March 5, 1963 T. R. THOMAS 3,079,867

CENTRALIZED ROTARY LUBRICATING PUMP

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Fig. 1.

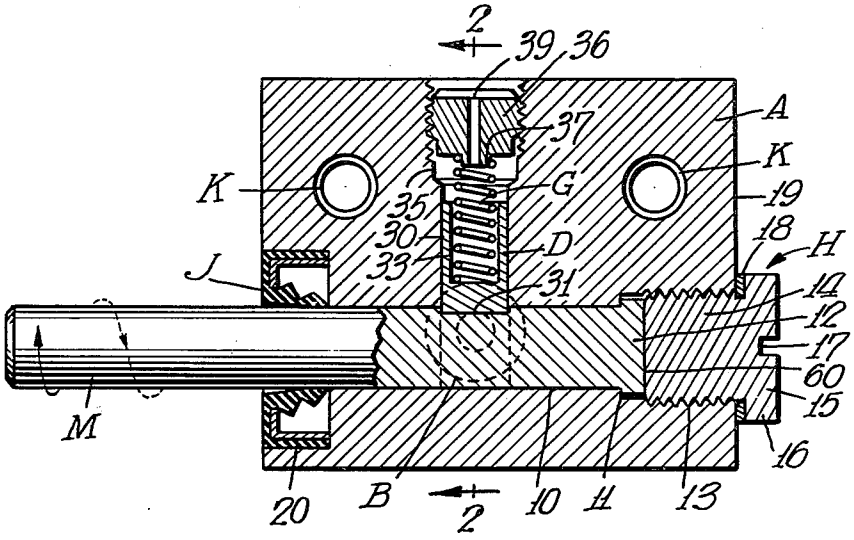
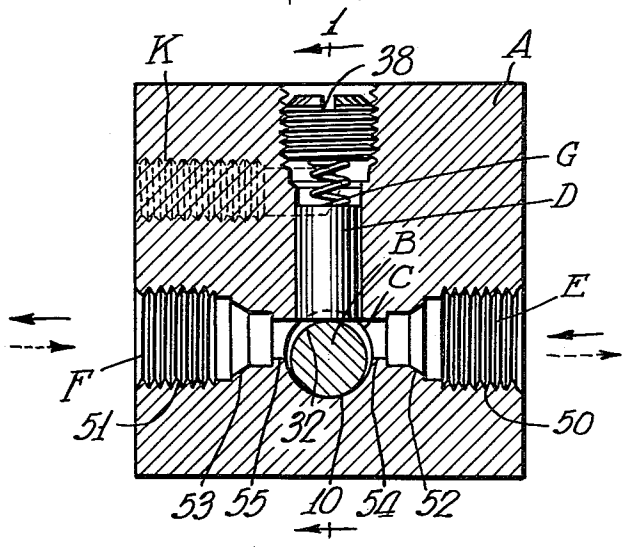


Fig. 2.

