1. Introduction

Due to the complexity of geological conditions, uniform layout design of shield thrust system cannot eliminate the partial load effect of jacking force acting on the rear segments (Deng et al., 2017), which, to a large extent, causes loose installation and sealing of segments or cracks that result in the tunnel leakage or even local collapse. Faced with this serious problem, many scholars have made the following three researches on how to eliminate the partial load.

An in-depth theoretical and experimental study was conducted on the design of construction parameters of thrust mechanism under the action of partial load: Snowdon and Temporal et al. studied the relationship between the variable stiffness of thrust mechanism and the thrust of the system, tunneling radius and partial load, and tried to find out the influence rule of the variable stiffness of thrust mechanism on partial load acting on the system (Snowdon et al., 1983) (Temporal and Snowdon, 1982). Parameter analysis and optimization design research of hydraulic system and control system in the propulsion mechanism: Mo et al. conducted in-depth analysis on the influence of attitude adjustment control parameters of the thrust system on construction quality of segments borne by partial load acting on the system (Snowdon et al., 1983) (Temporal and Snowdon, 1982). 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influence of longitudinal resistance torque $M_y$ and transverse resistance torque $M_x$ on the force transmission of the propulsion mechanism.

According to the principle of force balance, it can be known that if the longitudinal resistance torque $M_y$ or the transverse resistance torque $M_x$ cannot be well eliminated, it is difficult for the hydraulic cylinder in the thrust system to form the same jacking thrust $F_i$ to overcome the axial load $F_z$ in the driving direction under the coupling action of external force, thus causing partial load of the thrust system. However, at present the most effective methods to solve partial load, such as copying knife, overcutting knife, hinging device and bentonite assisted engineering method, cannot eliminate the longitudinal or transverse resistance torque formed by shield tunneling under composite stratum conditions. For example, the longitudinal resistance torque $M_y$ caused by the “top-heavy” feature of shield machine cannot be eliminated, because it is determined by the quality distribution attribute of shield structure itself. Therefore, it is necessary to thoroughly study the internal mechanism of thrust system under the condition of composite stratum and to explore new anti-bias load methods based on partial load mechanism.

This paper puts forward an adjustable structure of shield thrust system, and expects to use this structure to solve the problem of partial load caused by shield construction under composite stratum, and provide theoretical guidance and technical support for the design and selection of shield under composite stratum. The core of solving the partial load problem is to change the idea of eliminating the external resistance torque $M_y$ and $M_x$ to adapt to the uniform layout structure of the propulsion mechanism, but according to the coupling action characteristics of external loads $M_y$, $M_x$ and $F_z$ on the thrust system, the fixed connection between the traditional propelling hydraulic cylinder and the main machine is changed to the adjustable layout structure on the premise that the number of hydraulic cylinders and structural parameters remain unchanged. For example, under the condition of encountering heterogeneous rocks in the process of shield tunneling, after the shield tunneling at the distance of a segment ring is completed and during the segment assembly is carried out by the pipe assembly machine, the hydraulic cylinder is regulated by an adjustable mechanism, and the phase angle of the hydraulic cylinder in the circumferential sliding chute is correspondingly changed to eliminate partial load and adapt to external load.

2. Mechanical model construction of shield thrust system under uniform and non-uniform layout conditions

It is assumed that the external loads on the shield thrust system are shown in Fig. 1. The external loads are $\overline{F_y}$, $\overline{M_y}$ and $\overline{M_x}$ respectively. By taking the distributed geometric center of hydraulic cylinder in thrust system as the coordinate origin, the $z$-axis is parallel to each hydraulic cylinder and consistent with the direction of external resistance $\overline{F_y}$, and the $y$-axis is perpendicular to $z$-axis and points to the direction shown in Fig. 1, as well as the $x$-axis is determined by the right hand rule, which constructs the spatial rectangular coordinate system $\mathbf{o-xyz}$. In Fig. 1, $\overline{F_y}$ is the combined resistance in the positive direction of the $z$-axis; $\overline{M_y}$ is the combined resistance moment in the positive direction of the $x$-axis; $\overline{M_x}$ is the combined resistance moment in the positive direction of $y$-axis; $\overline{F_y}$ ($i = 1, 2, ..., N$) is the thrust of the $i^{th}$ hydraulic cylinder in the thrust system, whose direction is opposite to the positive direction of the $z$-axis; There are $N$ hydraulic cylinders in the shield thrust system.
According to the force acting on thrust system in Fig. 1, the force balance equations of (1), (2) and (3) can be acquired (Deng et al., 2011):

