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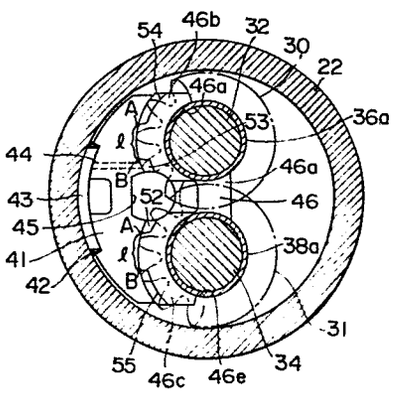
(54) **A tooth top sealing mechanism for a gear pump or motor.**

(57) A gear tooth top sealing mechanism for a gear pump or motor, comprising a pair of gears (30,31), being engaged with each other with their external teeth, the shafts (32,34) of the gears, being supported by bearings provided on both sides in a housing, a seal block (41), being provided between the tooth tops of said gears and the inside wall surface of the housing, arranged to be pressed to the tooth tops of the gears by a high pressure acting on the outside surface of the seal block, and side plates (46), being provided to cover both the sides of said gears at least on the high pressure side, to closely contact, at their outside ends, the inside surface of said seal block and to contact, at their inside ends, the circumferential surfaces of protrusions (36a, 38a) of said bearings, whereby the substantial tooth top sealing length ℓ of said seal is made almost equal to, but larger than the adjacent teeth pitch P of said gears, and throttle slots (54, 55) are formed in the positions adjacent to said tooth top sealing portions on the low pressure side.

FIG.5



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"A TOOTH TOP SEALING MECHANISM FOR A GEAR PUMP OR MOTOR"

The present invention relates to a gear pump or motor with gears engaged with each other with their external teeth.

In more detail, it relates to a gear pump or motor with a cover and a mounting flange provided on both sides of the housing, with the shafts protruded on both sides of said two gears and supported by the bearings provided in the cover and the mounting flange, and with a seal block arranged on the higher pressure side of the housing, to seal the tooth tops of said gears on the delivery side.

Brief explanation of the prior art:

As a gear pump of this kind, the one shown in Figs. 1 and 2 is known hitherto.

In the conventional gear pump, a cover and a mounting flange (not illustrated) are provided on both sides of a housing 1, and bushings 2 and 3 (bushings on the cover side are not illustrated) are pressed in the cover and

the mounting flange. The bushings 2 and 3 support the gear shafts 6 and 7 of gears 4 and 5 and have inward curves of a side plate 8 fitted on their circumferential surfaces.

The outward curves of said side plate 8 are made to coincide with the inward curves of a seal block 9, to support the seal block 9.

The seal block 9 supported as mentioned above has a loading area 10 formed on its outwardly curved surface which is a contact surface with the housing 1, allowing a high pressure to be applied to the loading area 10 from high pressure introducing holes 11 and 12.

Thus, the high pressure introduced into the loading area 10 is balanced by the pressure in the spaces of said gears, to seal the tooth tops of the gears by the inwardly curved surfaces of the seal block 9.

The disadvantage of this conventional mechanism is that if said pump is operated at a high speed at a high pressure, the load becomes excessive, causing the gear tracks of the seal block 9 to be deeper, and that later, the volumetric efficiency drops remarkably.

Such too deep gear tracks are caused by the fact that the substantial tooth top seal length ℓ of the

seal block 9 in the conventional mechanism, viz. the length from point A to point B is about double the adjacent teeth pitch P of said gears.

This will be described in detail in reference to the view showing the correlation between a gear and a seal block in Fig. 2.

Between the seal block 9 and the tooth tops of the gear 4, there is always a constant clearance S kept. If the gear rotates in the arrow direction in Fig. 2, the fluid in a space X leaks into a space Y, and the fluid in the space Y leaks into a space Z, causing the fluid to leak sequentially into the spaces arranged backward in the direction of rotation.

