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The Dynamic Comparative Analysis between Two TBM Main System Structures

Junzhou Huo[1], Hanyang Wu[1], Jing Chen[2*], Jingxiu Ling[1] and Xiaolong Sun[1]

[1]*School of Mechanical Engineering, Dalian University of Technology, Dalian116024, China*

[2]*School of Naval Architecture and Ocean Engineering, Dalian Ocean University, Dalian 116023, P.R. China*

Abstract

Excessive vibration and vibration damage are serious engineering problems in the TBM boring process. TBM dynamic characteristic analysis is essential to the anti-vibration design of the TBM mainsystem. This paper classifies the TBM thatmostly used in the practical engineering into TBM 1 and TBM 2, and comprehensively analyses the difference of cutter layout strategy, pinion layout strategy, shield support structure, etc between TBM 1 and TBM 2. A dynamic comparative analysis of TBM cutterhead was carried out based on a TBM dynamic model. The analysis results show that the cutterhead horizontal vibration of TBM 1 is around 0.3mm while that of TBM 2 is around 1.4mm.The vibration periodicity of TBM 2 significantly reduces. The influence of shield vertical support on the horizontal and vertical vibration of TBM 2 is analyzed. The results show that settinga vertical support at the bottom of the shield reduces the TBM cutterhead vibration by about 12%.

Keywords: *Comparative analysis; Dynamic model; Structure parameters*

1. Introduction

In hard rock, the construction process with big torque, thrust and large impact load. The excessive vibration of TBM will cause non-normal damage in critical components and shorten the life of TBM [1-2]. Among the component of TBM, the vibration situation of cutterhead is the most serious [3].This paper classifies the TBM that mostly used in the practical engineering into TBM 1 and TBM 2. The comparative vibration analysis of TBM 1 and TBM 2 can accurately simulate the vibration of each component. These analysis results determine the reasonable structure and provide a theoretical reference for TBM anti-vibration design.

Some foreign and domestic scholars have analyzed the TBM driving rotary system and TBM cutterhead system from the angle of dynamic vibration [4]-[6].K.Z.Zhang [7] *et. al.*, established a coupling dynamical model of shield machine considering redundant drive system, hydraulic propulsion system, geological conditions, *etc.*, and the dynamical characteristics of the rotary system was studied based on the dynamical model. J. X. Lin and W. Sun [8] established a nonlinear dynamical model of cutterhead system, and analyzed the dynamical characteristics of cutterhead. The comparative analysis based on the TBM 1 and TBM 2 in engineering is not that deep. By considering the difference of cutter layout strategy, pinion layout strategy, shield support structure, *etc.*, between TBM 1 and TBM 2, this paper determines more reasonable shield support structure, pinion layout strategy, *etc*.

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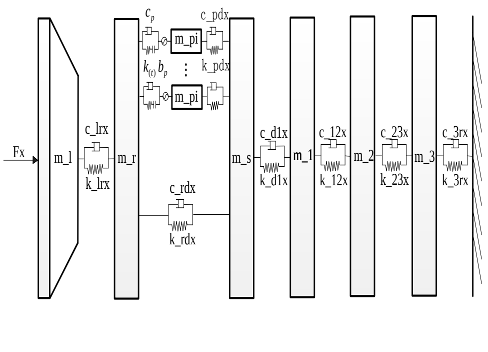
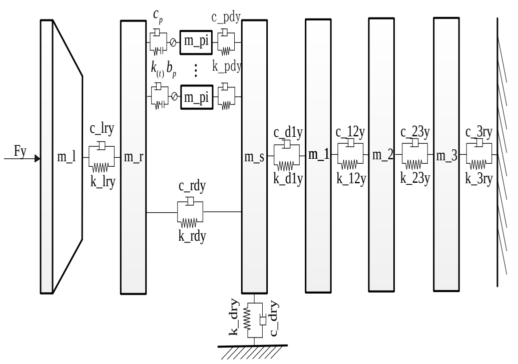
2. Comparative Analysis of Parameters