$$\sum_{i=1}^{N} F_i - F_z = 0$$

$$\sum_{i=1}^{N} F_i x_i + M_y = 0$$

$$\sum_{i=1}^{N} F_i y_i - M_x = 0$$

where \((x_i, y_i)\) is the force acting point coordinate of each hydraulic cylinder in the \(o-xy\) plane.

In order to evenly distribute the thrust of each hydraulic cylinder, optimization function of Eq. (4) is constructed on the ground of Eqs. (1) ~ (3) to acquire the minimum value of this function.

$$\Delta = \frac{1}{2} \sum_{i=1}^{N} \left( F_i - \bar{F} \right)^2$$

where \(\bar{F} = \frac{1}{N} \sum_{i=1}^{N} F_i = \frac{1}{N} F_z\).

If the minimum value of \(\Delta\) in optimization function (4) is obtained, it indicates that the difference between the thrust provided by each hydraulic cylinder in the thrust system is the minimum, and the jacking force acting on the rear segment tends to be uniform, which is beneficial to avoiding the crushing phenomenon of the rear segment caused by the partial load effect of jacking force.

Simultaneous optimization function (4) and three force balance equations (1) ~ (3), and Lagrange function (5) can be established:

$$L = \Delta + \lambda_1 \left( \sum_{i=1}^{N} F_i - F_z \right) + \lambda_2 \left( \sum_{i=1}^{N} x_i F_i + M_y \right) + \lambda_3 \left( \sum_{i=1}^{N} y_i F_i - M_x \right)$$

Fig. 1 Mechanical model of the thrust system
where \( \lambda_1, \lambda_2, \lambda_3 \) are the Lagrange parameters.

Partial derivatives of Lagrange equation (5) can obtain Eqs. (6) and (7).

\[
\frac{\partial L}{\partial F_i} = F_i - F_i^e + \lambda_1 x_i + \lambda_2 y_i = 0 \quad (i = 1, 2, ..., N) 
\]  
(6)

\[
\frac{\partial L}{\partial \lambda_i} = 0 \quad (i = 1, 2, 3) 
\]  
(7)

From Eqs. (6) ~ (7), the matrix equation can be gained as follows:

\[
\begin{bmatrix}
1 & \cdots & 1 & 0 & 0 & 0 \\
x_1 & \cdots & x_N & 0 & 0 & 0 \\
y_1 & \cdots & y_N & 0 & 0 & 0 \\
1 & \cdots & 0 & 1 & x_1 & y_1 \\
: & \cdots & : & : & : & \lambda_1 \\
0 & \cdots & 1 & 1 & x_N & y_N
\end{bmatrix}
\begin{bmatrix}
F_1 \\
\vdots \\
F_N \\
\lambda_1 \\
\lambda_2 \\
\lambda_3
\end{bmatrix}
= 
\begin{bmatrix}
F_z \\
-M_y \\
M_x \\
F_z \\
\frac{F_z}{N} \\
\frac{F_z}{N}
\end{bmatrix}
\]  
(8)

Assuming matrix \( A \) is the inverse matrix of coefficient matrix about Eq. (8), and let it be:

\[
A = 
\begin{bmatrix}
a_{11} & a_{12} & a_{13} & \cdots & a_{1N+3} \\
\vdots & \vdots & \vdots & \cdots & \vdots \\
a_{N+11} & a_{N+12} & a_{N+13} & \cdots & a_{N+1N+3} \\
a_{N+21} & a_{N+22} & a_{N+23} & \cdots & a_{N+2N+3} \\
a_{N+31} & a_{N+32} & a_{N+33} & \cdots & a_{N+3N+3}
\end{bmatrix}
\]  