When the fluid leaks through the clearance S as mentioned above, there always arises pressure drop, and the pressure Yp in the space Y positioned in the middle is theoretically $Yp = (Xp + Zp)/2$.

According to the pressures in the spaces, the loading area 10 is determined, to determine the load, but if said pump is operated at a high speed at a high pressure, said equation cannot be set up.

During high speed high pressure operation, since bubbles are generated in the sucked fluid, the leakage

q_1 from the space X into the space Y increases by the amount of the bubbles, to be larger than the leakage q_2 from the space Y into the space Z.

Therefore, the increase in the leakage q_1 increases the pressure drop, lowering the pressure in the space Y. That is, Yp becomes less than $(Xp + Zp)/2$.

If the pressure in the space Y drops in this way, the balance between the load determined based on said equation and the pressure in the space Y breaks, causing the load to be excessive. If the load becomes excessive, the seal block 9 is elastically deformed on the low pressure side, viz. gear side, and the gears return a little toward the seal block 9. Thus, both deformation and displacement together deepen the gear tracks of the seal block 9 instantly.

If the gear tracks become deep once, the high pressure on the delivery side steadily leaks into the space Y, and in the operation thereafter, the pressure in the space becomes incessantly large, making the load relatively small. Thus, since the load becomes small though the gear tracks become deep, the tooth top clearance S becomes very large.

For this reason, said conventional pump has a

disadvantage that the volumetric efficiency is remarkably lowered by the leak from said clearance S.

Object of the invention:

The object of the present invention is to provide a tooth top sealing mechanism for a gear pump or motor which prevents the generation of excessive load and keeps the clearance between the seal block and the tooth tops always in an optimum condition.

Summary of the invention:

To attain said object, in the present invention, the substantial tooth top sealing length ℓ of the seal block is made approximately equal to, but larger than the adjacent teeth pitch P of said gears, and throttle slots are formed in the positions adjacent to said tooth top sealing portions on the low pressure side.

Brief description of the drawings:

Fig. 1 is a longitudinal sectional view showing the tooth top sealing mechanism of a conventional gear pump.

Fig. 2 is a view showing the correlation between a gear and a seal block in the tooth top sealing mechanism of the conventional gear pump.

Fig. 3 is a longitudinal sectional view showing an example of the present invention.

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

Fig. 5 is a sectional view along the V-V line of Fig. 3.

Fig. 6 is an illustration showing the correlation between a seal block and a gear in another example.

Detailed description of the invention:

In Fig. 3, a gear pump 20 according to the present invention has a housing 21 consisting of a casing 22, a cover 23 and a mounting flange 24. Said casing 22 is cylindrical, and closed by the cover 23 on one side and the mounting flange 24 on the other side. The cover 23 and the mounting flange 24 are connected with both the ends of the casing 22 respectively through seal members 25 and 26 by fitting joints 27 and 28, being fastened firmly by bolts 29. In this joining, dowel pins as used in the ordinary gear pump are not required.

The casing 22 contains a pair of gears 30 and 31 engaged with each other as pumping components. The gears 30 and 31 are provided with shafts 32 & 33 and 34 & 35 respectively on both sides. These shafts 32 to 35 are supported rotatably by bearings 36 & 37 and 38 & 39 as bushing fitted in the cover 23 and the mounting flange 24. These bearings 36 to 39 are protruded at their inner

ends to some extent from the cover 23 and the mounting flange 24, to form protrusions 36a, 37a, 38a and 39a.

The shaft 35 on one side of said gear 30 is protruded outward through the mounting flange 24, and can be connected with a power source outside the gear pump 10.

In the portion of the mounting flange 24 through which the shaft 35 is passed through, an oil seal 40 is fitted, to seal the clearance between the shaft 35 and the mounting flange 24.

And as can be seen from Fig. 5, on the delivery side of the housing 21, a seal block 41 is provided between the inside wall of the casing 22 and the tooth tops of the gears 30 and 31.

