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Vibration response and parameter influence of TBM cutterhead system under extreme conditions †

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Abstract

A cutterhead endures complex random loads when the TBM tunnels, which leads to structure excessive vibration, due to the extreme working conditions. So anti-vibration design is key in the cutterhead system design. In this study we established a virtual prototype of TBM mainframe system, comprehensively considering the spatial multi-point random excitations, split cutterhead structure and multi-directional support stiffness of the main bearing, etc., based on multi-body system dynamics and virtual prototype technology. For verifying the virtual prototype model, a TBM cutterhead vibration field detection system was constructed, to test the cutterhead accelerations and compare the simulation and measured results. The results show that the acceleration change rule by the two approaches is similar, and the maximum relative error is 55 %, keeping on the same order of magnitude, which illustrates the established model has certain reliability. Accordingly, an orthogonal test scheme for cutterhead radial vibration analysis was designed, to analyze the influence of different parameters on cutterhead vibration, based on the virtual prototype model. As indicated in the analysis, the parameter of cutter force had the greatest influence on cutterhead vibration, and then the pinion support stiffness and cutterhead rotating speed. Last, the cutterhead vibration optimal schemes in different directions were confirmed. The research results and methods can provide reference for TBM system parameters matching.

Keywords: TBM cutterhead; Vibration responses; Parameters influence; Virtual prototype; Vibration measurement

1. Introduction

As a new and advanced large engineering equipment, the full face rock tunnel boring machine (TBM) is hailed as the king of construction machines, with the advantages of fast, high quality, safety and environmental protection, with the tunnel being excavated and crushed by the cutting tools. Cutterhead system is the core component of TBM, and its service life is equal to the whole machine life. During the tunnel excavation, the cutterhead always works in the conditions of superhard and fractured zone surrounding rock, water gush, as well as the multi-point impact loads with different types of cutters. Accordingly, the cutterhead endures extremely serious vibration, with the extremely complex stress state and load transfer law. For this reason, some engineering faults may appear, such as abnormal wear of cutting tools, cracking of cutterhead panel, main bearing failure and longtime stoppage [1, 2]. Therefore, the cutterhead should have strong strength, small vibration, high reliability and long service life, and it is necessary to analyze the dynamic characteristics of the TBM cutterhead system at the design stage, to provide reference for the matching design of system parameters.

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With the construction of railways, tunnels and water diversion projects, scholars have done extensive research on the TBM cutterhead system, such as rock breaking load detection, cutterhead system vibration characteristics and performance prediction. In respect of rock breaking load detection, the cutter force detection and analysis in different tunneling fields are carried out, to gauge the vertical force, lateral force and rolling force of cutters at different positions on the cutterhead, for studying the cutter load change regulation and distribution characteristics [3-5]. Similarly, Chang et al. did a linear cutting machine (LCM) test by using a cutting test platform, to analyze the relationship between cutter loads and cutting parameters, and then proceeded to establish the force prediction model of cutters [6, 7]. Geng et al. proposed a free-face-assisted rock breaking method based on the multi-stage TBM cutterhead, and carried out the rotary cutting tests in different free surface conditions [8]. Labra et al. built a rock breaking simulation model of cutters by coupling the FEM and DEM methods, and contrasted the simulation results with test results, to validate the validity of the numerical model [9]. Studies on the vibration characteristics analysis of cutterhead system are also concerned. A multi-degree-of-freedom coupled nonlinear dynamic model for TBM cutterhead system was built, and solved by a numerical iterative method, then the results were validated by the field test data [10-13]. Huo et al. proposed a generalized hierarchical modeling method to establish a dynamic model of the TBM cutterhead driving system, and then analyzed the system vibration characteristics and validated the results [14, 15]. Li et al. established a generalized nonlinear time-varying dynamic model for the cutterhead driving system of a shield tunnel boring machine, and analyzed the effects of system parameters on the dynamic response of driving system [16]. Liao et al. also established a general nonlinear dynamic model of the cutterhead driving system, and studied the torque control strategy of the driving system [17]. Many scholars have paid attention to performance prediction of the cutterhead system. Farrokh et al. calculated the TBM penetration rates (PR) with different models, and contrasted with the test data, and proceeded to put forward a new PR predictive model [18]. A two-stage cutterhead system was designed by using grey relational analysis method, and the mechanical performance between the two-stage cutterhead and flat-face cutterhead was compared [19]. Huang et al. presented an approach for the force analysis of an open TBM gripping–thrusting–regripping mechanism, that can be used to estimate the tunneling loads [20]. Xia et al. established a three-dimension FEM model of rock fragmentation induced by single TBM disc cutter, to investigate the influence of side free surface on the rock fragmentation pattern and cutting efficiency, and conducted a series of cutting experiments to validate the numerical results [21]. Based on three-dimensional theory, the theoretical formulas were developed for the fragmentation energy consumption of single traditional and newly designed disc cutters, and the actual field data was obtained to validate these formulas [22]. Han et al. carried out a series of sequential laboratory indentation tests, to study the influence of pre-set penetration depth and cutter spacing on sandstone breakage and TBM performance [23]. Moreover, many researchers studied the cutter plane layout design, cutter structure optimization, cutterhead life prediction, fatigue damage mechanisms, rock-breaking process numerical and experimental research, and obtained some useful conclusions [24-28].

