Finite Element Analysis on the Transmission Shaft of Downhole Electric Drill

Suling Wang [a], Yongjun Hou [b]

College of Mechatronic Engineering, Southwest Petroleum University, Chengdu, Sichuan 610500, China

[a]1206837997@qq.com, [b]hyj2643446@126.com

Abstract

As the development of drilling technology, drilling equipment applied in the drilling process are faced with more challenges to satisfy higher drilling efficiency. The underground drill is more efficient than other types of the drill, because it is hardly affected by those factors, like its operating depth, location and drilling mud, that the underground electric drilling tools deserved high attention in the oil exploration industry. While the drive shaft assembly is located in the bottom of the electric drill, and the structure of the transmission shaft is hollow whose load is relatively complex, it is necessary for us to carry out the finite element analysis on the drive shaft, which plays important role in downhole electric drill structure, to prevent the failure of the drive shaft. Firstly, the maximum stress and maximum deformation of the drive shaft were obtained by the static analysis. Secondly, the fatigue life was gained by the fatigue life analysis. Finally, the modal analysis was adopted to acquire the inherent vibration modes and natural frequency. The analysis results showed that the drive shaft of the designed electric drilling tool can operate safely and reliably under rated conditions.

Keywords

Electric Drill; Transmission Shaft; Static Analysis; Fatigue Analysis; Modal Analysis.

1. Introduction

Transmission shaft assembly(also called the output main shaft or main shaft)is located between planetary gear reducer and drill bit, which are belong to underground electric drill. The function of transmission shaft is making bit pressure reaction to the shell of electric drill by thrust bearing and avoid the rotor of the motor supporting the axial load. Then it can transmit rotation torque of motor to drill bit, in order to drive drill bit crushing wall rock. So transmission shaft is the link for underground electric drill to finish drilling operation. electric drill transmission shaft assembly is composed of thrust bearing series, alignment bearing, transmission shaft, shell, coupling and so on. Transmission shaft upper end through the coupling connects with reducer output shaft to input power. And the bottom through the thread connects with drill bits to improve rotation speed and torque crushing wall rock. The axial load of transmission shaft under different conditions is different [1, 2].

To understand the transmission shaft at work is under the action of drilling pressure, torque, etc. which areas are prone to damage, so making the static analysis under the maximum load effect to get the parts generating maximum stress and strain. It is also in order to facilitate the subsequent prevention or improvement; Failure is not caused by a load, but a continuous and gradual process of accumulation, If in the event of elastic or plastic bending deformation, every rotation will generate produces a stress, the stress concentration area will gradually formed micro cracks. When the shaft every rotation micro cracks would spread a bit, if this is not found in time, will eventually lead to failure. Therefore, mechanical fatigue life analysis was carried out on the transmission shaft is important; When transmission shaft works, it will be affected by dynamic load, the vibration of transmission shaft assembly parts is always closely connected with the transmission shaft vibration, the priority is to clear the dynamic performance of the transmission shaft assembly, particularly is the dynamic characteristics of the shaft itself, namely the inherent vibration mode and natural frequency of the transmission shaft, thus modal analysis was carried out on the drive shaft.

2. The static analysis of transmission shaft

We should know the bottom composition of the motor before the static analysis, as shown in Fig.1. The underground electric drill by the submersible motor through reducer to transmit the power to the output shaft, drive the drill cutting rock, when the power plant into the hole bottom direct drive rotary cutting bit. In the working process, the drive shaft in addition to transfer torque and rotational speed of the motor, but also bear axial force and radial force, axial force generated by the drilling pressure, the radial force is at the bottom of shaft lining [3]. In the normal working range, the maximum influence on the transmission shaft is the torque, axial force and lateral force secondly. The study found that: The fracture of the minor diameter of the drive shaft is mainly due to the torque transmitted power. Moreover, the torque is easier to get than the lateral force and axial force in the actual condition, so the torque analysis is available when the drive shaft is analyzed. In order to calculate simply, the coupling with the screw-thread of the drive shaft as a whole, the bottom of the drive shaft is connected with the drill bit by screw, because the influence of the screw-thread for the overall analysis can be ignored, so the thread surface is simplified into a cone. Then the simplified mechanical model of the transmission shaft, as shown in Fig.2. After the model is simplified, the calculation model is assumed as follows:

1. Assuming that the shaft is fixed constant linear system, and is the line elastomer (i.e. the corresponding linear relationship between elastic deformation and the load in the loading and unloading engineering, remove the load deformation of the elastic body restores its original size and shape when remove all load)

2. Assuming that the shaft the selected material is isotropic, density of uniform distribution, and continuous.

3. Assuming that the deformation and displacement are all small, that is, the case of small deformation [4].

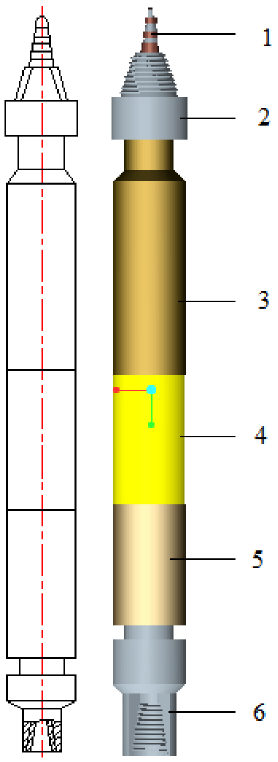

Fig. 1 Structure schematic diagrams of BHA

1. Cable contact rod; 2.upper joint; 3.electric motors; 4.reducer; 5.spindle; 6.bit joint

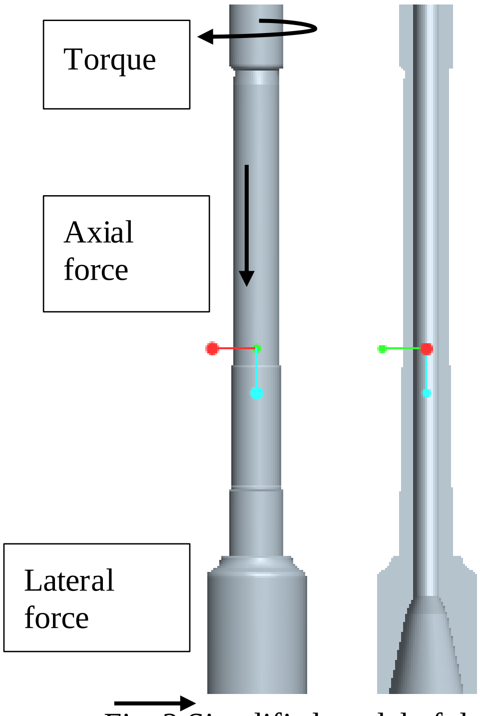

Fig. 2 Simplified model of drive shaft

The material of the drive shaft is selected as 40CrNiMoA, which is a kind of high quality tempered steel. The comprehensive mechanical properties were significantly improved after quenching and tempering. It have very high impact toughness at low temperature. There are a high fatigue limit and good hardenability, and its notch sensitivity is relatively low during Quenching, and then after low temperature tempering or high temperature tempering. It can be used as parts of larger cross-section, impact load and high strength [5]. The specific performance parameters of the material are as follows:

Tensile strength: $\sigma_b = 980Mpa$;

Yield strength: $\sigma_s = 835MPa$;

Fatigue strength: $\sigma_{-1} = 529MPa$;

Elastic modulus: $E = 209GPa$;

Shear modulus: $G = 81GPa$;

Poisson's ratio: $\mu = 0.3$;

Safety factor: $n_s = 2.2$;

Density: $\rho = 7830kg/m^3$;

Material ofallowable stress: $[\sigma] = \sigma_s / n_s = 379\text{MPa}$.

Because the main stress type of drilling tools is transferring torque, the torsional stiffness and strength were checked, due to the section shape of shaft is too complex to calculate the stiffness, so only the torsional strength is calculated.

$$\tau = \frac{T}{W} \le [\tau] \tag{0}$$

Where, τ is the torsion shear stress (MPa,Pa); T is the maximum torque in work(N m); W is the section modulus in torsion (cm^3,m^3).

$$W = \frac{\pi D^3}{16}\left[1 - \left(\frac{d}{D}\right)^4\right] \tag{1}$$

T=14040$N\ m$; D=85mm=8.5cm; d=46mm=4.6cm; put them into the formula(1)and then: $W = 110.24 cm^3 = 110.24 \times 10^{-6} m^3$, $\tau = 127 \times 10^6 Pa = 127 MPa$

$$[\tau] = 0.5[\sigma] = 0.5\frac{\sigma_s}{2.5} = 0.2\sigma_s = 0.2 \times 835 MPa = 167 MPa \quad (2)$$

The strength of drive shaft on maximum torque is enough due to $\tau < [\tau]$.

