Designing And Calculating The Stresses Induced In Scissors Jack For Three Different Materials

Jaideep Chitransh, Dilshad Hussain

Abstract: A Scissor Jack is a mechanical device used to lift a heavy vehicle from the ground for changing the wheel and for maintenance purpose. The most important fact of a jack is that, it gives the user a mechanical advantage by changing the rotational motion into linear motion and allowing user to lift a heavy car to the require height. In this work we have designed different components of scissors jack using CAE tools i.e. CATIA andCalculate stresses induced in its different part which are responsible for failure and To Reduce its cost, so that everyone can afford this. We have taken reference of Mahindra Bolero Scissors Jack. The Dimensions of scissors jack was measured by Vernier caliper. By measuring all the dimensions of components of scissors jack, we have designed it in CATIA, after that we assemble all the components of Jack to shape a model of scissors jack and calculated different parameters (Max. shear stress, Max. principal Tensile Stress, Total torque required to lift the vehicle etc) which is used in all components of scissors jack to avoid failure. To reduce the cost of jack we have taken two different sections of three different materials i.e. Mild Steel, AISI 1045 grade Steel, GS-52.3 Cast Steel and comparing which will be suited for above mentioned purpose. By using these materials we have calculate and compare the Strength to weight ratio and find which material is best suited for high load carrying capacity with minimum failure or deformation.

Index Terms: Scissors jack, CATIA, Factor of safety, Max Principal Tensile Stress, Max Principal Shear Stress, Strength to weight ratio.

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1. Introduction

Screw type jacks were very common for jeeps and trucks of World War 2. The jacks with wedge mechanism had introduced in Nigeria because they can't afford high priced Jack because of their low standard of living and this jack provide a lift with a self-locking capability for both small- and medium-sized vehicles (Ademola A. Dare and Sunday A. Oke, 2008) There is evidence of the use of screws in the Ancient world but it was the great Leonardo, who first gives the design of a screw jack for lifting loads. Leonardo da Vinci design used a threaded worm gear, supported on bearings that are rotated by the turning of a worm shaft to drive a lifting screw to move the load (Chinwuko Emmanuel Chuka et.al, 2014).The most notable inventor in mechanical engineering was mechanical engineer Joseph Whitworth, who found the need for precision had become as important part in industry There is design based 3D software Pro/E with 8m high scissors lift platform. The platform is designed to be folded away doors, to save more space for convenient storage (TianHongyu et.al, 2014). The idea of using a screw as a machine element was given by Archimedes in 250BC, with his device used for pumping water. In 2011, a paper was published in which an author uses a toggle damper brace system to calculate and analyze installation modes and experiments conducted to improve and Understand the influence of key factors and to provide design guidance (R. Zhang, 2011).In the next 20 years the Duff Manufacturing Company was the largest manufacturer of lifting jacks in the world.

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A new version of the ball bearing screw jack was introduced by Duff Company. A modified version of motor Screw jack has been introduced by using an electric motor in the Screw to make load lifting easier (A.S.Akinonmi et.al, 2012). In 1930, the First worm gear screw jack that is instantly recognizable was found. In Toggle jack different pairs of screw and nut have used to get induce stresses within the safe limit under the loading condition from 1KN to 5KN (Prof. N. R. Patel et. al, 2013). With the ability to linked mechanically and driven by either air or electric motors, the first model had a lifting capacity of 10 tons with a raise of 4 inch. A new design with a new methodology and an attempt was made to fabricate a new design that can produce greater capacity within the economic range (Vishesh Ranglani et.al, 2014). Today, screw jacks can be linked mechanically and also electronically and with the advances in motion control, loads can be positioned to any small unit like microns. Improvements in gear technology together with the addition of precision ball and roller screws means the applications for screw jacks today are endless and are real. The new type of jack which is operated by using the power of car battery (G. S. Udgirkar et.al, 2014). Various Developments in Lifting Devices are screw threads, hydraulics, wheels and axles. In a screw jack which uses motor is now referred to as a linear actuator which is essentially still a screw jack. Various failures have also been occurred in the jacks due to Different stresses induced in the jack (C.S.Dhamak et.al, 2015). The calculation made in this paper is used to reduce the cost of jack by using two different sections of three different materials and comparing which is best suited to reduce the cost with high strength to weight ratio. In this case of a jack, a small force applied in its horizontal plane is used to raise or lower large load. We use Double start square thread as it is very strong and can resist the large loads imposed on most scissors jack while not being weakened by wear over many rotations.

