Journal of Science and Technology in Civil Engineering, HUCE, 2024, 18 (1): 130–142

UTILIZING NUMERICAL SIMULATION TO DETERMINE THE REASONABLE PARAMETERS FOR THE DESIGN OF HYDRAULIC ROCK-SPLITTING HEAD

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Article history:
Received 17/10/2023, Revised 14/3/2024, Accepted 20/3/2024

Abstract

The hydraulic rock splitter with a wedge-type splitting head has various advantages, such as simple operation, high work efficiency, safety, shortened work time, and minimal environmental impact. The splitting head driven by a hydraulic cylinder, is the focus of this research, as it directly influences work efficiency. This paper introduces the general structure of the rock splitter machine and the hydraulic cylinder-driven splitting head, along with the method for selecting suitable parameters. To evaluate and select suitable variables, this study establishes procedures and calculation methods for two sets of problems: determining the parameters for the wedge and the parameters for the hydraulic cylinder. The numerical test results provide a design dataset applicable to real-world scenarios, including the wedge angle, cylinder stroke, cylinder diameter, pressure, minimum cylinder wall thickness, and piston rod diameter, corresponding to the respective rock-splitting forces. These parameters meet the set objectives of cylinder pushing force and minimize the cross-sectional area of the cylinder body, contributing to weight reduction and operational convenience.

Keywords: hydraulic cylinder; rock splitting; splitting force; pushing force; wedge-shaped.

https://doi.org/10.31814/stce.huce2024-18(1)-10

1. Introduction

Currently, there are various technologies and rock-splitting equipment that offer numerous advantages, including hydraulic rock-splitting devices. This technology has been widely adopted by countries around the world. Compared to traditional construction methods, using a rock splitter has several advantages, such as simple operation, high work efficiency, safety, shortened work time, and notably, minimal environmental impact [1–3]. A hydraulic rock splitter comprises a hydraulic power unit, control valves, and a wedge-type rock-splitting head. Fig. 1 illustrates the working principle of the rock-splitting head. Initially, drills are used to create holes in the rock. Subsequently, the rock-splitting head is inserted into the hole (1). The wedge is pushed down, causing the wedge wings to open and generate a separating force (2). This force will split the rock from within (3), and the rock will split in the predetermined direction [1, 2].

Worldwide, the initial guidance on using rock splitters originated in the United States through the work of Roy L. Campbell when exploring concrete demolition options for the military [2]. The designs during this time were relatively simple, with a small separating force. Subsequently, hydraulic wedge-type rock splitter demonstrated remarkable success in both the industrial and construction sectors [3]. In [4], a preliminary assessment of the potential application of hydraulic splitting cylinders was conducted to propose modifications for more convenient operation. Beyond researching the utilization

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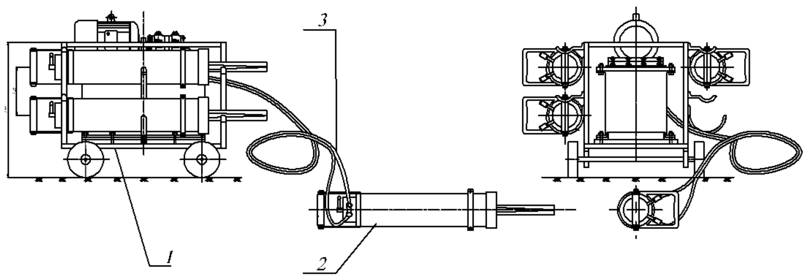
Figure 1. The working principle of the wedge-type rock-splitting head

of rock splitters or concrete in construction and mining, the authors in [5] also delved into their application for splitting rock in the production of schist.

The invention, described in [6], involves a wood-splitting axe that utilizes the wedge principle to split wood. It consists of two wedge wings with have cutting blade rotating around the ball joint, a movable wedge between the two wedge wings, and the splitting force generated by the kinetic energy when chopping wood. In [7], the patent addresses an improved hydraulic rock-splitting machine designed to prevent the deformation of the wedge and wedge wings. The innovation here involves the use of leaf springs in the connection between the wedge wings and the equipment body. However, the solution presented in [7] has a rather complex structure. A type of rock splitter combined with new tunneling technology is presented in [8]. The basic components of this machine include the rock-splitting mechanism, the support table, the lifting system, the hydraulic control system, the support components, and fixed parts. A fundamental design calculation for an industrial rock splitter is presented in [9]. This machine can be applied to both natural and artificial stones for the production of cobblestones, road curbs stone, and cladding stones. This machine operates based on Pascal's law and utilizes devices such as pressure compensators, check valves, directional control valves, dual-acting cylinders, pumps, and pressure-reducing valves. The hydraulic circuit design of the system is simulated using Fluidsim software. Structural durability is calculated and simulated using Solidworks software. Over time, rock splitters have continuously improved in structure, and are better applied in various fields. However, research on determining the optimum parameters for wedge-type hydraulic rock splitting heads is still not widely disseminated or remains the proprietary technology of manufacturers.

