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Development of a 120kg Load Lifting Capacity Scissor Elevator Platform

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Abstract: This work focused on the development of a 120kg load lifting capacity scissor elevator platform (SEP) with a horizontally positioned rack and pinion gear actuating mechanism which is driven by a DC motor. The time of lift to an elevated height of 0.9m is 30s. Simulation of a typical SEP structure in the 3D workspace of a Computer Aided Design (CAD) software package was carried out to investigate the balance of the SEP structure, the stresses experienced, the efficiency, and safety of operations. A prototype was also fabricated for the physical demonstration of SEP. The SEP can be used for a range of engineering applications such as making an adjustable workbench for workshop use, solving the problem of table adjustment for height-challenged personnel, or used as a load-transferring device if mobile to transfer loads between two or more elevated locations during construction or maintenance work. Calculated results give the platform weight as 136.693N, the scissor arms weight as 188.205N, the total structure weight as 1502.098N, the stress in the scissor arm at maximum platform elevation as 1.702MPa, the stress in the scissor arm at minimum platform elevation as 4.928MPa, the maximum actuation force as 4126.980N, and the power required to drive the mechanism as 26.963W. Autodesk Inventor Pro simulation results show that a wide range of data can be sourced when one considers the real-time behavior of SEP. The results also indicated the values of the reaction forces, reaction moments, stresses, strains, and displacements developed at every joint, link, hinged support, and every other point in a 3D workspace.

Index Terms: Scissor elevator platform, Rack and pinion gear, Actuating mechanism, Load lifting capacity, Engineering applications, Adjustable work bench.

1. Introduction

Load lifting has been one of the major concerns in both the construction and maintenance industries. The search for a load lifting device or means of achieving vertical transportation of load, to ease the stress of manual lifting of load has resulted in the invention of the ropes and pulley mechanism, scaffolding, crane lift, and scissors lifts responsible for the lifting of loads and working personnel to the desired height or elevation. The scissor lift as shown in Fig. 1 is one of the various types of aerial work platforms (AWPs) available in the AWP market. A scissor lift is a type of adjustable platform that can only move vertically with a unique but rather simple X-like structural pattern, meant to move the platform upwards or downwards. The upward movement is usually achieved by the application of force to the base sliding supports, with the aid of an actuating mechanism such as a hydraulic or pneumatic pump or mechanical systems such as the lead screw or the rack and pinion gear system. A scissor lift provides the most economical, dependable, and versatile methods of lifting loads; it has few moving parts which may only require lubrication [2,3,4,10]. Conventionally a scissor lift can be used to lift a vehicle to change a tire, to gain access to the underside of the vehicle, to lift the body to an appreciable height, and for many other applications. Such lifts can also be used for various purposes like maintenance and many material handling operations. It can be of mechanical, pneumatic, or hydraulic type [3]. Scissor lift platforms are sometimes mounted on a vehicle; usually described as a vehicle lift platform, which provides mobility and stability during operation, but is limited to vehicle-accessible locations unlike those designed for indoor use for decoration of buildings, electrical or plumbing maintenance, wall painting, glass cleaning, etc. Industrial scissor lifts are used for a wide variety of applications in many industries which include manufacturing, warehousing, schools, grocery distribution, military, hospitals, and printing [2].

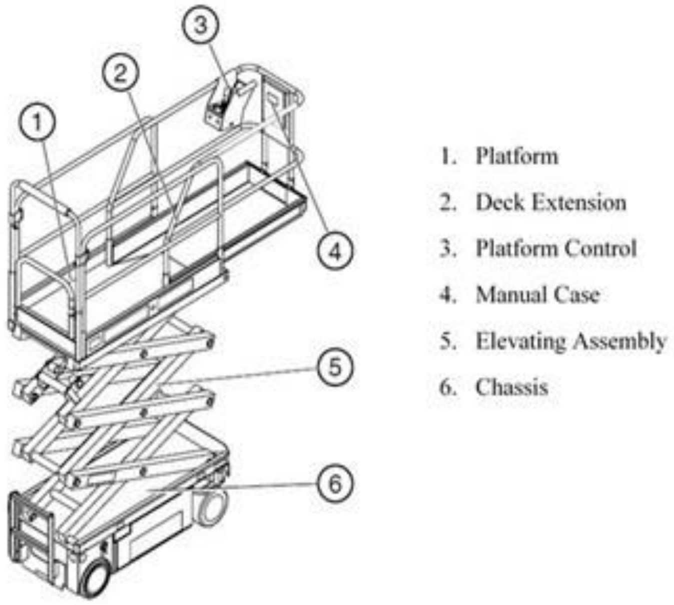

Fig. 1. A Typical scissors lift platform

This particular work focuses on the development of a 120kg load lifting capacity scissor elevator platform (SEP) which is also referred to as an aerial work platform (AWP), or equipment that helps individuals get access to unreachable elevations. The work describes the materials used for this development, the required analysis and calculations, and a prototype fabrication for physical demonstration. The work also considered the simulation of a typical SEP structure with a horizontally positioned rack and pinion gear system actuator driven by a DC motor in the 3D workspace of a computer aided design (CAD) software package. This will help in investigating the balance of the SEP structure, the stresses experienced, the efficiency, and safety of operations.