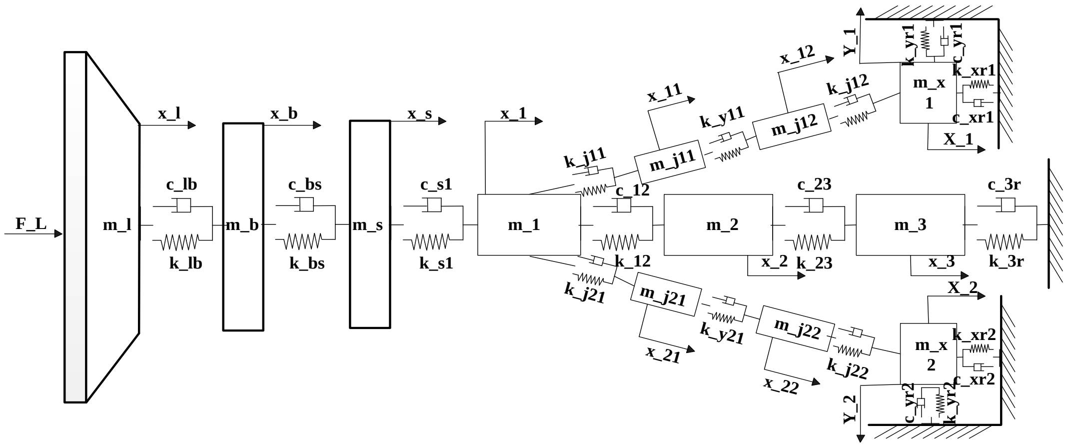
2.1 The Dynamic Model of the Two TBM

The dynamic model [9] of TBM is shown in Figure 1.

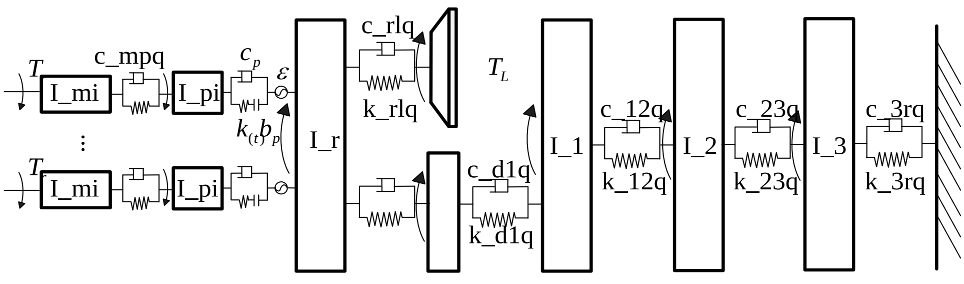




(a) The Horizontal Degree of Freedom (b)The Vertical Degree of Freedom



(c)The Axial Degree of Freedom



(d)The Torsional Degree of Freedom

Figure 1. The Dynamic Model of TBM

In dynamic model, $m_l, m_b, m_s, m_1, m_2, m_3, m_r, m_{pi}$ represent the mass of cutterhead, cutterhead bearing, cutterhead support, front frame, mid frame, end of frame, bull gear and each pinion. $m_{j11}, m_{j12}, m_{j21}, m_{j22}$ represent the mass of 4 hinges as shown in Figure 3 (c). m_{x1}, m_{x2} represent the mass of 2 gripper shoes as shown in Figure 3 (c). I_{mi}、 I_{pi}、 I_r、 I_l represent the rotary inertia of each motor, each pinion, bull gear and

