

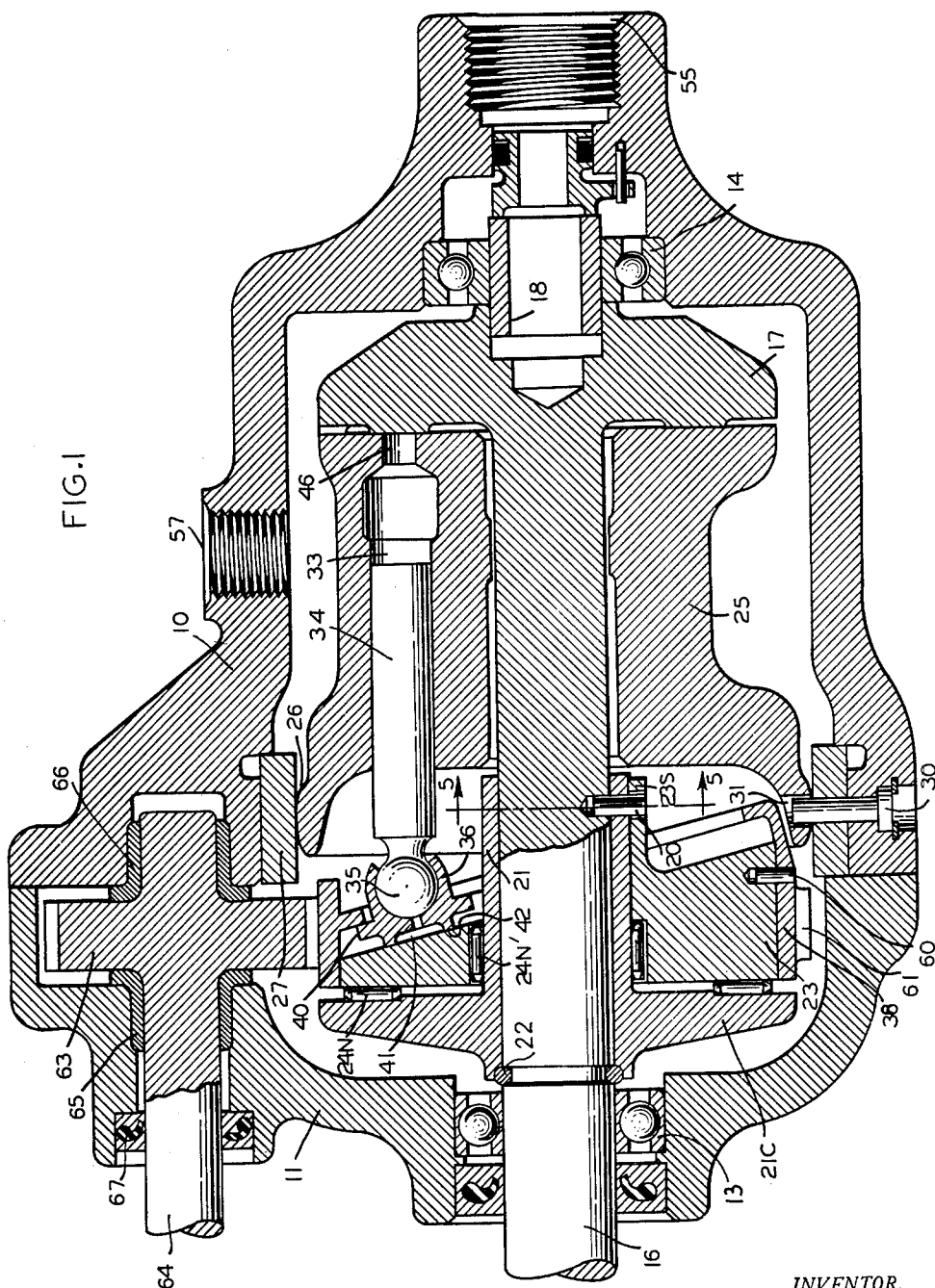
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T. BUDZICH
HYDRAULIC APPARATUS

