United States Patent [19]

Mayall

[11] **3,749,511**

[45] **July 31, 1973**

[54] **PRESSURIZED SEALING MEANS FOR A HYDRAULIC TURBODRILL**

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[30] **Foreign Application Priority Data**

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Apr. 30, 1971 Great Britain 12,446/71

[52] **U.S. Cl.** **415/113,** 415/502
[51] **Int. Cl.** **F01d 11/00**
[58] **Field of Search** 415/113, 107, 502; 418/202

[56] **References Cited**

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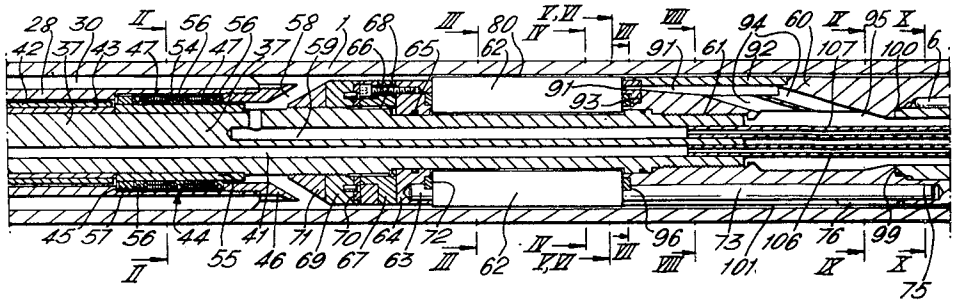
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Primary Examiner—C. J. Husar
Attorney—Bacon & Thomas

[57] **ABSTRACT**

A hydraulic motor of the turbodrill type including a housing, a rotor shaft rotatably mounted via bearings in said housing and connectible to a drill bit to transmit driving/torque thereto, sealing means between the rotor shaft and the housing arranged to oppose access of drilling mud to said bearing means, said sealing means comprising pressurized inner and outer sealing devices.

5 Claims, 13 Drawing Figures



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SHEET 1 OF 7

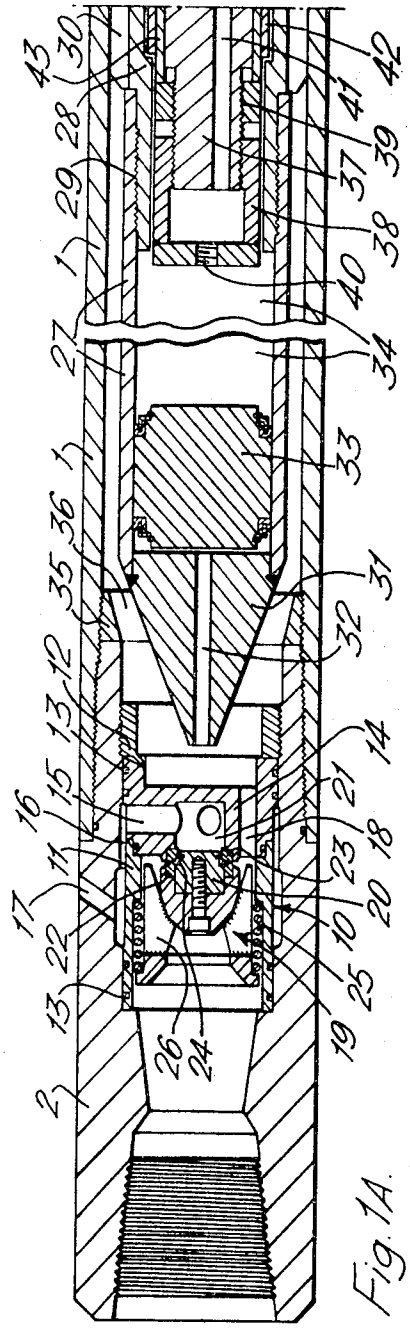


Fig. 1A.

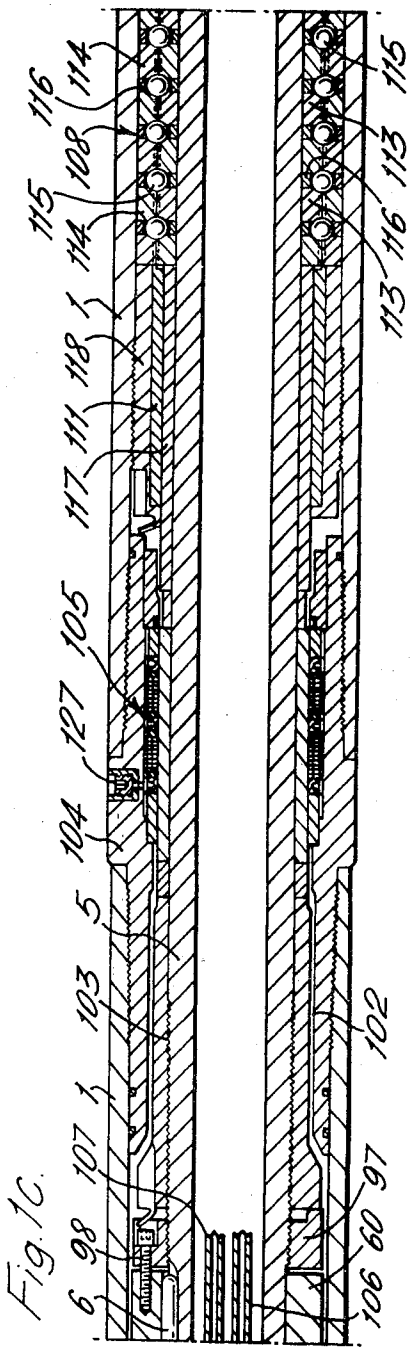


Fig. 1C.

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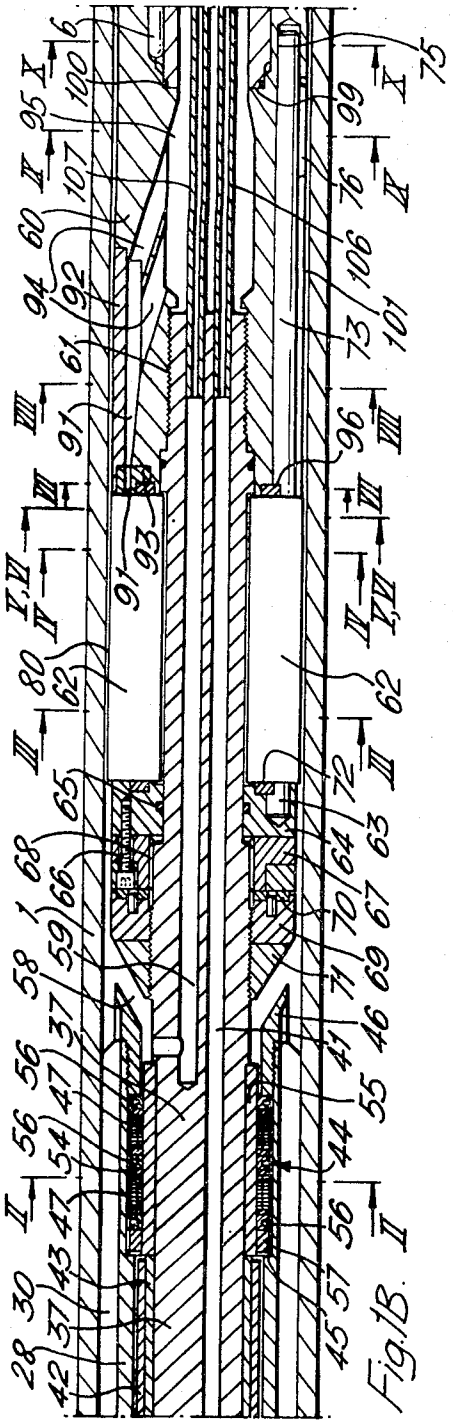


Fig. 1B.

