Column Car Jack
(Hinged Tongue Type)

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1.Introduction of Car Jack Design

1.1 Duty and Operation of Car Jacks

All carjacks have the same function: they raise the vehicle so that repairs may be made, usually changing the wheels when a puncture occurs. It is also necessary to lift the vehicle when changing the seasonal tyres or wheels. There is an emergency factor to changing wheels, particularly if the punctured tyre requires it. This signals a critical need for Car Jack: the wheel needs to be replaced quickly and safely while traffic is moving. The most crucial piece of equipment for a car is a functional car jack.

1.2 Design of Column Car Jack

Full assembly design of Column Car Jack is shown in detail in Fig. 1.1, meanwhile 2 detailed parts: A (Driving Handle) and B (U-Cup) are displayed in 2:1 scale, shown in Fig. 1.2 and Fig.1.3 respectively. the lifting tongue is placed in the lowest position where it can be fixed (slipped) into the holder socket which is fixed to the car body bottom (underside). The column inside the cup has square tube shape and slotted all along the column length in order for the free vertical moving of the nut holding bracket. The leg, the column and the locking device give solid shape of the jack. The driving handle (driving arm) operates the threaded shaft through a bevel gear couple inside the cup. The driven threaded shaft is suspended in the bracket through the bigger bevel gear and the thrust bearing. While the driving handle drives the threaded shaft, the nut generates movement for the locking device up or down in the column pending on the rotational direction of the driving arm. The lifting tongue moves together with the locking device where the lifting tongue is hinged on the pivot.

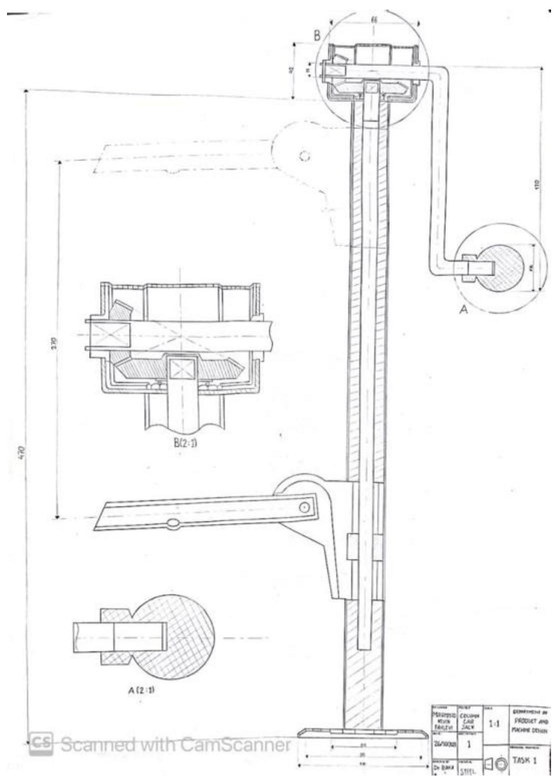


Fig. 1.1: The assembly view of the construction drawing of the Column Car Jack

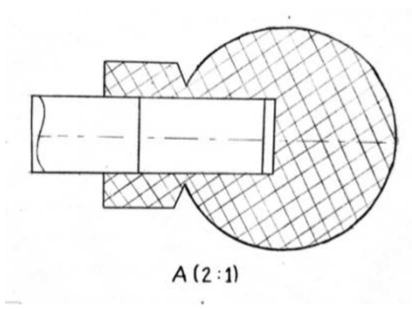


Fig. 1.2: The detailed view of the Driving Handle part

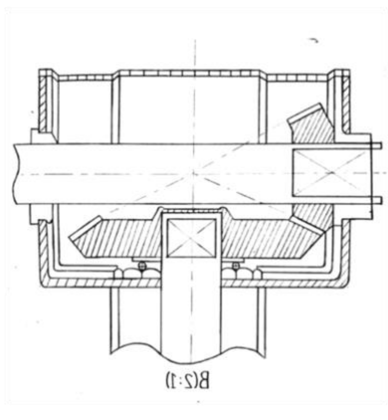


Fig. 1.3: The detailed section of the U-Cup

1.3 Parts and Material Specification of a Column Car Jack

There are 18 parts are connected in a Column Car Jack with a different mechanical function (purpose), and most of them are also made from various material technology with their own specification and standard. Three tables below provide information related to each parts and their specific description.

