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Design and Optimization of Mechanical Screw JackUsing Matlab Program

Akshay Shrishrimal[1], Pranav Patil [2], Ritesh Mane[3]

[1], *Dept. of Mechanical Engineering, MIT, Pune, India.*
[2], *Dept. of Mechanical Engineering, MIT, Pune, India.*
[3], *Dept. of Mechanical Engineering, MIT, Pune, India.*

Abstract:-*A screw jack serves to give mechanical advantage by changing rotational force to linear force thus allowing one to lift a load and support it at a given height. The aim of the paper is to come up with a design procedure for a simple screw jack and coding in Matlab to form a program that would require one to enter the mechanical properties of the materials to be used, lift and the desired load to be supported. This case study is divided into various sections that describes classification of screw threads, parts of the screw jack and selection of materials used for construction that are in agreement with current industry practice of screw jack design. The codingof the design procedures using Matlab was developed to serve the same function fast and efficient. This is realized as we obtained the similar theoretical and Matlab solutions as coded. Application of the procedure manually is tedious and extraneous since its lengthy and time consuming. Using Matlab for the same makes work easier, efficient and fast for designers since only material properties and specific design requirements such as lift and load are the only required input. The significance and purpose of this work is to modify the design methodology of screw jack in order to make the operation easier, safer and more reliable in order to reduce health risks.The designingof screw jack will also save time and requires less human energy to operate. The design when adopted will effectively curb the problems associated with Ergonomics - which is a fundamental concept of design process.*

Keywords- *Screw Jack, Components, Design procedure, Matlab script/code.*

I. INTRODUCTION

Screw jack is also called jack screw in other terms. A screw jack is an example of a power screw and referred to as a mechanical device that can increase the magnitude of an effort force. Screw jacks are used for raising and lowering platforms and they provide a high mechanical advantage in order to move moderately heavy and large weights with minimum effort. They function by turning the lead screw when raising or lowering of loads.

1.1 Operation

The jack can be raised and lowered with a metal bar that is inserted into the jack. The operator turns the bar with his/her hands in a clockwise direction. This turns the screw inside the jack and makes it go up. The screw lifts the small metal cylinder and platform that are above it. As the jack goes up, whatever is placed above it will raise as well, once the jack makes contact. The bar is turned until the jack is raised to the required level. To lower the jack the bar is turned in the opposite direction.

1.2 Factors to Consider in Selection of the Best Jack for Application Purposes

1. Consider the load carrying capacity of the lifting screw (column load) when jacks are loaded in compression. How high do you need to lift the load? One must choose a jack whose lifting screw is stout enough to handle the load at full rise.
2. Consider the travel speed of the dynamic load. The speed at which the load will be moved is a limiting factor. How fast do you need to move the load? Sometimes double lead machine screw jacks or ball screw jacks are a better choice in a given application.
3. How frequently will the jack need to move the load? Remember that heat builds up between the machine screw and nut during normal operation. Duty cycles for machine screw jacks must include periods of rest to dissipate that heat

1.3 Construction of a Screw Jack

A screw jack consists of a screw and a nut. The nut is fixed in a cast iron frame and remains stationary. The rotation of the nut inside the frame is prevented by pressing a set screw against it. The screw is rotated in the nut by means of a handle, which passes through a hole in the head of the screw. The head carries a platform, which supports the load and remains stationary while the screw is being rotated. A washer is fixed to the other end of the screw inside the frame, which prevents the screw from being completely turned out of the nut. Figure 1.1 below shows a screw jack and its parts and description as labeled.

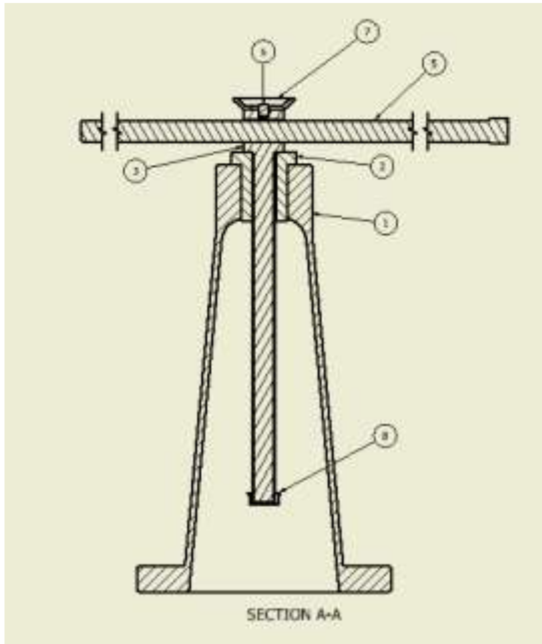

Figure 1.1 A screw jack and its parts.

