

Aug. 18, 1942.

L. FERSING

2,293,126

GEAR PUMP OR MOTOR

Filed April 24, 1939

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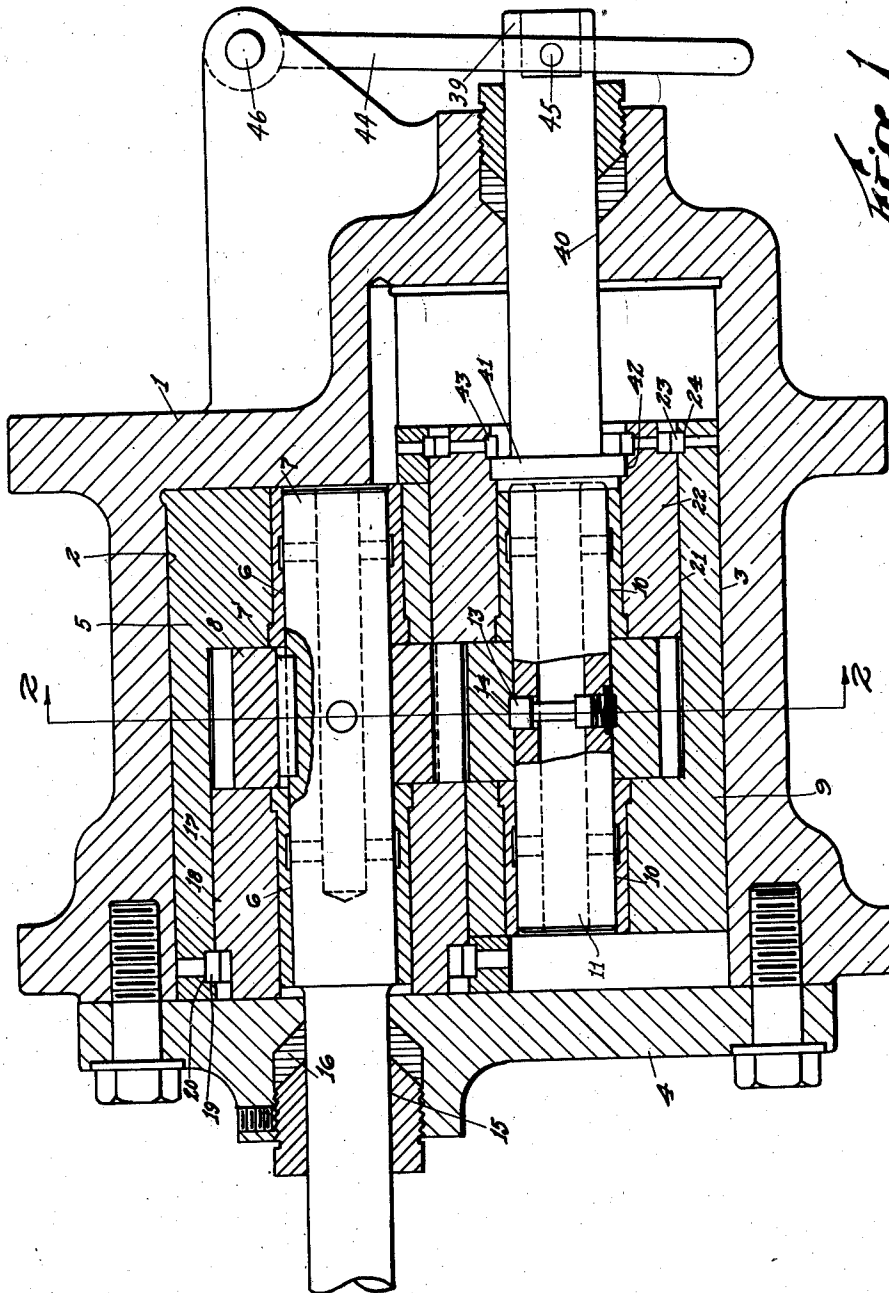


Fig. 1.

