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Determination of the cutterhead torque for EPB shield tunneling machine

Hu Shi, Huayong Yang, Guofang Gong *, Lintao Wang

State Key Laboratory of Fluid Power Transmission and Control, Zhejiang University, Hangzhou 310027, China

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ABSTRACT

Cutterhead torque is an important parameter for the design and operation of earth pressure balance (EPB) shields. Based on the analysis of several completed project cases from job sites, the conventional torque determination model based on experimentation proves rough enough to be improved. Composition and corresponding calculation method of cutterhead torque are presented, taking into account of cutterhead structure, cutting principle and the interaction between cutterhead and soil. Considering a Φ1.8 m EPB test machine in the lab, theoretical calculation following the improved model and test are carried out with three typical types of soils. Calculation and test results indicate that the cutterhead torque varies with geological conditions apparently, and the opening ratio of the cutterhead as well as earth pressure turns out to be the two most important factors in determining the cutterhead torque. The test results also show that the torque calculation formula for EPB shield tunneling can reasonably predict the excavation torque required by the cutterhead in clay soil tunneling, but for cohesionless tunneling, soil conditioning reduces the amount of torque necessary.

1. Introduction

Rapid economic development and urban population growth have been increasing the necessity for underground space exploration and utilization due to the need of upgrading and expanding the existing infrastructures. Tunneling plays a very important role in the underground engineering, providing a premium solution for those needs with minimum surface impacts [1]. Of all tunneling methods, EPB tunneling performed by EPB shield tunneling machines has attained the most extensive application due to its ability to adapt to a variety of geological conditions and discharge control. So EPB shield tunneling machine is of great importance to the tunnel construction for subway, highway, etc.

The two important tasks of the EPB machine are the cutting of frontal soils and face support with excavated soil by the cutterhead [2]. Because of this special task, it consumes a vast amount of energy accounting for more than half of the total required power of the machine. Therefore, when designing an EPB tunneling machine, more attention should be paid to the cutterhead drive and associated soils to determine the necessary power requirements. It is essential for engineers not only to estimate the loads, but also to know which factors may affect the loads [3].

Rotational speed and torque are two critical parameters of the cutterhead drive, and they are directly related to drive power. The former one is to be controlled as a constant during tunneling in a homogeneous layer while the latter varies with the different geological conditions. The torque capacity of cutterhead has to be considered in the design stage, taking into account a range of soils faced by the machine.

This paper gives a review on the empirical formula for calculating the cutterhead torque, then analyzes the factors resulting in the load torque and creates the mathematical models. Based on the theoretical modeling, the experiments are carried out on a tunneling test rig to verify the torque determination method for cutterhead drive of EPB shield tunneling machine.

2. Conventional model

At present, the torque equipped for cutterhead is empirically determined in terms of the diameter of shield machine [4]. The formula widely adopted by many designers is described as follows.

$$T = \alpha D^3 \tag{1}$$

where T is the provisional cutterhead torque (ton-m), D is the shield machine diameter (m), α is an empirical coefficient. For the EPB shield, it requires an α of 1–2.5. To clearly illustrate the relations between torque and diameter, the above equation is also shown in Fig. 1.

From Fig. 1, it can be concluded that the empirical equation just gives a quite rough estimate of the torque, as a reference for system design. In other words, the torque value may vary within the hatched area shown in Fig. 1, following variables α and D in design. Take the widely used Φ 6m EPB shield in metro tunnel construction as an example, its cutterhead torque can have an indeterminate value

* Corresponding author.
E-mail address: gfgong@zju.edu.cn (G. Gong).

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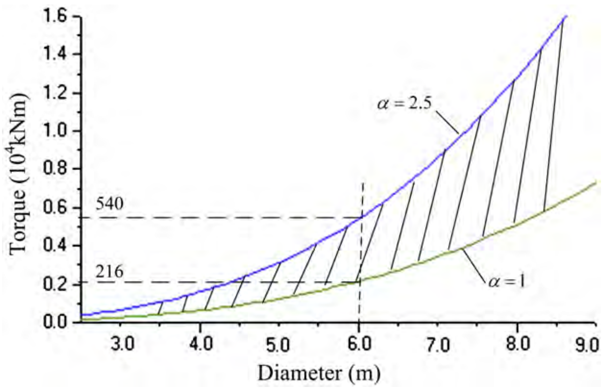

Fig. 1. Torque/diameter characteristics of EPB shield.

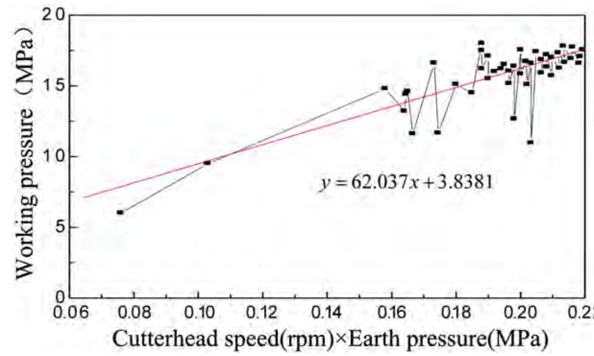

Fig. 2. Relation between torque and product of speed and earth pressure.

ranging from 2160 kNm to 5400 kNm, which makes design work blind to some extent. Therefore, it is unreasonable to conclude that α and D are the only vital ingredients to determine the equipped torque of EPB shield cutterhead.

In fact, from a number of finished construction projects as shown in Table 1, it can be seen that not only the diameter but other factors such as overburden depth and opening of cutterhead influence the torque [5,6]. Moreover, α is variable even with the same diameter and in the same soil layer, calculated through the empirical equations. The empirical coefficient α in different sites turns out to be varying between 0.48 and 3.69 according to the given projects listed in Table 1, far beyond the recommended rage. Thus the range of 1–2.5 for α is unnecessarily effective to all kinds of projects.

