

Aug. 25, 1964

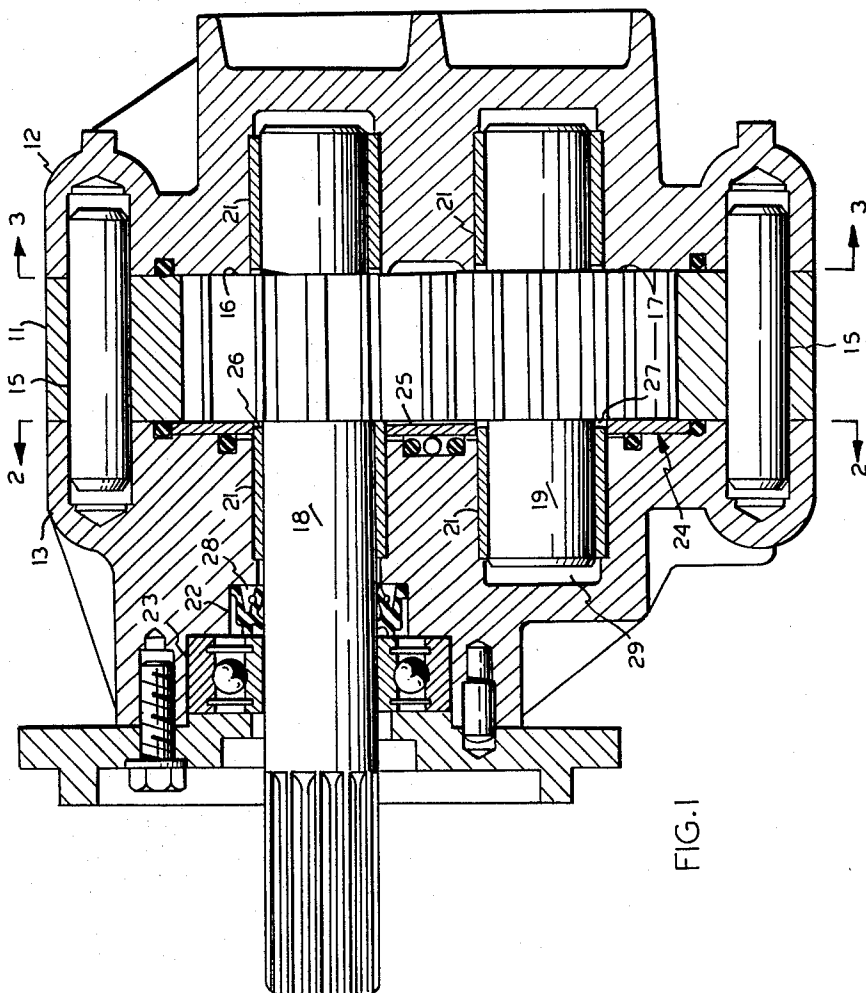
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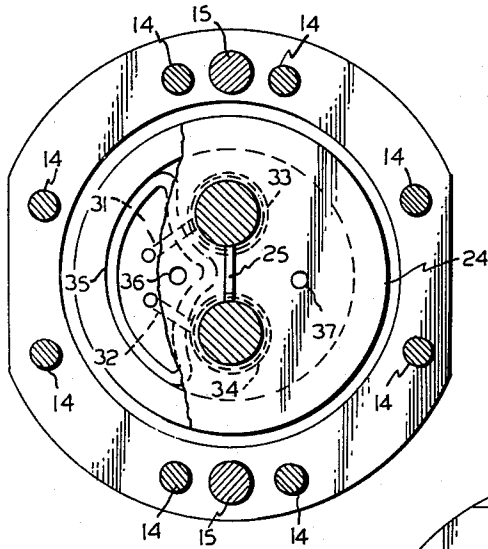