2. OBJECTIVE:

This paper includes the scissor jack of Automobile vehicle is to reduce the weight of the jack by changing the manufacturability, reduce the cost of the jack by using material of low cost, Remove welding (By using Rivets) to avoid distortion and Failure. This paper simply focuses on how to reduce the cost of the jack (By using link of different cross sections) and to avoid failure of jack for this we have calculated the various stresses acting on the various components of jack and checked whether it is falls under safe limit or not. To reduce the cost of the jack we have taken two different sections (I-section and C-section) of three different materials and by calculating the various stresses of each material and market survey of each material. We may choose that which material is best suited for maximum load and good strengthening capability with less failure.

3. MATERIALS AND METHODS:

The subject of research was scissor jack of Mahindra bolero. The mechanism is dedicated to passenger vehicles with an unload weight not exceeding 2 Ton. Initial figure of model was designed in CATIA V5 showing the correct operation of the jack and range of mobility. Numerical calculation was calculated with the help Machine Design Data Book and after that we have studied the different stresses induced in the components of scissors jack which may responsible for failure and find load carrying capacity of scissors jack of three different materials. The mechanism was analyzed numerically, at maximum lift height of 300mm. The jack is designed on the basis of measurements taken from a real jack. The spatial model designed in CATIA. Three Different Materials taken are:
1- ASTM A36 Mild Steel
2- AISI 1045 Aluminum Alloy
3- GS-52.3 Cast Steel

4. COMPOSITIONS OF MATERIALS

1. ASTM A36 Mild Steel

Aluminum(98.7%), Manganese(0.10%), Silicon(0.40%), Zinc(0.25%), chromium(0.10%), copper(.20%), Iron(.70%).

2. AISI 1045 graded steel

Iron(98.9%), Copper(.46%), Manganese(0.80%), Sulphur(<0.050%), phosphorus(<.04%).

3. GS-52.3 Cast Steel

Iron, carbon (0.35%), silicon, chromium (.25%), copper (.10%)

