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Study on design and modification of bend test apparatus

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Abstract: *In this study we have designed and carried out the modification of bend test apparatus. A bend test is used to determine whether a specific piece of metal in question will break or fracture under pressure. Various couplings and fittings are tested here. The current system of bend test uses an unbalanced weight loading which gives rise to less number of points of contact. In this study we design and modify this setup, so that the load application process is balanced and there is increase in number of points of contact. Thus this will lead to optimisation in testing time and man power. Different magnitudes of load are applied on the samples to accurately determine not only the quality of the part, but also the root cause of a variety of defects and it will also helps us to determine its bending strength. In this we are going to study two designs i.e. plate arrangement and screw jack and lever arrangement further optimization of the efficient arrangement will be carried out.*

Keywords: *Bend Test, Screw Jack, Lever, Flat Belt.*

1. INTRODUCTION

A bend test is used to determine whether a specific piece of metal in question will break or fracture under pressure. This is important in any project using metal otherwise the item being made could collapse from the immense pressure exerted on it. The bend test essentially measures a metal's ductility. Bend tests deform the test material at the midpoint causing a concave surface or a bend to form without the occurrence of fracture and are typically performed to determine the ductility or resistance to fracture of that material. The Project is concerned with the design and modification of bend test apparatus. Various couplings and fittings are tested over here. Our task is to modify the 'Bend' test setup. The basic aim of the bend test is to check the strength of the samples when it will go through practical application. Currently the setup used in the industry uses a weight loading system which is quiet unbalanced due to use of local equipments, which also leads to less number of point of contacts. So basically, we are going to balance this system by replacing it with a modified and newly designed setup. Also, this new design will help us to increase the number of point of contacts. We have proposed two methodologies to counter this problem.

2. LITERATURE REVIEW

Chetan S.Dhamak, D.S.Bajaj, V.S.Aher [1]

Manoj R Patil1* and S D Kachave [2]

Every engineering product involve cost effective manufacturing and its versatility in application maintaining its aesthetics as well as assign service life without failure keeping those parameters in mind they focused their intention on designing and analyzing the jack model for actual service loads for varying models of automobile L.M.V. sectors. Automobile sectors are very keen at their productivity and customer satisfaction. They also keen at reducing the weight of scissor jack at the same time maintaining its strength and service life. They made certain change in manufacturing process thereby made a new versatile jack that can be used for varying models of L.M.V automobile sector. Also the new design that made by Pro-e software can be tested by ANSYS software.

According to the study carried out in IIT Kharagpur [3] they came across this following information: correction factor for speed and angle of wrap are used to modify the belt maximum stress. This correction is required because stress value is given for a specified drive speed and angle of wrap of 180^0. Therefore, when a drive has different speed than the specified and angle of wrap is also different from 180^0 , then above mentioned corrections are required. The recommendations are; the centre distance should be greater than twice the sum of pulley diameters and the belt speed range should be within 15-25 m/s.

3. PROBLEM STATEMENT

Currently the setup used in the industry uses a weight loading system which is quiet unbalanced due to use of local equipments, which also leads to less number of point of contacts. So basically, we are going to balance this system by replacing it with a modified and newly designed setup. Also, this new design will help us to increase the number of point of contacts.

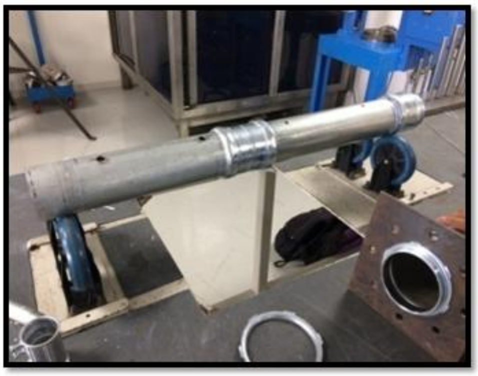


Figure 1: *Actual Bend set-up*

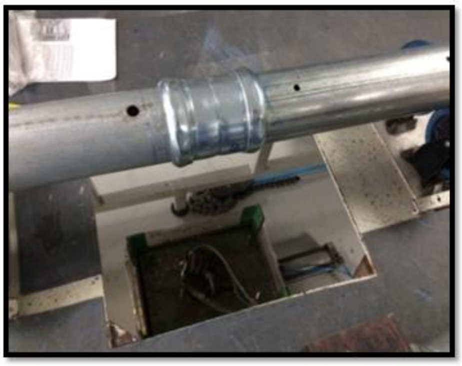
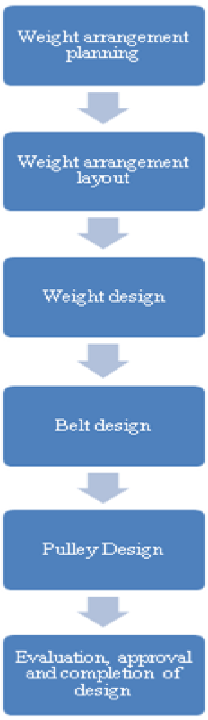