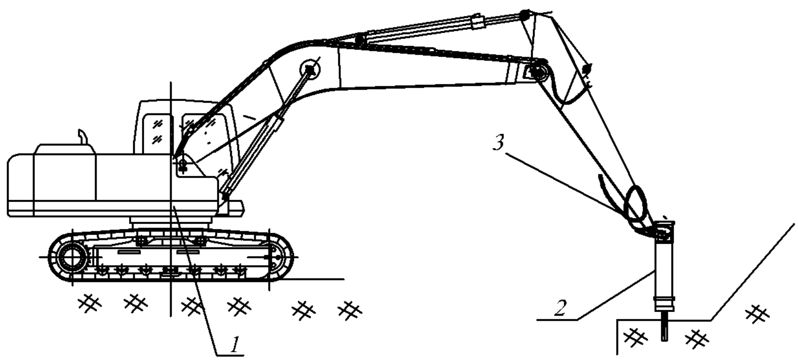
The foundational principles of calculations related to rock or concrete, mechanical design, and hydraulic transmission have been presented in [10–15]. In [10] a theoretical analysis of rock breaking containing hole is made, and then a numerical model of rock breaking under hole assistance is established. The rock-breaking mechanism is explained, including the reasons for crack formation, crack initiation, and propagation. The fracturing pressure and crack propagations are also investigated. From this paper, the splitting force of hydraulic splitter under hole assistance can calculated. In [11], general issues regarding rock or concrete materials are discussed, along with some models for crushing problems and wedge-clamping in machines for construction materials production. In [12], guidelines for designing elements of construction machinery are presented, including some applications of the wedge principle, such as safety brakes and working conditions of the mechanism. General hydraulic transmission system design is addressed in [13–15]. Mechanical detail design and applications are covered in [16–18]. Thus, it can be asserted that the foundational principles of designing wedge-type hydraulic rock splitter have been addressed in basic research.

Optimal design is currently a research area of significant interest among scientists. In optimization techniques, they are broadly categorized into two main groups: traditional methods and advanced optimization methods [19], [20]. Traditional optimization methods have been utilized for a long time, such as nonlinear programming, geometric programming, dynamic programming, and gradient methods. Although these methods can solve numerous problems in technology, they also come with various limitations. Traditional methods with analytical functions face challenges in solving complex problems with discrete variables and a large number of variables. With the robust development of computational software, numerical methods have solved this problem. Albeit not achieving complete optimality, numerical methods are suitable for engineering problems. Several optimization methods using numerical approaches and their fundamental applications in mechanical engineering are discussed in [20]. Additionally, [21] has applied the Taguchi method based on the theory of statistical probability to optimize the geometric parameters of a cone brake.



1- Hydraulic power unit, 2- Hydraulic-driven rock splitting head, 3- Hydraulic pipes and control valves

Figure 2. Hydraulic-driven rock splitter [3]



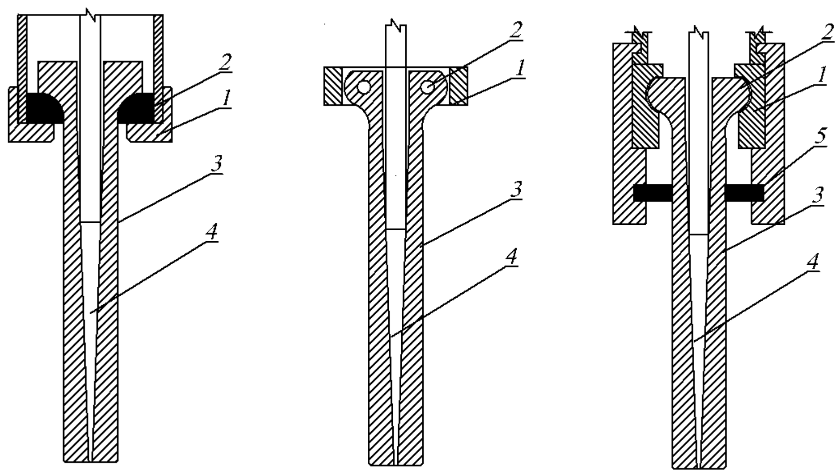
1- Base machine; 2- Hydraulic-driven rock splitting head; 3- Hydraulic pipes and control valves

Figure 3. Rock splitting head mounted on a self-propelled vehicle [3]

Through a comprehensive analysis, it is evident that hydraulic-driven rock splitters with wedge-type rock splitting heads offer numerous advantages. However, if the rock-splitting head is large and weighs, it can pose difficulties in practical utilization. On the other hand, research on determining reasonable parameters for hydraulic wedge-type rock-splitting heads remains not widely disseminated. Therefore, this paper will address the mentioned issues. Building upon existing studies, the paper introduces the general structure of rock splitters and hydraulic-driven wedge-type rock splitting heads. Subsequently, the research concentrates on establishing the design and calculation basis for the wedge-type rock-splitting head, including the process and methods for determining the optimum parameters of the wedge and the hydraulic cylinder. This study will employ numerical methods, investigated through software such as Minitab, Matlab, and Ansys. The numerical results provide a set of design data, such as the wedge angle, hydraulic cylinder stroke, cylinder diameter, pressure, minimum wall thickness of the cylinder, and piston rod diameter, corresponding to various rock-splitting forces. These parameters meet the set objectives of minimizing the hydraulic cylinder pushing force and minimizing the cross-sectional area of the cylinder body. The optimum parameters contribute to reducing the weight and size of the machine, making it convenient for operation. Other parameters such as the length of wedge wing, the width of wedge and wedge wing depend on dimension of hole assistance are not optimized in this paper.

