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Optimizing the weight of the two-level gear train in the personal rescue winch

In the previous study, we designed one personal rescue winch for high-rise building rescue. Its key requirement is to be small and light enough to suit users. In addition to using lightweight and reasonable materials as in the proposed winch design, in this study, we proceed to optimize the weight of one two-level gear train, which accounts for a large proportion of weight. The first stage is building a weight optimization problem model with seven independent variables, establishing one optimal algorithm, and investigating the variables by Matlab software. The other is replacing the web material of the gears and pinions with Aluminum 6061-T6 and optimizing their hole diameters and hole numbers through using Ansys software. The obtained result shows a significant weight reduction. Compared to the original design, the weight reduces by 10.21% and 52.40% after the first optimal and last stages, respectively.

1. Introduction

According to the development trend of society, high-rise buildings continue to be invested and built in big cities. Therefore, the risks of insecurity caused by fire and explosion are increasing, leading to serious damage if no timely response is taken. Research and development of personal rescue equipment for people living there, with which they can equip themselves and escape, are particularly effective for zones inaccessible by professional equipment and firefighters. Based on analysis and evaluation of the rescue solutions in [1, 2] and self-climbing system in [3], the research team proposed one personal rescue winch, as shown in Fig. 1 [4].

The rescue winch's construction includes the handwheel, the gear train, the screw shaft, the rope drum, the lifting rope, the winch housing, the safety brake,

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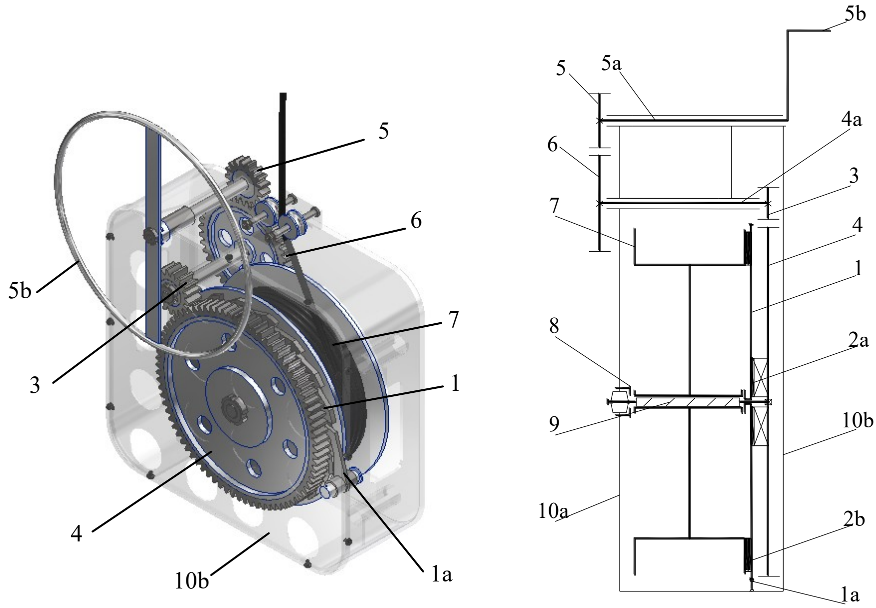


(a) 3D simulation of the winch (b) structure diagram of the winch

Fig. 1. Construction of the personal rescue winch: 1 – ratchet; 1a – pawl; 2a, 2b – friction surfaces; 3, 5 – pinions; 4, 6 – gears; 4a – intermediate shaft; 5a – input shaft; 5b – handwheel; 7 – rope drum; 8 – adjustment ring; 9 – screw shaft; 10a, 10b – housing

the ratchet, and the pawl. When turning the handwheel, the energy is transmitted through the gear train and the drum leading to lifting or lowering the load. The safety brake is the friction brake type. It is a combination of the screw shaft, the rope drum, the friction surfaces, the ratchet, and the pawl. The brake torque value depends on the weight of the suspended load. The gear train has a ratio to ensure that the lifting process is carried out by only one person who participates in the rescue or one being rescued.

Since the winch is designed either as a personal rescue device or installed in one rescue basket, its structure should be compact and it is needed to reduce the minimum weight for a normal person to carry. In this structure, the winch housing and the gear train occupy the most weight. For the first part, we use aluminum alloy to manufacture and make holes in the surface to reduce weight. As for the latter one, it is necessary to optimize the parameters and the structure according to the target function, which is the minimum weight, and ensure the binding conditions according to the design regulations.

Optimal design is now a research area that many scientists are interested in. Optimization techniques are divided into two method groups, including traditional and advanced optimization methods [5]. The first one has been used for a long time, such as nonlinear programming, geometric programming, dynamic programming, and generalized reduced gradient method. Although they can solve many optimization problems in technology, there are many limitations. For example, they are incapable to solve complex problems with discrete variables and large numbers of constraints. The latter is derived from natural phenomena such as Genetic Algorithms, Differential Evolution, Simulated Annealing, and Particle Swarm Optimization.

