

INTEGRATED DESIGN, ANALYTICAL MODELING, AND INITIAL ANGLE OPTIMIZATION FOR TORQUE-EFFICIENT SCISSOR LIFT SYSTEMS

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Abstract - In the present work is proposed an integrated methodology for the design, analytical calculations, and optimization of a high-load scissor lift mechanism with a focus on improving torque efficiency through initial angle selection. The proposed approach combines systematic mechanical design procedures with analytical modeling of the force transmission characteristics to evaluate the influence of the initial scissor geometry on torque requirements and overall system performance.

A full analysis of the initial configuration is conducted to investigate its effect on mechanical advantage and required effort. Based on the developed analytical framework, an optimization strategy is applied to identify an initial angle range that enhances torque efficiency while satisfying structural and operational design constraints. The complete mechanical design is developed in accordance with standard engineering practices, ensuring consistency between theoretical calculations and practical implementation.

The results highlight that the initial geometric configuration plays a critical role in determining the efficiency of force transmission within the mechanism. The proposed approach demonstrates that appropriate selection of the initial angle significantly improves torque utilization and contributes to a more efficient sizing process. The presented methodology offers a practical and scalable design framework for the development of torque-efficient scissor lift systems in industrial applications.

Keywords: Scissor lift mechanism; Mechanical design; Analytical calculations; Initial angle optimization; Torque efficiency; Parametric analysis; Structural design; Engineering optimization.

1. Introduction

Scissor lift mechanisms are widely used in industrial, construction, and material handling applications due to their compact structure, high load-bearing capability, and vertical lifting efficiency. Their ability to convert horizontal actuation into controlled vertical motion makes them a fundamental solution in lifting platforms, adjustable workstations, and automated handling systems. Despite their widespread adoption, achieving optimal mechanical efficiency and reduced actuator demand remains a key engineering challenge, particularly in high-load configurations.

The performance of a scissor lift system is strongly influenced by its geometric configuration

and design parameters. Among these, the initial scissor angle plays a critical role in determining mechanical advantage, force transmission characteristics, and actuator requirements throughout the lifting cycle. Ogbemhe et al. [1] investigated the multi-physical behavior of a planar translational scissor lift mechanism developed for maintenance applications in railway transmission systems. Their study demonstrated the importance of integrating kinematic and dynamic analyses for predicting system performance and actuator requirements under operational conditions. Yadav et al. [2] expanded the application of scissor mechanisms by developing a tri-axis motion system based on a uniaxial scissor unit and double universal joints, highlighting the influence of mechanism geometry on motion capability and structural

behavior. Similarly, Zhao et al. [3] explored deployable scissor mechanisms for robotic harvesting end-effectors, focusing on kinematic design and workspace optimization.

These studies confirm that geometric parameters strongly affect the performance, motion characteristics, and required actuation forces of scissor-based systems. However, the systematic optimization of the initial scissor configuration with respect to actuator force reduction and mechanical efficiency remains an important research challenge, particularly for high-load lifting applications.

Traditional design approaches often rely on empirical selection of geometric parameters combined with standard mechanical calculations for component sizing. While such methods ensure functional feasibility, they do not necessarily yield optimal configurations from a torque or energy efficiency perspective. As a result, systems may be overdesigned in terms of actuator capacity, leading to increased cost, weight, and energy consumption.

In recent years, analytical and optimization-based design methodologies have been increasingly applied to mechanical lifting systems. These approaches enable systematic evaluation of the relationship between geometric parameters and mechanical performance.

Shi et al. [4] investigated the geometric compatibility of scissor mechanisms for deployable space reflector structures, while Han et al. [5] analyzed the dynamic behavior of a scissors-based truss mechanism, emphasizing the influence of configuration parameters on system performance. Luo et al. [6] demonstrated the importance of accurate torque and motion parameter evaluation in mechanical system validation.

However, existing studies primarily focus on general mechanism behavior or dynamic response, while limited attention has been given to the systematic optimization of the initial configuration in high-load scissor lift systems with respect to torque efficiency.

In contemporary engineering research, a clear trend is observed towards the application of systematic experimental approaches and process optimization, particularly in the design and analysis of systems with complex geometry and stringent reliability requirements. This tendency is especially relevant for high-load mechanical systems, where the interaction between design parameters and operational performance must be thoroughly analysed and optimised at an early stage [7].

In parallel, a number of studies emphasise the influence of technological and design processes on the structural behaviour and mechanical performance of engineered systems, highlighting the necessity for precise parameter control in order to achieve optimal functional characteristics [8].

Motivated by this research gap, the present study focuses on the integrated design, analytical calculation, and optimization of the initial scissor angle in a high-load lifting mechanism. Tsuda et al. [9] investigated the composition of curvilinearly extendable tubular scissor mechanisms, demonstrating the influence of geometric configuration on deployability and structural behavior. Pan et al. [10] analyzed scissor lift operation from a safety and dynamic loading perspective, highlighting the importance of mechanism behavior under real operating conditions. However, a systematic methodology that directly links the initial geometric configuration with actuator torque requirements for efficient sizing of high-load lifting systems remains limited. Therefore, this study proposes an analytical framework for evaluating and optimizing scissor lift parameters with respect to actuator demand and mechanical efficiency.

In this work, the actuation system is based on a ball screw mechanism, which converts rotational motion into linear displacement and enables vertical lifting through the scissor linkage. Its selection is based on high transmission efficiency, low friction losses, and precise motion control capability. Jud et al. [11] investigated the influence of preload adjustment on ball screw performance, while Alberdi et al. [12] analyzed contact kinematics and force transmission characteristics. Cheng et al. [13] evaluated accuracy degradation under variable loading conditions, emphasizing the effect of operational factors on ball screw reliability. These studies support the application of ball screw mechanisms for precise load transmission and actuator performance evaluation in lifting systems.

The proposed approach introduces a parametric evaluation of the initial angle, enabling systematic comparison of different configurations in terms of mechanical efficiency. Previous studies have demonstrated the importance of parameter-based design approaches for mechanisms and actuator systems [14], [15], [16].

Based on analytical calculations, an optimized range of initial angles is identified that improves torque utilization while satisfying structural and functional design constraints. This methodology contributes to improved early-stage design decision-making and supports the development of more efficient scissor lift systems.

The remainder of this paper is structured as follows: Section 2 presents the mechanical design methodology and analytical formulation. Section 3 describes the angle selection parametric approach and the structural calculation of the mechanism. Section 4 concludes the study.

