United States Patent

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[54] **GEAR PUMP WITH LUBRICATING MEANS**

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[52] **U.S. Cl.** **418/102**

[51] **Int. Cl.** **F01c 21/04**

[58] **Field of Search** 418/102

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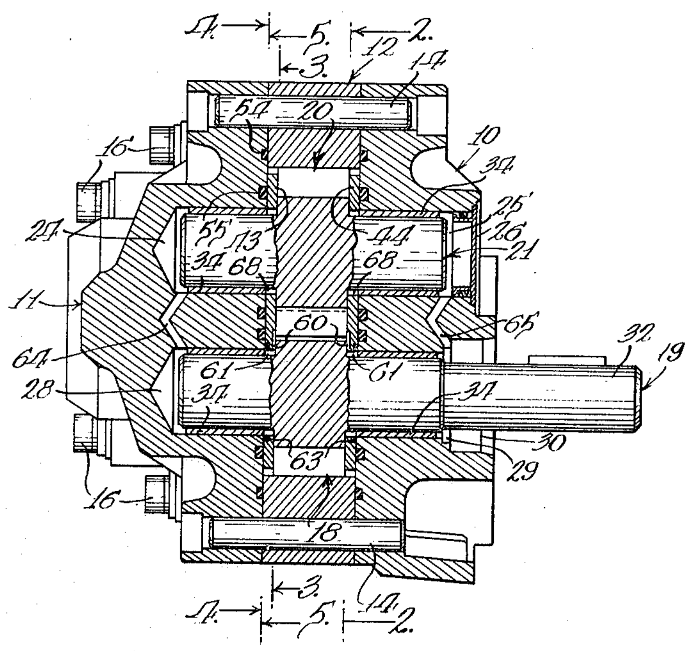
[57] **ABSTRACT**

A high pressure rotary spur gear pump including a pair of gear members intermeshing at a location between an inlet and an outlet, a pair of shafts carrying the gear members and having opposite ends mounted in bearing sleeves, together with means for pumping lubricating fluid under pressure from an area at the zone of intermeshing teeth, where the spaces between teeth are decreasing in volume, outwardly along the bearing sleeves at opposite ends of one shaft and back along the bearing sleeves at opposite ends of the other shaft, to the pump inlet.

4 Claims, 13 Drawing Figures



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3 Sheets-Sheet 1

Fig. 1.

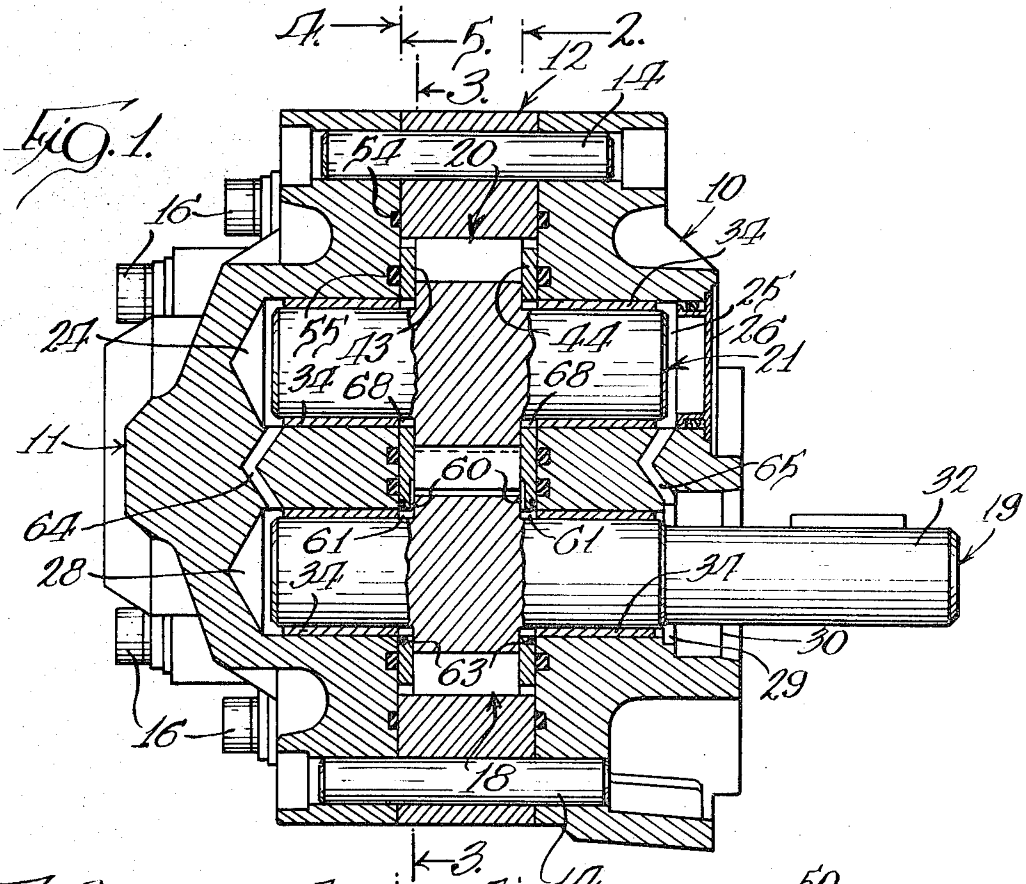


Fig. 2.