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cutterhead. k_{j11}、k_{j12}、k_{j21}、k_{j22} represent 4 stiffness of the corresponding hinges. k_{xr1}、k_{yr1}、k_{xr2}、k_{yr2} represent the horizontaland axial support stiffness of the corresponding gripper shoes. k_{lb}、k_{bs}、k_{s1}、k_{12}、k_{23}、k_{3r} represent the structural stiffness of cutterhead, cutterhead bearing, cutterhead support, front frame, mid frame, end of frame. k_{lrx}、k_{rdx}、k_{pdx}、k_{d1x}、k_{12x}、k_{23x}、k_{3rx} represent the horizontal structural stiffness of cutterhead, bull gear, cutterhead support, front frame, mid frame and end of frame. k_{lry}、k_{rdy}、k_{pdy}、k_{d1y}、k_{12y}、k_{23y}、k_{3ry} represent the vertical structural stiffness of cutterhead, bull gear, cutterhead support, front frame, mid frame and end of frame. k_{mpq}、k_{rlq}、k_{d1q}、k_{12q}、k_{23q}、k_{3rq}、$k(t)$ represent the torsional stiffness of transmission shaft, cutterhead, front frame, mid frame, end of frame and the time-varying damping stiffness. T_L、T_{pi}、F_x、F_y、F_L represent the load torque on cutterhead, the input torque of motor, the horizontal unbalanced force on cutterhead, the vertical unbalanced force on cutterhead and the axial force on cutterhead.

2.2 Different Cutter Layout Strategy

Different cutter layout strategies result in different equivalent force on cutterhead during boring process [10-11].The load on cutterhead contains axial force P_v, radial force P_r and torque T as shown in Figure 1. Each of the load is a resultant force which can be calculated according to the layout parameter of each cutter. The relationship between cutter layout position and angle is shown in Figure 2. In the figure, l represents the distance between each cutter axis and the center of cutterhead, θ represents the phase angle of each cutter on the cutterhead, β represents the angle between the vertical force of each cutter and the Y-axis, especially for the inner cutter and center cutter, the value of $\beta$is 0.

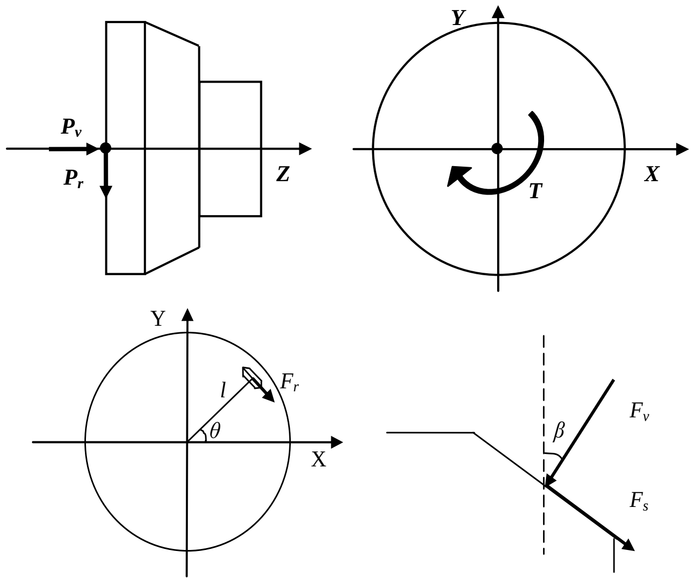

Figure 2. The Cutter Layout Position

The Axial Force on Cutterhead

The axial force on cutterhead is the sum of the Z component force on each cutter. The calculation formula is $\vec{P}_v = \sum_{q=1}^{t} F_{vq}(\cos\beta_q + k_s \sin\beta_q)$, where F_{vq} is the vertical force of the cutter numbered q,t is the number of cutter on cutterhead, $\beta_q$is the angle between vertical force of the cutter q and Y-axis.

The Radial Force on Cutterhead

The radial force on cutterhead is the sum of the Xand Y component force on each cutter. The calculation formula is $P_r = \sqrt{(\overline{F}_{vx\Sigma} + \overline{F}_{rx\Sigma} + \overline{F}_{sx\Sigma})^2 + (\overline{F}_{vy\Sigma} + \overline{F}_{ry\Sigma} + \overline{F}_{sy\Sigma})^2}$, where $F_{vx\Sigma}$ 、$F_{rx\Sigma}$ and $F_{sx\Sigma}$ are the sun of the X component force of the vertical force, rolling force and side force on each cutter. $F_{vy\Sigma}$ 、 $F_{ry\Sigma}$ and $F_{sy\Sigma}$ are the sun of the Y component force of the vertical force, rolling force and side force on each cutter.