FIG. 2

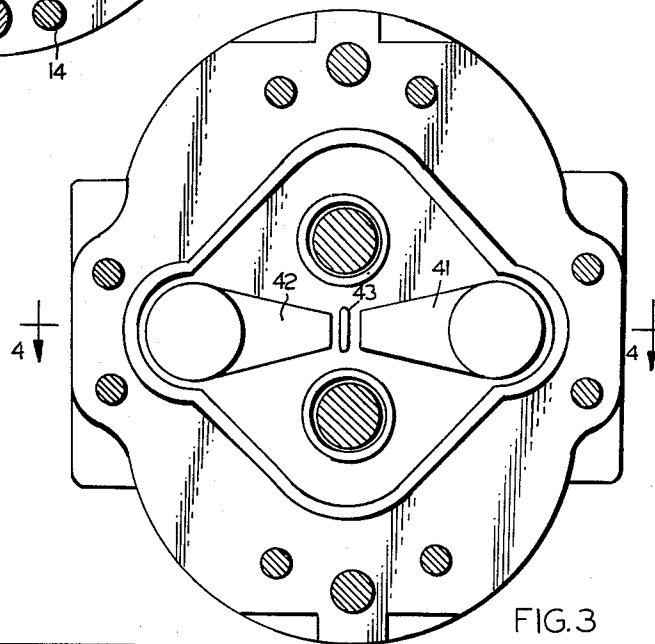


FIG. 3

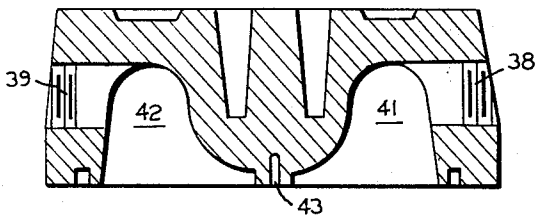


FIG. 4

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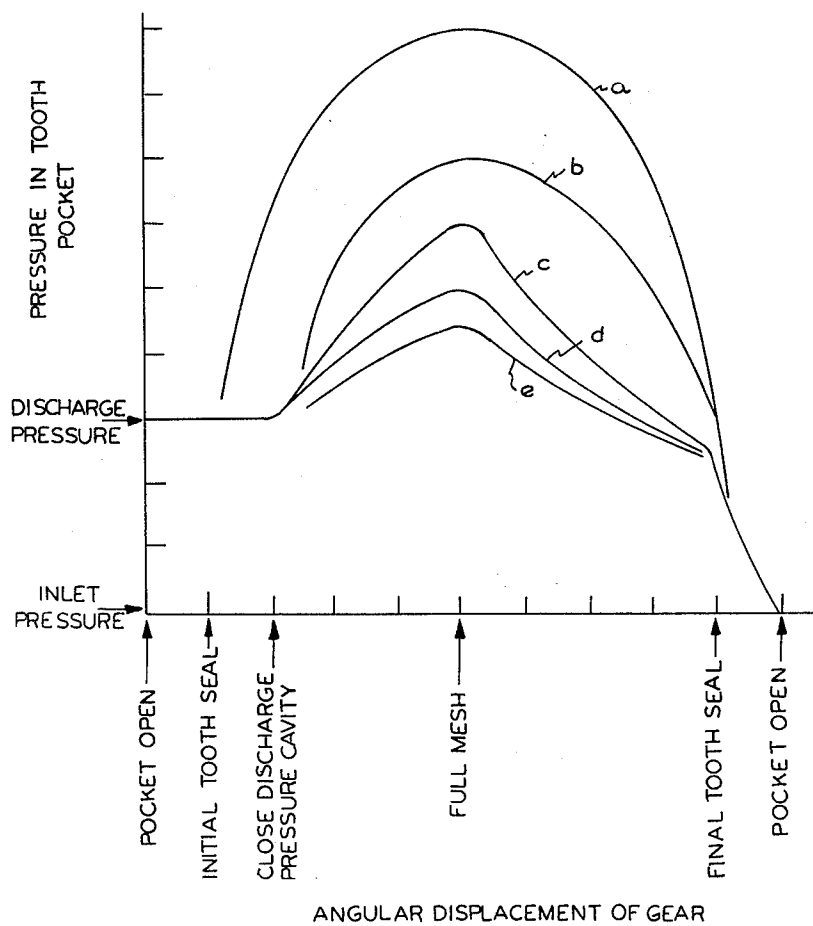