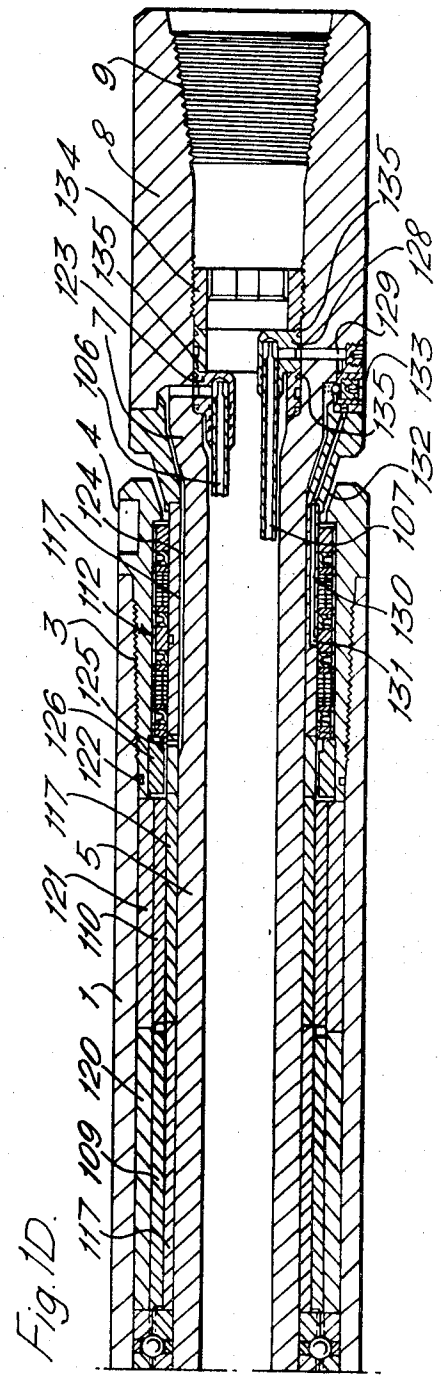


Fig. 1D.

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Fig. 2.

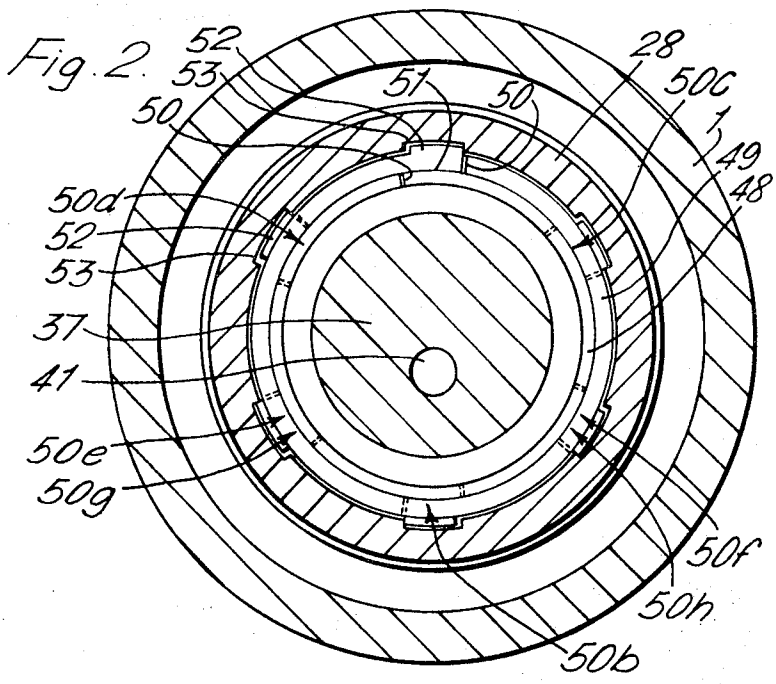
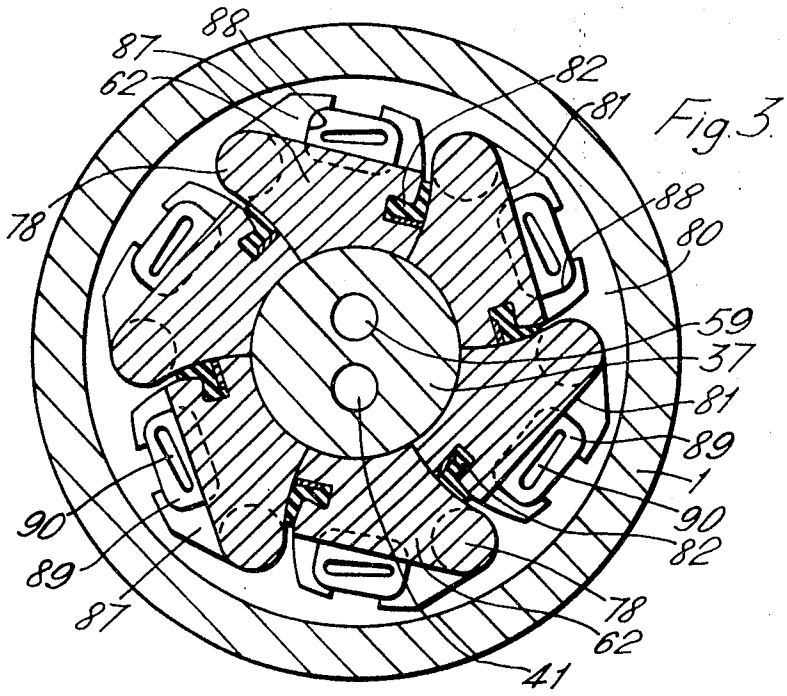


Fig. 3.



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Fig. 4.

