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Abstract: The cutting head is the core working mechanism of the roadheader for coal-rock materials cutting. The efficient and high performance design of cutting head is the key to improve the road head digging and mining technology. In this paper, based on cutting head design theory and virtual prototype technology, we propose a computer-aided structure design and performance optimization method for cutting head. We compile the calculation code and realize the reading and storing of relevant data through Excel. In particular, to obtain more realistic cutting performance data of the cutting head, we construct a coupling model of cutting head cutting rock wall based on virtual prototype technology, and then establish a database matching structural parameters, working parameters, coal-rock properties and cutting performance through extensive simulations. Based on the method, we complete the design of EBZ220 roadheader cutting head. We show that our method can realize the fast and efficient design of cutting head, and the designed cutting head has good working performance.

Keywords: Roadheader; Cutting head; Computer-aided design; Virtual prototype technology; Performance optimization

1. Introduction

The roadheader is mainly composed of walking mechanism, working mechanism, loading mechanism and reloading mechanism. In the process of tunneling and coal mining, the cutting head constantly crushes the coal-rock materials, which is the core working mechanism of the roadheader. Therefore, the efficient and reasonable design of the cutting head is the key to ensure the cutting performance and service life of the roadheader [1,2]. Since as early as the 1960s, a lot of research has been carried out on the design and performance analysis of the cutting head of roadheader. Hekimoglu [3,4] conduct a comparative analysis of the stresses on two types of roadheaders, longitudinal and transverse axis, and demonstrate that the equal circumferential spacing arrangement of the picks is an important factor to ensure the stability of the cutting head. Anon [5] develop a model for predicting the performance of a roadheader cutting hard rock based on experimental tests on rock samples. Acaroglu et al. [6] develop a software for stability analysis of roadheader and analyze the effect of pick inclination and cutting head shape on the stability. Ebrahimabadi et al. [7,8] investigate the effect of rock brittleness index on the performance of roadheader and establish a new empirical formula to predict the performance of roadheader under different cutting conditions. Moreover, they develop a database containing transient

cutting rates and parameters of coal-rock mechanical properties, and analyzed the database using statistical methods to obtain a prediction model for cutting performance of roadheader. Li et al. [9,10] simulate the cutting process of roadheader by finite element method, and analyze the stress of pick and coal-rock material. Dogruoz et al. [11] investigate the wear dullness of picks cutting rocks of different hardness developed a numerical model between the energy consumed per unit volume and the type of picks, wear conditions, and mechanical properties of the rock. Wang et al. [12] study the load fluctuation of cutting head of longitudinal axis roadheader by mathematical method and find that the frequency of low frequency fluctuation is the product of spiral number and rotation frequency, while the frequency of high frequency fluctuation is related to the circumferential spacing of cutting picks. Zhang et al. [13] study the influence of the number of spirals on the performance of the cutting head and find that compared with the two-spiral cutting head, the three-spiral cutting head has a larger average cutting area, less load fluctuation and less variation of the effective cutting picks. Huang et al. [14] propose a genetic algorithm-based cutoff spacing optimization design method with the goal of minimizing load fluctuation. Li [15] propose a design scheme for variable angle spiral line arrangement of picks based on TRIZ theory to improve the cutting performance of cutting head, and carry out experimental verification. Yuan [16,17] et al. explore the optimal interval of cutting angle and rotation angle of picks. Liu et al. [18] summarize the research trends of cutting head design analysis and propose that future research presents the characteristics of mutual verification of theory and experiment, two-way coupling of finite element and discrete element, and combination of intelligent algorithm and virtual simulation.

With the development of computer technology, the first cutting head design software package for roadheader appeared in the 1980s [19]. And then, scholars began to apply it to the cutting head design process continuously. By the 1990s, many scholars have developed a variety of software packages for cutting head design. Some of these software packages can generate cutting diagrams and calculate forces and moments of picks; some can obtain information on power, moments and circumferential forces under different cutting circumference angles; some can generate pick arrangement diagrams and draw curves of load and cutting specific energy consumption [20-26]. At the early 21st century, Tiryaki et al. [27] develop a new software for the design of cutting head for roadheader, which takes into account the equation of cutting force coefficient in the prediction of force on picks compared with the previous auxiliary design software. The software can reflect the difference of force and chip between one pick per line and three picks per line. However, the software does not consider the relationship between rock mechanical properties and the prediction of cutting specific energy consumption and the selection of cutting head design. Later, Li [28-31] compile a program to simulate the chip diagram of picks under the drilling condition of roadheader. In addition, He and other scholars [32,33] jointly prepare a design

program for spiral line and pick distribution of cutter head and a program to compare the force and chip shape of cutting head under different picks arrangement. Qin [34] develop an interactive parametric design system for cutting heads based on ANSYS APDL and VC++ language, which greatly improved the design efficiency and design quality. Wei [35] develop a longitudinal shaft roadheader dynamic stability analysis software.