3,056,387

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3 Sheets-Sheet 1



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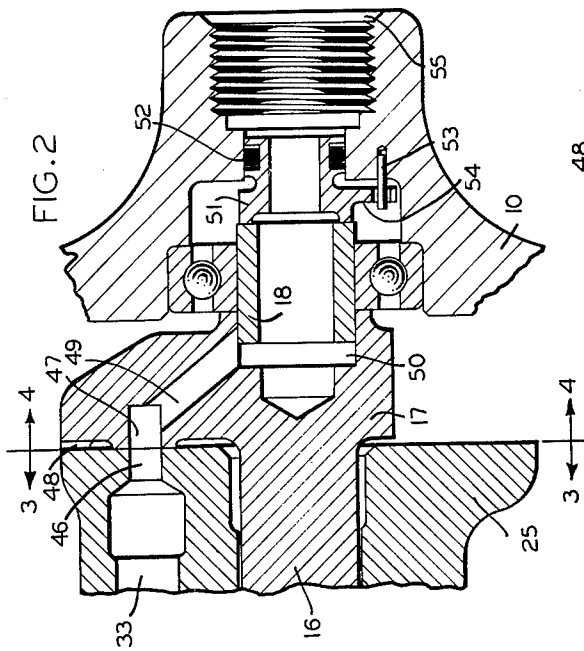
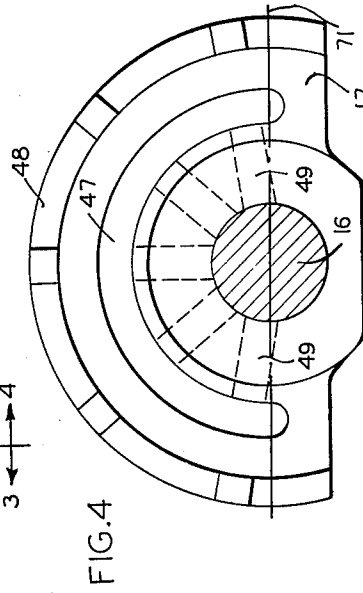
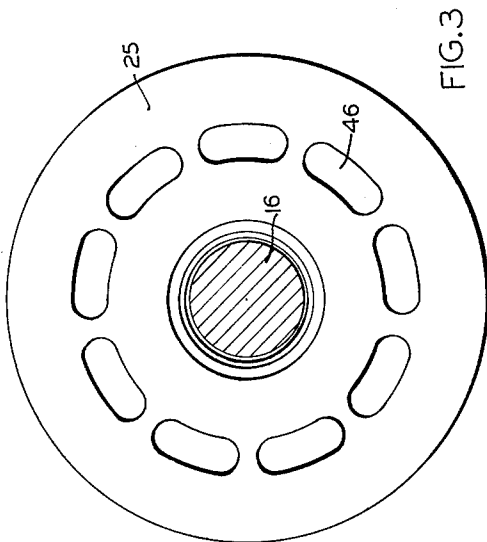
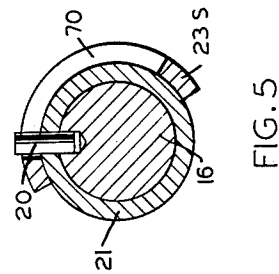
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3 Sheets-Sheet 3

