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Improves the Brake Working of the Winch to Control the Stop Brake Process

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ABSTRACT

High-altitude rescue is an issue that always receives great attention. Using rescue winches on the outside of structures is a popular solution. During the rescue process, when the brakes stop, it will affect the human body, causing discomfort, especially when the rescue subjects include the elderly and children. The purpose of the research is to find a reasonable brake structure solution and parameters to improve performance, allowing for increased speed while still ensuring smooth acceleration when braking and controlling shock. This article develops calculation formulas describing the braking process and develops a method for selecting design parameters to control the braking process including spring deformation, spring stiffness, braking time, etc. The study uses the orthogonal matrix, analyzing the influence of parameters on the response function, and spring stiffness. Applying numerical testing to the design to increase the speed by 300%, the result is that the acceleration when braking is reduced by 67.75% compared to the previous study, and the shock in the range is smaller than the allowable value.

Keywords: acceleration, allowable shock, braking process, braking time, rescue winch.

INTRODUCTION

Rescue at height is an issue that always receives great attention, especially in high-rise buildings. When an incident occurs, a type of problem can occur such as the limited capacity of stairs for emergency access during the evacuation of people, the evacuation of people cannot be carried out inside the building due to separation when stairs, elevators, escalators... are the only way out [1]. Therefore, there have been many published inventions and related research on rescue devices from the outside of buildings [2-4]. Using a rescue winch is a popular method for personal use or towing sliding baskets on rails [1-3]. The invention in [2] uses a fixed rail installed on the wall of the building, the fixed rail has a rack attached, the sliding motion follows the rail, and the gear of the device meshes with the rail and interlocks with the brake. The structural invention in [3] for both types of working principles of personal rescue winches is to use friction pulleys and cable reels, combined with automatic brakes. Using friction pulleys can give a large lifting height, however, the structure has many complex details. General characteristics of the rescue equipment in [2-4] have the advantage of being very compact, however, there are many limitations in terms of application.

The rescue winch in [4-6] has a service area of 28 m, a maximum design speed of 6 m/min, and a manual drive. The winch in [4] can be used as a personal rescue device for a normal healthy person and it is also a part of a complete rescue kit when combined with a sliding basket on a guide ladder. In [5], the gear system of the winch was also optimized, reducing the gear transmission by 46.98 % compared to the original, making the winch more compact. In addition, in [4], and [6] the vibration of the lifting basket and climbing ladder was evaluated when using the designed rescue winch during work. Surveyed in [6] is a system including a winch, basket, and ladder. The dynamic parameters of the rescue basket were investigated through Matlab software under different conditions, the maximum acceleration when lowering was 6 m/s^2 and the average acceleration was 4.8 m/s^2. In this study, the manual rescue winch had a maximum speed of only 6 m/min. If the winches designed in [4], and [6] are only driven by hand, it will not be able to meet the diverse tasks of reality, with limited operating speed. If this design has a faster-lowering speed, it will vibrate and have a large shock, causing negative effects on humans.

It can be seen that the brake structure of the winch in [4], and [6] is still limited and needs to be improved to control acceleration and shock. The internal brake [4] is based on the pressure of a screw-nut transmission, which converts rotational motion into reciprocating motion. In [7] also presented the basis and calculation of detailed mechanical designs including the screw and nut transmission, and the thrust of the screw when rotating. Instructions for designing and calculating parameters to optimize brakes are mentioned in [8], including brake surface pressure, brake torque for disc brakes, and clutch brakes. Research in [9] has designed a cone brake for manual winches at low speeds. In case of high speeds, this structure is difficult to ensure smooth braking, acceleration, and shock can exceed the limit, allowed term. In [10] applying the braking torque calculation method based on the change in total energy, this study considers the elements of the elevator balancing system, applied to the tower parking car system. Thus, calculating the braking torque can be based on the working principle of the brake or the total energy balance when braking.

Acceleration and shock when there is unstable movement are factors that need to be controlled because they affect many engineering problems [11, 12]. During the rescue process, when the brakes stop, it will affect the human body, causing discomfort, especially when rescuing the elderly and children. The vibration of the elevator cabin significantly affects the health condition as well as the stress level of the pupils during travel. The study in [13] measured and reported the results of vibration measurements taken during travel in a passenger elevator. There are guidelines on the possible effects of vibration on health, comfort, cognition, and motion sickness [14–16]. In [15] instructions are given to evaluate the level of human exposure to whole-body vibration, for example, if vertically in the frequency range from 1 Hz to 100 Hz, if the acceleration is 10 m/s^2, then the maximum limit that humans can withstand is in 1 minute. Regulations and methods for measuring the quality of elevator movement during movement are also presented in [16]. Regarding this issue, in [17] there was an experiment when emergency braking of the elevator cabin was done at five acceleration levels from 0.52 g to 0.62 g to evaluate human reactions. The results showed that at these levels there was no response adverse effects on human health. With passenger elevators, acceleration, and shock need to satisfy physiological conditions. This happens when the body is subjected to acceleration or deceleration. In [18] it is shown that acceleration values must be limited to less than 1.5 m/s^2 and shock values (rate of change of acceleration) at 2.0 m/s^3.

Developing new mechanical structures and building mathematical expressions to solve a given problem is a common trend of research in industrial engineering. The study in [19] describes the design of a pallet locking system of a multistory car parking system, presenting theoretical calculations of pallet brakes and a calculation model using SolidWorks software.

