United States Patent [19]

Ioanesian et al.

[11] **3,728,040**

[45] **Apr. 17, 1973**

[54] **TURBODRILL**

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[21] Appl. No.: **136,525**

[52] **U.S. Cl.** **415/123,** 415/502, 188/296

[51] **Int. Cl.**.....**F01d 15/12,** F03b 3/10, E21b 1/06

[58] **Field of Search**.....188/290, 296; 415/123, 55, 502, 123, 502; 175/107, 12

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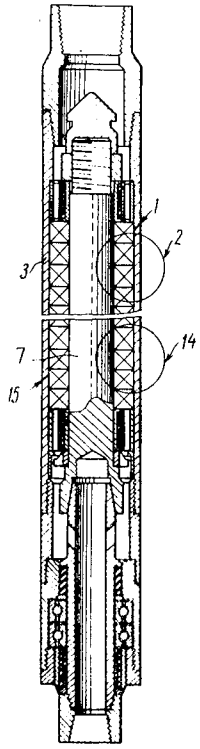
Primary Examiner—Henry F. Raduazo

Attorney—Holman & Stern

[57] **ABSTRACT**

A turbodrill for drilling wells having a turbine for the turbodrill shaft rotation together with a hydrodynamic braking unit, the stator and rotor blades of which extend in the same direction.