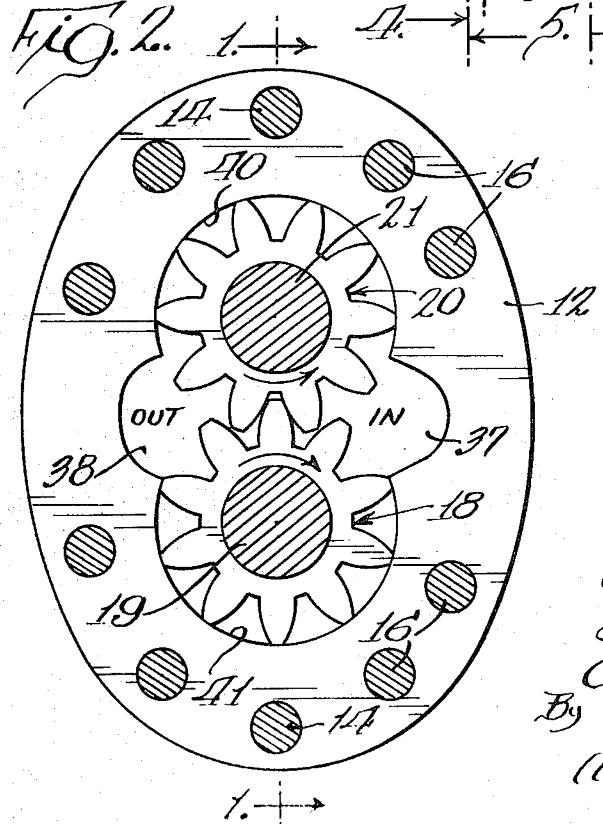
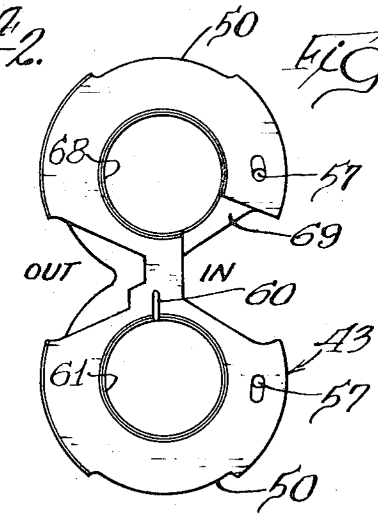


Fig. 3.



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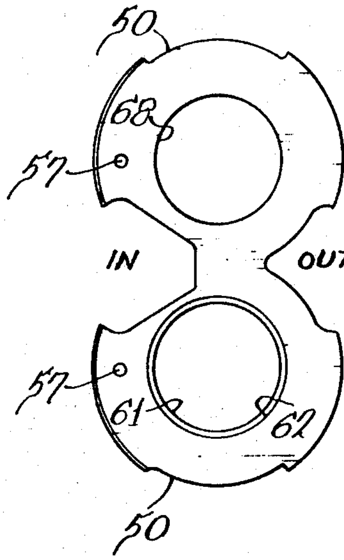

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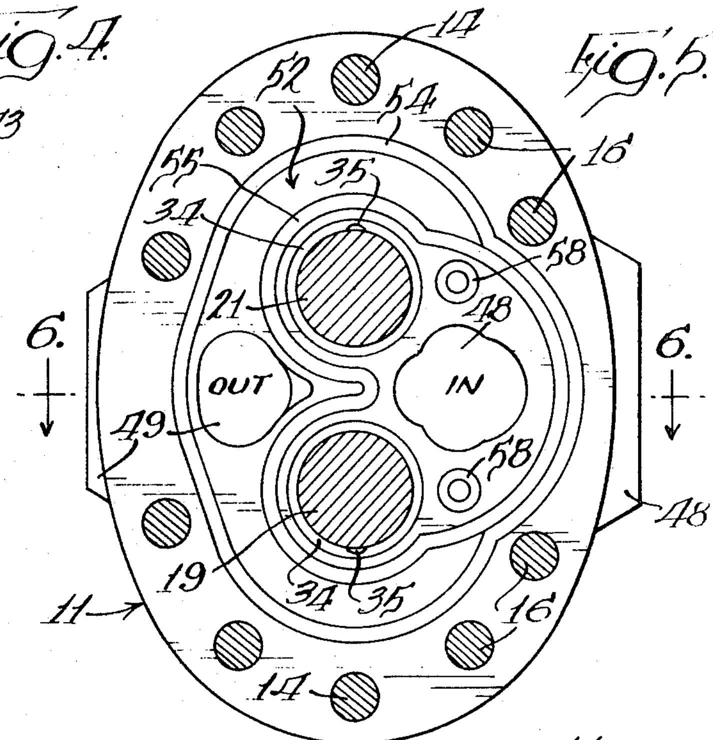


Fig. 5.

