July 27, 1965 J. F. HOFFER 3,196,800

GEAR PUMP

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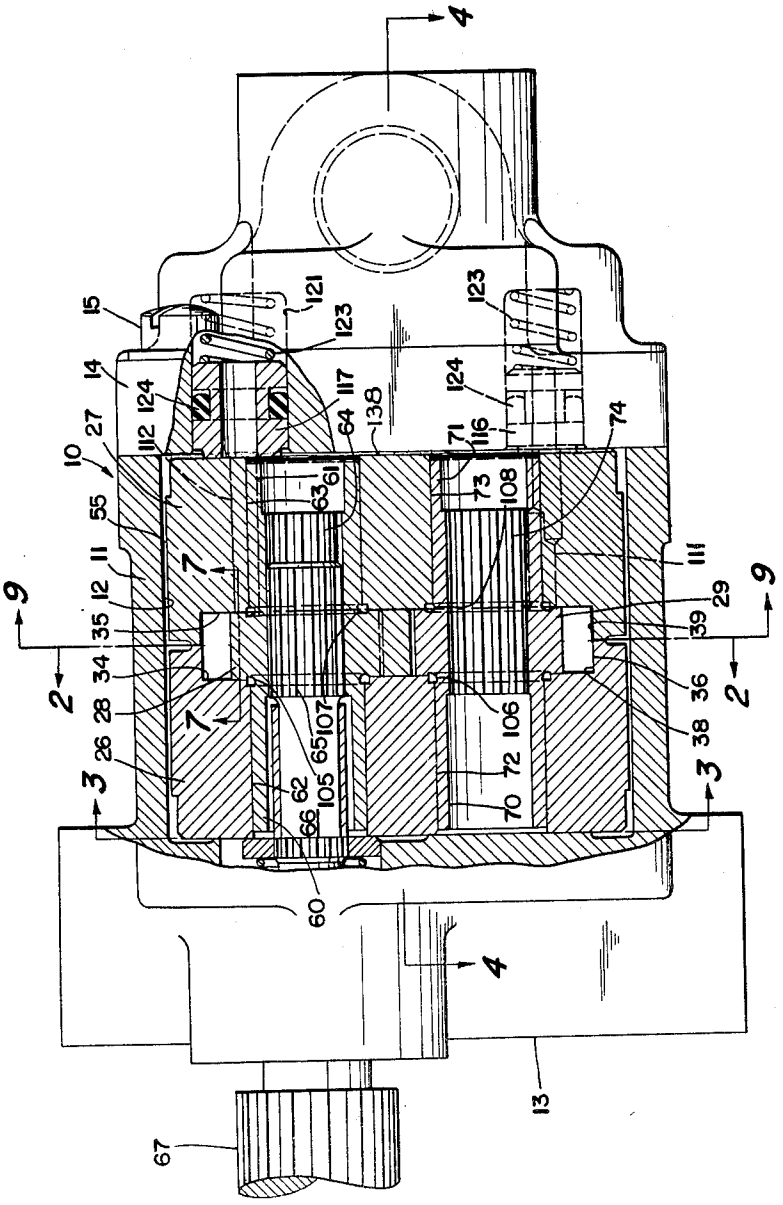


Fig. 1

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GEAR PUMP

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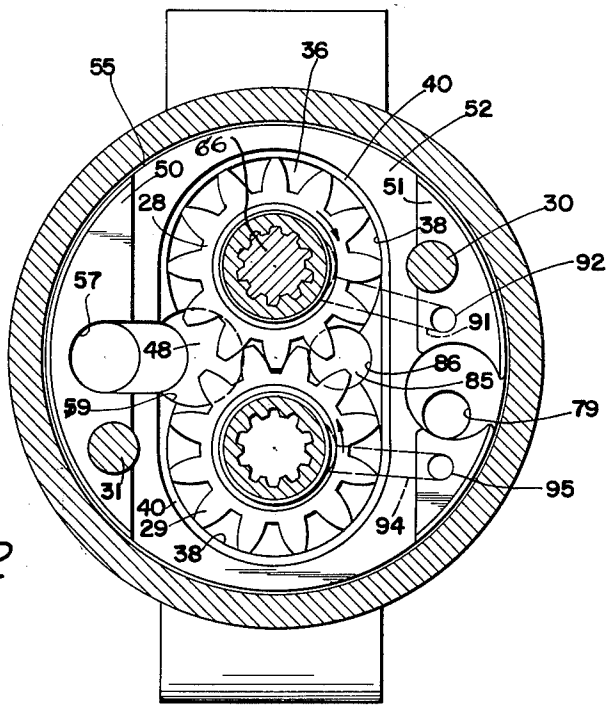


