April 17, 1962 J. L. NAGELY 3,029,739

GEAR PUMP OR MOTOR WITH RADIAL PRESSURE BALANCING MEANS

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Fig. 1.

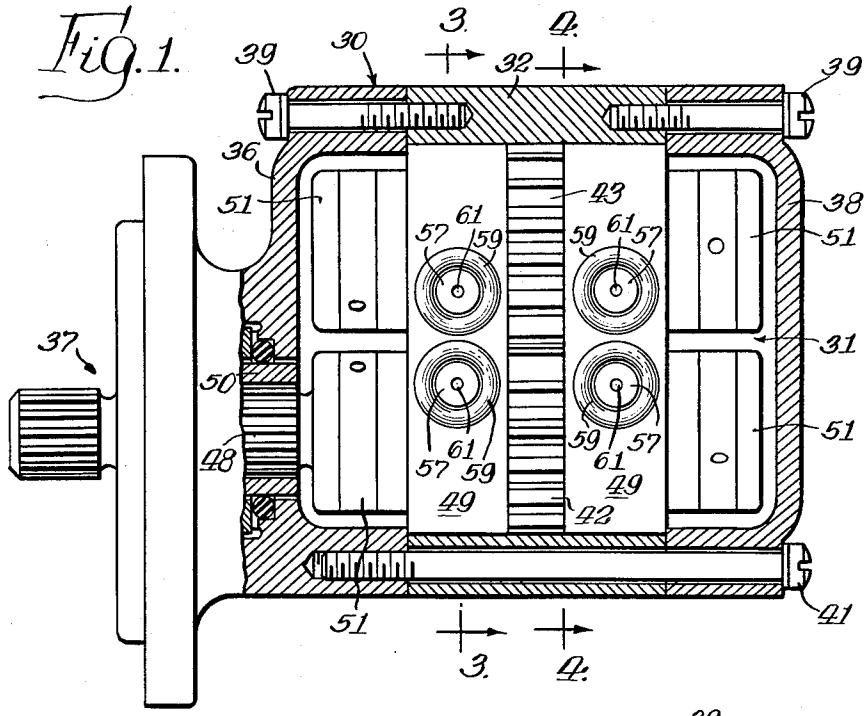


Fig. 3.

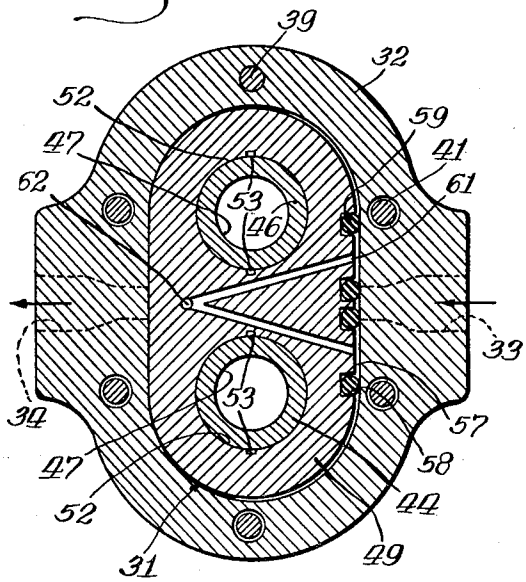
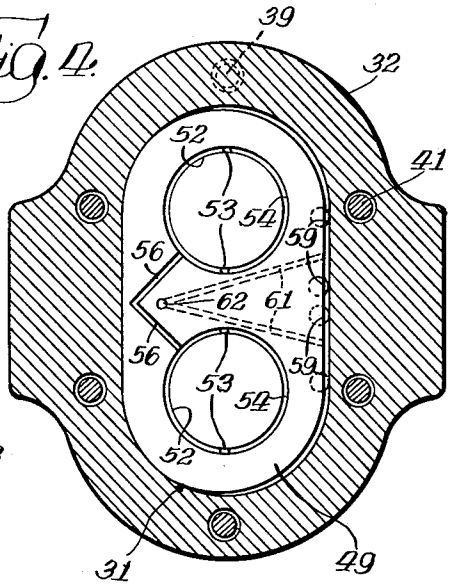


Fig. 4.



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Fig. 2.

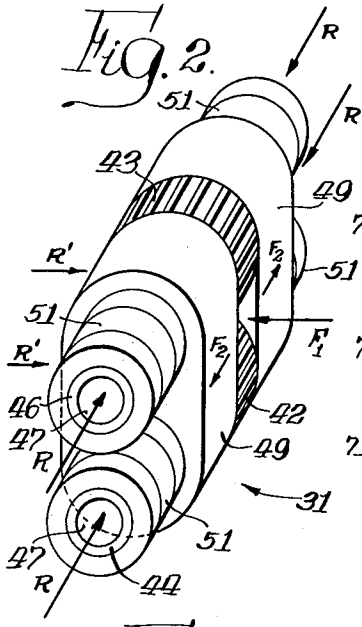


Fig. 5.

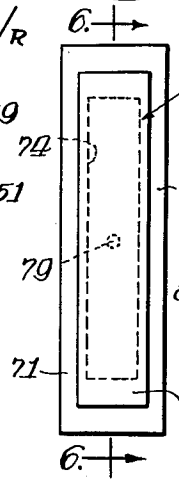


Fig. 6.

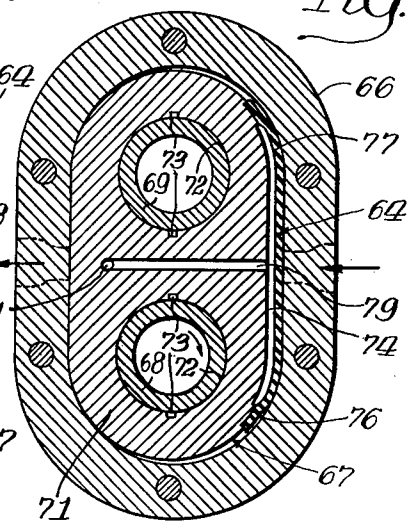


Fig. 9.

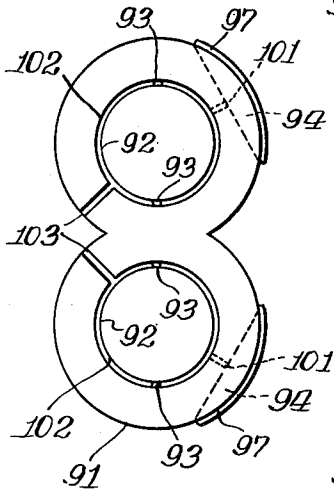


Fig. 7.

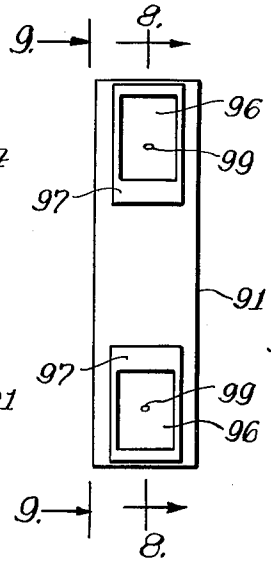
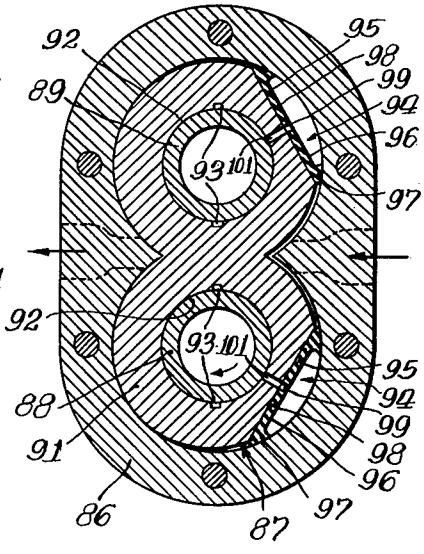


Fig. 8.