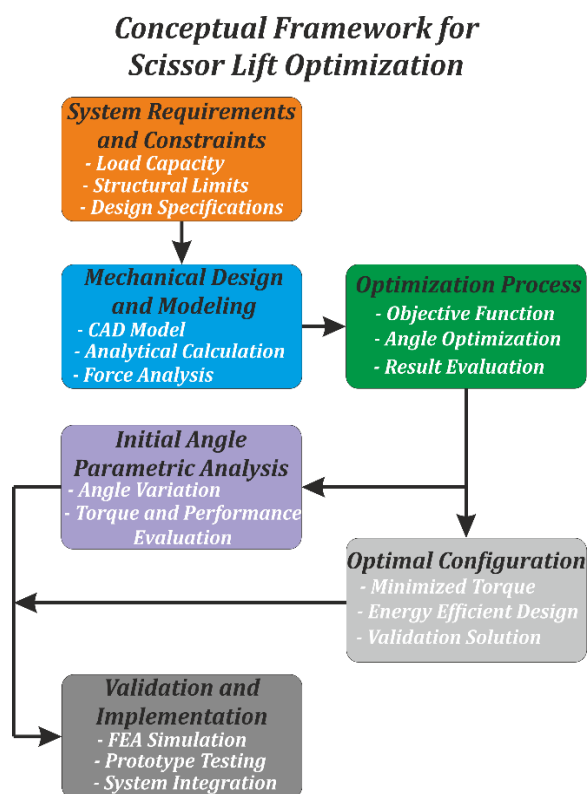


Figure 1: Conceptual Framework Optimization

Based on the mechanical behavior of scissor lift systems and the role of geometric configuration in force transmission, the following research hypotheses are formulated:

H1 — Influence of Initial Angle on Torque Demand

The initial scissor angle has a significant nonlinear influence on the torque required to initiate and sustain lifting in a high-load scissor lift mechanism.

H2 — Existence of an Optimal Geometric Configuration

There exists an optimal range of initial scissor angles that minimizes torque requirements while maintaining structural feasibility and load-carrying performance.

H3 — Efficiency Improvement Through Geometric Optimization

Optimizing the initial scissor angle leads to improved torque efficiency compared to non-optimized (arbitrary or conventionally selected) geometric configurations.

H4 — Sensitivity of Mechanical Performance to Initial Configuration

The mechanical performance of a scissor lift system, in terms of torque demand and force transmission efficiency, is highly sensitive to variations in the initial geometric configuration, particularly in the lower working position.

H5 — Design-Driven Analytical Predictability

The torque behavior of the scissor lift system can be accurately predicted using an analytical calculation-based model that incorporates geometric parameters, enabling reliable design-level optimization without requiring full experimental calibration.

Figure 1 highlights the conceptual framework for scissor lift optimization.

2. Integrated Design of the Machine

2.1 Main Points

From an engineering design perspective, the scissor lift is a classic example of transforming horizontal movement into vertical lifting through a geometrically determined kinematic scheme. The design is characterized by relatively simple mechanics, high load capacity and good stability during operation, which makes it a preferred solution in environments requiring safe access to a certain height.

The main structural elements include a supporting frame, scissor arms, articulated joints, a working platform and a drive mechanism with ball screw movement, translating the horizontal movement into vertical – Figure 2.

When designing such systems, factors such as load capacity, bending and torsional resistance, stability at maximum height, and operational safety are of key importance. Particular attention is required for material selection, arm geometry optimization, and load analysis at critical structural points. Vild and Bajer [17] investigated flexural buckling behavior using numerical analysis, demonstrating the importance of stability evaluation in loaded structures.

Jian et al. [18] and Zhao et al. [19] studied the mechanical behavior and load capacity of strengthened steel structures, emphasizing the influence of reinforcement and geometric parameters on structural performance. Lebastard et al. [20] analyzed lateral-torsional buckling of welded beams, highlighting the role of structural configuration in stability assessment.

These studies demonstrate the applicability of numerical methods, including finite element analysis (FEA), for evaluating stresses, deformations, and stability in load-bearing mechanical systems.

In the context of engineering design, the scissor lift platform is not considered just as a lifting mechanism, but as a complex system that integrates mechanics, safety, ergonomics and manufacturing processability. This makes it a suitable object for analysis and optimization within the framework of scientific research aimed at developing effective and sustainable engineering solutions. - Figure 2, 3 and Table 1.

Table 1. Construction elements

Position	Group
1	Base frame
2	Arms - internal
3	Arms - External
4	Working platform
5	Motor-reducer group
6	Linear guides
7	Support 1
8	Support 2

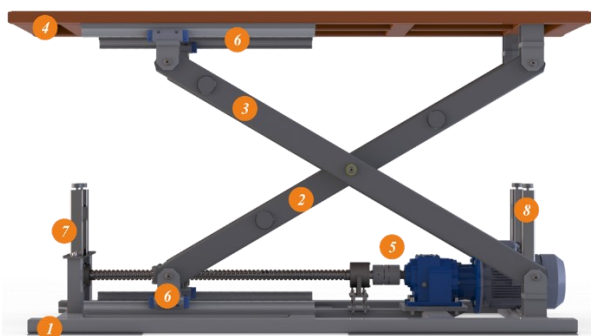


Figure 2: Main components of the mechanism



Figure 3: Scissor lifting mechanism Isometric View

2.2 Working Positions

The general dimension of the mechanism are highlighted in Figure 4. In the scissor lift platform, two main operating positions can be distinguished - lower (folded) and upper (unfolded), which determine the operating cycle and functional state of the system. - Figure 5.

In the lower position, the platform is in a fully retracted state, in which the scissor mechanism is maximally folded. In this mode, the height of the structure is minimal, which ensures compactness and facilitates transportation, positioning and storage of the equipment. The loads in the elements

are relatively low, and the center of gravity is located closer to the base, which increases stability during movement. This position is also typical for the initial phase of work, before starting the lift.

The upper position represents a fully extended state of the scissor mechanism, in which the platform reaches its maximum working height [21]. In this mode, the arms are spread out at a certain angle, and the forces in the hinge joints and supporting elements are significantly higher due to the increased bending and compressive loads [22]. System stability becomes a critical factor as the lifting height increases the influence of external influences such as eccentric loads and dynamic forces. Therefore, the structure is designed to ensure safe operation through stops, locking mechanisms and mechanical drive control [23].

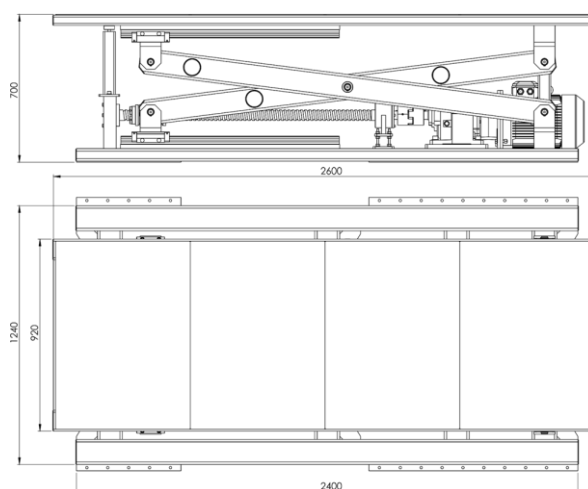


Figure 4: Overall dimensions of the platform and the supporting frame

A compact scissor lift platform with a working area of 2600 × 920 mm and a payload of up to 1000 kg is used in various industrial, logistics and service applications where a stable and safe lifting system with limited dimensions and high load capacity is required.

Its main purpose is to lift and position loads or work teams at different heights within production and service processes. Thanks to the relatively compact width and extended working surface, the platform is suitable for working in limited spaces, such as warehouses, production lines and service areas.

Main parameters:

- Payload – 1000 kg (on assignment);
- Lifting stroke – 1200 mm (minimum) – H (on basic);
- Lifting speed achieved – 0,06 – 0,1 m/s;
- General dimensions: L = 2600 mm; B = 920 mm (chosed - Figure 4);
- Maximum stroke from design: H = 2000 – 700 = 1300 mm; **H = 1300 mm**; (Figure 5)

$$H = H_2 - H_1 \quad (1)$$

The choice of overall dimensions of the work platform 2600×920 mm is justified by a combination of functional, structural and ergonomic requirements arising from the purpose of the facility and operating conditions.

