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(54) **Hydraulic gerotor motor and parking brake**

(57) A rotary fluid pressure device having a fluid displacement mechanism (19) and an output shaft (39) rotatable relative to a housing (15,21,41). The device has first and second brake discs (51,53), to brake the rotation of the output shaft. The brake discs are engaged by an actuation piston (59;91) biased by a Belleville washer (85) toward an engaged condition. The actuation piston (59;91) separates a first pressure chamber (73) from a second pressure chamber (75), the washer (85) being disposed in the first pressure chamber (73). When both pressure chamber are drained, the actuation piston is applied by only the washer (85), but when the pressure chamber (73) is pressurized, the actuation piston (59;91) is applied with a substantially greater force, resulting in a greater braking torque.

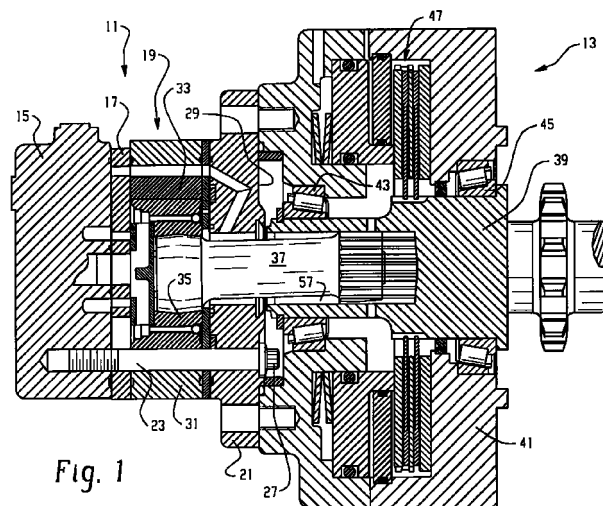


Fig. 1

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## Description

### BACKGROUND OF THE DISCLOSURE

**[0001]** The present invention relates to rotary fluid pressure devices, and more particularly, to such devices of the type including a fluid displacement mechanism, and even more particularly, to such devices in which the fluid displacement mechanism comprises a gerotor gear set.

**[0002]** Although the present invention may be included in a rotary fluid pressure device being utilized as a pump, it is primarily adapted for use in a motor, and especially for use in a low-speed, high-torque motor, and will be described in connection therewith. Also, even though the present invention may be utilized in axial piston motors, radial cam lobe motors, and other types of motors, it is especially suited for use in gerotor motors, and will be described in connection therewith.

**[0003]** In many vehicle applications for low-speed, high-torque gerotor motors, it is desirable for the motor to have some sort of parking brake or parking lock, the term "lock" being preferred in some instances because it is intended that the parking lock be engaged only after the vehicle is stopped. In other words, such parking lock devices are not intended to be dynamic brakes, which would be engaged while the vehicle is moving, to bring the vehicle to a stop. However, the term "brake" will generally be used hereinafter to mean and include both brakes and locks. The term "brake" is somewhat preferred to distinguish from a device which would operate either fully engaged or fully disengaged. In fact, the device of the present invention provides at least two different, distinct, engaged conditions.

**[0004]** For many years, those skilled in the art have attempted to incorporate brake and lock devices into gerotor motors, as opposed to merely adding a brake package on the motor output shaft. Examples of such devices are illustrated and described in U.S. Patent Nos. 3,616,882 and 4,981,423. In the device of U.S. 3,616,882, a braking element is disposed adjacent the forward end of the gerotor star, and is biased by fluid pressure into frictional engagement therewith. In the device of U.S. 4,981,423, there is a multi-disc brake assembly which is of the "spring-applied, pressure-released" type, and it has now become somewhat conventional in low-speed, high-torque motors, to utilize brake assemblies of the spring-applied, pressure-released type.

**[0005]** In the device of U.S. 4,981,423 the friction disc pack is in splined engagement with the output shaft, and therefore, must be able to brake or hold the full output torque of the motor, thus necessitating that the discs, the spring, and the apply/release piston all be relatively larger than is desirable.

**[0006]** One of the trends associated with the use of low-speed, high-torque gerotor motors is for the motor to be designed to operate at higher pressures, and

therefore, at higher output torques. One particular construction of gerotor motor which is especially adapted for higher pressures and torques is the VIS series of motors, sold commercially by the assignee of the present invention, such motors being illustrated and described in U.S. Patent Nos. 4,741,681 and 5,211,551, both of which are assigned to the assignee of the present invention and are incorporated herein by reference.