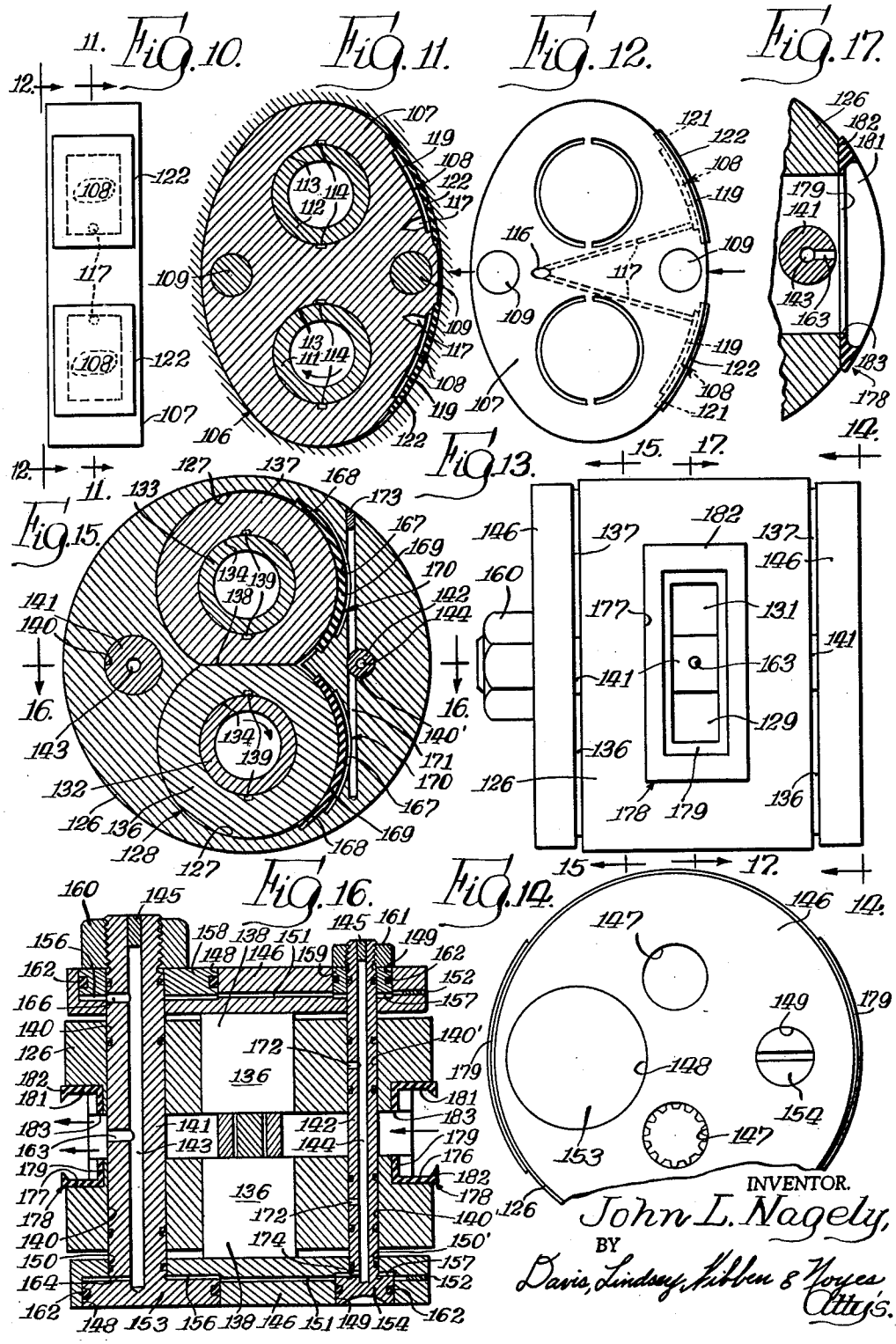


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Fig. 18.

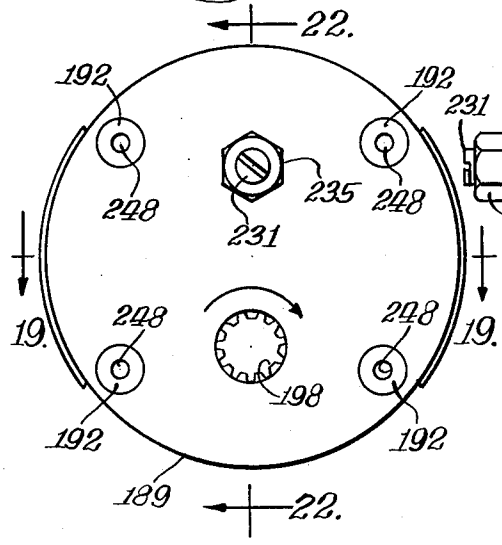


Fig. 20.

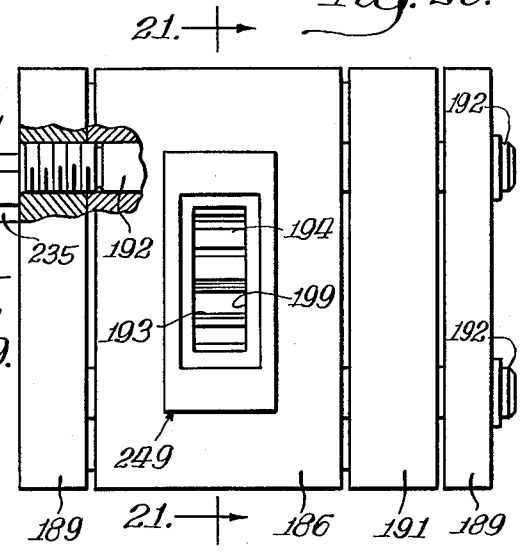


Fig. 19.

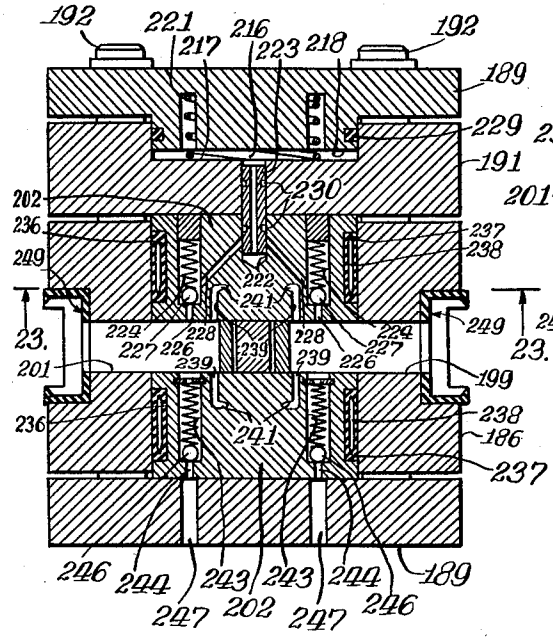
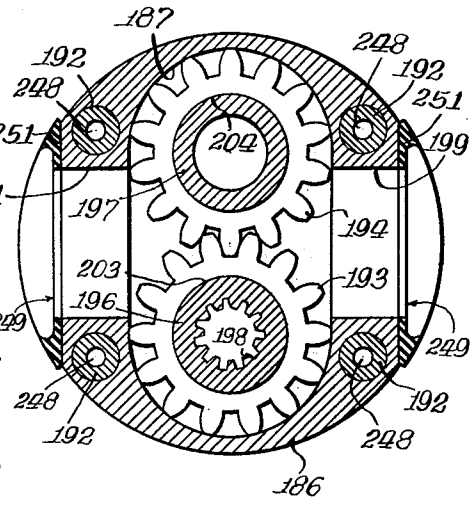


Fig. 21.



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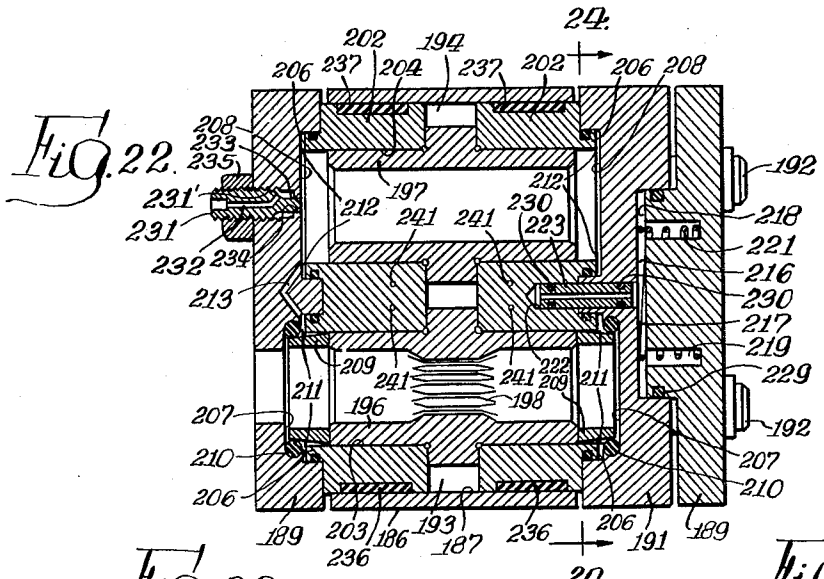
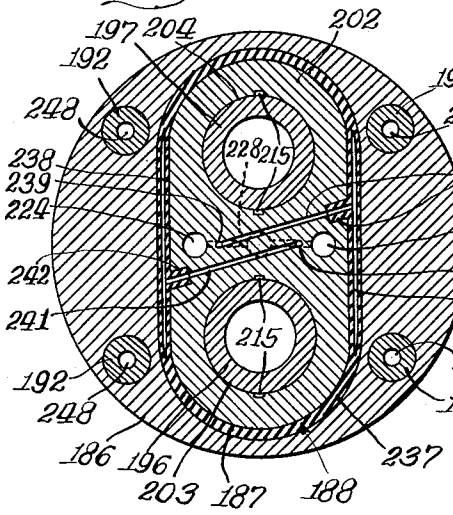
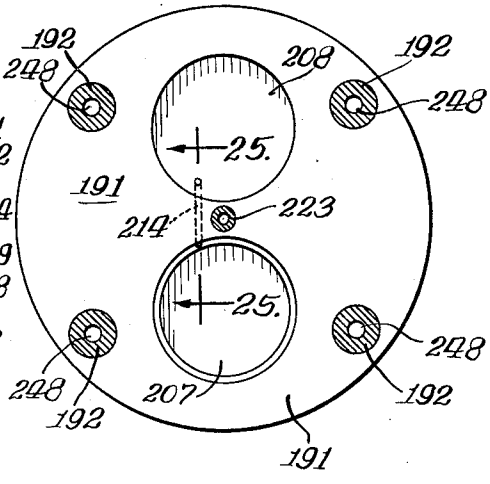