First of all, the length of 2600 mm is tailored to the need to work with standard load units, including Euro pallets (1200×800 mm), allowing the placement of one or two pallets depending on the load configuration. The extended working area provides sufficient space for cargo manipulation, tools or assembly operations, without the risk of compromising the stability or safety of the operator.

The width of 920 mm was chosen as an optimal compromise between compactness and working functionality. It allows passage through standard industrial corridors, doors and service areas, while providing sufficient working surface for safe positioning of the load and operator. The reduced width compared to more massive industrial platforms results in less dead weight and lower requirements for the drive mechanism, which improves the energy efficiency of the system.

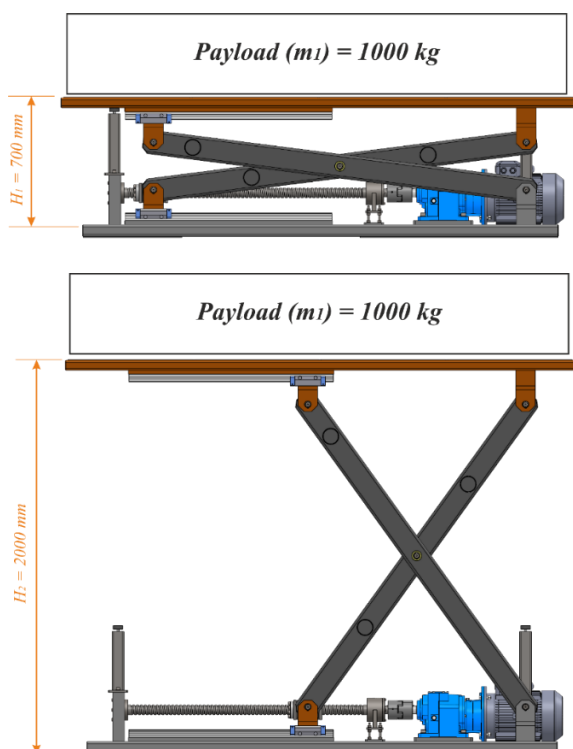


Figure 5: Lower position / Upper position

2.3 Motor-gearbox Group

The motor-gearbox group is the main drive unit in the specific configuration. It provides conversion of electrical energy into mechanical motion with appropriate torque and speed parameters, consistent with the load requirements.

In the considered configuration, the group includes the following elements (Figure 6).



Figure 6: Motor-gearbox group

- Gearbox – Helical Gear Reducer, H062 Series, $i = 7,39$ – Position 1; (Chapter 3 selection);
- Motor - AAH132M4BB5 – 9 kW – Position 2; (Chapter 3 selection);
- Coupling – ALS-095-R-40H-40H – 40 mm (Nominal shaft) – Position 3;
- Bearing – 30308T72_DB – Position 4;
- Ball Screw 50x20 R50-20K4-FSC – Working length – 1230 mm – Position 5;
- Bearing - NU_2308_ECML – Position 6.

2.4 Base Frame Design

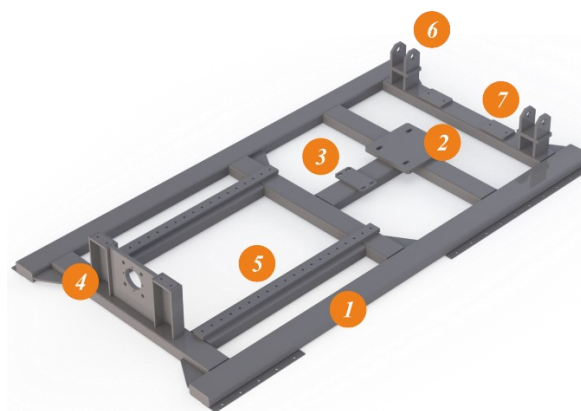


Figure 7: Base frame design

The base frame – Figure 7, is a basic structural element that determines the strength, stability and reliability of the entire system. In scissor lift platforms, it plays a key role in absorbing and distributing loads, ensuring safe and efficient operation of the mechanism. Its proper design is critical for the durability and functionality of the facility. In our case, the supporting frame is made of standard profiles and various structural elements that help secure the other components of the structure such as guides, arms and platform. It is manufactured as a single welded assembly, and then a suitable coating is applied, which will preserve its appearance and operability for a longer time – Table 2.

Table 2. Base frame components

Position	Component
1	Profile 120x60x6 (L = 11 140 mm)
2	Plate motor-reducer
3	Plate - 30308T72_DB
4	Support 1 base
5	Position linear guides
6	Position pin arms
7	Support 2 base

Main features:

- Material: S235JR (EN 10025-2);
- Heavy construction. Total weight: 250 kg;
- Possibility of direct attachment to the floor;
- Possibility of adjusting the position of the motor - reducer and axial bearing, thanks to elliptical mounting holes.

2.5 Arms Frame Design



Figure 8: Inside arms



Figure 9: Outside arms

The arms of the scissor lift platform – Figure 8 and Figure 9, are the main moving element of the mechanism by which the vertical lifting is realized. They form the characteristic “scissor” structure and play a key role in the conversion of the driving force into the elevation of the working platform.

The main purpose of the arms is:

- to provide the kinematics of lifting, by unfolding and folding;
- to transfer the load from the platform to the main frame;

- to maintain the stability of the structure at different heights;
- to distribute the forces evenly between the individual elements.

When the drive mechanism (for example, a ball screw or a hydraulic cylinder) operates, the arms unfold, which leads to an increase in the height of the platform.

The arms are arranged in pairs, connected by hinge joints in the middle, forming an X-shaped (scissor) system – Table 3.

Table 3. Arms frame components

Position	Component
1	Profile 120x50x6 (L = 8000 mm)
2	Profile, tube – 80x6 (L = 2310 mm)
3	Profile, tube – 50x5 (L = 780 mm)

Main features:

- Material: S235JR (EN 10025-2);
- Total weight of both arms: 140 kg;
- Closing of all possible openings of the profile and pipes with sheet metal – 3 mm;

2.6 Platform

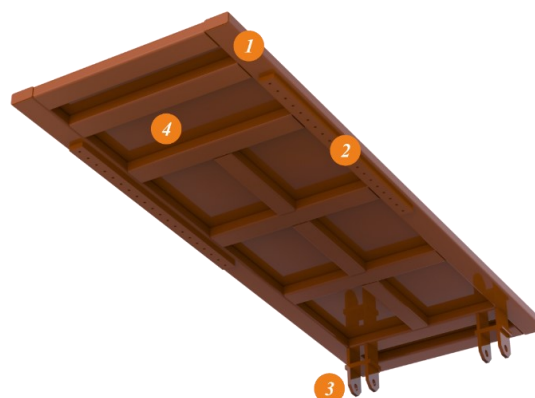


Figure 10: Platform

The platform of the scissor lift – Figure 10 - is the upper working element of the construction, designed to accommodate cargo, equipment or operators during lifting. It is directly connected to the scissor mechanism and plays a key role in the functionality and safety of the system.

The main purpose of the platform is:

- to provide a stable working surface;
- to receive and transfer the payload to the scissor mechanism;
- to ensure safety when working at height;
- to allow efficient performance of technological operations.

The platform is the final load point and is therefore a critical element in the design of the entire structure – Table 4.

