(54) Title: THE METHOD AND APPARATUS FOR PREVENTING THE FRICTION INDUCED ROTATION OF NON-ROTATING STABILIZERS

(57) Abstract: A method and apparatus prevents rotation of a stabilizer which rotates due to friction-induced forces of bearing seals disposed in a shaft. The apparatus includes a stabilizer having a ridged edge where the stabilizer is mounted to a drilling tool using a deformable member. The deformable member assists in creating friction of the stabilizer edge against a drilled bore hole wall and thereby reduces friction-induced rotation.
The Method and Apparatus for Preventing the Friction Induced Rotation of Non-Rotating Stabilizers

FIELD OF THE INVENTION

This invention provides a novel method and apparatus for preventing the friction induced rotation of non-rotating stabilizers which are attached to a drilling tool to control the direction and orientation of drilling. The invention can be utilized to prevent rotation at any combination of hole angle, curvature rate and bit load. The present invention provides a more efficient method of operating rotary steerable directional tools.

BACKGROUND

Most rotary steerable systems utilize non-rotating stabilizers to control the trajectory of the hole. The rotating friction between the non-rotating stabilizer and the shaft that turns the bit on conventional systems causes the non-rotating stabilizer to rotate in a clockwise direction. With conventionally surfaced stabilizers, the procession rate is related to the ratio of the rotational friction force between the shaft and the fixed stabilizer to the axial drag force between the fixed stabilizer and the borehole wall. The frictional rotation rate decreases as the hole angle, curvature rate, and/or bit weight increases. However, rotation rates may become excessive at low hole angles, low curvature rates and/or low bit weights. The worst conditions are most likely to occur at the kick off point in a vertical hole. This problem prevents the use of conventional rotary steerable systems on many directional drilling applications.

For example, Table 1 shows the expected frictional rotation rates for a 12 ft non-rotating stabilizer that includes a conventional smooth surfaced adjustable
stabilizer, a fixed stabilizer and which utilizes low friction sealed bearings between
the shaft and the non-rotating unit.

<table>
<thead>
<tr>
<th>Hole Angle deg.</th>
<th>Curvature Rate deg/100ft</th>
<th>Bit Weight kips</th>
<th>Lateral/axial Slide friction Ratio</th>
<th>Frictional Rotation rate deg/axial ft</th>
</tr>
</thead>
<tbody>
<tr>
<td>30.0</td>
<td>6</td>
<td>25</td>
<td>1</td>
<td>14.5</td>
</tr>
<tr>
<td>30.0</td>
<td>3</td>
<td>25</td>
<td>1</td>
<td>17.1</td>
</tr>
<tr>
<td>50.0</td>
<td>2</td>
<td>25</td>
<td>1</td>
<td>18.4</td>
</tr>
<tr>
<td>0.5</td>
<td>3</td>
<td>25</td>
<td>1</td>
<td>37.4</td>
</tr>
<tr>
<td>0.5</td>
<td>2</td>
<td>25</td>
<td>1</td>
<td>48.1</td>
</tr>
<tr>
<td>0.5</td>
<td>3</td>
<td>10</td>
<td>1</td>
<td>55.5</td>
</tr>
<tr>
<td>0.5</td>
<td>2</td>
<td>10</td>
<td>1</td>
<td>68.6</td>
</tr>
</tbody>
</table>

As suggested by the last column, the adjustable stabilizer blades must be
continuously adjusted to compensate for the friction-induced rotation of the non-
rotating unit. Even with a hole angle of 30 degrees the expected frictional rotation
rates would be a problem. With a .5 degree hole angle, the rotation rates are
unacceptable.

SUMMARY OF THE INVENTION

Applicant solves the frictional rotation problem by making the stabilizer
surface act like a drag bit and by increasing the contact forces between the stabilizer
and the bore wall formation. Rotation is prevented whenever the threshold torque
required to rotate the drag bit like contacts exceed the rotational driving torque. The
design is so effective that it can prevent frictional rotation by only applying the design
concept to the fixed stabilizer.