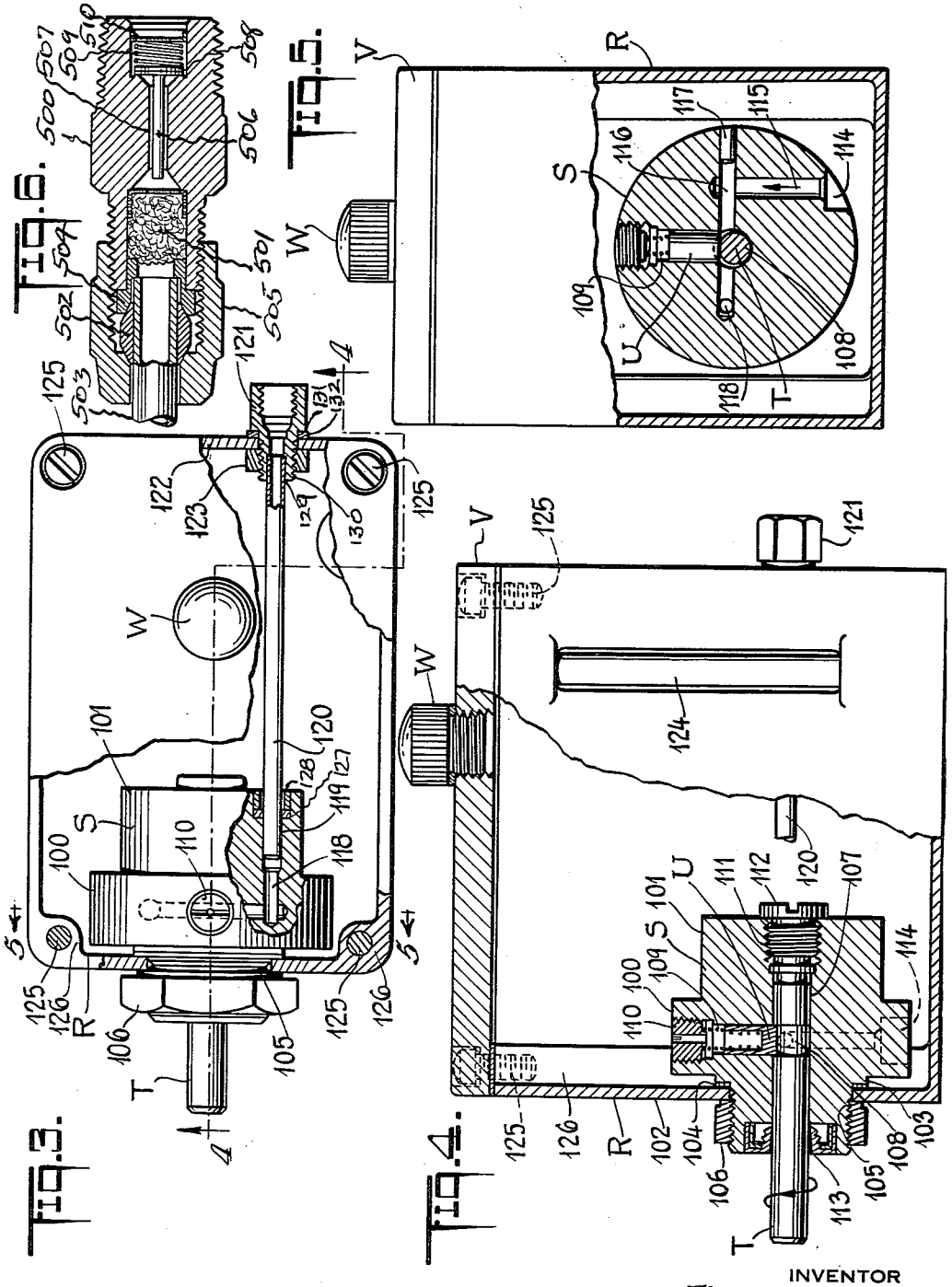


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CENTRALIZED ROTARY LUBRICATING PUMP

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CENTRALIZED ROTARY LUBRICATING PUMP

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The present invention relates to a centralized lubricating pump.

It is among the objects of the present invention to provide a durable, reliable, small rotary pump which will occupy small space and may be readily inserted in mechanisms.

Another object is to provide a simple rotary, reliable lubricating pump which may be utilized to supply a predetermined amount of lubricant to a lubricant distributing system and which will be at low cost reliable in operation.

Still further objects and advantages will appear in the more detailed description set forth below, it being understood, however, that this more detailed description is given by way of illustration and explanation only and not by way of limitation, since various changes therein may be made by those skilled in the art without departing from the scope and spirit of the present invention.

In accomplishing the above objects, it has been found most satisfactory according to one embodiment of the present invention to provide a small rotary pump which is provided with a rotary shaft having an eccentric groove tangent on one side in which will run a spring loaded piston type vane or reciprocating cylinder element.

The pump may be readily driven by means of a rotating shaft and the drive may be clockwise or counterclockwise. The pump element itself will consist of a sealed shaft which rotates in the pump body and has an eccentric groove whose diameter is less than the shaft and whose outer end is tangent with the shaft.

A spring loaded plunger type vane follows in the eccentric portion or eccentric groove of the sealed shaft and keeps in contact with the eccentric groove at all times.

This spring loaded plunger type vane cylindrical piston in essence behaves as a blower and divides the pump department into two separate chambers or sections.