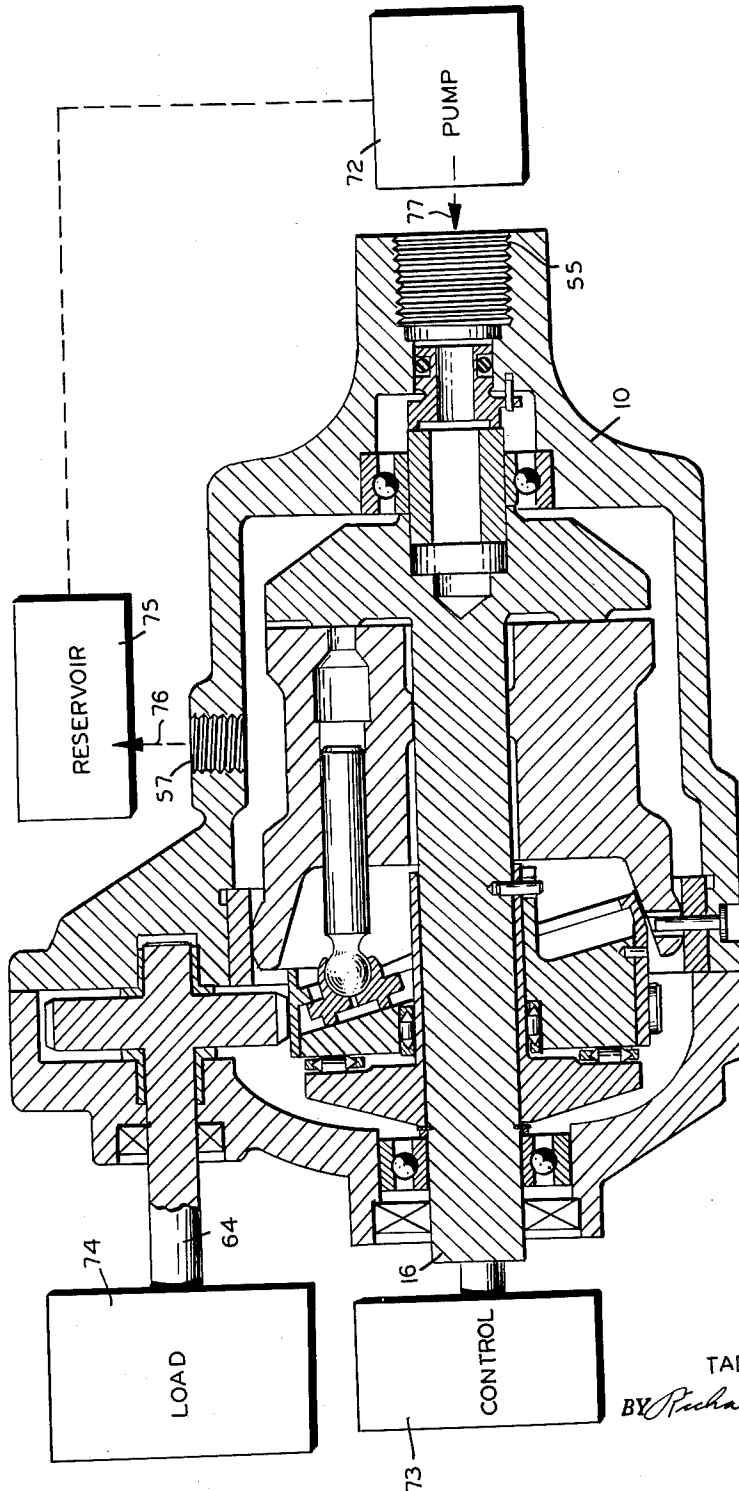


FIG. 6

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This invention relates to fluid pressure translating devices, and more particularly to torque multipliers for constant speed drives. This application is a continuation-in-part of my application, Serial No. 128 filed January 4, 1960, now abandoned.

For many applications it is necessary to drive a load, which may vary, at selectable speeds or selectable constant speeds irrespective of any variation of the magnitude of the load. It is therefore a principal object of this invention to provide a hydraulic actuated torque multiplier in which the speed of a driven load is regulated by the speed of a non-load carrying shaft.

A related object of this invention is to provide a hydraulic actuated torque multiplier which will drive varying loads at a constant speed.

It is a further object of this invention to provide a hydraulic actuated torque multiplier which will drive varying loads at a constant speed irrespective of pressure variations of the high pressure fluid.

It is a more particular object of this invention to provide a hydraulic actuated torque multiplier of the axial piston type which will drive varying loads at a constant speed and will use an amount of fluid proportional to the load magnitude and the pressure of the fluid.

Other objects and advantages will become apparent and the invention may be better understood from consideration of the following description of a preferred embodiment taken in conjunction with the accompanying drawings in which:

FIG. 1 is a longitudinal section view of a fluid motor operable in accordance with the present invention;

FIG. 2 is a longitudinal section showing a portion of the motor as in FIG. 1 but with a valve plate rotated 90°;

FIG. 3 shows end face of cylinder barrel (and cross-section of tie rod) as viewed along the line 3-3 of FIG. 2;

FIG. 4 shows end face of valve plate (and cross section of tie rod) as viewed along the line 4-4 of FIG. 2;

FIG. 5 is a cross-section of shaft, support sleeve, and cam plate-part-sleeve as viewed along the line 5-5 of FIG. 1; and

FIG. 6 is a diagrammatic representation showing surrounding apparatus useful in connection with operation of the hydraulic motor of FIG. 1.

Referring now to the drawings, and particularly to FIGURE 1, the torque multiplier has a housing 10 closed by an end cover 11. A tie rod 16 extends into the housing 10 through the end cover 11 and is journaled for rotation by bearings 13, 14. The tie rod defines the principal axis of the torque multiplier. The tie rod 16 carries a valve plate 17 at one end thereof, which valve plate will be described more fully presently.