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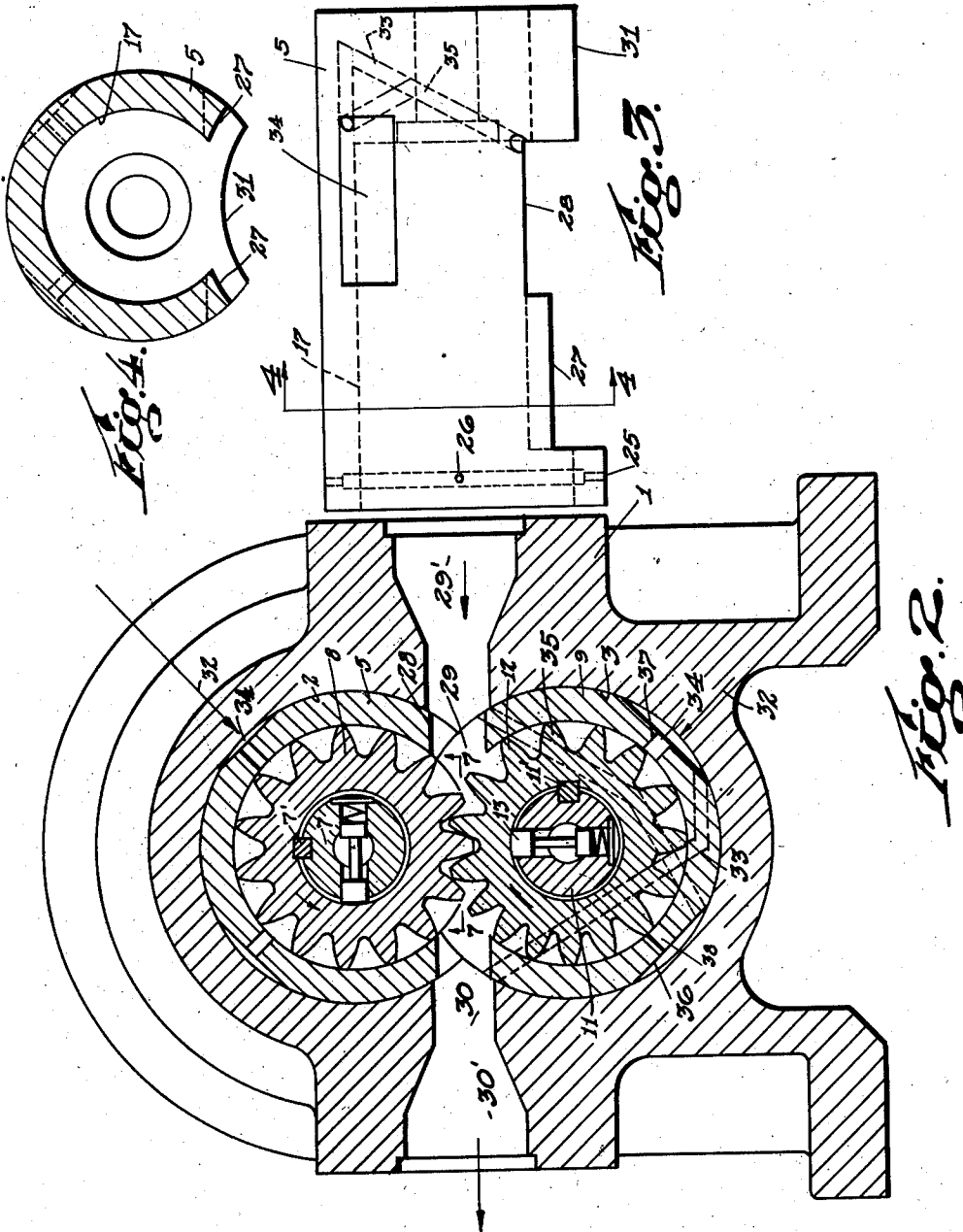
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4 Sheets-Sheet 2



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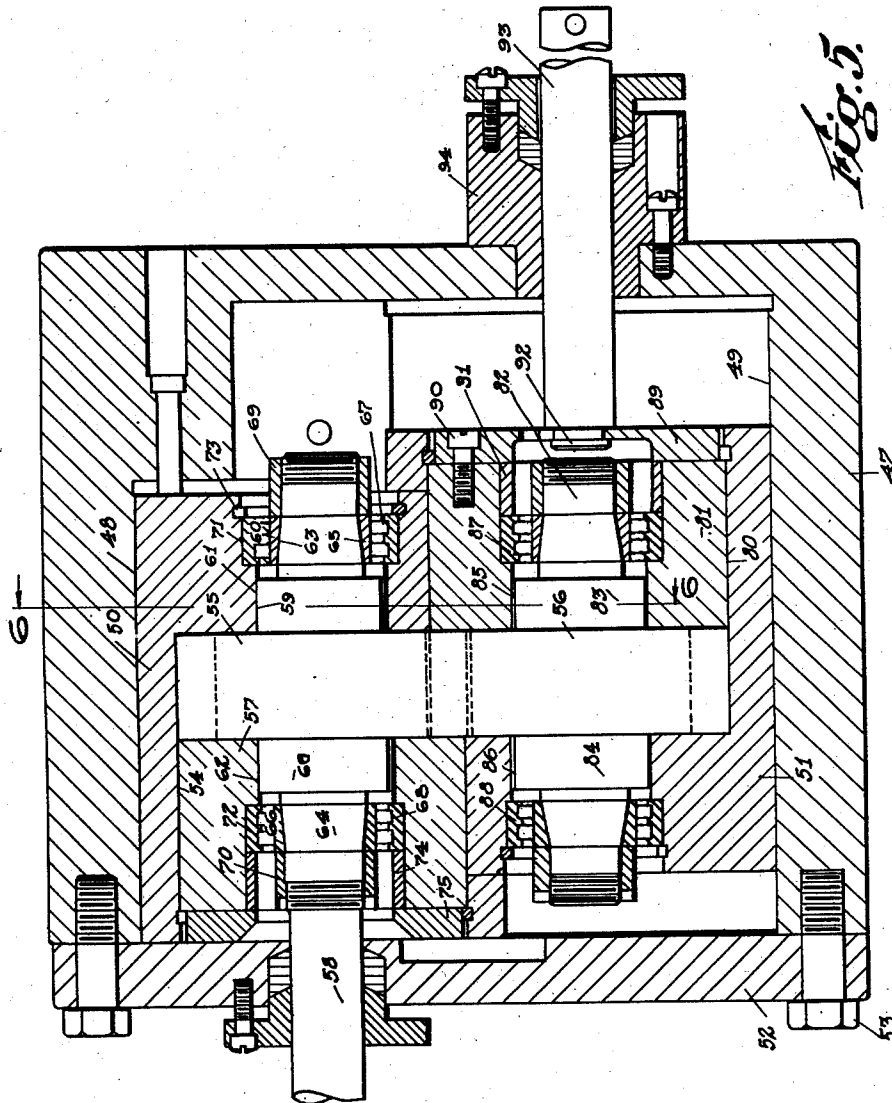
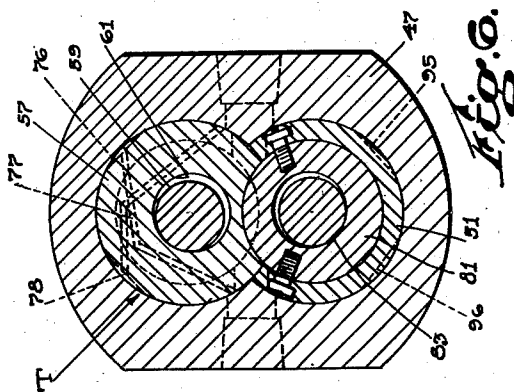
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4 Sheets-Sheet 3