2.0 Design procedure for the screw jack

The generalized adopted design procedure for a screw jack to raise a load of 2460 kg for example a large truck to a height of 200 mm.

Vehicle Class	Curb weight in kg
Compact car	1354
Midsize car	1590
Large Car	1985
Compact truck	1577
Midsize truck	1936
Large truck	2460

Table 2.1: Average Weight of Vehicles in India

2.1 Design for Screw Shaft

Material specification selected for the screw shaft is plain carbon steel to British Standard specification BS 970 080M30, Hardened and Tempered, whose properties are as shown in **Appendix B** and the material yield strength is 700 MPa both in tension and pure compression and 450 MPa in shear.

2.2 Core Diameter

The core diameter is determined by considering the screw to be under pure compression.

$$W=\sigma c\times Ac$$

σc = Pure compression stress = 700MPa
Ac = Cross sectional area of the screw shaft = $\pi/4*(dc)$
dc= Core diameter

$$W=\sigma c\times\pi/4(dc)^2$$
$$dc= \sqrt{}4W\sigma c\times\pi$$

Taking factor of safety $f.s$ = 5

$$dc= \sqrt{}4W\sigma cf.s\times\pi$$

$$dc = 0.0148147m$$
$$= 14.8147mm$$

For square threads of fine series, the following dimensions of screw are selected
The core diameter $dc = 16mm$, $do = 18mm$ and pitch $p = l = 2mm$.

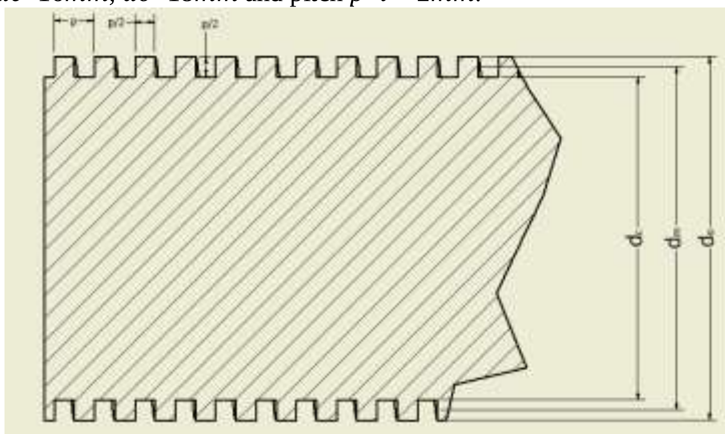

Figure 2.2: Section of screw spindle

2.3 Torque Required to Rotate the Screw

We know that torque required to rotate the screw is the same torque required to lift the load which is given by;

$$T1 = P \times dm2 = [W\tan(\alpha+\theta)]dm/2$$

We know that

$$dm = (do+dc)/2 = (18+16)/2 = 17mm$$

And

$$\tan\alpha = l\pi dm = 2\pi \times 17 = 0.03745$$

Assuming coefficient of friction between screw and nut,

$$\mu = \tan\theta = 0.1$$

Then

$$T1 = [24132.60\tan(2.1447+5.71)]0.0172$$
$$= 28.298Nm$$

2.4 Screw Stresses

Compressive Stress due to axial load using the new core diameter is,

$$\sigma c = WAC$$
$$= (dc)24$$
$$= 120.025MPa$$

And the shear stress due to this torque using the new core diameter is given by;

$$\tau = T1dc/2J$$

Where
$J = Polar\ moments = \pi dc^4/32$

$$\tau = 16T_1/\pi(dc)3$$
$$= 35.186MPa$$

2.5 Principal Stresses

Maximum principal stress is as follows:

$$(max) = 12[\sigma c + \sqrt{(\sigma c)2 + 4\tau 2}]$$

Substituting the stresses;

$$(max)=12[120.025+\sqrt{(120.025)2+4(35.186)2}]$$
$$(max)=129.579MPa$$

The design value of σc=700/5=140MPa
And maximum shear stresses as follows:

$$\tau max=12\sqrt{(\sigma c)2+4\tau 2} \quad (5.6)$$
$$\tau max=69.567MPa$$

The design value of τ=450/5=90MPa

Check: These maximum shear and compressive stresses are less than the permissible stresses, hence the spindle or shaft is safe.

