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T. BUDZICH
ENGINE

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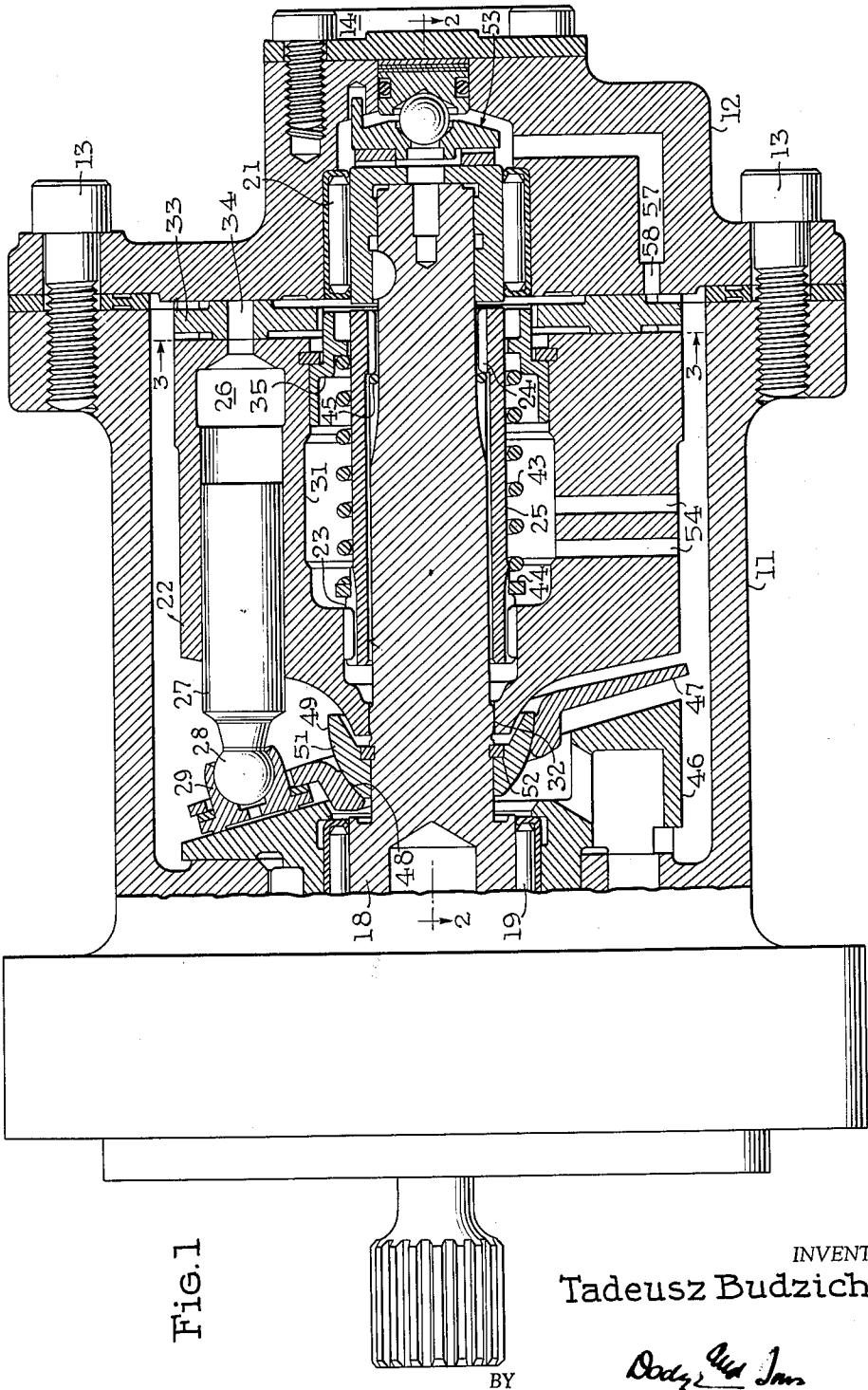