The shaft which is 1258mm long, has a maximum diameter of 184mm and weighs 74.6kg. When grid is formed. The number of Mesh units is 88799 and the number of nodes is 134969.

The rated torque of the output shaft is 5400$N\ m$ and braking torque is 14040 $N\ m$ when the electrodrill is under the rating condition.

Static analysis was carried out on the braking torque on the shaft in order to guarantee the safety of the shaft. Maximum torque exerted on the smaller end plane is 14040$N\ m$ and the larger were fixed.Finite element static analysis were carried on and the stress and the displacement change contours were obtained, as shown in Fig.3 and Fig.4.

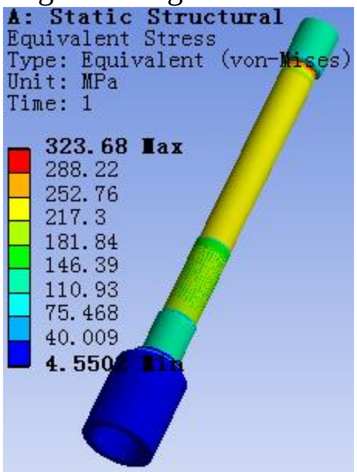

Fig. 3 The static analysis stress change contour of shaft

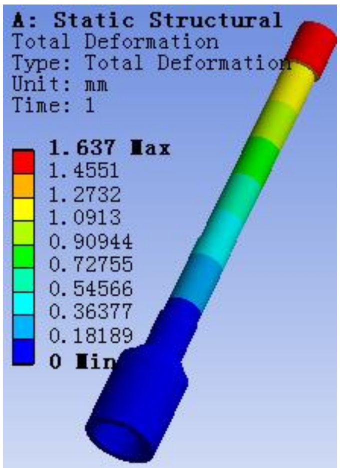

Fig. 4 The static analysis displacement change contour of shaft

In the static analysis above, the drive shaft which has a maximum stress of 323.68 MPa in the root between the drive shaft and the coupling under the maximum torque suggests that the earliest damage occurred in the root. When safety coefficient is 2.5, The allowable stress of the shaft $[\sigma]$=334MPa, so the shaft torsional strength is qualified. The shaft torsional strength also has a mutation when the shaft diameter changes, but the variation is small. the stress in small diameter parts of the shaft has a large value of 220 Mpa which is safe, the variation in the whole is more gentle. The deformation in small diameter parts of the shaft near the coupling are maximum, and the largest deformation is 1.637 *mm* as shown in drive shaft displacement change contours .

3. Fatigue Life Analysis of Driving Shaft

An important component of electrodrills is the drive shaft, which would contribute the drill to increase the power and speed. If the elastic plastic bending deformation has happened, the drive shaft rotated a circle will produce a stress, and the crack will occur in the stress concentration positions. If not discovered soon enough, it will lead invalidation of e drive shaft with the spread of the crack[6].

From the static analysis shows the maximum equivalent stress is 323.68*Mpa*, which less than the transmission shaft material yield strength (835*Mpa*). When the transmission shaft is located in normal working condition, the cloud diagram of the static stress analysis is shown in Fig. 5. The maximum equivalent stress has appeared along the junction of the transmission shaft and the coupling(132.41*Mpa*), which is much less than the yield limit of the material(835*Mpa*). Therefore it can be known that the fatigue type of the transmission shaft is the high-cycle fatigue(the stress fatigue). It is well known that fatigue limit and stress life curve (S-N curves) reflects the indicators of fatigue resistance[7]. The material information can be generated by ANSYS Wrorkbench material data manager, the S-N curve of material 40CrNiMoA is shown in Fig. 6.