Fig. 2

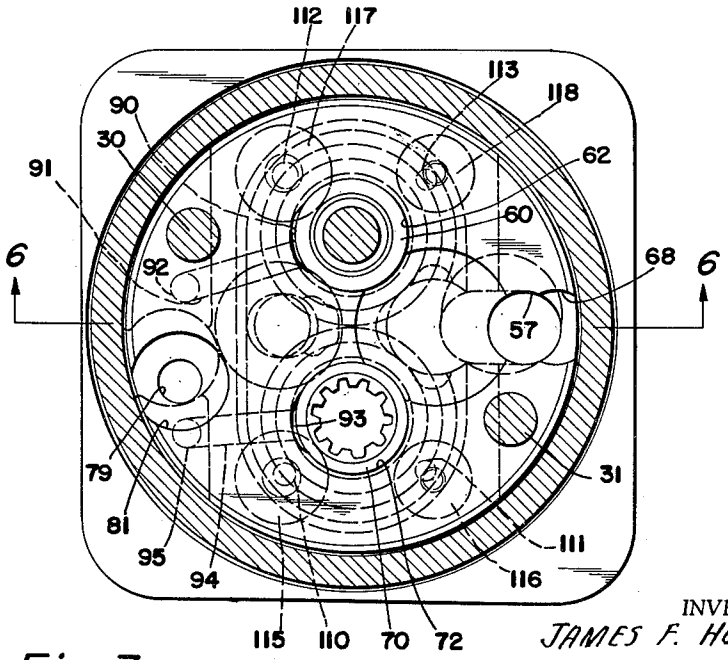


Fig. 3

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GEAR PUMP

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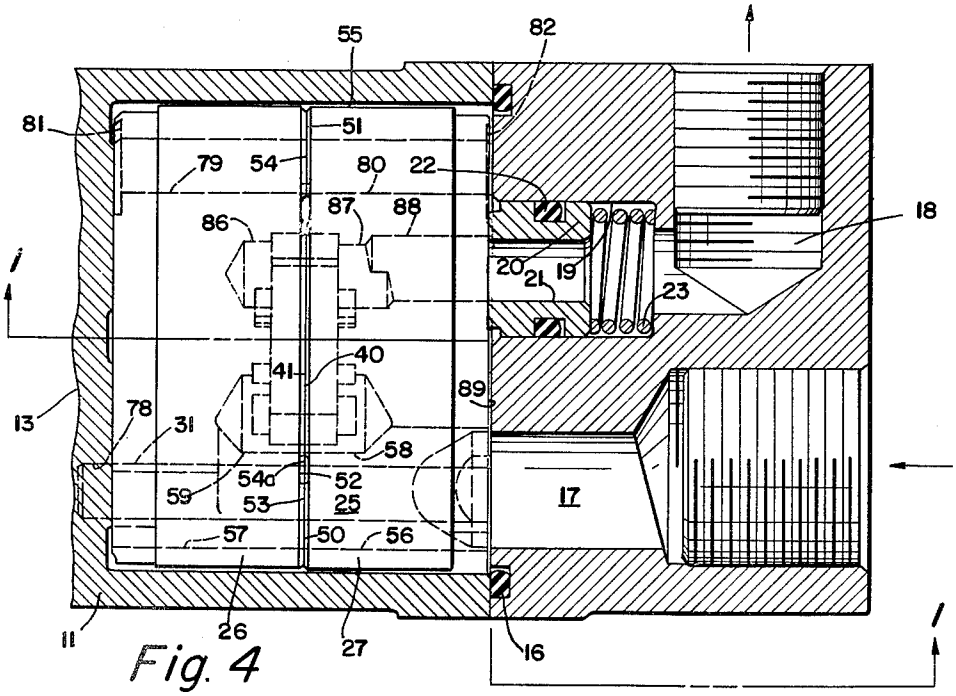


Fig. 4

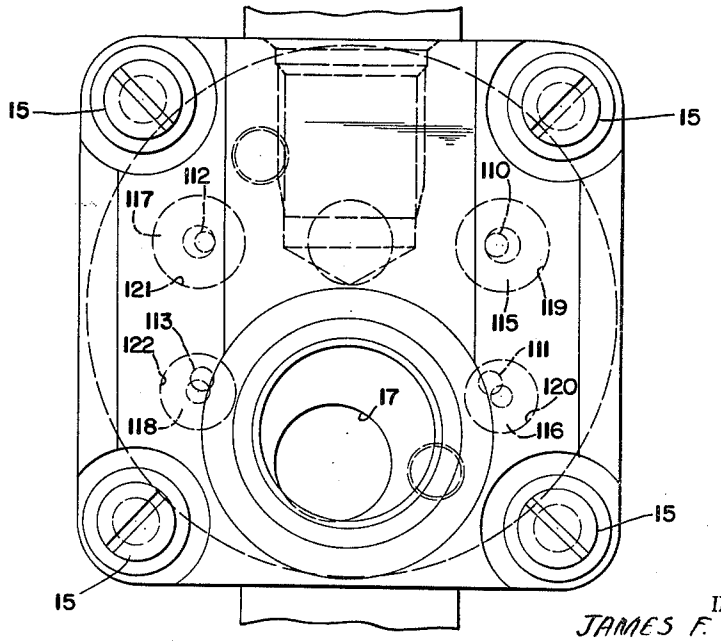


Fig. 5

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GEAR PUMP

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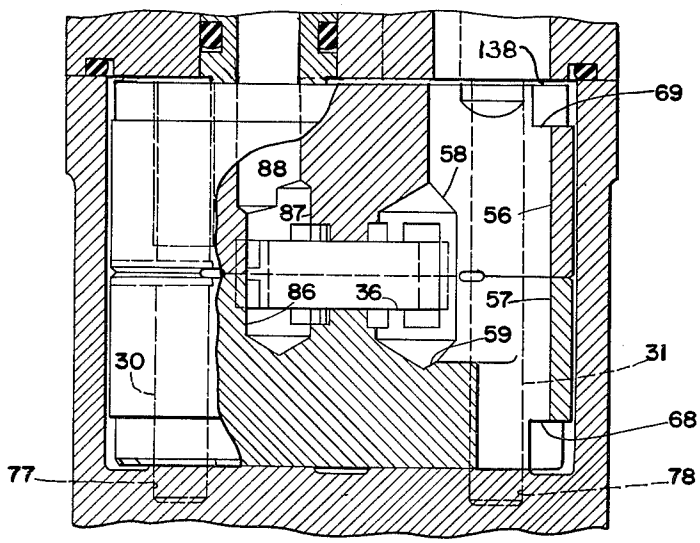


Fig. 6

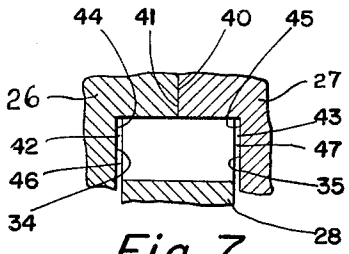


Fig. 7

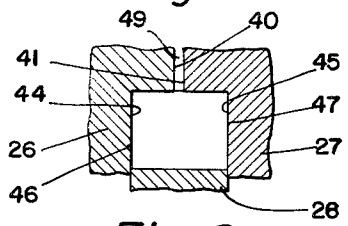


Fig. 8

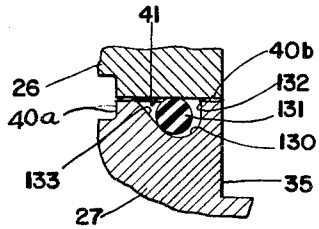


Fig. 10

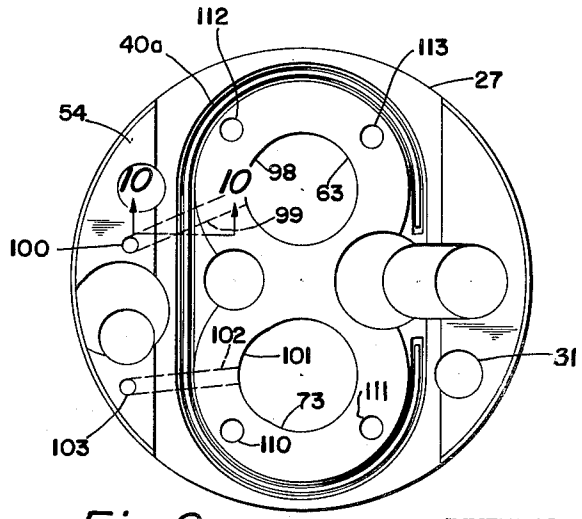


Fig. 9

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3,196,800

GEAR PUMP

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Filed Dec. 13, 1962, Ser. No. 244,317
25 Claims. (Cl. 103—126)