Table 4. Platform components

Position	Component
1	Profile 100x50x5 (L = 11 665 mm)
2	Position linear guides
3	Position pins axis
4	Sheet metal cover – 3 mm

Main characteristics:

- Material: S235JR (EN 10025-2);

- Total weight: 205 kg;

3. Structural Calculation Methodology

3.1 Power and Torque Assessment

Payload (as specified) – 1000 kg (m_1);

Load – Arms – 150 kg (m_2);

Platform – 205 kg (m_3);

Other elements: 45 kg (m_4);

Total load – $m = 1400$ kg;

Lifting height: 1200 mm (on assignment) – $H = 1300$ mm;

$$m = m_1 + m_2 + m_3 + m_4 \quad (2)$$

When designing a scissor lift platform, the selection of the motor-gearbox group is formulated as an optimization problem, which seeks the most appropriate combination of power, torque, lifting speed and design limitations of the mechanism. The main goal is to ensure reliable operation at the specified payload, while minimizing energy costs and avoiding oversizing the drive.

$$F = m \cdot g \quad (3)$$

where:

F – Force generated by the combined load [N];

m – Total mass of all components [kg].

$$F = 13\,734\,N$$

The maximum permissible load lifting speed is $V = 0,1$ m/s, and the preliminary calculations are based on this value.

$$P = F \cdot V \quad (4)$$

where:

P – Power required to lift the load [W];

V – Lifting speed [m/s].

$$P = 1373,4\,W$$

When choosing a motor with a specific power, it must be taken into account the principle of operation of the scissor mechanism. When retracted to the lower position, a significant decrease in the working angle - α is possible, which would lead to a multiple increase in force between 5 and 10 times depending on the angle value.

Taking this into account, we consider the various options for dealing with this load.

At the initial stage, two drive options are considered:

- electric motor with power 9 kW, 1500 rpm

- electric motor with power 11 kW, 1500 rpm

In both cases, the rotation frequency is the same, which allows a direct comparison of the results obtained at different gear ratios.

The optimization was carried out by sequentially studying the required lifting torque, the linear speed of the platform and the maximum load according to the variation of the minimum angle when positioning the platform in the lower position.

The reducer plays a key role, since it reduces the speed and, accordingly, increases the torque.

A typical compromise is observed - with a higher gear ratio, high torque is achieved but low speed, while with a lower gear ratio - higher speed but insufficient torque.

The torque is calculated using Formula 5.

$$T_1 = \frac{9550 \cdot P_M}{n} \quad (5)$$

where:

T_1 – Motor torque [Nm];

P_M – Nominal power [W];

n – Rounds per minute [rpm].

At 9 kW – $T_1 = 57,3$ Nm

At 11 kW – $T_1 = 70$ Nm

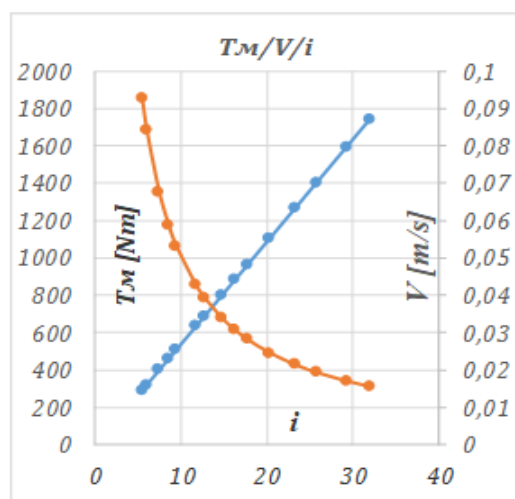
With this type of reducer, there are different values for the gear ratio, while maintaining the dimensions of the system. – Table 5.

Table 5. Reducer ratios

Position	i - ratio
1	5,38
2	5,93
3	7,39
4	8,5
5	9,39
6	11,69
7	12,67
8	14,75
9	16,29
10	17,67
11	20,28
12	23,27
13	25,7
14	29,33
15	32

Table 6. Torque values

i	T_M (9 kW) [Nm]	T_M (11 kW) [Nm]	V [m/s]
5,38	292,86	357,92	0,09294
5,93	322,8	394,51	0,08432
7,39	402,27	491,65	0,06766
8,5	462,7	565,49	0,05882
9,39	511,14	624,7	0,05325
11,69	636,35	777,72	0,04277
12,67	689,69	842,92	0,03946
14,75	802,92	981,3	0,0339
16,29	886,75	1083,75	0,03069
17,67	961,87	1175,56	0,0283
20,28	1103,94	1349,2	0,02465
23,27	1266,7	1548,12	0,02149
25,7	1398,98	1709,78	0,01946
29,33	1596,58	1951,28	0,01705
32	1741,92	2128,91	0,01563


 Figure 11: Graphic – relation $T_M / V / i$ at 9 kW

When choosing the gear ratio, it must take into account that as the gear ratio increases, the torque increases, but lose the speed of the system. Table 6 provides a comparative analysis of the torque obtained and the speed of the mechanism for different types of motors with different gear ratios

In Figure 11 and Figure 12 the relationship between T_M / i and V / i at 9 kW and 11 kW is clearly visible.

In this configuration the following dependencies and const. are available.

$$T_M = f(P_M, i, n, \eta) \quad (6)$$

$$T = f(F, p) \quad (7)$$

where:

T_M – Torque of the motor-gearbox group [Nm];

T – Torque obtained from the available load [Nm];

p – Screw pitch – [m];

η – Efficiency of the motor-gearbox group ($\eta = 0,95 = \text{const.}$);

t – Lifting time for full height [s].

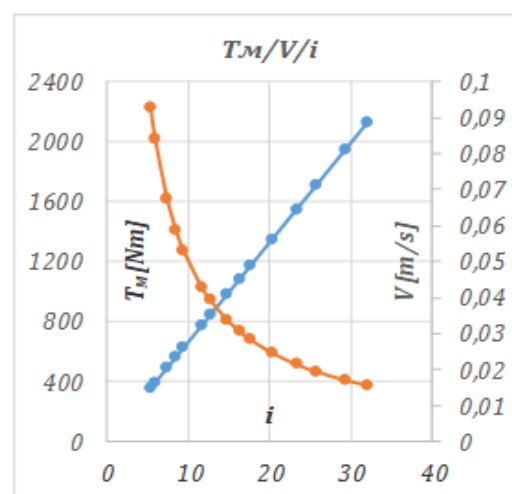
In this case, a screw with a diameter $d = 50$ mm and a pitch $p = \text{const.} = 20$ mm is chosen.

$$T_M > T \quad (8)$$

$$V = f(n, p, i) \quad (9)$$

Due to the restrictions set - $n = 1500$ rpm; $p = 20$ mm; $\eta = 0,95$; $p = 9 / 11$ kW – from (6) and (9):

$$T_M, V = f(i) \quad (10)$$


 Figure 12: Graphic – relation $T_M / V / i$ at 11 kW

From both graphs it is clear that the optimal options between the choice of speed, gear ratio and torque are:

- at 9 kW – $T_M = 689,7$ Nm; $V = 0,034$ m/s; $i = 14,74$.

- at 11 kW – $T_M = 842,92$ Nm; $V = 0,034$ m/s; $i = 14,74$.