Currently, many optimization algorithms are using numerical methods to solve problems. The approach is suitable for solving complex optimization problems such as discrete problems, integers, and global optimization. Genetic Algorithm and Simulated Annealing are good methods to solve the discrete and nonlinear variable problem [6]. However, they have disadvantages pointed out by [7], which are high requirements on both hardware and computing software, CPU runtime. Hence, when calculating, it is sometimes necessary to accept some limitations, such as that the solution found is not the best solution, simplify some problems, or accept other forced conditions.

The authors in [8] optimally designed the gear train weight with the five design variables, including the module, the tooth number, the gear shaft diameters, and the face width, through Advanced Genetic Algorithm. The authors in [9] set up the optimal problem with design variables as that in [8], used Genetic Algorithm, programed in Matlab, and compared the obtained result in both studies. The conclusion shows that the optimal gear weight in [9] is smaller. Research in [10] optimizes the design of the one-stage speed reducer with the design variables and the target function like that in [8]. Still, it is implemented in a two-level optimization. Level 1 finds optimal parameters by Genetic Algorithms. Level 2 uses CAD-CAE tools to survey and analyze the structure while the study considers other parameters that have not been mentioned before, such as the gear web and rim thicknesses.

Based on the statistics of natural mechanical phenomena in steelmaking, the Simulated Annealing method is presented in [11] as an optimization method used in engineering. [12] and [13] applied it to the problem of gear transmission optimization. In addition, [13] also uses the Particle Swarm Optimization method to solve that problem. The results show that both the Simulated Annealing method and the previous one provide a better solution than the Genetic Algorithm method. The authors in [14] used the Simulated Annealing method, Firefly Algorithm, and MATLAB solver Fmincon to optimize the design of the two-stage spur gear train. Its volume decreased by 7.2%, while the study considers additional constraints.

The above-mentioned studies show that the optimization methods inspired by nature give good efficiency in the calculation time as well as the result of the required target for optimization problems with many variables, many constraints, and a large search zone. However, the algorithms of these methods are often complex and depend on some characteristic factors of the respective ones. That leads to a high risk of errors in the implementation process and can lead to many different optimal results. Therefore, they may not be the final optimal results.

In studies [15–17], the topology optimization methods show a good effect on reducing the weight of gears. Nevertheless, these designed gears often require high technology to manufacture, resulting in high costs. Meanwhile, the cost of a personal rescue winch is an important factor for every household in any country wanting to equip themselves. In addition, the gear structure should also match the working principle of the winch. Hence, our research object is one two-level gear train with a traditional structure.

By analyzing specific technical factors and actual fabrication conditions, the number of variables and their range of values are reduced. The remaining calculations to optimize the gear transmission weight can be done effectively with the brute force method. The brute force algorithm is simple. It calculates and checks all possible cases to achieve the target. With problems having various variable numbers and large survey value areas, the calculation time is long. Nevertheless, with the rapid development of science and technology, computers with high configurations are commonly used in engineering to solve big data problems. Additionally, if the fast computation time is not the prerequisite goal, this method is a perfect choice. In this paper, we apply it to optimize the two-stage gear train of the personal rescue winch.

Our optimization problem is set up with eleven design variables. They are divided into two groups. The first one is seven variables that our proposed algorithm can optimize, and Matlab software is a tool to solve this problem. They consist of the modules, the tooth numbers, and the face widths. The other is four variables manually optimized with the help of structural analysis software such as Ansys. These include the hole diameters and hole numbers on the gears. Besides, the material replacement for the gear and pinion webs is also made to optimize the weight.

2. Describe the relationship of the parameters of the gear train

Gear pair 3-4 and 5-6 in Fig. 1 are spur gear train (helix angle $\beta = 0$, pressure angle $\alpha = 20°$). The gear structure in the transmission is selected as shown in Fig. 2, its parameters and design data are set up for Table 1. The gear ratio of the system must be ensured so that the handwheel drive 5b is suitable for the driving force of one average person.

$$i = \frac{M_{x4}}{M_q\, \eta_{34}\, \eta_{56}} = i_{34}\, i_{56}, \tag{1}$$

where M_{x4} is the torque in the shaft for which gear 4 is fitted, M_q is the torque rotated by the operator and η_{34}, η_{56} are efficiency of gear driver 3-4 and 5-6.

For spur gear pairs 3-4 and 5-6, the relationship between their parameters is shown in Table 2 [5].

