April 27, 1965 A. E. BROWN ETAL 3,180,178

VARIABLE STROKE RECIPROCATING MACHINE

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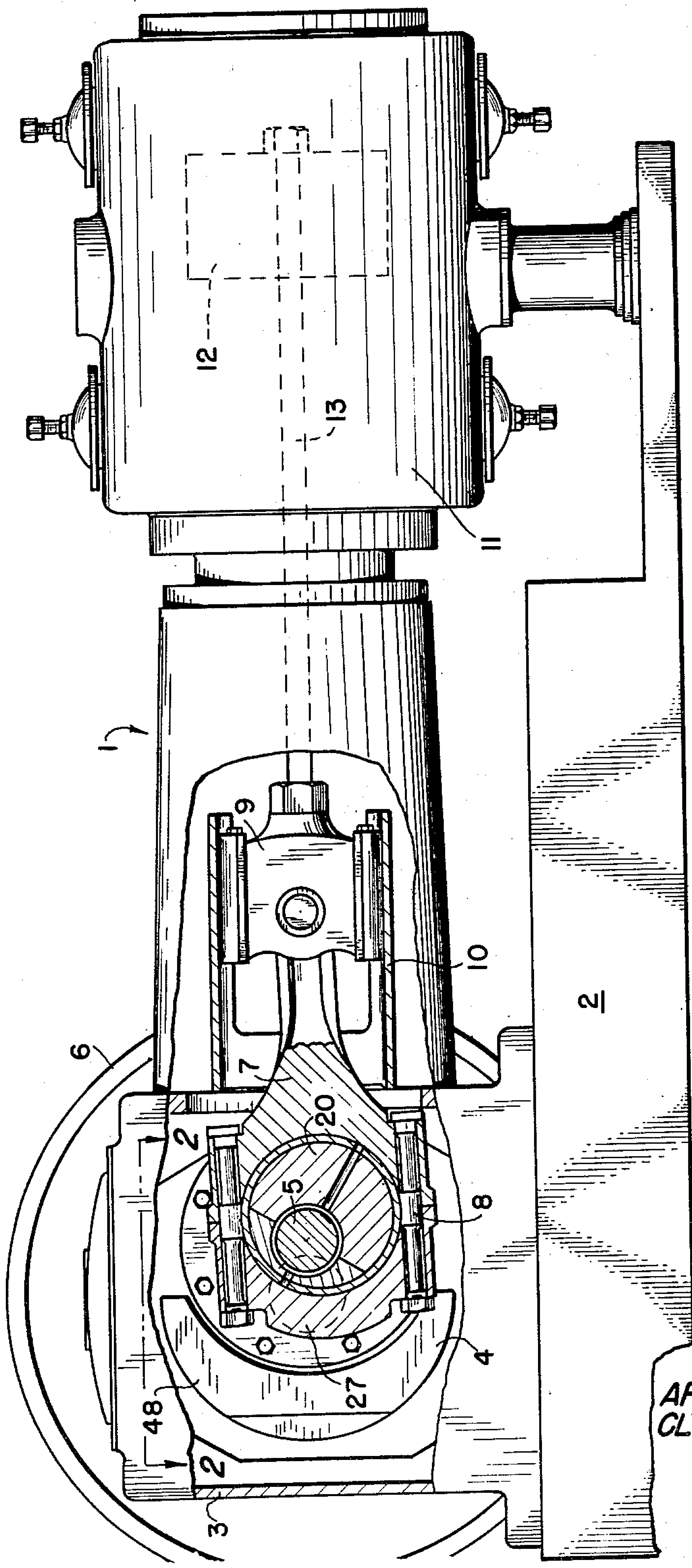


FIG. 1

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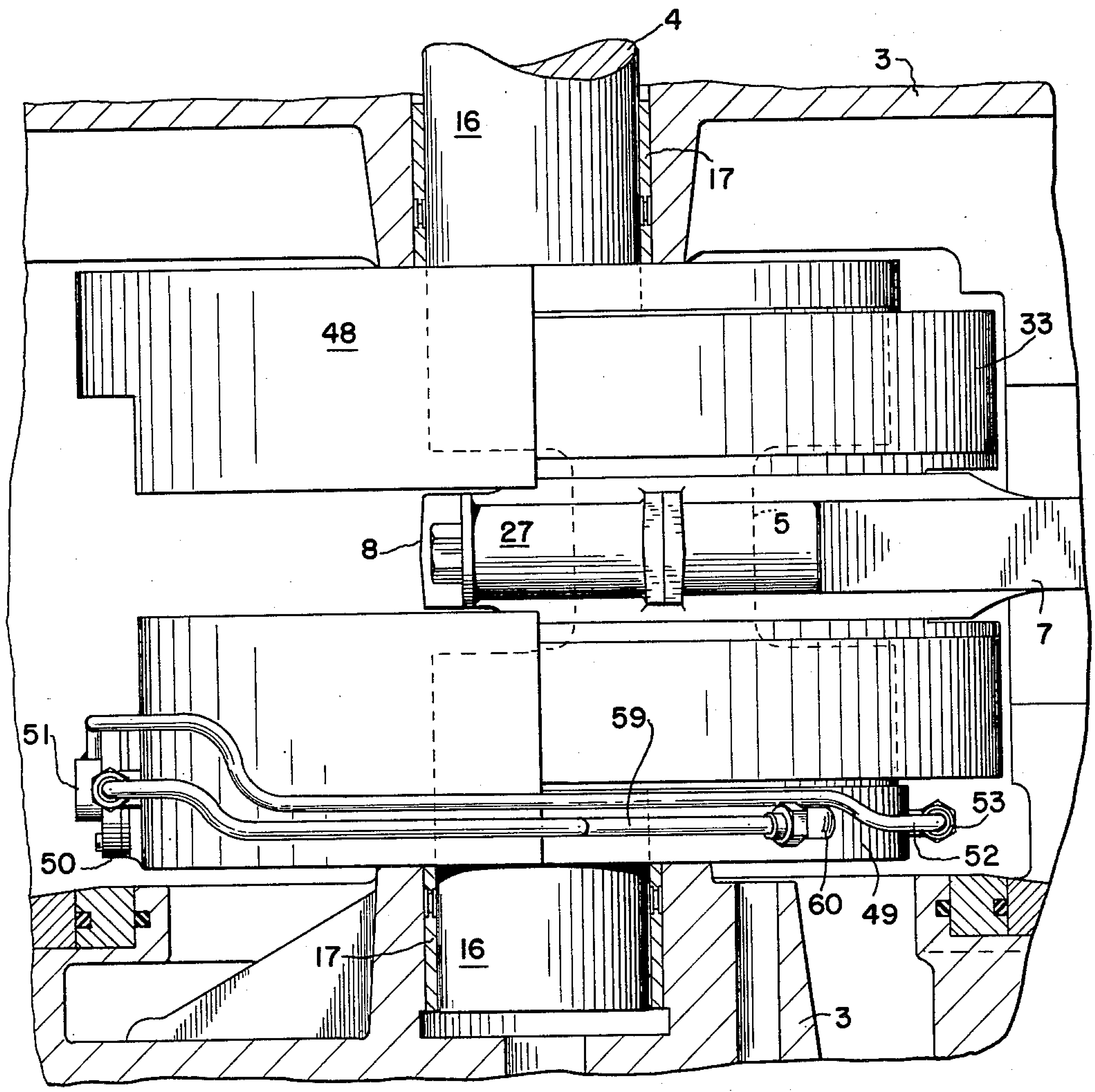


FIG. 2

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VARIABLE STROKE RECIPROCATING MACHINE