2. The structure of the hydraulic-driven wedge-type rock splitter

The structure of the rock splitter is illustrated in Fig. 2 and Fig. 3. The rock splitter consists of three parts: the base machine or hydraulic power unit (1), the rock splitting head (2), and the control valve (3). The hydraulic power unit (1) ensures the supply of pressurized oil and flow to the working components. The rock-splitting head (2) can be installed independently, as shown in Fig. 2, or mounted on a self-propelled vehicle, as depicted in Fig. 3. Consequently, it requires dimensions and weight that are suitable for usage conditions and the ability to operate manually.



(a) Through a cushion (b) Through a hinge (c) Using a bushing

1- machine body, 2- connecting part from wedge wing to wedge, 3- wedge wing, 4- wedge, 5- cushion

Figure 4. The rock-splitting head assembly the wedge connected to the wedge wing [3]

The rock-splitting head assembly proposed in [3] has the structure as depicted in Fig. 4. The wedge and wedge wing are arranged in various configurations, as illustrated in Fig. 4. In Fig. 4(a), the wedge (4) is connected to the wedge wing (3) through a cushion (2). The force from the cushion presses the wedge wing tightly against the wedge. The cushion is held in place by the machine body (1). The advantage of this configuration is that the force between the wedge and wedge wing remains stable during operation, reducing vibrations, and easy adjustment of the opening angle during work. The drawback of this approach is the complexity of assembly and intricate connections. Fig. 4(b) demonstrates the wedge (4) linked to the wedge wing (3) through a pin (2) via a hinge. This linking method has the advantage of simplicity in assembly but suffers from instability during operation due to the inconsistent force between the wedge and the wedge wing. In Fig. 4(c), the wedge (4) is connected to the wedge wing (3) using a bushing (2) and a cushion (5). Through the machine body, the bushing and cushion apply pressure to the wedge wing. The advantage of the configuration in Fig. 4(c) is stable operation, while the downside includes complexity, manufacturing challenges, and difficulty in assembly.

3. Determine the parameters of the hydraulic-driven wedge-type rock splitter head

3.1. Determine the parameters of wedge and wedge wing

Considering the model in Fig. 5, the parameters of the wedge and wedge wing include the wedge angle α (degrees), the wedge displacement x (mm) with the maximum value of H (mm) corresponding to the stroke of the hydraulic cylinder, and the opening of wedge wing a (mm). The frictional force between the wedge and wedge wing is caused by the force splitting rock:

$$F = fN \tag{1}$$

where f is the friction coefficient between the wedge and wedge wing.

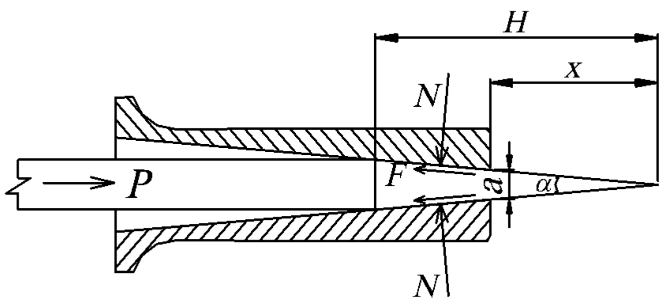


Figure 5. The model to calculate the force acting on the wedge

Balancing the forces along the hydraulic cylinder push direction P (N), and substituting the frictional force from formula (1), we have:

$$P = 2\left(fN\cos\frac{\alpha}{2} + N\sin\frac{\alpha}{2}\right) \tag{2}$$

where P (N) is the hydraulic cylinder pushing force, and N(N) is the rock splitting force. When the rock-fracturing mechanism using a hydraulic splitter under hole assistance, the rock splitting force can be calculated according to [10].

Formula (2) provides the relationship between the pushing force and the parameters of the wedge and wedge wing.