Fig. 1

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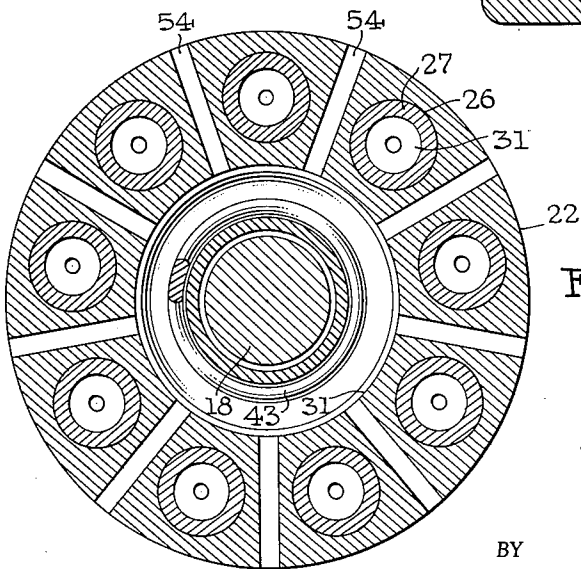
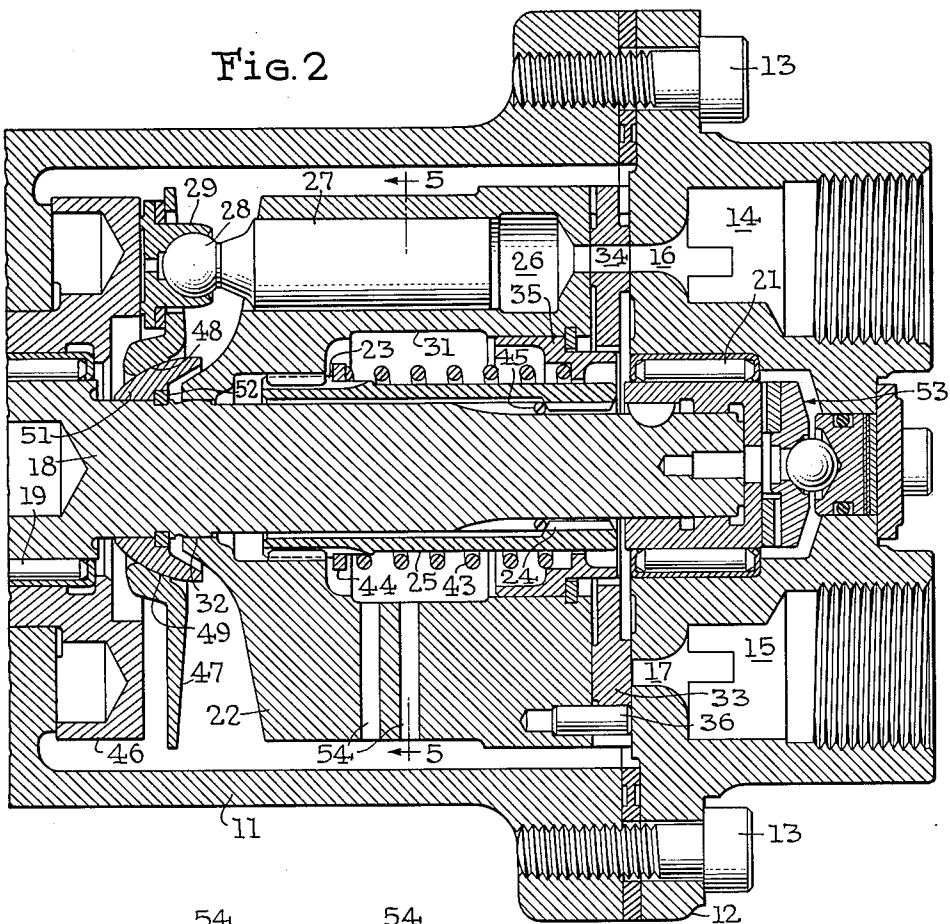
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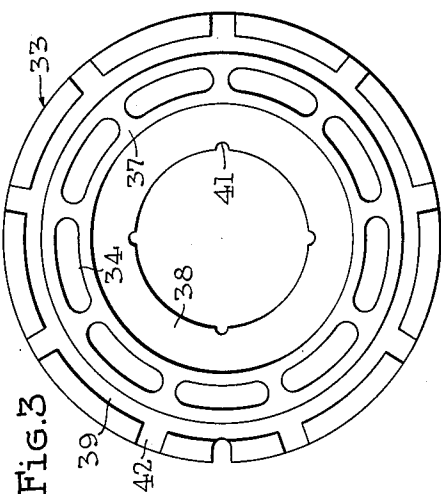
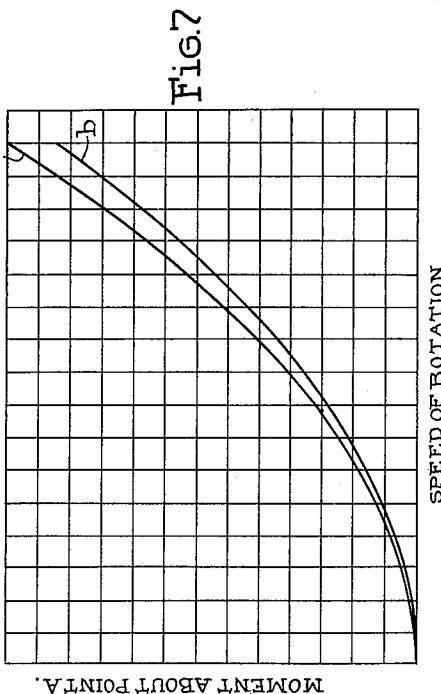
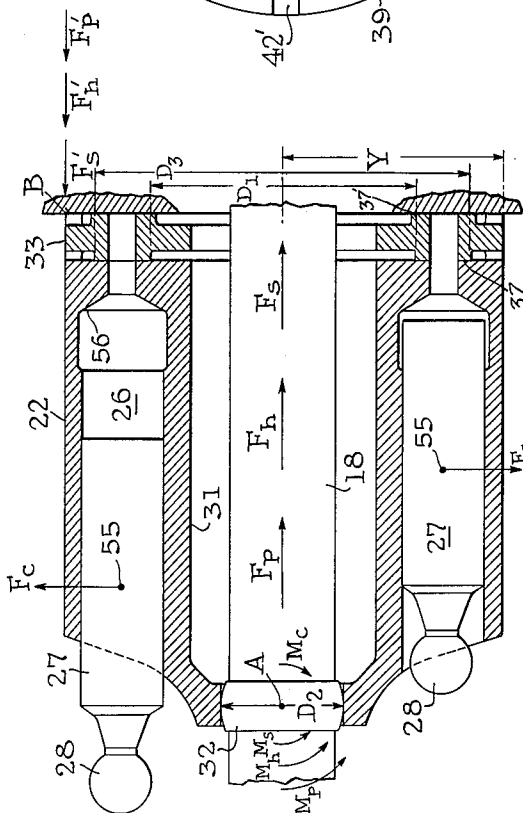
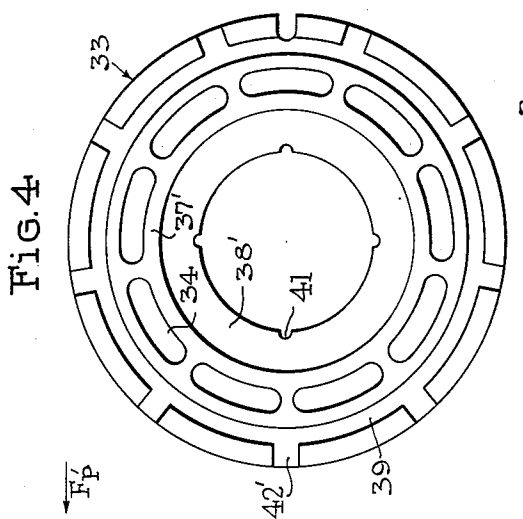


