Dec. 28, 1943. G. A. UNGAR 2,338,065

GEAR PUMP

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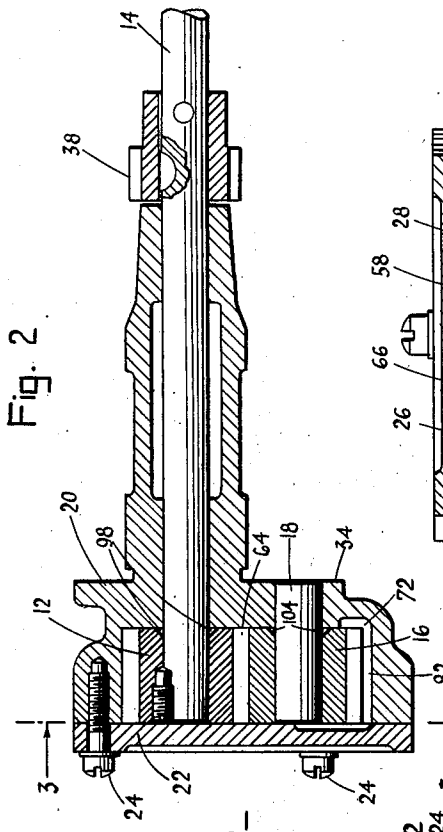


Fig. 2

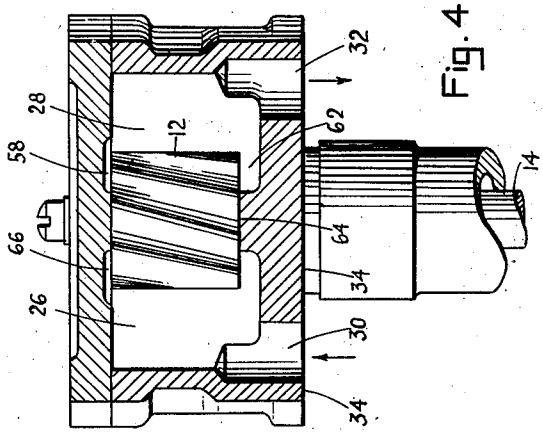


Fig. 4

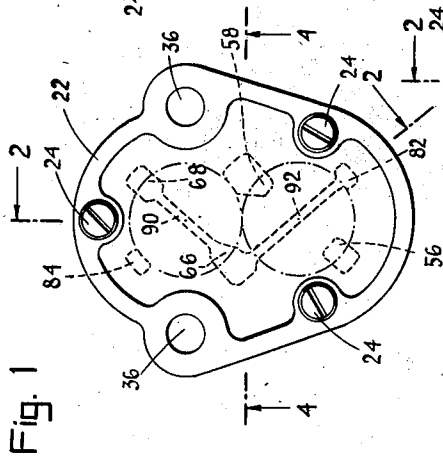


Fig. 1

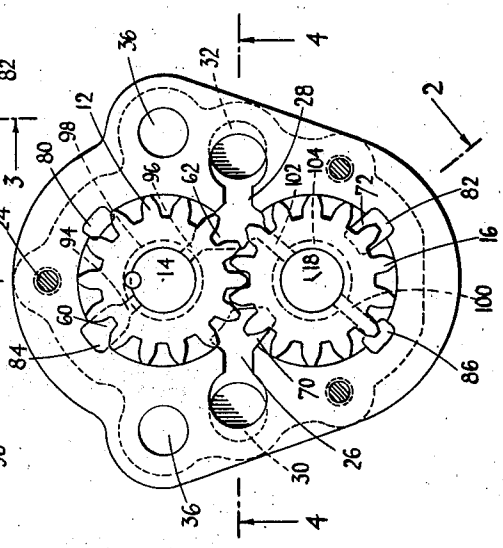


Fig. 3

INVENTOR
GUSTAVE A. UNGAR
BY James & Franklin
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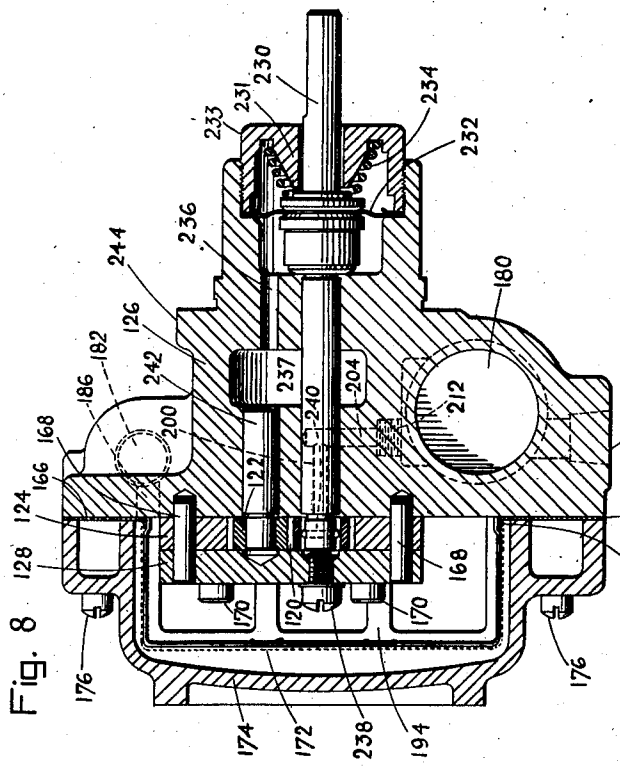