There is:

\[
\begin{bmatrix}
a_{11} & a_{12} & a_{13} & \cdots & a_{1N+3} \\
\vdots & \vdots & \vdots & \cdots & \vdots \\
a_{N+11} & a_{N+12} & a_{N+13} & \cdots & a_{N+1N+3} \\
a_{N+21} & a_{N+22} & a_{N+23} & \cdots & a_{N+2N+3} \\
a_{N+31} & a_{N+32} & a_{N+33} & \cdots & a_{N+3N+3}
\end{bmatrix}
\begin{bmatrix}
F_z \\
\vdots \\
F_N \\
\lambda_1 \\
\lambda_2 \\
\lambda_3
\end{bmatrix}
= 
\begin{bmatrix}
F_z \\
-M_y \\
M_x \\
F_z \\
\frac{F_z}{N} \\
\frac{F_z}{N}
\end{bmatrix}
\]  
(9)

Through solving Eq. (9), the following Lagrange parameters can be acquired:

\[
\lambda_2 = a_{N+21} F_z - a_{N+22} M_y + a_{N+23} M_x 
\]  
(10)
\[ \lambda_3 = a_{N+31} F_z - a_{N+32} M_y + a_{N+33} M_x \]  

(11)

From the relationship between matrix \( A \) and its own invertible matrix, we can get:

\[
\begin{bmatrix}
    a_{11} & a_{12} & a_{13} & \ldots & a_{1N+1} \\
    \vdots & \vdots & \vdots & \ddots & \vdots \\
    a_{N+11} & a_{N+12} & a_{N+13} & \ldots & a_{N+1N+3} \\
    a_{N+21} & a_{N+22} & a_{N+23} & \ldots & a_{N+2N+3} \\
    a_{N+31} & a_{N+32} & a_{N+33} & \ldots & a_{N+3N+3}
\end{bmatrix}
\begin{bmatrix}
    1 & \ldots & 1 & 0 & 0 & 0 \\
    x_1 & \ldots & x_N & 0 & 0 & 0 \\
    y_1 & \ldots & y_N & 0 & 0 & 0 \\
    1 & \ldots & 0 & 1 & x_1 & y_1 \\
    \vdots & \vdots & \vdots & \vdots & \vdots & \vdots \\
    0 & \ldots & 1 & 1 & x_N & y_N
\end{bmatrix}
\]  

(12)

From Eq. (12), the following equation can be deduced:

\[ Na_{N+21} + a_{N+22} \sum_{i=1}^{N} x_i + a_{N+23} \sum_{i=1}^{N} y_i = 0 \]  

(13)

\[ Na_{N+31} + a_{N+32} \sum_{i=1}^{N} x_i + a_{N+33} \sum_{i=1}^{N} y_i = 0 \]  

(14)

In the force uniform characteristic of the thrust system, if the thrust of each hydraulic cylinder is equal, the following equation is established:

\[ F_1 = F_2 = \cdots = F_i = \cdots = F_N = \frac{F_z}{N} \]  

(15)

For each hydraulic cylinder randomly distributed on the circumference of the thrust system, the force on each hydraulic cylinder can be expressed by the polar coordinate transformation of Eq. (6) as follows:

\[ F_i = \frac{F_z}{N} - \lambda_i - r \sqrt{\lambda_i^2 + \lambda_j^2} \sin (\theta_i + \varphi) \]  

(16)

where \( r \) represents the layout radius of each hydraulic cylinder in the shield thrust system, and \( \theta_i \) represents the included angle between the layout radius \( r \) and the positive direction of \( x \)-axis, and \( \varphi = \arctan (\lambda_2 / \lambda_3) \).
Through combining Eqs. (15) and (16), it can be derived that Eq. (15) can be established if and only if Lagrange parameters in Eq. (16) meet the following equation:

\[ \lambda_1 = \lambda_2 = \lambda_3 = 0 \]  
(17)

Through substituting Eq. (17) into Eqs. (10) and (11), it can be derived that:

\[ a_{N+21} F_z - a_{N+22} M_y + a_{N+23} M_x = 0 \]  
(18)

\[ a_{N+31} F_z - a_{N+32} M_y + a_{N+33} M_x = 0 \]  
(19)

Through comparing Eqs. (18) and (13) as well as Eqs. (19) and (14), it can be deduced that:

\[ F_z = \frac{-M_x}{1 \sum_{i=1}^{N} y_i} = \frac{-M_y}{1 \sum_{i=1}^{N} x_i} \]  
(20)

The above derivation indicates that Eq. (20) must be true when the thrust of each hydraulic cylinder in the thrust system is equal. Next, it will be demonstrated that the thrust of each hydraulic cylinder is equal under the condition that Eq. (20) is true.