In the model shown in Fig. 5, the opening of wedge wing a (mm) is determined by the wedge displacement x (mm) corresponding to the wedge angle α (degrees):

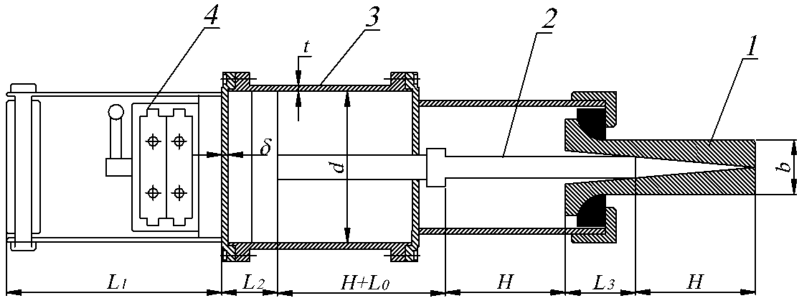
$$x = \frac{a}{2\tan\dfrac{\alpha}{2}} \tag{3}$$

When the displacement of the hydraulic cylinder maximum at H (mm), the opening a is maximum.

The parameters of the wedge, wedge wing, and hydraulic cylinder must be adjusted to ensure that the equipment's length remains below the specified value.

$$3H + L_1 + L_2 + L_3 + L_0 \leq [L_t] \tag{4}$$

where L_1 is the handle length (mm), L_2 is the distance from the outer bottom of the hydraulic cylinder to the piston (mm), $H + L_0$ is the length of the piston rod (mm), L_3 is the non-contact surface distance of the wedge (mm), $[L_t]$ is the maximum allowable length (mm). Fig. 6 presents the structure of the rock-splitting head and describes these distances.



1- wedge wing; 2- wedge; 3- hydraulic cylinder; 4- hydraulic control valves

Figure 6. Rock splitting head

3.2. Hydraulic cylinder design

A hydraulic cylinder is used to generate pushing force for the wedge. During operation, the hydraulic cylinder is under the pressure created by the working fluid and external loads. In this study, the problem of determining the parameters of the hydraulic cylinder is addressed for a slender cylinder, where the ratio of the outer diameter to the inner diameter is less than 1.2.

The stability conditions according to [14, 16] state that the pushing force required for the piston must satisfy Eq. (5):

$$P \leq \frac{P_{th}}{k_0 n_0} \tag{5}$$

where k_0 is the coefficient pertains to the ability to increase pressure within the system, under normal circumstances $k_0 \approx 1.15$, n_0 is the safety factor for stability, $n_0 \geq 3$ with steel material, and P_{th} is the limit force (N).

$$P_{th} = c\frac{\pi^2 EJ}{(H + L_0)^2}; \quad J \approx 0.05 d_p^4 \tag{6}$$

where c is the coefficient depend on piston rod linking, for the structure in Fig. 6, $c = 2$, E is the modulus of elasticity of the piston rod material, for steel material, $E = 2.1\times10^5$ N/mm^2. L_0 is piston rod length (mm), J is the moment of inertia of the piston rod, with a solid cylindrical $J \approx 0.05d_p^4$, and d_p is the piston rod diameter (mm).

Therefore, according to the conditions of strength and stability, the minimum required diameter of the piston rod must satisfy:

$$\begin{cases} d_p \geq 2\sqrt{\dfrac{P}{\pi[\sigma]}} \\ d_p \geq \left(\dfrac{Pk_0 n_0 (H + L_0)^2}{0.05c\pi^2 E}\right)^{0.25} \end{cases} \tag{7}$$

where $[\sigma]$ is the allowable stress of the piston rod material (N/mm^2).

The hydraulic oil pressure required to generate the pushing force P when neglecting the opposing force is:

$$p = \frac{4P}{\pi d^2} \tag{8}$$

where p is pressure in the cylinder chamber (N/mm^2), and d is the cylinder bore diameter (mm).

Stress caused by pressure in the cylinder is calculated according to the Hook's law, and must satisfy the condition [16]:

$$\frac{pd}{2t} \leq [\sigma] \tag{9}$$

where t is the cylinder wall thickness (mm).

In the case of a cylinder with a flat bottom, the minimum thickness of the cylinder end cap is [17]:

$$\delta = \sqrt{\frac{3pd^2}{32\,[\sigma]}(3+\nu)} \tag{10}$$

where δ is the cylinder end cap thickness (mm), and ν is Poisson's ratio of material, for steel $\nu = 0.3$.

The area of the cylinder body according to [19] is:

$$A = \pi t(d+t) \tag{11}$$

From formulas (8) to (11), it follows that:

$$\begin{cases} A = \dfrac{2P}{[\sigma]} + \dfrac{1}{\pi}\left(\dfrac{2P}{d[\sigma]}\right)^2 \\ \delta = \sqrt{\dfrac{3P}{8\pi[\sigma]}(3+\nu)} \end{cases} \tag{12}$$

So the problem model is formulated to determine the basic parameters of the hydraulic cylinder, including given values for X_0, design parameters X, and problem conditions:

$$X_0 = \{[\sigma], k, d_{\min}, t_{\min}, \delta_{\min}\} \tag{13}$$

$$X = \{d, t\} \tag{14}$$

$$\begin{cases} t = \dfrac{2P}{\pi d[\sigma]} \geq [t_{\min}] \\ \delta = \sqrt{\dfrac{3P}{8\pi[\sigma]}(3+\nu)} \geq [\delta_{\min}] \\ d \geq [d_{\min}] \end{cases} \tag{15}$$

where $[d_{\min}]$ is the minimum hydraulic cylinder bore diameter determined according to structural requirements (mm), $[t_{\min}]$ is the minimum thickness of the cylinder body to meet structural and technological requirements (mm), and $[\delta_{\min}]$ is the minimum thickness of the cylinder end cap to meet structural and technological requirements (mm).

