

Oct. 25, 1966

R. O. GORDON
GEAR PUMP OR MOTOR

3,280,756

Filed Dec. 21, 1964.

4 Sheets-Sheet 1

FIG. 1

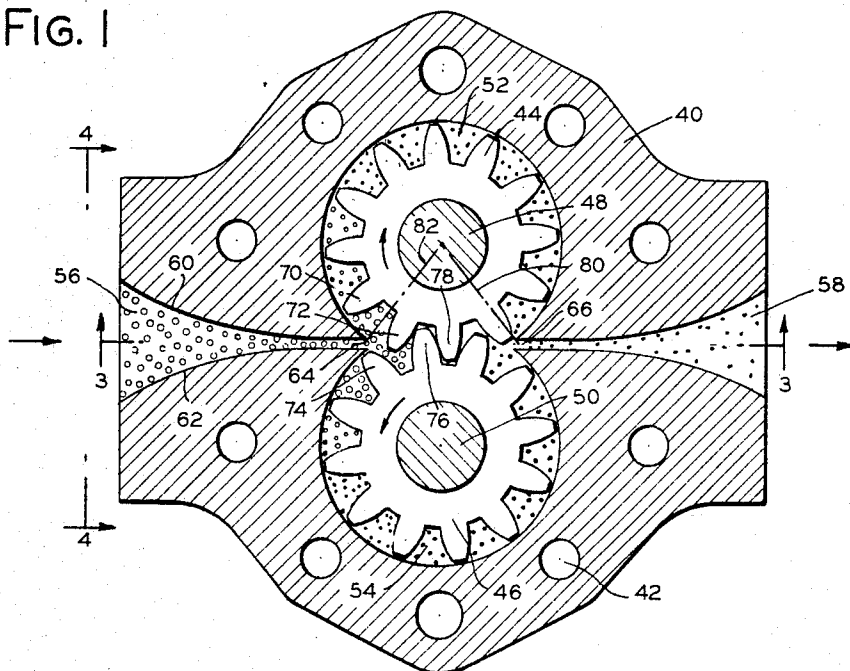
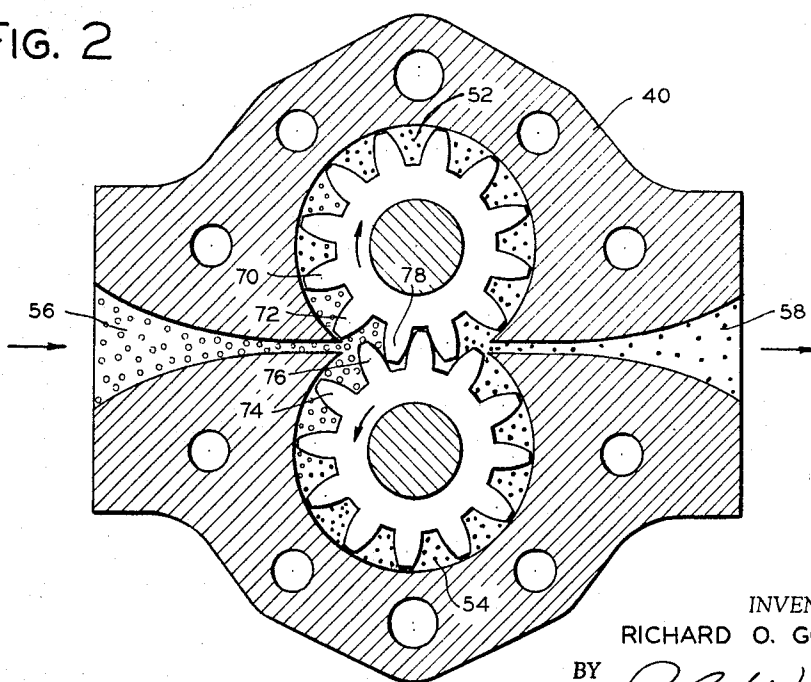


FIG. 2



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FIG. 3

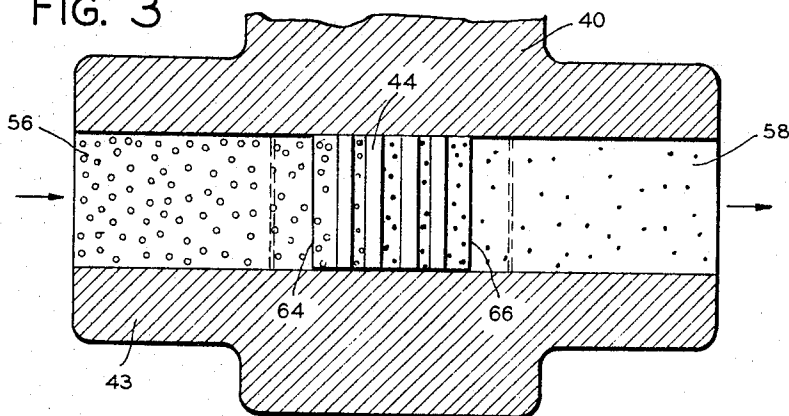


FIG. 4

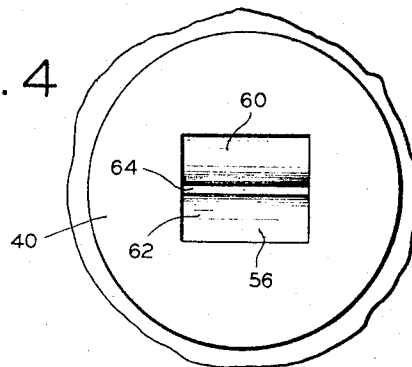
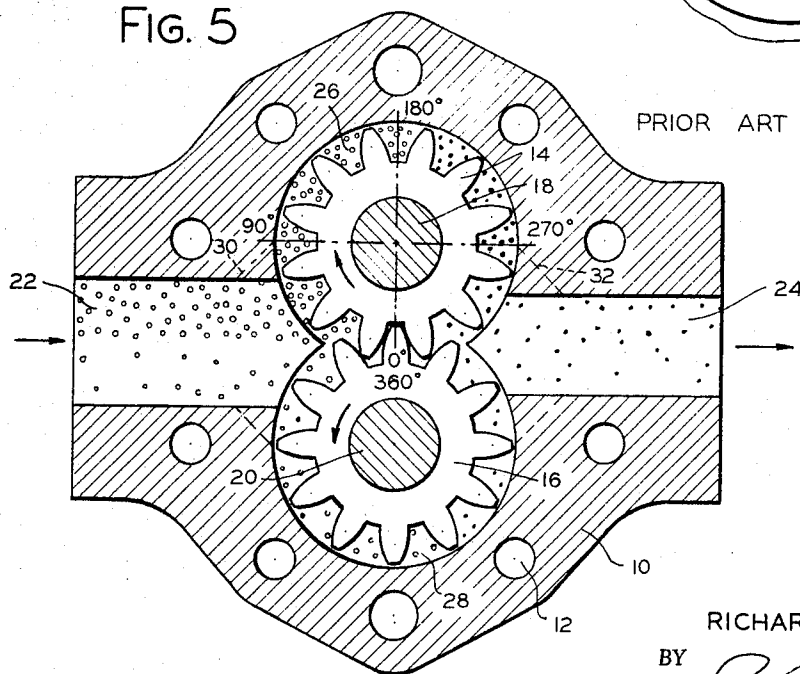