Through substituting Eq. (20) into Eqs. (10) and (11), and the below formula can be derived from Eqs. (13) and (14):

\[ \lambda_2 = \lambda_3 = 0 \]  
(21)

From Eq. (8), we can get:

\[ N\lambda_1 + \lambda_2 \sum_{i=1}^{N} x_i + \lambda_3 \sum_{i=1}^{N} y_i = 0 \]  
(22)

Substituting Eq. (21) into Eq. (22), we can get:

\[ \lambda_1 = 0 \]  
(23)

Through substituting Eqs. (21) and (23) into Eq. (6), Eq. (15) can be deduced to be established, that is, the thrust of each hydraulic cylinder in thrust system is equal.

When the hydraulic cylinder in thrust system is uniformly distributed, the following formula must be true:

\[ \sum_{i=1}^{N} x_i = 0 \]  
(24)

\[ \sum_{i=1}^{N} y_i = 0 \]  
(25)
Under the condition that Eqs. (24) and (25) are valid, the following formula can be obtained by combining Eq. (20):

\[
\begin{align*}
\sum_{i=1}^{N} x_i & \neq \frac{N M_y}{F_z} \\
\sum_{i=1}^{N} y_i & \neq \frac{-N M_x}{F_z}
\end{align*}
\] (26)

The results of the establishment of Eq. (26) show that: in the presence of transverse resistance torque \( M_y \) and longitudinal resistance torque \( M_x \), the thrust of each hydraulic cylinder in the uniform layout thrust system must be different, that is to say, the uniform layout thrust system must have the phenomenon of the partial load provided by the hydraulic cylinder.

When hydraulic cylinders in the thrust system are not uniformly distributed, if the layout coordinates of all hydraulic cylinders meet the condition of Eq. (20), the thrust of each hydraulic cylinder in the non-uniform thrust system will be equal, that is, the uniform force characteristics of the non-uniform thrust system. It can be seen from the above discussion that, under the condition of in the presence of transverse resistance torque \( M_y \) and longitudinal resistance torque \( M_x \), uniform thrust system must have partial load effect, while non-uniform thrust system can realize the anti-partial load effect. Therefore, in practical engineering applications, the uniform layout for the thrust system of shield tunneling machine under the condition of composite stratum will have serious off-load effect. The uniform force characteristics of the thrust system can be achieved by adjusting the layout position of the hydraulic cylinder to meet the condition of Eq. (20) through the layout adjustable mechanism.

3. Dimensional design of the adjustable layout mechanism of the shield thrust system

3.1 The angle design of the central angle corresponding to the sliding chute

![Diagram](image)

1-Shield head ; 2-Shield shell ; 3-Sliding chute ; 4-The middle joint ; 5-Partition of the chute ; 6-Dial ; 7-Propelling hydraulic cylinder ; 8-Supporting boot ; 9-Segment

Fig. 2 Structure distribution diagram of the shield thrust system
The specific structure of the shield thrust system is shown in Fig. 2. The propelling hydraulic cylinder slides in the chute, and the other end acts on the segment by means of a supporting boot. Since the maximum pressure of the supporting boot is 25 MPa, the maximum thrust of each hydraulic cylinder is 1500 kN. Therefore, the maximum angle of central angle corresponding to the chute in the thrust system can be determined by above two conditions, that is, the maximum adjustable range of adjustable layout mechanism, and the larger the adjustable range is, the more favorable it is for the non-uniform layout of the hydraulic cylinder in the thrust system, which lays a foundation for the later realization of the anti-bias load characteristics of the thrust system and provides greater possibility for the realization of the uniform force characteristics of the thrust system. And the design of the angle size is the basis of the related design size of the adjustable layout mechanism. Specific design calculation is as follows:

Fig. 3 shows that \( \alpha \) represents the included angle between two adjacent hydraulic cylinders. \( \beta \) represents the central angle corresponding to the supporting boot acting on the rear segment; \( \theta \) represents the angle between two adjacent supporting boots; \( r_1 \) and \( r_2 \) represent the inner and outer diameters of the rear segment ring. The minimum value of \( \beta \) can be obtained by the following formula:

\[
\frac{F_{\text{max}}}{S} \leq P_{\text{max}}
\]  

(27)

where \( F_{\text{max}} \) represents the size design under the maximum condition of meeting the thrust of hydraulic cylinder; \( P_{\text{max}} \) represents the maximum pressure that the supporting boots can bear; \( S \) represents the area of the supporting boot acting on the rear segment, and \[ S = \frac{1}{2} \beta_{\text{min}} (r_2^2 - r_1^2) \]; Through finding the relevant information, there are:

\( r_1 = 5400 \text{ mm}, r_2 = 6000 \text{ mm} \), \( F_{\text{max}} = 1500 \text{ KN} \), and \( P_{\text{max}} = 25 \text{ MPa} \). By substituting corresponding data, \( \beta_{\text{min}} = 4^\circ \) can be gotten.

On the ground of the relationship between \( \alpha, \beta \) and \( \theta \), there is:

\[
\theta = \alpha - \beta
\]  

(28)

where \( \alpha = 2\pi/N \), and here \( N = 24 \).

To design the maximum angle size of the center angle corresponding to the chute, the maximum value of \( \theta \) can be get
when and only when the center angle \( \beta \) corresponding to the rear segment acted by the supporting boot is minimized. From Eqs. (27) and (28) and the relevant data provided above, \( \theta_{\text{max}} = 11^\circ \) can be calculated.

### 3.2 The related size design of the layout adjustable mechanism

The adjustable layout mechanism designed in this paper drives the propelling hydraulic cylinder to slide in the chute through the expansion and contraction of the adjustable hydraulic cylinder, so as to realize the non-uniform layout of the propelling hydraulic cylinder. The specific mechanical schematic diagram is shown in Fig. 4.

![Mechanical schematic diagram of adjustable layout mechanism in the thrust system](image)

In this paper, the maximum transmission angle of the mechanism and the adjustable range of the adjustable hydraulic cylinder are combined for comprehensive design. The adjustable range of the adjustable hydraulic cylinder in this design is:

\[
l_2 - l_1 \leq l_1 + 100
\]

where \( l_2 \) represents the maximum length of the adjustable hydraulic cylinder; \( l_1 \) represents the length of the cylinder of the adjustable hydraulic cylinder.

As shown in Fig. 5, a circle is made through points \( B, C \) and \( O \). Point \( A \) is the position of the rack, which can move on the circle. The arc length \( \overrightarrow{BC} \) represents the length of the chute, and the center angle \( \theta \) corresponding to this arc length is the maximum value of the center angle corresponding to the chute, that is \( \theta = 11^\circ \). Point \( D \) is the midpoint of the arc length \( \overrightarrow{BD} \). Point \( E \) is the point where the linear vector \( \overrightarrow{OD} \) intersects the above circle.

According to the analysis in Fig. 5, \( \angle ABO, \angle ADO \) and \( \angle ACO \) are respectively the transmission angles of the layout adjustable mechanism at points \( B, D \) and \( C \). The relations among all angles are as follows:

\[
\angle ABO = \angle AEO = \angle ACO \quad (30)
\]

\[
\angle DAE + \angle AEO = \angle ADO \quad (31)
\]

From Eqs. (30) and (31), we can get:

\[
\angle ABO = \angle ACO < \angle ADO \quad (32)
\]
While point $D$ is the midpoint of the arc $\overline{BDC}$, it is easy to know that in the process of the adjustable hydraulic cylinder moving from point $B$ to point $C$, the transmission angle at point $D$ is the largest, namely $\angle DBO = \gamma_{\text{max}}$, and the transmission angle at point $B$ and $C$ is the smallest, namely $\angle ABO = \angle ACO = \gamma_{\text{min}}$.

The above analysis shows that, no matter how point $A$ moves in the circle shown in Fig. 5, the transmission angle of the mechanism at point $B$ and point $C$ is the smallest and the transmission angle at point $D$ is the largest. The next step is to determine exactly where point $A$ is on the circle.

