Nov. 7, 1961 T. BUDZICH 3,007,420

HYDRAULIC PUMP OR MOTOR

Filed Oct. 7, 1959 2 Sheets-Sheet 1

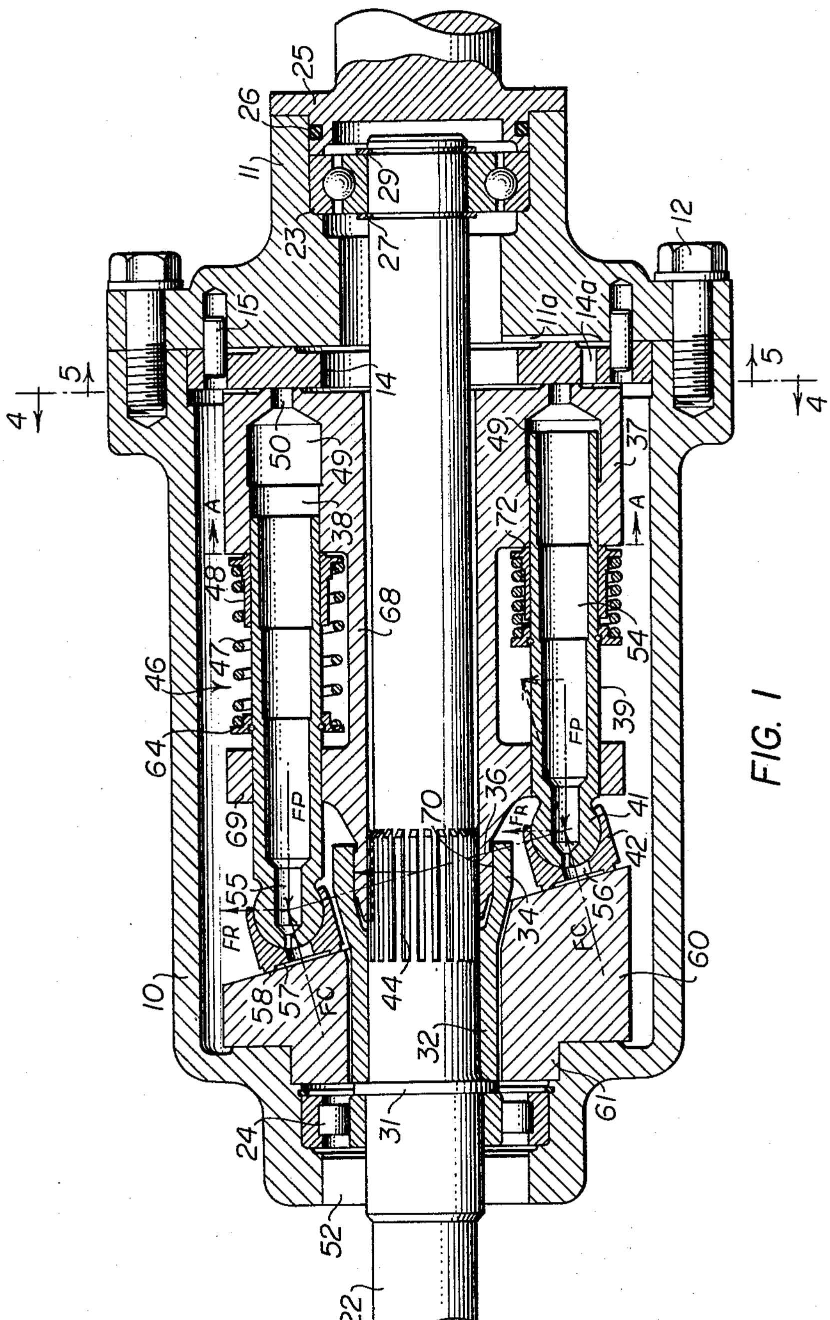

FIG. 1

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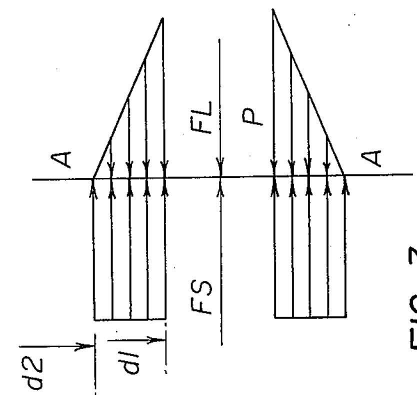

