Structural analysis of a trapezoidal scissor jack

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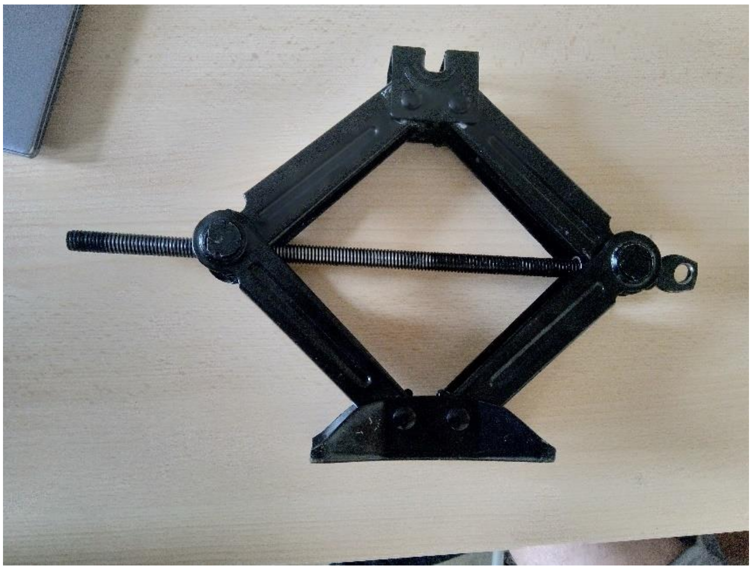
Introduction:

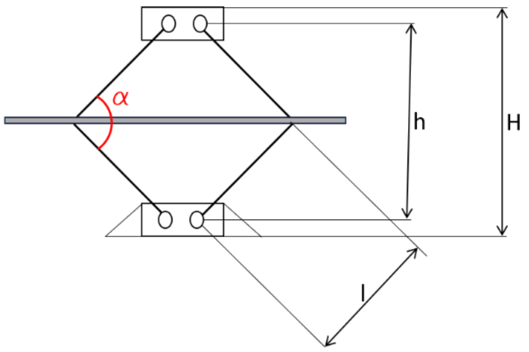
A trapezoidal scissor jack is mechanical device used to lift a road vehicle partially off the ground, usually for the purpose of changing a tyre or performing roadside repairs. It works on the basis of mechanical advantage, allowing small, gradual rotations of a screw to be converted into a linear, upward lifting force.



The safety considerations in the design of such a device are paramount. Any kind of structural failure of a car jack may leave the user trapped and/or seriously injured. Failures can through many mechanisms; the most common causes are the compressive yielding or buckling of the lifting arms, tensile yielding of the lifting screws, axial failure of the bearing or the stripping of the screw threads or driving teeth at the end of the lifting arms.

For this reason, this report will focus on assessing the safety of a scissor jack in these specific areas, and choosing the appropriate materials and dimensions to ensure it is fully able to comply with its nominal loading limits.

Parameters of the jack:



The following parameters have been measured from the selected scissor jack, or are nominal values provided by the manufacturer.

Parameter	Symbol	Value
Minimum lifting height	H_{min}	233mm
Maximum lifting height	H_{max}	322mm
Minimum height between the top and bottom joint pins	h_{min}	177mm
Maximum height between the top and bottom joint pins	h_{max}	266mm
Minimum arm angle	α_{min}	40 degrees
Maximum arm angle	α_{max}	150 degrees
Maximum mass rating	m	1T = 1000kg
Nominal lifting screw diameter	-	12mm
Arm length between pins	l	138mm

Acceptable minimum and maximum length of the lifting arms:

This sets the upper and lower limit of each lifting arm.

