Dynamic Characteristics Analysis of Cutterhead System with Uncertain Loads

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Dynamic Characteristics Analysis of Cutterhead System with Uncertain Loads

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Abstract. Tunnel boring machine (TBM) is a large and complex tunnel tunnelling equipment, which is widely used in underground works such as water conservancy, transportation, national defense and energy. In the construction of TBM, it is very difficult to predict the dynamic characteristics of the cutterhead system, because of its complexity and the uncertainty of the service environment, geological uncertainties and manufacturing process. To solve the above problems, first of all, this paper applies concentrated mass method, establishing a multi degree of freedom coupling dynamic model of TBM cutterhead system under uncertain excitation. Considering the changes between hob location and establishing the uncertain geological condition model, the load boundary with uncertainties is obtained and replaced by the load boundary. The dynamic characteristics of TBM cutter head system with uncertain load are obtained, which provides important basis for TBM vibration reduction and optimization design.

1. Introduction

Tunnel boring machine (TBM) is a complex large tunnelling equipment integrated with the functions of tunneling, ballast and lining. It is also a landmark product in the equipment manufacturing industry. Cutterhead is the most important core component of TBM. Because the cutterhead system has complex driving environment, the rock has the characteristics of high hardness, high wear resistance, high temperature and high confining pressure. In addition, the TBM hob has the characteristics of multi-point impact when rock breaking and there will be strong impact load when the hob is cutting rock. These loads will be transferred to the cutter head, which will cause severe vibration of the cutterhead during the working process, resulting in fatigue damage of the cutter head. Therefore, it is very necessary to analyze the dynamic characteristics of the cutter head system. However, due to the uncertainties in the geological distribution of tunneling, uncertainties in the tunneling process and manufacturing and assembly process, such uncertainties will result in uncertainties in the load of TBM, and consequently the dynamic characteristics can not be accurately predicted. Therefore, the dynamic characteristics analysis of the cutter system under the uncertainty factors is more close to the actual working environment, which is of great engineering significance for the structural design and vibration reduction optimization of the TBM cutter head system.

At present, domestic and foreign scholars have done a lot of analysis on the dynamic characteristics of TBM key components from theoretical modeling and experimental aspects. Tang Guowen and Xie Qijiang[1,2] set up the dynamic equations of TBM propulsion system, considering the load of the cutterhead and the supporting interface stiffness under variable constraints. At the same time, they also use interval numbers to describe the uncertainty of geological parameters, and analyze the dynamic characteristics of TBM propulsion system. The dynamic characteristics of TBM under the condition of uncertain geological distribution are evaluated according to the speed and speed of driving angle. Zhang Kaizhi [3] sets up a shield experimental platform in proportion to 10:1. By changing the position of the driving motor, he can get the torque and speed of each drive motor under all loads, and study the load sharing and stability of the driving system. Liu Ran [4] applies the simulated load to build the test bench of the cutter driving system. The validity of the adaptive load sharing control strategy is verified based on the testbed. Zhu Zhenhuan [5] studies the contact characteristics of disc hob during rock breaking process based on the finite element method, and analyzes the kinematics characteristics of disc hob and the three directional force characteristics of disc hob. Huo Junzhou [6] carries out a dynamic simulation of the TBM main engine system, and studies the influence of cutter speed, pinion speed fluctuation and other factors on the dynamic characteristics of the cutter head. Considering the nonlinearity of hob bearing, Huo [7] sets up the dynamic model of hob cutter holder system with multiple coupling surfaces, and modifies the model with experiments, and explores the difference of dynamic characteristics between different hob cutter holder systems. Li [8] established the shield dynamic model of the complex geological conditions, calculated the dynamic response of the cutter rotary system, and explored the influence of the key parameters. Sun [9] proposes a hierarchical modeling method for TBM complex cutter driving system, and deduces the dynamic equation of the cutter driving system with multi-source nonlinear factors, and verifies the model with the test bench.