Although the design software of cutting head for roadheader is constantly innovated and improved, the software functions and factors considered are not comprehensive, and it fails to consider the important indexes reflecting cutting performance such as load, chip area, square rate, cutting power, cutting specific energy consumption, etc., and also fails to consider the influence of structural parameters, kinematic parameters and coal-rock properties on cutting performance. Therefore, the computer-aided efficient design and performance optimization of cutting head is proposed in this study, which visualizes the cutting head design process by compiling calculation code, and reading and storing the relevant data in combination with Excel. It is worth mentioning that the cutting performance in the proposed approach is not completely calculated by theoretical equations, but based on the virtual prototype technology, the coupling model of cutting head cutting rock wall is constructed, and a database matching the structural parameters, working parameters, coal-rock properties and cutting performance is established through a large number of simulations. Through the design and application of EBZ220 cutting head, it is found that the design process is convenient and fast, and the cutting head has good performance, which indicates that the proposed approach can provide a clear quantitative basis for its structural design and performance optimization.

2. Kinematic theory of cutting head

The cutting head is composed of head body, spiral blade, pick and pick seat, as shown in Fig. 1. The spiral blade is welded to the head body in the form of segmental rolling or sheet metal lap, and the pick is installed in the pick seat and welded in the direction of the spiral blade according to the intercept distance. The shape of cutting head usually consists of cylindrical section, conical section and spherical section, and its structural parameters include cutting head length, diameter, isometric principle angle, spiral angle, etc.

Fig. 1. Structure composition of cutting head.

In the process of cross-swing cutting, the cutting head not only rotates around its own axis, but also swings with the cantilever, and the spatial position of the cutting head and the rotary hydraulic rod is shown in Fig. 2.

Fig. 2. Spatial position of the cutting head and the rotary hydraulic rod.

As shown in Fig. 2, the hydraulic rods A and B are arranged symmetrically, and the initial lengths are l . The distance from the rod extension to the center of the slewing mechanism is equal, i.e. $|O_A O| = |O_B O|$. The angle between the rod and the center of the slewing mechanism is equal, noted as δ . The cross-swing radius of the cutting head is R_c . The relationship between the cross-swing angle σ and the cross-swing speed v can be expressed as follows.

$$\sigma = \frac{vT}{R_c} \quad (1)$$

Where T is one-way cross-swing time.

At this moment, the expansion and contraction of the hydraulic rod are l_A , l_B , and the rod extensions

are noted as $O_A^{\cdot}$, $O_B^{\cdot}$. According to the geometric and kinematic relations, the following system of equations is obtained.

$$\begin{cases} |O_A O_A^{\cdot}|^2 + l^2 - 2|O_A O_A^{\cdot}| \cos(\delta + \theta_A) = (l + l_A)^2 \\ |O_A O_A^{\cdot}|^2 = |OO_A|^2 + |OO_A^{\cdot}|^2 - 2|OO_A||OO_A^{\cdot}| \cos \sigma \\ \theta_A = (180^\circ - \sigma) / 2 \end{cases} \quad (2)$$

$$\begin{cases} |O_B O_B^{\cdot}|^2 + l^2 - 2|O_B O_B^{\cdot}| \cos(\delta - \theta_B) = (l - l_B)^2 \\ |O_B O_B^{\cdot}|^2 = |OO_B|^2 + |OO_B^{\cdot}|^2 - 2|OO_B||OO_B^{\cdot}| \cos \sigma \\ \theta_B = (180^\circ - \sigma) / 2 \end{cases} \quad (3)$$

The distance from the pick J to the swing center of the cantilever is recorded as L_c , and the distance to the axis of the cutting head is recorded as L_r . Therefore, The equation of motion of the pick tip is derived as follows:

$$\begin{cases} x = L_c \sin(\omega_H T) + L_r \sin(\omega_r T) \sin(\omega_H T) \\ y = L_c \cos(\omega_H T) - L_r \sin(\omega_r T) \sin(\omega_H T) \\ z = L_r \cos(\omega_r T) \end{cases} \quad (4)$$

Moreover, the cutting power and cutting specific energy consumption can be described:

$$P = 2\pi n_r M_z / (60 \cdot 10^6 \eta) \quad (5)$$

Where M_z is the torque of cutting head; η is the mechanical transfer efficiency.

$$H_w = P / 60 \lambda A v \quad (6)$$

Where λ is the loose coefficient of coal-rock; A is the projected area of the cutting head in the vertical cross-swing direction; v is the cross-swing speed of the cutting head.

