United States Patent

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[33] **Great Britain**
[31] **33,142/69**

[54] **ROCK-DRILLING APPARATUS**
7 Claims, 12 Drawing Figs.

[52] **U.S. Cl.** **415/170,** 415/502, 415/193, 415/199
[51] **Int. Cl.** **F01d 11/08,** F01d 1/02
[50] **Field of Search** 415/502, 503, 170, 193, 199, 104, 105; 308/227

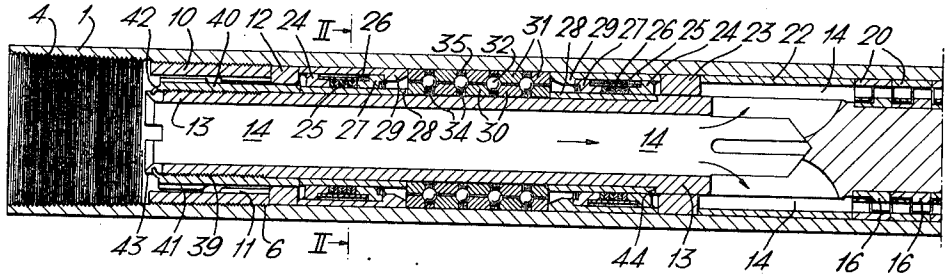
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Primary Examiner—Henry F. Raduazo
Attorney—Bacon & Thomas

ABSTRACT: A thrust-bearing structure for a turbodrill, and a turbodrill incorporating the same, comprising a stack of inner and outer ball bearing track members with sets of bearing balls running in a plurality of axially spaced tracks defined by said track members so as to be capable of transmitting axial thrust between the inner and outer track members by shear stress in the balls, the inner track members being arranged for the transmission of axial thrust between neighboring ones of such inner track members and the outer track members being arranged for the transmission of axial thrust between neighboring ones of such outer track members, whereby axial loads are distributed through the bearing structure in use.



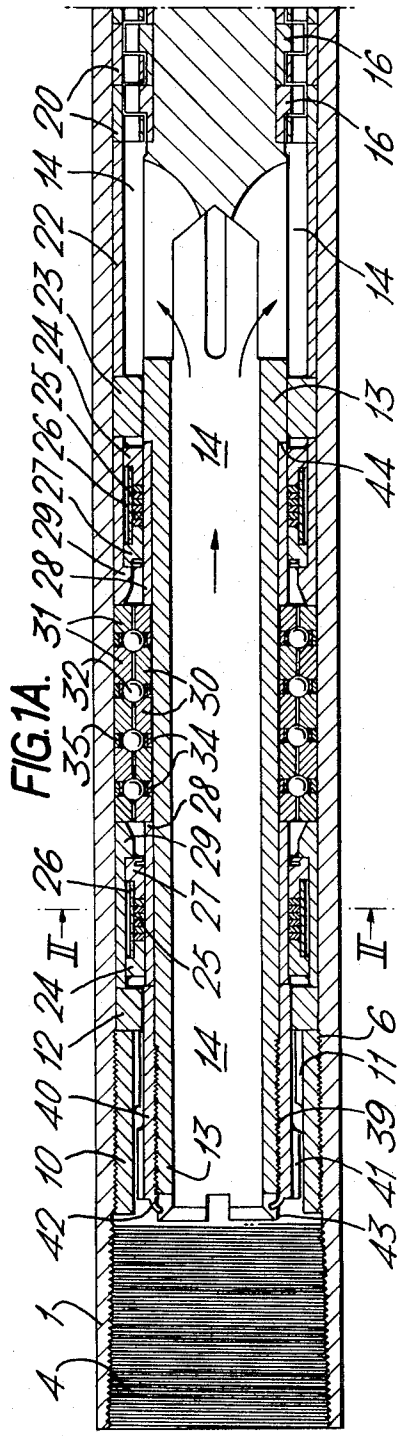


FIG. 1A.

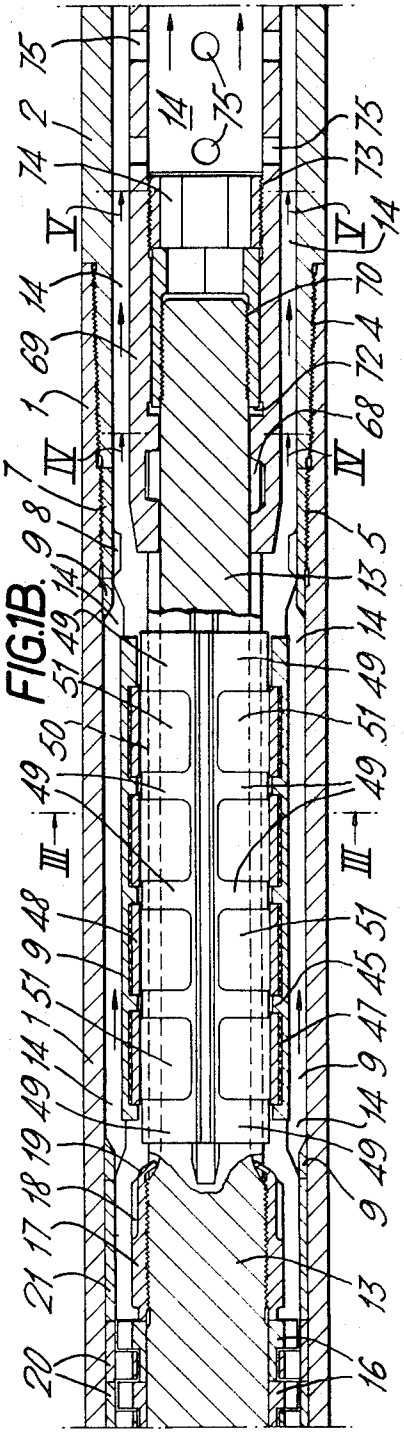


FIG. 1B.