This invention relates to gear pumps and more particularly to gear pumps of the cartridge type in which the cartridge comprises a set of plates forming a pumping chamber which contains the gears, the cartridge being mountable in a main housing which provides a driving connection and fluid inlet and outlet ports.

Cartridge type gear pumps may be of either fixed clearance or pressure loaded type. In the fixed clearance type end plates are fitted against the sides of the gears with a small predetermined or fixed clearance therebetween for sealing the side faces of the gears without having the plates in actual contact therewith. Thus there is only fluid friction on the gear faces as the gears rotate and there is relatively small loss of input power due to friction losses. However, the clearance is a fluid leakage path which affects the volumetric efficiency of the pump, and obviously, the greater the clearance the greater the leakage and the lower the volumetric efficiency. Thus it is desirable in high performance pumps to keep the clearance very small.

When the clearance becomes very small other problems are encountered. Thus localized or unequal heating or cooling of the plates and gears may cause expansion or contraction of the parts so as to take up the clearance and bring the plates and gear side faces into direct contact. This will not only reduce mechanical efficiency but is very apt to score or bind the parts so as to seriously damage or destroy the pump.

In high performance pumps, such as used for missiles and space craft where pressures may be as high as 3000 p.s.i. and rotative speeds upwards of 60,000 r.p.m., the problem becomes acute. The initial clearances must be very small to provide the volumetric efficiency required and the high pressures and speeds quickly cause localized heating and differential expansion of parts tending to take up such clearances.

In pressure loaded pumps the end plates are yieldably pressed against the gear side faces so that there is no predetermined or fixed clearance. This results in less leakage and high volumetric efficiency but has the disadvantage of lower mechanical efficiency because of the greater friction loss due to the gear side faces rubbing against the end plates. Friction losses are minimized as far as possible by applying just enough pressure on the end plates to press them against the gears with enough force to keep the parts from separating and not enough force to cause excessive friction and wear between the parts. However, when differential expansion or contraction of the gears or plates occurs the plates must be able to quickly move in a direction toward or away from the gears, as the case may be, in order to retain contact therewith and to maintain such contact without excessive pressure or friction between the parts.

Fixed clearance pumps are generally more suitable for low pressure operation because leakage through a given clearance is less than at high pressures and becomes a

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minor factor, and also because friction losses between the gear side faces and the end plates must be minimized since they are a relatively large percentage of the input power for driving the pump. Conversely, pressure loaded pumps are generally more suitable for high pressure operation because leakage past the gear side faces becomes a more important factor and is minimized by eliminating the clearance. Friction losses, on the other hand, become less important since they are a smaller percentage of the input power.

It is an object of the present invention to provide cartridge type gear pumps of either the fixed clearance or pressure loaded form in which the end plates can quickly move in axial directions without appreciable mechanical or frictional resistance to compensate for thermal expansion or contraction of the parts in such a manner that in fixed clearance pumps the clearance remains substantially constant and in pressure loaded forms the contact pressure between the plates and gear side faces remains substantially constant.

Another object is to avoid bolts or other mechanical means for clamping the end plates in position next to the gear side faces in the fixed clearance pumps, as has heretofore been the practice, but to utilize hydraulic forces only for this purpose, such hydraulic forces being carefully controlled and distributed so that the plates may move in either axial direction without radial displacement and without tilting.

Another object is to provide a cartridge type gear pump of either the fixed clearance or pressure loaded type in which the end plates are axially aligned with each other by dowels extending axially therethrough, the plates being symmetrically formed and hydraulically balanced in such a manner that the plates cannot move sideways or tilt so as to bind upon the dowels as this would interfere with the free axial movement of the plates axially along the dowels when compensating for differential expansion or contraction between the gears and plates.

It is another object to provide a cartridge type gear pump of either the fixed clearance or pressure loaded form in which the cartridge end plates are axially aligned by dowels passing therethrough and in which the plates are so balanced against forces tending to radially displace or to tilt the same that side loading of the plates upon the dowels is avoided whereby the dowels may be anchored in the pump housing at only one end thereof, thus greatly facilitating manufacture and assembly of the pump by making it unnecessary to provide opposed recesses aligned to a high degree of accuracy for receiving both ends of the dowels.

Another object is to provide a fixed clearance type gear pump in which an end plate is urged toward the gear side faces by controlled fluid pressure means, the plate being engaged by a stop to provide a predetermined clearance with the gear side faces, and with the plate being movable against the action of the fluid pressure means upon momentary differential expansion or contraction of the parts which takes up the clearance whereby the pump at that time will operate as a pressure loaded type without excessive frictional contact between the gear side faces and plates such as would bind or damage the parts.

Another object is to provide pumps of the types described in which a pair of end plates have opposed recesses to form a pumping chamber, the plates having

opposed faces with a sealing means therebetween for sealing the pumping chamber.

Another object is to provide cartridge type gear pumps of either the fixed clearance or pressure loaded form in which a pair of end plates have opposed recesses for receiving the gears and the plates are maintained in proper axial position with respect to the gears by fluid pressure.

Other objects of the invention will be apparent from the drawings and from further description.

Generally described, the invention provides a cartridge insertable into a main housing. The cartridge includes two end plates, each of which is recessed for receiving one-half the axial length of a pair of gears. In the fixed clearance type, the end plates butt against each other to provide initial fixed clearances of predetermined dimension between the gear side faces and the bottom walls of the recesses. In the pressure loaded form, the bottom walls of the recesses contact the gear side faces and there is a slight clearance between the plates where they surround the gears.