FIG. 5



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FIG. 6

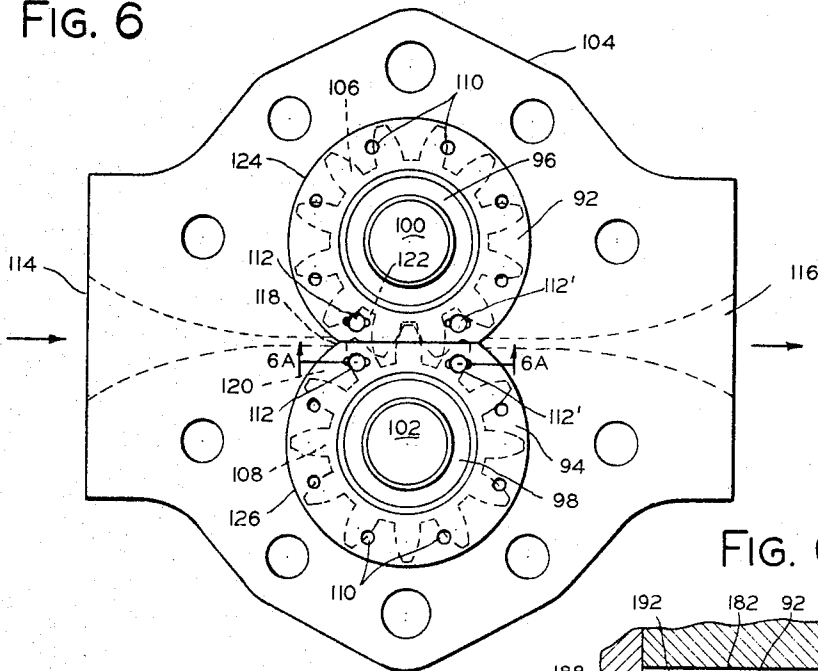


FIG. 6A

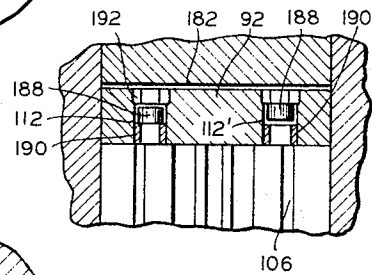
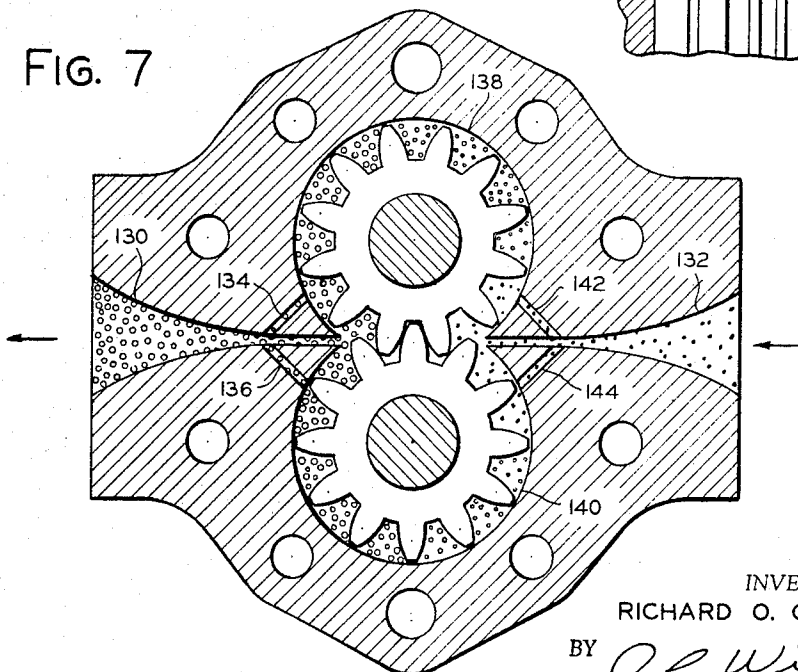


FIG. 7



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FIG. 8

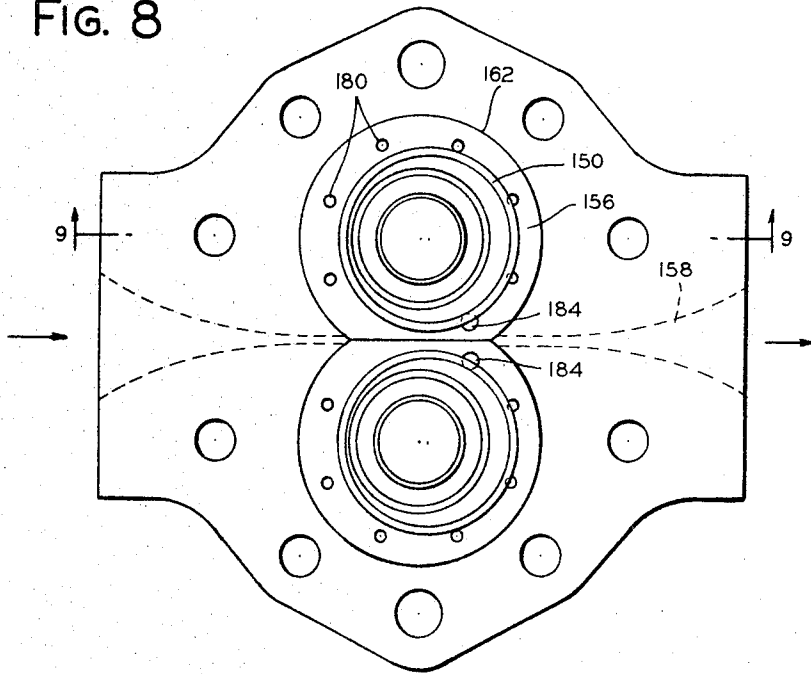
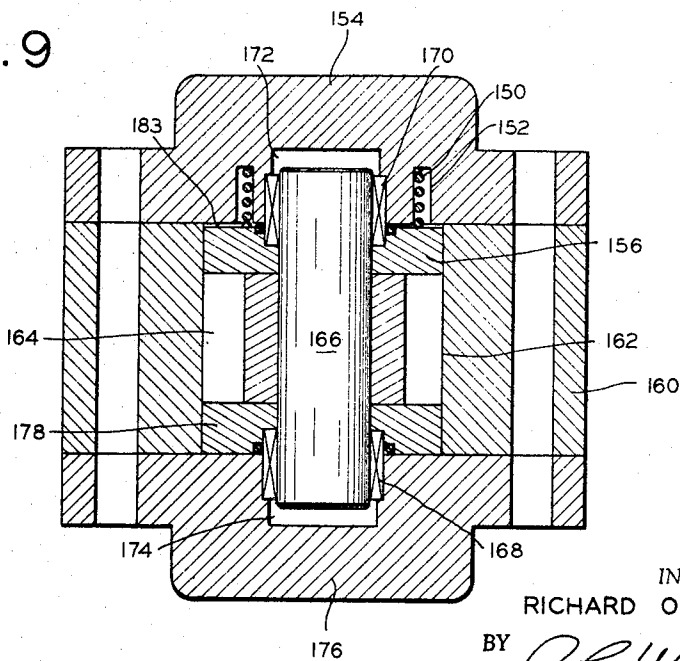


FIG. 9



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3,280,756

GEAR PUMP OR MOTOR

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Filed Dec. 21, 1964, Ser. No. 419,753

12 Claims. (Cl. 103-126)

This invention relates to fluid displacement pumps and motors, and more particularly to gear type pumps or motors.

