Problem Description:

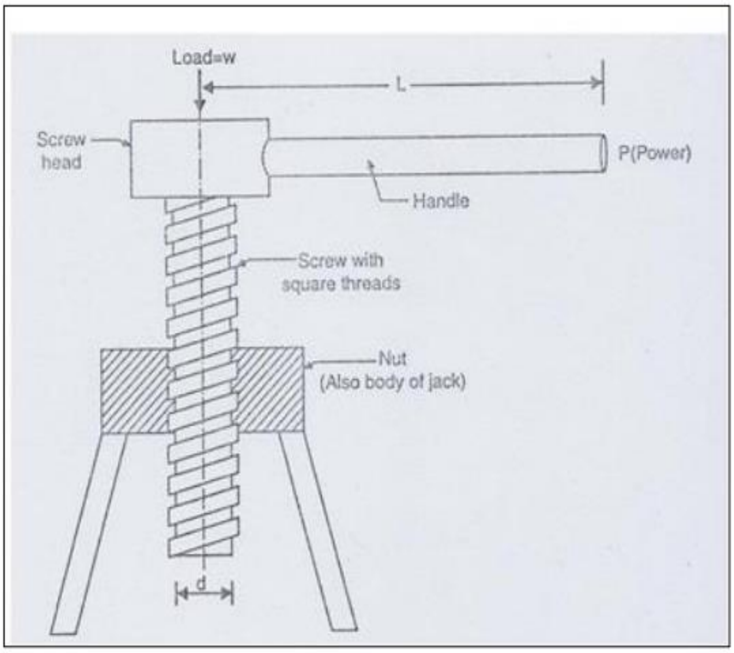


Figure 1. A simple screw jack

1-Definition and Mission of Screwjack

Screwjack: a screw-operated jack for lifting, exerting pressure, or adjusting position (as of a machine part).

Even by looking the definition we can figure out the mission of the screwjack.We are using screwjack in order to lift or adjusting the position of any kind of object has.So how this happens ?

The force which we imply on the handle creates a moment and by contribution of this moment a rotating motion occurs.For now these are the general knowledgements that we have to know about screwjack.

2-Assumptions and Values

S = Safety Factor = 2.5 ,

$\emptyset =$ angle caused by friction between the nut and screw

$\mu_s =$ friction coefficitient between nut and screw $= 0.15$

$\tan^{-1}(u_s) = \emptyset \cong 8.5308°$,

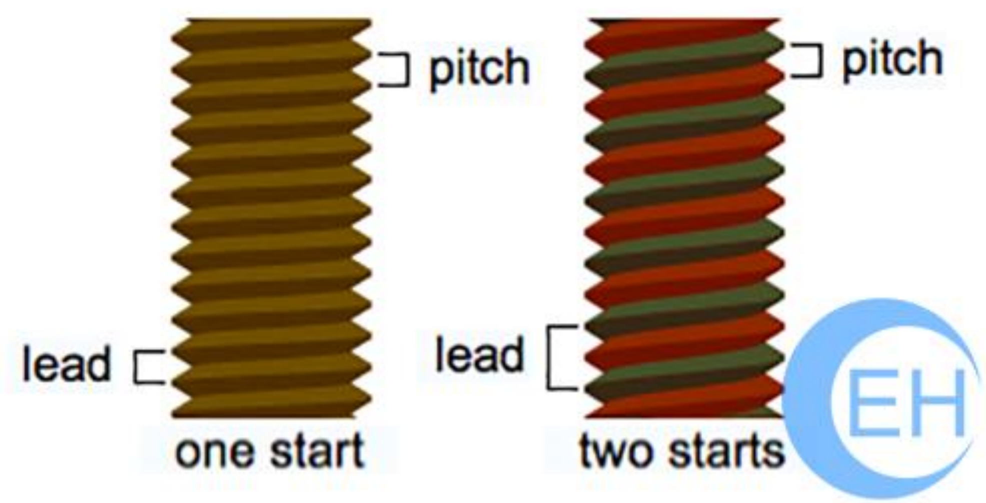
$p =$ pitch of screw thread $= 3mm$

$\alpha =$ helix angle of screw(degree) $\alpha = \tan^{-1}(\frac{\rho}{\pi d_m})$, $d_m = (d_c + d_0)/2$,

*Lead and pitch for two screw threads; one with one start and one with two starts.

Lead and pitch are closely related concepts. They can be confused because they are the same for most screws. Lead is the distance along the screw's axis that is covered by one complete rotation of the screw (360°). Pitch is the distance from the crest of one thread to the next. Because the vast majority of screw threadforms are single-start threadforms, their lead and pitch are the same. Single-start means that there is only one "ridge" wrapped around the cylinder of the screw's body. Each time that the screw's body rotates one turn (360°), it has advanced axially by the width of one ridge. "Double-start" means that there are two "ridges" wrapped around the cylinder of the screw's body. Each time that the screw's body rotates one turn (360°), it has advanced axially by the width of two ridges.

Screw Lead, Screw pitch, and Thread starts



T_1 = Torque required to rotate the screw $= W \tan(\emptyset + \alpha)\dfrac{d_m}{2}$

d_c = core diameter of screw → d_0 = outer diameter of screw $= d_c + p$

W = longitudinal force $= (1+8) * 1000 = 9000\ \mathrm{N}$

The screw part is made of St 60-2 steel , the nut part is made of St 50-2 steel and the handle is made of St 44-2 steel (Because screw is the most important material for this system.As for other parts it is assumed that they are made of St 37-2 material.

*We will assume that limits for tensile and compression stresses are equal.And also we will not modify yield stresses because we're assuming that diameters are small enough.

*Our analysis about safety of screwjack is terminated.This system is theoretically safe but it’s significant that to consider when it comes manufacturing there are some standarts that we have to obey.Since these materials are not standart elements i have disregarded the manufacturing standarts in this project.