2. Related Work

Several related works have been carried out on scissors lift platform. Yimer and Wang (2019) worked on the design, analysis and manufacturing of a hydraulic scissor lift having two levels elevated by one hydraulic cylinder. The scissor lifting machine is hydraulically operated which makes lifting simple. The overall objective of the paper was to design and manufacture a double scissors lift device elevated by one hydraulic cylinder that can be used in the automobile sector. The machine was tested by lifting different weights and it was successful and can lift up to the recommended weight of 270kg efficiently without any problem. Drafting and drawing of hydraulic system scissor lift was done using solid works. Dang and Nguyen [9] carried out investigation on the design of double-stage scissor lifts based on parametric dimension technique. The selection of a configuration of lift table system plays an important role since this depends on the working requirements of the systems and also the type of lift object. Based on the parametric dimension technique, the mathematical model of the configuration and the load calculation for the double-stage scissor lifts, which depends on the design parameters, were investigated in order to enhance the operation of scissor lift systems (e.g., lifting height, loading, and stability). A 2D-model of the system was constructed and simulated in the working model software to verify the accuracy of the proposed method. The results obtained from the simulation indicate that by adjusting the mounting positions of cylinders, the elevation of the platform and reactions on joints of the components can be calculated, which assists in improving the performance of the system. Furthermore, the results also prove the practical significance in the calculating process and dimensional design of scissor lifts, especially for double-stage structures.

Momin et al. [3] worked on the design, manufacturing, and analysis of hydraulic scissor lifts. The paper described the design as well as the analysis of a hydraulic scissor lift. The design described in the paper was developed keeping in mind that the lift can be operated by mechanical means by using a pantograph so that the overall cost of the scissor lift is reduced. The lift was designed to be portable and also work without consuming any electric power. This was achieved through the use of a hydraulic hand pump which powers the cylinder. Such a design can make the lift more compact and much more suitable for medium-scale work. The analysis of the scissor lift was done in ANSYS and all parameters were analyzed in order to check the compatibility of the design values. Jeyangel, Babu, and Balasubramani [5], Kumaresan, and Kumaran [6] worked on the design and kinematic analysis of gear-powered scissor lifts. The paper introduced scissor lifts powered by a gear mechanism. The components of gear powered scissor lift are spur gears, scissor arms, a platform, and a one-horsepower electric motor. These components were designed and the mechanism was simulated using the Automated Dynamic Analysis of Multi-Body Dynamics Simulation Solution (ADAMS 2013) software package. A detailed kinematic analysis was performed on the mechanism. For the different length-to-radius ratios, the maximum translatory displacement was measured. For the different speeds of the motor, the velocity of the link was studied. Different lift heights can be achieved by varying the number of links. This particular scissor lift design was expected to carry a load of around 2000 kilograms with a factor of safety equal to 4 and a lifting height of around 0.5 meters.

Raghavendra and Reddy [7] worked on the design and analysis of an aerial scissor lift. The paper described aerial scissor lifts generally as equipment used for temporary, flexible access purposes such as maintenance and construction work or by firefighters for emergency access, etc. which distinguishes them from permanent access equipment such as elevators. Aerial scissor lifts are designed to lift limited weights, usually less than a ton, although some have a higher safe working load (SWL). This is especially true when the work being accessed is raised off the floor and outside an operator's normal ergonomic power zone. In either case, it is much more economical to bring the worker to the work rather than bringing the work to the worker. The need for the use of lifts is very paramount and it runs across labs, workshops, factories, and residential/commercial buildings to repair street lights, fix billboards, electric bulbs, etc. expanded and less efficient, the engineers may run into one or more problems when in use. Considering the need for this kind of mechanism, estimating as well the cost of expanding energy more than the result gotten as well the maintenance, etc. it is better to adopt this design concept to the production of the machine. The initial idea of design considered was the design of a single hydraulic ram for heavy-duty vehicles and putting it underneath, but this has limitations as to the height and stability, and someone will be beneath controlling it. This particular design was done in Pro /E and analyzed in ANSYS. Solmazyiğit et al. [8] worked on the design and prototype production of a scissor lift platform with 25 Tons Capacity. In the study, an innovative 25-ton-capacity scissor lift was designed for the first time, and a prototype was produced. Within the scope of the study, a rolling bearing system was designed instead of the conventional welded scissor-hinge connection system. Static and strength calculations were made using finite element analysis (FEA) for comparison purposes. Based on the results of the analysis, it was determined that the stress distribution on the roller bearing system was more homogeneous and at lower values than the welded hinge system. In addition, a detachable (bolted) joint was obtained instead of a fixed (welded) joint with the designed rolling bearing system. With the design analyses carried out within the scope of the study, a particular joint-clamp system was made to expand the surface areas of the welded joints.

3. Materials and Method

Materials

Rack and Pinion Gear

Rack and pinion gear unlike any other gear type, comprises two gears; the traditional spur gear (round gear) and a straight bar with cog cut popularly known as the rack, is majorly used in machines due to its ability to convert rotary motion to linear and linear to rotary. For this reason, it has been applied in mechanical actuators such as the drilling machine up and down movement, steering rack of automobiles, and many more. Rack and pinion are classified into two types based on their cog cut, the two basic types are:

1. ***Helical Cog Rack and Pinion Gear***: this is linearly shaped to mesh with helical gear and designed to give rise to an infinite radius of rotation during operation.