The plates are aligned axially with respect to each other by dowels extending through the plates and having one end anchored in recesses in the housing for locating the angular position of the plates within the housing. The gears have shafts extending from either side thereof, the shafts having bearing support in openings in the plates. One plate is supported endwise by the body and the other plate is pressed toward the first mentioned plate by hydraulically actuated pistons or buttons which are so located and dimensioned that they not only counterbalance axial forces tending to separate the plates from the gears but also compensate for tilting forces applied to the plates. This, plus the face that the plates are symmetrically formed and so balanced that there is no tendency for the same to move sidewise with respect to the dowels, prevents side loading between the plates and dowels. As a result, the plates may move axially with respect to the dowels without appreciable interference due to friction or binding thereon so as to freely compensate for expansion or contraction of the gears or plates to maintain substantially the same clearance between the gears and plates in the case of fixed clearance pumps and to maintain substantially the same contact pressure between the gears and plates in the case of pressure loaded pumps.

In the drawings:

FIG. 1 is a side view, partly in vertical axial cross section,

FIG. 2 is a section view along the lines 2—2 of FIG. 1,

FIG. 3 is a section view along the lines 3—3 of FIG. 1,

FIG. 4 is a section view along the lines 4—4 of FIG. 1 but with the cartridge unsectioned,

FIG. 5 is an end view,

FIG. 6 is a fragmentary section view along the lines 6—6 of FIG. 3 with the cartridge shown partly sectioned and partly unsectioned,

FIG. 7 is a fragmentary section view along the lines 7—7 of FIG. 1 showing in exaggerated manner the clearances between the cartridge end plates and the side faces of the gear in the fixed clearance form of the pump,

FIG. 8 is a view similar to FIG. 7 but showing in exaggerated manner the clearance between opposed faces of the end plates in the pressure loaded form of the pump,

FIG. 9 is a view along the lines 9—9 of FIG. 1 but with the inner end of the end plate formed with a groove and a sealing element fitted within the groove, and

FIG. 10 is a fragmentary cross section along the lines 10—10 of FIG. 9 showing the groove and sealing element.

The pump has a main housing **10** comprising a body **11** having a cylindrical bore **12** therein which is closed at one end by a head structure **13** and which is closed at the other end by a cap **14** removably attached to body **11** by bolts **15** and sealed with respect to the body by a resilient sealing ring **16**.

Cap **14** has a fluid inlet port **17** and a fluid discharge port **18**. A portion of the latter is enlarged as at **19** to receive an annular port sealing piston or button **20** which has a discharge opening **21** therethrough in communication with enlarged bore portion **19**. Button **20** is sealed with respect to bore portion **19** by a resilient sealing ring **22** and is pressed toward body bore **12** by spring **23**.

Mounted within bore **12** is a cartridge assembly, generally designated **25**, which includes a pair of plates **26**, **27**, a pair of gears **28**, **29** and a pair of dowel pins **30**, **31**.

Plates **26**, **27**, have opposed recesses **34**, **35**, therein of equal depth which from a pumping chamber, generally designated **36** for receiving gears **28** and **29**. The gears are in mesh with each other and have a running fit with portions of radially outer walls **38**, **39** of recesses **34**, **35**, respectively.

Adjacent to radial outer walls **38**, **39** the plates have opposed flat raised surfaces or ribs **40**, **41**. These ribs normally contact each other in the fixed clearance form of pump, as shown in FIG. 7, to close pumping chamber **36**, there being very small clearances **42**, **43** between bottom walls **44**, **45** of the plates and the adjacent side faces **46**, **47** of the gears. In the pressure loaded form of pump, as shown in FIG. 8, there is a slight clearance **49** between ribs **40**, **41** when the gear side faces **46**, **47** are in contact with bottom walls **44**, **45** of the recesses in the plates. Ribs **40**, **41** surround the entire pumping chamber **36** except where the latter is intersected by an inlet opening.

Rib **41** is identical to rib **40**. Both are narrow for good sealing contact with each other and for keeping to a minimum the area therebetween which is subject to fluid pressure gradient.

Spaced from and co-planar with rib **40** are additional flat raised surfaces **50**, **51** on plate **26** with a channel **52** between rib **40** and raised surfaces **50**, **51**. Plate **27** likewise has raised surfaces **53**, **54** spaced from and co-planar with raised rib **41** with a channel **54***a* between such rib and surfaces.

Channels **52**, **54***a* communicate freely with an annular clearance **55** between the outer periphery of plates **26**, **27** and the wall of bore **12**, and also communicate with aligned low pressure inlet passages **56**, **57** in plates **27** and **26** respectively, passage **57** being in register with inlet port **17**.

Gear **28** has shafts **60**, **61** integral therewith, such shafts having bearing support or contact with openings **62**, **63** in plates **26**, **27**. The shafts are hollow and have a spline section **64** therein for receiving the splined end **65** of a drive shaft **66**. Shaft **66** has an external portion **67** for connection to a motor for driving the pump.

Gear **29** is identical to gear **28** and likewise has integral shafts **70**, **71** in bearing contact with openings **72**, **73** in plates **26**, **27** and there is a spline portion **74** which is utilized in the event cartridge **25** and cap **14** are assembled in a 180° rotative position with respect to body **11** for the purpose of using reverse direction of rotation of shaft **66**.

Dowels **30**, **31** extend through both plates **26**, **27** with a snug but sliding fit to axially and angularly align the same with respect to each other. One end of each dowel extends beyond the other face of plate **26** and into openings **77**, **78** in head portion **13** of body **11** for locating the plates **26**, **27** angularly within body **11**. Plates **26**, **27** also have openings **79**, **80** and counterbores **81**, **82** therein which serve to provide open communication between low pressure areas **12**, **52**, **54***a* and opposite sides of housing bore **12** for equalizing the pressure therein.

Passages **56**, **57** communicate with clearance **55** and with the ends of bore **12** by means of cutouts **68**, **69** in plates **26**, **27**. Inlet passages **56**, **57** intersect bores **58**, **59** which in turn intersect recesses **34**, **35** in the plates to form an inlet or low pressure portion, generally desig-

nated 48, of pumping chamber 36 which is exposed to gears 28, 29 on one side of a plane passing through the axis of the two gears. On the other side of such plane plate 26 has a bore 86 extending part way thereinto from recess 34. Opposite bore 86 is a bore 87 in plate 27 which intersects recess 35 also intersects a bore 88 which is slightly offset therefrom.