The quantity of oil which is drawn into the suction chamber is driven around and out of the outlet by the eccentric shape which traps and transfers the oil from one section to the other.

The vane, in contact with the eccentric, acts as a wall preventing the oil from being carried back to the inlet or suction side.

The pump is designed to operate as a constant volume pump until a peak pressure is reached which creates the maximum pressure for which the piston type vane spring was originally set. At this point the spring loaded plunger acts as a blowoff or safety valve.

When this maximum pressure is exceeded the pump will change characteristics from a constant volume pump to a constant pressure pump. Then, behaving as a constant pressure pump it will continue to function as a constant pressure pump until the pressure falls below the maximum for which the spring is pre-set. When this occurs the pump automatically reverts back to its operation as a constant volume pump.

The pump may be driven by spur or helical gears or it may be driven by a belt and pulley or a chain. When the pump shaft is driven by spur gears, belt and pulley or chain, there is no axial thrust.

On the other hand, when the pump is driven by helical gears, possible thrust due to the helix angle may be taken up either by an end face of a screw plug or by a shoulder at the bottom of the counter-bored hole which houses the screw plug, depending upon the direction of the drive.

The pump may be readily mounted by two holes in the body and these may either be tapped holes to take mounting screws or drilled through holes to take bolts.

In either case the screws or bolts will screw into the machine member on which the pump is operated and to which it is to supply lubricant.

With the foregoing and other objects in view, the invention consists of the novel construction, combination and arrangement of parts as hereinafter more specifically described, and illustrated in the accompanying drawings, wherein is shown an embodiment of the invention, but it is to be understood that changes, variations and modifications can be resorted to which fall within the scope of the claims hereunto appended.

In the drawings wherein like reference characters denote corresponding parts throughout the several views:

FIG. 1 is a transverse vertical sectional view upon the line 1—1 of FIG. 2.

FIG. 2 is a transverse vertical sectional view upon the line 2—2 of FIG. 1 at 90° to the section of FIG. 1.

FIG. 3 is a top plan view partly in section of an alternative form of the invention, showing the pump unit mounted at the side of a reservoir housing.

FIG. 4 is a side sectional view upon the line 4—4 of FIG. 3.

FIG. 5 is a transverse vertical sectional view upon the line 5—5 of FIG. 3.

FIG. 6 is a longitudinal sectional view of a high restriction flow metering outlet fitting which may be utilized at the outlets of the distributing system supplied with lubricant from the pump of FIGS. 1 to 5.

Referring to FIGS. 1 and 2 there is shown a pump body A having a rotating pump element B which is driven from the shaft M.

The rotating pump body B consists of an eccentric grooved portion of the shaft and the pump chamber C is divided into sections by means of the spring loaded plunger type vane D.

Depending upon the direction of drive, either opening E or F may serve as an inlet or outlet opening.

The spring G will determine whether the pump operates as a constant volume or constant pressure pump. The pump chamber C is sealed at the ends of the bearing by means of the sealing arrangements H at one side and J at the other side of the through passage for receiving the shaft M. The pump is mounted by means of the mounting openings K.

Referring specifically to FIGS. 1 and 2 the pump block A may consist of a non-porous metal into which is bored an opening 10 for receiving the shaft M.

The end of this bore is shouldered, as indicated at 11, to receive the enlarged portion 12 of the shaft M, and beyond the shoulder is provided the tapped portion 13 for receiving the threaded portion 14 of the plug 15, having the enlarged head 16 and the fillister slot 17.

The gasket 18 will be clamped in leak-tight fashion by the head 16 against the wall 19 of the block A.

The other end of the through bore 10 has an enlargement 20 in which is received an annular cup shaped seal J.

Transverse to the bore 10 is the bore 30 which receives the reciprocating vane D.

The reciprocating vane D has the lower portion 31 which rides in the eccentric groove 32 which forms the pump chamber C. The eccentric groove 32 will be seen in FIG. 2 to be crescent shaped and formed by a recessed cylindrical portion of the shaft and having a longitudinal axis offset from the longitudinal axis of the shaft end and having a periphery which is tangent to the periphery of

the shaft end at one point. The groove will therefore have a maximum portion at its top in FIG. 2 and a minimum portion where its periphery coincides with the periphery of the shaft end at its bottom in FIG. 2. The interior of the plunger is provided with an opening 33 which carries the coil spring G, tending to press the head 31 into the groove 32 and separate the pump compartment C into two chambers as best shown in FIG. 2.