Table 1.4/a: Parts Description of a Column Car Jack

Element's item number	Denomination	Material Technology	Specification	Loadings
1	Leg	Made of weldable steel plate, sheared &cold formed	Standard Number:MSz En 10025-2** Material: S235(R_{eh}=235 MPa, Rm=360-510 MPa	Stresses in the welded seam, leg pressure (from the ground) and bending
2	Column	Produced from steel plate by shearing &cold bending	Standard Number: MSz En 10025-2** Material: S235(R_{eh}=235 MPa, Rm=360-510 MPa) or S275N (Reh=275MPa&Rm=310-540MPa)	Compression, buckling, stresses in the welded seam at the leg
4	Cup	Zinc plated steel plate of 0.4mm	MSz EN 10139 DC01, Remin=310MPa Rm=310-540MPa	Loading does not exist
5	Bush*	Turned from bronze bar,semi-product	MSz 8579:1990 (Bzo 12) CuSn12 Rm=270MPa $R_{0.2}$=150MPa	Surface pressure
6	Threaded Shaft	Turned from bar.Semi-product of tempered carbon steel	MSz EN 10083-1 C35, Re=270MPa, Rm=520MPa C35E, Re=430MPa, Rm = 630MPa	Torsion & Tension. Shearing at the riveted head. Surface pressure between the contracting threads
7-8	Bevel Gear	Made of precision casting from white cast iron (or	MSz EN 1562 EN-GJMW-360-12 Rm=360MPa. $R_{p0.2}$=190MPa	Tooth root bending

		aluminium alloy)		
9	Thrust ball bearing	Purchased from stock as a complete element	**SKF** product series	Static axial (thrust) load

***May be produced from cast iron or polyamide too**

**MSz-Hungarian Standard

EN-Euro norm (Standard)

Table 1.4/b: Parts Description of a Column Car Jack

Element's item number	Denomination	Material Technology	Specification	Loadings
10	Driving arm	Made of semi-product steel bar by cutting to the proper length by chip formation & cold bending	MSz EN 10083-1 C25, Re=230MPa Rm=440MPa C25E, Re=370MPa, Rm=550MPa	Bending & Twisting
11	Retainer(snap) ring for the shaft	Purchased from stock catalogue	See more at:https://www.seeger-orbis.com/products/snap-rings/circlips/snap-rings/circlips-to-din-9927-/-9928	Loading does not exist
12	Handle	Made of Polyamide by injection moulding	PA-6 Re=103MPa Rm=90MPa	Loading is negligible
13	Retainer Ring	Made from spring wire by cutting and bending	MSz EN 10089 Material: 38Si7 $R_{p0.2}$=1150MPa Rm=1300-1600MPa	Practically no loading exists
3	Head (bracket or U shape)	Made of steek plate by shearing and bending	ASTM-A36 Steel Re=250MPa Rm=550MPa	Practically, no load exists
14-15	Locking Device	Consists of 2 mirrored parts and both parts are made by cutting & hot forming in die	MSz En 10025-2 S235JR Reh=275MPa Rm=430-580MPa	Surface pressure on the hinge pin & surface pressure on

		from steel plate. Then the 2 parts are fixed.		the top of the nut.
16	Tongue	It is cut from tempered steel rod; the hole is drilled, and the tongue body is forged.	C25E Re=370MPa, Rm=550MPa	

Table 1.4c: Parts Description of a Column Car Jack

Element's item number	Denomination	Material Technology	Specification	Loadings
17	Nut ***	Produced from tempered carbon steel by chip formation processes. It is advised to select better grade steel for the nut than for the threaded shaft	MSz EN 10083-1 C45, Re=305MPa, Rm=580MPa C35E, Re=430MPa, Rm=630MPa C25E, Re=370MPa, Rm=550MPa	Surface pressure on the threads
18	Pin	Made from carbon steel rod by cut into size & cold formed.	MSz EN 10083-1 C22, Re=240MPa, Rm=430MPa	Double shear &Surface Pressure

***Advised to select different material or different yield strength material for the nut and for the threaded shaft in order to avoid unfavourable friction behaviour between the two members.

2. Strength Calculations of a Column Car Jack

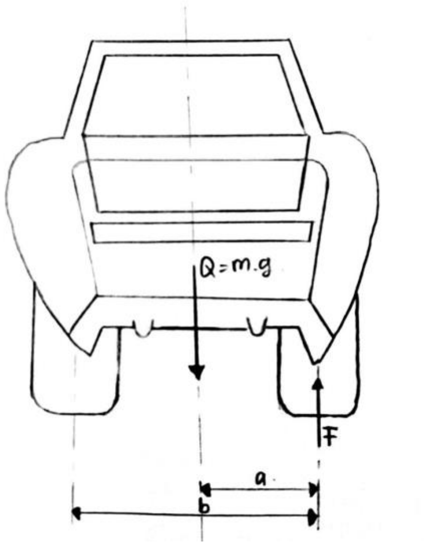
2.1 Basic Data of the Material

In order to calculate strength, stress, force, and other mathematical calculations related to a Column Car Jack, the initial data of the material must be recognized. These data are useful as important properties for the further calculations.

