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DESIGN OF TOGGLE JACK CONSIDERING MATERIAL SELECTION OF SCERW - NUT COMBINATION

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Abstract: Toggle jacks are simple mechanisms used to drive large loads short distances and to lift the heavy loads. It also have the advanced feature of rotating the parts about their axis where there is not enough space to move the load. The power screw design of a common Toggle jack reduces the amount of force required by the user to drive the mechanism. Most of the Toggle jacks are similar in design, consisting of eight main members out of whom four are driven by a power screw and rest of four by loading condition. In this report, a unique design of a Toggle jack is used to lift the heavy loads at the stable state with the unique condition. In toggle jack screw and nut are main components. A screw is moving part and nut is a stationary part. Both the parts always work in meshing condition. Therefore there are stresses like shear and tensile stresses induced in materials which are responsible for failure of screw. While a nut has bearing stress. Different materials of screw and nut can induce different magnitude of stresses so it is necessary to select a pair of material combination in such a way that a pair gives induced stress within safe limit. Here we have taken different material combinations of screw and nut & based on that analytical design is done for on different loading condition from 1KN to 5KN. It is also assumed that the nominal diameter of screw and material for link are same for all loads.

Keywords: critical load, max shear stress max tensile stress, screw – nut materials, Toggle jack

I. INTRODUCTION

A toggle jack is a device which lifts heavy equipment. The most common form is a car jack, floor jack or garage jack which lifts vehicles so that maintenance can be performed. Car jacks usually use toggle advantage to allow a human to lift a vehicle by manual force alone. More powerful jacks use hydraulic power to provide more lift over greater distances. Toggle jacks are usually rated for maximum lifting capacity.

- There is a one screw in the toggle jack which is rotating.
- There are two nuts which is fixed.
- There are four links connected to both nuts and eight pins to fix all links.
- There are two rings at both ends of the screw.
- There is a one platform which is connected to the upper two links for put load.

Working of toggle jack

- The jack can be raised and lowered with a metal bar that is inserted into the jack.
- The operator turns the bar with his hands in a clockwise direction for makes it go up.
- When the screw lifts the load on the platform which placed above will also be raised.
- The bar is turned until the jack is raised to the level needed.
- To lower the jack the bar is turned in the opposite direction.

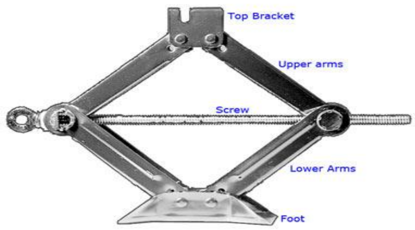

FIG. 1 Toggle jack

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II. FORMULATION OF DESIGN

A. DESIGN OF SCREW

The maximum load on the screw is when the jack is in the bottom most position. In this position of link CD is shown in fig.3.1

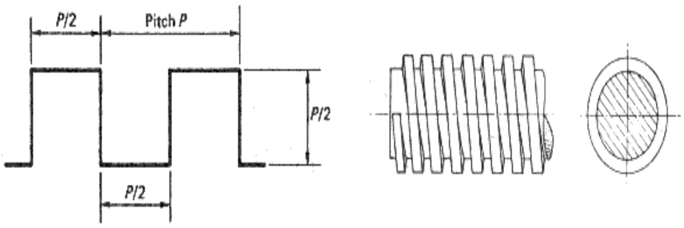

FIG. 2 Screw

- cosΘ=100-15/110 =39.4
- Pull in the screw, $F = \frac{W/2}{tan\theta}$

Where, F= force exerted on screw
W= load

- Total force in the screw due to both the nut, P1 = 2F

Where, P1= total force

- For the screw to be safe in tension,
  $dc=\sqrt{\frac{P1\times 4}{\pi \times \sigma t}}$
  where, dc= core diameter
  σt= tensile stress
- Now, we adopt pitch= 6 mm
- Outside diameter of screw,
  do= dc+p
- Mean diameter of screw,
  $dm=\frac{1}{2}(dc+do)$
- Helix angle,
  $\alpha = \tan^{-1}(\frac{p}{\pi dm})$
- Angle of friction,
  $\emptyset = \tan^{-1}\mu$
  Where, μ= co-efficient of friction
- Effort required to rotate the screw,
  $P = P1\times\tan(\alpha+\emptyset)$
  Where, P1= total force
- Torque, T=P × dm/2
- Torsional shear stress in screw,

  $\tau=\frac{16T}{\pi dc^3}$, Where, T= torque

- Direct tensile stress in screw,
  $\sigma t=\frac{4p1}{\pi dc^2}$, where, p1= total force
- Maximum principal stress in screw,
  $\sigma 1=\frac{\sigma t}{2}+\frac{1}{2}\sqrt{\sigma t^2+4\tau^2}$
  which should be less than 100 MPa, hence safe.