Beyond the transverse bore 30 is an enlargement 35 which receives the plug 36 having the nipple 37. The nipple 37 receives and centers the outside end of the spring G.

The adjustable threaded plug 36 has the filister slot 38 by means of which the position of the plug 36 may be varied, varying the tension on the spring G.

The screw plug 36 is provided with an axial opening 39 serving as a vent for the piston type vane D.

The openings K permit the pump block A to be mounted upon a machine structure.

The shaft M may be driven in either direction, clock-wise or counter-clock-wise, to send oil in the inlet E and out the outlet F, or in the inlet F and out the outlet E.

In FIG. 2, when the shaft B is driven clock-wise, the lubricant will flow with the solid arrows. When driven counter-clock-wise it will flow with the dotted arrows.

The tapped openings 50 and 51 form part of the inlet and outlet E and F and they have reduced lower portions 52 and 53 which have bores 54 and 55 leading to the pump compartment or chambers C. It will be seen that the upper part of the pump wall defining the bores 54 and 55 coincides with the lower portion 31 of the reciprocating vane D when the vane is in contact with the periphery of the recessed cylindrical portion of the shaft M at the maximum portion of the groove 32 as seen in FIG. 2. The lower portion of the pump wall defining the bores 54 and 55 will be seen to substantially coincide with the longitudinal axis of the shaft M.

Although there will be some leakage along the bearing at the shaft 10, the lubricant will be sealed into the chamber C by means of the gasket 18 and the annular seal J at the other end.

It will be noted that the pump chamber extends around the shaft M about 200 to 330° with the portion of the shaft between the ends of the groove sealing the chambers C.

In operation the follower D acts substantially as a vane to separate the chambers C or to separate the compartment C into two chambers.

As the shaft rotates the oil will be drawn into one of the chambers C as this chamber increases in volume with the rotation of the shaft M.

This oil which is then drawn in will be driven around and out of the outlet by the eccentric shape which traps and transfers the oil from one chamber C to the other chamber C.

The reciprocating piston vane D acts as a wall to be carried back from the inlet to the suction side.

As long as the pressure of the lubricant is less than the pressure of the spring G, the pump will function as a constant volume pump. However, as soon as the pressure exceeds the pressure of the spring G, this plunger or piston vane D will act as a blow-off or safety valve.

Then the pump will function as a constant pressure pump instead of a constant volume pump and this will continue until the lubricant pressure falls below that determined by the pressure or setting of the spring G. When this occurs, the pump automatically reverts back to being a constant pressure pump.

Normally, where the pump shaft M is driven by spur gears, belt or pulley or chain, there is no axial thrust. If driven by helical gears, however, there will be possible thrust due to the helix angle. In such case the thrust will be taken up either at the contact along the shoulder 11 or along the plug face 60.

The tapped holes K may take mounting screws or there may be through passageways in place of the tapped holes K to take bolts to screw into the machine member on which the pump is mounted.

In the embodiment of FIGS. 3, 4 and 5, the reservoir R is provided with a pump body S mounted on one side thereof.

The pump body is provided with a rotating drive member T and a reciprocating spring-pressed piston plunger U.

The reservoir is provided with a cover V having the removable filling cap W.

Referring particularly to FIGS. 3 to 5, the pump body S has a large diameter cylindrical portion 100 and a small diameter cylindrical portion 101 extending into the interior of the reservoir R.

At the other side of the large diameter portion 100 and adjacent to the wall 102 is the shoulder portion 103 with the gasket 104 which forms a lubricant-tight seal against the wall 102.

The body has a projecting threaded nipple portion 105 which receives the annular interiorly threaded collar or nut 106.

The pump body 100 has a through bore 107 which receives the shaft T. This shaft T has a reduced diameter eccentric portion 108 against which the bottom portion of the piston U rests.

The piston U consists of a cup and the interior of the piston receives a coil spring 109, the pressure of which may be adjusted by means of the screw plug 110.