FIG. 3

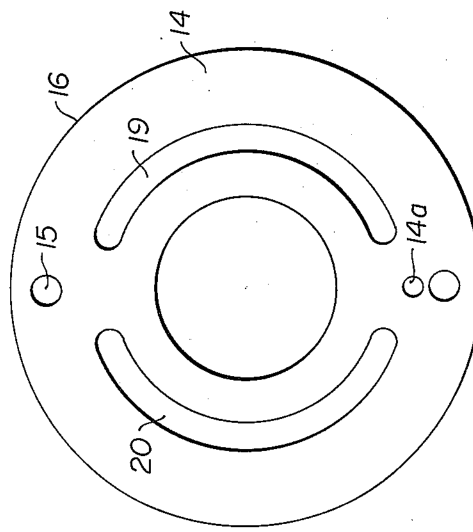

FIG. 5

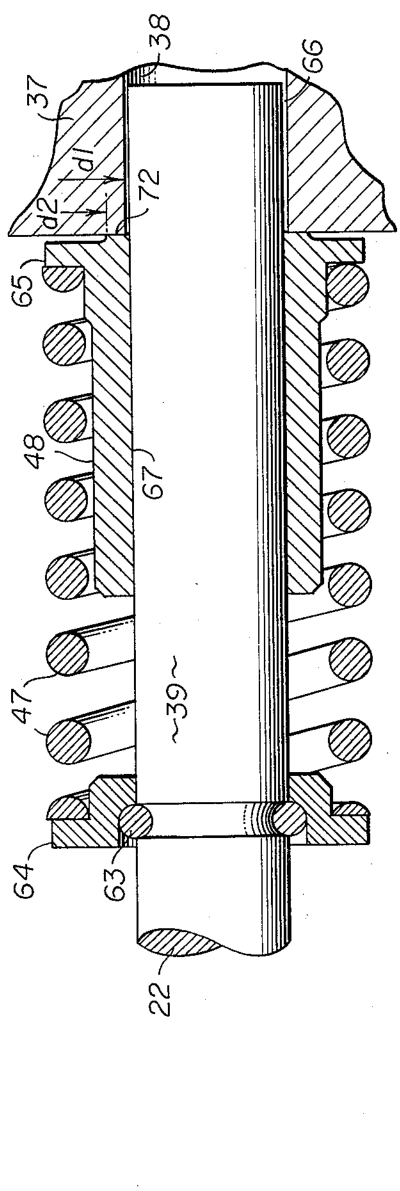

FIG. 2

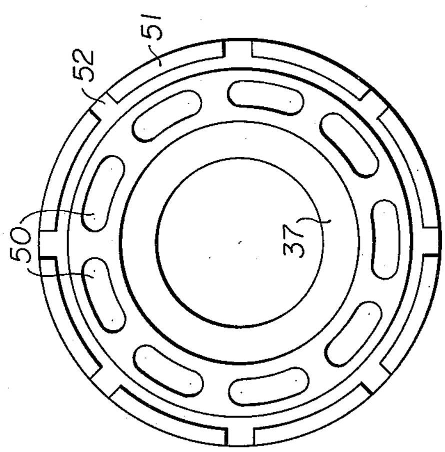

FIG. 4

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HYDRAULIC PUMP OR MOTOR

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8 Claims. (Cl. 103—162)

This invention relates to hydraulic apparatus and more particularly to fluid pressure energy translating devices commonly known as fluid pumps and motors.

In still more particular aspect the invention relates to fluid pumps and motors of the type having a cylinder barrel in which pistons are reciprocated as influenced by a cam plate device.

In still more particular aspects the invention relates to fluid pumps and motors in which the pistons transmit the driving torque to or from a cylinder barrel in the form of cantilever loads induced by transverse components of piston thrust when working against an inclined cam plate.

In a pump or motor of the above described types, transverse components of pistons thrust not only constitute the driving torque of the device but also result in setting up large transverse forces in the cylinder barrel, tending to displace it as from its abutment with a valve plate. This disturbs the distribution of requisite sealing pressures (as on the flat face of such a valve plate) reducing the efficiency of the unit and under extreme conditions causing separation of cylinder barrel and valve plate.

