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(54) **Pumping unit for rotary vane pump**

(57) Pumping unit (10) for rotary vane pumps, in particular for oil pumps, comprising a plurality of vanes (21,22,23) and means adapted to activate their movement. The pumping unit (10) comprises a rotor (16) and a stator (12) having a profile (12a) for vane sliding motion. Each vane (21,22,23) is inserted into a respective diametric opening (16b) provided in the rotor (16) so that the rotor (16) rotation generates the motion of the vanes with the ends (21a,22a,23a) of each vane(21,22,23) that remain in constant contact with the internal profile (12a) of the stator (12), thus defining a desmodromic translation movement of the vane (21,22,23) inside the diametric openings 16b) of the rotor (16). The number Np of the vanes is such that $Np \geq 3$, these vanes generating a number Nv of compartments (17a) so that $Nv = 2Np$.

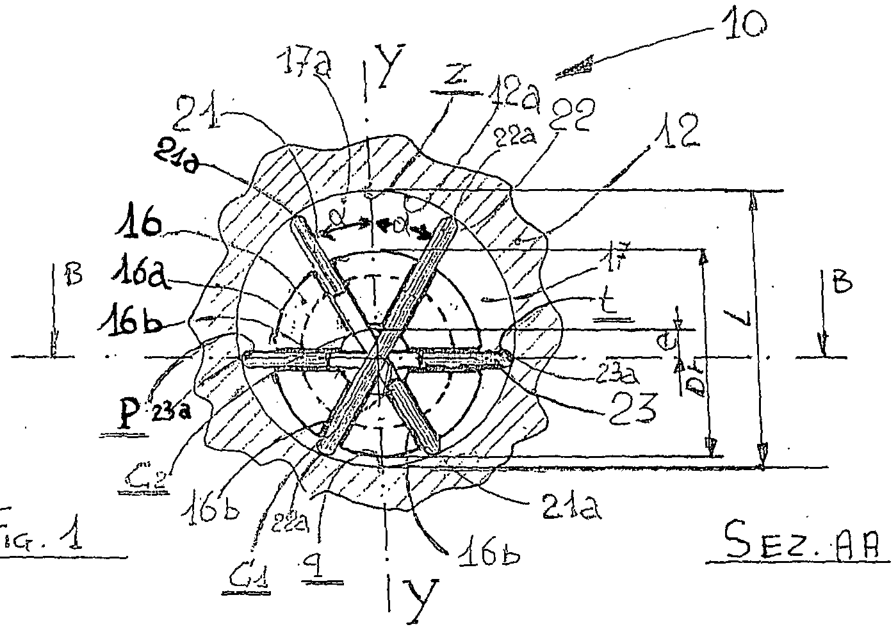

Fig. 1

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Description

[0001] The present invention relates to a pumping unit for a variable displacement vane pump.

[0002] In particular, the pumping unit of the present invention comprises a plurality of vanes, a rotor and a stator having an internal profile for vane sliding action, and is specifically provided for an oil pump for use on variable displacement radial vane vehicle pumps.

[0003] Throughout the following description, the pump as a whole will not be described, but exclusively the pumping unit: vanes, rotor and internal profile of the stator, and any elements that may replace the respective components generally present on all current radial vane pumps.

[0004] One of the main problems of radial vane pumps relates to the vanes which should guarantee constant contact with the internal surface of the stator under all working conditions, and therefore ultimately the hermetic sealing between the various pump areas, and especially the seal between the oil intake and the pressure delivery sections.

[0005] For this purpose, in radial vane pumps for vehicle use, the vanes are fixed, so that they can move in a desmodromic manner, by rings or possibly a piston that are floating and practically concentric with the stator ring, and on which the sides of the vanes opposite those that slide on the stator ring rest. Thus the vanes, mechanically constrained by both the rings/piston and the internal profile of the stator, can move desmodromically inside their own drive cavities present in the rotor.

[0006] A clear illustration of this type of pump is shown in the patent U.S. 4,702,083.

