Inversion Algorithm about Pre-tightening Force on Bearing Groups of Turbodrill

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Abstract

Turbodrill was used extensively for its many excellent properties. But there was a deadly weakness—its working life in the downhole was too short (100 to 250h). The main cause of turbodrill failure is the breakdown of thrust bearings. Turbodrill took large complete alternate load, and its unique structure caused the pre-tightening force of thrust bearings was difficult to determine. The trial and error method, and the experience method doesn’t work well. The inversion algorithm was taken in the essay, through which deformation of track and roller was gotten inversely based on pre-stress deformation of inner and outer rings of bearings, and was checked if it agrees with the Hertz theory to verify the probability of pre-tightening force, so as to obtain the suitable pre-tightening force. This essay made an instruction for the determination of pre-tightening force during turbodrill installation with inversion algorithm.

Keywords: Turbodrill, Bearing Groups, Pre-tightening Force, Inversion Algorithm, Hertz Contact Deformation Theory

1. INTRODUCTION

Turbodrill was the most dynamic downhole motor at present. Turbodrill was playing an increasingly important role in modern drilling operation and received universal attention from the countries. Although turbodrill had wide application and good progress, it had its peculiar weakness for its short working life in downhole. Now the working life of turbodrill in downhole was generally between 100 and 250 hours (Fleckenstein et al., 2016). It is difficult to determine the appropriate pre-tightening force, and then inappropriate pre-tightening force was the main cause of bearing groups failure of turbodrill.

Thrust bearings of turbodrill had very complicated working condition. It was lubricated badly (by drilling fluid, which had big granule); And the bearings also took large complicate alternating load ranging from 0 to 300KN (Beshoory, 2001; Simonyants, 2016). The special structure of turbodrill makes it’s difficult to inspect the right pre-tightening force.

The thrust bearing used in the turbodrill has a special type of angular contact thrust ball bearings. Due to the limited of shell diameter of turbodrill, the thrust ball bearing usually is multi-union structure, in order to enhance the carrying capacity. The structure of multi-union thrust ball bearing is shown in Figure 1.

Every row of multi-union thrust ball bearing is consists of the inner ring, outer ring and balls. 6~12 rows formed a multi-union thrust ball bearing. Unlike ordinary thrust ball bearings, rolling elements of each row contact with two inner rings and two outer rings at the same time. This is the so-called four-point contact ball bearings. During working hours, in order through the working medium and maintain bearing lubrication, there is approximately 2mm-gap between the inner ring and the outer ring. Bearings should be installed that both ends of the bearing inner and outer rings were placed in the same plane. The inner rings and outer rings were axial pre-tighten respectively in installation. The mechanical model of multi-union thrust ball bearing is as shown in Figure 2.
Conventional pre-tightening methods for turbodrill are extension & compression theory method and experience method. Extension & compression theory method was that: for decreasing influence of variables in design, design was based on that the sum of compression of bearings and elongation of axis which had the same length as bearings was between 0.5 to 1mm under the theoretical pre-tighten force F0. Theoretical pre-tighten force was just an approximate calculation. When turbodrill was pre-tightened, the inner ring was pre-tightened first and then the outer ring of the complicate structure of turbodrill. Bear groups had been set into the shell of turbodrill when they are pre-tightened, so deformation of balls and rings of bearings cannot be measured. Experience method was experience pre-tighten force value from long installation history of turbodrill. For example, pre-tighten force for top and bottom joint of turbodrill with diameter $\Phi 240$ had the rules called “top 7, bottom 8”, which means pressure for makeup of turbodrill for top joint need to achieve $7\text{MPa}$, and the one for lower joint need to achieve $8\text{MPa}$. But so-called rules “top 7, bottom 8” was not correct because different turbodrill had different shell thickness and ring diameter. In reality, some turbodrill often broke down before it is used because inner and outer rings of bearings collapse for higher pre-tighten force (Yu et al., 2015). As shown in Figure 3, inner and outer rings of bearings collapse for higher pre-tighten force. Some turbodrill may come loose and fell off because of lower pre-tighten force, or failed earlier for uneven wear of bearings inners, outer rings and balls which were also caused by lower pre-tighten force.
So, It’s important to make a study on the effective method of measuring pre-tighten force for prolonging bearing group life.