5 Claims, 13 Drawing Figures



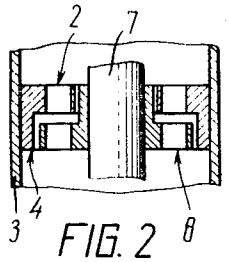


FIG. 2

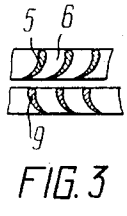


FIG. 3

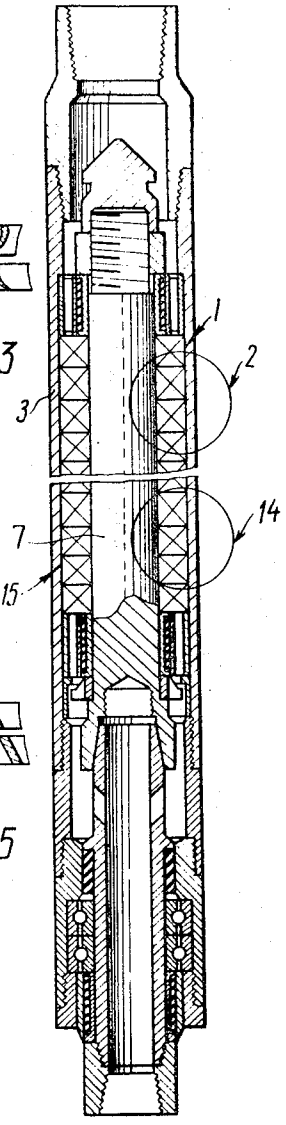


FIG. 1

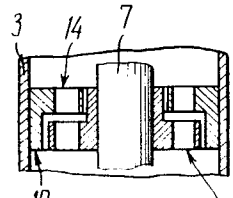


FIG. 4

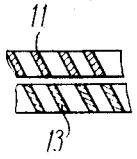


FIG. 5

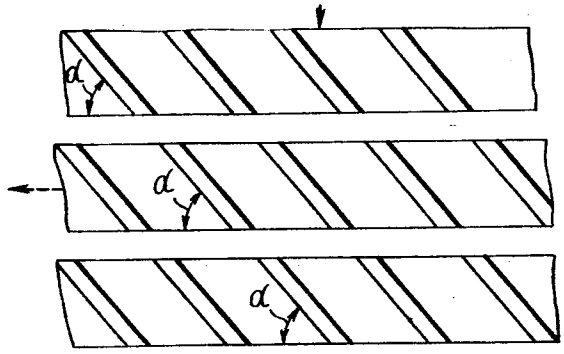


FIG. 6

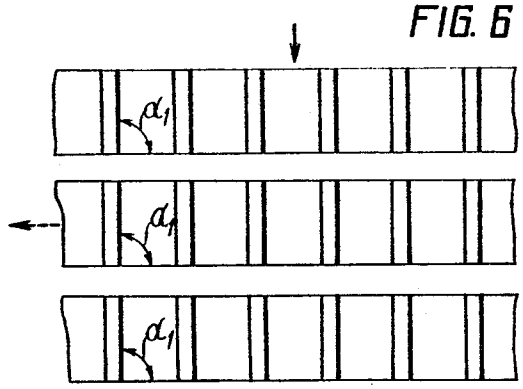


FIG. 7

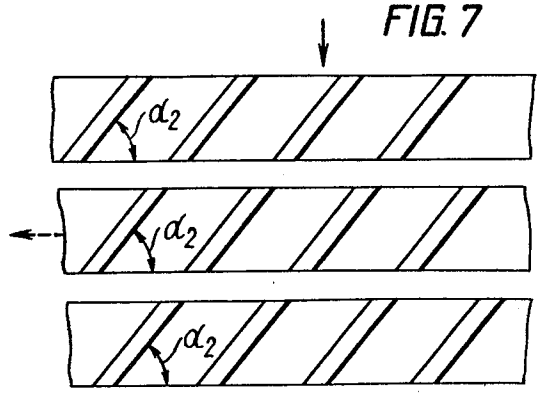


FIG. 8

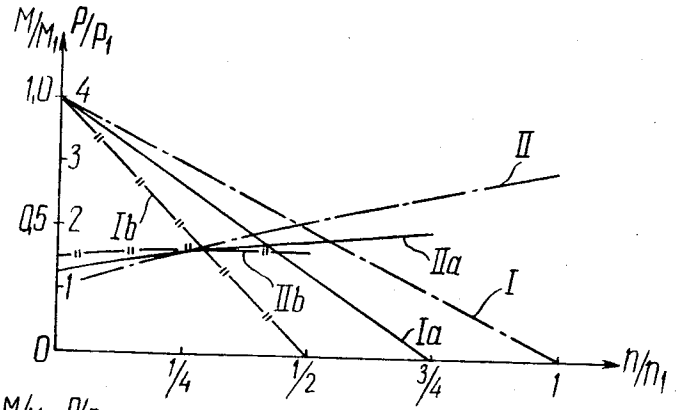


FIG. 9

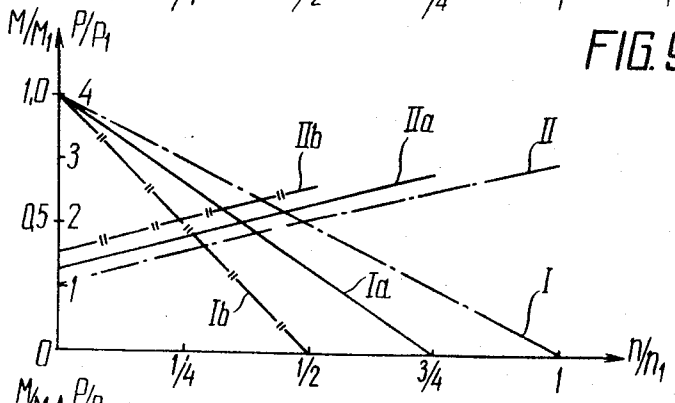


FIG. 10

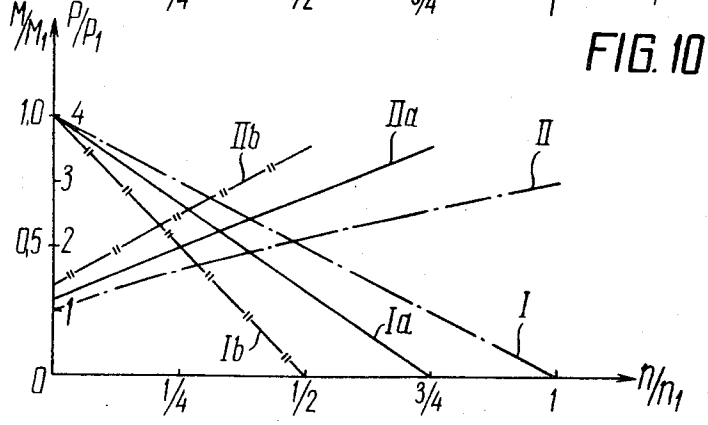


FIG. 11

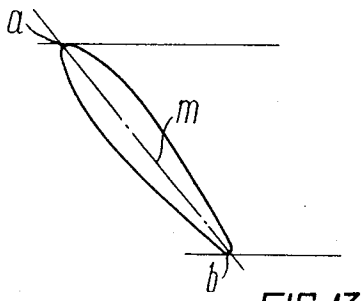


FIG. 13

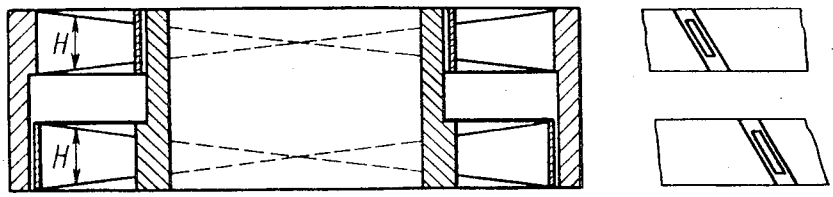


FIG. 12

TURBODRILL

BACKGROUND OF THE INVENTION

This invention relates to hydraulic bottom-hole motors for drilling deep wells in the surface of the earth with the object of obtaining oil, gas and other mineral resources, and more particularly to turbodrills.

PRIOR ART

Known in the art is a turbodrill for drilling oil or gas wells comprising a casing with fixed stators therein, the blades of which define guiding channels through which a drilling fluid passes, and a shaft mounted in ball bearings and provided with fixed rotors, the blades of which have a direction opposite to that of a stator and thereby change the direction of fluid motion, due to which a rotor, constituting with a stator the stage of a turbodrill turbine, rotates relative to the stator.

In the course of operation of the turbodrill, the shaft rotates with high speed (900 rpm) driving the rock breaking tool, for example a rock bit, into rotation. The drilling of wells by rock bits with a high speed of rotation results in a decrease in the footage per bit, and in view of the fact that in deep drilling, a substantial percentage of time is consumed by trips, the high speed drilling is economically inexpedient.