In addition, to investigate other tunneling parameters which may affect the cutterhead torque, a set of in situ data obtained from a metro construction site is processed and analyzed [7]. As shown in Fig. 2, take the working pressure of cutterhead hydraulic system which is proportional to the cutterhead torque and the product of cutterhead speed and earth pressure in the chamber as ordinate and abscissa respectively, the relations between them is obviously revealed. It seems as if there exists linearity. Fitting those data with a straight line, an equation is derived with a relatively satisfactory goodness of fit shown in Fig. 2.

The above data fitting shows good linearity, it provides another right reason for questioning the empirical formula. It is necessary to analyze the specific elements making up of the total cutterhead torque.

3. Composition of torque

It is estimated that several factors are responsible for the resistance torque applied on the cutterhead during tunneling, including soils, cutter frame, overburden depth, additives and other aspects.

1) Geological conditions. The EPB shield tunneling usually encounters complicated strata. Properties of different soils lead to various acting loads during cutting. For instance, it is well known that cutting in hard soil consumes much higher torque than in loose sand. Furthermore, even in the same cutting sectional face, the cutting torque may change randomly resulting from uneven distribution of soil characteristics [8].
2) Cutterhead shape and size. The opening ratio is a very important parameter which means the area ratio of opening to full face. Shield cutterhead may have small or large ratio with corresponding soil layers [9], as shown in Fig. 3. Different opening ratios decide different acting areas applied on the cutterhead by soil, resulting in different friction torques. The cutterhead diameter is also directly decisive of the cutting torque.
3) Overburden depth. The earth pressures acting on the cutterhead face and in the working chamber are proportional to the soil depth above the EPB shield, which cause the cutting resistance.
4) Additives. Adding additives can make the soil plasticized thereby facilitate the cutting process and reduce the friction [10,11].

Fig. 4 shows a typical structure of cutterhead of EPB shield. Based on the interaction between cutterhead and soil, the torque model is built as follows.

3.1. Friction torque on frontal surface

When EPB shield is advancing, the frontal face of the cutterhead resists the earth pressure from the soils against it. This applied pressure expressed in Eq. (2) causes the friction torque as the cutterhead is rotating.

$$p = K_0 \gamma H \tag{2}$$

where γ is the volume weight, H is the overburden depth, and K_0 is the coefficient of lateral earth pressure concerned with the property of soil. Fig. 5 gives the diagram of acting forces on cutterhead. Based on the diagram, the torque calculation is carried out.

Suppose that the frontal face of cuttterhead is normal to horizon, and neglect the extra load on the ground. Consider an acting point on the frontal face, note the angle of the point with respect to central

Table 1
Comparisons of α in different construction projects.

Projects	Soil classification	Depth (m)	Diameter (m)/opening (%)	α (T/D^3)	Torque (10^4 kNm)
Line 4 of Shanghai, China	Silty clay	9.0	6.34/40	0.70	177.4
A tunnel in Shanghai, China	Silty clay	9.72	7.65/60	0.91	232
Line l4 of Shanghai, China	Silty clay	14.24	6.34/40	1.22	310.4
Line 6 of Shanghai, China	Silty clay	14.38	6.34/30	0.48	121.2
Line M8 of Shanghai, China	Silty clay	14.948	6.34/40	0.50	128.6
Extended section of Line 2, Shanghai, China	Clay, silty clay	17.487	6.34/30	1.07	271.9
Line 12 of Tokyo metro, Japan	Gravel	14–24	8.66/43	0.92–1.69	600–1100
Section I of Sendai electric power tunnel, Japan	Sandy gravel	–	2.83/34	0.49–3.69	11.0–83.7

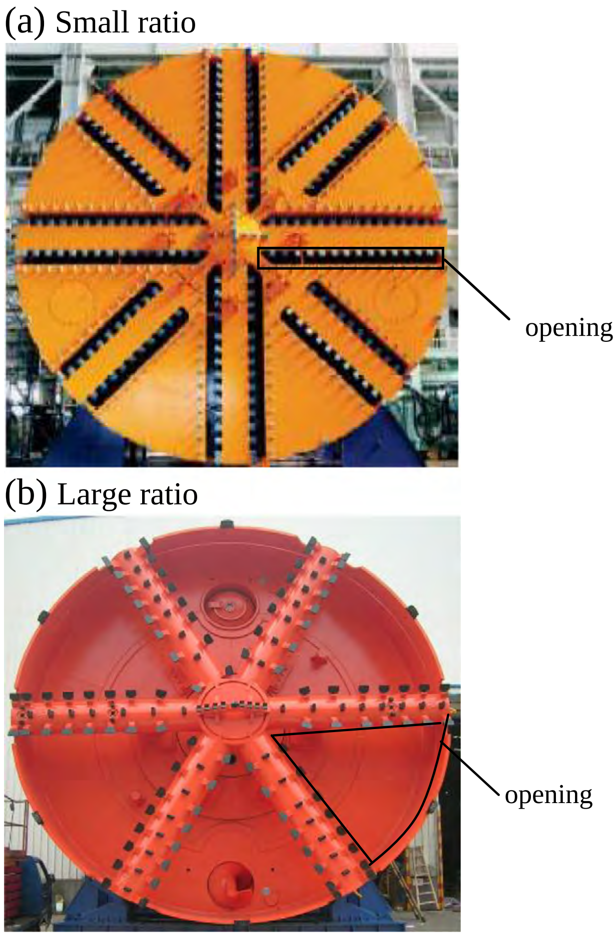

Fig. 3. Cutterhead of different opening ratios.