The experimental results and numerical simulation by finite element method of the new brake type CHP 2000 for elevators were studied in [20]. This study simulated numerical analysis for the roller gear system during braking from the neutral position to maximum displacement and alternate impact loads, evaluating the stresses in the clamping elements. Experimental studies and braking distance analysis verified the numerical calculation results. The study in [21] tested and compared different types of progressive safety gears to evaluate their reliability and compared with the newly proposed safety structure, the comparative evaluation parameter is braking distance length. In [22, 23] develop mathematical or experimental formulas, and use orthogonal planning matrices and Minitab software to optimize parameters, to bring higher efficiency to research subjects.

This study will overcome shortcomings in studies [4–6]. Results of evaluation and comparison with previous studies in [4–6], and [15–18] will demonstrate the effectiveness of the proposed brake structure, as well as the calculation method in reducing acceleration, and shock when braking to stop.

Based on the above analysis, this article will research to improve the efficiency of rescue winches. The article will develop calculation formulas to describe the braking process, representing the relationship between parameters. Develop a method to select design parameters, the purpose of which is to control the braking process. The influence of parameters on the response function is analyzed using Minitab software. The result of the research is to find a solution for the brake structure and reasonable parameters to improve performance, allowing for increased speed while still ensuring smooth acceleration when braking and controlling shock.

SOLUTION FOR BRAKE STRUCTURE AND DETERMINATION OF BRAKING TORQUE

Rescue winch and braking structure

The rescue equipment in [5, 6] as shown in Figure 1 includes a rescue winch 1 used to pull the basket 2 to move on the ladder (rail) 3. Basket 2 is lowered by winch 1 to necessary locations to rescue people, including children or the elderly. Therefore, during braking, it is necessary to limit excessive acceleration and shock. The rescue winch in Figure 2 is designed to inherit the research in [4, 6]. It can use both options

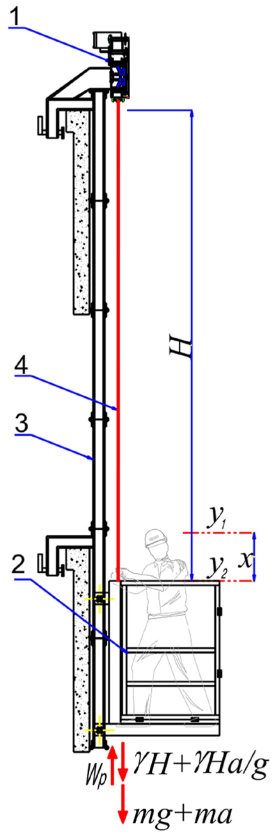

Fig. 1. Diagram of the braking process to lower the load of rescue equipment: 1. Winch, 2. Basket, 3. Ladder - guide rail, 4. Cable

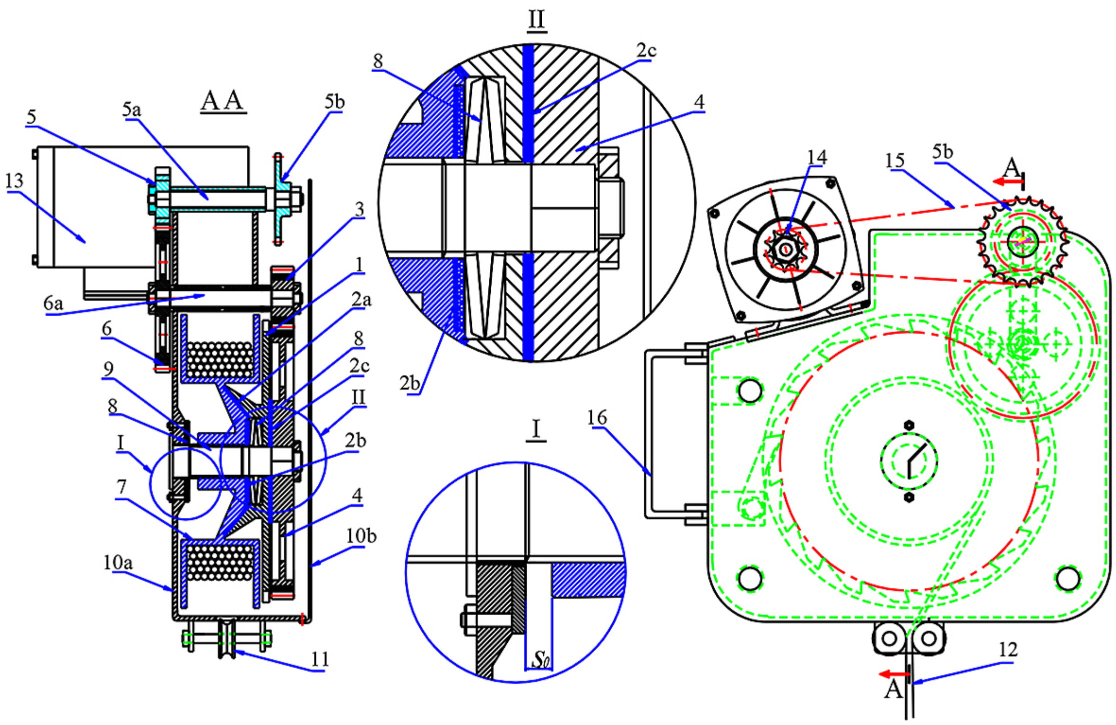

Fig. 2. Winches with braking structures to reduce acceleration and shock: 1. Ratchet, (2a, 2b, 2c). Brake friction surface, (3, 4, 5, 6). Transmission gears, 5a. Active shaft, 5b. Passive sprocket, 6a. Medial shaft, 7. Drum, 8. Spring, 9. Screw thread shaft, 10a, 10b. Winch housing, 11. Guiding roller, 12. Cables, 13. Motor, 14. Active sprocket, 15. Transmission unit, 16. Handle