Fig. 8

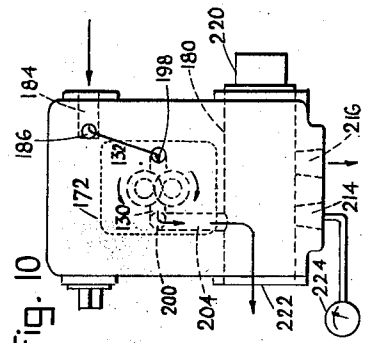


Fig. 10

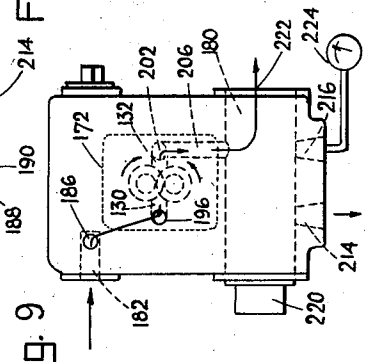


Fig. 9

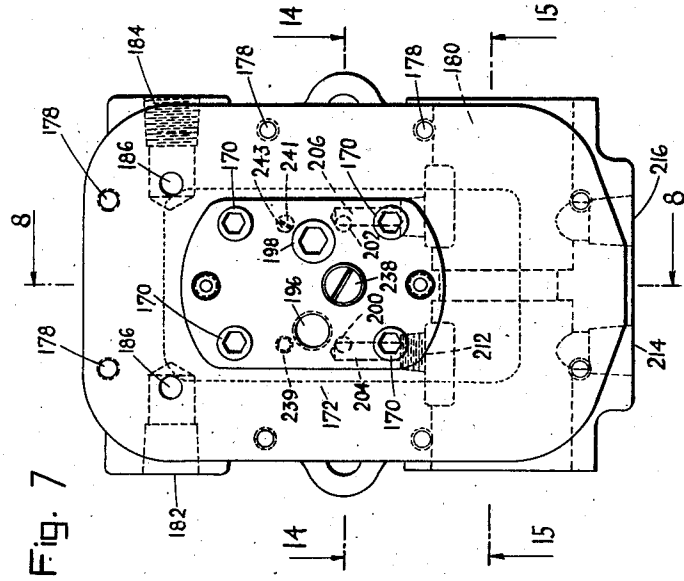


Fig. 7

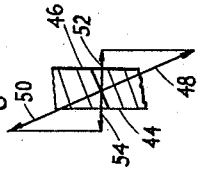


Fig. 6

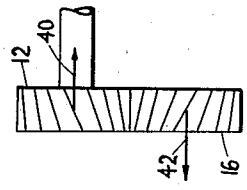


Fig. 5

INVENTOR
GUSTAVE A. UNGAR
BY James & Franklin
ATTORNEYS

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Fig. 11

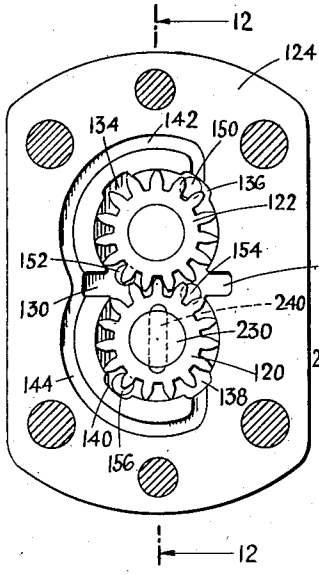


Fig. 12

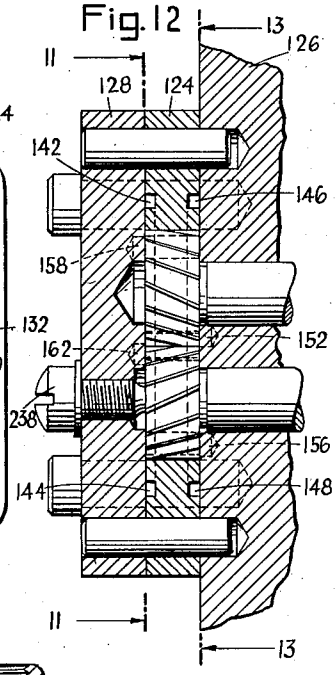


Fig. 13

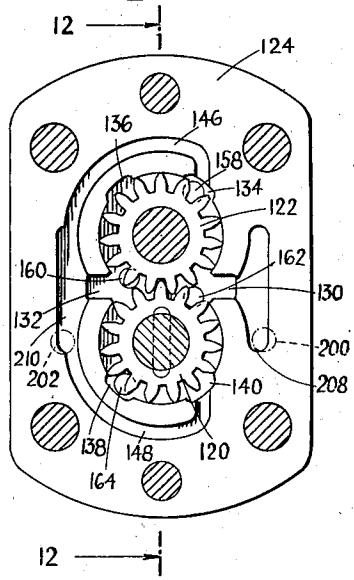


Fig. 14

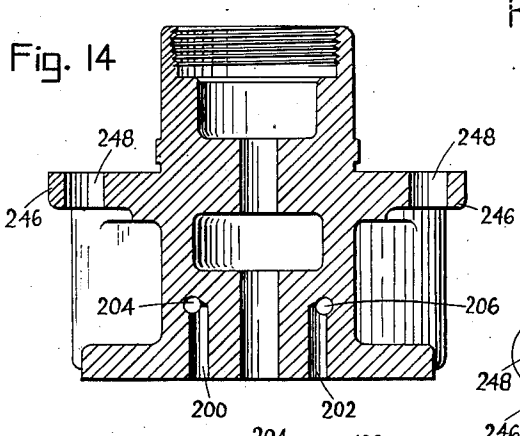


Fig. 15

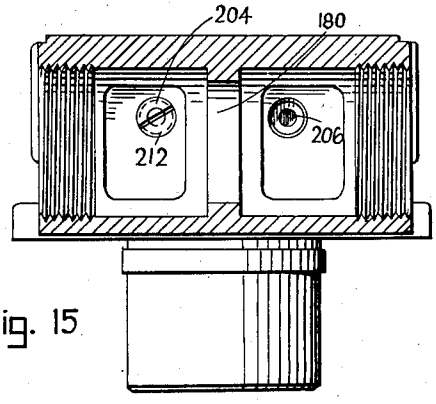
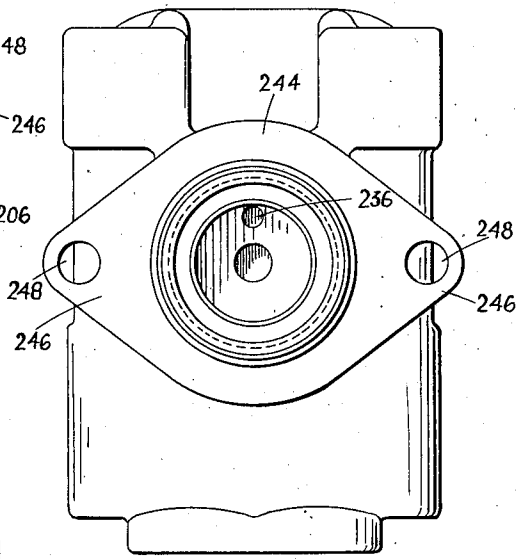


Fig. 16



INVENTOR
GUSTAVE A. UNGAR
BY James & Franklin
ATTORNEYS

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Fig. 17

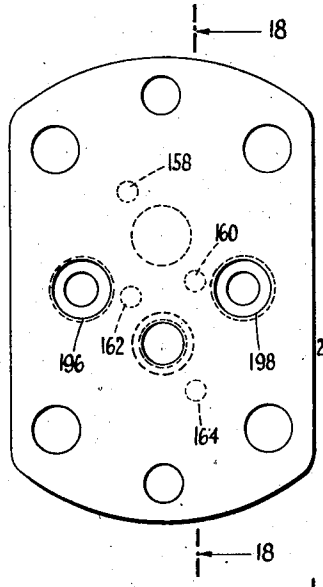


Fig. 19

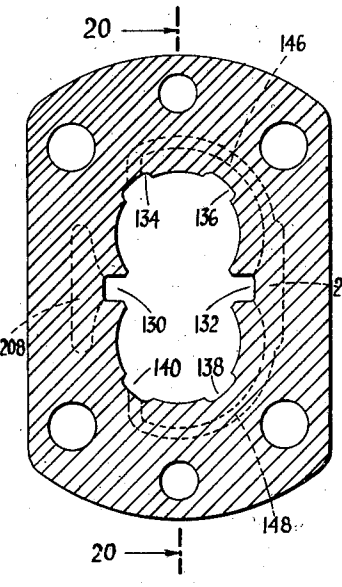
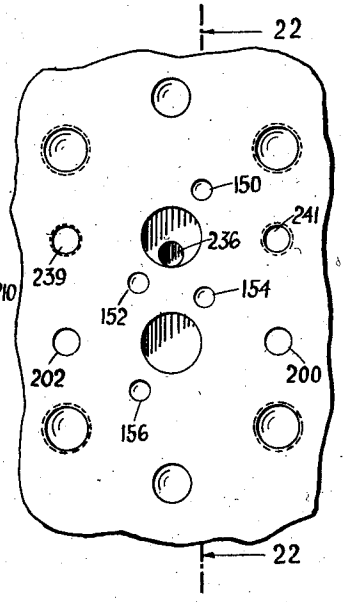