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Fig. 7.

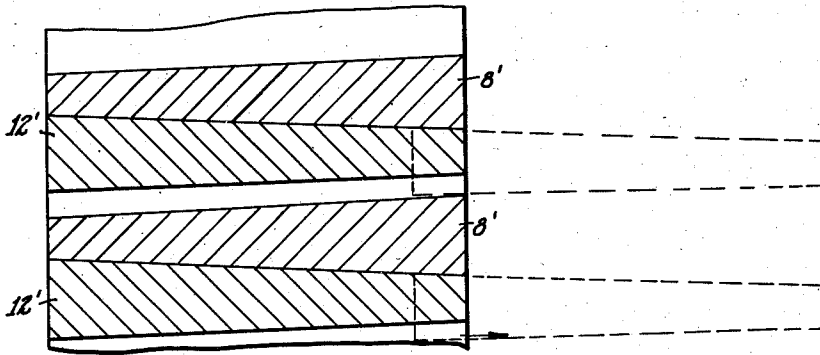


Fig. 8.

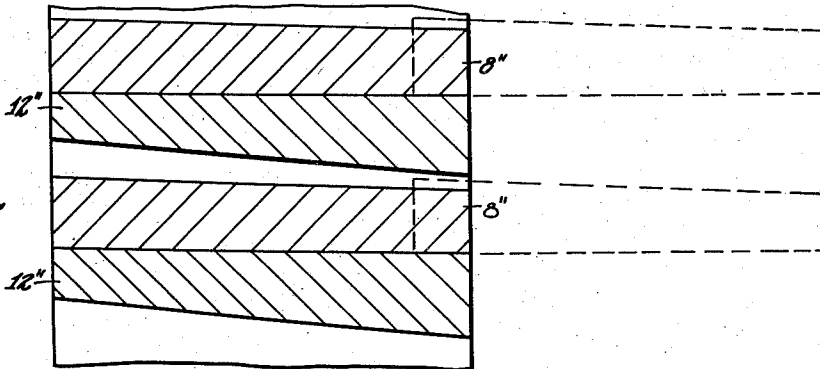
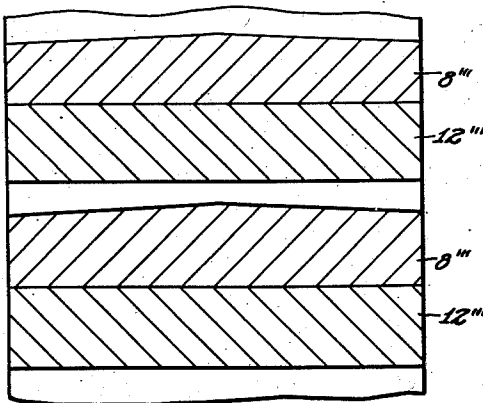


Fig. 9.



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UNITED STATES PATENT OFFICE

2,293,126

GEAR PUMP OR MOTOR

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Application April 24, 1939, Serial No. 269,659

8 Claims. (Cl. 103—120)

The present invention relates to gear pumps or motors, and particularly to a pump adapted for having a variable discharge when operated at a constant speed, or to a variable speed motor when driven by a constant volume of fluid.

In prior devices of this character, gear pumps have been utilized in which there is a provision for a relative lateral movement between the gears of the pump in order to vary the volume discharge by the pump. Such a device is shown, for example, in the Fraser Patent No. 815,522, in which one of the two pump elements is moved laterally relative to the other, thereby varying the volume of the discharge. This structure is satisfactory where only small pressures are developed, but where the pump operates under a relative high pressure differential the radial thrust exerted by the fluid pressure makes it very difficult to maintain alinement between the gears and to adjust the pump while in use. One of the objects of the present invention is to devise a structure by which the lateral shifting movement between the gears may take place independently of the pressure differential under which the pump operates and with a minimum of frictional resistance.