As can be seen from the previous studies, large amounts of research have concentrated on the force detection of disc cutters, dynamic characteristics and performance evaluation of cutterhead system, by using the test methods, theoretical derivation, numerical simulation and field test, and the corresponding results were achieved. Especially, many research achievements about multi-degree-of-freedom coupling TBM cutterhead model have been obtained, which are of valuable reference to our study [10, 11, 13]. However, there are still many problems in the vibration analysis and parameter influence about TBM cutterhead system, such as model simplification and parameter equivalence. Sun et al. [10] and Huo et al. [13] built multi-degree-of-freedom coupling dynamic models of TBM cutterhead system with different degree of freedom, respectively. Furthermore, the dynamic characteristics and field test of cutterhead were conducted, but the rationale of analysis results in Ref. [10] was not presented, and the influence of system sensitive parameters on cutterhead vibration was also not analyzed.

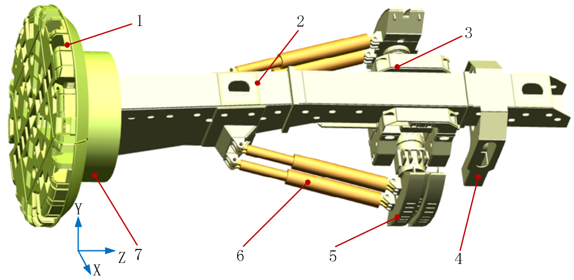


Fig. 1. The mainframe structure schematic of actual TBM: (1) Cutterhead; (2) main girder; (3) saddle; (4) rear supporting leg; (5) gripper shoe; (6) propel cylinder; (7) main driving system.

From the above discussion, it is necessary to further study the vibration responses of cutterhead under extreme conditions, as well as the influence of different system parameters on the cutterhead vibration, to provide a certain reference for TBM system parameter matching. For this purpose, a virtual prototype model of TBM mainframe system was established by the mechanical dynamics software ADAMS, and the cutterhead dynamic responses were obtained under the extreme conditions. To verify the correctness of the simulation results, a field vibration test of TBM cutterhead was conducted, and the simulation and test results were presented. Then, based on the virtual prototype model, the influence of system parameters on cutterhead vibration was analyzed, to obtain the reasonable parameter values.

2. Dynamic characteristics of TBM mainframe system and field test verification

2.1 Dynamic model of TBM mainframe system under the extreme condition

The TBM mainframe system is mainly composed of the cutterhead, main girder, saddle, rear supporting leg, gripper shoe, propel cylinder and the main driving system, as shown in Fig. 1. Wherein X, Y and Z are the horizontal, vertical and axial direction of TBM, respectively. TBM working parts mainly contain two kinds of power transmission path: Rotatory motion, which is provided by the main drive system; and translational motion, which is provided by the supporting and propulsion system. Thus, the cutterhead is supported by the main bearing, and bolted with the rotating part of the main bearing by the hydraulic tensioning expansion bolts. When the main drive system provides torque, the cutterhead rotates, to drive the cutters to excavate and crush the surrounding rock. On the other hand, the main girder is connected to the main bearing outer ring, and supported by the saddle, which can slide along the tunneling direction. The saddle is fixed to the tunnel wall with the gripper shoe and support cylinder, and the propel cylinder transmits thrust to the cutterhead, which provides the thrust that makes the whole machine move forward. Meanwhile, the saddle endures thrust, torque and overturning moment transmitted from the cutterhead system.

2.1.1 Definition of the extreme condition

Due to the complex variability of geological conditions and the uncertainty of control parameters, the cutter loads change randomly during excavation. In this paper, a 3D rock breaking simulation of cutters under typical geological conditions based on LS-DYNA simulation platform was carried out, and the results of cutters loads were corrected by the measured cutter loads, to obtain the dynamic loading history of each cutter, for providing the load excitation of system dynamic response analysis [10]. Moreover, from the perspective of safety and conservative design, it was assumed that all the cutters were in the state of rock breaking at the same time. That is, all the cutters on the cutterhead worked simultaneously, and the mean load of each cutter was equal to the nominal load, ignoring the bearing order in the rock breaking process.

2.1.2 Virtual prototype of TBM mainframe system

The equivalent model of TBM mainframe system was built in a 3D modeling software, and then it was imported into the dynamic simulation software ADAMS. A virtual prototype model of the mainframe system was built after the system joints, loads, contact forces, and the motions were added in ADAMS, as shown in Fig. 2. In the model, the forces of each cutter under the extreme condition were fitted by the fitting method of Akima spline curve, to simulate the load time history of cutterhead, as shown in Fig. 3.

In Fig. 2, the supporting bearings of pinions and main bearing of cutterhead were simulated by the bushing connectors, and the calculation method of each support stiffness and damping was shown in Sun et al. [10]. Between the main bearing and pinions, the contact pairs were added, and the contact stiffness and damping were calculated based on hertz theory, specific formulas [10]. Moreover, the relative displacements among each cutterhead block and girder were kept unchanged, so the fixed joints were applied to each par; also, the saddle and gripper shoes were fixed on the part of the ground. In addition, translational joints were added between the main girder and saddle, hydraulic cylinder and sleeve, and in the hinge positions of hydraulic cylinder, main girder and gripper shoe, eight revolute joints were added. For one thing, the rotary drive was supplied by ten evenly arranged pinions, and a point motion was applied to each pinion, which did not affect the free degree of system and kept floating support. For another, the propulsion power of the whole machine was provided by four hydraulic cylinders, and a translational joint motion was applied to the translational joint of each hydraulic cylinder.