Fig. 22.

Fig. 23.

Fig. 24.





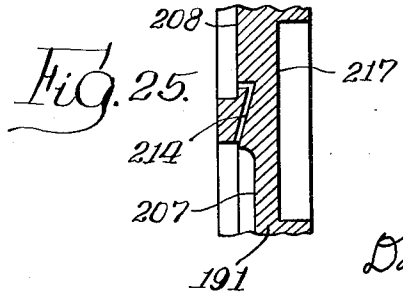


Fig. 25.

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United States Patent Office 3,029,739 Patented Apr. 17, 1962

3,029,739

GEAR PUMP OR MOTOR WITH RADIAL PRESSURE BALANCING MEANS

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Filed July 9, 1958, Ser. No. 747,410
14 Claims. (Cl. 103—126)

This invention relates to improvements in high pressure gear-type liquid displacement devices such as hydraulic pumps or motors. More particularly, the invention relates to a radial pressure balancing arrangement for a gear-type liquid displacement device employing an internal unit assembly construction.

One of the problems present in the high pressure gear pump art has been that of counterbalancing the effects of the discharge pressure force at the high pressure side of the pump, particularly the axial and radial components of this force, which tend to axially separate the adjacent bushings or end plates from the side faces of the gears.

In my prior United States Patents No. 2,772,638, issued December 4, 1956, and No. 2,848,953, issued August 26, 1958, I have disclosed and claimed a novel high pressure gear pump employing an internal unit assembly type construction in which the gears thereof are supported and rotatably journalled in a pair of adjacent spaced end plates, the internal unit assembly including thrust bearing means cooperating with the end plates to provide annular annular axial pressure loading chambers for counterbalancing the axial component of the discharge pressure force tending to separate the end plates from the side faces of the gears, the reaction forces from the pressure loading being taken up by the gear shafts or by separate tension members so that the axial forces are all resolved within the internal unit assembly independently of the pump casing. However, the pressure differential existing across the gears from the discharge side to the suction side of the pump results in a radial force component acting on the internal unit assembly as a free body and tending to displace the assembly bodily toward the suction side so as to create substantial frictional forces between the edges of the end plates of the pump and the casing bore at the suction or low pressure side. Such axial frictional forces tend to influence or interfere with proper positioning of the end plates relative to the gear faces and the thrust bearing means so that the axial counterbalancing force required to maintain the end plates engaged with the side faces of the gears are variable and oftentimes quite large. The present invention provides a simple and effective means for counterbalancing this radial force component tending to urge the internal unit assembly into frictional contact with the suction side of the casing bore.

Although the invention hereinafter to be described is primarily suited for use in high pressure gear-type pumps employing an internal unit assembly construction, it will be understood that the principles involved are applicable to liquid displacement devices generally, including both hydraulic pumps and motors.

Accordingly, a primary object of my invention is to provide a novel radial pressure balancing system for a liquid displacement device of the above-described type.

Another object of the invention is to provide a novel radial pressure balancing system for counterbalancing the radial force component of the discharge pressure force in a liquid displacement pump of the type previously described to thereby maintain the liquid displacement elements of the pump in their prescribed reference position in order to obtain maximum pump efficiency.

A further object of the invention is to provide a novel radial pressure balancing system for counterbalancing the radial force component of the discharge pressure force at the high pressure zone in a positive displacement pump employing an internal unit assembly construction in order to maintain a minimum clearance relationship between the internal unit assembly and casing bore on the discharge side of the pump and thereby minimize leakage around the internal unit assembly.

Still another object of the invention is to provide, in a high pressure gear pump employing axial pressure loading chambers and an internal unit assembly construction, a novel radial pressure balancing system for counterbalancing the radial force component of the discharge pressure which tends to urge the edges of the end plates of the internal unit assembly into frictional contact with the pump casing bore at the suction side, thereby to minimize such frictional contact and reduce the axial counterbalancing force required to counterbalance the axial component of the discharge pressure force tending to separate the end plates from the side faces of the gears.

Other objects and advantages of the invention will become evident from the following detailed description taken in connection with the accompanying drawings, in which:

FIG. 1 is a longitudinal sectional view, with some parts in elevation, showing a gear-type liquid displacement device that employs an internal unit assembly construction and radial pressure balancing means comprising one specific embodiment of my invention;

FIG. 2 is a reduced scale perspective view of the internal unit assembly of the liquid displacement device illustrated in FIG. 1 and showing graphically the various forces acting on the internal unit assembly as a free body;

FIG. 3 is a sectional view taken along the line 3—3 of FIG. 1;

FIG. 4 is a view similar to FIG. 3, taken along the line 4—4 of FIG. 1;

FIG. 5 is a side elevational view of an end plate of an internal unit assembly having radial pressure balancing means comprising another specific embodiment of the invention;

FIG. 6 is a sectional view along the line 6—6 of FIG. 5 but showing the end plate in assembled relation in the pump casing;

FIG. 7 is a side elevational view of an end plate of an internal unit assembly comprising another specific embodiment of the invention;

FIG. 8 is a cross-sectional view along the line 8—8 of FIG. 7 but showing the end plate in assembled relation in the pump casing;

FIG. 9 is an elevational view as seen along the line 9—9 of FIG. 7;

FIG. 10 is a side elevational view of still another end plate embodying another form of the invention;

FIG. 11 is a sectional view along the line 11—11 of FIG. 10 but showing the end plate in assembled relation in the pump casing;

FIG. 12 is an elevational view as seen along the line 12—12 of FIG. 10;

FIG. 13 is a side elevational view of an internal unit assembly comprising another embodiment of the invention;

FIG. 14 is an end elevational view of the internal unit assembly of FIG. 13 as seen along the line 14—14 thereof;

FIG. 15 is a sectional view taken along the line 15—15 of FIG. 13;

FIG. 16 is a sectional view along the line 16—16 of FIG. 15;

FIG. 17 is a fragmentary sectional view taken along the line 17—17 of FIG. 13;

FIG. 18 is an end elevational view of another internal unit assembly comprising another specific embodiment of the invention;

FIG. 19 is a sectional view taken along the line 19—19 of FIG. 18;

FIG. 20 is a side elevational view of the internal unit assembly of FIG. 18, and with a portion thereof broken away to show a structural detail;

FIG. 21 is a transverse sectional view taken along the line 21—21 of FIG. 20;

FIG. 22 is a sectional view taken along the line 22—22 of FIG. 18;

FIG. 23 is a sectional view taken along the line 23—23 of FIG. 19;

FIG. 24 is a sectional view taken along the line 24—24 of FIG. 22; and

FIG. 25 is a fragmentary sectional view taken along the line 25—25 of FIG. 24.

Briefly described, my invention comprises a novel radial pressure balancing system wherein high pressure working liquid is directed to one or more confined chambers or zones on the low pressure side of a liquid displacement device, such that the reaction force provided by high pressure liquid in these chambers counterbalances the radial displacing component on the high pressure side of the device. My invention is particularly suited for use in liquid displacement devices of the type employing an internal unit assembly construction having a pair of axially shiftable end plates disposed on the opposite side faces of the gears for rotatably journalling the shaft portions of the gears and which utilizes one or more pairs of axial pressure loading chambers disposed adjacent the outer end faces of the end plates for counterbalancing the axial force component of the discharge pressure force tending to separate the end plates from the gear faces, such a system being described and claimed in my United States Patents No. 2,772,638, issued December 4, 1956, and No. 2,848,953, issued August 26, 1958.