In summary, scholars at home and abroad have made some researches on the dynamic characteristics of TBM, and also achieved some results, however, they all have their own limitations. There are few studies on the dynamics of the TBM cutter system under the condition of working conditions, and the uncertainties in the TBM tunneling process are often neglected. Therefore, this paper will take the typical TBM cutter head system as the research object, consider the influence of uncertainty factors and establish a dynamic model of TBM cutter head system under uncertain excitation, so as to explore the dynamic behavior of TBM cutterhead system and make the theory more realistic.

2. Model Establishment

2.1. Geological Uncertainty Model

In the process of driving, there are many strata in the excavation interface of TBM. In the TBM excavation interface, the continuous change curve is used to represent the geological change of the excavation interface, thus forming a generalized geological feature model in the TBM cutterhead driving interface.

Assuming that the dividing line between strata is a continuous curve, the curve equation is $f(x,y)$, and xy as the end face of cutter head. For the sake of generality, if there are n strata distributed in the whole excavation interface, the composite geological model can be described by the $n-1$ divider line, and the combined geological model of the excavation interface can be obtained, as shown in Figure 1.

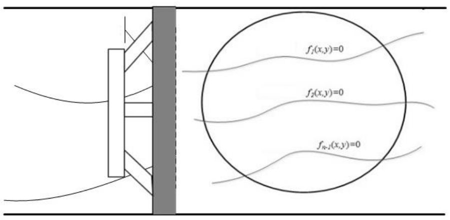


Figure 1. Geological uncertainty model

The t at any time when the cutter head is excavating, the function of the dividing line is $f_{i(t)}(x,y)$, that is, the stratum dividing line equation for the interface of the cutter head at the time. According to the dividing line equation, the stratum of each hob at the corresponding time can be determined, and the driving load of each hob can be obtained and the load of the cutter can be calculated by substituting the calculation formula of the load of the cutter.

According to the engineering statistics, the composite geology in the cutterhead excavation is usually a combination of soft rock and hard rock. Therefore, a composite stratum model consisting of hard rock and soft rock is established in this paper, that is, only one stratigraphic separation line is adopted, and two kinds of geotechnical models with the largest proportion are combined to model the strata.

2.2. Method for obtaining Interval Boundary of Uncertain Load Under Different Working Conditions

In the process of TBM tunneling, there will be different driving conditions. The external load of each cutter is different, and the vibration of cutter head will be very different. Therefore, it is necessary to calculate the vibration response of the cutterhead under different working conditions.According to the engineering experience and the design conditions of several TBM design companies, the typical driving conditions include 4 main types: maximum thrust condition, maximum overturning condition, correcting turning deviation condition and out of difficulty working condition. According to the analysis of different working conditions, and considering the uncertainty of tunnel geological distribution and uncertainty of hob driving process, the calculation flow of each load boundary of cutter head is as follows:

(1) According to the geological survey report and the measured data, the time domain diagram of the single hob is obtained;
(2) According to the driving conditions, the number and location of the hobs involved in the excavation are determined;
(3) We need to determine the formation of each hob at each time in the process of tunneling, and bring the position coordinates (xi，yi) of the i hob at any time to the surface equation of the hierarchical function, and calculate the y coordinates of the corresponding hob points at the current boundary surface. Compared with the yi of hob, the stratum of the hob is judged, and the load of the corresponding hob is extracted to calculate the load of the cutter head;
(4) The time history of the load of the cutter head under each working condition is obtained, and the minimum value of each load of the cutter head is obtained;
(5) In order to ensure the safety design, the maximum load of the cutterhead is generated under the nominal load of each hob, and the boundary of the cutter load can be obtained under different working conditions.

2.3. Equivalent Mechanical Model

The TBM cutter system mainly includes cutter head, big gear ring, drive gear, shield body and motor. The schematic diagram of the cutter head system is shown in Figure 2.