$$l_{max} = \frac{h_{min}}{2\sin\left(\frac{\alpha_{min}}{2}\right)} = 258.8mm$$

$$l_{min} = \frac{h_{max}}{2\sin\left(\frac{\alpha_{max}}{2}\right)} = 137.7mm$$

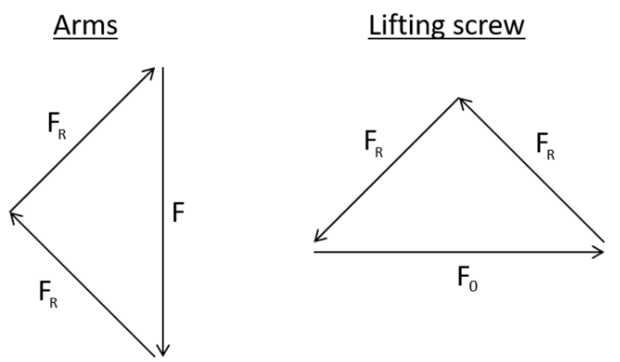
This means an arm length of 138mm between pins is acceptable.

Force exerted by the vehicle onto the jack, assuming the car carries half the vehicle's weight:

$$F = \frac{mg}{2} = 4903.3N$$

This is the maximum load that the jack is designed to carry.

Maximum load exerted onto the lifting arms and screw:



The reaction forces of the arm and lifting screw to the load are shown in the diagram to the left.

Maximum load on the lifting arms:

$$F_{R,max} = \frac{F}{2\sin\left(\frac{\alpha_{min}}{2}\right)} = 7168.2N$$

Maximum tensile force on the lifting screw:

$$F_0 = 2F_r\cos\left(\frac{\alpha_{min}}{2}\right) = 13471.8N$$

Dimensioning of the lifting screw:

$$T = F_0\frac{d_2}{2}\tan(\alpha + \rho')$$

Where:

d_2 = Trapezoidal thread diameter

α = Thread shape angle

ρ' = Correlated half cone of friction

For the material of the lifting screw, I chose C45E steel with a yield strength of R_c = 490MPa. I selected a reasonable safety factor of Z =1.5.

Finding the working stress:

$$\sigma_w = \frac{R_c}{Z} = 326.7MPa$$

Minimum core cross-sectional area and diameter:

$$A_{min} = \frac{F}{\sigma_w} = 41mm^2$$

$$d_{1,min} = \sqrt{\frac{4A_{min}}{\pi}} = 7.1mm$$

The thread standard I chose is TR12 x 3 from DIN 103. Its parameters are as seen in the table below.

Parameter	**Symbol**	**Value**
Nominal diameter	d	12mm
Pitch	P	3mm
Pitch diameter	$D_2 = d_2$	10.5mm
Screw core diameter	d_1	8.5mm
Outside diameter	D	12.5mm
Hole core diameter	D_1	9mm
Flank angle	β	30 degrees

We will assume a coefficient of friction of μ = 0.12 between the contacting parts of the threads.

Finding ρ' and α from the above:

$$\rho' = arctan\left(\frac{\mu}{cos\left(\frac{\beta}{2}\right)}\right) = 7.1°$$

$$\alpha = arctan\left(\frac{P}{d_2\pi}\right) = 5.1°$$

Since $\alpha < \rho'$, the thread is self-locking.

The cross-sectional area of the lifting screw:

$$A_1 = \frac{d_1^2\pi}{4} = 56.7mm^2$$

Normal stress induced in the lifting screw:

$$\alpha_1 = \frac{F_{0,max}}{A_1} = 237.4MPa$$

Required lifting torque:

$$T = F_0 \frac{d_2}{2} \tan{(\alpha + \rho')} = 15392.8Nmm$$

Shear stress:

$$\tau = \frac{T}{k_p} = \frac{T}{\left(\frac{d_1^3 \pi}{16}\right)} = 127.65MPa$$

Reduced stress:

$$\sigma_{red} = \sqrt{\sigma^2 + 3\tau} = 324.42MPa$$

The screw can bear the stress, since $\sigma_w > \sigma_{red}$.