2.6 Design for Nut
2.6.1 Height of the Nut

We find the height of the nut (h) by considering the bearing pressure Pb on the nut. The bearing pressure on the nut is given by;

$$Pb=W\pi 4[(do)2-(dc)2]n$$

Where
n = Number of threads in contact with screwed spindle

Material specification for the nut is phosphor bronze which has tensile stress = 150MPa, compressive stress = 125MPa, shear stress = 105MPa, safe bearing pressure not exceed 17MPa and a coefficient of friction of 0.1.
Assuming the load is uniformly distributed over the entire cross section of the nut and substituting for the known values we get the number of threads in contact,

$$17\times10^6=24132.60/\pi 4[(0.018)^2-(0.016)^2]n=451.861\times10^2/n$$
$$n=26.58$$

Say n=27
Then height of the nut is as follows;

$$h=n\times p \quad (5.8) \quad h=27\times2=54mm$$

Check: For a safe nut height $h\leq 4dc=64mm$

2.6.2 Stresses in the Screw and Nut

Shear stress in the screw is as follows;

$$\tau(screw)=W\pi n.dc.t$$

Where

$$t=Thickness\ of\ screw=p2=1mm$$
$$(screw)=24132.60\pi\times27\times0.016\times0.001=17.782MPa$$

And shear stress in the nut is as follows;

$$\tau(nut)=W/\pi n.do.t$$

Where

$$t=Thickness\ of\ screw=p2=1mm$$
$$(nut)=24132.60\pi\times27\times0.018\times0.001=15.806MPa$$

The given value of τ=105/5=21MPa

Check: These stresses are within permissible limit, hence, design for the nut is safe.

2.6.3 The outer diameter of Nut

Outer diameter $D1$ is found by considering the tearing strength of the nut.

$$\sigma t=W/\pi 4[(D1)2-(do)2]$$

Where
σt = $Tearing\ strength\ of\ the\ nut = Tensile\ stress$
σt =150/5=30MPa

Then we get $D1$ as follows;

$$30=24132.60\pi/4[(D1)2-(18)2]$$
$$D1=36.718mm,$$

Say $D1=37mm$

2.6.4 The outside diameter of Collar

Outside diameter $D2$ is found by considering the crushing strength of the nut collar.

$$\sigma c=W/\pi 4[(D2)2-(D1)2]$$

Where σc = $Crushing\ strength\ of\ the\ nut = Compressive\ stress$

σc =125/5=25MPa

Then we get $D2$ as follows;

$$25=24132.60\pi 4[(D2)2-(37)2]$$
$$D2=50.971mm,$$

Say $D2=51mm$

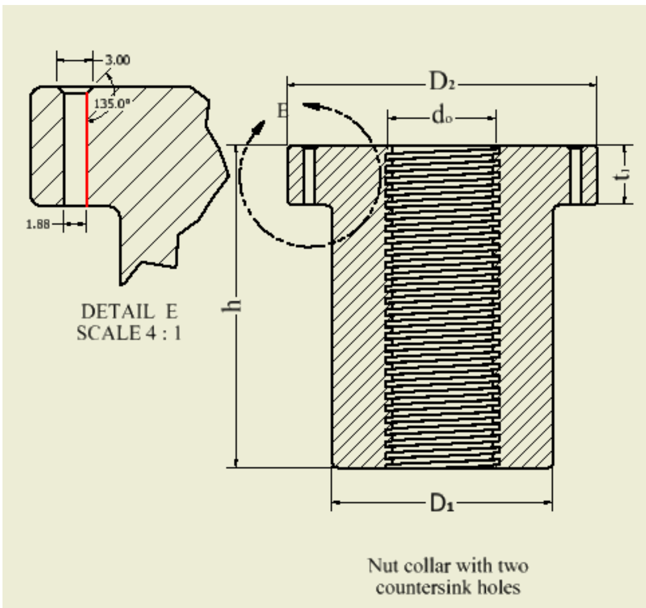

Figure 2.6.4: Section of Nut collar 1

2.6.5 Thickness of the Nut Collar

The thickness of nut collar $\boldsymbol{t1}$ is found by considering the shearing strength of the nut collar.

$$t1=W\pi D1.\tau \ (5.13)$$

Shearing strength of nut collar =105/5=21MPa

Therefore

$$t1=24132.60\pi\times 37\times 21$$
$$t1=9.88mm,$$

$Say\ t1=10mm$

2.7 Design for Head and Cup

2.7.1 Dimensions of Diameter of Head on Top of Screw and for the Cup $D3$

$$D3=1.75do$$
$$D3=1.75\times 18=31.50mm$$

$Say\ 32mm$

The seat for the cup is made equal to the diameter of the head and then chamfered at the top. The cup prevents the load from rotating and is fitted with pin of diameter $D4=D34⁄$ approximately. Therefore $D4=8mm$.

The pin should remain loose fit in the cup.