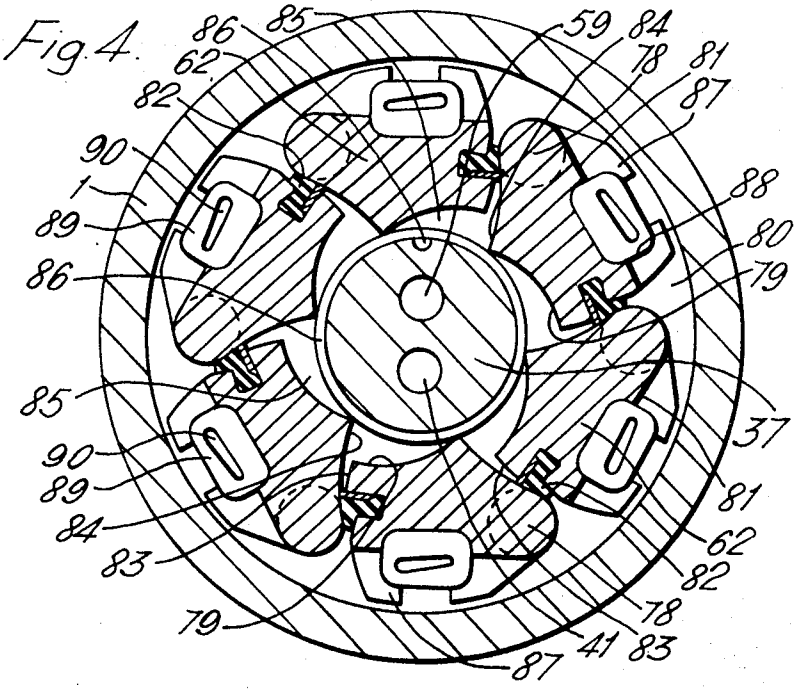
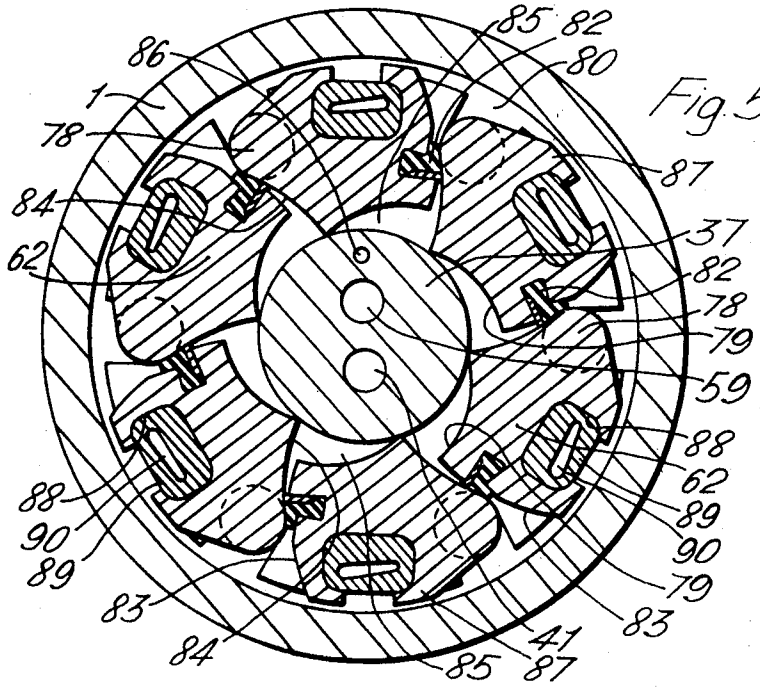
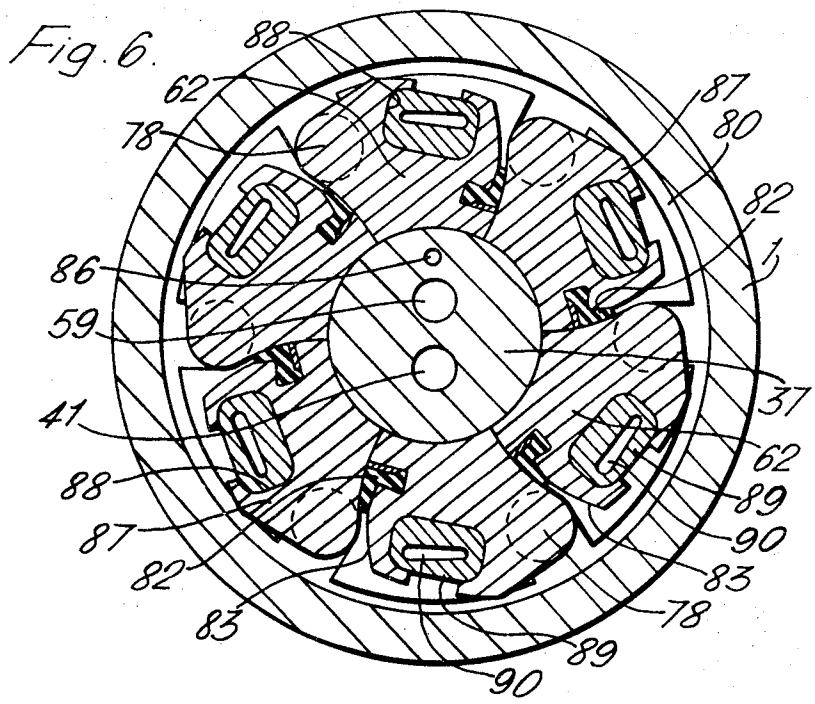


Fig. 5.



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Fig. 6.



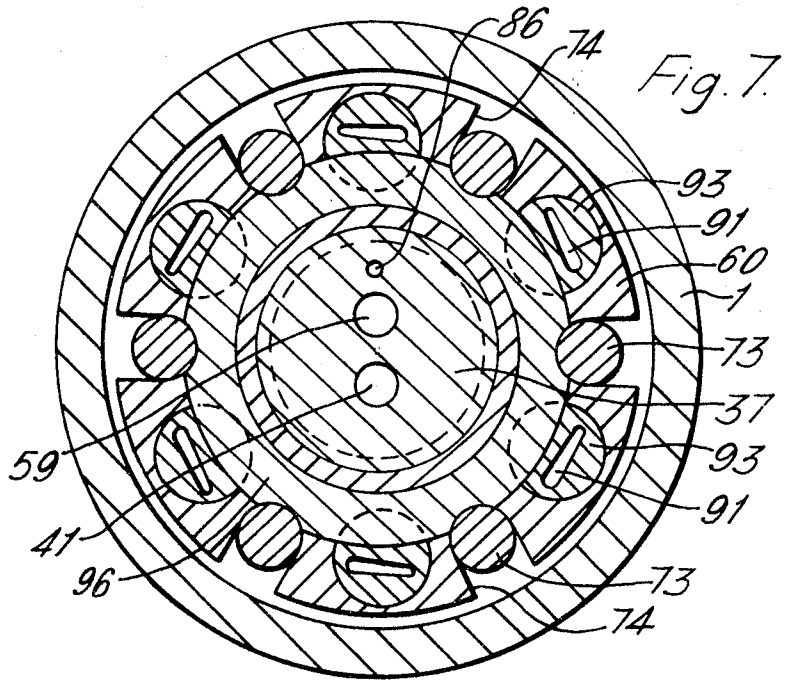
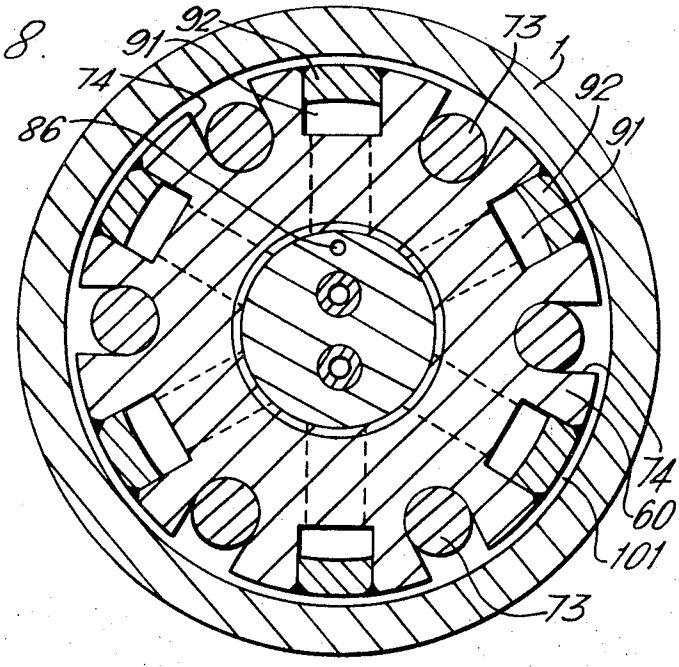