Table 1.
Parameters of the gear train

No.	Parameters	Symbol	Unit
1	Face width of pinions 3 and 5	b_3, b_5	mm
2	Pitch diameter of pinions 3 and 5	D_3, D_5	mm
3	Pitch diameter of gears 4 and 6	D_4, D_6	mm
4	Root diameter of pinions 3 and 5	D_{r3}, D_{r5}	mm
5	Root diameter of gears 4 and 6	D_{r4}, D_{r6}	mm
6	Tip diameter of pinions 3 and 5	D_{t3}, D_{t5}	mm
7	Tip diameter of gears 4 and 6	D_{t4}, D_{t6}	mm
8	Inside diameter of gear rims 4 and 6	D_{i4}, D_{i6}	mm
9	Outside diameter of the boss of gears 4 and 6	d_{o4}, d_{o6}	mm
10	Shaft diameter at the position of pinions 3 and 5	d_3, d_5	mm
11	Shaft diameter at the position of gears 4 and 6	d_4, d_6	mm
12	Rim thickness of pinions 3 and 5	l_{w3}, l_{w5}	mm
13	Thickness of pinion webs 3 and 5	b_{w3}, b_{w5}	mm
14	Diameter of the hole in gears 4 and 6	d_{p4}, d_{p6}	mm
15	Number of the hole in gears 4 and 6	n_4, n_6	
16	Center distance of gear pair 3-4 and of gear pair 5-6	A_{34}, A_{56}	mm
17	Module of a gear pinions 3 and 5	m_3, m_5	mm
18	Teeth number of pinions 3 and 5	Z_3, Z_5	
19	Teeth number of gears 4 and 6	Z_4, Z_6	
20	Weight of gear pair 3-4	W_1	kg
21	Weight of gear pair 5-6	W_2	kg

Table 2.
Relationship of design gear transmission parameters

Gear pair 3-4	Gear pair 5-6
$D_3 = m_3 Z_3$	$D_5 = m_5 Z_5$
$D_4 = m_3 Z_4$	$D_6 = m_5 Z_6$
$A_{34} = 0.5 m_3 (Z_3 + Z_4)$	$A_{56} = 0.5 m_5 (Z_5 + Z_6)$
$i_{34} = \dfrac{D_4}{D_3} = \dfrac{Z_4}{Z_3}$	$i_{56} = \dfrac{D_6}{D_5} = \dfrac{Z_6}{Z_5}$
$D_{r4} = m_3 (i_{34} Z_3 - 2.5)$	$D_{r6} = m_5 (i_{56} Z_5 - 2.5)$
$d_{o4} = d_4 + 25$	$d_{o6} = d_6 + 25$
$d_{p4} = 0.25 (D_{i4} - d_{o4})$	$d_{p6} = 0.25 (D_{i6} - d_{o6})$
$D_{i4} = D_{r4} - 2 l_{w4}$	$D_{i6} = D_{r6} - 2 l_{w6}$
$l_{w4} = 2.5 m_3$	$l_{w6} = 2.5 m_5$
$b_{w4} = 3.5 m_3$	$b_{w6} = 3.5 m_5$

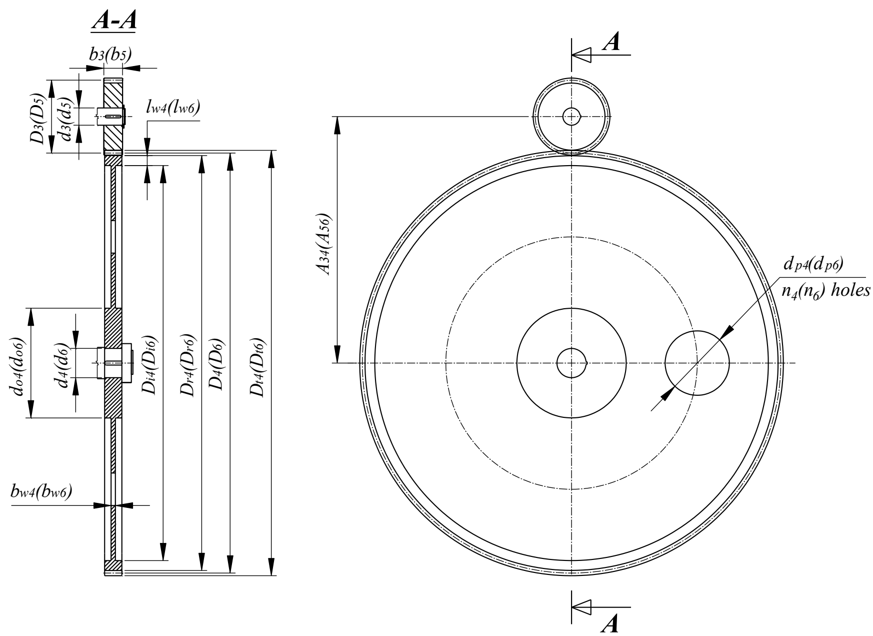


Fig. 2. The dimensions of gear pairs 3-4 and 5-6

The torques in the shafts for which gear 4, pinion 5 and gear 6 mounted with them are determined as follows:

$$M_{x4} = \frac{S_{\max} D_{tb}}{2}, \tag{2}$$

$$M_{x5} = M_q, \tag{3}$$

$$M_{x6} = M_{x3} = \frac{M_{x4}\, Z_3}{Z_4\, \eta_{34}}, \tag{4}$$

where $S_{\max}$ is the largest tension force when the rope is tangled into the drum, D_{tb} is the average diameter of the drum and the other parameters are identified as shown in Table 1.