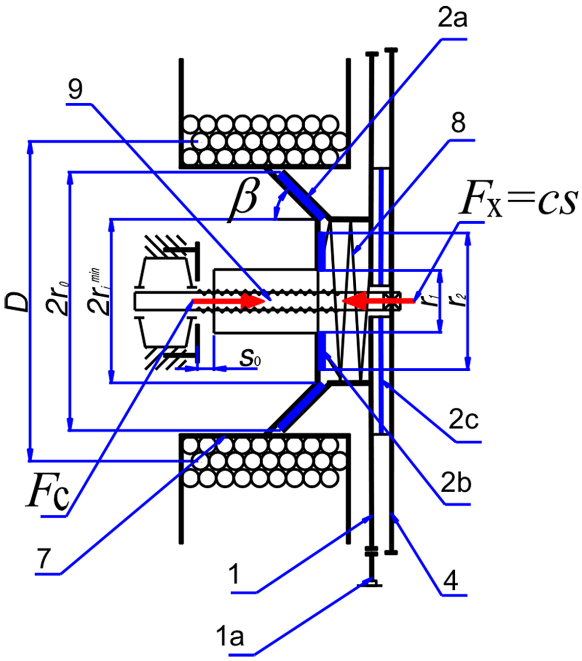

Fig. 3. Calculation model and brake structure diagram to reduce acceleration and shock: 1. Ratchet, 1a. Toad, (2a, 2b, 2c). Brake, 4. Gear, 7. Drum, 8. Spring, 9. Screw shaft

depending on specific conditions: manual drive similar to [4], [5] when installing the crank on shaft 5a or drive by motor 13 through transmission 15. Motor 13 is mounted directly into the winch housing. Spring 8 to reduce acceleration and shock when braking to lower the load. Drum 7 rolls cable 12 to lift the load because motor 13 rotates in the lifting direction. Motor 13 rotates in the downward direction, shaft 9 rotates, and the drum moves causing friction surface 2a to separate. Basket 2 lowers due to its weight, the drum rotates on screw thread shaft 9 with angular speed and presses against friction surface 2b with spring 8. The winch in Figure 2 is different from [5, 6], and [9] in that it can use a motor to increase working speed, and the brake has spring 8 and friction surface 2c to control the braking process.

The brake structure solution is described in Figure 3, the braking process has one to two stages. In the first stage with friction surface 2b working, the braking torque gradually increases due to the increased deformation of spring 8. Therefore, at this stage, the lowering acceleration will be controlled by friction surface 2b. In the second stage, if the drum rotates on the screw at the end of its stroke without stopping, friction surface 2a works. At this time, the friction surfaces 2a, 2b, and 2c are pressed tightly. At this stage, the torque will be created by both friction surfaces 2a and 2b. For the braking process when lowering the load smoothly, acceleration and shock are parameters that need to be controlled. Acceleration and shock when braking depends on the stiffness of the spring, spring deformation, and friction surface structural parameters.

Determine the braking torque according to the total energy conversion method

The change in kinetic energy of the system ΔDE consisting of reciprocating moving parts shown in Figure 1 is load 2 and steel cable 4. The structure of the winch is also described in Figure 2. The rotating moving part in Figure 3 includes drum 7 and cone brake 2a. Therefore, the kinetic energy change is determined according to formula (1):

$$\Delta DE = \frac{I_d^2}{2}\omega_m^2 + \frac{mv^2}{2} + \frac{\gamma H v^2}{2g} \tag{1}$$

where: I_d – the moment of inertia of the drum and clutch brake (kgm^2),
ω_m – the angular velocity of the drum (rad/s),
m – the mass of the basket and person (kg),
v – the descent speed (m/min),
H – the lifting height (m),
γ – the specific weight of the cable per meter long (N/m),
g – the gravitational acceleration, $g = 9.81$ m/s^2.

The change in potential energy ΔTE_r, due to the mass of the cable as it descends from the start of braking $y_1 = H - x$ to the end of braking $y_2 = H$ is described by equation (2):

$$\Delta TE_r = \gamma\int_{y_1}^{y_2} y dy = \frac{\gamma}{2}\left[H^2 - (H-x)^2\right] \tag{2}$$

where: x – the braking distance (m),
y – the position of the basket (m).

Potential energy change due to the mass of the basket and person ΔTE_p:

$$\Delta TE_p = mgx \tag{3}$$

The total change in potential energy of the system is described in Figure 1, from formulas (2) and (3) we have:

$$\Delta TE = mgx + \frac{\gamma}{2}\left[H^2 - (H-x)^2\right] \tag{4}$$

The change in energy ΔCE due to braking is the sum of changes in kinetic energy, potential energy, and other work (Nm):

$$\Delta CE = \Delta DE + \Delta TE + \Delta W_p \tag{5}$$

where: ΔW_p – the energy due to other resistance components, such as rolling resistance when sliding on rails.

From formulas (1) to (5) we have:

$$\Delta CE = \frac{I_d^2}{2}\omega_m^2 + \frac{mv^2}{2} + \frac{\gamma H v^2}{2g} + mgx + \quad H \quad H \quad x \\ + \frac{\gamma}{2}\left[H^2 - (H-x)^2\right] + \Delta W_p \tag{6}$$

The rotation angle α of the drum during braking can be expressed through the braking distance x as $\alpha = 2x/D$ (rad).