After the actual stress history and the distribution functions of stress can be gained by the computer simulation, the random fatigue loads of the shaft can be obtained through the application of circular computations by use rain-flow counting. Fatigue load spectra of the transmission shaft is symmetric circulant Fully Reversed constant-amplitude load, as shown in Fig. 7.

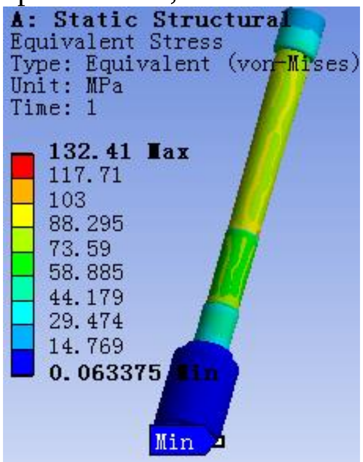

Fig. 5 Drill during normal drilling shaft stress nephogram

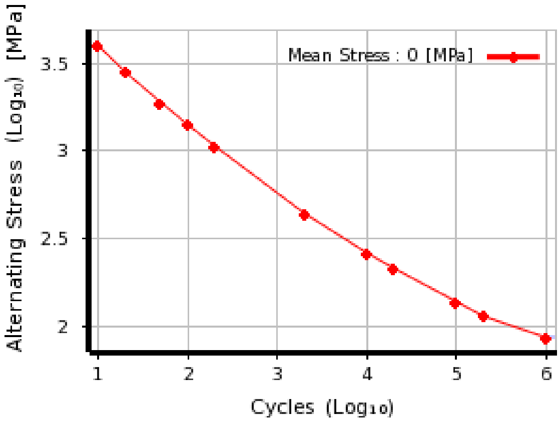

Fig. 6 Material 40CrNiMoA S-N curve for double-log coordinates

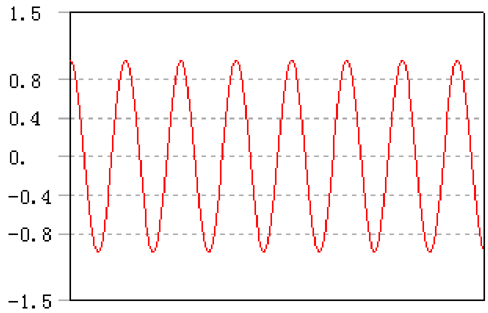

Fig. 7 Fully reversed constant amplitude load spectrum

As shown in Fig.8, the service life of the shaft cloud its value indicates that the shaft damage periodic load cycles. Statics analysis of the value of the maximum equivalent alternating stress more than material S-N curve defined in the lowest alternating stress value, so the maximum stress value of the life which is a minimum service life of the shaft, that is, the fatigue life of shaft under alternating load is 1.0401×10^{7} rotate. As shown in Fig.9 for the fatigue life of shaft sensitive curve, sensitive available fatigue life curve means the life in the critical region along with the change of load amplitude trend. An inflexion was found when the load amplitude is 0.575 times of the original load amplitude. That is, when the load amplitude is less than 0.575 times of the original load amplitude, the shaft for infinite life doesn't happen fatigue damage; when the load amplitude is greater than 0.575 times of the original load amplitude, the fatigue life decreases sharply. When the ratio of 2, life is about 9.24×10^{4} rotate. The Fig.10 shows that maximum damage location is the location of the maximum equivalent stress appeared drive shaft. Safety factor is the parts or components of material failure stress and design stress ratio, From the Fig.11 safety factor of the shaft cloud know that the location of the minimum safety factor, and the maximum equivalent stress of shaft in the same location, if the fatigue safety coefficient is more than 1, indicate that the shaft is safe.

