Sept. 22, 1964 J. A. MITCHELL ETAL 3,149,685

SHAFT BEARING FLUID SEAL FOR DOWNHOLE DRILLING ENGINES

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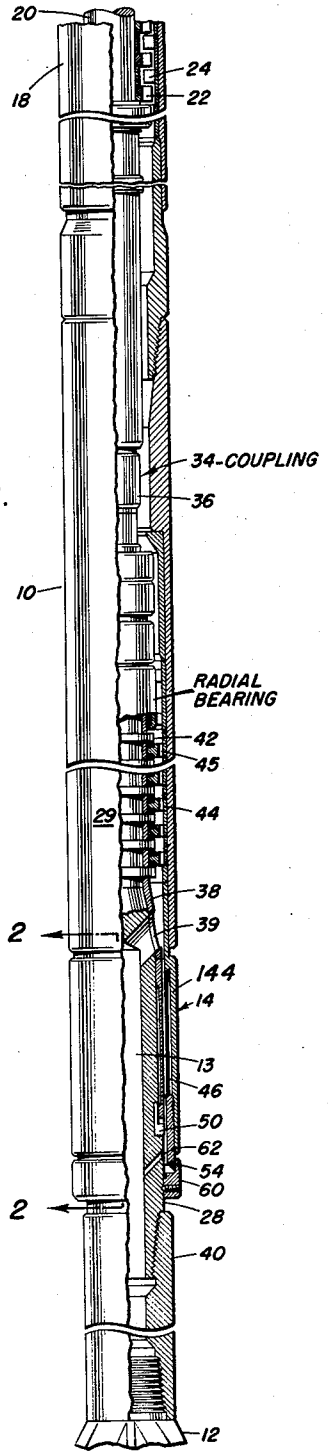


FIG. 1.

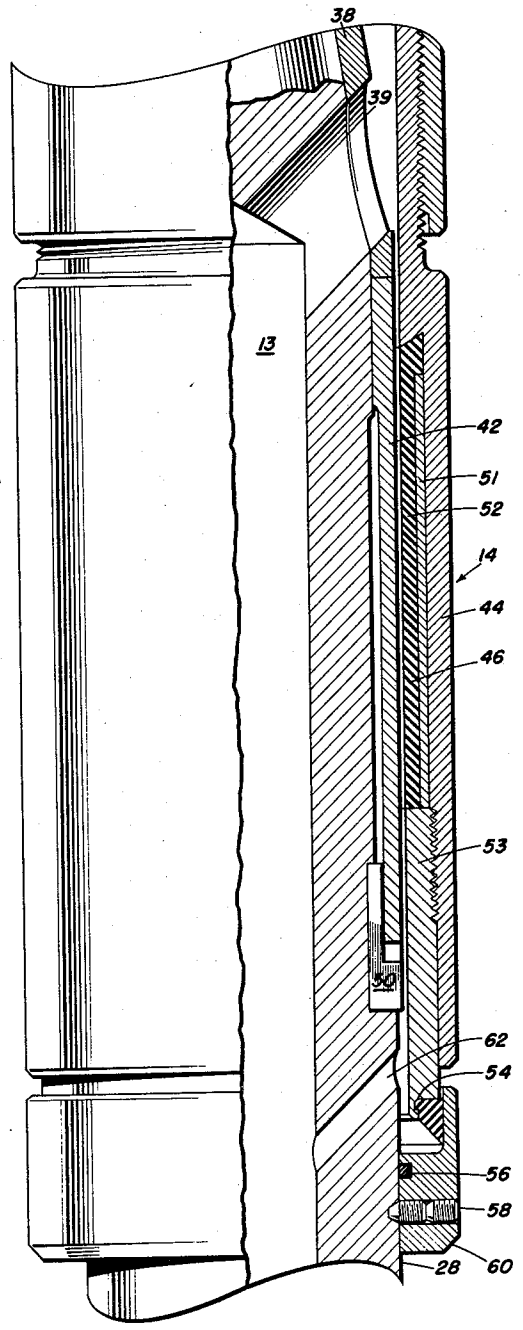


FIG. 2.

