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Analysis and simulation of the influence of computational loads and structural parameters on the optimal dimensions of the main girder of a double-girder box-type crane

Nguyen Anh Dung[1], Le Dang Ha[2], Nguyen Truong Giang[2] Nguyen Duc Cuong[2]
[1]*National Key Laboratory for Welding and Surface Treatment Technologies*
[2]*Ha Noi University of Industry*

Abstract
Along with the rapid development of the country, the number of vehicles has been continuously increasing over time, especially passenger cars. Therefore, improving quality and reducing breakdowns for these vehicles has become indispensable. Car lifts are essential equipment for the maintenance and repair of automobiles in garages and repair workshops. Consequently, designing and manufacturing car lifts is a highly practical endeavor to meet the current demands of the country.
Keywords: *Lifting bridge; automobiles; garage; scissor; two-post lift; lifting arm; lifting; lowering; simulation*

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I. INTRODUCTION

The rapid development of industrial machinery and automotive technology has led to increased demands for efficient and reliable maintenance equipment. Among such equipment, car lifts play a crucial role in ensuring safe and effective repair and servicing of vehicles.[1] The design of car lifts, particularly two-post and scissor lifts, must address various challenges, including structural stability, hydraulic efficiency, and adaptability to different vehicle types.[2]

The optimization of hydraulic transmission systems has become a significant focus in the development of modern lifting equipment.[3] These systems offer advantages such as high force transmission, compact size, and smoother operation compared to traditional mechanical systems. Recent studies have emphasized the role of weight optimization and structural analysis to improve the durability and performance of hydraulic lifts under repetitive loading conditions. Additionally, optimizing actuator placement and minimizing cylinder mass without compromising stability have been identified as critical areas of research, using advanced optimization algorithms.

This research aims to explore the influence of computational loads and structural parameters on the design and optimization of double-girder box-type crane systems, with a specific focus on hydraulic two-post lifts. It investigates hydraulic system configurations, geometric parameters, and material properties to ensure both strength and stability. Moreover, recommendations are provided for further improvement through advanced optimization techniques and consideration of dynamic loading conditions.

By addressing these aspects, this research contributes to the ongoing development of cost-effective and robust lifting solutions, meeting the growing needs of automotive repair workshops and industrial applications.

II. OVERVIEW OF LIFTING EQUIPMENT FOR AUTOMOBILES

A. Overview of Lifting Equipment

Automobile lifting equipment is a type of lifting machine used in maintenance and repair. Its primary purpose is to raise the vehicle, creating ample space underneath the chassis for technicians to work easily. There are various types of lifting equipment with different capacities and lifting power. Some use mechanical transmission, others use hydraulic transmission, or a combination of mechanical and hydraulic systems. Mechanical transmission can involve chain drives, screw-nut drives, or a combination of both. However, they all share the same primary power source: electrical energy. This energy, through an electric motor and gear or chain transmission, can convert the rotational motion of the motor shaft into rotational motion of a screw shaft. The screw-nut mechanism then converts this rotation into linear upward or downward motion of the lifting frame, thus lifting or lowering the vehicle. Alternatively, through a hydraulic pump, the rotational motion of the electric motor can be converted into potential energy in the form of high oil pressure. This oil pressure is then transformed into the linear motion of the lifting frame using hydraulic cylinders and parallelogram mechanisms.

B. Common Types of Car Lifts

a) Single-Post Lift

This type of lift has a simple design and uses hydraulic oil for transmission. However, its stability is low, and its lifting capacity is small, making it suitable only for small vehicles such as 4-seater passenger cars.

Figure 1. Single-post lift with four lifting arms

b) Two-Post Lift

The two-post lift is similar to the single-post lift but provides higher stability and a greater lifting capacity.

Figure 2. Two-post lift with overhead structure

c) Four-Post Lift

The four-post lift can handle heavier vehicles and offers high stability due to its large base area. However, it has limited working space and occupies a significant area in garages or workshops.

Figure 3. Four-post lift

d) Scissor Lift

This type of lift can raise heavy vehicles and offers high stability. It provides a relatively spacious working environment. Some models are highly portable and can be moved anywhere within a garage.

Figure 4. Scissor lift

III. RESEARCH RESULTS

With the advancement of modern technology, the trend toward hydraulic systems has increasingly dominated lifting machines in general and car lifts in particular. The use of hydraulic transmission systems offers the following advantages:

Capability to transmit large forces over long distances.