Braking torque according to the energy conversion method is determined according to formula (7).

$$W = \int_{t_1}^{t_2} (\omega_m - \alpha t) T_B dt = T_B(\omega_m - \frac{\alpha t}{2})t = \Delta CE; \\ T_B = \frac{\Delta CE}{(\omega_m - \frac{\alpha t}{2})t} \tag{7}$$

where: α – the rotation angle of the drum during braking time (rad),
t – the braking time from start to stop (s),
T_B – the braking torque (Nm).

Determine the braking torque according to the working principle

Assuming linear motion, the pulling force of the cable wound on the drum (Figure 1 and Figure 3) during the motion is unstable when braking.

$$S = (mg + \gamma H) + (m + \frac{\gamma H}{g})a \tag{8}$$

where: a – the acceleration during braking (m/s^2).

The axial force of the screw thread due to the torque caused by the torque on the drum (acting as a nut) when lowering, deploying, and applying to the problem, changes according to [7, 9] to obtain:

$$F_c = \frac{S\left(\pi d_m - \frac{fl}{\cos\varphi}\right)}{d_m\left(l + \frac{f\pi d_m}{\cos\varphi}\right)} \tag{9}$$

where: d_m – the average screw thread diameter (m),
f – the screw thread surface friction coefficient,
l – the screw thread pitch (m),
φ – the screw thread inclination angle (degrees).

Put:

$$A = \frac{\left(\pi d_m - \frac{fl}{\cos\varphi}\right)}{d_m\left(l + \frac{f\pi d_m}{\cos\varphi}\right)}$$

we can get the formula (10):

$$F_c = \left((mg + \gamma H) + (m + \frac{\gamma H}{g})a\right)A \tag{10}$$

Assume the relationship between force and deformation of the spring according to the formula:

$$F_x = cs \tag{11}$$

where: F_x – the spring compression force (N),
c – the spring stiffness (N/m),
s – the spring deformation (m).

The axial force of the screw thread to press the friction surfaces during braking from formula (10), (11):

$$F = F_c - F_x \tag{12}$$

In another way, the braking moment due to the axial force F depends on the structural parameters of the brake, applied to the problem and transformed according to [8, 9], we have the braking model with brake surfaces 2a and 2b as formula (13):

$$T_B = \begin{cases} \dfrac{0.7886\mu r_0}{\sin\beta\left(1 + \dfrac{\mu}{\tan\beta}\right)}F, & \text{for braking surface 2b} : 0 < \beta < 90^0 \\ 0.7886\mu r_2 F, & \text{for braking surface 2c} : \beta = 0^0 \end{cases} \tag{13}$$

where: r_0 – the radius of the large cone of the friction surface 2a (m),
r_2 – the radius of the friction surface 2b (m),
β – the cone angle (degrees),
μ – the coefficient of friction between the two cones in contact.

Put:

$$B = \frac{0.7886\mu r_0}{\sin\beta\left(1 + \dfrac{\mu}{\tan\beta}\right)}$$

Substituting equations (10), (12) into (13) we have:

$$T_B = \begin{cases} B\left\{A\left((mg+\gamma H)+(m+\dfrac{\gamma H}{g})a\right)-cs\right\}, & 0<\beta<90^0 \\ 0.7886\mu r_2\left\{A\left((mg+\gamma H)+(m+\dfrac{\gamma H}{g})a\right)-cs\right\}, & \beta=0^0 \end{cases} \tag{14}$$

Braking torque at the motion braking stage due to working surfaces 2b:

$$T_B = 0.7886\mu r_2\left\{A\left((mg+\gamma H)+(m+\frac{\gamma H}{g})a\right)-cs\right\} \tag{15}$$

In case of stopping motion when the braking torque due to friction surfaces 2a and 2b work together:

$$\begin{aligned} T_B = {} & B\left\{A\left((mg+\gamma H)+(m+\frac{\gamma H}{g})a\right)-cs\right\} \\ & +0.7886\mu r_2\left\{A\left((mg+\gamma H)+(m+\frac{\gamma H}{g})a\right)-cs\right\} \end{aligned} \tag{16}$$

CALCULATION METHOD TO SELECT PARAMETERS TO IMPROVE BRAKE OPERATION

Braking process

In the first stage when friction surface 2b works, the braking torque gradually increases due to increased spring deformation. It is assumed that during this period the movement slows down uniformly. So from equations (7), (15) and calculation assumptions, we have a system of equations (17):

$$\begin{cases} v = v_o - at_1, x = v_o t_1 - \dfrac{at_1^2}{2} \\ \alpha = \dfrac{2x}{D}, \omega_m = \dfrac{2v}{D} \\ \dfrac{\Delta CE_1}{(\omega_m - \dfrac{\alpha t_1}{2})t_1} = 0.7886\mu r_2\left\{A\left((mg+\gamma H)+(m+\dfrac{\gamma H}{g})a\right)-cs\right\} \\ c = F(X_0, X) \end{cases} \tag{17}$$

where: v_0 – the speed at the beginning of the lowering process (m/s),
t_1 – the braking time when friction surface 2b is working (s),
ΔCE_1 – the energy transformation in the first stage (Nm).

Changing the system of equations (17), we get the stiffness value of the spring $c = F(X_0, X)$. Where (X_0, X) are given values and design parameters.