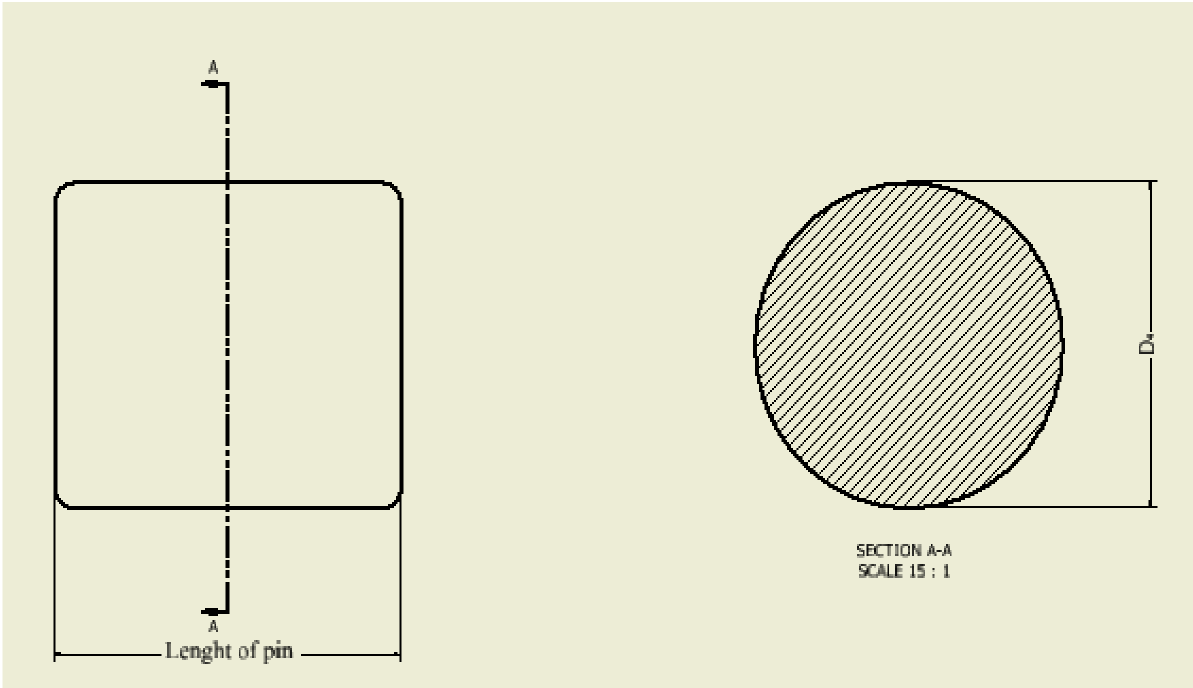

Figure 2.7.1: Section of Pin

Take length of pin to be 9mm.
Other dimensions for the cup are taken as:
Diameter at the top of the cup = Diameter of the head = 52mm
Height of cup = 9mm
Thickness of cup = 3mm
Fillet radii = 1mm

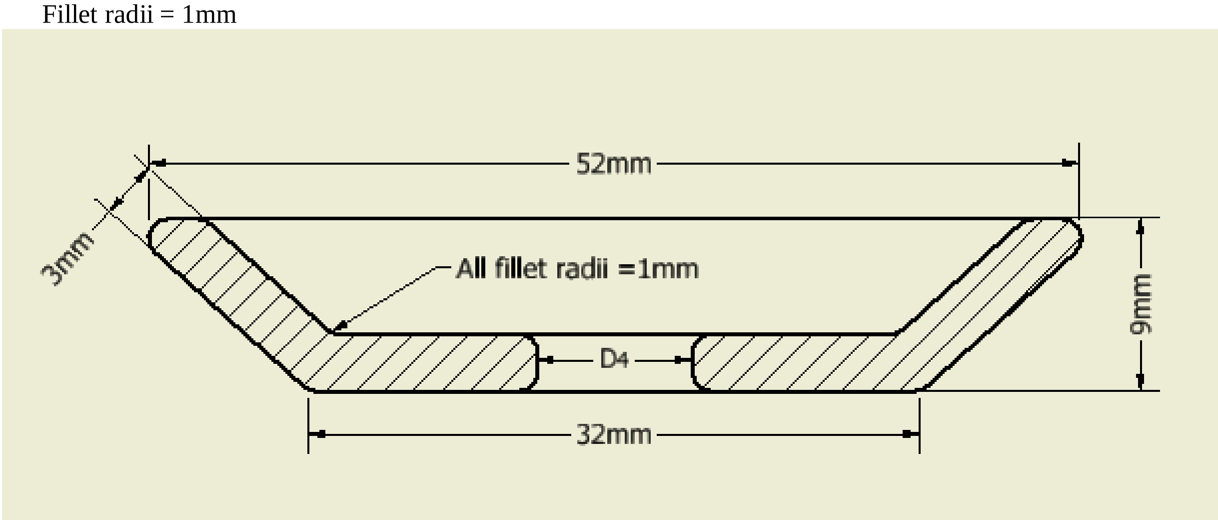

Figure 2.7.2: Section of Cup

2.7.2 Torque Required to Overcome Friction

We know that by assuming uniform pressure condition torque required to overcome friction is given as follows;

$$T2=13\times\mu1[(D3)3-(D4)3(D3)2-(D4)2]$$

Where
$D3$= Diameter of head = 32mm
$D4$= Diameter of pin = 8mm
Substituting for the known values we get; $T2=13\times0.1\times24132.60[(0.032)3-(0.008)3(0.032)2-(0.008)2]=27.0285Nm$

