June 22, 1965 T. BUDZICH 3,190,232

HYDRAULIC APPARATUS

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FIG. 1

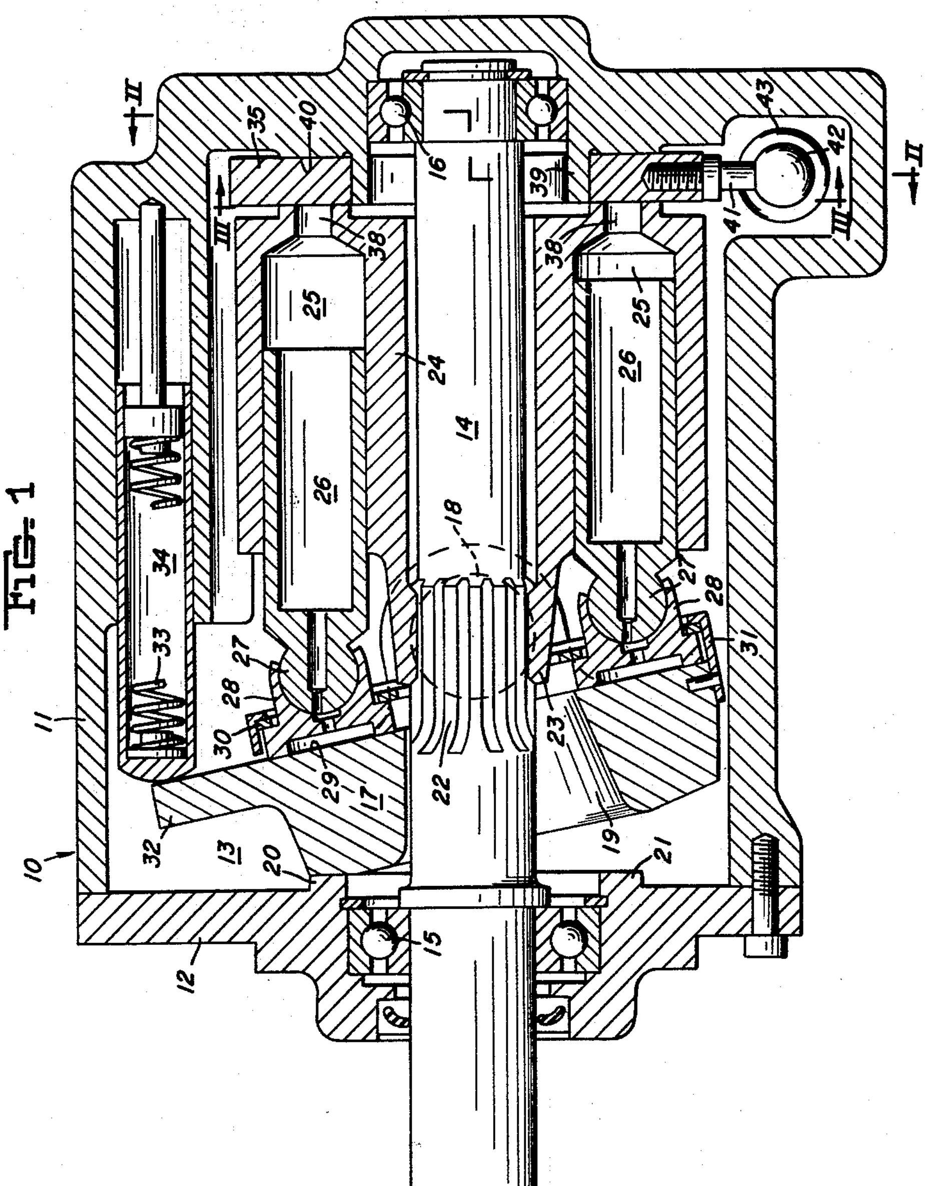

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FIG. 2

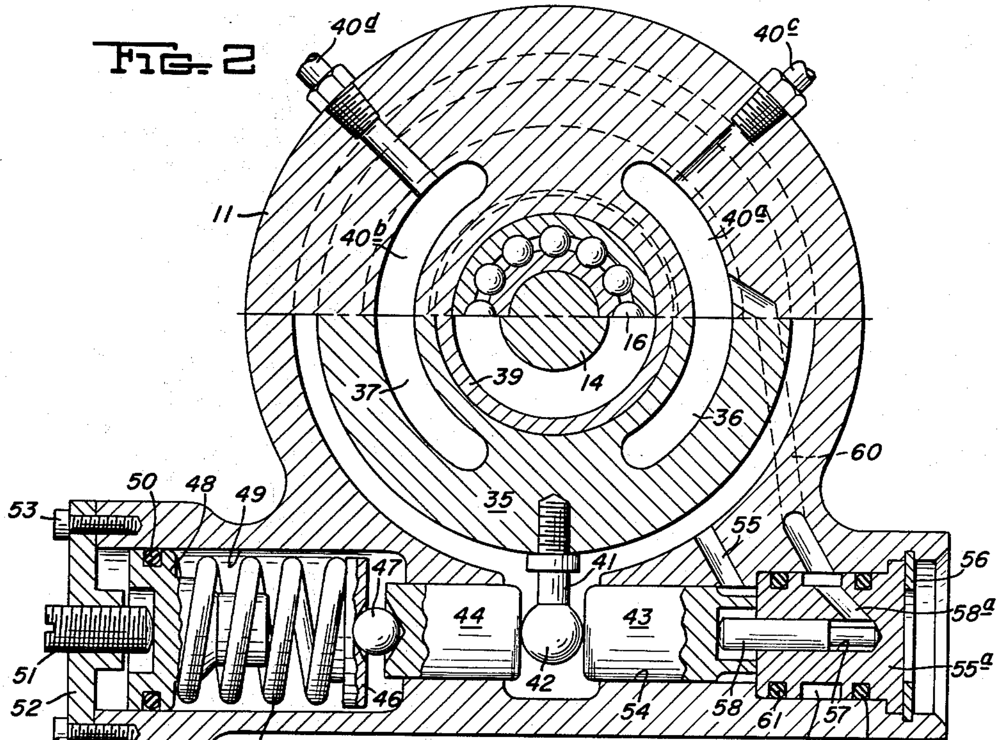


FIG. 3a