In either constant or variable volume gear pumps the fluid being pumped has a tendency to be locked at the bottoms of the grooves between adjacent gear teeth, and this locking or pocketing of fluid may cause the pump to vibrate. A further feature of the present invention resides in varying the amount of back lash between adjacent gear teeth from one end of the tooth to the other so that the fluid being pumped may be more readily moved toward relief pockets provided at the sides of the gears.

This same tapering of the gear teeth by which the pocketing of oil is avoided may be utilized in increasing the efficiency of a variable volume gear pump. In accordance with this feature of the invention, the gears are slightly tapered in such a manner that the amount of back lash between the gear teeth is increased as the length of gear teeth in mesh is increased. Thus, when the gear teeth are only slightly in mesh, the back lash is extremely small and leakage of fluid laterally of the gear teeth is prevented. When the gears are substantially entirely in mesh, the axial leakage of oil is negligible and the back lash may be substantially greater in order to prevent pocketing of the oil at the bottoms of the spaces between adjacent gear teeth.

In that type of gear pump in which there is a provision for relative lateral movement between

the gears of the pump, the movable gear is generally mounted in alinement with a sleeve having a notch in which the gear teeth of the opposite gear are received when the movable gear is shifted endwise to decrease the operative length of the gear teeth. The sleeve is effective to seal the tops of the gear teeth over a relatively small arc. In accordance with the present invention, the movable gear is mounted in a sliding housing having a diameter substantially larger than that of the gear, and this housing is cut away to receive the teeth of the opposing gear. This larger diameter housing provides for effectively sealing the tops of the teeth of the stationary gear over a much larger arc than in the usual type of pump, thereby producing a more efficient pump. By this larger sealing arc it is also possible to utilize helical gears as this larger arc will obviously cover more gear teeth at one time than the smaller arc of the conventional gear pump.

Other and further objects and advantages of the invention will appear from the following detailed description taken in connection with the accompanying drawings in which—

Fig. 1 is a sectional view through a pump or motor embodying the invention.

Fig. 2 is a sectional view along the line 2—2 of Fig. 1.

Fig. 3 is a side elevation of the gear housing in which the slidable gear is mounted.

Fig. 4 is a sectional view along the line 4—4 of Fig. 3.

Fig. 5 is a sectional view similar to Fig. 1, showing a modification.

Fig. 6 is a sectional view along the line 6—6 of Fig. 5.

Fig. 7 is a sectional view substantially along the line 7—7 of Fig. 2.

Fig. 8 is a sectional view corresponding to Fig. 7 and showing a modification.

Fig. 9 is a sectional view corresponding to Fig. 7, and showing a further modification.

With reference first to Fig. 1, the housing 1 in which the pump is positioned has parallelly extending intersecting bores 2 and 3, the open ends of which are closed by a cap 4. In the bore 2 is positioned a gear housing 5 having bearings 6 in which a shaft 7 is journaled. A gear 8 is secured, as by a key 7', to the shaft 7 for rotation therewith. The outer end of the shaft extends through the cap 4 to provide a driving connection for the pump or motor.

In the bore 3 is positioned a gear housing 9 having bearing surfaces 10 in which a shaft 11 is journaled. A gear 12 is secured to the shaft

11 for rotation therewith by a key 11', Fig. 2. Axial sliding of the gear on the shaft is prevented by a spring actuated plug 13 axially slidable in the shaft and engaging a groove 14 in the inner surface of the gear. The gears 8 and 12 are preferably helical gears, but they may be conventional spur gears, or they may be more nearly the type of gearing utilized in the pumps shown in the Fraser patent above referred to. In the construction shown, the shaft 7 is the drive shaft for the pump; if the device is utilized as a motor this shaft becomes the driven shaft. In either event the shaft extends through an opening 15 in the cap 4, leakage being prevented by suitable packing 16 surrounding the shaft.

For assembly of the shaft 7 and gear 8 in the housing 5, the latter has an axial bore 17 of a diameter equal to the outside diameter of the gear, this bore extending inwardly from one end of the housing and being concentric to the outside surface of the housing. After the gear 8 and shaft 7 have been positioned within the housing, a sleeve 18, which becomes a functionally integral part of the housing 5, is placed in the bore 17 and held therein by a snap ring 19 engaging in the groove 20 in the housing 5. One of the bearings 6 for the shaft 7 is located, as will be apparent, in the sleeve 18.

The housing 9, shaft 11, and gear 12 are assembled in a similar manner. The housing 9 has a bore 21, the diameter of which is equal to the outside diameter of the gear 12, and a sleeve 22 is positioned in the bore 21 after the gear and shaft have been positioned within the housing. The sleeve forms a functionally integral part of the housing, being held in position by a snap ring 23 engaging a groove 24 in the housing. One of the bearing surfaces 10 is provided by the sleeve 22, as shown.