**[0007]** Unfortunately, the substantially higher motor output torques, coupled with the friction disc pack being conveniently associated with the output shaft, would result in a brake assembly having a torque capacity requirement which may make the brake assembly not feasible economically in many vehicle applications. Typically, the brake discs are provided with some sort of friction material which, while effective in increasing the braking torque, also adds further to the cost of the brake discs.

**[0008]** On additional disadvantage of the conventional motor brake assembly, and specifically, of the spring-applied, pressure-released type, is that the brake assembly operates at only one specific level of braking torque. In reality, however, it would be more desirable if the brake assembly were able to provide more than one level of braking torque, to correspond to more than one operating condition.

### BRIEF SUMMARY OF THE INVENTION

**[0009]** Accordingly, it is an object of the present invention to provide a brake assembly adapted for use with a rotary fluid pressure device, in which the brake assembly overcomes the above-described disadvantages of the prior art.

**[0010]** It is a more specific object of the present invention to provide a brake assembly, for use with a low-speed, high-torque motor, wherein the brake assembly is of a design which can reduce the total number and/or size of the brake discs, thereby making the brake assembly more compact and less expensive.

**[0011]** It is another object of the present invention to provide such a brake assembly which achieves the above-stated objects and which is able to provide at least two different engaged conditions, each representing a different level of braking torque.

**[0012]** The above and other objects of the invention are accomplished by the provision of a rotary fluid pressure device of the type including housing means, a rotary fluid displacement mechanism, and an output member having a movement corresponding to an output movement of the device. A first brake disc (which may also be referred to as a "stator") is fixed relative to the housing means, and a second brake disc is fixed to rotate with the output member, the first and second brake discs being disposed for operable engagement by means of an actuation piston to brake rotation of the output member relative to the housing means. The actu-

ation piston is moveable between an engaged position and a disengaged position, and spring means biases the actuation piston toward the engaged position.

**[0013]** The improved rotary fluid pressure device is characterized by the actuation piston cooperating with the housing means to define a first pressure chamber and a second pressure chamber disposed on axially opposite sides of the actuation piston. The spring biasing means is disposed in the first pressure chamber. A first source of pressurized fluid is in fluid communication with the first pressure chamber and is selectively operable to apply a biasing force to the actuation piston, in addition to the spring biasing means, tending to move the actuation piston toward the engaged position. A second source of pressurized fluid is in fluid communication with the second pressure chamber and selectively operable to apply a biasing force to the actuation piston, tending to move the actuation piston toward the disengaged position.

#### BRIEF DESCRIPTION OF THE DRAWINGS

##### **[0014]**

FIG. 1 is an axial cross-section of a gerotor motor including a brake assembly made in accordance with the present invention.

FIG. 2 is an enlarged, fragmentary, axial cross-section, similar to FIG. 1, illustrating in greater detail the brake assembly of the present invention, including a schematic representation of the controls therefor.

FIG. 3 is an enlarged, fragmentary, axial cross-section, similar to FIG. 2, illustrating an alternative embodiment of the present invention.

#### DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

**[0015]** Referring now to the drawings, which are not intended to limit the invention, FIG. 1 is an axial cross-section of a low-speed, high-torque gerotor motor of the type with which the brake assembly of the present invention is especially advantageous. The motor shown in FIG. 1 may be of the general construction illustrated and described in above-incorporated U.S. 5,211,551.

**[0016]** The motor assembly shown in FIG. 1 comprises a modular motor assembly, generally designated 11, and a forward bearing package, generally designated 13. The brake assembly of the present invention is disposed within the forward bearing package 13.

**[0017]** The modular motor 11, which will be described only briefly herein in view of the incorporation of the above-cited patent, includes an end cap 15, a stationary valve plate 17, a gerotor gear seat, generally designated 19, and a flange member 21. The elements 15 through 21 are held in tight sealing engagement by means of a plurality of bolts 23, each of the bolts 23

including a head 27 disposed in engagement with a forward surface 29 of the flange member 21.