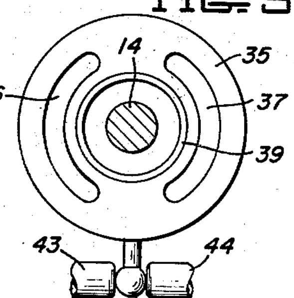


FIG. 4a

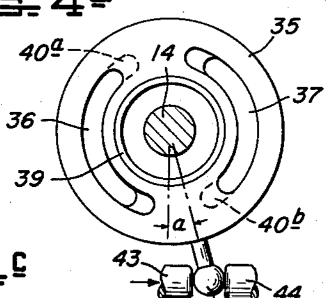


FIG. 3c

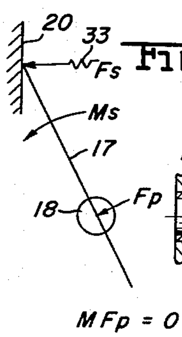


FIG. 3b

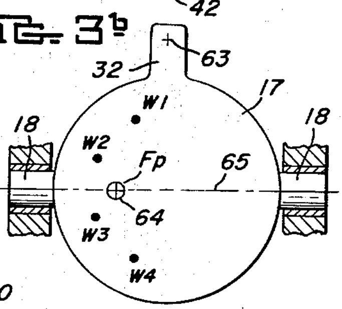


FIG. 4c

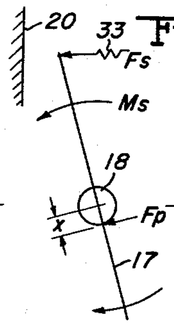
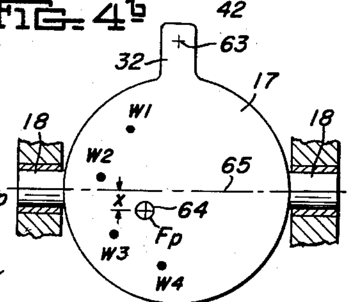