The gear housings 5 and 9 are similar in construction and a description of one will be sufficient. The housing 5 is shown in detail in Figs. 3 and 4, and with reference to these figures, it will appear that the housing is in general cylindrical and corresponds in diameter to the diameter of the bore 2. Since the gears 8 and 12 must be in mesh in order that the pump may function, both housings 5 and 9 must be correspondingly cut away in order that they may fit, one with the other.

As shown in Figs. 3 and 4, the housing 5 has a short section 25 at one end which is entirely cylindrical, and this section has a plurality of radial holes 26 which provide for the removal of the snap ring 19 from the groove 20. Adjacent to the portion 25, the housing 5 has an arcuate slot 27, the center of curvature of which coincides with the center of the shaft 11 which is journaled in the other housing, as will be apparent. The radius of curvature of this arcuate surface is equal to the outside radius of either of the housings 5 or 9, and thus this slot intersects with the bore 17.

The arcuate surface 27 extends from the full cylindrical portions 25 to the portion of the housing in which the gear 8 is positioned. Laterally of the portion where the gear is located, the housing 5 is horizontally cut away, as at 28, so that this surface defines, as shown in Fig. 2, the upper wall of the intake and discharge passages 29 and 30 for the pump substantially in line with the passages 29' and 30' in the main pump housing. On the same side of the housing as the arcuate cut away 27 and on the opposite side of the gear, the housing has an arc-

ate notch 31 therein, the center of curvature of which coincides with the axis of the shaft 11 when the pump is assembled and the radius of which corresponds to the radius of either of the bores 17 or 21. The arcuate surface 31 extends from the portion where the gear is located to the end of the housing 5.

When the device operates as a pump, the fluid moving in the direction of the arrows, Fig. 2, the housings 5 and 9 are subjected to a load which tends to force them toward the intake side of the pump, with the pressure concentrated approximately at areas indicated generally by the arrows 32, Fig. 2. To prevent the lateral pressure exerted on the housings from interfering with the sliding movement of the housing 9, a channel 33 is drilled in the housing 9 to provide an oil passage from the discharge opening 30' to a slot 34 formed on the outside of the housing adjacent the point of maximum load on the housing. The slot 34, which as shown in the drawings, is a flat surface, in effect, provides a recessed area on the outer surface of the housing 5 which is spaced from the surface of the bore 2; the entire periphery of this area is in contact with the surface of the bore, as will be apparent. Lubricant under pressure thus passes through the channel 33 to help to relieve the frictional drag between the housing and the main housing 1. Furthermore, a channel 35 in the housing 9 provides a fluid connection from the intake side of the pump to a slot 36 on the opposite side of the housing from the slot 34. This slot and channel provide for relieving any built-up pressures between the gear housing and the main housing. The slot or flat portion 36 constitutes a recessed area which is spaced from the surface of the bore in which the housing 5 is positioned and the entire periphery of this slot or area is in engagement with the surface of the bore, as will be apparent.

Not only is there a load on the gear housings at a point indicated by the arrows, but there is also a tendency for the gear, when pumping, to be forced toward the intake side, with the greatest pressure being substantially in line with the arrows 32. To overcome the friction drag resulting from this pressure the gear housing 9 has a channel 37 extending between the slot 34 and the bore in which the gear is positioned. A similar channel 38 extends radially inward from the slot 36.

In assembling the gear housings and associated structures to form a completed pump, the housings, with the gear positioned therein, are placed so that the gears, where they project from the housing, are in mesh with each other with the cut-away 31 of one of the housings engaging with the sleeve 22 of the other housing, and with the arcuate surfaces 27 engaging with the outside of the opposite gear housing adjacent the portion 31. This arrangement is best shown in Fig. 2.

To control the rate of discharge of the pump when operated at a predetermined rate of speed, one of the gear housings 5 or 9 is axially slidable in the main housing 1 to vary the effective tooth area of the gear. As shown in the drawings, the housing 9 is axially slidable, and to this end a stub shaft 39 extends through a bore 40 in the main housing in line with the gear shaft 11. The stub shaft has a head 41 positioned in a recess 42 in the end of the sleeve 22, being held against removal therefrom, as by a clamping ring 43. Movement of the shaft 39 axially,

as by means of a lever 44 secured by a pin 45 to the end of the stub shaft and by a pin 46 to a part of the main housing 1, will cause a corresponding movement of the gear housing 9 relative to the gear housing 5 and will shorten the length of each tooth which is in contact with the teeth of the opposite gear. This sliding movement of the gear housing 9 is possible since this housing has, as above stated, a cut-away portion 31 which leaves a surface in line with the outer surface of the gear 8 in the opposite housing. Obviously, the bore 3 is preferably enough longer than the housing 9 to permit a relative movement between the housing sufficient for entirely disengaging the teeth of the cooperating gear.

The output of the pump is proportional to the length of contact of the teeth of the cooperating gears and the output is accordingly dependent upon the position of the actuating lever 44. If the output of the pump is to be varied from a maximum, with the gear housings in the relative positions shown, to a minimum in which the gear teeth are entirely out of interengagement, it may be desirable to provide means for assuring a continued rotation of both gears 8 and 9 in unison when out of engagement, so that they may again be brought into mesh. Such a structure is well known and is shown, for example, in the Fraser Patent No. 815,522, above referred to.