Total torque to which the handle is subjected is given by

$$T=T1+T2 \quad (5.16) \quad T=28.298+27.0285=55.326Nm$$

Activity	Professional use	Domestic use
Pushing	200N (20.4kg)	119N (12.1kg)
Pulling	145N (14.8kg)	96N (9.8kg)

Table 5.2: Maximal Isometric Force by General European Working Population for Whole Body Work in a Standing Posture

Therefore taking the force of 96N in domestic use.Then the length of the handle required is $L=T/96$

Then $L=55.32696=0.5763m=576.30mm$ Say $L=580mm$

The length of the handle may be fixed by giving some allowance for gripping 70mm.

Therefore, the length of the handle/lever is 646.30mm.

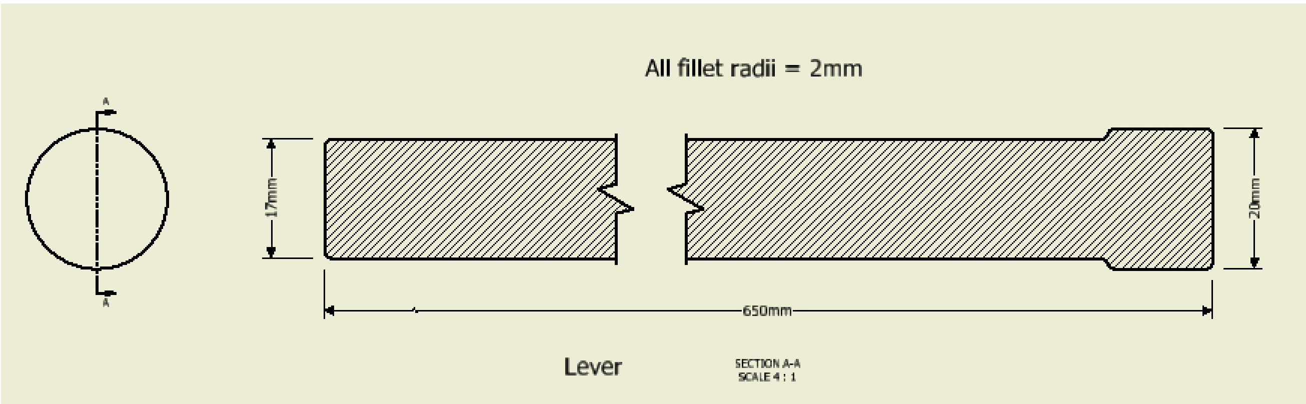

Figure 2.7.3: Section of Lever

2.7.3 Diameter of Handle/Lever

The diameter of the handle/lever, may be obtained by considering bending effects. We know that bending moment;

$$M=\pi/32\times\sigma b\times D3$$

While

$$\sigma b=\sigma t=\sigma c=7005=140MPa$$

And maximum bending moment on the lever/handle

$$M=Force\ applied\times Length\ of\ lever$$
$$M=96\times0.6463=62.0448Nm$$

Then

$$62.0448=\pi32\times140\times106\times D3$$
$$D=16.5269mm,$$
$$\text{Say } D=17mm$$

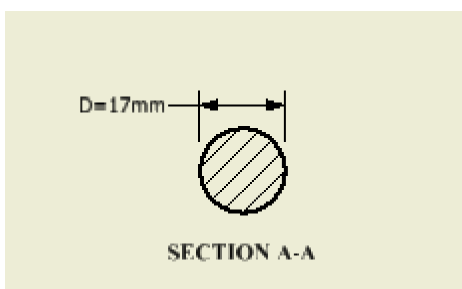

Figure 2.7.4: Section of Lever - Diameter

2.7.4 Height of Head

The height of head is usually taken as twice the diameter of handle.