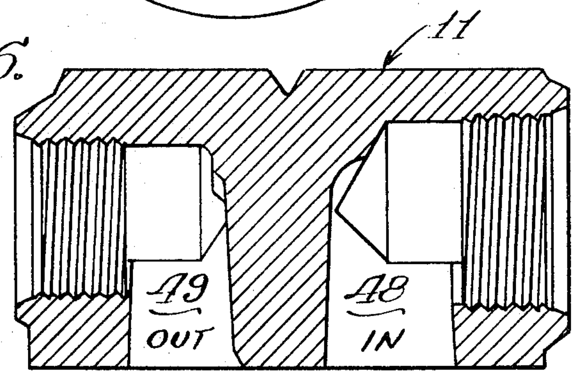


Fig. 6.

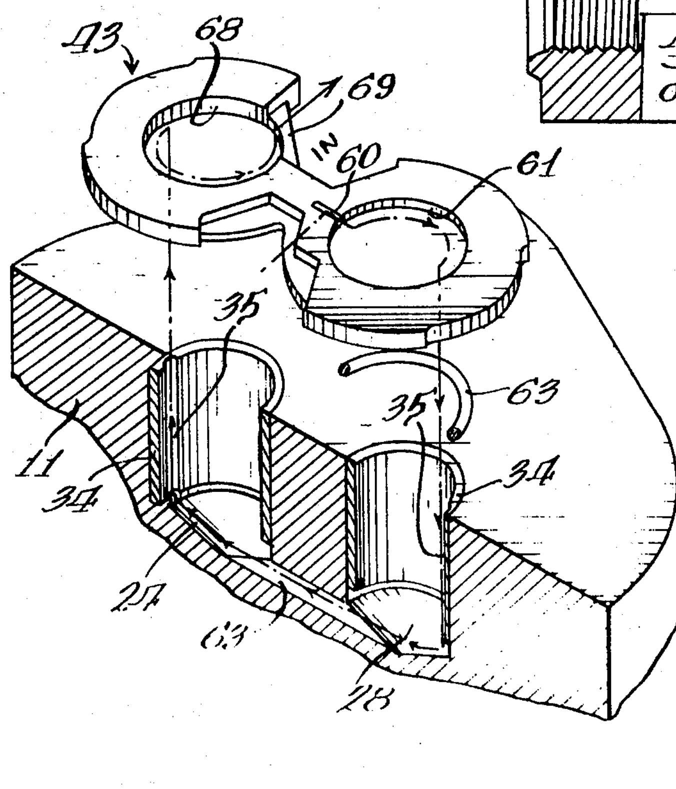


Fig. 7.

Fig. 8.

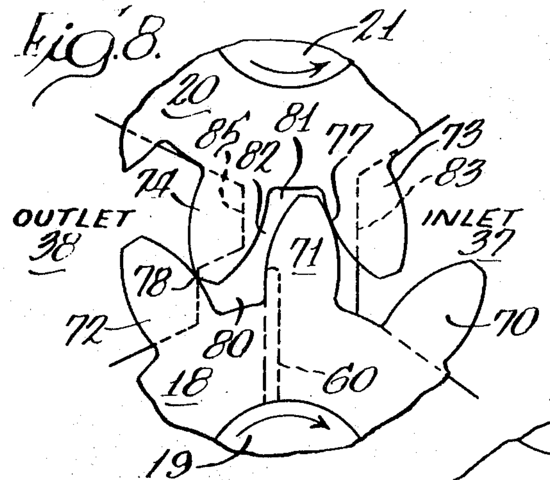


Fig. 9.

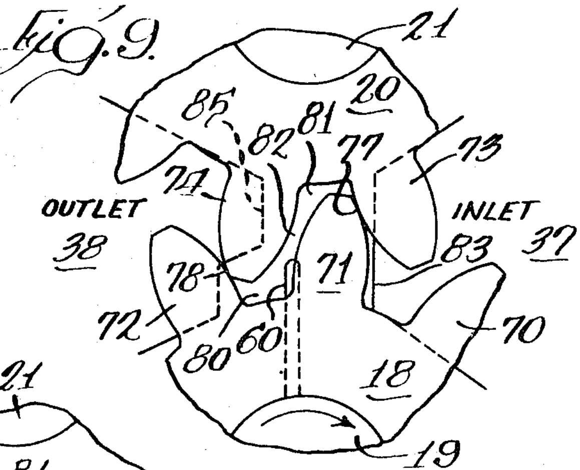


Fig. 10.

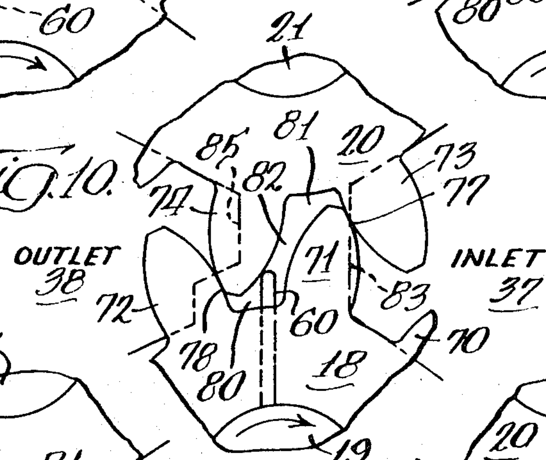


Fig. 11.