This paper adopts to TBM to analyze the model of the fractional cutterhead system. When establishing the dynamic model, the cutter disc body, the big gear ring, the driving gear, the shield body, the motor and the reducer are equivalent to the mass point, and the motor and the reducer are equivalent to a mass point. By analyzing the vibration of each mass point in radial, axial, overturning and torsional degrees of freedom, the equivalent mechanical model of typical cutterhead is derived by lumped mass method. Among them, X represents the horizontal and radial direction of the cutter head, Y represents the vertical direction of the cutter head, and Z is the axial direction, that is, the heading direction of TBM. The equivalent mechanical model is shown in Figure 3.

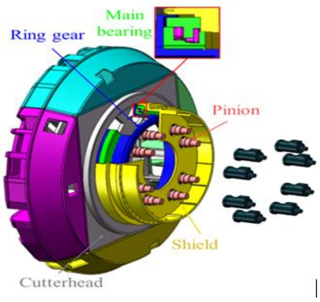
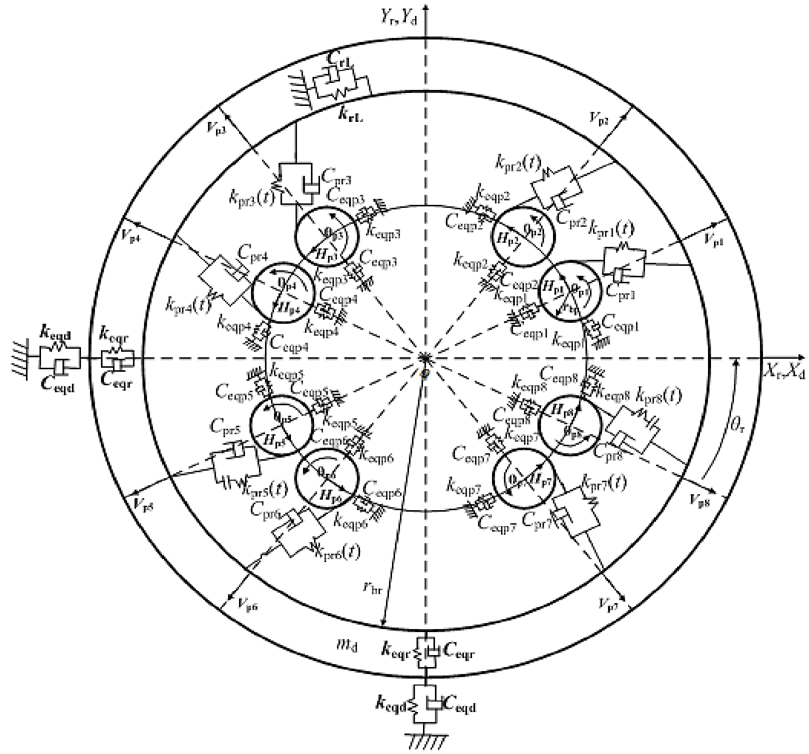
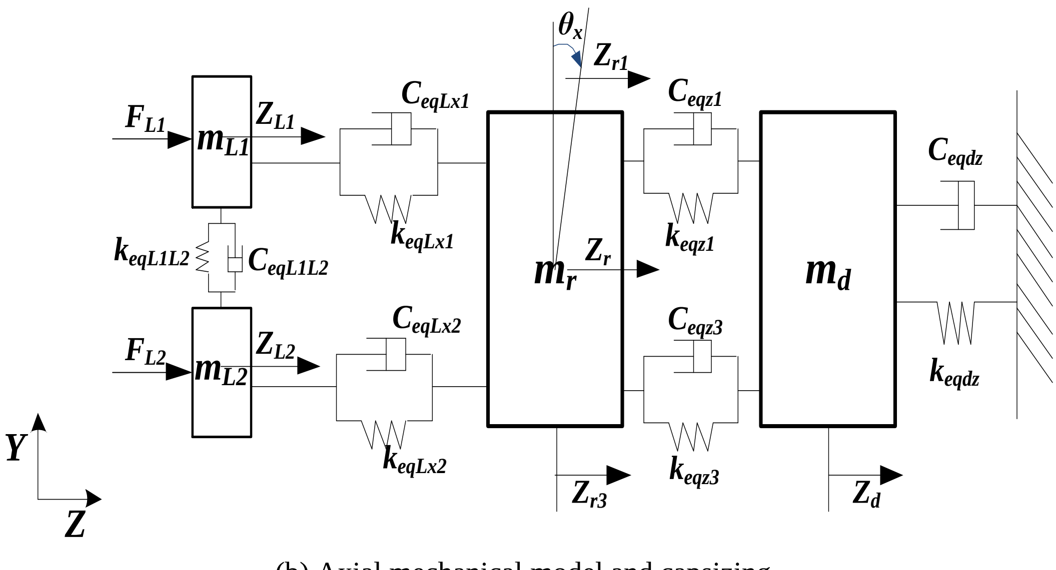