Let them be $AB = l_1$, $AC = l_2$, $OC = r$, and $r$ represents the layout radius of the thrust system and $r = 2815$ mm is set. According to the adjustable range of the adjustable hydraulic cylinder specified in inequality (29), when point $A$ approaches point $B$ on the circumference, the transmission angle $\angle ABO$ at point $B$ will increase, but at this moment $AB$ will decrease, and inequality (29) cannot be satisfied. And when point $A$ approaches point $O$ on the circumference, $AB$ will become larger, but at this time, the transmission angle $\angle ABO$ at point $B$ will become smaller. From the perspective of force transmission performance of the bar mechanism, it can be known that the larger the transmission angle is, the better the transmission performance will be. Therefore, considering the maximization of transmission angle and the adjustable range of the adjustable hydraulic cylinder, the position of point $A$ obtained under the condition that the equal sign in the inequality (29) is satisfied will be selected as the position of the rack in the adjustable layout mechanism.

At this time there are:

$$\angle BAC = \angle BOC = \theta$$  \hspace{1cm} (33)

$$|BC|^2 = 2r \sin \left( \frac{\theta}{2} \right)$$  \hspace{1cm} (34)

$$l_2 = 2l_1 + 100$$  \hspace{1cm} (35)

$$\cos \angle BAC = \frac{l_1^2 + l_2^2 - |BC|^2}{2l_1 \cdot l_2}$$  \hspace{1cm} (36)
By combining Eqs. (33) ~ (36) and substituting all the parameters in the equations, the solution is: $|AB| = 616\, mm, |AC| = 1132\, mm$. Furthermore, the transmission angle $\angle ABO = 72^\circ$ of the layout adjustable mechanism at point $B$ can be further solved. In the process of the mechanism moving from point $B$ to point $C$, there is always the transmission angle $\gamma \geq 72^\circ$. The final design sizes of the layout adjustable mechanism are shown in Fig. 6.

![Fig. 6 Schematic diagram of specific design size of the layout adjustable mechanism](image)

4 Case comparison analysis

Through the theoretical analysis in the second section, it can be known that under the premise of the existence of resistance torque, the uniform layout for hydraulic cylinder in thrust system will inevitably produce partial load effect, so the hydraulic cylinder needs to adopt the non-uniform layout to realize the anti-partial load. However, not all non-uniform layouts can effectively mitigate the impact of partial load effect, and the law of uniform force characteristics of the thrust system provides a possibility to find an arrangement under which the thrust of each hydraulic cylinder in the system is equal. It is easy to find that the layout of each hydraulic cylinder obtained by this law is adaptable to external loads generated under a composite stratum. Therefore, under the constraint of specific variation range of external load and the adjustable range of the designed adjustable layout mechanism, the layout phase angle of each hydraulic cylinder determined according to Eq. (20) and the elongation of the adjustable hydraulic cylinder in the adjustable layout mechanism are shown in Table. 1, while the change of external loads is shown in Fig. 7.

![Fig. 7 Schematic diagram about the change of external loads $F_z$, $M_x$, and $M_y$](image)
According to the layout phase angle of the hydraulic cylinder in original thrust system in Table 1, a virtual prototype model is established in SolidWorks, and then the established model is imported into ADAMS for virtual prototype simulation. The specific prototype simulation model is shown in Fig. 8, and the data of external loads applied in the simulation process are shown in Fig. 7. The thrust values of the 10th, 20th and 24th propelling hydraulic cylinders obtained from the simulation result are compared with the numerical simulation results calculated in MATLAB according to Eq. (8). The comparison results are shown in Fig. 9 - 11. It can be seen from the results in Fig. 9 - 11 that the thrust results of the above two kinds of simulations tend to be consistent, which verifies the correctness of the mechanical model of the shield thrust system built in the second section.
Fig. 9 Thrust comparison of the 10th jack of the uniform layout thrust system