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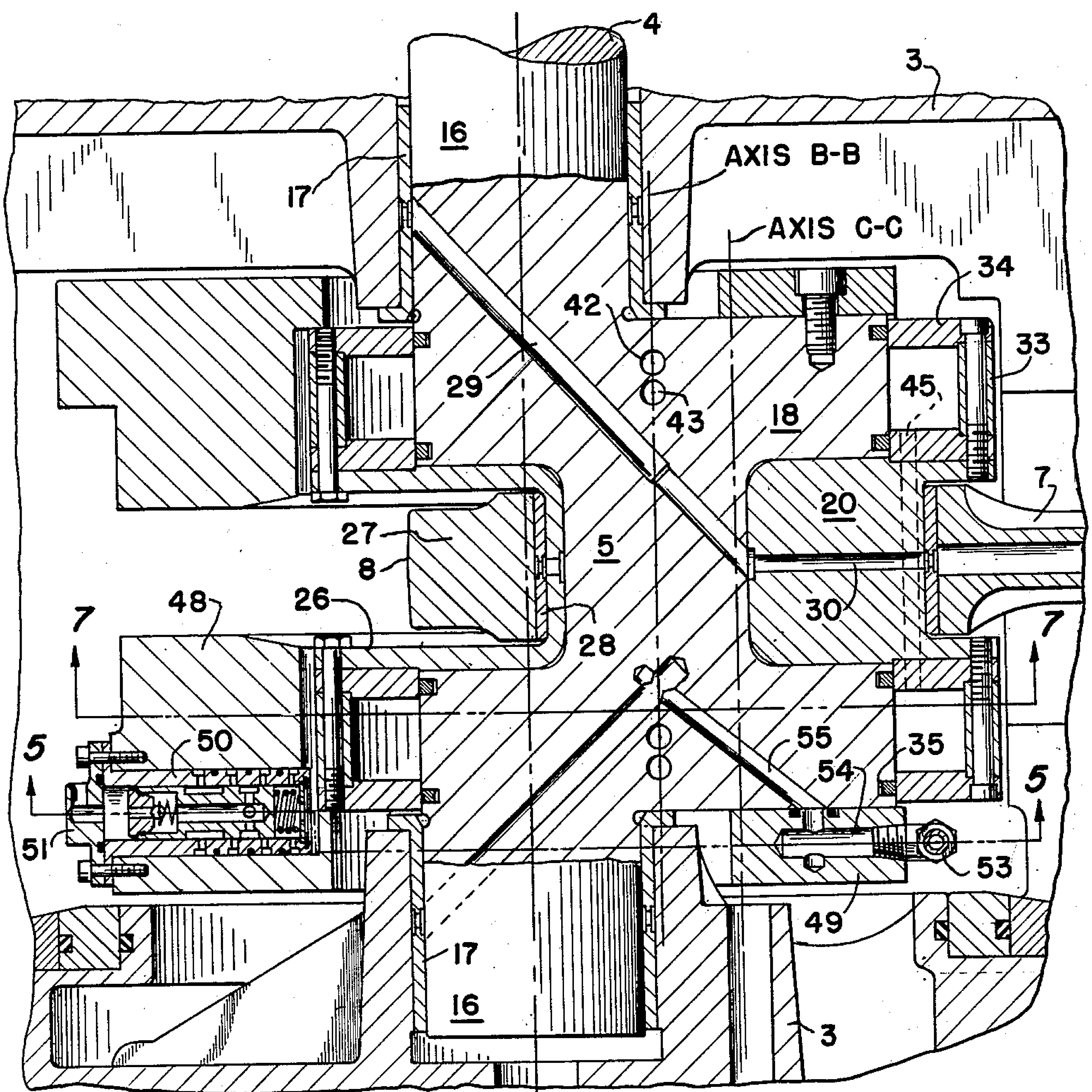


FIG. 3

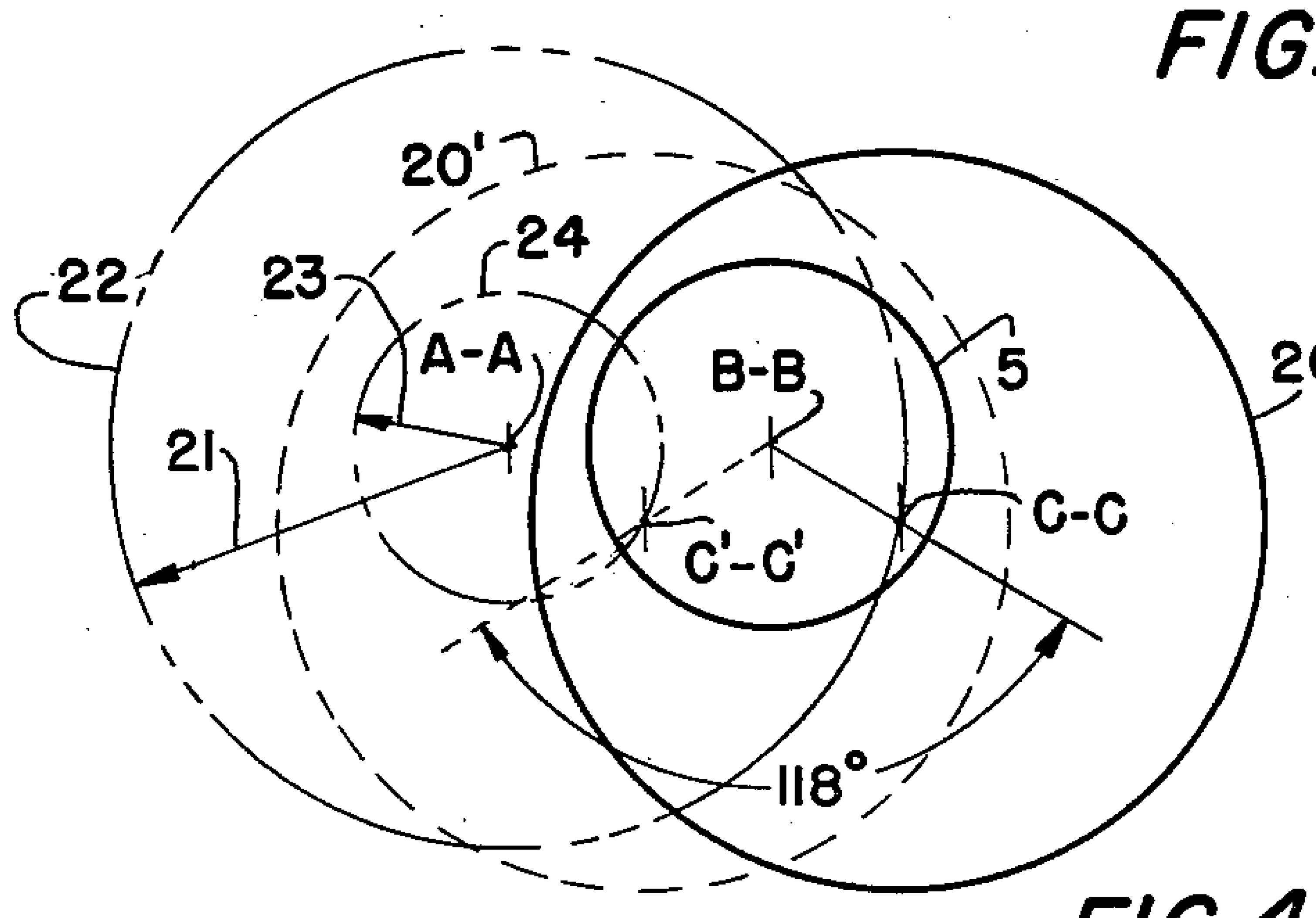


FIG. 4

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VARIABLE STROKE RECIPROCATING MACHINE