Due to the given constraints in the speed – $V = 0,06 - 0,1$ m/s a section of the graph that realizes similar speeds in analyzed. This is possible with gear ratios in the range between $i = 5 - 10$. Exactly 5 values fall into this range and therefore average value is chosen: $i = 7,39$.

The calculations have shown that the 11 kW provides greater starting torque and greater margin under dynamic loads, but results in higher energy consumption and larger dimensions and price.

The 9 kW motor is sufficient with a correctly selected gear ratio, covering the load capacity requirements, providing the required speed and being a more economical solution.

As a result of the optimization analysis, the **9 kW, 1500 rpm** motor option was selected.

$P_M = 9$ kW, $n = 1500$ rpm;

$$T_1 = 57,3 \text{ Nm}; i = 7,39; T_M = 402,27 \text{ Nm}$$

$$V = \frac{n \cdot p}{i \cdot 60} \quad (11)$$

$$V = 0,0677 \text{ m/s}$$

$$t = \frac{H}{V} \quad (12)$$

$$t = 19,2 \text{ s}$$

$$P = \frac{F \cdot V}{\eta} \quad (13)$$

$$P = 978, 14 \text{ W}$$

3.2 Analytical Kinematic Model of the Scissor Lift Mechanism

The angle at the base of the scissor platform (the angle between the arms and the horizontal - α) is one of the most critical parameters determining the power mode of operation of the mechanism. It has a direct impact on the required driving force and torque, and at small values leads to a significant increase in the required force.

In the scissor mechanism, vertical lifting is realized by opening the arms, which form a certain angle to the horizontal. The relationship between the applied force (from the screw or cylinder) and the lifted load is determined by the geometry of the system.

This effect is called force amplification due to the geometry of the mechanism, but in this case it is an "unfavorable amplification", since it requires a greater input force.

The practical consequences are a large load on the motor, gearbox, ball screw and occurs precisely at the beginning of the lifting. The system is sized precisely for this critical case.

The angle in the lower position is a key parameter in the optimization, because:

At a smaller angle:

- more compact design;
- much greater force required;

At a larger angle:

- less load on the drive;
- greater starting height.

A compromise is sought between:

- compactness;
- energy efficiency;
- requirements for the drive.

Optimization of this angle is a key element in the design, as it allows achieving a balance between compactness of the design and energy efficiency, while reducing the requirements for the motor-gearbox group.

The lifting mechanism is analyzed as a single-degree-of-freedom planar mechanism.

The actuator force is derived from the mechanism kinematics using the principle of virtual work, allowing the variation of the mechanical advantage throughout the lifting stroke to be taken into account.

The following parameters are introduced:

L – scissor arm length [m];

α – angle between the scissor arm and the horizontal axis;

H – platform height movement [m];

x – horizontal displacement of the actuator carriage;

W – total external load acting on the platform [N].

The platform height movement is expressed as:

$$H(\alpha) = 2L \cdot \sin \alpha \quad (14)$$

The horizontal position of the moving carriage is:

$$x(\alpha) = 2L \cdot \cos \alpha \quad (15)$$

Differentiating Eqs. (14) and (15) with respect to the scissor angle gives

$$\frac{dH}{d\alpha} = 2L \cdot \cos \alpha \quad (16)$$

$$\frac{dx}{d\alpha} = -2L \cdot \sin \alpha \quad (17)$$

The instantaneous mechanical advantage of the mechanism is:

$$MA(\alpha) = \left| \frac{dH}{dx} \right| = \left| \frac{dH/d\alpha}{dx/d\alpha} \right| = \cot \alpha \quad (18)$$

Equation (18) shows that the transmission ratio is not constant but varies continuously during the lifting motion.

The actuator force is determined using the principle of virtual work.

For an infinitesimal displacement:

$$\delta W = 0 \quad (19)$$

$$F \cdot dx = W \cdot dH \quad (20)$$

Substituting Eq. (18):

$$F(\alpha) = W \cdot \cot \alpha \quad (21)$$

where: $W = m \cdot g$

Equation (21) demonstrates that the required actuator force varies continuously during lifting and depends on the instantaneous scissor angle.

The torque required to drive the ball screw:

$$T(\alpha) = \frac{F(\alpha) \cdot p}{2 \cdot \pi \cdot \eta_{total}} \quad (22)$$

where: p – screw lead;
 η_{total} – total transmission efficiency

The torque required to drive the lifting mechanism depends not only on the actuator force but also on the cumulative efficiency of the mechanical transmission system. The total transmission efficiency is determined as the product of the efficiencies of the individual drivetrain components.

$$\eta_{total} = \eta_g \cdot \eta_c \cdot \eta_b \cdot \eta_s \cdot \eta_f \quad (23)$$

where: η_g – gearbox efficiency – 0.98;
 η_c – coupling efficiency – 0.99;
 η_b – bearing efficiency – 0.99;
 η_s – ball screw efficiency – 0.97;
 η_f – equivalent efficiency accounting for friction losses in the pin joints and linear guides – 0.98;

$$\eta_{total} = 0.91 \quad (24)$$

In practical operation, additional energy losses occur due to friction in the pin joints and linear guide system, particularly at small scissor opening angles where the actuator force reaches its maximum. Rather than modelling each friction interface individually, these losses are incorporated through an equivalent friction efficiency coefficient included in the overall drivetrain efficiency. This approach provides a realistic engineering approximation while maintaining a tractable analytical model.

Substituting Eq. (21) into Eq. (22),

$$T(\alpha) = \frac{W \cdot p \cdot \cot \alpha}{2 \cdot \pi \cdot \eta_{total}} \quad (25)$$

Although the lifting mechanism operates at a relatively low velocity, inertial forces occurring during start-up and stopping introduce additional loading that should be considered during the structural verification. In order to achieve dynamic analysis of the mechanism we should calculate the acceleration during start-up.

$$a = \frac{V}{t_a} \quad (26)$$

where: t_a – acceleration time [m/s²]

For heavy scissor lifts the recommended acceleration time is 2 – 4 s and we choose value – 3 s.

$$a = 0,0223 \text{ m/s}^2$$

$$F_i = m \cdot a \quad (27)$$

where: F_i – inertial force [N];

$$F_i = 31,2 \text{ N}; W = 13\,734 \text{ N}$$

$$K_d = \frac{F_i}{W} \quad (28)$$

where: K_d – dynamic coefficient; $K_d = 1,0023$ (0,23 %)

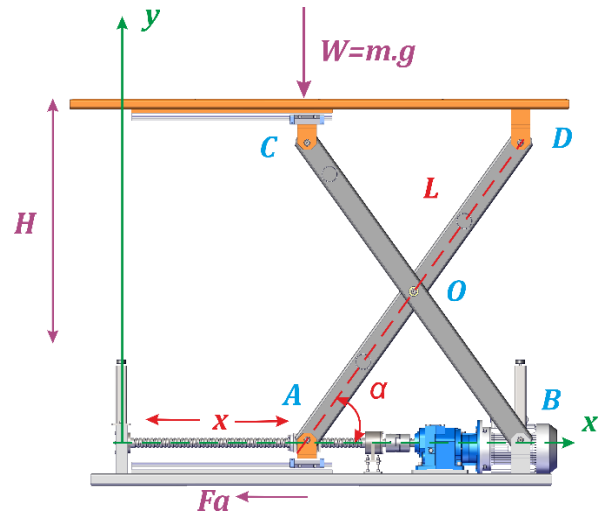


Figure 13: Kinematic model of the proposed scissor lift mechanism

Based on the specified lifting speed of 0.067 m/s and an acceleration time of 3 s, the calculated inertial force represents less than 0.3% of the static lifting load. Therefore, the mechanism operates under quasi-static conditions, and the static load governs the design calculations. The dynamic analysis confirms that inertia has a negligible influence on the actuator force and drive torque.