Eq. (15) indicates that the thickness of the cylinder end cap is independent of the design parameters. Therefore, to minimize the weight of the hydraulic cylinder, the cross-sectional area of the cylinder body must be minimized.

$$A = \frac{2P}{[\sigma]} + \frac{1}{\pi}\left(\frac{2P}{d[\sigma]}\right)^2 \to \min \tag{16}$$

4. Evaluate the influence of parameters

4.1. Method for evaluating the influence of parameters

The methods and procedure of this study are described in Fig. 7. The given parameters are indicated in Table 1 based on data from the fabrication material, technological requirements, and coefficients from specialized literature. Utilizing actual value ranges, this research employs four levels of values, as shown in Table 2.

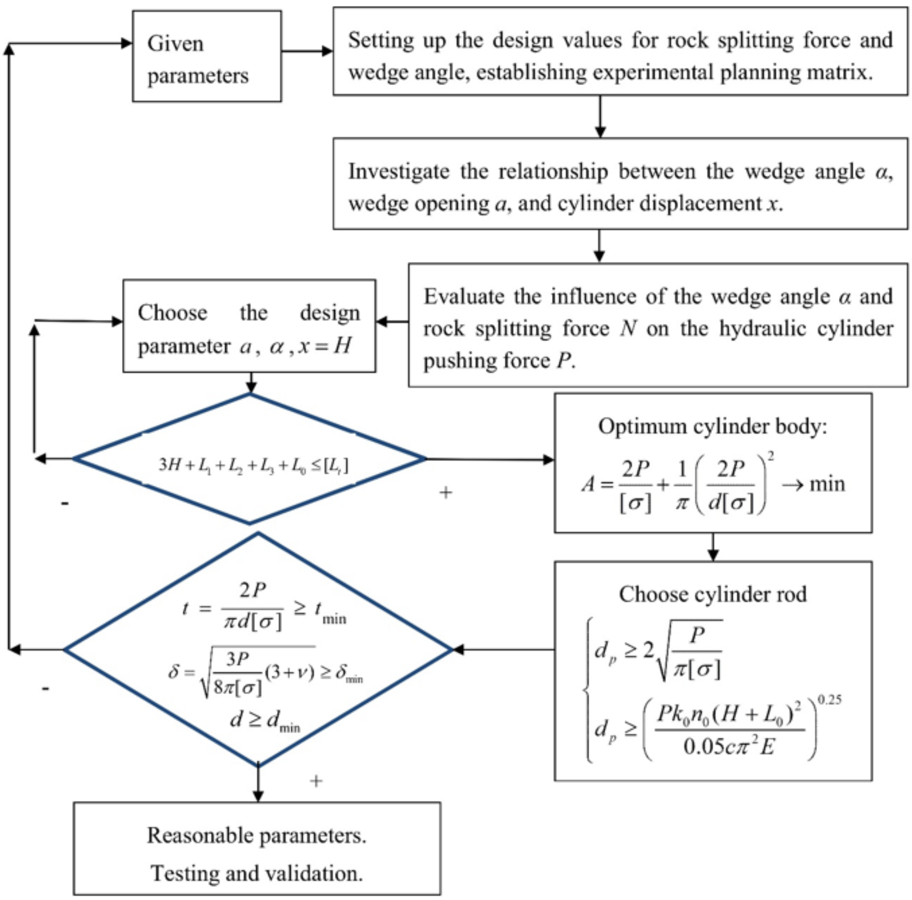


Figure 7. Methods and procedures for determining design parameters

In the first step, a procedure is conducted to select design parameters, including the wedge opening a, the wedge angle α, and the piston displacement $x = H$, satisfying formula (4) with $[L_t] = 1240$ mm [3]. The response function is the piston pushing force P according to formula (2), based on the use of an orthogonal matrix L16. The second step is designing the hydraulic cylinder, involves calculating the piston rod, and optimizing the piston body. Two design parameters are the diameter of the piston rod and the cylinder wall thickness. The response function is the cross-sectional area of the cylinder, to minimize this area.