The present invention contributes to the effective operation of the aforementioned axial pressure loading chamber system not only by minimizing leakage between the internal unit assembly and the casing bore, but also by maintaining the internal unit assembly substantially centralized in its casing bore so that the frictional forces between the peripheral edges of the end plates and the casing bore are minimized. To this end, several radial pressure balancing chamber arrangements are illustrated and described in conjunction with a series of related internal unit assembly constructions, each of which comprises an exemplary embodiment of the present invention. As will hereinafter be more fully described, one or more radial pressure balancing chambers are disposed between the internal unit assembly and the pump casing on the low pressure side thereof, the chambers being so located and having combined effective areas such that high pressure working liquid communicated to the chambers will effectively counterbalance the radial component of the discharge pressure and thus maintain the internal unit assembly centralized in the pump casing as a free body.

Referring now to FIGS. 1–4 of the drawings, the invention will first be described in connection with a gear-type liquid displacement device employing an internal unit assembly construction of the type shown in my aforementioned United States Patent No. 2,772,638, to which reference may be made for further details of the structure and operation thereof.

The liquid displacement device illustrated in FIG. 1, and in the succeeding figures, is of the high pressure gear pump type and generally comprises an external casing 30 and an internal unit assembly, indicated at 31, which constitutes the operating mechanism of the pump and is readily removable as a unit from the casing 30. The external casing 30 generally includes a central body portion 32 having a liquid inlet port 33 and an outlet port 34 (FIGS. 3 and 4) formed therein, a front cover 36 at the drive end of the pump including driving means and seal means, indicated generally at 37, and a rear cover 38 at the anti-drive end of the pump. The covers 36 and 38 are removably secured to the body portion 32 by screws 39 extending therethrough and threaded into the central body portion 32 and also by a plurality of elongated screws 41 extending through bores in the rear cover 38 and body portion 32 threaded into the opposite front cover 36. The construction of the driving and sealing means 37 at the drive end of the pump is conventional and further description thereof has therefore been omitted.

The internal unit assembly 31 generally comprises a pair of liquid displacing elements such as a drive gear 42 and driven gear 43 (FIGS. 1 and 2) carried integrally on gear shafts 44 and 46 (FIGS. 2 and 3), respectively, each having an axial bore 47 therethrough. The drive end of the gear shaft 44 adjacent the front cover 36 may carry an extension formed with a drive spline 48 for engagement with a complementary drive spline 50 in the driving means 37, though any other suitable drive connection could be utilized for transmitting torque from the driving means 37 to the internal unit assembly 31.

In order to support the gears 42 and 43 in meshed relationship and to maintain a pumping seal around the side faces of the gears, a pair of end plates 49 are disposed at the opposite side faces of the meshed gears, each end plate having axial bores therethrough, constituting bearing journals for the gear shafts 44 and 46. Encircling the outer ends of the gear shafts 44 and 46 are annular sealed thrust bearing means 51 which define annular axial pressure loading chambers for receiving working liquid at pump discharge pressure to urge the end plates inwardly toward the gears 42 and 43 and to counterbalance the hydraulic pressure forces at the discharge side of the pump tending to separate the end plates 49 from the side faces of the gears 42 and 43. Lubrication of the gear shaft journals, indicated generally at 52 in FIGS. 3 and 4, is provided by a pair of diametrically spaced axially extending grooves 53 which register with a pair of annular grooves 54 in the inner end faces of the end plates and receive high pressure working liquid from a pair of angularly intersecting grooves 56 having their common juncture point registering with the discharge zone of the pump.

The particular arrangement and construction of the thrust bearings 51 and integral axial pressure loading chamber system, including the manner in which high pressure working liquid is communicated to the pressure loading chambers, is set out in more detail in my aforementioned United States Patent No. 2,772,638 and further description thereof is unnecessary for purposes of the present invention.

It will be apparent that in addition to the axial component of the pump discharge pressure force tending to separate the end plates from the side faces of the meshed gears, a radial component of this force is also present and actionable to urge the internal unit assembly toward the low pressure side of the pump casing which introduces a leakage problem around the high pressure zone of the pump unless elaborate seals are provided. The displacement of the internal unit assembly toward the low pressure side of the casing bore also creates a frictional force resulting from the edge contact of the end plates with the housing bore which tends to resist the axial inward counterbalancing force of the axial pressure loading chambers. By the present invention, this displacement of the internal unit assembly is overcome, as described below, by imposing a radially inwardly directed counterbalancing force at the low pressure or inlet side of the casing bore that is substantially equal in magnitude but opposite in sense to the displacing radial force at the discharge side of the internal unit assembly, thereby to maintain the internal unit assembly substantially centralized as a free body in the casing bore.

The various generated and reaction forces acting on the internal unit assembly 31 as a free body are illustrated diagrammatically in the force diagram of FIG. 2. In FIG. 2, F_1 represents the resultant radially inwardly directed component of the discharge pressure force acting at the gears of the internal unit assembly 31. The arrows F_2 represent axial components of the discharge pressure

force acting to separate the end plates from the side faces of the gears. The opposed pairs of axially inwardly extending arrows R at the ends of each gear shaft 44 and 46 represent the axial counterbalancing force exerted by the pressure loading chambers in the thrust bearing means 51 whose summation is equal in magnitude but of opposite sense to the separating forces F_2. The force F_1 acts on the internal assembly 31 at the high pressure side thereof to urge the latter, as a free body, toward the low pressure side of the casing bore, this force being conventionally resisted as a bearing load. In the present invention, however, one or more pairs of radial counterbalancing forces R' are provided by the radial pressure balancing chambers on the low pressure side of the casing bore between the edges of the end plates and the adjacent wall of the casing bore to counterbalance the displacing force F_1. The forces R' are cumulatively equal in magnitude to the force F_1 but opposite in sense so that the internal unit assembly 31 is maintained in a substantially centralized equilibrium position in the casing as a free body.

In somes cases it may be advantageous to introduce a slight unbalance between the displacing and counterbalancing radial force components F_1 and R', respectively, to thereby maintain the internal unit assembly urged toward the high pressure or discharge side of the casing bore and thus reduce leakage at this point as well as the necessity for special seal means between these surfaces.

Referring again to FIG. 1 in conjunction with FIGS. 3 and 4, the radial pressure balancing means of the internal unit assembly 31 comprises a pair of annular chambers 57 formed at the low pressure edges of the end plates 49 and disposed substantially centrally thereof. The pressure balancing chambers 57 are defined by annular grooves 58 formed in the edge of each end plate and annular seal means in the form of resilient O-ring seals 59 seated in the grooves 58. The grooves 58 are sufficiently shallow so that portions of the seals 59 project beyond the edge surfaces of the end plates 49 for engagement with the adjacent inner side wall of the casing bore to form the confined annular chambers 57. The effective areas of the pressure balancing chambers 57 are such that the summation of the counterbalancing forces resulting from confined high pressure working liquid therein, and which is imposed on the edges of the end plates 49, will substantially counterbalance the radial force component of the discharge pressure force on the opposite side of the pump.

In order to subject the radial pressure balancing chambers 57 to pump discharge pressure and thus provide a counterbalancing force to centralize the internal unit assembly 31 in the bore of the casing 30, each end plate 49 includes a pair of intersecting angular bores or passages 61 that register at one end with the pressure balancing chambers 57 and at their other ends with the high pressure discharge side of the pump. Thus, the passages 61 extend convergently inwardly toward the high pressure discharge side of the pump and are intersected at their juncture point by a longitudinally extending bore 62 which communicates with the discharge pressure zone of the pump adjacent the outlet port 34 of the pump casing 30. It will be appreciated that confined pressurized working fluid in the radial pressure balancing chambers 57 provides a counterbalancing force to maintain the internal unit assembly 31 in radial equilibrium in the casing 30. In FIGS. 3 and 4, and also in the subsequent illustrated embodiments of the invention, the internal unit assembly is shown displaced completely toward the discharge side of the casing with the clearances being exaggerated for clarity. However, it is to be understood that the magnitude of the counterbalancing force may be arranged to provide a substantially central positioning of the assembly or any desired degree of unbalance.