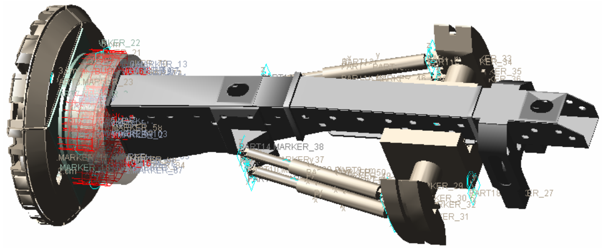


Fig. 2. Virtual prototype model of TBM mainframe system.

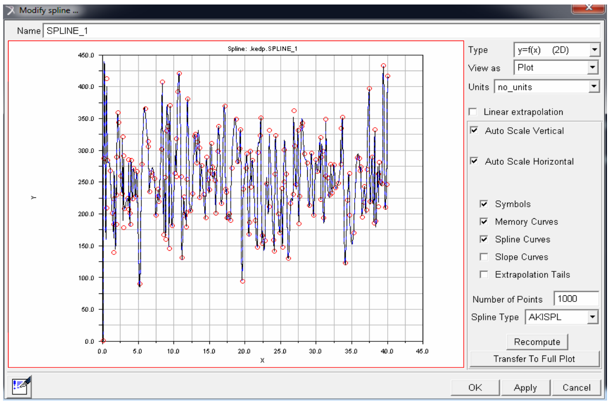


Fig. 3. Load spline curve of a cutter.

(a) (b)

Fig. 4. Surrounding rock in the tunneling field: (a) Surrounding rock of tunnel face; (b) rock detritus after excavation.

2.2 Field test verification

2.2.1 Engineering background

For validating the above virtual prototype model, a field vibration test was conducted to compare the test and simulation results. The engineering background of this test is a water diversion project in China, and the tunnel length is 230 km. Eight open type TBMs are used to construct simultaneously, and the surrounding rock at the excavation site is shown in Fig. 4. Combining the bidding documents with geological prospecting data, the surrounding rock of 8# section in tunnel is regarded as the geological boundary, with the statistical results of surrounding rock classification listed in Table 1. Wherein, the quartz ratio is about 38 %, the main categories of surrounding rocks are II and III, and the average compressive strength is about 90 MPa.

The net section diameter in TBM tunneling section of this water diversion project is 7.51 m, and the boring diameter is

Table 1. Statistics of the geological surrounding rock classification about section 8#.

Geological section		Surrounding rock classification								Compressive strength /Mpa
		II		III		IV		V		
Major lithological distribution	Segment length /m	Length /m	Percentage /%	Length /m	Percentage /%	Length /m	Percentage /%	Length /m	Percentage /%	
Migmatite	3658	914	25	2191	60	366	10	187	5	80~140
Monzonite	7950	1987	25	4770	60	795	10	398	5	60~120
Granite	1050	262	25	630	60	105	10	53	5	70~110
Quartz monzonite	5776	1444	25	3466	60	578	10	288	5	70~100
Granite	76					23	30	53	70	70~110

Table 2. Parameters of actual cutterhead system.

Number of cutters	Power of drive motor (kW)	Rated speed of cutterhead (rev/min)	Turnaround torque (kNm)
53	3300	5.6	9633
Rated thrust (kN)	Modulus (mm)	Teeth number of ring gear	Teeth number of pinion
16509	20	198	19

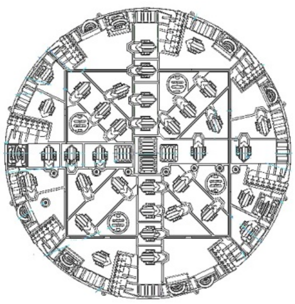


Fig. 5. The layout of actual cutterhead.

Table 3. Technical parameters of the wireless acceleration sensor.

Model number	Channel number	Frequency response ($Hz\cdot(-3db)^{-1}$)	Measurement range (g)
A302-EX	3	300	10
Transmission distance (m)	Transmission speed ($kbit\cdot s^{-1}$)	Synchronous precision (ms)	Bandwidth resolution (μg)
300	250	1	10

(a) (b)

Fig. 6. The sensor and connection base: (a) Acceleration sensor; (b) connection base.

8.53 m. The cutterhead system parameters are presented in Table 2.

The machine type of the actual TBM is a gripper, and the cutter layout is shown in Fig. 5. Since the bull gear bolts together to the cutterhead flange, when the bull gear of the main drive is driven by ten pinions synchronously, the cutterhead rotates. The whole mainframe is supported by the front side support and gripper shoes, with high structural stability, which is presented in Fig. 1.

