United States Patent

Pruvot

[15] **3,661,055**

[45] **May 9, 1972**

[54] **MULTI-CYLINDER BARREL HYDRAULIC PUMPS OR MOTORS**

[72] Inventor: **Francois C. Pruvot,** Billancourt, France

[73] Assignees: **Regie Nationale Des Usines Renault,** Billancourt; **Automobiles Peugeot,** Paris, France

[22] Filed: **May 26, 1970**

[21] Appl. No.: **40,541**

[30] **Foreign Application Priority Data**

June 12, 1969 France 6919519

[52] **U.S. Cl.** **91/486,** 91/505
[51] **Int. Cl.** **F01b 3/00**
[58] **Field of Search** 91/486–489, 504–507

[56] **References Cited**

UNITED STATES PATENTS

2,298,850	10/1942	Vickers	91/487
2,455,330	11/1948	Dennison	91/486
3,208,395	9/1965	Budzich	91/489
2,871,798	2/1959	Thoma	91/489
2,155,455	4/1939	Thoma	91/489

FOREIGN PATENTS OR APPLICATIONS

883,348	6/1942	France	91/507

Primary Examiner—William L. Freeh
Attorney—Stevens, Davis, Miller & Mosher

[57] **ABSTRACT**

Multi-cylinder barrel pump or motor of which the capacity is variable by controlling the displacement of the barrel in a support adapted to be inclined more or less in the pump casing about a pivot axis perpendicular to the axis of rotation of said barrel, the piston of the pump cylinders being provided with ball-shaped ends for connecting same to an impeller plate rotatably solid, together with said barrel, with the pump driving shaft, wherein said impeller plate is balanced in the axial direction by a hydrostatic bearing in alignment with the bearing slide face of said plate in the pump casing, and having a port corresponding to the delivery port of said distributor slide face and adapted to be connected to the delivery side of the pump through the medium of a check and balancing valve.