Smaller size and weight of the transmission system compared to other types, such as screw-nut drives.

Smooth operation with minimal noise.

Low inertia during transmission.

Ability to achieve high transmission ratios.

Built-in self-lubrication of the transmission system, extending the machine's lifespan.

Protection of the machine during overload conditions.

Flexible arrangement of the transmission system, allowing for aesthetically pleasing overall designs.

A. Defining objectives and input parameters for the hydraulic system of a two-post lift

The descriptive method used here disregards the effects of dynamic loads. In practice, during operation, dynamic loads occur whenever the mechanism moves, starts, or stops. The magnitude of these loads depends on the mass and acceleration during movement.

a) Hydraulic transmission diagram for a two-post lift

To meet the requirement for synchronized movement of the two hydraulic cylinders, the following hydraulic system diagram is chosen:

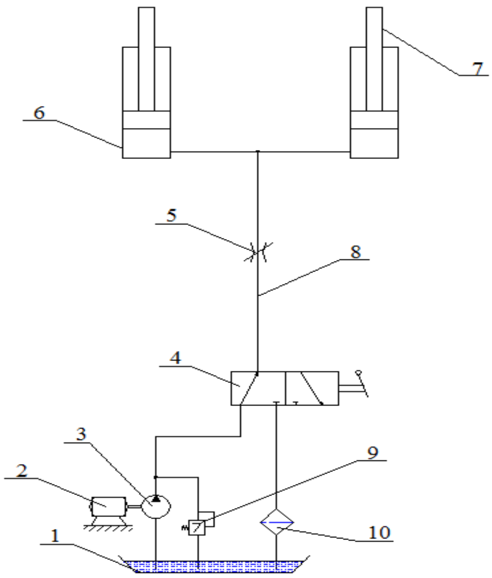


Figure 5. Hydraulic Transmission Diagram

1. Hydraulic oil reservoir; 2. Electric motor; 3. Hydraulic pump; 4. Distribution valve; 5. Throttle valve; 6. Hydraulic cylinder; 7. Hydraulic piston; 8. Oil pipes; 9. Safety valve; 10. Filter

b) Input parameters

The car lift must be capable of lifting all types of four-seater vehicles brought into the garage. Therefore, the specifications of the vehicle with the largest size and weight among the aforementioned cars were chosen, such as the Honda Civic or pickup trucks.

Wheelbase	Overall length	Overall width	Overall height	Ground clearance	Curb weight	Gross weight
2700 mm	4630 mm	1799 mm	1416 mm	133 mm	1317 kg	1740 kg

Table 1. Dimensions of the Honda Civic

B. Some calculation results

a) Geometric Parameters:

Based on the reference data, the input parameters are selected as follows:

- H1 - Average height of the technician (taken as 1800 mm).
- H2 - Safety height from the technician's head to the underside of the vehicle (taken as 260 mm).
- H3 - Height of the lifting platform (600 mm).
- H4 - Additional height for the top of the column (chosen as 160 mm).
- H5 - Height of the lift when in the lowest position (approximately equal to the ground clearance of the vehicle, taken as 100 mm).

From these dimensions, the following parameters for the column are determined:

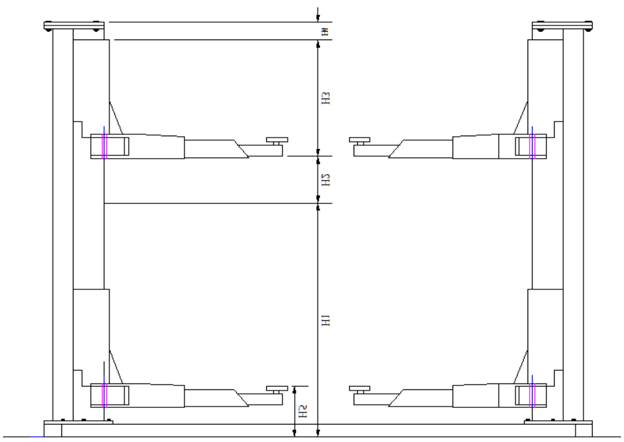
- Column height: $H_{column} = H1 + H2 + H3 + H4 + H5 = 2820$ mm
- Lifting height at the highest position: $H_{lift} = H1 + H2 = 2060$ mm
- Lifting height at the lowest position: H5 = 100 mm
- L1 - Average width of the vehicle (taken as 1900 mm).
- L2, L3 - Safety width from the side of the vehicle to the column surface (both taken as 460 mm).
- L4 - Maximum length of the lift arm.
- L5 - Minimum length of the lift arm.