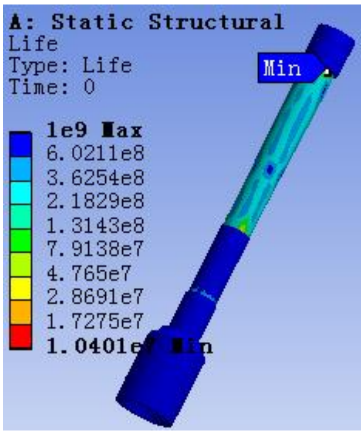

Fig. 8 drive shaft fatigue life cloud

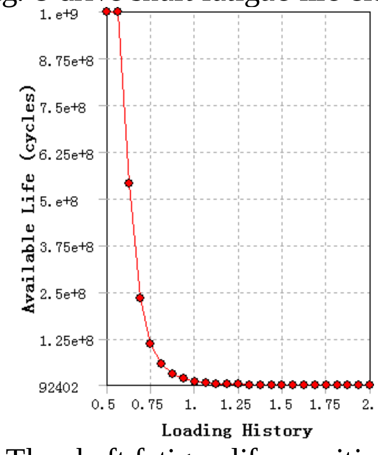

Fig. 9 The shaft fatigue life sensitive curve

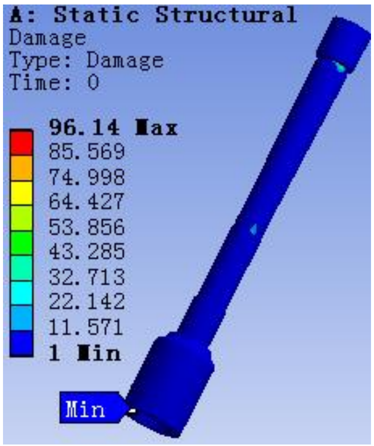

Fig. 10 The shaft damage cloud

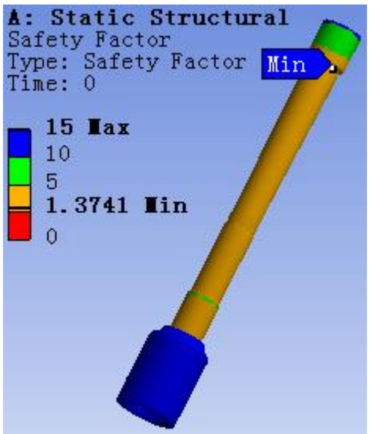

Fig. 11 Shaft safety coefficient cloud

4. Vibration modal analysis of transmission shaft

Structure modal analysis is, based on vibration theory and aiming at modal parameter analysis method, to research the physical parameter model, modal parameters model, and non-parameters model. It also study this models theory and application. Because most of mechanical structures is subject to dynamic load, dynamics design of mechanical structures should be conducted urgently[8].

Structure dynamics equations, to a system including n freedom, the vibration equation is experessed as:

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}=\{F(t)\} \tag{2}$$

Where{u}is displacement vector, [M] is mass matrix, [C] is damping matrix, [K] is stiffness matrix, and F(t) is dynamic load[9].

To dynamic structure analysis in Ansys, F(t) has four different types.

(1) Modal analysis: F(t)=0;

(2) Harmonic response analysis: F(t) is periodic loading;

(3) Transient dynamics analysis: F(t) is impulse load;

(4) Spectrum analysis: F(t) is random load.

To the modal analysis in ANSYS, structure damping matrix [C] is neglected, and the vibration equation of modal analysis is as follows:

$$[M]\{\ddot{u}\}+[K]\{u\}=\{0\} \tag{3}$$

Setting the solution of equation (5) is as following form, it means that each mass point does simple harmonic motion in the same frequency.

$$\{u\}=\{\phi_i\}\cos(\omega_i t) \tag{4}$$

Where ϕ_i is vibration model (Eigenvector), ω_i is the inherent frequency of vibration model i (1≤i≤n). substituted equation (6) into equation (5).

$$([K]-\omega_i^2[M])\{\phi_i\}=0 \tag{5}$$

According to matrix equation (7), the inherent frequency of different medal and the main vibration model vector can be achieved.

During the work situation, transmission shaft is subject to dynamic laod. In order to enhance the stability and reduce the fracture of transmission shaft assembly, it is significant to understand the dynamic property of transmission shaft assembly, i.e. the inherent frequency and inherent vibration model of transmission shaft[10].

Due to the finite element model of modal analysis is built on the basis of static finite element model, static structure analysis is used. Because the inherent characteristic of transmission shaft has nothing to do with external force, the external load can be neglected[11].