horizontal as θ, the radial distance between the point and the cutterhead center as r. By means of integration, the friction torque on the frontal face is obtained as:

$$\begin{aligned} T_1 &= \int_0^{2\pi}\int_0^{\frac{D}{2}} K_0 f\gamma(H-r\sin\theta)r^2 dr d\theta \\ &= \frac{\pi D^3}{12}K_0 f\gamma H \end{aligned} \tag{3}$$

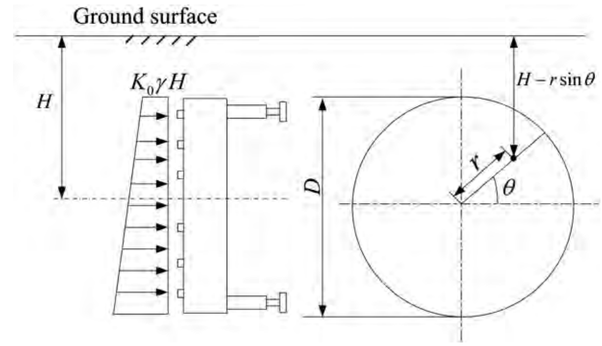

Fig. 5. Diagram of forces acting on the cutterhead.

where f is the coefficient of dynamic friction. Practically, considering the influence by the opening of cutterhead, the above equation is modified as:

$$T_1 = \frac{\pi D^3}{12}K_0 f\gamma H(1-\eta) \tag{4}$$

where η is the opening ratio of cutterhead. Here, the viscous friction is omitted given that the speed of cutterhead is very low. At the speed usually less than 2 r/min, the viscous friction is trivial compared with the Coulomb friction.

3.2. Friction torque on circular surface

The friction torque on circular surface is induced by the earth pressure composed of two parts: vertical component p_1 and lateral component p_2, as shown in Fig. 7. They are represented by the following equations:

$$p_1 = \gamma\left(H - \frac{D}{2}\sin\theta\right) \tag{5}$$

$$p_2 = \gamma K_0\left(H - \frac{D}{2}\sin\theta\right). \tag{6}$$

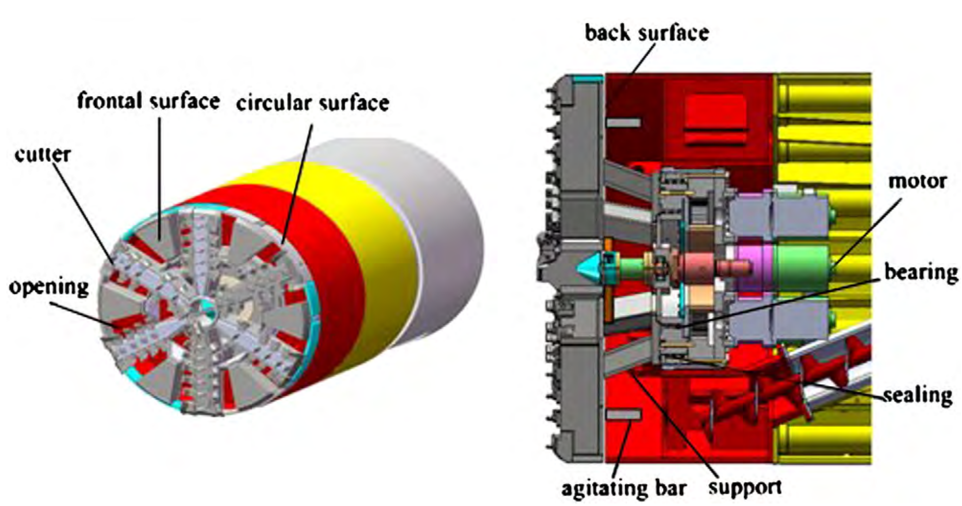

Fig. 4. Structure of cutterhead of EPB shield.

Accordingly, the friction torques generated due to vertical pressure and lateral pressure can be derived with integral calculation respectively as:

$$T_{21} = \int_0^{2\pi} \frac{D^2}{4} f W p_1 \ \sin^2\theta d\theta \tag{7}$$

$$T_{22} = \int_0^{2\pi} \frac{D^2}{4} f p_2 W \ \cos^2\theta d\theta \tag{8}$$

where W is the width of cutterhead. The total friction torque applied on the circular surface will be:

$$T_2 = T_{21} + T_{22} = \frac{\pi D^2}{4}(1+K_0) f \gamma H W. \tag{9}$$

3.3. Friction torque on back surface

Similar to T_1 in calculation, the friction torque applied on the back surface of the cutterhead is dependent on the earth pressure in the EPB chamber. The earth pressures in and out of the chamber should be balanced, as much as possible. In fact, to make sure that the soils cut down can enter into the chamber easily then be transported out, the inner pressure is slightly lower than the outer. Opening ratio is a vital factor influencing the setting of inner earth pressure, because it decides the contact area of soils in chamber and soils being cut. Apparently, the large opening makes earth pressure in the chamber closer to that of the cutting face. The friction torque on back surface can be described as:

$$T_3 = \frac{\pi D^3}{12} K_0 f \gamma H (1-\eta) f_{\Delta p} \tag{10}$$

where $f_{\Delta p}$ is the coefficient related to the difference between inner and outer pressures, it can be approximately seen as 1 in good earth pressure balance condition.

3.4. Cutting torque

EPB shield is widely used in soft ground tunneling, so the cutting process of soft ground is dealt with here. Usually, there are many cutters orderly distributed on the cutterhead. The total cutting torque is composed of the torque applied on every cutter. Fig. 6 shows the cutting process in sectional view.