In one arrangement of the prior art the cylinder barrel is supported by a radial bearing encircling it and transmitting the transverse forces directly to the pump housing. This solution gives bulky and heavy construction of limited maximum speed as dictated by characteristics of the large bearing.

In another prior art arrangement the cylinder barrel is supported on a shaft retained by bearings in the housing. The point of contact between the cylinder barrel and the shaft usually takes place at a spline, the looseness in the spline providing limited universal freedom. In this solution the transverse loads bearing against the spline teeth produce a binding effect and impair the efficiency of the universal action.

Heretofore, too, in pumps and fluid motors, leakage along the pistons is controlled by maintaining minimum clearances. But the tight clearances and long overlaps are synonymous with high costs, and also increase the dimensions of the device. Such minimum clearances are also disadvantageous because easily damaged by dirt particles in the pumped fluid and because of inability to carry away the heat generated at the points of high contact pressure. Another disadvantage of the conventional solution lies in the fact that due to cantilever loading of the piston, the piston assumes an eccentric position in relation to the cylinder bore. The leakage then, as is well known in the art, may be as much as three times greater than would be the case were the piston to remain concentric in the cylinder bore.

It is an object of the present invention to provide simple and inexpensive means for overcoming the above mentioned difficulties.

In accordance with one aspect of the present invention, transverse forces otherwise tending to upset the equilibrium of the cylinder barrel are balanced by proper selection of the point of its support along the longitudinal axis of the pump more or less independent of the splines and this feature cooperates with spring biased series leakage resistance means located in an opening or openings intermediate the cylinder bore ends and around each piston, to reduce binding, reduce weight, reduce leakage, reduce costs, and greatly improve operational efficiency.

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It is therefore another object of the invention to provide improved means for supporting a cylinder barrel on a shaft in which binding action due to transverse loads is eliminated from the driving spline.

A further object of this invention is to reduce leakage of high pressure fluid along the pumping (or motoring) pistons.

A still further object is to control leakage while increasing the clearance between piston and cylinder bore for economy.

Still another object of this invention is to provide separate readily replaceable means to reduce leakage about each piston, said means, if damaged being replaceable without replacement of the expensive cylinder barrel or piston.

A further object of this invention is to mount sleeves providing resistance to leakage in such a way as to eliminate all the transverse loads acting on them so that under action of the reciprocating motion of the piston they will take a central position thereabout further reducing leakage.

Other objects and advantages will become apparent and the invention may be better understood from consideration of the following description taken in connection with the accompanying drawing, in which:

FIG. 1 is a vertical cross-sectional view through an axial piston pump (or motor) provided with piston leakage sleeves according to the invention.

FIG. 2 is an enlarged view detailing the construction of one such leakage sleeve.

FIG. 3 is a graphical representation of force distribution in the plane *a—a* of FIG. 2 on the sealing area between leakage sleeve and cylinder barrel.

FIG. 4 is a view along the line **4—4** of FIG. 1 and showing an end view of a cylinder barrel **12** which may be conventional.

FIG. 5 is a view along the line **5—5** of FIG. 1 and showing an end view of a valve plate **24** which may also be according to the prior art, FIGS. 4 and 5 being included only to simplify description of the present invention which resides in other combinations of parts.

Description

Referring first to FIG. 1, I have shown a pump body comprising two sections, namely a pump housing **10** and a pump cover **11**, with the two suitably connected by bolts **12**. A valve plate **14** is secured to the pump cover **11** by dowel pins **15** and has its outer periphery **16** engaging a bored recess **17** in the housing **10**. Valve plate **14** is provided with kidney shape arcuate slots **19, 20** (see FIG. 5) which are in direct communication with pump cover inlet and outlet ports (not shown). If desired the valve plate may be provided with leakage drillings **14*a*** and the cover with communicating leakage passages **11*a***.

A drive shaft **22** extends longitudinally through the pump and is retained at the cover end in a bearing **23** and at the opposite end of the housing in a bearing **24**. Bearing **23** is retained in the cover by a shaft cover **25** having a sealing ring **26**. Bearing **23** is shown as a ball bearing which provides the end location for the shaft **22** retained with respect to the bearing by two snap rings **27** and **29**. The other bearing (**24**) has its inner race against a drive shaft shoulder **31** but may be of roller type (as shown) and provide radial location and carry radial loads only. A reaction sleeve **32** is tightly pressed on shaft **22** and against the shoulder **31**. Sleeve **32** has an enlarged end **34** spaced radially outward from the drive shaft **22** to slidably engage a cylindrical extension portion **36** of a cylinder barrel **37**.

