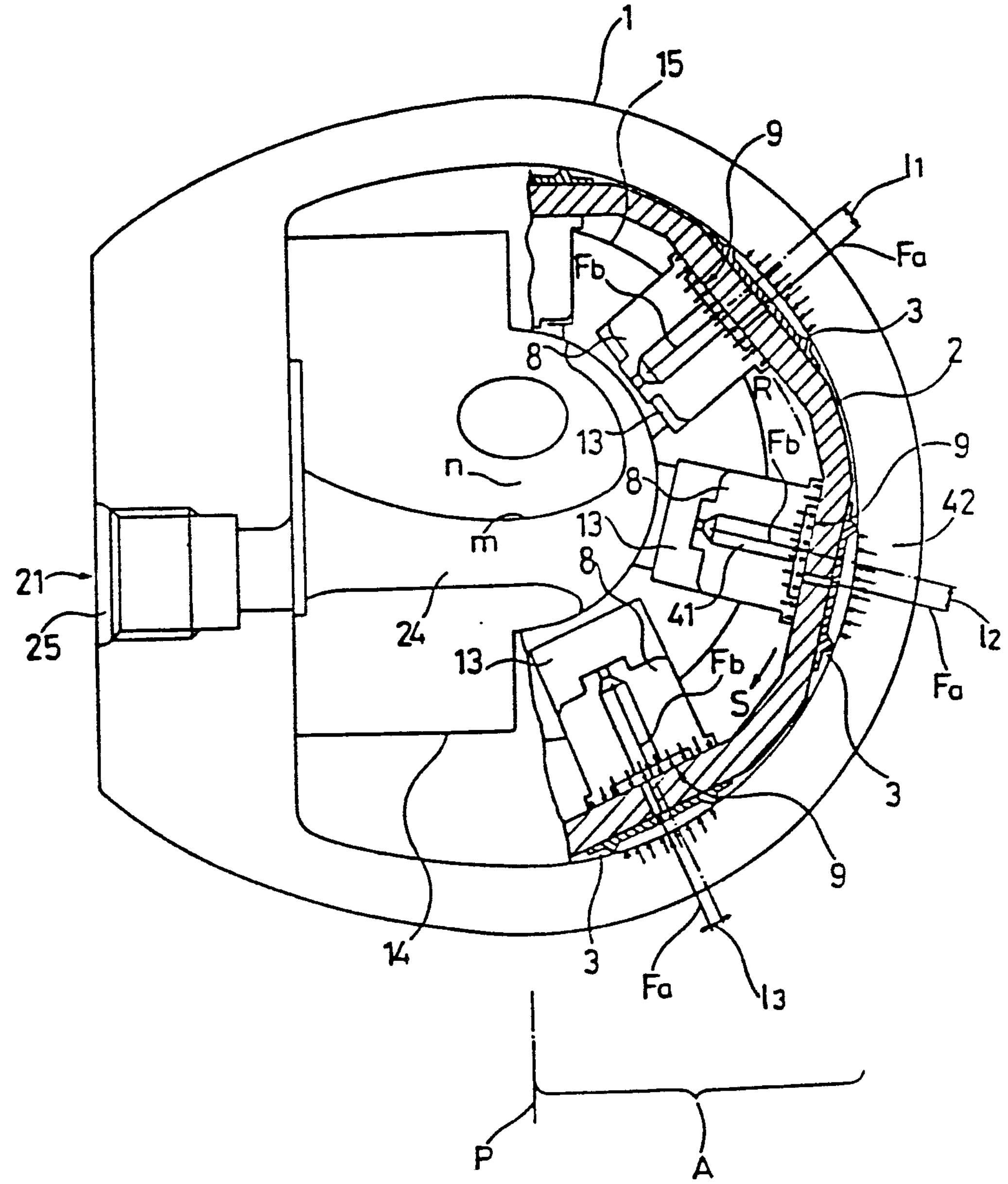
36
37
38
39
41
42a
42c
43
44
45
50
IV
A
B

fig.16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m
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S
3
9
14
3
Fa
l3
P
A