The balancing of the lateral pressures on the gear housings by the conduits extending from the discharge side of the pump to the point on the outside of the housing where the greatest pressure occurs, and also from the suction side of the pump to a pump where the least lateral pressure occurs on the gear housing, makes it easy to adjust the pump output manually, independently of the pressure differential between the intake and discharge sides of the pump.

The same arrangement is equally applicable when the device is used as a motor rather than a pump, since the balancing effect of the conduits is the same. Obviously, when the device operates as a motor, the maximum pressure is on the intake and the minimum is on the discharge side, so that the location of the maximum and minimum lateral pressures exerted by the gear housings is reversed.

With reference to Figs. 5 and 6, the device shown is intended for use only as a pump, although it will be apparent that the same structure can be so arranged that it can be utilized as a motor. As shown, a main housing 47 has parallel intercommunicating bores 48 and 49 which receive gear housings 50 and 51 respectively. A cap 52 is secured to the open end of the housing 47, as by bolts 53.

The gear housings 50 and 51 are, in general, similar to the housings 5 and 9 above described. In this embodiment of the invention, however, the radial thrust on the bearings is taken by a combination of both anti-friction bearings and plane bearings. Roller bearings are entirely satisfactory for predetermined loads and not too excessive rates of rotation, but before such loads and rates of rotation, it has been found desirable to utilize plane bearings to prevent overloading and thereby destruction of the anti-friction bearings.

As shown in Fig. 5, the housing 50 has a bore 54 of a diameter to receive one of the cooperating gears 55 and 56, and a sleeve 57 positioned in the bore 54 engages the side of the gear and holds

the assembled gear and shaft 58 on which the gear is mounted entirely in position within the gear housing 50.

The gear 56 is suitably secured to the shaft 58 for rotation therewith and the shaft is provided on opposite sides of the gear with plane bearing surfaces 59 and 60 engageable respectively with cylindrical bearing surfaces 61 and 62 in the housing 50 and sleeve 57 respectively. Adjacent to each of the bearing surfaces 59 and 60 the shaft 58 has tapered portions 63 and 64 on which are positioned the inner races 65 and 66 of anti-friction bearings 67 and 68. The inner races are clamped against movement on the shaft by threaded clamping rings 69 and 70, and the outer races 71 and 72 are held against movement in the gear housing 50 and sleeve 57 respectively by the split ring 73 and a collar 74, the latter being held in position by a secondary cap 75 held against the end of the sleeve 57 by suitable bolts, not shown. Tightening of the rings 69 and 70 will cause the inner races 65 and 66 to be expanded on the tapered portions of the shaft to take out practically any looseness in the bearing.

The surfaces 61 and 62 which cooperate with the plane bearing surfaces 59 and 60 on the shaft are slightly eccentric to the axis of the shaft, as shown in Fig. 6, in which figure the disclosure is greatly exaggerated. The corresponding diameters of the surfaces 60 and 62 or the surfaces 59 and 61 are such that there will be a clearance of approximately three thousandths of an inch between the surfaces along a center line connecting the gears, and at a point adjacent to the opposite gear; and on the side of the shaft away from the opposite gear the clearance will be only about two ten-thousandths of an inch. Each of the surfaces is truly cylindrical and obviously the center of the surfaces 61 and 62 must be slightly offset from the axis of the surfaces 59 and 60, which is the axis of the shaft, by a very small amount, this offsetting being toward the other gear 56.

The housing 50 has a flat portion 76 on the outside thereof corresponding to the portion 36 in Fig. 2, and this flat portion is connected by a channel 77 in the housing to provide a fluid connection between the inlet port and the flattened portion. A similar flat portion 78 corresponds to the flattened 34 of Fig. 2 and is connected by a channel 79 to the discharge side of the pump.

The gear housing 51, which is, in general, similar to the housing 9, has a bore 80 corresponding almost exactly to the dimension of the gear 56, and a sleeve 81 is positioned within said bore and engages one side of the gear. The gear is mounted on a shaft 82 which is journaled in the housing 51 and sleeve 81. On each side of the gear the shaft has cylindrical bearing surfaces 83 and 84 corresponding to the bearing surfaces 59 and 60 on the shaft 58. These bearing surfaces engage corresponding cylindrical surfaces 85 and 86 in the sleeve 81 and housing 51 respectively. The diameter of the surfaces 85 and 86 is slightly larger than the surfaces 83 and 84 and the center of the bearing surfaces 85 and 86 is offset from the center of the shaft 82 so that there will be a greater spacing between the surfaces on the side adjacent to the opposite gear. These corresponding surfaces are arranged substantially in the manner of the bearing surfaces provided for the shaft 58.