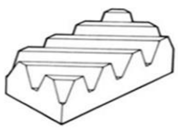

Fig. 2. Teeth profile of a helical rack gear

2. ***Straight or Spur Cog Rack and Pinion Gear***: This is usually a linear-shaped gear meshed with a spur gear at any number of teeth, and also serves as a portion of a spur gear with an infinite radius.

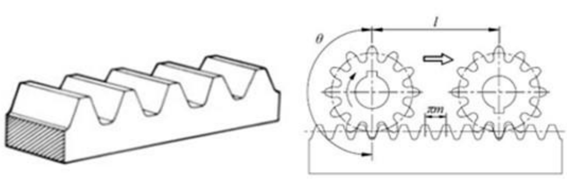

Fig. 3. Teeth profile of a spur rack gear

Displacement of the rack in a rack and pinion arrangement

For a rotation by an angle θ, in the pinion, the displacement d of the rack can be determined by the relation;

$$d = \frac{z\theta}{360} \times \pi m \tag{1}$$

Where,

z = number of teeth on the pinion
m = module $= \frac{25.4}{p}$
p = reference diameter or pitch

***Platform*:** The Platform is the uppermost point of the scissor lift which carries the payload and transfers the load weight and stresses developed to the support links (scissor arms).

***Scissor arms*:** The scissor arms are the major support for the lift platform which bears the load coming from the platform and then transfers it to the leg support at the base. The scissor arms are arranged in tiers to form a rigid but flexible mechanism, capable of lifting or lowering a specific load.

***Support rods*:** The support rods keep the lifting members in place during operation and try to prevent shearing of the structure.

***Base plate*:** The base plate serves as the major support for the whole system where the arms legs are rooted with ball bearings and pinned down with bolts and nuts.

***DC motor*:** To elevate the scissor platform upwards, an external force is required to be exerted on one side of the bottom support of the structure. The actuation was achieved using a DC motor which drives the rack and pinion gear system causing the elevation of the scissor platform.

Design Analysis and Calculations

The Platform

Design of the platform as shown in Fig. 4 is undertaken knowing the intended payload capacity, the length gap at the full stretch of the scissor arms. The platform does have similar requirements as the base plate.

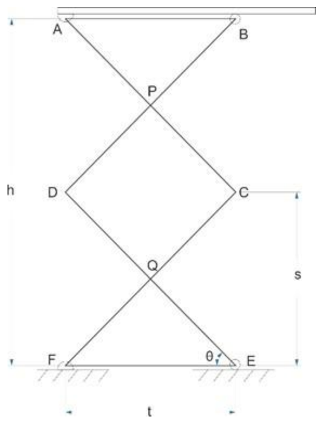

Fig. 4. Scissors lift platform

Considering the platform carrying a static load W, simply supported, and centrally loaded as shown in Fig. 5, the reaction at points A and B is determined as follows:

$$R_A + R_B = W \tag{2}$$

$$R_A = R_B = \frac{W}{2} \tag{3}$$

Shear force (SF) experienced on the platform

For section BC,

$$SF_{BC} = +\frac{W}{2} \tag{4}$$

For section AC,

$$SF_{AC} = \left(\frac{W}{2} - W\right) = -\frac{W}{2} \tag{5}$$

Bending moment (BM) experienced on the platform

BM at point B,

$$BM_B = 0$$

BM at point C,

$$BM_C = \frac{W}{2} \times \frac{L}{2} = \frac{WL}{4} \tag{6}$$

BM at point A, $BM_A = 0$
If the weight of the platform material W_{m1} and the payload W_P is considered, then,

$$R_A = R_B = \frac{W_{mi}+W_P}{2} \tag{7}$$

$$SF_{BC} = +\frac{W_{mi}+W_P}{2} \tag{8}$$

$$SF_{AC} = -\frac{W_{mi}+W_P}{2} \tag{9}$$

$$BM_C = \frac{(W_{mi}+W_P)L}{4} \tag{10}$$

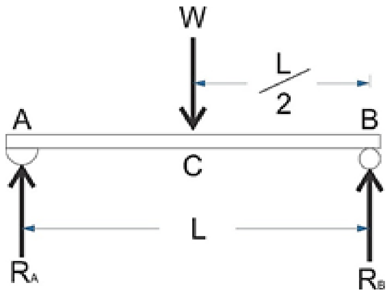

Fig. 5. The platform carrying a static load

The platform at dynamic loading, considering a critical point C as shown in Fig. 6, Reactions at point A and B is determined as follows

$$R_B \times x = W \times L \tag{11}$$

$$R_B = \frac{WL}{x} \tag{12}$$

$$R_A + R_B = W$$

$$R_A = W - R_B \tag{13}$$

$$R_A = W - \frac{WL}{x} \tag{14}$$

$$R_A = -\frac{Wy}{x} \tag{15}$$

SF experienced on the platform

For section BC,

$$SF_{BC} = -W \tag{16}$$

For section AB,

$$SF_{AB} = \left(\frac{WL}{x} - W\right) = +\frac{Wy}{x} \tag{17}$$

BM experienced on the platform

BM at point C,

$$BM_C = 0$$

BM at point B,

$$BM_B = W \times y \tag{18}$$

BM at point A,

$$BM_A = 0$$

If the weight of the platform material W_{m1} and the payload W_p is considered, then,

