Analysis and Design Optimization of Multi Arm Lift

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Abstract: Lift mechanisms are very much required in the production industries. To move the components from one place to other place or to lower or lift the equipment, the lifts are required. Since its main functioning is movement, they are mainly made of truss members. The trusses are pin jointed and so the loads are converted to axial loads. So the main loading is axial load in lift mechanisms. Due to length of the link members, main failure mode is buckling and so the design should be mainly based on buckling. Usage of certain basic principles of mechanics helps in better design of structures with lesser weight, lesser maintenance which helps in survival in the ever competing engineering industry.
Keywords: Moment of inertia, buckling, T-section, L-section, channel section

I. INTRODUCTION

A scissor lift is made of foldable mechanism in ‘x’ form. The name comes from the shape of ‘x’ or cross form of members for the mechanism. The force of drive may be hydraulic, manual, pneumatic or electrical. Even mechanical drives in the form of rack and pinion, belt drives also can be used. But most frequently the driving force is hydraulic due to its reliability and strength. In this mechanism, all the links has same length.

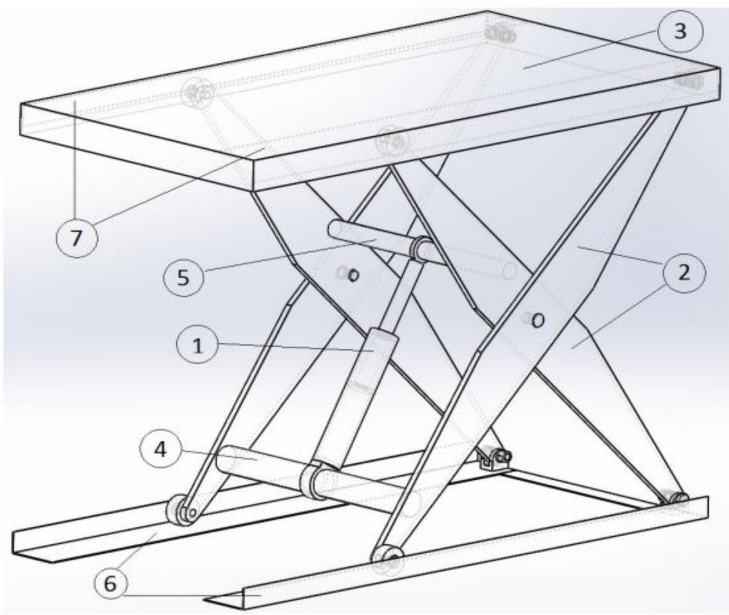

Fig 1: major parts of scissor lift

1) Hydraulic Cylinder
2) Leg
3) Table top
4) Supporting tube 1
5) Supporting tine 2
6) Base Plates
7) Top Plates

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A. Main drawbacks and problems with Scissor Lifts

Table 1: Main faults and Methods of Elimination

Type	Possible Reasons
Lift does not work	Cylinder Malfunction
Leaks in the joints of hydraulic tubes	Loose connections
The stock does not create sufficient pressure	Maladjustment of safety valve
Leakage under cylinder cover	Loosening of bolts
Insufficient pressure by pump	Malfunction of the pump

II. DESIGN SPECIFICATIONS

Maximum extension of the lift: 4000mm
Total number of tiers: 4
Each scissor extension =1000mm
Length of base =1600mm
Width of the base=910mm
Height of the base from the ground =500mm
Angle at height extension = 110degrees
At maximum extension distance between legs of scissors=800mm
Distance moved by sliding to the full extension =350mm
Platform specifications:
Total length of platform=1500mm
Total width of platform=800mm

A. Arm section

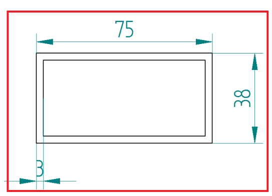

Fig 2: arm section

Cross section of the scissor arm = 75*38-72*35=330mm^2.
Length of the link considered L=1300
Volume of the link VL=1300*330=429000mm^3.
Weight of each link WL=429000*7800e-9=3.3462kgf or 33.46N.
Total number of links =16
Total weight =33.46*16=~536N.