The outwardly curved surface of the seal block 41 is formed to suit the inside wall surface of the casing 22, and inwardly curved surfaces are formed inside to suit the outside contours of the gears 30 and 31.

On the outwardly curved surface of the seal block 41, there is a high pressure area 43 sectioned by a seal member 42, and the high pressure area 43 communicates to the delivery side of the gears 30 and 31 through a radial high pressure introducing hole 44 and an axial high pressure groove 45 formed in the seal block 41.

The other portions of the circumferential surfaces of the gears 30 and 31 than the portions facing the inwardly curved surfaces of the seal block 41 have a space against the inside wall of the casing 22, being open to the suction side of the gears 30 and 31.

Said seal block 41 is almost equal to the overall width of the chamber in the casing 22 containing the gears 30 and 31 as can be seen from Fig. 4. Therefore, the seal block 41 is limited in its axial movement by the inside wall surfaces of the cover 23 and the mounting flange 24, but is arranged to be able to move radially to some extent in the range between the inside wall surface of the casing 22 and the tooth tops of the gears 30 and 31.

The gears 3 0 and 31 are made to be somewhat more narrow than the overall width in said casing 22. Thus, clearances are formed between the side surfaces of the gears and the cover 23 and the mounting flange 24, by narrowing the width of the gears 30 and 31, and in the clearances, side plates 46 and 47 made of steel are provided. The side plate 46 is positioned between the cover 23 and the side surfaces of the gears 30 and 31 facing it, while the side plate 47 is positioned

between the mounting flange 24 and the side surfaces of the gears 30 and 31 facing it.

Since the side plates 46 and 47 are the same in form, the side plate 46 only will be described below.

The side plate 46, as can be seen from Fig. 5, consists of a central portion 46a and branching portions 46b and 46c. The central portion 46a is positioned between said protrusions 36a and 38a of the bearings 36 and 38, and the branching portions 46b and 46c are formed to suit the circumferences of the protrusions 36a and 38a. The inside diameter of the curved surfaces 46e formed by the inward curves of the central portion 46a and the branching portions 46b and 46c exactly fits the protrusions 36a and 38a of the bearings 36 and 38. The outwardly curved surfaces of said branching portions 46b and 46c are the same in form as the outside contours of the gears 30 and 31, viz. as the inwardly curved surfaces of the seal block 41, and support the inwardly curved surfaces of the seal block 41. In this case, the outwardly curved surfaces of the side plates 46 and 47 keep arcs dimensionally somewhat smaller than the outside contours of the gears 30 and 31, when the curved surfaces 46d, 46e, 47d and 47e of the side plates

are supported by the circumferential surfaces of the protrusions 36a to 39a of the bearings 36 to 39.

Since both the sides of the gears 30 and 31 are sealed by the side plates 46 and 47 on the delivery side, pressure areas 50 and 51 are formed in the inside wall portions of the cover 23 and the mounting flange 24 respectively adjacent to the side plates 46 and 47.

As mentioned above, the side plate 46 has speed slots 52 and 53 formed on the outwardly curved surfaces on the high pressure side and throttle slots 54 and 55 formed on the outwardly curved surfaces on the low pressure side.

If the speed slots 52 and 53 and the throttle slots 54 and 55 are formed like this, the substantial tooth top sealing length in which the seal block 41 and the side plates 46 and 47 seal the tooth tops of the gears is from point A to point B excluding the speed slots and the throttle slots as shown in Fig. 5.

Said substantial tooth top sealing length ℓ is made to be approximately equal to, but larger than the adjacent teeth pitch P of the gears.

Thus, if the gear pump 20 is operated, the pressure medium is sucked toward the suction side of the chamber

in the casing 22 through an axial passage 57 from an inlet port 56 provided at one side of the cover (see Fig. 3), and is carried toward the delivery side by the gears 30 and 31 rotating mutually in the other directions.