Fig. 21



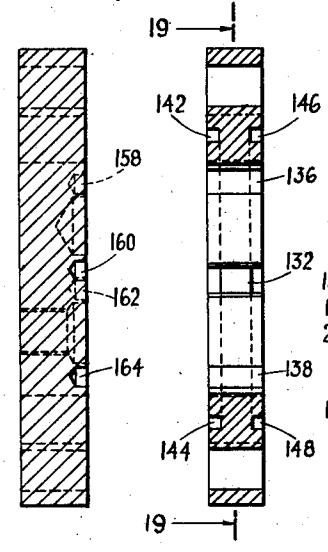


Fig. 18 Fig. 20

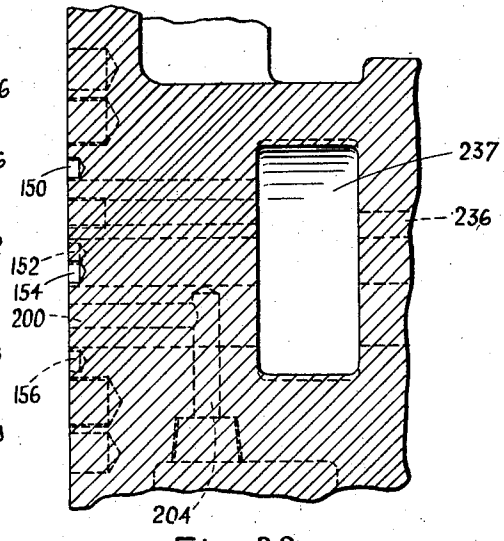


Fig. 22

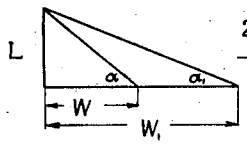


Fig. 23

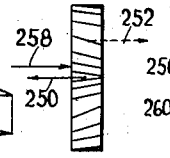


Fig. 24

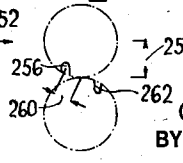


Fig. 25

INVENTOR
GUSTAVE A. UNGAR
BY James & Franklin
ATTORNEYS

Patented Dec. 28, 1943 2,338,065

UNITED STATES PATENT OFFICE

2,338,065

GEAR PUMP

Gustave A. Ungar, Pelham Manor, N. Y., assignor to Joseph F. Keller, New York, N. Y.

Application June 13, 1940, Serial No. 340,239

20 Claims. (Cl. 103—128)

This invention relates to gear pumps, especially helical gear pumps using single helical gears, and to an arrangement for hydraulically axially balancing the same.

To avoid noise in gear pumps, it is desirable to use helical gears rather than spur gears. For axial balance the helical gears are usually double helical gears or herringbone gears. However, double helical gears are expensive relative to single helical gears; are available in only a few helix angles, for production reasons; and are not commercially feasible at all for very small size pumps. If single helical gears are employed, difficulty arises because of axial thrust, one gear trying to move axially in one direction, and the other trying to move axially in the other direction. Unless special thrust bearings are employed, the clearance is eliminated on one side, and exaggerated on the other, exposing the end face to hydraulic pressure and so aggravating the unbalance and increasing gear wear.

The primary object of the present invention is to overcome this difficulty by hydraulically balancing the gear thrust. A corollary object is to provide a counterbalancing force which is proportional to the delivery pressure of the pump. For this purpose, I provide axial balance pressure ports in the pump casing at that side of the gear which tends to move toward the casing. I may further provide axial balance suction ports in the casing at that side of the gear which tends to move away from the casing. Of course, the position of the ports for the two gears is reversed, the pressure ports of one gear being on the same side of the casing as the suction ports of the other gear.

The area of the ports determines the counterbalancing force at any given pressure. It may be pointed out that the torque and consequently the tooth pressure is proportional to the hydraulic pressure, and that inasmuch as the axial force is proportional to the tangential force or tooth pressure, it follows that the axial gear thrust remains substantially proportional to the hydraulic pressure. However, for a given port area, the counterbalancing force is also proportional to the hydraulic pressure. Consequently, my improved pump remains axially balanced despite changes in the pumping pressure or hydraulic pressure.

The main pumping ports produce a radial thrust on the gears. It has already been proposed to counterbalance this radial thrust by appropriate radial balance ports extending outside the gears in a direction parallel to the axes of the gears, the radial balance pressure ports being located diametrically opposite the pumping pressure port, and the radial balance suction ports being located diametrically opposite the pumping suction port. In accordance with a further feature and object of the present invention, the axial balance ports are combined with the radial balance ports, one being, in effect, a direct continuation or extension of the other. For the sake of symmetry, and to avoid any tilting or wobble force on the gears, axial balance ports are located diametrically opposite one another, the inner ones being formed directly in communication with the main pumping ports. The other balance ports are connected to the appropriate pumping ports by suitable passages which may run to either the axial or radial balance ports, whichever may be more convenient, for these are in direct communication with one another.

To the accomplishment of the foregoing main objects and other more specific objects which will hereinafter appear, my invention consists in the single helical gear pump elements and their relation one to the other as hereinafter are more particularly described in the specification and sought to be defined in the claims. The specification is accompanied by drawings, in which:

Fig. 1 is an end elevation of a simple form of gear pump embodying features of my invention;

Fig. 2 is a longitudinal section taken in the plane of the line 2—2 of Figs. 1 and 3;

Fig. 3 is an end elevation with the cover plate removed, looking in the plane of the line 3—3 of Fig. 2;

Fig. 4 is a horizontal section taken in the plane of the line 4—4 of Figs. 1 and 3;

Figs. 5 and 6 are diagrams explanatory of the invention;

Fig. 7 is an end elevation of another pump embodying features of my invention;

Fig. 8 is a section taken in the plane of the line 8—8 of Fig. 7;

Fig. 9 is a schematic flow diagram with the pump piped in one manner;

Fig. 10 is a schematic flow diagram with the pump piped in a different manner;

Fig. 11 is a section through the gear housing taken in the plane of the line 11—11 of Fig. 12;

Fig. 12 is a section through the gear housing taken in the plane of the line 12—12 of Figs. 11 and 13;

Fig. 13 is a section through the gear housing taken in the plane of the line 13—13 of Fig. 12;

Fig. 14 is a horizontal section through the pump body taken in the plane of the line 14—14 of Fig. 7;

Fig. 15 is a horizontal section through the pump body taken in the plane of the line 15—15 of Fig. 7;

Fig. 16 is an end elevation of the pump body;

Fig. 17 is a front elevation of the gear housing cover;

Fig. 18 is a vertical section therethrough taken in the plane of the line 18—18 of Fig. 17;

Fig. 19 is a section through the gear housing taken in the plane of the line 19—19 of Fig. 20;

Fig. 20 is a vertical section through the gear housing taken in the plane of the line 20—20 of Fig. 19;

Fig. 21 is a front elevation of a part of the pump body;

Fig. 22 is a vertical section through the pump body taken in the plane of the line 22—22 of Fig. 21;

Fig. 23 is a diagram explanatory of a feature of the invention; and

Figs. 24 and 25 are diagrammatic views explanatory of an advantage of a modification.