2. THINKING LINE OF INVERSION ALGORITHM

2.1 Process of inversion algorithm

There was no way to measure deformation of bearing inner, outer rings and balls directly for bear group had been put into the shell of turbodrill when they were pre-tightened, so it was hard to check if its compression was in the range (Liu,2012; Liu et al., 2015; Xiao et al., 2015; Wang et al., 2015). Get the diameter of ball which could be put in deformed balls rail through pre-tightening without balls, and checked the deformation of bearing groups inner, outer rings and balls with Hertz contact deformation theory.

The flow diagram was shown in Figure 5.

![Flow Diagram of Inversion Algorithm](image)

2.2 Calculation of structure parameter and pre-tighten force of inner ring

Study the inversion algorithm with the instance of φ240 turbodrill.

Run calculation as left branch in flow diagram, turbodrill parameter was described as: diameter of center position of rail \(D_0=164\)mm, balls number \(Z=20\), axis diameter \(d_1=110\)mm, balls diameter \(D_g=25\)mm, shell thickness \(S=17\)mm, \(d_2=210\)mm.
Figure 6 Bearing Size in the Geometric Meaning

Take an instance that compression of the bearing inner ring is 0.8mm. Calculate the compression of the turbodrill axis and stator.

Elongation under pre-tighten force \( F_0 \) can be described as:

\[
l_1 = \frac{F_0 \times l}{E \times A_i} = \frac{F_0}{E \pi} \times \frac{l}{d^2 / 4}
\]  

(1)

Since the cross section area of stator varied along the axis, as shown in Figure 7 and Figure 8, the deformation could not direct calculate by formula 1, and the deformation should be gotten through integration.

Figure 7 Cross Section Area of Inner Ring Changed along Axis

Deformation of single bearing can be described as:

\[
l_2 = 2n_2 + (n + 1)l_{2j}
\]  

(2)

Where: \( l_{2n} \) is deformation of changeable cross section, \( l_{2j} \) is deformation of constant cross section of lower part of rail.

And they also satisfied the following relationship:
\[ l_1 + l_2 = 0.8 \] **(3)**

After calculation, theoretical pre-tighten force was:

\[ F_0 = 1.6493 \times 10^6 \text{N} \]
\[ P_n = 302.8153 \text{Mpa} \]

Where \( P_n \) should be the pressure exerted on the inner ring in the ansys model.

### 2.3 Calculation of outer ring theoretical pre-tighten force

Run the calculation with the method in chapter 2.2 “Calculation of structure parameter and pre-tighten force of inner ring”, outer ring theoretical pre-tighten force turned out to be:

\[ F_2 = 1.9342 \times 10^6 \text{N} \]
\[ p_w = 308.3343 \text{Mpa} \]

Where: \( p_w \) should be the pressure exerted on outer ring in the ansys model.

### 2.4 Stress analysis of bearing inner, outer rings in ANSYS

Like other finite element analysis steps, modeling and mesh are needed (ANSYS, 2016). As show in Figure 9.

![Figure 9 Set Model of One Quarter of Cross Section of Bearing Inner Ring and Gird](image)

Model deformation under pre-tighten force as showed in Figure 10.

![Figure 10 Original Model and Deformation Displacement Figure under pre-tighten Force 1.6493×10^6 N](image)
2.5 Data extraction

Extract the original coordinates value \( X \) and \( Y \) of 11 nodes on the inner raceway curve, also the displacement \( U_x \) and \( U_y \). Process those data through the Notepad, WORD editor and Excel sheet. Extracted data are as showed in the table 1.