fig.17

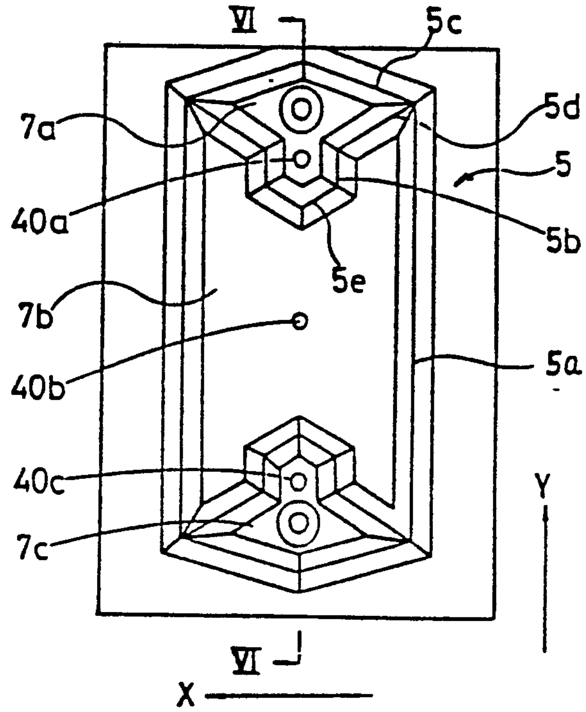


fig.18

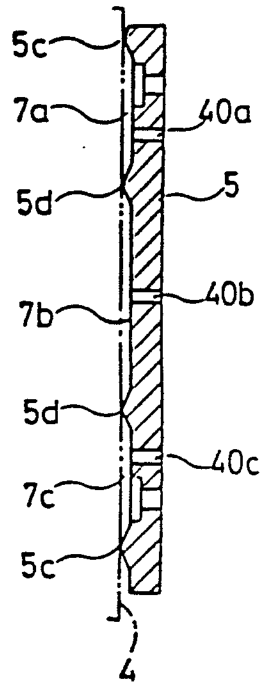


fig.19

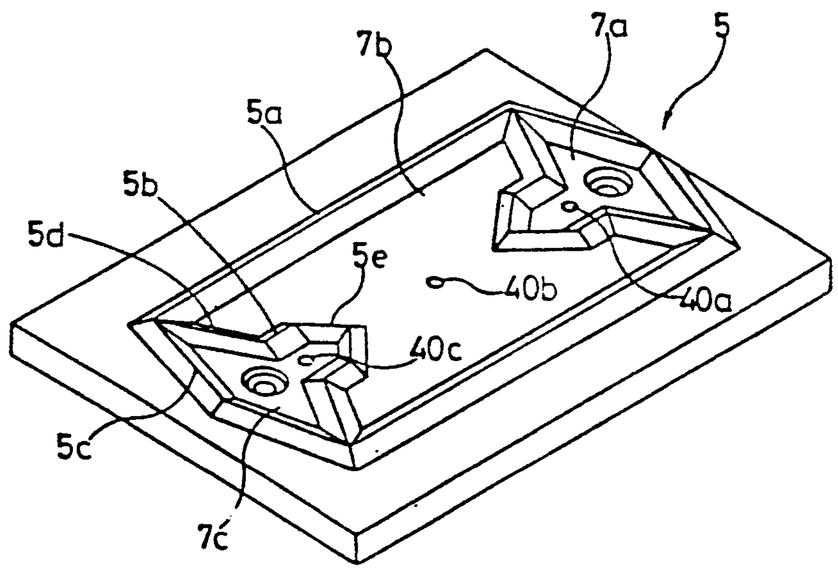


fig.20

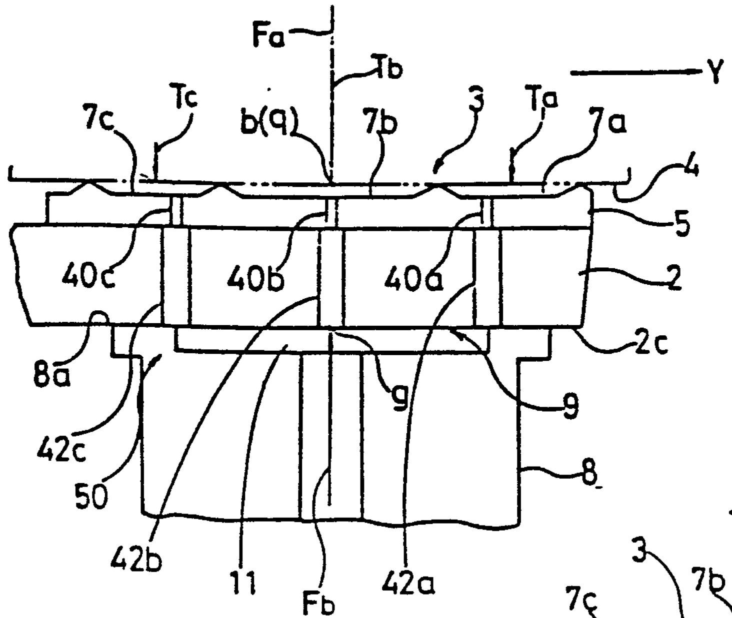


fig.21

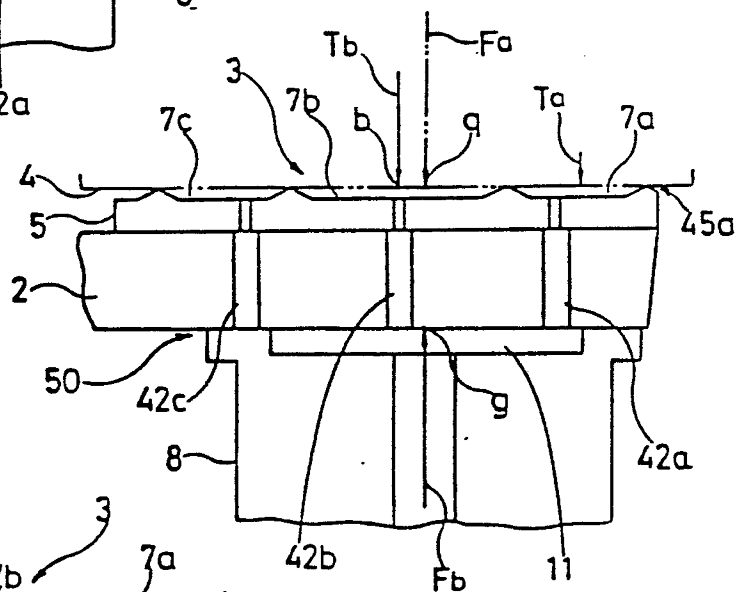


fig.22

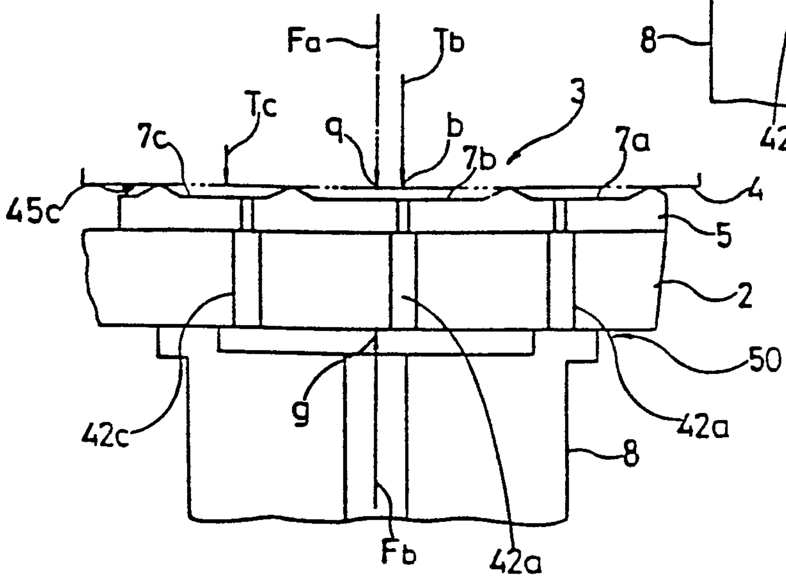


fig.23