- Maximum shear stress,
  $\tau_{max}=\frac{1}{2}\sqrt{\sigma t^2+4\tau^2}$

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which should be less than 50 MPa, hence safe.

B. DESIGN OF NUT

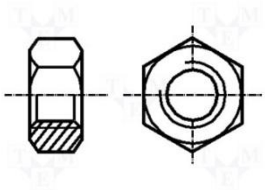

FIG. 3 Nut

- Let n= number of threads on nut.
  Designing the nut in bearing of threads,
- Thickness of threads, t = p/2
- σb=$\frac{P1}{\pi dm \times t \times n}$
  We adopt n=4
- Height of nut, h= np
  To prevent movement of nuts beyond 200 mm, rings of 8 mm thickness are provided on the screw on both sides and fixed by set screws.
  Length of screw=200+h+2×thickness of rings+30 mm for spanner
- Length of spanner:
  Let the operator apply a force of 50 N to the end of a spanner l mm long.
  L=T/50

C. DESIGN OF PINS IN NUTS

- The pins are in double shear. If dp is the diameter of pins then
  dp=$\sqrt{\frac{4F}{2\pi\tau}}$
- Diameter of pin head=1.5 dp
  Thickness of pin head is taken as 3 mm.
  Split pins are used to keep the pins in position.

D. DESIGN OF LINKS

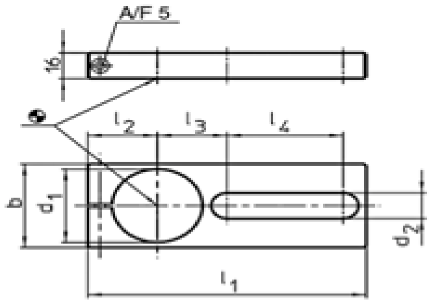

FIG. 4 Link

- Load on each link = F/2
  Let b1= width of link and t1= thickness of link
  Assuming b1=3t1
- Area of cross-section of link,
  A=$3t_1^2$
- Moment of inertia,
  $I_{yy}=\frac{t1b1^3}{12}$
- Least radius of gyration,
  $k^2_{yy}=\frac{I_{yy}}{A}$

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- For buckling of links in the vertical plane, the ends are considered hinged. Therefore, using Rankine-Gordon formula,
  For critical load,
  $P_{cr} = \frac{S_{yc} \times A}{1+a\left[\frac{l_e}{K}\right]^2}$
  For design load, P_{cr} =F.O.S×F
  Critical load should be more than design load so design is safe.

III. DESIGN CALCULATION

Calculation for Medium carbon steel (C35Mn75) - Phos.bronze material:
<u>Given data for sample calculation:</u>
Load= 3 KN
For Medium carbon steel (C35Mn75)(screw)
σ t=100 MPa, σs=60 MPa
For Phos. Bronze(nut)
σ t=50 MPa, σs=40 MPa, σc=45 MPa

A. DESIGN OF SCREW

The maximum load on the screw is when the jack is in the bottom most position. In this position,
cosΘ=100-15/110
Θ=39.4

- Pull in the screw,
  $F = \frac{W/2}{tan\theta}$
  =1.826 KN
- Total force in the screw due to both the nut,
  P1=2F
  =3.652 KN
- For the screw to be safe in tension,
  dc =12 mm
- Now, we adpot pitch= 6 mm
- Outside diameter of screw,
  do= dc+p
  =12+6 =18 mm
- Mean diameter of screw,
  $dm = \frac{1}{2}(dc + do)$
  $= \frac{1}{2}(12 + 18)$ = 15 mm
- Helix angle,
  $\alpha = \tan^{-1}(\frac{p}{\pi dm})$=7.25°
- Angle of friction,
  $\emptyset = \tan^{-1}\mu$
  $= \tan^{-1} 0.2$=11.310
- Effort required to rotate the screw,
  $P = P1 \times \tan(\alpha + \emptyset)$
  $= 3.653 \times \tan(7.25 + 11.31)$ =1.226 KN
- Torque,
  T=P×dm/2
  =1.226 ×10³× (15/2) =9.19 Nm
- Tornsional shear stress in screw,
  $\tau = \frac{16T}{\pi dc^3}$
  $= \frac{16 \times 9.19}{\pi(12)^3}$ =27.12 MPa
- Direct tensile stress in screw,
  $\sigma t = \frac{4p1}{\pi dc^2} = \frac{4 \times 3.653 \times 10^3}{\pi(12)^2}$ =32.30 Mpa
- Maximum principal stress in screw,