Fig. 6

Fig. 3

Fig. 7

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2 Claims. (Cl. 103-162)

This invention relates to fluid pressure engines and more particularly to engines of the rotary cylinder barrel longitudinally reciprocating piston type. While the invention could be used in pumps as well as motors, it finds its greatest utility in motors. For this reason, it will be described in relation to engines serving in this capacity.

In motors of this type, the pistons are carried by the rotary cylinder barrel in a circular series of longitudinal cylinder bores which sequentially register with inlet and exhaust ports in a valve member as the barrel rotates. The pistons usually coast with an inclined cam plate which moves them on their discharge strokes in timed relation to said sequential registration. Inherent in these devices is the fact that at any instant of time, the centers of gravity of the pistons lie in a plane extending in a direction parallel with the face of the cam plate. Since the pistons are rotating with the cylinder barrel they are subject to centrifugal forces which, because of this inherent characteristic, produce a moment on the cylinder barrel. The centrifugal force moment varies in magnitude with both the displacement of the motor, i.e., the angle of inclination of the cam plate, and the speed of rotation of the cylinder barrel. In those engines having a valve member which abuts against an end face of the cylinder barrel and in which the cylinder barrel is mounted on the drive shaft for universal and longitudinal movement relatively to the valve member, this centrifugal force moment tends to tilt the cylinder barrel relatively to the valve plate thereby causing intolerable leakage.