Figure 2. Schematic diagram of cutterhead system structure

This paper adopts to TBM to analyze the model of the fractional cutterhead system. When establishing the dynamic model, the cutter disc body, the big gear ring, the driving gear, the shield body, the motor and the reducer are equivalent to the mass point, and the motor and the reducer are equivalent to a mass point. By analyzing the vibration of each mass point in radial, axial, overturning and torsional degrees of freedom, the equivalent mechanical model of typical cutterhead is derived by lumped mass method. Among them, X represents the horizontal and radial direction of the cutter head, Y represents the vertical direction of the cutter head, and Z is the axial direction, that is, the heading direction of TBM. The equivalent mechanical model is shown in Figure 3.



(a) Torsional mechanical model of rotating system



(b) Axial mechanical model and capsizing

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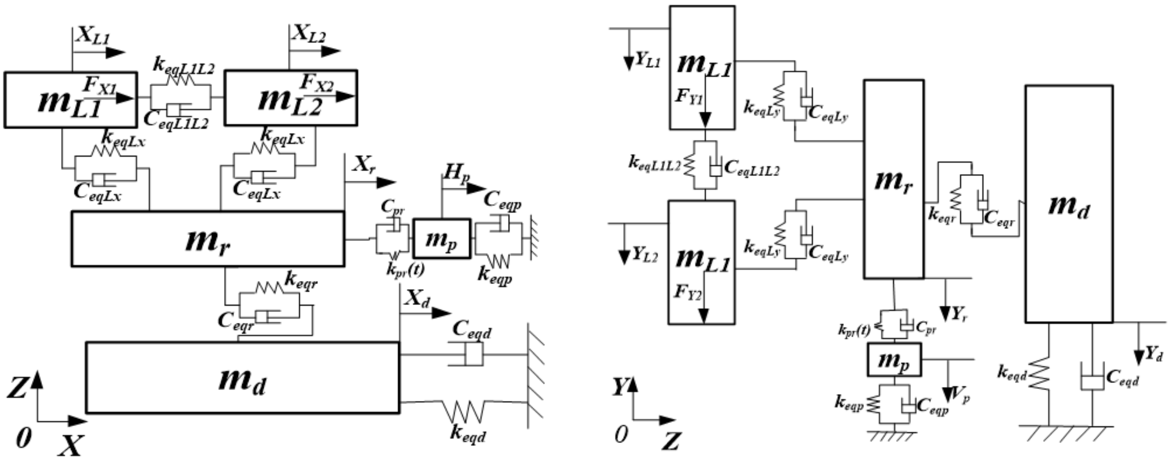

(c-1) Radial mechanical model (c-2) Radial mechanical model

Figure3. Equivalent mechanical model

2.4. Kinetic Equation

According to the established equivalent mechanical model and Newton's second law, this paper takes into account the load balance relationship of various components, derives the differential equations of cutter body split, cutter center block, big gear ring, shield body, pinion and driving motor, and then obtains the multi degree of freedom differential equations of cutter head system.

