DESIGN AND ANALYSIS OF PIPE BENDING MACHINE USING MOTORIZED SCISSOR JACK

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ABSTRACT

This paper is targeted to analyze the development in existing scissor car jack in order to apply the load on the rod and bend the rod to required angle using electric power as input by utilizing electricity as input power source. The contractions or expansion movement of scissor jack can be controlled by a joystick as per requirements. This modified scissor jack can be easily operated by any person and it saves time, hence reduce wastage of human efforts and time. The design of this scissor jack is being developed in Solid Works software. Manufacturing and fabrication work have been done using welding, milling, drilling, grinding and threading machines. The modified scissor jack is tested and implementing of design can solve ergonomics problems. Here also analyzing each object (scissor jack and pipe bending machine) with the help of cae tool Ansys workbench, and calculating maximum bearing capacity of the object, and knowing breakage point of each object. And also suggesting one new material which going to help us to increase the performance and durability of the object

INTRODUCTION

Jack

A jack is a mechanical lifting device used to apply great forces or lift heavy loads. A mechanical jack employs a screw thread for lifting heavy equipment. A hydraulic jack uses hydraulic power. The most common form is a car jack, floor jack or garage jack, which lifts vehicles so that maintenance can be performed. Jacks are usually rated for a maximum lifting capacity (for example, 1.5 tons or 3 tons). Industrial jacks can be rated for many tons ofload. The personal name Jack, which came into English usage around the thirteenth century as a nickname form of John, came in the sixteenth century to be used as a colloquial word for 'a man (of low status)' (much as in the modern usage 'jack of all trades, master of none'). From here, the word was 'applied to things which in some way take the place of a lad or man, or save human labour'. The first attestation in the Oxford English Dictionary of jack in the sense 'a machine, usually portable, for lifting heavy weights by force acting from below' is from 1679, referring to 'an Engine used for the removing and commodious placing of great Timber

SCISSOR JACK

Scissor car jacks usually use mechanical advantage to allow a human to lift a vehicle by manual force alone. The jack shown at the right is made for a modern vehicle and the notch fits into a jack-up point on a uni-body. Earlier versions have a platform to lift on a vehicle's frame or axle.

Electrically operated car scissor jacks are powered by 12volt electricity supplied directly from the car's cigarette lighter receptacle. The electrical energy is used to power these car jacks to raise and lower automatically. Electric jacks require less effort from the motorist for operation.

PIPE BENDING

Tube bending is any metal forming processes used to permanently form pipes or tubing. Tube bending may be form-bound or use freeform-bending procedures, and it may use heat supported or cold forming procedures. Form bound bending procedures like "press bending" or "rotary draw bending" are used to form the work piece into the shape of a die. Straight tube stock can be formed using a bending machine to create a variety of single or multiple bends and to shape the piece into the desired form. These processes can be used to form complex shapes out of different types of ductile metal tubing. Freeform-bending processes, like three-roll-push bending, shape the workpiece kinematic ally, thus the bending contour is not dependent on the tool geometry. Generally, round stock is used in tube bending. However, square and rectangular tubes and pipes may also be bent to meet job specifications. Other factors involved in the bending process are the wall thickness, tooling and lubricants needed by the pipe and tube bender to best shape the material, and the different ways the tube may be used (tube, pipe wires).

Literature review

V. SenthilRaja et al (2014) [1] designed and fabricated a mobile hydraulic pipe bending machine. They proposed that the hydraulic bender has higher productivity. Sometimes heat treatment is used during bending the pipe but the technique is unsafe because it causes many problems in the produced pipes namely wrinkles, curve formation, reduced thickness, hole forming, reduction in strength, makes it break easily. The hydraulic pipe bending machine based on press bending has superior characteristics as compared to one based on heat treatment methods. This type of bender is suitable for application in both industrial and domestic purposes.

E.O. Olafimihan (2015) [2] developed a bending machine based on hydraulic operation. He found the range of the levels upto which pipes were found to be oval to be in between 3% to 5%. The process of bending is economic when used for low & medium quantities due to less amount of tooling required. Portable bending machines make it convenient to perform multiple works on work pieces in theconstructional areas. The workforce involved in this field is not equipped with proper machine so as to provide uniformity in work piece instead they are using the tools which are harming as they are not able to provide the proper stress on the work piece.