5 Claims, 7 Drawing Figures

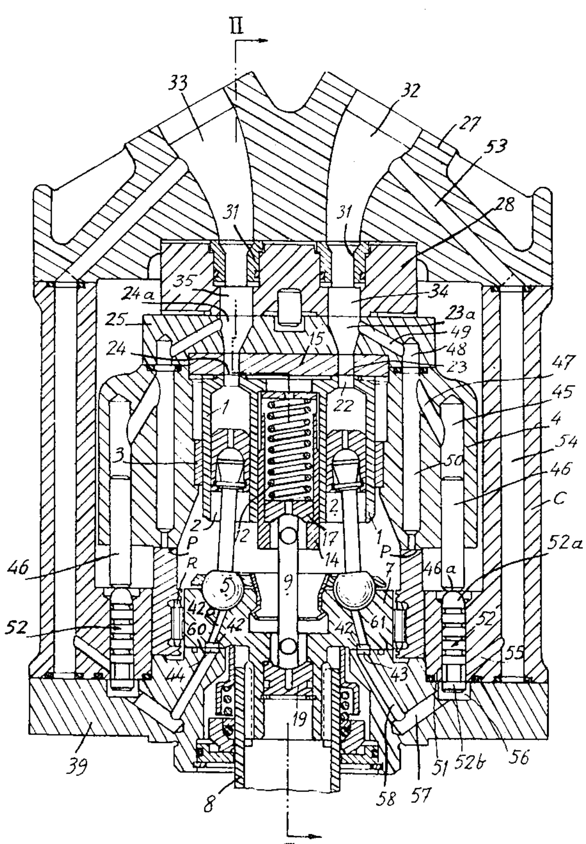


FIG.1

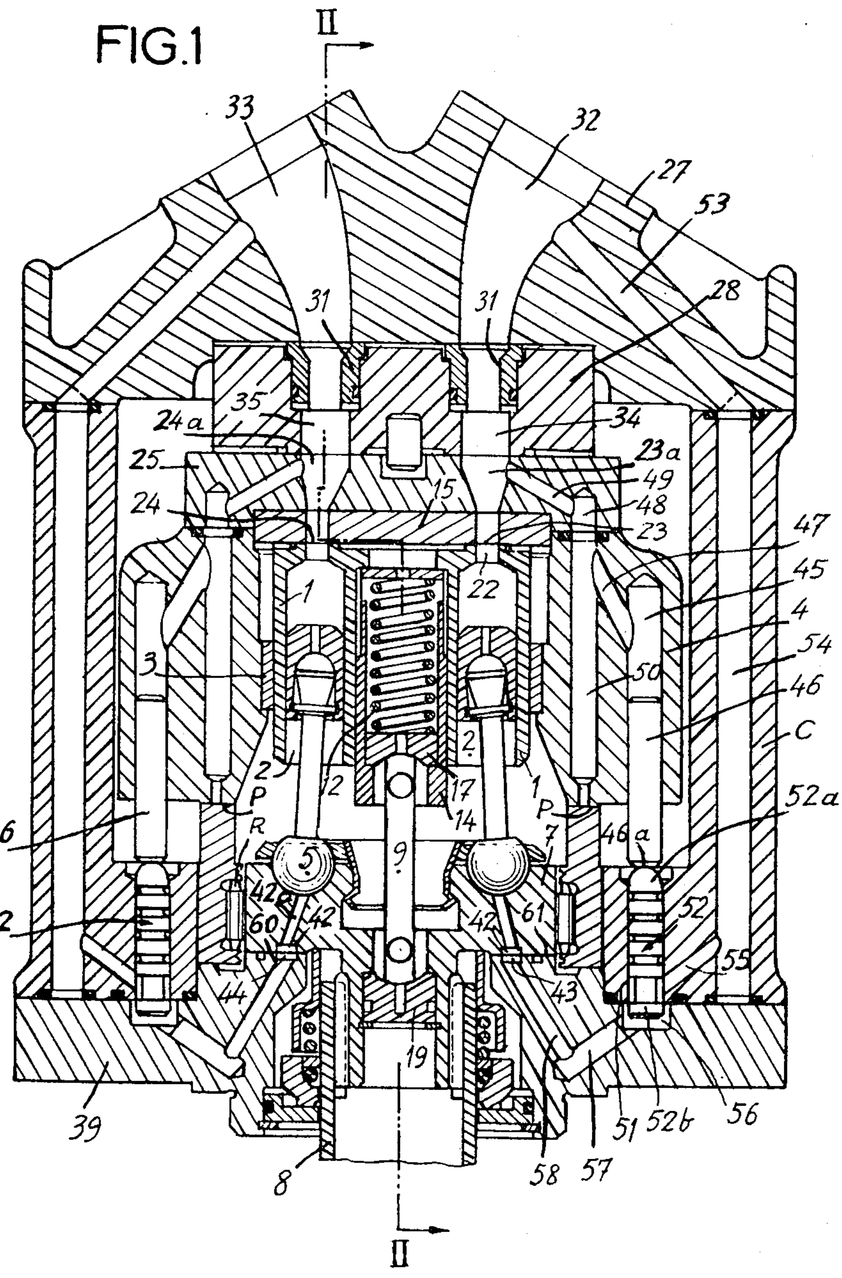

II
33
32
27
53
28
31
31
35
24a
34
23a
25
49
15
48
24
23
47
1
22
45
4
54
3
50
46
C
2
2
12
17
1
P
52a
46
P
R
14
9
7
46a
5
42a
42
42
61
52
52
60
55
44
43
19
39
51
52b
56
8
58
57
II