- The empty weight of the car to be lifted: m = 1000 Kg.
- The height of the connection point from the ground: h = 150 mm.
- Lifting height required for wheel change: H = 270 mm

2.2 Determination of the Acting Load (Lifting Force)

Figure 2.1 shows the car from behind. The mass, (m) acts at the car centre of gravity as a concentric load and the load Q= m.g acts in the direction of the gravity, while the reaction forces (F) act on the tires acting in the opposite direction (upward direction) from the ground.



Notes:
F: Force [N]
m: Mass of the Car [Kg]
g: Gravitational Force [m/s2]
= 9.81 m/s2
a: Distance between each wheel to the middle [m]
b: Distance between wheels [m]

When lifting, only one side of the car is lifted. With the car jack, we need to exert an upward lifting force of magnitude F, which can be calculated from the balance equation written for the point of contact of the left wheel with the ground.

-m•g•a+ F•b =0, and **F=m•g•a/b**

Due to the symmetry (b= 2•a)

F = m•g/2.

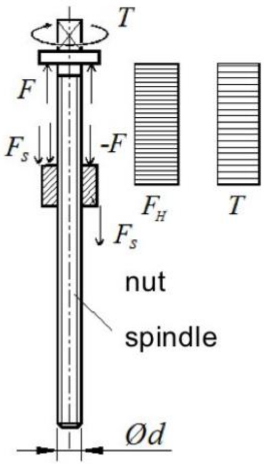
Assume, the mass of the car is 1000 kg. Thus, the lifting force is:

F = mg/2 = (1000•9.81)/2=4905N. The force is static and its direction is vertical.

2.3 Dimensions of the Threaded Shaft and Nut for Strength

A. Threaded shaft

The (d) diameter threaded shaft is loaded by the tensile lifting load (F). To lift the load by the nut requires a torque which overcomes the lifting and the friction load. If the friction is considered only on the thread and neglected at the thrust bearing and other places, then the required lifting torque is:



T = F•d •0.5•tan (α+ ρ'),

Where:

T = Torsion / lifting torque

F = Lifitng Force

d = the mean diameter of the thread

α = slope (lead) angle of the thread

ρ' =corrected half cone angle of the thread.

Strength dimensioning; the tensile stress loading the threaded shaft:

$\sigma = F/A$, where $A = d_1^2 \cdot \pi/4$ is the cross section of the thread at the root.

The permitted working stress is:

$\sigma_w = R_e/z$, where R_e, is the yield stress and z is the safety factor.

From Table. 1, the selected material is tempered carbon steel C35 E which has Re = 345 MPa (Minimum yield stress for C35E material) in tempered state and the safety factor is taken about z= 1.5. Thus, the permitted working stress here is:

$\sigma_w = 345/1.4 = 230$ MPa, substituting this value into the equation above for determine the needed thread root cross sectional area:

$A_{min} = F/\sigma_w = 4905/230 = 21.32\ mm^2$

The required minimum tooth root diameter from A_{min} is:

$$d_{min} = \sqrt{\frac{4Amin}{\pi}} = \sqrt{\frac{4*21.32}{\pi}} = 5.21\ mm$$

The cross-sectional area for the inner roller thread is:

$A = d_1^2\pi/4 = 10^2\ \pi/4 = 78.5\ mm^2$

The polar sectional modulus is:

$\mathbf{K_p = d_1^3\pi/16 = 10^3\ \pi/16 = 196.25\ mm^3}$

The selected material of the nut can be also steel some light greasing is considered between the threaded shaft and the nut providing the approximate friction coefficient μ=0.12.

The torque required for load lifting operation is:

$\mathbf{T=F*0.5*d_2*tan(\alpha+\rho') = 4905 * 7.5/2\ tan\ (3.6 + 6.6) = 3313.5\ N*mm}$

where $\alpha = \arctan (p/d_2*\pi) = \arctan 1.5/ 7.5*\pi = 3.6$ degree,

and $\rho' = \arctan \mu/\cos (30°/2) = 6.6$ degree.

According to the calculation result the thread is a self-locking one as $\alpha < \rho'$. Therefore, the lifted load keeps its position on the threaded shaft if the required torque is stopped to be applied and it means that the Car Jack is safe there is no chance for the car body to move downwards when the driving torque is not applied.