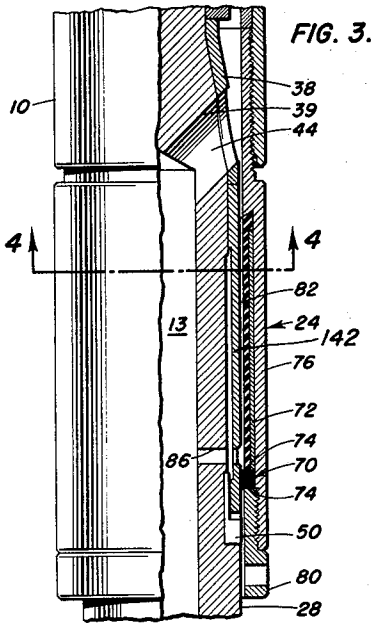
INVENTORS
JOSEPH A. MITCHELL
MADDEN T. WORKS
BY THOMAS E. TUCKER, JR.
Leech and Radue
ATTORNEYS

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FIG. 3.



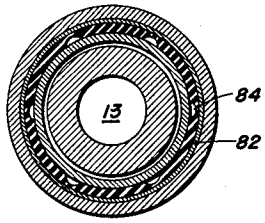


FIG. 4.

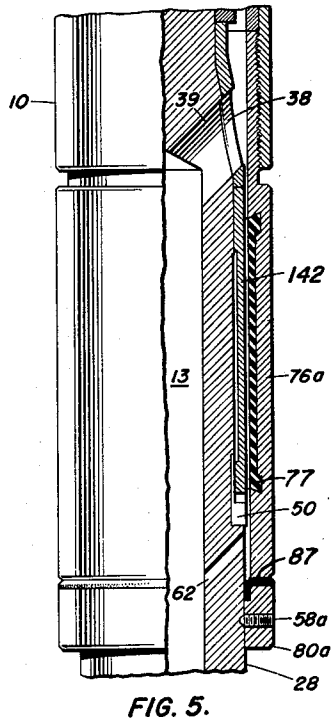


FIG. 5.

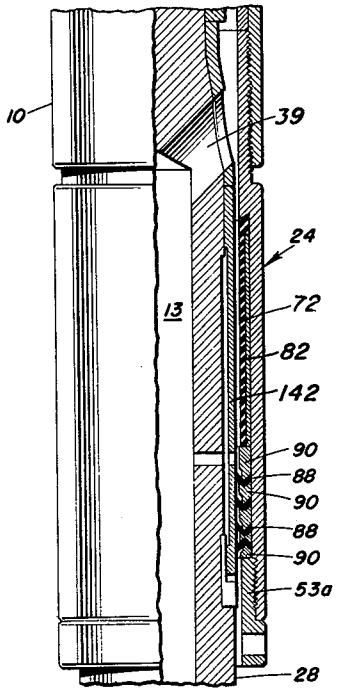


FIG. 6.

JOSEPH A. MITCHELL
MADDEN T. WORKS
THOMAS E. TUCKER, JR.
INVENTORS

BY
Leech and Radue
ATTORNEYS

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SHAFT BEARING FLUID SEAL FOR DOWNHOLE DRILLING ENGINES

Joseph A. Mitchell and Madden T. Works, Dallas, and Thomas E. Tucker, Jr., Midland, Tex., assignors to Dresser Industries, Inc., Dallas, Tex., a corporation of Delaware

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7 Claims. (Cl. 175—107)

The present invention relates to a turbodrill or fluid driven turbine that is carried on the end of a drill string and, actuated by the downflowing drilling fluid, drives a rotary drill bit utilized in the drill of oil wells and the like.

Although the turbodrill has been known for a number of years, as evidenced by Cross Patent No. 142,992 issued in 1873, and Westinghouse Patent No. 307,606 issued in 1884, it is only in very recent years that the basic turbodrill invention has advanced the turbodrill toward a state of commercial usage. There were various technical reasons for the delay, such as the early development of direct rotary drilling which, following its adoption around 1900, became the accepted procedure. Rotary drilling remains the standard procedure in this country, and from the standpoint of feet drilled per hour of operating time, it has not as yet been commercially superseded.

However, the early concepts of turbodrill operation and subsequent modifications have led the inventively minded in this field to give a great deal of consideration to the realization of the great economies which this form of drilling could afford. These are evident, because in rotary drilling there may be as much as 90% energy loss due to the friction of the rotary pipe in the bore hole. Moreover, the effect of the heavy loading on the bit, which now runs as much as 40,000 to 100,000 pounds, not only requires very heavy drill collar sections, but under many circumstances results in angular deviations in drilling that exceed the permisisble limits for well bore verticality. With a reduction of load on the bit, there is an attendant reduction in rate of penetration and frequently supplemental equipment such as whipstocks must be used to correct these deviations.