**[0018]** The gerotor gear set 19 may be of the type well known in the art, and includes an internally-toothed ring member 31 defining a plurality of generally semi-cylindrical openings, with a cylindrical roller member 33 being disposed in each of the openings, and serving as the internal teeth of the ring member 31. Eccentrically disposed within the ring member 31 is an externally-toothed star member 35, which typically has one less external tooth than the number of the internal teeth 33, thus permitting the star 35 to orbit and rotate relative to the ring 31, as is well known to those skilled in the art. The orbital and rotational movement of the star 35 within the ring 31 defines a plurality of expanding and contracting fluid volume chambers, not shown herein, also well known to those skilled in the art.

**[0019]** In view of the above-incorporation of U.S. 5,211,551, there will not be a complete description herein of the various fluid ports and passages involved in communicating pressurized fluid to the expanding volume chambers of the gerotor gear set 19, and communicating low pressure fluid exhausted from the contracting volume chambers. It is sufficient to note that the orbital and rotational movement of the star 35 is transmitted by means of a main drive shaft 37 into pure rotational movement of an output shaft 39, which comprises part of the forward bearing package 13.

**[0020]** The forward bearing package 13 includes a bearing housing assembly 41, which, in the subject embodiment, comprises two separate housing members joined together by any suitable means, not shown herein. The bearing housing assembly 41 is provided with a pair of tapered roller bearing sets 43 and 45 which receive, and rotatably support the output shaft 39.

**[0021]** Referring now primarily to FIG. 2, there will be a description of the brake assembly of the present invention, generally designated 47. The bearing housing 41 defines a disc chamber 49, and disposed within the disc chamber 49 is a plurality of externally splined friction discs 51, which are in splined engagement with the bearing housing 41, and a plurality of internally splined friction discs 53 which are in splined engagement with a set of external splines 55 defined by the output shaft 39. It should be understood by those skilled in the art from a reading and understanding of this specification that the particular construction details of the friction discs 51 and 53 are not essential features of the invention. As used herein and in the appended claims, the term "brake disc" will be understood to mean and include any sort of members which, upon frictional engagement, are effective to achieve a braking torque of the output shaft 39 relative to the stationary part of the motor, e.g., the bearing housing assembly 41.

**[0022]** As is well known to those skilled in the art of gerotor motors, the motor typically includes a case drain region which, in the subject embodiment, comprises the generally cylindrical space 57 (see FIG. 1) surrounding

the main drive shaft 37. It is into this case drain region 57 that leakage fluid from the gerotor gear set 19 flows and the fluid in the case drain region 57 serves various lubrication functions, such as lubricating the spline connections of the shaft 37. Preferably, the disc chamber 49 is in open fluid communication with the case drain region 57, and as is typical, both the chamber 49 and the case drain region 57 are maintained at relatively low fluid pressure, for example, in the range of about 0 to 200 psi.

**[0023]** Referring still primarily to FIG. 2, there is a generally annular actuation piston 59 disposed within the bearing housing 41 and including a radially outer seal member 61 and a radially inner seal member 63. The outer seal member 61 is in sealing engagement with an internal, generally cylindrical surface 62 of the bearing housing 41, while the inner seal member 63 is in sealing engagement with a cylindrical outer surface 64 of a cylindrical, axially extending portion 65 of the bearing housing 41.

**[0024]** Disposed adjacent the actuation piston 59 is a separation member 67, the primary function of which is to seal or separate the low pressure disc chamber 49 from the relatively higher pressure chamber in which the actuation piston 59 is disposed. The separation member 67 has an outside diameter and associated therewith, a seal member 68 which is in sealing engagement with an inside surface of the bearing housing 41. The separation member 67 also has an inside diameter 69 defining an opening, the opening also being referred to hereinafter by the reference numeral "69". Associated with the inside diameter 69 is a seal member 70 which is in sealing engagement with a cylindrical, outer surface of a generally cylindrical, axially extending engagement portion 71. Preferably, the engagement portion 71 is formed integrally with the actuation piston 59 and is dimensioned such that, with the actuation piston 59 in the disengaged position shown in FIG. 2, the engagement portion 71 may be touching the adjacent friction disc 51, but is not applying any substantial biasing force to the pack of friction discs 51 and 53.