$$R_B = \frac{(W_{mi}+W_P)L}{x} \tag{19}$$

$$R_A = -\frac{(W_{mi}+W_P)y}{x} \tag{20}$$

$$SF_{BC} = W_{mi} + W_P \tag{21}$$

$$SF_{AB} = -\frac{(W_{mi}+W_P)y}{x} \tag{22}$$

$$BM_B = (W_{mi} + W_P)y \tag{23}$$

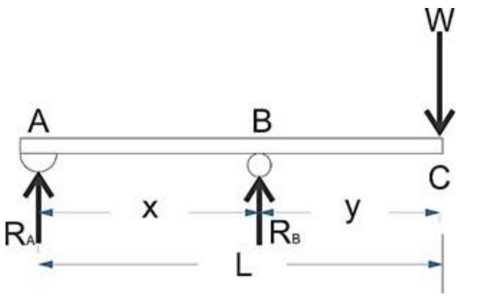

Fig. 6. The platform carrying a dynamic load

The Scissor Arms

To determine the forces acting within the arms (members), we consider Fig. 7.

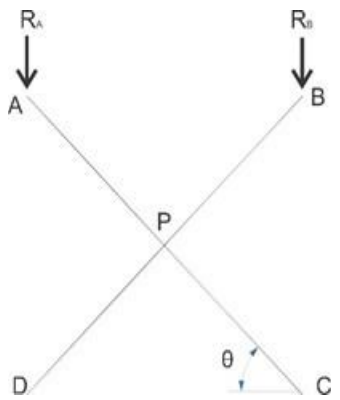

Fig. 7. The scissor arms

Considering the arm length APC as shown in Fig. 7,

$$F_{AC} = \frac{R_A}{sin\theta} = \frac{W}{2 \times sin\theta} \tag{24}$$

$$R_X = F_{AC} cos\theta = \frac{W}{2 \times tan\theta} \tag{25}$$

Where,
R_X = the horizontal reaction force at the scissor base support
Considering the arm length BPD as shown in Fig. 7,

$$F_{BD} = \frac{R_B}{sin\theta} = \frac{W}{2 \times sin\theta} \tag{26}$$

$$F_X = F_{BD} cos\theta = \frac{W}{2 \times tan\theta} \tag{27}$$

Where,
F_X = the horizontal actuating force at the scissor base support
If the weight of the platform material W_{m1}, the scissor arms material W_{m2}, and the payload W_p is considered, then,

$$R_X = \frac{W_{m1} + W_{m2} + W_p}{2 \times tan\theta} = F_X \tag{28}$$

Where,

$$W_p = 120kg \times 9.81m/s^2 = 1177.200N$$

The mass of the platform material M_{m1} is determined as follows:

$$M_{m1} = \rho_m \times V_{m1} \tag{29}$$

Where,
ρ_m = density of platform material = 7850kg/m^3
V_{m1} = volume of platform material

$$V_{m1} = 2A_1 \times (L_1 + w) + A_2 \times L_1 \tag{30}$$

Where,
A_1 = platform frame cross-section area = $3.048 \times 10^{-4} m^2$
A_2 = platform plate cross-section area = $1.840 \times 10^{-3} m^2$
w = platform width = 0.460m
L_1 = platform length = 0.610m
$V_{m1} = 6.096 \times 10^{-4} \times (0.610 + 0.460) + 1.840 \times 10^{-3} \times 0.610 = 1.775 \times 10^{-3} m^3$
$M_{m1} = 7850 \times 1.775 \times 10^{-3} = 13.934kg$

The weight of the platform material W_{m1} is therefore obtained as

$$W_{m1} = M_{m1} \times g = 13.934 \times 9.81 = 136.693N \tag{31}$$

The Length L_2 of the scissor arm and the Maximum angle of inclination θ of the arm from the horizontal at maximum platform elevation is determined as follows:

$$s = \frac{H}{n} \tag{32}$$

Where,
H = maximum height of the lift platform = 0.9m
n = Number of scissor arm tiers = 2
$s = \frac{0.9}{2} = 0.45m$
Considering triangle ACD as shown in Fig, 7, the length L_2 of one scissor arm member is obtained as

$$L_2 = \sqrt{0.450^2 + 0.330^2} = 0.558m$$

The maximum angle of inclination θ of the arm from the horizontal at maximum platform elevation is obtained as

$$\theta = sin^{-1}\frac{s}{L_2} = sin^{-1}\frac{0.450}{0.558} = 53.75^0 \tag{33}$$