When the EPB shield is advancing, the cutters installed on the rotating cutterhead apply cutting forces in both axial and radial directions on the soils. The cutters will suffer from resistance forces from the soil stratum in reverse. Neglect the friction on the side of the cutter, the cutting torque will be:

$$T_4 = \sum_{i=1}^{m} F_{ci} L_i \tag{11}$$

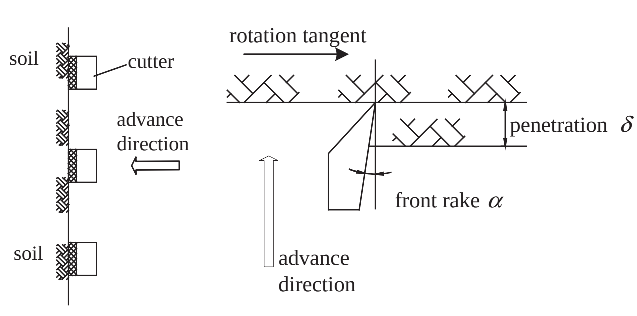

Fig. 6. Schematic of cutting process.

where F_{ci} is the resistance force applied on the cutter i, L_i is the distance between the cutter i and the center of cutterhead, m is the number of the cutter fixed on cutterhead. Suppose that v is the thrust speed of EPB shield and n is the rotational speed of cutterhead, then the cutting depth per revolution for cutterhead is:

$$t = \frac{v}{n} \tag{12}$$

and the cutting depth of one cutter can be:

$$\delta t = \frac{\beta}{360} \cdot t \tag{13}$$

where β is the angle between two adjacent cutters in the circular direction. The shear area of a cutter during cutting is:

$$A = w \delta t \ \tan\alpha = w \frac{\beta}{360} \frac{v}{n} \ \tan\alpha \tag{14}$$

where w is the width of cutter, α is the front rake. As can be seen from Fig. 7, the earth pressure applied on the cutter i can be written as:

$$\sigma_i = K_0 \gamma (H - L_i \ \sin\theta_i) \tag{15}$$

where θ_i is the angle of cutter i with respect to the horizontal plane. The shear strength of the soil around cutter i is:

$$\tau_i = \begin{cases} c + \sigma_i tg\varphi \text{ when soil is cohensive,} \\ \sigma_i tg\varphi \text{ when soil is cohensionless} \end{cases} \tag{16}$$

where c is the cohesion of soil. Thus the resistance force applied on the cutter i can be derived as:

$$F_{ci} = \tau_i \cdot A. \tag{17}$$

Substituting Eqs. (14)–(17) into Eq. (11), there will be:

$$T_4 = \begin{cases} \sum_{i=1}^{m} [c + K\gamma(H - L_i \ \sin\theta_i) tg\varphi] \cdot w \cdot \frac{\beta}{360} \cdot \frac{v}{n} \cdot tg\alpha \cdot L_i \\ \text{if soil is cohensive,} \\ \sum_{i=1}^{m} [K\gamma(H - L_i \ \sin\theta_i) tg\varphi] \cdot w \cdot \frac{\beta}{360} \cdot \frac{v}{n} \cdot tg\alpha \cdot L_i, \\ \text{if soil is cohensionless.} \end{cases} \tag{18}$$

3.5. Shearing torque on opening

When the excavated soils fall into the working chamber through the opening of cutterhead, the rotating cutterhead makes the slot shear the soils, thereby the shearing resistance is generated. This force is directly dependent on the opening ratio of cutterhead in inverse

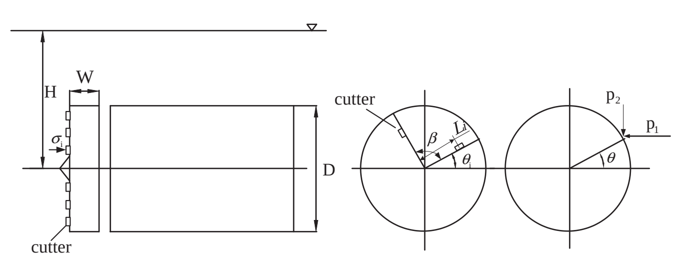

Fig. 7. Diagram of forces acting on cutters.

proportion. Consider an infinitesimal acting area of shear dA, the shear force acting on the slot can be expressed as:

$$F_q = \tau \cdot dA \quad (19)$$

where τ is shear modulus of soil. Taking into account of the opening ratio, the shearing torque will be obtained by integration as:

$$T_5 = \eta \int F_q = \frac{1}{12} \pi D^3 \cdot k_q \cdot \eta \cdot \tau \quad (20)$$

where k_q is a reduced coefficient related to shear area.

3.6. Agitating torque

The agitating bars mounted on the back of cutterhead and driven by the rotating cutterhead are used to stir the earth in the working chamber to prevent formation of clotty earth. It is assumed that the force acting on the bars is induced by earth pressure in the working chamber, as shown in Fig. 8.

Suppose that the number of the bars is n_b, then the agitating torque is:

$$T_6 = \sum_{i=1}^{n_b} \gamma \cdot (H - R_b \sin\theta_i) \cdot D_b \cdot L_b \cdot f_c \cdot n_b \cdot R_b \quad (21)$$

where R_b is the distance between the bar and the centerline of shield, θ_i is the angle of the plane through the axes of the bar and the shield with respect to the horizontal plane, D_b is the diameter of the bar, L_b is the length of the bar, f_c is the friction factor between the improved earth and the steel bar.

3.7. Torque of rotational bearing

There is a large bearing in the EPB shield to support the heavy cutterhead to rotate. The bearing bears both axial force because of thrust and radial force resulting from the cutterhead weight.

$$T_7 = F \cdot \mu \cdot R_t + G \cdot R_r \cdot \mu \quad (22)$$

where F is the thrust force of EPB shield, which can be also taken as the resistance force normal to the cutterhead approximately, R_t is the distance from the thrust acting point to the centerline of shield, μ is the coefficient of rolling resistance, G is the weight of cutterhead, and R_r is the radius of radial roller bearing.