From these dimensions, the following parameters for the distance between the two lift columns are determined:

- Width between the two columns: 2820 mm.
- Maximum length of the lift arm: 1170 mm.
- Minimum length of the lift arm: 762 mm.
- Maximum lifting capacity: 4000 kg.

The selected material for construction is CT38 steel, with the following properties:

- Density: $\gamma = 7800$ kg/m^3
- Yield strength: $\sigma_y = 3800$ kg/cm^2



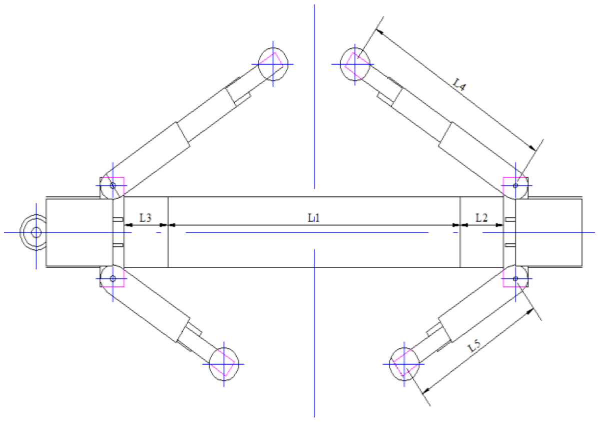


Figure 6. Height and Width Dimensions of the Car Lift

b) Determining the stroke and speed of the piston

Here, a pair of cylinders and pistons is used to lift and lower the car lift. These cylinders have identical lifting forces and properties, so the calculations are performed for a single cylinder.

The piston stroke is determined using Equation (1):

$$l_p = CO_2 - CO_1 \tag{1}$$

CO_1 - Length of segment CO in State 1 is the lowest lifting position: CO_1 = CO_2 / 2

CO_2 - Length of segment CO in State 2 is the highest lifting position: CO_2 = 2060 mm

Hence,

$$l_p = CO_2 - CO_1 = CO_2 - \frac{CO_2}{2} = \frac{2060}{2} = 1030\ mm$$

The piston's movement speed within the cylinder

Lifting time equals half of the total lifting cycle: $t = \frac{1}{2}\ minutes = 30\ seconds$

From this, the piston's speed is:

$$v_p = \frac{l_p}{t} = \frac{103}{30} = 3.43\ cm/s$$

c) Determining the force on the piston rod

Based on the structure of the lifting system, the force on the piston rod is determined as twice the force acting on the CO bar. Between the two states (lowest lifting position and highest lifting position), the force acting on the CO bar is greatest in State 1. Therefore, the force on the piston rod is calculated as follows:

$$F_p = 2 \times P_1 \tag{2}$$

According to the previous calculations, $P_1 = 2520\ kg$.

$$F_p = 2P_1 = 5040\ kg$$

d) Determining the lifting power of the cylinder piston

The lifting power of the cylinder piston is determined using the formula:

$$N = F_P \times v_P \tag{3}$$

F_P – Force acting on the piston rod, $F_P = 5040\ kg = 50400\ N$

v_P – Piston velocity relative to the cylinder, $v_P = 3.43\ cm/s = 3.43 \times 10^{-2}\ m/s$

Therefore,

$$N = 50400 \times 3.43 \times 10^{-2} = 1729\ W = 1.729\ kW$$

e) Piston and cylinder diameter parameters

Working pressure of the cylinder:

$$P_x = 16\ MPa = 16 \times 10^6\ N/m^2$$

Piston diameter:

$$F_p = \frac{P_x \pi D^2}{4} \tag{4}$$

$$=> D = \sqrt{\frac{4F_p}{P_x \times \pi}} = \sqrt{\frac{4 \times 50400}{16 \times 10^6 \times 3.14}} = 0.063\ m$$