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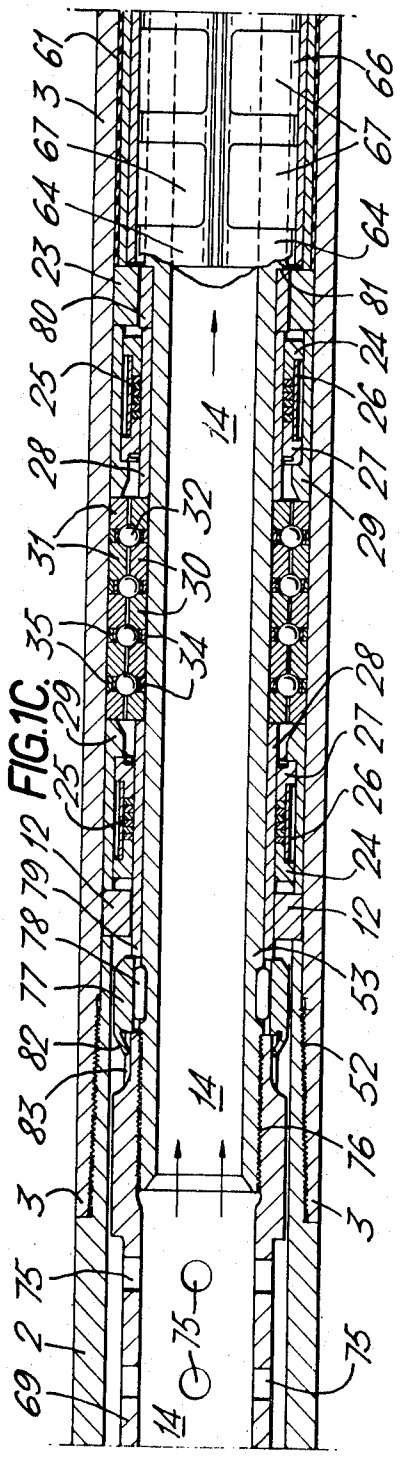


FIG.1C.

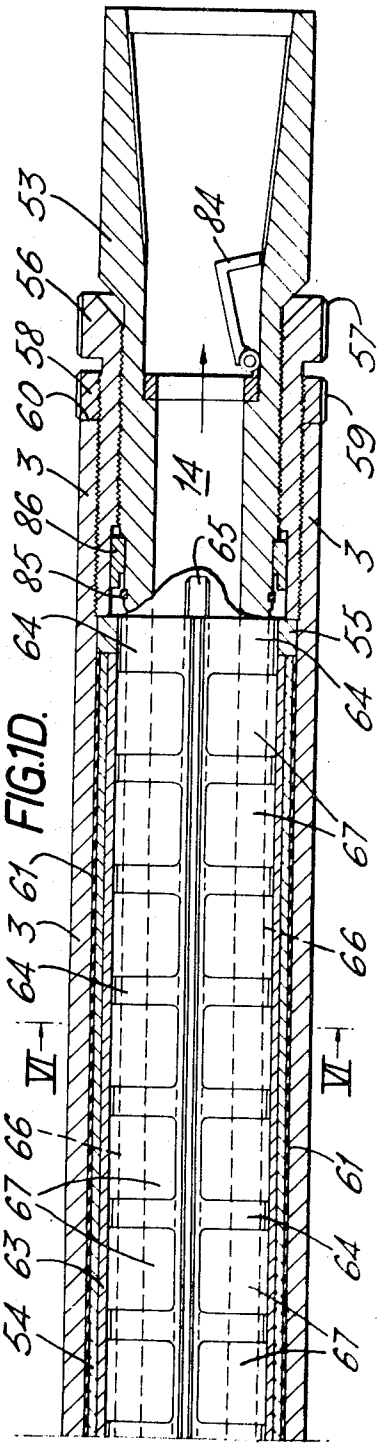
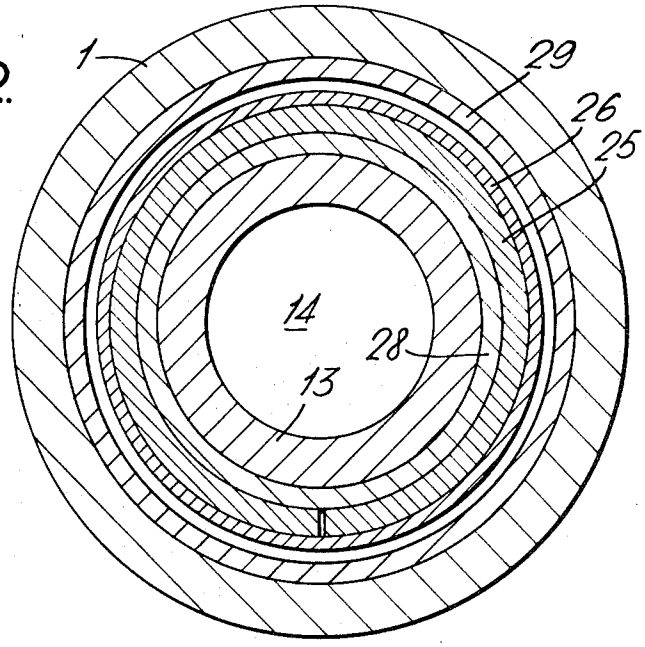


FIG.1D.

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FIG.2.



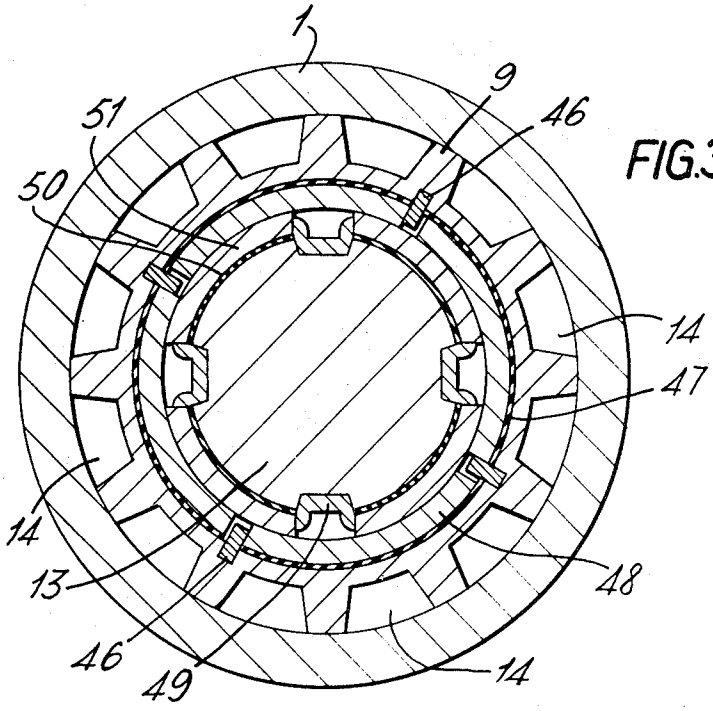


FIG.3.

