AXIAL PISTON PUMP DRIVE

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This invention relates to pumps and more particularly to constant pressure, variable volume output pumps of the type disclosed in my co-pending application Serial No. 825,005, filed July 6, 1959, now Patent No. 3,087,433, of which the instant application is a continuation-in-part.

Reference is also made to the pump control means for varying the displacement of the pumping cylinders as disclosed and claimed in my co-pending continuation-in-part application Serial No. 847,512, filed Oct. 20, 1959, now Patent No. 3,117,524. The instant application is directed more particularly to the structure provided for driving the pumping pistons.

It is among the objects of my invention to provide a pump having a cylinder block which is provided with a plurality of cylinders and wherein pistons are arranged to be reciprocated within the cylinders by a rotating cam and a non-rotating reaction member driven by the cam.

It is a further object of my invention to provide a pump according to the preceding object wherein the cylinder block is mounted for sliding movement on a guide within the pump housing and wherein the pumping pistons are reciprocated by a cam operatively mounted on a pump drive shaft and rotating within a bearing in the housing to tilt the reaction member and reciprocate the pistons.

It is a further object of my invention to provide a pump according to the preceding objects wherein the pumping pistons are spring-biased towards the inlet position or bottom of their stroke and wherein the reaction member is interposed between the driving cam and the pistons.

It is a further object of my invention to provide a pump according to the preceding objects wherein a supporting plate is mounted on the cylinder block guide and said supporting plate is apertured to loosely receive the pumping pistons and where a spring surrounds each of the pumping pistons and normally biases the piston to its inlet position or the bottom of its stroke and is driven in an opposite direction by the cam.

It is a further object of my invention to provide a pump according to the preceding objects wherein a driving shaft is mounted in the housing, a cam is splined to the driving shaft and a bearing is interposed between the cam and the housing and wherein a bearing is interposed between the opposite face of the cam and a reaction member wherein the reaction member is journaled with respect to the cam and the cam is journaled with respect to the housing.

It is a further object of my invention to provide a pump according to the preceding objects wherein the pump housing is provided with a channel at one side thereof and the reaction member is provided with a block riding in said channel whereby rotation of the reaction member relative to the housing is prevented during the rotation of the cam relative to the reaction member.

It is a further object of my invention to provide a pump according to the preceding objects wherein the bearing interposed between the driving cam and the housing is pinned to the housing and wherein the bearing interposed between the driving cam and the reaction member is pinned to the reaction member.

It is a further object of my invention to provide pump drive means according to the preceding objects wherein the driving cam carried by the drive shaft is splined to the drive shaft in a manner which permits a limited amount of tilting of the cam relative to the drive shaft to facilitate lubrication and load distribution to cam bearings.

It is a further object of my invention to provide pump drive means according to the preceding object wherein the drive shaft is provided with flexible drive means exterior of the pump housing such as, for example, a V-belt drive including a V-belt pulley which is constructed and arranged so as to introduce a force perpendicular to the drive shaft axis and such force is effective in a plane centrally located with respect to the tilttable spline connection between the driving cam and the drive shaft.

It is a further object of my invention to provide a bearing stack comprising a plurality of anti-friction roller bearing units arranged with respect to a floating plate to effect a differential action and thereby lower the bearing speed of the units with respect to the speed of the rotating pump parts.

It is a further object of my invention to provide an anti-friction bearing stack according to the preceding object wherein one set of roller bearings is arranged at one side of a floating bearing ring and another set of roller bearings is arranged at the other side of said floating ring and such assembly is arranged with respect to rotating pump parts whereby high loads may be transmitted at relatively low bearing speeds between the relatively high speed rotating pump parts.

Further objects and advantages relating to bearing lubrication, efficiency in operation, ease of assembly and low cost manufacture will appear from the following description and the appended drawings wherein:

FIG. 1 is a sectional elevation of a pump made according to my invention;

FIG. 2 is an elevation taken on a plane indicated at 2—2 of FIG. 1 showing a bearing ring employed in the drive mechanism made according to my invention;

FIG. 3 is an elevation taken on the plane indicated at 3—3 of FIG. 1 showing the driving cam employed in the drive mechanism;

FIG. 4 is an elevation taken at plane 4—4 of FIG. 1 showing the reaction member for driving the pistons of the pump;

FIG. 5 is a sectional view of a modified form of drive mechanism for the pump;

FIG. 6 is a sectional elevation showing a modified mounting for the inner end of the pump drive shaft;

FIG. 7 is a sectional elevation of a pump drive made according to my invention wherein anti-friction bearings are arranged to provide a differential bearing rotation;

FIG. 8 is an enlarged view with parts broken away taken on the plane as indicated at 8—8 in FIG. 7 and 8—8 in FIG. 9; and

FIG. 9 is an enlarged sectional view with parts broken away showing details of the bearing assembly.