FIG. 5

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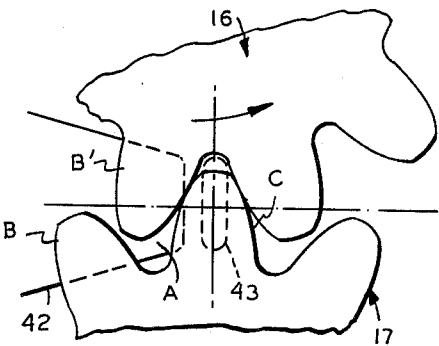


FIG. 6

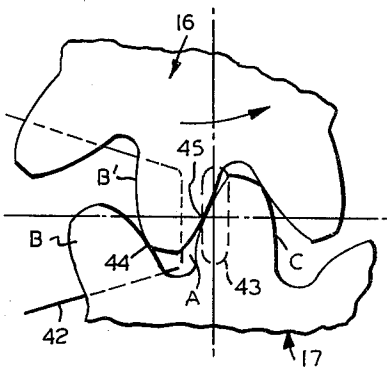


FIG. 7

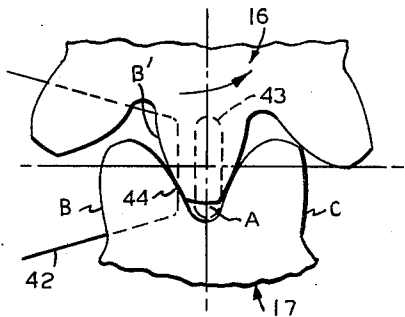


FIG. 8

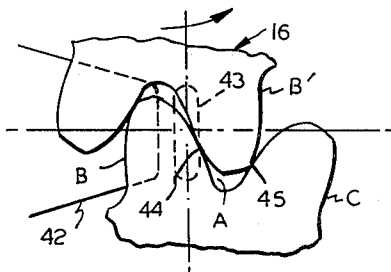


FIG. 9

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3 Claims. (Cl. 103—126)

This invention relates to hydraulic gear pumps and is particularly concerned with the problem of reducing noise.

It is known in the art that in pumps of this type in which there is a relatively small amount of backlash a closed pocket is formed alternately in the drive and driven gears as each gear tooth approaches the full mesh position. The oil trapped in these closed pockets is compressed to rather high pressures and it has been proposed to relieve these pressures by allowing the oil to escape through a bearing feed passage that leads to and through at least one of the bearings which journal the gear-supporting shafts. A scheme such as this provides an effective pressure release mechanism and also provides a convenient forced lubrication scheme for the bearings. However, it has been found that the rate of dissipation of the pressure in the closed pockets is so great at low discharge pressures, for example, pressures on the order of 100 p.s.i. and lower, that the gear teeth shift within the limits of the backlash and the gears oscillate back and forth each time the gears pass through the full mesh position. The frequency of this oscillation depends upon the rotary speed of the pump and on the number of teeth, and in some cases it is in the audible range and is quite objectionable.

The object of this invention is to provide a gear pump of the type employing a bearing feed passage which is considerably quieter in operation than similar pumps of the prior art. According to this invention, the pump includes a closed chamber that communicates with the pockets in the two gears as these pockets approach, pass through, and leave the full mesh position. The closed chamber increases the volume of fluid trapped in each gear pocket and thus decreases the percentage change in volume produced by movement of the gears into full meshing engagement. As a result, the peak pressure developed in the closed pockets of the gears is substantially reduced. Furthermore, as the gears move away from the full mesh position, the closed chamber acts as an accumulator and discharges compressed oil to the closed pockets. This pressurizing effect reduces the rate of decrease of pressure in the pocket as its volume increases. Finally, the closed chamber containing a trapped quantity of oil acts as a hydraulic cushion and aids in absorbing shock waves. In summary, the closed chamber greatly reduces impact velocity as the gear teeth move through the point of peak compression and thereby reduces the sound level of the resulting shocks.