The use of a turbodrill has the distinct advantage over rotary drilling in that the driving torque is close to the bit. It utilizes the drilling fluid which is necessary for removing the cuttings from the well and it can be made of adequate power to permit rapid drilling. The direct drive multiple stage type turbodrill does have a somewhat higher rotational speed as compared to normal rotary operation, but with improved bits, the turbodrill can be expected to drill considerably faster and cheaper than is possible with rotary drilling.

A turbodrill is, however, a down-hole engine which operates in a fluid-laden well bore, and the highly abrasive fluid stands in the way of any normal lubrication of the wear parts. The limited diameter of the well bore, which is usually less than twelve inches, also limits the internal design beyond that possible with turbines used above ground.

One of the serious problems that has developed in the operation of such downhole drilling engines is a tendency of the high velocity, downwardly moving drilling fluid to leak out through the lower radial bearing of downhole drilling engine such as the turbodrill and, due to this velocity effect, to force the bit cutting back down to the rotating rock bit, thus causing wear and erosion on the bearing race of the rotating bit elements and requiring the bit to recut these formation particles which reduces the bit's efficiency. Hence, one of the principal objects of this invention is to overcome this grave problem of effectively

protecting the rock bit against deterioration by eliminating the downward fluid flow from the bearing.

A further and specific object of this invention is to provide for cooling and lubricating the turbodrill shaft bearing structure by partial bypass of drilling fluid and its return into the turbodrill driven shaft above the lower bearing packing seal. The nature of these problems and the means of their solution will be more fully understood from the following detailed description and a consideration of the accompanying drawings, in which:

FIG. 1 is a side elevational view of the lower portion of a turbodrill having parts sectioned longitudinally to show the principal parts of a drill, its bearings, and a drill bit used therewith;

FIG. 2 is an enlarged side elevation of the lower part of the turbodrill housing partially sectioned to show a preferred form of seal and its retaining means;

FIG. 3 is a partially sectioned side elevation similar to that of FIG. 2 but embodying modified shaft sealing means;

FIG. 3 is a partially sectioned side elevation similar to that of FIG. 2 but embodying modified shaft sealing means;

FIG. 4 is a reduced scale cross-section taken on line 4—4 of FIG. 3;

FIG. 5 shows in partially sectioned side elevation another modification of shaft sealing assembly; and

FIG. 6 illustrates in partially sectioned side elevation an additional shaft sealing assembly.

Reference will now be made to the drawings and to FIG. 1 in particular, which shows a turbodrill unit 18 provided with a bearing assembly 10 and a conventional drill bit 12. The lower shaft 28 of the assembly has a tubular bore 13, and is surrounded on the lower end thereof by an annular bearing and sealing assembly 14. Particular attention will be directed herein to the novel features and improvements in a turbodrill for the operation of a rotary drill bit used in down-hole drilling of oil wells as described more fully in joint patent application, Serial No. 764,213, filed September 30, 1958, in the names of Otto Hammer and Joseph A. Mitchell, entitled "Turbodrill."

In common with the disclosure of the above-identified prior application, there are set forth herein several shaft sealing and packing devices for preventing high pressure leakage flow of abrasive drilling fluid from destructively eroding the drill shaft or the housing, or interfering materially with the upward return flow of drilling fluid.

With reference to FIG. 1 of the drawing, the turbodrill and bearing assembly include one or more turbine units 18, each provided with a shaft 20 having rotor blades 22 which are alternately spaced with stator blades 24 mounted in the turbodrill housing. The combination of the rotor and stator blades provides a multiple stage turbine unit which is actuated by the drilling fluid, and for typical operations has approximately one hundred pairs of blades or stages that develop from about one hundred fifty to three hundred horsepower. This, of course, will depend upon the drilling fluid circulation rate and is merely intended to reflect the approximate horsepower capacity.

A thrust bearing housing portion 29 extends donwardly of the turbodrill to a lower radial bearing housing section 144, with which it is connected.

Mounted below the turbine units 18 there is provided a separate bearing sub-assembly housing generally indicated at 10. This includes a shaft 28 which has rotating thrust bearing disks 42 which are alternately spaced with stationary rubberized bearing members 44. The construction is such that there is both support for and resistance to vertical movement of the shaft 28, as well as free passage for the drilling fluid in the passages 45, whereby

the drilling fluid after it drives the turbine and passes down over the bearings passes through the hollow portion 13 of the shaft 28, and then to and through the drill bit 12.

