Oct. 2, 1962 N. F. PEDERSEN ET AL 3,056,358

AXIAL PISTON PUMP WITH BALANCED RADIAL BEARING

Filed Jan. 20, 1960 2 Sheets-Sheet 1

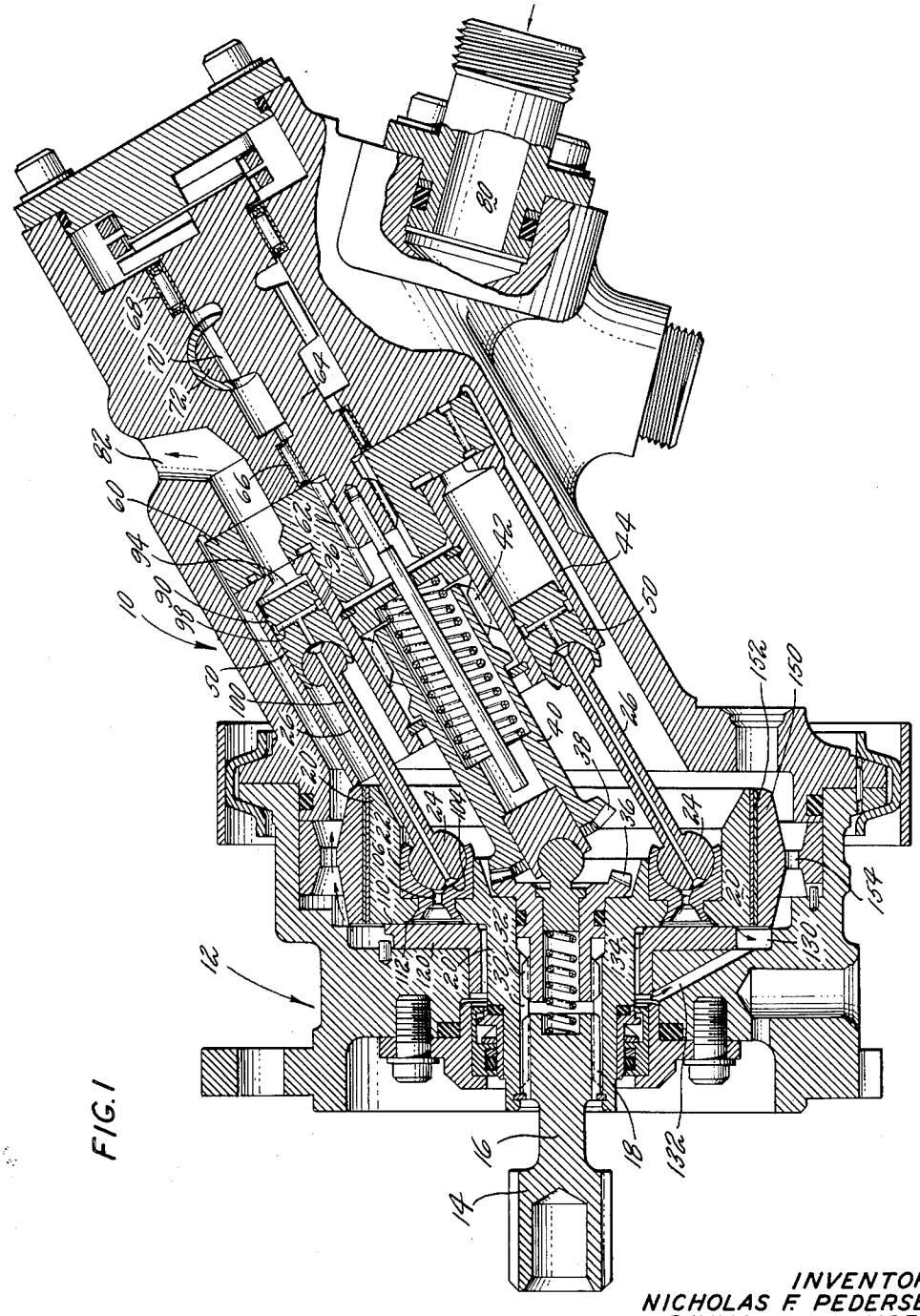


FIG. 1

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FIG. 3

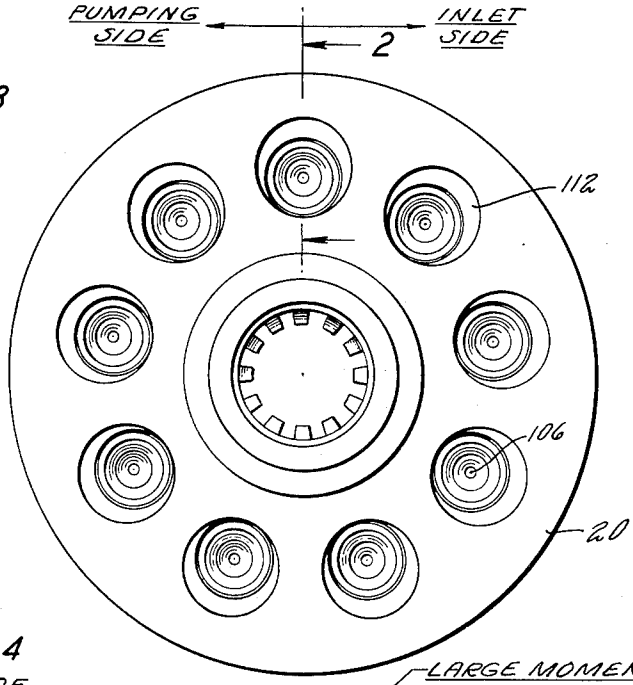



FIG. 4

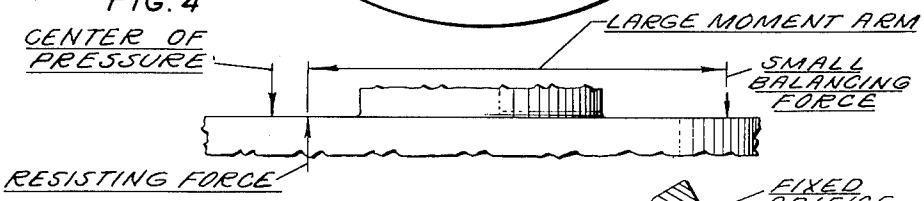



FIG. 2