3. Model of the weight optimization problem

3.1. The given parameters of the problem

The given parameters used to implement the weight optimization problem, their definitions, and values are listed in Table 3.

Table 3.

The given parameters

No.	Parameter	Symbol	Value	Unit
1	Torque at the handwheel 5b	M_q	16082	Nmm
2	Number of holes in gears 4 and 6	n_4, n_6	6, 4	
3	Allowable contact stress of the material of gear pair 3-4	$[\sigma_c]_{34}$	709	N/mm^2
4	Allowable contact stress of the material of gear pair 5-6	$[\sigma_c]_{56}$	709	N/mm^2
5	Density of material	ρ	7.85	mg/mm^3
6	Allowable bending stress of the material of gear pair 3-4	$[\sigma]_{34}$	251	N/mm^2
7	Allowable bending stress of the material of gear pair 5-6	$[\sigma]_{56}$	251	N/mm^2
8	Yield strength in tension of shaft material	$[\tau]$	40	N/mm^2
9	Maximum tension in the rope	$S_{\max}$	1680	N
10	Average diameter of drum 7	D_{tb}	170	mm
11	Efficiency of gear pairs 3-4 and 5-6	η_{34}, η_{56}	0.95, 0.95	
12	Overload factor of gear pairs 3-4 and 5-6	K_{o34}, K_{o56}	1, 1	
13	Dynamic factor of gear pairs 3-4 and 5-6	K_{v34}, K_{v56}	1, 1.02	
14	Size factor of gear pairs 3-4 and 5-6	K_{s34}, K_{s56}	1, 1	
15	Load distribution factor of gear pairs 3-4 and 5-6	K_{H34}, K_{H56}	1.279, 1.279	
16	Rim thickness factor of gear pairs 3-4 and 5-6	K_{B34}, K_{B56}	1, 1	
17	Bending strength geometry factor of gear pairs 3-4 and 5-6	Y_{J34}, Y_{J56}	0.34, 0.33	
18	Elastic coefficient of gear pairs 3-4 and 5-6	Z_{E34}, Z_{E56}	191, 191	(N/mm^2)$^{1/2}$
19	Surface condition factor of gear pairs 3-4 and 5-6	Z_{R34}, Z_{R56}	1, 1	
20	Geometry factor for pitting resistance of gear pairs 3-4 and 5-6	Z_{I34}, Z_{I56}	0.128, 0.115	

3.2. Design variables and objective function

The independent variables include the gear modulus of gear pairs 3-4 and 5-6; the width of pinions 3 and 5; and the tooth numbers of pinion 3, gear 4, and pinion 5. Their values are written in a vector format as follows:

$$\mathbf{m}_3 = \begin{bmatrix} (m_3)_1 & (m_3)_2 & \ldots & (m_3)_{i_{m3}} \end{bmatrix}^T, \quad i_{m3} = 1 - n_{m3}, \tag{5}$$

$$\mathbf{m}_5 = \begin{bmatrix} (m_5)_1 & (m_5)_2 & \ldots & (m_5)_{i_{m3}} \end{bmatrix}^T, \quad i_{m5} = 1 - n_{m5}, \tag{6}$$

$$\mathbf{b}_3 = \begin{bmatrix} (b_3)_1 & (b_3)_2 & \ldots & (b_3)_{i_{b3}} \end{bmatrix}^T, \quad i_{b3} = 1 - n_{b3}, \tag{7}$$

$$\mathbf{b}_5 = \begin{bmatrix} (b_5)_1 & (b_5)_2 & \ldots & (b_5)_{i_{b5}} \end{bmatrix}^T, \quad i_{b5} = 1 - n_{b5}, \tag{8}$$

$$\mathbf{Z}_3 = \begin{bmatrix} (Z_3)_1 & (Z_3)_2 & \ldots & (Z_3)_{i_{Z3}} \end{bmatrix}^T, \quad i_{Z3} = 1 - n_{Z3}, \tag{9}$$

$$\mathbf{Z}_4 = \begin{bmatrix} (Z_4)_1 & (Z_4)_2 & \ldots & (Z_4)_{i_{Z4}} \end{bmatrix}^T, \quad i_{Z4} = 1 - n_{Z4}, \tag{10}$$

$$\mathbf{Z}_5 = \begin{bmatrix} (Z_5)_1 & (Z_5)_2 & \ldots & (Z_5)_{i_{Z5}} \end{bmatrix}^T, \quad i_{Z5} = 1 - n_{Z5}, \tag{11}$$

where n_{m3}, n_{m5}, n_{b3}, n_{b5}, n_{Z3}, n_{Z4}, n_{Z5} are the number of corresponding values of m_3, m_5, b_3, b_5, Z_3, Z_4, Z_5.