The pressure medium carried as mentioned above is fed from the high pressure groove 45 formed in the seal block 41 through an axial passage 59 formed in the cover 23 and an outlet port 60 opening toward the other side of the cover 23 (see Fig. 3), to an actuator not illustrated.

The high pressure generated on the delivery side of the gears 30 and 31 in this way acts from the high pressure groove 45 to the seal block 41 onto the pressure areas 50 and 51 at the rear of the side plates 46 and 47.

This loads the side plates at the rear at a high pressure, causing them to adhere to both the sides of the gears 30 and 31 on the delivery side with a good sealing effect.

At the same time, said high pressure acts also on the pressure area 43 at the rear of the seal block 41 through the high pressure introducing hole 44, and the seal block 41, being loaded with the high pressure on the rear side, is pressed against the tooth tops of the gears 30 and 31. Thus, the seal block 41 is shaved

off on the inwardly curved surfaces by the tooth tops of the gears 30 and 31, and the tooth tops of the gears 30 and 31 slightly bite the seal block 41, for good sealing of the tooth tops of the gears 30 and 31. The depth of bite by the tooth tops of the gears 30 and 31 into the seal block 41 is as far as the inwardly curved surfaces of the seal block 41 are supported in contact with the outwardly curved surfaces of the side plates 46 and 47.

The above state is secured during the initial adaptation operation of the pump 20. Therefore, in the ordinary state of operation of the gear pump 20, the seal block 41 is always supported through the side plates 46 and 47 by the circumferential surfaces of the protrusions 36a to 39a of the bearings 36 to 39.

The tooth tops of the gears are sealed by the inwardly curved surfaces of the seal block 41 as mentioned above. The substantial tooth top sealing length ℓ is made to be approximately equal to P as mentioned above.

To keep said substantial tooth top sealing length ℓ, the throttle slots can be formed directly on the side plates as mentioned above, but for example as shown in Fig. 6, if a protrusion 61 corresponding to said length ℓ

is formed on each of the inwardly curved surfaces of the seal block 41, to bring the inwardly curved surface of the protrusion 61 into contact with the outwardly curved surface of the side plate, then throttle slots functioning like said throttle slot 54 or 55 can be formed in the other portion than said protrusion 61.

Thus, in Fig. 6, if the gear 30 rotates in the arrow direction, when a space T forward in the rotating direction opens toward the delivery side H, the space U adjacent to it positioned backward in the rotating direction opens toward the suction side L. In other words, the pressure Tp of the space T becomes almost equal to the delivery pressure, and the pressure Up of the space U becomes almost equal to the suction pressure. Therefore, in the length ℓ from point A to point B as the substantial tooth top sealing length of the seal block, the pressure in a space changes from the suction pressure to the delivery pressure, and in the process of the change, any unstable intermediate pressure does not occur. That is, the pressure in a space is either the suction pressure or delivery pressure, being very stable.

Since the pressure in a space is stable as mentioned above, optimum tooth top sealing can be always obtained

if the load is determined according to the pressure.

Furthermore, as mentioned before, since the side plates 46 and 47 have throttle slots 54 and 55 formed, the pressure in a space can be raised a little immediately before the tooth top sealing portion, viz. substantial tooth top sealing length ℓ is reached.

In the high speed high pressure operation which is liable to generate bubbles, the pressure oil in the space on the high pressure side leaks into the space on the low pressure side, but when the pressure oil passes the throttle slots 54 and 55, the pressure in the spaces corresponding to the slots rise a little.

Since the pressure in the space rises a little before the high pressure occurs in the space, the bubbles in the space are crushed beforehand, to prevent the generation of erosion caused by impact destruction of bubbles.