A cylinder barrel 25 is non-rotatively disposed in the housing and has a part spherical surface 26 contacting a mounting ring 27 carried in the housing 10. A pin 30 provides for a universal tilting movement while preventing rotation of the barrel 25 with respect to the housing. This type of mounting is more fully described in my co-pending application Serial No. 127 filed January 4, 1960. The cylinder barrel 25 is provided with a plurality of cylinder bores 33. Each cylinder bore 33 has a piston 34 disposed therein.

A cam plate 23 is provided against which the pistons work. The cam plate 23 surrounds the tie rod 16 at

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the lower end of the cylinder barrel 25 and is supported by a cam plate back-up support 21c which is carried by the tie rod 16. The cam plate back-up support 21c is connected to the tie rod 16 by a pin 20 extending through a longitudinal extension 21 of the cam plate back-up support, and is also connected to the tie rod 16 by a snap ring 22.

The cam plate 23 has a sleeve 23s which surrounds the extension 21 of the cam plate back-up support and is substantially coextensive therewith. The sleeve has an arcuate slot 70 (FIGURE 5) extending circumferentially and positioned to coact with the pin 20. As thus assembled the cam plate 23 is not drivingly connected to the tie rod 16, but rather is free to rotate with respect to the tie rod 16 within the limit of the arcuate slot 70. Either extreme position of the pin at one end of the slot or the other where it is abutting the end of the slot is beyond the operative range of the device, so, in all operative positions of the device, there is no direct driving connection between the tie rod 16 and the cam plate 23. The purpose for this freedom of rotation of the cam plate 23 with respect to the tie rod 16 will become readily apparent.

A plurality of needle bearings 24N, 24N' are disposed between the cam plate 23 and the cam plate back-up support 21c to permit easy relative rotation therebetween since the cam plate back-up support 21c is connected to the tie rod 16 and will rotate therewith.

Referring now to FIGURES 2, 3, and 4 a conventional fluid motor high pressure fluid inlet structure is provided. This includes a collector sleeve 18 communicating with a balance sleeve 51, which in turn communicates with a high pressure inlet port 55. A seal 52 is provided between the balance sleeve 51 and the housing 10. The balance sleeve 51 is prevented from rotating by a pin 53 extending from the housing 10 through a flange portion 54 of the balance sleeve 51.

This conventional structure also includes a high pressure side of the valve plate 17 having a kidney shaped port 47 connected to the collector sleeve 18 by a series of drillings 49 and a central bore 50. The cylinder bores 33 each terminate in kidney shaped openings 46 which sequentially register with the kidney shaped port 47 as the valve plate 17 rotates. The valve plate 17 also has a low pressure side which allows fluid to discharge from the cylinder bores 33 into the housing 10. The housing 10 has a low pressure fluid discharge port 57. Dynamic pads 48 are provided to stabilize the valve plate 17 and cylinder barrel 25.

As in a conventional motor, the pistons 34 have part spherical ends 35 closed over with shoes 36 which operate against the cam plate 23. The pistons are hydrostatically balanced by means of balancing lands 42 forming oil reservoirs 41 on the piston shoes. A nutating plate 38 is provided which coacts with flanges 40 on the piston shoes 36 to retain the pistons on their non-power strokes. The nutating plate 38 is connected to the cam plate 23 by pin 60.

An output shaft 64 extends through the end plate 11, and is journaled for rotation in the housing 10 by bearing 65, 66. The shaft 64 is sealed in the housing by shaft seal 67. The shaft 64 includes a pinion gear 63 located within the housing 10 which is meshed with teeth 61 around the periphery of the nutating plate 38. Hence, any rotation of the cam plate 23 will cause rotation of the output shaft 64.

A control 73 is diagrammatically shown connected to the tie rod 16 to rotate the tie rod, and a load 74 is shown diagrammatically on the output shaft 64 (FIGURE 6). The control 73 is preferably a variable speed motor, however for some applications a constant speed motor or even a hand crank would suffice. Also shown diagram-

matically are a high pressure fluid source 72, a discharge reservoir 74, and pipes 77, 76 connecting the source and reservoir respectively to the inlet port 55, and exhaust port 57.