$$\begin{cases} \vec{F}_{vx\Sigma} = \sum_{q=1}^{t} F_{vq} \sin\beta_q \cos\theta_q \\ \vec{F}_{rx\Sigma} = k_r \sum_{q=1}^{t} F_{vq} \sin\theta_q \\ \vec{F}_{sx\Sigma} = k_s \sum_{q=1}^{t} F_{vq} \cos\beta_q \cos\theta_q \end{cases} \quad \begin{cases} \vec{F}_{vy\Sigma} = \sum_{q=1}^{t} F_{vq} \sin\beta_q \sin\theta_q \\ \vec{F}_{ry\Sigma} = k_r \sum_{q=1}^{t} F_{vq} \cos\theta_q \\ \vec{F}_{sy\Sigma} = k_s \sum_{q=1}^{t} F_{vq} \cos\beta_q \sin\theta_q \end{cases}$$

The Torque on Cutterhead

The torque on cutterhead is the sum of all moments about Z-axis. The calculation formula is $\vec{T} = k_r \sum_{q=1}^{t} F_{vq} l_q$.

According to the layout parameters, the equivalent load on cutterhead of TBM 1 and TBM 2 can be calculated. The calculation results are shown in Table 1.

Tabe 1. The Equivalent Load on Cutterhead

	TBM 1	TBM 2
Axial force	14728359.28N	14587213.69N
Horizontal force	16885.80202N	88658.99331N
Vertical force	293285.8923N	44573.92857N
Torque	5640690N/m	5771487.9N/m

The loads are taken as external input to the dynamic model.

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2.3. The Pinion Layout Strategy

The TBM 1 and TBM 2 main drive system all contain 10 parallel pinions. The pinion layout strategy of TBM 1 is shown in Figure 3 while that of TBM 2 is shown in Figure 4.

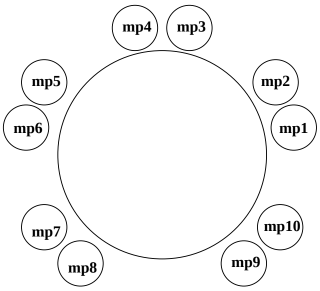

Figure 3. Pinion Layout Strategy of TBM 1

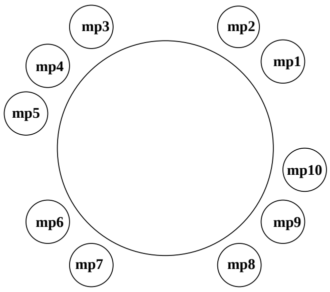

Figure 4. Pinion Layout Strategy of TBM 2

The phase angle of each pinion is as shown in Table 2.

Table 2. The Phase Angle of Each Pinion

pinion	TBM 1	TBM 2
1	11.25	34.67
2	33.75	57.35
3	78.75	122.34
4	101.25	145.51
5	146.25	168.24
6	168.75	214.74
7	213.75	237.32
8	236.25	303.02
9	303.75	325.08
10	326.25	348.24

2.4. The Mass of the Components

The mass of main components are listed in Table 3.

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Table 3. The Mass of Main Components

	TBM 1	TBM 2
Cutterhead	150t	193t
Front frame	37.674t	21.67t
Mid frame	22.546t	21.67t
End of frame	18.919t	21.67t
Gripper shoe	2.058t	2.25t

The Horizontal and Vertical Coupling Stiffness of Main Frame

The different parameters of the flange have great influence on the horizontal and vertical coupling stiffness of main frame. The coupling stiffness is calculated by following formula [9]. $k = Z(\frac{9EI'}{L^3} + \frac{3EA}{L})$ $I' = \frac{hb^3}{12}$ $A = bh$. In the formula, Z represents the number of lamination, L represents the distance between bolt holes and the center of flange, b represents the minimum width of the lamination, h represents the thickness of the lamination. The flange parameters of main frame are listed in Table 4.