It has long been a problem in the art to which this invention pertains to design gear pumps capable of generating relatively high discharge pressures and operable at high mechanical and volumetric efficiencies over an extended period of time with minimum maintenance requirements. Although gear pumps are of relatively simple basic construction, many difficult problems in design, construction and operation have been and continue to be encountered in the art. Relatively large expenditures have been and continue to be invested in research and development pertaining to gear pumps, despite the relatively advanced state of the art. One major problem which has consistently plagued the industry concerns the pressure gradient encountered in operation of pressure-loaded gear pumps which tends to cause the pump and the pump pressure sealing means, such as a thrust plate, to operate in an unbalanced condition with resulting short pump life, high maintenance costs, and reduced efficiency. Another serious problem encountered in the art concerns excessive bearing loads encountered in operation, particularly in high pressure gear pumps, which result primarily from unbalanced radial loads encountered as a result of the high pressure which exists around the discharge side of the pump with little counteracting force on the low pressure inlet side of the pump to balance the bearing load. Other exemplary problems encountered have concerned such things as excessive turbulence of fluid entering the pump on the inlet side, the inability of many such pumps to operate with acceptable efficiency and longevity in fluid systems in which a relatively large amount of air is present, and cavitation in the pump on the inlet side thereof.

Numerous possible solution have been proposed in the past for the above and other problems, with substantial success in some instances; frequently, however, solutions have required relatively complex structural changes, with consequent higher cost, and thus have to some extent compromised the basic simplicity of such pumps which is one of the primary advantages thereof.

One of the most recent advances in the art to which this invention relates is concerned primarily with a solution to the pressure gradient problem mentioned above, and is disclosed and claimed in my Patent Number 3,137,238, issued June 16, 1964. My present invention will operate to advantage with a pump of the type disclosed in my said patent, or in gear pumps and motors of less refinement than is therein disclosed, such as of the fixed clearance type wherein no pressure sealing or thrust plates are utilized.

Heretofore the art to which this invention relates has progressed on a basic erroneous assumption, viz, that a relatively large inlet area, as well as a relatively large outlet area, in the housing of the pump leading to and from the intermeshing gears is a requirement for efficient operation. Some manufacturers have gone to considerable lengths to provide outwardly flared inlet conduits adjacent the inlet side of the pump for minimizing the inlet velocity of the liquid entering the gears so that, it has been thought, turbulence of the liquid is minimized upon entry in the gear tooth pockets and erosion of the gears and other internal parts is avoided. As a matter of fact, in my above-mentioned patent is shown typical

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inlet and discharge conduit design of relatively large dimension which has been rather consistently used heretofore in the art. Another example of typical prior practice is disclosed in Newmeier Patent No. 2,781,730, issued February 19, 1957, wherein large inlet and discharge conduits are provided in addition to additional inlet area portions formed in the thrust plates for directing the liquid around the sides of the gears and into the expanding gear pockets on the inlet side during pumping. The prior teaching of the gear pump art has been consistently contrary to my discovery that the use of a small area nozzle or venturi leading to the inlet of the pump in communication with the gear teeth pockets as they open into the inlet section of the pump housing effects significant improvements in operation of such pumps and solves or significantly alleviates all of the aforementioned problems heretofore encountered.

In conventional gear pump inlet conduit design the velocity of the entering fluid is normally a value which is only a fraction of the tip velocity of the gear teeth. Frequently, a substantial amount of air is entrained in the fluid being pumped. The relatively low ratio of entering velocity to gear tip velocity creates a condition of substantial turbulence at the inlet of the pump, which effects a reduction in volumetric efficiency and tends, over a period of time, to raise the temperature of the fluid significantly. Also, as is well known, gear pumps are installed in hydraulic systems in a variety of attitudes relative to a horizontal plane, and even when installed in a horizontal plane in a mobile system, such as is encountered in vehicular equipment, will assume various attitudes relative to a horizontal plane when such equipment is operated on rough terrain, such as in the earth moving construction or machinery field. I have found that installation or use of such pumps in a location in which one gear is higher in elevation than the other, the extreme situation being when the gears are in a vertical plane, effects a gross misdistribution as between the gears of air which may be entrained in the liquid during operation. Entrained air entering the pump at a low velocity relative to the tip speed of the gears will tend to rise through a given liquid at a velocity substantially greater than the velocity at which the liquid enters the pump, and, when the gears are in a vertical plane, for example, substantially all of the air tends to be transported to the discharge conduit through the upper gear. The lower gear, therefore, tends to pump almost solid oil while the upper gear pumps a mixture of oil and air. The greater the quantity of air in the system the more serious the problem becomes.

The problem of pumping air with liquid in gear pumps is discussed in detail in my above-mentioned patent. In this situation the upper thrust plate and gear, in a pressure-loaded type pump, tend to show serious wear and scoring long before the occurrence of such wear at the lower gear and thrust plate, due to the relatively large pressure gradient across the gear face. In addition, in pumps having a large area inlet conduit, significant pressure is not produced until the liquid is transported a substantial distance around the periphery of the gears, which results in a seriously unbalanced radial loading condition at the bearings which support the gear shafts, due to discharge pressure fluid acting on each gear radially of the bearing on the discharge side of the pump which is countered on the inlet side of the pump only by substantially atmospheric pressure fluid. Furthermore, the upper bearing wears more rapidly than the lower bearing for the same reason as mentioned above in respect of the upper gear.

My invention tends to minimize the effect of all of the foregoing problems, and is readily applicable to gear pumps and motors of either the non-reversible or reversible

type, of either the fixed clearance type in which the gears are not pressure-loaded or of the pressure-loaded type, with or without pressure gradient or other controls.

In carrying out my invention in a preferred embodiment thereof I provide a gear pump or motor having a pair of intersecting bores in a housing for receiving intermeshing gears of the pump or motor, and inlet and discharge conduits in the housing connected to the inlet and discharge sides of the bores, respectively, the inlet conduit converging from a relatively large opening at one end of the housing to a relatively narrow slot at the intersection with the bores to provide preferably a streamline flow of entering fluid and to inject fluid alternately into the pockets of the gears at a relatively high velocity. The invention resides primarily in the concept of injecting fluid on the inlet side of gear pumps and motors through a relatively narrow slot and at a relatively high velocity.