The cam plate 23 and the valve plate 17 are so arranged that at one extreme position of relative rotation of the cam plate 23 and valve plate 17, i.e. a position where the pin 20 is contacting one end of the slot 70, the pistons subjected to the high pressure fluid are disposed symmetrically about the lowest point of the cam plate 23. When from this position the tie rod 16 is rotated (clockwise as shown in FIGURE 5) the pin 20 will move out of contact with the end of the slot 70 and commence moving therein. The rotation of the tie rod 16 will cause identical rotation of the valve plate 17, but its rotation will not per se cause the rotation of the cam plate 23 since there is no driving connection therebetween. The rotating of the valve plate 17 out of the position described above will change the angular relationship of the valve plate 17 with respect to the cam plate 23 and hence the pistons subjected to the high pressure fluid will not be those symmetrically disposed about the low point of the cam plate 23, but rather there will be more pistons on one side of the low point of the cam plate subjected to high pressure fluid than on the other side of the low point of the cam plate. This imbalance of force will tend to rotate the cam plate in the same direction that the valve plate is being rotated. However, the load 74 on the output shaft 64 tends to resist the rotating motion of the cam plate 23 because of the drive connection of the pinion 63 to the nutating plate 38 which is connected to the cam plate 23. Hence, no rotation of the cam plate 23 will occur until the valve plate 17 has been sufficiently rotated to generate enough torque on the cam plate through the imbalance of the pistons subjected to high pressure fluid to overcome the resistance of the load 74. When the valve plate 17 has rotated sufficiently to cause a force through the pistons acting on the cam plate 23 to overcome the resistance of the load 74 then the cam plate will commence rotating in the direction of and with the speed of the valve plate 17. The amount of the angular displacement of the valve plate 17 with respect to the cam plate 23 is referred to as the lead angle.

If the load 74 remains constant and the pressure of the high pressure fluid remains constant it is apparent that the speed of the cam plate 23, and hence the speed of the output shaft 64 will remain constant. If the magnitude of the load increases, then momentarily the speed of the output shaft will decrease, this will cause an increase in the lead angle of the valve plate until sufficient force is produced by the pistons acting on the cam plate 23 to cause the cam plate 23 to rotate at the speed of the tie rod 16. Conversely, if the load 74 should decrease, the speed of the output shaft would momentarily increase, this will decrease the lead angle of the valve plate and reduce the force exerted by the pistons on the cam plate 23, and the speed of the cam plate 23 will be the same as the speed of the tie rod 16. Within the operating limits of the torque multiplier, the speed of the cam plate 23 is the same as the speed of the valve plate 17, and the speed of the valve plate is regulated by the non-load carrying tie rod 16.

Similarly, the speed of the cam plate 23 will be the same as that of the tie rod 16 irrespective of pressure variations in the high pressure fluid. If the pressure increases, then the lead angle of the valve plate 17 will decrease due to a momentary increase in the speed of the cam plate 23 due to the additional force exerted thereon by the pistons subjected to high pressure and the condition of force equilibrium will be established at this new lead angle. If the pressure of the high pressure fluid decreases, the lead angle of the valve plate 17 will increase due to the reduced force on the cam plate 23, by the pistons subjected to high pressure and the condition

of force equilibrium will be established at this new lead angle.

From the foregoing it is apparent that the average speed of the cam plate is the same as the speed of the tie rod but there is no driving connection therebetween. Only enough force to overcome the frictional losses need be supplied to the tie rod 16 and a great load can be driven with very little force needed to drive the tie rod 16.

Another feature of this invention is that the amount of high pressure fluid required to drive the cam plate 23 is proportional to the lead angle which in turn is proportional to the magnitude of the load being driven. When the load is small the lead angle is small and hence a small amount of high pressure fluid is required. As the load increases the lead angle increases and hence the amount of high pressure fluid required increases. Similarly, with a constant load 74, as the pressure of the high pressure fluid increases the lead angle decreases reducing proportionally the amount of high pressure fluid needed.

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, the combination of a generally non-rotatable cylinder barrel having a plurality of cylinder bores and pistons reciprocable therein, a powered rotatable shaft, a valve plate carried by said shaft and rotatable therewith, said valve plate being positioned at one end of said cylinder barrel and having at least one port which sequentially registers with each cylinder bore as the shaft rotates, a cam plate arranged at the opposite end of the cylinder barrel and operatively associated with the pistons, said cam plate being drivingly connected to a power transmission means, and support means secured to said rotatable shaft to prevent axial while permitting rotational movement of cam plate with respect to the valve plate.