Referring now to FIGS. 5 and 6, a modified radial pressure balancing means is illustrated and designated generally at 64, which comprises a second specific embodiment of the present invention. The pressure balancing means 64 is illustrated in conjunction with a gear type liquid displacement device that is substantially identical in construction and operation to the liquid displacement device described in FIGS. 1–4. Thus, the present embodiment includes an external casing 66 and a removable internal unit assembly 67. The internal unit assembly 67 includes a pair of meshed gears (not shown) that are carried on integral driving and driven gear shafts 68 and 69. A pair of end plates 71 are provided for rotatably journalling the shaft portions 68 and 69 of the gears, the end plates 71 and enclosing casing 66 being effective to define a high pressure zone adjacent the outlet of the device on operation thereof. A pair of axially extending grooves 73 are provided in the journalling bores 72 which register with the discharge pressure zone and the axial pressure loading chambers of the device to insure adequate lubrication at this point as previously described. The radial pressure balancing means 64 for the internal unit assembly 67 differs from the first embodiment of the invention in that the edges of the end plates 71 are each formed with a single vertically elongated rectangular cavity or recess 74 adjacent the inlet side of the pump. A second circumscribing marginal recess 76, of a somewhat shallower depth than the rectangular cavity 74, is provided to receive the marginal edges of a rectangular shaped resilient member 77 which encloses the cavity 74 and may be made of rubber or other suitable elastomer materials. The marginal edges of the resilient member 77 are bonded or otherwise suitably fastened in the circumscribing recess 76 such that the edges of the end plates 71 include what I term a "boot" type pressure balancing chamber.

It should be understood, however, that the "boot" type construction illustrated in FIGS. 5 and 6, which contemplates a single resilient member or strip 77 overlying and marginally fastened to the recess 74 to define the pressure balancing chamber therebetween, could be modified to include a double-walled resilient member construction with pressure sealed perimetrical edges, if desired, to eliminate the necessity of marginally connecting the resilient member 77 around the cavity 74. The latter construction would in effect provide an "innertube" type arrangement having certain maintenance and replacement advantages over the preferred form of construction of the resilient member 77 previously described.

High pressure liquid is communicated to the pressure balancing cavity 74 through a passage 79 which extends transversely through the end plate 71 and an axially extending bore 81 that registers with the high pressure zone of the pump. The operation of the radial pressure balancing system 64 is in all other respects similar to that of the first embodiment of the invention.

Referring now to FIGS. 7–9, a modified radial pressure balancing means is illustrated which comprises another embodiment of the invention and is shown in conjunction with a somewhat different internal unit assembly having a "figure-8" configuration. The general construction and operation of the liquid displacement device illustrated in FIGS. 7–9 is substantially the same as in the two previously described embodiments of the invention to the extent that it includes a casing 86 and an internal unit assembly 87 of the type employing axial pressure loading chambers. The internal unit assembly 87 includes a pair of meshed gears (not shown) that are carried integrally on a driving gear shaft 88 and driven gear shaft 89. A pair of "figure-8" shaped end plates 91 having a pair of axial bores 92 therethrough are provided to journal the gear shafts 88 and 89 and to provide a pumping seal with the side faces of the gears. Each end plate also includes a pair of diametrically spaced axially extending grooves 93 in the journalling bores 92 which register with the axial pressure loading chambers in the thrust means of the internal unit assembly 87 to provide a flow path for pressurized working liquid to lubricate

the gear shaft journals 92 as previously described. The position and location of the radial pressure balancing chambers in the internal unit assembly 87 and the orientation of the flow passages for directing working liquid at pump discharge pressure to the pressure balancing members differs, however, from the previously described embodiments of the invention as will now be described.

As will be apparent from FIGS. 7 and 8, the radial pressure balancing means in this embodiment of the invention comprises a pair of generally segmental chambers 94 disposed on the low pressure or inlet edges of the end plates 91 and angularly displaced toward the top and bottom ends of the internal unit assembly and casing. The angular orientation of the pressure balancing chambers 94 is necessitated by the "figure-8" construction of the internal unit assembly and casing bore, in that the high pressure zone adjacent the outlet of the pump tends to separate the gear shafts 88 and 89 as well as to urge the internal unit assembly toward the inlet of the pump, which effect is counterbalanced by angularly displacing the radial pressure balancing chambers 94, as shown in FIG. 8. While no specific angular relationship of the pressure balancing chambers 94 is contemplated, relative to a transverse axis through internal unit assembly 87, an included angular range of between 15° to 30° for each chamber is preferable. The pressure balancing chambers 94 are defined by a pair of rectangular resilient members 95 of rubber or the like having a planar base portion 96 and perimetrical upraised lips 97 for engagement with the adjoining walls of the casing bore, the resilient members 95 preferably being bonded at their base portions 96 to flattened chordal portions 98 on the end plates 91. The resiliently deflectable lips 97 project outwardly beyond the plate 91 and are arcuately shaped to conform to and tightly engage the inner walls of the casing bore in fluid pressure sealed engagement to thus localize the position and direction of the generated counterbalancing force component.

Working fluid substantially at pump discharge pressure is supplied to the pressure balancing chambers 94 through a passage system that includes a centrally disposed opening 99 in the base portion 96 of each resilient member 95, a radial connecting passage 101 in each end plate 91 which registers with an annular groove 102 (FIG. 9) in the side face of each end plate adjacent the gears, and a pair of radial passages 103 which register with the high pressure or discharge zone of the pump and extend into the grooves 102. Thus, it will be appreciated that high pressure working fluid is supplied from the high pressure or discharge zone of the liquid displacement device of FIG. 8 to the pressure balancing chambers 94 through a path around the side face of the end plates 91 to augment lubrication of the gear shaft journal bores. In addition, the angularly offset arrangement of the pressure balancing chambers 94 is also advantageous to counterbalance the separating action of this high pressure zone of the pump when a "figure-8" type internal unit assembly and casing construction is used.

Referring now to FIGS. 10–12, another radial pressure balancing means is illustrated in conjunction with a liquid displacement device employing an oval type internal unit assembly 106 which comprises another embodiment of the invention. The internal unit assembly 106 includes a pair of oval or rounded contour end plates 107 having a pair of angularly displaced pressure balancing chambers 108 disposed on the low pressure edges of each end plate. The structural details and mode of operation of a liquid displacement device employing an oval or rounded contour internal unit assembly construction is set out in my Patent No. 2,848,953, issued August 26, 1958. It will suffice for the purposes of the description of the present invention to state that a pair of separate longitudinally extending tension members 109 are provided in the internal unit assembly 106 to resist the reaction forces from the axial pressure loading chambers. The internal unit assembly 106 is in all other respects substantially identical to the previously described embodiments and thus comprises a pair of meshed driving and driven gears (not shown) that are carried integrally on gear shafts 111 and 112, respectively, and which are rotatably journalled in bores 113 in each end plate 107. A pair of axially extending grooves 114 are formed in the journalling bores 113 to provide fluid communication between the inner side faces of the end plates 107 within the root circle of the meshed gears and the axial pressure loading chambers of the assembly.

The radial pressure balancing chamber means utilized on the low pressure or inlet side of each end plate 107 is of the direct communication type in that working fluid under pressure is directly supplied through an axial bore or passage 116 registering with the high pressure zone of the device to the common intersection of a pair of diverging passages 117 which communicate with the chambers 108.

The pressure balancing chambers 108 are similar in construction to the pressure balancing chambers 74 illustrated in FIGS. 5 and 6 in that they are of the inflatable "boot" type and have an elongated rectangular shape. As best shown in FIG. 12, each pressure balancing chamber 108 is defined by a rectangular shaped cavity or recess 119, formed in the low pressure edges of the end plates 107, the marginal edge portions of each cavity 119 having a shoulder or recess 121 therearound to which the peripheral edge portions of a complementary resilient member such as a trip 122 of rubber or the like, is joined to form the pressure balancing chamber 108 therebetween. Working fluid pressure is thus communicated through the connecting passages 117 to the pressure balancing chambers 108 to inflate the resilient members 122 into contact with the surrounding inner walls of the casing bore to urge the internal unit assembly 106 toward the discharge side of the device.

It should be understood that while an inflatable "boot" type pressure balancing chamber construction has been illustrated and described in conjunction with the end plates 107 of the internal unit assembly 106, a perimetrically upraised lip type resilient member construction could also be employed, similar to the resilient members 95 of FIG. 8, if so desired.

The operation of the pressure balancing means illustrated in FIGS. 10–12 is substantially identical to that described in the previous embodiments of my invention, particularly the modification illustrated in FIG. 8, and further description thereof is thus deemed unnecessary.

Referring now to FIGS. 13–17, the invention is illustrated in connection with another internal unit assembly and casing employing a modified "figure-8" type cross-sectional configuration and a separate hollow tension member construction. The radial pressure balancing means shown in FIGS. 13–17 differs somewhat from the previous embodiments of the invention in that the pressure balancing chambers thereof are formed in the inner walls of the casing bore, rather than in the edges of the end plates.