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FIG.4.

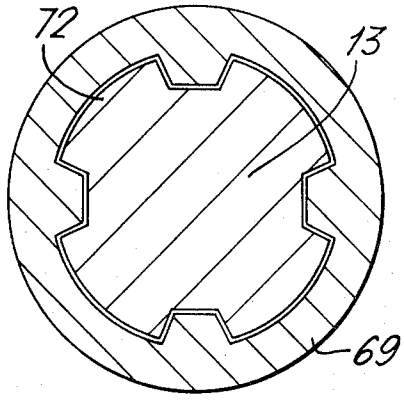


FIG.5.

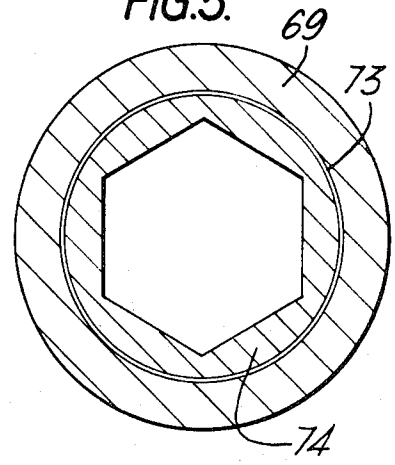
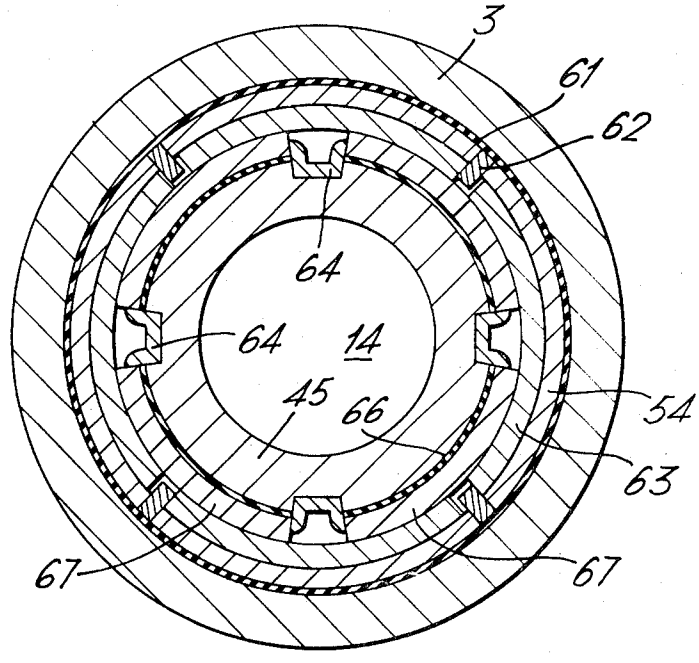


FIG.6.



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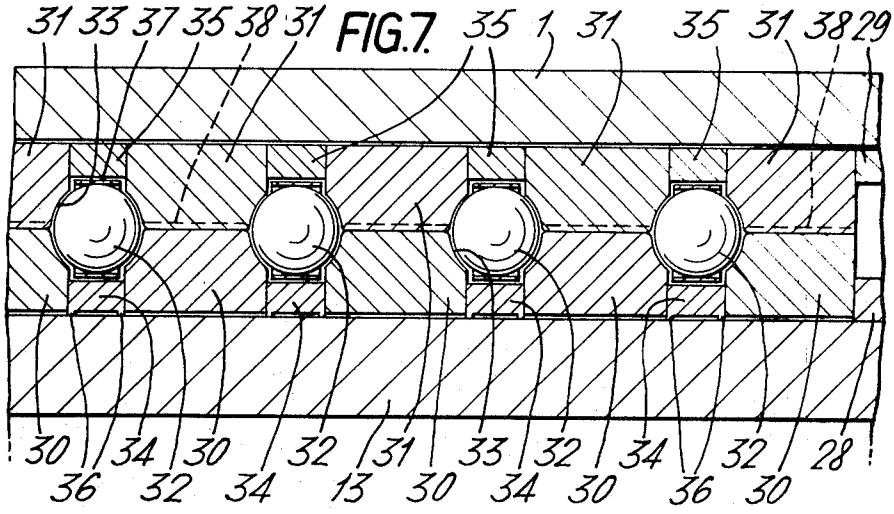


FIG.7.

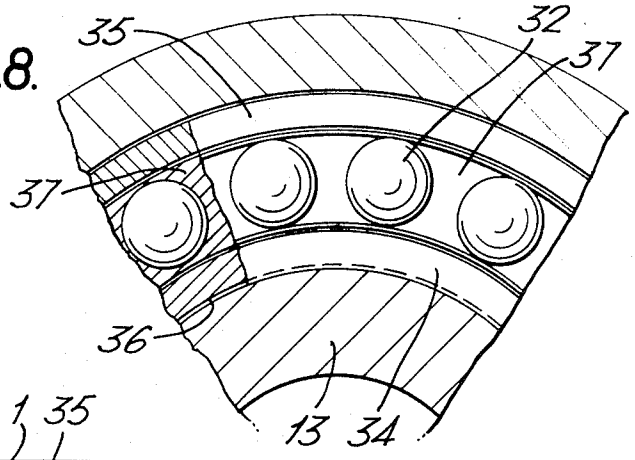


FIG.8.

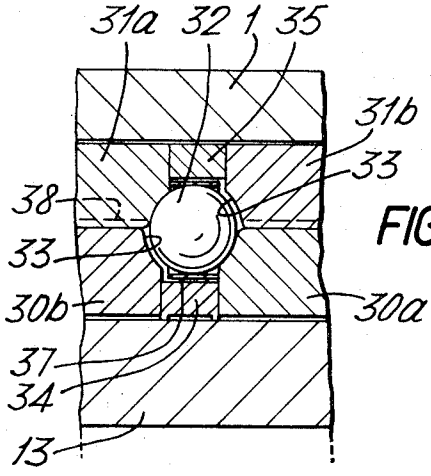


FIG.9.

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ROCK-DRILLING APPARATUS

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 lowermost 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 drill string pipes must also be straight and accurately formed so as to transmit torque without deflection that would cause the drill bit to "wander" and produce a nonrectilinear borehole. In addition, 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 borehole-flushing functions.