Referring to the drawings, the pump includes a housing 19 having a mounting flange 11, a pump inlet at 12 and a pump outlet as at 13. The housing 10 is provided with a drive shaft 15 which projects from the right-hand end of the housing as viewed in FIG. 1 and the projecting end is adapted to have a pulley or the like mounted thereon for driving the pump. The inner end of the shaft 15, as at 16, is journaled within a plate 17 having a boss 18 thereon pressed into the tubular end of guide member 19 carried within the housing. The cylinder block 20 is mounted for reciprocating movement on the guide 19.
The cylinder block 20 is provided with a series of axial bores. In the instant embodiment there are seven axial boreswhereby each bore forms a cylinder 21 having an inlet port 22. It will be understood as the description proceeds that the interior of the housing is filled with oil at inlet pressure by way of the inlet 12 and that whenever the pistons carried in the cylinder 21 are retracted as shown in FIG. 1, the cylinders are filled with hydraulic fluid by way of the inlet ports 22.

Each of the cylinders 21 is provided with a reaction piston 23 which projects forwardly of the cylinder 21 and bears against the end of a tubular sleeve 24. The outermost end of the tubular sleeve 24 terminates in a check valve 25. The check valve 25 is spring-biased and opens in response to fluid pressure moving from the cylinder 21 into the outlet passageway 26 and thence to outlet chamber 27 terminating in outlet 28. Each of the reaction pistons is provided with a coil spring 28 interposed between a flange on the end of the reaction piston and the end of the cylinder block 20 so that the reaction pistons 23 are normally biased toward the check valve member 24.

As more particularly described in my said co-pending application Serial No. 847,512, filed Oct. 20, 1959, the cylinder block is moved to the left by fluid-responsive control means so as to reduce the fluid output volume in response to increases in pressure at the outlet. A ring 30 is secured to the exterior of the guide member 19 and forms with a counterbore on the cylinder block a chamber 31. A passageway 32 opens from the interior of the guide member 19 into the chamber 31 and when fluid under pressure is admitted to the chamber 33 at the interior of the guide member 19, such fluid pressure is effective in the chamber 31 to overcome the bias of the springs 28 and thus move the cylinder block 20 to the left, a position of reduced displacement.

As also more fully described in my said co-pending application Serial No. 847,512, filed Oct. 20, 1959, a control plunger 34 is mounted in the guide member with its outermost end extending to the outlet 27 so that the plunger 34 is responsive to fluid pressure at the outlet. A spring 35 at the interior of the guide member 19 normally biases the plunger 34 to a position so as to open a conduit leading from the housing interior to the interior of the guide member, that is, to the chamber 33. The control plunger 34 is shown in a balanced position in FIG. 1. However, when the pressure conditions are such that the spring 35 plus fluid in chamber 33 may overcome the outlet pressure acting on the plunger, the plunger 34 will move to a position so as to discharge fluid pressure from the chamber 33 and passageway 36 back to the interior of the housing. Accordingly the position of the cylinder block is controlled accurately so as to maintain a substantially constant output pressure. Relatively small fluid pressure changes in chamber 33 are utilized to move the cylinder block 20 to different control positions.

It will be understood that the cylinder block is free of any forces causing it to turn or bind on the guide 19 and that it floats free and independent of the positions of the pumping pistons 40 or the reaction pistons 23.

The pumping pistons 40 are preferably formed with a tubular section 41 having a flange 42 secured to the inlet free end thereof. A spring 43 is interposed between the flange 42 and the bottom of the recess 44 formed in the plate 17 around each of the openings in the plate 17 which is to receive a pumping piston 40. Accordingly the piston 40 is biased to the right as viewed in FIG. 1 and the limiting position in response to such bias is determined by the piston rods 45 and the reaction member engaged by the rods 45.

The inner end of the tubular section 41 of the piston is provided with a hemispherical recess which receives the spherical end 47 of the piston rod 45. The other spherical end 46 of the piston rod 45 rides in a hemispherical recess formed at the bottom of a cup 48 in the driving cam 61.