FIG. 2

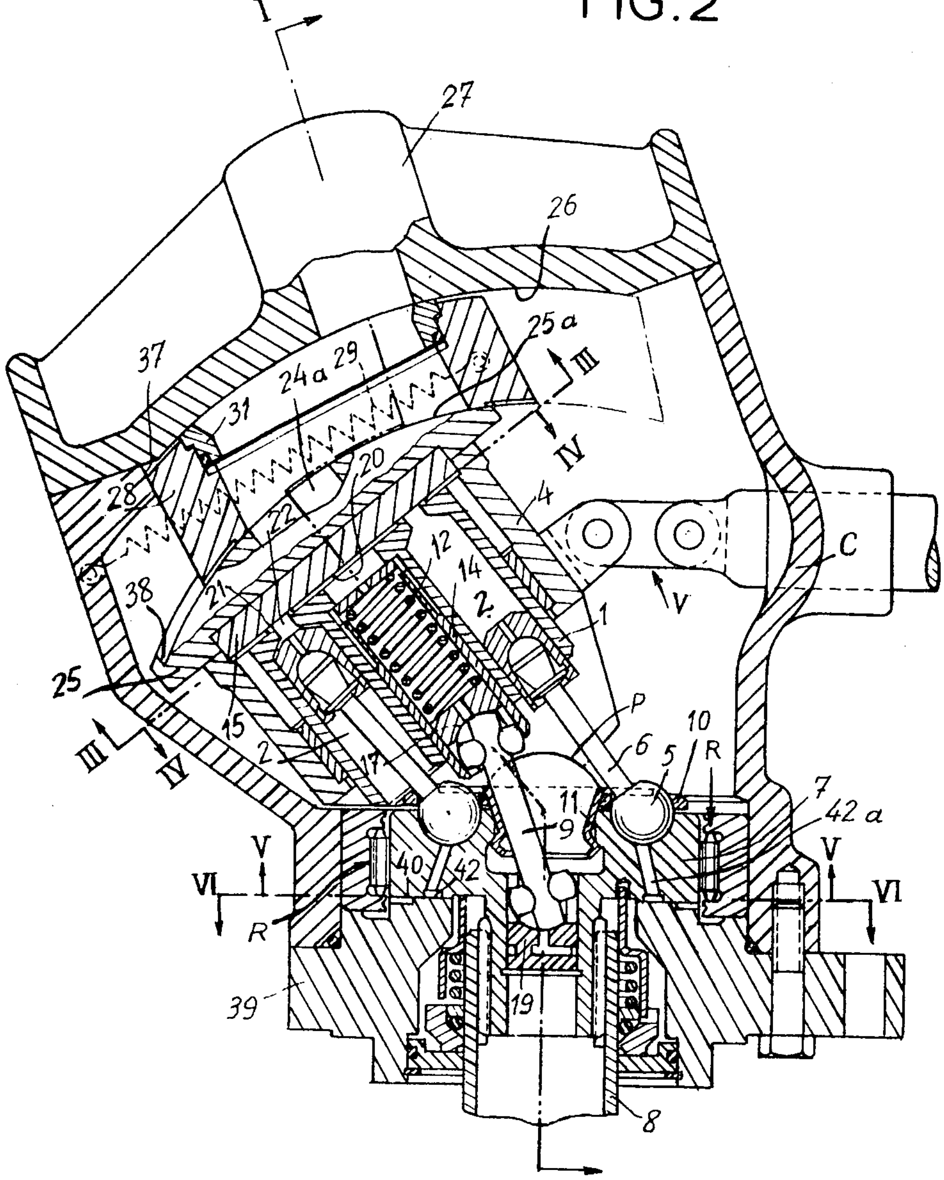

I
27
26
25a
III
IV
37
24a
29
31
28
20
22
12
14
4
2
1
C
V
21
38
15
25
III
IV
2
17
P
6
10
5
R
11
9
7
42a
V
VI
40
42
V
VI
R
39
19
8
I

FIG. 3

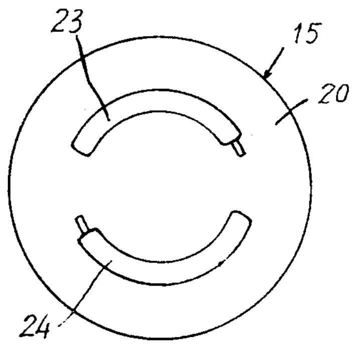


FIG. 4

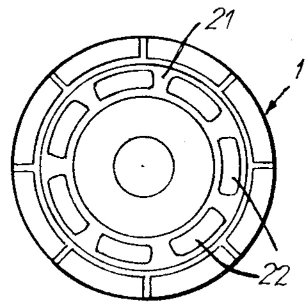


FIG. 5

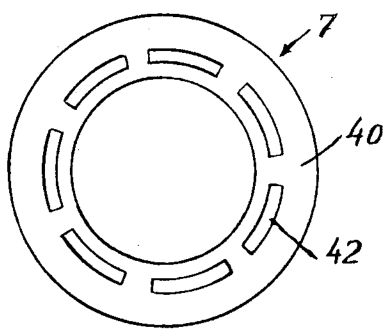


FIG. 6

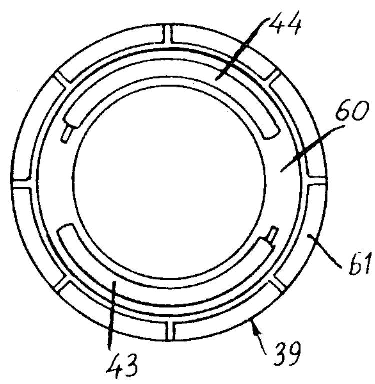
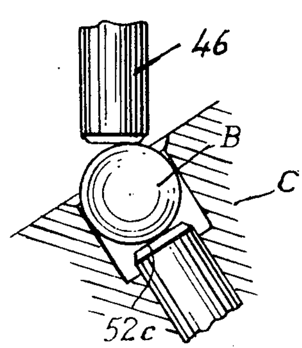