Turbodrill designs are known in which the r.p.m. is reduced by means of a mechanical reducer. However, these turbodrills are very complicated to manufacture, and are unreliable in operation because of the heavy-duty conditions of a reducer.

Also known are turbodrill designs in which a decrease in rotational speed of the shaft is attained by adjusting the flow rate of a fluid which is supplied to the turbodrill. Among these are the turbodrills provided with reducing valves and flow rate ejector-multipliers which effect a decrease in rotational speed of the shaft to that in rotary drilling. However, turbodrills with reducing valves and flow rate ejector-multipliers are also complicated to manufacture.

A principal object of the present invention is to provide a turbodrill of a simple design and which ensures a high reliability of a drilling bit operation at the rpm characteristic of rotary drilling.

SUMMARY OF THE INVENTION

The turbodrill of the present invention comprises a casing with immovably fixed stators, the blades of which define guiding channels through which a drilling fluid passes and a shaft mounted in a ball bearing means provided with immovably fixed rotors, the blades of which have a direction opposite to that of the stator blades and thereby change the direction of fluid motion, due to which a rotor constituting with a stator, the stage of a turbodrill turbine, rotates as related to a stator, and a means for reducing the shaft rotational speed. According to the invention, the means for reducing the rotational speed of a shaft is constituted by further stators fixed in the turbodrill casing and further rotors secured to the shaft, with both components constituting hydrodynamic braking stages in which the blades of the further stator and further rotor have, mainly, one and the same direction.

It is expedient that the stator and rotor blades, reducing the shaft rotational speed, be disposed at the same angles to a plane normal to the turbodrill shaft axis.