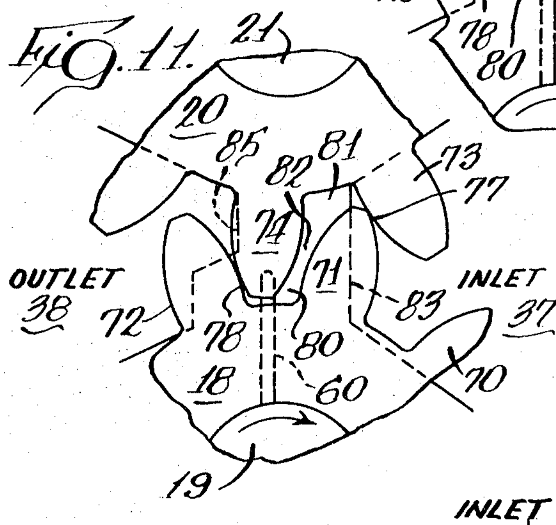


Fig. 12.

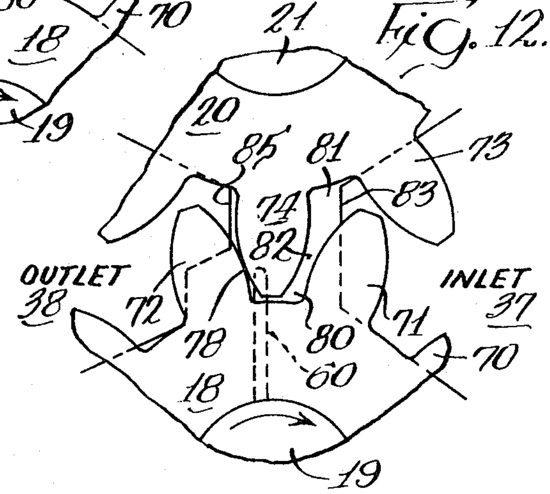
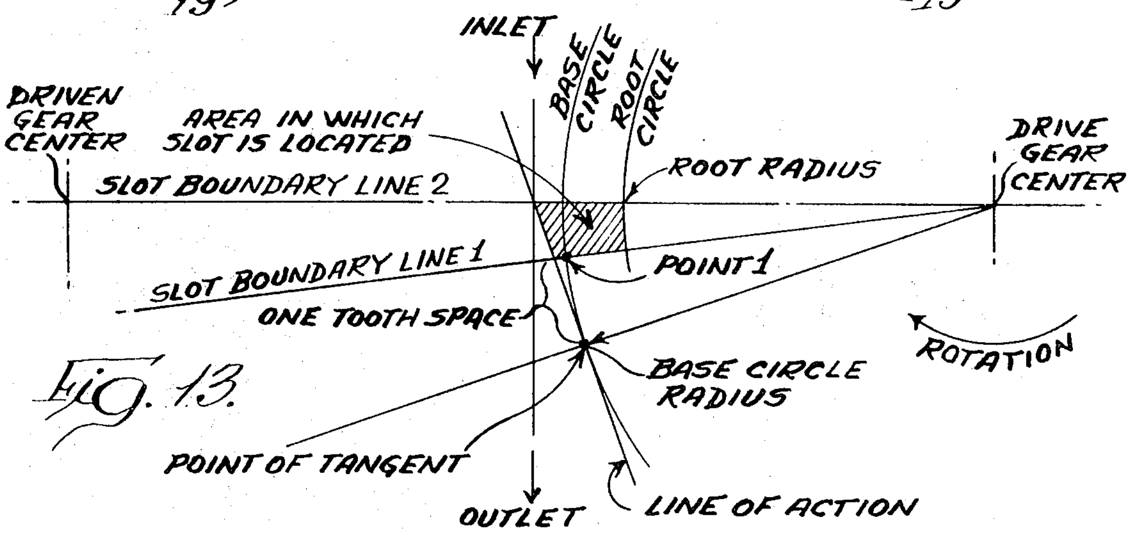


Fig. 13.



GEAR PUMP WITH LUBRICATING MEANS

BACKGROUND OF THE INVENTION

The present invention relates to rotary spur gear pumps including a pair of pump rotors or gear members with intermeshing teeth, and in the preferred construction illustrated herein, the gear members are supported by means of a pair of parallel shafts having the gear members fixed thereon and having opposite ends extending from the gear members into bearing sleeves which mount the shafts for rotation. Because the gear shafts rotate with the gears, and the sleeve bearings are stationary, it is necessary to provide lubrication at the engaged surfaces of the shafts and the bearings.

In the past, various systems have been contemplated for purposes of providing lubrication for the bearing surfaces providing for rotation of the gears, but the known systems have been subject to certain disadvantages. For example, shaft lubrication has been provided by static delivery of fluid to the bearing surfaces to be distributed by the adherence of lubricating fluid to the moving surface. Such a system lacks positive direction of the fluid in predetermined quantity.