The structure of two gear pairs selected is shown in Fig. 2. From Table 1, Table 2, and equation (1), their weight can be determined by

$$\begin{aligned} W_1 = \frac{\pi\rho}{4\cdot 10^6}\Big[& b_3 m_3^2 Z_3^2 \left(1 + i_{34}^2\right) - \left(D_{i4}^2 - d_{o4}^2\right)(b_3 - b_{w4}) \\ & - n_4 d_{p4}^2 b_{w4} - \left(d_3^2 + d_4^2\right) b_3 \Big], \end{aligned} \tag{12}$$

$$\begin{aligned} W_2 = \frac{\pi\rho}{4\cdot 10^6}\Big[& b_5 m_5^2 Z_5^2 \left(1 + i^2/i_{34}^2\right) - \left(D_{i6}^2 - d_{o6}^2\right)(b_5 - b_{w6}) \\ & - n_6 d_{p6}^2 b_{w6} - \left(d_5^2 + d_6^2\right) b_5 \Big], \end{aligned} \tag{13}$$

where ρ is the density of the material used to manufacture the gear (pinion); W_1 is the weight of gear pair 3-4; W_2 is the weight of gear pair 5-6.

Hence, the objective function is

$$W = W_1 + W_2 \tag{14}$$

and the goal of the work is to minimize the total weight W.

3.3. Constraints and boundary conditions

The shafts fitted with the gears must satisfy strength conditions, such as withstand bending torque and torsion. With the structure and the lifting capacity of the winch, the stress on the shaft at the gear-mounted location caused by the bending moment is tiny compared to the one caused by the torque. Therefore, to simplify the calculation and still satisfy the research results' reliability, the diameters of the holes where the shafts are fitted are determined as follows:

$$d_4 = \frac{1}{10}\left(\left\lfloor 10\sqrt[3]{\frac{0.5\,S_{\max} D_{tb}}{0.2[\tau]}}\right\rfloor + 1\right), \tag{15}$$

$$d_5 = \frac{1}{10}\left(\left\lfloor 10\sqrt[3]{\frac{0.5\, S_{\max} D_{tb}}{0.2[\tau]\, i\, \eta_{34}\, \eta_{56}}}\right\rfloor + 1\right), \tag{16}$$

$$d_6 = d_3 = \frac{1}{10}\left(\left\lfloor 10\sqrt[3]{\frac{0.5\, S_{\max} D_{tb} Z_3}{0.2[\tau]\, Z_4\, \eta_{34}}}\right\rfloor + 1\right), \tag{17}$$

where $[\tau]$ is the yield strength in tension of shaft material.

Based on the load capacity of the gear train and fitness for use in the winch, the gear width should be selected within a range of values, such as

$$10 \leqslant b_3 \leqslant 40, \tag{18}$$

$$10 \leqslant b_5 \leqslant 30. \tag{19}$$

Similarly, the tooth modules' value should be the standard ones specified in [18] and not greater than 5 mm. This condition is to ensure that the gears are not too big for the winch housing.

The above conditions and the equations for determining the web thicknesses in Table 2 show that the thickness values can be larger than the gear width values. To avoid this and under the structural conditions of the winch, b_{w4} and b_{w6} are determined by the following constraints:

$$b_{w4} = \begin{cases} 3.5m_3 & \text{if } b_3 > 3.5m_3\,, \\ b_3 & \text{if } b_3 \leqslant 3.5m_3\,; \end{cases} \tag{20}$$

$$b_{w6} = \begin{cases} 3.5m_5 & \text{if } b_5 > 3.5m_5\,, \\ b_5 & \text{if } b_5 \leqslant 3.5m_5\,. \end{cases} \tag{21}$$

For the tooth numbers of the gears (pinions), they are positive integers and should be not less than 17. Because of the personal rescue winch's purposes, the housing's outer dimensions are designed to be 360 mm × 350 mm × 125 mm. The number of teeth is the maximum corresponding to the gear module of 1 mm, and the mounting condition with the cover should be satisfied. Thus, the maximum value of Z_4 is 310 and that of the others is 150.

Besides, Z_3 and Z_4 must also satisfy the condition (22). That is the structural condition. The center distance between the pinion and the gear must be sufficient for the rope drum to be installed.

$$0.5m_3\,(Z_3 + Z_4) \geqslant 145. \tag{22}$$

The values of seven selected variables must ensure the conditions of contact strength and bending strength.

The constraint conditions for contact strength are

$$\sigma_{c34} = Z_{E34}\sqrt{\frac{2M_{x3}}{D_3} K_{o34} K_{v34} K_{s34} \frac{K_{H34}}{D_3 b_3} \frac{Z_{R34}}{Z_{I34}}} \leqslant [\sigma_c]_{34} \,, \tag{23}$$

$$\sigma_{c56} = Z_{E56}\sqrt{\frac{2M_{x5}}{D_5} K_{o56} K_{v56} K_{s56} \frac{K_{H56}}{D_5 b_5} \frac{Z_{R56}}{Z_{I56}}} \leqslant [\sigma_c]_{56} \,. \tag{24}$$

The constraint conditions for bending strength are

$$\sigma_{34} = \frac{2M_{x3}}{D_3} K_{o34} K_{v34} K_{s34} \frac{1}{b_3 m_3} \frac{K_{H34} K_{B34}}{Y_{J34}} \leqslant [\sigma]_{34} \,, \tag{25}$$

$$\sigma_{56} = \frac{2M_{x5}}{D_5} K_{o56} K_{v56} K_{s56} \frac{1}{b_5 m_5} \frac{K_{H56} K_{B56}}{Y_{J56}} \leqslant [\sigma]_{56} . \tag{26}$$

The values of M_{x3} and M_{x5} in equations (23) to (26) are determined from equations (3) and (4). The other elements are defined as shown in Tables 1, 2 and 3.