2. In an energy translating fluid pressure device, the combination as in claim 1 further characterized by an arcuate clearance drive means inter-connecting said cam plate with said shaft while providing limitations in the freedom of rotation of said cam plate with respect to the valve plate.

3. In an energy translating fluid pressure device, the combination as in claim 2 further characterized by the valve plate, the powered rotatable shaft, and a separate cam plate support sleeve being secured to rotate together; the cam plate being mounted with partial freedom of rotation with respect to valve plate, shaft and sleeve while the powered rotatable shaft acts as a tie rod connecting the valve plate and the cam plate support sleeve to absorb hydraulic reaction loads therebetween while connecting the valve plate and the cam plate at predetermined limits rotationally.

4. In an energy translating fluid pressure device, the combination as in claim 3 further characterized by anti-friction bearings interposed between the cam support sleeve and the cam plate whereby the cam plate may lead or lag according to load with minimum friction.

5. In an energy translating fluid pressure device, the combination as in claim 4 further characterized by an outer housing for the device and anti-friction bearings interposed between housing and powered rotatable shaft and for supporting power rotatable shaft, valve plate, cam plate, and cam plate support sleeve.

6. In an energy translating fluid pressure device, the combination as in claim 2, further characterized by said power transmission means including a second shaft and gear means thereon, and gear teeth arranged to be rotatable with cam plate, said gear teeth and said gear means being drivingly engaged.

7. In an energy translating fluid pressure device, the combination as in claim 6, further characterized by rota-

tive control means connected to the first shaft to move valve plate with resistance only equivalent to the bearing losses.

8. In an energy translating fluid pressure device, the combination as in claim 2, further characterized by the pistons having part-spherical piston ends, piston shoes engaging said part-spherical piston ends, means arranged around said cam plate and piston shoes and for securing the shoes against the cam plate during intake stroke, gear teeth rotative with the cam plate, a pinion arranged to be driven by said teeth, and a second shaft arranged to be driven by said pinion to provide output torque from the device, while the first mentioned shaft serves to interconnect the valve plate and cam plate longitudinally and also serves to permit application of a rotative control input to the valve plate.

9. A torque converting device comprising, in combination, a power unit and a converting unit, said power unit having a housing including an internal chamber, a valve plate shaft extending into said chamber and journaled for rotation in said housing, power means drivingly connected to said valve plate shaft to rotate said valve plate shaft, a valve plate carried by and rotatable with the valve plate shaft in said chamber, a cylinder barrel non-rotatably mounted in said chamber, said cylinder barrel having a plurality of cylinder bores, each bore having a piston reciprocally mounted therein, said valve plate positioned adjacent the opening of said cylinder bores and having passage means to sequentially register with each of said bores, a cam plate surrounding said shaft and positioned to coast with said pistons, said valve plate shaft and said cam plate being rotatable independently, said cam plate and said valve plate shaft having coacting means to permit but limit relative rotation therebetween and limit axial movement therebetween, high pressure fluid inlet means communicating with the passage means of said valve plate, low pressure outlet means communicating with the chamber, said converting unit having a power transmission shaft journaled for rotation in said housing, said power transmission shaft being drivingly connected to said cam plate.

10. In the combination of claim 9, said coacting means of said cam plate and said valve plate shaft including an arcuate slot in said cam plate and a pin carried by said valve plate shaft projecting into said slot.

11. A torque converting device comprising, a housing having a cavity therein, a control shaft journaled in the housing and extending into the cavity, means defining a plurality of piston chambers generally paralleling the control shaft and disposed in spaced relationship therewith, said housing including a fluid inlet port, valve means connected to the control shaft and interposed between the inlet port and at least some of the piston chambers, the valve means sequentially registering with each piston chamber, said valve means being controllable by rotation of said shaft, a cam plate in the cavity surrounding said shaft, a piston in each of the chambers and coactable with the cam plate, an output shaft connected to the cam plate, and means interposed between the control shaft and the cam plate to permit relative rotation within a predetermined range and thereby permit the control shaft to rotate relative to the cam plate during operation whenever there is a variation of torque load and whenever there is a variation of the fluid under pressure supplied through said input port.

12. The device of claim 11 wherein the means defining the piston chambers is a barrel surrounding the shaft and disposed within said housing cavity, and wherein a means connects the barrel to the housing to prevent relative rotation therebetween.

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