3. Structure design of cutting head

3.1. Picks arrangement

According to the shape characteristics of the cutting head, the picks arrangement is designed in steps according to the cylindrical section, the conical section and the spherical section.

(1) cylindrical section

According to the cutting requirements of roadheader, the relationship between the intercept distance of the pick and the maximum cutting depth of the cutting head can be expressed as follows:

$$t_1 = 2h_{\max} (\tan \varphi - \tan \beta) \quad (7)$$

Where t_1 is the intercept distance; $h_{\max}$ is the maximum cutting depth; φ is the caving angle; β is the spiral angle.

The number of intercept lines is:

$$n_1 = \text{floor}[(L_1 - L_f) / t_1 + 1] \quad (8)$$

Where *floor* is calculated by rounding down; L_1 is the length of the cylindrical section; L_f is the distance between the first intercept line and the large end face of the cutting head.

Thus, the remaining length of the cylindrical segment L_{1s} can be described as:

$$L_{1s} = L_1 - L_f - t_1 n_1 \quad (9)$$

The axial distance of the pick from the large end face is derived as:

$$l_1(i) = L_f + (n_i - 1) \cdot t_1 \quad (10)$$

(2) conical section

The intercept line arrangement of the conical section can be divided into two types: equidistance or equidifference, which are chosen according to the relationship between the remaining length of the cylindrical section and the intercept distance.

The number of intercept lines for each of the two types is:

Equidistance:

$$n_2 = \text{floor}[(L_2 + L_{1s}) / t_2] \quad (11)$$

Equidifference:

$$n_2 = (1 - \sqrt{1 - 8 \text{floor}(L_{1s} + L_2) / d}) / 2 \quad (12)$$

Where L_2 is the length of the conical section, d is the tolerance, t_2 is the intercept distance of the conical section.

Thus, the remaining length of the cylindrical segment L_{2s} can be described as:

Equidistance:

$$L_{2s} = L_{1s} + L_2 - n_2 t_2 \quad (13)$$

Equidifference:

$$L_{2s} = L_{1s} + L_2 - (t_2 + n_2(n_2 - 1)d / 2) \quad (14)$$

The axial distance of the pick from the large end face is derived as:

Equidistance:

$$l_2(i) = L_1 - L_{1s} + (n_i - n_1) \cdot t_2 \quad (15)$$

Equidifference:

$$l_2(i) = L_f + (n_1 - 1) \cdot t_1 + (n_i - n_1 - 1) \cdot t_2 - (n_i - n_1 - 1)^2 \cdot d / 2 \quad (16)$$

(3) spherical section

There is a strong compression tension effect in the coal-rock crushing process. In order to avoid deformation or damage of the pick due to excessive force, the density of the pick at the spherical section should be gradually increased. Therefore, the pick arrangement of the spherical section adopts the equal degree principle, and the number of intercept lines is:

$$n_3 = (90 - \phi) / \alpha \quad (17)$$

Where ϕ is the inclination of the first intercept line; α is the isometric principle angle.

The axial distance of the pick from the large end face is derived as:

$$l_3(i) = R \cdot \sin \theta - l_f + L_1 + L_2 \quad (18)$$

Where R is the radius of the spherical section; θ is the inclination angle of picks; l_f is the distance from the first intercept line of the spherical section to the spherical center.

The formula of the circumferential angle γ_b determining the pick installation position is as follows:

$$\gamma_b = \frac{360^\circ l_i}{\pi D_y \tan \beta} \quad (19)$$

Where D_y is the outer edge diameter of spiral blade.

By compiling and running the program code, the pick arrangement diagrams under the above two types were obtained as shown in Fig. 3.

Fig. 3. Pick arrangement diagram: (a) equidistance; (b) equidifference.