The end of bore 107 has an enlarged tapped portion 111 which receives the plug 112.

At the external end of the shaft T the shaft is sealed by means of the cup-shaped sealing members 113.

The pump has an inlet recess 114 with an inlet bore 115 leading into the transverse bore 116, which transverse bore 116 is plugged at 117.

The lubricant will flow through the bore 116 from right to left as shown in FIG. 5 and into the transverse bore 118.

From the transverse bore 118 the lubricant will flow into the socket 119, the tube 120 and thence into the outlet fitting 121 mounted in the wall 122 of the reservoir.

It will be noted that the pump unit of FIGS. 3 to 5 differs from the pump unit of FIGS. 1 and 2, in that the pump body has a threaded nipple portion 105 which extends through the wall of the reservoir R and is held in position by the nut 106 with the gasket 104 preventing leakage.

With the inlet port 114 in the position as shown in FIG. 5, and the direction of drive clock-wise as shown in FIG. 5, the lubricant will be forced from the right of underside of passageway 116 to the left underside of passageway 118, tubing 120 and the outlet connection 121. The tubing will be sealed by an O ring seal 127 held in position by the bushing or insert 128 (see FIG. 3).

The other end of the tube 120 will be held in lubricant-tight fashion at 129 in the threaded nipple 130 of the outlet connection 121. The threaded nipple 130 extends through the wall 122 of the reservoir R and is held in position by the nut 123 which clamps the shoulder 131 against the gasket 132, which gasket is compressed against the wall 122 of the reservoir R.

In the reservoir shown in FIG. 4, there is a tubular oil level indicator which indicates the oil level.

Although the size and dimension of these constant pressure rotary lubricators may be widely varied, the eccentric groove may be 0.05 inch deep and precision ground into a 5/16 diameter drive shaft.

The spring-loaded plungers D and U will act as a separator between the inlet bore 115 and the outlet bore 118 and will maintain a pressure of about 20 pounds per square inch.

As the shaft rotates counter-clock-wise, lubricant is drawn into the sweep area of the eccentric groove and is discharged through the outlet passage.

If the outlet pressure tends to rise above 20 pounds per square inch, due to higher external resistance, the plunger valve D or U will rise slightly against the spring, allowing some oil to by-pass back to the inlet side and thus maintain the outlet pressure at the desired level.

The discharge from these lubricators may range from zero to 20 cc. per minute at 20 pounds per square inch, depending upon system resistance and the oil viscosity.

In the preferred applications, the pump discharge should be in the range of 5 cc. per minute or less and this volume may be distributed among either few or a large number of points.

Normally, before the inlet passage **115** there will be positioned an inlet filter and an inlet check valve.

The check valve when used prior to the inlet passageway will aid in the prevention of any tendency toward reverse flow of lubricant from the outlet of the pump to the inlet of the pump.

The adjustment at **110** in FIG. 4 or **38** in FIG. 2 will enable a correct setting of the desired outlet pressure.

The constant pressure rotary lubricators of the present invention provide a compact unit which will provide fully automatic and continuous lubrication at a very low flow rate for machines with precision ways and other bearing surfaces.

There will be no danger of rise and fall of the table of such machines due to periodic hand oiling or cyclic lubrication and objectionable dripping of oil from the ways will be avoided, as occurs when a high volume flood system is employed.

Thus the present lubricator avoids variations in precision cuts caused by rise and fall because of variations in oil film thickness and it also avoids constant oil dripping from ways which causes dirty and unsafe working conditions.

In addition to lubrication of precision ways, the units of the present invention are particularly suitable for the lubrication of medium speed, anti-friction bearing, optical and testing machinery, gauging equipment or any other mechanisms where oil film thickness variations may affect operational precision.

The outlet of the pump of FIGS. 1 to 5 may supply flow metering restriction fittings of the type shown in FIG. 6.

These fittings have a hexagon body **500** with an inlet strainer **501**. The inlet end of the body **500** is connected by the compression coupling sleeve **502** to the inlet **503** which abuts the insert **504** holding the strainer **501** in position. The compression coupling nut **505** will compress the sleeve **502** around the tube **503**.