2.2.2 Test device selection and scheme construction

1) Sensor selection

Due to the serious vibration environment in the excavation site, and the installation space limitation of the test device, the test circuit cannot be arranged effectively. Consequently, a wireless acceleration sensor was chosen to gauge the cutterhead's three-dimensional accelerations. The wireless acceleration sensor device integrates the power module, data acquisition module and wireless transceiver module, which can collect the three-dimensional accelerations simultaneously. The data can be transmitted to the computer in real-time, and can also be stored in the internal memory, which guarantees the accuracy of data acquisition. Besides, all modules are packaged in a special shell, and the shell is manufactured by a special anti-vibration and waterproof material. Moreover, a super magnet is placed in the shell, and then the sensor is bolted to the cutterhead by a connection base. The sensor and connection base are shown in Fig. 6, and the sensor technical parameters are listed in Table 3.

2) Test scheme construction

Considering the actual working condition of the TBM cutterhead, the sensor must be installed in an enclosed space, to avoid the damage caused by rock impact.Therefore, the acceleration sensor is placed in the enclosed space located at the

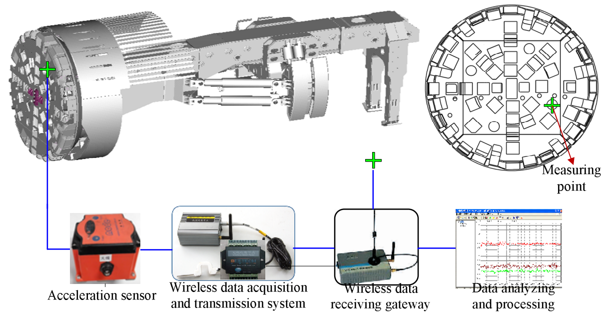
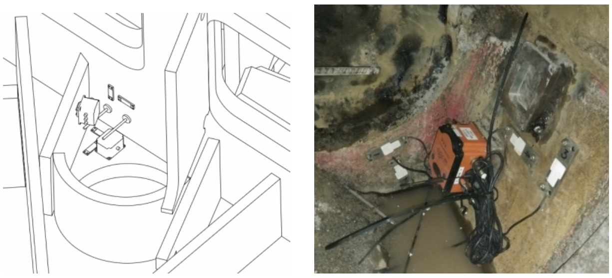


Fig. 7. Vibration test schematic of TBM cutterhead.



(a) (b)

Fig. 8. Layout of the sensor: (a) Sensor layout design; (b) actual layout of sensor.

Fig. 9. The composition of vibration testing system.

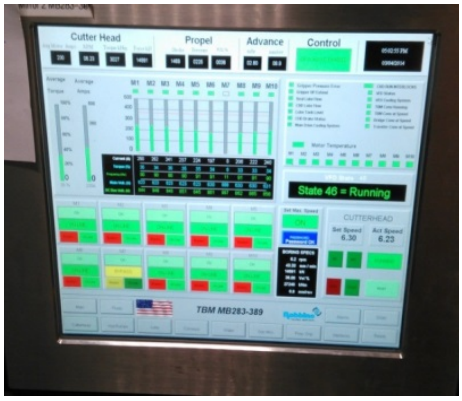
cutterhead manhole, to measure the three-dimensional accelerations. The accelerations signal is transmitted to computer via a sensor node and receiving gateway, and then recorded and stored by the data acquisition software. The constructed cutterhead vibration test schematic is shown in Fig. 7, and the sensor layout is presented in Fig. 8.

2.2.3 Data acquisition and post processing

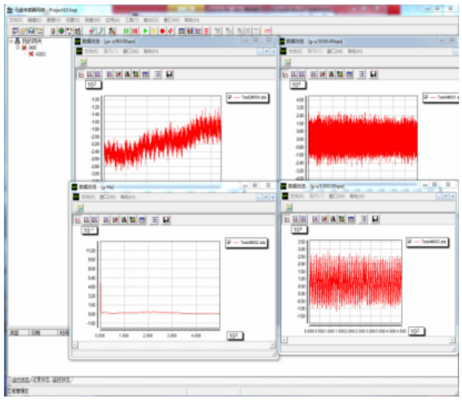
After the wireless acceleration sensor is installed to the cutterhead, the acceleration signal is sent to the receiver, and then it is stored and post-processed. The compositions of whole vibration detection system are displayed in Fig. 9, and the monitoring interface of TBM tunneling parameters, data post-processing interface are presented in Fig. 10.

2.2.4 Data analysis and discussion

The cutterhead three-dimensional translational accelerations



(a)



(b)

Fig. 10. Interface of the data monitoring and collection: (a) Tunneling monitoring interface; (b) post-processing interface of sensor data.

were collected by the above vibration detection system, and recorded by an acquisition and processing software BeeData. Then the data was treated by fast Fourier transform (FFT), to obtain the accelerations time domain and frequency domain results as shown in Fig. 11.

To further analyze the vibration intensity of test accelerations, the minimum value, maximum value, mean square deviation, and the variation coefficient of each direction were calculated, as the results presented in Table 4.