3-Design of Screw

1) Design of Screw:

Direct compressive stresses are induced in the screw due to application of a load ‘W’. Core diameter of screw can be determine,

$$\sigma_c = \frac{W}{\frac{\pi}{4} \times d_c^2}$$

Torsional shear stress induced in the screw,

$$\tau = \frac{16 \times T_1}{\pi \times d_c^3}$$

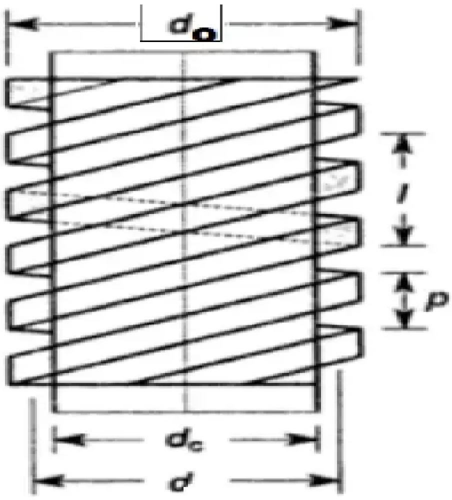
Torque required to rotate the screw,

$$T_1 = W \times \tan(\emptyset + \alpha) \times \frac{d_m}{2}$$

Check whether principal stresses less than the permissible stresses for safe dimensions of screw,

$$\tau_{max} = \frac{1}{2} \times \sqrt{(\sigma_c^2) + 4\tau^2}$$

$$\sigma_{c\,max} = \frac{\sigma_c}{2} + \frac{1}{2} \times \sqrt{(\sigma_c^2) + 4\tau^2}$$



do= Nominal diameter {Outside diameter of screw}
dc= Core diameter of screw
d= Mean Diameter of the screw
p= Pitch of the screw.

1) Mean diameter of Screw (d)

Mean diameter

$d = d_o - \frac{p}{2}$,

core diameter

$d_c = d_0 - p$

Befor
e doing the calculations firstly we have to devise our method about appropriate safety calculations.So for the inside part of screw we have 3 conditions:

1-Maximum shear stress , 2-Maximum tensile stress , 3-Analytical Method

We will start from analytical method and get a safe core diameter value.

(it's essential to start from the analytical method.First we have to determine safe core diameter variable according to the analytical method then test it for maximum stress theories.Let's start it.)

N/mm^2

MALZEME CİNSİ		STATİK (d < 16 mm)				DİNAMİK		
		Kopma Dayanımı	Akma Dayanımı			Yorulma Dayanımı		
			Çekme	Eğilme	Burulma	Çekme	Eğilme	Burulma
		σ_K	$\sigma_{AK(ç)}$	$\sigma_{AK(e)}$	τ_{AK}	$\sigma_{D(ç)}$	$\sigma_{D(e)}$	τ_D
DIN17100	St 37-2	370	240	330	140	150	170	100
Genel	St 44-2	440	280	380	160	180	200	120
Konstr.	St 50-2	500	300	410	170	210	240	140
Çelik-	St 60-2	600	340	470	190	250	280	160
leri[1]	St 70-2	700	370	510	210	300	330	190

(1) : $d > 16$ için, Akma Dayanımı değerleri $\approx \sqrt[3]{K_b}$ ile çarpılmalıdır. (Kopma Dayanımı değerleri değişmez.)

(2) : $d > 16$ için, Kopma Dayanımı değerleri $\approx \sqrt{K_b}$ ile,
Akma Dayanımı değerleri $\approx K_b$ ile çarpılmalıdır.

(3) : $d > 16$ için, Kopma ve Akma Dayanımı değerleri $\approx K_b$ ile çarpılmalıdır.

(4) : "B" grubunun dayanım değerleri, yaklaşık olarak, A ve C grubunun ortalamasıdır.

Not : K_b ile çarpılarak elde edilen değerler, sonu 0 (sıfır) olan tamsayılara yuvarlatılır.

Bileşik gerilmeler halinde statik ve dinamik gerilmeler ayrı ayrı ele alınır. Statik gerilmeleri σ_o , dinamik gerilmeleri σ_g temsil eder. "Ş.D.E" hipotezine göre, σ_o ve σ_g de şöyle hesaplanır.

(Not : Mevcut olmayan terimlerin yerine "sıfır" konulmalıdır.)

$$(\sigma_v)_{sta} = \sigma_o = \sqrt{\sigma_{sta}^2 + 3 \cdot \tau_{sta}^2}$$

$$(\sigma_v)_{din} = \sigma_g = \sqrt{\sigma_{din}^2 + 3 \cdot \tau_{din}^2}$$

C-3) ANALİTİK YÖNTEM ($\sigma_o \neq 0$ ise tercih edilir.)

Bu yöntem, grafik yöntemin analitik hesaba dönüştürülmüş şeklidir. Bu yöntemde hesaplamalar için, Bölüm 1.3.9 (B-3)' e göre, aşağıda verilen dizayn bağıntılarının oluşturulması gerekir.

$$1°)\quad \frac{\sigma_g}{\sigma_D^* / S} + \frac{\sigma_o}{\sigma_K / S} \leq 1 \quad , \quad 2°)\quad \frac{\sigma_{max}}{\sigma_{AK} / S} \leq 1$$

Ancak millerde burulma ve eğilme momentleri birbiriyle ilişkilidir. Dolayısıyla ortalama gerilme, gerilme genliğinden bağımsız olarak, yüksek değerler alamaz. Yani pratikte akma doğrusu için yazılan 2. bağıntı kullanılmaz

Buna göre bu yöntemde hesaplar şöyle yapılır.