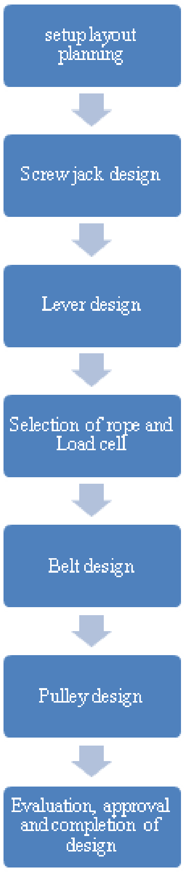
Figure 2: *Actual Bend set-up*

4. METHODOLOGY

4.1 Plate Design



4.2 Screw jack and Lever design



5. DESIGN AND CALCULATIONS

In design method, firstly we understood the problem statement and then we found various solutions to overcome the problem.

Table 1: *Weight layout for set-1*

Plate no	Weights	
	Lbs	Kg
1	16.81	15
2	2.27	5
3	4.54	10
4	2.27	5
5	6.81	15
6	4.54	10
7	6.81	15

Table 2: *Weight layout for set-2*

Plate no	Weights	
	Lbs	Kgs
1	2.27	5
2	27.22	60
3	9.08	20
4	13.61	30
5	4.54	10
6	18.14	40

Set 1:-

Keeping height =0.02m, breadth = 0.144 m

Density (ρ) =7870 kg/m^3

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For 15lbs (M=6.81kg)

Without considering holes:-

Density ($_\rho$) = mass (M)/volume (v)

Volume = 8.65*10-4 m^3

Volume = l*b*h

l = 0.3 m

Considering 4 holes:-

m=6.574 kg

Difference between mass without hole (M) and with hole (m) -

M-m = 6.81-6.5474 = 0.236 kg

Density ($_\rho$) = mass/volume

7870 = (6.81+0.236)/ (volume)

Volume = $8.9529*10^{-4}$ m^3

l*b*h = $8.9529*10^{-4}$

l*0.144*0.02 = $8.9529*10^{-4}$

l= 310.5 mm = 0.3105m

Similarly, the remaining plates for set 1 are designed as above.

For set 2-

Keeping height (h) = 0.04m and breadth (b) =0.190m as constant

Density ($_\rho$) =7870 kg/m^3

*For plate no 8- 5lbs plate we are going to use hanging weight

Similar method is used as in set 1, to find the remaining lengths.

Table 3: *Results for set-1*

Plate no	Mass (lbs)	Length(m)
1	15	0.3105
2	5	0.11
3	10	0.2105
4	5	0.11
5	15	0.3105
6	10	0.2105
7	15	0.3105

Table 4: *Results for set-2*

Plate no	Mass(lbs)	Length(m)
*8	5	-
9	60	0.462
10	20	0.1585
11	30	0.234134
12	10	0.1518
13	40	0.31013