The cylinder barrel **37** is provided with cylinder bores

38 and pistons 39 work in operational contact with these bores. Each piston 39 terminates in a spherical end 41 and a plurality of shoes 42 closes each over one of the spherical ends 41.

As shown, the shaft 22 has a male spline portion 44 engaging a corresponding spline located in the extension 36 of cylinder barrel 37. It will be seen that cylinder barrel 37 is radially spaced from shaft 22 except in the region of the splines. The longitudinal bores 38 are interrupted each by the same circular recess indicated generally at 46 and for housing piston return springs 47 and leakage sleeves 48. At the right end, as viewed in FIG. 1, the cylinder bores 38 communicate with recessed reliefs 49 terminating in slots 50 which pass through the endmost flat face of the cylinder barrel 37 (see FIG. 4). This flat face also has usual dynamic pads 51 and leakage slots 52 and works in operational contact with a flat face of the valve plate 14.

Within the pistons the various drillings such as 54 and 55 connect the cylinder bores 38 with an opening 56 in the respective piston shoe 42 and with the balancing area 57 thereof. The piston shoes 42 with the balancing areas 57 and balancing lands 58 work in sliding engagement with the angled flat face of a cam plate 60 radially located at one end of the housing 10 by a flange portion 61 engaging a corresponding recess in the housing 10.

The piston return springs 47 are anchored at one end each to the respective piston 39 by a snap ring 63 and a spring retainer 64 (see FIG. 2). The other end of each piston return spring 47 engages a flange portion 65 of the corresponding leakage sleeve 48. The leakage sleeve 48 encircles the piston 39 and, in cooperation with the spring retainer 64, radially locates the piston return spring 47 while the return spring 47 keeps the leakage sleeve 48 in abutment with the cylinder barrel 37. With this construction, and as hereafter explained, any leakage oil, flowing along a clearance 66 (between piston 39 and cylinder bore 38 as indicated in FIG. 2) must also flow through a clearance 67 between the leakage sleeve 48 and the piston 39. Since resistance to leakage flow is proportional to length of flow path, the leakage sleeve is purposely configured to extend a material distance longitudinally to reduce leakage volume.

Operation

Although the device may function either as a pump or as a motor, its operation will be described as that of a pump. Accordingly the drive shaft 22 may be assumed connected to a suitable prime mover, not shown, and the inlet and outlet ports of the pump connected to a hydraulic system. The drive shaft 22, revolving in the bearings 23 and 24 will induce rotation of cylinder barrel 37 through the male and female splines. The cylinder barrel, while revolving, will sequentially register each cylinder bore with inlet and discharge ports of the pump. In a well known manner the cam plate 60 will cause the pistons to reciprocate in timed relation with this sequential registration and thereby will produce a pumping action. The proper sealing between the mating faces of the cylinder barrel 37 and valve plate 14 is affected by hydraulic pressure, acting within cylinder bores 38, and force due to the piston return springs 47.

The floating cylinder barrel 37 is free to slide in a longitudinal direction and align itself to the valve plate. The longitudinal forces, acting on it, will produce an effect of rapidly closing any gap existing between these mating members. The flat face of cylinder barrel 37, mating with valve plate 14, is hydrostatically balanced in a well known manner so that the actual contact pressure, existing between these two parts, is caused by only a small fraction of the hydraulic reaction force available.

The cylinder barrel circular recess 46 defines a central tubular part 68 connecting a cylinder barrel front flange part 69 with a rear part housing the pressure bores. While each of the circumferentially spaced cylinder bores 38 extend from the front flange 69 to the main body of the cylinder barrel, each is interrupted by the circular recess 46 housing piston return springs 47.