$$H=2D$$

Therefore

$$H=2\times17=34mm$$

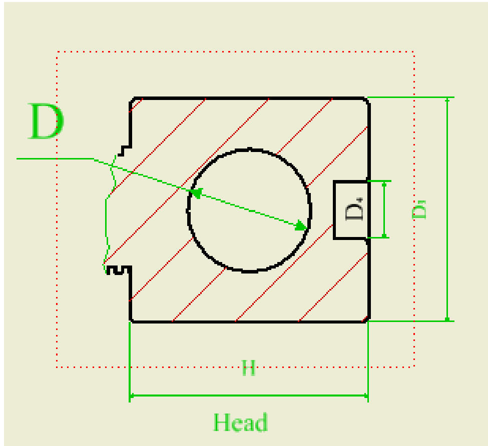

Figure 2.7.5: Section of Screw Head

2.7.5 Design Check against Instability/Buckling

Effective length of screw, $Leff$=$Lift\ of\ screw$+ 12 $of\ height\ of\ nut$

$$Leff=H1+h2 \quad (5.19)$$

$$Leff=200+542$$

$$Leff=227mm$$

When the screw reaches the maximum lift, it can be regarded as strut whose lower end is fixed and the load end is free. Therefore, buckling or critical load for this given condition is as follows

$$Wcr=Ac.[1-\sigma y4C\pi 2E(Leffk)2]$$

Where

σy= Yield stress = 385MPa

C = End fixity coefficient. The screw is considered to be strut with lower end fixed and load end free.

Therefore C = 0.25

k = The radius of gyration = $\sqrt{}IA$= 0.25dc= 0.004

I= Moment of inertia of the cross section.

The buckling load as obtained by the above expression and must be higher than the load at which the screw is designed. Substituting for the known values:

$$Wcr=\pi 4(dc)2.\sigma y[1-\sigma y4C\pi 2E(Leffk)2]$$

$$Wcr=28784.55N$$

While

$$W=24132.60N$$

Wcr>, hence there is no chance for the screw to buckle.

2.8 Design of Body

2.8.1 Dimensions for the body of the screw

The dimension of the body may be fixed.

1. Diameter of the Body at the Top

$$D5=1.5D2 \quad (5.22)$$

$$D5=1.5\times 51=76.50mm$$

2. Thickness of the body

$$t2=0.25do \quad (5.23)$$

$$t2=0.25\times18=4.5mm,$$
$$say\ t2=5mm$$

3. inside Diameter at the Bottom

$$D6=2.25D2$$
$$D6=2.25\times51=114.75mm$$

4. Outer Diameter at the Bottom

$$D7=1.75D6$$
$$D7=1.75\times114.75=200.8125mm$$

5. Thickness of Base

$$t3=2t1$$
$$t3=2\times10=20mm$$

6. Height of the Body Hb

$Height\ of\ the\ body$=Max$lift$+$Height\ of\ nut$+$Extra$ 50mm Hb = 200+54+50=304mm
Finally, the body is tapered in order to achieve stability of the jack.

2.8.2 Efficiency of the Screw Jack

Efficiency of screw jack is given as follows:
$\eta=Torque\ required\ to\ rotate\ screw\ with\ no\ friction Total\ torqure\ output=ToT$ (5.27a)

$$\eta=ToT$$
$$\text{But } To=W\tan\alpha\times dm2$$
$$To=24132.60\times0.03745\times0.0172$$
$$To=7.682Nm$$
$$T=55.326Nm$$