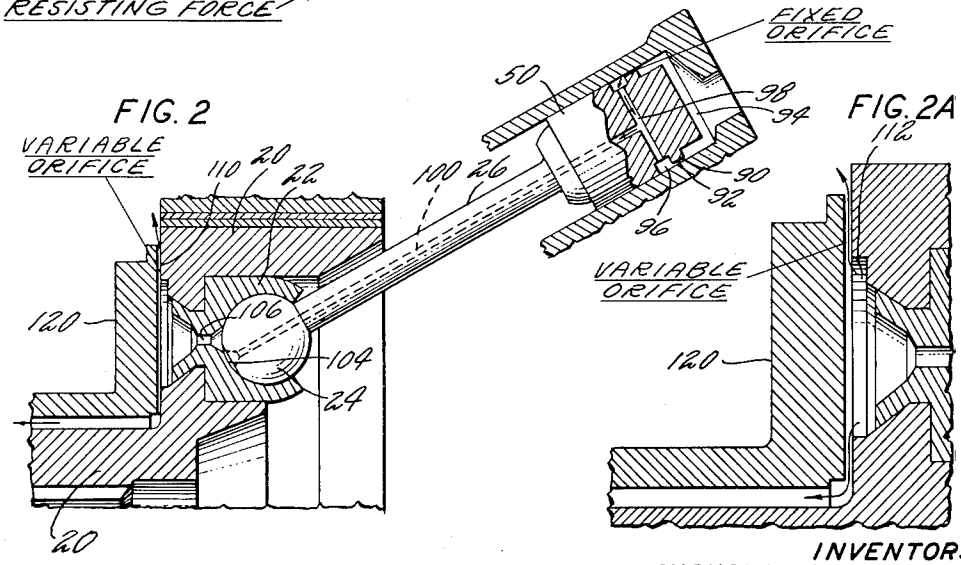



FIG. 2A




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# AXIAL PISTON PUMP WITH BALANCED RADIAL BEARING

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This invention relates to piston pumps and more particularly to fluid pumps having a rotor carrying the pistons and in which the pistons reciprocate.