4. Weight optimization

4.1. Optimizing the weight with the first seven variables

The proposed algorithm to optimize the weight and the corresponding parameters is shown in Fig. 3. All vector indices of the variables are assigned an initial value of 0. The first value of each variable corresponds to the one of the first element in the vector. It is the smallest in the value group of that variable.

Based on the brute force method, in principle, each variable's value is combined with all other variable's values in turn to calculate the weight and compare the conditions from (22) to (26). If any of the conditions is not met, the weight is assigned a value of 0. Otherwise, it is assigned the calculated value. These values are sorted into a weight matrix according to a certain principle. The goal of this arrangement is to determine the values of the variables when we know the position of the weight value in the matrix.

After calculating the weight with all combinations of values of the variable and completing the creation of matrix **W**, the gear train's minimum weight is non-zero and is the smallest value among the weight values found.

Using Matlab software to program the weight optimization program according to the algorithm and above data, we get the result. The weight obtained after optimization is 4.461 kg instead of 4.968 kg, as in the previous design. The parameters corresponding to that minimum value are $m_3 = 3$ mm, $m_5 = 2.5$ mm, $b_3 = 16.5$ mm, $b_5 = 15$ mm, $Z_3 = 19$, $Z_4 = 78$, $Z_5 = 17$ and $Z_6 = 40$. The comparison between the weight and strength values at the original design and that at this