Dimensioning the nut:

The surface pressure p between the spindle and the nut is given by:

$$p = \frac{F_0}{A_p}$$

Where

$$A_1 = \frac{d^2 D_1^2}{4} i\pi$$

Where i = m / P = The number of threads

m = The height of the nuts

P = Thread pitch

Z = 1.5 = Our safety factor.

Using the material C225E with a yield strength of R_e = 340MPa, the maximum press allowed to be exerted on each thread:

$$p_w = \frac{R_e}{Z} = 226.7MPa$$

Finding the number of threads needed to bear this load is:

$$i = \frac{F_0}{\frac{d^2 - D_1^2}{4} \pi p_w} = 1.20 \approx 2$$

Which means we need at least 2 threads (since we need to round the number up).

Finding the minimum length of the threaded part of the nut given that we need at least two threads:

$$Minimum\ length\ of\ the\ threaded\ part\ of\ the\ nut = iP = 6mm$$

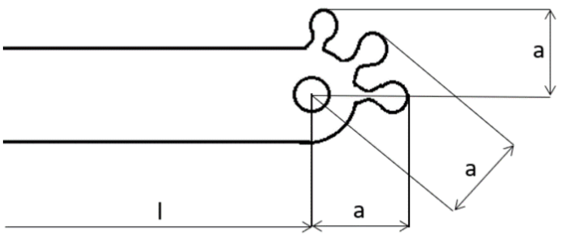
Choosing an appropriate bearing:

The spindle will require a thrust bearing, as the loads going through it will be primarily axial. The SKF51201 standard thrust bearing suits the dimensional requirements of the car jack. To see if its static load rating of c_0 = 20.8kN = 20800N is enough, we must check its static load ratio:

$$S_0 = \frac{c_0}{F_0} = 1.2322$$

S_0 = 1.2322 > 0.5, which means the bearing checks out.

Checking the loading on the end of the lifting arm:



Number of teeth n = 6

l = 138mm

a = 20mm

The teeth experience parallel ($\tau_{||}$) and perpendicular ($\tau_{\perp}$) shear stresses given by:

$$\tau_{||} = \frac{F_0}{4(l-a)a} = 1.4MPa$$

$$\tau_{\perp} = \frac{T}{d \times 2a(l-a)} = 0.4MPa$$

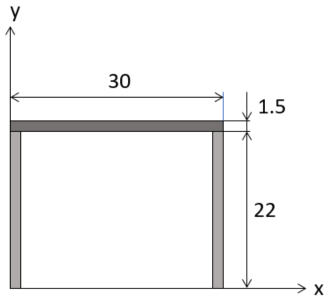
Reducing this stress into a value σ comparable to the material's properties:

$$\sigma = \sqrt{3\left(\tau_{\perp}^2 + \tau_{||}^2\right)} = 2.5\ MPa$$

Recalling that the yield stress is R_c = 490MPa, we can see that teeth at the end of the lifting arm easily meet the required strength value.

Dimensioning the lifting arm:

Modelling the bar's cross-section as a u-profile beam:



Area A = 111mm^2

The displacement of the centroid of from the x-axis S_y = 15.76mm

The second moment of area I_y = 6365mm^4

Compressive stress acting on the beam:

$$\sigma = \frac{F_R}{A} = 64.6\ MPa$$

Calculated margin of safety:

$$Z = \frac{R_c}{\sigma} = 7.59$$

The beam's margin of safety is higher than our safety factor of 1.5, therefore it is acceptable at its current size.

To see whether the arms need to be checked for buckling, we have to calculate the slenderness ratio λ. To do this, we need the radius of gyration i:

$$i = \sqrt{\frac{I_y}{A}} = 7.272mm$$

The arm comes under a risk of buckling if the slenderness ratio exceeds 60:

$$\lambda = \frac{l}{i} = 18.225\ MPa < 60\ MPa$$

Which means that the beam's slenderness well under a value which would pose a risk of buckling. This means it is unnecessary to check the beam's buckling characteristics.