Choose $D = 70\ mm$

The inner diameter of the cylinder is equal to the piston diameter D:

$$D_t = D = 70\ mm$$

Select the scaling factor $\varphi = 1.25$

We have:

$$\varphi = \frac{D_t^2}{D_t^2 - d_n^2} \tag{5}$$

$$d_n = D_t\sqrt{\frac{\varphi - 1}{\varphi}}$$

$$d_n = 70\sqrt{\frac{1.25 - 1}{1.25}} = 31.3$$

Select the piston rod diameter: $d_n = 35\ mm$

Select the inner diameter of the piston rod

$$\sigma = \frac{N}{F} < [\sigma] \tag{6}$$

In which

$N = F_P = 50400N$

$[\sigma] = 2100\ daN/cm^2$

$$=> F > \frac{N}{[\sigma]}$$

Therefore,

$$F_{min} = \frac{N}{[\sigma]}$$

$$\frac{\pi}{4}(d_n^2 - d_{tmax}^2) = \frac{N}{[\sigma]}$$

$$d_{tmax} = \sqrt{d_n^2 - \frac{4N}{\pi[\sigma]}} = \sqrt{3.5^2 - \frac{4 \times 5040}{\pi \times 1600}} = 2.87\ cm$$

We select $d_t = 25\ mm$

Determine the outer diameter of the cylinder

To determine the outer diameter of the cylinder, we have:

$$\frac{\pi}{4}(D_{nmin}^2 - D_t^2) = \frac{N}{[\sigma]} \tag{7}$$

$$D_{nmin} = \sqrt{D_t^2 + \frac{4N}{\pi[\sigma]}} = \sqrt{7^2 + \frac{4 \times 5040}{\pi \times 1600}} = 7.28\ cm$$

Choose $D = 900\ mm = 9\ cm$

e) Stability and durability of the piston and cylinder

The working length factor of the piston rod: We consider the piston rod with one end fixed into the cylinder and the other end connected to the lifting arm through a chain combined with a roller. Therefore, the conversion factor for the calculated length is $\varphi = 1$.

The stability condition of the piston rod under axial compression is:

$$\sigma = \frac{N}{\varphi \times F_{ng}} < R_{tt} \tag{8}$$

With a cross-sectional area in the shape of a ring, $d_t = 25\ mm$, $d_n = 35\ mm$.

The cross-sectional area of the section:

$$F_{ng} = \frac{\pi}{4} \times (d_n^2 - d_t^2) = \frac{\pi}{4} \times (3.5^2 - 2.5^2) = 4.71\ cm^2$$

The moment of inertia of the cross-section:

$$J_x = 0.05 \times (d_n^4 - d_t^4) = 0.05 \times (3.5^4 - 2.5^4) = 5.55\ cm^4$$

So, the radius of gyration of the cross-section:

$$i = \sqrt{\frac{J_x}{F_{ng}}} = \sqrt{\frac{5.55}{4.71}} = 1.08\ cm$$

The slenderness ratio of the piston rod:

$$\lambda = \frac{\mu l}{i} = \frac{1 \times 103}{1.08} = 95.37$$

When $\lambda = 95.37$ then $\varphi = 0.64$. Then,

$$\sigma = \frac{5040}{0.64 \times 4.71} = 1671\ daN/cm^2 < R_{tt}$$

⇨ Ensure stability

Check the cylinder wall strength

- The maximum oil pressure in the cylinder is 25 Mpa
- The inner diameter of the cylinder D_t = 70 mm
- The outer diameter of the cylinder D_n = 90 mm

Based on the formula for determining the stress in a pressure vessel, the stress in the oil cylinder is calculated as follows:

$$\delta_r = -\frac{pa^2}{b^2 - a^2}\left(1 - \frac{b^2}{r^2}\right) \tag{9}$$

$$\delta_t = -\frac{pa^2}{b^2 - a^2}\left(1 + \frac{b^2}{r^2}\right) \tag{10}$$

In which:

δ_r - Compressive stress

δ_t - Tensile stress

p - Pressure in the cylinder, p = 25 MPa = 25 × 10^6 N/m^2

b - Outer radius of the cylinder, b = 35 mm

a - Inner radius of the cylinder, a = 45 mm

r - Distance from the center to the point where stress needs to be determined

For a pressure vessel, the stresses are maximum at the inner edge of the cylinder, where r = a

Maximum tensile stress:

$$\sigma_{tmax} = p\frac{b^2 + a^2}{b^2 - a^2} = 25\frac{45^2 + 35^2}{45^2 - 35^2} = 101.5\ MPa$$

Therefore, the cylinder wall strength condition is ensured.