$\sigma 1 = \frac{\sigma t}{2} + \frac{1}{2}\sqrt{\sigma t^2 + 4\tau^2}$

$= 16.15 + \frac{1}{2}\sqrt{(32.30)^2 + 4(27.12)^2} = 47.23$ MPa

- Maximum shear stress,
  $\tau_{max} = \frac{1}{2}\sqrt{\sigma t^2 + 4\tau^2}$
  $= \frac{1}{2}\sqrt{(32.30)^2 + 4(27.12)^2} = 31.56$ MPa

B. DESIGN OF NUT

- Let n= number of threads on nut.
  Designing the nut in bearing of threads,
- For the nut to be safe in tension,
  dc =12 mm
  Here, pitch, p=6
  Outside diameter=10+6=16 mm
  Mean diameter=1/2(10+16)= 13 mm
- Thickness of threads, t = p/2=3
- $\sigma b = \frac{P1}{\pi dm \times t \times n} = \frac{3.652 \times 10^3}{\pi \times 15 \times 3 \times 4} = 6.46$ MPa
- Height of nut, h= np=4×6=24 mm
  To prevent movement of nuts beyond 200 mm, rings of 8 mm thickness are provided on the screw on both sides and fixed by set screws.
- Length of screw=200+h+2×thickness of rings+30 mm for spanner =270 mm
- Length of spanner:
  Let the operator apply a force of 50 N to the end of a apanner l mm long.
  L=T/50=148.9 mm

C. DESIGN OF PINS IN NUTS

- The pins are in double shear. If dp is the diameter of pins then
  $dp = \sqrt{\frac{4F}{2\pi\tau}}$
  $= \sqrt{\frac{4 \times 1826}{2\pi(40)}} = 5.39 = 6$ mm
- Diameter of pin head=1.5 dp= 1.5×6=9 mm
  Thickness of pin head be taken as 3 mm.
  Split pins are used to keep the pins in position.

D. DESIGN OF LINKS

- Load on each link=F/2= 913 N
  Let b1= width of link and t1= thickness of link
  Assuming b1=3t1
- Area of cross-section of link,
  $A = 3{t_1}^2$
- Moment of inertia,
  $I_{yy} = \frac{t1 b1^3}{12} = 2.25 t1^4$
- Least radius of gyration,
  $k^2_{yy} = \frac{I_{yy}}{A} = \frac{2.25 t1^4}{3t1} = 0.75 t1^2$
- For buckling of links in the vertical plane, the ends are considered hinged. Therefore, using Rankine-Gordon formula,
  For critical load,
  $Pc = \frac{S_{yc} \times A}{1 + a\left[\frac{l_e}{K}\right]^2}$ Taking a=1/7500
  $= \frac{100 \times 3t1^2}{1 + (1/7500)\left[\frac{110}{0.75t1}\right]^2} = \frac{300 t1^4}{t1^2 + 2.15}$
  For design load,
  F.O.S=2
  P_{cr} =F.O.S×F

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=2×913=1826 N

$$P_{cr} = \frac{300t1^4}{t1^2 + 2.15}$$

t1=4 mm

b1=12 mm

The link is considered to be fixed ends for bulking in a plane perpendicular to the vertical plane.

Therefore, l_e=1/2=55 mm

$$P_{cr} = \frac{100\times 3t1^2}{1+1/7500\left[\frac{55}{0.75t1}\right]^2} = \frac{100\times 3\times 4^2}{1+1/7500\left[\frac{55}{0.75\times 4}\right]^2} = 4643.39 \text{ KN}$$

Critical load should be more than design load so design is safe.