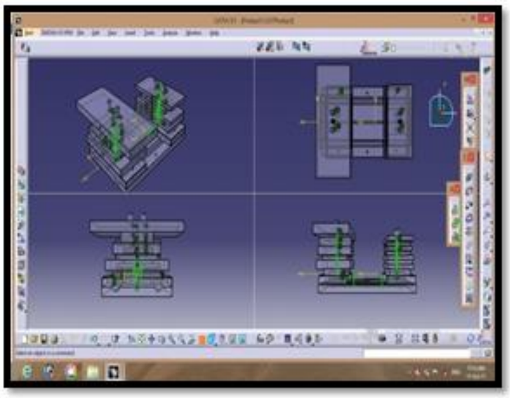

Figure 3: *All views of assembly*

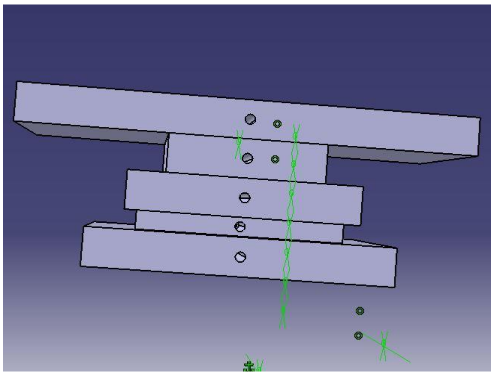

Figure 3: *Set two- 6 plates*

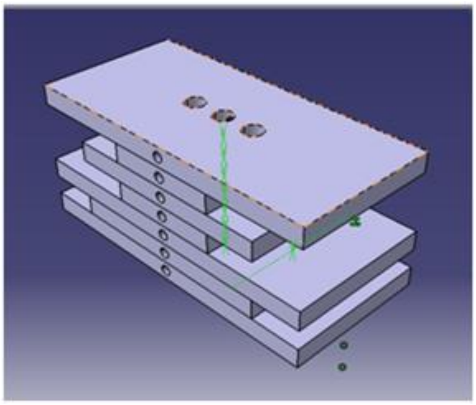

Figure 5: *Set one- 7 plates*

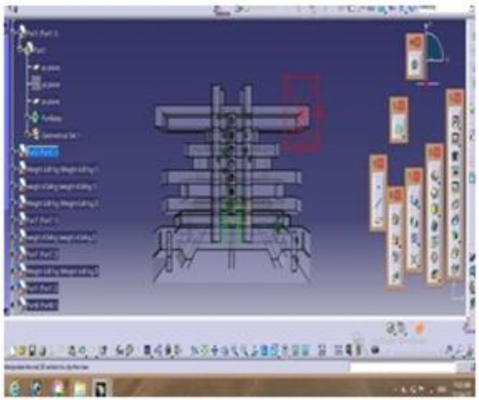

Figure 6: *Cut section*

6. DESIGN OF FLAT BELT

For coupling 3 to 4 inch

- Standard width = 50mm
- 3 PLY
- Maximum belt speed = 10m/s
- Minimum pulley diameter = 90mm
- Belt length :-

L= 2C+ (π/2)*(D+d) + ((D-d) ²)/ (4*C)

Where, C= Centre distance = 510mm

D= Outer race diameter = 90mm

D= Inner race diameter = 85mm

L= 1294.90mm

- For belt tension: Belt of 3 plies- 1.5% of 15mm per m length shorter Steady load and load correction factor is 1.2 Arc of contact factor

= 180-{(D-d)/C}*60

= 179.41degrees.

For coupling of 1 to 2 inch

For this belt standard width is 25mm and all the other specifications are same as above

7. DESIGN OF SCREW JACK

- **Problem specifications :-**

It is required to design a screw jack for applying pulling force on a coupling through a lever.

1. Load carrying capacity = 2.5 kN
2. Lifting height = 0.2m
3. Components of screw jack and materials used :-

Sr. No	Name of component	Material
1	Screw	Steel 30C8
2	Nut	Phosphor Bronze Grade1
3	Handle	Steel 30C8
4	Cup	Grey Cast Iron FG200

4. Maximum hand force applied is of 200N.
5. It is operated by a single worker.
6. Coefficient due to reduction in force due to inconvenience:-

P= 0.9*1*200

= 180N

- **Design of Screw (PCS) :-**

$S_{yt} = S_{yc} = 400\ N/mm^2$

Taking FOS as 5

Compressive Stress = S_{yc}/FOS

= 400/5

= 80 MPa

Compressive Stress

is 4W/ (π*d_c^2)

Where dc = core diameter of screw 80 = 4*2500/ (π* d_c^2)

d_c = 6.307mm

taking standard value,

d_c = 8mm

Therefore, for square threads Nominal diameter= 22mm, Pitch= 5mm

d = 24m

p = 5mm

d_c = d-p = 25 = 19mm

d_m = d-0.5p = 24-2.5 = 21.5mm

Assuming single start threads,

l=p=5mm

tan (α) = l/ (π* d_m)

α = 4.23 degrees.

Taking μ = tan (ϕ) = 0.18

ϕ = 10.20 degrees.