The output rotate speed is about 200 r/min, and the working frequency is between 0 Hz to 4 Hz. In order to analyze transmission shaft and other parts synthetically, the computing spectrum is choose between 0 Hz to 300 Hz to satisfy the transmission shaft's computing spectrum.

As shown in the Fig.12-16, it is drawn a conclusion by the analysis of ANSYS Workbench, there are five different elastic mode included in the calculated frequency within 0-300 Hz. Each order inherent frequency and modal characteristics are shown in Table 1.

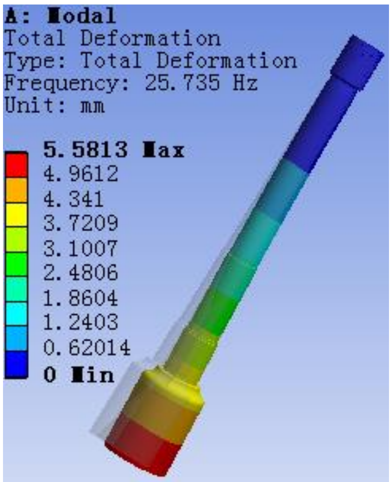

Fig. 12 transmission shaft first-order natural vibration mode

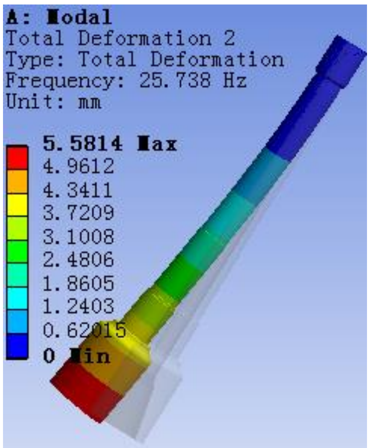

Fig. 13 Second-order natural vibration mode of transmission shaft

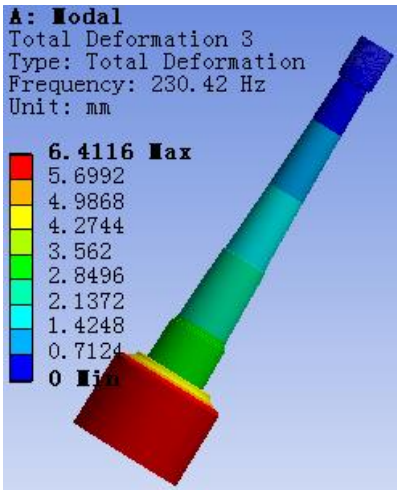

Fig. 14Third-order natural vibration mode of transmission shaft

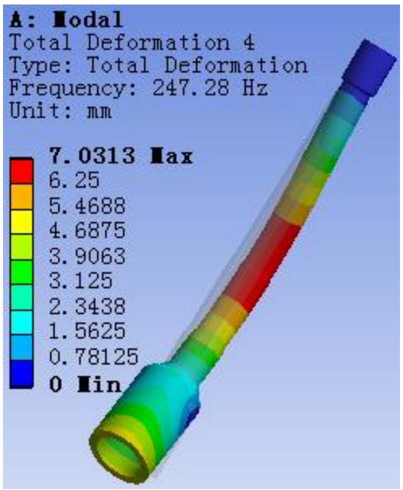

Fig. 15 Fourth-order natural vibration mode of the transmission shaft

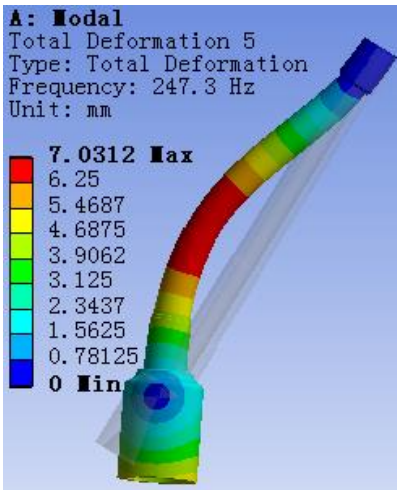

Fig. 16 The fifth-order natural mode of the transmission shaft

In conclusion, there are three kinds of natural mode of vibration of the transmission shaft within 0-300 Hz: the first kind is the first-order polarization mode. The second kind is the radial stretch and torsional vibration mode. The third kind is the first bending mode shape. So within 0-300 Hz, there is more local vibration instead of simultaneous vibration. So from the finite element analysis, we can arrive at the conclusion that the rigidity of the transmission shaft would rely largely on the position of minimum diameter, and when electric drill is under the rated conditions, the vibration frequency of the transmission shaft is within 0-300 Hz, which is far away from the first order natural frequency. At the same time, it is working speed is less than the minimum critical speed of the transmission shaft, so under normal working state, the transmission shaft would not produce resonance phenomenon.