(1)Cutterhead

The multi degree of freedom dynamics model of the Chinese five-fraction cutterhead is shown in Figure 3.4. The center of the cutterhead, the big gear ring and the shield body are in the generalized coordinate system. And the cutter coordinate system ζi, ηi are relative coordinate systems. With the rotation of the generalized coordinates, coordinate transformation is needed.

$$
\begin{cases}
m_{L1}\ddot{X}_{L1}+C_{eqLx}(\dot{X}_{L1}-\dot{X}_{r})+C_{eqL1L2}(\dot{X}_{L1}-\dot{X}_{L2})+k_{eqLx}(X_{L1}-X_{r})+k_{eqL1L2}(X_{L1}-X_{L2})=F_{X1}^{I}\\
m_{L1}\ddot{Y}_{L1}+C_{eqLy}(\dot{Y}_{L1}-\dot{Y}_{r})+C_{eqL1L2}(\dot{Y}_{L1}-\dot{Y}_{L2})+k_{eqLy}(Y_{L1}-Y_{r})+k_{eqL1L2}(Y_{L1}-Y_{L2})=F_{Y1}^{I}\\
m_{L2}\ddot{X}_{L2}+C_{eqLx}(\dot{X}_{L2}-\dot{X}_{r})+C_{eqL1L2}(\dot{X}_{L2}-\dot{X}_{L1})+k_{eqLx}(X_{L2}-X_{r})+k_{eqL1L2}(X_{L2}-X_{L1})=F_{X2}^{I}\\
m_{L2}\ddot{Y}_{L2}+C_{eqLy}(\dot{Y}_{L2}-\dot{Y}_{r})+C_{eqL1L2}(\dot{Y}_{L2}-\dot{Y}_{L1})+k_{eqLy}(Y_{L2}-Y_{r})+k_{eqL1L2}(Y_{L2}-Y_{L1})=F_{Y2}^{I}\\
m_{L1}\ddot{Z}_{L1}+C_{eqLx1}(\dot{Z}_{L1}-\dot{Z}_{r1})+C_{eqL1L2}(\dot{Z}_{L1}-\dot{Z}_{L2})+C_{eqLy1}(\dot{Z}_{L1}-\dot{Z}_{r2})+C_{eqL1L2}(\dot{Z}_{L1}-\dot{Z}_{L2})\\
\qquad +k_{eqLx1}(Z_{L1}-Z_{r1})+k_{eqL1L2}(Z_{L1}-Z_{L2})+k_{eqLy1}(Z_{L1}-Z_{r2})+k_{eqL1L2}(Z_{L1}-Z_{L2})=F_{L1}^{I}\\
m_{L2}\ddot{Z}_{L2}+C_{eqLx2}(\dot{Z}_{L2}-\dot{Z}_{r3})+C_{eqL1L2}(\dot{Z}_{L2}-\dot{Z}_{L1})+C_{eqLy2}(\dot{Z}_{L2}-\dot{Z}_{r4})+C_{eqL1L2}(\dot{Z}_{L2}-\dot{Z}_{L1})\\
\qquad +k_{eqLx2}(Z_{L2}-Z_{r3})+k_{eqL1L2}(Z_{L2}-Z_{L1})+k_{eqLy1}(Z_{L2}-Z_{r4})+k_{eqL1L2}(Z_{L2}-Z_{L1})=F_{L2}^{I}\\
I_{Lx}\ddot{\theta}_{Lx}+r_{br}[C_{eqLz1}(\dot{Z}_{L11}-\dot{Z}_{r1})-C_{eqLz3}(\dot{Z}_{L33}-\dot{Z}_{r3})+k_{eqL1}(Z_{L11}-Z_{r1})-k_{eqLz3}(Z_{L33}-Z_{r3})]=M_{X}^{I}\\
I_{Ly}\ddot{\theta}_{Ly}+r_{br}[C_{eqLz2}(\dot{Z}_{L22}-\dot{Z}_{r2})-C_{eqLz4}(\dot{Z}_{L44}-\dot{Z}_{r4})+k_{eqLz2}(Z_{L22}-Z_{r2})-k_{eqLz4}(Z_{L44}-Z_{r4})]=M_{Y}^{I}\\
I_{L}\ddot{\theta}_{L}+C_{rLQ}(\dot{\theta}_{L}-\dot{\theta}_{r})+k_{rLQ}(\theta_{L}-\theta_{r})=-T_{L}^{I}
\end{cases}
\tag{1}
$$