The preferred embodiment of the invention is described herein in detail with reference to the accompanying drawings in which

FIG. 1 is an axial sectional view of a gear pump incorporating the invention.

FIG. 2 is a sectional view, on reduced scale, taken on line 2—2 of FIG. 1.

FIG. 3 is a sectional view, on reduced scale taken on line 3—3 of FIG. 1.

FIG. 4 is a sectional view, on reduced scale, taken on line 4—4 of FIG. 3.

FIG. 5 is a graph illustrating the pressure variation in a gear pocket during various conditions of operation at low discharge pressure.

FIGS. 6—9 are schematic diagrams illustrating the se-

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quence of events which occur as the two gears approach and leave the full mesh position.

As shown in FIG. 1, the pump comprises a casing including a central housing 11, an end cover 12 and an adapter 13 which are held together by bolts 14 (see FIG. 2) and which are maintained in alignment by pins 15. The central housing 11 encloses a generally oval pumping chamber which contains a pair of meshing gears 16 and 17 which are formed in one piece with shafts 18 and 19, respectively. The shafts 18 and 19 are journaled in plane bearings 21 and the drive shaft 18, which projects from the housing through the adapter 13, is provided with a seal 22 and an additional bearing 23.

The opposite ends of the pumping cavity in central housing 11 are closed by a pair of transverse end walls defined by the inner face of cover 12 and by the front face of a deflectable wear plate 24 which is clamped around its outer periphery between the central housing 11 and the adapter 13. The front face of wear plate 24 is provided with a narrow slot 25 that is centered with respect to the plane containing the axes of shafts 18 and 19 and which extends between the bores 26 and 27 through which the shafts extend. This slot 25 registers with the closed pockets in the two gears as the pockets approach the full mesh position and serves to transmit oil under pressure from these pockets to the bearings 21 located at the left side of the pumping chamber. This slot 25, therefore, constitutes part of a bearing lubrication passage for these bearings. The oil which flows outward along the outer peripheries of shafts 18 and 19 collects in chambers 28 and 29 and is returned to the inlet side of the pump through passages 31 and 32. Similar passages (not shown) drain the spaces at the right ends of the gear shafts.

Wear plate 24 is urged outward away from the side faces of the gears by the force developed by the pressure gradient in the pumping cavity. In accordance with the usual practice in this art, means are provided for developing on the rear face of the wear plate an inward directed pressure force that slightly exceeds the outward directed force developed by the pressure gradient. In the illustrated embodiment, this means includes resilient sealing members 33 and 34 which are mounted in grooves formed in the face of adapter 13 and arranged coaxially with the shafts 18 and 19, and a resilient sealing member 35 which is mounted in an irregularly shaped groove formed in the face of adapter 13 and positioned on the inlet side of the pump. The space enclosed by sealing member 35 communicates with the inlet side of the pumping cavity through a bore 36 formed in wear plate 24. Thus in the inlet region of the pump, the pressure forces acting on the front and rear faces of the wear plate are balanced. The remaining portion of the rear face of the wear plate, that is, the portion not enclosed by the sealing members 33, 34 and 35, is in communication with the discharge side of the pump through a second bore 37 formed in the wear plate. The area of this remaining portion is so selected that the pressure force developed on the rear face of the wear plate exceeds the outward directed force developed by the pressure gradient in the pumping cavity and deflects the wear plate 24 inward into sealing engagement with the side faces of the gears.