Check the stability condition of the cylinder. The cylinder has:

- D_t = 70 mm
- D_n = 90 mm

Similar to the piston rod, the conversion factor for the calculated length of the cylinder is also similar to the length calculation factor for the piston rod $\mu = 1$.

$$\sigma = \frac{N}{\varphi \times F_{ng}} < R_{tt} \tag{11}$$

In which:

$$N = F_p = 5040\ daN$$
$$F_{ng} = \frac{\pi}{4} \times (D_n^2 - D_t^2) \tag{12}$$
$$F_{ng} = \frac{\pi}{4} \times (9^2 - 7^2) = 25.13\ cm^2$$

φ - The reduction factor

The moment of inertia of the cross-section:

$$J_x = 0.05 \times (D_n^4 - D_t^4) = 0.05 \times (9^4 - 7^4) = 208\ cm^4$$

So, the radius of gyration of the cross-section:

$$i = \sqrt{\frac{J_x}{F_{ng}}} = \sqrt{\frac{208}{25.13}} = 2.87\ cm$$

The slenderness ratio of the piston rod:

$$\lambda = \frac{\mu l_x}{i} = \frac{1 \times 103}{2.87} = 35.89$$

In which,

l_x - The stroke of the cylinder is also the stroke of the piston, $l_x = l_p = 103\ cm$

When $\lambda = 35.89$, using the interpolation method, we obtain: $\varphi = 0.928$.

$$\sigma = \frac{5040}{0.928 \times 25.13} = 216\ daN/cm^2 < R_{tt}$$

Thus, the cylinder ensures the stability condition.

f) Simulation results

In this section, we present the outcomes of the finite element analysis (FEA) conducted using CAD/CAE software. The results provide critical insights into the structural integrity and performance of the two-post lift under loading conditions. The simulations were carried out with a maximum lifting capacity of 4,000 kg, and the findings are summarized as follows:

The Von Mises stress distribution (*Figure 7*) reveals the critical stress zones within the lifting arms and columns. The maximum stress occurred at the joint between the lifting arms and the baseplate. This stress value remains below the yield strength of the material (CT38 steel, 380 MPa), confirming the structural safety under static load conditions.

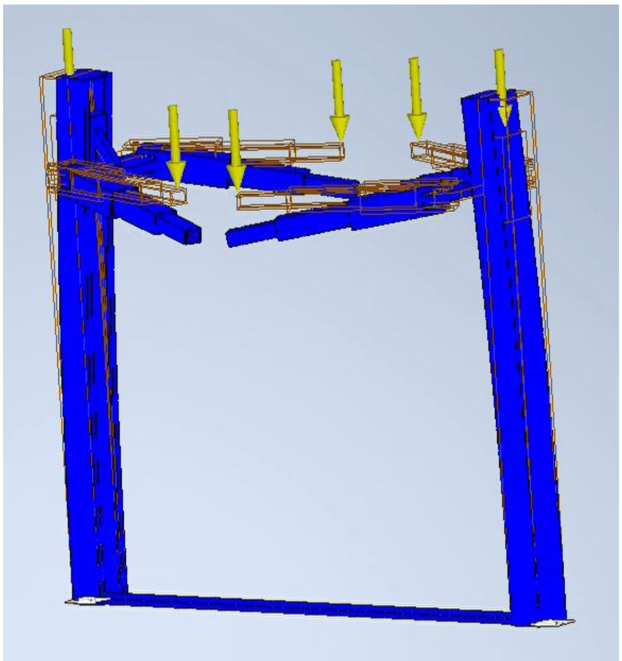


Figure 7. Result of Von Mises Stress

The 1st principal stress analysis (*Figure 8*) identifies the areas subjected to maximum tensile stress. These stresses are predominantly concentrated near the upper section of the lifting columns. The results demonstrate adequate resistance to tensile loading, with no risk of material failure.