DESCRIPTION OF THE DRAWINGS

Preferred embodiments of the invention will be described below in reference
to the appended drawings wherein,

Fig. 1A-1D illustrate the relationship between rotational drive torque, bearing forces and frictional rotation rate as observed by the present inventor;

Fig. 2 illustrates a fixed stabilizer as mounted on a drilling tool according to a preferred embodiment;

Figs. 3A-B illustrate the fixed stabilizer according to a preferred embodiment;

Figs. 4A-B illustrate the forces on a PDC cutter which models drag behavior of the fixed stabilizer according to the present invention.

DESCRIPTION OF PREFERRED EMBODIMENT

Referring to the accompanying Figures, a preferred embodiment of the invention is described as follows.

Referring to Figs. 1A-1D, the present inventor determined that the mechanics of frictional rotation are defined generally by the following equations:

\[ RDT = \frac{bd \cdot ff \cdot SBF}{2} + 2 \cdot st \]  

(1)

Where:

- \( RDT \) = Rotational driving torque  
- \( bd \) = Shaft bearing diameter  
- \( ff \) = Bearing friction factor  
- \( SBF \) = Sum of all the bearing forces  
- \( st \) = Rotating seal torque  

\( \text{inlbs.} \)  
\( \text{in.} \)  
\( ^\circ \)  
\( \text{lbs.} \)  
\( \text{inlbs} \)

The present invention is specifically applicable to drilling tools including two to four bearings. However, the invention may be applied to a drilling tool with a different number of bearings as long as the summation of the lateral forces on the
bearings (SBF) is taken into account. The rotational driving torque comes from the frictional torque in the Karsi seals and the lateral contact forces in the bearings between the shaft and the (non-adjustable stabilizer) (NAS). The resisting forces are generated by the lateral contact forces between the stabilizers and the hole. Whenever the resisting forces are larger than the frictional forces rotation is prevented.

Fig. 2 illustrates the placement of fixed stabilizer blades 1, 1' on a non-rotating stabilizer according to the present invention. Reference number 2 corresponds to an adjustable stabilizer to position the drilling tool in the bore hole.

The present inventor noted that the calculated cases for Table 1 (conventionally surfaced stabilizer) assumed that the axial sliding friction factor and the rotating friction factor were equal. If the sliding surface of the stabilizer blade were modified to increase the rotating friction factor, the rotation rate would be reduced.

Table 2 shows the effect of utilizing a blade surface that provides a rotational friction factor that is 3 times the axial sliding friction factor. The desired effect is enhanced by aligning the edges of the ridges parallel to the axis of the bore hole and making the ridges sharp.
This improvement makes the frictional rotational rates acceptable in 30 degree holes but still presents a significant problem in .5 degree holes, especially at reduced bit weight and curvature rate.

In the present invention, the contact surface of the stabilizer blade is modified to inhibit lateral movement. The preferred modification places axial ridges on the surface of fixed stabilizer blades. The lateral forces on the stabilizer push the ridges into the bore wall, thereby preventing lateral rotation of the drilling assembly whenever the resisting shear forces in the formation wall exceed the rotational friction force.

Referring to Fig. 3A, each stabilizer fin has 6 sharp drag bit shaped cutters 3a-3f. The cutters are equally spaced from the tool center. The cutters are curved along the axial direction. Under low loads only a single cutter will contact the wall of the hole. Referring to Fig. 3B, the stabilizer fins are supported on load springs. The allowable radial travel is set to provide an under gauge diameter (relative to the bore hole) when the blades are fully collapsed and an over gauge diameter when fully extended. The trailing edge will be 30° (angle A) below the tangential surface.
The cutters act like polycrystalline diamond compact (PDC) cutters on a PDC bit. The rotational mechanics of this design can be modeled using technology developed for the drill bit industry. Figs. 4A-4B illustrate known configurations for PDC cutters. An excellent source of useful information was published by Glowka of Sandia Nat'l Labs in the Society of Petroleum Engineers Journal of Petroleum Technology in August 1989 pgs 797-849.

Glowka used a variety of single PDC cutters to measure the mechanics of drilling in three kinds of rock. The test included flat faced cutters as well as sharp edge cutters. Most of the tests measured the axial cutter loads and the penetration forces as a function of the depth of cut.