Fig. 10 Thrust comparison of the 20th jack of the uniform layout thrust system

Fig. 11 Thrust comparison of the 24th jack of the uniform layout thrust system

On the ground of the layout phase angle of the hydraulic cylinder in thrust system after adjustment in Table. 1, a virtual prototype model is established in SolidWorks, and then the established model is imported into ADAMS for virtual prototype simulation. The specific prototype simulation model is shown in Fig. 12. Similarly, the thrust values of the 10th, 20th and 24th adjusted propelling hydraulic cylinders obtained from the simulation results are compared with the numerical simulation results calculated in MATLAB according to Eq. (8). The comparison results are shown in Fig. 13 - 15. According to the results obtained in Fig. 13 - 15, the thrust values of the above two simulations tend to be consistent, which verifies the correctness of the conclusion obtained from Eq. (20).
Fig. 12. The model in ADAMS of adjusted layout for hydraulic cylinder in thrust system

Fig. 13 Thrust comparison of the 10th jack of adjusted layout thrust system

Fig. 14 Thrust comparison of the 20th jack of adjusted layout thrust system
In Fig. 9 - 11 and Fig. 13 - 15, the forces applied by jacks of the same number in two different layout thrust systems are compared. It is easy to find that, the change of forces exerted by hydraulic cylinders No. 10, 20 and 24 in adjusted non-uniform thrust system is 685 KN to 715 KN, while the change of forces exerted by hydraulic cylinders No. 10, 20 and 24 in the original is 660 KN to 770 KN, which indicates that the thrust of the hydraulic cylinder in adjusted non-uniform thrust system can appear more evenly. Fig. 10 shows that the maximum thrust of the 20th hydraulic cylinder in the uniform thrust system is close to 770 KN, much larger than the maximum thrust of 715 KN for the hydraulic cylinder in adjusted non-uniform thrust system (as shown in Fig. 14), which manifests that adjusted non-uniform thrust system can protect the segments better and save energy effectively.

Based on the Ref. (Deng et al., 2016), it can be known that the CV (variation coefficient) value is an effective way to evaluate the thrust uniformity of hydraulic cylinders in thrust system. The smaller CV value is, the closer the thrust value of each hydraulic cylinder is. The thrust value of each hydraulic cylinder can be solved by substituting the original and adjusted layout phase angle of hydraulic cylinder in Table. 1 and the value of external loads shown in Fig. 7 into Eq. (8), and then substituting these values into Eq. (10) in Ref. (Deng et al., 2016) to solve CV value. The change of CV value is shown in Fig. 16.

From the results shown in Fig. 16, the thrust CV value of adjusted hydraulic cylinder in the thrust system compared to that of the original is smaller, indicating that the thrust of adjusted hydraulic cylinder in thrust system tends to be more uniform, more conducive to offset the negative influence of partial load, and can better prevent the rear segment rupture.
or crushing.

5 Conclusions

In order to avoid serious partial load effect, based on the uniform force characteristics of thrust system, non-uniform distribution of the hydraulic cylinder in thrust system can be realized through the adjustable layout mechanism. However, in the composite stratum, the external load acting on the shield thrust system is not a constant, so a certain non-uniform layout can only achieve complete anti-bias effect under a certain external load condition. When the external loads are changing in a certain interval, this particular non-uniform layout cannot completely offset the influence of partial load. But compared to the uniform layout thrust system, the non-uniform distribution effectively weakens the effect of partial load effect. It can be seen from the result of the virtual prototype simulation: compared with the original thrust system, the variation coefficient of the non-uniform thrust system adjusted by the adjustable layout mechanism is much smaller, which indicates that the force transmission performance of the adjusted non-uniform thrust system is better and the anti-bias effect is better. The specific conclusions are as follows:

(1) Under the action of transverse resistance torque and longitudinal resistance torque, when the hydraulic cylinders of the shield thrust system are uniformly distributed, the thrust provided by the hydraulic cylinders must have partial load effect.

(2) Under the condition of composite stratum, when the layout for hydraulic cylinder in the shield thrust system satisfies the equation of uniform force characteristic law, the thrust provided by the hydraulic cylinder will be the same everywhere, so as to achieve complete anti-partial load effect.

(3) The adjustable layout mechanism designed in this paper has the smallest transmission angle when the adjustable hydraulic cylinder pushes the propelling hydraulic cylinder from the starting to ending positions of the chute, that is, the mechanism can always ensure that the transmission angle is larger than the starting and ending positions in the moving process of the chute. This design method greatly improves the force transmission performance of the bar mechanism.

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