Table 4.The Flange Parameters of Main Frame

	TBM 1	TBM 2
The type and number of link bolt	M36 48	M42 66
The section parameters of flange	1191x1319, 1722x2150	3100x3245mm
The thickness of flange	100mm	100mm

The main frame section of TBM 1 is a square while that of TBM 2 is a circle. The intermediate parameters for calculating the stiffness are listed below.

Table 5. The Intermediate Parameters and the Coupling Stiffness

	TBM 1	TBM 2
L	133mm	150mm
b	265.5mm	72.5mm
h	100mm	100mm
I'	$1.56 \times e^{-4}$	$3.18 \times e^{-6}$
A	0.02655	0.00725
The coupling stiffness	$3.2 \times e^{11}$	$6.446 \times e^{10}$

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According to the calculation results, the horizontal and vertical coupling stiffness of TBM 1 main frame are $3.2 \times e^{11}$, the horizontal and vertical coupling stiffness of TBM 2 main frame are $6.446 \times e^{10}$. These values of coupling stiffness are taken as dynamic parameters in the model.

2.5. The Cutterhead Flange Radius

Cutterhead flange radius greatly affects the connection strength between cutterhead and the bull gear. The horizontal and vertical coupling stiffness are mainly determined by the cutterhead flange radius. The stiffness is calculated by the formula described in 2.6. The flange parameters are listed in Table 6.

Table 6. The Cutterhead Flange Parameters

	TBM 1	TBM 2
Cutterhead flange radius	IØ4166mm/OØ4900mm	IØ3537mm/OØ4320mm
Cutterhead flange mass	4113kg	3803kg
The number of bolt	72	76

The intermediate parameters for calculating the stiffness are listed below.

Table 7. The Intermediate Parameters and the Coupling Stiffness

	TBM 1	TBM 2
L	197.8mm	162.4mm
b	367mm	391.5mm
h	100mm	100mm
I'	$4.12 \times e^{-4}$	$5.00 \times e^{-6}$
A	0.0367	0.03915
The radial coupling stiffness	$4.35 \times e^{11}$	$3.08 \times e^{11}$

According to the calculation results, the horizontal and vertical coupling stiffness of TBM 1 cutterheadare $4.35 \times e^{11}$, the horizontal and vertical coupling stiffness of TBM 2 main frame are $3.08 \times e^{11}$. These values of coupling stiffness are taken as dynamic parameters in the model.

2.6 The Shield Support Type

There are 3 supports at the shield of TBM 1 while there is one vertical support at the shield of TBM 2. This paper assumes that the pressure of the gripper shoe is directly proportional to the support stiffness. In dynamic calculation, the pressure of the gripper

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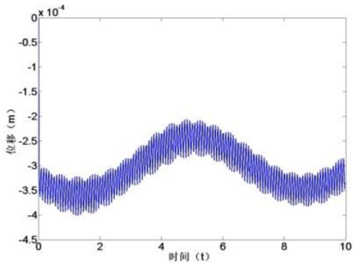
shoe at main frame is 2.95MPa, and the support stiffness is $1 \times e^{11}$ N/m. The pressure of the gripper shoe at shield is 1.2Mpa, therefore, the corresponding support stiffness is $4.06 \times e^{10}$ N/m according to the proportional relation. The horizontal and vertical stiffness of shield is defined by the sum of structural stiffness and support stiffness: $K = K_{\sup\ port} + K_{structure}$. The structural stiffness of shield $5 \times e^{10}$ N/m. The horizontal and vertical stiffness of shield is listed in Table 8.