Bore 88 extends through the plate to the outer surface 89 thereof and in register with opening 21 through button 20. The latter bears on surface 89 to seal passages 88 and 21 from the interior of body bore 12 and also to help provide force for holding plates 26, 27 in position with respect to the gears, as is hereinafter further described. Button 20 is held against surface 89 by spring 23 plus the effect of fluid pressure in counterbare 19 acting on the outer end of the button.

The outer end area of button 20 exposed to high pressure fluid in counterbore 19 is larger than the area of the other end of the button which is in contact with plate surface 89 so that the fluid pressure gradient across the inner end of the button can never unseat the button from plate surface 89.

Bores 86, 87, together with the portions of recesses 34, 35 therebetween, provide a high pressure zone, generally designated 85, of the pumping chamber and fluid therein at full discharge pressure is exposed to adjacent portions of the two gears and to the adjacent portions of outer radial walls 38, 39 of the pumping chamber.

As shown in FIG. 3, openings 62 in plate 26 has a lubrication groove 90 at one side thereof running throughout its length and there are drilled passages 91, 92 connecting groove 90 with inner raised face 51 of the plate. Likewise, opening 72 has a lubrication groove 93 which is connected by drilled passages 94, 95 with the inner face of plate 26. In like manner, and as shown in FIG. 9, opening 63 in plate 27 has lubrication groove 98 connected by drill passages 99, 100 with inner face portion 54 and opening 73 has a lubrication groove 101 connected by drill passages 102 and 103 with inner face portion 54. Passages 92 and 100 register with each other and passages 95, 103 register with each other whereby groove 90 is connected with groove 98 and groove 93 is connected with groove 101 to provide approximately equal flow of fluid through such connected grooves for substantially equally lubricating and cooling the gear shafts in their bearings.

Grooves 90, 93, 98 and 101 are also open to the low pressure clearance spaces at the outer ends of the cartridge, these clearance spaces being in turn open to clearance 55. In addition to supplying lubricant to the gear shaft bearings, grooves 90, 93, 98 and 101 also serve to prevent build-up of pressure in undercuts 105, 106, 107 and 108 by fluid leaking past the gear side faces and bottom walls of recesses 34, 35 by venting such undercuts to low pressure areas of the pump.

Plate 27 has passages 110, 111, 112 and 113 therethrough connecting recess 35 at points traversed by the gear teeth and the gears rotate with the outer end face 89 of the plate where they register with openings through respective annular pistons or buttons 115, 116, 117 and 118 located in recesses 119, 120, 121 and 122 in cap 14. Each of the buttons is constantly urged toward plate 27 by a spring such as 123 and is sealed with respect to its respective recess by a packing such as 124. Fluid under pressure from recess 35 passes through the buttons and acts upon the rear face thereof to press the buttons into sealing engagement with end face 89 of plate 27. The inner end of each button has a reduced diameter face, similar to that of button 20, to present a smaller area of contact against plate 27 than the area of the rear face so that there is always an overbalance area acted upon by fluid pressure for mintaining leak-proof abutment of the buttons against plate 27.

In operation, shaft 66 is rotated in a clokwise direction as viewed in FIG. 2. This causes gears 28 and 29 to rotate as indicted by the arrows. Fluid under low pressure entering port 17 passes through passage 56 into low pressure chamber 48 where it is picked up by the rotating gears and carried in the gear pockets to be discharged under high pressure into high pressure chamber 85 and through bore 87, passage 88 and bore 21 of button 20 to be discharged through outlet port 18. Spring 23 and the pressure of the fluid in outlet bore portion 19 acting on the rear face of button 20 presses the latter into sealing engagement with end face 89 of plate 27 to seal the high pressure fluid from bore 12 exteriorly of plates 26, 27.

As the gears rotate and high pressure is developed in chamber 85 there will be some leakage past the radially outer edges and side faces of the gear teeth as the teeth approach chamber 85 so that fluid within the gear pockets that are closed by radially outer walls 38, 39 of the pumping chamber gradually takes on a higher and higher pressure as the tooth pockets approach high pressure chamber 85. Thus, as a given tooth pocket sweeps across low pressure chamber 48 it becomes filled with fluid at low pressure. When the pocket is closed off by walls 38, 39 the pressure of the fluid therein gradually increases as the pocket approaches high pressure chamber 85 due to leakage past the gear teeth from the direction of high pressure chamber 85. The fluid pressure gradient between walls 38, 39 and the outer ends of the gear teeth, plus the fluid pressures within the closed gear teeth pockets and within chamber 85 impose radial loads upon the gears tending to force the same generally toward the low pressure side of the pump. There are, of course, lesser and oppositely acting side loads on the gears from the low pressure side of the pump but the net effect is to impose side loads on the gears tending to move them away from the high pressure side of the pump. These net side loads are transmitted by the gear shafts to plates 26, 27 at bearing contact points in openings 62, 63, 72 and 73. On plates 26, 27 the side loads are counterbalanced in radial directions by the pressure of fluid acting on outer walls 38, 39 of recesses 34, 35 in plates 26, 27. Thus the side or radial loads imposed upon plates 26, 27 are balanced so that there are no forces acting internally of the plates tending to move the plates transversely of the housing. Likewise, low pressure fluid has access to the complete outer periphery of plates 26, 27 so that there are no unbalanced forces acting externally on the plates tending to move the same transversely of the housing. As a result of such internal and external balancing of radial forces acting on the plates, there are no side loads imposed by the plates upon dowels 30, 31.

The fluid pressures within recesses 34, 35 act in directions upon plates 26, 27 tending to axially separate them from each other. This separating pressure is greater on the high pressure side of the pump than on the low pressure side so that these pressures also tend to tilt the plates with respect to each other, that is, to separate them more on the high pressure side than on the low pressure side.

In addition to the tendency for the plates to tilt because of the higher axial force applied to the plates on the high pressure side than on the low pressure side, there is another set of forces acting on each plate also tending to tilt the same and acting so as to augment the first mentioned tendency to tilt. Thus, the fluid pressures within plate recesses 34, 35 which act in radial outward directions on walls 38, 39 impose a net force on such walls generally in the direction of the high pressure side of the pump, whereas the net counteracting bearing loads applied to openings 62, 63, 72 and 73 act on the plate in an opposite direction and at points axially offset from the net fluid pressure force acting on walls 38, 39. Since these net forces are opposite and axially offset, they apply a tilting force to the plates which tends to separate the plates on the high pressure side and bring them together on the low pressure side. Since the forces are opposite and equal there is no tendency for the plates to shift radially.