It is also expedient that the stator and rotor blades reducing the shaft rotational speed be of variable height, which increases proportionally to a radial distance of a blade section from the turbodrill shaft axis. This provides for the drilling fluid flow without surging shock through the stator and rotor blades and thereby reduces pressure losses across the braked turbodrill and aids in reducing head losses when the turbodrill is idle. The stator and rotor blades reducing the shaft rotational speed may also be of constant height. These blades have the advantage of being simple to manufacture.

The stator and rotor blades reducing the rotational speed of the shaft are preferably of a streamlined shape, formed by mold curves, with their middle lines being disposed at the same angle to a plane normal to the turbodrill shaft axis. This provides a decrease of hydraulic losses in the hydrodynamic braking stages.

According to this invention, a turbodrill of a simple design is developed, which provides the high reliability of a drilling bit operation at the rpm characteristic of rotary drilling.

The invention will be further described with reference to an embodiment shown in the accompanying drawings, in which:

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a longitudinal section through a turbodrill according to the invention;

FIG. **2** is a fragmentary sectional view of a turbine stage;

FIG. **3** is a fragmentary cross-sectional view of the stator and rotor blading of a turbine stage;

FIG. **4** is a fragmentary sectional view of a hydrodynamic braking stage;

FIG. **5** is a fragmentary cross-sectional view of the stator and rotor blading of a hydrodynamic braking stage;

FIG. **6** is a view illustrating an angle at which the stator and rotor blades of a turbodrill hydrodynamic braking stage, according to the invention, are mounted on the plane; at this angle the hydrodynamic braking stages act not only as a brake, but also as a pressure regulator;

FIG. **7** is a view illustrating the same angle at which the hydrodynamic stages offer the greatest braking effect;

FIG. **8** is a view illustrating the same angle at which the hydraulic resistance increases more intensively from braking regime to idling regime than at above said angles;

FIG. **9** is a diagrammatic view of turbine curves of the turbodrill with hydrodynamic braking stages and with the setting angle of blades according to FIG. **6**;

FIG. **10** is a diagrammatic view with the setting angle of blades according to FIG. **7**;

FIG. **11** is a diagrammatic view with the setting angle of blades according to FIG. **8**;

FIG. **12** is a longitudinal section of a hydrodynamic braking stage with blades the height of which increases proportionally to a radial distance of a blade section from the turbodrill shaft axis;

FIG. **13** is a cross-sectional view of a shaped rotor and stator blade of a hydrodynamic braking stage, the blade being of streamlined shape formed by mold curves.

DETAILED DESCRIPTION OF THE INVENTION

The turbodrill shown in FIG. 1, in its upper part has a turbine 1 comprising stages 2, with each being constituted by a stator 4 (FIG. 2) fixed in casing 3 (FIG. 1,2,4), with blades 5 (FIG. 3) thereof defining guiding channels 6, through which a drilling fluid passes, and a rotor 8 (FIG. 2) secured to a turbodrill shaft 7, with blades 9 thereof having a direction opposite to that of the blades 5 thereby changing the direction of fluid motion. As a result thereof, the rotor 8 rotates as related to the stator 4 and transmits the torque of rotation to the turbodrill shaft 7.

In the bottom part of the turbodrill, further stators 10 (FIG. 4) fixed in the casing 3 have blades 11 (FIG. 5) and further rotors 12 (FIG. 4) secured to the shaft 7 have the blades 13 (FIG. 5) with the stators 10 and rotors 12 being disposed at the same angles to a plane normal to the axis of the turbodrill shaft 7.

The blades 13 of the rotor 12 are represented as if they are an extension of the blades 11 of the stator 10 and vice versa. The stator 10 and rotor 12 constitute a stage 14, and stages form a hydrodynamic braking unit 15, which reduces the speed of rotation of the turbodrill shaft 7.

With a braked turbodrill shaft, a drilling fluid passes freely through the stator and rotor blades of the hydrodynamic braking stages. In this case, a minimum pump head is consumed, and hydraulic resistance in rotors and stators is due to the surface irregularities of the channels and inaccuracy of the blade shape.

