PRESSURE-BALANCED GEAR PUMP

Inventor:
John C. Lee

By
Horton, Davis
Brewer + Brugman
Attys.
Inventor:
John C. Lee

By: Horton, Davis,
Brewer, Burrows
Attys.
The present invention relates to gear pumps, and more particularly to means for pressure-balancing internal-gear pumps.

A principal object of the invention is to provide a construction and arrangement in an internal-gear pump in which fluid pressure is utilized for counteracting the forces otherwise tending to cause deflection of moving parts and consequent wear to those parts and other parts between which there is relative movement.

Another object is to provide in a gear pump of the character noted, greater sealing effect between relatively moving parts.

Another object is to provide a gear pump in which, because of the novel arrangement of fluid-pressure balancing of the character referred to, the loads imposed on the moving parts are reduced, higher operating pressures can be accommodated, and a more compact structure results with consequent reduction in weight and cost.

Another object is to utilize the pressure-balancing system referred to for positively lubricating certain internal relatively moving parts of the pump.

A still further object is to utilize the pressure-balancing arrangement referred to, to provide more effective lubrication to the bearing for the drive shaft of the pump and to prevent leakage through the seal for the drive shaft.

Other objects and advantages of the invention will appear from the following detailed description taken in conjunction with the accompanying drawings in which:

FIG. 1 is a longitudinal axial sectional view, taken on a staggered line, of a pump embodying the features of the present invention; and

FIG. 2 is a view taken substantially on line 2—2 of FIG. 1.

Referring now in detail to the drawings attention is directed first to FIG. 1 showing the main parts of the pump of the present invention. The pump illustrated is of the overhanging or cantilever type, as applied to which the present invention is found to be unusually effective.

The pump includes a housing 10 made up of a casing 12 and a head 14. Within the housing 10 are a pair of gears which include a ring gear or rotor 16 and a pinion gear or idler 18. The ring gear or rotor 16 is connected to a drive shaft 20 for rotation by an external source, such as a motor (not shown), and the idler gear 18 is mounted on a fixed pin 22 mounted in the head 14. Upon rotation of the gears in a first direction, which in the example assumed throughout this specification is counterclockwise as viewed in FIG. 2, the fluid to be pumped enters into the suction passage 24 and emerges through the discharge passage 26 formed in the casing 12. Further details of these main parts will be described herein below.

The housing 10 includes a gear chamber 28 which is defined by the casing 12 and the head 14, in which the gears 16 and 18 are confined and positioned for rotation.

The casing 12 includes a main body portion 30 having a shaft opening 32 for receiving the drive shaft 20 and opening into the chamber 28. This opening includes a recess 34 next adjacent the chamber 28, for receiving the hub 36 of the ring gear or rotor 16; next thereto is a reduced diameter portion 38 for receiving a casing bearing 40 which directly supports the drive shaft 20; the opening 32 also includes a counterbore portion 42 rearwardly of the portion 38 in which is disposed a shaft seal 44 of conventional construction, held in place by a follower 46 having a flange 48 through which bolts 50 are inserted into tapped holes in the body 30 of the casing. The casing body 30 has an extension 52 in which is disposed a conventional ball bearing 54 for supporting the drive shaft 20 at this point, the shaft then being supported at two widely spaced points, namely, the bearings 40 and 54, for maintaining the drive shaft, and thus the ring gear or rotor 16, accurately in position.

Formed in the body 30 of the casing 12 are a pair of return vent passages 56 and 58 communicating with the opening 32 at a point approximately between the bearing 40 and seal 44 at one end, and the chamber 28 of the casing at the other end. One of the passages, such as the passage 56, communicates with the chamber 28 in the suction passage 24, and the other return vent passage 58 communicates with the chamber 28 in the discharge passage 26. A plug 60 is removably positioned in one of the passages, such as by threading it in place, so that it can be positioned selectively in the passages for blocking that passage, it being understood that the pump is reversible and the suction and discharge passages are such depending upon the direction of operation of the gears. However, in the example assumed herein, the passage 24 will be regarded as the suction passage and therefore the passage 56 is connected to its side of the interior of the pump. If it should be desired to reverse the direction of the pump, the insert plug 60 is removed from the passage 58 and placed in the opposite passage 56.