Fig. 7.

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Fig. 8.



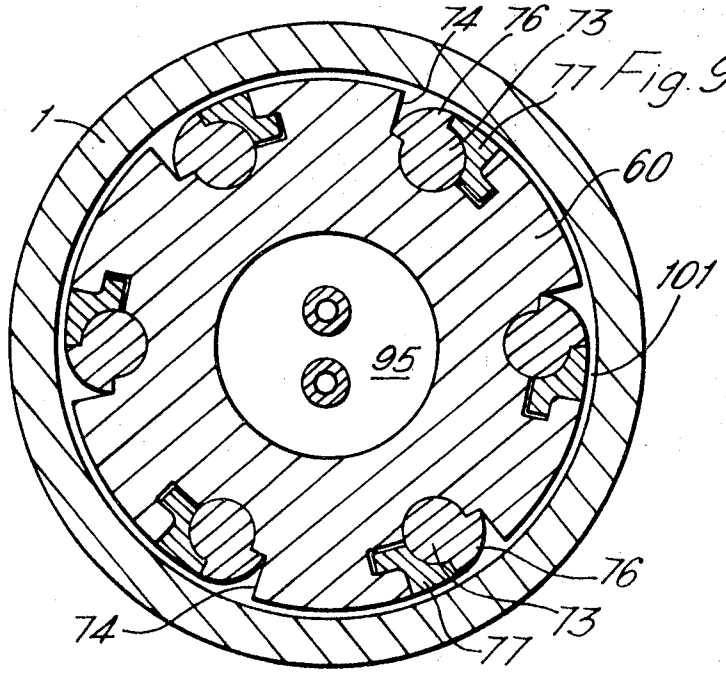


Fig. 9.

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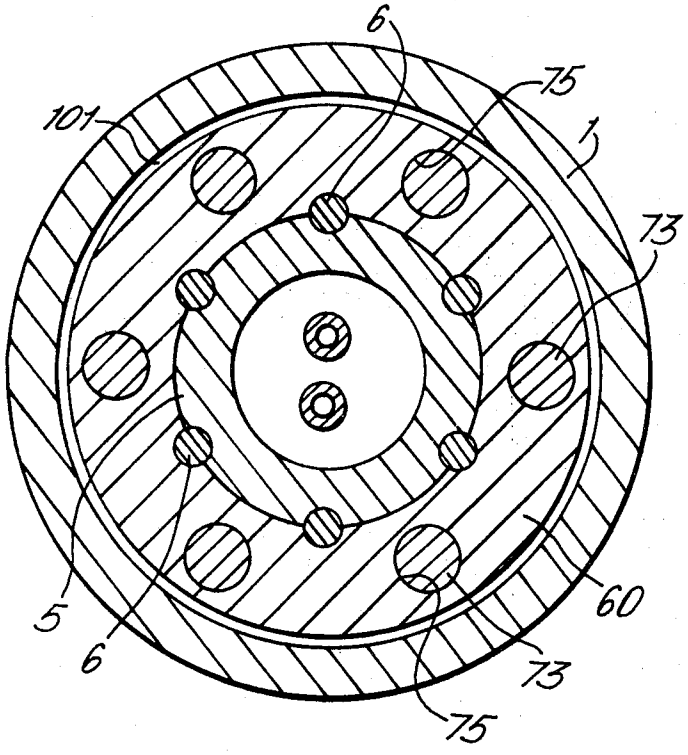
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Fig. 10.



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PRESSURIZED SEALING MEANS FOR A HYDRAULIC TURBODRILL

This invention concerns rock drills and more particularly drills for penetrating the earth's crust to great depths, for instance to reach oil or gas-bearing strata. Commonly, such drilling is accomplished by the use of a drilling rig adapted to rotate a suitable drill bit by rotation of a so-called drill-string of coupled pipes that extend from the rig on the surface down to the drill bit which is fixed at the bottom of the lower-most pipe of the drill-string. Chips and rock particles produced by the action of the rotating bit on the borehole bottom are flushed to the surface via the annular space between the drill-string and the borehole wall by a stream of liquid that is conveyed to the bottom of the borehole through the connected pipes constituting the drill-string. This liquid, usually called mud, also serves the purpose of cooling the drill bit.

The torque needed to rotate the drill bit is substantial and, as indicated, is commonly transmitted to the drill bit through the drill-string from the rig on the surface; the energy stored in the drill-string when transmitting this torque is therefore considerable, especially when drilling at great depths, giving rise to difficulties when the bit penetrates discontinuities that cause fluctuations in the torque requirements of the bit. Moreover, the transmission of the required torque from the rig to the bit requires the drill-string pipes to be of substantial wall thickness and rigidity and thus both heavy and expensive. The requirement for the drill-string pipes to have substantial wall thickness conflicts with the other requirement of these pipes, namely to have a large bore cross section for conducting the mud in large volumes to the bottom of the borehole to accomplish its bit-cooling and scavenging functions. In addition, if a rectilinear borehole is to be cut, the drill-string pipes must be straight and accurately formed so as to transmit torque without deflection that would cause the drill bit to "wander" and produce a non-rectilinear borehole; on the other hand the formation of a borehole having a planned deviation from the rectilinear, as is sometimes desirable, is difficult if not impossible.