As the rotational speed of a turbodrill rotor increases, which may be caused by a decrease of bit weight or by a reduction of torque load applied to a turbodrill rotor, an eddy zone is developed behind the blading. The rotor blades of the hydrodynamic braking stages start operating as a rotor of an axial pump, taking up a part of the rotor power and building up an effective fluid head which assists counteracts the mud pumps in forcing the fluid through the operating turbine of the turbodrill.

The rotor and stator blades of a hydrodynamic braking stage are shown in FIGS. 6, 7, 8. The stator and rotor blades of each stage have or extend in the same direction and are disposed at the same angles (α, α_1, α_2). The course of the drilling fluid motion is shown by solid arrows, and the direction of rotor motion, by dashed arrows.

The indicated angles distinguish three preferable variants of blades setting in rotors and stators of hydrodynamic braking stages, each of which provides for a specific characteristic of the turbodrill.

The blades setting in rotors and stators of hydrodynamic braking stages according to the first variant (FIG. 6) is expedient to use with turbines, the hydraulic resistance of which reduces as the rpm is decreased.

The initial characteristic curve of the turbodrill with such a turbine and the characteristic curve which is obtained as a result of setting up hydrodynamic braking stages with blades shown in FIG. 6, is shown in FIG. 9 in which

n_1 — is the idling rpm of the turbodrill shaft,
N — is the operational rpm of the turbodrill shaft,
M_1 — is the braking torque of the turbodrill,
M — is the operational torque of the turbodrill,
P_1 — is the braking pressure drop across the turbodrill,
P — is the operational pressure drop across the turbodrill,
I — the torque line $M/M_1 = f(n/n_1)$ for turbodrills without hydrodynamic braking stages,
II — the head loss line $P/P_{P1} = f(n/n_1)$ for a turbodrill without hydrodynamic braking stages;
I_a — the torque line $M/M_1 = f(n/n_1)$ for a turbodrill with K_1 number of hydrodynamic braking stages,
I_b — the same for K_2 number of hydrodynamic braking stages, where $K_2 > K_1$,
II_a — the head loss line $P/P_1 = f(n/n_1)$ for a turbodrill with K_1 number of braking stages
II_b — the same for K_2 number of hydrodynamic braking stages, where $K_2 > K_1$.

As illustrated in FIG. 9, in this case the hydrodynamic braking stages function not only as a brake impeding the turbodrill speeding up, but also as a pressure regulator which prevents the pressure from building up beyond an allowable value. By setting up a sufficient number of hydrodynamic braking stages, it is possible not only to decrease the idling rpm of the turbodrill shaft, but also narrow the operational rpm range.

With these hydrodynamic braking stages being used, the pressure loss across the turbodrill will be minimum.

In some cases, it is expedient to use the hydrodynamic braking stages with vertical rotor and stator blades, the second variant (FIG. 7) and experience has shown that such blades have maximum braking capacity. The characteristic curve of a turbodrill with such hydrodynamic braking stages is shown in FIG. 10. The designations in FIG. 10 correspond to the designations in FIG. 9.

Such hydrodynamic braking stages increase the hydraulic resistance factor of the turbodrill as a whole and particularly as compared to the braking stages made in accordance with the first variant (FIG. 6).

Vertical blades cannot be used as a head loss regulator in the turbodrill, but they are very simple and low-priced to manufacture: therefore they may find wide industrial use.

For a low number of turbodrill shaft operational rpm (in the order of tens rpm) to be obtained, the third variant of the hydrodynamic braking stages embodiment (FIG. 8) holds much promise.

In this case, with the turbodrill rotor being immovable, hydrodynamic braking stages, as well as in the first variant (FIG. 6), hardly result in an increase of pressure loss across the turbodrill, that is increase slightly the total pressure drop across the braked turbodrill as a whole.

However as the rotational speed of a turbodrill shaft increases, the turbodrill starts to operate more and more as an axial pump producing counterpressure for a main pump.

The hydraulic resistance factor of a turbodrill with a turbine, the hydraulic resistance of which decreases with a reduction of rotational speed of a turbodrill shaft increases the most intensively with an increase in speed of rotation as compared with the two first variants. The characteristic curve of a turbodrill with such hydrodynamic stages is shown in FIG. 11. The designations in FIG. 11 correspond to the designations in FIG. 9 and FIG. 10.