1- Kontrol Hesabı : Bu hesapta, milin emniyetli olması için, dizayn bağıntılarının 1. si sağlanmalıdır. Yani,

$$\frac{\sigma_g}{\sigma_D^* / S} + \frac{\sigma_o}{\sigma_K / S} \leq 1$$

olmalı veya emniyet katsayısına göre düzenleme yapılırsa, şu bağıntı sağlanmalıdır.

$$S_{mev} = \frac{1}{\dfrac{\sigma_g}{\sigma_D^*} + \dfrac{\sigma_o}{\sigma_K}} \geq S_{değ}$$

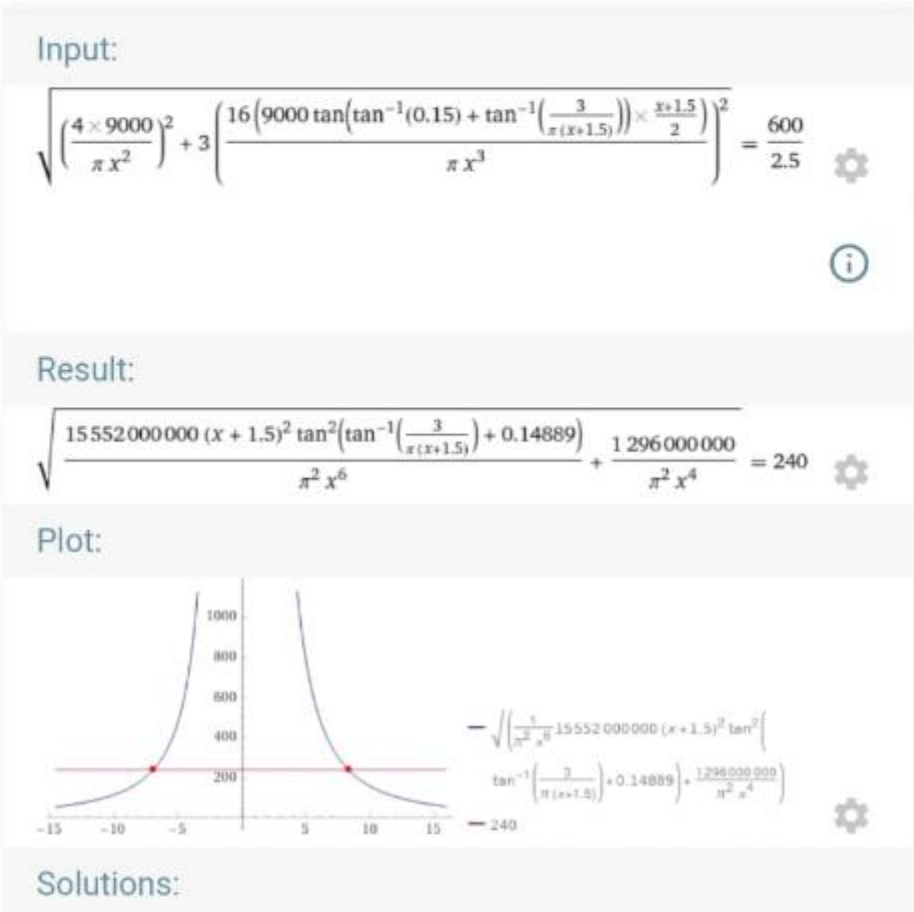
2- Boyutlandırma Hesabı : Bu hesapta mildeki minimum çap belirleneceği için, 1. dizayn bağıntısı çapa bağlı olarak yazılır ve buradan çap değeri çekilirse, aşağıdaki sonuca ulaşılır.

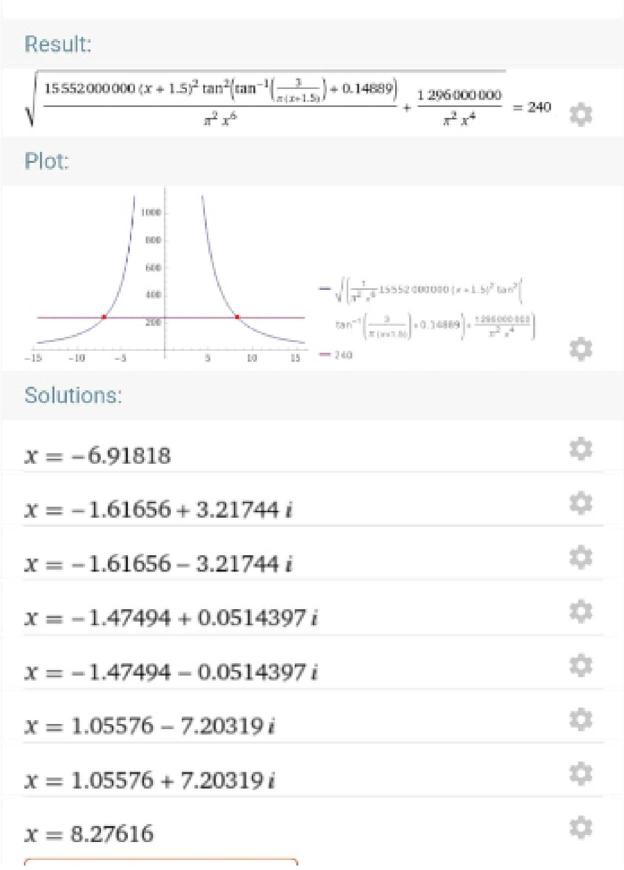
$$d \geq \sqrt[3]{\frac{32}{\pi} \cdot S_{değ} \left(\frac{M_e}{\sigma_D^*} + \frac{\sqrt{3}}{2} \cdot \frac{M_b}{\sigma_K} \right)}$$

We don't need to modify the fracture stress.Because our material is in group 1.And since all stresses are static stresses we don't have to calculate σ_D^*.

Because that part of the equation will be equal to 0.

$\frac{\sigma_g}{\sigma_D^* / S} + \frac{\sigma_0}{\sigma_K / S} \leq 1$ When we simplify this formula as a function of core diameter we get





obviously there are more than one root but we are interested in real and positive roots so safe core diameter is $8.27616 \cong 12$.

$$d_0 = 12 + 3 = 15$$

$$d_m = \frac{12+15}{2} = 13.5$$

$$\alpha = \tan^{-1}\left(\frac{3}{\pi 13.5}\right) \cong 4.0461°$$

$$\tau = \frac{16T_1}{\pi d_c^3} = \frac{16x13553.520}{\pi 12^3} \cong \frac{40N}{mm^2} < 76\ (\frac{190}{2.5}\ \text{safe torsion shear stress})$$

$$T_1 = 9000\tan(8.5308° + 4.0461°)\frac{13.5}{2} = 13553.520\ \text{Nmm}$$

$$\sigma_c = \frac{4W}{\pi d_c^2} = \frac{4x9000}{\pi 12^2} \cong 79.58\ \text{N/mm^2}$$