The chamber 28 has axially opposite end faces 62 and 64, the rear one of which, 62, is formed on the casing 12 while the front one, 64, is formed on the head 14. These end faces are accurately finished for producing effective sealing surfaces with the corresponding and adjacent surfaces of the gears 16 and 18. The rear end face 62 is annular in shape, surrounding the recess 34 and is provided with recessed portions 66 providing free flow of fluid from the respective return vent passages 56 or 58 into the respective suction or discharge passage 24 or 26. These recessed portions 66 may be of any desired circumferential extent. The fluid pumped is free to flow past the periphery of the rotor 16 through the recessed portions 66, or through the space 68, into the space 70, namely, the space between the rotor 16 and the end face 62. The rear surface of the disc 72 of the rotor 16 is accurately finished, and the space 70 is of small dimension and acts as a sealing space against the free flow of fluid therethrough. However, as in a pump of this general nature, there is of course, a certain minor amount of flow of fluid through all such spaces.

The hub 36, referred to above, is received in the recess 34, being secured to the drive shaft 20 by any suitable means such as a pin 74. The space 76 between the hub 36 and the recess 34 is preferably of relatively great dimensions, for relatively freer flow of fluid therethrough.

Extending forwardly from the periphery of the disc 72 are a plurality of teeth 78, in the form of pump utilized, are eight in number, as seen in FIG. 2. These teeth extend into effective sealing engagement with the front end face 64 on the head 14 for forming a space 80 which is of dimensions similar to those of the space 70.

The space 80 serves as an effective seal against the free flow of fluid therethrough. Between the teeth 78 are cavities 82 for receiving fluid from the suction passage 24 and carrying it through the pump, as described herein below.

The suction passage 24 includes a chamber-like portion 84 surrounding a substantial portion of the periphery of the rotor 16 and a terminal portion 86 formed in a tubular element 88 of the casing serving as an adapter for connection to a conduit. In a similar manner the discharge passage 26 includes a chamber-like portion 90 and a terminal portion 92 formed in a tubular portion 94,
The casing is shaped for engagement by the gears, on one side by a surface portion 96 (FIG. 2), and on the opposite side by a surface portion 98, both engaged by the teeth 78. Other elements cooperate to effectively separate the suction and discharge passages, as described below.

The rotor or ring gear 16, as will be understood, meshes with the pinion or idler 18. The idler is mounted on the pin 22 in a conventional manner, having a bearing 100 interposed therebetween. The outer portion 102 of the body of the idler is accurately finished on the axially opposite surfaces for effective sealing engagement with the corresponding surfaces of the disc 72 and front end face 64, for forming spaces 104 therewith, serving as a fluid free flow passage. The idler 18 has teeth 106 forming cavities 108 therewithin, the idler in the design utilized for illustration herein having six teeth and cavities.

The rotor 16 and idler 18, as will be observed from FIG. 2, are relatively eccentrically mounted and the meshing or engagement between the gears takes place at one side of the pump, the right side—FIG. 2. The sectional view of FIG. 1 is taken on a staggered line, the right and left portions, as measured by the length of the shaft 20 and pin 22, being taken substantially on the axes of the pinion and the disc 72, respectively. Adjacent the opposite side is a crescent 110 secured to or integral with the head 14 and extending into effective sealing engagement with the opposed surface of the disc 72 for providing effective sealing engagement against the free flow of fluid therefrom.

The gears 16 and 18 engage the crescent in their rotation in such a way that closed cavities indicated at 108a, and 108x and 82b, are formed for carrying fluid from the suction passage to the discharge passage. In the discharge passage side of the pump, the teeth intermesh and force the fluid from the cavities against the free flow of fluid therefrom.

Holes 112 are formed in the disc 72 for establishing communication on opposite sides of the disc 72, for accomplishing an advantage explained more fully hereinbelow. These holes preferably are located adjacent to and radially inwardly of the respective teeth 78.

Means is provided for balancing the pinion or idler 18 through the instrumentality of supplying fluid under pressure to axially opposite sides of the idler. Formed in the head 14 are a pair of return vent passages 114 and 116 communicating respectively with the suction passage 116 and the discharge passage 118 in the head 14 surrounding the pin 22. The pin is provided with a bore 120 communicating between the groove 118 and the space between the pin and the bearing 100.

The bearing 100 and the adjacent portion 122 of the idler are of axially dimensioned greater in height than the axial spacing between the end faces 62 and 64, leaving spaces 124 of substantially greater dimension than the sealing spaces 104. The passage 114 is provided with a removable plug 126 closing that passage from the suction side, but the passage 116 from the discharge side remains open. Such a removable plug is utilized for enabling its use alternatively in either of the passages according to the direction of operation of the pump, as explained above in connection with the passages 56 and 58. In the operation of the pump, the greater pressure of the fluid in the discharge passage 58 of the pump is forced through the passage 116, the groove 118 and the bore 120 into the space between the pin and the bearing 100, from which it enters into the spaces 124 on the axially opposite sides of the pinion. This fluid in the spaces 124 produces a balancing action as explained below, and a positive lubricating effect on the pinion or idler 18.