### Table 1 Detailed Values of 11 Nodes (Unit: mm)

<table>
<thead>
<tr>
<th>Node</th>
<th>( X )</th>
<th>( Y )</th>
<th>( U_x )</th>
<th>( U_y )</th>
<th>( x1 )</th>
<th>( y1 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>81</td>
<td>7.866356698</td>
<td>0.0094647</td>
<td>0.00012314</td>
<td>81.0094647</td>
<td>7.86647984</td>
</tr>
<tr>
<td>2</td>
<td>79.15878006</td>
<td>8.302430706</td>
<td>0.0093706</td>
<td>-0.00038538</td>
<td>79.16815066</td>
<td>8.30204533</td>
</tr>
<tr>
<td>3</td>
<td>77.39002033</td>
<td>8.974547058</td>
<td>0.0090987</td>
<td>-0.0010136</td>
<td>77.39911903</td>
<td>8.97353346</td>
</tr>
<tr>
<td>4</td>
<td>75.7238402</td>
<td>9.871260591</td>
<td>0.0086452</td>
<td>-0.0019343</td>
<td>75.7324854</td>
<td>9.86932629</td>
</tr>
<tr>
<td>5</td>
<td>74.18861229</td>
<td>10.97730158</td>
<td>0.0080427</td>
<td>-0.0032757</td>
<td>74.19665499</td>
<td>10.9740259</td>
</tr>
<tr>
<td>6</td>
<td>72.8104793</td>
<td>12.27383575</td>
<td>0.0073407</td>
<td>-0.0050927</td>
<td>72.81782</td>
<td>12.2687431</td>
</tr>
<tr>
<td>7</td>
<td>71.61290884</td>
<td>13.73878503</td>
<td>0.0065758</td>
<td>-0.0073497</td>
<td>71.61948464</td>
<td>13.7314353</td>
</tr>
<tr>
<td>8</td>
<td>70.61629377</td>
<td>15.34720345</td>
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<td>70.62205267</td>
<td>15.3372518</td>
</tr>
<tr>
<td>9</td>
<td>69.83760501</td>
<td>17.07170198</td>
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<td>-0.012751</td>
<td>69.84242011</td>
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</tr>
<tr>
<td>10</td>
<td>69.29010249</td>
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<td>69.29369049</td>
<td>18.8673919</td>
</tr>
<tr>
<td>11</td>
<td>68.98310938</td>
<td>20.75</td>
<td>0.0014145</td>
<td>-0.01826</td>
<td>68.98452388</td>
<td>20.73174</td>
</tr>
</tbody>
</table>

Note:
The first 5 columns of data can extract from the finite element analysis software ANSYS. The last 2 columns of data were processed by excel sheet. Each column of data represents meaning as follows.

- \( X, Y \)——Original coordinates value without pre-tighten force
- \( U_x, U_y \)——Displacement of the nodes on the raceway under pre-tighten force
- \( x1, y1 \)——coordinates value under pre-tighten force (CVUPF)

### 2.6 Curves fitting

Fit into a curve by the 11 nodes coordinates of an inner ring. The fitting process is performed by MATLAB soft.

The inner ring is a part of a circle. Connect 11 nodes in line. The line is close to the original arc, so the fitting model selects the quadratic curve.

The fitting function of MATLAB is:

\[
polyfit(x1, y1, 2)
\]

Where:

- \( x1, y1 \)——the row vector of deformation under pre-tighten of 11 nodes. Such as \( x1 \) and \( y1 \) in table 1.

The result is:

\[
y_a = 0.1077 \times x_a^2 -17.0731 \times x_a + 684.6983
\]

(4)

The range of the independent variable is from 81.0094647 to 68.98452388. The unit of independent variables and variable is millimeter.

Plot the fitting curve according to the fit formula (4). As shown in figure 11.
Notes:
Marked “+” in blue dotted line is actual coordination of inner ring under pre-tighten force.
Marked “*” in red solid line is fitting curve coordination of inner ring under pre-tighten force.

How to evaluate the fitting function? It is a good method to compare the coincidence degree between the exact curve and the fitting curve, also it is indicated in Figure 11. The indicator of measure the fit uses R2. R2 is goodness of fit. The closer the value of R2 is to 1, the better the fitting degree of the fitting line to the observed value.