The driving shaft 15 is splined as indicated at 60 and such splines 60 are received within cooperating grooves on the splines 60 formed at the inner end of the driving cam 61.

The driving shaft 61 is thus free to float axially along the drive shaft splines 60 and to tilt within limits to adjust to various driving loads and oil film pressures during operation.

A thrust bearing, indicated at 65, is pinned to the housing at 66. This bearing 65 is provided with oil grooves 67 at that face thereof adjacent the driving cam 61.

The hub of the cam 61 rides at the inner periphery of the bearing 65 and an oil bore 68 extends through the thinned area of the driving cam 61 from the juncture of the annular surface on the hub of the member 61 and the radial face thereof which bears against the bearing 65.

It will be understood that as the cam 61 rotates, hydraulic fluid within the driving cam 61 and adjacent the drive shaft 15 is pumped outwardly by centrifugal force along the oil bore 68 and is thus pumped under pressure to the radial and annular surfaces of the bearing 65.

The driving cam 61 is provided with an annular bearing groove 70 at its forward face to receive a bearing 71. The bearing 71 is constructed substantially as the bearing 65 and is pinned to the driving cam 61 as at 72. The bearing 71 is provided with radial oil grooves 74 on the forward face thereof which bears against the reaction member 50. The reaction member 50 provides oil flow to the inner ends of radial oil grooves 74.

The reaction member 50 is provided with a radial bore 76 in which is secured one end of the pin 77. The lower end of the pin 77, as viewed in FIG. 1, is fitted with a square block 78 which is adapted to ride between the side walls 79 and 80 of the member 81 secured to the lower part of the housing. The interior of the block 78 is bored to receive and journal the pin 77 and thus the reaction member 50 is prevented from rotating in response to rotation of the driving cam 61.

The reaction member 50 provides about a point such as the point 200. The pump as shown in FIG. 5 is provided with a V-belt pulley 201 mounted on the end of the drive shaft 103 projecting beyond the end wall of the pump housing 100. The pulley is keyed to the drive shaft 103 as at 202 and is secured by a nut 203 threaded on the end of the drive shaft 103. Preferably the V-belt groove in the pulley 201, as indicated at 204, is located axially so as to run the V-belt generally in the plane indicated at 205. The V-belt 206 provides a drive which introduces a force which is perpendicular to the longitudinal axis of the pump. This force is indicated in FIG. 5 by the arrow 207.

The bearing 65 in FIG. 1 and the bearing 104 in FIG. 5 is substantially aligned with the axial center of the spline connection between the driving cam and the drive shaft. The spline connection where constructed to have an axial extent of about three-fourths of an inch preferably has a clearance on the spline teeth of about four thousandths to three-thousandths of an inch. This permits a limited tilting of the cam with respect to the drive shaft about a point indicated at 209. The transverse load due to the V-belt 206 can then be transmitted directly through the teeth of the spline connection to the bearing 65 (FIG. 1) or to the bearing 104 (FIG. 5) without introducing a transverse moment which would affect the alignment of the cam relative to the bearing carried in the pump housing.

It will be noted that the drive component indicated at 267 in FIG. 5 is transmitted from the pulley in one direction while the radial component resulting from the piston 267.
drive revolves with the cam. Accordingly in one extreme position the load from the piston drive will be added to the load component from the V-belt drive. In another extreme position the component resulting from the V-belt drive will be subtracted from the load resulting from the piston drive.

It will be understood that other forms of flexible drives, such as a chain gear eccentric of the drive shaft, will produce a component of force like that indicated at 207 in FIG. 5.

It will also be understood that the radial reaction force resulting from the piston drive is much greater than the component such as 207 induced by the V-belt drive and that a minute wobbling action thus occurs in the space provided for the oil film between the cam and the bearing mated in the pump housing. When the load resulting from the piston drive is added to the load resulting from the V-belt drive, the oil film is being thinned out to minimum in the radial space between the face of the cam and the adjacent face of the bearing. The oil film is continuously replenished from the radial oil grooves in the radial face in the bearing.

The freedom of alignment of the cam in relation to the bearing carried in the pump housing and, therefore, the freedom of generation of the squeeze oil film is affected by both the clearance of the spline teeth in relation to the bearing and in the clearance that must be provided in the spline teeth. Since the clearance or freedom of alignment between the bearing faces and the cam would be measured in microns for the generation of the wedge-type oil film, a very small clearance in the spline is sufficient for correct operation of the bearing. It will be understood that the shorter the spline teeth, the greater the freedom of alignment and, accordingly, the length of the spline teeth is selected as may be determined by the torque requirements.