The forces tending to axially separate and to tilt the plates are counteracted by buttons **20**, **115**, **116**, **117** and **118**. The force applied by each button is proportional to the force tending to separate and tilt the plates and acting in the vicinity of the particular button. Thus buttons **116** and **118**, which are on the low pressure side of the pump, receive their fluid through passages **111** and **113**, respectively. These passages are exposed to tooth pockets toward the low pressure side of the pump and generally opposite the particular button. The fluid pressures tending to separate the plates in these areas are relatively low and therefore the fluid pressure behind each button **116** and **118** is likewise relatively low. The net area of these buttons acted upon by this fluid for urging the buttons toward the plates, that is, the area of the outer end face of each button between the hole through the button and the projected outside diameter of the packing ring, is selected so as to provide a force somewhat greater than necessary for counterbalancing the axial separating forces on the portions of the plates served or counterbalanced by the particular buttons less a proportionate part of the force with which the plates tend to be brought together in this region as a result of the tendency for the plates to tilt.

In like manner buttons **115**, **117** on the high pressure side of the pump receive fluid pressure through passages **110**, **112** which intersect the pumping chamber in regions where the fluid pressure in the tooth pockets is relatively high and thus the fluid pressure behind such buttons is also relatively high. Button **20** is exposed to full discharge pressure in counterbore **19**. The area of each of these three buttons **115**, **117** and **20** acted upon by fluid pressure urging them toward the plates is so selected that the buttons press against plate **27** with a force somewhat greater than necessary for counterbalancing the axial separating forces on the corresponding portions of plates **26**, **27** plus a force for counterbalancing the separating force applied to these portions of the plates due to the tendency of the plates to tilt.

With all of the axial separating and tilting forces on the plates counterbalanced by the buttons in the manner just described, the plates will have substantially uniform contact pressure at all points of their meeting faces or ribs **40**, **41** in the case of the fixed clearance arrangement of FIG. 7.

Since recesses **34**, **35** are of equal depth and of like size and shape, the radial and tilting forces acting on the plates are identical and opposite. It is important that they be so because the force applied to plate **27** by the buttons to oppose tilting is also transmitted to plate **26**. If the recesses were of different depth, size or shape, the tilting force on one plate would be different than on the other plate and balancing of the force on one plate by means of the buttons would not result in balancing of the forces on the other plate and the unbalanced plate would impose side loads on the dowels. Also, the radial loads on each plate would be unbalanced and result in side loads on the dowels. With both plates balanced radially and counterbalanced against tilting forces, as described, there is no appreciable side load or binding upon the dowels.

In the fixed clearance type pump ribs **40**, **41** normally remain in contact and the side faces of the gears are normally out of contact with the bottom wall of the recesses so as to maintain clearances **42**, **43**. When temperature changes occur in a manner for momentarily taking up clearances **42**, **43** the gears will contact the plates but will not bind thereagainst since plate **27** will move slightly in a direction away from the gears against the holding force of the buttons, there being a clearance **138** between plate **27** and cap **14** to permit such movement. In prior forms of fixed clearance pumps in which the end plates are mechanically clamped or bolted in position next to the gears, the clamping or bolting prevents such outward movement of the end plate and hence the clearances must be sufficiently large to permit any differential expansion or contraction which is apt to occur if the pump is to be protected against binding and other damage. Since the plates in the present invention may move away from the gears as described, the fixed clearances may be very small, as for example from .0002" to .0005" total for both sides, or they may even be zero, without danger of damage being done to the pump. During the intervals in which clearances **42**, **43** are taken up and the gear side faces contact the plates, ribs **40**, **41** may be separated slightly and the pump will function as a pressure loaded type with the plates pressing against the gears with controlled force from the buttons.

At no time in either the fixed clearance or pressure loaded forms, will the plates be moved in a direction axially away from the gears by fluid pressure, they are always either in contact with each other at ribs **40**, **41** or in contact with the gear side faces. Moreover, ribs **40**, **41** are sufficiently narrow so that changes in fluid pressure gradient therebetween as the ribs move toward or away from each other, in either fixed clearance or pressure loaded types, do not prevent the buttons from maintaining the plates in contact with each other or with the gears, as indicated.

It is important that plate **27** be able to move freely axially along dowels **30**, **31** in the manner just described and at no time should the dowels offer frictional resistance to such movement. This is especially true in ultra high speed pumps as required for missile and space vehicle applications since such high speeds can induce extremely rapid differential thermal expansion and contraction and the plates must respond extremely rapidly with compensating sliding movement in order to prevent binding of the gears against the end plates and consequent pump failure. By balancing out the radial and tilting forces on the plates as herein described there are no appreciable side loads imposed between the plates and dowels and hence no appreciable frictional resistance to axial movement of the plates.

It may be noted that the pressure of the gear shafts against the bearing surfaces of the plates in openings **62**, **63**, **72** and **73** does not offer appreciable frictional resistance to axial movement of the plates because of the fact that the shafts are rotating within such openings.

Since plate **26** butts against an inner transverse wall of head **13**, it is apparent that when it shrinks or expands in axial length its one end remains against the wall of head **13** and the axial shortening or lengthening is reflected at the end adjacent the gears. Such axial shortening and lengthening results in axial sliding movement with respect to dowels **30**, **31** and thus it is important that it also have no side loading or binding upon the dowels which would interfere with such axial sliding. Since it is necessary for both plates to be freely movable upon the dowels it is important that recesses **34**, **35** be of like depth, size and shape so that tilting forces will be identical and may be simultaneously counterbalanced by the action of the buttons, as described.

In the pressure loaded arrangement of FIG. 8, the buttons cause plates **26**, **27** to press against the gear side faces with a selected force that is proportional to the output pressure of the pump. When expansion or contraction of the gears and/or plates occurs the buttons and plate **27** will move in a direction toward or away from plate **26** and a slight increase or decrease in clearance **49** will result. Upon reversal of the conditions inducing this action the direction of movement of the buttons and plate **27** will likewise reverse and the predetermined minute clearance **49** will again be established.