Height of base from ground = 150mm
At maximum platform elevation, the distance between two scissors feet = 0.328m
At minimum platform elevation, the distance between two scissors feet = 0.524m
Distance moved by sliding foot to achieve maximum platform elevation d = 0.196m
The weight of scissor arm members W_{m2} is obtained as follows:

$$A_3 = b^2 - (b-2t)^2 = 5.456 \times 10^{-4} m^2 \tag{34}$$

$$V_{m2} = A_3 \times L_2 \times 8 = 5.456 \times 10^{-4} \times 0.56 \times 8 = 2.444 \times 10^{-3} m^3 \tag{35}$$

$$M_{m2} = \rho_m \times V_{m2} = 7850 \times 2.444 \times 10^{-3} = 19.185kg \tag{36}$$

$$W_{m2} = M_{m2} \times g = 19.185 \times 9.81 = 188.205N \tag{37}$$

Where,
A_3 = area of cross-section of the scissor arm member
V_{m2} = volume of the scissor arm members
M_{m2} = mass of the scissor arm members
The total weight of the SEP structure W_{ST} is obtained as

$$W_{ST} = W_{m1} + W_{m2} + W_P = 136.693 + 188.205 + 1177.200 = 1502.098N \tag{38}$$

Stress on each support arms

The force within the scissor arm at maximum platform elevation is obtained as

$$F_{AC} = \frac{R_A}{sin\theta} = \frac{751.049}{sin54} = 928.348N$$

Stress on each support arm at maximum platform elevation is obtained as

$$\sigma_{AC} = \frac{F_{AC}}{A_3} = \frac{928.348}{5.456\times10^{-4}} = 1.702 \times 10^6 N/m^2 \tag{39}$$

The force within the scissor arm at minimum platform elevation is obtained as

$$F_{AC} = \frac{R_A}{sin\theta} = \frac{751.049}{sin20} = 2195.920N$$

The stress on each support arm at minimum platform elevation is obtained as

$$\sigma_{AC} = \frac{F_{AC}}{A_3} = \frac{2195.920}{5.456\times10^{-4}} = 4.928 \times 10^6 N/m^2 \tag{40}$$

The overall force required at maximum platform elevation is obtained as

$$F_X = \left(\frac{W_S}{2 \times tan\theta}\right) \times n = \left(\frac{1502.098}{2 \times tan54°}\right) \times 2 = 1091.338N$$

The overall force required at minimum platform elevation is obtained as

$$F_X = \left(\frac{W_S}{2 \times tan\theta}\right) \times n = \left(\frac{1502.098}{2 \times tan20°}\right) \times 2 = 4126.980N$$

Rack and Pinion Gear Design

The following considerations were made for the design of the actuating rack and pinion gear system:

1. ***The Need*:** the gear configuration is meant to perform a horizontal push of one side of the bottom support, therefore causing an upward stretch of the scissor lift. The transfer of load from the ground level to the

maximum attainable height of the scissor lift is intended to be convenient and efficient, without failure during operation.

2. ***Availability*:** most of the time rack and pinion gears are readily available in the market in a wide range of specifications that will suit different purposes and individual projects. For the purpose of this work, a well-fitted rack and pinion gear which can easily be applied was bought.
3. ***Safety and Maintenance*:** the need for interval maintenance and safe operation is a major concern after development. Wear and tear of the scissor elevator parts are not negligible due to the kind of operation the elevator performs and the material it is made up of. Therefore, an access opening for the gear configuration must be created in the base portion of the scissor elevator, in order to give way for constant lubrication and routine check.

The following mathematical relationships are expected to serve as a guide in the gear selection process. The design procedure for a spur gear as a pinion in the rack and pinion gear arrangement is as follows [1]:

Tangential tooth load W_T

$$W_T = \frac{P}{v} \times C_S \tag{41}$$

Where,
P = power transmitted (Watts)

$$\text{v = pitch line velocity (m/s)} = \frac{\pi DN}{60} \tag{42}$$

D = pitch circle diameter (m) = mT
m = module (m)
T = number of teeth
N = speed (rpm)
Cs = service factor as given in Table 1

Table 1. Service factor

Type of load	Type of Service		
	Intermittent 3 hours per day	8-10 hours per day	Continuous 24 hours per day
Steady	0.8	1.00	1.25
Light stock	1.0	1.25	1.54
Medium stock	1.25	1.54	1.80
Heavy Stock	1.54	1.80	2.00

Lewis equation

$$W_T = \sigma_W \times b \times P_c \times y = \sigma_W \times b \times \pi m \times y = (\sigma_\circ \times C_v) b \times \pi m \times y \tag{43}$$

The Lewis equation is applied only to the weakest of the gears. If the pinion and the driven gear are made of the same material, the weak gear is considered to be the pinion but if the pinion is made of a different material; then the deciding factor is either the product of $(\sigma_w y)$ or $(\sigma_o y)$. Therefore, the equation is then used on the wheel where $(\sigma_w y)$ or $(\sigma_o y)$ is less. The product of $(\sigma_w y)$ is called the strength factor of the gear. And the face width (b) may be taken as 3pc to 4pc (9.5m to 12.5m) for cut teeth and 2pc to 3pc (6.5m to 9.5m) for cast teeth.

