The Mechanical Performance Analysis of DZ101 TBM Main Beam

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Abstract. Taking the DZ101 TBM(Tunnel Boring Machine) main beam as research object, with CAD/CAE used, a three-dimensional mechanical analysis model of the main beam was established. And then the static mechanical performance of the main beam was studied. The results show that: the maximum stress of main beam is 132MPa and the maximum deformation is 12.48mm, its safety factor is 2.6, which meets the requirements of strength and the stiffness. The results from modal analysis shows that the frequency of main drive-0.13Hz is far less than the lowest frequency of main beam-8.86Hz. Thus, resonance will not occur. By comparing the results from the field tests, simulation results is verified.

1. Introduction

TBM is a main construction equipment in tunnel excavation [1]. Support system is the key component of TBM,which consists of main beam, thrust cylinders and gripper shoes [2]. Under the complicated uneven geological conditions, TBM main beam is easy to vibrate and shock [3,4,5,6], its reliability directly affects the tunneling efficiency [7,8,9].

Recently, more and more scholars began to focus on the structural design of main beam. Huang Wei [10] researched TBM support system and analyzed force transmission characteristics; Zhang Gao Feng [11] studied the relationship between TBM main machine and other parts by Catia software; Tao Lei [12] analyzed the mechanical performance of main beam under different working conditions and obtained its stress distribution, but don't consider the dynamic performance of main beam. This paper takes the main beam of DZ101 TBM in a tunnel project as the research object. The stress and deformation distribution are obtained by static FEM. The natural vibration characteristics of main beam are acquired by modal analysis and the modal parameters are identified.

2. The analysis of main beam structure

2.1 The introduction of main beam structure

As shown in Fig.1, the main beam is arranged above the center line of the tunnel and consists of the front and rear section. The front section that connects with gripper shoes and main drive transmits force from thrust cylinders to main drive during the advancement process of TBM, to push cutterhead forward. Meantime, the counterforce from cutterhead is transmitted to gripper shoes through the front section. The front one helps to reduce vibration of the front parts of TBM by using the rectangular beam with variable cross section. The rear section connects horseshoe-shaped saddle and rear supports. During stoke changing, the rear main beam and rear supports bear the gravity of TBM main machine together, and the saddle slides forward along the slide rails of main beam. And rectangular cross-section beam structure is employed to match with saddle through slides. Both sections adopt hollow beam structure to install conveyer belt for transporting rocks.

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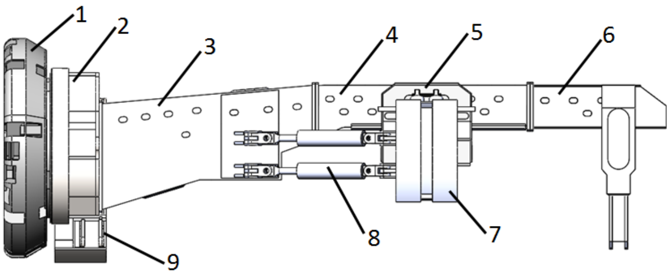


1.cutterhead 2.main drive 3.front main beam 4.rear main beam
5.saddle 6.rear support 7.gripper shoe 8.thrust cylinder 9.shield
Fig.1 The structure of DZ101 TBM

2.2 Force analysis of the main beam

(1)Thrust force of main beam

The thrust from each thrust cylinder can be described as:

$$F_d = F_D / (i \cdot \cos\alpha) \tag{1}$$

Where: F_D is the total thrust of cutterhead, i is amount of thrust cylinders; α is the angel between thrust cylinder and center line which is decided by cylinder stroke, as seen in Fig.2, from $11.17°$ to $16.35°$.

Taken the extreme working condition into consideration as α is $11.17°$, the thrust F_d is $4450\,kN$.

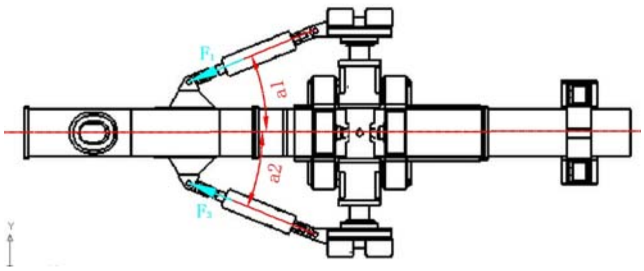


Fig.2 Thrust force of main beam

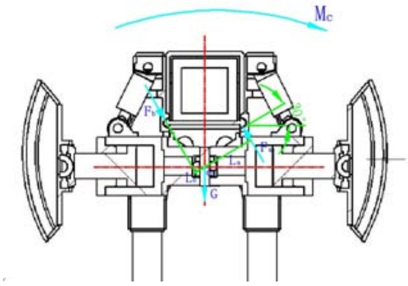


Fig.3 Torque of main beam

(2)Torque of main beam

As seen in Fig.3, when the main beam bears reverse torque, each side of main beam will provides different support forces to balance the reverse torque. According to Newton's Law, the load balance equations can be described as follows:

$$F_a \cos 30° - F_b \cos 30° - G = 0 \tag{2}$$

$$M_C - F_a L_a - F_b L_b = 0 \tag{3}$$

Where: M_C is the reverse torque, $M_C = 8915\,kN$; F_a and F_b are the support force of both sides of main beam; L_a and L_b are force arms, $L_a = 1250\,mm$ and $L_a = 250\,mm$; G is The gravity of main machine born by the rail, $G = 2058kN$.

Then, $F_a = 6339\,kN, F_b = 3963\,kN$ can be acquired.

3. Establishment of FEM model

The actual model of main beam was simplified for mechanical analysis, therefore, some small parts was neglected in the FEM model, such as some bolts holes. The whole model was meshed by 10-node tetrahedral element SOLID187. Finally, the FEM model consisted of 145132 elements and 50883 nodes.

Taken the structure and motion features of main beam into consideration, the full constraints were employed to the front face connected with main drive. The thrust force from thrust cylinders was applied to bolts which connected main beam with thrust cylinders. And the torque was applied to the contact face between the slide of main beam and the saddle of TBM.

4. Simulation result and discussion

Based on the basic theory of FEM, static and dynamic performance analysis was carried out in ANSYS. The stress and deformation distribution of main beam were obtained to determine whether the strength requirement was satisfied. Modal results can determine whether resonance will occur.

4.1 Results of static analysis

Fig.4 and Fig.5 illustrated the stress and deformation distribution of main beam respectively. The maximum stress of main beam, appeared on the contact face between rails and main beam, is 132MPa. It can be explained that the torque induced the relative stress concentration and some mitigation measures such as fillets and chamfers were neglected. In addition, the stress concentration also appeared on the connection part between propulsion base and main beam. As a whole, the stresses on the majority regions of the main beam were less than 70MPa and the safety factor is 2.6, which indicated that the total strength can satisfy design requirements and the structure can be further optimized. The maximum deformation amount, appeared on the rear flange face of main beam, reached to12.48mm.

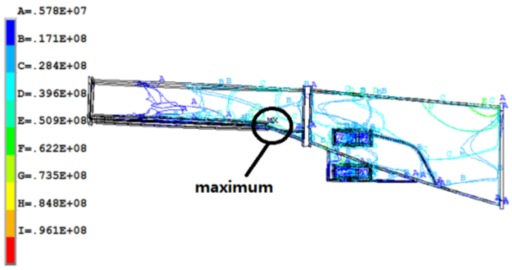


Fig.4 Stress contour

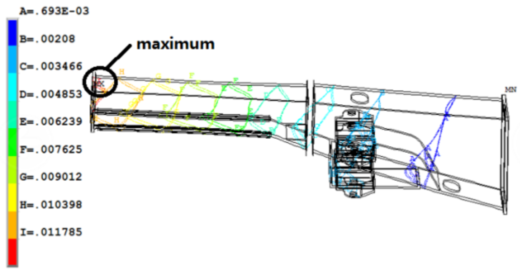


Fig.5 Deformation contour

4.2 Modal results

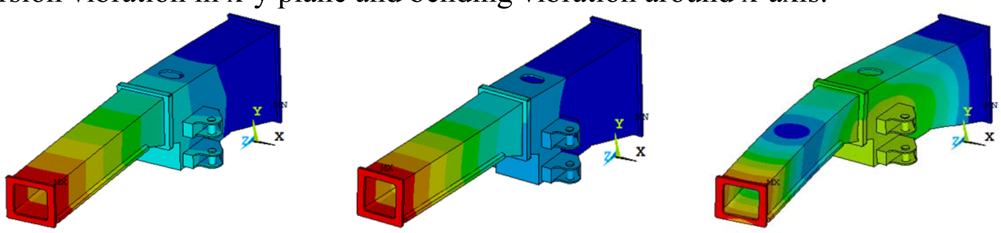
According to the actual working condition, constraints were imposed on the front flange face of the main beam and modes are extracted by Block Lanczos method. Low order modal is the main factor affecting the dynamic characteristics of main beam. Within a certain range, the higher the natural frequency is, the better dynamic performance of main beam is and the less possibility to resonate. The first 10 modes frequency (PREQ) and its displacement deformation (SMX) were shown in Table 1.