[0007] In fact, in pumps without rings/piston, under low revolution rates and under low temperature conditions, the vanes, due to the lack of the action exercised by the centrifugal force and because of the sticking effect with their respective seat in the rotor caused by high oil viscosity, tend to remain stationary and completely inserted in their cavities, consequently annulling the specific pumping function of the pump. The rings, or possibly the piston, solve the problems described above by imposing the translation motion of the vanes against any hindrance to movement. But, because the rings or piston act against any type of action that attempts to prevent vane translation inside the cavities, said rings or piston in turn become a critical aspect for the pump. In fact, any action that prevents vane translation (such as grit or excessive friction between the vane and its seat) generates forces between vane and ring that are discharged, generally amplified by the ring in question, to the other vanes (due to the fact that the vanes are activated in directions that differ from that which generates the overload action) and thus to the sliding surface of the stator, creating wear both between the vane and the rings/piston as well as between the vane and the stator surfaces. However, the rings, or possibly the piston, do not prevent the vanes whose centre of gravity is at a certain distance from the centre of rotation of the rotor, also because of the space occupied by the rings, from exercising a strong contact pressure between the vane heads and the stator at high revolution speed. Such strong contact pressure is provoked by the centrifugal forces which are consequently also high. This leads to extreme wear on both vanes and stator.

[0008] A further problem often encountered in radial vane pumps due to the use of rings/piston is that the rings/piston occupy radial space in the pump and therefore prevent pump use for certain applications where an extreme compactness in the direction of the diameter is required.

[0009] Therefore the object of the present invention is to provide a pumping unit, in particular for oil pumping and for vehicle application, free of the drawbacks described previously.

[0010] This object is achieved with a rotary multiple vane pumping unit having the features claimed in claim 1.

[0011] Therefore, a pumping unit has been conceived in order to eliminate the aforesaid problems wherein the drawbacks connected with wear on the vane and stator profile caused by vane friction have been basically eliminated. In particular, in the present invention, there is a considerable reduction in the centrifugal force and force of inertia acting on the vanes and causing these problems. In fact, thanks to the fact that the vanes of the unit of the present invention cross the rotor diametrically, and both ends of each vane are simultaneously in contact with the stator profile, their centre of gravity is located very close to the rotor centre of rotation, and consequently centrifugal action and low level inertia are generated, these being proportional to the distance between the centre of gravity and the centre of rotation. Furthermore, the present invention eliminates the need to use rings/piston, and consequently also all problems involving wear and failure; in fact, the desmodromic function performed by the rings/piston in traditional radial vane pumps is now intrinsically obtained through the combination of diametric vanes and stator profile. The constraint of maintaining the two ends of the vane simultaneously in contact with the stator profile allows during the rotor rotation a desmodromic motion to be performed by the translation of the vane inside its cavity on the rotor. The absence of rings which have to be housed within the rotor contributes towards reducing diametral rotor size considerably and consequently, this means that the diameter size of the pump which contains this type of pumping unit is also reduced to a large extent.

[0012] The present invention will now be described with reference to the appended figures which illustrate a non-limiting embodiment wherein:

- Figure 1 shows a cross-section A-A of the pumping unit according to the present invention;
- Figure 2 shows a longitudinal section B-B taken on the cross-section of figures 1;

- Figure 3 shows the view of a first possible embodiment of the vanes on the pumping unit according to the present invention;

- Figure 4 shows the view of a second possible embodiment of the vanes on the pumping unit according to the present invention;

- Figure 5 shows the view of a third possible embodiment of the vanes on the pumping unit according to the present invention.

[0013] In the appended figures the numeral 10 is used to indicate the pumping unit of the present invention.

[0014] This pumping unit 10 comprises a stator 12 inside which a cavity (figures 1, 2) having a surface with a profile 12a is formed. In turn the profile 12a is defined by two profile portions: a first portion belonging to the sector defined by the points p, q, t, having any possible continuous decreasing curve up point q and increasing curve up to point t, among which also the semicircular curve shown in figures 1 and 2, and a second portion belonging to the sector defined by points t, z, p having a non-semicircular curve; aim and shape of this geometry will be described more clearly further on.