As can be known from the above vibration curves in Fig. 11, the fluctuation center of horizontal and vertical accelerations was all about 10m·s^{-2}, and the axial value was about 0m·s^{-2}. This is because the horizontal and vertical accelerations were tested under the rotating coordinate system of the cutterhead, all of which were affected by gravity, and the axial acceleration was not influenced. So the test results were basically in accordance with the actual situation, which showed that the detection scheme was reasonable and effective to a certain extent. Besides, as a whole, the horizontal and vertical acceleration curves exhibited periodic impact characteristics, and the period was about 12 s, which was basically consistent with the cutterhead rated speed 5.6 rev/min. It was illustrated that the radial acceleration fluctuation was caused by the cutterhead rotation, and the local oscillation was due to the rock

Table 4. Statistical results of cutterhead accelerations.

Index	Statistical results in all directions /m·s^{-2}		
	Horizontal	Vertical	Axial
Minimum value	-10.37	-25.28	-27.88
Maximum value	28.89	36.51	27.99
Mean square deviation	6.99	7.34	5.55
Coefficient of variation	0.94	0.85	-88.91

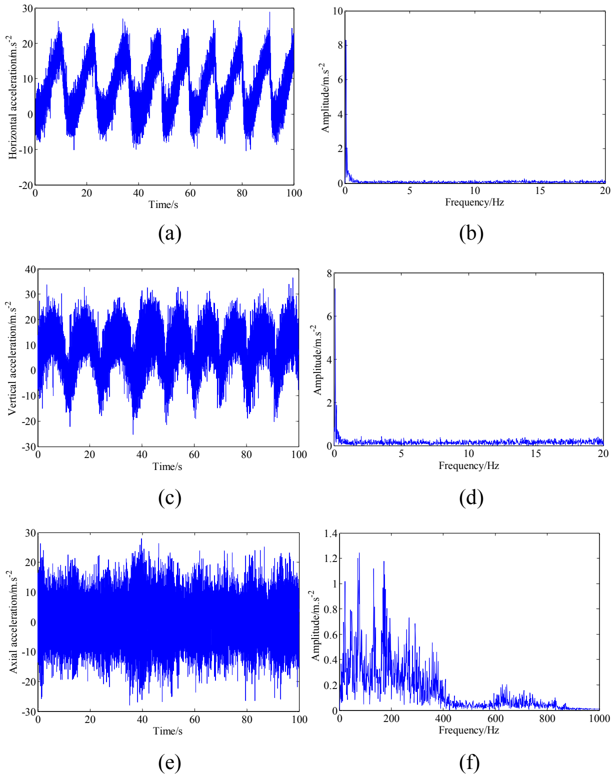


Fig. 11. The time and frequency domain curves of cutterhead accelerations: (a) Horizontal acceleration; (b) frequency response of horizontal acceleration; (c) vertical acceleration; (d) frequency response of vertical acceleration; (e) axial acceleration; (f) frequency response of axial acceleration.

breaking impact. Nevertheless, the axial acceleration changed very exquisitely and stochastically, which was caused by the rock excavation. Moreover, the horizontal and vertical acceleration maximum amplitude frequency was concentrated near the cutterhead rotating frequency, while the other amplitude frequencies were negligible. The axial acceleration frequency was mainly below 400 Hz, and the maximum amplitude frequency was about 75 Hz, the secondary energy frequency was around 200 Hz. The frequency results were approximately closed to the calculated external excitation frequency of cutterhead system [10], which was in accordance with the rules of forced vibration.

From the statistical results in Table 4, the acceleration range of each direction was 39.26 m·s^{-2}, 61.79 m·s^{-2} and 55.87 m·s^{-2}, respectively, wherein the vertical range was the maximum and the horizontal value was the minimum. The difference in the mean square deviation of three directions was almost the same, and the vertical value was the maximum, with the value of 7.34 m·s^{-2}, then the horizontal and axial value. In addition, the absolute value of variation coefficient at axial direction was much larger than the radial value, which illustrated that dispersion degree of the cutterhead axial acceleration was the highest, and the vibration was the most intense.

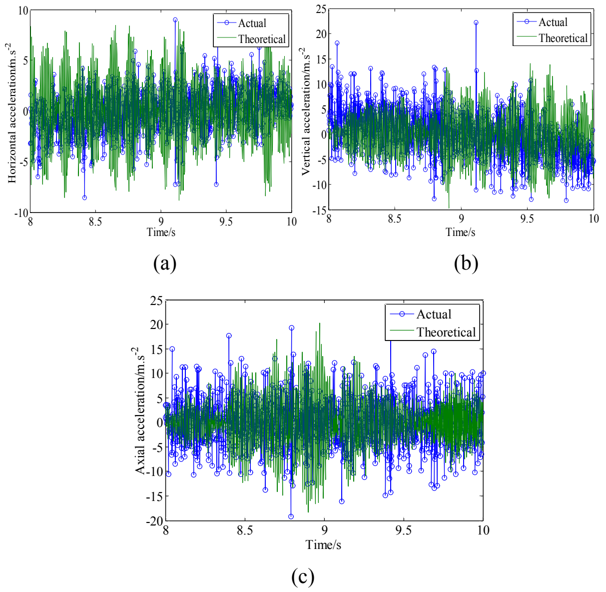


Fig. 12. Comparations of the actual and theoretical results about cutterhead accelerations: (a) Horizontal acceleration; (b) vertical acceleration; (c) axial acceleration.