They developed the following empirical relationships for cutting dry rock at the surface.

\[
\begin{align*}
\text{FDB} &= \text{FA} \cdot (0.90 + 2.2 \cdot D) \\
\text{FDT} &= \text{FA} \cdot (0.65 - 0.58 \cdot D) \\
\text{FDS} &= \text{FA} \cdot (0.63 + 0.88 \cdot D)
\end{align*}
\]

where

- \( \text{FDB} \) = Cutter drag force in Berea Sandstone (lbs.)
- \( \text{FDT} \) = Cutter drag force in Tennessee Marble (lbs.)
- \( \text{FDS} \) = Cutter drag force in Sierra White Granite (lbs.)
- \( \text{FA} \) = Downward force on the dull cutter (lbs.)
- \( D \) = Depth of cut (in.)

The tests that used sharp cutters required larger cutter drag forces than observed with the dull cutter tests. They also ran tests with drilling fluid. These tests showed that the drilling fluid acted as a lubricant and reduced the cutter drag forces by 10 percent.
The inventor notes that the cutter drag forces are greater in softer rocks. Using the performance in granite should underestimate the cutter drag forces in all oilfield formations. Combining all these factors gives the following safe estimate for the rotational resistance of the stabilizer design of the invention:

\[
RT = FS(0.63)(0.9)\frac{sd}{2}
\]  
(2)

Where:

- \(RT\) = Rotation resisting torque \(\text{inlbs.}\)
- \(FS\) = Total lateral loads on all of the stabilizer blades \(\text{lbs.}\)
- \(sd\) = stabilizer diameter \(\text{in.}\)

By using a bit cutter like contact, the mechanics are changed from a two dimensional sliding problem to establishing a threshold resisting load that prevents any rotation whenever it exceeds the driving torque \(RDT\) defined above in equation (1).

Referring back to Fig 3B, the fixed stabilizer adjusts from 8 3/8 in. outer diameter to 8 5/8 in. outer diameter for drilling 8 1/2 in. holes. The fixed stabilizer load springs apply 50 to 60 pound loads across the travel limits. The shaft uses three low friction bearings. Both the cutter like contacts and the spring assisted fixed stabilizer blades are needed to completely eliminate frictional rotation. The rotational torque in this situation is modified from equation (2) above to further include the sum of the spring forces.

\[
RT_s = \frac{sd}{2} (FS + \Sigma Fspring)(0.63)(0.9).
\]  
(3)

Where:

- \(RT_s\) = Rotation resisting torque with spring loaded contacts \(\text{inlbs.}\)
FS = Total lateral loads on all of the stabilizer blades lbs.

sd = stabilizer diameter in.

ΣFspring = Sum of all of the spring forces lbs.

Table 3 shows the expected performance of using cutter type contacts without spring loaded stabilizer blades.

<table>
<thead>
<tr>
<th>Hole Angle deg.</th>
<th>Curvature Rate deg/100ft</th>
<th>Bit Weight kips</th>
<th>Rotational Driving Torque Inlbs</th>
<th>Rotational Resisting Torque Inlbs</th>
<th>Avoidance Design Factor</th>
</tr>
</thead>
<tbody>
<tr>
<td>30.0</td>
<td>6</td>
<td>25</td>
<td>145</td>
<td>1349</td>
<td>9.3</td>
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<tr>
<td>30.0</td>
<td>3</td>
<td>25</td>
<td>131</td>
<td>902</td>
<td>6.9</td>
</tr>
<tr>
<td>30.0</td>
<td>2</td>
<td>25</td>
<td>126</td>
<td>750</td>
<td>5.9</td>
</tr>
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<td>0.5</td>
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<td>375</td>
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<td>227</td>
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<td>25</td>
<td>120</td>
<td>73</td>
<td>0.6</td>
</tr>
<tr>
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<td>128</td>
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<td>0.2</td>
</tr>
<tr>
<td>0.5</td>
<td>2</td>
<td>10</td>
<td>124</td>
<td>23</td>
<td>0.2</td>
</tr>
<tr>
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<td>0.9</td>
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<tr>
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<td>5</td>
<td>120</td>
<td>37</td>
<td>0.3</td>
</tr>
</tbody>
</table>