3.8. Torque of sealing

In order to separate the driving mechanism of cutterhead from the muck in the working chamber to prevent abrasion even failure, several sealing rings are installed usually. The drive torque consumed on overcoming the friction caused by sealing is:

$$T_8 = 2\pi R_s^2 \cdot F_s \cdot n_s \cdot \mu_s \quad (23)$$

where F_s is the positive pressure applied on the sealing rings, R_s is the radius of the sealing ring, n_s is the number of the sealing rings, and μ_s is the frictional coefficient between sealing material and steel.

4. Improved model

In order to investigate the components of total torque and how much proportion they account for, eight parts discussed above need to be distinguished between those of crucial importance and those that are not. Calculation and model test will be carried out by the aid of a simulator test rig as shown in Fig. 10. The cutterhead to be considered is given in Fig. 9, its diameter is 1.8 m and width is 0.3 m. Four detachable parts and four stationary parts are positioned on the cutter disk. By installation and removal of stationary parts, the opening of cutterhead can attain a changing range from 30% to 70%. The maximum number of cutters on the disk is 80, and the cutter is 7 cm wide while the front rake is 30°. The four agitating bars, with a diameter of 200 mm, are 320 mm long and at a distance of 700 mm apart from the cutterhead center.

Because there are three types of soils involved in the lab test, torque calculation is also conducted based on these soils. Their properties are provided in Table 2. Substituting the mechanical structure parameters and the soil properties in Table 2 into the torque calculation equations, the above assumed eight parts T_1 to T_8 in three typical types of soils are derived as shown in Tables 3, 4 and 5. Results of two different opening ratios are given.

From the above calculation results and by comparisons, it can be seen that:

1) T_1, T_2, T_3, T_5 and T_6 are the main constituent parts of cutterhead torque. The sum of the other three parts accounts for about 1% of total amount in all soils involved in this work. Occupying nearly half of the total torque in proportion, T_2 turns out to consume the most power of the cutterhead drive. It is more pronounced in the cases with larger opening ratio.
2) Opening ratio is an influential factor. Because the larger opening can reduce the friction surface between earth and steel, the cutterhead torque will be smaller. As shown in the tables, there is a reduction by about 10%.
3) The kind of soil makes a great difference in cutterhead torque. Excavating in clay, sandy soil and sandy gravel, the cutterhead torque increases by degrees.

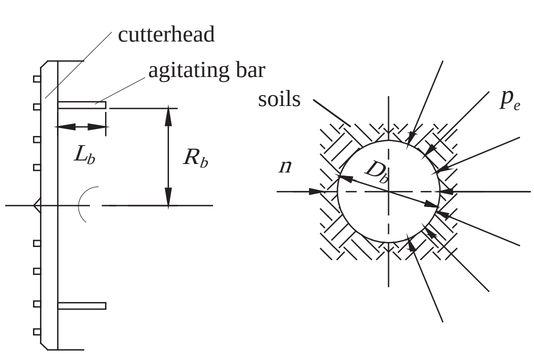

Fig. 8. Diagram of force acting on the agitating bar.

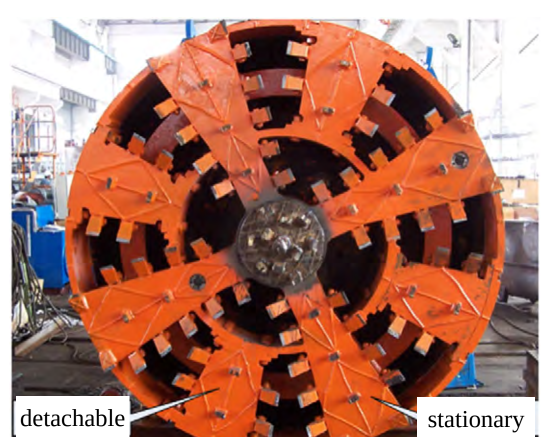

Fig. 9. Cutterhead used in experiment.

Table 2
Physicomechanical properties of soils.

Items	Clay	Sandy soil	Sandy gravel
Equivalent depth (m)	7	7	7
Water content (%)	47.2	18.4	18
Unit weight (kN/m^3)	18.5	20.2	20.2
Lateral pressure coefficient	0.6	0.6	0.6
Friction angle (°)	10	30	37.8
Cohesion (kN/m^2)	10	0	0.1
Void ratio	1.26	0.634	0.559
Frictional coefficient	0.2	0.3	0.35
Shear modulus (N/m^2)	3.1×10^4	2.1×10^4	3×10^4

4) Friction mostly determines the torque required to the cutterhead drive. The torque caused by cutting soil is far less than that by friction in the same soil. According to calculation, compared with the sum of frictional torques T_1, T_2 and T_3, cutting torque T_3 is insignificant enough to be excluded from consideration during design.

Based on those analyses and comparisons, the empirical formula $T=\alpha D^3$ can be improved as the following expression:

$$T = k_1D^3 + k_2D^2 + k_3 \tag{24}$$

where $k_1 = \frac{\pi}{12}\left[K_0(1+f_{\Delta p})f\gamma H(1-\eta) + k_q\eta\tau\right]$, $k_2 = \frac{\pi}{4}(K_0+1)\times f\gamma HW$, $k_3=\gamma HD_bL_bf_cR_bn_b$, and the term D^3 are embedded, in agreement with the empirical formula that T is a function of D^3. Besides, other factors are also taken into consideration in this improved model. Apparently, γH dominates the whole expression, representing the earth pressure resulting from the overburden depth. When earth pressure balance is achieved, the earth pressure p_e in the working chamber is approximately equal to the lateral earth pressure $K_0\gamma H$ shown in Eq. (24) which coincides with the fitting results [12].