B. Bush Dimensions:

Inner diameter of bush d1=25mm
Outer diameter of bush d2=38mm
Length of bush L=55mm
Volume bush V=35378mm^3.
Weight of each bush =33378*7800e-9*9.81=2.75N
Total number of bushes =48
Weight of all bushes: 48*2.75=132.45N.
Mass of each pin =Volume * Density
Pin length =100mm

Volume $=\Pi Ld^2/4=3.14*100*25^2/4$
Mass of each pin =3.82N
Weigh of cotter pins=0.15N
Mass of 24 cotter pins=3.6N
Weight of each shaft for bearings53.6N
Weight of each pipe used to connect the scissors=22.05N
Weight of two bearings =5.8N
Weight of double acting cylinder=300N
Weight of brackets=75N
Weight of mounting plates=42N
Weight of the platform =350N
Let two persons of 75 kg each along with accessories of equal weight
Total load on the platform is 75*4=300kg or 3000N
Considering a load factor of 1.5
Total load on the platform =4500N
Considering all the weight on the cylinder
Total load on cylinder with all the factors =6682N
For design simplicity the load is considered equal to 7000N with fluctuations on the loading platform.
Cylinder power required to lift the same load : 7000/sin35 =12204N

C. Material Specification:
Material: Mild steel
Elastic Modulus: 200Gpa
Poison's ratio=0.3
Density=7800kg/m^3.
Yield Stress: 250Mpa

D. Design Checks
Maximum Load on the Pin: 12204N
Area of the pin $A=\pi r^2=3.14*12.5^2=491mm^2$.
Shear stress developed in the pin =12204/491=24.85Mpa
So factor of safety of the pin material =250/24.85=~10
So pin design is safe.

E. Buckling strength for the link:
Assuming the links are hinged at both the ends
Critical Load $Pcr=\pi^2*E*I/L^2$
Here moment of inertia 'I" for box section from Ansys software is

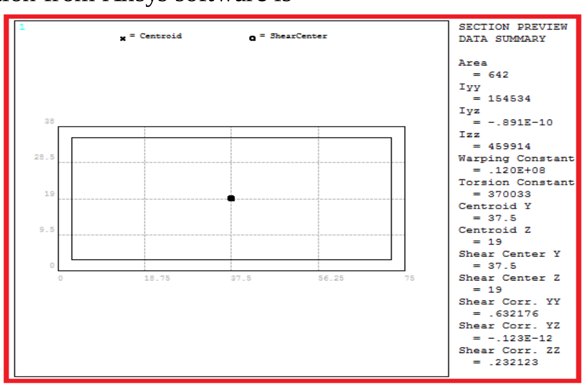

Fig 3: Moment of inertia in the major and minor axis

Since buckling can take place in any axis, minor moment of inertia is considered for the problem

Imin=154534mm^3.

$P_{cr}=3.14^2*200000*154534/1300^2$

P_{cr}=180495N

Since applied load is only 12204N, the considered section is very safe and can be optimised for the dimensions.

1) Buckling analysis from Ansys :