In my aforementioned Patent No. 2,848,953, I have described a similar internal unit assembly construction employing separate tension members which are axially bored to define passages for communicating working liquid pressure from the discharge pressure zone of the pump to the axial pressure loading chamber of the internal unit assembly, to which reference should be made for the structural details and operation of this type of internal unit assembly construction. Inasmuch as the details of the construction and operation of the separate tension member internal unit assembly construction and their relationship to the axial pressure loading chamber system form no part of the present invention, only a brief description thereof will be included.

The internal unit assembly and casing construction illustrated in FIGS. 13–17 comprises a circular casing 126 having a pair of centrally intersecting bores 127

therethrough which receive a "figure-8" type internal unit assembly 128 therein. The internal unit assembly 128 includes a meshed pair of driving and driven gears 129 and 131 that are carried on shafts 132 and 133, respectively. Each gear shaft is rotatably journalled in bores 134 in separate pairs of lower and upper end plate halves 136 and 137 adjacent the side faces of the gears 129 and 131, the end plate halves 136 and 137 being flattened along their line of engagement, as at 138, to maintain their assembled orientation in the casing bore 127. A pair of diametrically spaced axial grooves 139 are provided in each journalling bore 134 which communicate working liquid pressure from the high pressure zone of the device to the axial pressure loading chambers of the internal unit assembly for the purpose and function previously described. The casing 126 is axially bored as at 140 and 140' to receive a pair of longitudinally extending tension members 141 and 142 having axial bores 143 and 144, respectively, therethrough. The tension members 141 and 142 resist the reaction forces of the axial pressure loading chambers and the bores 143 and 144 provide a liquid pressure path to the axial pressure loading chambers and radial pressure balancing chambers of the device, as will hereinafter be described.

The end plate halves 136 and 137 differ from those described in the previous embodiments in that they are formed with an axial length such that their outer ends will extend somewhat beyond the end faces of the internal unit assembly casing 126 when their inner side faces are in sealed engagement with the side faces of the meshed gears 129 and 131. A pair of circular external thrust plates 146 (FIG. 14) are provided for engagement with the projecting ends of the end plate halves 136 and 137 and are thus maintained in axially spaced relation to the end faces of the internal unit assembly casing 126. As best shown in FIGS. 14 and 16, each external thrust plate 146 includes a pair of bores 147 which provide coaxial registry with the interiors of the hollow gear shafts 132 and 133, a pair of bores 150 and 150' coaxial with and equal in diameter to the bores 140 and 140', respectively, and a pair of axial counterbores or recesses 148 and 149 formed in the outer ends of the bores 150 and 150', the purpose and function of which shall be hereinafter more fully described. A diametrical bore or passage 151 interconnects the counterbores 148 and 149, and plug means, indicated generally at 152, seals the outer end of the passage 151 from communication with the exterior of the thrust plates 146.

In order to define a pair of axial pressure loading chambers at the drive end of the internal unit assembly 128, the tension members 141 and 142 are formed with integral disc-shaped head portions 153 and 154, respectively, which are received in close-fitting relation in the respective counterbores 148 and 149 of the plate 146 at the drive end. The axial depths of the head portions 153 and 154 are somewhat less than the depths of the respective counterbores 148 and 149 so as to define one pair of axial pressure loading chambers 156 and 157 therebetween. It should be noted that the diameter of the counterbore 148 and tension member head 153 is substantially greater than the diameter of the counterbore 149 and tension member head 154, and that the counterbore 148 and tension member head 153 are eccentric to the longitudinal axis of the tension member 141 in order that the resultant force components and moments acting on the internal unit assembly 128 be maintained in equilibrium. Another pair of pressure loading chambers 156 and 157 are defined in the external thrust plate 146 at the opposite or anti-drive end of the internal unit assembly 128 by means of a pair of annular ring-like disc members 158 and 159 mounted at the opposite ends of the tension members 141 and 142, respectively, and seated in close-fitting relation in the counterbores 148 and 149. A pair of lock nuts 160 and 161 are threaded onto the ends of the tension members 141 and 142 to maintain the various components of the internal unit assembly 128 and casing 126 in assembled relationship. Annular seal means 162 are disposed around the circumferential edges of the tension member heads 153 and 154 and the members 158 and 159 for sealing coaction with the annular side walls of the counterbores 148 and 149. Thus, the reaction forces from the axial pressure loading chambers 156 and 157 at the drive and anti-drive end of the device are transmitted to the shaft portions of the tension members 141 and 142 by the head portions 153 and 154 and annular disc members 158 and 159 and are thus resisted as tensile stress elongation of the shafts in the manner described and claimed in my aforementioned Patent No. 2,848,953.

In order to communicate working fluid at pump discharge pressure to the pressure loading chambers 156 and 157 the tension member 141 is provided with a centrally disposed radial bore 163 (FIGS. 16 and 17) which communicates with the high pressure zone of the device and registers with the axial bore 143 therethrough. A second pair of axially spaced radial passages 164 and 166 are provided in the tension member 141 and serve to communicate high pressure working liquid to the axial pressure loading chambers 156 at the drive and anti-drive ends, respectively, of the internal unit assembly 128. The passages 151 serve to direct high pressure liquid to the adjacent pressure loading chambers 157.

Referring particularly to FIGS. 15 and 16 the structural details and operation of the radial pressure balancing means for the internal unit assembly 128 will now be described. Unlike the preceding embodiments of the invention, the radial pressure balancing chambers for the internal unit assembly 128 are formed in the inner walls of the casing bore 127 at the inlet side thereof and receive high pressure working liquid through an indirect flow path that includes the working liquid flow path to the axial pressure loading chambers 156 and 157.

As best shown in FIG. 15, the casing bore 127 is formed with two pairs of cavities or recesses 167, which are preferably rectangular and circumferentially elongated in shape and have their perimetrical edges formed with a shoulder or recess 168, in the manner previously described. A pair of similarly shaped resilient strip members 169 may be bonded to the shoulders 168 to close the cavities 167 and define a pair of radial pressure balancing chambers 170 therebetween. The chambers 170 are thus of the "boot" type as were heretofore described in connection with FIG. 6.

In order to communicate working liquid pressure to the pressure balancing chambers 170, the casing 126 includes a pair of bores 171 (FIG. 15) disposed adjacent each set of end plate halves 136—137 which communicate with radial bores 172 (FIG. 16) in the tension member 142 for interconnecting each pair of pressure balancing chambers 167 with the axial bore 144 in the tension member 142. The bores 171 are blind and have their open ends closed, as by plug means 173. Pressurizing working liquid from the axial pressure loading chamber 157 at the drive end of the device is communicated by a passage 174 adjacent the head 154 of the tension member 142 to the axial bore 144 and thence to the radial pressure balancing chambers 170 by means of the path previously described. Liquid pressure in the chambers 170 is imparted through the resilient members 169 to the edges of the end plate halves 136—137 for urging the latter toward the discharge side of the device in the manner previously explained.

As best shown in FIGS. 16 and 17, the casing 126 is formed with a pair of diametrically disposed recesses 176 and 177 in its circumferential periphery around the inlet and discharge ports thereof, respectively, to which are bonded a complementary pair of channeled, circumferen-

tially elongated, generally rectangular resilient members 178 having planar base portions 179 and radially outwardly extending wall portions 181 which form liners for the recesses 176 and 177. The wall portions 181 have marginal lips 182 for sealingly engaging the inner walls of an enclosing housing or bore (not shown) in which the internal unit assembly casing 126 may be inserted, e.g. in a so-called buried installation. The planar base portions 179 are formed with openings 183 to permit liquid flow through the inlet and discharge ports of the casing 126.

Referring now to FIGS. 18–25 of the drawings, another modification of the invention is illustrated which differs from the previously described embodiments in that it employs an "inner tube" type resilient member having a pair of radial pressure balancing chambers formed integrally therein and disposed at opposite edges of each end plate of the internal unit assembly. A system of check valves is also provided to permit selective communication of high pressure working liquid to a particular pair of pressure balancing chambers so that the device can operate in either direction without structural rearrangement of the various components. This particular modification of the invention also employs a novel intermediate thrust plate at the anti-drive end of the internal unit assembly to provide an auxiliary axial pressure loading chamber for augmenting the counterbalancing force of the axial pressure loading chambers adjacent the ends of the gear shafts and outer side faces of the end plates.