3.2 Pick cutting diagram

The cutting diagram is an important basis for considering the size of the coal-rock. The larger the cutting area, the larger the rock lumpiness and the smaller the amount of dust, and the key to drawing the cutting diagram is to determine the upper and lower caving lines of the picks. The intersection point of the crumbling line of the pick on the i th intercept line during the cross-swing cutting of the sequential cutting head is shown in Fig. 4.

Fig. 4. Schematic diagram of intersection point of the crumbling line.

Where E_1 , E_2 are the upper intersection point and lower intersection point respectively.

According to the geometric and kinematic relationship, the following equation can be obtained.

$$|A_1B_1| = \frac{\cot(\varphi - \theta_i) \cdot (t \cot(\varphi + \theta_{i+1}) + (1000v_i \Delta\gamma_{i+1} / 360n_r))}{\cot(\varphi - \theta_i) + \cot(\varphi + \theta_{i+1})}$$

$$|A_2B_2| = \frac{\cot(\varphi - \theta_{i-1}) \cdot (t \cot(\varphi + \theta_i) + (1000v_i \Delta\gamma_{i-1} / 360n_r))}{\cot(\varphi - \theta_{i-1}) + \cot(\varphi + \theta_i)}$$

The distance from the point C to the center axis of the cutting head is:

$$L_c = R_i + \frac{1000v_i \gamma_{bi}}{360n_r} + (n-1) \frac{1000v_i}{m} \quad (n=1, 2, L)$$

Where $\Delta\gamma_{i+1}$, $\Delta\gamma_{i-1}$ are the difference in circumferential angle of the picks on the adjacent intercept line; n_r is the rotation speed; R_i is the cutting radius; γ_{bi} is the circumferential angle of the pick tip; m is the number of blade heads.

Therefore, the equations of the upper and lower caving lines can be described as:

$$y_s = \tan(\theta_i - \varphi)(x_s - L_c) + l_i \quad L_c - |A_1B_1| \leq x_s \leq L_c$$

$$y_x = \tan(\theta_i + \varphi)(x_x - L_c) + l_i \quad L_c - |A_2B_2| \leq x_x \leq L_c$$

By compiling and running the program code, the cutting diagrams under the above two types were obtained as shown in Fig. 5.

Fig. 5. Cutting diagram:(a) equidistance; (b) equidifference.

4. Construction and verification of virtual prototype model

4.1 Construction of coupling model

In this study, the cutting performance information is obtained based on the virtual prototype technology. Firstly, a 3D model of the cutting mechanism of the roadheader is established using Pro/e, and an arc-shaped rock wall model with the slewing mechanism as the center and the cutting arm as the radius is established according to the cross-swing trajectory of the roadheader. Secondly, the rock wall model is imported into the discrete element software for particle filling and bonding to generate a discrete element model of the rock wall. Then, the 3D model of the cutting mechanism is imported into the

multi-body dynamics software. According to the working principle of the cutting mechanism, constraints and drives are added to the parts to establish the dynamics model of the cutting mechanism. Finally, the cutting head is used as the coupling part with the rock wall to establish the coupling model. The coupling model and data transfer principle are shown in Fig. 6 [36].

Fig. 6. Coupling model and data transfer principle.

Through simulation, the load curves of the cutting head is obtained as shown in Fig. 7.

Fig. 7. Load curves of the cutting head.

From the Fig. 7, it can be seen that in the cutting process, the load in X, Y, Z direction of the cutting head presents nonlinearity and fluctuation, and the load amplitude in Z direction is large, and the load amplitude in Y direction is relatively small.

Based on the load data obtained from simulation, the cutting power and cutting specific energy consumption can be calculated according to Eq. (5)~(6).

4.2 Experimental validation

In order to verify the accuracy of the load information obtained based on virtual simulation, an cutting

test bench for cutting head was built. Firstly, the physical experiment model of rock wall was established, and the similarity between the rock wall model and the actual rock wall downhole was proved by uniaxial compression test on the rock wall specimens (Fig. 8). Secondly, a physical model of roadheader, the hydraulic control system, the electric control cabinet and the control consoles were built. Then, the cutting head is controlled by the computer control platform to cut the rock wall, and the load information of cutting head is obtained through the data acquisition system (Fig. 9).

Fig. 8. Rock wall construction and specimens: (a) filling; (b) compacting; (c) rock wall model; (d) specimens.

Fig. 9. Cutting test bench.