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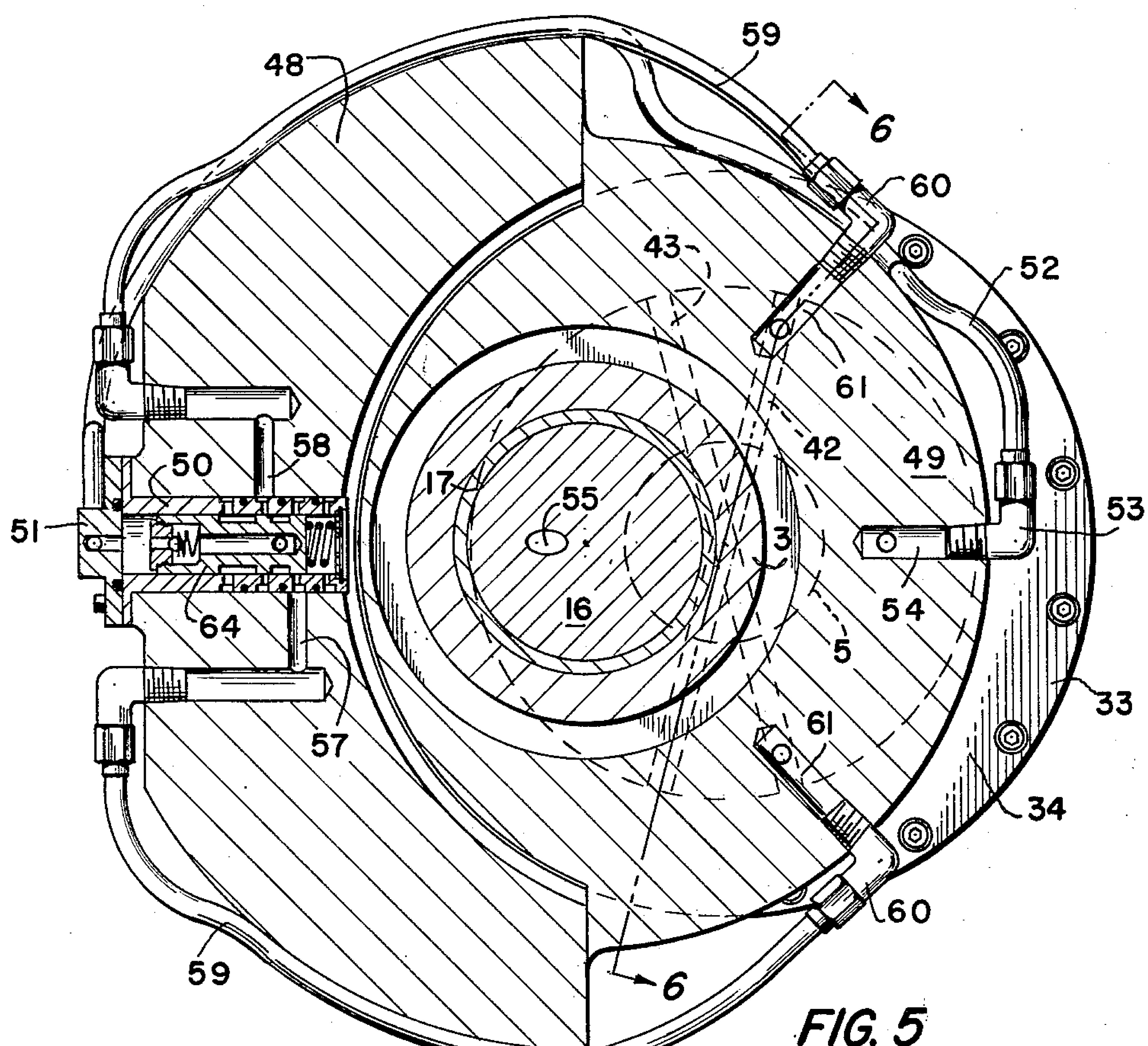


FIG. 5

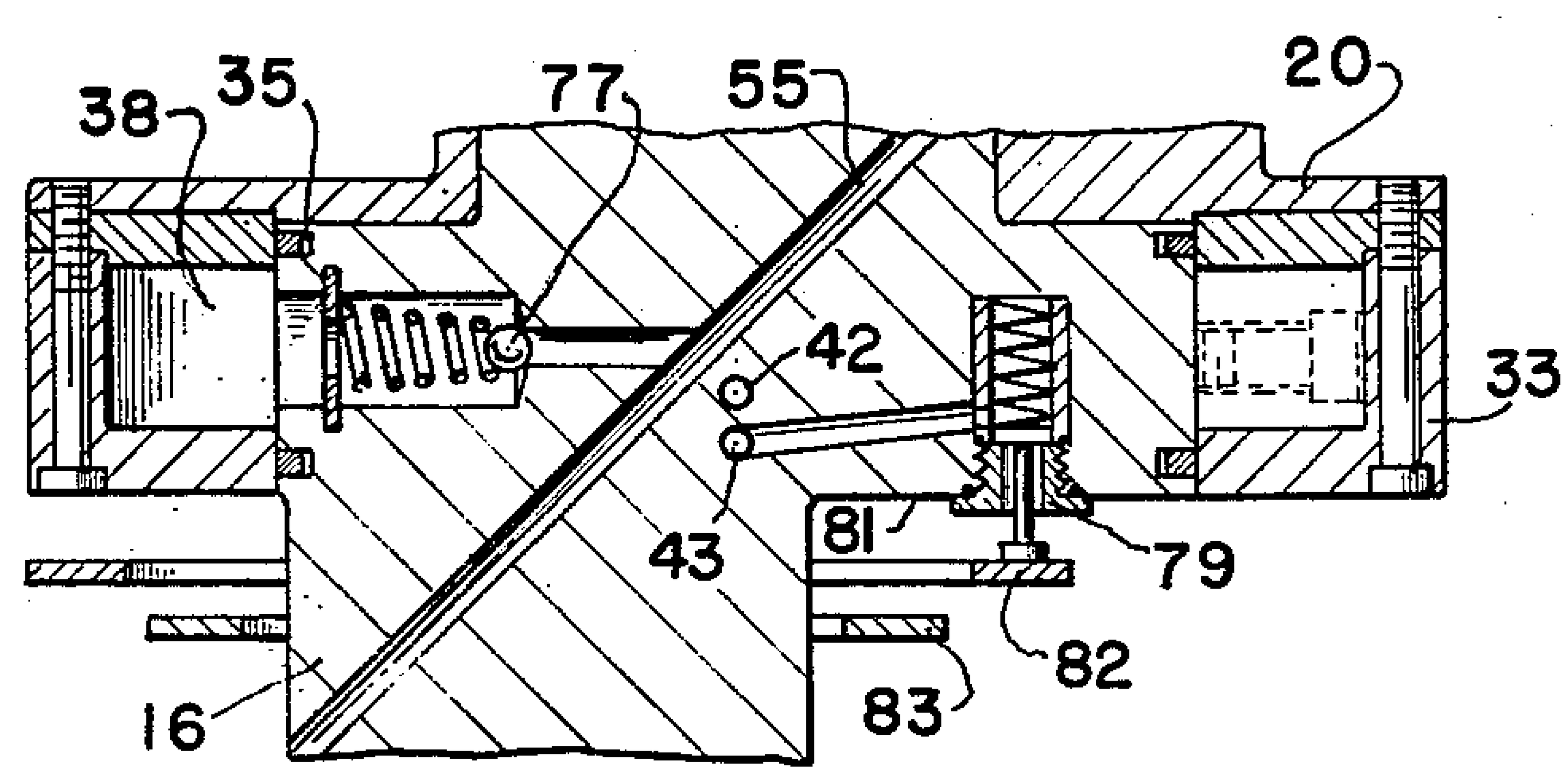


FIG. 10

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VARIABLE STROKE RECIPROCATING MACHINE