This design easily prevents rotation at both 30 degree and .5 degree hole with bit weights of 25,000 lbs and curvature rates of 2deg/100ft or more. However, the last five cases in the table would not stop frictional rotation. As shown in Table 4, adding 50 to 60 lb. springs to each of the stabilizer blades completely eliminates any chance of frictional rotation. Five blades are contemplated for the preferred embodiment. At a minimum, the present invention can minimize rotation to 1-3° of rotation per foot drilled, even for a verticle hole.
Table 4
NON-ROTATING STABILIZER WITH ROTATION
AVOIDANCE CUTTERS AND 50/60 POUND SPRINGS
ON THE FIVE FIXED STABILIZER BLADES

<table>
<thead>
<tr>
<th>Angle</th>
<th>Curvature Rate</th>
<th>Bit Weight</th>
<th>Rotational Driving Torque</th>
<th>Rotational Resisting Torque</th>
<th>Avoidance Design Torque</th>
<th>Design Factor</th>
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<tr>
<td>deg.</td>
<td>deg/100ft</td>
<td>kips</td>
<td>Inlbs</td>
<td>Inlbs</td>
<td></td>
<td></td>
</tr>
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<td>25</td>
<td>131</td>
<td>1567</td>
<td>11.9</td>
<td></td>
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<td>699</td>
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<td></td>
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<td>10</td>
<td>124</td>
<td>691</td>
<td>5.6</td>
<td></td>
</tr>
<tr>
<td>0.5</td>
<td>1</td>
<td>10</td>
<td>120</td>
<td>679</td>
<td>5.7</td>
<td></td>
</tr>
<tr>
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<td>5</td>
<td>129</td>
<td>779</td>
<td>6.0</td>
<td></td>
</tr>
<tr>
<td>0.5</td>
<td>2</td>
<td>5</td>
<td>125</td>
<td>740</td>
<td>5.9</td>
<td></td>
</tr>
<tr>
<td>0.5</td>
<td>1</td>
<td>5</td>
<td>120</td>
<td>700</td>
<td>5.8</td>
<td></td>
</tr>
</tbody>
</table>

The combination of cutter like contacts and spring loaded blades provides a rotational resistance force that is at least 5 times greater than the frictional driving force under all conditions.

While a preferred embodiment has been described above, one skilled in the art would recognize that the invention can be modified and still fall within the scope of the appended claims. For instance, the load spring can be replaced by alternative mechanism to exert a lateral force against the wall such as a hydraulic system.
I CLAIM:

1. A stabilizer assembly mountable on a surface of a drilling tool having a longitudinal shaft, said stabilizer comprising:
   a plurality of stabilizer blades, each blade having multiple ridges and being independently mounted on the drilling tool.

2. The stabilizer assembly according to claim 1, further comprising a displacement member disposed between each stabilizer blade and the drilling tool, said displacement member deformable to provide a minimum radius and a maximum radius, different from the minimum radius, from a center of the longitudinal shaft to an outer extremity of at least one of the multiple ridges.

3. The stabilizer assembly according to claim 2, wherein when the displacement member provides the minimum radius, at least one of the multiple ridges are exposed.

4. The stabilizer assembly according to claim 2, wherein the displacement member comprises a load spring.

5. The stabilizer assembly according to claim 4, wherein the load spring provides a rotational resistance RR sufficient to counterbalance frictional rotation RDT according to an expression:

   \[ RDT = \frac{bd \cdot ff \cdot SBF}{2} + 2 \cdot st \]

where:

   \( RDT \) = rotational driving torque  \( \text{inlbs.} \)

   \( bd \) = shaft bearing diameter  \( \text{in.} \)

   \( ff \) = bearing friction factor  \( * \)

   \( SBF \) = sum of all the bearing forces  \( \text{lbs.} \)

   \( st \) = rotating seal torque  \( \text{inlbs.} \)
6. The stabilizer assembly according to claim 4, wherein the load spring has a spring load in a range of 50-60lbs across a travel distance of 1/8 inch.

7. A stabilizer assembly mountable on a surface of a drilling tool having a longitudinal shaft, said stabilizer comprising:
   at least a first stabilizer blade having multiple ridges and movably mounted on the drilling tool.