The ζi is the tangential ηi of each component of the cutterhead, which is normal; the pinion coordinate system is Hp, Vp, Hp is the pinion tangent, and Vp is the radial of the pinion. In the formula $I(k=L,r,Lx,Ly,rx,ry,pj,mj)$, they respectively represent the torsion of the center block, the twist of the big gear ring, the cutter coiling around the x axis and the cutter coiling around the y axis, the large gear ringing around the x axis and the big gear ringing around the y axis, the pinion twisting, and the inertia moment of the motor torsion. In the formula $m_{\Theta}(\Theta=i,L,r,d,pj)$, they respectively represent each component of cutter head, the center of the cutterhead, the big gear ring, the shield body and the equivalent mass of pinion. In the formula $k_{L\varsigma i},k_{L\eta i},k_{Li}$, they represent the tangential equivalent supporting stiffness of the cutter body, the normal equivalent supporting stiffness and the axial equivalent supporting stiffness respectively. In the formula $k_{eq\vartheta}(\vartheta=x,y,L,r,z,d,dz,p)$, they represent the lateral equivalent supporting stiffness of the cutter head, the longitudinal equivalent supporting

stiffness of the cutter head, the radial equivalent supporting stiffness of the gear ring, the axial stiffness of the gear ring, the lateral equivalent supporting stiffness of the shield body, the longitudinal equivalent supporting stiffness, the axial equivalent supporting stiffness, and the equivalent supporting stiffness of the pinion. In the formula $k_{rLQ}, k_{mpQ}, k_{prt(t)}$, they represent the torsional stiffness of the cutterhead, the torsional stiffness of the pinion connecting shaft and the mean mesh stiffness of the gear respectively. In the formula $C_{eq\vartheta}(\vartheta = x, y, L, r, z, d, dz, p)$, they represent the lateral damping coefficient of the cutter center block, the longitudinal damping coefficient of the cutter center block, the radial damping coefficient of the big gear ring, the axial damping coefficient of the big gear ring, the transverse damping coefficient of the shield body, the longitudinal damping coefficient, the axial damping coefficient, and the damping coefficient of the pinion. In the formula $C_{rLQ}, C_{mpQ}, C_{prt(t)}$, they represent cutterhead damping, torsional damping coefficient of pinion connecting shaft and gear mesh damping respectively. In the formula $T_L, T_{mj}, F_X, F_Y, M_X, M_Y, F_L$, they represent cutter torque, motor torque, lateral load of center block, longitudinal load of center block, cutterhead overturning moment and axial load of cutterhead. In the formula $F_{\zeta i}, F_{\eta i}, F_{Li}$, they separately represent the tangential load of the cutterhead body, the normal load of the cutterhead body and the axial load of the cutterhead body. In the formula b_{prj}, e_{prj}, they represent the meshing error of pinion and large gear tooth side clearance as well as pinion and large gear tooth clearance. In the formula r_{br}, r_{bp}, they represent the base circle radius of the big gear ring and the radius of the pinion base circle respectively.
According to the equivalent mechanics model, the generalized displacement matrix of cutter head system $\{\delta\}$ is:

$$\begin{aligned}\{\delta\} = \{&\zeta_i, \eta_i, Z_i, X_L, Y_L, Z_L, \theta_{Lx}, \theta_{Ly}, \theta_L, X_r, Y_r, \\ &Z_r, \theta_x, \theta_y, \theta_r, X_d, Y_d, Z_d, H_{pj}, V_{pj}, \theta_{pj}, \theta_{mj}\}^T\end{aligned} \tag{2}$$