Referring to the drawings, and more particularly to Figs. 1 through 4, the gear pump there shown is a miniature pump intended for use as a lubricating oil pump for an internal combustion engine.

It comprises a driving gear 12 mounted on a driving shaft 14, and meshing with a driven gear 16 freely rotatable on a pin 18. The pump housing comprises a body 20 which forms a double chambered gear housing closed by a cover plate 22 held by screws 24. The pumping ports are clearly shown at 26 and 28 in Figs. 3 and 4, these being located in conventional fashion and communicating, in the present case, with rearwardly extending passages 30 and 32. For some purposes these passages may be threaded for pipe connections, but this is not done in the present instance because the surface 34 of the pump is intended to be bolted against an appropriate part of the engine body having openings registering with the passages 30 and 32. The holes 36 (Figs. 1 and 3) are provided to receive the bolts which hold the pump against the engine body. The drive shaft 14 is driven by a suitable pinion 38 meshing with a gear in the engine as, for example, on the cam shaft or other convenient part thereof.

The gears are helical gears for silence, but are so tiny that they are preferably single helical, rather than double helical or herringbone gears. The result is an opposite axial thrust which corresponds to the axial component of the gear tooth pressure. Thus, referring to Fig. 5, the driving gear 12 will be urged in one direction, as is indicated by the arrow 40, while the driven gear 16 will be urged in opposite direction, as is indicated by the arrow 42. The reason for this is diagrammatically shown in Fig. 6 which is supposedly taken in a plane extending transversely between the gears at their point of contact. The numeral 44 represents a driving tooth bearing against a driven tooth represented at 46. The reaction on the driving gear tooth 44 is represented by vector 48, while the direct tooth pressure on the driven gear tooth 46 is represented by the opposite vector 50. Vector 48 has an axial component 52, and vector 50 has an axial component 54. These axial components produce an undesirable axial thrust which the present invention compensates by hydraulic means.

The general principle underlying the present invention is the provision of pressure ports in the casing at that face of the gear which tends to move toward the casing. In the present case, I shall assume the direction of rotation to be such that pumping port 26 (Fig. 3) is a suction port and pumping port 28 a pressure port. Specifically, I provide axial balance pressure ports 56 and 58 (Fig. 1) on the outer face of driven gear 16, both ports being shown in Fig. 1, and one of the ports being shown in Fig. 4. The ports are formed by recesses in cover plate 22 and are preferably located diametrically opposite one another in order not to introduce any tilting force on the gear. The axial balance pressure ports for the driving gear 12 are shown at 60 and 62 in Fig. 3, the latter port being also shown in Fig. 4. These ports are formed by recesses in the side wall 64 (Figs. 2 and 4) of the gear housing, and are preferably located diametrically opposite one another.

I prefer to also provide suction ports in the casing at that face of the gear which tends to move away from the casing. This is desirable for hydraulic reasons, and also so that the pump will be reversible. Thus, in the present case, I provide suction ports 66 and 68 (Fig. 1) in the cover plate 22 at the outer face of driving gear 12, these ports being shown in Fig. 1, and the port 66 being shown also in Fig. 4. I further provide suction ports 70 and 72 on the inner side of driven gear 16, these ports being shown in Fig. 3, and the port 72 also being shown in Fig. 2.

It will be noted that the pressure ports 56 and 58 (Fig. 1) for driven gear 16 are located on the same side of the pump as the suction ports 66 and 68 for the driving gear 12, these ports all being in the cover plate 22, and that the pressure ports 60 and 62 (Fig. 3) for the driving gear 12 are located on the same side of the pump as the suction ports 70 and 72 for the driven gear 16, these all being in the face wall 64 of the pump body. The axial balance ports thus create hydraulic thrusts opposite to the mechanical gear thrusts.

For any given pump pressure, the axial balance force depends on the effective port area of the axial balance ports. When this area has been properly selected the pump will remain in balance, even though operated at different pressures. This will be clear from the following formula, in which:

The diametrical pitch is DP

The depth of the gear tooth is

$$\frac{2}{DP}$$

The helical gear thrust is G_T lbs.

The tangential gear load is T_P lbs. (the force required to drive the driven gear)

The width of the gear tooth is W

The helix angle is a

The hydraulic pressure is p lbs. per square inch

The mechanical efficiency of the pump is E_m

$G_T = T_p \tan a$ disregarding mechanical efficiency

$$T_p = \frac{2}{DP}\frac{1}{2} W_p = \frac{W}{DP} p$$

$$G_T = \frac{W}{DP} \tan a p \frac{1}{E_m}$$

G_T is therefore proportional to P. However, the compensating force is also proportional to p. Hence, the pump remains in balance, despite changes in pump pressure.

I have so far discussed the matter of axial balance which is the primary purpose of the present invention. However, the gears in a gear

pump are also subject to radial thrust or unbalance which is preferably compensated by appropriate radial balance ports. Referring to Fig. 3, it will be seen that part of the peripheries of the gears 12 and 16 are exposed to pump pressure port 28, and that part of the peripheries of the gears are exposed to suction pressure port 26. These create radial unbalance forces which are preferably counteracted by appropriate radial balance ports extending axially of the gears at points diametrically opposite the pressure ports. Thus, the effect of suction port 26 on gear 12 is preferably counteracted by a radial balance suction port 80, and its effect on gear 16 is preferably compensated by a radial balance suction port 82. The effect of pump pressure port 28 on gear 12 is preferably counteracted by a radial balance pressure port 84, and its effect on gear 16 is counteracted by radial balance pressure port 86. The radial balance suction port 82 is clearly shown in Fig. 2 of the drawings.

In accordance with a feature and object of the present invention, the radial and axial balance ports are preferably combined in a convenient manner by simply locating the axial balance ports at the ends of the radial balance ports. Thus, an axial balance port may be considered a continuation of a radial balance port or vice versa, the complete port extending along the outside of the gear and turning around the side face of the gear at one end or the other. Specifically, axial balance pressure port 60 is formed at the inner end of radial balance pressure port 84. Axial balance pressure port 56 is formed at the outer end of radial balance pressure port 86. Axial balance suction port 72 is formed at the inner end of radial balance suction port 82. Axial balance suction port 68 is formed at the outer end of radial balance suction port 80. The other axial balance ports are formed as direct continuations of the main pumping ports 26 and 28. Thus, axial balance pressure port 58 is formed at the outer end of pump pressure port 28, while axial balance pressure port 62 is formed at the inner end of pump pressure port 28. Axial balance suction port 66 is formed at the outer end of pump suction port 26, and axial balance suction port 70 is formed at the inner end of pump suction port 26.