In the next stage, if the rotating drum on the screw thread shaft moves without stopping, friction surface 2a works. At this stage, the braking torque will be generated by both friction surfaces 2a and 2b. Assume the hardness of friction surfaces 2a is very hard. So from equations (7), (16), and calculation assumptions we have a system of equations (18):

$$\begin{cases} \alpha_0 = 0, \omega_0 = \dfrac{2v_{01}}{D} \\ \dfrac{\Delta CE_2}{(\omega_0 - \dfrac{\alpha_0 t_2}{2})t_2} = B\left\{A\left((mg+\gamma H_0)+(m+\dfrac{\gamma H}{g})a\right)-cs\right\}+0.7886\mu r_2\left\{A\left((mg+\gamma H)+(m+\dfrac{\gamma H}{g})a\right)-cs\right\} \\ v_{01} = v_o - a_0 t_1 \\ a = \mathrm{F}(\mathrm{X}_0, t_2) \end{cases} \quad (18)$$

where: v_{01} – the final velocity of the previous stage (m/s),
ω_0 – the final drum angular velocity of the previous stage (rad/s),
a_0 – the acceleration in the previous stage (m/s^2),
t_2 – the braking time when friction surface 2a works (s),
ΔCE_2 – the energy transformation in the second stage (Nm).

Transforming the system of equations (18), we get the acceleration function $a = F(X_0, t_2)$, where (X_0, t_2) are the given values and braking time.

The discomfort and instability for the person on the basket is that the acceleration causes the shock to be too great. The shock level j is the derivative of the acceleration (m/s^3).

$$j = \frac{da}{dt} \quad (19)$$

According to the principle of acceleration structure, the shock when braking depends on the stiffness of the spring, spring deformation, and friction surface structure parameters 2b. The descent acceleration will be controlled by friction surface 2b and must satisfy the constraint related to human ability.

$$\begin{cases} a \le [a] \\ j \le [j] \end{cases} \quad (20)$$

where: $[a]$ is the maximum allowable acceleration (m/s^2),
$[j]$ is the allowable shock (m/s^3).

For passenger elevators or similar equipment according to [15, 18] during braking or unstable movement, the acceleration value $[a]$ = 1.5 m/s^2 and shock $[j]$ = 2 m/s^3.

Model of the problem and numerical testing method

Parameters that need to be determined include spring stiffness, spring deformation, braking time, and friction surface parameter 2b, these are very important factors to control the braking process in the first stage. The desired spring stiffness is as small as possible for convenience in use, exploitation, and manufacturing.

The given parameters, including data related to the technical characteristics of the rescue system, winch parameters, coefficients... according to [4-6, 9, 18] are represented by the set.

$$X_0 = \{m, d, l, i, H, [a] ...\} \quad (21)$$

In the test example in Table 1, design rated speed v_0 = 0.3 m/s, acceleration $a = [a]$ = 1.5 m/s^2. From the conditions according to formula (21) and the structure of the brake, the design parameters and constraint conditions are expressed by expressions (22), (23):

$$X = \{\mathrm{r}_2, t, s\} \quad (22)$$

$$\begin{cases} \mathrm{r}_2^{\min} \le \mathrm{r}_2 \le \mathrm{r}_2^{\max} \\ t_{\min} \le t_1 \le t_{\max} \\ s_{\min} \le s \le s_{\max} \\ a = [a] \\ j \le [j] \end{cases} \quad (23)$$

In which the constraint values related to the brake structure are the minimum and maximum large radius of friction surface 2b ($r_2^{\min}$, $r_2^{\max}$), and minimum and maximum spring deformation ($s_{\min}$, $s_{\max}$) it depending on the brake structure. Maximum braking time when working surface 2b is $t_1 = t_{\max}$ determined according to conditions $t_{\max} = v_0/[a]$. In the test example, there are value levels for Table 2.

The response function is determined from the system of equations (17):

$$c = F(X_0, X) \to \min \quad (24)$$

Table 1. Design data for rescue equipment's winch [4-6, 9, 18]

TT	Parameter	Symbol	Value	Unit
1	Load volume	m	168	kg
2	The average diameter of the drum	D	198	m/min
3	Average screw thread diameter	d_m	0.028	m
4	Screw thread tilt angle	a	14.5	degree
5	Screw thread pitch	l	0.003	m
6	Friction coefficient of screw thread surface	f	0.06	
7	Large end radius cone brake	r_0	0.066	m
8	Cone angle of brake	β	50	degree
9	Braking surface friction coefficient	μ	0.3	
10	Lift height	H	18	m
11	Lifting speed	v_0	0.3	m/s
12	Permissible acceleration	$[a]$	1.5	m/s^2
13	Permissible shock level	$[j]$	2	m/s^3
14	Moment of inertia of drums and brakes	I_d	0.0145	kgm^2
15	Secondary resistance	W_p	15	N

The method of selecting braking parameters to control the braking process of rescue equipment is shown in Figure 4. To evaluate the influence of the parameters, this article uses the Taguchi method for the minimization problem. The S/N ratio (Signal/Noise) represents an efficiency indicator. The optimal set of parameters gives the largest S/N. The response function is the spring stiffness c_i, and the design parameters are $X = \{r_2,\ t_1,\ s\}$.