In the operation of the turbodrill, drilling fluid under high pressure flows downwardly through the turbodrill housing wherein it acts against the turbine rotor blades 22 and reacts against the stator blades 24 to effect a substantially continuous driving force on the driving shaft 20 of the turbodrill.

The hydraulic fluid thus applied produces a substantially continuous driving force on the shaft 20, which acting through the coupling 34 causes rotation of the lower shaft 28 and operation of the drill bit 12.

Before reaching the bottom end of the turbodrill, the drilling fluid passes through the thrust bearings and thence into a ported lantern ring 38 having a large fluid flow passage 39 extending between the lowermost end of the thrust bearing assembly and the tubular bore 13 of the shaft 28.

In order to support the turbine shaft by the lower shaft 28 with the necessary spacing between the rotor and stator blades 22 and 24, the upper shaft 20 is coupled to the lower shaft 28 through a coupling generally indicated by numeral 34 in FIG. 1. It includes two interengaging toothed clutch members, each of which has a shank portion appropriately threaded into an adjacent threaded bore of the respective shafts, such shank portions ending in a collar or shoulder section. These clutch members are in turn surrounded by a sealing collar or spacing ring 36 which is of extremely accurate length so as to precisely space the shoulder sections of the clutch members one from the other. The coupling unit referred to above comprises the subject matter of patent application Serial No. 768,474, filed October 29, 1958 in the names of Madden T. Works, Joseph A. Mitchell and John Russell Mooney now Patent No. 2,990,895.

As more clearly shown in FIG. 2, the ported lantern ring 38 affords a large fluid flow passage 39 between the lowermost end of the thrust bearing assembly and the tubular bore 13 of the shaft 28.

From the lower end of the shaft 28 the drilling fluid enters the bit sub 40 and is discharged to the borehole after passing through the drill bit 12. A radial bearing sub-assembly 14 encircles the lower end of the shaft 28 immediately above the upper extremity of the bit sub 40.

The sub-assembly 14 comprises a lower radial bearing housing 144 provided with a cartridge and sleeve type bearing element 46 held in place by a threaded retainer ring 53. A metallic key member 50 engages the sleeve 142 which thus rotates with the shaft 28. The bearing element 46 which is the same general assembly employed in the embodiments illustrated in FIGURES 3, 4 and 6, is made up of a rubber sleeve 52 reinforced by a metal sleeve 51. A resilient lip seal 54, which may be of rubber or the like, interlocks at its inner periphery with the adjacent end of the lower bearing retainer ring 53 and is surrounded by a support ring 60. The sealing of the lower end of this bearing assembly is completed by a seal member 56 which may be an O ring engaging the shaft 28. The support ring 60 is held securely in place by a plurality of setscrews 58.

From the preceding description of the construction and functions of the bearing that has been set forth in detail, it will be understood that the principal part of the high pressure fluid passes downward and discharges into the bore 13 of shaft 28 through the port 39. Some of the fluid, however, is used to cool and lubricate the bearing, and this portion passes between the cartridge type bearing 46 and the bearing sleeve 142. The fluid reaching a point below the bearing will discharge through the downwardly inclined port 62 and also enter into the shaft 28 for discharge through the drill bit.

The lip seal 54 interengaging with the adjacent lower outturned end of the threaded retainer ring 53, and the upper inner periphery of the support ring 60 together with the ring seal 56 acting on the shaft 28 will effectively prevent leakage from the bearing and packing assembly 14. The mounting setscrew means 58 provide for ready servicing when necessary.

The relatively minor modification of this invention as represented by the embodiments of FIGS. 3, 5 and 6 will be described in view of the detailed description previously made of the principal embodiment of FIGS. 1 and 2.

The most significant difference in the embodiment of FIG. 3 is the substitution of a T type seal 70 for the lip type one of FIG. 2. The leg of the T-ring seal 70 is positioned inwardly beneath a sleeve bearing assembly 72, extends inwardly and is supported by enclosing back-up rings 74 on the top and bottom, while its annular head portion bears outwardly against the bearing housing 76. A threaded retainer ring 80 holds the sleeve bearing assembly 72 and the T ring elements 70 and 74 in operative relation in the housing.