With just this variant of the hydrodynamic braking being used in turbodrills with flow rate ejector-mul-

tipliers, as well as in turbodrills with reducing valves, that is in all turbodrills in which the flow rate of a drilling fluid supplied to the turbine depends on the turbodrill operating conditions, the minimum operating rpm may be obtained. With the blades of hydrodynamic braking stages being inclined to a plane normal to the turbodrill axis (FIG. 6 and 8), the drilling fluid flow without surging shock through the stator and rotor blades is of great importance in reducing the hydraulic resistance factor of a braked turbodrill as well as the rate of head loss decrease with the turbodrill idle rotation, which is possible only when the outlet blades of the rotor and stator are directed along the radius. The last-mentioned condition is impossible with inclined blades being made of a constant axial height independently of the distance, at which the blades section is disposed from the turbodrill axis.

Therefore, for the head loss with a braked turbodrill to be reduced, the blades of the hydrodynamic braking stages according to the first and third variants (FIGS. 6 and 8) should be made of variable axial height along the radius, as shown in FIG. 12. In cases in which the axial height H of a blade increases proportionally to the distance of the blade section from the turbodrill axis, it is possible to fulfill two most important conditions, that is: the inclination of the blade axis to a plane normal to the turbodrill axis is the same along its full height and, in addition, the input and output edges of the blade are directed along the radius, that is the projections of the input and output blade edges on a plane normal to the turbodrill axis coincide with the radius directions.

To facilitate the hydrodynamic braking stages blades casting, they may be given a streamlined shaped, formed by mold curves. The shape of such a blade is shown in FIG. 13. In this case, the inclination of the blade to a plane normal to the turbodrill shaft axis, is determined by the inclination of the middle line "*m*," and the projections of input "*a*" and output "*b*" blade edges on a plane normal to the turbodrill shaft axis coincide with the radius directions.

We claim:

1. A turbodrill for drilling wells including: a casing; stators secured to said casing, the stators having blades, the blades of the stators defining guiding channels through which a drilling fluid passes; a shaft; ball bearing means on which said shaft is mounted; rotors secured to said shaft, said rotors having blades, the blades of the rotors having a direction opposite to that of the blades of said stator thereby changing the direction of fluid motion; due to which said rotor constituting with said stator a stage of a turbodrill turbine, rotates as related to said stator; and means for reducing the rotational speed of said shaft defined by further stators fixed in said turbodrill casing and further rotors secured to said shaft providing hydrodynamic braking stages said further stators and rotors having blades, the blades of said further rotors and further stators extending in the same direction, with the angles of the inlet and outlet of all blades being the same.

2. A turbodrill for drilling wells including: a casing; stators secured to said casing, the stators having blades, the blades of the stators defining guiding channels through which a drilling fluid passes; a shaft; ball bearing means on which said shaft is mounted; rotors secured to said shaft, said rotors having blades, the blades of the rotors having a direction opposite to that of the blades of said stator thereby changing the direction of fluid motion; due to which said rotor constituting with said stator a stage of a turbodrill turbine, rotates as related to said stator; and means for reducing the rotational speed of said shaft defined by further stators fixed in said turbodrill casing and further rotors secured to said shaft providing hydrodynamic braking stages in which the blades of said further rotors and further stators extend mainly in the same direction, with the blades of said stators and rotors reducing the rotational speed of said shaft being disposed at the same angles to a plane, normal to the axis of said turbodrill shaft.

3. The turbodrill as claimed in claim 1, in which the blades of said stators and rotors, reducing the rotational speed of said shaft, are of variable height increasing proportionally to a radial distance of a blade section from the axis of said turbodrill shaft.

4. The turbodrill as claimed in claim 1, which the blades of said stators and rotors, reducing the rotational speed of said shaft, are of a constant height.

5. The turbodrill as claimed in claim 1, in which the blades of said stators and rotors, reducing the rotational speed of said shaft, are of a streamlined shape, defined by mold curves, and their middle lines being disposed at the same angle to a plane normal to the axis of said turbodrill shaft.

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