It is a feature of this invention to provide a pressure-balancing arrangement whereby fluid under pressure is directed from a high-pressure source through a fixed laminar restriction and to the antipumping end of the piston where it is conducted to a reaction surface such as a bearing so that bearing forces are reduced.

It is another feature of this invention to provide the above pressure balancing in an axial piston pump having a fixed reaction bearing surface.

It is another feature of this invention to provide a predetermined spacing between the piston and piston walls to form a fixed restriction which connects in turn to an annulus around the piston for subsequent communication with passages in the piston rod which lead to the reaction surface.

It is still a further feature of this invention to provide means for maintaining a fluid film between the fixed reaction bearing surface and unitary drive plate surface so that the high friction normally associated with bearing surfaces is reduced to a minimum during starting regimes of the pump.

It is another feature of this invention to provide a unitary drive plate that has axial motion relative to the fixed thrust bearing so that relative distance therebetween forms a variable fluid escape path.

It is another feature of this invention to provide an axial piston pump having a thrust member which operatively receives the antipumping ends of the piston rods and includes a surface comprising a plurality of pressure wells receiving the metered fluid under pressure for pressure balancing the pump, these wells forming pressure chambers in relation to the adjacent flat bearing or reaction surface.

It is a further feature of this invention to provide pressure wells which are so located with respect to their respective piston rods such that the resultant center of pressure is coincident with or radially spaced outwardly with respect to the location of the major force resisting this pressure load.

It is a further feature of this invention to provide a plain axial bearing and a plain radial bearing with the radial bearing being particularly aligned to avoid eccentric loading.

These and other features of this invention will become readily apparent from the following detailed description of the drawings in which:

FIG. 1 is a detailed cross-sectional illustration of the pump assembly embodying the features of this invention;

FIG. 2 is an enlarged detailed cross-sectional showing of the piston rod assembly including its connection to the thrust plate;

FIG. 2*a* is an enlarged detail of a portion of FIG. 2 showing the parts in a slightly exaggerated position;

FIG. 3 is an enlarged detailed showing of the bottom of the unitary thrust plate; and

FIG. 4 is a partial showing of a side view of FIG. 3 schematically illustrating the tilt-balancing forces.

Referring to FIG. 1, the pump is shown as having an



upper casing 10 and a lower base casing 12 from which protrudes at the left side a toothed member 14 for driving the pump. The member 14 includes a shaft 16 having splines 18 for driving the unitary thrust or drive plate 20. The unitary thrust plate 20 includes a number of peripherally spaced ball-receiving sockets 22 which receive the balls 24 at the antipumping end of each piston rod 26. The unitary drive plate adjacent its left end includes an elongated spline 30 which also through the spline 32 drives a coupling 34 carrying a beveled gear 36. The beveled gear 36 engages a cooperating gear 38 for driving the shaft extension 40 which via the splines 42 drives barrel-like rotor 44. The axis of the rotor 44 is at an angle with respect to the drive plate 20 and its adjacent bearing 120 such that as the barrel rotates the piston rods 26 are reciprocated thereby inducing a pumping motion to the pistons 50. A valve plate 60 is suitably held into engagement with the top pumping end of the rotor 44 and is held against continued rotation by means of a spline connection 62 leading to a control shaft 64. The control shaft 64 is mounted on suitable bearings 66 and 68 and carries a pinion 70 which is engaged by one or more racks 72 which can in turn be actuated by a suitable servo unit. The servo unit is intended to rotate the valve plate 60 so as to interconnect one or more pistons either to the inlet 80 or the outlet 82 or to connect certain pistons to each other. A control mechanism of this type is more clearly illustrated and described in copending patent application Serial No. 821,180, filed June 18, 1959, by Richard N. Sullivan.