Fig. 10 shows the comparison curves between experimental load and simulation result of the cutting head. From Fig. 10, it can be seen that the simulation results are consistent with the experimental load. Accordingly, we can conclude that it is feasible to obtain the load information of the cutting head based on the virtual prototype technology. Therefore, the virtual simulation can be used to obtain the load information of the cutting head under different working conditions, and then obtain the cutting performance.

Fig. 10. The comparison curves between experimental force and simulation result: (a) X-directional; (b) Y-directional; (c) Z-directional.

5. Cutting performance of the cutting head

By performing a large number of simulations, the relationship surfaces for cutting performance under different parameters are constructed. The following is the examples of the isometric principle angle, spiral angle, cross-swing speed and rotation speed.

5.1 Isometric principle angle and spiral angle

The cutting head with the isometric principle angles of 4° , 4.5° , 5° , 5.5° and 6° and the spiral angles of 14° , 16° , 18° and 20° were established respectively, and the cutting load, cutting specific energy consumption, maximum cutting area and cutting power were obtained by simulation and calculation, and the relationship surfaces of each cutting performance index with the isometric principle angle and spiral angle were fitted, as shown in Fig. 11.

Fig. 11. Relationship surfaces of each cutting performance index with the isometric principle angles and spiral angles: (a) cutting load; (b) cutting specific energy consumption; (c) maximum cutting area; (d) cutting power.

From Fig. 11 (a), it can be seen that with the increase of spiral angle, the cutting load increases and then decreases; with the increase of isometric principle angle, the cutting load has a non-linear decreasing

trend.

From Figs. 11 (b) and (d), it can be seen that with the increase of spiral angle, the cutting specific energy consumption and cutting power of cutting head first decreases and then increases. When the spiral angle is 15.5° , both of them reach the minimum value; with the increase of the isometric principle angle, the cutting specific energy consumption and cutting power show a non-linear and gentle decreasing trend.

From Fig. 11 (c), it can be seen that with the increase of the spiral angle, the maximum cutting area decreases. This is due to the fact that the increase of the spiral angle can reduce the distance between the picks on the two intercept lines in the circumferential direction, thus shortening the length of the upper and lower caving lines, resulting in a smaller cutting area; with the increase of the isometric principle angle, the maximum cutting area shows an approximately linear increasing trend. The effect of the angle of the isometric principle on the maximum cutting area is more significant.

5.2 cross-swing speed and rotation speed

By adjusting the cross-swing speed and rotation speed of cutting head, the cutting load, cutting specific energy consumption, maximum cutting area and cutting power were obtained by simulation and calculation, and the relationship surfaces of each cutting performance index with the cross-swing speed and rotation speed were fitted, as shown in Fig. 12.

Fig. 12. Relationship surfaces of each cutting performance index with the cross-swing speeds and rotation speeds: (a) cutting load; (b) cutting specific energy consumption; (c) maximum cutting area; (d) cutting power.

From Figs. 12 (a) and (c), it can be seen that the maximum cutting area and cutting load increase with the increase of the cross-swing speed and decrease with the increase of the rotation speed. This is due to the fact that increasing the cross-swing speed can increase the cutting distance of the picks per unit time, resulting in the increase of the cutting area. When the volume of rock material to be cut is the same, the higher the rotation speed, the shorter the cutting cycle of single pick, and the more cutting times, which leads to the reduction of cutting area.

From Fig. 12 (b), it can be seen that the cutting specific energy consumption is non-linearly decreasing with the increase of the cross-swing speed, and the change rate is gradually decreasing; the cutting specific energy consumption is non-linearly increasing with the increase of the rotation speed. The effect of the cross-swing speed on the cutting specific energy consumption is more significant.

From Fig. 12 (d), it can be seen that the increase of the cross-swing speed and the rotation speed can increase the cutting power in a nearly linear trend, and the influence of the cross-swing speed on the cutting power is more obvious. This is because increasing the cross-swing speed can increase the cutting distance of the picks, and once the cutting distance increases, the power consumed by the cutting motor will increase; while the rotation speed determines the number of picks involved in cutting per unit time, and the more the number of picks, the higher the cutting power required.

Cutting performance data under other parameters are obtained, so that a database matching cutting performance with structural parameters, working parameters and coal-rock properties is established for performance optimization in the cutting head design process.