FIG. 4b



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HYDRAULIC APPARATUS

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12 Claims. (Cl. 103—162)

This invention relates to controls for variable displacement pumps and more particularly to a constant pressure control for a variable displacement pump.

In still more particular aspects this invention relates to a constant pressure control for a variable displacement pump in which the volume changing mechanism is actuated through the forces developed in the pumping mechanism.

Axial piston pumps employing tiltable cam plates to vary the pump displacement are well known in the art. The tilting of the cam plate is usually achieved by a fluid pressure piston cylinder arrangement for tilting the cam plate against the bias of a spring. A pressure signal from a control is introduced into fluid cylinder to vary the pump displacement. This type of arrangement suffers from the following disadvantages. The size of the control pistons used in tilting the cam plate is limited by the size of an acceptable envelope. This limits the level of the operating force transmitted to the cam plate resulting in a comparatively slow responding control. With large control pistons, although the actuating force level is substantially increased, time is lost in supplying large volume of oil to actuate such a control which also results in a slow responding control.

In the present invention the tilting of the cam plate is obtained without the use of a conventional control piston cylinder arrangement directly operating the cam plate, but rather by changing the centroid of the pumping reaction forces normally developed on the tiltable cam plate in respect to the axis of rotation of the tiltable cam plate. The centroid of the reaction pumping forces is moved by rotating (through a small angle) a control plate interposed between the valve plate and cylinder barrel of the pump. In this way large tilting forces can be developed on the cam plate while utilizing the basic components of the pumping mechanism.

Pumps employing the principle of valve plate rotation to divert the oil from the pumping cylinders to the suction cylinders are well known in the art. In those devices when rotating the valve plate through 90° while maintaining the stroke of the pistons constant, a change from zero to maximum effective pump displacement is possible.

In the present invention a control plate between the barrel and valve plate is rotated through a very small angle not to bypass the fluid from the pump discharge to the suction zone, but rather to tilt the cam plate by unbalancing the pumping reaction forces acting on the cam plate in respect to the axis of its suspension.

It is therefore an object of the present invention to provide a constant pressure control of a variable displacement pump in which the signal to change pump displacement to maintain a constant output pressure changes the force distribution in the pumping mechanism to actuate the displacement changing mechanism.

Another object of this invention is to provide a displacement changing mechanism of a variable displacement pump which is actuated by the pumping forces developed inside the pumping cylinders.

Still another object of this invention is to provide actuation of the displacement changing mechanism of a pump by moving the centroid of the pumping forces developed inside the pumping cylinders in respect to the axis of suspension of the cam plate, whereby large turning moments can be developed in the displacement changing mechanism.

These and other objects as well as complete understanding of the invention will be apparent from a consideration of the following description and drawings in which:

FIGURE 1 is a longitudinal sectional view through an axial piston variable displacement pump embodying this invention.

FIGURE 2 is a sectional view taken substantially along the line 2—2 of FIGURE 1 showing the features of the control.

FIGURES *3a*, *3b* and *3c* are diagrammatic representations of the forces developed on the cam plate by the position of the control plate when the pump is delivering full flow; and

FIGURES *4a*, *4b* and *4c* are diagrammatic representations of the forces developed on the cam plate by the position of the control plate rotated through a small angle to reduce the flow of the pump.