Table 1. The given parameters

f	E (N/mm^2)	$[\sigma]$ (N/mm^2)	k_0	n_0	c	$[d_{\min}]$ (mm)	$[t_{\min}]$ (mm)	$[\delta_{\min}]$ (mm)
0.2	2.1×10^5	480	1.15	3	2	80	5	5

Table 2. Four levels values of splitting force and wedge angle

Parameter	Symbol	Level of values			
		1	2	3	4
Splitting force (kN)	N	2000	3000	4000	5000
Wedge angle (degree)	α	5	10	15	20

The finite element method is also employed by Ansys software to assess reasonable parameters for the cylinder body and wedge. The pressure in the cylinder chamber is tested according to formula (8). It is assumed that the force applied is uniformly distributed on the wedge and wedge wing surfaces. Here, the pressure p_{CH} (N/mm^2) acting on the surfaces of the wedge and wedge wing is calculated as:

$$p_{CH} = \frac{N}{A_{ch}} \tag{17}$$

where A is the contact area between the wedge and wedge wing surface (mm^2).

4.2. Calculation results

Fig. 8 presents the influence of the wedge angle and rock splitting force on the hydraulic cylinder pushing force, with response values generated by the L16 orthogonal matrix and Minitab software. Within the investigated range of wedge angles from 5 degrees to 20 degrees and rock splitting forces from 2000 kN to 5000 kN. The required hydraulic cylinder pushing force varies from 973 kN (at 5 degrees and 2000 kN rock splitting force) to 3705 kN (at 20 degrees and 5000 kN rock splitting force). A smaller wedge angle results in a lower hydraulic cylinder pushing force, but Fig. 9 depicts that the cylinder stroke is substantial enough to achieve the required wedge opening.

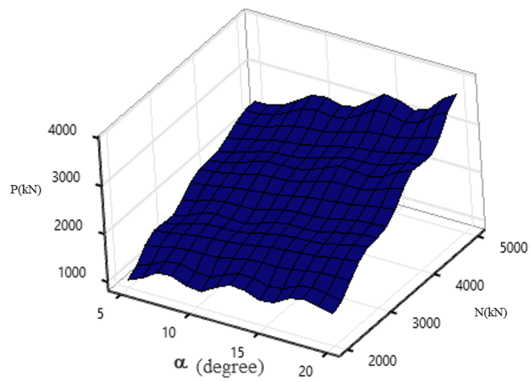


Figure 8. Influence of the wedge angle and rock splitting force on hydraulic cylinder pushing force

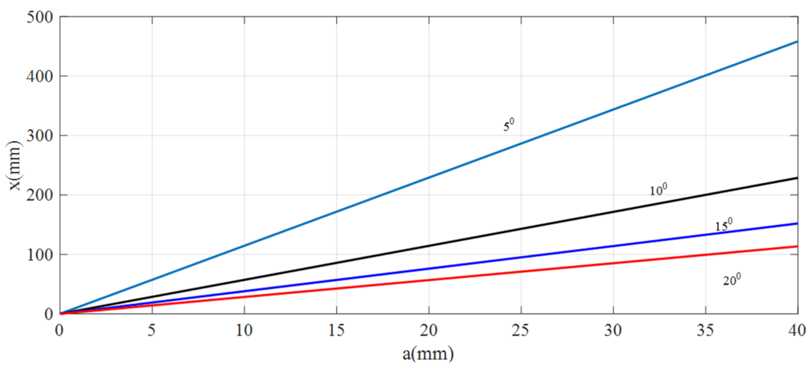


Figure 9. Influence of the wedge opening on the hydraulic cylinder stroke at various wedge angle values

Assuming the design and usage conditions with L_1 = 350 mm, L_2 = 90 mm, L_3 = 116 mm, L_0 = 80 mm, designed wedge opening of a = 35 mm (Fig. 6). The calculated results from Fig. 9 provide essential data. With wedge opening of a = 35 mm, we have the maximum hydraulic cylinder stroke H = 400 mm at wedge angle 5 degrees, H = 200 mm at wedge angle 10 degrees, H = 130 mm at wedge 15 degrees, and H = 100 mm at wedge angle 20 degrees. The overall length condition for transportation and installation must satisfy $[L_t]$ = 1240 mm. In this case, the hydraulic cylinder stroke $H \leq 200$ mm satisfies this condition. When H = 200 mm, the total length of the device is 1236 mm; when H = 130 mm, the total length is 1026 mm; and when H = 100 mm, the total length is 936 mm. Fig. 8 illustrates that a smaller wedge angle results in a lower hydraulic cylinder pushing force, requiring a larger cylinder stroke to achieve the designed wedge opening. A larger wedge angle provides a smaller cylinder stroke but demands a higher pushing force, leading to increased pressure and cylinder diameter. Therefore, we choose a cylinder stroke of H = 200 mm, as it corresponds to the minimum hydraulic cylinder pushing force according to the response function.

Calculations and graphical tools in Matlab software provide calculation results, as shown in Fig. 10. The results indicate that as the hydraulic cylinder diameter increases, the cross-sectional area of the cylinder body decreases. However, beyond a certain value, the cross-sectional area of the cylinder body remains almost constant, which represents a reasonable value. This relationship is a common pattern when calculating hydraulic cylinders. From the reasonable hydraulic cylinder diameter value, we can determine the appropriate cylinder wall thickness and required pressure. Table 3 presents the hydraulic cylinder design dataset with corresponding rock-splitting forces. Depending on the usage requirements, a suitable set of parameters can be selected for the design.