In other systems, provision has been made for inducing flow of lubricating fluid as a result of suction in the vicinity of the inlet port. Still other arrangements have relied upon minor differences in pressure at different locations in the inlet chamber. Such approaches lack significant pressure differences and are believed to be less effective than positive distribution of lubricating fluid. Further, the systems are subject to variations in inlet conditions which tend to vary the quantity of lubricating fluid distributed.

Other apparatus has included the use of lubricating fluid directed from the pump outlet chamber under pressure, but under conditions which are understood to not consistently provide flow of predetermined quantities adequate for optimum lubrication of all bearing areas.

SUMMARY OF THE INVENTION

The present invention involves a lubrication system for a gear pump with parallel gear shafts wherein lubrication fluid is positively directed under pressure from a port under pressure at the zone of intermeshing teeth where the spaces between the teeth are decreasing in volume in a manner to force lubricating fluid in predetermined quantities to the bearing surfaces.

As illustrated, provision is made for pumping lubricating fluid from the outlet chamber outwardly along one shaft to the end thereof, laterally to the adjacent other shaft end, and back along the other shaft to the inlet chamber, as a result of which there is a predetermined significant pressure differential which positively insures flow of lubricating fluid in adequate quantities to the bearing areas of both shafts.

In the preferred construction illustrated, lubricating fluid is simultaneously directed outwardly along opposite ends of one shaft, laterally to the adjacent ends of the other shaft and back inwardly along the other shaft to the pump inlet chamber.

Preferably the lubricating fluid is pumped from the zone of intermeshing teeth at a location where spaces between the teeth are momentarily sealed by tooth contact from both the pump inlet chamber and the pump outlet chamber, trapping fluid under pressure, and forcing the trapped fluid in predetermined metered amounts to the bearing areas.

As shown herein, the specific passages for flow of lubricating fluid include a longitudinal groove along the inside of each bearing for distributing fluid to the interengaged bearing surfaces, an annular channel around each shaft at the inner end of the bearing groove, passages in the housing communicating outer ends of adjacent bearing grooves, a groove in each seal plate at opposite sides of the gears leading from the zone of intermeshing teeth to the annular channel on one shaft and a groove in each seal plate leading from the annular channels on the other shaft to the inlet chamber.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal sectional view through a pump embodying the present invention, taken at about the line 1—1 of FIG. 2;

FIG. 2 is a cross-sectional view taken at about the line 2—2 of FIG. 1;

FIG. 3 is an elevational view of a pressure balance seal plate taken at about the line 3—3 of FIG. 1, viewing the pressure balance plate from the side in contact with the gears;

FIG. 4 is an elevational view of the pressure balance plate taken at about the line 4—4 of FIG. 1, viewing the opposite side of the plate;

FIG. 5 is a cross-sectional view taken at about the line 5—5 of FIG. 1, viewing the inner surface of the end housing member;

FIG. 6 is a sectional view through the housing member taken at about the line 6—6 of FIG. 5 showing the inlet and outlet ports in the housing member;

FIG. 7 is a fragmentary exploded view with a diagrammatic illustration of the path of flow of lubricating fluid;

FIGS. 8–12 are enlarged fragmentary illustrations of the gear teeth at the zone of intermesh, with the teeth in various successive positions, demonstrating decreasing volumes in the spaces between teeth, resulting in positive displacement of lubricating fluid; and

FIG. 13 is a diagrammatic illustration of the manner in which the lubricating fluid pickup groove in the face of the seal plate is located.

DESCRIPTION OF PREFERRED EMBODIMENT

Referring now to the drawings in more detail, a pump construction embodying the present invention is illustrated as including a housing comprised of three outer members including an end housing member **10** suitably constructed for mounting the pump upon an appropriate support, an end housing member **11** formed with an inlet and an outlet, and a central housing member **12** in the form of a spacer plate. The housing members **10**, **11** and **12** are suitably aligned as by pins **14** at the top and bottom of FIGS. **1** and **2**, and held in assembled relationship by machine bolts as at **16**. The housing members define an interior cavity for the pump gears, sealing means and inlet and outlet chambers.

Drive gear **18** is formed on a drive shaft **19**, and idler gear or driven gear **20** is formed on a driven shaft **21**. The shafts **19** and **21** are arranged on parallel spaced axes in the housing with the teeth on the gear members meshing at a zone centrally located in the housing cavity. Opposite ends of the shaft **21** project into bore recesses **24** and **25** in the housing members **11** and **10**. The bore **24** ends in the housing member **11** and the recess **25** in the housing member **10** is closed by an appropriate seal **26**. Opposite ends of the shaft **19** project into bore recesses **28** and **29** in the housing members **11** and **10**. The bore **28** ends in the housing member **11** and the recess **29** in the housing member **10** is closed by an appropriate seal means **30** including provision for a projecting end portion **32** on shaft **19** adapted to be connected with a suitable prime mover. Opposite end portions of the drive shaft and driven shaft are mounted in similar bearing sleeves **34** fixed in the housing recesses. In order to provide for lubrication, the interior surface of each bearing sleeve **34** includes a longitudinal groove **35** preferably located as shown in FIG. **5** and referred to in more detail hereinafter.