[0015] A rotor 16 is housed inside the stator cavity 12. The external profile 16s of rotor 16, together with the internal profile 12a of stator 12, define a pumping space 17. The rotor 16 is driven in rotation by a motor shaft 16d integral with the rotor 16, partially shown in figure 2, and supported inside the pump by means that are not illustrated, but which operate on said shaft 16d and on an upper portion 16c of the shaft 16d which is always integral with the rotor 16. The shaft 16d rotates around a Z-Z axis (centre C1 in figure 1).

[0016] The rotor 16 and the portion 16c of the shaft 16d have a plurality of diametric openings 16b, which are three in number in the embodiment illustrated, but not necessarily limited to the number in this example. A vane 21, 22, and 23 passes through each one of these diametric openings 16b. As to the geometry of the vanes 21, 22, 23, see figure 3 or the equivalent vanes shown in figures 4-5, identified respectively by numerals 31, 32, 33 and 41, 42 43.

[0017] Each of said vanes has a geometrical conformation conceived to permit a simultaneous free crossing action through centre C1 of rotor 16. More particularly, the vanes 21, 22, 23 in figure 3, and those in figures 4-5, have central openings 21b, 22b, 23b that form connecting branches between the two symmetrical parts of the vane, each one having a height of H/3, more generally, if the number of vanes is represented by Np, and the height by H, the height of the lowered branches will be H/Np.

[0018] The vanes 21, 22, 23 (hereafter only these vanes will be referred to in the description since the same conditions apply throughout to the vanes in figures 4-5 in the same way) are inserted by vertical mounting only in the diametric openings 16b of rotor 16, and at the same time in the upper appendage 16c of the shaft 16d in the same sequence shown in figure 3, in such a manner that branch H/3 of the vane 22 is positioned centrally in relation to the branches H/3 of the vanes 21, 23.

[0019] In figure 2 it is shown that the sectors of the portion 16c of shaft 16d forming a single piece with rotor 16 are centrally threaded so that they can be pushed by means of a conical screw 19 onto a reinforcement ring 18. This solution creates a continuous surface on the portion 16c of shaft 16d because of the presence of the ring 18, this surface being conceived to cooperate successively with a bushing, thus making the rotor 16 more rigid and strong. This also permits vane disassembly and reassembly.

[0020] With reference to figure 1, those features of the profile 12a of the stator cavity 12 such that during rotor rotation the vanes 21, 22, 23 which perform a sliding action inside their seats 16b maintain the two ends 21a 22a 23a of each vane constantly in contact with the points of the stator surface 12a are described.

[0021] To this end, the profile 12a is realized in such a manner that all the points of the profile of the sector t, z, p are generated from the ends 21a 22a 23a of any one of the vanes when the corresponding opposite end slides on the generator profile of sector p, q, t moving from p to t. Among all the infinite number of curve profiles available, this profile can also be an arc of circumference tangent to rotor 16 in point q. In this way, the distance of points p, t and q, z of the profile 12a will be the same as the length L of the vane. Therefore during the rotor rotation the end of each vane will be constrained to remain in constant contact with the profile 12a, thus forcing the vane in question to slide inside the opening 16b with a translation movement which can be defined as desmodromic. According to the generation of the profile described above, it is possible to define also for the pumping unit 10 an eccentricity e between rotor and stator as the distance between the rotor centre C1 and the stator centre C2, defined as that point, distant from C1 as half the difference between the vane length L and the diameter of the rotor Dr.

[0022] From this description it is apparent that during rotor rotation, the centre of gravity G of each vane will rotate around the centre C1 maintaining a distance that is equal to or less than the eccentricity e, this condition being extremely advantageous for the force of inertia and centrifugal force that operate on the vane, since these are in proportion with the distance between G and C1 (in fact in traditional floating radial vane pumps it is necessary to add Dr/2 to this distance).

[0023] According to the consideration provided above it is obvious that when comparing pumping units having diametric vanes of the type according to the present invention with pumping units of the same size with radial vanes, in the pumping units of the present invention the centrifugal force and the force of inertia are reduced by about 1/4, when the pumps rotate at the same number of revolutions per minute. Moreover, considering the fact that in pumping units according to the invention the vanes 21, 22, 23 rest on the two end points of the rotor cavity 16b and are no longer fitted in a floating condition at one end, the thickness s of the vanes can be reduced to approximately half the size.