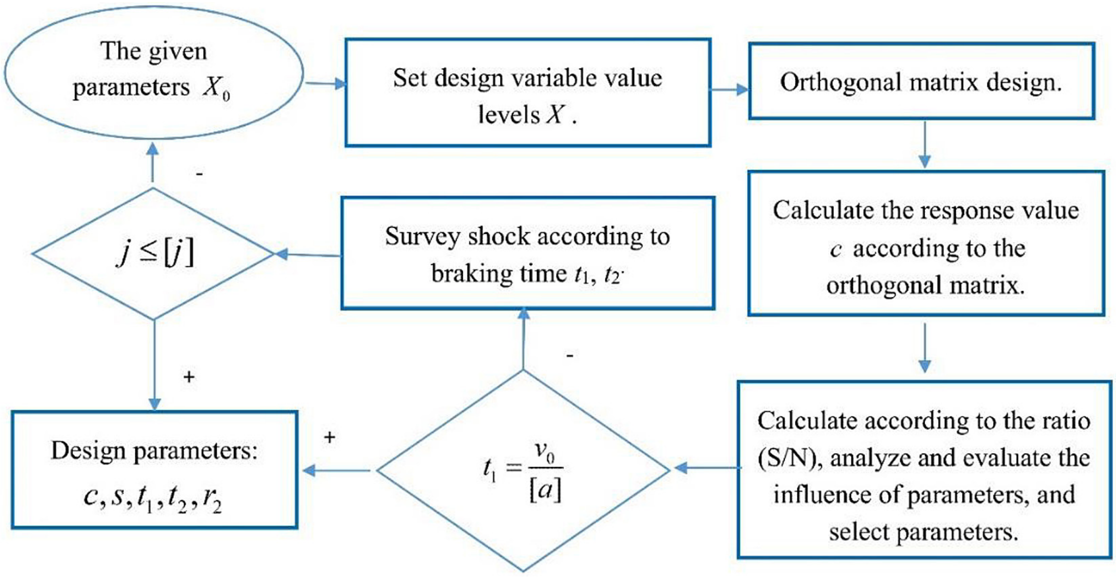

Fig. 4. Method of selecting design parameters to control the braking process

Table 2. Design parameters and value levels

Design parameters	Code	Value level			
		1	2	3	4
Friction surface radius, r_2 (m)	x_1	0.02	0.03	0.04	0.05
Braking time, t_1 (s)	x_2	0.05	0.1	0.15	0.2
Spring deformation, s (m)	x_3	0.005	0.01	0.015	0.02

The experimental sequence number is k, the number of experiments is n.

$$S/N = -10\lg\left(\frac{1}{n}\sum_{k=1}^{n} c_{ik}^{2}\right) \quad (25)$$

RESULTS AND DISCUSSION

Numerical testing for brakes with structural solutions as shown in Figures 1 and Figures 3. The given parameters are given in Table 1. Table 2 shows the design parameters and four value levels. In the early stages of the braking process, the response jaw is spring stiffness *c*. Experimental design using L16 orthogonal matrix and Minitab software results for Table 3. The influence of parameters on the response function through the S/N ratio is shown in Figure 5.

The allowable acceleration when braking is designed to be $[a] = 1.5$ m/s^2. Spring deformation has the most impact on spring stiffness, friction surface radius 2b, and braking time have a negligible effect on spring stiffness. Figure 6 shows the

Table 3. Orthogonal matrix and response value

N	x_1 (m)	x_2 (s)	x_3 (m)	c (N/m)
1	0.02	0.05	0.005	338626
2	0.02	0.1	0.01	169405
3	0.02	0.15	0.015	112954
4	0.02	0.2	0.02	84719
5	0.03	0.05	0.01	169484
6	0.03	0.1	0.005	339152
7	0.03	0.15	0.02	84800
8	0.03	0.2	0.015	113069
9	0.04	0.05	0.015	113104
10	0.04	0.1	0.02	84873
11	0.04	0.15	0.005	339535
12	0.04	0.2	0.01	169770
13	0.05	0.05	0.02	84914
14	0.05	0.1	0.015	113277
15	0.05	0.15	0.01	169936
16	0.05	0.2	0.005	559874