The object of the present invention is to provide a device for counteracting the tilting tendency of the cylinder barrel under the action of the centrifugal force moment. In its preferred form, the invention comprises a closed cavity formed in the cylinder barrel, an inlet passage for delivering fluid to the cavity, opposed reaction surfaces located on the cylinder barrel, one being subject to the pressure in the cavity and urging the cylinder barrel away from the valve member and the other being subject to the pressure in the interior of the motor housing, and a centrifugal pump rotating in unison with the cylinder barrel and arranged to draw fluid from the cavity and discharge it into the housing. The areas of the two reaction surfaces are so selected that the pressure differential created by the centrifugal pump produces a net pressure force tending to urge the cylinder barrel toward the valve member. Since this pressure differential is proportional to the square of the speed of rotation, the net pressure force or counteracting force will vary with speed in the same manner as the centrifugal force. By proper selection of the magnitude of the counteracting force, it is possible to prevent tilting of the cylinder barrel at all speeds within the operating range.

In most engines of this type, a pressure land is provided between the cylinder barrel and the valve member so that oil from the high pressure port in the valve member which leaks radially inward and outward across the land forms a bearing film which minimizes metal-to-metal contact between these members. Another feature of the invention relates to an arrangement of the inlet passage, whereby the leakage fluid passing radially inward across the pressure land is transmitted to the cavity. In this way, the leakage fluid performs two separate functions.

Another feature of the invention relates to a circuit

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utilizing the pressure differential created by the centrifugal pump for forcing fluid through one or more of the bearings supporting the drive shaft. Since both the heat generated in the bearings and the rate of flow there-through vary directly with the speed of rotation, the invention provides a simple means of adequately cooling and lubricating these bearings.

The preferred embodiment of the invention will now be described in relation to the accompanying drawings, in which:

FIG. 1 is a partial axial sectional view of a constant displacement motor embodying the invention.

FIG. 2 is a sectional view taken on line 2-2 of FIG. 1.

FIG. 3 is a sectional view taken on line 3-3 of FIG. 1, showing the front face of the wear plate in reduced size.

FIG. 4 is a view showing the opposite or rear face of the wear plate.

FIG. 5 is a sectional view taken on line 5-5 of FIG. 2, showing the arrangement of the elements of the centrifugal pump.

FIG. 6 is a schematic diagram illustrating the various forces and moments treated in the present invention.

FIG. 7 is a graph showing the variations of the centrifugal force moment and the counteracting moment with speed of rotation of the cylinder barrel.

Referring to FIGS. 1 and 2, the motor comprises a housing having separable sections 11 and 12 formed with mating flanges which are connected together by bolts 13. Formed in housing section 12 are inlet and discharge ports 14 and 15 which communicate with conventional kidney ports 16 and 17 located in the inner face of this section and arranged symmetrically with respect to the longitudinal axis of the motor. A drive shaft 18, journaled in the housing by needle bearings 19 and 21, is connected in driving relation with the cylinder barrel 22 by splines 23 and 24 and torque tube 25. The cylinder barrel contains a circular series of nine longitudinal cylinder bores 26 which extend through the barrel and receive pistons 27. Each piston 27 carries a spherical head 28 at one end for universally supporting a piston shoe 29. The cylinder barrel contains an axial bore 31 which, at its left end, rests on a spherical enlargement 32 carried by the shaft and centered at the point of intersection of the axis of the shaft and the plane of the centers of spherical piston heads 28. This enlargement is in line contact with the surface of bore 31 and thus permits the cylinder barrel to tilt and move longitudinally relatively to the shaft. The method of supporting and driving the cylinder barrel is disclosed and claimed in applicant's application Serial No. 656,574, filed May 2, 1957, now Patent No. 2,925,046 granted Feb. 16, 1960. In accordance with the conventional practice in this art (see U.S. Patent 1,925,378 granted September 5, 1933), the housing is provided with a drain port (not shown).