[0024] The profile 12a conceived as previously described is extremely advantageous when the pumping unit 10 is used also for variable displacement pumps. It is common knowledge that there is a variation in displacement according to the variation in eccentricity e and that displacement will be zero when the stator centre C2 is set over the position of the rotor centre C1. In order to maintain efficient the pumping function for any eccentricity value, the compartment 17a set over the Y-Y axis that separates the intake area from the pumping unit delivery must be hermetically sealed, that is, the clearance between the head of the vanes 21 and 22 and the profile 12a must be very small and less than 5/100 of millimetres, this being a value for which the fluid dynamics function of the pumping unit, even when the oil temperature is very high, is still guarantee. With maximum eccentricity, that is when the distance between C1 and C2 is equal to e, the clearance will always be zero, and this is obtained according to the way in which the profile 12a has been constructed, while for distances C1-C2 less than e increased clearance is generated according to the reduction in working eccentricity, but in any case, decreasing as the vanes 21, 22, 23 progressively approach the Y-Y axis until this clearance is annulled with any eccentricity value when the vanes are set over the Y-Y axis. In fact, in this position, the distance of points q, z belonging to profile 12a is equal to the length L of the vanes.

[0025] In the pumping unit 10, as shown in figure 1 illustrating the non-limiting example of three vanes (Np=3) that generate six compartments (Nv=6), more generally the number of compartments equals doubles the number of vanes (Nv=2Np). Advantageously, compared to current radial vane pumps where Np=Nv, the angular distance between the vanes of each compartment is $2\alpha=360/2Np$, which is $2\alpha=60°$ where Np=3, resulting in $\alpha=30°$. This means that the vanes 21, 22 set on Y-Y axis, being not very distant from the axis, will have very little clearance with respect to profile 12a.

[0026] From calculations based on the construction geometry of profile 12a, it is possible to estimate the clearance of the vanes set on Y-Y axis that form compartment 17a according to the ratio e/L (vane eccentricity/length) and it is possible to verify that for $e/L\leq 0.08$ and with $Np\geq 3$, when the working eccentricity is zero, these are comprised between 5/100 and 2/100 of millimetres, and the clearance values are even smaller, for eccentricity between zero and e. It should be remembered that in most applications for which this invention is intended, the e/L is less than 0.08.

[0027] The ratio $e/L>0.08$ could generate in compartment 17a clearance between vanes and stator profile higher than those necessary for a good volumetric performance of the pumping unit 10.

[0028] In relation to this aspect, figure 4 shows a set of vanes 31, 32, 33 geometrically similar to those in figure 3 but having the prerogative of eliminating any clearance between vanes and the stator profile for any work eccentricity value of pumping unit 10 and in any angular position the vanes may assume in relation to the Y-Y axis.

[0029] In more detail, the vanes 31, 32, 33 are each composed of three overlaid layers having a thickness of s/3 (where s is the thickness of the vane). The two external layers 31a, 32a, 33a which are identical on all vanes, are geometrically configured like the vanes 21, 22, 23 but are slightly shorter by approximately $0.2 \div 0.3$ mm, while the internal layer contained between the previous two layers is composed of three elements: the two end sliding shoes 35 which are conceived to recover, when subjected to the centrifugal force, the clearance between vanes and stator profile and the central element 31b 32b, 33b that acts as a spacer for the sliding shoe and that together with the sliding shoe has a geometrical configuration equal to the vanes 21, 22, 23 and with the same length L. As shown in greater detail in figure 4, the central body 31b, 32b, 33b of each vane has two rectangular openings, indicated with 35c, where the sliding shoes 35 are positioned by insertion of the respective rectangular projections 35a. In this way, the sliding shoes are bound axially to the central body 31b, 32b, 33b of the vanes, being free to recover the radial clearance due to their sliding motion within the two container layers 31a, 32a, 33a of each vane.

[0030] The solution with layer vanes allows using steel for the external layers and plastic materials for the central element, and different suitable plastic materials for the spacer element and for the sliding shoes.