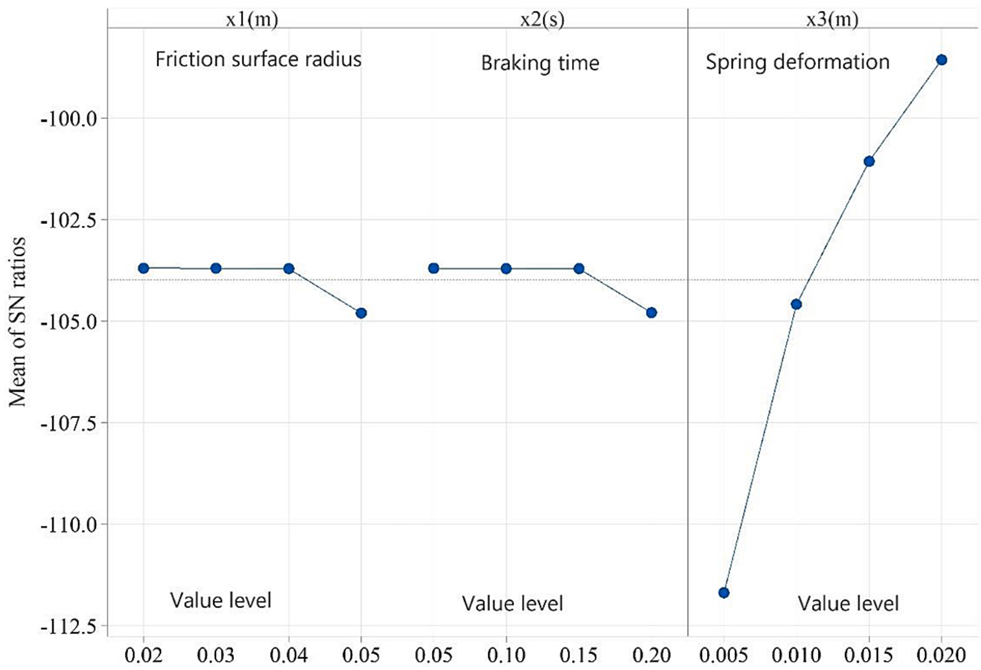

Fig. 5. Analysis of the influence of parameters through signal/noise ratios

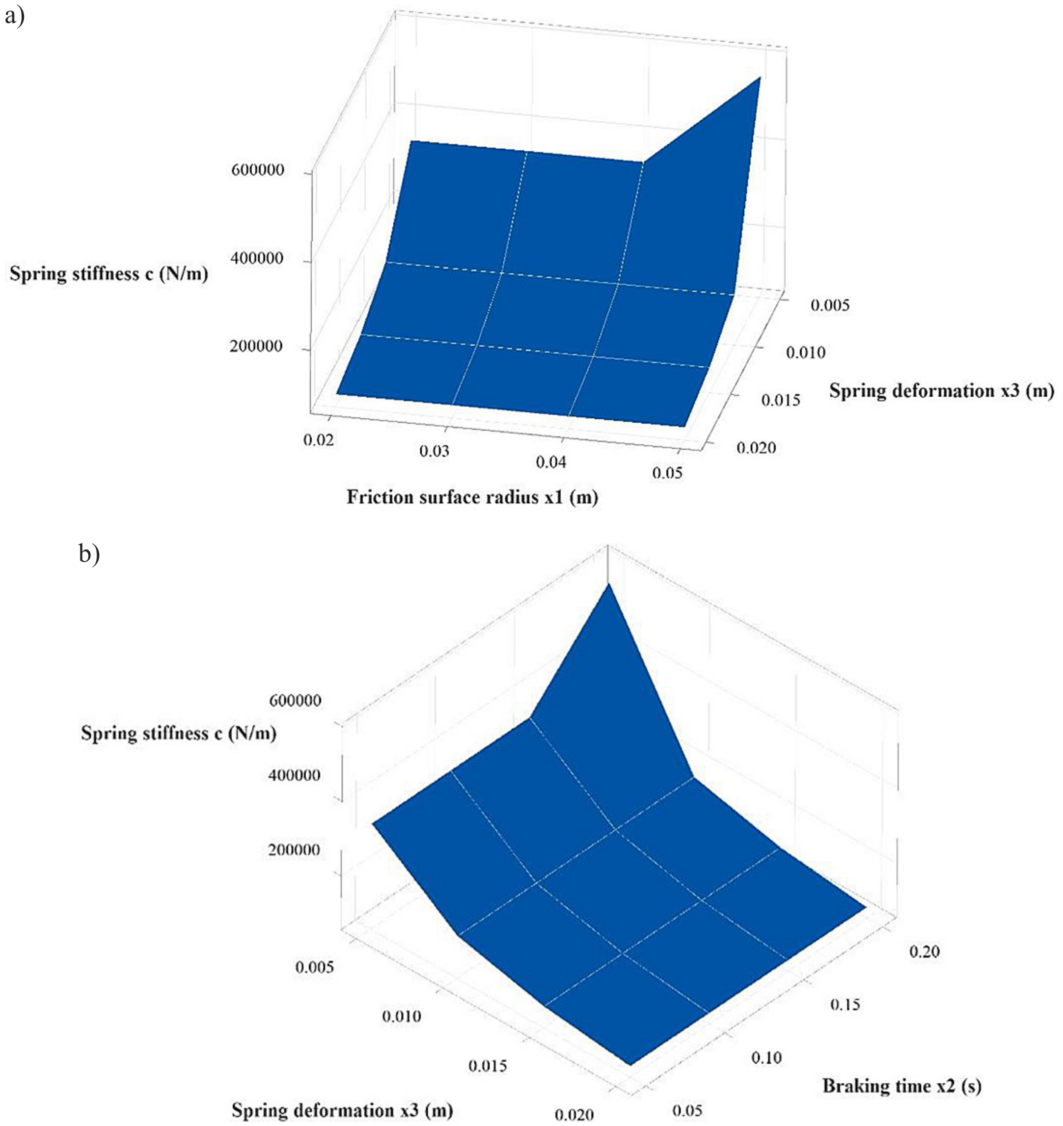

Fig. 6. Graph of spring stiffness and design parameters: a) with friction surface radius 2b and spring deformation; b) with braking time and spring deformation

relationship between these parameters. The spring stiffness is greatest when the spring deformation is smallest and the size of friction surface 2b is largest. When the braking time is $t_1 = x_2 < 0.15$ s, it does not affect the spring stiffness much.

Using Minitab software gives us the relationship between stiffness and deformation of the spring as shown in Figure 7 and the regression equation as (26).

$$c = 926872 - 1.47*10^8 x_3 + \\ + 8.96*10^9 x_3{}^2 - 1.86*10^{11} x_3{}^3 \tag{26}$$

During use, we can adjust the gap $s_0 < s = 0.02$ m (Figure 2 and Figure 3) to adjust the lowering speed. This adjustment does not greatly affect the acceleration and shock when braking to a stop because the deformation and spring compression also change.

The ideal maximum braking time is $t_1 = v_0/[a] = 0.2$ s so that at the end of the first stage the speed is $v_{01} = 0$ m/s. Analyzing according to the S/N ratio of the remaining parameters according to the response function, we choose friction surface radius $r_2 = 0.04$ m, spring deformation $s = 0.02$ m, spring stiffness $c = 84885.5$ Nm, and braking distance $r_2 = 0.03$ m. When the acceleration is constant, in this stage the shock $j = 0$ m/s^3. These parameters meet the requirements for acceleration and shock.