FIG. 7



MULTI-CYLINDER BARREL HYDRAULIC PUMPS OR MOTORS

The present invention relates to hydraulic motors or pumps of the multi-cylinder barrel type having a variable capacity.

To simplify the disclosure, only pumps will be referred to in the following description, but it will be readily understood that the data concerning hydraulic pumps as given herein apply as well to hydraulic motors.

This invention is directed more particularly to improvements in multi-cylinder barrel pumps of the type broadly mentioned hereinabove, wherein a change in the cylinder capacity is obtained by modifying the angle formed between the barrel axis and the axis of the pump driving shaft, a zero angle corresponding to a zero output or cylinder capacity.

Due to the particular mode of construction of pumps of this type, the impeller or swash plate rigid with the driving shaft receives the resultant thrust from the pistons which comprises a radial component and an axial component.

As a rule the radial effort is absorbed by one or a plurality of roller or ball-bearings, although pumps are already known wherein plain bearings, hydrodynamic or hydrostatic bearings have also been used.

The axial effort is absorbed in most cases by ball, needle or roller bearings. In other, more seldom encountered cases, this effort is absorbed by plain bearings, or hydrodynamic or hydrostatic bearings.

In another pump type the plate is not connected directly to the driving shaft and the connecting rods extend at right angles thereto.

In this known arrangement the radial efforts are absorbed by the barrel in which the cylinders are formed, and relatively very moderate radial efforts are exerted on the swash plate, so that the use of rolling-contact bearings can be dispensed with for absorbing these forces.

In the above-mentioned constructional examples it is clear that the radial and axial efforts applied to the impeller plate vary as a function of the "breaking" angle of the pump, even under constant-pressure conditions.

However, in broken-axis pumps wherein the impeller swash plate is connected directly to the pump shaft, the axial effort supported by the multi-barrel body is constant irrespective of the breaking angle since it is dependent only on the assumedly constant pressure, and on the number and diameter of the multiple pistons.

This made it possible to balance the stresses supported by the barrel (at least as far as axial stresses are concerned) by using relatively simple hydrostatic bearings. As a rule, it is the outline of the high-pressure port formed in the distributor plate that is shaped to balance at any time the axial forces produced in the pump cylinder by the fluid pressure. A slight degree of unbalance preserved therein is subordinate to the magnitude of the permissible leakages.

This arrangement is ill adapted to the impeller or swash plate of the pump for the axial force received by this plate varies substantially as the cosine of the breaking angle. Now the necessity of combining a low cost with a moderate weight and a high efficiency leads to more and more increasing of the breaking angle. Thus, in certain recently constructed pumps a breaking angle of 45° has been adopted which gives an axial thrust on the plate that varies in proportion to the cosine of an angle of 0° to 45° (1 and 0.707).

Under these conditions it is obvious that the force necessary for hydrostatically balancing the impeller plate cannot exceed 0.707 times the sum of the thrust exerted by the pistons (about 0.65 times this thrust since a certain residual pressure must constantly be maintained to keep the working faces in mutual contact). The axial bearing should then be capable of absorbing an additional effort representing 0.35 times the piston thrust when the pump capacity is zero. Of course, this is true only for a breaking angle of 45°.

Modern high-efficiency multi-cylinder barrel pumps cannot be provided, due to their moderate dimensions, with a swash plate bearing for axially balancing the barrels. The state of unbalance between the bearing forces and the axial reaction would actually develop excessive friction and therefore cause a rapid wear of the parts in mutual contact. This inconvenience is particularly serious in the case of a hydraulic motor developing a maximum torque at low or zero speed, that is, when the coefficients of friction have their highest values.

The inconveniences of the above-mentioned balancing methods are well known in the art.