**[0025]** The actuation piston 59 cooperates with the bearing housing 41 and the adjacent separation member 67 to define a first pressure chamber 73 and a second pressure chamber 75. The bearing housing 41 defines a fluid port 77 and a fluid port 79. The first pressure chamber 73 is in communication, by means of the fluid port 77, with a first source 81 of pressurized fluid, while the second pressure chamber 75 is in communication, by means of the fluid port 79, with a second source 83 of pressurized fluid. The sources of pressurized fluid 81 and 83 are illustrated herein schematically as each comprising a fixed displacement pump and a simple solenoid valve. However, those skilled in the art will understand from the subsequent description of the operation of the brake assembly 47 that there would typically be provided some sort of appropriate intercommunication between the valves and the vehicle

microprocessor (not shown herein) or some other form of control logic. It is believed that the necessary valving and controls is well within the ability of those skilled in the art. For example, each of the valves shown in FIG. 2 includes a flow "blocked" position, whereas it would be likely that the valve, when not energized, would be spring biased to a position in which the valve would drain the respective port and chamber. Also, although the valves shown in FIG. 2 each merely have two discrete positions, it would be apparent to those skilled in the art that the valves could be of the "infinitely variable" type, operable to modulate the fluid pressure in the respective chamber 73 or 75, thus gradually applying or releasing the brake.

**[0026]** As was mentioned in the BACKGROUND OF THE DISCLOSURE, the brake assembly 47 of the present invention is generally of the "spring-applied, pressure-released" type. Therefore, disposed within the first pressure chamber 73 is a set of Belleville washers (or springs) 85, tending to bias the actuation piston 59 from the disengaged position shown in FIG. 2 toward an engaged position. As is understood by those skilled in the brake and clutch art, there is frequently very little difference apparent between the disengaged and engaged positions of a set of friction discs. For example, in the disengaged position of FIG. 2, the friction discs 51 and 53 may actually be in contact with each other, but without sufficient actuation or biasing force applied to the discs, the contact of the discs does not generate any substantial braking torque. The actuation piston 59 is maintained in the disengaged position of FIG. 2 by draining pressurized fluid from the first pressure chamber 73 and communicating pressurized fluid to the second pressure chamber 75, thus exerting sufficient biasing force on the actuation piston 59 to bias it to the left in FIG. 2 until the piston 59 engages a shoulder surface 87 defined by the bearing housing 41.

**[0027]** In order to achieve "normal" braking torque, both of the pressure chambers 73 and 75 are drained, such that the only biasing force on the actuation piston 59 is that of the Belleville washers 85, thus moving the actuation piston 59 slightly to the right in FIG. 2 such that the engagement portion 71 applies an engagement load to the friction discs 51 and 53. The result is sufficient frictional engagement of the discs 51 and 53 to provide a predetermined, standard level of braking torque applied to the output shaft 39. When it is necessary to achieve a substantially greater level of braking torque, pressurized fluid is communicated to the first pressure chamber 73, while the second pressure chamber 75 is drained, such that the actuation piston 59 is now biased by both the Belleville washers 85 and the pressurized fluid in the chamber 73. As a result, there is substantially greater engagement force applied to the friction discs 51 and 53 by the engagement portion 71, thus substantially increasing the braking torque capacity of the brake assembly 47.

**[0028]** It is believed to be within the ability of those

skilled in the art to select the Belleville washers 85, and the dimensions of the various other elements of the brake assembly 47, in order to achieve the desired levels of braking torque. Typically, the greater level of braking torque would correspond approximately to the maximum, continuous torque rating of the motor, whereas the normal braking torque would be, by way of example only, that torque required to prevent a fully loaded vehicle from rolling downhill on a twenty-five degree slope.

**[0029]** Referring now primarily to FIG. 3, there is illustrated an alternative embodiment of the invention, in which the same, or similar elements, bear like numerals, and new or greatly modified elements bear reference numerals in excess of "90". Thus, in the FIG. 3 embodiment, there is an actuation piston 91 which is generally similar to the piston 59 in the main embodiment, but differs from the piston 59 in that the piston 91 moves rearwardly (to the left in FIG. 3) whenever the brake is being applied, and moves forwardly (to the right in FIG. 3) whenever the brake is being released. Also, in the main embodiment, there is a separation member 67, whereas, in the FIG. 3 embodiment, the flange member 21 includes a preferably integral separation portion 93. The separation portion 93 carries the seal members 68 and 70, and serves the function of separating the second pressure chamber 75 from the chamber containing the friction discs 51 and 53, which is open to the case drain region 57, as was explained previously.