With high pumping loads, normally, the antipumping end of the piston, piston-rod assembly would have to be engaged and supported by thrust bearings of substantial weight and size. However, provision is made for pressure balancing the assembly so that a lightweight plain bearing can be utilized in place of the much heavier antifriction-type ball bearings or the like. To this end, each piston 50 (see also FIG. 2) has a land portion 90 which is meticulously machined and lapped with respect to its adjacent cylinder wall so as to form a first fixed orifice 92, better seen in FIG. 2. The fixed orifice 92 thus formed is continuously wiped during the piston reciprocation which tends to make it a self-cleaning capillary restriction. The restriction 90 is of a predetermined size and length so that under maximum pressure operation of the pump a predetermined pressure drop will occur thereacross. During operation, high-pressure fluid from the pumping chamber 94 flows through the fixed restriction 90 and thence to an annulus 96 leading to a transverse passage 98 in the piston. This passage in turn connects with a passage 100 drilled centrally of the piston rod 26. The ball joint 24 at the antipumping end of the piston rod 26 includes a flat portion 104 so that continuous communication is provided with the drilled passage 106 in the socket 22, regardless of the rotary or tilted position of the piston rod in the pumping cycle.

The bottom or reaction face (FIG. 2) 110 of the unitary drive plate 20 contains a plurality of circular wells or recesses 112 which, as seen also in FIG. 3, may have their centers eccentrically located or spaced in an outward direction with respect to the axis of the bleed passage 106. The wells or recesses 112 form pressure-balancing areas which form a liquid-type bearing surface to reduce the friction loading on the adjacent cooperating plain bearing 120.

The construction of the pump is such that there is permitted a limited relative axial movement of the thrust drive 20 with respect to the plain bearing 120 so that there is provided therebetween a variable orifice depending upon the pressure and the loading of the pump parts.

The pressure wells 112 and the remaining radial areas

of the juxtaposed surface 110 and the bearing 120 form the total balancing force depending upon the pressure across this region. Thus, the maximum balancing pressure will occur in the wells 112 but there is a drop across the cooperating surfaces as the leakage flow passes outwardly to the substantially low casing pressure. The total force is an integration of the particular pressures over their respective areas which, at the balanced position of the parts, are equal to the total force acting on the piston heads. Thus, the variable orifice formed by the space in between the reaction surface 110 of the drive plate 20 and the bearings 120 determines the outflow of the bleed fluid and hence the pressure level necessary to achieve equilibrium and balancing.

For example, if the combined fluid pressure in the well and between the cooperating surfaces is at a level whereby the force acting on the thrust bearing is less than the force generated by the piston, the unitary drive plate will be urged toward the thrust bearing and effectively reduce the area of the variable orifice which effects a reduction in fluid flow and hence bulids up the pressure in the well and the cooperating surfaces until the forces created by this newly increased pressure will substantially equal and cancel out the forces generated by the piston. And conversely should the combined fluid pressure in the well and between the cooperating surfaces generate a force which is greater than the force generated by the pistons, the unitary drive plate will be urged away from the thrust bearing, effecting an increase in effective area of the variable orifice so as to increase the flow therethrough and hence reduce the pressure in the well and cooperating surfaces. In this manner the movement of the unitary drive plate with respect to the thrust bearing will automatically adjust itself to maintain a fluid pressure level in the well and between the cooperating surfaces so that the forces generated thereby will substantially match or equal the forces generated by the piston.

It is thus apparent that during starting, the unitary drive plate will be moved away from bearing 120 so as to form a lubricant film immediately to avoid starting friction.

The bleed fluid from the radial variable orifice can flow through a chamber 130 leading to the casing. Another passage 132 is provided for receiving leakage fluid from the annular chamber 134 leading from the inside or axial end of the variable orifice.

Returning to FIG. 1, the radial pump forces are absorbed through an annular plain bearing 150 which engages an outer peripheral surface 152 on the drive plate 20. The size of the bearing 150 is materially reduced by locating its supporting pedestal 154 in a plane containing the centers of pivotation of each of the balls 24 at the antipumping end of the piston rods 26. In this way, the radial loads are symmetrical, thus avoiding any cocking tendency which would require larger and higher strength bearings and supports. With this particular bearing arrangement and the pressure balancing, the size and weight of the pump of this invention are materially reduced.

In order to further reduce the weight requirements of the pump assembly, reference is made to FIGS. 3 and 4. As previously stated, the pressure wells 112 which are disposed about the center of the unitary drive plate 20 have their centers radially spaced from the centers of the bleed passages 106 leading thereto. This moves the total center of pressure radially outwardly so that on the pumping side of the thrust plate, the resisting force will be located inwardly thereof as seen in FIG. 4. This resisting force and the center of pressure produce a force couple which can be balanced readily by a relatively small balancing force, as indicated in FIG. 4, because of its relatively large moment arm. It should be noted, however, that as viewed in FIG. 4 the center of pressure will travel as the valve plate 60 is adjusted since only certain pistons will be contributing to the final pumping output. In any case, the center of pressure should be coincident with or outboard of the major resistance vector.