As ϕ>α, screw is self locking

μT = {W*d_m*tan (ϕ +α)}/2

= 6915.32 Nmm

τ = (16* μT)/ (π* d_c^3)

τ = 5.13 N/mm²

Compressive stress = 4W/ (π*d_c^2)

= 8.817 N/mm²

Bending moment (M_B),

$M_B = P*l_h$ M_B = 48600 Nmm

Bending stress = $(32* M_B)/(\pi* d_c^3)$

= 72.17 N/mm²

Principal shear stress(Z_{max}) :-

Z_{max} = sqrt{(bending stress/2)2+ τ^2}

= 36.44 N/mm²

Factor of safety(FOS) :-

FOS = $\{S_{sy}/Z_{max}\}$

= $\{0.5*S_{yt}/Z_{max}\}$

FOS = 5.48

As our assumed FOS is less than actual FOS Therefore, our design is safe.

- Buckling considerations :-

  During the raising of load through a distance of 200mm, the portion of screw between the nut and handle acts as a column.

Length of column(L_C) = 200+50 = 250mm

Polar Moment of Inertia(I) = $(\pi*d_c^4)/64$

I = 6397.1171 mm⁴

Area(A) = $(\pi*d_c^2)/4$

A = 283.52 mm²

K = sqrt(I/A)

K = 4.75 mm

Slenderness ratio of screw

= $(L_C/K)_{actual}$ = 250/4.75

= 52.63

Since one end of the screw is fixed in the nut and the other end is free, the end fixity coefficient is 0.25. The border line between short and long column is given by:-

$\{S_{yt}/2\} = (n*\pi^2*E)/(L_C/K)^2_{critical}$

$(L_C/K)_{critical}$ = 50.53

As $(L_C/K)_{actual}$ is greater than $(L_C/K)_{critical}$

Hence the screw should be treated as long column and by using Euler's Equation:-

$P_{CR} = (n*\pi^2*E*A)/(L_C/K)^2_{actual}$

P_{CR} = 52278.97 N

FOS for buckling :-

FOS = P_{CR}/W

= 20.91

Therefore, the design is safe against buckling.

- **Design of nut**

The permissible bearing press between the steel screw and bronze is 10N/mm²

The number of threads required to support the load is z

$Z = (4W)/\pi*S_b*(d^2-d^2_c)$

Z = 1.4805(approx) = 2

Axial length of the nut (H)

H = Z*p = 2*5 = 10mm

The transverse shear stress at the root of the threads in the nut is given by

$Z_n = W/(\pi*d*Z*t)$

Here, t = 3

Z_n = 5.526 N/mm2

FOS = S_{sy}/Z_n

FOS =17.4

The outer diameter of the nut is assumed to be twice of the nominal diameter of the thread

- **Design of cup**

The annular area of the collar friction has outer diameter of 1.6d and the inner diameter is assumed to be 0.8d

Do=1.6d=1.6*24=38.4mm
Di = 0.8*24 = 19.2mm

The collar friction torque

$(Mt)_c=(\mu_c*W/4)*(D_o+D_i)$

$(M_t)_c$ = 7200 Nmm

The total torque (Mt) t required to raise the load is given by

$(Mt)t = (Mt)_c + (M_t)_t$

(Mt) t= 14115.32 Nmm

- Length of the handle(L_h)

.The external torque which is exerted by the worker is given by

$(M_t)_t=0.9*1*200*L_h$
L_h=78.41 mm

- **Design of bearing**

The procedure for the selection of ball bearing from manufacturer's catalogue

By selecting single deep groove ball bearing The handle is rotated manually, it is thus not possible to find out the speed accurately. For the purpose of bearing selection, it is assumed that the handle rotates at 10rpm

Life of bearing in million revolutions is given L = 2.4 million revolutions

Also,

P = W = 2500N

C= dynamic load capacity

$C = P*L^{1/3}$

C = 3347.16 N

It is assumed that bore diameter of the bearing is 50 mm

Designation of the bearing =51110

Bearing no 51110 with dynamic load capacity of 25500 N is selected for jack

The dimensions of the bearing are as follows

d =50 mm

D = 70mm

H = 14mm

D_1 = 52mm

8. CONCLUSION

In this paper we have studied above two methods- 'plate arrangement' and 'screw jack and lever arrangement'. So we have come to a conclusion that the plate arrangement method is the efficient in terms of cycle time as well as economical consideration due to the ease of operation, increase in number of point of contacts, compact setup and cost. The screw jack and lever arrangement is not cost efficient and has complexity in setup.

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