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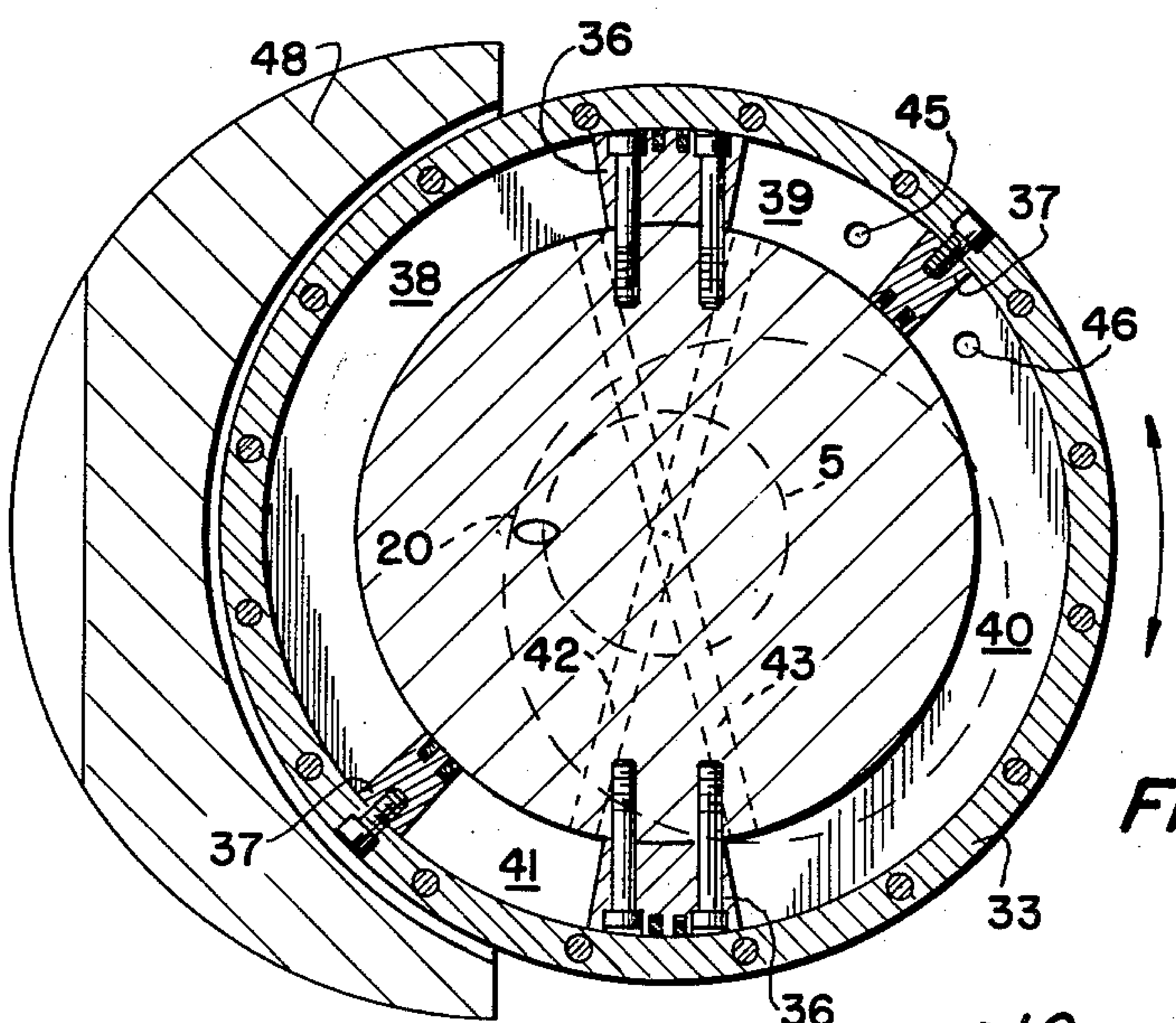