Located between cylinder barrel 22 and housing section 12 is a wear plate 33 having front and rear faces which abut the cylinder barrel and housing section, respectively, and containing nine small kidney-shaped passages 34 which sequentially connect each cylinder bore 26 with kidney ports 16 and 17 as the cylinder barrel rotates. The wear plate 33 is located radially by a sleeve 35 carried by the cylinder barrel and is connected to the cylinder barrel by pin 36. Because of this arrangement, the wear plate rotates with the cylinder barrel but is free to move longitudinally relatively to it and to housing section 12.

As shown in FIGS. 3 and 4, the kidney passages 34 open through annular lands 37 and 37' formed on the front and rear faces, respectively, of the wear plate 33. These lands are defined by annular leakage grooves 38, 39, 38' and 39' which are coaxial with the axis of rotation of the motor. The inner grooves 38 and 38' are inter-

connected by longitudinal slots 41 and the outer grooves 39 and 39' are connected with the interior of housing section 11 by radial slots 42 and 42'. During operation, high pressure fluid in the kidney passages 34 leaks radially inward and outward across lands 37 and 37' thereby providing one hydraulic film between the cylinder barrel and the wear plate and another such film between the wear plate and housing section 12. The lands 37 and 37' are so dimensioned that their hydraulic films carry 85% and 95%, respectively, of the hydraulic reaction forces transmitted from the cylinder barrel 22 to housing section 12. This arrangement serves to maintain the cylinder barrel in contact with the wear plate and to maintain the wear plate in contact with housing section 12. A contact-maintaining force is also developed by spring 43 which reacts between sleeve 35 and a ring 44 carried by the spline teeth 23 formed on torque tube 25. The reaction force for spring 43 is transmitted through drive shaft 18, snap ring 45 and spline teeth 24.

The piston shoes 29 are in sliding contact with an inclined cam plate 46 which, together with nutating plate 47, functions to reciprocate pistons 27 in a known manner. The nutating plate 47 is formed with a central spherical recess 48 arranged to seat on the spherical outer surface 49 of the collar 51. The center of the spherical surface 49 is coincident with the center of spherical enlargement 32. Collar 51 is restrained against axial movement by split ring 52 which is seated in an annular groove formed in the drive shaft 18. The piston inertia loads which are transmitted from the nutating plate to the shaft by collar 51 and ring 52 are, in turn, transmitted to the housing section 12 by thrust bearing 53. This method of handling the piston inertia loads is disclosed and claimed in applicant's application Serial No. 665,387, filed June 13, 1957, now Patent No. 2,953,099 issued Sept. 20, 1960.

As seen in FIGS. 1 and 5, the cylinder barrel 22 contains a plurality of radial passages 54 which extends from axial bore 31 through the peripheral surface of the cylinder barrel. These passages function as a centrifugal pump and, when the cylinder barrel is rotated, draw fluid from the bore 31 and discharge it into the interior of housing section 11. The parts of the motor are so arranged that the leakage fluid passing radially inward across wear plate lands 37 and 37' is transmitted to the bore 31 through the radial clearance between sleeve 35 and torque tube 25.

When the motor is operating, the pistons 27 are subjected to centrifugal forces F_c which act through their centers of gravity indicated (in FIG. 6) at points 55. Since these centers of gravity always lie in a plane extending in a direction parallel with the surface of cam plate 46, it is apparent that these forces will produce a moment M_c on cylinder barrel 22 tending to tilt it in a clockwise direction (as viewed in FIG. 6) about an axis extending in a direction normal to the axis of drive shaft 18 and passing through the center A of spherical enlargement 32. Furthermore, since centrifugal forces are proportional to the square of the rotational speed, it will be apparent that the centrifugal force moment will vary with the speed of the motor in the manner indicated by curve *a* in FIG. 7.

In order for it to tilt relatively to housing section 12, the cylinder barrel 22 must move to the left (as viewed in FIG. 6) so that point B, on wear plate 33, can move into contact with and slide radially inward along the inner face of the housing section. This movement of the cylinder barrel is resisted by three separate forces which exert counteracting moments about point A. The first of these is the force developed by spring 43. This force, labeled F_s , acts along the axis of shaft 18 and, when tilting is impending, is balanced by an equal and opposite reaction force F_s' passing through the point B. The resulting counteracting moment about point A, M_s ,

equals $F_s' \times Y$ where Y is the radial distance between points A and B.