It has therefore been recognized 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 to 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 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 relatively to the drill string. The art is replete with proposals for turbodrill constructions but, so far as I am aware, no turbodrill construction yet proposed has demonstrated an ability to compete economically with a drill of the conventional construction discussed above. The reason for this is that when drilling at great depths, the ability to maintain continuous operation is of paramount importance; in the case of a turbodrill, therefore, any failure thereof involving its withdrawal to the surface for adjustment, repair or replacement is extremely costly in terms of loss of drilling time.

The "mud" that is used to cool the drill bit and to flush chippings from the borehole, and which in a turbodrill provides the energy source for driving the drill bit, 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 it should be understood that this fluid, 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. Because of the abstraction of energy from this mud, the pressure differential across the turbodrill and hence across various seals and bearings therein is often of the order of 600 to 1,400 pounds per square inch.

The bearings of a turbodrill therefore are prone to become contaminated with the highly abrasive mud that is used to drive the drill; since these bearings are subject to extremely high loadings and large load variations, it will be understood that bearing failure has usually been the main cause of unreliability in turbodrills so far proposed.

An object of this invention is therefore to provide a bearing structure capable of reliably operating under service conditions such as are encountered in a turbodrill, a further object being the provision of a turbodrill having such a bearing structure.

In one aspect therefore the present invention provides a thrust-bearing structure comprising a stack of inner and outer ball bearing track members with sets of bearing balls running in a plurality of axially spaced tracks defined by said track members so as to be capable of transmitting axial thrust between said inner and outer track members by shear stress in the balls, the inner track members being arranged for the transmission of axial thrust between neighboring ones of such inner track members and the outer track members being arranged for the transmission of axial thrust between neighboring ones of such outer track members, whereby axial loads are distributed through the bearing structure.

Ball bearings that operate in shear, as in the thrust-bearing structure of the invention, have a surprisingly high resistance to wear by abrasive particles coming between the ball and track surfaces; moreover in a stacked arrangement, as in the bearing structure of the invention, a high axial load will be distributed through the structure and should one or more of the track members or sets of bearing balls wear to a greater extent than the others then the load thereon will automatically be redistributed through the stack so as to avoid the breakup of the worn element(s) and total failure thereof and of its neighbors. Accordingly a thrust-bearing structure in accordance with the invention may be designed to accept a total load substantially greater than that likely to be encountered in service and will be capable of continuing to support the service loads in the event that wear of one or more elements of the bearing structure has resulted in the unloading of the worn element(s).

A thrust-bearing structure in accordance with the invention is inherently capable of supporting thrust loads in either direction along the axis thereof and has the same load-carrying capacity for either direction of applied thrust. Therefore such a thrust-bearing assembly is inherently suited for handling the thrust loads in a turbodrill since such thrust loads fluctuate and repeatedly reverse during many practical drilling circumstances.

In accordance with an important preferred feature of the invention, the said outer track members may be journaled on the said inner track members; with such an arrangement the structure constitutes a combined thrust and journal bearing.

Preferably the said ball tracks are defined by arcuately sectioned annular faces of said track members.

In one simple form of the invention the said thrust-bearing structure may be made up of a stack of individual ball thrust bearings each including inner and outer track members defining a ball track, with the inner and outer track members arranged as aforesaid for the transmission of axial thrust therebetween to distribute axial loads through the bearing structure.

Preferably each of the said ball tracks is defined by two neighboring inner track members and two neighboring outer track members. In these circumstances, other than when the structure is made up of a stack of individual ball thrust bearings as mentioned above, it is preferred that each said track member other than the terminal ones of the stack partly define two neighboring ball tracks.

Although the said neighboring pairs of inner track members could engage directly with one another for the transmission of axial thrust, as could the outer track members, in a preferred construction each said neighboring pair of said inner track members is separated by an inner spacer ring which defines the axial spacing of said pair of track members and constitutes an inner closure of the ball track partly defined by said pair of track members. If the said inner track members of the as-

sembly are to be mounted on means to be axially supported, e.g. a rotary shaft, which is made of a material having a higher coefficient of thermal expansion than the material of the said track members, it is essential that the track members be mounted radially spaced outwardly from such means. This may be achieved, in the preferred construction just mentioned, by making the minimum internal diameter of each of said inner spacer rings less than the minimum internal diameter of each of said inner track members and securing each such inner spacer ring to one of the two inner track members that it separates so that, in use, all of said inner track members are radially located by said inner spacer rings in positions spaced radially outwardly from means, e.g. a rotary shaft, mounting said spacer rings.

Similarly it is preferred that each said neighboring pair of said outer track members is separated by an outer spacer ring which defines the axial spacing of said pair of track members and constitutes an outer closure of the ball track partly defined by said pair of track members, each such outer spacer ring being secured to one of the two outer track members which it separates so as to be radially located by said track member.

Preferably, and in accordance with an important preferred feature of the invention, all of the said track members are so arranged as to define ball tracks of slightly greater width than the diameter of the said bearing balls, whereby upon the application of axial thrust to the bearing structure in either direction each of the balls will be pinched in rolling shear between those two respective track surfaces, forming diametrically opposite portions of the ball track concerned, whose axial spacing is reduced by thrust in the direction in question, while being held spaced from the other two track surfaces whose axial spacing is increased.

The combined effect of the above-described preferred relative arrangements of the track members and spacer rings is to ensure that all of such elements, and the sets of bearing balls, are maintained precisely coaxial in operation, whatever the direction of axial thrust on the bearing assembly.

The track members and bearing balls are preferably made of an extremely hard wearing material such as tungsten carbide; the spacer rings can be of a rather less hard wearing material, such as high-tensile steel, if desired.

Preferably the assembly is formed with passages for the flow of lubricating liquid through the assembly from end to end thereof, via each of the said ball tracks. When the outer track members are slidably and rotatably mounted on neighboring inner track members, as aforesaid, such passages for lubricant may conveniently be defined by and between the inner and outer track members.

In another aspect the invention provides a turbodrill comprising a housing having a rotary assembly supported therein against axial thrust loads by at least one thrust-bearing structure as set forth above.

Preferably there are at least two such thrust-bearing structures arranged to support the said rotary assembly of the turbodrill, the rotary assembly being arranged to divide axial thrust loads between said bearing structures irrespective of the direction of such thrust loads. Thus in a preferred arrangement the said rotary assembly comprises a turbine rotor which is supported by one said thrust-bearing structure, and a drill shaft that is supported by another said thrust-bearing structure, the drill shaft and the turbine rotor being coupled by an adjustable coupling sleeve adapted for the transmission of torque and axial thrust.