FIG. 7

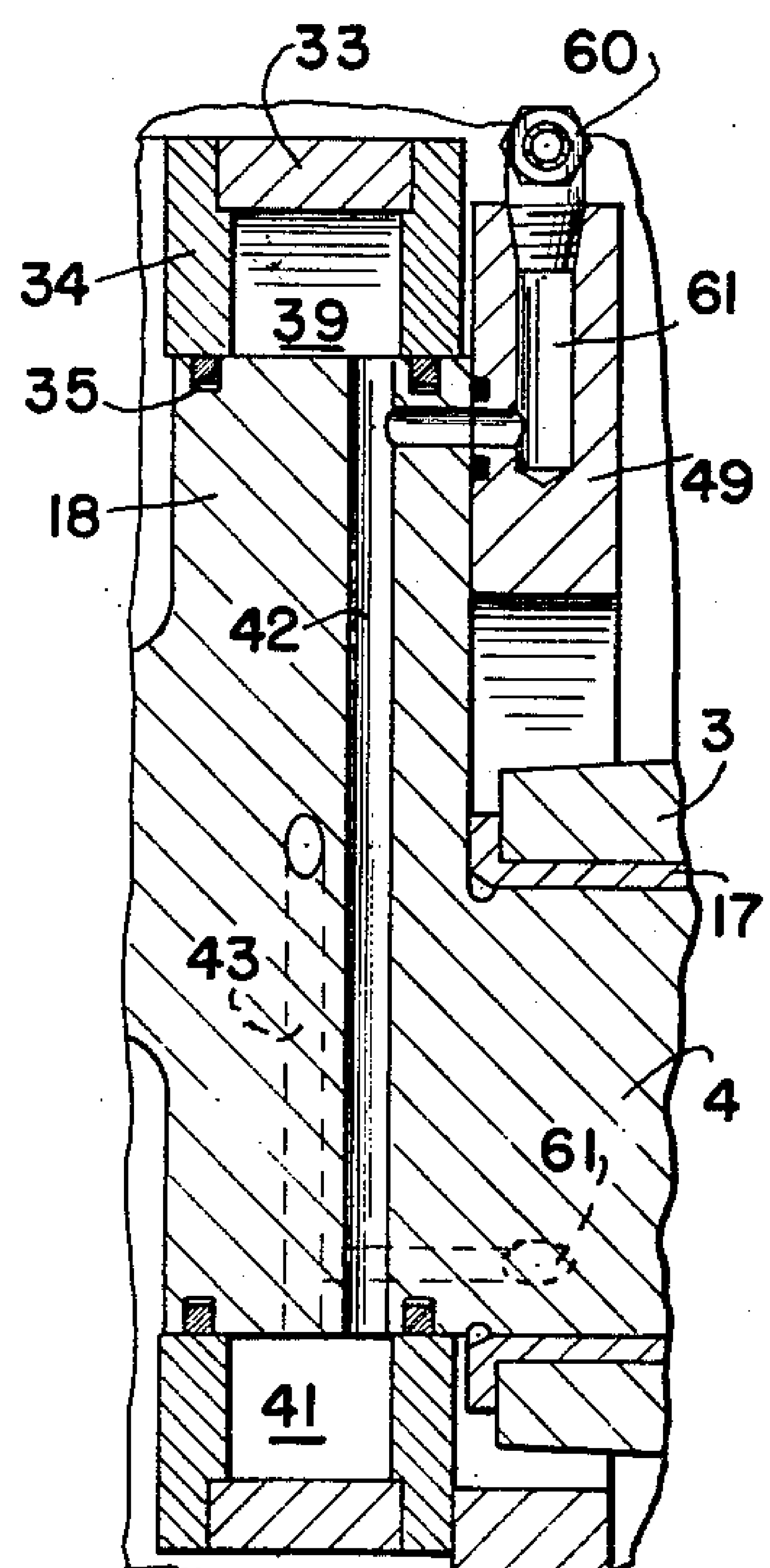


FIG. 6

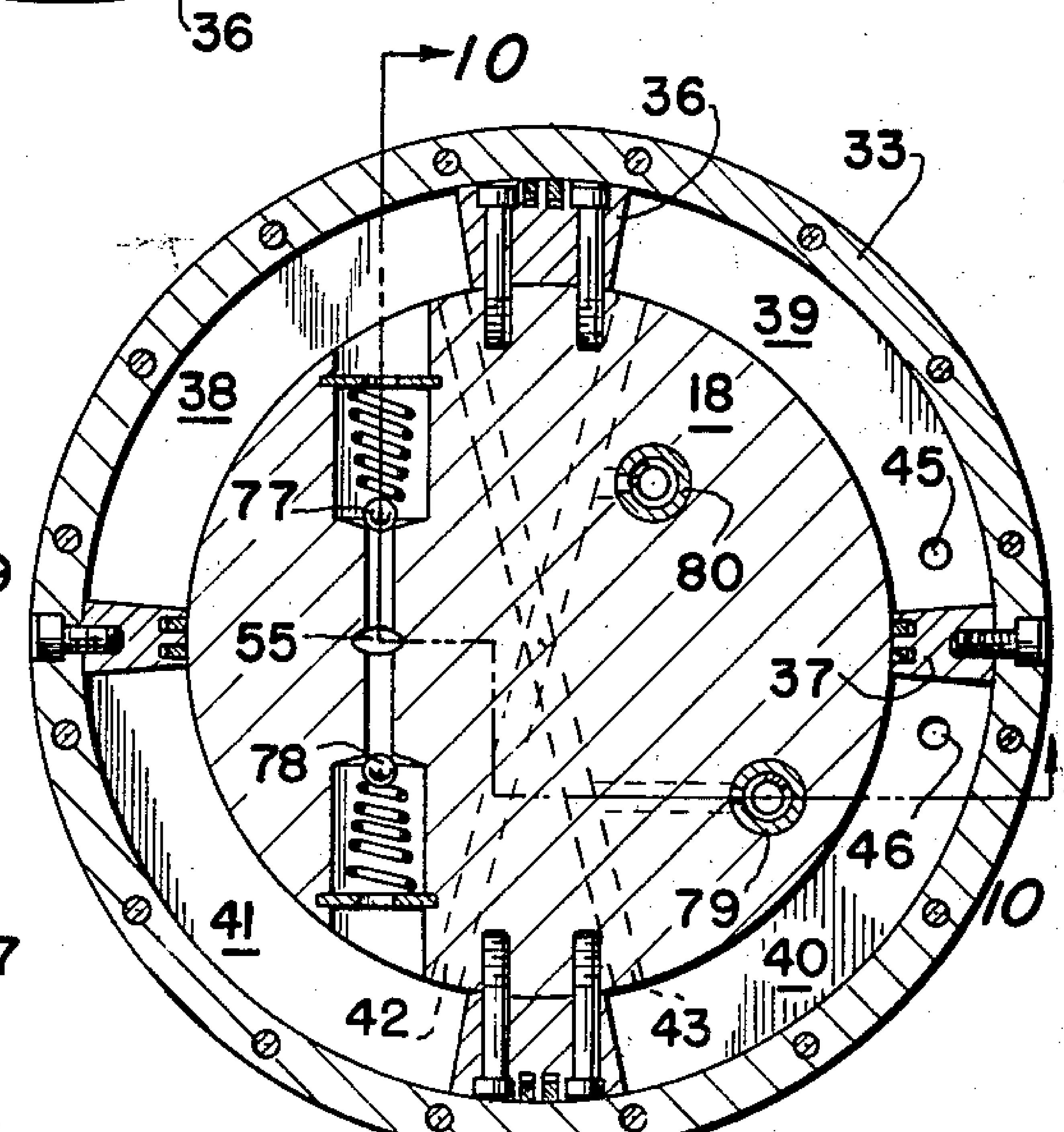


FIG. 9

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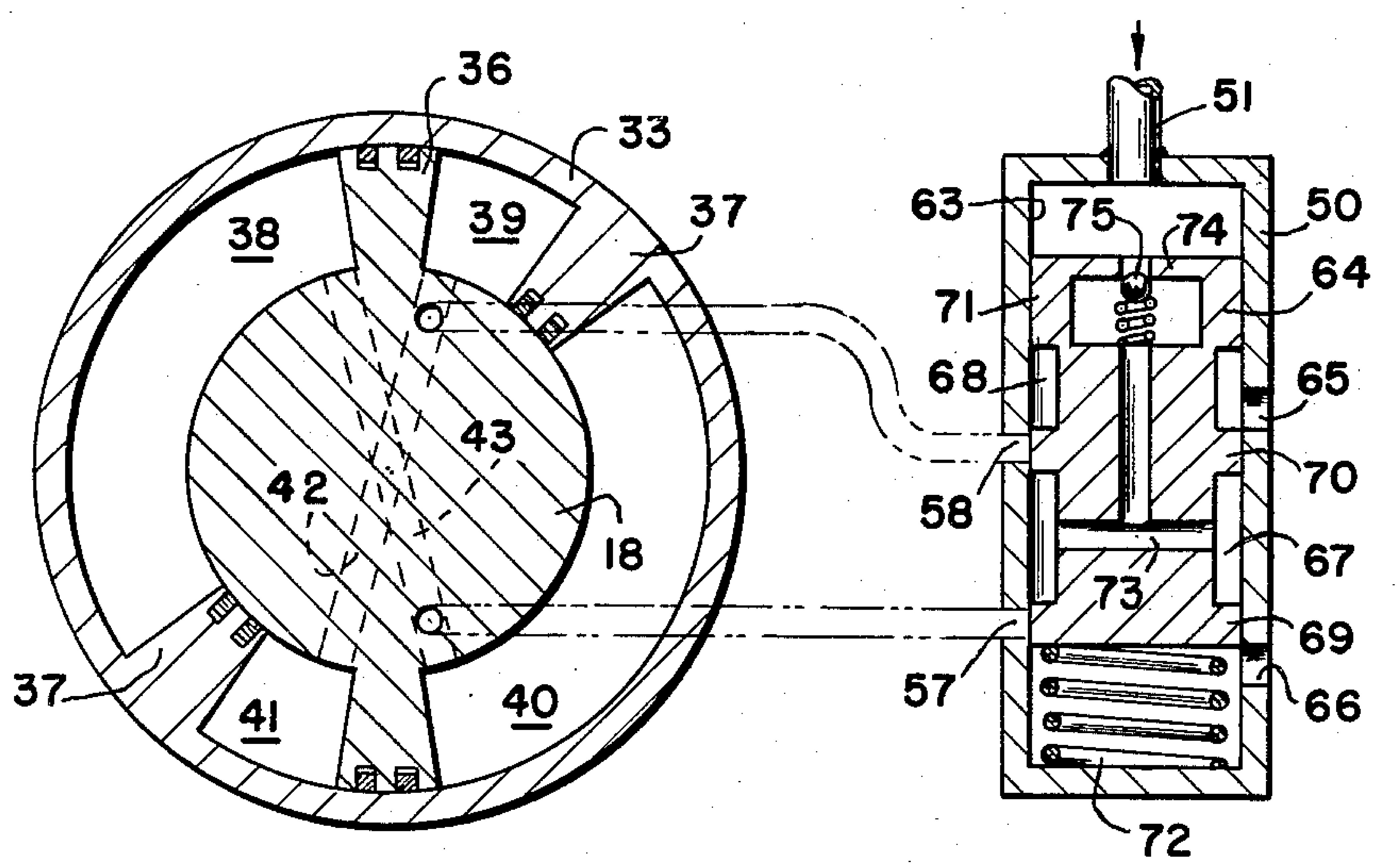


FIG. 8

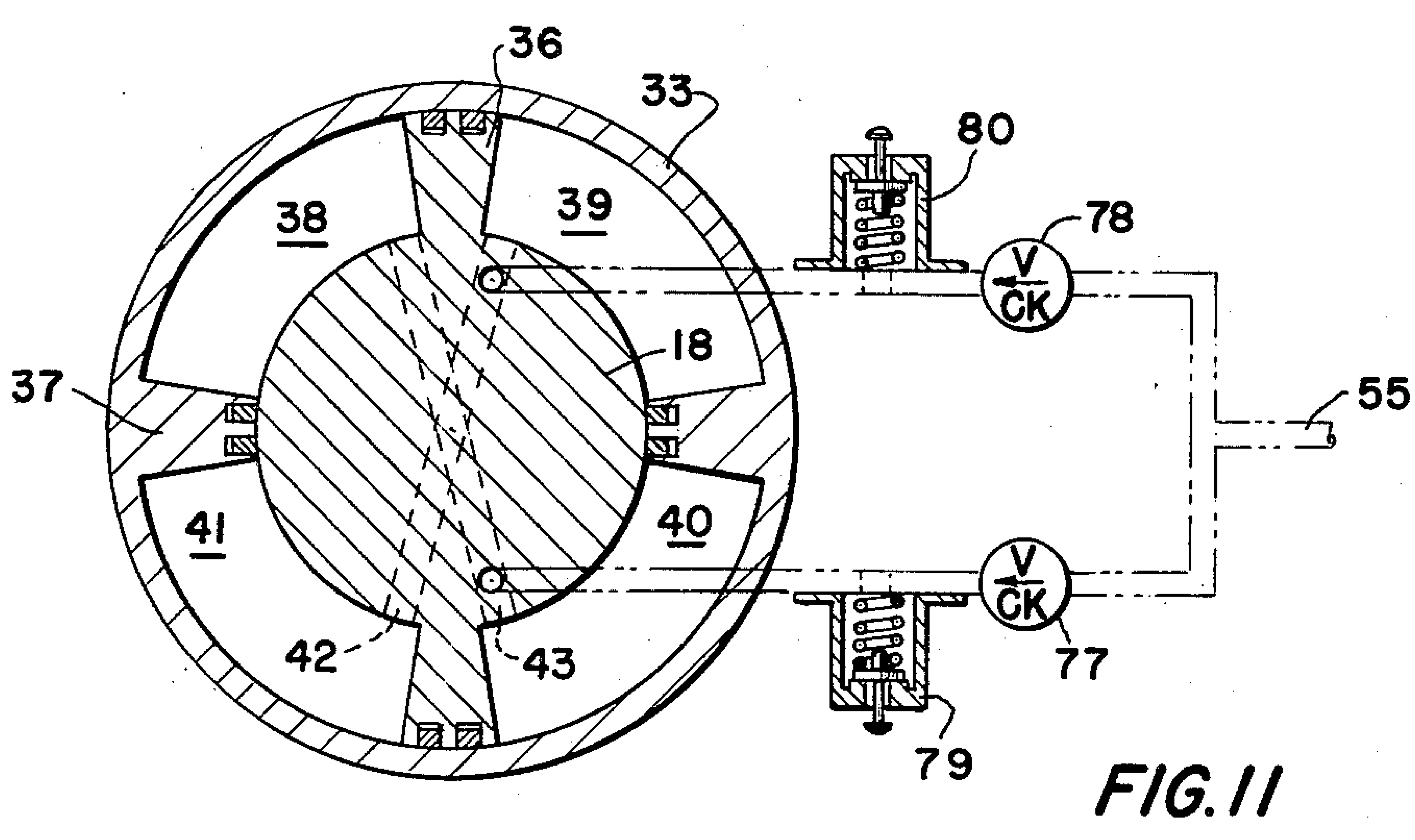


FIG. 11

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# VARIABLE STROKE RECIPROCATING MACHINE