[0031] Figure 5 shows another embodiment of the vanes 41, 42, 43 wherein, compared to the vanes 21, 22, 23 in figure 3, vertical projections 45 having a semicircular shape are provided protruding from opposite sides of the vane; the generatrices 45b of the highest points of these projections become the supporting and sliding points for the vanes inside the diametric openings 16b on the rotor. This vane embodiment provides the transformation of the type of contact of the prismatic couple defined by the vane with its own seat 16b, from surface to surface contact to the contact between a surface and a line 45b. This specific line-surface coupling, which forms a cavity between the vane and hollow 16b, is capable of collecting small impurities that may be present in the machine oil without causing malfunction which prevents free vane sliding in their seats, as would, occur in the case of surface to surface contact.

[0032] The advantages of the present invention are apparent from the previous descriptions. In particular:

- the pumping unit has intrinsically desmodromic vanes, and therefore there is no need for rings or pistons to be used for this function, thus eliminating a component that can cause very critical problems for pump reliability;

- the diametric vanes locate the centre of gravity of each vane close to the centre of rotor rotation, in this way drastically reducing the centrifugal force and force of inertia and thus making it possible to use the pumping unit at high rpm;

- the number of the pump compartments is double the number of the vanes, thus reducing the costs with respect to standard pumps, where the number of compartments and vanes are the same;

- the communicating compartments between oil intake and oil delivery, that is, those set on the Y-Y axis, provoke very slight, almost negligible clearance between vanes and profile, and this maintains the pumping unit very efficient as far as volumetric performance is concerned, and for any eccentricity value under which the pumping unit works;

- the absence of the rings or pistons inside the rotor, used as spacers for the vanes, reduces the transversal size of the pump drastically;

- the possibility of using vanes configured in order to create a cavity between the vanes and their seats in the rotor, and therefore conceived to collect solid dirt in the oil without blockage occurring;

- the possibility of creating small diameter rotors, which are extremely advantageous for their low cost construction in a single piece with their own driving shaft.

Claims

1. Pumping unit (10) for rotary vane pumps, comprising a stator (12), a rotor (16) and a plurality of vanes (21,22,23,31,32,33,41,42,43) driven in rotation by said rotor (16) and defining between one another a plurality of pumping compartments (17a), the vanes of said plurality of vanes (21,22,23,31,32,33, 41,42,43) each having a set of openings (21b,22b, 23b) in a central position thereof, and being able to cross the rotor (16) diametrically, reciprocally intersecting one another freely in the centre (C1) of said rotor (16), each vane (21,22,23,31,32,33,41,42,43) being inserted in a respective diametric opening (16b) provided in said rotor (16) in such a manner that the rotation of said rotor (16) generates the motion of said vanes (21,22,23,31,32,33,41,42,43) with the two ends (21a,22a,23a) of each vane remaining constantly in contact with a profile (12a) inside the stator (12), **characterised in that** said plurality of vanes (21,22,23,31,32,33,41,42,43) can generally be composed of a number Np of vanes such that Np≥3, these vanes generating a number Nv of compartments (17a) such that Nv=2Np.

2. Pumping unit (10) according to claim 1, wherein a portion (t,z,p) of said profile (12a) is generated by any one of said ends (21a,22a,23a) of the vanes when the other corresponding opposite end describes a semicircular curve (p,q,t) which is tangent at point (q) on said rotor (16), determining a stator centre (C2) as that point with a distance from the centre (C1) of said rotor (16) which measures half the difference between the length (L) of the vanes and the diameter (Dr) of said rotor (16), and therefore eccentric from (C1) by an amount e=C1-C2.

3. Pumping unit (10) according to claim 1 or 2, wherein said rotor comprises an external profile (16a) and wherein the vanes (21,22) which, together with said profile (12a) of the stator (12) and the profile (16a) of the rotor (16), define a communicating compartment (17a) between the oil intake and the oil delivery, have a clearance (C1-C2*) with said profile (12a) of the stator (12) between zero and a maximum of 0.05 for any working eccentricity value (e), with 0≤(C1-C2*)≤e when Np≥3 and when the ratio between maximum eccentricity (e) and vane length (L) is ≤0.08.

4. Pumping unit (10) according to any one of the previous claims, wherein said vanes (21,22,23) are lowered centrally by means of said openings (21b,22b, 23b) and in the lowered zones have a height equal to H/3, where H is the vane height.