It has therefore been recognised that many of the disadvantages and design compromises in drills of the above described construction might be avoided by generating the required torque for rotating the drill bit by means disposed adjacent the drill bit and powered by energy conducted to such means. The so called "turbodrill" is one well-known proposal for accomplishing this objective, the turbodrill comprising a hydraulic motor in the form of a turbine adapted to be fitted to the bottom of a drill-string and to extract energy from the mud flowing down the latter so as to produce torque to rotate the drill bit relative to the drill-string. The art is replete with proposals for turbodrill constructions but, so far as I am aware, no turbodrill yet proposed has demonstrated an ability to compete in practical circumstances with a drill of the conventional construction discussed above.

Drilling mud is commonly a highly abrasive slurry or suspension of sand and rock particles: the mud is continuously circulated through the drill-string, the borehole and a settling pond or the like in which the bulk of the larger chippings are deposited. This extremely abrasive fluid is understandably damaging to any relatively moving surfaces between which it may penetrate

and, as previously mentioned, the mud at the bottom of a deep borehole is under a very substantial hydrostatic pressure so that its power of penetration into bearing structures and between relatively moving parts is very high. The bearings of a hydraulic motor for a drill are therefore prone to become contaminated with the highly abrasive mud that is used to drive the motor; since these bearings are subject to extremely high loadings and large load variations, it will be understood that bearing failure has been a notable cause of unreliability in hydraulic motors of the turbodrill type so far proposed.

The present invention provides a hydraulic motor for driving a rock drill, comprising a housing connectible to a drill-string and a rotor shaft rotatably mounted via bearing means in said housing and connectible to a drill bit, sealing means between the rotor shaft and the housing on each side of said bearing means, the space containing the bearing means between said sealing means being adapted to be filled with lubricant, at least one cylinder housing a piston being in communication with said space, and means being provided for leading drilling mud, at a pressure at least as high as the mud which will be present outside of the said sealing means in operation, to the side of said piston remote from said space so as to pressurise the lubricant therein. The fact that the drilling mud led to the said cylinder is at a pressure at least as high as the mud which is on the outside of the said sealing means in operation, so that the lubricant in its turn is maintained at a pressure at least as high as the mud outside the sealing means, keeps the mud out of the sealing means and out of the space containing the bearing means. In a preferred arrangement the said cylinder is in communication with the said space containing the bearing means via an inlet to said space closely adjacent one of the said sealing means, preferably the lower one. Preferably means are also provided for leading pressurised mud to a point between the ends of at least one of the said sealing means (preferably that adjacent which the said lubricant inlet is situated) and a path for the leakage of mud out of the said housing is provided from that point to an outlet outside of the sealing means so that mud, rather than lubricant, will be caused to leak under the pressure in the space containing the bearing means.

In one form of the invention the cylinder and piston assembly for pressuring the lubricant is of annular configuration, the cylinder being an annular one defined between the said rotor shaft and its housing. Preferably however the cylinder and piston assembly is of conventional configuration and is positioned near to what will be the top end of the motor in use, above the rotor, that chamber of the cylinder which is intended to contain lubricant in operation being connected to the said space containing the bearing means via a conduit extending along and within the rotor. The cylinder is preferably dimensioned to be capable of housing a substantial quantity of lubricant, e.g. about 9 gallons of grease, so that despite the small leakages which must inevitably occur in operation it will only need re-charging at infrequent intervals.

It will be appreciated that thrust bearing means must be provided between the housing and the rotor assembly of a hydraulic motor according to the invention, capable of supporting thrust loads applied in either direction and which are such as to fluctuate and repeatedly reverse during many practical drilling circumstances.

Preferably a hydraulic motor according to the invention incorporates at least one thrust bearing structure of the kind described and claimed in U.S. Pat. No. 3,630,634. In its preferred form such a thrust bearing structure also constitutes a radial bearing but, even so, such a hydraulic motor will preferably also incorporate further radial bearing means for centering the rotor assembly in the housing.

A preferred form of the present invention is a hydraulic motor incorporating the principal aspects of the invention as set forth above.

In order that the invention may be more readily understood and further features and advantages thereof appreciated, one embodiment of the invention will now be described by way of example and with reference to the accompanying drawings, in which:

FIGS. 1*a* to 1*d* are a view in axial cross-section of a hydraulic motor according to the invention; and

FIGS. 2 to 10 are transverse cross-sectional views on the respective lines indicated in FIG. 1.

The general arrangement of the hydraulic motor will first be briefly described with reference to FIGS. 1*a* to 1d.

The motor comprises a housing in the form of a cylindrical casing tube 1 which extends from end to end of the motor and is provided with an internally screw-threaded fitting 2 at its top end for attachment to the bottom pipe of a drill-string. The lower end of the casing tube is internally screw-threaded at 3 for attachment to an end fitting 4. The casing tube 1 will not, of course, rotate in operation and it constitutes the stator of the motor assembly.

A rotor assembly is supported for rotation within the casing tube 1 and includes a rotor shaft 5 which is mounted in a sealed bearing system over the great part of its length. The rotor shaft 5 is tubular and has its upper end connected by six drive pins 6 to the bottom end of the drive means assembly of the rotor, whilst the lower end of the rotor shaft is radially enlarged at 7 and is then formed as an integral end fitting 8 formed with an A.P.I. internal thread 9 for the connection of a drill bit (not shown).

The construction of the hydraulic motor will now be described in detail beginning at its top end, i.e. the left-hand end of FIG. 1*a*.

The top end fitting 2, for attachment to the bottom pipe of a drill-string, houses a dump valve 10 which is automatically operative to provide an exit for the column of mud in the drill-string when the mud becomes de-pressurized, e.g. when the drill-string is to be withdrawn from the borehole. The dump valve comprises a body 11 secured in the fitting 2 by a lock nut 12 and sealed to the interior of the fitting 2 by sealing rings 13, the body 11 including a transverse portion 14 formed with three exit passages 15 for mud, leading via an annular recess 16 in the inside wall surface of the fitting 2 to an exit opening 17 extending through such wall. The transverse portion 14 of the dump valve casing is also provided with three passages 18 for the flow of pressurized mud therethrough, which are always open. Entry of mud to the exit passages 15 is however prevented, when the mud is pressurized, by a valve member 19 slidably mounted in the dump valve body and provided with a central plug 20 engageable in a central aperture 21 of the transverse portion 14 of the body, leading to the exit passages 15. Tungsten carbide wear rings 22 and 23 are provided on the valve member 19 and the transverse portion 14 of the dump valve body respectively, which inter-engage when the valve is closed, and a rubber sealing ring 24 which seals against the wear ring 23 is mounted around the plug member 20. The valve member 19 is urged upwardly in the dump valve body by a helical compression spring 25 so as to cause the valve to open automatically when the mud is depressurized, but is held in its closed position, against the action of the spring 25, by pressurized mud. Passages defined between fins 26 of the valve member permit the flow of pressurized mud past it and through the passages 18 in the transverse portion 14 of the dump valve body during operation.