Referring now to the drawings and particularly to FIGURES 1 and 2, a pump generally designated as **10** is shown having a housing **11** and a cover **12**, which together define a chamber **13**. A shaft **14** extends into the chamber **13** and is journalled by bearings **15** and **16**, located in the cover **12** and housing **11** respectively. A trunnion **17** is journalled on a pin **18** carried by the housing **11** and is provided with an opening **19** through which the shaft **14** extends. Angular tilting movement of trunnion **17** is limited by stops **20** and **21** provided on the cover **12**. The shaft **14** is equipped with splines **22** drivingly engaging a splined extension **23** of a cylinder barrel **24**. The cylinder barrel **24** has a multiplicity of cylinder bores **25** in which pistons **26** are disposed. Pistons **26** have part spherical ends **27** universally mounting piston shoes **28** which operationally engage flat face **29** of the trunnion **17**, the trunnion thus operating as a cam plate. Piston shoes **28** are maintained against the trunnion by a nutating plate **30** and retainer **31**. The trunnion **17** has an extension **32** against which a biasing spring **33** works through a guide **34**. The cylinder barrel **24** works in abutment with a control plate **35** having kidney shaped timing slots **36** and **37** (FIGURE 2). Cylinder bores **25** each terminate in kidney shaped slots **38** which communicate with timing slots **36** and **37**. The control plate **35** is journalled for limited rotation in respect to the housing **11** by a tubular support **39** and works in abutment with a sealing face **40** formed on the end of the housing **11**. Sealing face **40** is provided with a pair of kidney shaped passages **40*a*** and **40*b*** similar to timing slots **36** and **37**. Passage **40*a*** is connected to high pressure port **40*c*** and passage **40*b*** is connected to low pressure port **40*d***. The control plate **35** has an actuating arm **41** equipped with a part spherical end **42** engaging a control piston **43** and a biasing piston **44**. The biasing piston **44** is biased by a control spring **45** through control spring guide **46** and a ball **47**. The control spring **45** is located by control stop **48** which is slidably mounted in control spring bore **49** and sealed by an O ring **50**. An adjusting screw **51** engaging the control stop **48** is retained in a cover **52** mounted by bolts **53** to the pump housing **11**. The control spring bore **49** is vented by a passage not shown to chamber **13**. The control piston **43** is slidably guided in control bore **54** which is vented by passage **55** to chamber **13**. A plug **55*a***, retained in the pump housing **11** by snap ring **56**, is equipped with a cylinder bore **57** in which is slidably disposed force piston **58**. The cylinder bore **57** is connected by passage **58*a***, annular space **59** and drilling **60** (shown partially in broken lines) with the high pressure timing slot **36** located in control plate **35** through port **40*a***. Annular space **59** is suitably sealed by O rings **61** and **62**.

Operation

The trunnion **17** under the action of biasing spring **33** will tend to rotate around pin 18 and engage stop 20 arriving at a position as shown in FIGURE 1. The rotary motion from the shaft 14 is transmitted through splines 22 to cylinder barrel 24. The rotation of cylinder barrel 24, in a well known manner, will induce the reciprocating motion in the pistons 26 which with piston shoes 28 will follow the flat surface 29 of trunnion 17. The reciprocating motion of the pistons 26 will displace fluid in cylinder bores 25 causing a pumping action. The displaced fluid is phased in timed relation to timing slots 36 and 37 by the rotation of the cylinder barrel 24, working in abutment with the control plate 35. Since the inlet and outlet pump ports 40*c* and 40*d* are connected with the timing slots 36 and 37 through slots 40*a* and 40*b*, a pumping action will be originated by the rotating shaft 14. The volume of the fluid taken from and displaced into the timing slots 36 and 37 per one revolution of the shaft 14 will be proportional to the length of the stroke of the pistons 26 and therefore proportional to the angle of inclination of the trunnion 17 in respect to the axis of rotation of the shaft 14. The angular inclination of the trunnion 17 as shown in FIGURE 1 corresponds to the maximum displacement of the pump. When the trunnion 17 is rotated in a clockwise direction around pin 18 to a position where the flat face 29 is normal to the axis of rotation the pump output will become zero.