Table 1 The first 10 modes frequency and its displacement deformation

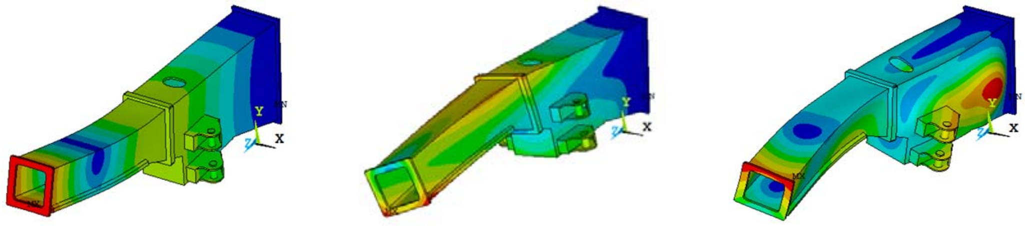
Mode	1	2	3	4	5	6	7	8	9	10
PREQ(Hz)	8.86	12.50	31.26	38.78	47.04	73.41	77.97	83.17	89.08	92.48
SMX(mm)	0.0076	0.0086	0.0066	0.0068	0.0077	0.0073	0.0053	0.010	0.021	0.0074

From Table 1, the first order modal frequency of the main beam is 8.86Hz. Within a certain range, corresponding deformation of main beam presented periodicity during vibrating. The displacement of each modal has a few difference. While the rated rotating speed of main drive is 8r/min and the excitation frequency is 0.13Hz, which is far less than the minimum natural frequency of main beam. So the main beam will not resonate.

Main beam with different modes will vibrate in different ways. There is a corresponding vibration natural frequency and mode shape to every vibration mode. As shown in Fig.6, the first and second mode shape is rigid body rotation. The third and forth mode shape are bending vibration around x-axis and y-axis respectively. The fifth mode shape is torsion vibration in x-y plane. The sixth mode shape is torsion vibration in x-y plane and bending vibration around x-axis.



(a)the first mode (b)the second mode (c)the third mode



(d)the forth mode (e)the fifth mode (f)the sixth mode

Fig.6 The first six modes shape of main beam

5. Engineering verification

Three positions are selected as test points for stress measurement, as shown in Fig.7. Fig.8 shows the change law of stress with time in the horizontal direction at point 1 and the average stress is 6.22MPa. The stress results from simulation are in good agreement with those from field tests, as shown in Fig.9. The stresses in the horizontal direction and driving direction are gradually reduced from the front main beam to the rear one. The measured values decreased by 72% from point 3 to point 1, while the simulated values reduced by 81%. The maximum relative error between test results and simulation results is 10%, which validates the simulation model effectively.

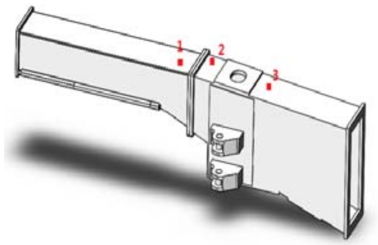


Fig.7 Positions of field test points

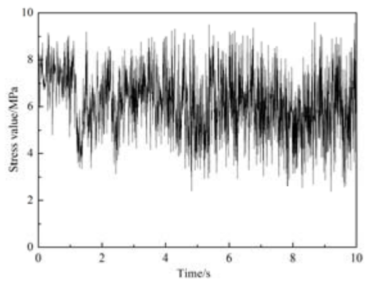


Fig.8 Stress results of point 1

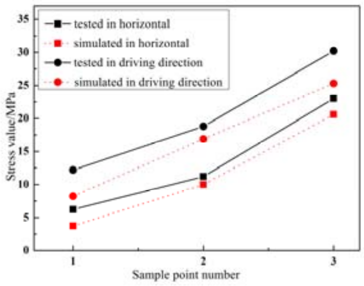


Fig.9 Comparison between test results and simulation results

6. Conclusion

(1) Under the extreme working condition, The maximum stress of main beam appeared on the contact face between rails and main beam, up to 132MPa, and the maximum deformation appeared at the rear flange face of the main beam, up to 12.48mm, which fulfilled the requirements of the strength and stiffness.

(2) The lowest natural frequency of the main beam is 8.86Hz, while the frequency of the main drive is 0.13Hz. The resonance between main beam and main drive will not occur.

(3) The simulated results are in good agreement with test results and the maximum relative error is 10%.

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