2.2.5 Results comparison

After the influence of gravity acceleration was eliminated, the stable simulation and test results were intercepted, and the contrast results are presented in Fig. 12. At the same time, the rain flow counting method was applied to analyze the acceleration results, respectively, as shown in Fig. 13.

The acceleration mean values in each direction were all around 0 after elimination of gravity effect, so the amplitude fluctuation information was mainly concerned. Then further statistical analysis was carried out on the amplitudes in each direction, to obtain the mean value and mean square deviation, as the results shown in Table 5.

From the above comparison results in Fig. 12, first, the cutterhead acceleration simulation and test results in each direction basically show the same random change law, and the simulation results coincide with the test results to a great extent, which illustrates that there was a certain reliability and correctness about the established TBM mainframe system virtual prototype. Secondly, from the rain flow statistical results in Table 5, there was a certain error in the mean value of amplitude, and the vertical value was the maximum, with the relative error value of 39 %, axial value was the minimum, only 7 %. Similarly, the relative error of acceleration amplitude mean square deviation in vertical was also the maximum,

Table 5. The statistical results of test and theoretical cutterhead accelerations.

Index	Direction	Theoretical result ($m \cdot s^{-2}$)	Test result ($m \cdot s^{-2}$)	Relative error
Mean value of amplitude	Horizontal	0.97	0.72	35 %
	Vertical	1.04	1.71	39 %
	Axial	1.64	1.76	7 %
Mean square deviation of amplitude	Horizontal	0.81	0.75	8 %
	Vertical	0.85	1.88	55 %
	Axial	1.59	1.81	12 %

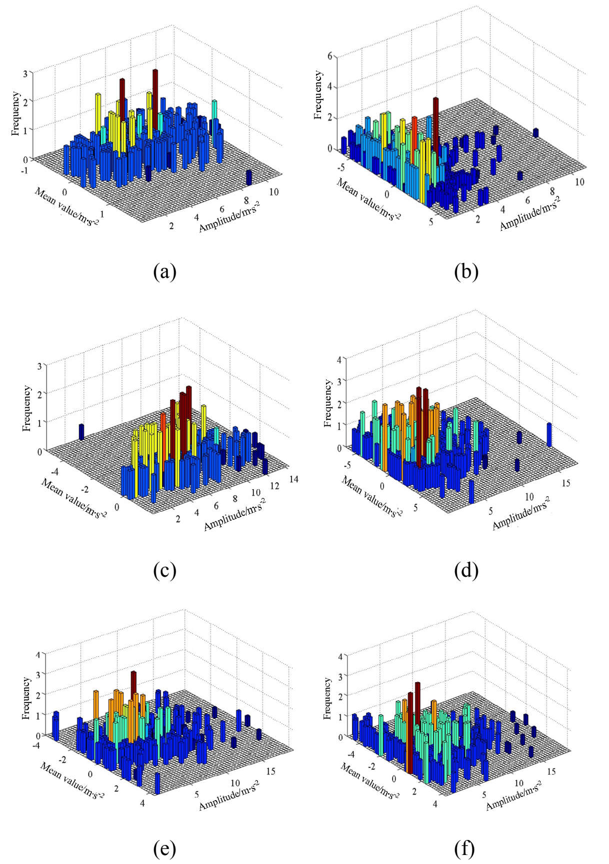


Fig. 13. The statistical results comparisons of cutterhead accelerations rain flow counting: (a) Theoretical result of horizontal direction; (b) test result of horizontal direction; (c) theoretical result of vertical direction; (d) test result of vertical direction; (e) theoretical result of axial direction; (f) test result of axial direction.

which was 55 %, then the horizontal and axial value, with the value of 8 % and 12 %, respectively. Finally, considering the complexity of the TBM cutterhead system structure and the worse environment at the driving site, many practical factors were difficult to take into account in the dynamic model, such as the influence of geology sudden change and vibration impact on test accuracy. The errors between simulation analysis and test results were unavoidable, and the statistical results showed that the two results can basically be kept at the same order of magnitude, with the similar change law, which further illustrated that the established analytical model had certain theoretical and practical significance.

Table 6. Table of factors and levels.

Level	Factor		
	Rotating speed *A* ($rev \cdot min^{-1}$)	Axial mean force *B* (kN)	Support stiffness of pinion *C* ($N \cdot mm^{-1}$)
1	3.6	125	3.6E6
2	5.6	250	3.6E7
3	7.6	375	3.6E8

3. Influence of vibration response parameters on TBM cutterhead system

3.1 Influence parameters selection

The selection of influence parameters can directly affect the rationale and correctness of the analysis results, so it is necessary to select the parameters which have significant influence on the cutterhead vibration. From the simulation comparison analysis it can be seen that the cutterhead rotating speed directly affects the cutterhead vibration, and the slight fluctuation of the rotating speed will aggravate the system vibration. Therefore, the cutterhead rotating speed can be considered as one of the influence parameters on cutterhead vibration.

During the actual tunneling process, the hard rock or soft soil often appears alternately on the cutterhead boring section, so the cutter loads increase or decrease sharply, which has great influence on the system vibration. Hence, the cutter loads can also be considered as one of the influence parameters on cutterhead vibration. Besides, the driving device may be damaged due to the severe impact of rock broken. For example, the bearing of a pinion is damaged due to the long term overloading. And then the bearing stiffness of the bearing is insufficient, which can also cause severe vibration of the cutterhead. Therefore, the support stiffness of the pinion is also considered as one of the influence parameters on cutterhead vibration.