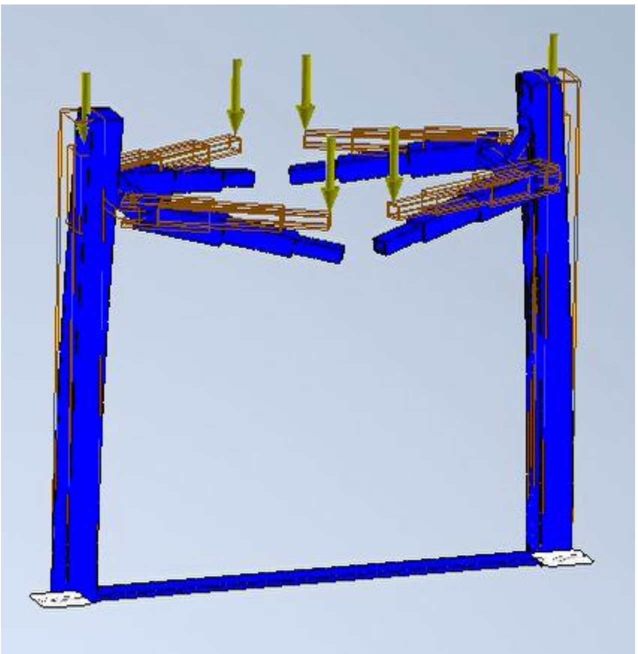


Figure 8. Result of 1st Principal Stress

The 3rd principal stress (*Figure 9*) highlights compressive stress zones. The highest compressive stress is located at the base of the lifting columns. This result is within acceptable limits, indicating that the structure can effectively withstand compressive forces during operation.

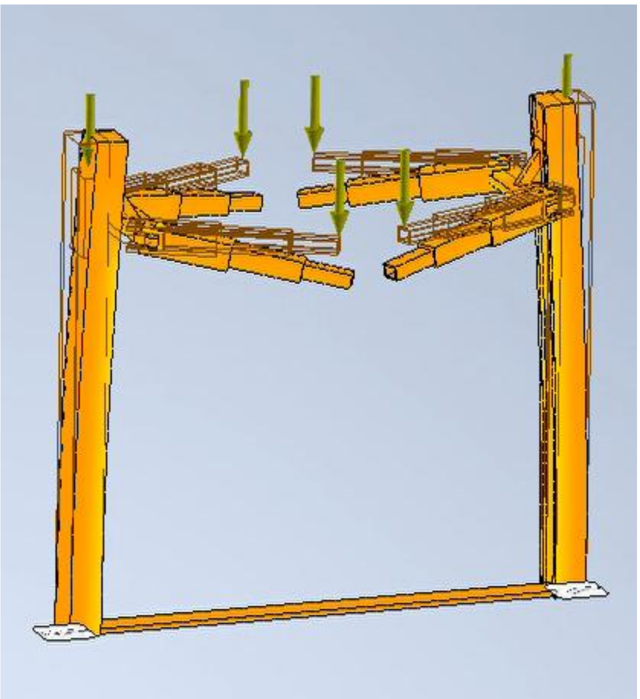


Figure 9. Result of 3rd Principal Stress

The displacement simulation (*Figure 10*) shows the deformation profile of the two-post lift under maximum load. This minimal deformation confirms the rigidity of the design and ensures consistent performance under operational conditions.

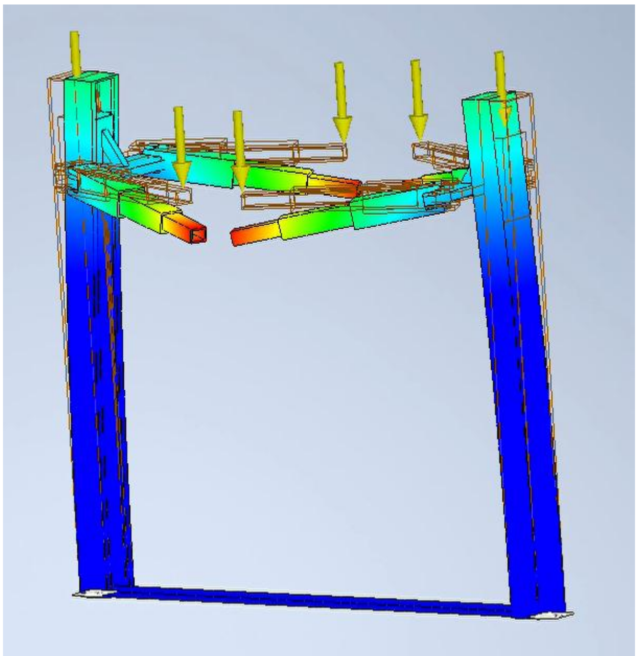


Figure 10. Result of Displacement

IV. CONCLUSIONS AND RECOMMENDATIONS

A. Conclusions

a) The results of the calculations show that the hydraulic system meets the working conditions, ensuring both strength and stability conditions.
b) The optimization problem to reduce material costs, weight, and size has not been addressed yet.

B. Recommendations

a) Further research on calculation methods and the use of artificial intelligence (optimization for continuous variables) should be conducted, and improvements made, to apply to the optimization problem for reducing size and weight.

b) The problem considered at special points and static loads still does not fully account for the damping time and other dynamic loads and natural vibrations. Attention should be given to calculating these parameters to achieve more accurate results.

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