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8 Claims. (Cl. 74—600)

This invention relates to apparatus for varying the stroke of a reciprocating piston machine, such as a pump or compressor, during its operation, to vary the displacement volume of such machine. When employed in a pump or compressor, the variable stroke apparatus of this invention is useful for varying the fluid handling capacity of the pump or compressor.

Conventional variable stroke mechanisms for reciprocating piston machines employ a linkage between the crankshaft and piston of the machine. The use of the linkage complicates the machine and substantially increases the weight, bulk and cost of the machine, as compared to a constant stroke machine. Furthermore, the linkage-type of variable stroke mechanism normally cannot be installed in a conventional constant stroke reciprocating piston machine, to convert it to a variable stroke machine, since the linkage is much too bulky to go into the normal crankcase.

The principal object of this invention is to substantially eliminate or minimize the foregoing disadvantages and objections to the prior art variable stroke mechanisms.

Other important objects are: to provide a variable stroke mechanism which is effective to vary the stroke of a reciprocating piston machine while it is continuously operating; to provide a variable stroke mechanism which can be readily installed in a conventional constant stroke reciprocating machine; to provide a variable stroke mechanism which is relatively simple, small and economical; to provide a variable stroke mechanism which is operated by the lubricating system of the reciprocating piston machine containing such mechanism; to provide a variable stroke mechanism which will fit into the crankcase of a conventional reciprocating piston machine; and to provide a variable stroke mechanism which is particularly adapted for use in varying the fluid handling capacity of a constant speed compressor.

In general, the above objects are attained by pivoting an eccentric on the crank pin of the crankshaft in a reciprocating piston machine, journaling the connecting rod of the machine on the circumference of the eccentric and providing a hydraulic fluid actuated means for rotating the enccentric relative to the crank pin through a range of limited angular movement. This hydraulic means is mounted on the crankshaft, rotates with it and normally holds the eccentric in a stationary relative position to the crank pin except during the time that the stroke length of the reciprocating machine is being changed.

The invention is illustrated in the accompanying drawings wherein:

FIG. 1 is an elevational view with portions being broken away and shown in section of a variable stroke reciprocating compressor constructed in accordance with this invention;

FIG. 2 is an enlarged fragmentary view taken on the line 2—2 in FIG. 1;

FIG. 3 is a sectional view of FIG. 2 taken along the axis of the crankshaft;

FIG. 4 is a diagrammatic view illustrating the eccentric stroke adjusting mechanism in two different adjusted positions on the crank pin;

FIG. 5 is a section taken along the line 5—5 of FIG 3;



FIG. 6 is a section taken along the line 6—6 of FIG. 5;

FIG. 7 is a section taken along the line 7—7 of FIG. 3;

FIG. 8 is a diagrammatic view of the stroke adjustment system of the embodiment shown in FIGS. 1 to 7;

FIG. 9 is a sectional view similar to FIG. 7 of another embodiment of the invention;

FIG. 10 is a section taken on line 10—10 of FIG. 9; and

FIG. 11 is a diagrammatic view similar to FIG. 8 of the embodiment shown in FIGS. 9 and 10.

The reciprocating air compressor **1** shown in FIG. 1 includes the following conventional structure: a base or foundation **2**; a crankcase **3**; a crankshaft **4** rotating in the crankcase **3** and having a crank pin **5**; a flywheel **6** fixed on one end of the crankshaft **4**; a connecting rod **7** having its big end **8** surrounding the crank pin **5** and its eye end pivoted to a crosshead **9** sliding in the crosshead guides **10**; a cylinder **11** containing a sliding piston **12**; and a piston rod **13** interconnecting the crosshead **9** and the piston **12**.

Looking at FIGS. 2 and 3, the crankshaft **4** includes a pair of axially spaced main journals **16** rotating in main bearing shells **17** about an axis A—A. The bearing shells **17** are mounted in the crank case **3**. The crank pin **5** is located along the axis B—B, which is radially offset from the axis A—A, and is supported at its ends by a pair of integral crankshaft webs **18** extending radially between the main journals **16** and the crank pin **5**. The crankshaft webs **18** are circular in circumference and concentric with the crank pin axis B—B. The reason for the crankshaft webs **18** having a circular circumference is seen later.

The stroke of the piston **12**, crosshead **9** and connecting rod **7** is rendered adjustable or variable in length by pivoting an eccentric **20** on the crank pin **5** of the crankshaft **4** for limited rotary movement and journaling the lower end of the connecting rod **7** on the circumference of the eccentric **20**. The eccentric has an axis C—C which is shown in FIG. 3 relative to the crankshaft axis A—A and the crank pin axis B—B. The stroke of the piston **12** is varied by rotating the eccentric **20** on the crank pin **5** through a range of limited rotary movement to vary the distance between the crankshaft axis A—A and the eccentric axis C—C.

FIG. 4 diagrammatically illustrates how the eccentric **20** varies the length of the piston stroke. In FIG. 4, the eccentric is shown in two different adjusted positions on the crank pin **5**. In the solid line showing of the eccentric **20**, its axis C—C is spaced from the crankshaft axis A—A by the radius **21** and the path of the axis C—C about the crankshaft axis A—A, as the crankshaft **4** rotates, is indicated by the phantom line circle **22**. In this postion of the eccentric **20** on the crank pin **5**, the stroke of the piston **12** corresponds to the diameter of the phantom line circle **22**, or double the radius **21**.

The second adjusted position of the eccentric in FIG. 4 is shown in dotted lines and indicated by the reference number **20'**. In this second eccentric position **20'**, the axis C'—C' of the eccentric is rotated approximately 118 degrees about the crank pin **5** from the axis C—C of the solid line position of the eccentric **20**. The eccentric axis C'—C' of the dotted line eccentric position **20'** is located at a radius **23** from the crankshaft axis A—A and travels about the axis A—A along the phantom line circle **24**, as the crankshaft rotates. In this eccentric position **20'** on the crank pin **5**, the stroke of the piston **12** corresponds to the diameter of the circle **24**, or double the radius **23**. It is readily apparent that the stroke of the piston **12** is much smaller in the dotted line position **20'** of the eccentric. It also should be apparent that the stroke can be varied from maximum to minimum by rotating the eccentric 20 on the crank pin through an angle of 180° and that the eccentric can be adjusted to provide any length stroke desirable between the maximum and minimum stroke lengths available by adjusting the eccentric to some point located between the 180° angular range of maximum and minimum stroke lengths. It should be understood that once the eccentric 20 is adjusted properly to provide the desired stroke length, it is locked in that position on the crank pin 5 and remains fixed relative to the crank pin 5 until further adjustment of the stroke length is desired. Hence, once the eccentric 20 is fixed in relative stationary position on the crank pin 5, the eccentric 20, in effect, serves as the crank pin which reciprocates the connecting rod 7 and its connected elements back and forth.

The eccentric 20 is formed in half sections to enable it to be mounted on the crank pin 5 and includes a pair of side flanges or rims 26 which extend radially outwardly from the opposite sides of the circumference of the eccentric 20. The rims 26 seat against the opposed end faces of the crankshaft webs 18 holding the crank pin 5, as shown in FIG. 3, project radially beyond the crankshaft webs 18 and have outer circumferential edges which are circular and concentric with the circumferences of the crankshaft webs 18. Hence, the circumferences of the rims 26 are eccentric to the circumference of the eccentric 20.