CLAIMS:

1. A tooth top sealing mechanism for a gear pump or motor, comprising a pair of gears (30, 31), being engaged with each other with their external teeth, the shafts (32 to 35) of the gears, being supported by bearings (36 to 39) provided on both sides in a housing (21), a seal block (41) being provided between the tooth tops of said gears and the inside wall surface of the housing, with its inside surface formed to suit the outside contours of the gears and arranged to be pressed to the tooth tops of the gears by a high pressure acting on the outside surface of the seal block, and side plates (46, 47), being provided to cover both the sides of said gears at least on the high pressure side, to closely contact, at their outside ends, the inside surface of said seal block, being the same in form as the outside contours of the gears, and to contact, at their inside ends, the circumferential surfaces of the protrusions of said bearings supporting the shafts of the gears, thereby positioning said seal block at a predetermined position in relation to the protrusions of the bearings, wherein the substantial tooth top sealing length ℓ of said seal block is made almost equal to, but larger than the adjacent teeth pitch P of said gears, and throttle slots (54, 55) are formed in the positions adjacent to said tooth top sealing portions on the low pressure side.

2. A tooth top sealing mechanism for a gear pump or motor, according to Claim 1, wherein said throttle slots (54, 55) are formed on the side plates (46, 47).

3. A tooth top sealing mechanism for a gear pump or motor, according to Claim 1, wherein protrusions (61) corresponding to said substantial tooth top sealing length ℓ are formed on the inside surface of the sealing block (41) to form said throttle slots (54, 55).