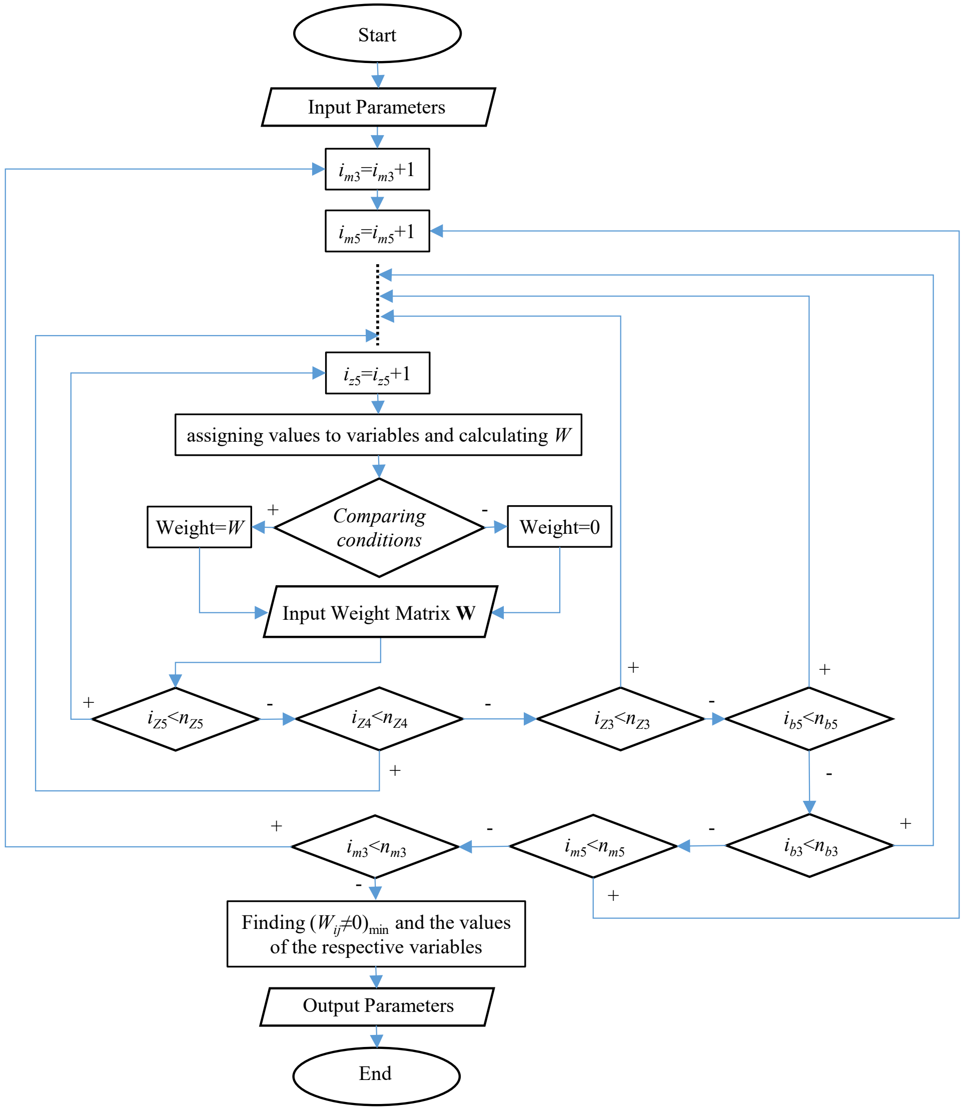


Fig. 3. Optimal algorithm

optimal stage is shown in Table 4. In there, negative signs (–) and positive signs (+) represent decrease and increase. The total weight of the gear train is reduced by 10.21%. The reduction ratio is almost evenly divided between both transmission levels. The bending stress value is still small compared to the allowable value. Meanwhile, the contact stress value is close to the limit.

However, intending to minimize the weight of the winch so that the average person can easily carry it, we continue to reduce the weight of the webs.

Table 4.

Results after optimal stage 1

Parameters	Unit	Value		Comparison
		Original design	Optimal stage 1	
m_3	mm	3	3	
m_5	mm	3	2.5	
b_3	mm	20	16.5	
b_5	mm	13	15	
Z_3	tooth	17	19	
Z_4	tooth	80	78	
Z_5	tooth	17	17	
W_1	kg	4.017	3.619	-9.91%
W_2	kg	0.951	0.842	-11.46%
W	kg	4.968	4.461	-10.21%
σ_{34}	N/mm^2	78.53	97.63	+24.32%
σ_{c34}	N/mm^2	669.08	705.66	+5.47%
σ_{56}	N/mm^2	64.92	81.26	+25.17%
σ_{c56}	N/mm^2	632.27	707.40	+11.88%

4.2. Optimizing the web weight

To further reduce the gear weight, the construction of the proposed gears (pinions) is composed of two parts. Each gear (pinion) includes the rim and web. They are joined together by an interference joint. The rim is fabricated from high-strength steel as used for the calculation in the above items. The material of the webs is replaced by a high-strength and lightweight one such as Aluminum alloy 6061-T6. For gears 4 and 6, there are some holes on the gear web to reduce weight and easily assemble and disassemble. Thus, to reduce the weight of the webs, two processes are performed. The first is to calculate the pressure generated on the contact surface between the rim and web when assembling them together with an interference fit. The second is to increase the hole number and hole diameters while still ensuring durability, stability, and other fabrication conditions.

In an interference joint, a torque is transmitted by friction force. Hence, to transmit torque M in each gear (pinion), the minimum pressure on the contact surface between the rim and web is

$$p = \frac{M}{2\pi f B c^2}, \tag{27}$$

where f is the coefficient of friction between aluminum alloy and steel, B is the web thickness at the contact position, and c is the contact radius of two materials.

The minimum force to press them to bond together is

$$F = 2\pi f c B p. \tag{28}$$

The minimum diametral interference to produce pressure p is

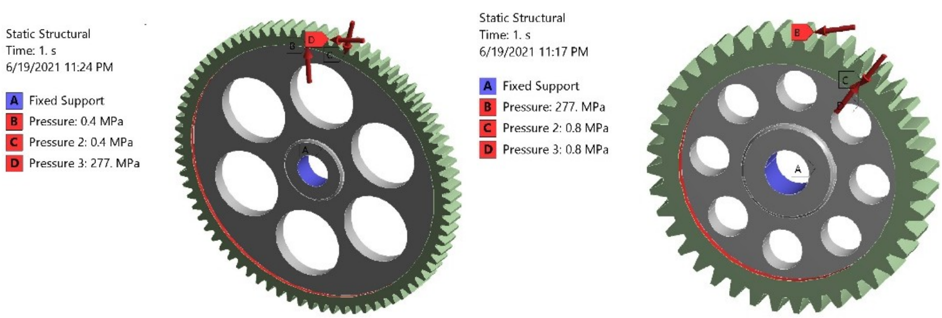
$$\Delta = pc\left\{\frac{1}{E_{Al}}\left(\frac{c^2+a^2}{c^2-a^2}-\mu_{Al}\right)+\frac{1}{E_{St}}\left(\frac{b^2+c^2}{b^2-c^2}+\mu_{St}\right)\right\}, \tag{29}$$

where a is the hole radius where is fitted with the shaft, b is the tooth root radius, E_{St} is the modulus of steel elasticity, E_{Al} is the modulus of aluminum alloy elasticity, μ_{St} is the Poisson coefficient of steel, μ_{Al} is the Poisson coefficient of aluminum [19].

The stress and displacement of two gear pairs are investigated when the hole diameters on the webs of gears 4 and 6 change between 40–55 mm and 10–20 mm, respectively, and the hole number changes from 4 to 8. The input parameters and calculation results for the case of the maximum hole number and the maximum hole diameters that still satisfy the required conditions are shown in Figs. 4–6. The required conditions are stress and displacement conditions and the dimensional relationship conditions between the holes and the gear web to satisfy the working principle of the winch.