The dynamic load W_D

Using Buckingham's equation,

$$W_D = W_T + W_I \tag{44}$$

$$W_D = \frac{21v(b.C + W_T)}{21v + \sqrt{b.C + W_T}} \tag{45}$$

The tangential tooth load (W_T) can be determined if service factor is neglected, using this relation;

$$W_T = \frac{P}{v} \tag{46}$$

The Static Tooth load W_S

This is referred to as the beam strength or endurance strength of the tooth and may be obtained using the equation

$$W_S = \sigma_e.b.p_c.y = \sigma_e.b.\pi m.y \tag{47}$$

For safety against breakage, W_S should be greater than W_D

Wear Tooth Load W_W

This can be determined using this relation

$$W_W = D_P \times b \times Q \times K \tag{48}$$

Where wear load W_w should not be less than the dynamic load W_D
Considering the linear motion of a rack and pinion gear
Tangential Acceleration force

$$F_r = mg\mu + ma \tag{49}$$

Where,
m = mass (kg)
μ = coefficient of friction.

$$\text{Torque } T = \frac{9950.p}{n} = F \times R \tag{50}$$

Where,
F = force
R = radius of the pinion

Module of the rack

The Module of the rack to achieve the maximum elevation of the scissors lift at $d = 0.190m$ and $z = 12$ is obtained as

$$m = \frac{d}{\pi z} = \frac{0.196}{\pi \times 12} = 5.199 \times 10^{-3} m$$

Required Power

The power P required to actuate or elevate the scissor platform is obtained as

$$P = \frac{WD}{t} = \frac{F_X \times d}{t} = \frac{4126.980 \times 0.196}{30} = 26.963W$$

Where,
WD = work done
t = rise time

Simulation

The design simulation was carried out using Autodesk AutoCAD, Autodesk Inventor. The AutoCAD software is used for the 2D drafting while Autodesk Inventor Pro is used for the 3D modeling and simulation. Each follows a series of steps or procedures to achieve the desired result.

Tools used for the computer simulation

Input Devices

1. Keyboard
2. Optical Mouse
3. Hp Laptop Duo Core Processor 2.0GHz, 8GB RAM.

Output Devices

1. Monitor display (Laptop Screen0
2. Printer (Plotter)

2D Drafting Using AutoCAD

AutoCAD drafting was accomplished using the drawing, modifying, and annotation tools of the software to perform a structural draft of the scissor lift platform, showing the top, side, and hidden view and component positions of the mechanism.

3D Modeling Using Autodesk Inventor Pro

The 2D drafts and dimensions are transferred (Imported) into the Autodesk Inventor Pro Interface by a DWG file format, to perform 3D solid modeling activity on the draft drawn in AutoCAD. The first of the modeling process is the part modeling; this aspect involves importing a 2D draft of the scissor elevator platform from AutoCAD into the inventor part model interface, then 3D face extrusion process began aided by a tool called EXTRUDE, with other tools such as SWEEP, MOVE, TRIM, FILLET, CHAMFER, etc. which requires selecting a face or group of faces or edges of the 2D draft to form the desired 3D shaped object. The part modeling was carried out using solid geometric modeling command; one of three possible types of geometric model useable in the 3D workspace (geometric models such as surface, wireframe, and solid modeling geometrics). All part models such as the scissor arms (links), the top platform, the base plate, the hinged ball bearing, pins, support rods, roller guide, bolts and nuts, etc. were all coupled (assembled) together in the assembly model interface to form the scissor elevator with fitting dimensions of various parts of the scissor elevator mechanism. This process involves placing the object, collecting them from their various files using the PLACE command, then trying to position each object in their right location, using command tools such as JOINT, CONSTRAINT, ROTATE, etc. to complete the assembly task.

3D Model Simulation Using Autodesk Inventor Pro

A simulation of the structural stresses was carried out and analyzed using the stress analysis tool. The following steps were taken to complete the simulation task:

1. Material assignment: Applying the type of material a solid primitive object is, in the simulation process MILD STEEL, WELDED is used. This enables the software to determine the physical and mechanical properties of the object or geometry
2. Setup simulation setting
3. Assign Constraints: This process involves joint allocation (FIXED, PIN, FRICTIONLESS)
4. Application of load (1177.200N) on the platform relative to the mechanism's structure.
5. The simulation is run (RUN) for stress analysis.
6. After, the result is obtained and saved as a Word document or HTML data to be opened in an internet browser application.

4. Results and Discussion

This section contains the summary of the result obtained from calculations and that obtained from the simulation performed on Autodesk Inventor.

Results

With the results obtained, a simulation of the design parameter is carried out in Autodesk Inventor Pro. Using its FEA (Finite Element Analysis) capabilities to check for stress concentration, deformation, and analysis on the frame structure by simulating the 3D model designed within the 3D workspace. Fig. 8 gives the image showing the area of displacement during the stress analysis simulation of SEP. Fig. 9 gives the photograph showing the assembly of the base support, scissor arrangement, and the platform for the prototype of SEP. Table 2 gives the summary of calculation results, Table 3, 4, and 5 gives the operation conditions for 120kg loading, the reaction force and moment on constraints, and the summary of simulation results respectively.

