Sept. 11, 1962 R. P. LAMBECK 3,053,197

AXIAL PISTON PUMP WITH BALANCED DRIVE PLATE

Filed Dec. 24, 1959 2 Sheets-Sheet 1

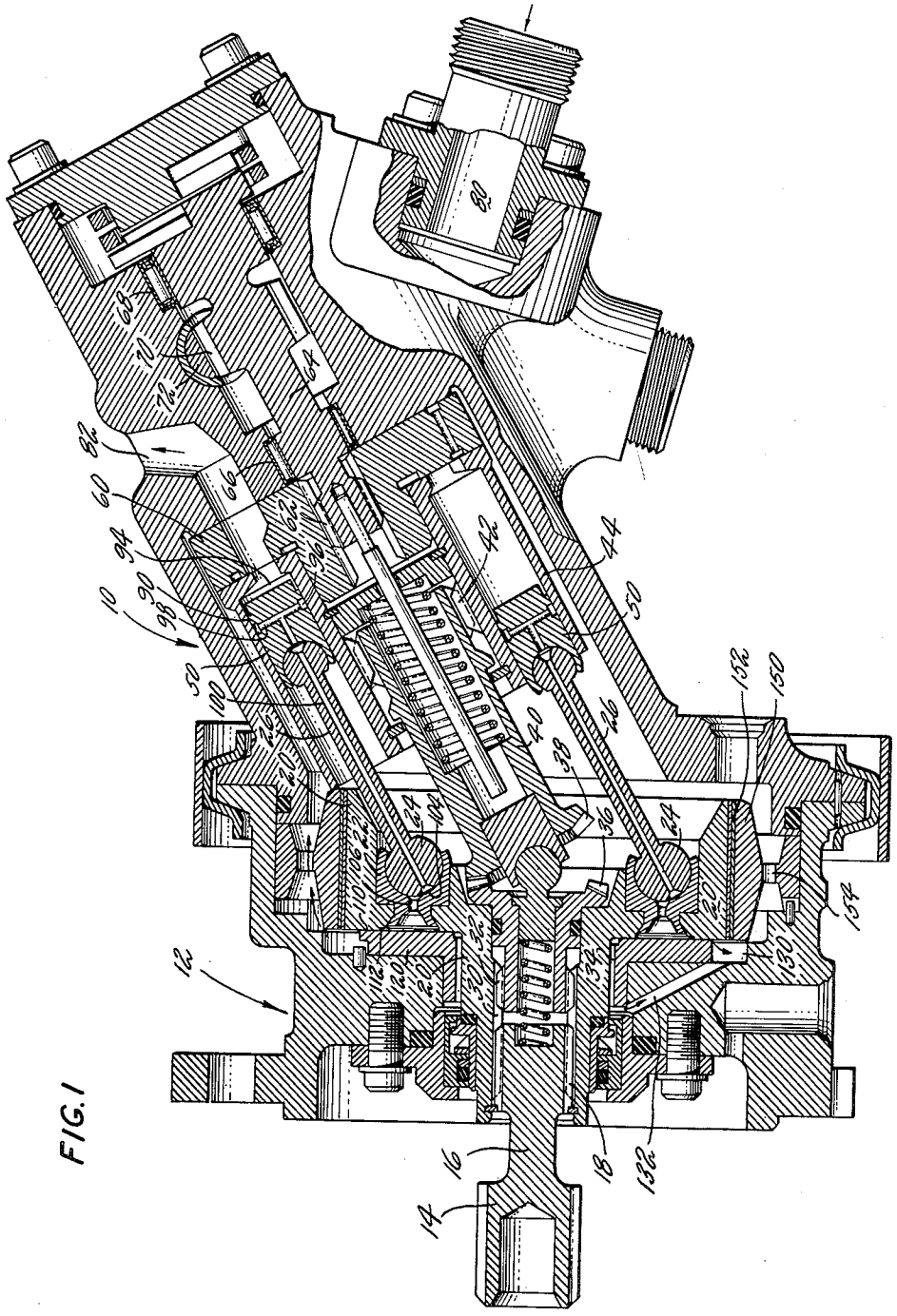


FIG. 1

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FIG. 3

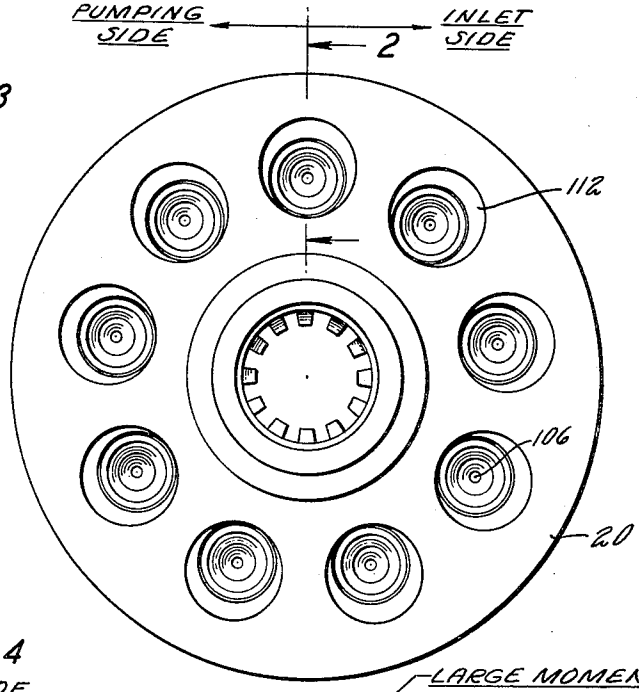


FIG. 4

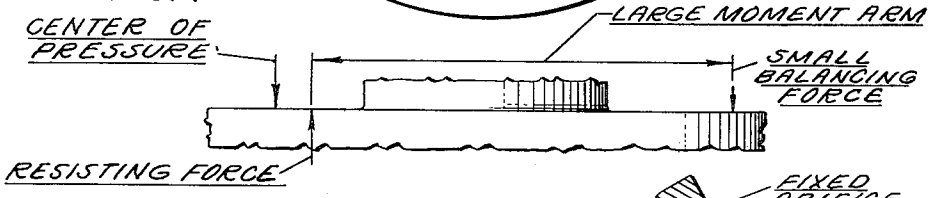


FIG. 2

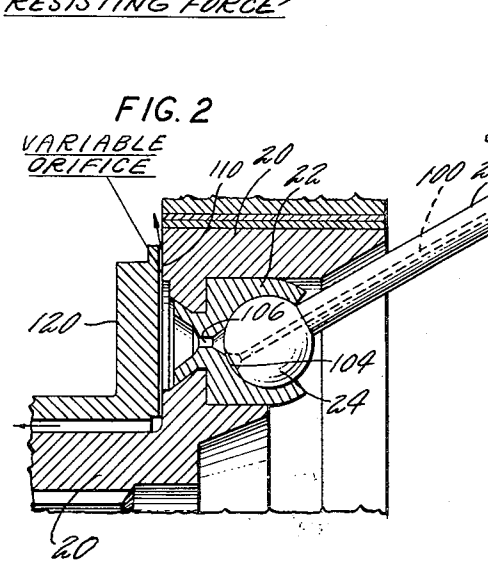
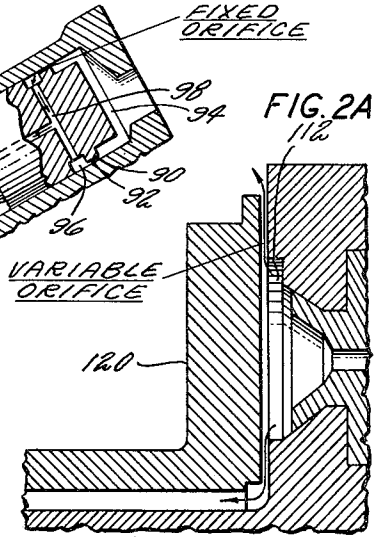


FIG. 2A



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3,053,197

AXIAL PISTON PUMP WITH BALANCED DRIVE PLATE

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6 Claims. (Cl. 103—162)

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 builds 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.

Another advantage of the radial bearing described is that the rotor of the pump does not have to absorb these radial loads. Instead the side loads are taken at the antipumping end of the connecting rods so that piston wear is avoided since they are not subject to eccentric loading.

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 the center of pressure will travel radially 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.

I claim:

1. In a multiple piston pump having a rotatably mounted rotor, a plurality of cylinders formed in said rotor, a corresponding number of pumping pistons mounted in said cylinders, a plate member connected to the anti-pumping end of said piston, 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 the 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, and means providing recesses in said plate member corresponding to said pistons respectively, for maintaining alignment of said plate member relative to said bearing member, said recesses being disposed eccentrically outwardly relative to the resultant axial forces of their respective pistons.

2. In a multiple piston pump according to claim 1, wherein said fixed restriction is defined by each piston having a portion of its length of reduced diameter smaller than that of its cylinder communicating with said passage means.

3. In a multiple piston pump according to claim 1, wherein each piston has a rod connected thereto, each rod having an end remote from its piston connected to said plate member by a swivel joint.

4. In a multiple piston pump according to claim 1, wherein each passage means includes a passage disposed transversely of said piston and a passage disposed axially of said piston.

5. In a multiple piston pump according to claim 1, wherein each piston has a rod connected thereto, each rod having an end remote from its piston connected to said plate member by a swivel joint, and said passage means

includes passages extending transversely of said piston, axially of said rod and through said swivel joint.

6. In a multiple piston pump according to claim 1 wherein each piston has a rod connected thereto, each rod having an end remote from its piston connected to said plate member by a movable joint, said passage means extending through said piston, said rod and said joint.

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