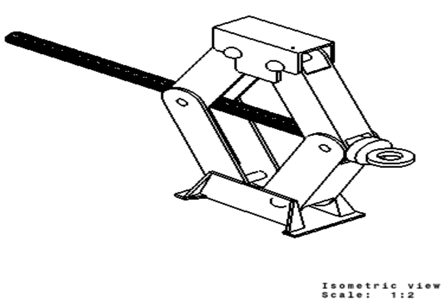

Fig 1. *Isometric view of scissors jack*

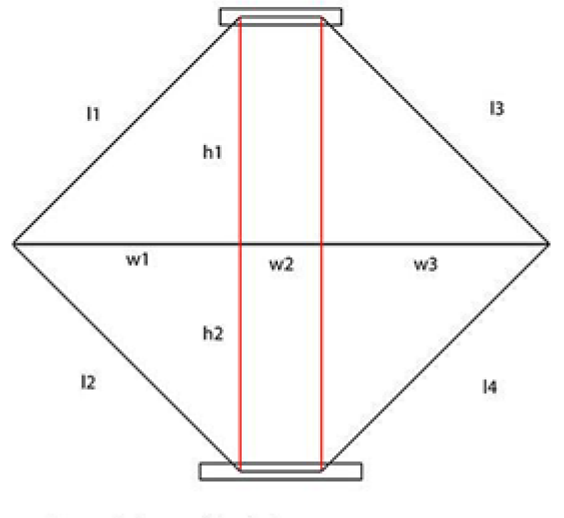

Fig 2. *Line diagram of scissors jack*

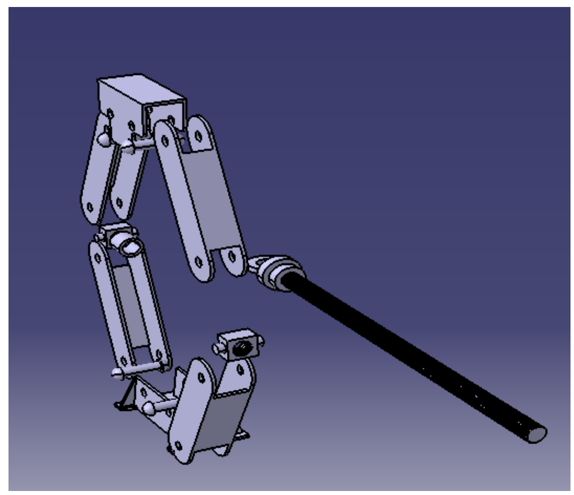

Fig 3. *Exploded View of Scissors Jack*

5. DESIGN CALCULATIONS

5.1 Design of Power Screw

Let the weight of the car is considered as 2 ton, then the weight acting on rear and front axle is 40% and 60% of its total weight, hence the weight acting on front axle is 1200 kg. A weight of 600 kg exerts on each wheel and the maximum load on screw act when jack is at its lowest position. Let us assume that, the thread on screw is a Double Start Square thread.

Coefficient of friction between threads= 0.25. The whole calculation is made by assuming material of cast steel.
Let L_1, L_2, L_3, L_4 be the length of each link.
Such that $L_1 = L_2 = L_3 = L_4 = 160$mm and $W_1+W_2+W_3$ be the Length of power Screw
So, $W_1 = W_3 = 150$ mm
$W_2 = 50$ mm
Max. Lift = $(h_1+h_2) = 300$ mm
θ is the angle between Link with the horizontal when jack is at its lowest position.
W = (Mass * g) = (600*10) = 6000 N = 6 KN
The tension T acting on the power screw

Cos θ = (175-25)/160 = 20.36˚
Tension, T = W/2*tan θ
Total tension = 2*T = W/tan θ

Since we have assumed that the material for power Screw is a Cast Steel, so σ_t = 130N/mm^2

Let dc be the core diameter of the screw.
Load = (π/4)* dc² * σ_t
2*T = W/Tan θ = (π/4)* dc² * σ_t
2*T = 6000/Tan (20.36˚) = 16168.053 N
dc² = [W/ Tan (θ)*4]/ [(π* σ_t)]
Hence, dc = 12.58 mm

Since the screw is subjected to torsion shear stress we take, dc = 12 mm, pitch P = 1.5mm
Mean diameter, d = do - P/2 = 13.5-1.5/2 = 12.75 mm
Outer diameter, do = dc + P = (12+1.5) = 13.5 mm
Check for self-locking
Tan (α) = Lead/π*d; α= helix angle
Lead L = 2*P; since the screw has a double start square thread.
Tan (α) = 2*p/π*d = 2*1.5/ π*12.75 = 0.0606
Helix angle; α= 3.47˚
Coefficient of friction; μ = Tan ϕ = 0.25; friction angle: ϕ= 14.03˚
Since, ϕ >α hence the screw is self-locking
Effort required to support the load = 2*T Tan (θ+α)
= 16168.05 (Tan (α) + Tan (θ))/ (1- (Tan (α) * Tan (θ)))
= 5099.04 N
Torque required to rotate the screw = effort*d/2
= 32506.42 N-mm
Tensile stress σ_t = 2*T/ (π/4) * dc² = 16168.05/ (π/4) * 12^2
= 142.95 N
Shear stress in the screw due to torque ζ= 16*T/ (π* dc³)
= 23.82 N/mm^2
Maximum shear stress ζ_{max} = √ (σ_t^2 +ζ²)/2
= 72.460N/mm^2
Maximum principal stress $\sigma_{t\,max}$ = σ_t/2 + √ (σ_t^2 + ζ²)/2
= 143.93 N/mm^2
Since the maximum stresses $\sigma_{t\,max}$ and ζ_{max} within the safe limits.

5.2 Design of Nut
Let n is the number of threads in contact with the screw Let us assumed that the load is Uniformly Distributed over the cross sectional area of the nut. Let P_b be the Allowable Bearing pressure between the threads.
Bearing pressure is assumed as 65 N/mm^2
P_b = (2*T)/ ((π/4)*(do²-dc²)*n)
65 = (16168.05)/ ((π/4)*(13.5^2-12^2)*n)
Number of threads, n = 9.79 ≈ 10
In order to have good stability let n=10
Thickness of Nut = n*p = 10*2=20 mm
Width of Nut b =1.5*do =1.5*13.5=20.25 mm
To control the movement of nuts beyond 300 mm the rings of 8 mm thickness are fitted on the screw.The length of screw portion = 300 + (8*2) + 20
= 336 mm ≈ 350 mm
Total length of screw is 350 mm.