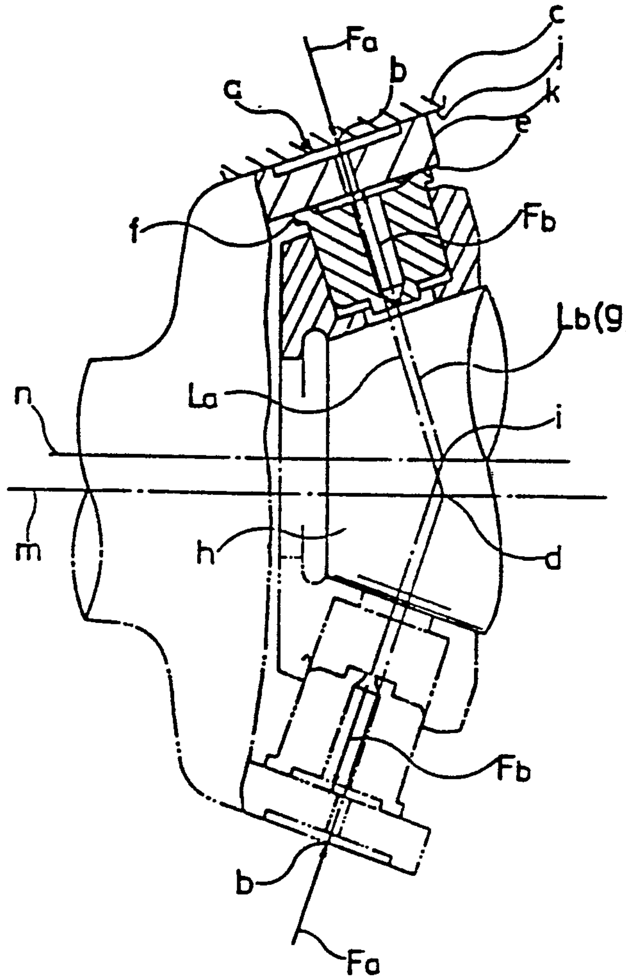


fig.24

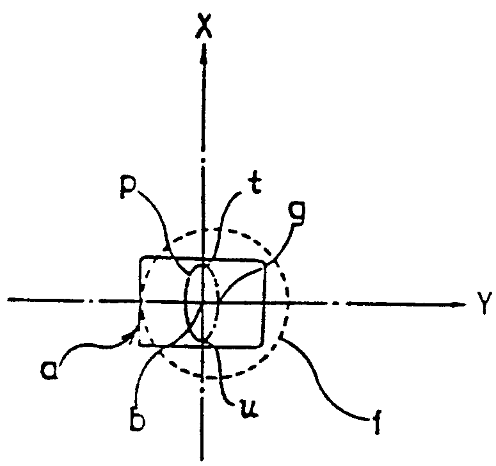
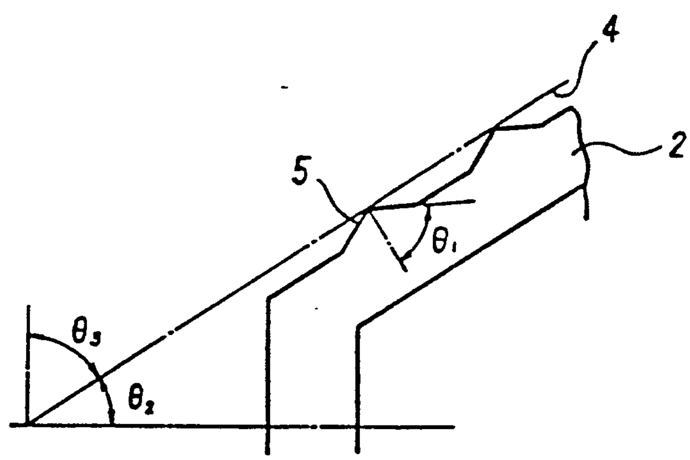


fig.25



European Patent Office

EUROPEAN SEARCH REPORT

Application number

EP 86 40 0385

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.4)
A	EP-A-0 078 513 (SHIMADZU) * Whole document * & JP-A-58 77 179 (Cat. D)	1	F 04 B 1/10

A	DE-A-2 416 772 (VOITH) * Page 8, last paragraph - page 9, first paragraph *	1	

TECHNICAL FIELDS SEARCHED (Int. Cl.4)
F 04 B
F 03 C
F 01 B

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	22-10-1986	VON ARX H.P.

CATEGORY OF CITED DOCUMENTS

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