As seen in FIG. **2**, an inlet chamber **37** is provided in the spacer plate **12** at one side of the zone of intermeshing gear teeth and an outlet chamber **38** is provided in the spacer plate **12** at the other side of the zone of intermeshing teeth. Both chambers extend outwardly from the spacer plate partially into the end housing members. The inlet chamber **37** is always sealed from the outlet chamber **38** by means of tooth contact in the zone of intermeshing teeth and by sealing means engaging the outer edges of the gear teeth. While the seal between the inlet chamber and the outlet chamber provided by tooth contact at the zone of intermeshing teeth is constantly changing along the line of action tangent to the base circles of the gears during rotation of the gears, the inlet chamber is generally defined at one side of the intermeshing teeth and the outlet chamber is generally defined at the other side of the intermeshing teeth. The outer tips of the gear teeth rotating from inlet to outlet are sealed by contact with inner surfaces of the spacer plate **12** as at **40** and **41** which define generally cylindrical gear cavities intersecting at the zone of intermeshing teeth. The sides of the gear teeth are sealed by means of pressure balanced seal plates **43** and **44** generally of 8-shaped configuration fitted around the shafts and against opposite sides of the gears in cavities **40** and **41** in the spacer plate.

As best seen in FIGS. **5** and **6**, the inlet chamber **37** communicates with an axially disposed inlet passage **48** in the end housing member **11** having a laterally directed internally threaded terminus, and outlet chamber **38** communicates with an axially disposed outlet passage **49** in the end housing member **11** having a laterally directed internally threaded terminus. If desired, the threaded ends of passages **48** and **49** may be axially disposed.

In order to urge the pressure balance plates **43** and **44** into sealing engagement with the sides of the gear teeth, fluid is supplied to the outside of the plates as explained in more detail in our copending application Ser. No. 110,178, filed Jan. 27, 1971. **In general, as best seen in FIG.** 3, **the plate** 43 **is formed at each end on the periphery with a portion of reduced radius as at** 50 **to permit uniform distribution of the high pressure of outlet fluid along a predetermined portion of the gear side of the plate and behind the plate into an area** 52 **(FIG.** 5**) between a pair of sealing rings** 54 **and** 55 **in housing member** 11, **so that the area** 52 **is subjected to pressure corresponding generally to that in the outlet chamber. At opposite sides of the inlet chamber** 37, **the plate** 43 **is formed with ports** 57 **which admit fluid to the outside of the plate in areas within circular O-rings** 58 **in housing member** 11 **at pressure intermediate the outlet pressure and the inlet pressure. The gear side of the plate** 44 **is a mirror image of the gear side of the plate** 43, **and the outside of the plate** 44 **is subjected to fluid pressure similar to that supplied at the outside of plate** 43.

In order to provide for lubrication of the opposite ends of the drive shaft and driven shaft in the bearings **34**, lubricating fluid is pumped from trapped fluid in a sealed chamber at the zone of intermeshing teeth where the spaces between teeth decrease in volume, outwardly along the end portions of drive shaft **19**, across to the end portions of driven shaft **21**, and back inwardly along the latter to the inlet chamber.

In order to take lubricating fluid under pressure from the zone of intermeshing teeth, each of the pressure balanced plates **43** and **44** is formed on the gear side adjacent the drive shaft **19** with a groove as at **60** (FIGS. **1, 3** and **7**) which extends from the intermeshing teeth inwardly toward the shaft. The groove **60** terminates at the inner end in communication with an annular channel as at **61** around the shaft defined radially by the inner surface of the pressure balance plate and the outer surface of the shaft, and defined axially by the inner end of the bearing sleeve and the outer surface of the gear. The annular channel **61** communicates with the inner end of longitudinal groove **35** in the adjacent bearing sleeve **34** so that lubricating fluid may flow longitudinally outwardly along the drive shaft to the bearing recesses **28** and **29**. In order to seal the high pressure fluid in the annular channel **61**, a groove **62** (FIG. **4**) in the outer surface of each pressure plate around the drive shaft includes an O-ring **63** engaging the adjacent housing member.

In order to communicate the outer end of the bearing recess **28** with the outer end of the bearing recess **24**, the end housing member **11** is formed with a passage **64** which conducts lubricating fluid from the recess **28** to the recess **24**. End housing member **10** includes a passage **65** communicating the bearing recess **29** and the bearing recess **25** to permit lubricating fluid to flow to the latter. Lubricating fluid in the recesses **24** and **25** flows inwardly along the driven shaft through the grooves **35** in bearing sleeves **34** to annular channels as at **68** surrounding the driven shaft at opposite sides of the idler gear, like those at **61** on the drive shaft. Each annular channel **68** communicates with the pump inlet chamber **37** by means of a recess as at **69** in the face of the pressure plate adjacent the idler gear extending from annular channel **68** radially outwardly to the inlet chamber.