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FIG.1

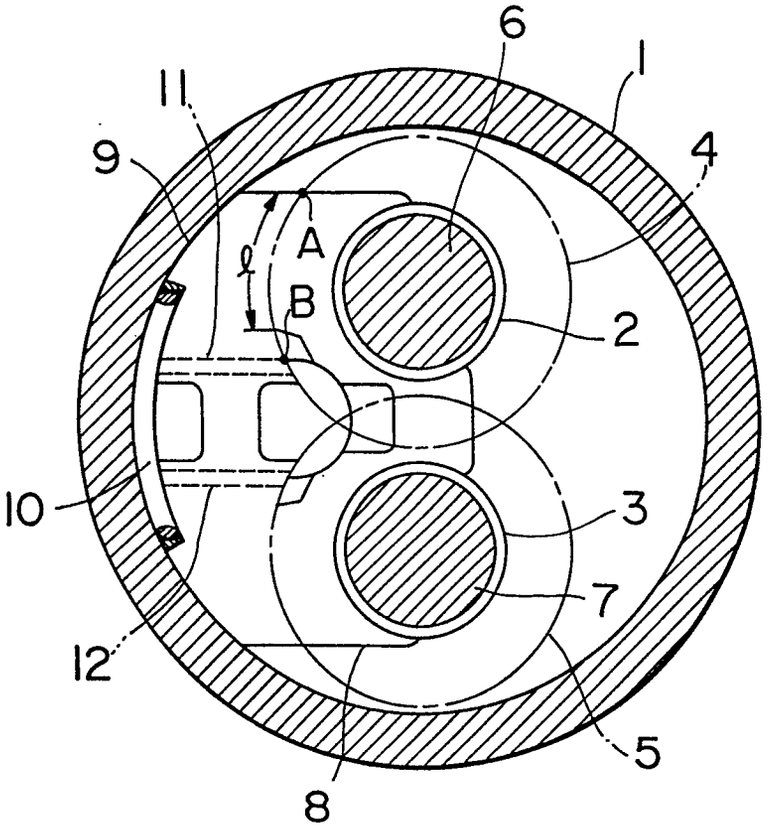