The second force is that produced by the fluid pressure within cylinder bores 26 acting against the shoulders 56 and urging the cylinder barrel toward the housing section 12. The resultant of these forces, force F_h , also acts along the axis of rotation and, like the spring force, is balanced by an equal and opposite reaction force F_h' which acts through the point B. The counteracting moment M_h produced by this force equals $F_h' \times Y$. Most engines of this type include the structure which develops the moments M_s and M_h and, during certain operating conditions, these two moments alone are sufficient to prevent tilting of the cylinder barrel. For instance, when the pressure in the cylinder bores 26 is high and the rotational speed of the cylinder barrel is low, the sum of the moments M_s and M_h exceeds the centrifugal force moment and tilting is prevented. However, when the motor is unloaded and consequently its rotational speed is high and the pressure in cylinder bores 26 is low, the centrifugal force moment predominates and tilting can occur. Of course, the spring force can be increased by using a larger spring, but this is an unsatisfactory solution because there is rarely sufficient room in the motor to accommodate a spring of adequate size and furthermore, even if sufficient space were available, the large spring force would produce undue friction between the wear plate and the housing section 12 which would make starting of the motor extremely difficult, if not impossible.

The third counteracting force, and the one provided by this invention, varies with rotational speed and thus prevents tilting of the cylinder barrel at high speeds without impairing starting torque characteristics. This force results from the pressure differential between the bore 31 and the interior of housing section 11 which is created by the centrifugal pump 54. The pressure within bore 31, P_0 , acts on a first reaction surface having an area equal to

$$\frac{\pi}{4}(D_1^2 - D_2^2)$$

where

D_1 is the diameter of the circle enclosed by the inner edge of wear plate land 37, and

D_2 is the diameter of spherical enlargement 32

and produces a force tending to move the cylinder barrel 22 away from the housing section 12. The pressure P_1 , in the interior of the housing acts on a second reaction surface, located on the front face (i.e., the face on the end opposite the wear plate 33) of the cylinder barrel, and having an effective area equal to

$$\frac{\pi}{4}(D_3^2 - D_2^2) - A_c$$

where A_c represents the combined cross-sectional areas of the cylinder bores 26 and D_3 is the diameter of the circle enclosed by the outer edge of wear plate land 37. The force acting on the second reaction surface tends to maintain the cylinder barrel in contact with the housing section 12 and the areas of the two reaction surfaces are so selected that this force always predominates. The net force, F_p , developed by these reaction surfaces can be expressed by the following equation:

$$F_p = P_1 \left(\frac{\pi}{4}(D_3^2 - D_2^2) - A_c \right) - P_0 \left(\frac{\pi}{4}(D_1^2 - D_2^2) \right)$$

which in turn can be expressed in the form

$$F_p = (P_1 - P_0) \left(\frac{\pi}{4}(D_1^2 - D_2^2) \right) + P_1 \left(\frac{\pi}{4}(D_3^2 - D_1^2) - A_c \right)$$

In most engines, the term

$$P_1 \left(\frac{\pi}{4}(D_3^2 - D_1^2) - A_c \right)$$

in this equation is small and can be neglected and, there-

fore, the net force F_p is basically a function of the pressure differential $P_1 - P_0$. Since the pressure differential created by a centrifugal pump is proportional to the square of the speed, this counteracting force F_p will vary in the same way as the centrifugal forces F_c . The force F_p , like the forces F_h and F_s , acts along the axis of rotation, is balanced by an equal and opposite reaction force F_p , acting through point B, and exerts a moment M_p about the point A equal to $F_p \times Y$. The variation of this moment with speed is represented by curve *b* in FIG. 7. From the graphs in this figure, it will be seen that the centrifugal force moment, M_c , is greater than the counteracting moment, M_p , but the difference between these two moments is more than made up by the moments M_h and M_s . By properly selecting the areas of the reaction surfaces on which the pressures P_0 and P_1 act, the tendency of the cylinder barrel to tilt under the action of the centrifugal force moment may be counteracted or prevented over the entire speed range of the motor.