Outside of the plane bearings incorporating the bearing surfaces 83 and 84 are anti-friction bearings 87 and 88 corresponding in structure to

the bearings 67 and 68. A cap 89 is positioned on the right hand end of the housing 51 and is held there by suitable bolts 90. This cap holds the outer race ring of the bearing 67 in position by means of a sleeve 91. The center of this cap has connected thereto, as by a centrally positioned stud 92, the adjusting shaft 93 by which the relative axial position of the gears 55 and 56 may be adjusted. The shaft 93 extends through a member 94 mounted on the housing and the outer end of the shaft is available for manual adjustment of the position of the gears.

The housings 50 and 51 are cut away in the same manner as the housings 5 and 9 above described and interengage with each other in a similar manner. The presence of the housing 51 which surrounds the gear 56 and supports the gear shaft makes a very stable support for the gear, so that its axis will always be in predetermined relation to the axis of rotation of the gear 55. It will be noted from Fig. 6 that the outer surface of the housing 51 is flattened, as at 95 and 96 respectively, and that these surfaces are connected to the suction and pressure sides of the pump so that the lateral pressure exerted on the housings during the rotation of the gears may be counterbalanced to a great degree by the fluid pressure.

The operation of this gear pump is entirely similar to that of the pump disclosed in Figs. 1 to 4 inclusive. When the pump is operating at maximum volume, the gears are in the position shown in Fig. 5, and the entire tooth surface on one gear is in contact with the entire tooth surface of the opposite gear. The gears may be either ordinary spur gears, or any of the type of involute gear used in gear pumps. Obviously the gears may be helically cut to assure a more uniform pumping action.

The anti-friction bearings for the two gear shafts are so designed that the inner races may be slightly expanded so that these bearings will have substantially no play, and accordingly, at low speeds, the anti-friction bearings carry the gear shafts and assure a free rotation thereof. As the rate of rotation increases with the gears rotating in the directions indicated by the arrows, Fig. 6, the lubricant between the bearing surfaces of the plane bearings tends to build up in the form of a wedge angle, due to the eccentricity in the bearings, so that the radial thrust on the bearings resulting from the pumping action and represented by the arrow T, Fig. 6, will be counterbalanced by the oil pressures built up between the bearing surfaces of the plane bearings, with these bearings carrying substantially all of the load. This has been found desirable since anti-friction bearings under the excessive high speeds and excessive pressures necessary for some gear pump installations have been found to fail from overloading.

It will be understood that the disclosure of Figs. 5 and 6 is directed entirely to a gear pump with the power input applied through the shaft 58. The column is controlled obviously by axial shifting of the housing 51, together with the gear 56, thereby controlling the amount of tooth contact of the gears 55 and 56. Since the gear is supported on opposite sides by the housing 51 (of which the sleeve 81 is functionally an integral part) the axially sliding movement of the housing 51 can be effected readily even though the pressure differential of the pump is large, and while the pump is in operation. It will be understood that the flattened portions of the hous-

ing 51 aid in distributing pressure and relieving the frictional drag on the housing during the axial shifting movement.

As above stated, the structure of Figs. 5 and 6 is intended for use as a pump, and when this is true, the radial thrust load, resulting from a pumping action, is in the direction of the arrow T. When the device is used as a motor the radial thrust is in the opposite direction, and in order that the plane bearings may function if the device is used as a motor, it is essential that the relative position of the eccentric outer bearing surface be shifted. Thus, if the device of Figs. 5 and 6 were to be used as a motor, the centers of the outer bearing surfaces 61 and 62 and 85 and 86 would be on the sides of the gear axes remote from the opposite gear in order that the minimum clearance between the inner and outer surfaces of each bearing might be adjacent to the point of maximum radial thrust.

With reference to Fig. 7, the teeth 12' of the gear 12 are axially tapered, as shown, and the teeth 8' of the gear 8 are similarly tapered axially so that the width of each tooth at the pitch line varies from end to end of the tooth. Furthermore, the width of each of the teeth 8' and 12' is so arranged that there will be a back-lash between the gears. That is to say, the width of each tooth is less at the pitch line than the space between adjacent teeth on the opposite gear at the pitch line. Thus, as shown in this figure, there is a space between the leading edge of each tooth on the driven gear and the rearward face of the adjacent tooth on the driving gear.

It will be apparent from Fig. 7 that when the gear 12 is shifted to the right in the direction of the arrow, thereby varying the amount of effective gear tooth contact, the amount of back-lash between the teeth becomes smaller, as will be apparent from the dot-dash showing of the tooth 12'. By this arrangement it is possible to materially reduce the amount of leakage when there is only a small effective tooth contact. At the same time as the gears become more nearly fully engaged the back-lash increases to prevent locking of oil between adjacent teeth of the gears.

Although the amount of back-lash shown in Fig. 7 is uniform the entire width of the gears when they are fully in mesh, it may be desirable to arrange the gear tapers so that when the gears are fully in mesh the back-lash itself will be tapered from end to end so that there will be a tendency for the oil between adjacent gear teeth to be forced toward one end of the gears rather than to be trapped in the spaces between the teeth. To this end, as shown in Fig. 8, the gear teeth 8'' are conventional and at the pitch line are of a uniform width throughout the length of each tooth. The teeth 12'', however, taper from one end to the other with the result that when the gear carrying the tooth 12'' is moved endwise to place the teeth in the dot-dash position shown, the relatively small back-lash is obtained to correspond to the diminishing amount of back-lash of Fig. 7. At the same time the advantage of a tapering back-lash, which prevents trapping of oil, is also assured, independently of the amount that the gear teeth on opposite gears are in engagement.