The big end 8 of the connecting rod 7 is conventional and includes a removable cap 27 bolted to the connecting rod body and holding a shell bearing 28 which embraces the circumference of the eccentric 20.

The lubrication system for the compressor is conventional. The main bearing shells 17 are supplied with lubricant under pressure via a passage (not shown) from a conventional lubricant pump. From the main bearing shells 17, the lubricating fluid, normally oil, flows through the passageway 29 in the crank shaft 4 to the surface of the crank pin 5, thence through the hole 30 in the eccentric 20 to the connecting rod shell bearing 28 and into a rifle-drilled passage in the connecting rod 7 to lubricate the crosshead 9 and various other elements attached to the outer end of the connecting rod 7.

The eccentric 20 is moved through a limited range of angular movement on the crank pin 5 and locked in adjusted position by hydraulic motor means, utilizing the lubricating pressure of the lubricating system as a motive force. An annular hollow housing or cage 33 is rotatively mounted on the circumference of each crankshaft web 18 and is bolted to the adjacent flange or rim 26 of the eccentric 20. Each cage 33 is U-shaped in cross-section, opens radially inwardly and includes a pair of side walls 34 which seat and rotate on seal rings contained in annular sealing grooves 35 formed in the circumference of each crankshaft web 18.

The annular space in each cage 33 is segregated into four chambers by two pair of vanes 36 and 37. The pair of vanes 36 are fixed on the crankshaft web 18 at diametrically opposite points and slidably engage the interior of the cage 33 while the vane pair 37 are fixed to the cage 33 at diametrically opposite points and slidably engage the crankshaft web 18. The semi-arcuate shaped chambers formed by the vanes 36 and 37 are designated 38, 39, 40 and 41. Looking at FIG. 7, it will now become clear that hydraulic pressure introduced to the chambers 38 and 40 will act on the vanes 37 and cause the cage 33 to rotate around the crankshaft web 18 in a counterclockwise direction and hydraulic pressure in the chambers 39 and 41 will cause the cage to rotate in a clockwise direction. The pair of chambers 38 and 40 are interconnected by a diametrical passage 43 bored in the crankshaft web 18 and the chambers 39 and 41 are interconnected by another diametrical passage 42 which is axially spaced from the passage 43. The diametrical passages 42 and 43 open into their chambers near the crankshaft mounted vanes 36 so that they are not obstructed by the movement of the cage vanes 37. It will be understood that each of the cages 33 and the passages 42 and 43 are identical on both crankshaft webs 18.

Since the flanges or rims 26 of the eccentric 20 are attached to each cage 33, the rotation of the cages 33 move the eccentric 20 around the crank pin 5 over a range of limited angular movement. Due to the use of four vanes in each cage 33, the angular movement of the cage 33 is limited to a little less than 180° about the crankshaft web 18, which is sufficient to adjust the eccentric 20 through a range providing the maximum stroke length and the minimum stroke length to the compressor piston 12.

The chambers 39 of both cages 33 are interconnected for communication by a longitudinal passage 45 extending through the thick part of the eccentric 20 and the chambers 40 are also interconnected by a like longitudinal passage 46 extending through the eccentric 20. The passages 45 and 46 extend through the cage side walls 34 and open into their respective chambers 39 and 40 near the cage vane 37 as shown in FIG. 7. Thus, all of the counter-clockwise rotary passages 38 and 40 in both cages 33 are interconnected together for communication and the clockwise rotary passages 39 and 41 in both cages are likewise interconnected. For example, hydraulic pressure introduced to the clockwise chamber 39 of one cage 33 will flow through the diametrical passage 42 to the chamber 41 of that cage and also through the longitudinal passage 45 to the corresponding chambers 39 and 41 of the other cage 33.

When all of the chambers 38 to 41 of each cage 33 are filled with hydraulic fluid under equal pressure, the cages are locked in whatever position they may be in. Thereafter, the cages 33 may be moved in one direction or the other by releasing hydraulic fluid from one set of chambers and adding fluid under pressure to the other set of chambers. This is the method that is used for moving the cages 33 between various adjusted positions on the crankshaft webs 18.

One embodiment for introducing and releasing hydraulic fluid from the chambers of the cages 33 is illustrated in FIGS. 2 to 8. The crankshaft 4 includes a pair of counterweights 48 carried by rings 49 which are bolted to the outer faces of the crankshaft webs 18. A spool valve 50 is contained in one of the counterweights 48 and includes a hydraulic fluid inlet 51 which receives lubricating fluid under pressure from the lubricating system of the compressor. The inlet 51 is connected by a pipe 52 running from an angle fitting 53 which is threaded into a passageway 54 located in the counterweight ring 49. The passageway 54 communicates with a lubricating fluid passage 55 bored in the crankshaft 4 and running from a main bearing shell 17. Thus, the lubricating fluid under pressure in the main bearing shell 17 can flow, in sequence, through the crankshaft passage 55, the passageway 54 in the counterweight ring 49, the angle fitting 53, the pipe 52 and into the inlet 51 of the valve 50.

The spool valve 50 includes a pair of outlets 57 and 58 which communicate with the chambers 38 to 41 of the cages 33, with the outlet 57 connected to the counterclockwise chambers 38 and 40, as seen in FIG. 8, and the outlet 58 connected to the clockwise chambers 39 and 41. As shown in FIGS. 3 and 6, the outlets 57 and 58 are connected by pipes 59 to respective angle fittings 60 which are threaded into respective passages 61 formed in the counterweight ring 49 and communicating with the adjacent diametrical passages 42 and 43 located in the crankshaft web 18. In operation, the spool valve 50 acts to close both of the outlets 57 and 58 to hold the cages 33 in their adjusted position, or to vent one of the outlets 57 and 58 to release and dump its hydraulic pressure while feeding hydraulic pressure to the other outlet to cause the cages 33 to move on the crankshaft webs 18.

The construction and operation of the spool valve **50** is best shown in FIG. 8. The spool valve includes a cylindrical bore **63** slidably containing a spool **64**. The outlets **57** and **58** open into the bore **63** through the side of the valve **50** at longitudinally spaced points and the inlet **51** opens into the end wall of the valve **50**. The side wall of the valve **50** also contains a pair of vent ports **65** and **66** which are longitudinally spaced from each other and from both of the outlets **57** and **58**.

The circumference of the valve spool **64** contains two spaced annular grooves **67** and **68** bordered by three spaced enlarged portions or annular ridges **69, 70** and **71**. The spool **64** is biased outwardly toward the inlet **51** by a spring **72** at its other end and contains a passageway **73** leading from its inlet end face **74** to the groove **67** on its circumference. A ball check valve **75** is mounted in the passageway **73** to allow hydraulic fluid to flow into the passageway **73** from the inlet **51** and to prevent its reverse flow.

Normally, when the fluid pressure in the inlet **51** is relatively low or zero, the biasing force of the spring **72** is effective to force the spool **64** outwardly against the inlet **51**. In the position shown in FIG. 8, the spool **64** is in an intermediate position between its two extreme positions of movement in the bore **63**. This intermediate position is obtained by providing sufficient pressure to the inlet **51** to act against the inlet end face **74** of the spool **64** with a force which counterbalances the spring **72** in the intermediate position of the spool **64**. It will be noted that in the intermediate position shown, the spool ridges **69** and **70** are closing both of the outlets **57** and **58**, resulting in holding the fluid in the cage chambers in a static position. Thus, in the intermediate position of the spool **64**, the cages **33** and eccentric **20** are locked in whatever position they may be in.

The spool valve **50** may be actuated to supply fluid pressure to one of the outlets **57** and **58** while simultaneously venting the other outlet by changing the fluid pressure in the inlet **51** in either direction. For example, assume that it requires, say, 60 p.s.i. pressure in the inlet **51** to obtain the intermediate position shown in FIG. 8. If the pressure is then reduced to, say, 50 p.s.i., the spool **64** will be biased outwardly by the spring **72** so that pressure in the passageway **73** and the groove **67** of the spool **64** flows to the outlet **58**. Simultaneously, the outlet **57** is opened to the vent port **66**.