Thus in the FIG. 8 form it is likewise important that the plates be freely axially movable on the dowels for rapid response to the changing conditions so that the pump may continue to operate efficiently and without damage. This form is preferred for pump discharge pressures of over 1,500 p.s.i. Clearance **49** is kept very small so that leakage therethrough is minor, it being less than

it would be through a like clearance at the side face of the gear since leakage between stationary parts is less than between relatively moving parts when the clearances are of equal size.

In the form of the invention illustrated by FIGS. 9 and 10, the raised rib 40*a* on plate 27 is provided with a groove 130 in which a flexible sealing member 131 is located thus providing an auxiliary seal in addition to the seal formed by the ribs. Preferably, groove 130 is formed with one vertical wall 132 and a tapered wall 133, wall 132 being vertical to keep the width of the groove to a minimum and wall 133 being tapered to provide a wedge shaped space into which sealing member 131 may be tightly forced by fluid pressure from pumping chamber recess 35. Groove 130 is of a depth slightly greater than the corresponding dimension of the sealing member so that the latter does not contact the bottom of the groove.

Although several forms of the invention have been shown and described, it is obvious that many other modifications may be made within the scope of the invention as defined by the claims.

I claim:

1. A pump comprising a housing, first and second plates within said housing and having opposed recesses therein forming a pumping chamber, means supporting said plates in said housing, at least one of said plates being axially movable relative to the other, inlet and outlet passages in communication with said pumping chamber, rotary pumping means in said pumping chamber operative when rotated for forcing fluid from said inlet passage to said outlet passage, and fluid pressure means forcing said one of said plates axially toward the other.

2. A pump in accordance with claim 1 in which there is an auxiliary sealing means surrounding at least a portion of said pumping chamber and in sealing contact with both said plates.

3. A pump in accordance with claim 1 in which said one of the plates has a groove surrounding at least a portion of the chamber, and a sealing member in said groove and in sealing contact with both said plates.

4. A pump in accordance with claim 3 in which said groove has a tapering wall on the low pressure side of said sealing means, said sealing means being movable by fluid pressure in said chamber into wedging engagement with said tapered wall and the other of said plates.

5. A pump in accordance with claim 3 in which said groove has a tapering wall on the low pressure side of said sealing means, said sealing means being shaped and dimensioned so as to be normally out of contact with the bottom of said groove whereby a greater area of said sealing means is exposed to the high pressure side thereof, said sealing means being movable by fluid pressure acting on said area into wedging engagement with said tapered wall and a surface on the other of said plates.

6. A pump comprising a housing, first and second plates within said housing, means supporting said plates within said housing, said plates having opposed recesses forming a pumping chamber, a pair of meshed rotatable gears in said chamber, each gear having a running fit with sections of the radially outer walls of the recesses so as to divide the pumping chamber into a high pressure portion and a low pressure portion, an inlet passage leading to said low pressure portion, an outlet passage leading from said high pressure portion, said gears when rotated serving to deliver fluid from said low pressure portion to said high pressure portion, and fluid pressure means for utilizing fluid from said chamber for urging one of said plates axially toward the other for sealing the side faces of the gears.

7. A pump in accordance with claim 6 in which said plates have opposed faces surrounding at least a portion of said pumping chamber, said opposed faces being normally held in contact with each other by said fluid pressure means for closing said chamber, the bottom walls of said recesses having a slight clearance with the side faces of the gears.

8. A pump in accordance with claim 6 in which said plates have opposed faces surrounding at least a portion of said pumping chamber, the bottom walls of said recesses being in engagement with said gear side faces so as to provide a slight clearance between said opposed faces.

9. A pump in accordance with claim 6 in which there is a dowel means axially aligning said plates, said dowel means having one end only thereof engaged with the housing for locating the plates radially with respect to said housing.

10. The pump of claim 6 in which said plates have opposed faces surrounding said recesses, at least one of said faces has a raised rib adjacent the pumping chamber and cooperating with said other face to close said chamber, said rib defining an area subject to a pressure gradient of fluid from said chamber which tends to separate said plates, said rib being narrow so as to minimize the area subject to such pressure gradient, and a section of said one face outwardly of the rib being connected to a low pressure zone of said pump and being spaced from said other face to provide for free flow of fluid leaking past said rib to said low pressure zone whereby buildup of fluid pressure on said faces outwardly of said rib is avoided.

11. A pump in accordance with claim 10 in which said other plate has a raised rib opposed to the first mentioned rib and cooperating therewith to close said chamber.

12. A pump comprising a housing, first and second plates within said housing, means supporting said plates within said housing, said plates having opposed recesses forming a pumping chamber, first and second openings through the bottom walls of each recess, a pair of meshed rotatable gears in said chamber and having a running fit with sections of the radially outer walls of the recesses so as to divide the pumping chamber into a high pressure portion and a low pressure portion, said gears when rotated serving to deliver fluid from the low pressure portion to the high pressure portion, means for utilizing fluid from said high pressure portion for urging one of said plates axially toward the other for closing said pumping chamber, said gears having shafts in bearing contact with the walls of said openings for transmitting to said plates radial loads impressed upon said gears by fluid in said chamber, the radially outer walls of said recesses being subject to radial loads from fluid in said chamber so as to counterbalance the radial loads applied to said plates by said shafts, said recesses being symmetrical and of equal depth and said gears extending into said recesses equal distances whereby the radial loads transmitted by the gears and by fluid pressure in said chamber are the same on both plates.

13. A pump comprising a housing, first and second plates within said housing, means mounting said plates within said housing, said plates having opposed recesses forming a pumping chamber, a pair of meshed rotatable gears in said chamber, the sides of said gears having a sealing fit with the bottom walls of said recesses and each gear having a running fit with sections of the radially outer walls thereof so as to divide the pumping chamber into a high pressure portion and a low pressure portion, an outlet passage leading from said high pressure portion, said gears when rotated serving to deliver fluid from said low pressure portion to said high pressure portion, said gears having shafts in bearing contact with said plates, said plates being acted upon by fluid in said chamber and by said shafts in a manner tending to axially separate and to tilt said plates, and fluid pressure means for applying force to selected portions of at least one of the plates for precisely counteracting the tilting forces acting on said one plate and for preventing axial separation of said one plate from said gear sides.