As a result of this invention, it will be apparent that an extremely lightweight pump has been provided which can be made relatively small in size. This compactness makes it readily adaptable for aircraft and missiles where weight saving is of the essence. Thus, the bearings are reduced materially in size and weight permitting the entire structure to be manufactured at low cost.

Although one embodiment of this invention has been illustrated and described herein, it will be apparent that various changes may be made in the construction and arrangement of the various parts without departing from the scope of the novel concept.

We claim:

1. In an axial piston pump having a casing and a rotor carried in said casing, a plurality of cylinders carried by said rotor, an axis of rotation for said rotor, means for rotating said rotor, a plurality of cylinders in said rotor and forming pumping chambers, pistons in said cylinders, at least a portion of each piston being slightly smaller in diameter than the bore of said cylinder providing a fixed orifice connected to said pumping chamber, a first passage means leading from the side wall of said piston and communicating with said fixed orifice, a piston rod having one end connected to each of said pistons, a swivel joint at the other end of said piston rod, a unitary plate receiving said swivel joint, a unitary plain bearing in juxtaposed relation with said plate, a rounded balancing chamber between said plate and bearing, second passage means in said piston rod leading from said first passage means and through said swivel joint and terminating in an opening at said balancing chamber, said balancing chamber being eccentrically disposed and having its center radially spaced outwardly from said axis with respect to the axis of said second passage opening, plain bearing means surrounding said plate and having an inner circumferential surface engaging the outer periphery of said plate, a support for said plain bearing means, the center of said support and the center of rotation of said swivel joint lying in a plane extending transversely of said axis, wall means forming inner and outer chambers surrounding the juxtaposed surfaces of said plate and bearing for receiving fluid flowing from said rounded chamber and from between said juxtaposed surfaces, said plate and bearing being relatively movable toward and from each other when pressure is supplied from said pumping chamber via said passage means to said balancing chamber whereby said juxtaposed surfaces form a variable area orifice, means for conducting fluid from said inner chamber to said outer chamber, and means for conducting fluid from said outer chamber to the area between said bearing means and the outer periphery of said plate.

2. In a multiple piston pump having an axis, a rotatable rotor mounted about said axis, a plurality of cylinders formed in said rotor, a corresponding number of pumping pistons mounted in said cylinders, plate means connected to the anti-pumping end of said piston, a driving member, a swivel joint connecting said driving member to said rotor, a bearing member having a first bearing surface and a second bearing surface surrounding the first bearing surface, chamber means located between said bearing member and said plate means for receiving fluid from said cylinders so that said fluid acts on said first bearing surface, passage means extending through said piston including a fixed restriction connected to said cylinders for communicating with said chamber means, said plate member being movable rectilinearly with respect to said bearing member for defining a variable orifice, said variable orifice serving to control the leakage of fluid from said chamber means so that leakage fluid acts on said second bearing surface, the combined first bearing surface and second bearing surface serving to counterbalance the thrust force created by said pumping piston, in combination with an annular bearing means surrounding said plate member and having an inner circumferential surface engaging the outer periphery of said plate member, a support for said

annular bearing means, the center of said support and the center of rotation of said swivel joint lying in a plane extending transversely of said axis.

**References Cited** in the file of this patent

UNITED STATES PATENTS

| | | |
|---|---|---|
| 1,817,063 | Carrie et al. | Aug. 4, 1931 |
| 2,241,701 | Doe | May 13, 1941 |
| 2,608,159 | Born | Aug. 26, 1952 |
| 2,875,701 | Ebert | Mar. 3, 1959 |
| 2,918,012 | Lucien | Dec. 22, 1959 |
| 2,967,491 | Wiggermann | Jan. 10, 1961 |

FOREIGN PATENTS

| | | |
|---|---|---|
| 604,168 | Great Britain | June 29, 1948 |
| 1,114,641 | France | Dec. 19, 1958 |