Thus, if rolling-contact bearings are used for absorbing the axial and radial forces involved, their weight and overall dimensions are considerable, and these components are very expensive.

An attempt was made to avoid these inconveniences by using roller thrust bearings but their insufficient characteristics limit the efficiency values, pressure values and useful life of the pumps.

In certain cases the impeller or swash plate is balanced by hydrostatic means, but to avoid the inconveniences described hereinabove only a moderate breaking angle was adopted, in order to keep its cosine close to 1.

In this case, one of the essential advantages characterizing the broken-axis pump, namely the small size of its component elements, is lost due to the wide breaking angle.

The present trend towards the use of increasingly higher pressure and speed values is observed in pumps and motors used in hydrostatic transmissions of vehicles, or in power control systems of public works equipments and vehicles such as hydraulic power shovels.

In this case the pumps must be driven directly from diesel engines capable of operating at increasingly higher speeds (e.g. 3,000 r.p.m. and more) and high pressures must be resorted to if it is desired to use actuators and pipe lines of reasonable size. But then it will be seen that the thrust bearings necessary for absorbing the axial stresses on the pump swash plates are not available commercially, and conversely the bearings having a sufficient capacity have a low operating speed, and vice versa. Therefore, pumps must be used wherein the axial forces are balanced hydrostatically, such as the pumps wherein the impeller plate is not coupled to the driving shaft. However, this type of pump is confronted with so-called "in line" pumps which are both simpler and smaller in size. As a consequence, broken-axis pumps are gradually abandoned except for applications requiring very high efficiencies, wherein their low weight and high efficiency compensate more or less the high cost of the bearings, although these pumps are seldom operated under very high pressures (their pressure is frequently limited to a maximum of 210 bars = 3,045 p.s.i.).

It is the essential object of the present invention to bring a satisfactory solution to the various problems set forth hereinabove by providing a variable-capacity multi-cylinder barrel pump or motor of the broken-axis type, wherein the impeller plate is balanced in the axial direction by a hydrostatic bearing of which the action varies with the pump operation.

More particularly, this invention is directed to a multi-cylinder barrel pump motor of which the capacity is varied by means of a controlled displacement of the barrel in a support adapted to be inclined in the pump body about a pivot axis perpendicular to the axis of rotation of said barrel, the pistons being provided with ball-headed rods connecting said pistons to an impeller plate rotatably solid, together with said barrel, with the pump driving shaft, said barrel comprising a slide face formed with ports communicating with apertures formed in the cylinder bottoms, said slide face being in rotary contact with the distributing slide face of a distributor plate having formed therethrough two fluid passages each opening into said distributing slide face by means of a crescent-shaped port communicating with suction and delivery ducts of the pump, characterized in that the impeller plate is balanced in the axial direction by a hydrostatic bearing comprising a port registering with the bearing face of said plate in the pump body and corresponding to the delivery port of said distributing slide face, and adapted to be connected to the delivery side of the pump, through the medium of a balancing valve comprising in the barrel support a control piston responsive to the pump delivery pressure and in the pump body a check valve controlling the communication between said port of said hydrostatic bearing and the pump delivery, said control piston and said check valve operating in mutual opposition whereby the balancing pressure applied to said hydrostatic bearing of said impeller plate varies as a function of the inclination of the barrel support during the pump operation, and therefore as a function of the axial stress to be compensated.

This structure is characterized by moderate overall dimensions and permits the construction of very high pressure and high speed pumps having a satisfactory useful life due to the combination of a hydrostatic bearing with the known advantageous features of broken-axis pumps which are essentially the low inertia of rotating parts and the moderate stress supported by moving parts.

Other features and advantages of this invention will appear as the following description proceeds with reference to the accompanying drawing illustrating diagrammatically by way of example a typical form of embodiment of a pump constructed according to the teachings of the invention. In the drawings:

FIG. 1 is a sectional view of the pump, the section being taken along the line I—I of FIG. 2;

FIG. 2 is another section taken along the line II—II of FIG. 1;

FIG. 3 is a view taken in the direction of the arrows III—III of FIG. 2 showing the distributor plate of the pump;

FIG. 4 is a similar view of the barrel, as seen in the direction of the arrows IV—IV of FIG. 2;

FIG. 5 is a similar view taken in the direction of the arrows V—V of FIG. 2;

FIG. 6 is a view taken in the direction of the arrows VI—VI of FIG. 2, and

FIG. 7 is a modified form of embodiment of a detail of the construction shown in FIG. 1.