As already mentioned longitudinal passages such as 54 and 55 extend through each piston and connect with a passage 56 and balancing area 57 of the respective piston shoe 42 to affect in a well known manner a hydrostatic balance. The distribution of hydraulic forces, acting on the flat faces of piston shoes 42 is so arranged that during the discharge stroke there exists a small resultant force keeping the piston assembly against the flat face of the cam plate 60. During the suction stroke, the piston assembly is maintained in contact with the cam plate 60 by the piston return springs 47 working in abutment with the cylinder barrel 37 through the leakage sleeve 48. For maximum pumping efficiency the cylinder barrel 37 must be free to align itself against the corresponding face of the valve plate 14, irrespective of deflection in the housing 10 and out-of-squareness of sealing surfaces due to manufacturing tolerances.

Referring to FIG. 1, a force FP is induced in the piston assembly by pressure generated in the cylinder bore 21, during the discharge stroke, acting on the piston area. This force is opposed by a force FC, perpendicular to the surface of cam plate 60 and a force FR, acting through the center of spherical surface 41, and transverse to the axis of the piston. The forces FR, acting on all pistons subjected to the discharge pressure, are transmitted in the form of cantilever loads to the cylinder barrel extension 36 and to the cylinder bores 38. These forces produce the driven or driving torque depending on whether the device is acting as a pump or as a motor. At the same time the forces FR produce a transverse moment acting on the cylinder barrel, tending to unseat it from its abutment with the valve plate 14. In known manner these moments can be eliminated from the cylinder barrel by arranging the point of support between cylinder barrel and drive shaft at the point of intersection of a plane connecting the centers of the spherical piston ends with the center line of the cylinder barrel. Cylinder barrel cylindrical extension 36, with its flat land 69, engages in a slidable manner the inner periphery of the enlarged portion 34 of the reaction sleeve 32. The sum of the forces FR, represented by a resultant forces FRR, see FIG. 1, is transmitted from the cylinder barrel to the inner surface of the reaction sleeve 32 and then to drive shaft 22. The center of contact pressure between cylinder barrel 37 and reaction sleeve 32 is shown at point 70. The splined connection between the cylinder barrel 37 and shaft 22 is located directly inside the reaction sleeve 34, the length of this splined engagement being equally spaced on each side of the center of contact pressure represented by point 70.

In a conventional type of solution the cylinder barrel is supported directly on the shaft in the region of the splines without any such reaction sleeve to assist. This type of cylinder barrel suspension suffers from very serious disadvantages. Large transverse loads, acting on the involute profile of the spline teeth, tend to produce a binding action, which very seriously affects the freedom of alignment of the cylinder barrel. At the same time these transverse loads produce an eccentric location of the female spline in relation to the male spline by the amount of clearance available in the drive. Since the load is supported on the full length of the spline teeth any adjustment in the position of the cylinder barrel, in relation to its longitudinal axis, will not only result in a binding action but it will induce an additional couple disturbing the equilibrium of the cylinder barrel. The present invention eliminates this binding action on the spline by removing from the splines the transverse loads acting on the cylinder barrel, as transmitted from the cantilever piston assemblies. The great circle engagement of the cylindrical extension 34 of the reaction sleeve 32 locates the cylinder barrel 37 in relation to the drive shaft 22 and absorbs the resultant force FRR. The reaction sleeve 32 ensures that the female spline is kept concentric with the male spline and this results in the spline teeth carrying torque loads only. This gives a much greater freedom of alignment of the cylinder barrel. The land 69 provided on the cylindrical extension 36 of the cylinder barrel 37 is made to engage the reaction sleeve 32 directly at the mid-length position of the spline. Only in this type of cylinder barrel suspension can the maximum of movement of cylinder barrel be obtained for any specified clearance in the spline teeth.

Thus the transverse forces upsetting the equilibrium of the cylinder barrel are balanced by proper selection of the point of its support along the longitudinal axis of the pump. The point of support of the cylinder barrel is substantially at the point of intersection of its longitudinal axis with the plane connecting the centers of the spherical piston ends so that the cylinder barrel mounted around this point will be capable of aligning itself against the flat surface of the valve plate irrespective of deflection of the housing and out-of-squareness of sealing surfaces due to manufacturing tolerances, but unlike prior art arrangements in the described arrangement this does not involve the splines.