According to the above analysis, the cutterhead rotating speed, axial mean force of cutter and the support stiffness of pinion were all selected as the influence parameters to carry out the cutterhead vibration analysis, which was denoted as *A*, *B* and *C*, respectively. Due to the device rate limitation, the change of the cutterhead rotating speed is not obvious, and the value of 2 rev/min was regarded as the variation in parameter influence analysis based on engineering experience. In addition, the nominal load 250 kN of 17 inch cutter was used as the benchmark for parameter *B*, and the value of 50 % was regarded as the variation, since the cutter load impact factor is frequently 1.5-times in engineering applications. Beyond that, the support stiffness of pinion varies with different damage degree, and the value of ten-times was regarded as the variation for parameter *C* in this study. Consequently, the three values were designed for each level of parameters, and the designed level table of factors is shown in Table 6, wherein the factor values in level 2 were the system parameter values under the above extreme condition.

Table 7. Orthogonal test scheme.

Sequence number	A	B	C	Empty column	Scheme
1	1	1	1	2	$A_1B_1C_1$
2	2	1	2	1	$A_2B_1C_2$
3	3	1	3	3	$A_3B_1C_3$
4	1	2	3	1	$A_1B_2C_3$
5	2	2	1	3	$A_2B_2C_1$
6	3	2	2	2	$A_3B_2C_2$
7	1	3	2	3	$A_1B_3C_2$
8	2	3	3	2	$A_2B_3C_3$
9	3	3	1	1	$A_3B_3C_1$

3.2 Orthogonal test design

There were three parameters and three levels in this study above-mentioned, for improving the analysis efficiency, an orthogonal experimental design method was used to carry out the test. Since it was an experiment with three factors and three levels, and the interaction between each factor was not considered, a standard orthogonal table $L_9(3^4)$ was chosen to meet the demands. Simultaneously, to improve the reliability of the analysis results, an empty column was added to the factor design, and the designed orthogonal test scheme is presented in Table 7, wherein each level is denoted as 1, 2 and 3, respectively, and the empty column level was assigned with a random value, which had no effect on the test scheme.

As can be known, the result of an orthogonal test is calculated based on a comprehensive scoring point. In this paper, the orthogonal test design was to calculate and analyze the mean value and range of the cutterhead vibration amplitude in the three translational directions, to estimate the optimization levels of the influence factors and obtain the optimum scheme.

3.3 Parameter influence analysis of vibration responses

Based on the above orthogonal test scheme, the corresponding three parameter values in the virtual prototype of TBM mainframe system were changed during the dynamic simulation, and the simulation time was defined as 40 s, the step was set to 8000. From the engineering statistics, the cutterhead radial vibration has a great influence on the main bearing and pinion support bearing, and excessive radial vibration may result in engineering failure, such as the bearing seal failure and roller raceway spalling. Accordingly, the influence of different parameters on the cutterhead radial vibration was analyzed. After the simulation was finished, the horizontal and vertical displacements of the cutterhead centroid were extracted from the post processing interface in ADAMS, and the results were statistically analyzed.

3.3.1 Horizontal vibration

Through simulation on different schemes, the horizontal displacements of cutterhead centroid were obtained by nine orthogonal test schemes, as shown in Fig. 14.

Table 8. Statistical results of horizontal vibration.

Sequence number	Factor				Mean square deviation /mm
	A	B	C	Empty column	
1	1	1	1	1	36.77
2	2	1	2	2	45.86
3	3	1	3	3	51.67
4	1	2	3	2	23.92
5	2	2	1	3	36.86
6	3	2	2	1	39.07
7	1	3	2	3	49.43
8	2	3	3	1	35.79
9	3	3	1	2	34.22
K_1	110.12	134.30	107.85	111.63	-
K_2	118.51	99.85	134.36	104.00	-
K_3	124.96	119.44	111.38	137.96	-
k_1	36.71	44.77	35.95	37.21	-
k_2	39.50	33.28	44.79	34.67	-
k_3	41.65	39.81	37.13	45.99	-
R	4.94	11.49	8.84	11.32	-
Influence order	$B>C>A$				
Optimum level	A_1	B_2	C_1	-	-
Optimum scheme	$A_1B_2C_1$				

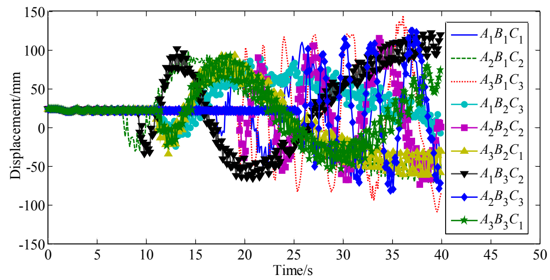


Fig. 14. Horizontal displacements about cutterhead centroid of each scheme.

The quantitative analysis results of each parameter were compared by using the direct analysis method in orthogonal experiment, to obtain the optimum scheme of cutterhead vibration. For analyzing the cutterhead vibration severity, the performance index of mean square deviation was employed. So the horizontal data on the above mentioned schemes were statistically analyzed, as the results listed in Table 8.