Conversely, if the pressure in the inlet **51** is increased to, say, 70 p.s.i., the spool **64** will be moved inwardly from its intermediate position to cause the fluid pressure to flow into the outlet **57** and to simultaneously dump the fluid from the outlet **58** through the vent **65**. Hence, the valve **50** is operable to actuate the cages **33** and eccentric **20** to any desired adjusted position merely by varying the pressure of the lubrication system of the compressor **1**. The variance of the lubrication pressure can be performed by a manually-operated pressure regulator (not shown) of conventional type, or by an automatically-operated pressure regulator controlled by the load on the air compressor **1**.

*Operation*

Assuming at the start of the description of operation, the compressor **1** is operating at constant speed, the eccentric **20** is located in the position shown in FIGS. 1 and 7, and shown in solid lines in FIG. 4, the lubrication pressure on the lubrication system is 60 p.s.i. and the valve spool **64** is in its intermediate position, as shown in FIG. 8, wherein it is closing both of the outlets **57** and **58**. In this position, all of the chambers **38** to **41** are filled with oil and the cages **33** and eccentric **20** are locked relative to the crank pin **5** so that the stroke length of the compressor **1** is constant or fixed at a relatively long stroke. As the crankshaft **4** rotates, the fluid pressure in the chambers **38** to **41** will alternately rise to high pressures and decrease to zero due to the torque loads exerted on



the crankshaft **4** by the piston **12**, but the cages **33** will not move on the crankshaft webs **18** because the oil in the chambers **38** to **41** is relatively incompressible.

Next, assume that as the compressor **1** continues to operate, its production of compressed gas exceeds the demand for the gas and that it is desirable to reduce the quantity of gas being compressed. This is carried out by reducing the length of the stroke of the piston **12** by moving the eccentric **20** to the dotted line eccentric position **20'** shown in FIG. 4. To reach the dotted line position **20'**, the eccentric **20** must be rotated in a clockwise direction and this is done by reducing the pressure of the compressor lubrication system to 50 p.s.i.

Looking at FIG. 8, reducing the lubrication pressure from 60 to 50 p.s.i. at the inlet **51** of the spool valve **50** causes the spool **64** to move from its intermediate position, shown in FIG. 8, outwardly toward the inlet **51**. This outward movement of the valve spool **64** simultaneously opens the outlet **58** to the lubrication pressure and the outlet **57** to the vent or drain **66**. As a result, hydraulic pressure is drained from the cage chambers **38** and **40** while additional hydraulic fluid flows into the cage chambers **39** and **41**, causing the cages **33** and the eccentric **20** to move in a clockwise direction, looking at FIGS. 4, 7 and 8. When the stroke length is reduced to the desired point, the lubrication pressure is again raised to 60 p.s.i. and the valve spool **64** returns to its intermediate position to close both of the valve outlets **57** and **58** and again lock the cages **33** and eccentric **20** in a relatively stationary position on the crank pin **5**.

During the time when the outlet **58** is open to the lubrication pressure, the crank torque on the crankshaft **4** may cause the pressure in the chambers **39** and **41** to exceed the lubrication pressure during certain angles of rotation of the crankshaft. When this occurs, the check valve **75** in the valve spool **64** will prevent the hydraulic fluid from reversing its flow direction and leaving the chambers **39** and **41**. This high pressure in the chambers **39** and **41**, caused by crankshaft torque, will decrease when the crankshaft is passing through its top dead center, and bottom dead center positions. At the latter positions of the crankshaft **4**, the lubrication pressure can readily change the position of the eccentric **20** on the crank pin **5**. Furthermore, in certain angles of the crankshaft **4**, the crank torque load should aid the movement of the eccentric **20** to its new relative position on the crank pin **5**. Hence, it is easily seen how the relatively low pressure of the lubrication system can be utilized for actuating the eccentric **20** between adjusted positions on the crank pin **5**.

When it is desired to lengthen the stroke length of the compressor **1**, this is done by moving the eccentric **20** in a counter-clockwise direction as seen in FIGS. 7 and 8. The lubrication pressure is raised from 60 to 70 p.s.i. and the valve spool **64**, in FIG. 8, is pushed inwardly away from the valve inlet **51** to simultaneously drain the valve outlet **58** to vent **65** and feed hydraulic pressure to the valve outlet **57**. Thus, hydraulic fluid is drained from the cage chambers **39** and **41** while it is fed to the cage chambers **38** and **40**. When the eccentric **20** arrives at the new desired position, the lubrication pressure is again reduced to 60 p.s.i. and the valve spool **64** returns to its intermediate position shown in FIG. 8.

*Second Embodiment FIGS. 9–11*

The embodiment shown in FIGS. 9 to 11 provides another means for controlling the hydraulic pressure in the cages **33** to move the eccentric **20** to various angular positions on the crank pin **5**. In this embodiment, the lubricating pressure is conducted through the crakshaft passage **55**, from the main bearings shell **17**, through respective ball check valves **77** and **78** to the groups of cage chambers **39** to **41**, with the ball check **77** feeding the chambers **38** and **40** and the ball check **78** feeding the chambers **39** and **41**.

Pressure is vented from one or the other set of chambers to cause the pressure in the non-vented chambers to move the cage **33**. This release of pressure is perfomed by a pair of poppet valves **79** and **80** mounted on the crankshaft web **18** and extending from its outer end face **81**. Each of the poppet valves **79** and **80** are in communication with one of the diametrical passages **42** and **43** so that pressing one of them dumps or releases hydraulic fluid from one diametrical passage and its communicating cage chambers.

The poppet valves **79** and **80** are located at different radial distances from the axis A—A of the crankshaft **4** and are opened by engagement with individual concentric rings **82** and **83**, one of which is pressed axially toward the web end face **81** when one of the respective poppet valves **79** and **80** is to be opened.

It will be understood that although only two embodiments of the invention are specifically described, the invention may embrace various other embodiments which are obvious from an understanding of the described embodiment and are embraced within the claims of the invention.

Having described our invention, we claim:

1. A variable stroke reciprocating piston machine comprising: a rotary crankshaft having an eccentrically mounted crank pin thereon and a circular web located beside said crank pin and having an axis aligned with the crank pin axis; an eccentric rotatively mounted on said crank pin; a connecting rod having one end rotatively mounted on said eccentric; and hydraulic means including a rotor fixed to said eccentric and rotatively mounted on the circumference of said circular web, said rotor being operative to rotate said eccentric on said crank pin during the rotation of said crankshaft for varying the stroke of said connecting rod.

2. The machine of claim 1 including: a hydraulic control system for rotating said rotor about said crank pin axis through a rotary travel path of about 180° relative to said crank pin, said hydraulic control system being operative to hold said rotor stationary relative to said crank pin at any position throughout its rotary travel path.

3. The machine of claim 2 wherein: said hydraulic motor includes a pair of hydraulic chambers; said rotor includes fluid pressure surfaces forming portions of each of said chambers; said fluid pressure surfaces being arranged to be acted on by fluid pressure in said chambers for forcing said rotor in opposite rotary directions; means for admitting hydraulic pressure to both of said chambers; and means for individually releasing fluid pressure from each of said chambers for causing said rotor to rotate in one direction or the other.

4. The machine of claim 3 wherein said rotor is a hollow annular member mounted on and rotating about the circumference of said circular web.

5. The machine of claim 4 wherein: said hollow annular rotor contains an annular space therein; and said annular space is divided into said pair of hydraulic chambers by a pair of vanes extending radially across said annular space, one of said vanes being fixed on said web and the other being fixed on said rotor.

6. The machine of claim 5 wherein; said hydraulic control system includes a control valve operative at one predetermined hydraulic pressure to hold the hydraulic fluid in both of said chambers, said valve being operative in response to a reduction in hydraulic pressure to release fluid from one of said chambers and operative in response to an increase in hydraulic pressure to release fluid from the other of said chambers.

7. The machine of claim 6 wherein: said hydraulic control system receives hydraulic fluid pressure from the lubrication system of the machine.

8. The machine of claim 5 wherein said hydraulic control system includes: means for continuously supplying hydraulic pressure fluid to both of said hydraulic chambers; and exhaust means for selectively dumping pressure fluid from each of said chambers.

**References Cited by the Examiner**

UNITED STATES PATENTS

| | | | |
|---|---|---|---|
| 546,419 | 9/95 | Deam. | |
| 1,876,129 | 9/32 | Arnold. | |
| 2,335,645 | 11/43 | Celio. | |
| 2,550,180 | 4/50 | Allen | 121—99 |
| 3,037,355 | 6/62 | Brathwaite | 121—99 |


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