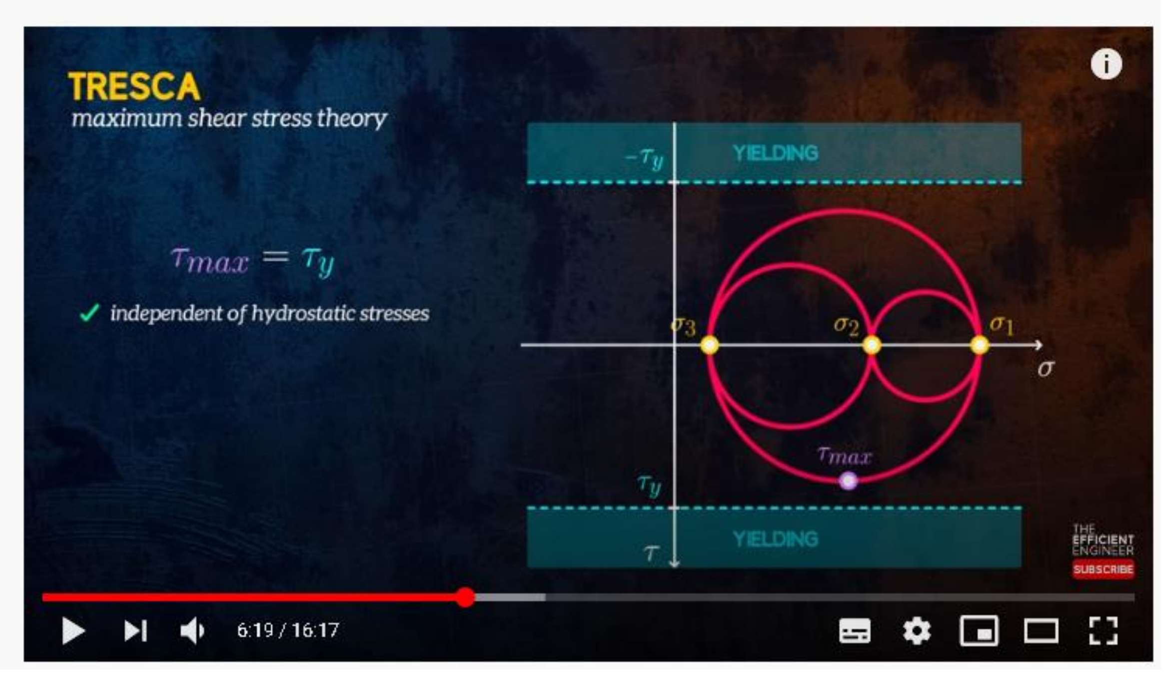
$$\sigma_0 = \sqrt{(\sigma_c^2 + 3\tau^2)} = \sqrt{(79.58^2 + 3(40)^2)} \cong 105.51\ \text{N/mm^2}$$

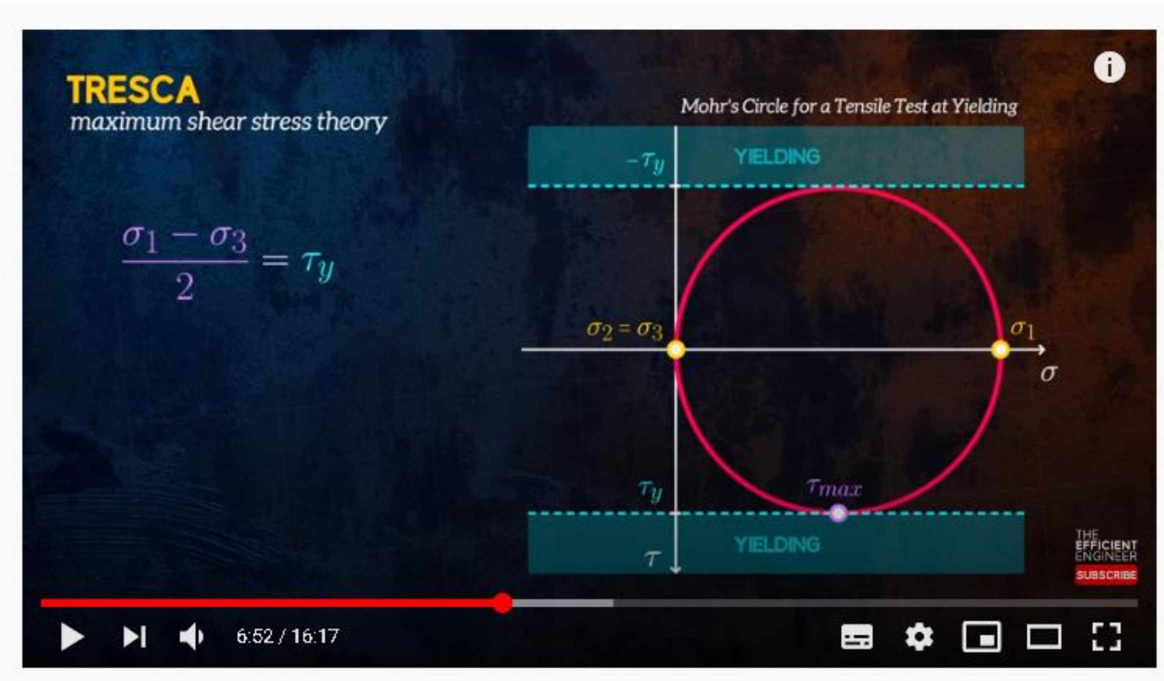
$\frac{\sigma_g}{\sigma_D^*/S} + \frac{\sigma_0}{\sigma_K/S} = 0 + \frac{105.51}{600/3.5} \cong 0.615 < 1\ \textit{so safety checked}$.And also:

$$S_{current} = \frac{1}{\sigma_g/\sigma_D^* + \sigma_0/\sigma_K} = 0 + \frac{1}{105.51/600} = \cong 5.68\ > 2.5$$

Curent safety factor is greater then the assumed one so thus we controlled first formula.

As we see we don't need to modify yield stresses now we will calculate maximum stresses.