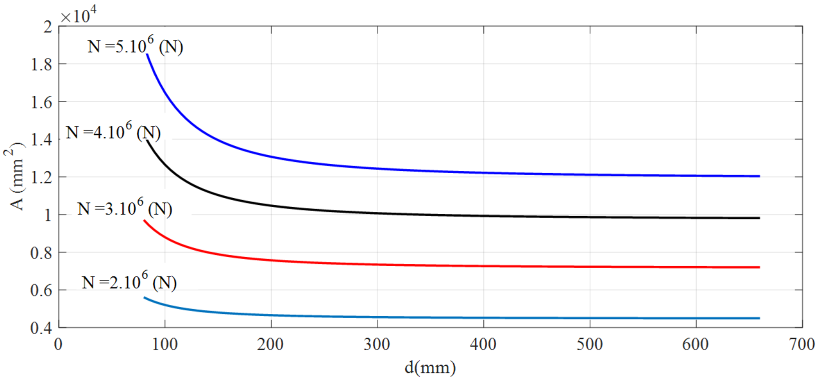


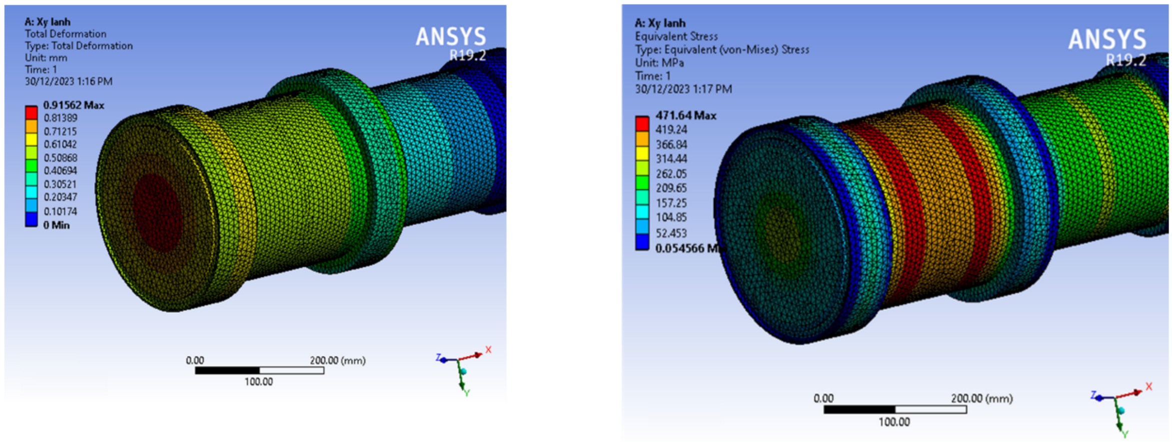
Figure 10. Influence of the hydraulic cylinder diameter on the cross-sectional area of the cylinder body at various rock splitting force values

Table 3. Hydraulic Cylinder Design Data

N (N)	d (mm)	A (mm^2)	t (mm)	δ (mm)	p (N/mm^2)	P (N)
2000000	200	4953	7.6	30.5	36	1145000
3000000	250	7419	9.1	37.5	35	1718000
4000000	300	9863	10	43.1	32	2290000
5000000	350	12299	10.8	48.7	30	2863000

Data in Table 3 determines the required cylinder wall thickness corresponding to design pressure, with an allowable stress of $[\sigma]$ = 480 N/mm^2. The objective is to fully utilize the material's ability with the specified safety factor. To validate the calculation results, the article provides an example calculation for the hydraulic cylinder design experiment with rock splitting force of N = 3000 kN, wedge opening a = 35 mm, cylinder stroke H = 200mm, and wedge angle α = 10 degrees. Using the data in Table 3, we select hydraulic cylinder diameter d = 250mm, cylinder wall thickness t = 9.1 mm, required pressure p = 35 N/mm^2, and cylinder end cap thickness δ = 37.5 mm. The piston rod diameter is d_p = 40 mm, and the piston rod length is $H + L_0$ = 280 mm. The pressure on the wedge surface, according to formula (17), is p_{CH} = 166 N/mm^2.