It is therefore a primary object of the invention to provide improved pumps and motors of the types contemplated wherein fluid is received on the inlet side from a restricted passageway and at a substantial velocity.

Another object of the invention is to provide in gear pumps and motors both restricted inlet and discharge conduits providing relatively high velocity streamline fluid flow both to and from the pump or motor.

A further object of the invention is to generally improve and simplify the construction of gear pumps and motors of either the pressure-loaded type or the fixed clearance type, and whether reversible or non-reversible in operation.

Another objection of the invention is to provide a gear pump or motor construction which minimizes unbalanced bearing loads.

Another object of the invention is to provide an improved gear pump or motor of the pressure-loaded type which operates under a substantially balanced condition of loading of the thrust plate without requiring special provision for pressure gradient control.

Other objects of the invention include the minimizing of turbulence in the fluid entering and leaving the gear chambers, provision for approximating an equal distribution of entrained air as between the gears of the pump or motor and an equal distribution of oil under conditions of pump cavitation, and providing greatly simplified construction of gear pumps and motors.

Other objects, features and advantages of the present invention will become apparent from the following detailed description taken in conjunction with the drawings wherein:

FIGURES 1 and 2 are schematized longitudinal sectional views of a gear pump which embodies my invention;

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

FIGURE 4 is a partial sectional view taken along line 4—4 of FIG. 1;

FIGURE 5 is a schematized view which illustrates a typical prior art construction;

FIGURE 6 is a longitudinal sectional view of another embodiment of the structures shown in FIGS. 1 and 2;

FIGURE 6A is a partial sectional view taken along line 6A—6A of FIG. 6.

FIGURE 7 is a longitudinal sectional view of another modification of the structures shown in FIGS. 1 and 2;

FIGURE 8 is a longitudinal sectional view of still another modification of the pump construction; and

FIGURE 9 is a sectional view taken along line 9—9 of FIG. 8.

Referring now in detail to the figures, and first to FIG. 5, I have illustrated a typical schematized prior art gear pump construction in longitudinal section which, to operate relatively efficiently, utilizes pressure-loaded thrust plates, not shown, to seal the side faces of the gears in well-known manner, such as is illustrated in the above-mentioned Newmeier patent. Illustration of the thrust plates is not re-

quired to explain problems inherent in such prior art constructions, which, as related to the present invention, are primarily concerned with the design of inlet and discharge fluid conduits associated with the pump.

A suitably chambered gear pump housing is denoted at numeral 10 having a series of openings 12 for receiving bolts which are adapted to hold housing and cover portions of the pump in a mating relation. Within housing 10 are rotatably mounted driven and driving gears 14 and 16 mounted upon a driven shaft 18 and a drive shaft 20 which are adapted to be journaled in bearing means, not shown, in a well-known manner. Suitable seals, not shown, are provided to prevent leakage of pressure fluid from the pump. Pressure responsive thrust plate means for sealing the side faces of the gears may be of a type such as is disclosed in detail in my aforementioned Patent No. 3,137,238, or any other suitable construction. A conventional inlet conduit 22 is provided in the housing and communicates upper and lower gear chambers 26 and 28 with a fluid source, such as a sump, not shown, connected to inlet conduit 22, and a fluid motor means or other work producing device is adapted to be connected to high pressure discharge conduit 24. The quadrant markings in FIG. 5 denote degrees of rotation of gear 14 from inlet to outlet in clockwise direction from the zero degree position indicated, which is the location at which the gear teeth are fully in mesh on the vertical center line along the vertical axis of the pump, as shown, which location is herein referred to as the meshing axis.

It has been conventional practice heretofore to provide a relatively large area of opening at the intersection of both the inlet and discharge conduits with the gear chambers. Conventionally, inlet areas may vary in pumps of different design such that the gear teeth rotate an amount which may vary from 50° to 90° in different pumps prior to mating of the gear teeth with the gear chamber wall to form a substantially closed gear pocket. It has been considered advantageous heretofore to provide a relatively large opening at the intersection of said conduits with the gear chambers so that fluid enters the gear pockets as a relatively low velocity, which has been deemed to be important to the efficient operation of gear pumps. In the FIG. 5 representation of the prior art, each gear tooth rotates approximately 72° from the zero degree position prior to mating with the wall of gear chamber 26, and approximately 67° on the discharge side from the position of movement into the discharge opening to the 360° position of tooth rotation. Dotted lines 30 on the inlet side and 32 on the discharge side represent flared openings which may alternatively provide for approximately 90° of gear tooth rotation on both the inlet and discharge sides prior to the occurrence of mating relation with the wall of the respective gear chambers. Such flared openings are provided in some conventional pumps in order to further reduce the velocity of fluid flow to and from the gear chambers.

In actual tests of such pumps, as mentioned hereinabove, I have found that most of the air which may be entrained in the fluid being pumped rises in the inlet passage during entrance to the gear chambers and passes through the upper gear in situations in which the pump is located in an upright position, as shown in FIG. 5, or in a semi-upright position. In tests of 21 gallons per minute (g.p.m.) capacity pumps provided with transparent covers, for example, I have found that a gear speed of 2000 revolutions per minute (r.p.m.) the gear tip velocity is about 21 feet per second (f.p.s.), and it is conventional to design the inlet and discharge conduits of such pumps so that the velocity of fluid entering the pump is approximately 4 to 6 f.p.s. and the velocity at the discharge of the pump is about 10 to 12 f.p.s. Using number 10 hydraulic oil at 160° F., entrained air bubbles rise through the oil at 12 f.p.s. If flared openings are used at the inlet of the pump, as represented at numeral 30, the velocity of entering oil is

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decreased to about 4 f.p.s. in the flared opening portion as the oil approaches the gear chamber. It will therefore be apparent that the entrained air bubbles tend to rise more rapidly than the velocity of the oil entering the pump, which results in misdistribution of entrained air as represented in FIG. 5 by the relatively large number of air bubbles entrained in the oil in gear chamber 26 as compared with gear chamber 28. In addition, there results a considerable amount of turbulence at the inlet of the pump as a result of the relatively high gear tooth velocity in relation to the entering fluid velocity. Also, as the fluid pressure and/or temperature increases, the air bubbles become smaller in size and tend to increase the proportion thereof of flowing with the oil through the upper gear chamber. Some of the other problems encountered in prior pump designs utilizing relatively large inlet and discharge conduits as aforesaid are enumerated hereinabove, one of the most serious of which concerns the pressure gradient encountered in operation of such pumps, particularly under conditions of operation in which air is entrained in the hydraulic system fluid, as well as the unbalanced radial forces which tend to overload the bearings which support shafts 18 and 20.