The tensile stress in the threaded shaft at the thread is:

$\mathbf{\sigma= F/A = 4905/78.5 = 62.5\ MPa}$

Torsional shear stress in the threaded portion is:

$\mathbf{\tau= T/ K_p= 3313.5/ 196.25 = 16.8\ MPa}$

According to the energy (or HMH= Muber- Mises- Hencky) theorem the equivalent stress

(σ_{eq}) is:

$\mathbf{\sigma_{eq}=\sqrt{\sigma^2 + 3 * \tau^2} = \sqrt{162.4^2 + 3 * 16.8^2} = 68.9\ MPa.}$

B. Nut

During loading in between the threaded shaft and the nut surface stress will be developed and its magnitude is:

$\mathbf{P= F/ A_p}$, Where:

$\mathbf{A_p = [(d_1^2- D_1^2)*\pi/4]*i,}$ for the number of connecting threads "i" and the thread number can be assumed or calculated from the selected mut height m and the pitch of the thread p, so i = m/P.

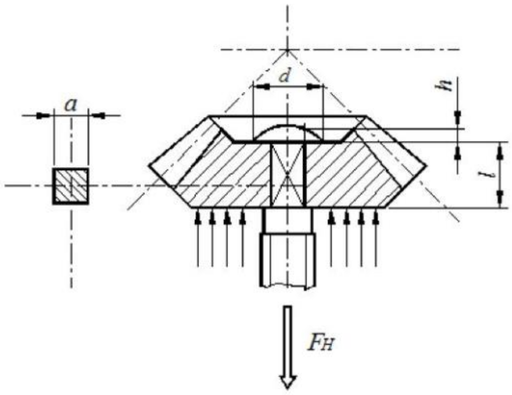
As the nut material C25 E is selected a grade lower (then the threaded shaft), the yield stress for the nut material is R_e= 370 MPa (see Tab XX). The safety factor may be taken 1.5 <i<2 and selected one is z = 1.8, the permitted surface stress is:

$\mathbf{P_w = R_e/z = 370/1.8 = 205\ MPa,}$ then substitute this value to the above equation.

$\mathbf{i = F/[1/4(d_1^2- D_1^2\ )*\pi*P_w] = 4905/[1/4\ (8^2 - 6.5^2)*\pi*205 = 1.4}$, this value is valid for the surface pressure.

2.4 Dimensions of the Threaded Spindle Support

To transmit the torque between the threaded shaft and the bevel gear a torque transmitting joint should be formed which can be for example a square cross section polygon joint, the circle drawn around the square is taken d_2=7.5 mm, the mean diameter of the thread selected.



According to the Fig.2.4, the magnitude of the sheared cross section is about:

A_τ= 4ah.

The average shear stress in the cross section is:

$\tau = F/A_\tau = F/4ah$

Expressing the required riveted head height:

$h_{min} = F/ 4a\tau_W$

The permitted working shear stress is taken as half of the permitted tensile (normal) stress of the selected material. C30E steel:

$\tau_W = \sigma_W/2 = 230/2 = 115$ MPa.

Substituting into the equation above:

$h_{min} = F/4a\tau_W = 4905/4*5.3*1 = 2.01$ mm

Where a = 5,3 mm is the square side can be drawn within the circle d_2=7.5 mm.

2.5 Dimensions of the Torque Connection of a Threaded Spindle

The torque transmission joint (Fig. 2.4) is a polygon shaft joint. The transmittable torque by the joint is:

$T= 0.5*(D- d) *l*i*p_w [(D+d)/4]$

Where:

D = 7,25 mm is the circle diameter which can be drawn in the polygon cross section.

d = 5,4 mm is the circle diameter can be drawn in the polygon cross section.

l = The length of the polygon shaft end.

i = 4 the number of the polygon shaft sides.

The required shaft end length-from the above equation is:

$l_{min} = 8T/ (D^2-d^2)*i*p_w$.

In case of the carbon steel shaft and a selected temper steel material for the bevel gear (symbols: EN-GJMW-360-12 and MSz EN 1562, having the accepted yield strength at ε=0,2% $R_{p0.2}$= 170 MPa) the bevel gear material has lower strength. Therefore, the calculation is continued with the parameters of the bevel gear material. So $R_{p0.2}$= 170 MPa, at elongation A%= 15% for the brake (ultimate) strength. The permitted surface strength is:

$P_w = R_{p0.2}/z$ =170 MPa /1.5 = 113.3 MPa

If z =1.5 is selected as the required safety factor.