Tensile yield stress from the material table is represented as sigma1 in this example.And the yield shear stress equals to half of this value.(sigma2 and sigma 3 = 0.)

$$\tau_{\max} = 0.5x\sqrt{(79.58^2} + 4x40^2) \cong 56.42\text{N/mm\^{}2}$$

$$\sigma_{max} = \frac{79.58}{2} + 0.5x\sqrt{(79.58^2} + 4x40^2) \cong 96.21\text{N/mm\^{}2}$$

$$\tau_{max} = \frac{1}{2} \times \sqrt{(\sigma_c^2) + 4\tau^2}$$

$$\sigma_{c\,max} = \frac{\sigma_c}{2} + \frac{1}{2} \times \sqrt{(\sigma_c^2) + 4\tau^2}$$

now we will compare these maximum stresses with the yield stresses.

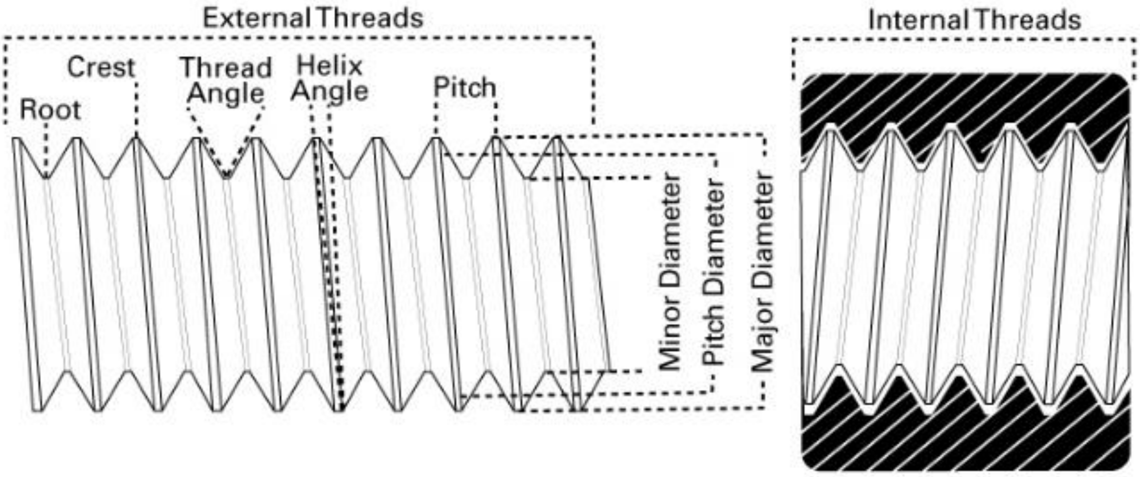
$$\frac{\sigma_{\text{tensileyield}}}{\text{S}} = \frac{340}{2.5} = 136 \ > 96.21 \ safe$$

$$\frac{\tau_{\text{yield}}}{\text{S}} = \frac{340}{2x2.5} = 68 > 56.42 \ safe$$

4-Design of Nut

This part is a little bit confusing.We will try to understand stresses by using figures.

First let's start with bearing stress.Outside of screw and inside of nut are pressing each other during the load.So this situation creates a stress and it is called as bearing stress.



white trimmed black part is nut.In every single pitch screw is applying a pressure on that nut part.And the circumferance of the same part occurs a shear stress on the lateral area of nut.

2) Design of Nut:

The threads of nut and screw are engaged to each oth there is relative motion between them under compr load, hence nut may fails for the bearing pressure,

$$p_b = \frac{W}{\frac{\pi}{4} \times (d_o^2 - d_c^2) \times n}$$

$$\tau_{nut} = \frac{W}{\pi \times d_o \times n \times t}$$

$$p_b = \frac{9000x4}{\pi(15^2 - 12^2)n}, \tau_{nut} = \frac{9000}{\pi 15n1.5}$$

n = number of threads engaged with the nut.Now we will execute the same maximum stresses principle.

t = thickness of screw = p/2

$$p_b = \frac{9000x4}{\pi(15^2-12^2)n} \leq \frac{300}{2.5} \rightarrow n \geq 1.179, \tau_{nut} = \frac{9000}{\pi 15n1.5} \leq \frac{300}{2.5x2} \rightarrow n \geq 2.12$$

*Material is subjected to these stresses separately

it seems that to take n as 5 is a proper choice for us.

The height of nut can be calculated by,

$$H = n \times p$$

Height of the nut =5x3=15 mm.

and $p_b = \frac{9000x4}{\pi(15^2 - 12^2)5} \cong \frac{28.30N}{mm^2} <$ Safe longitudional stress $(\frac{300}{2.5})$

$\tau_{nuttransverse} = \frac{9000}{\pi 15x5x1.5} \cong \frac{25.465N}{mm^2} <$ Safe shear stress $(\frac{300}{5})$

Remember that we have used the analytical method in order to assign a proper core diameter.Normally when maximum stresses are in the safe region our current static total stresses are lower than the fracture stress.

Now we will calculate the transverse shear stress which is occured by the rubbing reaction between helixes and inside part of screw.

$$\tau_{screw} = \frac{W}{\pi \times d_c \times n \times t}$$

t = thickness of screw and it is equal to

p/2 =1.5mm

$\tau_{screwtransverse} = \frac{9000}{\pi x12x5x1.5} \cong 31.83 \frac{N}{mm^2}$

$< Safe\ shear\ stres \left(\frac{340}{5}\right)$, Now we will determine a safe inside diameter for the nut.Firstly let's talk about the stresses.

1-

Inner diameter (D_1) fails under tensile stress,

$$\sigma_t = \frac{W}{\frac{\pi}{4} \times (D_1^2 - d_o^2)}$$

$\rightarrow \frac{4x9000}{\pi(D_1^2 - 15^2)} \leq \frac{300}{2.5} \rightarrow D_1 \geq 17.90.$ So we can assign D_1 as 20mm.

2- Outer diameter (D_2) fails under crushing stress,

$$\sigma_{ck} = \frac{W}{\frac{\pi}{4} \times (D_2^2 - D_1^2)}$$

$\rightarrow \frac{4x9000}{\pi(D_2^2-18^2)} \leq \frac{300}{5} \rightarrow$ D_2 ≥22.7. Same again.We can assign D_2 as 25mm.