In FIGS. 1-4 there is illustrated one embodiment of the present invention in a fixed clearance pump wherein no pressure plates are utilized to seal the side faces of the gears, thereby decreasing significantly the cost and complexity of such pumps. Fixed clearance gear pumps have not been generally used since the advent of pressure-loaded gear pumps primarily for the reason that such pumps have heretofore operated at inherently low efficiency as a result of the running clearance provided between the sides of the gear chambers and the side faces of the gears providing a leakage passage from the discharge side of the pump to the inlet thereof. In use of my invention, however, it is feasible to utilize fixed clearance gear pumps in some applications wherein very high efficiency is not required for reasons which will become apparent.

A gear pump housing is illustrated at numeral 40 in which are provided a plurality of bolt opening 42 so that a cover plate 43 can be tightly secured to the housing 40 during assembly of the pump. Gears 44 and 46 are mounted upon shaft 48 and 50 which are supported in bearing means, not shown, in a manner similar to the mounting of the gears of FIG. 5 for rotation in gear chambers 52 and 54. A running clearance should be provided between the inner surfaces of the housing and cover plate which are contiguous to the gear chambers so that the gears can rotate at high speed without scoring or burning while operating with a close a clearance as feasible to improve efficiency. An inlet conduit is provided in the housing at numeral 56 and a discharge conduit at numeral 58. FIG. 4 illustrates a broken-away end view of inlet conduit 56 which is constructed to provide a substantially constant dimension transverse of the gears preferably equal substantially to the width of the gears while the conduit converges in a vertical direction, as shown, along walls 60 and 62 to provide a narrow transverse open slot 64 at the intersection of conduit 56 with gear chambers 52 and 54. Likewise, discharge conduit 58 forms a narrow transverse slot 66 at the intersection of the conduit with the gear chambers, which slot is preferably substantially equal to the width of the gears.

The major advantages of this invention result from the use of a narrow slot or other equivalent construction at the opening of the inlet conduit into the gear chambers, as will become more apparent as the description proceeds, and although I prefer the use of a discharge conduit construction similar to such inlet conduit construction, it should be clearly understood that the use of a small discharge opening at the gear chambers is not necessary to the practice of this invention which may provide a relatively large gear chamber opening at the discharge. It should also be understood that while FIGS. 1-4 illustrate

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walls 60 and 62 of the inlet conduit, as well as the corresponding walls of the discharge conduit, as being of converging unbroken smooth wall construction to provide streamline flow of fluid, such construction is not necessary to the practice of this invention. It may be found more feasible in practice, for example, to provide a step-like conduit construction to effect the necessary reduction in conduit cross-sectional area from the end of the housing to the slotted area 64 which communicates with the gear chambers, and it is not intended that the invention be restricted to streamline flow conduit construction. Likewise, in respect of the other exemplary embodiments of the invention disclosed hereinafter.

Inlet conduit 56 is constructed to direct hydraulic fluid with or without air entrained therein, to fluid injection slot 64 which is designed to inject relatively high velocity inlet fluid alternately into the gear pockets of the respective gears as the gear teeth move out of meshing relation from the zero degree position thereof to the positions illustrated in FIGS. 1 and 2. In FIG. 1 the pocket formed between gear teeth 70 and 72 has previously filled with inlet pressure oil while the gear pocket formed between gear teeth 74 and 76 is almost full of inlet pressure oil in the position shown. In FIG. 2 the pocket formed between gear teeth 70 and 72 has already closed within gear chamber 52 and the fluid pressure therein has increased substantially as a result of the cascading of high pressure fluid around the outer periphery of the gear through the clearance provided between the gear teeth and the wall of the gear chamber, whereas the gear pocket between gear teeth 74 and 76 is full of inlet pressure fluid and approaches a closed condition within gear chamber 54, while the gear pocket formed between gear teeth 72 and 78 is almost fully open and full of inlet pressure fluid injected at said relatively high velocity through slot 64. As exemplary only of my design, I have provided in one embodiment, such as shown in FIGS. 1-4, a cross-sectional area of slot 64 which provides for an oil velocity of 8 f.p.s. at the entrance to conduit 56 and an oil velocity of 21 f.p.s. at the slot 64 which latter velocity is equal to the tip speed of the gears in the aforementioned example. As represented in these figures, the gear pockets are filled alternately and the high speed fluid flow through the small slot 64 maintains a mixture of hydraulic fluid and air so that as between the pockets of the respective gears an approximately equal division of the entrained air necessarily is effected. Tests of this construction indicate that hydraulic fluid having entrained air is injected smoothly through the inlet slot 64 across the entire width of the gear at a velocity which is preferably approximately equal to the gear tip velocity to produce an impact pressure upon entrance of the fluid into the gear pocket into which fluid is being injected which prevents any tendency of the pump during operation at high speed to eject part of the entering fluid back into the inlet conduit as a result of the centrifugal force produced by the speed of rotation of the gears. This latter tendency, it should be pointed out, has been found to be a problem in some gear pumps, particularly those of relatively large capacity which have gears of relatively large diameter and consequent high gear tip velocity which, in pump constructions having prior conventional inlet conduit design, produce sufficient centrifugal force to actually eject a substantial portion of the entering fluid back into the inlet conduit. This result tends to starve the pump in operation thereby accentuating existing pressure gradient and bearing loading problems in pressure-loaded gear pumps inasmuch as the gear pockets may rotate as much as 180° or even more before being completely filled with pressure fluid. This problem is avoided in the use of the present invention.

In addition, my invention permits the use of higher pump speeds, and therefore greater output, since the impact pressure of the entering hydraulic fluid, as aforesaid, counteracts the centrifugal force which tends to eject

the fluid in the gear pockets. For example, I have found in tests of a particular pump design having a 21 g.p.m. capacity, that the gear pockets filled without substantial cavitation up to 3600 r.p.m. and a gear tip velocity of as much as 30 f.p.s. whereas the velocity of entering oil was about 21 f.p.s. Above 3600 r.p.m., it was found that excessive cavitation occurred. Gear pumps of similar design and capacity using prior conventional inlet conduit design encounter serious cavitation at a much lower r.p.m. because the oil tends to be ejected, as aforesaid, back into the inlet conduit at a much lower critical gear tip velocity than 30 f.p.s. In the above-mentioned tests of said 21 g.p.m. pump, satisfactory results were obtained in the use of various size inlet slots which were varied from $\frac{1}{16}$ " to $\frac{1}{4}$ " in width.

I prefer to use a design of discharge slot 66 which is somewhat smaller in area than the area of slot 64 so that if the velocity of the oil at slot 64 is 21 f.p.s., as aforesaid, the velocity of the oil entering slot 66 from the gear chambers is higher than said inlet velocity, such as 25 f.p.s., so that as each gear pocket rotates past slot 66 the oil is discharged positively and directly from the pocket into the outlet conduit.

To further exemplify the divergence in practice between the prior art and the present invention, it is noted that it is common practice in the manufacture of gear pumps to limit the vacuum at the pump inlet to about 4" of mercury, whereas I have operated pumps embodying the present invention with good results while employing a vacuum at the inlet of as high as 28" of mercury. The gears in pumps using this invention will fill at substantially higher r.p.m. and the pump will operate at much greater inlet vacuum than heretofore.