5. Pumping unit (10) according to claim 4, wherein at least two of said vanes (21,23) have the same shape and at least one other vane (22) has a shape that is different from the previous vanes.

6. Pumping unit (10) according to claim 5, wherein if Np is the number of the vanes, the lowered height is equal to H/Np and the number of vanes having a different shape is Np/2 for even Np and (Np-1)/2)+1 for uneven Np.

7. Pumping unit (10) according to any one of the claims from 3 to 6, wherein said vanes (31,32,33) are capable of recovering the clearance between the vane ends and said profile (12a) of the stator (12) when e/L>0.08 and wherein inside each of said vanes (31,32,33), contained between two layers (31a,32a, 33a) of each vane (31,32,33) each one having a thickness of s/3, two sliding shoes (35) are housed, in an axial position with respect to the vanes (31,32,33) by means of projections (35a) that are housed in respective seats (35c) of the central body (31b,32b,33b) of the vanes (31,32,33), said sliding shoe (35) being able to slide in a radial direction with respect to said central body (31b,32b,33b), this being also able to slide with respect to said two vane layers (31a,32a,33a).

8. Pumping unit (10) according to any one of the claims from 1 to 6, wherein said vanes (41,42,43) have a pair of specifically shaped protruding parts (45) with generatrices (45b) which rest on the respective diametric openings (16b) of said rotor (16).

9. Pumping unit (10) according to any one of the previous claims, wherein said rotor (16) comprises a first portion (16d) on one side that acts as a driving shaft, integral with said rotor (16), and on the opposite side, once again integral with said rotor (16), a second portion (16c) configured in sectors defined by the openings (16b) of said rotor (16)., inside which is housed a threaded conical cap (19) conceived to push said sectors of said second portion (16c) onto a reinforcing ring (18) adapted to cooperate with a bushing.