The balance ports should, of course, be connected to the pumping ports, the balance pressure ports being connected to the pump pressure port, and the balance suction ports being connected to the pump suction port. The necessary connections may be made by complex passages formed within the pump castings, but in the present case the desired connections are made in very simple fashion by means of open channels formed in the castings and effectively closed by only the side faces of the gears. Specifically, referring to Fig. 1, the axial balance suction ports 66 and 68 are connected to one another and to the pump suction port by means of a diametrically extending passage 90. This is formed on the inner face of cover plate 22, and is closed by only the outer face wall of driving gear 12. Similarly, the radial balance suction port 82 is connected to the pump suction port by means of a passage 92 extending diametrically of the driven gear and closed solely by the outer face of the driven gear 16.

The pressure ports are connected by diametrical passages at the opposite end of the pump. Specifically, and referring now to Fig. 3, the axial balance pressure ports 60 and 62 are connected to one another and to the pump pressure port 28 by means of a diametrical passage 94, 96, the two parts being interrupted by the shaft 14. To connect the parts 94, 96, I employ an annular passage 98 extending around shaft 14. It is more convenient to form this annular passage in the gear 12, as is shown at 98 in Fig. 2, than to form it in the body of the pump, but either location may be used. The diametrical passage 94, 96 is, of course, an open channel formed in the face wall 64 of the pump body and closed solely by the inner face of driving gear 12.

The radial balance pressure port 86 is connected to the pump pressure port 28 by an open diametrical channel 100, 102 shown in Fig. 3. These are connected by an annular passage 104 extending around the pin 18 of the driven gear. The annular passage 104 is preferably formed in driven gear 16, as is best shown in Fig. 2. The diametrical passage 100, 102 is an open channel which is closed by the inner face of driven gear 16.

It will be understood that while it is very convenient and inexpensive to employ communicating passages of the type here shown, they, however, affect the balance of the pump. Thus, the suction passage 90 adds to the axial balance suction port area, while the suction passage 92 operates in negative sense, it being located at the pressure port side of the driven gear. Similarly, the pressure port passage 94, 96 adds to the axial balance pressure port area, while the passage 100, 102 acts in a negative sense, for it is located on the suction port side of the pump. The resulting error is readily taken care of by appropriately modifying the size or area of the axial balance ports, the ports 66, 68, 60, and 62 being reduced somewhat in area, or the ports 56, 58, 70 and 72 being enlarged somewhat in area, or both.

As an illustration of the small size of the pump here under discussion, I may point out that in one specific case the helical gears had a pitch diameter of 1.167 inches, an axial dimension of one inch, and were provided with fourteen teeth having a helix angle of fifteen degrees and a pressure angle of twenty-eight degrees. The outside diameter of the gears was 1.333 and the root diameter 0.990. Such a pump may be used to meet a minimum test performance of 1.25 gallons per minute at 650 revolutions per minute with an oil gauge pressure of twenty-five pounds per square inch, while using an oil such as Mobil "A" at 160 degrees Fahrenheit. It will be understood that these dimensions are given solely by way of illustration and not in limitation of the invention.

Referring now to the remaining sheets of drawings, I there show a pump intended for use as a fuel pump for a household oil burner installation.

Referring first to Figs. 11, 12, and 13 the pump comprises a driving gear 120 meshing with a driven gear 122, these gears being housed in chambers formed in a flat plate-like gear housing 124. Plate 124 is enclosed between a pump body 126 and a gear housing cover 128. Gear housing 124 is provided with pumping ports 130 and 132 (Figs. 11 and 19). It is also provided with radial balance ports 134, 136, 138, and 140. All of these ports pass entirely through the gear housing 124 (see Fig. 19) and their positions are reversed in Fig. 13 relative to Fig. 11, because these show opposite sides of gear housing 124.

The radial balance ports 136 and 138 are con-

nected to pumping port 130 by channels or passages 142 and 144, as is shown in Figs. 11 and 12. The radial balance ports 134 and 140 are connected to pumping port 132 by channels or passages 146 and 148, as is shown in Figs. 12, 13, and 19. The passages are located on opposite faces of the gear housing plate 124, and are closed when plate 124 is secured between the pump body 126 and the cover 128.

As before, the gears are single helical gears which produce opposite axial thrusts. These are counterbalanced by axial balance ports, and it is convenient, in the present case, to locate these in pairs which are diametrically opposite one another, and which are located at the ends of radial balance ports or pumping ports. In the present case, the axial balance ports are circular, being formed by drilling blind holes into the face walls enclosing the gear housing plate 124. Altogether there are eight such axial balance ports, two at each side of each gear. Four of these are visible in Figs. 11, 21, and 22, these being numbered 150, 152, 154, and 156. These are visible through the gear tooth spaces in Fig. 11, and are drilled into the pump body 126, as is indicated in Figs. 12 and 22. The ports 150 and 152 communicate with pumping port 130, the connection of port 150 being by way of the radial balance port 136, while the ports 154 and 156 communicate with the pumping port 132.

The other four axial balance ports are indicated in Figs. 13 and 17, these being marked 158, 160, 162, and 164. In Fig. 13 these ports are seen through the gear tooth spaces. They are drilled into the cover plate 128, as is indicated in Figs. 12 and 18. The ports 158 and 160 communicate with the pumping port 132, while the ports 162 and 164 communicate with the pumping port 130, the connection of port 164 being by way of the radial balance port 138.

The relation of the gear housing to the complete pump is best shown in Figs. 7 and 8, in which it will be seen that the gear housing plate 124 and the gear housing cover 128 are secured to the face wall 166 of pump body 126. The parts are properly located by dowels 168 and are secured in position by bolts 170.

The entire gear housing is enclosed within a large area strainer 172, which in turn is housed within a strainer cover 174, the latter being shown in Fig. 8 but omitted in Fig. 7. The cover 174 is secured to the pump body 126 by suitable fastenings, in this case, six screws 176, the threaded holes for receiving which are shown in the pump body at 178 in Fig. 7.

All piping is, for convenience, connected to the pump body. This includes an inlet pipe, a discharge pipe, and a by-pass pipe for return of excess fuel oil to the storage tank. The pump body is made with symmetrical passages, so that the piping may be connected in either direction, the unused passages being blocked by suitable plugs. For this reason, there are six pipe connections instead of three. Moreover, some of the internal passages within the pump may be reversed to accommodate rotation of the pump in either direction, there being four passages to and from the pump, instead of two, two of the passages being normally closed by appropriate plugs. The pump body is provided with a large cylindrical passage 180 for receiving a by-pass unit which is made up separately and need not be described in detail here, except for brief mention of flow connections to the same.

Considering the arrangement in greater detail, the pump body 126 is provided with inlet passages 182 and 184 near the upper corners thereof. These communicate with holes 186 extending in the direction of the axis of the pump and leading to points outside the strainer 172, as is clearly shown in Fig. 7. It will be understood that the strainer is soldered or otherwise sealed to the turned edge 188 (Fig. 8) of a plate or strainer flange 190, the latter being locked between the strainer cover 174 and the pump body 126. Suitable gaskets may be used at each side of the strainer flange 190 to insure a fluid tight connection. The strainer 172 may, if desired, be supported against bending or collapse by a suitable frame or support 194. The incoming liquid flows into the space between strainer cover 174 and strainer 172, and then flows inwardly through strainer 172 to the gear housing.