IV. RESULT AND DISCUSSION

In the present work, a toggle jack is design analytically. In this jack, screw and nut are the most significant components. A screw is designed based on maximum tensile stress and maximum shear stress. For maximum load it is necessary to keep both values within limit for safe design. Nut is a stationary part in which a screw rotates. Therefore a bearing pressure is also considered. For both the components, if we take combination of different material for each pair of screw and nut so we can find best suitable material for design at maximum load. For simplicity of design, screw parameters like, core diameter, mean diameter and helix angle are taken same throughout design for all loads. Also for link design standard dimensions of link cross section are considered. For material of link, same kind of material i..e Mild steel is taken. In analytical design, we have taken different six pairs of standard materials for screw and nut. By taking each pair design is calculated for 1KN to 5KN load in step of 1KN. In analytical design, for screw maximum tensile and maximum shear stresses are checked with given value of corresponding stresses, for nut bearing stress is checked with standard value of bearing stress of material and a link is checked in critical load with design value. In the design of toggle jack different stresses are found for applying each load from 1 KN to 5 KN.

The table 1 shows overall summery of total force and torque transmitted through screw, induced stresses in screw and nut, cross section of link and comparison of critical load acting on link with design value for applying load of 1KN to 5KN. Table 2 shows Comparison of calculated and design stresses acting on screw and nut. Also graph is plotted for σ max and τmax v/s Load for screw by taking different standard materials like MS, MCS (C35Mn75), MCS (C55Mn75), and MCS (C35), MCS (30C8), STEEL (40NI14) in Fig. 5 and for nut Pb V/s Load in Fig.6 are shown below.

TABLE:1 Overall Summery

Material combination for		Load (KN)	Screw					Nut	Link				Pcr design (N)	Pcr Actual (N)
Screw	Nut		Total force p1 (KN)	P (KN)	Torque t (Nm)	σmax (Mpa)	σs max (Mpa)	σb (N/mm²)	dp (mm)	F (KN)	t1 (mm)	b1 (mm)		
M.S	M.S	1	1.217	0.408	3.066	15.89	10.51	2.15	3	0.304	3	9	1529.57	2549.5
		2	2.434	0.817	6.13	31.78	21.02	4.30	4	0.608	3	9	3043	2549.5
		3	3.652	1.126	9.19	47.68	31.53	7.45	5	0.913	5	15	4565.25	7342
		4	4.87	1.635	12.26	63.5	42	8.61	6	1.217	5	15	6087.5	7342
		5	6.087	2.044	15.33	79.49	52.58	10.7	7	1.521	5	15	7607.5	10650
MCS (C55Mn75)	CI	1	1.217	0.448	3.06	15.89	10.51	2.15	5	0.304	3	9	1826.1	2039.1
		2	2.434	0.818	6.13	31.78	21.02	4.30	7	0.608	5	15	3652.2	5876.59
		3	3.652	1.126	9.19	47.68	31.53	7.45	8	0.913	5	15	5478.3	7342
		4	4.87	1.635	12.26	63.5	42	7.45	8	1.217	8	24	7305	15232
		5	6.087	2.044	15.33	79.49	52.58	10.7	10	1.521	9	27	9126	19311
MCS (30C8)	PHO. BRONZE	1	1.217	0.408	3.06	15.89	10.51	2.15	5	0.304	3	9	912.9	2165
		2	2.434	0.817	6.13	31.78	21.02	4.30	7	0.608	3	9	1826.1	2165.5
		3	3.652	1.126	9.19	47.68	31.53	7.45	8	0.913	4	12	2739	3947.3
		4	4.87	1.635	12.26	63.5	42	8.61	10	1.217	4	12	3652.5	3947.3
		5	6.087	2.044	15.33	79.49	52.58	10.7	11	1.521	5	15	4561.5	6240.3
MCS (C35Mn75)	PHO. BRONZE	1	1.217	0.408	3.06	15.89	10.51	2.15	2	0.304	2	6	608	1057.78
		2	2.434	0.817	6.13	31.78	21.02	4.30	5	0.608	3	9	1217	2545.7
		3	3.652	1.126	9.19	47.68	31.53	7.45	6	0.913	4	12	1826	4643.3
		4	4.87	1.635	12.26	63.5	42	8.61	6	1.217	5	15	2434	7343
		5	6.087	2.044	15.33	79.49	52.58	10.7	8	1.521	6	18	3043	10641.04
STEEL (40NI14)	PHO. BRONZE	1	1.217	0.408	3.06	15.89	10.51	2.15	3	0.304	2	6	1520	2802.5
		2	2.434	0.817	6.13	31.78	21.02	4.30	4	0.608	3	9	3042.5	5108.3
		3	3.652	1.126	9.19	47.68	31.53	7.45	5	0.913	7	21	4565	15194.3
		4	4.87	1.635	12.26	63.5	42	8.61	7	1.217	7	21	1217.0	15994.7
		5	6.087	2.044	15.33	79.49	52.58	10.7	7	1.521	7	21	15105.0	26554.01
MCS (C35)	PHO. BRONZE	1	1.217	0.408	3.06	15.89	10.51	2.15	4	0.304	3	9	912	2038.22
		2	2.434	0.817	6.13	31.78	21.02	4.30	6	0.608	4	12	1825.5	3719.6
		3	3.652	1.126	9.19	47.68	31.53	7.45	7	0.913	4	12	2739.5	3719.6
		4	4.87	1.635	12.26	63.5	42	8.61	8	1.217	5	15	3611.0	5873.6
		5	6.087	2.044	15.33	79.49	52.58	10.7	9	1.521	5	15	4665.4	5873.6