**[0030]** In operation, the FIG. 3 embodiment is substantially the same as the main embodiment, except for the opposite direction of movement of the actuation piston 91. To release the brake discs 51 and 53, the pressure chamber 73 is drained, while the pressure chamber 75 receives high pressure, thus moving the piston 91 to the right in FIG. 3, to a disengaged condition. For normal braking, both pressure chambers 73 and 75 are drained, and the Belleville washer 85 provides the only apply force to the piston 91. For greater braking torque, the pressure chamber 73 is pressurized, while the pressure chamber 75 is drained, so that the piston 91 is biased toward the left in FIG. 3, to an engaged condition, by both the washer 85 and the pressure in the chamber 73.

**[0031]** It has been determined, in connection with the development of the present invention, that the FIG. 3 embodiment permits the entire forward bearing package and brake assembly to be substantially smaller and more compact than does the main embodiment. In addition, the FIG. 3 embodiment requires fewer parts, thus having at least the potential to be somewhat less expensive to manufacture.

**[0032]** The invention has been described in great detail in the foregoing specification, and it is believed that various alterations and modifications of the invention will become apparent to those skilled in the art from a reading and understanding of the specification. It is intended that all such alterations and modifications are

included in the invention, insofar as they come within the scope of the appended claims.

## Claims

1. A rotary fluid pressure device of the type including housing means (15,21,41), a rotary fluid displacement mechanism (19), and an output member (37,39) having a movement corresponding to an output movement of said device; a first brake disc (51) fixed relative to said housing means (15,21,41), and a second brake disc (53) fixed to rotate with said output member (37, 39), said first (51) and second (53) brake discs being disposed for operable engagement by means of an actuation piston (59;91) to brake rotation of said output member (37,39) relative to said housing means, said actuation piston being moveable between an engaged position and a disengaged position (FIG. 2), and spring means (85) biasing said actuation piston (59;91) toward said engaged position; characterized by:

(a) said actuation piston (59;91) cooperating with said housing means (41) to define a first pressure chamber (73) and a second pressure chamber (75) disposed on axially opposite sides of said actuation piston (59;91);

(b) said spring biasing means (85) being disposed in said first pressure chamber (73);

(c) a first source (77,81) of pressurized fluid being in fluid communication with said first pressure chamber (73) and selectively operable to apply a biasing force to said actuation piston (59;91), in addition to said spring biasing means (85), tending to move said actuation piston toward said engaged position; and

(d) a second source (79,83) of pressurized fluid being in fluid communication with said second pressure chamber (75) and selectively operable to apply a biasing force to said actuation piston (59;91), tending to move said actuation piston toward said disengaged position (FIG. 2).

2. A rotary fluid pressure device as claimed in claim 1, characterized by said actuation piston (59;91) including seal means (61,63) in sealing engagement with said housing means (41), thereby isolating said first (73) and second (75) pressure chambers from each other.

3. A rotary fluid pressure device as claimed in claim 2, characterized by said actuation piston (59;91) comprises an annular member including an outer cylindrical surface and an inner cylindrical surface and said seal means comprises a first seal member (61) associated with said outer surface and a second

seal member (63) associated with said inner surface.

orbital and rotational movement to said output shaft (39).

4. A rotary fluid pressure device as claimed in claim 1, characterized by said housing means (15,21,41) defining a disc chamber (49), said first (51) and second (53) brake discs being disposed in said disc chamber (49), said rotary fluid pressure device defining a case drain region (57), and said disc chamber (49) being in open fluid communication with said case drain region (57). 5 10
  
5. A rotary fluid pressure device as claimed in claim 4, characterized by said housing means (15,21,41) including a separation member (67;93) disposed to separate said disc chamber (49) and said second pressure chamber (75), and defining an opening (69), said actuation piston (59;91) including an engagement portion (71) disposed to extend axially through said opening (69) for operable engagement with one of said first (51) and second (53) brake discs when said actuation piston (59) is in said engaged position. 15 20
  