In Table 8, where K_j is the sum of test indexes at each level in column j factor (j = 1, 2, 3), k_j is the mean value of K_j, $k_j = K_j/3$, which can judge the factor optimum level in column j and R_j is the index range in column j factor, $R_j = \max(k_j)-\min(k_j)$, which can judge the influence order of each factor.

In the above analysis, the mean square deviation of cutterhead vibration was used as the evaluation index. The smaller the index value, the more stable the vibration in the direction,

Table 9. Statistical results of vertical vibration.

Sequence number	Factor				Mean square deviation /mm
	A	*B*	*C*	Empty column	
1	1	1	1	1	38.74
2	2	1	2	2	37.27
3	3	1	3	3	50.41
4	1	2	3	2	33.48
5	2	2	1	3	36.01
6	3	2	2	1	31.29
7	1	3	2	3	43.60
8	2	3	3	1	36.64
9	3	3	1	2	36.90
K_1	115.82	126.42	111.65	106.67	-
K_2	109.92	100.78	112.16	107.65	-
K_3	118.60	117.14	120.53	130.02	-
k_1	38.61	42.14	37.22	35.56	-
k_2	36.64	33.59	37.39	35.88	-
k_3	39.53	39.05	40.18	43.34	-
R	2.89	8.55	2.96	7.78	-
Influence order	$B>C>A$				
Optimum level	A_2	B_2	C_1	-	-
Optimum scheme	$A_2B_2C_1$				

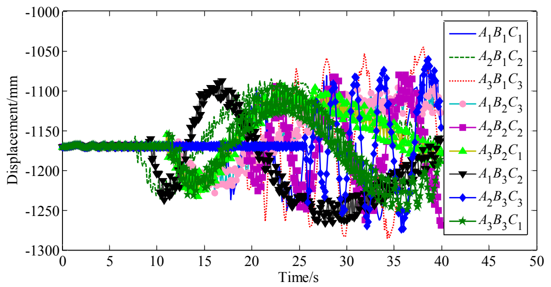


Fig. 15. Vertical displacements about cutterhead centroid of each scheme.

and the corresponding scheme is better. Hence, to obtain the scheme with stable vibration, the level value corresponding to minimum value of k_j in each factor should be selected. From Table 8, the range value of each factor is arrayed as follows: $R_2 > R_3 > R_1$, which illustrates that the influence order of the three factors on cutterhead horizontal vibration is *BCA*. In addition, the k_i value of each factor can be compared as follows, to the factor *A*: $k_1 < k_2 < k_3$, to the factor *B*: $k_2 < k_3 < k_1$, and to the factor *C*: $k_1 < k_3 < k_2$, which can identify the optimization level of each factor.

In summary, the preliminary conclusion is as follows: the influence of *B* on cutterhead horizontal vibration is greater than that of *C*, and the influence of *C* is greater than *A*. Thus, among the three influencing factors, the effect of cutter load on cutterhead vibration is the most obvious, and the influence of cutterhead rotating speed is the minimum. Besides, an optimum scheme is determined, that is $A_1B_2C_1$.

3.3.2 Vertical vibration

Similarly, the vertical displacements of cutterhead centroid with different schemes were obtained, as shown in Fig. 15.

Statistics were also conducted on the above schemes, with the results listing in Table 9.

Likewise, from Table 9, the range value of each factor is also arrayed as follows: $R_2 > R_3 > R_1$, and the influence order of the three factors on cutterhead vertical vibration is *BCA*. And the k_i value can also be compared as follows, to the factor *A*: $k_2 < k_1 < k_3$, to the factor *B*: $k_2 < k_3 < k_1$, and to the factor *C*: $k_1 < k_2 < k_3$. Then an optimum scheme of the cutterhead vertical vibration can be obtained, that is, $A_2B_2C_1$.

4. Conclusions

A dynamic model of the TBM mainframe system was established based on virtual prototype technology, and a TBM of an actual project was taken as an instance. After the dynamic model was simulated, the cutterhead translational vibration accelerations of each direction were obtained. Then a field test for measuring cutterhead accelerations was carried out, and the simulation results were compared with the test values for the model validation. Comparison results showed that the two results basically remained at the same order of magnitude, and the acceleration relative error in vertical direction was the maximum, that was only 55 %. And the simulation and test acceleration change regulation was similar, which illustrated that the established virtual prototype model had certain reliability and practical significance. Accordingly, based on the TBM mainframe virtual prototype model, an orthogonal test scheme with three factors and three levels was built, to analyze the influence of different parameters on cutterhead radial vibration. The analysis results showed that the influence of cutter force on cutterhead vibration was the most obvious among the three factors, and the influence of cutterhead rotating speed was not evident relatively. Meanwhile, the optimal schemes of cutterhead vibration in the horizontal and vertical directions were determined. Next, the research on the vibration of TBM reduced scale cutterhead in the laboratory will be carried out, to verify the rationality of the theoretical model and correct the model's parameter values.

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Nomenclature

A : Rotating speed
B : Axial mean force
C : Support stiffness of pinion
K : Test indexes sum of each level
k : Mean value of *K*
R : Range of *k*

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