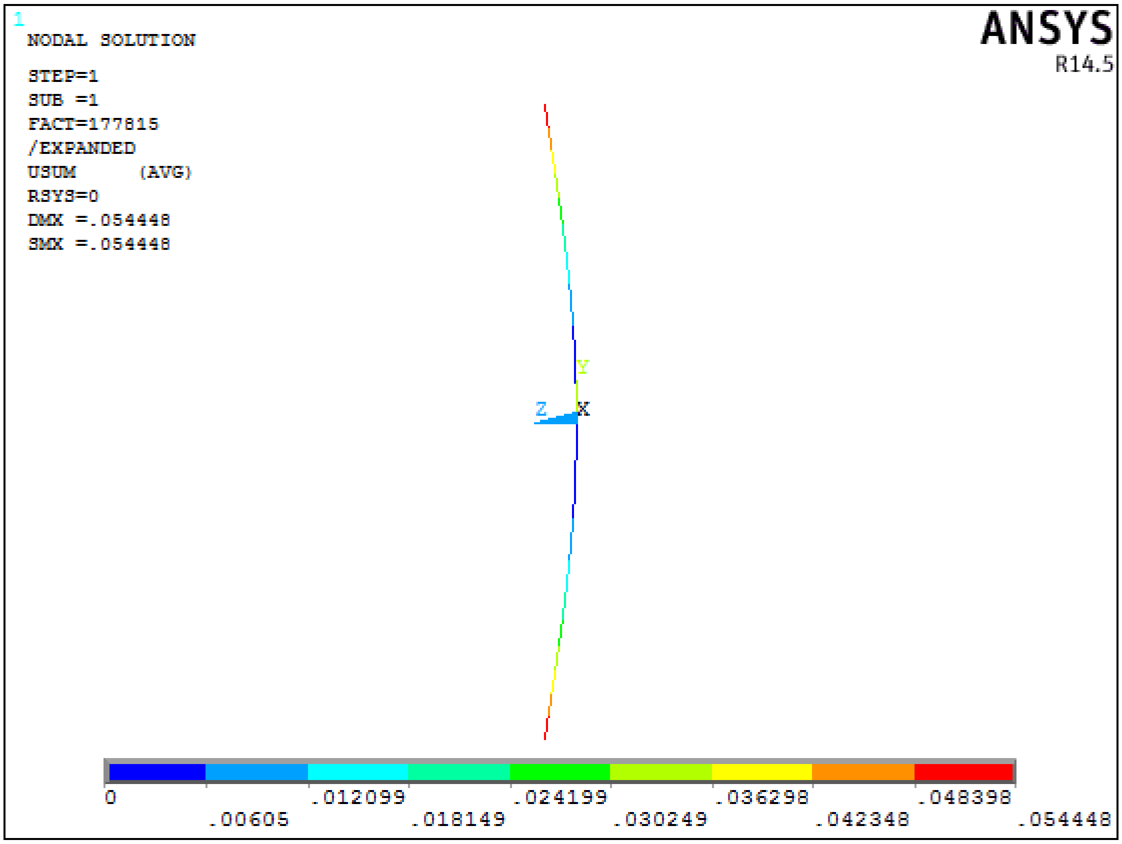

Fig 4: Critical Buckling Load from Buckling Analysis

F. Comparison of Finite Element and theoretical Calculation

Table 2: Comparative values from Theoretical and Finite element Solution

Description	Theoretical Solution(N)	Finite Element Solution(N)	Error
Buckling Load(Pcr) Newtons	180495	177815	1.5%

III.RESULTS & DISCUSSION

A. Design optimisation of the problem

Optimisation is carried out based on moment of inertia of the problem.

Table 3: Iteration summary for finding the best dimension for the problem

Iterations	Deformation(mm)	Vonmises Stress (Ma)	Weight (kg)	Moment of inertia(mm^4)	Minimum Value Required(mm^4)
1(75X38X3)	0.115	0.66	370.5	154534	52245
2(60X30X3)	0.1493	0.792	300	72792	52245
3(50X25X3)	0.185	0.9799	249	39954	52245
4(50X28X3)	0.178	0.933	264.42	52424	52245

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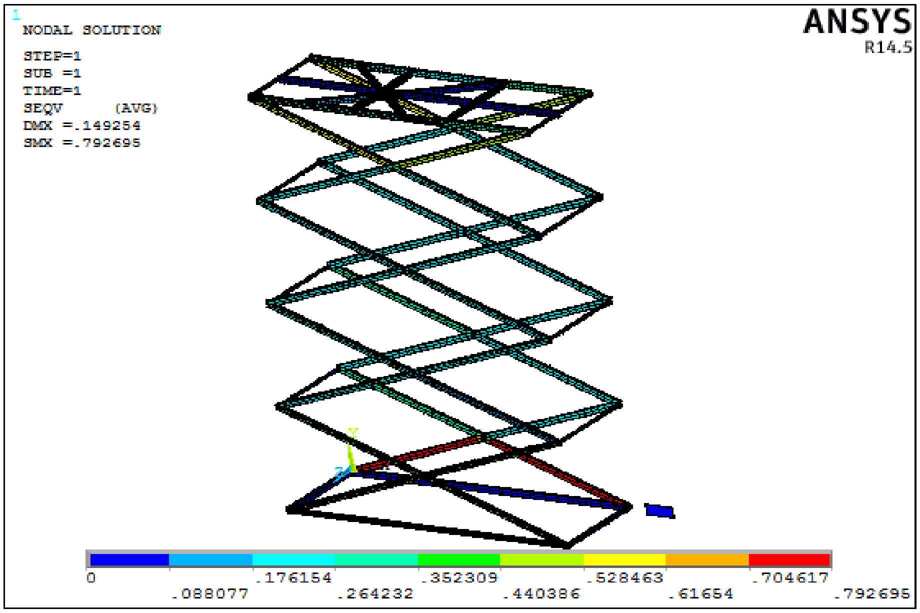