As illustrated in the drawing the pump comprises a cylindrical rotary barrel in which a number of parallel cylinders 2 disposed on a common concentric circle are formed. This barrel 1 revolves in babbit or like bearings 3 rigid with a support 4. This support 4 is pivotally mounted in a pump body or outer casing C by means of a pair of pivot faces P formed on the outer race of a rolling-contact bearing R fitted in the casing C; thus, this support 4 can pivot under the action of a control device designated by the reference letter V (FIG. 2) about an axis coincident with the center of said pivot faces P and positioned in the plane containing the centers of the ball-shaped heads 5 of the rods of pistons 6, said ball-shaped heads being mounted in an impeller or swash plate 7 rotatably solid in turn through splines with the pump driving shaft 8. This shaft 8 drives the barrel 1 through the medium of a universal joint comprising a link 9 provided at its ends with driving studs adapted to slide in axial grooves formed on the one hand in said plate and on the other hand in a socket 14 also assembled through splines with said barrel 1.

The ball-shaped heads 5 are retained in part-spherical cavities of plate 7 by means of a retainer plate 10 and a spring washer 11 common to all the ball-shaped heads. A spring 12 extending axially in a cavity of the fixed socket 14 of babbit metal or like material holds the barrel 1, in conjunction with the assistance of inner pressures, against the distributor plate 15 secured to the support 4 together with an auxiliary distributor member 25. The socket 14 is held in position by means of the link 9 constituting the barrel driving member. This link 9 is provided with a pair of opposite ball-shaped heads engaging the one a corresponding part-spherical cavity formed in a piston-like member 17 slidably fitted for engagement by a compression spring 12 in the bore of socket 14 and the other a similar cavity formed in a member 19 rigid with plate 7.

The distributing slide face 20 of distributor plate 15 engages the slide face 21 of barrel 1 into which open the ports 22 formed in the bottoms of cylinders 2. In the distributor slide face 20 a pair of crescent-shaped delivery and suction apertures 23, 24 communicate with a pair of holes 23*a* and 24*a* respectively, which are formed through the said auxiliary distributor member 25 of which the outer surface constitutes a convex cylindrical slide face 25*a*.

This convex slide face 25*a* is parallel to a concave cylindrical slide face 26 formed on the inner surface of the cover 27 of pump casing C; as disclosed in the U.S. Pat. application, 6,717, filed on Jan. 29, 1960, now U.S. Pat. No. 3,589,224, the convex slide face 25*a* is separated from the concave slide face 26 by an intermediate cylindrical member 28 mounted for rotational sliding movement in the cover 27. The axis of the convex and concave slide faces 25*a* and 26, as well as the axis of said intermediate member 28, are merged into the pivot axis of support 4. Under these conditions, said member 28 can pivot about this last-mentioned axis.

Openings formed in member 28 and receiving tubular members 31 constitute intermediate ducts 34, 35 permitting the communication of holes 23*a* and 24*b* respectively with the inlet and outlet ducts 32 and 33 respectively formed in said cover 27.

It may be noted that the intermediate ducts 34 and 35 constantly connect the holes 23*a* and 24*a* to the inlet and outlet ducts 32 and 33, respectively, in all the relative positions of assembly 15–25, intermediate member 28 and pump cover 27.

On the other hand, at least one traction spring shown only diagrammatically at 29 in FIG. 2 connects the casing C to said intermediate member 28 and urges the latter towards a first stop forming portion 37 formed in the casing and a second stop-forming element 38 provided on one end of the convex slide face 25*a* of said auxiliary distributor member 25.

This specific form of embodiment corresponds to a pump with output reversal, this reversal being obtained by moving the barrel from one side to the other side of its zero-capacity position in which it is aligned with the axis of the impeller plate, i.e. when the suction and delivery ducts are inverted. Of course, as will be emphasized presently, this invention is applicable a fortiori to any similar pump, whether of the output reversal type or not, that is, with a barrel adapted to be inclined on one or both sides of its zero-capacity position.