(i=1~4，j=1~number of pinion) Matrix in 3.1, the formula ς_i, η_i, Z_i, they represent the tangential displacement, normal displacement and axial displacement respectively. In the formula $X_L, Y_L, Z_L, \theta_{Lx}, \theta_{Ly}, \theta_L$, they represent the lateral displacement, longitudinal displacement, axial displacement, overturning vibration displacement and torsional vibration displacement of cutter head respectively. In the formula $X_r, Y_r, Z_r, \theta_x, \theta_y, \theta_r$, they represent the lateral displacement, longitudinal displacement, axial displacement, overturning vibration displacement and torsional vibration displacement of the gear ring respectively. In the formula X_d, Y_d, Z_d, they represent lateral displacement, longitudinal displacement and axial displacement respectively. In the formula $H_{pj}, V_{pj}, \theta_{pj}$, they represent the tangential displacement, radial displacement and torsional vibration displacement of each pinion, respectively. The symbol θ_{mj} represents the angular displacement of each motor's torsional vibration.

3. Interpretation of Results

Through the detailed analysis of many uncertain complex influence relationships such as geological factors, this paper establishes the uncertain geological condition model and takes account of the relationship between the hob position, and finally obtains the cutterhead load boundary under different conditions by interval method, and introduces it into the dynamic equations established and calculates the dynamic response boundaries of the five-fraction cutterhead in each working condition of each part of the project, respectively, as shown in Figure 4. In this paper, the response is analyzed, and then the vibration condition of the cutter head system during tunneling is theoretically understood.