The pumping path of gear pumps may be defined as 360° of gear rotation minus the number of degrees of gear rotation in which the fluid being pumped is not trapped in the gear pockets between the gear teeth and the wall of the gear chamber on the inlet side of the pump. In FIG. 5, for example, the inlet conduit 22 is designed such that the gears rotate approximately 73° from the meshing axis of the gears before the fluid is trapped in the gear pockets between the gear teeth and the chamber wall, so that the pumping path is about 287°. If, as is common practice, the inlet and discharge conduits are flared outwardly into the gear chambers, as indicated by dotted lines at numerals 30 and 32, then the pumping path, as above-defined, is only 270°. The present invention, on the other hand, utilizes a much longer pumping path, as illustrated in FIG. 1, wherein the broken lines 80 and 82 each forms with a vertical line at the zero degree position or meshing axis of the gears an angle of approximately 38°. The pumping path is, therefore, approximately 322°. Depending upon pump requirements and design, I have found that the pumping path utilized in my invention may vary substantially, preferably such that each gear rotates from a zero degree position through a range of about 30° to 40° prior to beginning the pumping path. As any given pump design is varied to meet particular operating requirements, the pumping path utilized may vary as required, provided, however, that a relatively high inlet velocity and inlet vacuum results so as to provide fluid injection at the inlet with a conversion of velocity pressure to impact pressure, as previously discussed.

It should be understood that all values stated herein, such as the dimensions of parts, volume capacity r.p.m., velocities, temperatures, angles degrees, and the like, are exemplary only and are for the purpose of clarifying the invention in terms of specific results in relation to a particular pump design; I do not intend that this invention be limited in any sense or manner by such specific examples as are stated herein.

It will now be apparent to persons skilled in the art that fixed clearance pumps utilizing this invention may be designed to operate in a relatively efficient manner, and

provide a practical, inexpensive and extremely simple pump construction with minimum maintenance requirements. Such pumps may be used in many applications wherein high speeds, pressures, volumes and efficiencies are not required.

Referring now to FIG. 6, there is illustrated schematically a pressure-loaded type gear pump utilizing upper and lower pressure responsive thrust plates 92 and 94 having annular recesses 96 and 98, respectively, which are adapted to seat upon suitable bearings supporting the drive and driven shaft 100 and 102 in the pump housing 104, all in a well-known manner. The thrust plates are preferably of the so-called "floating" construction, mounted in chambers in the housing and subjected to pumping pressure on the sides remote from side faces of the gears 106 and 108 for sealing the gears from leakage to the inlet of the pump, as occurs in some degree in fixed clearance pumps as described above by virtue of the running clearance between the side faces of the gears and the contiguous housing walls.

Pressure-loaded gear pumps of many different designs have been manufactured and proposed heretofore, but it has always been deemed essential to provide a masked-off area on the inlet side of the pressurized side of the thrust plate adjacent the inlet pressure side of the pump because of the pressure gradient which is always present between the inlet and discharge sides of the pump and which will, among other things, cause the plates and gears adjacent the inlet side of the pump to burn and score because of the excessive differential pressure which is applied to the thrust plates adjacent the inlet unless the area is sealed from discharge pressure fluid which is normally applied to the unmasked portions of the thrust plates. The nature of the pressure gradient problem which has plagued the art for many years is discussed in detail in my aforementioned Patent No. 3,137,238, and need not be repeated herein. For the purpose of illustration of one preferred embodiment of this invention utilizing thrust plates, as aforesaid, it is my intention that the pump structure illustrated in FIG. 6 be generally similar to the embodiment of the gear pump of my aforementioned patent particularly in respect of the thrust plates which are generally similar to that shown in FIGS. 5 and 6 of said patent, including the location of pressure gradient openings 110 herein which are spaced circumferentially of each of thrust plates 92 and 94, as shown, from a location near the inlet to a location near the outlet of the pump. The particular location, size and spacing of pressure gradient openings 110 in each thrust plate may be varied to suit requirements, so long as the openings provided communicate with the gear pockets, i.e., are located in the thrust plates circumferentially intermediate the root and tip diameters of the gear teeth. A discharge pressure opening 112' is provided in each thrust plate for communicating discharge pressure fluid to the pressure chambers on the top of the thrust plates and producing in the chamber, in cooperation with gradient openings 110, a variable pressure chamber condition for maintaining a suitable sealing of the side faces of the gears by the thrust plates without excessive force, also described in detail in my aforementioned patent. Many other thrust plate constructions can be utilized with this invention, the above generally described construction being preferred, although merely exemplary.

The construction of FIG. 6 hereof is readily adaptable for use as a reversible gear pump merely by providing an opening 112 in each thrust plate on the inlet side of the pump similar to opening 112' on the discharge side and providing check valves in the said openings 112 and 112' such that when fluid is flowing in the direction illustrated in FIG. 6, check valves in openings 112' are opened under discharge pressure to supply gradient openings 110 while the check valves in openings 112 remain closed, whereas with a reversal in said flow the check valves in

openings 112' close and check valves in openings 112 open to supply discharge pressure fluid as aforesaid to the gradient openings 110.

As shown in FIG. 6A, each of openings 112 and 112' house a floating check valve member 188, a ring member 190 on which the check valves are adapted to seat and close the opening therein, and a ring member 192 having a slot therein elongated to extend beyond the periphery of check valve 188. Thus, with fluid flowing in the direction illustrated in FIG. 6, opening 112' receives discharge pressure which unseats valve 188 from ring member 190 and seats the valve on ring member 192, thus permitting pressure fluid to flow into pressure chamber 182 which, in conjunction with pressure gradient openings 110 provide the required compensation for gradient variations, while also causing check valve 188 in opening 112 to seat upon ring 190, thus preventing flow of pressure fluid to the pump inlet. If it is desired to operate the pump in a reverse direction, conduit 116 becomes the inlet, and opening 112 is then subjected to the discharge pressure which reverses the flow to the check valves 188 so that discharge pressure fluid flows through opening 112 into pressure chamber 182 and closes check valve 188 in opening 112'. In order that the reversible pump operates with the same characteristics in either direction, it is, of course, important that the pressure gradient openings 110 be symmetrically disposed in respect of the horizontal and vertical axes of the gear with which the thrust plate is associated, and that the correspondingly located openings on opposite sides of the vertical axis be of the same area.