5. Tests and discussions

5.1. Test system

Fig. 10 shows the experimental system. The test rig consists of a cylindrical soil simulator box for geoenvironment simulation, the tunneling machine and the condition monitoring system. The soil simulator box, with an inner diameter of 4 m and an axial length of 6 m, can be filled with a variety of soils. The soils will be pressurized by the bag filled with high pressure water. The water bag loading system can provide the soils with a pressure up to 0.4 MPa so that the tunneling machine will be able to go through this artificial underground condition [13].

In the test rig, there is an EPB test shield machine for tunneling test, with a screw conveyor discharging the muck cut down by a cutterhead with a diameter of 1800 mm. The thrust system is composed of six hydraulic cylinders of the same stroke of 1500 mm. The cylinder body is fixed on a backrest while the piston rod is movable to push the machine forward as shown in Fig. 10. When the driving distance is beyond the stroke of the cylinder, the additional blocks are needed to be set between the machine and the hydraulic cylinder to relay the jacking process.

5.2. Test results

In the tests, soil properties are just as those listed in Table 2, corresponding to theoretical calculation. A lot of other main parameters and variables involved in the tests can be found in Ref. [5]. Moreover, the cutterhead speed and the thrust speed are kept at 1 r/min and 1.5 cm/min respectively, the test results are obtained as follows.

The cutterhead torques under 30% and 70% opening ratios in three different soils are shown in Fig. 11. It is obvious that the cutterhead torque varies with the soil classification, especially remarkable between clay and sandy soil. As a whole, the torque value in sandy soil and gravel is larger than in clay, which is in good agreement with the calculation results in Tables 3 and 4. The experiment results also verify that the cutterhead torque required under 30% ratio is about 1.1 times as much as that under 70% ratio, which coincides with the comparisons between the total torques in Tables 3 and 4.

Additionally, the opening ratio has considerable effect on the cutterhead torque. It averages 76.68 kNm in 30% opening ratio and 62.98 kNm in 70% opening ratio, irrespective of various geological conditions. The test results proved that larger opening ratio can reduce the torque required by EPB shield cutterhead. Accordingly, small opening ratio is unfavorable to soil excavation.

Compared with the calculation results, the experimental data agreed on the whole in clay but are severely lower in other soils. Especially in sandy gravel, the difference between prediction and test reaches up to 30 kNm. The reduction is caused by additives, such as bentonite, foam, added to condition the soils so as to decrease friction applied on cutterhead [14,15]. In fact, soil conditioning is also adopted to make excavation easier in tunneling job site. In the former calculation, this improvement on cutting behavior is not considered. The disagreement also indicates that soil conditioning is a quite effective way to cope with the sandy tunneling practice, consistent with test results that suitably mixed soils with additives permit a reduction by 25% in torque values, obtained by Ref. [16].

Fig. 12 shows the cutterhead torque and the earth pressure in the working chamber in the opening ratio of 30%. The data are obtained in

Table 3
Results of 30% opening ratio.

Torque (kNm)	Clay	Sandy soil	Sandy gravel
T_1	21.63	31.57	35.09
T_2	27.56	40.21	48.79
T_3	10.38	15.16	18.76
T_4	0.32	0.32	0.4
T_5	5.69	3.86	6.4
T_6	4.31	6.13	10.53
T_7	0.28	0.28	0.28
T_8	0.27	0.27	0.27
Total torque	70.44	97.8	120.52

Table 4
Results of 70% opening ratio.

Torque (kNm)	Clay	Sandy soil	Sandy gravel
T_1	12.13	17.7	21.9
T_2	27.56	40.22	49.79
T_3	5.82	8.49	10.52
T_4	0.32	0.32	0.4
T_5	13.28	8.9	14.9
T_6	4.42	9.56	11.1
T_7	0.28	0.28	0.28
T_8	0.27	0.27	0.27
Total torque	64.08	85.9	109.36

Table 5
Calculation results of coefficients in improved model.

Soils	T (kNm)					
	At 30% ratio			At 70% ratio		
	k_1	k_2	k_3	k_1	k_2	k_3
Clay	6.68	8.51	4.31	5.31	8.51	4.42
Sandy soil	8.57	12.41	6.31	6.00	12.41	9.56
Sandy gravel	11.14	15.36	10.53	8.05	15.36	11.1

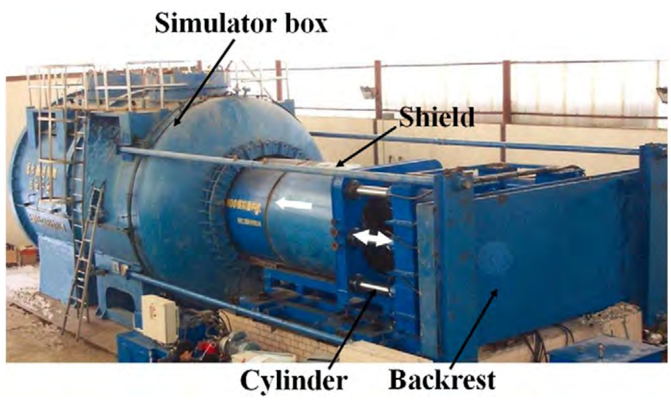

Fig. 10. Tunneling test rig.