Table 1The five order natural frequency and mode description of the transmission shaft

Modal order	Natural frequency (Hz)	Modal characteristics	Displacement (mm)
1	25.735	the part of connecting the drill shaft has a significant portion of the first order offset (see Fig. 12)	5.5813
2	25.738	he part of connecting the drill shaft has a significant portion of the first order offset (see Fig. 13)	5.5814
3	230.42	Along the radial stretch and accompanied by twist phenomenon (see Fig. 14)	6.4116
4	247.28	String of bearing shaft section of the site have significant first-order bending (see Fig. 15)	7.0313
5	247.3	String of bearing shaft section of the site have significant first-order bending (see Fig. 16)	7.0312

5. Conclusion

1. According to the static analysis, the maximum stress, which is less than the allowable stress of the drive shaft, occurs at the root of the connection parts between the transmission shaft and coupling, when the drive shaft is acted by the maximum torque. Then the maximum deformation value of transmission path near the coupling part is 1.637mm, which is enough small that the shaft rotation axis in the torsional strength is qualified to operate safely.

2. On the basis of static analysis, fatigue analysis was carried out on the drive shaft, the result showed that the fatigue life of the drive shaft is 1.0401×10^{7} times under the alternating load. According to the fatigue theory, when the stress-number of cycles of the material reach 1×10^{6} times, the material can still work normally, that can be considered able to withstand the infinite time. The position of the minimum safety factor is the same as that of the maximum equivalent stress of the drive shaft, and the fatigue safety factor is greater than 1, so that the axis of rotation of the drive shaft can be operated for a long time under alternating load conditions.

3. Under the rated operating condition, the vibration frequency of the drive shaft is 0-4Hz, which is far away from the first-order natural frequency. Its working speed is less than the minimum critical shaft speed, so the drive shaft, when it operates at the normal working state, will not produce resonance phenomenon.

References

[1] LIU Chunquan,XU Maolin and TANG Pinghan.The Present Situation and Development of Underground Electric Drill[J]. Drilling& Production Technology, 2008, 31(5) : 115-117.

[2] ZHU Zhaoqing,LI Guangyong,CHANG Qingzhong,ZHU Qingguo. A Screw Drill Tool Shaft Assembly[P].Jiang Shu: CN204139964U, 2015-02-04.

[3] B. I. Abyzbayev, N. K. Baibakov, B. V. Baidyuk, Y. Gelfgat, M. Gelfgat. Electrodrilling: Past Experience and Present Opportunities[J] . Electric Drilling, 1997 : 573-574.

[4] WANG Jun. *The Static and Dynamic Characteristics Analysis of Screw Drilling Tools*[D]. Shenyang University of Technology,2013.

[5] DONG Junguo, *Practical Material Manual*[M]. Metallurgical Industry Press,2000.

[6] YAO Weixing,*Fatigue Life Prediction of Structures*[M]. National Defend Industry Press,2003.

[7] ZHAO Xi,ZHU Li,LAI Xide,etc.,Fatigue Life Analysis of the Turbine Shaft Based on Workbench[J].China Academic Journal Electronic Publishing House,2015(9):169-171.
[8] KONG Deming. The Research of Modal Parameters Identification for Vibrant System[D]. Yanshan University,2010.
[9] ZHANG Ce.*Machinery Dynamics(Second Edition)*[M].Higher Education Press,2008.
[10] Robert G. Joyner,ZHOU Shijian. The Shaft Can Be Used As a Vibration Source to Study[J].China Academic Journal Electronic Publishing House,1980(4).
[11] PU Guangyi.*The Basic Tutorials and Example Explanation*[M]. China Academic Journal Electronic Publishing House,2010.