End cover 12 is formed with inlet and discharge ports 38 and 39, respectively, which communicate with recessed inlet and discharge cavities 41 and 42 that open through the inner face of the cover. The discharge cavity 42 is so arranged that it connects the closed pockets in the gears with the discharge port 39 after the mating teeth of the two gears have been brought into sealing contact and the pockets have been brought into registration with slot 25. With this arrangement, a portion of the fluid discharged by the pump is made available for bearing

lubrication and the peak pressure which is created in the closed pockets is reduced. The inner margin of cavity 42 is spaced from the plane containing the axes of the shafts in order to minimize leakage between the high and low pressure sides of the pump.

The inner face of cover 12 also contains a chamber 43 which is bisected by the plane containing the axes of the shafts 18 and 19 and which is symmetrical about an axis which is normal to that plane and located midway between the shaft axes. Unlike the slot 25 formed in the front face of the two plates, this chamber 43 does not communicate with the bores in cover 12 that receive the bearings 21 but is sealed from them by the side faces of the gears. Therefore, for all practical purposes, this chamber 43 is closed when the pump is assembled and is operating.

The operation of the invention will be apparent from an inspection of FIGS. 6-9. When the gears 16 and 17 are in the positions shown in FIG. 6, the pocket A between the teeth B and C of driven gear 17 overlies the discharge cavity 42, and the teeth B and B' have not been brought into sealing engagement. As the two gears rotate to the positions shown in FIG. 7, a seal is established along line 44 between teeth B and B', but pocket A still registers with discharge cavity 42 and thus is subjected to discharge pressure. Continued rotation of the gears causes the seal line 45 between teeth B' and C to overtravel the edges of slot 25 and chamber 43 so that fluid at discharge pressure is delivered to the left-hand bearings 21 of FIG. 1 through slot 25 and to chamber 43. Although the volume of pocket A decreases during movement of the gears from the FIG. 7 position, in which the initial tooth seal at line 44 is established, to the FIG. 8 position, in which seal line 44 is about to cross the inner edge of cavity 42, the pocket remains at discharge pressure because it is in continuous registration with cavity 42. When the gears move past the positions shown in FIG. 8, the seal line 44 between teeth B and B' crosses the edge of cavity 42 with the result that pocket A is isolated from the discharge side of the pump. Therefore, as the gears move from this position to the full mesh position, the oil in pocket A and in chamber 43 is compressed and its pressure rises above discharge pressure. As the gears move away from full mesh position, the volume of pocket A increases and the oil in this space and in chamber 43 is allowed to expand. After the gears pass the positions shown in FIG. 9, pocket A is isolated from slot 25 and chamber 43 and the seal at line 45 between teeth B' and C is broken. Therefore, from this point on, pocket A is connected with the inlet side of the pump.

The graphs of FIG. 5 illustrate the pressure variation in pocket A as the gears move to and through the full mesh position. The curve *a* shows the normal pressure change in the pocket resulting from the changes in its volume; the pressure rising from the level of the discharge pressure, at a point just prior to the creation of the initial tooth seal at line 44, to a maximum at the full mesh position, and then decreasing to inlet pressure as the gears move away from that position. Curve *b* illustrates the effect of adding to the pump the discharge cavity 42. Since the inclusion of this cavity has the effect of maintaining communication between the gear pocket and the discharge side of the pump for a portion of the movement of the gears from the point at which the initial tooth seal is created to the full mesh position, it reduces the percentage change in volume of the pocket after the latter is closed, and thus reduces the maximum pressure created at the full mesh position. As in the case of the conventional gear pump, the rising and falling portions of the pressure curve are generally symmetrical.

The addition of a bearing feed passage, i.e., a passage such as the one defined by slot 25 and the associated paths which lead fluid to and through bearings 21, further reduces the peak pressure created at full mesh posi-

tion (see curve *c*) because this passage, though restricted, does allow oil to escape from the gear pocket. However, since this escape takes place during movement both toward and away from the full mesh position, the slope of the descent portion of curve *c* is steeper than the slope of the rise portion. In other words, the inclusion of the bearing feed passage has the effect of increasing the rate of decrease of pressure in the gear pocket as the gears move away from full mesh position. It is believed that this rapid dissipation of pressure is what causes the noise frequently characterizing gear pumps that use bearing feed passages.