In addition to creating a counteracting moment on the cylinder barrel, the pressure differential established by the centrifugal pump is also used to supply cooling oil to two of the shaft bearings. This is accomplished by providing a passageway 57 in housing section 12 which transmits fluid from the interior of the housing section 11 to bore 31 through the thrust bearing 53, the radial bearing 21, and the radial clearance between torque tube 25 and sleeve 35. Since the heat generated in these bearings increases with the speed of rotation and since the rate of flow through the bearings depends on the pressure differential created by the centrifugal pump, which also increases with the speed of rotation, the invention affords a satisfactory and simple means for adequately cooling these two bearings. If desired, a similar arrangement for cooling the front radial bearing 19 could be employed.

It should be noted that the centrifugal pumping action of the cylinder barrel represents an expenditure of energy; the energy consumed varying directly with the pressure differential between bore 31 and the interior of the housing, and with the rate of flow through radial passages 54. Since a low rate of flow (on the order of several cubic inches per minute) is sufficient to adequately cool bearings 21 and 53, the overall efficiency of the motor can be increased, without risk of overheating of the bearings, by limiting the flow through passageway 57. In the illustrated embodiment, this is accomplished by metering orifice 58.

It should also be observed that the centrifugal pump cannot create a pressure differential between bore 31 and the interior of housing section 11 if these two spaces are in free communication with each other. In the preferred embodiment of the invention, such communication is prevented by the mating surfaces of bore 31 and spherical enlargement 32 (between which there is a radial clearance of about 0.0005 inch), by the mating faces of cylinder barrel 22, wear plate 33 and housing section 12, and by the restriction to flow through passageway 57 which is afforded by orifice 58, bearings 21 and 51, and the radial clearance between sleeve 35 and torque tube 25.

As stated previously, the drawings and description relate only to a preferred embodiment of the invention. Since many changes can be made in this embodiment

without departing from the inventive concept, the following claims should provide the sole measure of the scope of the invention.

What is claimed is:

1. In an engine of the type including a housing, a drive shaft, bearings mounted in the housing for supporting the drive shaft, a rotary cylinder barrel driven by the shaft and mounted thereon for universal and longitudinal movement, a circular series of longitudinally reciprocable pistons mounted in the cylinder barrel, a distributing valve having mating faces carried by the cylinder barrel and by the housing which extend in planes perpendicular to the axis of rotation, high and low pressure ports formed in the valve face carried by the housing, and a pressure land formed in one of the valve faces for supporting an hydraulic film produced by the leakage of high pressure fluid radially inward and outward across the land, and in which the pistons are moved on their discharge strokes by a cam plate inclined with respect to the axis of rotation of the cylinder barrel, whereby the centers of gravity of the pistons lie in a common plane inclined with respect to a plane normal to said axis so that the centrifugal forces generated by the pistons produce a tilting moment on the cylinder barrel, the improvement which comprises a closed cavity formed in the cylinder barrel and isolated from free communication with said high and low pressure ports, the cavity encircling the axis of rotation; a flow passage for transmitting to the cavity the fluid which flows radially inward across the pressure land; a centrifugal pump driven in unison with the cylinder barrel and arranged to draw fluid from the cavity and discharge it into the interior of the housing; and two opposed reaction surfaces on the cylinder barrel, one of the surfaces being subject to the pressure in the housing and arranged to produce a force which urges the valve faces into contact with each other, and the other surface being subject to the pressure in the cavity, the areas of the reaction surfaces being so correlated with the pressure differential created by the centrifugal pump that the force acting on said one reaction surface predominates, whereby the net pressure force developed by the reaction surfaces counteracts the tendency of the cylinder barrel to tilt under the action of the centrifugal force moment generated by the pistons and the mating faces of the distributing valve are maintained in engagement.

2. The improvement defined in claim 1 including a restricted passage connecting the cavity with the interior of the housing, said passage passing through at least one of the shaft bearings.

References Cited in the file of this patent

UNITED STATES PATENTS

1,749,682	Weldy	Mar. 4, 1930
2,292,125	Ifield	Aug. 4, 1942
2,429,005	Watson et al.	Oct. 14, 1947
2,661,700	Towler et al.	Dec. 8, 1953
2,674,191	Ifield	Apr. 6, 1954
2,733,666	Poulos	Feb. 7, 1956
2,776,628	Keel	Jan. 8, 1957

FOREIGN PATENTS

311,938	Great Britain	May 23, 1929
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