To reduce leakage and therefore increase efficiency of a piston pump, tight clearances between the pistons and cylinder bores are usually maintained. The tight clearances of the prior art, although beneficial from the standpoint of performance, carry certain inherent disadvantages. They are expensive to produce and susceptible to damage by dirt particles contained in the fluid pumped. Tightly fitted pressure bores must not only seal but also carry the side loads to which the pistons are subjected. In cylinder bores working with tight clearances it is difficult to cool the cylinder walls subjected to high localized pressures. A strict compromise between efficiency and life of the sealing surface becomes necessary. There is one additional disadvantage common to all piston pumps in which the pistons are carrying transverse loads. Under the influence of these loads the pistons assume an eccentric position in the clearance provided in the cylinder bore. It is well known in the art that the leakage past these eccentrically located pistons will be as much as three times larger than would be the case were these pistons to remain concentric in the cylinder bores. The present invention eliminates all these disadvantages, as listed above, by providing additional resistance to leakage in series with the resistance to flow in the clearance between each piston and its cylinder bore in form of a leakage sleeve 48. Each leakage sleeve 48 encircles the respective piston and under action of the piston return spring 47, is kept in abutment with the flat face of the cylinder barrel 37. The actual decrease in the leakage volume will be proportional to the length of the leakage sleeve, length of the piston overlap in the cylinder bore, and respective clearances in leakage and in cylinder bore. Because of its construction the leakage sleeve can be more cheaply manufactured to closer tolerances and therefore the clearance between this leakage sleeve and the piston can be kept to a minimum. At the same time the clearance between the piston 39 and the cylinder bore 22 can be increased, thus reducing the cost of a large and complicated part. This increase in clearance between the piston and the cylinder bore, with the use of leakage sleeves, does not decrease the efficiency of the pump and at the same time it provides a more favorable condition of cooling and loading of the rubbing surfaces. An important advantage of this solution is that (although, under action of the catilever transverse load, the piston will take an eccentric position in the cylinder bore) the leakage sleeve can remain concentric, further reducing the total leakage along the piston. The ability of the leakage sleeve 48 to maintain its central position in respect to pump piston is obtained by slideably mounting a sleeve sealing surface 72 along the cylinder barrel face AA (see FIG. 2).

The leakage oil when forced to flow by the pressure gradient through the clearances 66 and 67 (FIG. 2) along the surface of the piston 39 affects the pressure distribution on sealing surface 72 at the point of contact between leakage sleeve 48 and cylinder barrel 37. The pressure distribution on sealing surface 72 is shown as P on the right of line AA in FIG. 3. The pressure P acting on the inner edge of the sealing area is equal to discharge pressure less than pressure drop in clearance 66. The resultant force FL caused by the pressure distribution on sealing face 72 tends to separate leakage sleeve 48 from the face AA of cylinder barrel 37. This separation in prevented by a force FS representing the minimum preload in the piston return spring 47. The force distribution on the sealing face 20 is so arranged that at all times force FS is larger than force FL resulting in effective sealing. This construction also permits replacement of individual leakage sleeves, if damaged, making wear in the cylinder bores 38 of the cylinder barrel 37 less important. At the same time this solution permits a wider use of materials in construction of the cylinder barrel and leakage sleeves.

There is thus provided a device of the character described capable of meeting the objects above set forth and whereby improved mounting of cylinder barrel and additional spring biased resistance means to prevent leakage (despite large tolerances between each piston and piston bore) as well as the large openings between portions of each cylinder bore cooperate to minimize or obviate binding action due to transverse force couples, to reduce weight of rotating parts, to minimize leakage of high pressure fluid and thus to increase efficiency of the device, and to afford substantial economies both in manufacture and in maintenance.

While I have illustrated and described a particular embodiment, various modifications may obviously be made without departing from the true spirit and scope of the invention which I intend to have defined only by the appended claims taken with all reasonable equivalents.

I claim:

1. In an energy translating fluid pressure device comprising a housing, a drive shaft journalled with respect to the housing, a cylinder barrel rotatable with said shaft and having a plurality of cylinder bores and pistons reciprocable therein, valve structure having inlet and discharge ports which sequentially register with each cylinder bore as the cylinder barrel rotates, spherical piston ends associated with the pistons outside the cylinder barrel, shoes closing one over each of the spherical piston ends while universally mounted with respect thereto, and a cam plate operably connected with the shoes and mounted with respect to the housing, the combination of said barrel and said drive shaft having drivingly interconnected splines, said shaft including constraining means, said constraining means being in axially-slidable engagement with said barrel at a region radially spaced from said spline interconnection, whereby said constraining means permits axial movement between said shaft and said barrel but limits transverse movement therebetween.