FIG.2

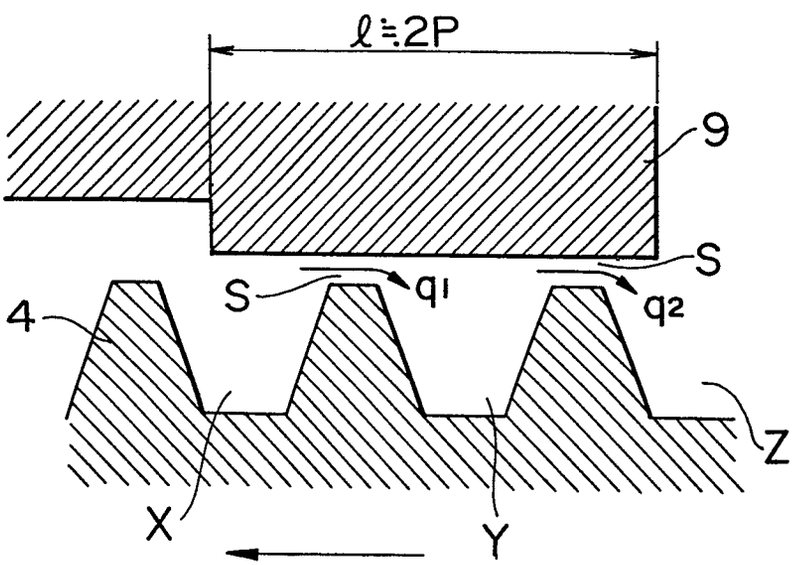
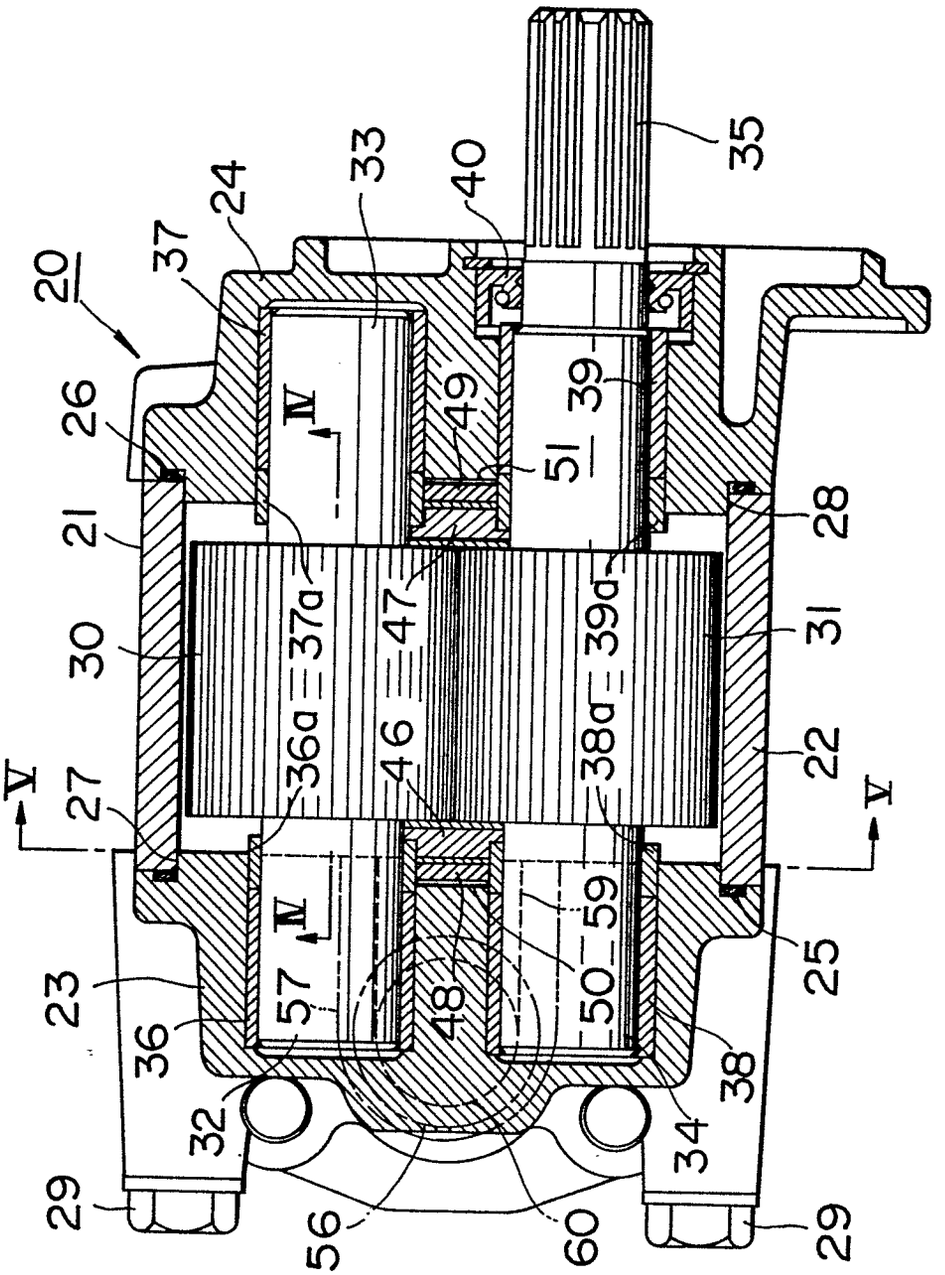