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TABLE: 2 Comparison of calculated and design stresses

Material Combination for: Screw	Nut	Load (KN)	σt allow (Mpa)	σ1 max (Mpa)	Diff.	τ allow (Mpa)	τ max (Mpa)	Diff.	Pb Standard (N/mm²)	σ b calculated (N/mm²)	Diff.
M.S	M.S	1	100	15.89	-84.11	50	10.51	-39.49	30	2.15	-27.85
		2		31.78	-68.22		21.02	-28.98		4.305	-25.69
		3		47.68	-52.32		31.53	-18.47		7.45	-22.55
		4		63.5	-36.5		42	-8		8.61	-21.49
		5		79.49	-20.51		52.58	2.58		10.76	-19.24
MCS (C55Mn75)	CI	1	80	15.89	-64.11	40	10.51	-29.49	13.5	2.15	-11.35
		2		31.78	-48.11		21.02	-18.98		4.305	-9.19
		3		47.68	-32.32		31.53	-8.47		7.45	-6.5
		4		63.5	-16.5		42	2		8.61	-4.89
		5		79.49	-0.51		52.58	12.58		10.76	-2.74
MCS (30C8)	PHO. BRONZE	1	85	15.89	-69.11	60.4	10.51	-49.89	17	2.15	-14.85
		2		31.78	-53.22		21.02	-39.38		4.305	-12.69
		3		47.68	-37.32		31.53	-28.87		7.45	-9.55
		4		63.5	-21.5		42	-18.4		8.61	-8.39
		5		79.49	-5.51		52.58	-7.82		10.76	-6.3
MCS (C35Mn75)	PHO. BRONZE	1	100	15.89	-84.11	60	10.51	-49.49	18	2.15	-15.83
		2		31.78	-68.22		21.02	-38.98		4.305	-13.69
		3		47.68	-52.32		31.53	-28.47		7.45	-10.55
		4		63.5	-36.5		42	-18		8.61	-9.39
		5		79.49	-20.51		52.58	-7.42		10.76	-7.30
ALLOY STEEL (40NI14)	PHO. BRONZE	1	200	15.89	-184.11	85	10.51	-74.49	15	2.15	-12.85
		2		31.78	-168.22		21.02	-63.98		4.305	-10.69
		3		47.68	-152.32		31.53	-53.47		7.45	-7.55
		4		63.5	-136.50		42	-43		8.61	-6.39
		5		79.49	-120.51		52.58	-31.42		10.76	-4.30
MCS (C35)	PHO. BRONZE	1	80	15.89	-64.11	45	10.51	-34.49	15	2.15	-12.85
		2		31.78	-48.11		21.02	-23.98		4.305	-10.69
		3		47.68	-32.32		31.53	-13.47		7.45	-7.55
		4		63.5	-16.5		42	-3		8.61	-6.39
		5		79.49	-0.51		52.58	7.58		10.76	-4.30