Curves *d* and *e* illustrate the manner in which the chamber 43 combats the effect of the bearing feed passage, the curve *d* showing the pressure variation in the gear pocket when the volume of chamber 43 is approximately one-half the volume of the pocket at full mesh position, and the curve *e* showing this variation when the volume of chamber 43 equals the volume of the pocket at full mesh position. As shown by these curves *d* and *e*, the chamber 43 reduces both the peak pressure at full mesh position and the rate of decrease of pressure in the gear pocket as the gears move away from this position. This in turn results in a reduction in the magnitude of the shocks encountered each time a gear tooth moves through full mesh position and in a reduction in noise level. Since the volume of oil trapped in the gear pocket is increased by the addition of chamber 43, it will be apparent that the pressure rise created in the gear pocket as the gears move to full mesh position depends upon the size of chamber 43. It is for this reason that the curve *e* of FIG. 5 lies below curve *d*.

As stated previously, the drawings and description relate only to the preferred embodiment of the invention. Since many changes can be made in the structure of this embodiment without departing from the inventive concept, the following claims should provide the sole measure of the scope of the invention.

What is claimed is:

1. In a hydraulic pump of the gear type including a housing formed with inlet and discharge ports and with a working chamber containing a pair of meshing gears, the meshing teeth of the two gears forming a closed pocket alternately in one gear and then in the other as the two gears rotate, shafts for the gears journaled in bearings supported in the housing, and means defining a restricted bearing feed passage which registers with said closed pockets but otherwise is isolated from free communication with the intertooth spaces of the gears for leading fluid from the closed pockets to and through at least one of the bearings, the improvement which comprises means defining a noise reduction chamber that is isolated from free communication with the inlet and discharge ports and communicates with said closed pockets as they approach, move through, and leave the full mesh position of the gears.

2. A hydraulic pump of the gear type including a housing formed with inlet and discharge ports and with a working cavity provided with two transverse end walls; a pair of meshing gears mounted in the working chamber cavity and having side faces in sealing engagement with the end walls, the meshing teeth of the two gears forming a closed pocket alternately in one gear and then in the other as the two gears rotate into full mesh position; a pair of parallel shafts supporting the two gears and extending through openings formed in the two transverse end walls; bearing means journalling each end of each shaft; a bearing feed groove formed in one of the transverse end walls and extending between the openings therein, said groove being so positioned that it is intersected by a plane containing the axes of the two shafts; and a chamber formed in the other of the transverse end walls and arranged along said plane containing the axes of the two shafts, said chamber being so positioned that it registers with said closed pockets and is isolated from

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free communication with the inlet and discharge ports as the pockets approach, pass through, and leave the full mesh position of the gears.

3. A hydraulic pump as defined in claim 2 which includes a second chamber formed in one of said transverse end walls and located on the discharge side of the working cavity, the parts being so arranged that each gear pocket moving to the full mesh position of the gears passes in sequence through a first zone of movement wherein it registers with the second chamber and is isolated from the first chamber and the bearing feed groove, a second zone of movement wherein it registers simultaneously with the second chamber, the first chamber and the bearing feed groove, and a third zone of movement wherein it registers simultaneously with both the first

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chamber and the bearing feed groove and is isolated from the second chamber.

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References Cited in the file of this patent

UNITED STATES PATENTS

1,271,970	Wood	July 9, 1918
1,644,817	Conant et al.	Oct. 11, 1927
1,682,842	Hamer	Sept. 4, 1928
1,972,271	McIntyre	Sept. 4, 1934
2,627,232	Lauck	Feb. 3, 1953
2,718,758	Minshall et al.	Sept. 27, 1955
2,885,965	Haberland	May 12, 1959

FOREIGN PATENTS

977,730	France	Nov. 15, 1950
716,772	Great Britain	Oct. 13, 1954