3 - We also have a shear stress in the inside lateral area of nut

Thickness of nut collar (t_1) fails under shear stress,

$$\tau = \frac{W}{\pi \times D_1 \times t_1}$$

$$\rightarrow \frac{9000}{\pi 20 t_1} = \frac{300}{5} \rightarrow t_1 \geq 2.39. \text{ We can assume it as 5mm.}$$

$* \; t_1$ is representing the height of the nut′s outer part which is a member of jack

5-Design of Handle and Screw Head

It's already mentioned that T_1 = Torque required to rotate the screw but what about T_2?

Screwhead is placed on the cup(which is a platform where the load stands on it) of the screwjack and thus prevent the load to rotate.But it's also necessary to overcome the friction between handle and cup.So T_2 is the required torque to overcome this friction.

3) Diameter of screw head:

$$D_3 = 1.75 d_o$$

4) Diameter of screw head collar pin

$$D_4 = \frac{D_3}{4}$$

$$T_2 = \frac{2}{3} \times \mu_1 \times W \times \frac{(R_3^3 - R_4^3)}{(R_3^2 - R_4^2)} \quad \text{(Uniform Pressure Condition)}$$

5) Length of Handle:

Total torque transmitted by the hanc

$$T_2 = \mu_1 \times W \times \frac{(R_3 + R_4)}{2} \quad \text{(Uniform Wear Condition)}$$

$$T = T_1 + T_2$$

$$P \times L = T_1 + T_2$$

*Let's say $\mu_1 = 0.10$

$$T_1 = W \times \tan(\varnothing + \alpha) \times \frac{d_m}{2}$$

$D_3 = 1.75x15 = 26.25mm\, , D_4 = \frac{26.25}{4} = 6.5625mm$

Mean radius R = $((26.25/2)^3 - (6.5625/2)^3/((26.25/2)^2 - (6.5625/2)^2$

= 13.78125 mm

D_3 assumed as 28mm and D_4 assumed as 8mm.

$T_2 = 0.10X9000((26.25/2)^3 - (6.5625/2)^3)/((26.25/2)^2 - (6.5625/2)^2)$

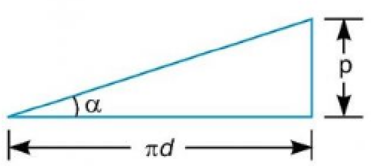
$T_2 = 12403.125$ Nmm $\rightarrow T_{total} = 12403.125 + 13553.520$

$= 25956.645$ Nmm

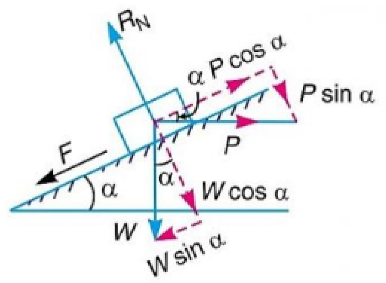
At this part we have to reconsider screw and determine the P and handle force F respectively.

The torque required to lift the load by a Screw Jack

If one complete turn of a screw thread by imagined to be unwound, from the body of the screw and developed, it will form an inclined plane as shown in below figure



Development of screw



Forces acting on the screw

- p = Pitch of the screw,
- d = Mean diameter of the screw,
- α = Helix angle,
- P = Effort applied at the circumference of the screw to lift the load,
- W = Load to be lifted, and

- W = Load to be lifted, and
- $\mu = \tan \varphi$ Coefficient of friction, between the screw and nut (= $\tan \varphi$), where φ is the friction angle

From the geometry shown above, we can write

$$\tan \alpha = p/\pi d$$

Since the principle on which a screw jack works is similar to that of an inclined plane, therefore the force applied on the lever of a screw jack may be considered to be horizontal as shown above figure(right side).

Since the load is being lifted, therefore the force of friction ($F = \mu.R_N$) will act downwards. All the forces acting on the screw are shown in the above figure(right side).

Resolving the forces along the plane,

$$P \cos \alpha = W \sin \alpha + F = W \sin \alpha + \mu.R_N \text{ ---(a)}$$

and resolving the forces perpendicular to the plane,

$$R_N = P \sin \alpha + W \cos \alpha \text{ --(b)}$$

Substituting (b) (value of R_N) in equation (a),

$$P \cos \alpha = W \sin \alpha + \mu (P \sin \alpha + W \cos \alpha)$$

$$P \cos \alpha = W \sin \alpha + \mu P \sin \alpha + \mu W \cos \alpha$$

$$P \cos \alpha - \mu P \sin \alpha = W \sin \alpha + \mu W \cos \alpha$$

$$P (\cos \alpha - \mu \sin \alpha) = W (\sin \alpha + \mu \cos \alpha)$$

$$P = (W (\sin \alpha + \mu \cos \alpha))/ (\cos \alpha - \mu \sin \alpha)$$

Substituting the value of $\mu = \tan \varphi$ in the above equation, we get

$$P = (W (\sin \alpha + \tan \varphi \cos \alpha))/ (\cos \alpha - \tan \varphi \sin \alpha)$$

Multiplying the numerator and denominator by cos φ,

$$P = (W (\sin \alpha \cos \varphi + \tan \varphi \cos \alpha \cos \varphi))/ (\cos \alpha \cos \varphi - \tan \varphi \sin \alpha \cos \varphi)$$

$$P = (W (\sin \alpha \cos \varphi + \sin \varphi \cos \alpha))/ (\cos \alpha \cos \varphi - \sin \varphi \sin \alpha)$$

$$P = (W (\sin (\alpha + \varphi))/ (\cos (\alpha +\varphi))$$

$$P = W \tan (\alpha+\varphi)$$

Torque required to overcome friction between the screw and nut

$$T_1 = P \times (d/2)$$

$$T_1 = W \tan (\alpha+\varphi) \times (d/2)$$

When the axial load is taken up by a thrust collar or a flat surface, as shown in above figure, so that the load does not rotate with the screw, then the torque required to overcome friction at the collar,

$$T_2 = \mu_1 \times W \times (R_1+R_2)/2$$

$$T_2 = \mu_1 \times W \times R$$

- R1 and R2 = Outside and inside radii of the collar
- R = Mean radius of the collar, and
- μ_1 = Coefficient of friction for the collar

∴ Total torque required to overcome friction (i.e. to rotate the screw),

$$T = T_1 + T_2$$

$$T = (P \times (d/2)) + (\mu_1 \times W \times R)$$

If an effort P1 is applied at the end of a lever of arm length *l*, then the total torque required to overcome friction must be equal to the torque applied at the end of the lever, i.e.