The liquid then flows through gear housing cover 128 to the suction port of the pump. There are two openings for this purpose, marked 196 and 198 in Fig. 7, the latter being closed by a plug and the former acting as an inlet. Referring to Figs. 7, 11, and 17, it should be understood that the inlet hole 196 leads directly to the pumping port 130, and that the hole 198 leads to the pumping port 132. In the present case, with hole 198 blocked, the pumping port 130 acts as a suction port, while pumping port 132 acts as a pressure port.

The liquid discharged by the gears flows in opposite direction, that is, toward the right or into the pump body 126, as viewed in Fig. 8. Specifically, the pump body is provided with holes 200 and 202 (Fig. 7) extending axially of the pump until they run into vertical holes 204 and 206 which lead downwardly to the by-pass chamber 180. The horizontal passages 200 and 202 leading to vertical passages 204 and 206 are clearly shown in Fig. 14, looking downward, while the lower ends of passages 204 and 206 leading into the by-pass chamber 180 are clearly shown in Fig. 15, looking upward. The hole 200 communicates with an enlargement 208 (Figs. 13 and 19) at one end of the pumping port 130, and similarly the hole 202 communicates with an enlargement 210 at the outer end of pumping port 132. These enlargements do not go through the gear housing plate 124. They are localized at the inner wall thereof, much as is the case with the connecting passages 146 and 148. They are used merely to permit the horizontal flow passages 200 and 202 to be located at downwardly offset points.

As before, only one or the other of these communicating passages is used at any one time. It is convenient to plug the passage at a point near the by-pass chamber 180. In the present case, the vertical passage 204 is plugged at 212, as is shown in Figs. 7, 8, and 15. Thus, the passages 202 and 206 are effective, the passage 202 communicating with the pumping port 132, which is as it should be.

The pump body 126 is provided with two pipe connections 214 and 216 (Figs. 7 and 8). These passages open directly into the by-pass chamber 180 at a suitable point to receive by-pass fluid from the by-pass unit. One or the other hole is plugged or, if desired, may be connected to a pressure gauge, the latter being connected temporarily for servicing purposes, or permanently if desired.

The flow arrangement may be clarified with reference to the schematic diagram shown in

Fig. 9. Liquid is piped into the pump body at 182 and then flows forwardly through hole 186 to the region surrounding the strainer 172. The liquid then flows through the strainer and rearwardly through hole 196 to pumping port 130. The liquid is pumped around with the gears to pumping port 132, and then flows rearwardly through hole 202, and then downwardly at 206 to a suitable by-pass unit 220 in the by-pass chamber 180. The discharge pipe is connected to the end 222 of by-pass unit 220, the opposite end being a closed end. By-pass fluid flows downwardly through passage 214. A pressure gauge 224 may be connected at passage 216, or passage 216 may be simply blocked by a plug.

If it is more convenient to connect the piping in opposite fashion, then the arrangement will operate as shown in Fig. 10, there being an inlet pipe connected at 184 leading through forward passage 186 to the region surrounding strainer 172, whereupon the liquid flows through the strainer to passage 198 leading to pumping port 132. The liquid then moves around with the gears to pumping port 130, and then flows rearwardly through passage 200 and then downwardly through passage 204 to by-pass unit 220, inserted in by-pass chamber 180 in reversed position, the closed end being at 220 and the pipe connection for the discharge pipe being at 222. By-pass liquid flows to the storage tank at 216, while passage 214 is either plugged or connected to a pressure gauge 224.

Figs. 9 and 10 should not create the impression that the inlet pipe connections are necessarily related to the direction of rotation of the pump. This is not at all the case. Once the inlet liquid surrounds the strainer, it is accessible to either pumping port. The location of the discharge passage and the by-pass unit is related to the direction of rotation, but in Fig. 9 the inlet pipe may be connected at the upper right-hand corner, instead of the upper left-hand corner, and in Fig. 10, the inlet pipe may be connected at the upper left-hand corner, instead of the upper right-hand corner, according to greatest convenience in making the necessary pipe connections. To express the matter in a different manner, the direction of rotation determines the character of the pumping ports, which in turn determines the location of the plugs located within the pump-specifically, the plugs 198 and 212 in the present case. The pipe connections determine the location of the external plugs.

Referring now to Fig. 8, the driving gear 120 is driven by a driving shaft 230, the latter passing through the pump body and being sealed by an appropriate seal, the diaphragm of which is numbered 232. The seal may be of the conventional mechanical surface sealing type and need not be described in detail. This seal includes a spring 234. The inner side of the sealing membrane is exposed to the suction side of the pump through a passage 236. This leads to a hollow space 237, which in turn is connected to the pumping ports by horizontal passages 239 and 241 (Figs. 7 and 21), one of which, that leading to the pressure port, is blocked off by screw 243. The passages 239 and 241 lead to the upwardly extending parts of the enlargements 208 and 210 (Figs. 13 and 19).

These things lead to an axial thrust on the shaft which, when the pump is not operating under pressure, may be counteracted by an end thrust bearing or screw 238. When it is under pressure, the shaft tends to move to the right and there is no thrust on the screw. In all normal cases the delivery pressure exceeds the suction, and the force tending to move the shaft to the right exceeds the force tending to move the shaft to the left. The movement to the right is limited by shoulder 231 of end cover 233. It is desirable to eliminate any effect of these elements on the axial balance of the driving gear 120. Accordingly, in the present pump the shaft 230 is slidably keyed to the driving gear 120. Referring to Figs. 11 and 13, the driving shaft 230 is provided with a transverse pin or key 240, the ends of which are received in splines or keyways formed on the inside of driving gear 120.

The driven gear 122 is freely rotatable and freely axially slidable on a pin or stationary shaft 242, the latter being received with a force fit in the pump body 126.

Referring to Fig. 8 the pump body may be bolted to the rest of the machine by a suitable flange located at the point 244. The projections 246 of this flange are best shown in Figs. 14 and 16, these being provided with holes 248 for suitable mounting bolts. Figs. 14 and 16 show the pump body alone, the shaft and the parts of the seal being removed.

In a specific case, some of the leading dimensions of the pump heretofore described were as follows:

The gears have an outside diameter of 0.703 inch, a root diameter of 0.5292 inch, and a pitch diameter of 0.625 inch, a face width or axial dimension of 0.3084 inch. The gears have fifteen teeth with a helix angle of twenty-three degrees, and a pressure angle of twenty-eight degrees. The diametral pitch in the plane of rotation is twenty-four. This pump has a capacity of twenty-two gallons per hour without the by-pass action at 1750 R. P. M. and zero pressure.

It will be understood that these dimensions are given solely by way of illustration and not in limitation of the invention. They show the small size of the gears used, thus indicating the difficulty or impracticability of providing double helical gears in such small dimensions.