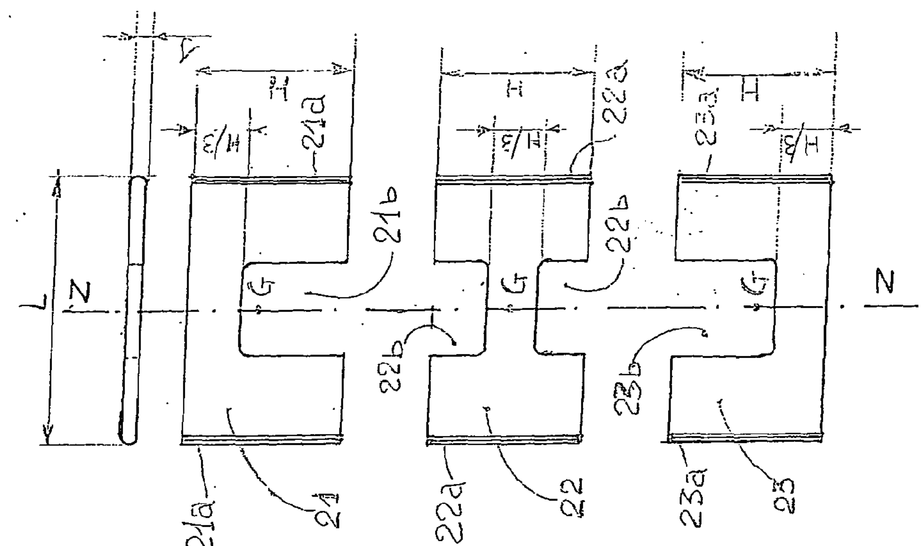

Fig. 3

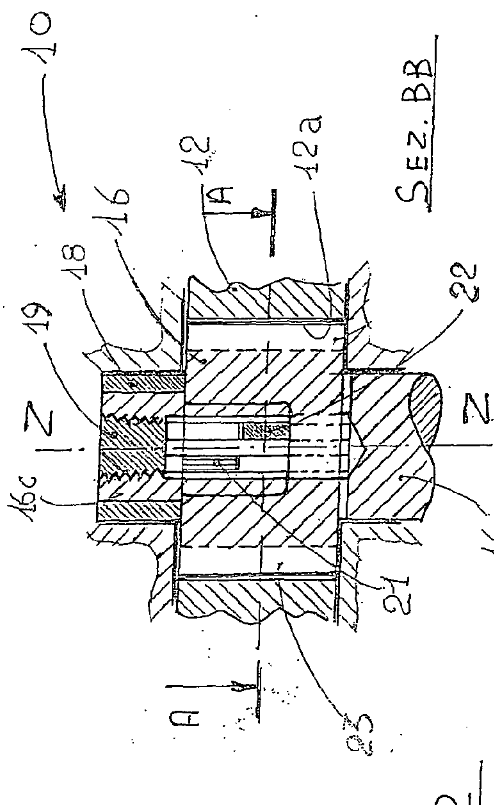

Fig. 2

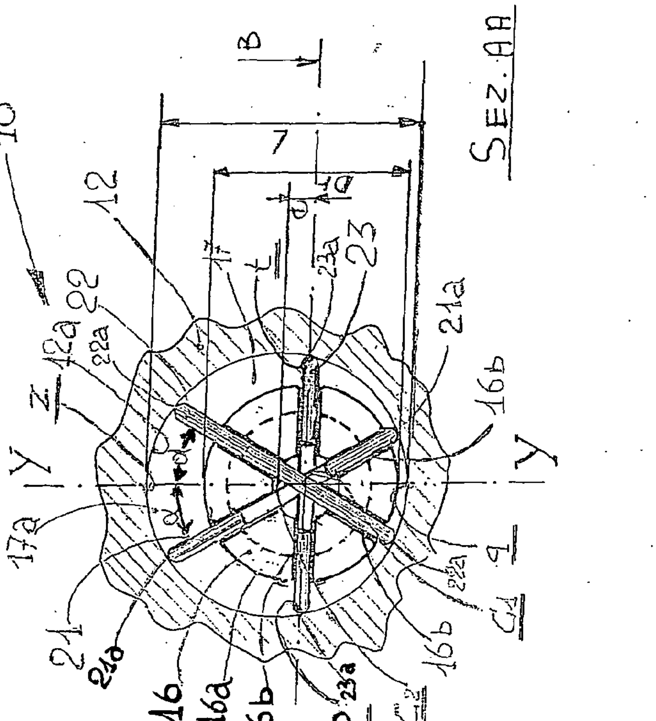

Fig. 1

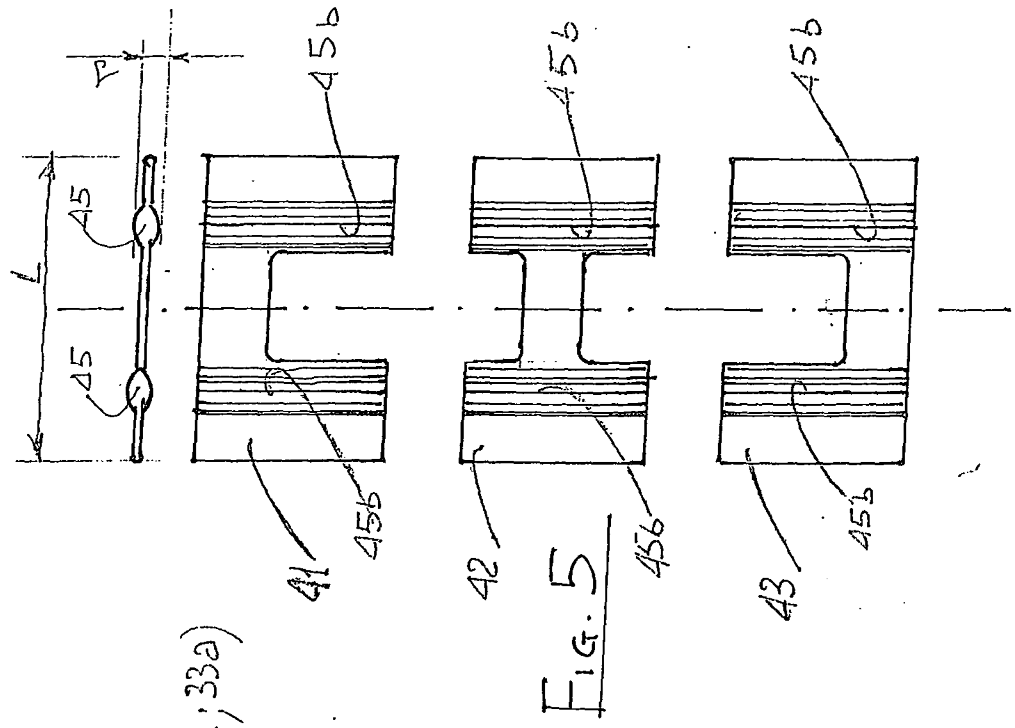

Fig. 5

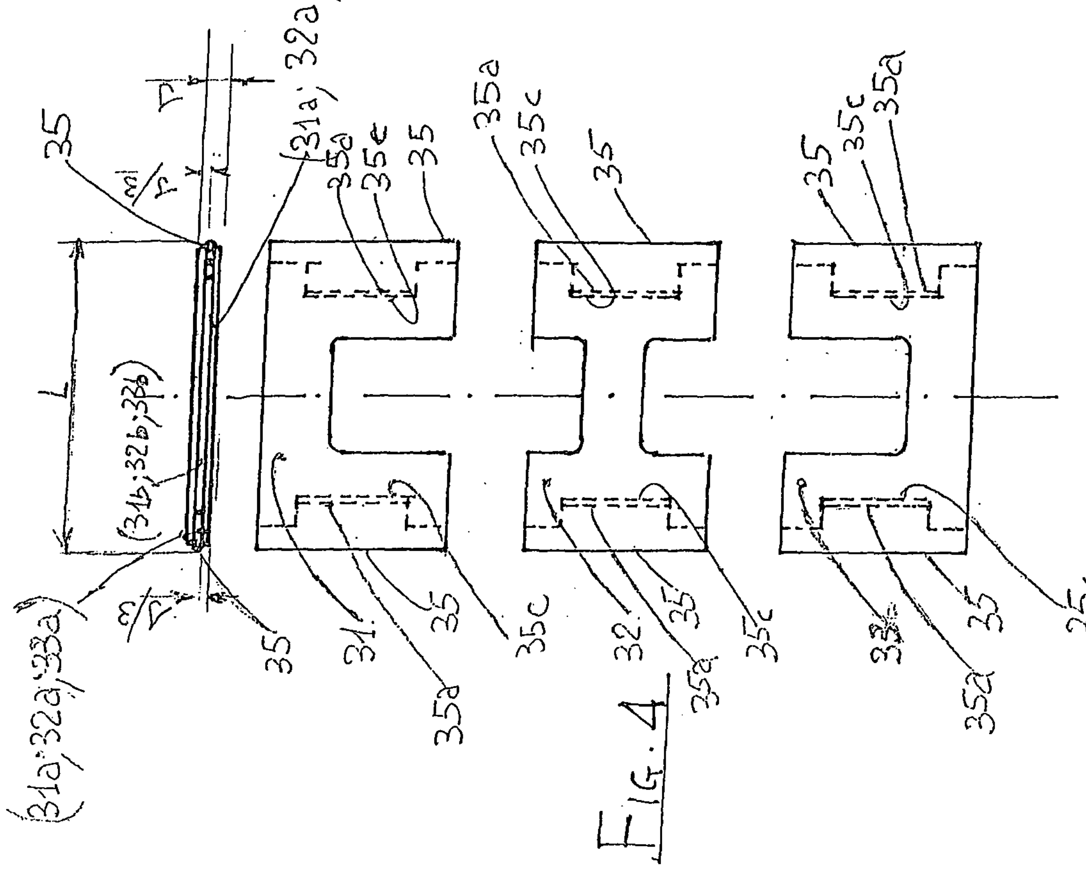

Fig. 4

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EUROPEAN SEARCH REPORT

Application Number
EP 07 42 5310

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The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
The Hague	10 April 2008	Biloen, David

CATEGORY OF CITED DOCUMENTS

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EPO FORM 1503 03.82 (P04C01)

8

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 07 42 5310

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REFERENCES CITED IN THE DESCRIPTION

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