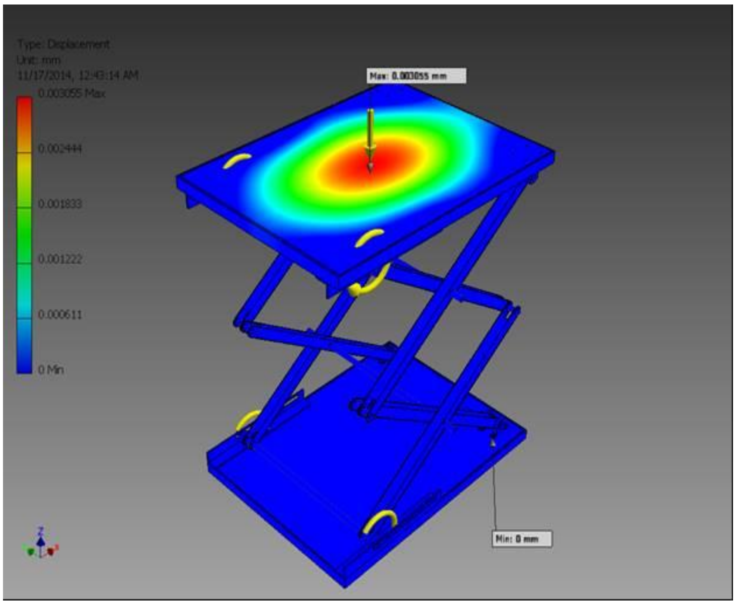

Fig. 8. Image showing the area of displacement during the stress analysis simulation

Fig. 9. Photograph showing the assembly of the base support, scissor arrangement, and the platform.

Table 2. Summary of calculation results

QUANTITY	VALUE	UNIT
Weight of the platform materials	136.693	N
length of one scissor arm member	0.558	m
Weight of the scissor arm members	188.205	N
Weight of the SEP structure	1502.098	N
Force required at maximum platform elevation	1091.338	N
Force required at minimum platform elevation	4126.980	N
Stress on each support arms at maximum platform elevation	1.702×10^6	N/m^2
Stress on each support arms at minimum platform elevation	4.928×10^6	N/m^2
Distance between two scissors feet at maximum platform elevation,	0.328	m
Distance between two scissors feet at minimum platform elevation,	0.524	m
Diameter of center pin	0.010	m
Power required	26.963	W

Table 3. Operation conditions for 120kg loading

Load Type	Force
Magnitude	1177.200 N
Vector X	-11.201 N
Vector Y	0.000 N
Vector Z	-1177.147 N

Table 4. Reaction force and moment on constraints

Constraint	Reaction Force		Reaction Moment	
	Magnitude (N)	Component (X,Y,Z)	Magnitude (Nm)	Component (X,Y,Z)
Pin Constraint:1	0.120396	0.1193 N	0.00301426	-0.00119469 Nm
		0.00536029 N		0 Nm
		-0.0153014 N		-0.00276739 Nm
Pin Constraint:2	0.253975	0.139368 N	0.00119007	0.00111208 Nm
		-0.0305065 N		0 Nm
		0.210118 N		-0.000423732 Nm
Pin Constraint:3	0.0033217	0.00316869 N	0.0000228732	0.0000147353 Nm
		-0.000222746 N		-0.000000313161 Nm
		-0.000971327 N		0.0000174915 Nm
Pin Constraint:4	0.00511068	0.00494489 N	0.0000178677	-0.0000114882 Nm
		0.000210767 N		0.00000095469 Nm
		-0.00127386 N		-0.0000136515 Nm
Pin Constraint:5	45.9573	45.9546 N	0.25576	0.144598 Nm
		0 N		0 Nm
		-0.495711 N		0.210962 Nm
Fixed Constraint:1	4.02264	4.00533 N	0.0976292	0.00538978 Nm
		0.0358413 N		0.0900705 Nm
		0.371096 N		-0.0372789 Nm
Pin Constraint:6	0.0084185	0.00811358 N	0.00000506964	0.00000324718 Nm
		-0.0000119789 N		0.000000641529 Nm
		-0.00224519 N		0.00000383999 Nm
Pin Constraint:7	0.169996	0.124861 N	0.039241	-0.0390667 Nm
		0.0253389 N		0 Nm
		-0.112545 N		0.00369478 Nm
Pin Constraint:8	45.9573	45.9546 N	0.25576	0.144598 Nm
		0 N		0 Nm
		-0.495711 N		0.210962 Nm
Pin Constraint:9	0.40066	-0.195403 N	0.0823858	-0.0729387 Nm
		0.0553657 N		0 Nm
		0.345371 N		0.038306 Nm
Pin Constraint:10	0.0412333	0.00529637 N	0.00344154	-0.00342806 Nm
		-0.00193295 N		0 Nm
		0.0408461 N		-0.000304229 Nm
Pin Constraint:11	0.0359747	0.0012678 N	0.00302457	0.000549029 Nm
		0 N		0 Nm
		-0.0359523 N		0.00297432 Nm
Pin Constraint:12	234.259	-41.3214 N	2.99631	2.99222 Nm
		69.7921 N		0 Nm
		219.77 N		0.156668 Nm
Pin Constraint:13	220.766	-40.0022 N	3.19674	-3.19609 Nm
		-55.8733 N		0 Nm
		209.799 N		-0.064185 Nm
Pin Constraint:14	51.529	-14.3064 N	4.75103	3.7893 Nm
		-13.9705 N		0 Nm
		47.491 N		2.86592 Nm
Frictionless Constraint:1	4.05602	4.04244 N	0.0918431	0.00981215 Nm
		0.0408576 N		0.0834794 Nm
		0.329112 N		-0.0370143 Nm
Frictionless Constraint:2	0.746787	0 N	0.0583604	0.0000448161 Nm
		0 N		0.0583604 Nm
		0.746787 N		0 Nm
Frictionless Constraint:3	0.585351	0 N	0.0496512	0.0019752 Nm
		0 N		-0.0496119 Nm
		-0.585351 N		0 Nm
Frictionless Constraint:4	699.842	6.71532 N	5.78875	-2.02031 Nm
		0 N		-5.42475 Nm
		699.81 N		0 Nm