Before concluding the description, it may be pointed out that once the balance ports and end plates have been worked out for a pump using gears of a certain diameter, the capacity of the pump may be changed by changing the axial dimension or width of the gears without changing the diameter. In such case, redesign of the port area may be avoided by appropriately changing the helix angle of the gears. This may be shown with reference to Fig. 23 of the drawings, in which gears of different width, W and W_1, are assumed to have a different helix angle A and A_1, such as to produce the same lead L. Thus

$$\tan a = \frac{L}{W}$$

$$\tan a_1 = \frac{L}{W_1}$$

Revering to the formula given earlier in the specification:

$$G_T = \frac{W}{DP} \tan ap \frac{1}{E_m}$$

This may be rewritten as:

$$G_T = \frac{W}{DP} \frac{L}{W} \frac{p}{E_m} = \frac{L}{DP} \frac{p}{E_m}$$

Now for a modified gear having a greater axial width, but a smaller helix angle, we may write

$$G_{T_1}=\frac{W_1}{DP}\frac{L}{W_1}\frac{p}{E_m}=\frac{L}{DP}\frac{p}{E_m}$$

which is the same as before. Thus the thrust to be balanced is independent of the gear width, and is the same when the lead is the same.

It is not essential to use diametrically opposite axial balance ports. In some cases it may be preferred to use a single axial balance port and in such case it would preferably be the port near the contact point of the gears, that is, at the end of the pumping port. This arrangement may be used to help reduce a tilting force on the gear caused by the axial component of the gear tooth pressure. Thus, referring to Figs. 24 and 25, if the axial component of the gear tooth pressure is represented by arrow **250**, this would ordinarily be counteracted at the shaft as is indicated by arrow **252**, thus providing a tilting moment, the lever arm of which is indicated at **254**. If, however, the axial thrust **250** is counteracted by an axial balance port **256** creating a force **258**, the lever arm of the resulting moment is indicated at **260** which may be made substantially less than the distance **254**. The axial balance port **262** for the lower gear would, of course, be a suction port, assuming that port **256** is a pressure port. Referring to Fig. 21, it is somewhat as though ports **150** and **156** were eliminated, retaining only the ports **152** and **154**, these, however, being appropriately increased in area.

It is believed that the construction, operation, and principles underlying my new pump, as well as the many advantages of the same, will be apparent from the foregoing detailed description thereof. The advantage of silent operation is obtained while using single helical gears. This lowers the cost of the pump and makes it possible to employ helical gears in relatively tiny pumps despite their small size. However, the invention is not limited to very small pumps. For example, in gasoline dispensing pumps, it has been proposed to eliminate all noise due to the mating of metallic gears by using one gear made of metal, while the other is made of a plastic. Double helical pump gears are usually standardized at a helix angle of approximately thirty degrees. The plastic material tends to chip at the sharp corners of the teeth. With a single helical gear, various other helix angles may be used, for example, fifteen degrees or eighteen degrees. This avoids chipping of the plastic material. The invention is, therefore, of value for freedom in the selection of helix angle, as well as for economy, because single helical gears are much cheaper than double helical gears. They are also available in a greater range of sizes and angles. Also, the molding of double helical gears presents considerable practical difficulties, compared to the molding of single helical gears.

It will be apparent that while I have shown and described my invention in several preferred forms, many changes and modifications may be made in the structures disclosed without departing from the spirit of the invention defined in the following claims.

I claim:

1. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears having side faces and being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including an axial balance port of limited area in the casing opening into that face of the gear which tends to move toward the casing, the axial balance port for one of the gears being on the opposite side of the casing from the axial balance port for the other gear, said ports being located near the peripheries of the gears, and non-restrictive passages freely connecting the pump pressure port to the axial balance ports.

2. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including an axial balance pressure port in the casing at that face of the gear which tends to move toward the casing, an axial balance suction port in the casing at that face of the gear which tends to move away from the casing, the axial balance pressure port for one of the gears being on the same side of the casing as the axial balance suction port for the other gear, non-restrictive passages freely connecting the pump pressure port to the axial balance pressure ports, and non-restrictive passages freely connecting the pump suction port to the axial balance suction ports.

3. A pump as defined in claim 2, in which the connecting passages are open channels formed in the end wall of the casing and extending diametrically of the gears, whereby the adjacent faces of the gears function to close the open side of the channels, there being an annular groove around the shafts of the gears where the aforesaid diametrical passages are interrupted by a shaft.

4. A pump as defined in claim 2, in which the driving gear is splined to the driving shaft so that the gear is independent of axial movement of the shaft.

5. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears having side faces and being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including a plurality of axial balance ports of limited area in the casing opening onto that face of the gear which tends to move toward the casing, the axial balance ports for one of the gears being on the opposite side of the casing from the axial balance ports for the other gear, said ports being located near the peripheries of the gears, and non-restrictive passages freely connecting the pump pressure port to the axial balance pressure ports.

6. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including a plurality of axial balance pressure ports in the casing at that face of the gear which tends to move toward the casing, and a plurality of axial balance suction ports in the casing at that face of the gear which tends to move away from the casing, the axial balance pressure ports for one of the gears being on the same side of the casing as the axial balance suction ports for the other gear, non-restrictive passages freely con-

necting the pump pressure port to the axial balance pressure ports, and non-restrictive passages freely connecting the pump suction port to the axial balance suction ports.

7. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears having side faces and being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including a plurality of axial balance ports of limited area in the casing opening onto that face of the gear which tends to move toward the casing, the axial balance ports being located near the peripheries of the gears and the parts for each gear being located diametrically opposite one another with respect to the gear so as not to produce any tilting moment on the gear, the axial balance ports for one of the gears being on the opposite side of the casing from the axial balance ports for the other gear, and non-restrictive passages freely connecting the pump pressure port to the axial balance ports.

8. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including a plurality of axial balance pressure ports in the casing at that face of the gear which tends to move toward the casing, and a plurality of axial balance suction ports in the casing at that face of the gear which tends to move away from the casing, the axial balance ports for each gear being symmetrically located with respect to the gear so as not to produce any tilting moment on the gear, the axial balance pressure ports for one of the gears being on the same side of the casing as the axial balance suction ports for the other gear, non-restrictive passages freely connecting the pump pressure port to the axial balance pressure ports, and non-restrictive passages freely connecting the pump suction port to the axial balance suction ports.

9. A pump as defined in claim 8, in which the connecting passages are open channels formed in the end wall of the casing and extending diametrically of the gears, whereby the adjacent faces of the gears function to close the open side of the channels, there being an annular groove around the shafts of the gears where the aforesaid diametrical passages are interrupted by a shaft.

10. A pump as defined in claim 8, in which the driving gear is splined to the driving shaft so that the gear is independent of axial movement of the shaft.

11. A gear pump comprising driving and driven gears, a casing surrounding said gears and including suction and pressure pumping ports leading to said gears, said ports producing a radial thrust on said gears, said gears having side faces and being single helical gears which develop an axial thrust, hydraulic balance means for counteracting the radial thrust, said means including radial balance ports extending in the direction of the axes of the gears at points diametrically opposite the pressure pumping port, and hydraulic means for hydraulically counterbalancing the axial gear thrust, said means including axial balance ports of limited area in the casing opening onto that face of the gear which tends to move toward the casing, the axial balance ports of one gear being on the opposite side of the casing from the axial balance ports of the other gear, said ports being located near the peripheries of the gears, and appropriate non-restrictive passages for freely connecting the balance ports to the pump pressure port.