14. A cartridge for use in a pump housing, said cartridge comprising a pair of plates having opposed recesses

therein forming a pumping chamber, inlet and outlet passages at opposite sides of said plates communicating with said pumping chamber, a pair of gears within said chamber, dowel means for axially aligning said plates, at least one of said plates being axially movable on said dowel means toward the other of said plates for closing said pumping chamber.

15. A pump comprising a housing, first and second plates within said housing and having opposed recesses forming a pumping chamber with a low pressure inlet zone and a high pressure outlet zone, means for supporting said plates within said housing, at least one of said plates being axially movable relative to the other, gears rotatable in said chamber for forcing fluid from said inlet zone to said outlet zone, fluid pressure means for forcing said one plate axially toward the other, said recesses and gears having opposed surfaces subject to radially acting forces from fluid in said pumping chamber tending to radially displace the gears relative to the plates, said gears having shafts in bearing contact with the plates at points axially offset from said recesses for preventing such radial displacement, such bearing contact applying radial forces to said plates in opposition to and axially offset from the radial forces acting on said recess surfaces, said fluid pressure means counteracting the tendency of said plates to tilt due to said axially offset radial forces acting oppositely thereon.

16. The pump of claim 15 in which the recesses are of equal depth and the gears extend the same axial distance into each recess whereby the forces tending to tilt each plate are the same.

17. A pump comprising a housing, first and second plates within the housing having opposed recesses therein and axially movable relative to each other, means for supporting said plates within said housing, said recesses forming a pumping chamber with a low pressure inlet zone and a high pressure outlet zone, meshed gears within said chamber and rotatable therein for forcing fluid from said inlet zone to said outlet zone, said plates having radially extending areas respectively subject to high pressure fluid in said outlet zones and to low pressure fluid in said inlet zones whereby fluid pressure in said zones tends to axially separate said plates, said plates and gears having axially extending opposed areas respectively subject to radially acting forces from high pressure fluid in said outlet zone and from low pressure fluid in said inlet zone, the net effect of said radially acting forces tending to radially displace the gears toward the low pressure inlet zone, said gears having shafts in bearing contact with the plates at points axially offset from said opposed areas for preventing such radial displacement, such bearing contact applying radial forces to said plates in opposition to and axially offset from the net radial forces acting on said axially extending plate areas, said axially offset forces tending to tilt said plates, and fluid pressure actuated means acting on at least one of said plates to counteract both said axial separating and plate tilting forces.

18. The pump of claim 17 in which the axes about which the plates tend to tilt are so located that the net effect of the tilting forces tends to move the plates in a separating direction at the high pressure zones and in a converging direction at the low pressure zones, and said fluid pressure means compensates for said tilting tendency by applying greater force to said one plate at the high pressure zones than required for counteracting said axial separating forces at said high pressure zones and by applying less force to said one plate at the low pressure zones than selected for counteracting said axial separating forces at said low pressure zones.

19. The pump of claim 18 in which the force applied to said one plate at the high pressure zone by said fluid pressure actuated means is greater than a force selected for overbalancing said axial separating force at said high pressure zone by an amount substantially equal to the plate separating force applied at the high pressure zone due to said tilting forces and the force applied to said plates at the low pressure zone by said fluid pressure actuated means is smaller than a force selected for overbalancing said axial separating force at said low pressure zone by an amount substantially equal to the plate converging force applied at the low pressure zone due to said tilting forces.

20. A pump comprising a housing, first and second plates within the housing having opposed recesses therein, means supporting said plates within said housing, said recesses forming a pumping chamber with a low pressure inlet zone and a high pressure outlet zone, meshed gears within said chamber and rotatable therein for forcing fluid from said said inlet zone to said outlet zone, dowel means for axially aligning said plates, at least one plate being axially slidable on said dowel means toward and away from the other plate to accommodate thermal expansion and contraction of the gears and plates, said plates having radially extending areas respectively subject to high pressure fluid in the outlet zone and to low pressure fluid in the inlet zone whereby fluid pressure in said zones tends to axially separate said plates, said plates and gears having axially extending opposed areas respectively subject to radially acting forces from high pressure fluid in the outlet zone and low pressure fluid in the inlet zone, the net effect of said radially acting forces tending to radially displace the gears toward the low pressure inlet zone, said gears having shafts in bearing contact with the plates at points axially offset from said opposed areas for preventing such radial displacement, such bearing contact applying radial forces to said plates in opposition to and axially offset from the net radial forces acting on said axially extending plate areas, said axially offset forces tending to tilt said plates in a manner which would cause at least said one plate to impose a radial load upon said dowel means, and fluid pressure operated means for counteracting said axial separating forces but yieldable to permit said axial sliding of said one plate on said dowel means due to said thermal expansion and contraction, said fluid pressure operated means also counteracting said tilting forces whereby radial loading of said dowel means by said one plate is avoided.

21. The pump of claim 20 in which both of said plates are axially slidable on said dowel means and said fluid pressure means causes counteraction of said tilting forces on said other plate.

22. The pump of claim 20 in which said dowel means is engaged in housing recess means at one end only of said plates for locating said plates within said housing.

23. A pump comprising a housing having a pumping chamber with inlet and outlet passages, gear means rotatable within the chamber for forcing fluid from the inlet passage to the outlet passage, axially movable plate means mounted for sealing side faces of said gear means, said plate means being acted upon at different points by differing fluid pressures in said chamber urging the plate means in a direction axially away from the gears, a stop for limiting movement of said plate means toward said gears for establishing an initial predetermined clearance between the gear side faces and said plate means, a multiple of fluid pressure actuated means for urging the plate means against said stop, each of said fluid pressure actuated means being acted upon by a fluid pressure corresponding to that at a selected one of said points.

24. A pump comprising a housing, a pair of plates mounted within said housing, said plates having opposed recesses forming a pumping chamber with inlet and outlet passages, gears rotatable within said housing for forcing fluid from said inlet passage to said outlet passage, said plates sealing the side faces of said gears, at least one of said plates being movable in axial directions toward and away from said gears, said one plate being acted upon at different points by differing fluid pressures in said chamber urging said one plate in a direction axially away from the gears, a multiple of fluid pressure actuated

means for urging said one plate toward said one gear with greater force than said one plate is urged away from said gears by fluid pressure within said chamber, each of said fluid pressure actuated means being acted upon by fluid from a selected one of said points.

25. The pump of claim 24 in which said one plate is also subject to forces tending to tilt the same, and said fluid pressure actuated means counterbalances said tilting tendency.

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