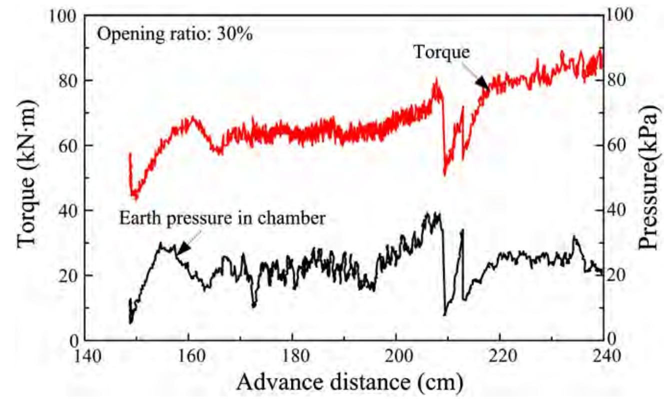

Fig. 12. Relation between torque and earth pressure in chamber.

clay test section. As can be seen, the parameter variations trend in the same way, indicating that the earth pressure in the working chamber influences the torque strikingly. As is known, the earth pressure in the working chamber approaches that of excavated face when earth pressure is balanced. Following the fundamental of soil mechanics, the earth pressure concerned in EPB shield tunneling process is directly in relation to the quantity γH. The results illustrated in Fig. 12 correspond closely to the calculating expression in Eq. (24).

Thrust force, as another important tunneling parameter, is dependent upon the earth pressure acting on the EPB shield as well. The thrust force and the torque variations in the clay tunneling section between 80 cm and 150 cm with the opening ratio of 30% are shown in Fig. 13. They mostly change synchronously, showing considerable linear relation in clay soil. This relationship can also be confirmed from the experimental data fitting result in Fig. 14. The great majority of measured points fall into the area formed by $\pm$95% offset of red straight line, showing that those parameters have good linearity. Actually, friction is the source of thrust force and cutterhead torque in essence, and the frictional force is generated by earth pressure applied on the shield machine surface. Thus, both of them are the reflection of earth pressure, and influenced jointly. It is proved in another respect that γH is a critical factor for determination of cutterhead torque.

5.3. Further analysis

In fact, tunneling is a dynamic process involving the acts of cutting, thrust, discharge and other procedures. So the cutterhead torque will be also related to some other tunneling parameters, and the coordination control of cutting and thrust should be paid much attention.

As we know, the excavated face before the cutterhead, being squeezed when the shield machine is thrust forward, will generate a force to counter this thrust action. That is released by the rotating cutterhead cutting down the pressurized soils. The relationship between the cutterhead torque and the thrust speed is shown in Fig. 15(a), indicating that the higher thrust speed requires higher cutterhead torque. It is obvious that the increase and decrease of thrust speed means the presence of active and passive earth pressures.

Defining the ratio of thrust speed to cutterhead speed as cutter penetration, then we obtain the experiment results in Fig. 15(b). Under the perfect condition, the advance distance of the shield machine should be equal to a cutter length when the cutterhead completes one revolution. However, the cutter penetration can not be kept in that level all the time. In the experiment, the cutter is 30 mm long. It can be seen from the figure that the penetration is mostly smaller than the cutter length through this tunneling section.

Additionally, the rotational speed change of the cutterhead also exerts an influence on the cutterhead torque, as shown in Fig. 15(c), which is in agreement with the related results presented in the past [17]. According to in-situ data analysis, the increased torque results from this factor is mainly consumed by stirring the pressurized earth in the chamber, due to the stirring action shearing through the earth. Therefore, the cutterhead speed should be as small as possible on the basis of meeting the requirement of cutting which is achieved by a lot of trials in a given soil layer before construction.

Compared with the above theoretical model, factors such as thrust speed and cutter penetration are illustrated in Fig. 15. Because tunneling is a dynamic process related to advance speed while analysis

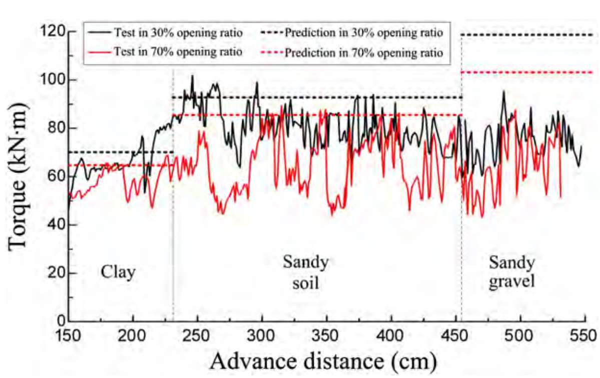

Fig. 11. Torque variations with different opening ratios.

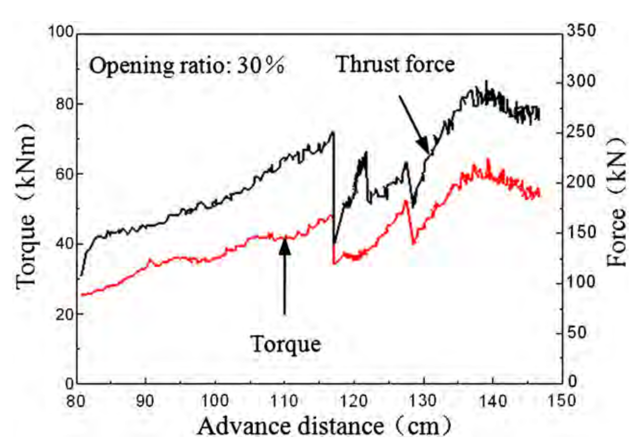

Fig. 13. Variations of torque and thrust force in clay.