The bearing R retained in the casing C under an end cover 39 is adapted to act as a radial guide to the impeller plate 7 and therefore to the shaft 8. According to a specific feature of this invention the face 40 of plate 7 bears against a hydrostatic bearing similar to the one formed between the barrel 1 and the distributor plate 15; in other words, this face 40 is formed with distributing cavities 42 identical with those of holes 22 and in this case the cavities 42 are connected through passages 42*a* to the part spherical recesses receiving the ball-shaped heads 5 in order to ensure the proper lubrication thereof.

These cavities 42 register with a pair of ports 43, 44 formed in the cover 39 of casing C, the shape of these ports as seen in plane view in FIG. 6 being identical with that of ports 23, 24; more specifically, the same relationship exists between the holes 22 and apertures 23, 24 on the one hand, as between cavities 42 and ports 43, 44 on the other hand. The ports 43, 44 are furthermore connected to apertures 23, 24 as will be seen presently, are insulated from the low pressure prevailing in the pump casing by a flat land 60, sliding shoes 61 compensating the residual unbalance of the plate.

In the barrel support 4 two cylinders 45 each receiving a sliding piston 46 communicate respectively with the pump ducts 32, 33 through holes 47, 50 formed in support 4 and 48, 49 in member 25. These holes 50 are also used for directing lubricating oil to the pivot faces P of support 4.

A pair of piston type check valves 52 of same diameter as the pistons 46 mounted in a pair of cylindrical bores 51 of casing C engage with their ball-shaped head 52*a* the flat end 46*a* of the corresponding piston 46. As an alternative, it will be noted that this piston 46 may also co-act with a piston check-valve through the intermediary of a ball B as illustrated in FIG. 7, the valve having in this case a flat end face 52*c*.

Ducts **53**, **54** and **55** connect the ducts **32**, **33** to the corresponding cylinders **51** so that the valves **52** are fed under the same pressure as pistons **46**. Each valve **52** is provided with a spool or like seating member **52*b*** adapted to close the inlet orifice of a chamber **56** formed in the cover **39** in alignment with the cylinder **51** for feeding through ducts **57**, **58** that port **43**, **44** which registers with the delivery port of the pump, which may be port **23** or **24** according to the direction of inclination of the barrel **1**.

The principle of operation of barrel pumps is well known to those conversant with the art and therefore it is not deemed necessary to describe in detail the specific mode of operation of the pump elements between the driving shaft **8** and the concave slide face **26** of cover **27**.

It is obvious on the one hand that the pump output depends on the inclination of barrel **1**, supports **4** and distributor plate **15** on either side of the zero-capacity position which is the position in which the axis of rotation of the barrel is coincident with that of the driving shaft, and that on the other hand the fact of passing from one to the other side of this zero-capacity position is attended by the reversal in the direction of flow of the fluid through the ducts **32** and **33** of the pump cover **27**.

In the specific position illustrated in FIG. **2**, the intermediate member **28** urged by spring **29** engages the stop portion **37** of casing C and as the barrel **1** has its maximum inclination the pump operates under maximum output conditions. The liquid sucked into the cylinders **2** and forced by the pistons flows through the pair of apertures **23**, **24** of distributor plate **15** and the pair of intermediate ducts **34**, **35** of member **28**, this liquid being for example sucked through the inlet duct **32** and forced through the outlet duct **33** of cover **27**.

As the barrel **1** is pivoted the output decreases and eventually a zero-output condition is attained when the axis of rotation of the barrel is coincident or aligned with that axis of the impeller plate **7**. If the barrel support **4** is further pivoted about its axis by actuating its control device V, the abutment or stop portion **38** of distributor member **25** moves the intermediate member **28** until the barrel support **4** is in its endmost position on the right-hand side of FIG. **2**.

To ensure a satisfactory operation of the hydrostatic bearing with respect to the slide face **40** of the impeller plate, irrespective of the "breaking" angle of the pump, it is only necessary that its feed pressure varies as the axial force acting upon this plate.

Now as already emphasized in the foregoing, under constant pressure conditions the axial force of the plate varies as the cosine of the "breaking" angle. It is therefore only necessary that the feed pressure of the fluid acting upon the hydrostatic bearing varies likewise as the cosine of the "breaking" angle.

It will now be assumed that the barrel is positioned very close to the axis of the pump shaft so as to deliver nevertheless a very low fluid output, the delivery pressure being exerted for instance on the piston **46** on the right-hand side of FIG. **2**. The pressure prevailing in chamber **56** will be exerted against the end of valve **52** which is of same diameter as piston **46**. If the pressure produced in chamber **56** is too low, the piston **46** will drive the corresponding valve **52**, thus permitting the supply of fluid from port **55** to this chamber **56** in order to increase the pressure therein until the state of balance is obtained.