Figure 13 highlight the kinematic model of the mechanism.

$$F_d = W + F_i \quad (29)$$

$$T(\alpha) = \frac{F_d \cdot p \cdot \cot \alpha}{2 \cdot \pi \cdot \eta_{total}} \quad (30)$$

$$F_d = 13\,765,2 \text{ N}$$

$$F(\alpha) = F_d \cdot \cot \alpha \quad (31)$$

In this case, we envisage that the design can be constructed with an angle value α at the following degrees: 4°, 6°, 8°, 10°, 12°, 14°, 16°, 18°, 20°.

Table 7. Torque / Angle optimization

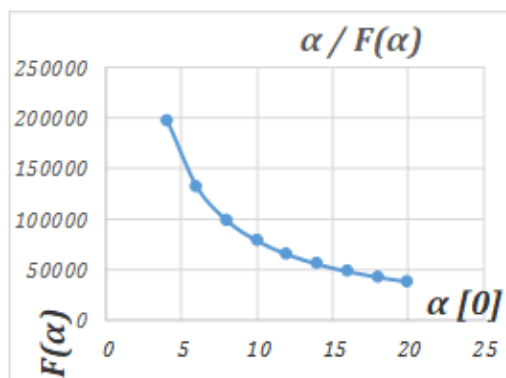
Number	α [°]	$F(\alpha)$ [N]	$T(\alpha)$ [Nm]
1	4	196 853	688,3
2	6	130 955	457,9
3	8	97 952	342,5
4	10	78 065	273
5	12	64 766	226,4
6	14	55 206	193,1
7	16	48 010	167,9
8	18	42 365	148,1
9	20	37 823	132,2

Table 8. Angle / Force decline

Angle change	% F Decline
4° – 6°	≈ 33,5 %
6° – 8°	≈ 25,2 %
8° – 10°	≈ 20,3 %
10° – 12°	≈ 17 %
12° – 14°	≈ 14,76 %
14° – 16°	≈ 13%
16° – 18°	≈ 11,76 %
18° – 20°	≈ 10,7 %

Table 7 shows the different starting angle with the calculated force / torque.

Table 8 demonstrates that increasing the initial scissor angle substantially reduces the actuator force required to initiate lifting. The largest improvement is observed between 4° and 6°, where the actuator force decreases by approximately 33.5%. As the initial angle increases, the percentage reduction gradually diminishes, reaching approximately 10.7% between 18° and 20°. This behaviour confirms the nonlinear relationship between the initial scissor angle and the mechanical advantage of the lifting mechanism, highlighting the importance of selecting an appropriate initial configuration during the design stage. Figure 14 highlights the change in the force according to the angle variation.


Figure 14: Graphics – relation $\alpha / F(\alpha)$

This leads to the conclusion that the scissor platform is most critically loaded and most sensitive to geometric changes at very small angles, where even small variations in the angle lead to large changes in the force.

Therefore:

- The initial lifting phase (small angles) determines the design.
- The correct sizing of the reducer, screw and motor must be based on the value of the angle (respectively the critical force).
- Operation or starting of the system at very small angles should be minimized or carefully controlled due to the high requirement for force and sensitivity.

From the built design of the structure – Figure 2 and 3 we determine that: $\alpha = 8^\circ$ on the designed mechanism.

At $\alpha = 8^\circ$

$$F(\alpha) = 97\,952\text{ N}; T(\alpha) = 342,5\text{ Nm}; T_M = 402,27\text{ Nm}$$

$$k = \frac{T_M}{T(\alpha)} \quad (32)$$

where: k = safety factor; $k = 1,17$

The relatively low operating speed, the high structural stiffness of the welded frame, and the use of a ball screw drive with continuous mechanical engagement substantially reduce the possibility of dynamic excitation.

Furthermore, the presence of bronze guide bushings in the pin joints and rolling-element bearings at the rotating interfaces minimizes friction-induced stick-slip behaviour and suppresses local vibration generation during the lifting cycle.

Considering these operating conditions, no significant resonance or vibrations are expected within the normal working range of the mechanism. Therefore, the structural design may be evaluated using a quasi-static loading approach, while the finite element verification performed under the maximum loading condition provides a conservative assessment of the structural integrity of the system.

3.3 Ball Screw Design Verification

From the calculations made earlier, the following input data are collected when calculating the screw: Ball – screw: $d = 50\text{ mm}$, pitch $p = 20\text{ mm}$; $d_{\text{root}} = 40\text{ mm} = 0,04\text{ m}$;

$$\sigma = \frac{F(\alpha)}{A} \quad (33)$$

$$A = \frac{\pi \cdot d_{\text{root}}^2}{4} \quad (34)$$

$A = 0.001256 \text{ m}^2$; $\sigma F = 77,95 \text{ Mpa}$
 Material - 42CrMo4;
 Yield strength - $Re = 650 \text{ MPa}$;
 S – safety factor; **$S = 8,34$**

$$S = \frac{Re}{\sigma F} \quad (35)$$

In addition to the stresses, we need to check the screw for buckling – (36).

$$F_B = \frac{\pi^2 \cdot E \cdot I}{(K \cdot L)^2} \quad (36)$$

where:

F_B – Critical buckling force [N];
 E - Young's modulus of the material (hardness); $E = 210 \text{ Gpa}$;
 L – Length of the working part of the screw; $L/2 = 615 \text{ mm} = 0,615 \text{ m}$;
 K – Effective column length factor; $K = 0,7$. (with this type of bearing connection);
 I – Moment of inertia [m^4];

$$I = \frac{\pi \cdot d^4}{64} \quad (37)$$

$$F_B = 689 \text{ kN}; F(\alpha) = 97,95 \text{ kN};$$

$$S_B = \frac{F_B}{F(\alpha)} \quad (38)$$

where: S_B – safety factor: **$S_B = 7$**

The effective length factor $k=0.7$ was chosen based on the actual support arrangement of the ball screw. The drive end of the screw is supported by a matched pair of tapered roller bearings (30308T72/DB), providing both radial and axial constraint, while the opposite end is supported by a single-row cylindrical roller bearing (NU 2308 ECML), which provides radial support while allowing axial displacement to accommodate thermal expansion.

This bearing arrangement corresponds to a **fixed-supported** boundary condition, for which an effective length factor of $K=0.7$ is commonly recommended in engineering design practice and machine design literature. Consequently, the effective buckling length is determined as, providing a realistic representation of the structural behaviour of the screw under compressive loading.

3.4 Bearing Calculation Check

3.4.1 Matched tapered roller bearings - 30308T72/DB

Permissible dynamic load – $C = 182 \text{ kN}$;
 Permissible static load – $C_0 = 190 \text{ kN}$.

$$F(\alpha) = 97\,952 \text{ N};$$

$$S_0 = \frac{C_0}{F(\alpha)} \quad (39)$$

$$S_0 = 1,94$$

where: S_0 – safety factor;

The actuator force decreases continuously during lifting because the mechanical advantage increases.

$$F_{eq} = \frac{F(\alpha)_8 + F(\alpha)_{10} + F(\alpha)_{12} + F(\alpha)_{14} + F(\alpha)_{16}}{5} \quad (40)$$

The maximum axial load acts only during the initial lifting stage corresponding to the minimum scissor opening angle. As the platform rises, the scissor angle increases, resulting in a significant reduction in the actuator force due to the increasing mechanical advantage of the mechanism.