During operation of the pump the pressure at the suction side of the pump is, of course, reduced relative to other parts of the pump and particularly the discharge passage. In those cases where the pressure behind the disc 72 is uniform, as in pumps herebefore known, the variation of the pressure on the front side caused binding, tilting and cocking of the ring gear or rotor. In addition, the pressures in the cavities of the gears vary from one side of the pump to the other and tend to produce a further binding or cocking effect on the rotor 16.

The present invention counteracts such an effect by establishing a fluid pressure on the rear side of the disc 72 similar to that on the front side, varying in accordance therewith; the spaces 68 and 66 enable relatively free flow of fluid from the suction and discharge passage into the space 70 between the disc 72 and the rear end face 62, and therefore the fluid pressures throughout this space 70 assume a pattern closely similar to the pattern of distribution of pressures on the front-side of the disc, i.e., adjacent the suction passage 24 where the fluid enters in the pump are relatively low, there is a relatively low fluid pressure on the rear side of the disc; similarly, at the discharge side of the pump there is a higher fluid pressure on the front side of the disc, and since the fluid can flow relatively freely from the discharge passage into the space behind the disc, the disc at the discharge side has relatively high fluid pressure on both sides.

To aid in establishing these equal pressures on opposite sides of the disc, and thus balancing the ring gear 16, the holes 112 in the disc 72 are provided which establish direct connection between the cavities in the gears and the space 70 behind the disc. In addition to the general and overall increase of fluid pressure from the suction side to the discharge, there is an irregular pattern or distribution of fluid pressures through the gears, or in the different cavities. The holes 112 provide direct communication between each of the cavities individually and the adjacent portion of the space on the rear side of the disc so that at each portion of the disc at any location around the disc, the fluid pressures on opposite sides of the disc are substantially equal.

Due to these equal fluid pressures on opposite sides of the disc, the ring gear one can be held in a perfectly true and accurate position, there being no binding or cocking effect thereon as was produced in pumps of the kinds heretofore known. In such known pumps the binding or cocking action was so great as to cause bending of the drive shaft and hence unequal wear of the bearings such as 40 and 54, as well as deflection of the seal 44. Moreover, and what was more serious, the various parts of the pump were required to be much heavier and stronger to withstand the binding and cocking forces. As a consequence, the pump was required to be more massive and heavy than the other and tend to produce a pump made according to the present invention may be made of much smaller and lighter parts. An important aspect of the balancing effect is that the discharge pressures of the pump are not limited by reason of limitations necessarily placed on the strength of the parts in the pump as in the case of prior pumps.

Due to the return vent passage 56, an additional balancing effect is produced on the ring gear or rotor 16. A reduced pressure is produced in the space 76, which counteracts any counter-force tending to be produced by the closed cavities 106c and 82t and 82b (FIG. 2), in which there is, of course, a reduced fluid pressure. The holes 112 associated with these closed cavities are at the time closed by the crescent 110, the holes being located adjacent the radially inner surfaces of the teeth 78 so as to prevent any blocking effect of the fluid in the cavities by the meshing of the teeth—see extreme right FIG. 2.

The reduced fluid pressure in the shaft opening 32 produced by the return vent passage 56 produces a more effective lubricating action in the bearing 40. It will be recalled that there is a relatively high pressure in the space 76 at the discharge side of the pump and fluid therefrom flows from this location through the space 76 to the bearing 40.