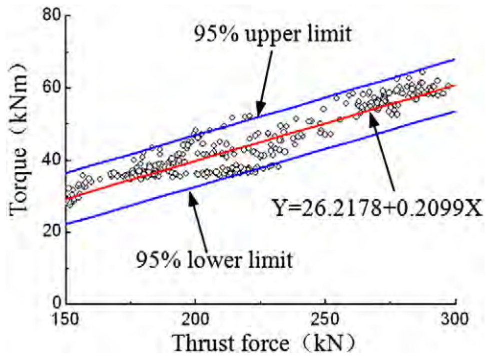

Fig. 14. Relation between torque and thrust force.

on the composition of torque and improved model is based on a steady state of tunneling, it is quite difficult to keep constant speed in view of soil linearity. Therefore, when we design a shield cutterhead drive system, these factors should also be considered.

It should be noted that the thrust speed and the rotational speed of the cutterhead have some fluctuations compared with the set values. These are caused by nonuniform compactness of the soils filled in the box, owing to the noncontinuously distributed loading bags.

6. Conclusions

EPB tunneling requires proper cutterhead torque to obtain a correct excavation control particularly in complex geological conditions. Unfortunately, widely adopted empirical calculation methods are shown to be experience dependent.

A calculation model of cutterhead torque for EPB shield machine is presented based on the comparisons between various types of tunneling projects and the analysis of working and cutting principle of the cutterhead, taking eight main concerned components into account. The calculation allows more accurate and suitable torque capacity than rough estimate by empirical method, consequently, reduces the waste of power to some extent.

Based on the improved model, theoretical calculation and experiments are carried out. The following conclusions can be drawn:

1) The traditional method to use the empirical equation $T = \alpha D^3$ to determine the torque required by the cutterhead shows inaccurate evaluation results in various conditions, and needs to be improved.
2) Friction is the most influential one of all parts that constitute the total cutterhead torque of EPB shield machine. Reducing friction is very useful to soil excavation.
3) Opening ratio of cutterhead has a great influence on the torque determination. The larger the ratio is, the smaller the torque is.
4) Earth pressure produced by overburden depth and unit weight is decisive to determining the cutterhead torque, it directly contributes to frictions applied on the cutterhead.
5) The torque model shows satisfactory predicted results in clay tunneling. To deal with the torque calculation in cohesionless soil tunneling, soil conditioning has to be taken into consideration to acquire improved results.
6) Penetration is a dynamic influential factor affecting the cutterhead torque. The coordination control between thrust speed and cutterhead speed is an effective way to restrain the total cutterhead torque. Smaller thrust and cutting speed is helpful to reduce the torque and save energy under the conditions of soft soil and relatively steady load applied on the thrust system.

These conclusions will definitely provide some positive guidelines and be helpful to design a better EPB machine, especially one that operates in clay. Our subsequent research goal will focus on other drive parameters of the cutterhead regarding power transmission, as well as, adapting the machine to different geological conditions.

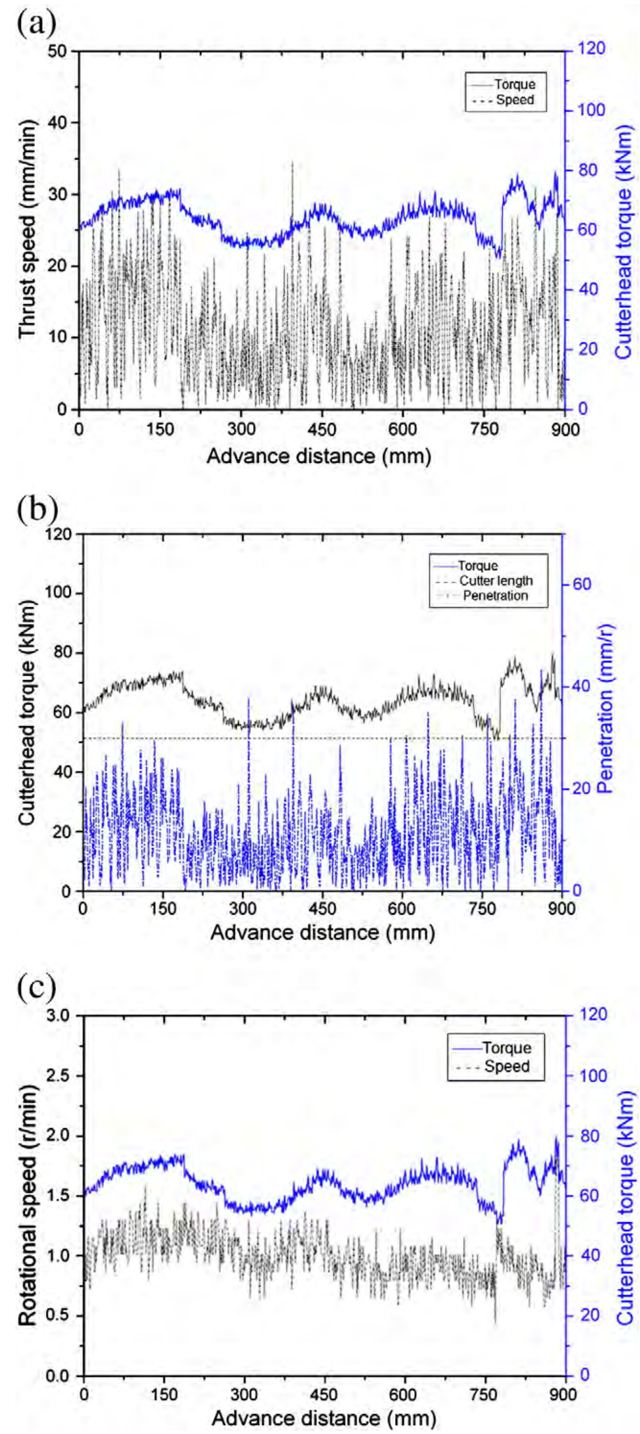

Fig. 15. Effects of thrust and cutting on cutterhead torque.

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