Therefore, the maximum load was adopted for the static verification of the bearings, whereas the fatigue life assessment was performed using the equivalent operating load calculated over the complete lifting cycle - F_{eq} . This methodology provides a realistic representation of the actual bearing loading conditions and avoids excessive conservatism in the fatigue life estimation. The obtained results confirm that the selected SKF bearing arrangement provides sufficient static load capacity and satisfactory operational durability for the intended lifting application.

In the calculation – Equ. (40) only the first 5 values are chosen, because of the higher actuator force.

$$F_{eq} = 68,8 \text{ kN}$$

$$L_{10} = \left(\frac{C}{F_{eq}}\right)^p \quad (40)$$

where: L_{10} – Bearing life (number of revolutions / hours);

$p = 10/3$;

$$L_{10} = 3,22 \cdot 10^7 \text{ revolutions}$$

$$n_B = \frac{n}{i} \quad (41)$$

where: n_B – screw speed during operation;

$$n_B \approx 200 \text{ rpm}$$

$$L = \frac{L_{10}}{60 \cdot n_B} \quad (42)$$

where: L – working hours;

$$L = 2644 \text{ working hours}$$

3.4.2 Single row cylindrical roller bearings - NU 2308 ECML

Permissible dynamic load – C = 129 kN;
Permissible static load – C₀ = 120 kN.

This bearing is only subjected to radial load, which is much lower than axial load, so inspection is not necessary.

3.5 Shoulder Joints Dislocation

$$F_s = \frac{\pi^2 \cdot E \cdot I}{(K \cdot L)^2} \quad (43)$$

where:

F_s – Critical shoulder dislocation force [N];

$$I = \frac{B \cdot H^3 - b \cdot h^3}{12} \quad (44)$$

where:

I – Moment of inertia in the threatened section [m⁴];

H – External dimension of the profile (length) [m];

h – Internal dimension of the profile (length) [m];

B – External dimension of the profile (width) [m];

b – Internal dimension of the profile (width) [m].

For the arms we use a rectangular profile with dimensions 120x60x6:

H = 120 mm; h = 108 mm; B = 60 mm; b = 48 mm

$$I = 1,16 \cdot 10^{-6} \text{ m}^4; F_s = 600 \text{ 450 N} = 600,45 \text{ kN}$$

$$S_s = \frac{F_s}{F(\alpha)} \quad (45)$$

where: S_s – safety factor: S_s = 6,13

3.6 Pins Calculation

- on shear stress:

$$\tau = \frac{F(\alpha)}{2 \cdot A} \quad (46)$$

$$A = 2 \cdot \frac{\pi \cdot d^2}{4} \quad (47)$$

τ – Shear stress [MPa];

A – Shear area [m²]; (double shear stress)

d – pin diameter [mm]; d = 30 mm;

τ_{FP} – Allowable shear stresses. [MPa], At C45 Steel;

τ_{FP} = 200 MPa;

$$A = 0,0014137 \text{ m}^2; \tau = 69,3 \text{ MPa}$$

$$\tau < \tau_{FP} \quad (48)$$

- bearing stress (contact):

$$\sigma_H = \frac{F(\alpha)}{d \cdot L} \quad (49)$$

σ_H – Contact stress [MPa];

L – Pin length [m]; L = 0,06 m

$$\sigma_H = 54,42 \text{ MPa}$$

τ_{HP} – Allowable contact stress. At C45 Steel; σ_{HP} = 450 MPa;

$$\sigma_H < \sigma_{HP} \quad (50)$$

The pin connections are reinforced by welded steel bushings with an effective bearing length of 65 mm. In addition, a bronze guide bushing is installed inside each steel bushing to reduce friction, minimize wear during repeated lifting cycles, and improve the service life of the joint.

Consequently, the bearing pressure was evaluated using the effective bearing length of the welded bushing rather than the wall thickness of the structural tube. Based on the maximum calculated actuator force, the resulting bearing pressure equals 54,42 MPa, which is well within the allowable contact stress for the selected steel-bronze bearing pair, confirming the suitability of the adopted joint design.

4. FEA Analysis

To complement the analytical calculations and provide structural validation of the proposed design, a finite element analysis (FEA) was performed on the most highly loaded region of the scissor lift mechanism. The objective of the analysis is not to reproduce the global kinematic behaviour of the complete lifting system, but rather to verify the structural integrity of the actuator attachment under the maximum loading condition obtained from the analytical model.

The complete scissor lift assembly consists of multiple components connected through pin joints, bearings, and sliding interfaces. Accurate numerical representation of the entire mechanism would require nonlinear contact definitions, joint clearances, frictional interactions, and large-displacement kinematic analysis. Since the primary contribution of the present work is the development of an analytical design methodology and the optimization of the initial scissor angle, such a comprehensive numerical model is beyond the scope of this study and would not provide additional insight.

Instead, a local finite element submodel is adopted following a widely accepted engineering

validation approach, in which the most critical structural region is isolated and analysed under the maximum service load. This methodology enables accurate evaluation of the stress distribution in the actuator attachment while representing the stiffness contribution of the remaining mechanism through equivalent boundary conditions.

The finite element model comprises the actuator attachment assembly, including the Inside arms of the scissor mechanism with the welded cylindrical bushing, and the corresponding pin connection. The surrounding structural members were represented by fixed boundary conditions applied at the interfaces connecting the analysed region to the remaining mechanism. This approach replaces the global structural stiffness of the complete lifting system while maintaining the correct load-transfer path through the analysed component.

The loading conditions were defined using the maximum actuator force obtained from the analytical calculations presented in Section 3.2. Under the most unfavourable operating configuration, corresponding to the minimum initial scissor angle, the total actuator force reaches 49 kN (in this section of the arm), which was applied as two equal bearing loads of 24.5 kN acting on the actuator connection pins (lower pins). Since this force already represents the equivalent load required to lift the maximum rated platform load, no additional external platform loading was introduced in the numerical model, thereby avoiding double counting of the applied loading.