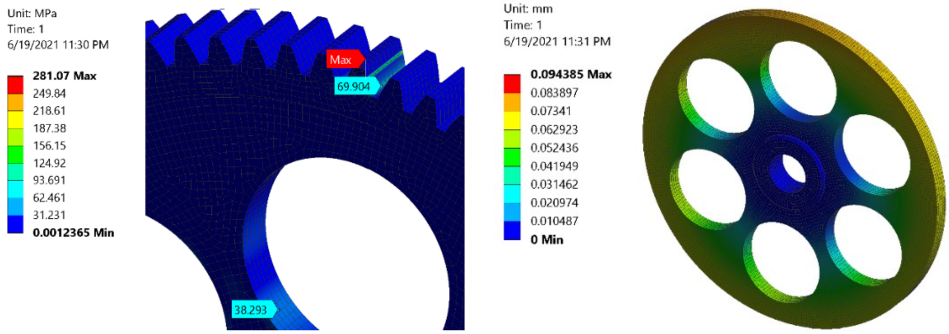
In Fig. 4, the force acting on the tooth is placed in an area calculated according to the gear design theory. The pressure caused by the interference joint is also applied. Using Ansys software, the computational model meshed with the size of 0.15 mm × 0.15 mm, 0.5 mm × 0.5 mm, and 2 mm × 2 mm on where the force is applied on the tooth surface, the neighborhood of the first place, and others, respectively. The maximum stresses and displacements on the webs and gear roots in Figs. 5 and 6 show that they are within allowable limits. The maximum stress at the contact position between the teeth (position of force) when working is the local stress. It does not reflect the nature of this study.

At this stage, the optimal results obtained are listed in Table 5. The total weight is greatly reduced and is only 2.365 kg instead of 4.461 kg as in the previous optimization. The weight reduction percent of gear pair 3-4 (49.77%) is larger than that of gear pair 5-6 (35.04%).



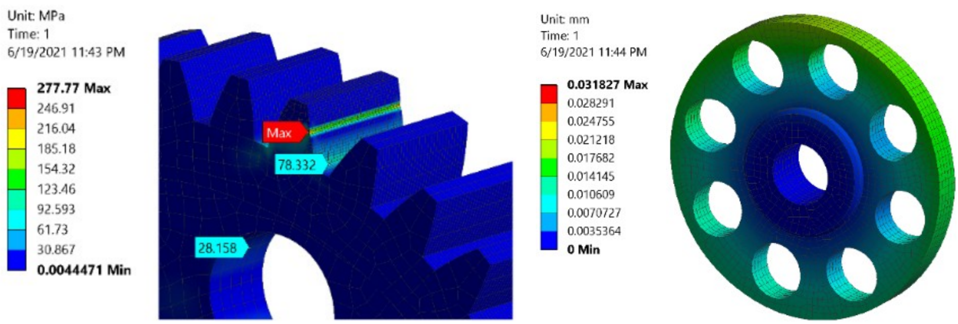
(a) gear 4 (b) gear 6

Fig. 4. Input parameters for stress and displacement testing on the webs



(a) stress (b) displacement

Fig. 5. Stress and displacement of the web of gear 4 under the loads



(a) stress (b) displacement

Fig. 6. Stress and displacement of the web of gear 6 under the loads

Table 5.

Results after optimal stage 2

Parameters	Unit	Value		Comparison
		Optimal stage 1	Optimal stage 2	
n_4	hole	6	6	
n_6	hole	4	8	
d_{p4}	mm	40.075	55	
d_{p6}	mm	9.888	15	
Material of gear (pinion) webs	–	high-strength steel	Aluminum 6061-T6	
W_1	kg	3.619	1.818	–49.77%
W_2	kg	0.842	0.547	–35.04%
W	kg	4.461	2.365	–46.98%

5. Conclusion

The paper presents the weight optimization of one two-level gear train in the personal rescue winch. The aim is to optimize the winch so that it is small in weight, and it has a reasonable construction. The research results have been achieved:

- The model of the optimization problem of the spur gear transmission system, including two levels, has been established with the case of eleven design variables in two optimal stages.
- By means of the brute force method, Matlab software, and Ansys software to solve the optimization problem, the total weight of the gears is reduced by 52.40% compared to that of the original design. The percent reduction weight at optimal stages 1 and 2 is 10.21% (compared to the original design) and 46.98% (compared to optimal stage 1).
- The proposed algorithm in this study can be used to solve similar technical optimization problems well. The smaller the values of the variables are divided, the more considerable the amount of calculation. That leads to longer computation times. However, if the divarication is satisfied with the necessary technical conditions, this method is a great choice.
- The combination of Aluminum 6061-T6 and high-strength steel on the gears helps them reduce weight significantly while ensuring the transmission requirements.
- The parameters that cannot be automatically optimized by conventional algorithms, such as the hole diameters and hole number, can be manually optimized with the help of commonly structural calculation programs in engineering.

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