Table 8. The Horizontal and Vertical Stiffness of Shield

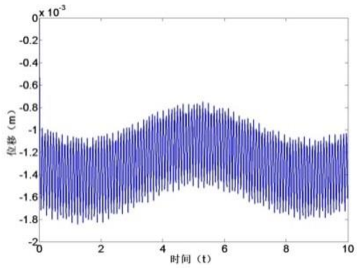
	TBM 1	TBM 2
Equivalent horizontal support stiffness	$7.03 \times e^{10}$	0
Equivalent vertical support stiffness	$8.12 \times e^{10}$	$4.06 \times e^{10}$
Equivalent horizontal stiffness of shield	$1.2 \times e^{11}$	$5 \times e^{10}$
Equivalent vertical stiffness of shield	$1.3 \times e^{11}$	$9 \times e^{10}$

3. The Analysis Results

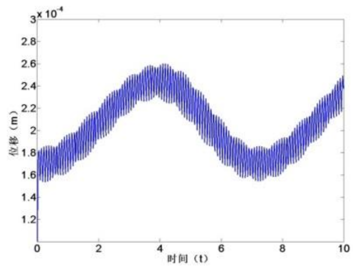
The above parameters are taken as dynamic parameters, and the vibration responses are obtained. Take the vibration responses of cutterhead inaxial, horizontal, vertical and torsional DOF as example. The analysis results are shown in Figure 5.



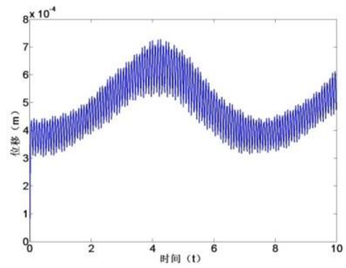
The Horizontal Cutterhead Vibration of TBM 1



The Horizontal Cutterhead Vibration of TBM 2



The Vertical Cutterhead Vibration of TBM 1



The Vertical Cutterhead Vibration of TBM 2

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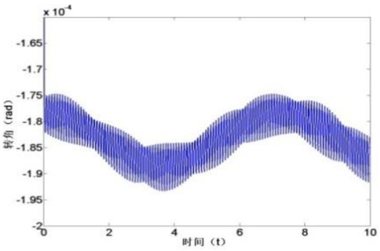
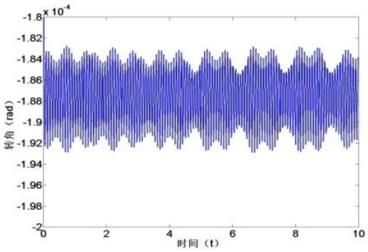
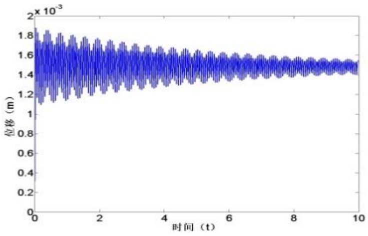


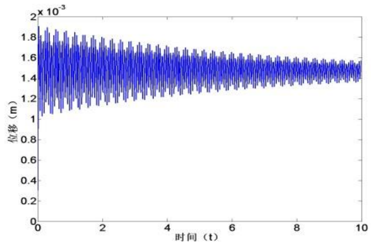
The Torsional Cutterhead Vibration of TBM 1



The Torsional Cutterhead Vibration of TBM 2



The Axial Cutterhead Vibration of TBM1

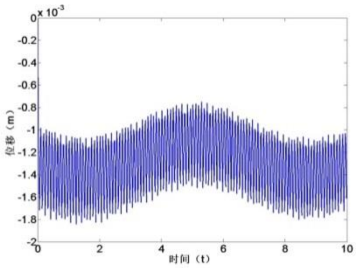


The Axial Cutterhead Vibration of TBM 2

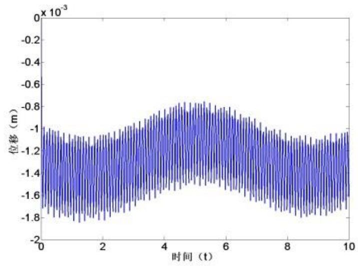
Figure 5. The Analysis Results

It can be seen that:(1)the cutterhead vibration of TBM 1 is significantly smaller than the TBM 2 in horizontal and vertical DOF. The mean of the horizontal cutterhead vibration of TBM 1 is around 0.3mm while that of the TBM 2 is 1.4mm which is 4 times of the former one. The mean of the vertical cutterhead vibration of TBM 1 is around 0.2mm, and that of the TBM 2 is 0.5mm. (2) The horizontal vibration situation of TBM 1 is similar to the vertical one. The horizontal vibration situation of TBM 2 is obviously more serious than the vertical one. (3) The torsional cutterhead vibration of TBM 1 is similar to the TBM 2. The maximum vibration amplitude reaches $1.8 \times 10^{-4} rad$. The vibration periodicity of TBM 2 significantly reduces. (4) The aixalcutterhead vibration of TBM 1 is similar to the TBM 2, and the maximum vibration reaches around 1.6mm.