Consequently we get : $25956.645/P = L$

We can assume that an average person applies 300N force on device by hand.Then handle length can be calculated as 86.52215 mm ≅ 90 mm

Handle may fails under bending stress, diameter can determine,

$$M = P \times L = \frac{\pi}{32} \times D_H^3 \times \sigma_b$$

$$\rightarrow \left(\frac{32x25956.645}{\frac{\pi 380}{2.5}}\right)^{\left(\frac{1}{3}\right)} \leq D_H \geq 12.06 \text{ we can assume it as 13mm .}$$

then handle diameter can be determined as 15mm.(Just like we have done before bending yield stress value have been taken from the table above.)

6) Height of Screw Head:

$$H = 2 \times D$$

*D refers to 2xmean radius

H = 4x13.78125 = 55.125 ≅ 56 mm

6-Controling the Buckling of Screw

C – Flambaj (Burkulma)

Basmaya çalışan uzun çubuklarda görülen yana doğru bel verme olayına "flambaj" veya "burkulma" adı verilir. Makine elemanlarında bu olayın meydana gelmesi istenmez.

Flambajı oluşturan kuvvet, HOOKE kanunundan hareketle, EULER tarafından şöyle hesaplanmıştır.

$$F_f = \frac{\pi^2 E \cdot I}{\ell_f^2}$$

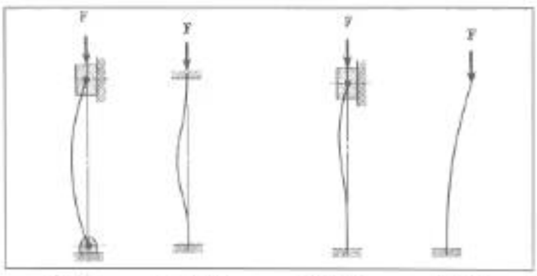
Burada, F_f : Flambajı oluşturan basma kuvvetini,

E : Malzemenin elastisite modülünü,

I : Çubuk kesitinin atalet momentini,

ℓ_f : Flambaj uzunluğunu göstermektedir.

Çubuğun mesnetleme şekline göre, " ℓ_f " nin hangi değerleri alması gerektiği, Şekil-1.19 da gösterilmiştir.



$\ell_f = \ell$	$\ell_f = 0.5\,\ell$	$\ell_f = 0,7\,\ell$	$\ell_f = 2\,\ell$	Teorik
$\ell_f = \ell$	$\ell_f \cong 0.9\,\ell$	$\ell_f \cong 0,9\,\ell$	$\ell_f = 2\,\ell$	Pratik

	Elastiklik Modülü
Metal Alaşım	*GPa*
Alüminyum	69
Pirinç	97
Bakır	110
Megnezyum	45
Nikel	207
Çelik	207
Titanyum	107
Volfram	407

$$I = \text{Inertia moment of screw section} = \frac{\pi x 12^4}{64} \cong 1.018\ mm^4$$

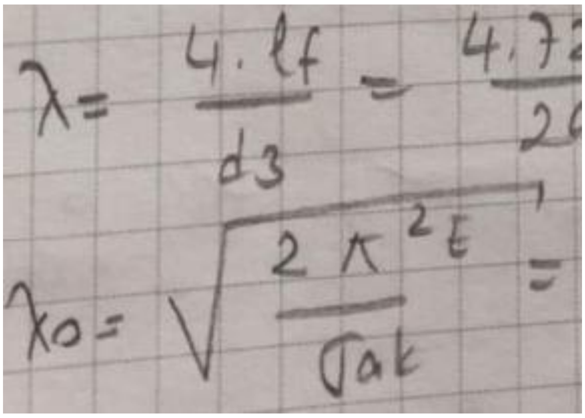
E = Elasticity Modulus , can be assumed as 207GPa for steel materials.

L = Total height of screw , l_f = effective height of screw

$$l_f \leq \left(\frac{20700x12^4x\pi^3}{64x9000}\right)^{0.5} \rightarrow l_f \leq 480.68 \text{ mm}$$

$l_f = 0.9$ L , L $\leq$ 534.1 mm we can assume it as 380 mm

Now we will compare fragility values



σ_y = yield stress of St 60 − 2 = 340N/mm^2

$\lambda = ((4x380x9)/(10x12)) = 114$

$$\lambda_0 = \left(\frac{2x\pi^2 207000}{340}\right)^{0.5} \cong 109.62$$

Since $\lambda > \lambda_0$ buckling is under control.

.

*Our analysis about safety of screwjack is terminated.This system is theoretically safe but it's significant that to consider when it comes manufacturing there are some standarts that we have to obey.Since these materials are not standart elements i have disregarded the manufacturing standarts in this project.

Now we will draw technical drawing of screwjack and it's members.But before the drawings i want to share my references.