As seen in FIGS. 19, 22 and 23, the embodiment of the invention illustrated therein comprises a cylindrical internal unit assembly casing 186 having an elongated bore 187 therethrough which removably receives an internal unit assembly 188 therein. A pair of external thrust plates 189 are disposed adjacent the outer ends of the internal unit assembly 188 and an intermediate thrust plate 191 is interposed between the internal unit assembly and the external thrust plate 189 at the anti-drive end of the unit. Four separate tension members 192 are utilized to resist the reaction forces from the axial pressure loading chambers, the tension members being radially symmetrically arranged and extending longitudinally or axially through the internal unit assembly casing 186. The tension members 192 function in a manner similar to the tension members in the previously described embodiment of the invention to accommodate the reaction forces from axial pressure loading chambers disposed around the inner faces of the end plates of the internal unit assembly and from the auxiliary pressure loading chamber between the intermediate and external thrust plates at the anti-drive end of the unit, as will hereinafter be more fully described.

As best shown in FIG. 22, the internal unit assembly 188 comprises a drive gear 193 and a meshed driven gear 194, each carried integrally on respective hollow gear shafts 196 and 197, the driving gear shaft 196 being centrally internally splined, as at 198, to facilitate the application of driving torque thereto. The gears 193 and 194 are enclosed by the internal unit assembly casing 186 to define a pair of diametrically arranged ports 199 and 201 (FIGS. 19 and 21) for respectively supplying and receiving working liquid. A pair of elongated end plates 202 (FIG. 23) are disposed adjacent and engaging the opposite side faces of the meshed gears, each end plate having a lower bore 203 and an upper bore 204 therein constituting bearing journals for the ends of the gear shafts 196 and 197. The outer side faces of the end plates 202 are formed with a pair of annular axially outwardly extending bosses 206 (FIG. 22) which cooperate with a complemental pair of coaxial counterbores 207 and 208 adjacent the ends of the lower and upper gear shafts 196 and 197, the counterbores being formed in the opposing faces of the adjoining external thrust plate 189 at the drive end of the unit and the intermediate thrust plate 191 at the anti-drive end of the unit. At the driven gear shaft 197 the axial ends of the bosses 206 cooperate with the counterbores 208 to define a pair of annular pressure loading chambers 212 for urging the end plates 202 inwardly toward the side faces of the gears 193 and 194. At the driving gear shaft 196 a corresponding pair of pressure loading chambers 211 are defined between the bosses 206, a pair of annular seal rings 209, and a pair of resilient support rings 210. The elements 209 and 210 cooperate to provide a compensating seal arrangement more fully described and claimed in my copending application Serial No. 709,045, filed January 15, 1958, now Patent No. 2,967,487, which should be referred to for a more complete explanation of this aspect of the structure. A passage 213 in the thrust plate 189 at the drive end of the internal unit assembly and a passage 214 (FIGS. 24 and 25) in the intermediate thrust plate 191 at the anti-drive end of the internal unit assembly provide liquid communication between the pressure loading chambers 211 and 212. High pressure working liquid is supplied to the axial pressure loading chambers of the device by passages (not shown) in the end plates 202. A pair of diametrically spaced axially extending grooves 215 (FIG. 23) are provided in the gear shaft journalling bores 204 and connect the pressure loading chambers 211 and 212 to the high pressure zone of the device and contribute to the lubrication of the journalling bores in addition to the leakage flow therethrough, as previously described.

The auxiliary axial pressure loading chamber at the anti-drive end of the internal unit assembly 188 is indicated at 216 (FIGS. 19 and 22) and is formed between the spaced faces of a central counterbore 217 in the outer face of the intermediate thrust plate 191 and an axially inwardly extending boss portion 218 on the inner face of the adjacent thrust plate 189. The boss portion 218 includes an annular recess 219 in which is seated a coil spring 221 that bottoms in the recess 219 and engages the base of the counterbore 217 to urge the intermediate thrust plate 191 into pressure sealed engagement with the adjacent end plate 202. The intermediate thrust plate 191 and adjacent end plate 202 are centrally axially bored, as at 222, to loosely receive a tubular bushing 223 which serves to maintain axial alignment of these members and to conduct pressurized liquid from the high pressure side of the device to the auxiliary pressure loading chamber 216.

In order to provide selective liquid communication between the bore 222 and the high pressure zone of the device, the end plate 202 at the anti-drive end of the device has a pair of diametrically spaced axial bores 224 (FIG. 19) having reduced diameter metering passages 226 at their inner ends which are in registry with the port passages 199 and 201 of the internal unit assembly 188. Spring biased ball check valves 227 are provided to normally close the metering passages 226 to provide selective unidirectional flow therethrough depending on the direction of rotation of the gears. A pair of angular passages 228 connect the bores 224 with the bore 222 so that working liquid at pump discharge pressure is communicated around and through the tubular loosely fitting bushing 223 to the auxiliary pressure loading chamber 216. Annular seal means 229 (FIG. 22) is provided around the boss 218 to prevent leakage from the auxiliary pressure loading chamber 216 between the intermediate thrust plate 191 and its external thrust plate 189, and a pair of spaced annular seal means 230 are provided around the periphery of the tubular bushing 223 to prevent leakage of pressurized working liquid between the anti-drive end plate 202 and the intermediate thrust plate 191.

Thus it will be appreciated that the axial pressure loading system employed in the internal unit assembly 188 is effective to counterbalance the pressure forces tending to separate the end plates 202 from the side faces of the gears 193 and 194, even at extremely high pressure operation, due to the inclusion of the auxiliary axial pressure loading chamber 216. Moreover, the device is read-

ily reversible without rearrangement of the structural components of the device, through the provision of the spring biased check valves 227 that insure unidirectional communication of liquid pressure to the auxiliary pressure loading chamber 216. As described in my prior United States Patent No. 2,772,638, pressurized working liquid from the axial pressure loading chambers ultimately finds its way to the hollow interiors of the gear shafts 196 and 197, to be returned to the low pressure flow of working liquids entering the gears at the pump inlet.

In order to regulate the flow of working liquid from the hollow interiors of the gear shafts to the pump inlet and thus control the flow of liquid through the bearings and bearing surfaces of the pump, the external thrust plate 189 at the drive end of the pump is provided with an adjustable bleed means in the form of an adjustable needle valve 231 (FIG. 22) threadedly mounted in an opening 231' in the thrust plate 189 and having a bore 232 therethrough and a tapered inner end 233. The tapered end 233 of the valve member 231 coacts with a valve seat 234 in the thrust plate 189, which communicates with the hollow interior of the gear shaft 197, to control the flow through the bore 232. A lock-nut 235 maintains the desired setting of the needle valve 231 for a given operating condition. The adjustable bleed means is necessitated due to the high seal efficiencies around the pressure loading chambers 211, 212 and 216.

Referring now more particularly to FIGS. 19, 22 and 23, the radial pressure balancing means for maintaining the internal unit assembly 188 substantially centralized as a free body in its casing 186 will now be described. As seen in FIGS. 19 and 22, each end plate 202 is formed with a peripheral edge groove 236 in which is seated a surrounding one-piece "inner-tube" type resilient member 237 having elongated, generally rectangular, confined cavities or pockets 238 formed therein adjacent the edges of the end plates 202. For providing liquid pressure communication with the high pressure zone of the device, each end plate 202 is formed with a pair of short axial passages 239 (FIG. 19) registering with the ports 199 and 201 adjacent the meshed gears 193 and 194 and extending axially outwardly therefrom. Intersecting diagonal passages 241 (FIGS. 19, 22 and 23) in the end plates 202 connect the cavities 238 with the passages 239. Each passage 241 is counterbored at its outer end in the edges of the end plates 202 to receive a tubular stem portion 242 formed on the inner side face of the resilient member 237 thereby providing a liquid pressure sealed passage for directing pressurized working liquid to the interior of the cavities 238. It will be apparent from the location of the axially extending passages 239 that one pair of pressure balancing cavities 238 at one side of the device will be subjected to pump discharge pressure via one pair of connecting passages 241 for a given direction of rotation of the gears 193 and 194, and an opposite pair of pressure balancing cavities 238 will be so pressurized on a reversal of the direction of rotation of the gears 193 and 194. As previously described, the size, shape and location of the cavities 238 is such as to effectively counterbalance the radial component of the displacing force at the high pressure side of the pump either with a balanced or slightly unbalanced force, thereby to insure minimum leakage at the high pressure side of the casing and to maintain a more favorable high pressure distribution around the gears.