In summary, the flow of computer-aided efficient design and performance optimization of the cutting

head for roadheader is shown in Fig. 13. During the design process, the design parameters need to be input into EXCEL, and then the program code is executed to read the relevant parameters through “xlsread” to calculate the corresponding pick arrangement and cutting performance data, and finally stored in EXCEL. If the design result is not satisfactory, the design parameters can be adjusted to modify the structure of the cutting head to achieve the purpose of optimizing the performance. Parts of the interface and program code are shown in Fig. 14.

Fig. 13. The flow of computer-aided efficient design and performance optimization of the cutting head for roadheader.

Computer-aided efficient design and performance optimization software of cutting head for roadheader					
Design parameters	Value	Unit	Design parameters	Value	Unit
cross-swing speed	1.11	m/min	length of cylindrical section	110	mm
rotation speed	46.7	r/min	length of conical section	400	mm
isometric principle angle	4	°	conical section half cone angle	16	°
spiral angle	13	°	length of spherical section	403	mm
diameter of cylindrical section	1120	mm	number of blade heads	3	
conical segment arrangement	1		caving angle	46	°
firmness coefficient of rock	7		rock capacity	2.7	t/m ³

(a)

Text output					
Items	MAX	MIN	AVG	Variance	Fluctuation coefficient
X-directional force(N)	87014.8	73011.7	79794.0	3008.1	0.038
Y-directional force(N)	-55710.1	-71363.2	-63750.6	3197.5	0.050
Z-directional force(N)	17294.9	15144.9	16263.9	592.8	0.036
Cutting power(kW)	214.0	186.6	199.7	6.57	0.033
Cutting specific energy consumption (kW·h/m ³)	4.99	4.35	4.66	0.15	0.03
Maximum cutting area(mm ²)	860.87				

(b)

%Cylindrical section designing(part)
1:n1=floor((Lcylin-Wpb)/tmax);
2:Lcylin_s=Lcylin-Wpb-n1*tmax;
3:N1=n1+1;
4: for i=1:N1
5: z(i)=Wpb+(i-1)*tmax;
6: r(i)=D/2;
7: if mod(i-1,3)==0
8: angle(i)=yzj*(i-1)/3.0;
9: if (angle(i)>720&&angle(i)<1080)
10: angle(i)=angle(i)-720;
11: elseif (angle(i)>360&&angle(i)<720)
12: angle(i)=angle(i)-360;
13: elseif (angle(i)<0&&angle(i)>=-360)
14: angle(i)=angle(i)+360;
15: elseif (angle(i)<-360&&angle(i)>=-720)
16: angle(i)=angle(i)+720;
17: elseif (angle(i)<-720&&angle(i)>=-1080)
18: angle(i)=angle(i)+1080;
19: else
20: angle(i)=yzj*(i-1)/3.0;
21: end
...

(c)

Fig. 14. Parts of the interface and program code: (a) partial parameter input interface; (b) partial text output interface;(c) partial program code.

6. Engineering Applications

We designed the EBZ220 roadheader cutting head by the computer-aided structure design and performance optimization method, and obtained the optimal design scheme and performance data as shown in Table 1.

Table 1

Optimal scheme for the design of the cutting head.

Projects	value
cross-swing speed (m/min)	1.106
rotation speed (r/min)	46.7
spiral angle (°)	15.5
isometric principle angle (°)	4
cutting specific energy consumption (kW·h/m ³)	4.9
maximum cutting area (mm ²)	854.95
cutting power (kW)	217.49
cutting load (kN)	112.20

The cutting head was designed and manufactured according to the structural parameters in Table 1 and used in downhole mining (Fig. 15). Through long-term monitoring of the working stability and cutting performance of the roadheader, we find that the cutting head no obvious breakage phenomenon and is highly reliable. The production goals of high efficiency, economy and quality are achieved, which shows that the proposed approach is feasible.

Fig. 15. Cutting head of EBZ220 roadheader in underground mining.

7. Conclusion

We present a method for the structure design and performance optimization of cutting head. The core components of the proposed approach include compiling the calculation code jointly with EXCEL and obtaining the cutting head load information and other performance data based on the virtual prototype

technology. The establishment and simulation of the coupling model of the cutting head cutting rock wall plays a crucial role for the accuracy and effectiveness of the proposed approach. The coupled model of cutting head cutting rock wall is verified by physical experiments. The cutting head designed by the proposed approach in this paper was also validated by downhole mining tests, which showed that the cutting head created by the proposed approach works reliably and has high performance. For future work, if the wear characteristics can be considered in the performance optimization of the cutting head, it is significant to extend its service life and improve the economic efficiency.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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