In case the braking time is $t_1 < 0.2$ s, according to the analysis of the S/N ratio, we can choose a reasonable set of parameters $r_2 = 0.04$ m, $s = 0.02$ m, $c = 84884$ Nm. The shock when

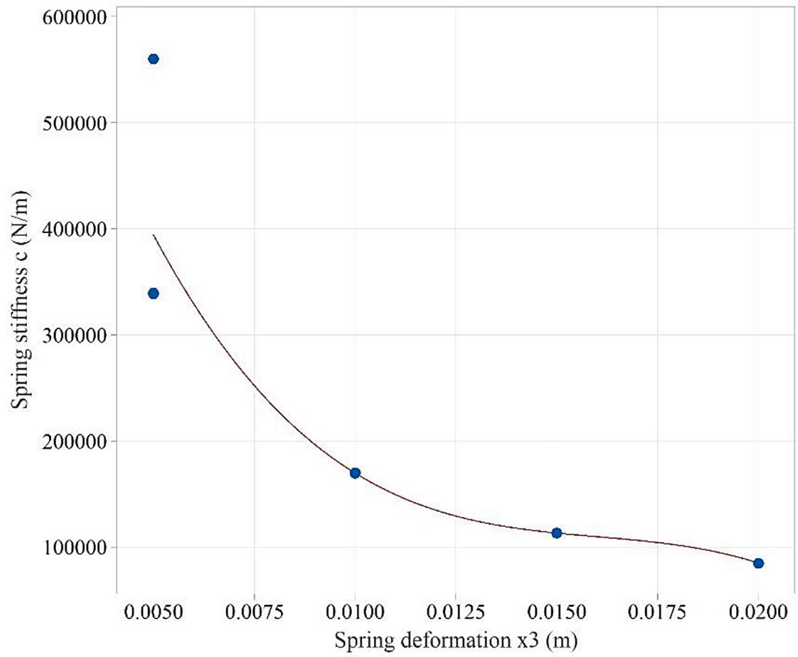

Fig. 7. Select spring stiffness and deformation to control braking

friction surface 2a works from the system of equations (19) is $j = 0.00589t_2^{-2}$. With the selected parameters, the results of the braking time and shock survey are shown in Figure 8. The results show that when braking $t_2 \geq 0.055$ s, the shock ranges from 0 to 1.947 m/s^3, smaller than the allowed value $[j] = 2$ m/s^3. The acceleration time exposed to whole-body vibration is less than 1 minute at the design acceleration level of 1.5 m/s^2, satisfying the evaluation guidelines of ISO 2631 [15]. These results are also smaller than the acceleration value in the tests of the study [17] when emergency braking of the elevator cabin, ensures no negative impact on human health.

Thus, the design parameters we can choose to design to control the braking process of rescue equipment are $r_2 = 0.04$ m, $s = 0.02$ m, $c = 84884$ Nm, $t_1 \leq 0.2$ s, $t_2 \geq 0.055$ s. Table 4 is a comparison with the research and survey results in [4-6]. These new brake design and structure

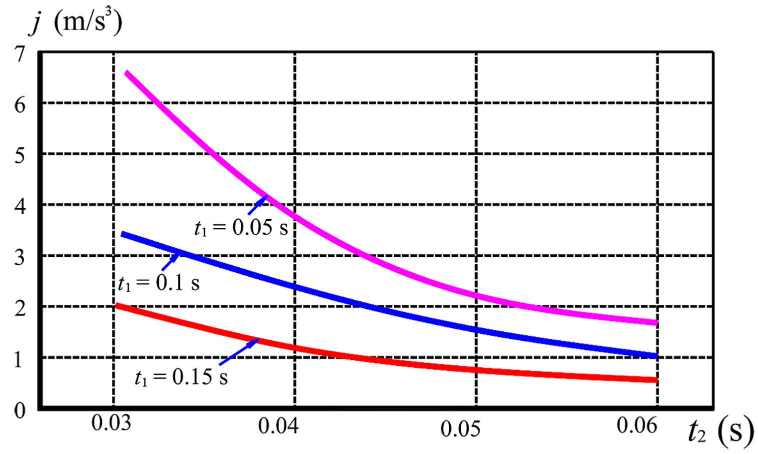

Fig. 8. Relationship between shock and braking time

Table 4. Compare and evaluate

Parameter	Design parameters	Parameters in [4-6]	Parameter improvement
Speed (m/s)	0.3	0.1	+ 300%
Acceleration (m/s^2)	1.5	4.8 (6)	- 67.75%

parameters bring higher efficiency, the speed increases by 300%, while the design acceleration is 1.5 m/s^2 and the shock is less than 2 m/s^3. In this article, the stiffness of the friction surfaces is assumed to be large, in addition, the temperature generation at the friction surface when braking is an issue that needs to be further discussed in other studies.

CONCLUSIONS

The article has researched to improve the working ability of winches in rescue equipment, including finding solutions to brake structures to control the braking process and methods to select those parameters. Formulas describing the braking process and problem models for selecting parameters to control the braking process have also been developed. Reasonable parameters to choose from in the example include braking time, spring deformation, spring stiffness, and braking radius. Research helps reduce shock when braking thanks to the reasonable design of the brake structure, which allows for increased lowering speed. The reasonable parameters satisfy the conditions related to human health in studies related to acceleration and shock when stopping. In addition to controlling spring stiffness and geometric parameters as mentioned in this study, the brake structure is also elastic, which can affect acceleration and shock when lowering. Long braking times and long braking distances to ensure acceleration and shock within allowable limits will increase friction surface temperatures. The increase in temperature will affect wear and change the surface friction coefficient of the brake. Reasonable selection of braking parameters is essential to ensure safety during the rescue. The influence of structural elasticity and temperature factors will be studied in the next stage.

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