Where the pump is a constant volume pump the tapered back-lash may be obtained by tapering the teeth of one or both gears from the center toward the end of each gear tooth. As shown in Fig. 9, the teeth 8''' are tapered from the center

of each tooth toward the end thereof along the pitch line and the gear teeth 12" are of a uniform width throughout their length. By this arrangement the fluid being pumped may be urged endwise into the relief pockets at the ends of the gears, thereby preventing locking of oil, with resultant vibration in the pump.

It will be understood that the tapered arrangement of Fig. 8 is applicable not only to a constant volume pump, but also to a variable volume pump of the type shown in Figs. 1 and 5. It may be further noted in connection with these latter figures, that although the tapered gear tooth arrangement is described in connection with the disclosure of Figs. 1, 2 and 3, it is equally applicable to the arrangement of Fig. 5.

Further, in connection with Figs. 1 and 5, it may be noted that in either of these structures the slidable housing in which each gear is mounted may be eliminated in connection with the stationary gear without interfering with the functioning of the housing so far as the housing provides for easy axial sliding movement of the movable gear. It is desirable, however, in certain instances to utilize the sliding housing with the stationary gear in the manner shown, as this arrangement provides for easier assembly of the gear and its supporting structure within the main housing in which the gears are placed.

From the foregoing it will be apparent that the invention resides in a housing for supporting either of the gears of a gear pump, with this housing so arranged as to provide a bearing support for the gear shaft on each side of the gear. One of the gears being axially slidable for adjustment of the volume of fluid pumped (or for adjustment of the rate of rotation of the motor when the device is used for a motor) it is clear that the housing being substantially longer than the axial thickness of the gear will provide a very stable support for the gear and by its movement axially, will cause a corresponding axial movement of the gear without in any way affecting the accurate axial spacing between the gears or the precision alinement of the gear shafts.

I claim:

1. In a device of the class described, a casing having intercommunicating bores therein, said casing also having intake and discharge ports, a pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gear, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having a channel extending from the discharge port in the casing to the outside of the housing at a point approximately opposite to the discharge port.

2. In a device of the class described, a casing having intercommunicating bores therein, said casing also having intake and discharge ports, a pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gear, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having a channel extending from the discharge port in the casing to the outside of the housing at a point approximately opposite to the discharge port, said housing having a flat portion on its

outer periphery communicating with the end of said channel.

3. In a device of the class described, a casing having intercommunicating bores therein, said casing also having intake and discharge ports, a pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gear, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having channels therein extending respectively from the intake and discharge ports of the casing to points on the periphery of the housing substantially diametrically opposite to the intake and discharge ports to partially counterbalance the axial thrust on the housing, whereby axial sliding movement of the housing while the gears are rotating is possible.

4. In a device of the class described, a casing having intercommunicating bores therein, said casing also having intake and discharge ports, a pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gear, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having channels therein extending respectively from the intake and discharge ports of the casing to points on the periphery of the housing substantially diametrically opposite to the intake and discharge ports, said housing having flat portions on its periphery communicating with the ends of the channels, thereby partially counterbalancing the axial thrust on the gears and housing and making possible the endwise shifting of the housing when the gears are rotating.

5. In a device of the class described, a casing having intercommunicating bores therein, said casing also having intake and discharge ports, a pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gear, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having a channel extending from the discharge port in the casing to the outside of the housing at a point approximately opposite to the discharge port and having a flat portion on its outer periphery communicating with the end of said channel, said housing also having a channel from said flat portion to the opening in the housing in which the gear is turnable.

6. In a device of the class described, a casing having interconnecting bores therein, said casing also having intake and discharge ports, a pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gear, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having a recessed portion on its outer periphery at a point approximately opposite to the discharge port in the casing, and a passage for fluid between the discharge port in the casing and said portion.

7. In a device of the class described, a casing having intercommunicating bores therein, said casing also having intake and discharge ports, a

pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gears, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having areas on its periphery substantially diametrically opposite to the intake and discharge ports, said areas being recessed in the housing, and passages for fluid from the intake and discharge ports to the substantially diametrically opposed areas, whereby the fluid pressure in said areas partially counterbalances the axial thrust on the gears and housing and makes possible the endwise shifting of the housing when the gears are rotating.

8. In a device of the class described, a casing

having intercommunicating bores therein, said casing also having intake and discharge ports, a pair of intermeshing gears, one of which is positioned in each bore, at least one of said bores having a housing slidable endwise therein, said housing extending around and on opposite sides of said gear, and a shaft supported by said housing on opposite sides of the gear and on which the gear is mounted, said housing having a channel extending from the discharge port in the casing to the outside of the housing at a point approximately opposite to the discharge port and having a recessed portion on its outer periphery communicating with the end of said channel, said housing also having a channel from said portion to the opening in the housing in which the gear is turnable.

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