In the reduced size cross-section 4—4 of FIG. 3, as shown in FIG. 4, it will be noted that the inner rubber face 82 of bearing 72 is formed with spaced longitudinal grooves 84 to permit fluid to discharge into the lateral port 86 opening to the shaft bore 13. The portions of inner-rubber face 82 between grooves 84 are in contact with sleeve 142, the fluid in said grooves serving to lubricate this bearing arrangement. It will be noted that in this embodiment and that illustrated in FIGURE 6, the seal 70 lies above the lower end of the sleeve 142 and further that the lateral port 86 is positioned slightly above the seal. Therefore, there is a possibility of a very small leakage occurring at the lower end of the sleeve 142. However, any such leakage would be of insignificant proportion. The important matter is the preventing of leakage through the relatively large area of the fluted passage formed by the spaced longitudinal grooves 84 in the inner face 82 of sleeve bearing assembly 72. Obviously, if the leakage passing the lower end of sleeve 142 was ever considered serious, the arrangement illustrated in FIGURES 1, 2, and 5 could be employed wherein the lateral portion 62 is positioned below the lower end of sleeve 142.

Referring now to the embodiment of FIG. 5, it will be observed that an annular face seal 87 of L-shape in cross-section has its shorter leg portion interposed between the lower end of the bearing housing 76*a* which carries rubber sleeve bearing 77 on its inner face and the shaft 28 below the downwardly inclined port 62, and its longer leg portion extending between the end face of housing 76*a* and a retainer ring 80*a* secured in place on shaft 28 by a setscrew 58*a*.

Drilling fluid passing through these shaft bearings will likewise be removed through a port 62 to the bore 13 of shaft 28.

The only principal differences between the modification of FIG. 6 as compared with the T-type seal construction of FIG. 3 are the substitution of a plurality of chevron or V-type seal rings 88 spaced by interengaging support rings 90, and the positioning of the seal construction above the lower retainer ring 53*a* as clearly shown.

The novel packed off lower bearing assemblies for turbodrills as described herein afford numerous advantages in the operation and maintenance of turbodrills, such as the provision of effective cartridge type bearing assembly units that are replaceable with a minimum of time and labor.

A still further improvement contributing substantially to the bit cutting efficiency is obtained from the utilization of an auxiliary port below the bearing assembly and above a shaft seal arrangement to redirect the drilling fluid utilized for lubrication and cooling of the bearing means into the drill bore and thereby maintaining a high fluid pressure differential across the bit.

Having thus described our invention, what we claim as novel and desire to secure by Letters Patent of the United States is:

1. A bearing sub-assembly for a drilling fluid driven downhole drilling motor assembly having a hollow driven shaft rotated by said motor, tubular means connected to the motor and surrounding the upper portion of the driven shaft thereby forming an annular passage between said tubular means and the driven shaft adapted to receive drilling fluid discharged from said motor, and a bearing sub-assembly for and surrounding the lower portion of the driven shaft, said bearing sub-assembly comprising a cylindrical housing adapted to encircle the lower portion of the driven shaft, said housing being adapted to be secured to the lower end of the tubular means, a bearing sleeve carried within the housing and adapted to engage the encircled driven shaft, said sleeve having a plurality of spaced longitudinally extending grooves in the face adapted to engage the driven shaft so that drilling fluid may be passed down through the grooves to lubricate and cool the sleeve and the driven shaft riding thereagainst, a port in the driven shaft adjacent the lower end of the bearing sub-assembly adapted to discharge drilling fluid passed between the driven shaft and the bearing sub-assembly, and seal means adapted to prevent leakage of drilling fluid between the lower end portion of the housing and the encircled driven shaft.

2. The invention as described in claim 1 in which the seal means is rubber-like and has a lip seal formation.

3. The invention as described in claim 1 in which the seal means includes a packing ring having a T formation in cross-section and the end of the leg of the T is in sealing engagement with the driven shaft.

4. The invention as described in claim 1 in which the seal means includes a face seal of L formation having one leg of the L in sealing engagement with the driven shaft.

5. The invention as described in claim 1 wherein the seal means comprises a plurality of chevron type seal rings spaced by interengaging support rings.

6. The invention as described in claim 1 and wherein the bearing sleeve is rubber-like and is molded into the housing.

7. The invention as described in claim 1 and wherein the bearing sleeve comprises an inner rubber-like sleeve and an outer metallic reinforcing sleeve.

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