As shown in FIG. 6, inlet and discharge conduits 114 and 116, respectively, may be similar in configuration to conduits 56 and 58 in FIG. 1. I have found that in a properly designed pressure-loaded gear type pump utilizing my invention, the masking-off of an area on the inlet side of the pump can be readily dispensed with, and that when such pumps utilize pressure plates such as is disclosed in FIG. 6, pressure gradient compensation is automatic and proper sealing forces at the side faces of the gear result, thus providing a greatly simplified pump construction at reduced cost, while affording an efficient, high speed, high discharge pressure pump. These advantageous results are accomplished primarily by the use in such pumps of a narrow slotted inlet 118 so as to provide the functional advantages discussed hereinabove in respect of FIG. 1. I have found, for example, that in a pump having a design similar to that shown in FIG. 6 wherein the tip speed of the gears is about 21 f.p.s. and the inlet slot 118 is designed to produce a velocity of fluid of about 21 f.p.s., that the immediate filling of adjacent gear pockets 120 and 122 at the velocity pressure of the incoming fluid during the breaking of mesh of the corresponding gear teeth effects an instantaneous pressure rise in said gear pockets which is carried over by the fluid filled gear pockets into the pumping path defined by the walls 124 and 126 of the gear chambers. With pressure gradient openings 110 communicating with a modified discharge pressure fluid which enters the pressure chambers through openings 112 under pressure gradient conditions, such as may be caused by air entrained in the oil, as previously discussed, it will be apparent that relatively high pressure fluid will be present in the gear pockets throughout most of the pumping path, and that the area of thrust plate unbalance adjacent the inlet of the pump resulting from the low inlet pressure being carried into the pumping path by gear pockets 120 and 122, for example, will be minimized. In standard gear pump practice utilizing relatively large inlet openings, and particularly under conditions in which the pressure gradient may vary as a result of such things as varying in quantity the air entrained in the fluid pumped, relatively large variations in temperature of the fluid, and other factors are encountered, all as discussed in my aforementioned patent.

In pumps utilizing standard inlet conduit design, as

described above, there is no pumping action until the gear teeth and gear chamber wall close the pockets in the 70° to 90° range, as shown in FIG. 5, following which the fluid pressure increases as the gear rotates toward the outlet. If the pump is operating on solid oil, i.e., little or no entrained air, at normal temperature, discharge pressure will be present from about 135° of gear rotation clockwise to the 360° position in the outlet. Under extreme pressure gradient conditions resulting from, for example, substantial amounts of entrained air, the fluid may not reach discharge pressure until the gear has rotated as much as 270°, unless a pressure gradient control, such as disclosed in my aforementioned patent is utilized. The masked-off section normally utilized in thrust plates is determined empirically for each pump design, and is accurate for only one particular condition of pump operation, so that the effectiveness of the masked-off area to compensate for low pressure adjacent the inlet side varies substantially as pump operating conditions vary.

The present invention on the other hand, inherently provides a very small area of pressure unbalance adjacent the inlet of the pump which tends to generate a greater sealing force on the gear side faces than is required, since the pumping path in the FIG. 6 embodiment begins at a location of about 38° in advance of the meshing axis or zero degree position of the gear teeth. The narrow slotted high velocity fluid inlet 118, combined with a suitable pressure gradient compensated thrust plate construction, such as illustrated in FIG. 6, therefore minimizes the necessity for masked-off thrust plate areas adjacent the inlet in such pumps.

It may be found advantageous in some applications of my invention to provide for eccentric loading of the thrust plate to compensate for the relatively small pressure unbalance which is present at the inlet of the pump as above discussed. The pump illustrated in FIGS. 8 and 9 is similar to the embodiment of my invention disclosed in FIG. 6, except that a spring 150 is adapted to be retained in an annular groove 152 in a cover plate 154 of the pump for imposing on the pressure plate 156 an eccentric force in the direction of the discharge conduit 158. The pump body or housing is shown at numeral 160 having a gear chamber 162 and upper gear 164 contained therein mounted upon a shaft 166 which is rotatable in bearing 168 and 170 mounted in chambers 172 and 174 of cover plates 154 and 176. A fixed thrust plate is shown at numeral 178; the floating thrust plate 156 having gradient openings 180 therein is essentially the same in structure as the thrust plates described in FIG. 6. The pressure chamber which motivates the thrust plate 156 into sealing relation with the side face of gear 164 is shown at numeral 183, and it is from this chamber that pressure fluid communicates with the gear pockets through gradient openings 180. In addition, FIGS. 8 and 9 show a discharge pressure vented opening 184 in each of the floating thrust plates.

It will also be apparent that the bearing loads are significantly less than heretofore in pumps using this invention, inasmuch as the pumping path is substantially longer, and the fluid tends to reach significantly higher pressures than previously in the first 120° of gear rotation which produce fluid forces near the inlet slot 118 which tends to balance or counteract radial forces on the bearings resulting from the high pressures present on the discharge side of the pump, all as will be apparent to persons skilled in the art.

Referring now to FIG. 7, there is disclosed schematically a reversible hydraulic motor, which may be in all respects built in accordance with the teaching of either FIGS. 1 or 6 above, except that the inlet conduit 132 is provided with auxiliary passages 142 and 144 which connect the inlet conduit with gear chambers 138 and 140, respectively, and outlet conduit 130 is provided with auxiliary passages 134 and 136 which connect the outlet conduit with said gear chambers. Preferably, the aforementioned

tioned connecting passages between the inlet and outlet conduits and the gear chambers are formed at an angle of about 45° in the direction of the gear chambers from the longitudinal axis of the inlet and discharge conduits. The purpose of these connecting passages is to provide better pressure distribution in the motor and to provide for injection of pressure fluid from conduit 132 at an angle relative to the gear chambers so as to provide a velocity pressure acting on the gear teeth which tends to rotate the gears in the direction desired, thereby overcoming inertia of the gears either at rest or during increases in pump r.p.m. The passages 134 and 136 on the discharge side of the motor also provide an additional area for exhausting fluid from the motor. More importantly, utilization of passageways on both sides of the gear chambers permits the motor to be used as a dual-rotation motor wherein the velocity pressure is applicable in the direction of gear rotation whether fluid flows in the direction illustrated in FIG. 7, or in the opposite direction.

The multiple advantages of my invention will now be appreciated by persons skilled in the art, which include (1) approximately equal distribution of fluid entrained air as between the gears regardless of the position of mounting of the pump; (2) distribution of hydraulic fluid substantially equally to both gears when the pump cavitates; (3) inherent approximate hydraulic balance of the thrust plates in a pressure-loaded pump or motor with minimum area of unbalance adjacent the inlet such that masked-off areas are not required; (4) injection of relatively high velocity inlet fluid to provide initial impact pressure and build-up of pressure immediately upon closure of the gear pockets adjacent the circular wall of the gear chamber; (5) minimum turbulence adjacent both the inlet to and discharge from the gear chambers; (6) reduction of load on the bearings which support the gear shafts by maximizing the length of the pumping path; (7) simplification and lower cost of pump construction; (8) capability of pump operation at acceptable efficiencies for some applications in a fixed clearance type pump construction without use of thrust plates; (9) utilization in combination with the pressure gradient control means of my Patent No. 3,137,238 to effectuate some of the foregoing advantages, and, finally, (10) a greatly simplified and wholly adequate solution to the pressure gradient problem in gear pumps.

Although a few embodiments of my invention have been described herein, the disclosure hereof is primarily for the purpose of illustration and not as a limitation of the scope of the invention. It is therefore to be expressly understood that the invention is not limited to the specific embodiments shown, but that various modifications may be made to suit different requirements, and that other changes, substitutions, additions and omissions may be made in the construction, arrangement and manner of operation of the parts without necessarily departing from the scope of the invention as defined in the following claims.