It will be understood that the interior of the pump housing is normally filled with hydraulic fluid having lubricating characteristics and that a film of such oil is disposed between the bearing 65 and the radial face 67 of the driving cam 61. It will also be understood that in the form of drive in FIG. 1 an annular film of oil is disposed between the face of the bearing 71 and the annular face of the reaction member 58 adjacent the bearing in the radial face 15 at a rate such that the oil film is never completely squeezed out of the space between the bearing surfaces during the normal operation of the pump.

When the pump is idle the springs 43 will continuously bias the reaction member 59 toward the bearing 71 and the cam member 61 toward the bearing 65 so as to squeeze the oil film from between adjacent bearing surfaces. The radial oil grooves such as 67 and 74 provide a supply of oil available to the bearing surfaces and upon rotation of the cam such oil from the grooves will replenish the film between the bearing surfaces. The pump is preferably unloaded when idle and at the start of operation as disclosed in my said co-pending application Serial No. 847,512.

In that form of invention illustrated in FIG. 5, the housing 100 is provided with annular races 101 adjacent the opening 102 for the drive shaft 103. An annular bearing 104 is arranged in the chamber 101 and is pinned at 105 to the housing. The bearing 104 is constructed substantially as the bearing 65 of the first embodiment. The bearing 114 includes radial oil grooves 116 arranged to receive the oil film from between the bearing surfaces. The drive shaft 103 is provided with an annular groove to receive a snap-ring 107 which prevents axial displacement of shaft 103. In this form of my invention, however, the bearing 110 interposed between the driving cam 108 and the reaction member 111 and the bearing is pinned to the reaction member as at 112 rather than being pinned to the driving cam as in the embodiment of FIG. 1. The bearing 116 being pinned to the reaction member 111 is provided with oil grooves 113 bearing against the radial face of the driving cam 108.

In this form of my invention the pistons 120 and the piston spring 121 are constructed and arranged substantially as in the earlier embodiment. Similarly, the reaction member 111 is provided with a block 122 which rides in a channel 123 in the guide member 124 so as to prevent rotation of the reaction member 111 during the drive of the pump. In this form of my invention the driving cam 108 is splineled to the shaft 103 and is provided with oil passageway 109 to move the lubricant by centrifugal force through the radial grooves 106 when the cam 108 is rotated. Similarly an oil passageway 108 effects oil flow along radial grooves 113 in bearing 110.

Referring again to the fact that the load applied by the pumping pistons is eccentric of the driving shaft 103, it will be found that this driving load is applied progressively to a different bearing surface on the bearing 110 as the cam turns. This is to be distinguished from the arrangement in FIG. 1 where the same surface of the bearing 71 is subjected to the driving loads during pump operation. Thus in the form of FIG. 5 the load revolves in relation to the bearing surface 110 and an improved replenishment of oil film with resultant longer pump life is provided.

In connection with all forms of drive it will be understood that the clearance between the driving shaft and the cam is sufficiently large to permit a limited amount of tilting of the cam with respect to the drive shaft. This limited tilting results in an accommodation of the cam and bearings to the eccentricity about the axis of the driving shaft. This insures against a break-down of the oil film between the bearing surfaces during pump operation and also prevents the occurrence of localized high bearing loads.

In the form of invention shown in FIG. 6, the shaft 15 is provided with a reduced end portion 16 as in the embodiment of FIG. 1. In this form, however, the disc 150 is provided centrally with a tubular boss 151 which is pressed into the open end of the tubular guide member 19. The central opening in the boss 151 is provided with a counterbore 152 which is adapted to receive a flanged bronze bushing 153. In this manner the driving shaft 15 is journaled for free rotation in the tubular guide member 19. The pistons 40 and springs 43 are constructed and arranged as in the first embodiment.