Fig 5: Stress in the 2nd iteration (Maximum Stress: 0.792MPa)

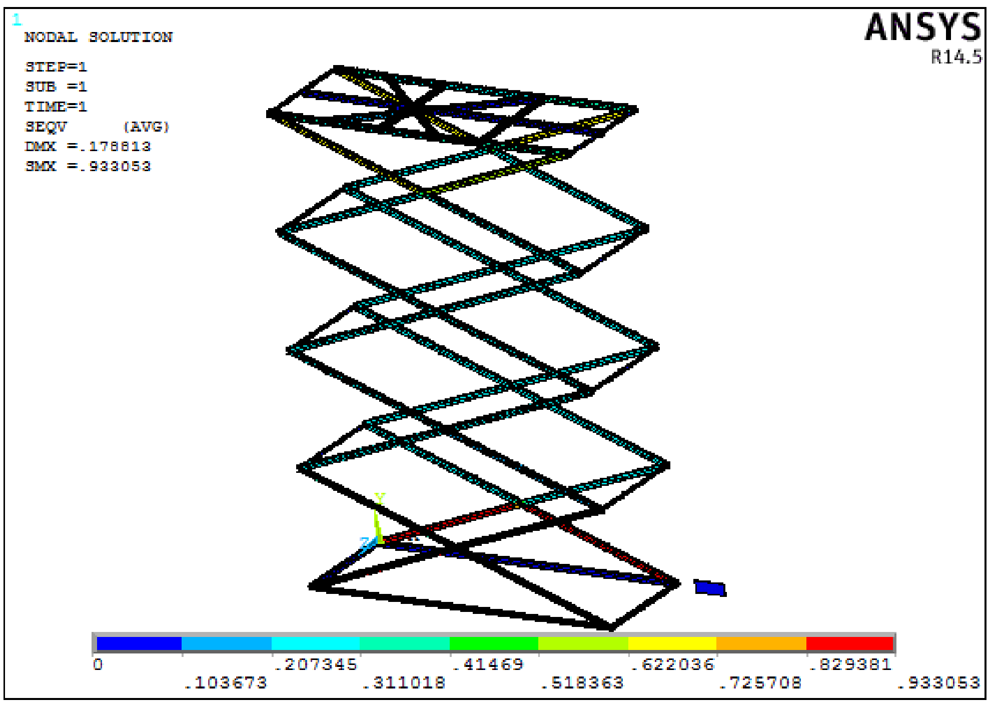

Fig 6: stress in 4th iteration (max stress 0.933 MPa)

B. Analysis with Various Sections

Table 4: Comparison of Moment of inertia with various Sections

Specification	Dimensions	Area	Moment of Inertia
Box	50X28X3	432	52424
Hallow Circular Section	25X22X3	442	122588
L Section	75X75X3	441	248455
C- Section	30X30X90	432	488916