Preferably, in such a turbodrill, the maximum external diameter of the said outer track members of the or each said thrust-bearing structure is slightly less than the minimum internal diameter of the said housing, so as to allow for tolerances in the inside diameter of the element or elements forming the housing.

Preferably the or each said thrust-bearing structure has its said inner track members secured in longitudinal compression on said rotary assembly by sleeves mounted on and locked to said assembly, and its said outer track members secured in longitudinal compression on said housing by sleeves mounted in and locked to said housing.

Preferably a rotary seal structure is provided between the said housing and the said rotary assembly above and below the or each thrust-bearing structure, to minimize to ingress of drilling mud to the thrust bearing.

The invention further comprises a seal structure for providing a seal between relatively rotating parts, for instance a shaft and a bearing housing, such seal structure comprising a pack of slit rings having their individual splits indexed out of alignment, said rings being disposed in an outer sleeve adapted to apply radially inwards thrust upon the rings of said pack, said sleeve being fixed and sealed to end rings respectively abutting the opposite ends of said split ring pack, the inner faces of said split rings being adapted to engage and seal against an internal cylindrical member.

The invention also provides a shaft-bearing assembly comprising inner and outer relatively rotatable cages respectively defining external and internal opposed recesses each housing a shell-like bearing pad defining a segment of a cylindrical surface, at least the working surface of each such bearing pad being formed of a hard, wear-resistant material and the recesses in at least one said cage also housing resilient pads backing the bearing pads therein.

Conveniently the said bearing pads are formed of ceramic or cermet material, or ferrobestos.

Such a shaft bearing assembly with its resiliently backed bearing pads is capable of operating in very adverse environmental conditions and in particular is capable of functioning for a considerable time when exposed to drilling "mud" and consequently such a bearing assembly is particularly suitable for supporting the radial loads of the rotating assembly of a turbodrill. Thus the invention further provides a turbodrill having its rotating assembly radially supported at least partially by a shaft-bearing assembly as above defined.

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

FIGS. 1A, 1B, 1C and 1D are consecutive portions of an axial sectional view of a turbodrill embodying the invention;

FIG. 2 is a transverse cross-sectional view at station II—II of FIG. 1A;

FIG. 3 is a transverse cross-sectional view at station III—III of FIG. 1B;

FIGS. 4 and 5 are respectively transverse cross-sectional views at stations IV—IV and V—V of FIG. 1B;

FIG. 6 is a transverse cross-sectional view at stations VI—VI of FIG. 1D;

FIG. 7 is enlarged view in axial section of part of a thrust-bearing structure of the turbodrill;

FIG. 8 is a transverse cross-sectional view at station VIII—VIII of FIG. 7; and

FIG. 9 is another enlarged view, in axial section, of part of the thrust-bearing structure showing the manner in which the structure operates under axial thrust.

FIGS. 1A, 1B, 1C and 1D together illustrate, in axial section, a turbodrill embodying the present invention, FIG. 1A representing the upper end of the turbodrill and FIG. 1D showing the lower end thereof that is adapted, in operation, to be fitted with a suitable drill bit.

For convenience of construction and to facilitate assembly, the outer housing of the turbodrill illustrated in the drawings is formed in three sections, an upper section 1, an intermediate section 2, and a lower section 3. The upper casing section 1 is formed at each end with an internal taper thread 4, the upper such thread being adapted to receive the mating thread provided at the lower end of a drill pipe (not shown) while the lower taper thread 4 receives a taper-threaded spigot at the upper end of the casing section 2. The casing section 1 further has internal parallel threads 5 and 6 at its lower and upper

ends, respectively, the thread 5 receiving a thrust sleeve 7, having spanner slots 8, which abuts the lower end of a shaft-bearing assembly outer cage 9 hereinafter described and also abuts against and locks on final assembly against the upper end of the casing section 2. The thread 6 at the upper end of the casing section 1 receives a lockring sleeve 10 having spanner slots 11 and which abuts against a distance piece 12.

The upper casing section 1 houses the turbine of the turbodrill. This turbine comprises a rotor in the form of a shaft 13 the upper section of which is hollow to form a passage 14 communicating with the bore of a drill pipe to which the casing section 1 is fixed, thereby to enable the drilling mud to pass through this passage 14 and, via ports 15, into the annular space between the shaft 13 and the casing 1 downstream of an upper thrust-bearing structure and its seals hereinafter described. The intermediate portion of shaft 13 is fitted with a stack of bladed rings 16 slidably fitted to the shaft and held in place by a screwed collar 17 having spanner slots 18 and locking tags 19, the collar 17 urging the stack of rings 16 upwardly along the shaft 13 against a shoulder thereon. The blades on rings 16 coact with stator blades on a set of rings 20 fitted within the casing 1, the rings 20 being located between a distance ring 21, that engages the upper end of the bearing outer cage 9, and a distance sleeve 22 that in turn abuts against a distance piece 23.

The shaft 13 is supported within the casing section 1 by means of a thrust-bearing structure disposed between a pair of seal structures. The two seal structures are identical in construction, each comprising an end ring 24 and a set of split rings 25 encircled by a sleeve 26 and trapped between the end ring 24 and a similar end ring 27. The pack of rings 25 is expanded to fit over an inner sleeve 28 and is maintained under radially inwards directed compression upon the sleeve 28 by shrink fitting of the sleeve 26 and welding this to the end rings 24 and 27, so that the pack of rings 25 is also under axial compression between the end rings 24 and 27. The sleeve 28 is fitted to the shaft 13 and the end rings 24 and 27 are mounted within a stepped sleeve 29 that fits within the casing 1.