The angular position of the trunnion 17 is dependent upon the balance between the moment of the reaction forces of the pistons on the trunnion and the moment caused by the force exerted by the spring 33. The control plate 35 is provided and adapted to change the net moment of the reaction force of the pistons on the trunnion by changing the centroid of these forces with respect to the axis of rotation (tilt) of the trunnion. This change in the net moment of the reaction forces of the pistons is induced by rotation of the control plate 35. Rotation of the control plate will change the position of the timing slots 36 and 37 with respect to the axis of rotation (tilt) of the trunnion which will in turn change the position of the net resulting moment resulting in a tilting or turning movement. This change in net moment is shown diagrammatically in FIGURES 3*a*, 3*b* 3*c*, 4*a*, 4*b*, and 4*c*.

In FIGURE 3*b* the forces acting on the trunnion 17 are shown, the axis of rotation (tilt) of the trunnion being used as a reference line designated by the reference character 65. The extension of the trunnion 17 is subjected to a force from the biasing spring 33, this force being designated as F_s working at point 63. The points W1, W2, W3, and W4 denote the reaction forces transmitted to trunnion 17 by pressure developed on the pistons 26 which are on the pumping stroke. The force F_p located at point 64 denotes the resultant force obtained from summation of forces W1, W2, W3 and W4 and constitutes the total hydraulic reaction force transmitted to the trunnion 17 by the pumping pistons 26. FIGURE 3*a* shows the angular relationship of the control plate 35 and timing slots 36 and 37 to provide the forces on the trunnion 17 in the position shown in FIGURE 3*b*. FIGURE 3*c* shows diagrammatically the force distribution acting on the trunnion 17 with control plate 35 angularly disposed as shown in FIGURE 3*a* and the trunnion angularly disposed as shown in FIGURE 1. Since in this position of the control plate 35 the point 64 of resultant force F_p coincides with the axis of rotation 65 of the trunnion pin 18 of trunnion 17 it will be directly carried in the form of a bearing pressure on the pin 18 without inducing any turning moment to the trunnion 17. The force F_s, transmitted to the trunnion 17 from the biasing spring 33 will produce a turning moment M_s around the trunnion pin 18 maintaining the trunnion 17 against stop 20. Referring now to FIGURE 4*a* the control plate 35 is shown having been rotated by an actuating arm 41 through a small angle $a°$. The resultant rotation of the timing slot 36 angularly displaced the position of force W1, W2, W3 and W4 acting on trunnion 17 in respect to its axis of rotation 65 which moves the position 64 of the resultant force F_p away from trunnion axis of rotation 65. FIGURE 4*c* shows diagrammatically the force distribution acting on the trunnion 17 with control plate 35 angularly disposed as shown in FIGURE 4*a*. Since the position 64 of resultant force F_p is now displaced by a distance x from the axis of rotation 65 of the trunnion 17, it will induce a counter clockwise turning moment MF_p as shown which will decrease the trunnion angle with respect to the pump axis when it exceeds moment M_s thus decreasing the pump output. Rotation of the trunnion about the pin 18 will change the magnitude of the moment M_s by compressing the spring 33. This will establish a balance between the moment M_s and moment M_p for any given position and magnitude of resultant force F_p. The amount of rotation (tilt) caused by any given turning moment MF_p will be dependent upon the force of biasing spring 33 and the trunnion 17 will be rotated in a clockwise direction away from the stop 20, thus reducing the pump output. Thus a small rotation of the control plate 35 will change the position of the resultant force F_p acting on the trunnion 17 in relation to the axis of rotation 65 of trunnion 17 regulating its angular inclination in respect to the pump axis against a register of biasing spring 33. The change in angular inclination of trunnion 17 will proportionally change the displacement of the pump. To increase the volume when the pump is delivering a reduced volume, the control plate is rotated the other way which will move the position of the resultant force F_p back toward the axis 65. And, if fast response is desired the control plate can be configured and arranged to move the force F_p past the axis 65.