The reduced pressure just referred to also has the advantage of reducing or eliminating possible leakage.
through the seal 44 because of the fact that the such re-
duced pressure is less than atmospheric pressure.
The balancing action on the pinion or idler 18 is
essentially a separate aspect from the balancing action
on the rotor 16. The fluid from the discharge side of
the pump, as explained above, is forced through the pas-
sage 116, groove 118, bore 120 through the space be-
tween the pin 22 and the bearing 100 into the spaces
124. Because of the relatively greater pressure of the
fluid in these spaces, fluid is maintained therein at all
times and results in substantially equal pressures on both
sides. Any tendency of the pinion or idler to float for-
wardly or rearwardly is resisted by the fluid in the cor-
sponding space 104 and as a result the idler is maintained
out of actual contact with both the disc 72 and the front
end face 64. Consequently, wear between these parts is
reduced and substantially eliminated. There is no ten-
dency for the idler 18, as made according to the above
considerations, to bind or tilt or cock either from any
similar inaccurately directed forces from the rotor 16 or
from other forces including any resulting from the flow
of fluid or its mounting on the gear 22.
Due to the greater sealing area in the space 70 as com-
pared with prior known pumps there is less leakage and
consequent higher volumetric efficiency. However, not-
withstanding such greater sealing area, the viscous drag
torque is reduced because the area is closer to the axis
of rotation, as compared with the sealing areas in prior
pumps around the periphery of the rotor.
While I have disclosed herein a preferred form of the
invention, it will be understood that variations may be
made therein without departing from the spirit and scope
of the appended claims.
I claim:
1. An internal-gear pump comprising a housing hav-
ing a chamber with axially opposite end faces, and an inlet
passage leading to the chamber and an outlet passage lead-
ing therefrom, a rotor gear having a disc and axially
extending teeth at its periphery, said rotor gear having
passageways through said disc adjacent the innermost ex-
treme of said teeth mounted for rotation in said cham-
ber with its disc disposed adjacent one of said end faces,
the space between said disc and the adjacent said face
being of substantially uniformly sealing-effect dimensions
entirely therearound, a drive shaft connected with said
rotor gear and extending to the exterior, an idler gear
within the rotor gear and having its axially opposite sides
adjacent said disc and the opposite end face respectively,
said gears having intermeshing engagement at one side
throughout their axial extent and forming cavities there-
between into which fluid is drawn from the inlet passage
and from which it is expelled into the outlet passage, the
housing having elements in cooperation with the teeth of
the gears closing the cavities to the inlet and outlet pas-
sages at intermediate points therebetween with the rotor
gear engaging the idler gear so that said passageways
through the disc are uncovered in all positions of engage-
ment to prevent any blocking effect of the fluid in the
cavities formed by the meshing of the teeth whereby
said passageways provide communication between said
inlet and outlet passages and the respective adjacent por-
tions of said space, and between said cavity and the seal-
ing space between the outer end wall of the disc and said
one end wall of the housing chamber at all times during
the engagement of said teeth for establishing substan-
tially equal fluid pressures on opposite sides of said disc
at all points therearound.
2. The invention set forth in claim 1 which said equal
pressures are established by said passageways through
said disc, one such passageway being disposed adjacent to
and radially inwardly of each of the teeth of the rotor
gear and being disposed in the closed cavity formed by
the same said teeth of the rotor gear and the two ad-
jacent teeth of the idler gear, the teeth on the two gears
being so dimensioned that the rotor gear teeth do not
completely fill the space between adjacent idler gear
teeth and the passageways remain constantly open at the
side of the gears at which they mesh.
3. An internal-gear pump comprising a housing hav-
ing a chamber with axially opposite end faces, an inlet
passage leading to the chamber and an outside passage lead-
ing from the chamber, a rotor gear having a disc with
axially extending teeth at its periphery and mounted for
rotation in said chamber with the disc in sealing rela-
tionship adjacent one of said end faces and the extended
ends of said teeth in sealing relationship adjacent the
other of said end faces, a drive shaft connected to said
rotor gear and extending through a fluid seal to the ex-
terior of said housing, a pinion gear within said housing
having one axial side in sealing relationship against said
disc and the other axial side in sealing relationship
against said other end face, said pinion gear having a
smaller diameter than said rotor gear, being mounted
on an axle parallel to and eccentric to the axis of said
rotor gear, having its teeth intermeshing in fluid sealing
engagement with the teeth of said rotor gear on one side
and having indentations between adjacent teeth that are
sufficiently deep so that the teeth of said rotor gear do
not extend to the bottom thereof, means associated with
said housing located diametrically opposed to the en-
gaged portions of said gears and shaped to maintain
sealing contact with the tips of the pinion gear teeth on
one side thereof and with the tips of the rotor gear
teeth on the other side thereof to provide a fluid seal be-
tween the inlet passage and outlet passage, vent holes
through said disc, said vent holes located immediately
adjacent the tip of each of said rotor gear teeth so as
to prevent any blocking effect of the fluid in the cavities
formed by the meshing of the teeth to provide open
communication through said disc into the sealing space
between the outer end wall of the disc and said one end
wall of the housing chamber at all times during the
engagement of the rotor gear teeth with the pinion gear
teeth.
4. The internal-gear pump of claim 3 further char-
acterized in that channel means in said housing and in
the axle of said pinion gear form an open communi-
cation between the outlet passageway and a point midway
axially of said pinion gear and between said pinion gear
and its axle.
5. The gear pump of claim 4 further characterized in
that axially opposite sides of said pinion gear have shal-
low annular cavities adjacent the axle of said pinion gear.
6. The internal gear pump of claim 3 further char-
gerized in that channel means in said housing forms an
open communication between the inlet passageway and
the fluid seal of the drive shaft.

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