The next part of the motor below the dump valve 10 is a cylinder and piston assembly whose purpose is to supply pressurized lubricant, more specifically grease, to the seals and bearings lower down the motor. The cylinder 27 of this assembly (only a small part of whose length is shown in FIG. 1*a*) is mounted at its lower end from a sleeve 28 to which the cylinder is screw-threadedly secured at 29. The sleeve 28 is formed with radially extending fins 30 which locate against the inside of the casing tube 1 to centralise both the cylinder 27 and other parts which are mounted within the sleeve and are described hereinafter. The cylinder 27 has welded to its top end a conical fitting 31 formed with an axial passage 32 for the flow of pressurized mud into the cylinder, and the cylinder contains a piston 33 arranged to be urged downwardly in the cylinder by the mud pressure. In use, the whole of the cylinder space 34 below the piston 33 will be filled with grease for pressurized application to the seals and bearings of the motor as previously mentioned.

The cylinder 27 is located at its top end by a locking sleeve 35 screwed in the casing tube and integrally connected to the fitting 31 by webs 36 defining mud flow passages therebetween. The sleeve 35 is connected to the casing tube by a left-hand thread in case the cylinder 27 should tend to rotate clockwise (as seen from above) during operation.

The sleeve 28 on which the cylinder 27 is mounted houses and centralises the top end of the rotor assembly of the motor, in the form of a shaft 37 having a top end cap 38 screw-threadedly secured thereto at 39. The cap 38 is formed with an aperture 40 for the passage of pressurized grease from the cylinder space 34 into a bore 41 of the shaft 37. Pressurized grease can also pass into the narrow annular gap between the outside cylindrical surface of the cap 38 and the inside surface of the sleeve 28, to lubricate bearing means for the top end of the shaft 37, provided within the sleeve 28. Such bearing means takes the form of a hardened steel sleeve 42 mounted and secured in a recess within the sleeve 28 and rotatable on a bearing sleeve 43 secured on the shaft 37 by the end cap 38. A seal structure generally indicated at 44 (see also FIG. 2) is secured between a shoulder 45 of the sleeve 28 and a lock nut 46 screw-threadedly mounted in the lower end of the sleeve, and is formed by two axially spaced packs 47 of pairs of inner and outer split rings 48 and 49. Each pack 47 contains eight such pairs of split rings and the rings of each pair made a close fit one within the other (see FIG. 2). The splits 50 of the rings of each pair are slightly circumferentially spaced from one another and the two rings are fixed together at the interface 51 extending between the slits. The connection between the rings can be by brazing, welding or soldering depending

on the material from which they are formed. The successive pairs of rings of each pack have their pairs of splits **50** indexed out of alignment. In the particular arrangement shown in FIG. **2** the splits **50***b* of the second pair of rings (from the top) are indexed 180° away from the splits **50** of the top pair, whilst the splits of the remaining pairs of rings are positioned as indicated by the successive reference numerals **50***c* to **50***h* in FIG. **2.**

Each outer ring **49** is formed with a radial enlargement **52** (FIG. **2**) engaging in one of six key-ways **53** formed in the sleeve **28** so that all of the rings **48** and **49** are thereby secured against rotation. The packs of rings **47** are separated by spacer rings **54** which float on a sleeve **55** secured on the shaft **37** and all of the spacer rings **54** and packs of sealing rings **47** are urged together axially by spring steel rings **56.** The inner split rings **48** of each pack are urged into sealing engagement with the sleeve **55** on the shaft **37** by grease which is fed to the inside of the bearing sleeve **28.** A distance piece **57** is provided between the uppermost spring steel ring **56** and the shoulder **45** of the sleeve **28.**

The purpose of the seal structure **44** is to exclude pressurized mud from the bearing **42.** The mud flows from the top of the cylinder **27** down the annular space defined between the cylinder and the casing tube **1** and thence through the passages defined between the fins **30** on the sleeve **28.** When the mud reaches the lower end of the lock nut **46** on the sleeve **28,** its flow divides into two parts, some of the mud flowing through an annular passage **58** into a second bore **59** in the shaft **37** and the remaining part of the mud flowing around the outside of the drive assembly of the rotor to drive the same in a manner to be described below.

Referring now to the drive assembly of the rotor in detail, the lower part of the shaft **37** constitutes an upper body portion of such assembly whilst a lower body portion thereof is constituted by a generally tubular member **60** screw-threadedly mounted at **61** on the lower end of the shaft **37.** The structure of the drive assembly is also shown in the cross-sectional views of FIGS. **3** to **10,** of which FIGS. **5** and **6** are taken in the same plane, FIG. **5** showing the rotor driving blades (to be described below) in their rest positions and FIG. **6** showing the blades in the positions which they occupy when the rotor is rotating. The shaft **37** and the tubular member **60,** together with the various elements which they mount and support, form the top part of the rotor assembly of the hydraulic motor, the stator being constituted by the casing tube **1.**

A ring of six rotor driving members in the form of blades **62** are mounted around the shaft **37** near to its lower end. Each blade **62** is provided with a cylindrical top mounting stud **63** which is rotatably received in one of six bores in a top blade mounting ring **64** rotatably mounted on the shaft **37** and sealed thereto by an O-ring **65.** The ring **64** is secured in position on the shaft **37** by six Allen screws **66** mounted in a sleeve **67** splined on the shaft **37** at **68.** The screws **66** thus hold the blade mounting ring **64** against rotation on the shaft **37**; nevertheless, in order to permit the circumferential positioning of the blades so as correctly to locate their lower ends in further mounting means to be described below, the passages in the sleeve **67** through which the screws **66** pass are somewhat circumferentially enlarged. A nut **69** provided with an inset wear ring **70** is screwed down on top of the blade mounting ring **64** and held in position by a lock ring **71** which, like the nut **69,** is screwed on the shaft **37.** A rubber sealing ring **72,** formed with suitable arcuate cut-outs to receive the blade mounting studs **63,** is inset in the bottom surface of the blade mounting ring **64.**

It will thus be seen that the blades **62** are rotatable at their top ends in the mounting ring **64,** by way of their mounting studs **63.** At their lower ends, however, the blades are held against rotation in a manner now to be described.

A mounting and torsion shaft **73** is integral with the bottom end of each blade **62** and is received in one of a ring of six slots **74** in the outer surface of the lower tubular body part **60** of the drive assembly, the lower end extremities of the torsion shafts being received in bores **75** in the member **60** to locate the shafts circumferentially. The torsion shafts **73** are not intended to rotate when the motor is being driven but rather to twist so as to permit a limited degree of rotation of the blades **62.** Each torsion shaft is therefore held against rotation by the engagement of a sector-shaped portion **76** thereof (see FIG. **9**) in a correspondingly shaped aperture defined in its receiving slot **74** in the tubular member **60.** As will be explained hereafter, the blades are urged to rotate clockwise (as seen from above) in operation and a suitably shaped wear piece **77** is provided on the appropriate side of the sector-shaped portion **76** of each torsion shaft **73,** in the shaft-receiving slot of the member **60**; the wear pieces **77** are easily replaceable when worn. It will thus be understood that each torsion shaft **73** is held against rotation at its lower end but is able to twist over the length of the shaft extending between the region of the FIG. **9** cross-section and the bottom end of the associated blade **62,** so as to permit rotation of the blade about the axis of the torsion shaft as previously mentioned; the mounting studs **63** at the top ends of the blades are coaxial with the torsion shafts **73** and this axis is the "twisting axis" of the blade previously referred to herein.