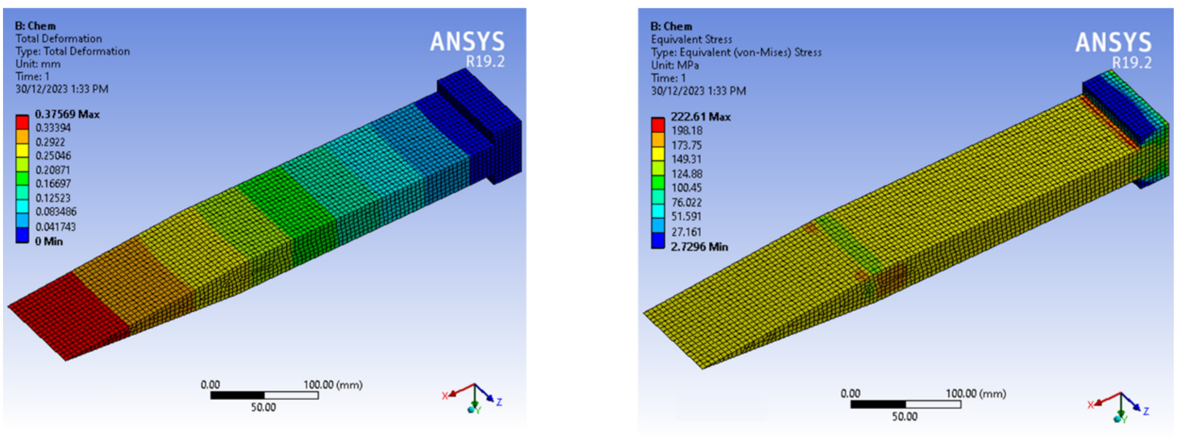
The simulation results using Ansys for the structural strength of the cylinder body and wedge are presented in Figs. 11 and 12, respectively. In Fig. 11, the maximum von Mises stress in the cylinder body is 471.64 N/mm^2, with a maximum deformation of 0.92 mm. The highest stress occurs in the cylinder wall, and the greatest deformation is at the bottom of the cylinder end cap. The stress deviation value is 1.9% compared to the allowable design stress indicating that the calculating results in Table 3 are accurate and appropriate. For the simulation result of the wedge in Fig. 12, the maximum von Mises stress is 222.61 N/mm^2, ensuring below the allowable stress, and the maximum deformation is 0.38 mm.



(a) Deformation

(b) von Mises stress

Figure 11. Simulation results of hydraulic cylinder



(a) Deformation

(b) von-Mises stress

Figure 12. Simulation results of wedge

4.3. Discussions

In this study, an investigation has been conducted for common hydraulic cylinders in the rock-splitting head. While the calculated values have not yet reached optimal results, they align well with engineering requirements. The research outcomes have established a set of hydraulic cylinder design data for use in the design of rock-splitting equipment. In the example experiment, the simulation using Ansys software exhibits some deviation from the allowable design stress. To enhance accuracy, adjusting the meshing in the program may be considered.

Table 3 provides reasonable hydraulic oil pressure values within a low-pressure range. Higher pressures would result in a reduction in the cylinder diameter, an increase in the cylinder wall thickness, and consequently, an increase in mass. The data in Table 3 shows that if the rock-splitting force is significant, the required cylinder diameter is large, making it challenging to meet installation and manual operation conditions. Therefore, this method is more suitable for working heads with rock-splitting forces of less than 300 tons. The impact on the hydraulic cylinder pushing force has only been considered with the wedge angle and predefined friction coefficient. However, from formula (2), it is evident that the influence of the friction coefficient between the wedge and wedge wing surface is substantial. To reduce the hydraulic cylinder pushing force, mechanical machining of the wedge and wedge wing surfaces is necessary to achieve the minimum friction coefficient. The external dimensions of the wedge wing, and the wedge wing opening, depend on the hole. The mechanism for separating rock from boreholes is presented in [10]. In case the wedge wing sizes do not match the hole, it will affect working efficiency and reduce productivity.

The hydraulic oil pressure in the system is calculated as static pressure in this study. Computing dynamic pressure during operation and oscillation frequency would enhance equipment reliability. This is a relatively complex issue when the rock-splitting head is mounted on a self-propelled machine, requiring continued research based on theory and experimentation.

5. Conclusions

This paper investigates the hydraulic rock-splitting device with the wedge-type head. This study establishes procedures and calculation methods for two problem sets: determining suitable parameters for the wedge and determining suitable parameters for the hydraulic cylinder. Numerical test results provide a set of technical design data applicable to real-world scenarios, including the wedge angle, hydraulic cylinder stroke, cylinder diameter, pressure, minimum cylinder wall thickness, and piston diameter, corresponding to various rock-splitting forces. These parameters aim to minimize hydraulic cylinder pushing force and minimize the cross-sectional area of the cylinder body. In the test example, designing a rock-splitting head with a splitting force of 3000 kN and a wedge opening of 35 mm. The results show that reasonable parameters are: hydraulic cylinder stroke of 200 mm, wedge angle of 10 degrees, cylinder diameter of 250 mm, and suitable cylinder wall thickness of 9.1 mm. These reasonable parameters have also been durability-tested using Ansys software. To reduce hydraulic cylinder pushing force, the mechanical surfaces of the wedge and wedge wing need machining to achieve the lowest friction coefficient. The hydraulic oil pressure in the system is statically calculated in this research. Considering dynamic pressure during operation and oscillation frequency could enhance equipment reliability, providing a direction for further research.

Acknowledgment

The research was developed based on project code 51/2017/KHXD. The author would like to thank the Hanoi University of Civil Engineering for allowing this research.

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