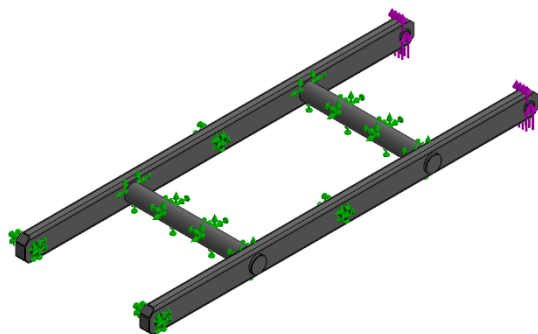


Figure 15: Fixed geometry and force appliance

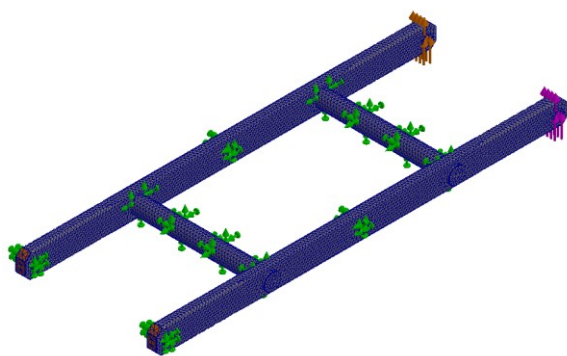


Figure 16: Applied mesh

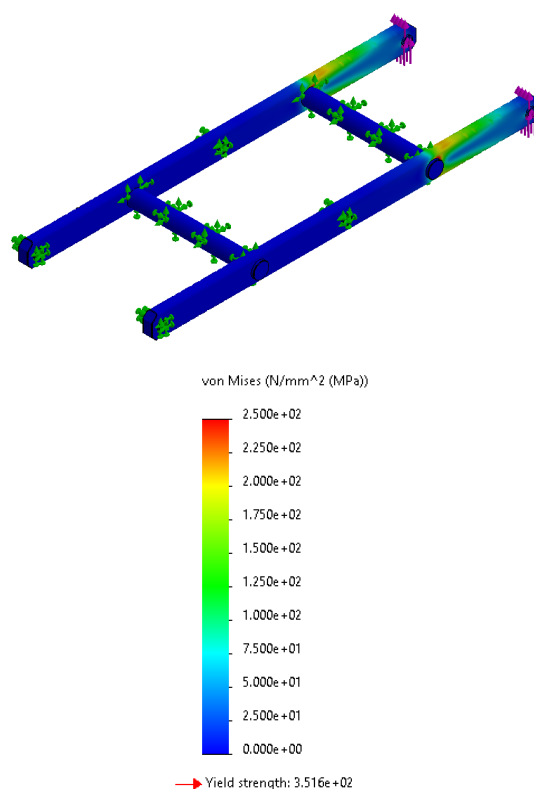


Figure 17: Stress results

Bonded contact conditions were defined between the welded bushing and the arm structure, while the remaining contacts were represented by continuous structural connections – Figure 15. The fixed support on the the two middle tubes is necessary for the correct mesh and model verification when running the analysis. The fixed supports are applied at the middle and top pins, otherwise the model loses position and the results are not adequate.

A refined finite element mesh was generated in the vicinity of the welded bushing and pin connection to accurately capture the stress concentration within the critical load-transfer region – Figure 16.

The numerical results were evaluated in terms of equivalent von Mises stress, total displacement, and factor of safety. Particular attention was given to the welded actuator attachment, which represents the location of the highest internal loading during the lifting process. The obtained results provide structural verification of the proposed design and confirm that the actuator attachment is capable of safely transmitting the maximum analytically predicted actuator force – Figure 17 and Figure 18.

The stress distribution demonstrates that the actuator force is effectively transferred from the welded bushing into the scissor arm through the expected load path. The highest structural stresses associated with the actuator attachment were observed in the vicinity of the welded bushing and remained within the allowable limits of the selected material.

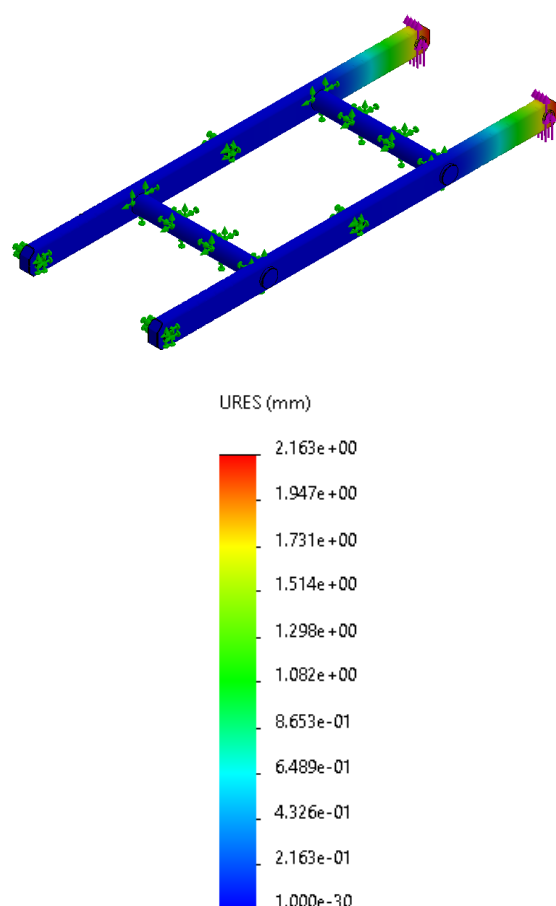


Figure 18: Stress results

Localized stress peaks were identified at the constrained regions of the numerical model, particularly around the intermediate support, as a consequence of the adopted boundary conditions used to represent the stiffness of the remaining mechanism. These localized numerical effects were not considered representative of the actual operating stresses and therefore were not used for structural assessment.

The maximum total displacement of 2.1 mm indicates adequate structural stiffness of the actuator attachment under the maximum analytically predicted actuator force. Consequently, the finite element analysis confirms the structural integrity of the actuator attachment and supports the validity of the proposed analytical design methodology.

5. Conclusions

This study presented an integrated approach for the design, analytical calculation, and initial angle optimization of a high-load scissor lift mechanism, with the primary objective of improving torque efficiency through geometric configuration. By combining systematic mechanical design procedures with a parametric evaluation of the initial scissor angle, a structured framework was developed to

support informed actuator selection and efficient system configuration.

The conducted analysis confirmed that the initial scissor angle is a governing design parameter with a significant influence on actuator torque requirements. Through the proposed parametric approach, the relationship between geometric configuration and torque demand was clearly established, demonstrating that appropriate selection of the initial angle leads to improved mechanical advantage and more efficient force transmission within the mechanism.

The main results of the study can be summarized as follows:

A consistent analytical framework was developed linking geometric parameters with actuator torque requirements in scissor lift systems.

A parametric evaluation strategy was successfully applied to assess the influence of initial angle variations on system performance.

An optimal range of initial angles was identified, enabling improved torque efficiency and more effective actuator sizing.

The integrated design and calculation approach ensured compatibility between theoretical analysis and practical mechanical implementation.

The study demonstrated that early-stage geometric optimization can significantly enhance the overall efficiency of scissor lift systems without requiring major structural modifications.

In relation to the formulated research hypotheses, the findings of the study provide the following confirmations:

H1 is supported, as the results show that the initial angle has a strong and nonlinear influence on actuator torque requirements.

H2 is validated through the identification of an optimal initial angle range that minimizes torque demand while maintaining design feasibility.

H3 is confirmed, demonstrating that optimized configurations achieve improved torque efficiency compared to non-optimized designs.

H4 is supported, indicating that system performance is highly sensitive to variations in the initial geometric configuration, particularly in the lower operating position.

H5 is validated, as the analytical calculation-based approach proved effective in predicting actuator requirements and supporting design-level optimization.

Overall, the proposed methodology contributes to the advancement of design-oriented optimization strategies for scissor lift mechanisms by emphasizing the importance of geometric configuration in determining system efficiency. The results provide practical guidance for engineers in the development of high-performance lifting systems, particularly in applications where minimizing actuator demand and improving energy efficiency are of primary importance.

Future work may extend the presented approach by incorporating dynamic effects, advanced optimization algorithms, FEA and experimental validation to further enhance the robustness and applicability of the methodology.

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