The lubricating circuit is illustrated diagrammatically by the broken line with arrows in the exploded view of FIG. **7**. In particular, it should be noted that the lubricating grooves **35** in bearings **34** are preferably located in the vicinity of the showing of FIG. **5** where the associated shaft is loaded least. That is to say, the shafts are usually forced inwardly toward each other by fluid pressure, and the lubricating grooves should be located outwardly from the place where the shafts are pressed hardest against the bearing sleeves.

FIGS. 8–12 illustrate the spaces between intermeshing teeth including teeth 70, 71 and 72 on drive gear 18 and teeth 73 and 74 on driven gear 20. In FIG. 8, contact of tooth 71 with tooth 73 at 77 provides a seal against the inlet chamber 37. Contact of tooth 72 with tooth 74 at 78 has just been made, providing a seal against the outlet chamber 38. Fluid under pressure has been trapped in the space between gear teeth 71 and 74 including a chamber 80 adjacent gear 18, a chamber 81 adjacent gear 20 and a passage 82 connecting the chambers 80 and 81.

As seen in FIG. 8, the lubrication pickup groove 60 is substantially covered by the gear 18 and the tooth 71. As the drive gear 18 rotates in a clockwise direction to the succeeding positions of FIGS. 9–12, the groove 60 is first uncovered slightly by the tooth 71 as seen in FIG. 9. At this time, the chamber 80 is decreasing in size, and fluid under pressure is forced from the chamber 80 into the lubrication groove 60. In FIG. 10, the end of the lubrication groove 60 is totally uncovered and the chambers 80 and 81 are of substantially equal volume. As the teeth rotate from the position of FIG. 8 to the position of FIG. 10, and the volume of chamber 80 is reduced, the volume of chamber 81 is increased, but to a lesser extent than the decrease in volume of chamber 80, as a result of which the net change in volume is a reduction which results in pumping lubrication fluid into the groove 60.

As the gears move from the position of FIG. 10 to the position of FIG. 11, the chamber 81 increases in volume and the chamber 80 decreases in volume, so that there is a tendency for fluid to flow out of the chamber 80 and out of the groove 60 into the chamber 81. However, the tooth 74 covers a substantial portion of the groove 60, and the passage 82 becomes restricted. Also, by the time the teeth reach the position of FIG. 11, chamber 81 reaches the edge 83 of the seal plate at the boundary of the inlet chamber, and by the time the teeth reach the position of FIG. 12, the contact at 77 is broken, and fluid flows into the chamber 81 from the inlet 37 so that all the fluid for the increasing volume of chamber 81 need not come from the chamber 80. Chamber 81 is opened to inlet 37 promptly by location of the edge 83 of the seal plate close to the line connecting the gear centers while still maintaining adequate seal between inlet and outlet. Thus, only a small percentage of fluid, on the order of 5 percent, flows out of the slot 60 compared to that which was pumped into it.

In order to assure that the lubrication slot 60 provides positive flow of a predetermined metered amount of lubricating fluid to the bearings 34, the slot must be properly located. The principles for locating the groove 60 are set forth in FIG. 13. As illustrated in FIG. 13, the uncovered end portion of groove 60 as seen in FIG. 10 should be located within the crosshatched area of FIG. 13. The radically inner boundary of the crosshatched area lies on the root circle of the gear 18. The radially outer boundary of the crosshatched area in FIG. 13 is defined by the line of action which connects the base circles of the two gears at the points of tangency. The upper boundary of the crosshatched area in FIG. 13 (the leading edge in terms of gear rotation) is defined by a line connecting the centers of the two gears. The lower boundary of the crosshatched area in FIG. 13 (the trailing edge in terms of gear rotation) is a line drawn from the center of the drive gear to a place labeled "POINT 1" which lies on the base circle and is located one tooth space from the point of tangency of the line of action to the base circle of the drive gear. The end of the slot 60 need not conform identically with the shaded area in FIG. 13, but may be varied somewhat in order to facilitate machining which might not necessarily conform with the shaded area when performed in the most expeditious manner.

In operation, the pump operates at relatively high pressures, on the order of 3,000 to 3,500 psi, and in one embodiment has a capacity of 20 GPM. It is expected that lubricating fluid on the order of 0.5 percent (0.1 GPM) of the theoretical pump capacity will be pumped at about 5 psi into pickup grooves 60. Actually, the pressure in the trapped volume where lubricating flow originates might be greater than that in the outlet in the absence of relief and sometimes leads to chatter of movable parts. The pumping of lubricating fluid from the trapped volume therefore relieves pressure at that point and reduces noise. Adjacent the outlet, on the gear side of the seal plate, it is recessed as at 85 to reduce the trapped volume between teeth near to gear 20 and shaft 21.

It will be seen that the pump embodies a lubricating system utilizing a predetermined quantity of fluid which is positively pumped from the sealed chamber between intermeshing teeth where fluid is trapped between the inlet chamber and the outlet chamber. The system has the advantage that a predetermined quantity of lubricating fluid is pumped regardless of the pressures developed in the pump. In pressure lubricating systems where lubrication is dependent merely upon pressure drop rather than positive pumping, the amount of lubricating fluid flow varies with variations in pressure, and therefore the efficiency of the pump varies. In the present system, the lubricating flow bears a predetermined relationship to the over-all pump capacity and the volumetric efficiency as affected by lubrication flow remains substantially constant regardless of variations in pressure.