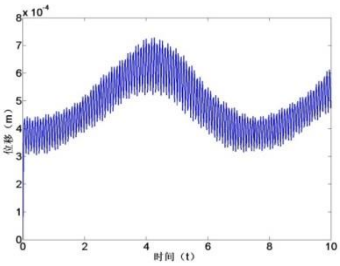
In order to further analyze the influence of shield vertical support on the horizontal and vertical vibration of TBM 2, this paper calculates a dynamic model with unbraced structure. The analysis results are shown in Figure 6.



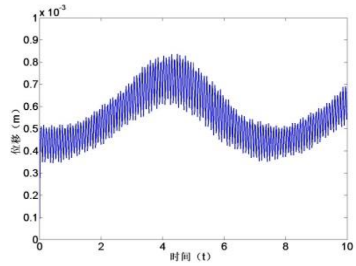
Horizontal Vibration of TBM 2 with Braced Structure



Horizontal Vibration of TBM 2 with Unbraced Structure



The Vertical Vibration of TBM 2 with Braced Structure



The Vertical Vibration of TBM 2 with Unbraced Structure

Figure 6. The Vibration Situation of TBM with Braced and Unbraced Structure

The statistical parameters of the cutterhead vibration is listed in Table 9.

Table 9. The Statistical Parameters of the Cutterhead Vibration

	TBM 2 with braced structure	TBM 2 with unbraced structure
The mean of horizontal vibration	-0.0013	-0.0013
The amplitude of horizontal vibration	0.0018	0.0018
The mean ofvertical vibration	4.7398×10^{-4}	5.4215×10^{-4}
The amplitude ofvertical vibration	7.2861×10^{-4}	8.3630×10^{-4}

It can be seen that: (1) The vertical support has no obvious influence on the horizontal vibration of cutterhead. (2) The vertical support affects the vertical vibration of cutterhead to some extent. The mean of vertical vibration reduces by12.57%, and the amplitude of vertical vibration reduces by12.88%.

4. Conclusion

The cutterhead dynamic responses of TBM 1 and TBM 2 are obtained by considering the parameters and structure of these two TBM. The results show that the mean of the horizontal cutterhead vibration of TBM 1 is around 0.3mm while that of the TBM 2 is 1.4mm which is 4 times of the former one. The mean of the vertical cutterhead vibration of TBM 1 is around 0.2mm, and that of the TBM 2 is 0.5mm. It indicates that the shield support structure of TBM 1 is more reasonable than TBM 2. A dynamic model with unbraced structure is analyzed and the results show that a vertical shield support can reduce the cutterhead vertical vibration by about 12%.

Acknowledgment

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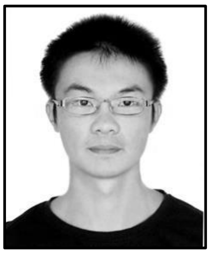
Authors

JunzhouHuo, he is currently a master supervisor at the School of Mechanical Engineering at Dalian University of Technology in China. Huo's research interests include layout optimization and TBM cutter head design.



Hanyang Wu, is currently a master at the School of Naval Architecture Engineering at Dalian University of Technology in China. Wu's research interests include optimization design.

Jing Chen, she is currently an assistant professor in the School of Naval Architecture and Ocean Engineering in Dalian Ocean University. Her current research interests include ships' subdivision design, loading design, stability and strength design, and their applications in ship design and shipping.

Jingxiu Ling, he is currently a Ph.D. candidate at the School of Mechanical Engineering at Dalian University of Technology in China. Ling's research interests include system dynamics and life prediction of TBM cutter head.

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