5.3 Design of Top Arm
Since σ_{yt} for cast steel be 260 N/mm² and assume Factor of safety (FOS) be 3.
Then,
σ_t = σ_{yt}/FOS=260/3=86.66 N/mm²
σ_c = 1.25*σ_t = 1.25*86.66 = 108.33 N/mm²
Moment of Inertia I_{xx} = 48007.56 mm^4, I_{yy}= 51009.38 mm^4
Radius of Gyration R_x= 15 mm, R_y= 13 mm
Rankine constant (a) =1/7500 (For Steel)
=1/9000 (For wrought iron)
= 1/1600(For cast iron)
a=σ_c/п^2*E
Cross section area (A) = (40*3) + (24*3) + (40*3) = 312 mm²

Crippling Load in horizontal plane
Both Ends are Fixed (L_{eff} = L/2)
Crushing stress can be calculated by the formula
=П^2*E*I/L^2
=300N/mm^2
P_{cr} = (σ_c*A)/ [1+a*(L/2*R_x) ²]
= (300*160*40)/ {1+ (1/7500)*(160/2*15)²}
= 97660.2523 N

Crippling Load in vertical plane
Both Ends are hinged (L_{eff} = L)
Crushing stress can be calculated by the formula
=П^2*E*I/L^2
σ_c = 300 N/mm²
P_{cr}= (σ_c*A)/ {1+a*(L/R_y) ²}
= (300*312)/ {1+ (1/7500)*(160/13)^2}
= 91764.70 N
Since the Buckling load is greater than Design load, dimensions of the link safe.

5.4 Design of Bottom Arm
Since σ_{yt} for cast steel be 260 N/mm² and assume Factor of safety (FOS) be 3.
Then,
σ_{yt} for cast steel = 260 N/mm²
Factor of safety (F.S) = 3
σ_t = σ_{yt}/F.S=248/3=86.66 N/mm²
σ_c = 1.25*σ_t = 1.25*82.66 = 108.33 N/mm²
Radius of Gyration
R_x= √I_x/A= 14.778 mm, R_y=√I_y/A= 12.784 mm
Rankine constant (a) =1/7500 (for Steel)
=1/9000 (For wrought iron)
= 1/1600(For cast iron)
a=σ_c/п^2*E
Ends are hinged (L_{eff} = L/2) (Both ends are Hinged)
Cross section area (A) = (40*3) + (30*3) + (40*3) = 330 mm²
Moment of Inertia I_{xx} = 72720 mm4, I_{yy}= 54567.31 mm^4

Crippling load in horizontal plane
Ends are hinged (L_{eff} = L/2) (Both ends are Fixed)
Crushing stress can be calculated by the formula
σ_c =П^2*E*I/L^2
=300N/mm^2
P_{cr}= (σ_c*A)/ {1+a*(L/2*R_x)²}
= (300*160*40)/ {1+ (1/7500)*(160/2*14.778)²}
= 102589.15N

Crippling load in vertical plane

L_{eff}=L (Both ends are hinged)
Crushing stress can be calculated by the formula
$\sigma_c = \Pi^2 * E * I / L^2$
$= 300 N/mm^2$
$P_{cr} = (\sigma_c * A) / (1 + a * (L / R_y)^2)$
$= (300*330) / \{1 + (1/7500)*(160/12.784)^2\}$
=96963.76 N
Since buckling load is greater than the Design load, dimensions of the link safe.

5.5 Top Plate (Loading Platform)

P = 6000 N, L = 50 mm, B = 36 mm, H = 40 mm
Moment, M = (P*L)/4
M = (6000*50)/4 =75000N-mm
$Z = (B*H^2)/6 = (36*402)/6 = 9600\ mm^3$
$\sigma_b = M/Z = 75000/9600 = 7.8125 N/mm^2$

The permissible stress for cast steel is 130 N/mm^2 and it is greater than $\sigma_b = 7.81\ N/mm^2$

The top plate design is safe.