The bore **507** is substantially completely filled by the pin **506**. The outlet end of the body **500** has an external pipe thread permitting mounting upon a bearing with an outlet check valve **508** in the outlet socket.

The outlet check valve is sealed by a spring **509** and held in position by the spring retainer **510**.

The rate of drive speed of the lubricators may be about 1725 r.p.m. with and directly connected 1/50 H.P. motor or by a belt or chain drive from a rotating shaft of the machine.

With the inlet valve positioned at **114** in FIG. 5 there can be no reverse flow of lubricant passed through the eccentric recess or groove when it is at a position of 180° from that shown in FIG. 2. However even without an inlet check valve the inertia of the lubricant flow is such and the speed of the pump is so high and the crevice offered to reverse flow is so restricted that even though there may be an opening through the crevice at the inlet and outlet ports, there will be no reverse flow of lubricant. It is true that there will be a pulsation at the outlet with maximum flow taking place at one part of the cycle and minimum flow at the other part of the cycle when the position of the parts is 180° from that shown in FIG. 2 but at no time will there be any substantial reverse flow in view of the inertia and viscosity of the lubricating oil which is being fed through the pump. This inertia effect is increased by the position of the bores **54** and **55** which cover the upper half of the shaft B as shown in FIGS. 1 and 2 and leave only a narrow crevice when the eccentric recess is 180° away from the position shown. As a result there will be no tendency toward blow-back.

The lubricators are equipped with either a one-pint or six-pint reservoir and with a one-pint lubricator. The overall dimension may be 4⅛" x 4½" x 3" exclusive of drive shaft extension.

With a six-pint lubricator, the overall dimensions may be 5⅛" x 6½" x 9½".

The approximate stroke of the piston or valve will be about ¼" and the piston or valve may be stepped if desired.

It is thus apparent that the applicant has provided a small, compact, reliable rotary pump which may supply either at constant volume or constant pressure to give a reliable lubricant feed in association with a machine structure in accordance with its lubricant requirements.

While there has herein been illustrated and described the preferred embodiment of the invention, it is to be understood that applicant does not limit himself to the precise construction herein disclosed, and the right is reserved to all changes and modifications coming within the scope of the invention as defined in the appended claims.

Having now particularly described and ascertained the nature of the invention, and in what manner the same is to be performed, what is claimed is:

1. A rotary pump of a central lubricant distributing system, comprising a pump body having a first bore extending therethrough, said first bore having a diameter and a longitudinal axis, a rotatable shaft in said first bore, said shaft having a longitudinal axis, two ends and a periphery, means to rotate said shaft, said shaft further having a groove intermediate the ends thereof, said groove being crescent-shaped and having a maximum portion and a minimum portion and being formed by a recessed cylindrical portion of said shaft having an axis offset from the longitudinal axis of said shaft and having a periphery tangent to the periphery of said shaft at a point corresponding to the minimum portion of said groove, said groove and said first bore forming a pump compartment, a second bore having a longitudinal axis transverse to the axis of said first bore and leading to said pump compartment, a vane reciprocatable in said bore and having a bottom end extending into said groove, a spring biasing said vane in the direction of said groove and maintaining said bottom end in continuous contact with the periphery of said recessed portion of said shaft to divide said pump compartment into two separate chambers, an inlet passageway communicating with one of said chambers, an outlet passageway communicating with the other of said chambers, said inlet and outlet passageways each having a smaller diameter than said first bore and each having a longitudinal axis transverse to the axis of said first bore and to the axis of said second bore, the upper portion of the pump body wall defining each of said inlet and outlet passageways being in line with the bottom end of said vane when said bottom end is in contact with the periphery of said recessed cylindrical portion of said shaft at the maximum portion of said groove, and the lower portion of the pump body wall defining each of said inlet and outlet passageways substantially coinciding with the axis of said first bore.

2. A rotary pump according to claim 1, comprising, further, means operatively connected to said spring for adjusting the tension thereof.

3. A rotary pump according to claim 1, wherein said shaft has a shoulder at one end forming a thrust bearing in one direction, and said pump body has a plug at the other end of said shaft forming a thrust bearing in an opposite direction.

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