In order to facilitate the calculation, compile a program, the code is as follows.

```matlab
x1=[…]; % Row vector of X – coordination of the nodes under pre-tighten force
y1=[…]; % Row vector of Y – coordination of the nodes under pre-tighten force
[p, s]=polyfit(x1,y1,2); % Fitting curve with coordination under pre-tighten force
y_poly=polyconf(p, x1, s); % Find the predicted value of the fitting curve at x variable
y=mean(y1); % Averaging
resquare=sum((y_poly - y).^2)./sum((y1-y).^2) %The calculation of curve fitting degree
```

Result:

```
resquare=0.9801
```

The results were satisfactory.

Get the 11 nodes coordinate at corresponding positions on outer ring.

Fit curve with the 11 nodes coordinate values after deformation.
According to the structure of four-point contacted thrust ball bearing, there are 4 curves around a ball. The maximum allowable ball diameter is the diameter of the largest enveloping circle of 4 curves. When the inner and outer rings were pre-tightened without balls, all the inner rings had the same deformation, and so did the outer rings. According to Figure 12, two rows of inner rings in contact and two rows of outer rings in contact were symmetric about contact surface A-B. So only deformation of one group of inner, outer rings need to be considered. Deformed arcs of inner and outer rings formed a circle. Circle center coordinate $Y_0$ was on the contact surface. Circle center coordinate $X_0$ after deformed could be obtained by moving the original circle center a small distance along radial direction since the inner and outer rings had different deformation.

For the symmetry about plane AB, the maximum circle diameter of the envelope curve can be obtained by 2 fit curves. That is, diameter of balls could be fitted in between 2 curves was shown in Figure 13.

Solved the equations, we could get the rail center coordinate $X_0$, and $D' = 164.00187\text{mm}$, which was $0.00187\text{mm}$ larger than the original one $164\text{mm}$. The maximum diameter of balls could be placed in the deformed rails was $r = 12.1912\text{mm}$. The pre-designed balls diameter was $25\text{mm}$, so the radius was $12.5\text{mm}$ which was larger than $12.1912\text{mm}$. Calculated its Hertz contact stress, compared it with the allowable value.

2.7 Calculation of maximum Hertz contact stress on the base of deformation simulation experiment

According to reference(Cheng, 1999; Cheng, 1993), factors in contact ellipse were:

$$A = \frac{1}{2} \left( \frac{1}{R_1} - \frac{1}{R_2} \right), \quad B = \frac{1}{2R_1}$$

(3)

Where $R_1 < R_2$, $R_1$ was balls radius, $R_2$ was rail radius.

The maximum contact compression stress could be expressed as:
\[ \sigma_{\text{max}} = \alpha \sqrt{p E (R_2 - R_1)} \]

Run calculation with Hertz contact theory, we could get the total deformation of elastic body \( \xi = 0.3277 \) and the stress \( q_0 = 3726.907 \text{ Mpa} \).

Maximum deformation from the simulation experiment was 12.5 - 12.1913 = 0.3087 < \xi. According to reference[10], the maximum allowable bearing stress \( [\sigma] = 1.3 \sigma = 1.3 \times 3800 = 4940 \text{ Mpa} > q_0 \).

The results also mean the change of clearance between bearing race and balls was in millimeter scale, which was far larger than the compensation provided by tolerance of balls and rail; the bearing pre-tighten force was far larger than its working load, which was 5~15 times of working load.

3. CONCLUSION

Inversion algorithm provided evidence for the settlement of pre-tighten force for installation of turbodrill bearing groups, and avoid bearing group collapse or fall-off caused by unsuitable pre-tighten according to extension & compression theory and experience theory. It would extend the life of bearing groups, and improved the life of turbodrill also, and improved the efficiency and saved the cost.

The inversion algorithm method of pre-tightening force solved the deformation cannot be measured after being mounted. This method has successfully been used by the Research Institute of Downhole Tools and obtained satisfactory results.

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