Between the two seal structures is disposed a thrust-bearing structure for supporting the rotor shaft 13 relative to the turbodrill casing, which bearing structure is illustrated on a larger scale in FIGS. 7 to 9. The thrust-bearing structure comprises a stack of inner and outer bearing ball track-defining members comprising inner ring members 30 and outer ring members 31, all of tungsten carbide, defining between themselves six axially spaced tracks for sets of tungsten carbide bearing balls 32. The ball tracks are defined by arcuately sectioned annular face portions 33 of the inner and outer track members 30 and 31 and it will be noted that all of the track members except the terminal ones of the stack partly define two neighboring ball tracks. Each neighboring pair of inner track members 30 is axially spaced from one another by an inner spacer ring 34 of high-tensile steel and the neighboring outer track members 31 are similarly spaced by outer spacer rings 35 again of high-tensile steel. Each of the inner spacer rings 34 is secured to the inner track member 30 above it (i.e. to the left of it as seen in FIG. 7) to constitute an outer closure of the adjacent ball track, and is formed internally with a pair of annular ring portions or pips 36 of slightly smaller internal diameter than the inner track members 30 so as to space the inner track members slightly radially outwardly from the shaft 13 on which they are mounted. The outer spacer rings 35 are similarly secured to the outer track members 31 above them so as to be radially located thereby and thus to form outer closures for their adjacent ball tracks. The outer track members 31 are slidably and rotatably mounted on their respective neighboring inner track members 30, so that the bearing structure in fact constitutes a combined thrust and journal bearing; it will thus be seen that the inner track members 30 are radially located by the inner spacer rings 34, the outer track members are radially located by the inner track members, the bearing balls are radially located by the inner and outer track members, and lastly the outer spacer rings 35 are radially located by the outer track members 31; thus all of such elements are maintained in precisely coaxial relationship in operation, all coaxial with the turbine rotor shaft 13.

The inner track members 30 and inner spacer rings 34 are axially located in longitudinal compression, and thus held to the shaft 13, between the two sleeves 28 of the seal structures, while the outer track members 31 and outer spacer rings 35 are axially located in longitudinal compression, and thus held to the casing 1, between the sleeves 29 of the seals. The downstream sleeve 29 abuts against the distance piece 23 with the result that the lockring sleeve 10 is effective to locate, against axial displacement, the sleeves 29 of the seals, the outer track members 31 of the thrust-bearing structure, the stator rings 20, and the bearing cage 9 against thrust sleeve 7 which is screwed into the thread 5 and locked back against the end of casing section 2 at the lower end of the casing section 1. Thus the inner track members 30 are arranged for the transmission of axial thrust between neighboring ones of such track members, and the outer track members 31 are similarly arranged for the transmission of axial thrust therebetween, whereby axial loads are distributed through the bearing structure in use.

All of the track members 30 and 31 are so spaced by the spacer rings 34 and 35 as to define bearing ball tracks of slightly greater width than the diameter of bearing balls 32. In the illustrated embodiment, in which the bearing balls are 0.60 inch in diameter, the width of the ball tracks (by which is meant the maximum dimension thereof measured parallel to the axis of the bearing structure) is about 0.015 inch greater than the diameter of the bearing balls. As a result, and referring now particularly to FIG. 9, upon the application of axial thrust to the bearing structure in either direction the balls of each set will be pinched in rolling shear between those two respective inner and outer track members defining the track in question (designated 30*a* and 31*a* in FIG. 9) whose axial spacing is reduced by thrust in the direction in question, while being held spaced from the other two track members (30*b* and 31*b*) whose axial spacing is increased. Thus the axial thrust on the shaft 13 is transmitted to the casing 1 by rolling shear stress in the bearing balls 32. Upon the direction of axial thrust changing, the slight freedom of the balls in their tracks permits momentary "freewheeling" of the balls before they are pinched between the other pair of track members.

The slight radial spacing of the inner track members 30 from the shaft 13, maintained by the annular pips 36 of the inner spacer rings 34, permits thermal expansion of the (steel) shaft 13 which will occur at a substantially higher rate than will expansion of the tungsten carbide track members. It will be noted that the outer track members 31 and spacer rings 35 are also slightly radially spaced inwardly from the casing 1, the purpose of this being to allow for manufacturing tolerances in the interior diameter of the casing.

The space between the two seal structures is filled with a suitable lubricating oil and the thrust-bearing structure is formed with passages 38, defined by and between the neighboring pairs of inner and outer track members 30 and 31, for flow of oil through the bearing structure via the bearing ball tracks.

The upper end of the shaft 13 has an external screw thread 39 on which is fitted a lockring 40 having spanner slots 41 and locking tags 42 that engage in castellations 43 at the end of the shaft 13. The lockring 40 abuts against the upstream seal inner sleeve 28 which in turn abuts against the uppermost inner track member 30 of the ball thrust-bearing structure, the lowermost inner track member 30 abutting against the downstream seal inner sleeve 28 which in turn engages a shoulder 44 on the shaft 13; thus the lockring 40 is effective to secure the seal sleeves 28 and the inner track members 30 of the thrust-bearing structure in a fixed axial position on the shaft 13.

The lower end portion of the shaft 13 is supported within the casing section 1 by a shaft bearing the function of which is to locate this end of the shaft 13 against radial displacement.

This shaft-bearing assembly comprises the cage 9 that, as previously described, fits within the casing section 1 and is located between the thrust sleeve 7 and the distance ring 21.

As best seen in FIG. 2, the cage 9 has, in effect, external longitudinal splines that engage the inner wall of the casing section 1 and thus define a series of longitudinal passages for the flow of drilling mud that has passed through the blades of the turbine. These passages are marked 14 in the drawings to indicate that they constitute a continuation of the mud flow path that starts with the passage 14 in the hollow upper section of the shaft 13.

The cage 9 is formed at intervals along its length with internal flanges 45 and is further fitted with internal longitudinal ribs 46 in the form of keys. The flanges 45 and ribs 46 thus define the boundaries of recesses in the internal surface of the cage 9 and each of these recesses is fitted with a rubber backing pad 47 and a bearing pad 48 of part cylindrical shell-like form constructed from a hard, wear-resistant material such as a ceramic, a cermet, or ferrobestos the inner face of which provides the actual bearing surface.

Within the cage 9 the shaft 13 is fitted with a complementary cage 49 longitudinal ribs of which are located in grooves in the periphery of shaft 13 so as to fix the cage 49 to the shaft for rotation therewith. This inner cage 49 defines external recesses complementary to those in the bore of cage 9 and each of which houses a rubber backing pad 50 and a bearing pad 51 of part cylindrical shell-like form and formed of a hard, wear-resistant material such as a ceramic, a cermet, or ferrobestos. The outer surfaces of the pads 51 thus constitute the bearing surfaces for the shaft 13 and engage the internal bearing surfaces of the bearing pads 48.