Table 5. Summary of simulation results

Name	Minimum	Maximum
Volume	13077700 mm^3	
Mass	102.791 kg	
Von Mises Stress	0.000000000347287 MPa	2.55252 MPa
1st Principal Stress	-0.423515 MPa	2.84413 MPa
3rd Principal Stress	-2.92682 MPa	0.696442 MPa
Displacement	0 mm	0.00305526 mm
Safety Factor	15 ul	15 ul
Stress XX	-1.33854 MPa	1.13103 MPa
Stress XY	-0.549446 MPa	0.563511 MPa
Stress XZ	-0.809855 MPa	0.774491 MPa
Stress YY	-0.956788 MPa	2.82032 MPa
Stress YZ	-1.36263 MPa	0.679854 MPa
Stress ZZ	-2.2049 MPa	0.869284 MPa
X Displacement	-0.000176193 mm	0.000257688 mm
Y Displacement	-0.000562835 mm	0.000516029 mm
Z Displacement	-0.00305434 mm	0.000212248 mm
Equivalent Strain	0.0000000000000186401 ul	0.0000102923 ul
1st Principal Strain	-0.000000056726 ul	0.0000111038 ul
3rd Principal Strain	-0.0000119909 ul	0.000000121094 ul
Strain XX	-0.00000488123 ul	0.00000425203 ul
Strain XY	-0.00000318429 ul	0.0000032658 ul
Strain XZ	-0.00000469348 ul	0.00000448853 ul
Strain YY	-0.00000327861 ul	0.0000110904 ul
Strain YZ	-0.00000789708 ul	0.00000394006 ul
Strain ZZ	-0.0000079303 ul	0.00000360576 ul
Contact Pressure	0 MPa	11.9026 MPa
Contact Pressure X	-1.04797 MPa	2.20316 MPa
Contact Pressure Y	-1.47309 MPa	1.81716 MPa
Contact Pressure Z	-6.02644 MPa	11.7079

Discussion

From the calculation results, the starting angle of the scissor lift is paramount to the amount of force required to get the mechanism moving from its static position; the lower the angle of inclination at the static position, the more the force required to move the mechanism. The result indicated that the actual force required to move the SEP increases with a decrease in platform elevation. The result as tabulated in Table 2 also shows that the force required to move the SEP from its static position at the scissor arm inclination angle of 20 °to the horizontal support plane is 4126.980N, while that required to get the SEP up to its maximum elevation of 0.9m at the scissor arm inclination angle of 54 °to the horizontal support plane is 1091.338N. The system is designed for a load lifting capacity of 120kg (1177.200N). Calculated results as summarized in Table 2 gives the platform weight as 136.693N, the scissor arms weight as 188.205N, the total structure weight as 1502.098N, the stress in the scissor arm at maximum platform elevation as 1.702MPa, the stress in the scissor arm at minimum platform elevation as 4.928MPa, the maximum actuation force as 4126.980N, and the power required to drive the mechanism as 26.963W. These values are within acceptable limits.

The results obtained in the Autodesk Inventor Pro simulation as tabulated in Tables 4 and 5 shows that a wide range of data can be sourced when one considers the real-time behavior of the SEP. The tabulated results also indicated the values of the reaction forces, reaction moments, stresses, strains, and the displacements developed at every joint, link, hinged support, and every other point in a 3D plane (X,Y,Z coordinate).

5. Conclusion

The work focused on the development of a scissor elevator platform (SEP) with a horizontally positioned rack and pinion gear actuating mechanism. The SEP was designed to carry a load of 120kg (1177.200N) to a maximum platform elevation of 0.9m in 30s. After simulation in a 3D workspace, a proto-type was fabricated for a physical demonstration. The SEP is workable for a range of engineering purposes such as making a collapsible workbench for workshop use, and many more, solving the problem of table adjustment for height-challenged personnel or used as a load-transferring device if mobile. Autodesk Inventor Pro simulation results show that a wide range of data can be sourced when one considers the real-time behavior of the SEP. The results also indicated the values of the reaction forces, reaction moments, stresses, strains, and displacements developed at every joint, link, hinged support, and every other point in a 3D workspace. Calculated results give the platform weight as 136.693N, the scissor arms weight as 188.205N, the total structure weight as 1502.098N, the stress in the scissor arm at maximum platform elevation as 1.702MPa, the stress in the scissor arm at minimum platform elevation as 4.928MPa, the maximum actuation force as 4126.980N, and the power required to drive the mechanism as 26.963W.

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