I claim:

1. In a gear pump or motor, the combination of a pair of meshing rotary gears forming fluid containing pockets between the gear teeth, a housing including a perimetric wall surrounding the gears and forming a pump chamber in which the gears are adapted to rotate, said housing including inlet and outlet fluid conduits at opposite sides of the meshing axis of said gears, said inlet conduit providing a progressive reduction in flow area in the direction of fluid flowing therethrough and being formed at the exit thereof to provide an elongated dimension which extends transversely of the gears a distance which permits fluid flowing through said exit to communicate directly with substantially the entire adjacent projected area of the gear pocket and a dimension normal to said elongated dimension which together with said latter dimension forms an exit flow area of such restricted size as to permit fluid to be injected and substantially enclosed between each gear pocket and the perimetric wall following rotation of a gear tooth through not more than 44° from the axis of meshing

of the gears whereby to effect a relatively high velocity of pressure fluid flowing through said exit area which in combination with the resultant long pumping path, minimizes leakage of discharge pressure fluid to the inlet conduit, said perimetric wall forming with the tips of the gear teeth a peripheral or annular clearance for the gears comprising a flow path through which pressure fluid migrates during operation from the high pressure side of the pump chamber toward the low pressure side thereof to a location near said exit area of the inlet conduit for producing radially directed fluid pressure forces in the gear pockets near the said exit area such that the resultant radial forces on one side of the axis of gear meshing tend to counteract the resultant radial forces on the opposite side of said axis, said exit area of the inlet conduit being of such a size as to also effect a velocity of the fluid flowing therethrough which produces a fluid impact pressure at said exit area of such magnitude as to substantially counteract a tendency of the fluid in the pump chamber to be ejected therefrom into the inlet conduit as a result of the rotation of the gears therein.

2. A fluid pump or motor as claimed in claim 1 wherein axially movable thrust plate means having a sealing surface for sealing the adjacent side faces of said gears are located between the one side faces of the gears and the adjacent housing, the side of said thrust plate means remote from said sealing surface forming a portion of a motive pressure chamber means, passage means conducting pump generated pressure fluid to said pressure chamber means for moving said thrust plate means axially into sealing engagement with the said side faces of said gears, and a plurality of openings in said thrust plate means communicating said passage means and pressure chamber means with gear pockets on the sealing side of said thrust plate means for minimizing in conjunction with the long pumping path the pressure gradient in the pump chamber.

3. A fluid pump or motor as claimed in claim 2 wherein said plurality of openings are spaced circumferentially of said thrust plate in symmetrical relation thereto and in symmetrical relation as between the opposite sides of the axis of meshing of said gears, said passage means comprising a pair of passages adjacent the inlet and outlet conduits, respectively, and valve means in each of said passages operable in combination to provide a reversible pump or motor.

4. A gear motor as claimed in claim 1 wherein a pair of passages connect the inlet conduit to opposite portions of the motor chamber, each of said passages being connected to the inlet conduit at a location upstream of the exit area thereof and to each respective portion of the motor chamber at a location displaced from said exit area in the direction of rotation of the respective gears.

5. A gear motor as claimed in claim 4 wherein each of said passages communicates with the respective portion of the motor chamber at a location displaced not less than 40° and not more than 50° from the meshing axis of the gears.

6. A fluid pump or motor as claimed in claim 1 wherein the velocity of fluid flowing through said exit area is not substantially less than the rotational velocity of the tips of the gear teeth.

7. A gear pump or motor as claimed in claim 1 wherein said inlet conduit is of substantially rectangular cross-section and includes upper and lower converging substantially smooth walls from the inlet thereof to said exit area such that the exit area has a configuration of an elongated narrow rectangular slot.

8. A gear pump as claimed in claim 1 wherein the exit area of the inlet conduit may be of such restricted size as to produce a vacuum in the restricted portion of the inlet conduit of as high as 28" of mercury.

9. A gear pump or motor as claimed in claim 1 wherein the said normal dimension of said exit area for gear pumps and motors having a capacity of approximately 21

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gallons per minute is not more than $\frac{1}{4}$ ", and for pumps of different capacities said maximum dimension may be varied in proportion to the capacity of the pump.

10. A gear pump or motor as claimed in claim 1 wherein the configuration of said exit area of the inlet conduit is such as to effect an injection of relatively high velocity fluid alternately into the pockets formed between the teeth of the respective gears in such a manner as to cause an approximately equal division of any gas bubbles entrained in the fluid flowing as between the portions of the pump chamber which house the respective gears and irrespective of the position of the pump relative to a vertical plane.

11. A gear pump as claimed in claim 1 wherein the inlet area to the outlet conduit is of a configuration similar to the corresponding exit area of the inlet circuit.

12. In a gear pump or motor, the combination of a pair of meshing rotary gears forming fluid containing pockets between the gear teeth, a housing including a perimetric wall surrounding the gears and forming a pump chamber in which the gears are adapted to rotate, said housing including inlet and outlet fluid conduits at opposite sides of the meshing axis of said gears, said inlet conduit providing a progressive reduction in flow area in the direction of fluid flowing therethrough and being formed at the exit thereof to provide an elongated dimension which extends transversely of the gears a distance which permits fluid flowing through said exit to communicate directly with substantially the entire adjacent projected area of the gear pocket and a dimension normal to said elongated dimension which together with said latter dimension forms an exit flow area of such restricted size as to permit fluid to be injected and substantially enclosed between each gear pocket and the perimetric wall following rotation of a gear tooth through not more than 44° from the axis of meshing of the gears, whereby to effect a relatively high velocity of pressure fluid flowing through said exit area which is not substantially less than the rotational velocity of the tips of the gear teeth, and which in combination with the resultant long pumping path, minimizes leakage of discharge pressure fluid to the inlet conduit, the configuration of said exit area being also such as to effect an injection of relatively high velocity fluid alter-

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nately into the pockets formed between the teeth of the respective gears in such a manner as to cause an approximately equal division of any air bubbles entrained in the fluid flowing as between the portions of the pump chamber which house the respective gears and irrespective of the attitude or position of the pump, said perimetric wall forming with the tips of the gear teeth a peripheral or annular clearance for the gears comprising a flow path through which pressure fluid migrates during operation from the high pressure side of the pump chamber toward the low pressure side thereof to a location near said exit area of the inlet conduit for producing radially directed fluid pressure forces in the gear pockets near the said exit area such that the resultant radial forces on one side of the axis of gear meshing tend to counteract the resultant radial forces on the opposite side of said axis, said exit area of the inlet conduit being of such a size as to also effect a velocity of the fluid flowing therethrough which produces a fluid impact pressure at said exit area of such magnitude as to substantially counteract a tendency of the fluid in the pump chamber to be ejected therefrom into the inlet conduit as a result of the rotation of the gears therein, and said outlet fluid conduit having an inlet area of a configuration generally similar to the corresponding exit area of said inlet conduit.

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