The rubber backing pads 47 and 50 urge the respective bearing pads 48 and 51 into firm engagement with one another. It will be noted that the shaft-bearing structure is exposed to the drilling mud and indeed relies upon the latter for lubrication and cooling. The bearing pads may be made of a material such as a ceramic, a cermet, or ferrobestos the bearing surface at least of which is appreciably harder than the rock particles suspended in drilling mud so as not to be extensively damaged by the presence of such particles between the surfaces of cooperating bearing pads. Moreover the individual pads are able to accommodate themselves to wear and, when worn, may be replaced with relative ease and economy.

Of the three outer casing sections of the turbodrill, the intermediate section 2 is relatively short and constitutes a coupling between the sections 1 and 3. As previously described the upper end of the section 2 has a taper-threaded spigot mating with the thread 4 at the lower end of the casing section 1. The lower end of the casing section 2 also has a taper-threaded spigot that is received by an internal taper thread 52 in the upper end of the lower casing section 3.

The lower casing section 3 houses a hollow drill shaft 53 the lower end of which protrudes from the end of the casing section 3 and is formed with an A.P.I. taper socket threaded to receive a drill bit (not shown).

The drill shaft 53 is supported within the casing section 3 by two bearing assemblies, one of which is a thrust-bearing structure identical in construction to that which supports the upper end of the turbine shaft 13 in the casing section 1 and previously described in relation to FIGS. 1A and 7 to 9; the thrust-bearing structure for the drill shaft 53 will, therefore, not be further described. In FIG. 1C, parts of the thrust-bearing structure for the drill shaft 53 have been marked with the reference numerals used in FIGS. 1A and 7 to 9 to designate the same parts of the thrust-bearing structure for the turbine shaft 13.

As in the case of the thrust-bearing structure for the turbine shaft 13, the thrust-bearing structure for the drill shaft 53 is disposed between a pair of seal structures that serve to protect the thrust-bearing from ingress of drilling mud. These seal structures are identical with those used to protect the thrust-bearing assembly for the turbine shaft 13 and accordingly will not be further described herein. The parts of these seal structures shown in FIG. 1C are marked with the same reference numerals as the corresponding parts of the seal structures shown in FIG. 1A.

It should be noted in FIG. 1C that the lower end of the intermediate casing section 2 abuts against a thrust ring 12 fitted within the casing section 3, the thrust ring 12 in turn abutting against the outer sleeve 29 of the upstream seal structure, this sleeve 29 in turn abutting against the uppermost outer track member 31 of the thrust-bearing structure, the lowermost outer track member 31 engaging the outer sleeve 29 of the downstream seal structure which sleeve 29 in turn abuts against a distance piece 23.

The distance piece 23 is engaged by the upper end of the outer cage 54 of a shaft bearing to be described, the lower end of cage 54 abutting against a distance ring 55 which is engaged by a lockring 56, having spanner slots 57, that screws into a parallel internal thread at the lower end of casing section 3 and is locked in place by means of locking ring nut 58, having spanner slots 59, that is tightened against the lower end surface 60 of the casing section 3 to secure the locking ring 56.

The lower portion of the drill shaft 53 is further supported against radial displacement by a shaft-bearing structure that is similar in construction to the previously described shaft-bearing structure that supports the lower portion of shaft 13 in casing section 1, but differs from that structure in three respects: in that its outer bearing cage 54 is fitted with an external rubber sleeve 61 that engages the bore of casing section 3 to provide a degree of resilience for the outer cage 54; in that there are no resilient backing pads for the bearing pads in the cage 54; and in that the cage 54 has no internal flanges equivalent to the flanges 45 of the cage 9. However, the cage 54 is fitted internally, like cage 9, with longitudinally extending keys 62 that define internal ribs between which are disposed part cylindrical shell-like bearing pads 63 formed of a hard, wear-resistant material such as a ceramic, a cermet, or ferrobestos.

Within the bearing cage 54, the drill shaft 53 is fitted with an inner bearing cage having circumferential and longitudinal ribs 64 that locate within longitudinal grooves 65 in the periphery of shaft 53, the circumferential and longitudinal ribs of the cage defining recesses each of which is fitted with a rubber backing pad 66 and a part cylindrical shell-like bearing pad 67 formed of a hard, wear-resistant material such as ceramic, a cermet or ferrobestos; the outer faces of the pads 67 constitute the bearing surface for the shaft 53 and engage the bearing surface formed by the internal faces of the bearing pads 63 in the outer cage 54.

As in the case of the shaft-bearing structure that supports the lower end of the turbine shaft 13, the shaft-bearing structure for the drill shaft 53 is particularly well adapted to withstand its environment, the bearing pads 63 and 67 withstanding abrasion by the rock particles suspended in drilling mud and being readily and economically replaceable when worn.

The drill shaft 53 is coupled to the turbine shaft 13 in order that these two shafts shall together constitute a single rotating assembly and in such manner that the thrust loads on the two shafts of such assembly may be combined and the resultant thrust load distributed between the two thrust-bearing structures that have been described. The turbine shaft 13 and the drill shaft 53 are coupled together by coupling components that are mainly housed within the intermediate casing section 2, the upper part of which is shown in FIG. 1B and the lower part in FIG. 1C.

Referring first to FIG. 1B, it will be seen that the lower portion of the turbine shaft 13 has splines 68 complementary to internal splines in the upper end of a coupling sleeve 69. The lower extremity of the shaft 13 has a screw thread 70 that receives an internally threaded sleeve 71 adapted to abut against a shoulder 72 at the lower end of the splined section of shaft 13.

The coupling sleeve 69 has an internal thread 73 to receive an externally threaded locking ring 74 that abuts against the sleeve 71.

The coupling sleeve 69 has ports 75 in its wall to enable drilling mud to flow from the annulus between the casing section 2 and the sleeve 69 into the bore of the latter and thence to the bore of the drill shaft 53. Thus the said annulus and the bores of the sleeve 69 and the drill shaft 53 constitute a further continuation of the mud passage 14 in the turbine shaft 13 and are accordingly marked 14 in the drawing.

The lower end of the coupling sleeve 69 is internally screw threaded to screw upon a thread 76 at the upper end of the drill shaft 53. As seen in the FIG. 1C, the lower end of the coupling sleeve 69 abuts against a tag sleeve 77 that is keyed to shaft 53 by keys 78. The lower end of tag sleeve 77 abuts against a distance sleeve 79 which in turn abuts against the inner seal sleeve 28 of the upper seal for the drill shaft thrust bearing, this sleeve 28 abutting against the uppermost inner track member 30 of the thrust-bearing, the lowermost inner track member 30 of which abuts in turn against the inner sleeve 28 of the lower bearing seal. This lower sleeve 28 abuts against a distance ring 80 that seats against a shoulder 81 on the shaft 53.