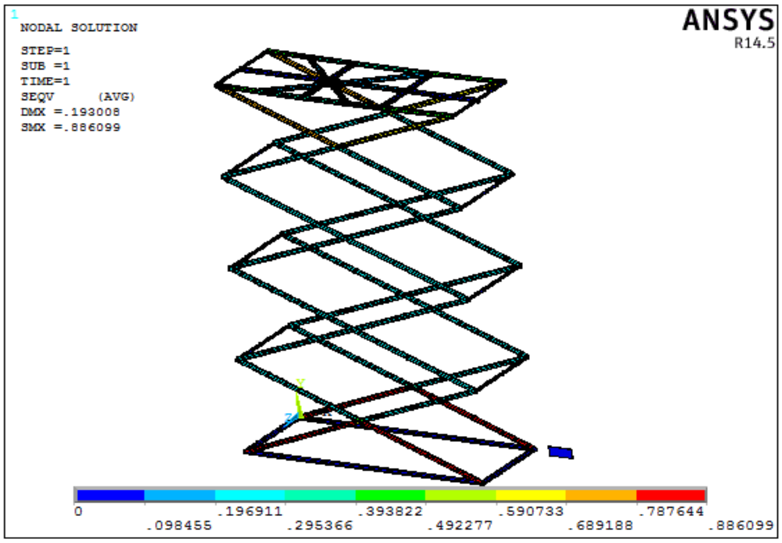

Fig 7: Vonmises Stress with circular section(Maximum Stress: 0.886099Mpa)

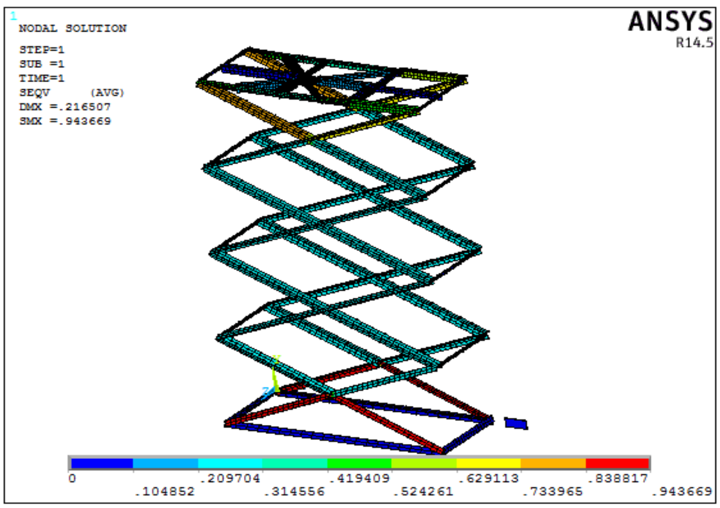

Fig 8: Vonmises Stress with L – Section

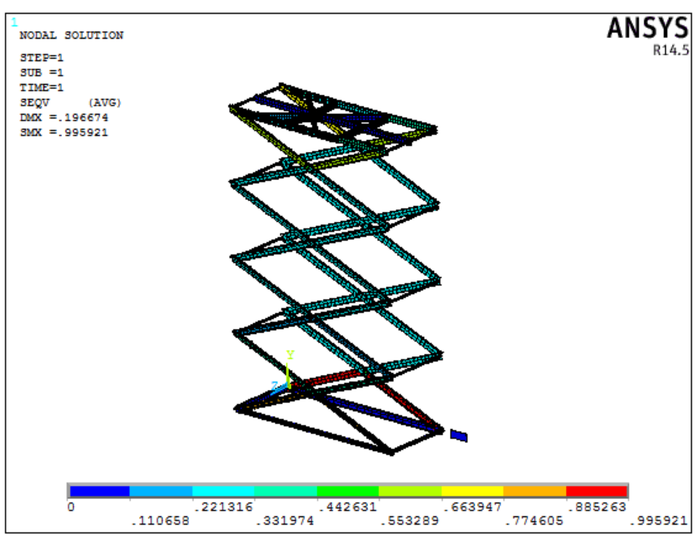

Fig 9: vonmises Stress with C-Channel

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C. Alternative Arrangement to Reduce the Load:

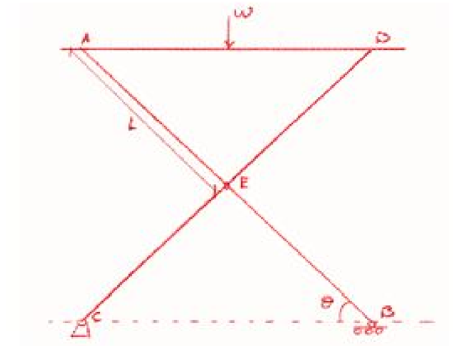

Fig 10: Free Body Diagram

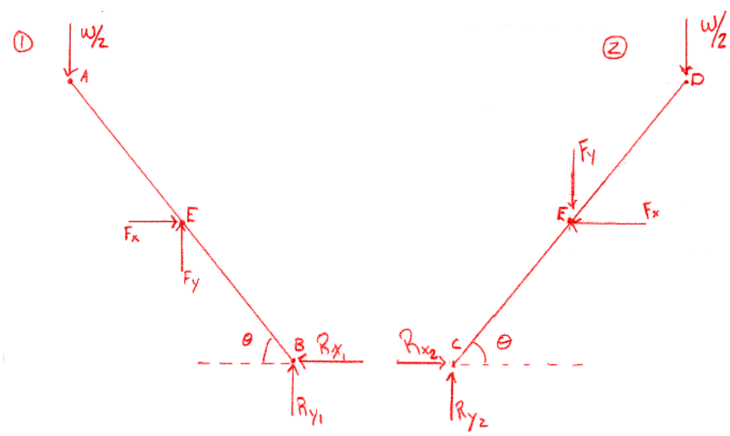

Fig 11: Reaction Forces

$$R_{x_1} = R_{x_2} = \frac{W}{\tan\theta}$$

$$F_x = \frac{W}{\tan\theta}$$

$$R_{y_1} = R_{y_1} = \frac{W}{2}$$

$$F_y = 0$$

Fig: So load acting on the actuator: Rx2=W/tanθ = 7000/tan(25) =15011 N
For Lowest Configuration Rx2=W/tan5=80010N
So lifting this may not be suitable arrangement for the problem.
If vertical drive is provided , no loss or amplification of force is required. But the vertical space requirement is more.
If vertical drive is used , angle equals to 90 degrees.
Force required to lift R=W/tan(90)=7000/tan(90)=7000N

Table 5: Comparison of Load Requirements with Hydraulic Load Direction

Description	Force for Driving(N)
Inclined Load	12204
Horizontal Load	15011
Vertical Load	7000

vertical loading is better for lifting the loads. But it requires vertical space during closure conditions or unloaded conditions

IV. CONCLUSIONS

An existing multi-scissor mechanism is analysed for actual stress conditions and finite element analysis is carried out to improve the design reducing the weight. The overall summary is as follows.

A. Initially cross sections are checked for sufficiency to with stand the load using theoretical calculations based on strength material basic concepts like bending, axial stress, buckling etc.
B. The truss members are checked for axial strength and pins are checked for shear strength.
C. The link section calculation is carried out for moment of inertia based on euler's buckling formulae for both ends hinged members. This moment of inertia required to take the load without buckling failure is taken as the reference value for optimisation.
D. Further structural stress condition is analysed for both open and closed condition. The closed condition load is different from open or extended condition load. The modelling and meshing is carried out using one dimensional line elements(link180) and analysis is carried out for closed and open condition due to variation of loads. The frame structure modelling is done by Beam188 element and the load is applied after bottom members are fixed in position.
E. Design optimisation is carried out based on moment of inertia which is calculated based on minimum buckling strength for hinged configuration. The dimensions are modified keeping the minimum buckling thickness 3mm. The geometry is optimised for the weight.
F. Further design optimisation is carried out based on the moment of inertia of the problem as the moment of inertia plays important role in the buckling strength. Different sections like box, hallow circular, L-angle and C-channel are checked for the strength. C-Channel has better strength for the same weight from finite element analysis. But Hallow circular and L-angle has equal moment of inertia on both major and minor axis.
G. Finally load required for driving is compared with various design formats. The existing force mode is better then horizontal model and the vertical model is better compared to other models. But the space requirement is a prohibitive term for adopting vertical drive. So existing drive is design is better when comparing with space and compactness.
H. All the designs are represented with necessary graphical and pictorial plots.

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