FIG.3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37a
38a
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3/4

FIG.4

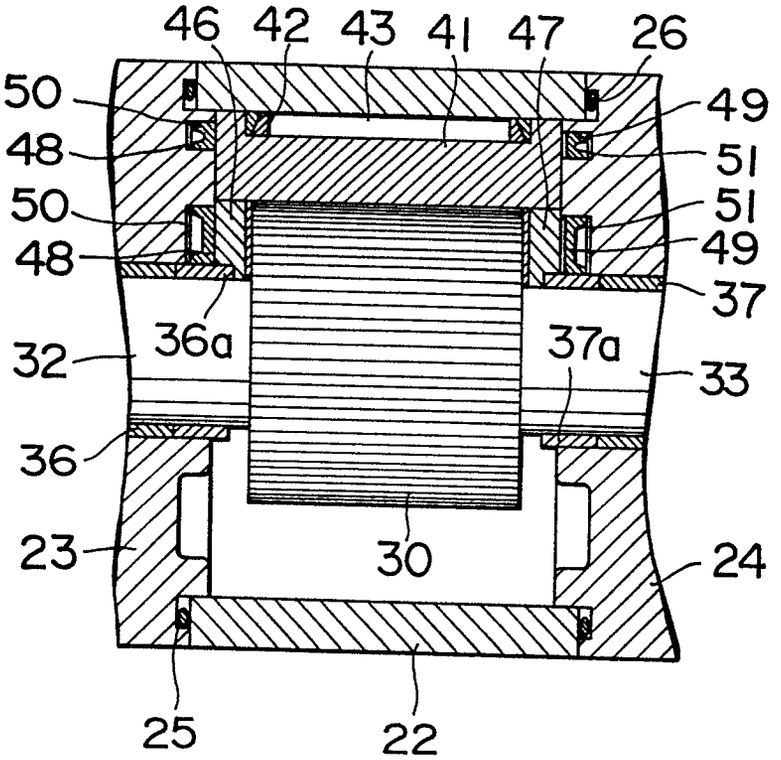
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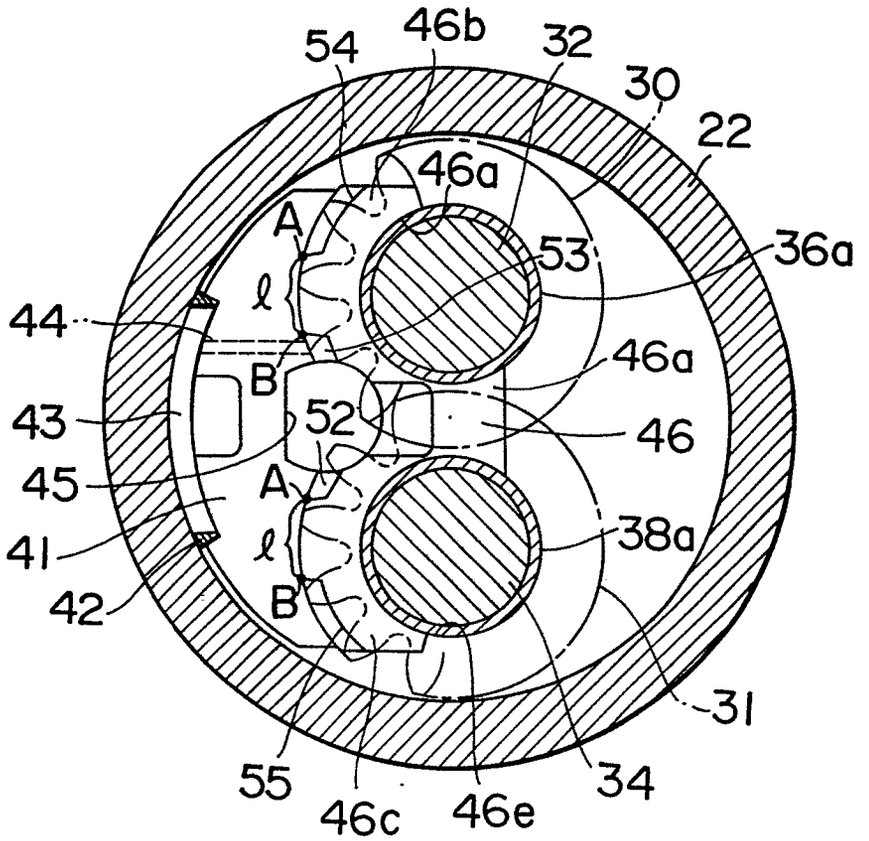
FIG.5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31
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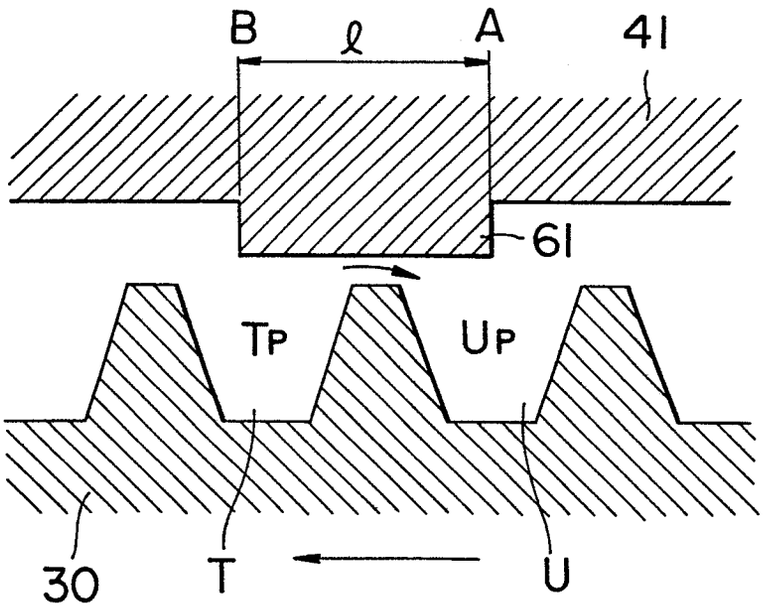
46c
46e
34

FIG.6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EUROPEAN SEARCH REPORT

0081293
Application number

EP 82305755.9

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. ³)
A	US - A - 2 993 450 (WEIGERT) * Fig. 1-4 * --	1,3	F 04 C 2/08 F 04 C 15/00
A	GB - A - 1 311 868 (CLAYTON DEWANDRE CO. LTD.) * Totality; especially page 2, lines 31-34 * --	1	
A	GB - A - 1 198 805 (THE PLESSEY CO. LTD.) * Fig. 1-3,5; claim 2 * --	1	
A	US - A - 3 381 619 (MINGOT) * Totality * ----	1	

TECHNICAL FIELDS SEARCHED (Int. Cl. ³)

F 04 C 2/00
F 04 C 15/00

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
VIENNA	18-03-1983	WITTMANN

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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