The cross-sectional shape of the major part of the length of each blade **62** is that shown in FIGS. **3** and **4** and the twisting axes of the blades are indicated in those Figures at **78.** The blades are shown in FIGS. **4** and **5** in their rest positions, i.e. when the rotor is stationary, but clockwise rotation of the blades during operation to drive the rotor brings them into the positions shown in FIGS. **3** and **6** in which their arcuate inner faces **79** precisely engage face-to-face with the surface of the shaft **37.**

Referring now to the manner in which pressurized mud drives the rotor, it will be recalled that there is a flow of mud down the outside of the rotor assembly past the top end mountings of the blades **62,** so that pressurized mud presents itself in the annular space **80** defined between the outer faces **81** (FIGS. **3** and **4**) of the blades and the inside of the casing tube **1.** The mud pressure causes clockwise rotation of the blades about their twisting axes as seen from above, accompanied by twisting of the torsion shafts **73** in the manner previously described, and a mud pressure in excess of a predetermined magnitude causes rotation of the blades to their positions indicated in FIGS. **3** and **6,** with their inner faces **79** engaging the shaft **37.** In these circumstances the torsion created in the shafts **73** is transmitted to the tubular body member **60** of the rotor assembly via the sector-shaped portions **76** of the shafts and the wear pieces **77,** so as to drive the rotor assembly in clockwise rotation. The blades have rubber sealing

strips **82** inset in their end faces **83**, which strips are squeezed between their associated blades and the rear faces **84** of the adjacent blades when the blades are rotated inwardly by the mud pressure, so as to exclude mud from the spaces **85** (FIGS. **4** and **5**) which initially exist between the inside faces **79** of the blade and the shaft **37**. A vent passage **86** is provided connecting the spaces **85** behind the blades to a region of reduced mud pressure below the drive assembly, to release the vacuum which will be created when the mud is depressurized, so as to enable the blades to open and thus to stop the rotor. Such a vent passage also prevents any build-up of high pressure mud behind the blades if the seals **82** should leak slightly, so as to maintain the necessary pressure differential across the blades during operation.

Means are provided for permitting, in flow-restricting fashion, the escape of pressurized mud from the annular space **80** surrounding the blades **62** when the rotor assembly is rotating. Thus each blade **62** is formed with an integral radially outwardly extending step **87** at its lower end, which step is formed with an axially extending slot **88** which receives a tungsten carbide insert **89** formed with a restricted passage **90** for the flow of mud therethrough. The slot **88** extends into the outer face **81** of the blade to some extent so that the insert **89** is somewhat inset relative to the face **81** (see FIGS. **3** and **4**). As the blades rotate inwardly in operation each of the passages **90** gradually becomes fully aligned with one of a ring of six passages **91** defined in the tubular body member **60** and closed at their outer sides by strips **92** (FIGS. **1** and **8**). Referring particularly to FIGS. **1** and **7** it will be seen that the inlets to the passages **91** of the body member **60** are defined by tungsten carbide inserts **93** and thence lead into oblique passages **94** for the flow of mud into the interior of the member **60**, i.e. to the central space **95** thereof indicated in FIGS. **1** and **9**. A rubber sealing ring **96** is inset in the top surface of the tubular body member **60** to seal against the bottom faces of the blades **62**.

The mud which has been employed to rotate the rotor assembly loses only a relatively small part of its pressure in so doing, in this particular embodiment of the invention the pressure lost being in the region of 250 p.s.i. This mud, having performed its driving function, flows freely through a central passage of the remaining length of the apparatus to the drill bit to perform the conventional cooling and scavenging functions. The rate of rotation of the rotor assembly, and thus of the drill bit, produced in the above described manner is of the order of 200 to 250 r.p.m., i.e. the desired and conventional rate of rotation of a drill bit in this kind of apparatus. At the same time, however, much of the power extracted from the mud is not lost but is converted into torque so that as well as rotating at the desired speed the drill bit is also provided with the necessary power.

The remaining length of the rotor assembly of the motor, below the drive assembly just described, is rotatably mounted in the casing tube **1** by sealed bearing means to be described below. This lower part of the rotor assembly is basically constituted by the tubular rotor shaft **5** previously referred to, which as already mentioned is connected by six drive pins **6** to the bottom end of the drive assembly of the rotor, specifically to the bottom end of the tubular member **60** constituting the lower body part of that assembly. The body member **60** is further located relative to the top end of the rotor shaft **5** by a lock nut **97** screw-threadedly mounted on the rotor shaft **5** and carrying a ring of six Allen screws **98** engaged in threaded bores in the lower end of the member **60** to hold the latter firmly against rotation relative to the rotor shaft. The top end of the rotor shaft **5** is thus firmly secured against an internal shoulder **99** of the tubular member **60** and a rubber sealing ring **100** is provided at this point.

Because the drive assembly is of course rotatable within the casing tube **1**, an annular passage **101** exists between the lower end region of the drive assembly and the inside of the casing tube, and there is pressurized mud in this passage in operation. This mud, which has not passed through the restricted passages **90** of the blades **62**, is at the same pressure as when it enters the top of the motor, and this pressure is hereinafter referred to as "top pressure". This mud thence fills a further annular passage **102** defined between an alongate nut **103** screw-threadedly mounted on the rotor shaft **5** and the inside of a connector member **104** whose function is to join together upper and lower parts of the outer casing tube **1**. This top pressure mud eventually reaches the top end of a seal structure **105** whose purpose is to exclude mud from the radial bearings and thrust bearings of the rotor shaft. The seal structure **105** is of similar construction to the seal structure **44** previously described.

The purpose of the division of the casing tube **1** into two parts interconnected by the connector member **104** is to enable the upper part of the casing tube to be easily removed for access to the drive assembly, and the removal of that assembly from the rotor shaft **5** when desired.

The construction and manner of lubrication of the radial bearings and thrust bearings of the rotor shaft **5** will now be described with reference to FIG. **1**.

It will be recalled that the shaft **37** constituting the upper part of the body of the drive assembly is provided with two internal bores **41** and **59** which respectively receive pressurized grease (at top pressure) from the cylinder **27** and top pressure mud. A tube **106** is secured in the lower end of the grease bore **41** of the shaft **37** and extends right through the interior of the rotor shaft **5** to the lower end thereof. Similarly a tube **107** is secured in the lower end of the mud bore **59** in the shaft **37** and again extends to the lower end of the rotor shaft **5**. The purpose of the tube **106** is to deliver grease to the radial and thrust bearings to lubricate the same, whilst the purpose of the delivery of top pressure mud via the tube **107** to the lower end of the motor is to prevent leakage of lubricant from the bearings at such lower end, in a manner to be described below. Although not shown in the drawings, the tubes **106** and **107** do in fact cross each other in their passage through the rotor shaft.