We claim:

1. A high pressure rotary gear pump, comprising,

a. a housing,

b. a pair of spur gears in the housing having gear teeth intermeshing at a zone centrally of the housing,

c. said housing including seal plates at opposite sides of the gears and walls around the peripheries of the gears,

d. an inlet chamber at one side of the zone of intermeshing teeth,

e. an outlet chamber at the other side of the zone of intermeshing teeth,

f. a pair of shafts respectively carrying said gears and having adjacent ends extending into recesses in the housing,

g. bearings in the housing recesses respectively rotatably mounting the shaft ends,

h. a lubricating groove along each bearing, and

i. means for pumping lubricating fluid under pressure from the zone of intermeshing teeth outwardly along one bearing groove and back along an adjacent bearing groove to the inlet, including i-1. a passage in one seal plate originating at the zone of intermeshing teeth where the space between intermeshing teeth is decreasing in volume and connecting with said one bearing groove,

i-2. a passage in the housing connecting outer ends of said bearing recesses, and

i-3. a passage in the seal plate communicating said other bearing groove with the inlet chamber.

2. A pump as defined in claim **1**, wherein

a. said passage in said one seal plate includes a groove in the seal plate originating at the zone of intermeshing teeth and leading radially inwardly toward one shaft, and an annular channel around said one shaft connecting said radial groove and the inner end of said one bearing groove, and

b. the passage in said other seal plate includes a second annular channel around said other shaft and the inner end of said other bearing groove, and a passage in the seal plate communicating the second annular channel with the inlet chamber.

3. A high pressure rotary gear pump, comprising,

a. a housing,

b. a pair of spur gears in the housing having gear teeth intermeshing,

c. the intermeshing teeth forming a seal between the inlet and outlet and said housing including seal plates at opposite sides of the gears and walls around the peripheries of the gears,

d. an inlet chamber at one side of the zone of intermeshing teeth where the spaces between intermeshing teeth increase in volume,

e. an outlet chamber at the other side of the zone of intermeshing teeth where the spaces between the intermeshing teeth decrease in volume,

f. a pair of shafts respectively carrying said gears and having ends extending from the gears,

g. bearings in the housing respectively rotatably mounting the shaft ends,

h. a lubricating groove along each bearing,

i. means for pumping lubricating fluid under pressure from the zone of intermeshing teeth outwardly along one lubricating groove including

i-1. a groove in one seal plate originating at the zone of intermeshing teeth where the space between intermeshing teeth is decreasing in volume and leading radially inwardly toward one shaft, and

i-2. an annular channel around said one shaft connecting said radial groove and said one lubricating groove.

j. said groove in the seal plate originating where a space between intermeshing teeth is sealed from the outlet chamber and sealed from the inlet chamber with fluid from the outlet trapped therein,

k. the origin of the groove in the seal plate falling within an area the radially inner boundary of which falls on the root circle of the drive gear, the outer boundary of which falls on the line of action tangent to the base circles of the drive and driven gears, the leading boundary of which falls on a line extending between the centers of the gears, and the trailing boundary of which falls on a line from the center of the drive gear to a point one tooth space advanced from the point of tangency of the line of action to the base circle of the drive gear.

4. A high pressure rotary gear pump, comprising,

a. a housing,

b. a pair of external spur gears in the housing having gear teeth intermeshing at a zone centrally of the housing,

c. an inlet chamber at one side of the zone of intermeshing teeth where the spaces between intermeshing teeth increase in volume,

d. an outlet chamber at the other side of the zone of intermeshing teeth where the spaces between the intermeshing teeth decrease in volume,

e. said housing including floating seal plates against opposite sides of the gears forming with the intermeshing teeth a seal between the inlet and outlet and walls sealing the gear teeth rotating from the inlet chamber to the outlet chamber,

f. a pair of shafts respectively carrying said gears fixed thereon and having opposite ends extending into housing recesses,

g. sleeve bearings in the housing recesses respectively journalling the shaft ends,

h. a lubricating groove along each shaft bearing,

i. means for pumping lubricating fluid from the zone of intermeshing teeth outwardly along the lubricating grooves at opposite ends of one shaft end and back through the adjacent bearing grooves at opposite ends of the other shaft to the inlet including,

i-1. a groove in each seal plate leading from the zone of intermeshing teeth where a space between intermeshing teeth is decreasing in volume and extending radially inwardly toward said one shaft,

i-2. an annular channel around said one shaft at each side of the gear thereon connecting the adjacent radial groove and bearing groove,

i-3. a passage in the housing connecting outer ends of adjacent bearing recesses,

i-4. an annular channel around said other shaft at each side of the gear thereon at the inner end of the adjacent bearing groove, and

i-5. a passage in the seal plate communicating each last recited annular channel with the inlet chamber.

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