The angle of the control plate 35 is regulated by a control signal which is developed by and proportional to the output pressure of the pump. This pressure signal is developed in the pumping mechanism and is transmitted to the control mechanism shown in FIGURE 2. The control plate 35 is connected to actuating arm 41, equipped with spherical end 42. Fluid under pressure from the timing slot 36 which is the high pressure slot is conducted through the drilling 60 to annular space 59 and through passage 58*a* to cylinder 57 slidably engaging force piston 58. Since the end of force piston 58 in contact with the control piston 43 is subjected to a pressure in control cylinder 54 which is vented by passage 55 to chamber 13, the force piston 58 working as a fluid motor will transmit an axial force to control piston 43 proportional to the pressure difference between timing slot 36 and chamber 13. This force acting through the spherical end 42 of actuating arm 41 will tend to induce a turning moment in control plate 35 journaled on tubular support 39. The biasing piston 44 will transmit to spherical end 42 of actuating arm 41 an opposing biasing force of control spring 45. With the force developed due to pressure existing in timing slot 36 on cross-section area of force piston 58 smaller than the preload in the control spring 45, the control piston 43 will rest against the plug 55*a* and the control plate 35 will be maintained in a position as shown in FIGURES 2 and 3*a*. This is a position equivalent to maximum pump displacement as shown in FIGURE 1. Once the force developed on the force piston 58 is sufficient to overcome the preload in the control spring 45, the control plate 35 will be rotated in a clockwise direction, the angle of rotation being proportional to the rise in pressure in timing slot 36. As explained when referring to FIGURES 3*a*, 3*b*, 3*c*, 4*a*, 4*b* and 4*c* rotation of the control plate 35 will change the equilibrium of forces acting on the trunnion 17 turning it towards a position of reduced pump displacement. In this way, the pump will supply the maximum flow until the pressure developed in timing slot 36 reaches a predetermined value equivalent to the preload in the control spring 45, which is adjustable by adjusting screw 51. Once this pressure is reached the pump will automatically reduce its flow with rise in discharge pressure, the flow of the pump becoming zero with the trunnion 17 reaching a perpendicular position in respect to axis of rotation of the shaft 14. Since the resultant force F_p in a pump constitutes the total hydraulic pumping reaction and therefore is very large, the unbalanced forces developed on the stroke changing mechanism, regulating the output of the pump are very large producing a fast responding control.

While one embodiment of this invention has been shown and described, various modifications and adaptations can be made without departing from the spirit and scope of the appended claims.

What is claimed is:

1. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction, valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing means interposed between said stationary support means and said cylinder barrel, mounting means for said fluid distributing means to permit movement of said fluid distributing means in respect to said tiltable cam plate to change the force moment of the pistons on the cam plate in respect to the axis of tilt of the cam plate, and actuation means to impart movement to said fluid distributing means to change the cam angle of the tiltable cam plate in response to change in the force moment of the pistons coactable with the cam plate.

2. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction; valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing means interposed between said stationary support means and said cylinder barrel, means journalling said fluid distributing means for rotation in respect to said cam plate to change the force moment of the pistons on the cam plate in respect to the axis of tilt of the cam plate, and actuation means to rotate said fluid distributing means to change the cam angle of the tiltable cam plate in response to change in the force moment of the pistons coactable with the cam plate.

3. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction, valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing means interposed between said stationary support means and said cylinder barrel, said distributing means including fluid conducting passages defining high and low pressure zones communicating with said inlet and outlet ports, means journalling said fluid distributing means for rotation in respect to said cam plate to change the force moment of the pistons on the cam plate in respect to the axis of tilt of the cam plate, and actuation means to rotate said fluid distributing means to change the cam angle of the tiltable cam plate in response to change in the force moment of the pistons coactable with the cam plate.

4. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction, valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing means interposed between said stationary support means and said cylinder barrel, said distributing means including fluid conducting passages defining high and low pressure zones communicating with said inlet and outlet ports, a plurality of said pistons being connected to the high pressure zone with each exerting a force on said cam plate, the resultant force of said pistons acting on the cam plate being located by the position of the high pressure zone, means journalling said fluid distributing means for rotation in respect to said cam plate to change the position of said high pressure zone with respect to the axis of tilt of the cam plate, and actuation means to rotate said fluid distributing means to change the position of said resultant force in respect to the axis of tilt of said cam plate.

5. The device of claim 4 further characterized by the resultant force of the pistons which are communicating with the high pressure zone being movable in respect to axis of tilt of the cam plate with rotation of the fluid distributing means from said axis to at least one side thereof.

6. The device of claim 5 further characterized by the biasing means being located on the other side of the axis of tilt from the side to which said resultant force is moved.

7. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction, valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing means interposed between said stationary support means and said cylinder barrel, mounting means for said fluid distributing means to permit movement of said fluid distributing means in respect to said tiltable cam plate to change the force moment of the pistons on the cam plate in respect to the axis of tilt of the cam plate, fluid actuated control means to impart movement to said fluid distributing means to change the cam angle of the tiltable cam plate in response to change in the force moment of the pistons coactable with the cam plate.

8. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction, valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing means interposed between said stationary support means and said cylinder barrel, mounting means for said fluid distributing means to permit movement of said fluid distributing means in respect to said tiltable cam plate to change the force moment of the pistons on the cam plate in respect to the axis of tilt of the cam plate, pressure responsive control means adapted to move said distributing means responsive to change in the fluid pressure in device to vary the cam angle of the cam plate to maintain a substantially contant fluid pressure at the outlet port.

9. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction; valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing

means interposed between said stationary support means and said cylinder barrel, said distributing means including fluid conducting passages defining high and low pressure zones communicating with said inlet and outlet ports, a plurality of said pistons being connected to the high pressure zone with each exerting a force on said cam plate, the resultant force of said pistons acting on the cam plate being located by the position of the high pressure zone, means journalling said fluid distributing means for rotation in respect to said cam plate to change the position of said high pressure zone with respect to the axis of tilt of the cam plate, and actuation means to rotate said fluid distributing means, said actuating means including a fluid actuated motor operably connected to said distributing means to rotate said distributing means in one direction, and second biasing means operably connected to said distributing means opposing the action of said fluid motor.

10. The combination of claim **9** further characterized by said fluid motor including a fluid operated piston and said second biasing means including a spring.

11. A fluid pressure energy translating device comprising a housing having inlet and outlet ports, a cylinder barrel rotatably mounted in said housing, pistons reciprocally disposed in said cylinder barrel, a cam plate mounted for tiltable movement in respect to the housing about an axis, biasing means normally urging said cam plate to tilt in one direction, valving means to sequentially connect said pistons with the inlet and outlet ports, said valving means including stationary support means, and fluid distributing means interposed between said stationary support means and said cylinder barrel, said fluid distributing means including first and second arcuate slots defining respectively high and low pressure zones communicating with the outlet and inlet ports, means on the support means journalling said fluid distributing means for rotation with respect to said cam plate, said fluid distributing means abutting said support means and said cylinder barrel, control means to impart rotation to said fluid distributing means, said control means including a fluid operated piston operably connected to said distributing means to rotate the distributing means in one direction and spring means operably connected to said distributing means opposing said fluid operated piston.

12. The device of claim **11** characterized by fluid conducting means connecting said fluid operated piston to the high pressure zone of the device to operate said fluid operated piston responsive to the pressure of the device, whereby the fluid distributing means are rotated to vary the cam angle of the cam plate to maintain substantially constant fluid pressure at the outlet port.

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