The bearing assembly for the rotor shaft **5** includes a main axial thrust bearing structure generally indicated at **108**, two radial sleeve bearings **109** and **110** below the thrust bearing structure, and a third radial sleeve bearing **111** above it. The seal structure **105** already mentioned, located above the radial bearing **111**, is arranged to prevent the ingress of top pressure mud to the top end of the bearing assembly, whilst a second seal structure **112** (again similar to the seal structure **44**) has the purpose of preventing the ingress of mud to the lower end of the bearing assembly.

The thrust bearing structure **108** is of the kind described in detail in U.S. Pat. No. 3,630,634. Thus such thrust bearing structure comprises inner and outer stacks of rings **113** and **114** secured respectively on the rotor shaft **5** and the casing tube **1** and having arcuately shaped annular corner faces defining six axially spaced tracks for sets of bearing balls **115**, the arrangement being such that thrust is transmitted from the rotor shaft **5** to the casing tube **1** by shear stress in the bearing balls. The neighbouring pairs of track-defining rings **113** and **114** (both inner and outer) are separated by spacer rings **116** so dimensioned that the width of each ball track is slightly greater than the ball diameter. As a result, during the application of thrust to the bearing structure, the balls of each set are pinched between one pair of diametrically opposite track-defining rings whilst being free of the other pair. Each inner track-defining ring **113** rotates within and in contact with its adjoining outer track-defining ring **114** so that the bearing structure is in fact a journal bearing in addition to its main function as a thrust bearing.

Each of the radial bearings **109** to **111** comprises a hardened steel bearing sleeve secured to the casing tube **1** and rotatable on respective ones of a series of sleeves **117** mounted on the rotor shaft **5** and secured thereto by the elongate nut **103** previously mentioned. Referring more specifically to the upper radial bearing **111**, the bearing sleeve of this bearing is secured in position by a locking sleeve **118** screw-threadedly secured to the casing tube **1** at **119**, which locking sleeve also engages the top of the outer stack of rings **114** of the thrust bearing structure **108** to locate the same within the casing tube **1**.

The two lower radial bearings **109** and **110** again comprise hardened steel bearing sleeves secured to the casing tube **1** within a pair of sleeve **120** and **121** which are themselves secured in position between the lower end of the outer stack of rings **114** of the thrust bearing structure and the end fitting **4** of the casing tube **1** and sealed to the latter by an O-ring **122**. The lower seal structure **112** of the bearing assembly is mounted within the end fitting **4**.

The grease for lubrication of the bearing assembly is delivered from the tube **106** through an end fitting **123** of such tube and thence through three passages **124**, defined between the lower end of the rotor shaft **5** and the lowermost of the sleeves **117**, to an annular channel **125** defined in a packing ring **126** at the top end of the seal structure **112**. From this point the grease is able to flow in two directions. Firstly it flows around the outside of the seal structure **112** as far as the bottom end thereof, and secondly it passes up through the whole bearing assembly to the top end of the top seal structure **105**. It will be appreciated that the grease delivered in this way is at top pressure because it is pressurized by top pressure mud applied to the piston **33** at the top of the motor. Thus the grease which reaches the top end of the top seal structure **105** of the bearing assembly is at the same pressure as the mud which, as already described, presents itself to the top end of that seal structure via the passage **102**. As a result there is no substantial tendency for the top pressure mud presenting at the top end of the seal structure **105** to enter that seal structure and find its way into the bearing assembly. An air vent plug **127** is provided in the connector member **104** adjacent the top end of the seal structure **105** to enable the motor to be wholly filled with grease.

The purpose of the delivery of top pressure mud to the lower end of the motor via the tube **107** is to substantially prevent leakage of grease to the exterior through the bottom seal structure **112**. This mud is delivered via an end fitting **128** of the tube **107** through a passage **129** in the end fitting **8** of the rotor shaft to a passage in the lowermost sleeve **117** and thence to an annular groove **131** defined in that sleeve within the seal structure **112**. Thus top pressure mud is delivered to a point inside the seal structure **112** between the ends thereof and, from this point, is allowed to leak to the exterior of the motor along the inside of the seal structure to an outlet **132**, such leakage being brought about by the fact that the pressure of the mud flowing up the outside of the motor will be in the order of 1,000 p.s.i. less than top pressure. As the mud delivered to the inside of the seal structure **112** is at the same top pressure as the grease delivered to the top end of the seal structure it will substantially prevent the leakage of grease through the seal structure, substantially all of the leakage which occurs being mud which is of course readily expendable.

A plug **133** is provided for feeding grease into the passages **124** for the purpose of refilling the motor with grease when it is withdrawn from a borehole to change the drill bit. The end fittings **123** and **128** of the tubes **106** and **107** are held in position by a locking sleeve **134** screw-threadedly mounted in the rotor shaft end fitting **8**, and sealing lugs **135** are provided on both of the end fittings **123** and **128** which distort when the sleeve **134** is screwed tight so as to seal the end fittings relative to the fitting **8**.

It will thus be seen that the invention, at least in its preferred embodiment described above and illustrated in the accompanying drawings, provides pressurised lubrication system for the rotor bearing assembly.

What I claim is:

1. A hydraulic motor for driving a rock drill, including a housing connectible to a drill-string, a rotor shaft rotatably mounted via bearing means in said housing and connectible to a drill bit to transmit driving torque thereto, sealing means between the rotor shaft and the housing arranged to oppose access of drilling mud to said bearing means, and means effective in operation to apply the pressure of drilling mud, at a high pressure substantially that at which it is delivered to the motor via the drill-string, to pressurize lubricant present in said bearing means characterized by means effective in operation to convey drilling mud, at or near to said high pressure, through the said rotor shaft to the bottom end region of the said bearing means for application to the outside of the said sealing means at said bottom end region.

2. A hydraulic motor as claimed in claim **1**, wherein the said sealing means comprises inner and outer sealing devices and the said mud conveying means is arranged to convey the said mud to a point in said sealing means between said inner and outer sealing devices, a path being provided for the leakage of the said mud from the said sealing means out of the said housing.

3. A hydraulic motor as claimed in claim **2**, wherein each of the said sealing devices comprises a plurality of interengaging sealing rings.

4. A hydraulic motor as claimed in claim **1**, wherein the said mud conveying means is arranged to receive high pressure drilling mud directly from the drill-string without any substantial loss of pressure.

5. A hydraulic motor as claimed in claim **1**, wherein the said mud conveying means comprises a conduit extending through the hollow rotor shaft, an end fitting on said conduit which end fitting is secured relative to the rotor shaft, and a passage leading along the rotor shaft to the said sealing means.

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