The turbodrill as so far described is assembled by first assembling the sections thereof constituted by the upper casing part 1 and the turbine components housed therein, on the one hand, and the casing section 3 and the drill shaft components housed therein, on the other hand, so as to produce two subassemblies each comprising an outer casing section with a rotating unit supported therein by a sealed thrust-bearing assembly at its upper end and a shaft-bearing assembly at its lower end. These two subassemblies are then coupled together by means of the coupling components that in the assembled turbodrill are housed mainly within the intermediate casing section 2.

In coupling these said subassemblies, the coupling sleeve 69 is first fitted internally with the internally threaded sleeve 71 and the externally threaded locking ring 74, whereafter the lower end of the coupling sleeve 69 is screwed onto the upper end of the drill shaft 53, being locked in place thereon by bending tags 82 on the tag ring 77 into slots 83 provided for this purpose at the lower end of the coupling sleeve 69. At the same time, the intermediate casing section 2 is fitted to the lower end of the upper casing section 1, thrust sleeve 7 (which is backed off during fitting of the casing section 2 so as to enable this fully to enter the section 1) being thereafter tightened back upon the upper end of section 2 whereafter the lockring 10 is fully tightened to secure the bearings and seals in the casing section 1.

The lock 56 is now backed off and the two subassemblies with the respective coupling components added as described are brought together, the lower end of the turbine shaft 13 being introduced into the upper end of the coupling sleeve 69 and the respective splines on these two components mated; then the lower casing section 3 is connected to the intermediate casing section 2 through the taper thread 52, the turbine shaft 13 sliding down within the coupling sleeve 69 as these two casing sections are screwed together. The lockring 56 is then tightened and locked by the ring nut 58.

Finally, a suitably extended hexagon key is inserted through the hollow drill shaft 53 and the coupling sleeve 69 into a hexagon socket at the lower end of the sleeve 71 so that the latter may be screwed up upon the thread 70 on the lower extremity of the turbine shaft 13. Finally the said hexagon key is sufficiently withdrawn to enable it to engage only the locking ring 74, which is of internal hexagon shape, and this locking ring is screwed into engagement with the lower end of the sleeve 71.

From the foregoing description it will be understood that the coupling sleeve 69 constitutes an extension of the drill shaft 53, torque being transmitted from the turbine shaft 13 via the splined connection between the latter and the sleeve 69. The sleeve 71 and the locking ring 74 together serve to connect the coupling sleeve 69 to the turbine shaft 13 in such manner that the drill shaft 53 may transmit upward axial thrust to the turbine shaft 13 to be distributed between the two thrust-bearing structures while the normal downwards thrust reaction on the turbine shaft 13 may be transmitted to the drill shaft 53 to be distributed between the two thrust-bearing structures. In practice therefore, the sleeve 71 and the locking ring 74 are adjusted to apply a preload to the thrust-bearing structures so that the normal axial loads on the respective shafts of the rotating assembly may be balanced against each other and the resultant distributed between the two thrust-bearing structures.

As illustrated in FIG. 1D, the lower end of the bore of the drill shaft 53 is fitted with a nonreturn valve 84 that is shown in its open position in the drawing in which it permits the flow of drilling mud to the bit (not shown). The valve 84 will however close to prevent a reverse flow of drilling mud (or other material) from the borehole.

Moreover, as also shown in FIG. 1D, the lower end of the drill shaft 53 is fitted with a circlip 85 and a loose retaining ring 86 that may engage an internal shoulder on the locking ring 56 to retain the drill shaft (and bit) in the casing section 3 in the event of a breakage of the drill shaft 53 at a point below the thrust-bearing structure in the casing section 3.

Although the bearing structures and seal structures embodied in the illustrated turbodrill have been designed and developed for duty in a turbodrill it should be understood that such structures have utility in other fields; for instance in marine propeller shaft arrangements.

I claim:

1. A turbodrill comprising a housing having a rotary assembly supported therein against axial thrust loads by at least one thrust-bearing structure comprising a stack of inner and outer ball bearing track members with sets of bearing balls running in a plurality of axially spaced tracks defined by said track members so as to be capable of transmitting axial thrust between the inner and outer track members by shear stress in the balls, the inner track members having means for the transmission of axial thrust between neighboring ones of such inner track members and the outer track members having means for the transmission of axial thrust between neighboring ones of such outer track members, whereby axial loads are distributed through the bearing structure in use.

2. A turbodrill as claimed in claim 1, including at least two of said thrust-bearing structures arranged to support the said rotary assembly of the turbodrill, the rotary assembly having means to divide axial thrust loads between said bearing structures irrespective of the direction of such thrust loads.

3. A turbodrill as claimed in claim 2, wherein the said rotary assembly comprises a turbine rotor which is supported by one said thrust-bearing structure, and a drill shaft that is supported by another said thrust-bearing structure, the drill shaft and the turbine rotor being coupled by an adjustable coupling sleeve adapted for the transmission of torque and axial thrust.

4. A turbodrill as claimed in claim 1, wherein the maximum external diameter of the said outer track members of the said thrust-bearing structure is slightly less than the minimum internal diameter of the said housing.

5. A turbodrill as claimed in claim 1, wherein the said thrust-bearing structure has its said inner track member secured in longitudinal compression on said rotary assembly by sleeves mounted on and locked to said assembly, and its said outer track members secured in longitudinal compression on said housing by sleeves mounted in and locked to said housing.

6. A turbodrill as claimed in claim 1, wherein a rotary seal structure is provided between the said housing and the said rotary assembly above and below the thrust-bearing structure, to minimize the ingress of drilling mud to the thrust-bearing.

7. A turbodrill as claimed in claim 6, wherein each said rotary seal structure comprises a pack of split rings having their individual splits indexed out of alignment, said rings being disposed in an outer sleeve adapted to apply radially inwards thrust upon the rings of said pack, said sleeve being fixed and sealed to end rings respectively abutting the opposite ends of said split ring pack, said end rings sealingly engaging said housing and the inner faces of said split rings sealingly engaging the said rotary assembly.

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