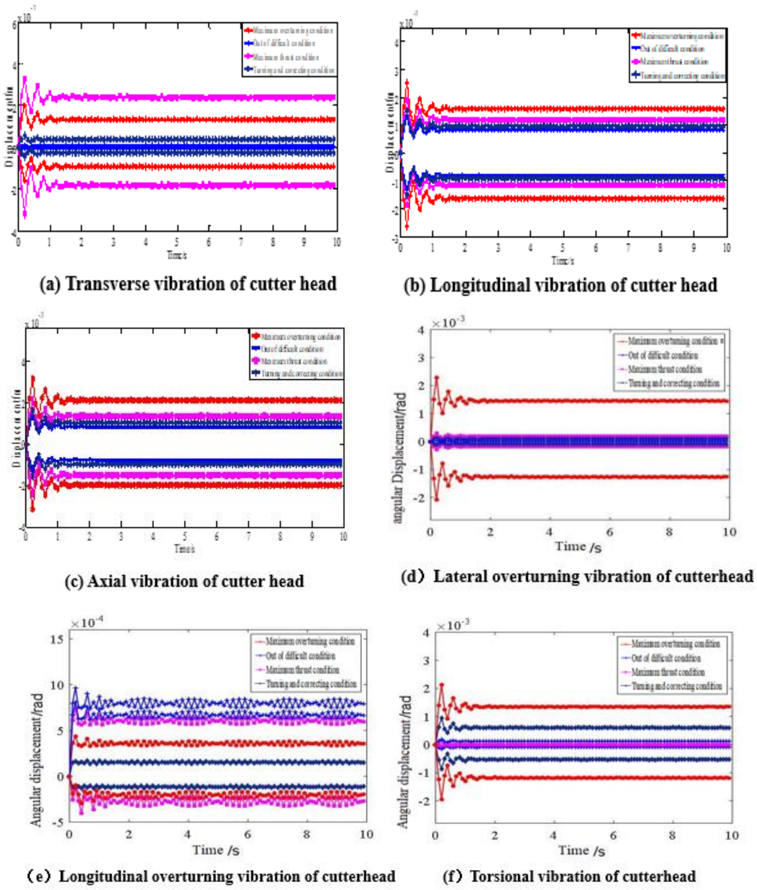

Figure 4. Dynamic response boundary of various degrees of freedom of cutterhead system under different working conditions

As shown in the figure above, the free vibration response of cutterhead can be divided into two stages, namely, non-stationary stage and stationary stage. At the initial stage of cutterhead excavation, the vibration response boundary is non-stationary, oscillating with time, and the vibration response interval enters a stationary stage and oscillates with time. By analyzing the maximum response interval and the steady-state response interval of one circle cutterhead under different conditions, the stationary response intervals of the degrees of freedom of the cutterhead are all narrower than those of the maximum response interval, and the vibration response interval of the cutter body translation is wider than that of the center block translational vibration response area.
Through the dynamic analysis of different degrees of freedom on different degrees of operation, it can be obtained that under different degrees of freedom, which kind of working condition has greater response, therefore, designers can comprehensively consider the vibration response of different working conditions and different forces when designing the cutter head, and aim at the vibration reduction design of the cutter head structure.

4. Conclusion

This paper first considers the geological uncertainty and establishes the geological uncertainty model, and analyzes the uncertainty factors of the cutter head system, and solves the external load interval boundary of the TBM cutter head system. Then, based on multi-body dynamics, this paper establishes a typical TBM cutter system dynamics model under uncertain excitation, the dynamic equation, and obtains the dynamic characteristics. The analysis results can provide a theoretical reference for the structural design and vibration reduction optimization of the cutterhead, and further improve the damage of the cutterhead and ensure the quality of the engineering excavation.

Acknowledgments
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References
[1] Tang G, Haidong Y U, Xie Q. Evaluation of Dynamic Characteristics of Tunnel Boring Machine Excavating in Uncertain Geological Parameters[J]. Journal of Shanghai Jiaotong University, 2016.
[2] Xie Q J. Research on Modeling and Dynamic Behavior of TBM with Variable Constraints [D]. Journal of Shanghai Jiaotong University, 2015.
[3] ZHANG Kaizhi. Study on dynamic characteristics of redundantly driven revolving system of shield TBM[D]. Shanghai:Shanghai Jiao Tong University,2011.
[4] LIU R.Research on Adaptive Load Sharing Control for synchronous Driving System of cutterhead in shield machines [D].Dalian University of Tecnology, 2013.
[5] Zhu Z H. Numerical Simulation of Wear Evolution of disc cutter of TBM [D]. Central South University, 2013
[6] Huo J, Ouyang X, Wang Y, et al. Analysis of influencing factors of vibration behaviors of TBM cutterhead under heavy impact loads[J]. Journal of Harbin Engineering University, 2015, 36(4):555-559.
[7] Huo J, Hou N, Sun W, et al. Analyses of dynamic characteristics and structure optimization of tunnel boring machine cutter system with multi-joint surface[J]. Nonlinear Dynamics, 2017,87(1):237-254.
[8] Xian H L, Hai B Y, Ming Z Y, et al. Dynamic Modeling and Analysis of Shield TBM Cutterhead Driving System[J]. JOURNAL OF DYNAMIC SYSTEMS MEASUREMENT AND CONTROL-TRANSACTIONS OF THE ASME, 2010, 132(4):1-14.
[9] Sun W, Ma H, Song X, et al. Modeling and Dynamic Analysis of Cutterhead Driving System in Tunnel Boring Machine[J]. Shock and Vibration, 2017:1-12.