References

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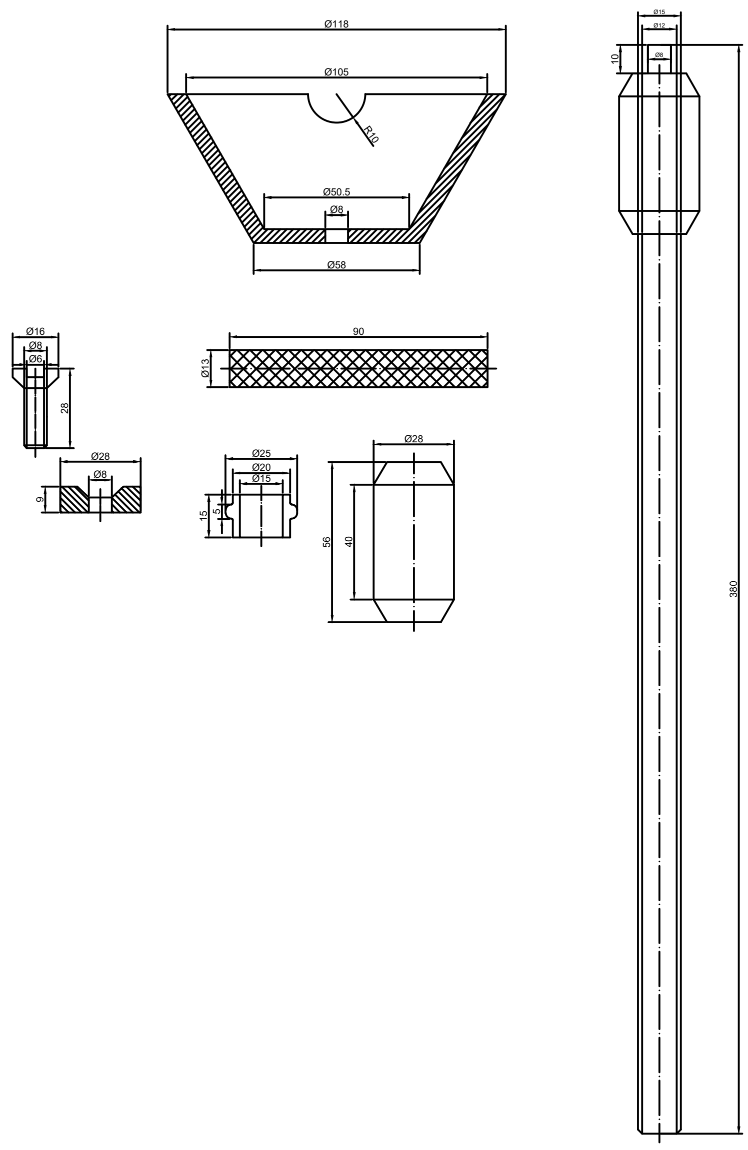
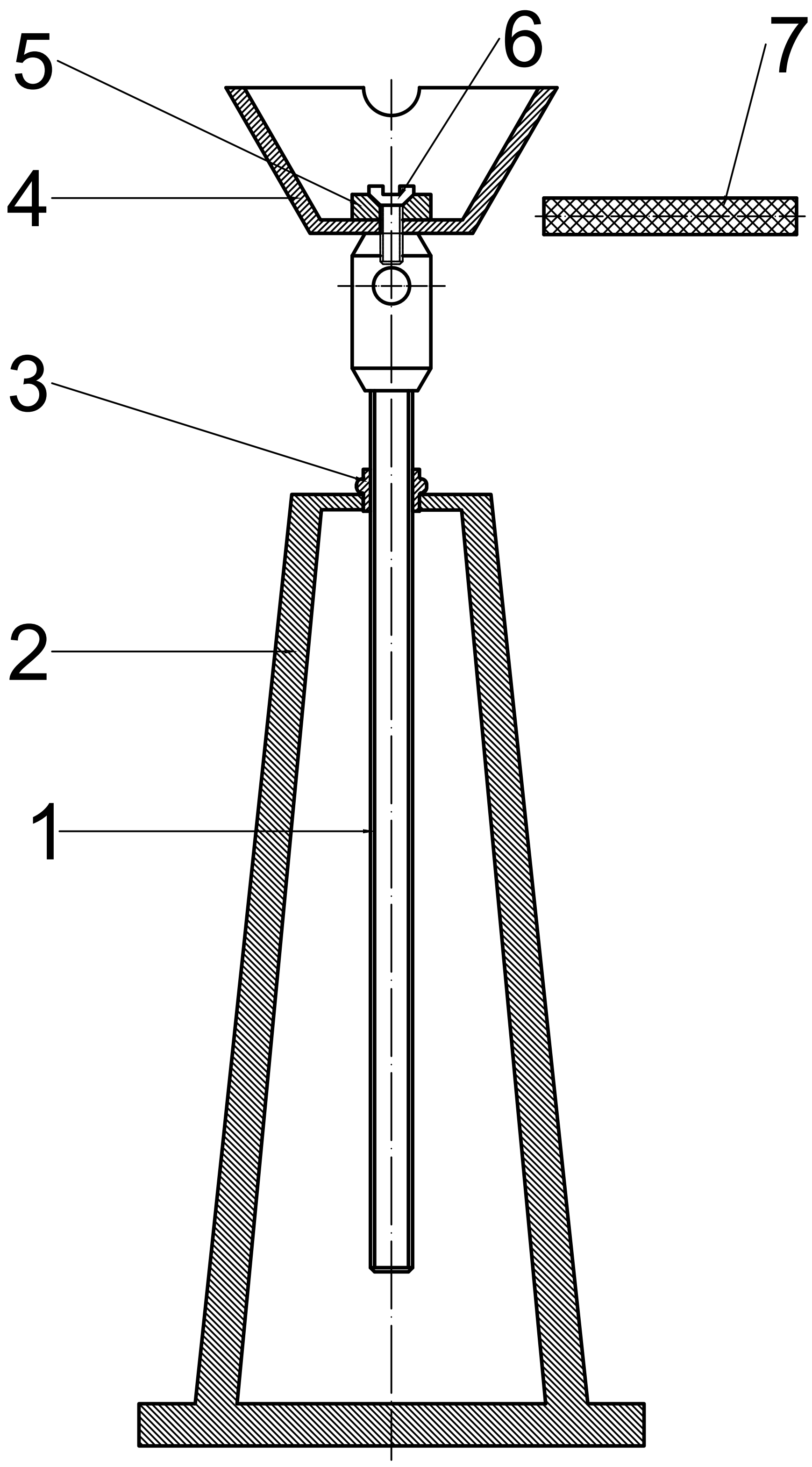


Drawer	Hakan Aktaş	Scale		
Student Number	18065037			
Date	2.01.2021			
Controller		Scale 1:1		Drawing Name
				Screw Jack Project



Assembly of Simple Screw Jack

Part Number	Quantity of Part	Name of Part	Material
1	1	Screw Spindle	St60-2
2	1	Body	St 37-2
3	1	Nut	St 50-2
4	1	Cup	St 37-2
5	1	Special Washer	St 37-2
6	1	Screw	St 37-2
7	1	Handle	St 44-2

	Name	Date
Drawer	Hakan Aktaş	2.01.2021
Controller		
	Scale 1:1	