6. A rotary fluid pressure device as claimed in claim 5, characterized by said separation member (67;93) comprises a generally annular member including an outer cylindrical surface in sealing engagement with an adjacent surface of said housing means (41), and an inner cylindrical surface defining said opening (69), said inner cylindrical surface being in sealing engagement with said engagement portion (71) of said actuation piston (59;91), as said engagement portion (71) moves axially, relative to said separation member (67;93). 25 30 35
  
7. A rotary fluid pressure device as claimed in claim 6, characterized by said housing means comprises a flange member 21 and said separation member (93) is formed integrally with said flange member (21), said flange member (21) and said separation member (93) cooperating to engage said first brake disc (51). 40
  
8. A rotary fluid pressure device as claimed in claim 1, characterized by said fluid displacement mechanism comprises a gerotor gear set (19), and said rotary fluid pressure device includes an output shaft (39), said output member comprising said output shaft (39). 45 50
  
9. A rotary fluid pressure device as claimed in claim 7, characterized by said gerotor gear set (19) includes an internally-toothed ring member (31) and an externally-toothed star member (35) eccentrically disposed within said ring member (31) for relative orbital and rotational movement, said device including a drive shaft (37) operable to transmit said 55

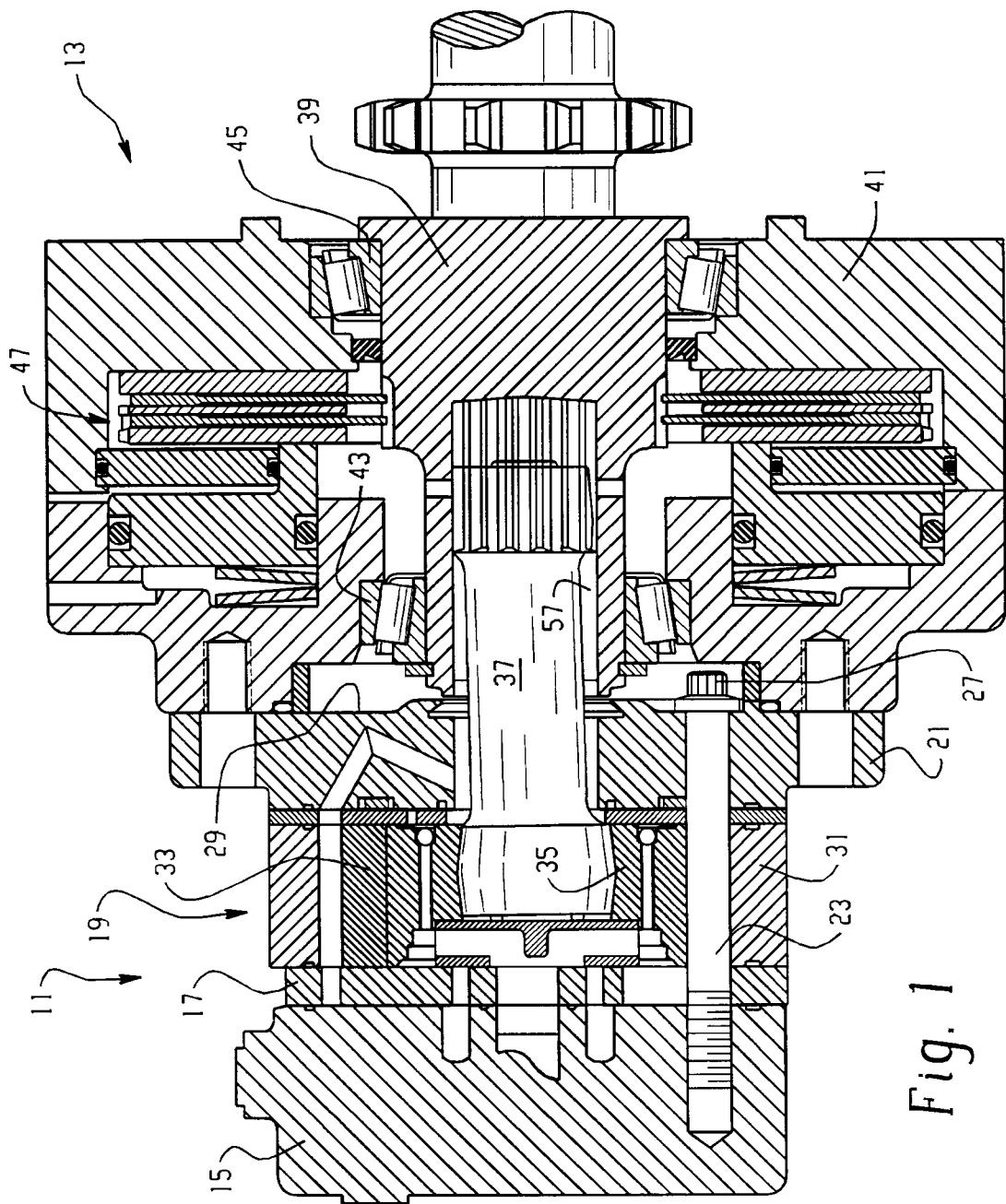


Fig. 1

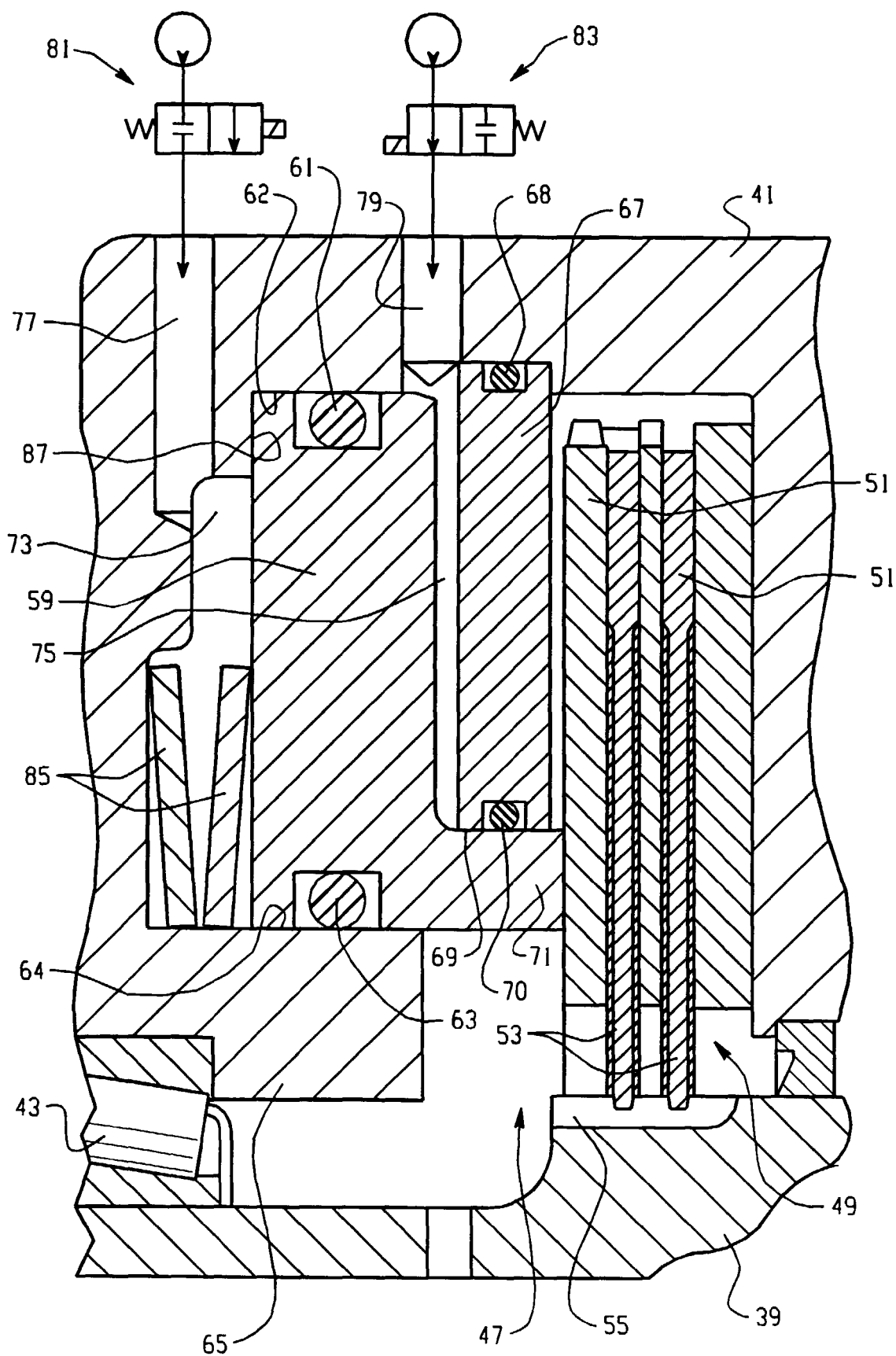


Fig. 2



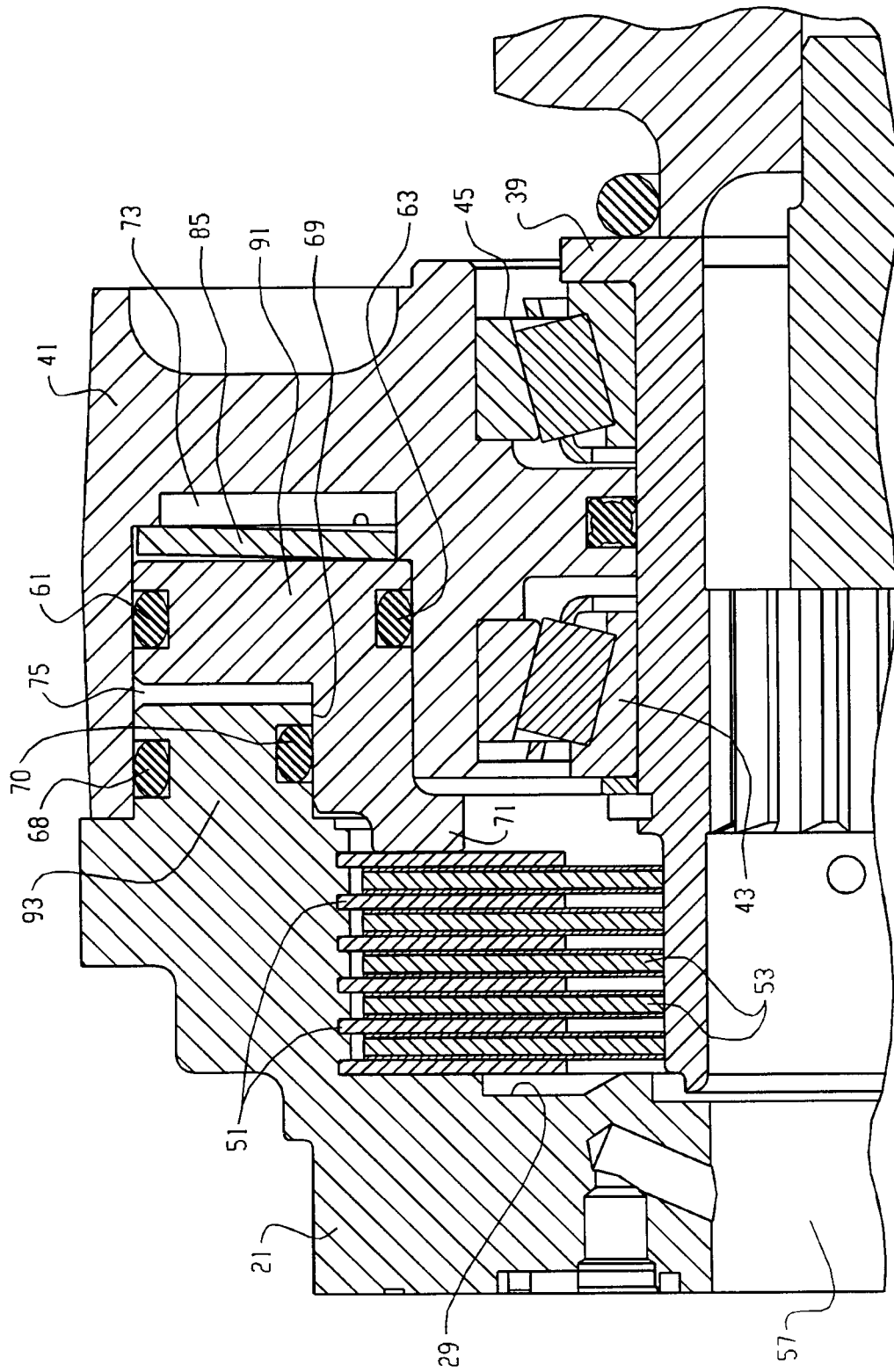


Fig. 3