8. The stabilizer assembly according to claim 7, wherein the stabilizer assembly includes a second stabilizer blade having multiple ridges, said second stabilizer blade independently and movably mounted onto the drilling tool from the first stabilizer blade.

9. The stabilizer assembly according to claim 8, wherein each of the first and second stabilizer blades is attached to the drilling tool by a displacement member disposed between the stabilizer blade and the drilling tool, said displacement member deformable to provide a minimum radius and a maximum radius from a center of the longitudinal shaft to an outer extremity of at least one of the multiple ridges.

10. The stabilizer assembly according to claim 9, wherein when the displacement member provides the minimum radius, at least one of the multiple ridges are exposed.

11. The stabilizer assembly according to claim 9, wherein the displacement member comprises a load spring.

12. The stabilizer according to claim 11, wherein the load spring provides a rotational resistance $RR$ sufficient to counterbalance frictional rotation $FR$ according to an expression: This equation must be replaced by equation 3

\[ RTs = \frac{sd}{2} (FS + \Sigma Fspring)(0.63)(0.9) \]

where:
RTs = rotation resisting torque with spring loaded contacts  in lbs.

FS = total lateral loads on all of the stabilizer blades  lbs.

sd = stabilizer diameter  in.

\( \Sigma F_{spring} = \text{sum of all of the spring forces} \)  lbs

13. The stabilizer according to claim 11, wherein the load spring has a spring load in a range of 50-60 lbs across a travel distance of 1/8 inch.

14. A method of forming a stabilizer assembly for a drilling tool including a plural number of bearings disposed in a longitudinal shaft, said method comprising:

- determining a lateral force of each plural number of bearings SBF;
- determining a shaft bearing diameter bd;
- determining a frictional seal torque st;
- determining a bearing friction factor ff;

and determining a resistance torque RTs for the stabilizer sufficient to counterbalance a frictional rotation RDT according to an expression:

\[
RTs = \frac{sd}{2} (FS + \Sigma F_{spring})(0.63)(0.9)
\]

where:

RTs = rotation resisting torque with spring loaded contacts  in lbs.

FS = total lateral loads on all of the stabilizer blades  lbs.

sd = stabilizer diameter  in.

\( \Sigma F_{spring} = \text{sum of all of the spring forces} \)  lbs.
FIG. 1A

FIG. 1B
FIG. 2

ADJUSTABLE STABILIZER FORCE

FIXED STABILIZER FORCES
FIG. 3A

PRECESSION FORCE

4.188 in. RADIUS

3a
3f

LOCK

FIG. 3B

4 in.

.125 in. TRAVEL

2.00 in. RADIUS

4.313 in. EXPANDED OD

50-60 TOTAL SPRING FORCE
**INTERNATIONAL SEARCH REPORT**

**A. CLASSIFICATION OF SUBJECT MATTER**

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<th>IPC(7)</th>
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According to International Patent Classification (IPC) or to both national classification and IPC

**B. FIELDS SEARCHED**

Minimum documentation searched (classification system followed by classification symbols)

U.S.: 166/241.1-241.7, 117.5, 117.6, 117.7

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

Please See Continuation Sheet

**C. DOCUMENTS CONSIDERED TO BE RELEVANT**

<table>
<thead>
<tr>
<th>Category</th>
<th>Citation of document, with indication, where appropriate, of the relevant passages</th>
<th>Relevant to claim No.</th>
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<td>US 5,467,834 A (HUGHES ET AL) 21 November 1995, see entire document.</td>
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<td>US 4,729,438 A (WALKER ET AL) 8 March 1988</td>
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* Further documents are listed in the continuation of Box C.

* Additional categories of cited documents:

  - **T** document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
  - **X** document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
  - **Y** document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
  - **&** document member of the same patent family

Date of the actual completion of the international search: 08 June 2005 (08.06.2005)

Date of mailing of the international search report: 06 JUL 2005

Authorized officer: KENNETH J. DORNER

Telephone No. 571-272-3600

Form PCT/ISA/210 (second sheet) (January 2004)
Continuation of B. FIELDS SEARCHED Item 3:
USPT, USOCR, EPO, JPO
non-rad3. spring, teeth, ridge, blade, fin, stabilizer, centralizer