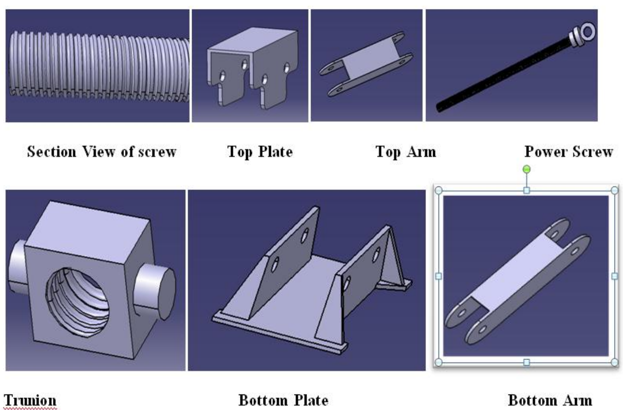

Fig 4. *Design of components in CATIA*

6. RESULT

On calculating the designs of various components of scissors jack by taking three different materials. we concluded that out of three materials AISI 1045 graded Steel is also good for carrying maximum load but in comparison to mild steel it is more.GS -52.3 cast steel is also falls under the safe limit so it can also be consider for manufacturing purpose of scissors jack. In this Paper we concluded that all three materials falls under the safe limit which is very important to avoid the failure and its top and bottom arm should be made of C-section to reduce the product of cost because it uses low quantity of material to manufacture and also have great stability. The calculation is made manually and comparing it with ANSYS software to get better and reliable result.

TABLE 1

MECHANICAL PROPERTIES OF MATERIAL

S.No	Ultimate tensile strength	Ultimate yield strength	Modulus of elasticity	Shear modulus	Poisson's ratio
ASTM A36 mild steel	450MPa	250MPa	200GPa	79.3GPa	.260
A1045 grade steel	565MPa	310MPa	201GPa	80GPa	.29
GS-52.3 cast steel	650MPa	360MPa	200GPa	80GPa	.30

TABLE 2

STRESSES INDUCED IN DIFFERENT SECTIONS OF THREE DIFFERENT MATERIALS

S.No	C-Section			Box Section		
	σ_{tmax} [MPa]	τ_{max} [MPa]	σ_b [N/mm^2]	σ_{tmax} [MPa]	τ_{max} [MPa]	σ_b [N/mm^2]
ASTM A36 Mild steel	102.1	62.7	6.5	87.6	48.3	4.1
A 1040 grade steel	128.4	73.8	6.8	109.3	56.7	4.9
GS-52.3 cast steel	143.3	72.4	7.8	118.7	63.4	5.8

Where σ_{tmax} – Max.principal tensile stress
τ_{max}—Max principal shear stress
σ_b—bearing stress

7. CONCLUSION

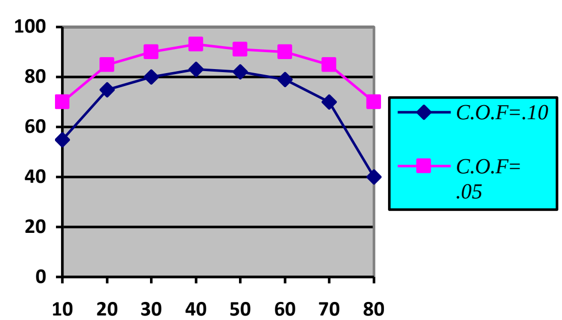

Fig5. *Efficiency v/s Helix angle Diagram*

This jack has the reduced weight (by changing the manufacturability). Only rivet joints are induced (Removal of welding to avoid distortion).Following conclusions have been made while calculating the above equation. The efficiency of square threaded screw increase rapidly up to helix angle of 20°, after which the increase in efficiency is low. The efficiency is maximum for helix angle between 40° to 45°and start decreasing when maximum value is achieved (decreases rapidly when helix angle exceeds60°).The η of scissors jack starts decreasing rapidly when coefficient of friction increases. This is due to the fact the normal thread force becomes large and thus the force of friction and the work of friction become large as compared with the useful work. This results in low efficiency. To increase the efficiency of square threaded screws, reduce the coefficient of friction between screw and nut by proper lubrication and to Increase the helix angle up to 40 to 45° by using Double Square start threads.

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