The anti-friction bearing arrangement for my pump drive is illustrated in FIGS. 7, 8 and 9. In this form of the invention the pump housing 160 is provided with a drive shaft 161 and a cam member 162 is splined to the driving shaft as at 163. The driving shaft 161 is restrained against withdrawal from the pump housing by means of a snap-ring 164 arranged in a groove within the driving shaft and adapted to bear against a shoulder at the interior of the cam 162. The interior of the pump housing 160 is provided with a fixed bearing ring 165 which is preferably pinned to the housing as at 166. The driving cam 162 is provided with a reduced hub portion 167 which is arranged within a roller bearing assembly 167a carried by the ring 165. A radially disposed anti-friction bearing stack 166 is arranged between the radial face on bearing 165 and the rear radial face of cam 162. A similarly constructed bearing assembly 169 is arranged between the inclined face of the cam 162 and the rear radial face of the reaction member 170. Preferably the reaction member 170 is provided with a block 171 journalled on a pin 172 and the block is retained against the pin 173 carried by the member 174 of the pump housing. The reaction member 170 is thus restrained against rota-
tion within the pump housing and the forward face of the reaction member is preferably provided with rounded sockets 175 to receive the rounded end on piston rods such as 176.

Recent developments in the anti-friction bearing art have resulted in the production and general use of needle roller thrust bearings of the type disclosed in U.S. Patent No. 2,724,625 to R. H. White, dated Nov. 22, 1955. In this type of bearing a pair of complementary annular discs of sheet metal are provided with radial apertures to receive the outer periphery of the radially arranged rollers. Referring to FIG. 9, a needle roller thrust bearing unit of the type referred to is shown at 180, 181, 182 and 183. It will be illustrated that the units at 182 and 183 are arranged with their inner peripheries adapted to bear against the cylindrical hub portion 162a of the cam 162. Each of the bearing units 182 and 183 is surrounded by floating spacer rings indicated at 184 and 185. The rings 184 and 185 thus provide an abutment for the inner diameter of the units 180 and 181, respectively.

It will be understood by reference to said U.S. Patent No. 2,724,625 that each of the bearing units, such as 180, includes complementary annular disc members 186 and 187 which are provided with radial openings which receive and expose therethrough portions of the rollers 188. The ends of the rollers are rounded, as shown best in FIG. 9, and the rollers are retained between the plates 186 and 187 by reason of the fact that the width of the opening in the plate is less than the diameter of the roller 188.

According to my invention a free-floating, flat annular bearing ring 190 is interposed between the bearing units 180–183 and the complementary bearing units 181–182. The annular ring 190 is free with respect to the hub portion 162a of the cam 162.

The rolling resistance of the rollers in units 189 and 183 is substantially equal to the rolling resistance of the rollers in units 181 and 182 and accordingly the bearing plate 190 will revolve at approximately half of the speed of the rotating cam. Since the cam 162 rotates at the speed of driving shaft 161, the relative speed of the rollers in the bearing units is half of the speed that such rollers would have if one layer of anti-friction rollers were used. Accordingly with the bearings arranged as disclosed herein, the anti-friction bearings may be used with pump speeds higher than the conventional design speeds for such type of bearings. The basic advantage of the bearing units is the extreme flatness and the high thrust load carrying capacity at relatively low revolutions per minute. By using the floating plate 190 and bearing units as disclosed herein, the bearing units may be operated well within their designed speed of rotation, and yet the pump may be driven at speeds higher than the bearing design speeds.

Another advantage of the anti-friction bearing drive assembly which I have provided is that one layer of bearing units, such as the layer 181–182, may be scored or its resistance to rolling increased and thus cause the floating plate 190 to turn at a higher speed, yet the other layer of bearings 180–183 will permit useful operation of the pump. Such operation would continue with the layer 180–183 turning at a lower speed than a single unit. The concentric arrangement of the bearing units 182 and 181 and the units 183 and 180 provides a high load carrying capacity at high speeds within minimum space.

The cam 162 is provided with a hub 1675 which is disposed within the reaction member 170 and is journaled with respect to the reaction member 170 by a roller bearing 169.

It will be understood that where the driving torque is applied coincident of the axis of the drive shaft, the bearing between the cam and the housing is not required to carry a load such as indicated at 207. Such drive applied coincident of the drive shaft would not be usual or ordinary and I have illustrated, as a preferred form, the belt drive.

Although I have illustrated and described in consider-
means on said housing and on said reaction member to prevent rotation of the reaction member in response to rotation of the driving cam to permit said reaction member to be progressively tilted to reciprocate the pistons during rotation of the driving cam, a driving shaft extending axially into one end of said housing, said driving shaft extending through said driving cam and being journaled at its inner end in said housing, and a loose spline connection between said driving cam and said driving shaft to cause said driving shaft to rotate said driving cam while allowing limited universal rocking motion of said driving cam relative to said shaft and said bearings to permit the differential reaction forces exerted on said reaction member to cause a progress rocking action of said driving cam with respect to said fluid film bearings to create a rotating wedge shaped oil film at said bearings.

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