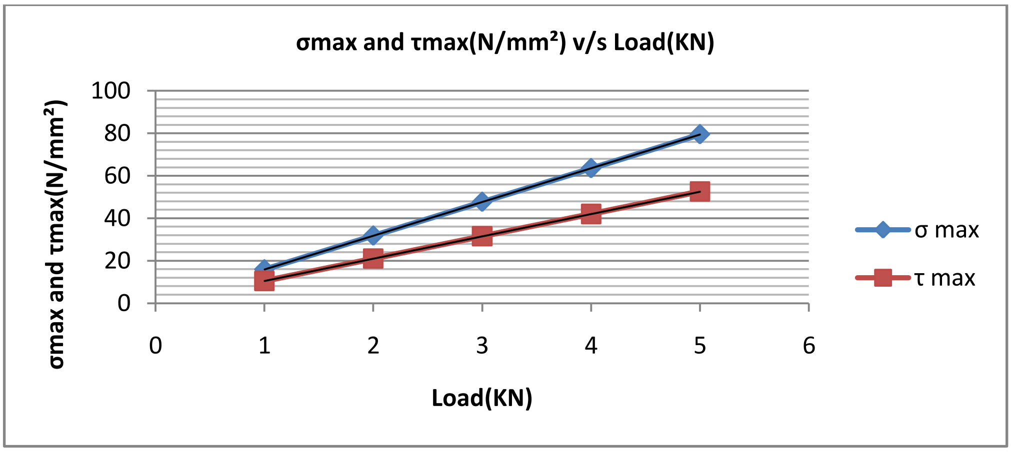

FIG. 5 σ max and τmax v/s Load

The above graph shows that the magnitude of σ1 max is higher than τmax and also it increases with increment of load.

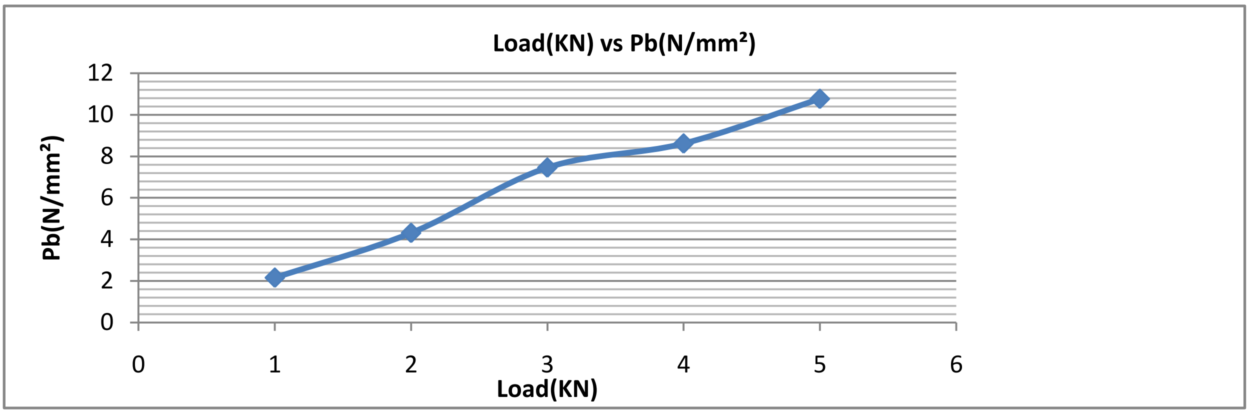

FIG. 6 Pb v/s Load

Same as bearing stress induced in material of nut also increases with increment of load.

V. CONCLUSION

From summery of calculated values and graph, it shows that for low load condition at 1KN, difference between σt allowable and σ1max is higher. i.e -184.11 MPa, while for higher load at 5KN , it is higher. i.e -120.52MPa. Same way, for load condition at 1KN, difference between τ allowable and τ max is higher. i.e -74.49 MPa and at 5KN, it is -43 MPa. The result shows that alloy steel for screw and phosphorus bronze for nut is the best suitable combination for pair. The value shows that such kind of materials keep induced stresses (σ1max and τmax) in screw within limit for safer design as compare to other materials. While difference between Pb Standard and σb calculated for a nut has higher values at 1KN i.e -27.85 N/mm² and at 5KN i.e -19.24 N/mm² for Mild steel material. The value shows that if there is a combination of MS – MS, it induces less magnitude of bearing stress in nut.

REFERENCES

1) R.S.Khurmi, J.K.Gupta, “Machine Design Book”, fourteenth edition, S.Chand & company ltd, New Delhi.
2) B.L.Singhal, F.B.Sayyad, “Machine Design Book”(GTU),second edition,Tech- max publication,Pune.
3) P.S.G.Design Data Book, second editon, Koimbtoor.
4) Richard G. Budynas, J. Keith Nisbett, “Mechanical engg. Design”,ninth edition, Mcgraw hill publication, New Delhi.
5) P.C Sharma and D. K. Aggarwal “ Machine Design”, S.K. Kataria & Sons 2009.
6) P.J Shah, “Machine Drawing”, S.Chand Publication.
7) Joseph Edward Shigley and Charles R. Mischke, "Mechanical Engineering Design", McGraw Hill International Edition.
8) Abdulla Shariff, “Hand Book of Properties of Engineering Materials & Design Data for Machine Elements”, Dhanpat Rai & Sons Publication.

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