Therefore

$$\eta=7.68255.326=0.1388\ or\ 13.88\%$$

3.0 MATLAB Coding

The Screw jack MATLAB design Script is

```
% MATERIAL PROPERTIES %
W = input ('Enter load capacity of the screw jack in kg');
W = W * 9.81;
H1 = input ('Enter the required lift of the screw jack in mm');
u = input ('Enter coefficient of friction between materials');
E = input ('Enter the modulus of elasticity of the screw material in MPa');
y = atan (u);
TstrYP = input ('Enter yield point for tension in MPa for screw and nut material as 1*2 matrix respectively');
CstrYP = input ('Enter yield point for compression in MPa for screw and nut material as 1*2 matrix respectively');
SHstrYP = input ('Enter yield point for shear in MPa for screw and nut material as 1*2 matrix respectively');
% DESIGN FOR SCREW SHAFT %
d1 = (4*W/(pi*CstrYP(1,1)/5))^0.5; % Minor Diameter %
disp ('CALCULATED VALUE OF MINOR DIAMETER in mm =')
d1 = (d1) %#ok<NOPTS>
d2 = input ('SELECT A STANDARD MAJOR DIAMETER FROM LIST 10 12 14 16 18 20 22 24 26 28 30 32 34
36 38 40 42 44 46 48 50 52 55 58 60 62 65 68 70 72 75 78 80 82 85 88 90 92 95 98 100 105 110 115 120 125 130
135 140 145 150 155 160 165 170 175');
disp ('STANDARD MAJOR DIAMETER in mm =')
disp (d2)
if 10<=d2 && d2<20 % Pitch %
p = 2;
elseif 22<=d2 && d2<62
p = 3;
```

```
elseif 65<=d2 && d2<110 45
p = 4;
elseif 115<=d2 && d2<175
p = 6;
else disp ('SELECT CORRECT NOMINAL DIAMETER in mm =')
break
end
disp ('PITCH')
disp (p)
d1 = d2 - p; % Standard Minor Diameter %
disp ('STANDARD MINOR DIAMETER in mm =')
disp (d1)
dm = (d2+d1)/2; % Mean Diameter %
helix = atan (p/(pi*dm)); % Helix Angle %
disp ('HELIX ANGLE in radians =')
disp (helix)
if helix >= y
disp ('Screw Nut Pair is Not Self Locking')
break
end
T1 = W*(dm/2)*tan (helix + y); % Torque Required to Rotate the Screw %
disp ('TORQUE TO ROTATE THE SCREW in Nmm =')
disp (T1)
% DESIGN FOR NUT %
Pb = input ('Enter Allowable Bearing Pressure in MPa');
if Pb == [] %#ok<BDSCA>
Pb = 17; % Default value for Pb for the Phosphor Bronze %
end
n =4*W/ (pi*(d2^2-d1^2)*Pb); % Number of Screw Thread %
n = round (n);
disp ('NUMBER OF THREAD =')
disp (n)
h = n*p; % Height of the Nut %
disp ('HEIGHT OF THE NUT in mm =')
disp (h)
% CHECKING FOR SAFE NUT HEIGHT %
if n <= 27 && h <= 4.0*d1
disp ('NUT DESIGN CHECK IS SATISFIED')
else
disp ('NUT DESIGN CHECK IS NOT SATISFIED')
break
end
% CHECKING FOR SCREW STRESSES %
CstrB = 4*W/ (pi*d1^2); % Compressive stress %
SHstrB = 16*T1/ (pi*d1^3); % Shear stress % 46
SHstrmax = ((CstrB/2) ^2 + SHstrB^2) ^0.5; % Maximum shear stress %
f = SHstrYP (1, 1)/SHstrmax; % Factor of safety %
if f >= 5
disp ('SCREW DESIGN IS SAFE')
else
disp ('SCREW DESIGN IS NOT SAFE')
break
end
% OTHER ASPECTS OF NUT DESIGN %
D1 = ((4*W/ (pi*TstrYP (1, 2)/5)) +d2^2) ^0.5; % Outer diameter of the nut %
D1 = ceil (D1);
disp ('OUTER DIAMETER OF THE NUT in mm =')
```

```
disp (D1)
D2 = ((4*W/(pi*CstrYP(1,2)/5))+D1^2)^0.5; % Outside diameter of the collar %
D2 = ceil (D2);
disp ('OUTSIDE DIAMETER OF THE COLLAR in mm =')
disp (D2)
t1 = W/ (pi*D1*SHstrYP (1, 2)/5); % Collar thickness %
t1 = round (t1);
disp ('COLLAR THICKNESS in mm =')
disp (t1)
%CHECKING FOR SCREW AGAINST INSTABILITY %
Leff = H1+h/2; % Length of column considered %
k = 0.25*d1; % Radius of gyration %
R = Leff/k; % Slenderness ratio%
C = 0.25; % End condition for fixed free column %
Ac = (pi*d1^2)/4;
Rcr = ((2*C*pi^2*E)/TstrYP (1, 1)) ^0.5; % Critical slenderness ratio%
if R < Rcr
Wcr = Ac*TstrYP*(1-0.5*(R/Rcr) ^2); % Short column formula%
else
Wcr = pi^2*E*k^2*Ac/ (4*Leff^2); % Long column formula%
end
if Wcr >= W
disp ('THE SCREW IS STABLE')
else
disp ('THE SCREW IS NOT STABLE REDUCE THE LIFT ')
break
end
% DESIGN FOR HANDLE, PIN AND CUP %
D3 = 1.75*d2; % Diameter for the Cup %
D3 = round (D3);
disp ('CUP DIAMETER in mm =')
disp (D3)
D4 = D3/4; % Diameter of Pin %
disp ('PIN DIAMETER in mm =')
disp (D4)
T2 = 1/3*u*W*((D3^3-D4^3)/ (D3^2-D4^2)); % Torque required to overcome friction %
disp ('TORQUE TO OVERCOME FRICTION in Nmm =')
disp (T2)
T = T1 + T2; % Total Torque the handle is subjected to %
disp ('TOTAL TORQUE in Nmm =')
disp (T)
L = T/96; % Length of the Handle %
L = L+70; % Length of handle + Allowance for gripping 70mm %
disp ('LENGTH OF THE HANDLE in mm =')
disp (L)
M = 96*L; % Maximum bending moment on the lever%
disp ('MAXIMUM BENDING MOMENT in Nmm =')
disp (M)
D = ((32*M*5)/ (pi*CstrYP (1, 1))) ^0.33; %Diameter of handle%
D = round (D+1);
disp ('DIAMETER OF HANDLE in mm =')
disp (D)
H = 2*D; % Height of Head %
disp ('HEIGHT OF HEAD in mm =')
disp (H)
% DESIGN OF BODY %
D5 = 1.5*D2; % Diameter of the body at the top %
```

```
disp ('DIAMETER OF BODY AT THE TOP in mm =')
disp (D5)
t2 = 0.25*d2; % Thickness of the Body %
t2 = round (t2);
disp ('THICKNESS OF BODY in mm =')
disp (t2)
D6 = 2.25*D2; % Inside Diameter at the Bottom %
disp ('INSIDE DIAMETER AT THE BOTTOM in mm =')
disp (D6)
D7 = 1.75*D6; % Outside Diameter at the Bottom %
disp ('OUTSIDE DIAMETER AT THE BOTTOM in mm =')
disp (D7)
t3 = 2*t1; % Thickness of Base %
disp ('BASE THICKNESS in mm =')
disp (t3)
Hb = H1+h+50; % Height of body %
disp ('HEIGHT OF BODY in mm =')
disp (Hb)
```