Let us now assume that the pump (or motor) is given any breaking angle a; it will be seen that as a consequence of the mutual engagement between the spherical end **52*a*** of valve **52** and the flat end face of piston **46** the latter will exert on valve **52** only a force F (cos a), wherein F is the force exerted on the piston **46** by the delivery pressure. Under these conditions it will be seen that if P is the delivery pressure the piston **46** and valve **52** will cause a pressure P (cos a) to build up in chamber **56** and port **44**, thus permitting an impeller plate operation identical with that of barrel **1**. This barrel is balanced in the known fashion, that is, in short, the fluid pressure exerted on the bottoms of the barrel cylinders is substantially balanced by the pressure exerted in the delivery port **23** or **24** and on the auxiliary distributor member **25** in the corresponding intermediate duct **34** or **35**.

It will be noted that this device is particularly compact and of reduced overall dimensions. Only the driving connection between impeller plate **7** and shaft **8** through its rotary joint take some space. Moreover, it will be seen that the arrangement does not require any particularly elaborate component ; thus, the hydrostatic bearing may consist preferably of sintered metal and constitute an insert fitted in the casing cover. However, this bearing may also be pressed from a bimetallic bronze-steel plate.

It will also be seen that the connecting rods have not to be formed with bores for directing lubricating oil to the ball-joints **5** as usually observed, this constituting a substantial advantage in the case of pumps and motors of very small dimensions.

FIG. **1** illustrates a developed view of the pump and it is clear that the holes **54** may lie outside the radial plane containing the pistons **46** in order to reduce the transverse dimensions of the pump.

In the case of a pump in which the high-pressure port is invariable a single assembly comprising a piston **46** and valve **52** is sufficient between the pump delivery side and the hydrostatic bearing of impeller plate **7**.

Although a single form of embodiment has been described and illustrated herein, it will readily occur to anybody conversant with the art that various modifications and changes may be brought thereto without departing from the spirit and scope of the invention as set forth in the appended claims.

What is claimed is:

1. A pump or motor of the multi-cylinder barrel type, of which the capacity is adapted to be varied by displacing the barrel in a support adapted to be inclined in the pump body about a pivot axis perpendicular to the axis of rotation of the barrel, the pistons being provided with ball-headed rods positively connecting said pistons to an impeller plate which positively connects the barrel with the pump driving shaft, said barrel comprising a slide face into which open the cylinder ports, said slide face being in rotary contact with a distributing slide face of a distributor plate in which two fluid passages are formed which open each into said distributing slide face through a port having the shape of a sector of an annulus and communicating with suction and delivery ducts of the pump, wherein the impeller plate is balanced axially by a hydrostatic bearing having an alignment with the bearing face of the impeller plate in the pump body at least one port corresponding to the delivery port of said distributing slide face and adapted to be connected to the pump delivery side through a balancing valve which comprises in the barrel support a control piston responsive to the delivery pressure and in the pump body a valve controlling the communication between the aforesaid port of the hydrostatic bearing and the delivery side of the pump, said control piston and said valve coacting in opposition with each other whereby the balancing pressure exerted against the hydrostatic bearing of said impeller plate varies as a function of the inclination of the barrel support during operation, and therefore as a function of the axial stress to be balanced at the impeller plate.

2. The pump or motor of claim **1**, comprising an impeller plate bearing face having in alignment with the aforesaid port of the pump body, openings corresponding to the aforesaid cylinder ports opening into the barrel slide face.

3. The pump or motor of claim **1**, wherein at least one of the pump ports connects with a cylinder formed in the barrel support, the axis of said cylinder being parallel to the barrel axis but secant to the pivot axis of the barrel support, said cylinder receiving a piston having its free end located in close proximity to said pivot axis.

4. The pump or motor of claim **1**, comprising a bearing surface of spherical configuration through which said control piston and said valve coact with each other.

5. The pump or motor of claim **1**, comprising a ball bearing disposed between said control piston and said valve, through which said control piston and valve coact, said ball bearing loosely guided in a recess and bearing against the faces of said piston and said valve which faces are perpendicular to the directions of morement of said piston and said valve, respectively.

* * * * *