The required length of the joint is:

$l_{min} = 8T/ (D_2-d_2) *i*p_w = 8*3313.5/(7.25^2-5.4^2)*4*113.3=2.5$ mm.

The result obtained is very small and it permits to use only one face, which needs four-time longer length for the polygon shaft end. Thus:

$l_{min,i}$ = 4*2.52 = 10 mm, this value is valid and accepted for the design.

2.6 Selection of the Thrust Rolling Bearing

In order to reduce the friction loss between the thrusting (lower surface) of the bevel gear and upper surface of the U-shape bracket - due the acting load F in the threaded shaft - a thrust ball bearing is applied.

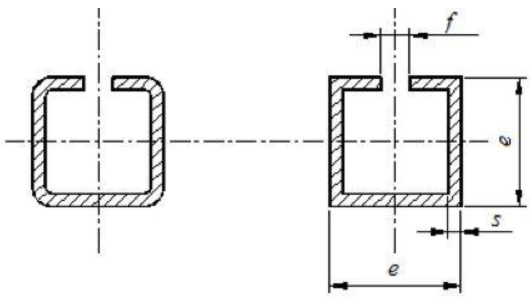
In this Column Car Jack, a thrust bearing number BA8 is selected, where the static load rating:

C_0 = 3800N, which provides a safety factor of

So = C_0/F= 3800/ 4905 = 0.77.

<u>This safety factor is all right</u> as it is much bigger than the recommended minimum one (S_0 = 0.5) by the manufacturer.

2.7 Dimensions of Strength of the Column



Real cross-section Simplified c.s. for calculation

The cross section consists of rectangles having the side lengths a and b and the centre (point) of gravity is y measured from the lower edge (Fig. 2.6). Notations concerned: (A) is the area of cross section, the coordinate of the centre of gravity is (y_s) and the moment of inertia (I_{x0}) values for (x_s) axis - most unfavourable for buckling- calculated in tabulated form are given in Table.2.1.

I_0= ab^3/12, and A=ab.

Table 2.7: Calculation of cross-sectional characteristics:

a	b	y	I_0	A	$A \cdot y$	y_s	$y-y_s$	$A \cdot (y-y_s)^2$
4	20	10	2666,67	80	800		0,585799	27,45282
16	2	1	10,67	32	32		-8,4142	2265,561
11,6	2	19	7,73	23,2	440,8		9,585799	2131,791
			2685,07	135,2	1272,8	9,414201		4424,805

The sectional modulus calculated for the axis at the centre of gravity:

$I = \Sigma\ [\ I_0 + A\ (y-y_s)^2] = 2685 + 4425 = 7110$ mm.

For determining the column length, the position points of lifting height and the minimum, position point height from the ground should be known. From the given data:

H = 150 mm and the total height of lifting from the ground is:

$H+\Delta H = 420$ mm.

where $\Delta H = 270$ mm is the way of load travelling.

Considering the structural characteristics, i.e. adding the height of the leg and the upper distance (H_{leg} +H_{dist} are together about 30mm), the total length of the column is:

1= 420 mm.

Checking the column for buckling the slenderness ratio:

$\boldsymbol{\lambda = l/i_x = 420/7.25 = 57.93}$

where the radius of gyration is:

$$\mathbf{i_x} = \sqrt{\frac{I}{A}} = \sqrt{\frac{7110}{135.2}} = \mathbf{7.25\ mm}$$

Here, the column is a compressed element under buckling and the slenderness ratio λ = 57.93 makes this case to the buckling case calculated straight line formula. Checking for buckling depends on the value of slenderness ratio. If $\lambda < 60$, then buckling calculation is neglected and only the yield strength is considered for compression. When $60<\lambda<100$ then the buckling calculation is done according to the straight-line formulae. If $\lambda > 100$, then the buckling calculation goes on with the Euler (hyperbola) equation.

Remark: the column is considered as hinge supported both ends and the bending moment provided by the locking device due to outer loading is neglected. Due to this neglecting the safety factor for buckling is advised to be at least z = 5. (In general cases of engineering, the safety factor z bigger than 3 and smaller or equal to 5.)

From the above calculation, only the yield strength is considered for compression. Applied material for column is S275N which provides 275 N/mm^2 yield strength. The factor of safety (z) will be 3.5.

$\boldsymbol{\sigma_W = R_{eh}/z = 275 / 3.5 = 78.57\ N/mm^2}$

$\boldsymbol{\sigma = F/A = 4905 / 80 = 61.31\ N/mm^2}$

As you can see, $\sigma_W > \sigma$, thus the column is all right for buckling.

References

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3. www.seeger-orbis.com