12. A gear pump comprising driving and driven gears, a casing surrounding said gears and including suction and pressure pumping ports leading to said gears, said ports producing a radial thrust on said gears, said gears being single helical gears which develop an axial thrust, hydraulic balance means for counteracting the radial thrust, said means including radial balance pressure ports extending in the direction of the axes of the gears at points diametrically opposite the pump pressure port, and radial balance suction ports extending in the direction of the axes of the gears at points diametrically opposite the pump suction port, hydraulic means for hydraulically counterbalancing the axial gear thrust, said means including axial balance pressure ports in the casing at that face of the gear which tends to move toward the casing, and axial balance suction ports in the casing at that face of the gear which tends to move away from the casing, the axial balance pressure ports of one gear being on the same side of the casing as the axial balance suction ports of the other gear, and appropriate non-restrictive passages for freely connecting the balance pressure ports to the pump pressure port, and for freely connecting the balance suction ports to the pump suction port.

13. A gear pump comprising driving and driven gears, a casing surrounding said gears and including suction and pressure pumping ports leading to said gears, the fluid in said ports producing a radial thrust on said gears, said gears being single helical gears which develop an axial thrust, hydraulic balance means for counteracting the radial thrust, said means including radial balance ports extending in the direction of the axes of the gears at points diametrically opposite the pressure pumping port, and hydraulic means for hydraulically counterbalancing the axial gear thrust, said means including axial balance ports in the casing at that face of the gear which tends to move toward the casing, the axial balance ports of one gear being on the opposite side of the casing from the axial balance ports of the other gear, the axial balance ports being formed directly at the ends of the radial balance ports and the pump pressure port, and appropriate passages for connecting the balance ports to the pump pressure port.

14. A gear pump comprising driving and driven gears, a casing surrounding said gears and including suction and pressure pumping ports leading to said gears, the fluid in said ports producing a radial thrust on said gears, said gears being single helical gears which develop an axial thrust, hydraulic balance means for counteracting the radial thrust, said means including radial balance pressure ports extending in the direction of the axes of the gears at points diametrically opposite the pump pressure port, and radial balance suction ports extending in the direction of the axes of the gears at points diametrically opposite the pump suction port, hydraulic means for hydraulically counterbalancing the axial gear thrust, said means including axial balance pressure ports in the casing at that face of the gear which tends to move toward the casing, and axial balance suction ports in the cas-

ing at that face of the gear which tends to move away from the casing, the axial balance pressure ports of one gear being on the same side of the casing as the axial balance suction ports of the other gear, the axial balance pressure ports being formed directly at the ends of the radial balance pressure ports and the pump pressure port, the axial balance suction ports being formed directly at the ends of the radial balance suction ports and the pump suction port, and appropriate passages for connecting the balance pressure ports to the pump pressure port, and for connecting the balance suction ports to the pump suction port.

15. A pump as defined in claim 14, in which the connecting passages are open channels formed in the end wall of the casing and extending diametrically of the gears, whereby the adjacent faces of the gears function to close the open side of the channels, the pressure channels being in one end wall and the suction channels being in the opposite end wall of the casing, there being an annular groove around the shafts of the gears where the aforesaid diametrical passages are interrupted by a shaft.

16. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including an axial balance pressure port in the casing at that face of each gear which tends to move toward the casing, an axial balance suction port in the casing at that face of each gear which tends to move away from the casing, whereby each gear is subjected to both pressure and suction forces operating in the same direction, the axial balance pressure port for one of the gears being on the same side of the casing as the axial balance suction port for the other gear, passages freely connecting the pump pressure port to the axial balance pressure ports, and passages freely connecting the pump suction port to the axial balance suction ports, the pump balance being maintained substantially independently of the pressure and suction heads at which the pump is operated, and the suction and pressure ports being so interchangeably arranged in the casing that the pump is reversible.

17. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including two axial balance ports of circumferentially limited area in the casing near the peripheries of the gears and opening onto that face of each gear which tends to move toward the casing, the two axial balance ports for each gear being recessed into the casing wall and being diametrically opposite one another, with a first one of said balance ports being located at and communicating directly with the pressure pumping port, and the second of said balance ports communicating with the first, the two axial balance ports for one of the gears being on the opposite side of the casing from the two axial balance ports for the other gear, and passages freely connecting the pump pressure port to the axial balance pressure ports.

18. A gear pump comprising a driving gear, a driven gear, a casing surrounding said gears and providing suction and pressure pumping ports leading to said gears, said gears being single helical gears resulting in axial thrust, and means to counterbalance said axial thrust, said means including two axial balance pressure ports in the casing at that face of each gear which tends to move toward the casing, and two axial balance suction ports in the casing at that face of each gear which tends to move away from the casing, whereby each gear is subjected to both pressure and suction forces operating in the same direction, the two axial balance ports on each side of each gear being diametrically opposite one another, the axial balance pressure ports for one of the gears being on the same side of the casing as the axial balance suction ports for the other gear, passages freely connecting the pump pressure port to the axial balance pressure ports, and passages freely connecting the pump suction port to the axial balance suction ports, the pump balance being maintained substantially independently of the pressure and suction heads at which the pump is operated, and the suction and pressure ports being so interchangeably arranged in the casing that the pump is reversible.

19. A gear pump comprising driving and driven gears, a casing surrounding said gears and including suction and pressure pumping ports leading to said gears, the fluid in said ports producing a radial thrust on said gears, said gears being single helical gears which develop an axial thrust, hydraulic balance means for counteracting the radial thrust, said means including radial balance ports extending generally in the direction of the axes of the gears at points diametrically opposite the pressure pumping port, and hydraulic balance means for hydraulically counterbalancing the axial gear thrust, said means including two axial balance ports of circumferentially limited area in the casing opening onto that face of each gear which tends to move toward the casing, the axial balance ports of one gear being on the opposite side of the casing from the axial balance ports of the other gear, said axial balance ports being located so near the peripheries of the gears that they open onto the tooth ends of the gears, the two axial balance ports on one side of a gear being diametrically opposite one another, with one of them located at and communicating directly with the pump pressure port, and the other of said axial balance ports being located at and communicating directly with one end of a radial balance port, and appropriate passages for freely connecting the balance ports to one another and the pump pressure port.

20. A gear pump comprising driving and driven gears, a casing surrounding said gears and including suction and pressure pumping ports leading to said gears, the fluid in said ports producing a radial thrust on said gears, said gears being single helical gears which develop an axial thrust, hydraulic balance means for counteracting the radial thrust, said means including radial balance pressure ports extending generally in the direction of the axes of the gears at points diametrically opposite the pump pressure port, and radial balance suction ports extending generally in the direction of the axes of the gears at points diametrically opposite the pump suction port, hydraulic means for hydraulically counterbalancing the axial gear thrust, said means including two axial balance pressure ports in the casing at that face of each gear which tends to move toward the casing, and two axial balance suction ports in the casing at that face of each gear which tends

to move away from the casing, whereby each gear is subjected to both pressure and suction forces operating in the same direction, the two axial balance ports on each side of each gear being diametrically opposite one another, with one of said two axial balance ports located at and communicating directly with a pumping port, and the other of said two axial balance ports being located at and communicating directly with one end of a radial balance port, the axial balance pressure ports of one gear being on the same side of the casing as the axial balance suction ports of the other gear, and appropriate passages for freely connecting the balance pressure ports to one another and the pump pressure port, and for freely connecting the balance suction ports to one another and the pump suction port.

GUSTAVE A. UNGAR.