2. In a device as in claim 1, the combination thereof further characterized by the constraining being a sleeve having an inner bore at one end engaging the shaft and an enlarged opposite end with an inner bore engaging a portion of the cylinder barrel radially spacing said cylinder barrel from the shaft.

3. In a device as in claim 1, the combination thereof further characterized by the cylinder barrel having a cylindriral extension having inner peripheral female splines for mating with male splines provided on the shaft, while said cylinder barrel extension has an outer peripheral land, and constraining means, said constraining means comprising a sleeve engaging the shaft at a region axially spaced from the male splines at one end and with its other end slideably engaging said peripheral land of said cylinder barrel.

4. A pressure fluid mechanism comprising a housing, a drive shaft journalled in said housing, a rotatable cylinder barrel having a plurality of axially extending cylinder bores each interrupted and made into fore and aft support and pressure portions by a circular recess extending around the cylinder barrel, pistons mounted for sliding in said cylinder bores and each having a part-spherical surface at one end, piston shoes universally mounted one on each of said part-spherical piston ends, a cam plate having an inclined face arranged to engage the shoes and thus to move the pistons in at least one direction with relative rotation of cylinder barrel with respect to the cam plate, resilient biasing means located in the barrel circular recess and resiliently biasing each piston with respect to the cylinder barrel for aiding motion of the pistons in one direction, a relatively stationary valve plate arranged in abutment with an end of the cylinder barrel and having inlet and outlet ports for sequentially registering with each cylinder bore as the barrel rotates, cylinder barrel rotation means comprising a male spline on the drive shaft and a co-operating female spline on the cylinder barrel, and a sleeve mounted for rotation with the drive shaft and slidably engagnig a portion of the cylinder barrel containing the female splines and radially spaced therefrom with the midpoint of engagement coinciding with a plane passing through the midpoint of the female splines transverse to the axis of the shaft.

5. A pressure fluid mechanism as in claim 4 further characterized by a substantial clearance provided between male and female splines, the cylinder barrel outside the splines being radially spaced from the shaft to ensure requisite universal action between cylinder barrel and drive shaft, while the constraining means maintains a mid-point of the splines concentric to obviate binding action under transverse load.

6. A pressure fluid mechanism as in claim 4 further characterized by a plurality of sleeves each encircling a different piston and located in the circular recess adjacent the cylinder barrel pressure portion, said sleeves and barrel having mating pressure sealing faces, the sealing face of each sleeve having an inner diameter less than the diameter of its associated cylinder bore, and the biasing means comprising a plurality of springs one about each of said sleeves and against a portion thereof to urge said mating faces into engagement while the opposite end of each spring is secured with respect to the

associate piston, with the area of said faces and the preload force of said springs so selected that the force due to pressure gradient acting on said faces is less than the preload in said springs at any piston position.

7. In the combination of claim 1, the centers of the spherical piston ends being co-planar in all operative positions and wherein the plane of the centers of the spherical piston ends intersects axial sliding engagement of the constraining means and the cylinder barrel and also intersects the driving interconnection of the splines of the barrel and the shaft in all operative positions of the device.

8. In an energy translating fluid pressure device comprising a housing, a drive shaft journaled with respect to the housing, a cylinder barrel rotatable with the shaft and having a plurality of cylinder bores and pistons reciprocable therein, valve structure having inlet and discharge ports which sequentially register with each cylinder bore as the cylinder barrel rotates, spherical piston ends associated with the pistons outside the cylinder barrel, shoes closing one over each of the spherical piston ends while universally mounted with respect thereto, and a cam plate operably connected with the shoes and mounted in the housing, the combination of the barrel being disposed around at least a portion of the shaft, the shaft and the barrel each having splines drivingly interconnected, said shaft having an annular constraining sleeve, said sleeve ahving an inner cylindrical surface radially spaced from the shaft spline and defining an annular recess between the shaft spline and the sleeve, said barrel having an annular projecting portion, the splines of the barrel being located on the inner rim of said projecting portion, the outer rim of said projecting portion being an annular surface mated to coact with the inner surface of said sleeve in axially sliding engagement, said projecting portion projecting into and terminating in said recess, whereby axial movement between the barrel and the shaft is permitted but transverse movement therebetween is limited.

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