3.1 Program Run of the Design Script/Code & Solution

The script was run and the value were added as input. The proceedings were-

```
Enter load capacity of the screw jack in kg 2460
Enter the required lift of the screw jack in mm 200
Enter coefficient of friction between materials 0.1
Enter the modulus of elasticity of the screw material in MPa 200000
Enter yield point for tension in MPa for screw and nut material as 1*2 matrix respectively [700 150]
Enter yield point for compression in MPa for screw and nut material as 1*2 matrix respectively [700 125]
Enter yield point for shear in MPa for screw and nut material as 1*2 matrix respectively [450 105]
CALCULATED VALUE OF MINOR DIAMETER in mm =d1 = 14.8147
SELECT A STANDARD MAJOR DIAMETER FROM LIST 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40 42
44 46 48 50 52 55 58 60 62 65 68 70 72 75 78 80 82 85 88 90 92 95 98 100 105 110 115 120 125 130 135 140 145
150 155 160 165 170 175 18
STANDARD MAJOR DIAMETER in mm =18
PITCH = 2
STANDARD MINOR DIAMETER in mm = 16
HELIX ANGLE in radians = 0.0374
TORQUE TO ROTATE THE SCREW in Nmm = 2.8300e+04
Enter Allowable Bearing Pressure in MPa = 17
NUMBER OF THREAD = 27
HEIGHT OF THE NUT in mm = 54
NUT DESIGN CHECK IS SATISFIED
SCREW DESIGN IS SAFE
OUTER DIAMETER OF THE NUT in mm = 37
OUTSIDE DIAMETER OF THE COLLAR in mm = 51
COLLAR THICKNESS in mm = 10
THE SCREW IS STABLE
CUP DIAMETER in mm = 32
PIN DIAMETER in mm = 8
TORQUE TO OVERCOME FRICTION in Nmm = 2.7029e+04
TOTAL TORQUE in Nmm = 5.5329e+04
LENGTH OF THE HANDLE in mm = 646.3422
MAXIMUM BENDING MOMENT in Nmm = 6.2049e+04
DIAMETER OF HANDLE in mm = 17
HEIGHT OF HEAD in mm = 34
DIAMETER OF BODY AT THE TOP in mm = 76.5000
THICKNESS OF BODY in mm = 5
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INSIDE DIAMETER AT THE BOTTOM in mm = 114.7500
OUTSIDE DIAMETER AT THE BOTTOM in mm = 200.8125
BASE THICKNESS in mm = 20
HEIGHT OF BODY in mm = 304

II. CONCLUSION

1. The Matlab program designed ran with following proper sort of theorems and equations. The MATLAB coding and results are tabulated.
2. Self-locking of the screw is not possible when the coefficient of friction (μ) is low. The coefficient of friction between the surfaces of the screw and the nut is reduced by lubrication. Excessive lubrication may cause the load to descend on its own.
3. The self-locking property of the screw is lost when the lead is large. The lead increases with number of starts. For double-start thread, lead is twice of the pitch and for triple threaded screw, three times of pitch. Therefore, the single threaded screw is better than multiple threaded screws from self-locking considerations.

III. RECOMMENDATION

1. Further research should be carried out onhow to minimize vibration and noise duringoperation.
2. From the case study, we concentrated on design of a simple mechanical screw jack where the nut is fixed in a cast iron frame and remains stationary while the spindle is being rotated by the lever. This design can only work for light loads hence when a screw jack is needed for heavy load application a different design is required where the nut is rotated as the spindles moves. We therefore recommend design of a screw jack for the heavy loads.

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