In order to prevent excessive pressure between the internal unit assembly casing 186 and its enclosing housing (not shown), the drive end plate 202 includes a pair of axial bores 243 (FIG. 19), each of which has a reduced diameter outlet passage 244 formed in its outer end and which is normally closed by a spring biased ball check valve 246. The drive end plate 189 includes a pair of registering bores 247 in alignment with the passages 244. A liquid pressure passage is thus defined from the low pressure zone of the pump to a reservoir or chamber formed between the internal unit assembly casing 186 and its enclosing housing (not shown), such that the latter chamber may be vented to the low pressure zone of the pump whenever excessive pressure occurs. Communication between the reservoir at the drive and anti-drive ends of the internal unit assembly casing 186 and its enclosing housing is provided by axial bores 248 (FIGS. 18, 21, 23 and 24) extending through the tension members 192.

In order to prevent leakage around the ports 199 and 201 of the internal unit assembly casing 186, a pair of resilient seal assemblies 249 (FIGS. 19, 20 and 21) are provided which are seated in appropriately formed recesses 251 in the sides of the internal unit assembly casing 186, the seal assemblies 249 being of the same type as the seals 178 illustrated and described in connection with the FIGS. 13–17 embodiment previously described.

While only certain specific embodiments of my invention have been herein illustrated and described, it should be understood that modifications and variations thereof may be effected without departing from the scope of the novel concepts herein disclosed, as set forth in the appended claims.

I claim:

1. In a liquid displacement device of the character described, the combination of a casing having a bore therethrough, and an internal unit assembly in said casing bore comprising support means, at least one liquid displacement element rotatably supported by said support means, said element being operable to define separate zones of low and high pressure working liquid in said casing, said high pressure zone providing a radial force tending to urge said internal unit assembly toward the low pressure side of said casing, and means for opposing said radial force and defining at least one chamber between the support means supporting said element and said casing at the low pressure side of the latter, said opposing means including a passage providing fluid communication between said chamber and said high pressure zone so that high pressure working liquid in said chamber provides a force opposing said radial force whereby to maintain said internal unit assembly in a predetermined position in said casing bore, said chamber having means for confining the high pressure fluid therein.

2. In a liquid displacement device of the character described, the combination of a casing having a bore therethrough, and an internal unit assembly in said casing bore including a pair of meshed gears, a pair of spaced support means disposed in sealing relation at the faces of said gears and rotatably supporting the latter, said gears being operable to define separate zones of low and high pressure working liquid at said gears, and radial pressure balancing means for opposing the force of said high pressure zone tending to urge said internal unit assembly toward the low pressure side of said casing bore, said radial pressure balancing means having at least one chamber disposed between each of said pair of support means and said casing at the low pressure side of the latter and having a flexible wall portion, said balancing means also having a passage connecting said high pressure zone with said chamber to provide communication therebetween so that high pressure working liquid in said chamber causes said wall portion to oppose said force whereby to maintain said internal unit assembly in a predetermined position in said casing bore, said chamber having means for confining the high pressure liquid therein.

3. The combination according to claim 2, in which said chamber is so constructed and arranged as to provide a resultant counterbalancing force substantially equal and opposite to said force at the high pressure zone whereby said internal unit assembly is substantially centralized in said casing bore.

4. In a liquid displacement device of the character described, the combination of a casing having an inlet and an outlet and a bore therethrough, and an internal unit

assembly in said bore comprising a pair of meshed gears between said inlet and said outlet, a pair of spaced end plates rotatably supporting said gears and being disposed in sealing relation at opposite faces of said gears, said gears being operable to define zones of low and high pressure working liquid adjacent said inlet and said outlet, respectively, and radial pressure balancing means for opposing the radial component of the discharge pressure force of said high pressure zone, said radial pressure balancing means comprising flexible means defining a plurality of chambers between the peripheral edges of said end plates and said casing at the inlet side of the latter, said flexible means being constructed and arranged to confine high pressure working liquid in said chambers, said balancing means including passages interconnecting said chambers with said high pressure zone whereby high pressure working liquid in said chambers opposes said radial component to maintain said internal unit assembly in a predetermined position in said casing bore.

5. In a liquid displacement device of the character described, the combination of a casing member having an inlet and an outlet and a bore therethrough, and an internal unit assembly in said casing bore comprising a pair of meshed gears between said inlet and said outlet, a pair of spaced end plate members rotatably supporting said gears and being disposed in sealing relation at opposite faces of said gears, said gears being operable to define zones of low and high pressure working liquid adjacent said inlet and said outlet, respectively, and radial pressure balancing means for opposing the radial component of the high pressure force in said high pressure zone, said radial pressure balancing means comprising a resilient seal interposed between and engaging the peripheral surfaces of said end plate and casing members to define with at least one of said peripheral surfaces chambers for confining high pressure working liquid, said balancing means including passages interconnecting said chambers with said high pressure zone whereby to maintain said internal unit assembly in a predetermined position in said casing bore.

6. The combination according to claim 5, in which said chambers are surrounded by annular recesses formed in the low pressure side of one of said members, and annular resilient seals are seated in said recesses and have portions thereof engaging the adjacent wall of the other of said members in pressure sealed relationhip.

7. The combination according to claim 5, in which said chambers are defined by flattened portions on the peripheral surfaces of said end plates and segmentally shaped resilient seals carried on said portions, said resilient seals having a lip therearound for engaging said wall in pressure sealed relationship, said lip including arcuately shaped portions to conform to the adjacent wall of said casing bore.

8. In a liquid displacement device of the character described, the combination of a casing having an inlet and an outlet and a bore therethrough, a pair of meshed gears in said casing between said inlet and said outlet, a pair of spaced end plates rotatably supporting said gears and being disposed in sealing relation at opposite faces of said gears, said gears being operable to define zones of low and high pressure working liquid adjacent said inlet and said outlet, respectively, and radial pressure balancing means for opposing the radial component of the high pressure force in said high pressure zone, said radial pressure balancing means comprising at least one elongated resilient member peripherally bonded to the edge of each end plate at the inlet side thereof to define a chamber therebetween, said balancing means including passages interconnecting said chambers with said high pressure zone, said resilient member being expansible in response to confined high pressure working liquid communicated to said chamber to engage the adjacent wall of said casing bore, whereby high pressure working liquid in said chambers provides a force opposing said radial component to maintain the gears and end plates in a predetermined position in said casing bore.

9. The combination according to claim 8, in which said resilient member comprises a single elongated strip of resilient material extending substantially across the full edge of each of said end plates.

10. The combination according to claim 8, in which each of said end plates has a pair of circumferentially spaced elongated strips of resilient material on the edges thereof.

11. In a liquid displacement device of the character described, the combination of a casing having an inlet and an outlet and a bore therethrough, a pair of meshed gears in said casing between said inlet and said outlet, a pair of spaced end plates rotatably supporting said gears and disposed in sealing relation at opposite faces of said gears, said casing bore having at least one cavity formed in the wall of said bore adjacent the edges of said end plates on the inlet side thereof, said gears being operable to define zones of low and high pressure working liquid adjacent said inlet and said outlet, respectively, and radial pressure balancing means for counterbalancing the radial component of the high pressure force in said high pressure zone, said radial pressure balancing means comprising resilient means defining at least one chamber between the peripheral edge of each end plate and said casing at the inlet side of the latter, said resilient means being an elongated strip of resilient material peripherally bonded around the edge of its associated cavity, said balancing means including passages interconnecting said chambers with said high pressure zone whereby high pressure working liquid in said chambers provides a force opposing said radial component to maintain the gears and end plates in a predetermined position in said casing bore.

12. The combination according to claim 11, in which the wall of said bore has a pair of cavities for each end plate, and elongated strips of resilient material are peripherally bonded around the edges of each cavity.

13. In a liquid displacement device of the character described, the combination of a casing having a reversible inlet and outlet and a bore therethrough, a pair of meshed gears in said casing between said reversible inlet and outlet, a pair of spaced end plates rotatably supporting said gears and being disposed in sealing relation at opposite faces of said gears, each of said end plates having a shallow recess formed in each of the side edges thereof, said gears being operable to define zones of low and high pressure working liquid adjacent said reversible inlet and outlet dependent upon the direction of rotation thereof, and radial pressure balancing means for opposing the radial component of the high pressure force in said high pressure zone, said radial pressure balancing means comprising resilient means defining a pair of chambers between the respective side edges of each end plate and said casing at the inlet and outlet sides, said resilient means being a strip of resilient material seated in each of said recesses and having pockets formed therein at the edges of said end plates, said balancing means including passages selectively interconnecting the pockets opposite a high pressure zone with high pressure liquid, depending upon the direction of rotation of said gears, whereby high pressure working liquid in said pockets provides a force opposing said radial component to maintain the gears and end plates in a predetermined position in said casing bore.

14. The combination according to claim 13, in which each of said end plates has a shallow recess extending around the periphery of the end plate, and said resilient means is seated in said recess and extends around the end plate.

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