CALCULATIONS

4.1 PIPE BENDING MACHINE DESIGN CALCULATIONS

Maximum Load =1500N.

Taking into account the heat losses a high value of 2000 was chosen Force to cause bending applied to the rollers (FB) =2000N Frictional force (Fr) = μ N Taking μ = 0.06 and N = 1500

The normal force being equal to the force to cause bending Total Force (Ft) = Force to cause bending + frictional force = (0.06 ×1500) + 1500 = 2090 N r = radius of the rollers Taking r to be 100mm for the radius of the pulleys

T=Torque applied to the rollers T = Fr = 2090N × 0.5m = 1045.0Nm t= thickness of the sheet L= roll gap

Size of the shaft subjected to gradually applied load.With little or no bending

force on the shaft, the available weight is the weight of the shaft • Km=0 , M =0
Hence we remain with Te = Kt ×T =1045 ×103Nmm and Me = Kt ×T = 1045 ×103Nmm We also know that equivalent twisting moment (Te).
Taking the value of the allowable shear stress to be equal to 100 Mpa.
Torque transmitted of the shaft from the motor = 1045 ×103Nmm 1045 ×103 = ×100×d3 = d = √(33221.4) = 37.60 mm
We know that the equivalent bending moment, Me= ×110×d3 =1045 ×103 = ×110×d3
d= 36.4mm
Taking the larger of the two values and from the standard sizes of shafts d = 40mm

SCISSOR JACK CALCULATIONS

- For Safe Design we take maximum capacity of jack = 1000kg
- Since while jack is used other three wheels are in contact with ground so assumption of
- 1000 kg is safe.
- Maximum Height of Jack = 355.4 mm
- Minimum height of jack = 254 mm
- Maximum Capacity of Jack is 1000 kg,
- So that Load (W) =1000□9.81=9810 N=10,000N(Approx.)
- Load (W) = 10,000 N
- Let, Coefficient of friction between threads(μ)= 0.25.
- The whole calculation is made by assuming material of cast steel. • Let L1, L2, L3, L4 be the length of each link.
- Such that L1 =L2 =L3 =L4 =160mm and W1+W2+W3 be the Length of power Screw • So, W1 = W3 = 150 mm W2 = 50 mm
- Max. Lift = (h1+h2) = 300 mm
- θ is the angle between Link with the horizontal when jack is at its lowest position.
- W = (Mass * g) = (1000*10) = 10,000 N = 10KN The tension T acting on the power screw
- Cos θ = (175-25)/160 = 20.36˚
- Tension, T = W/2*tan θ = 13473.37N
- Total tension = 2*T = W/tan θ
- Since we have assumed that the material for power Screw is a Cast Steel, so
- σt = 130N/mm2 Let dc be the core diameter of the screw.
- Load = (π/4)* dc^2 * σt 2*T
- = W/Tan θ = (π/4)* dc^2 * σt 2*T
- = 10000/Tan (20.36˚) = 26946.75598 N
- dc^2= [W/ Tan (θ)*4]/ [(π* σt)] = 263.9205794
- Hence, dc = 16.245mm
- Since the screw is subjected to torsion shear stress we take, dc = 16.2 mm, pitch P = 1.5mm
- Outer diameter, do = dc + P = (16.2+1.5) = 17.7 mm
- Mean diameter, d = do - P/2 = 17.7-1.5/2 = 16.95 mm
- Check for self-locking Tan (α) = Lead/π*d; α= helix angle Lead L = 2*P;
- since the screw has a double start square thread.
- Tan (α) = 2*p/π*d = 2*1.5/ π*16.95 =0.056338
- Helix angle; α= 3.22˚
- Coefficient of friction; μ = Tan ϕ = 0.25 & Friction angle: ϕ=14.03˚ • Since, ϕ >α hence the screw is self-locking Effort required to support the load is
- 2*T Tan (θ+α) =11761.54N
- Torque required to rotate the screw = effort*d/2 = 32506.42 N-mm
- Tensile stress σ t = 2*T/ (π/4) * dc^2 = 16168.05/ (π/4) * 12 2 = 142.95 N
- Shear stress in the screw due to torque ζ= 16*T/ (π* dc^3) = 23.82 N/mm2

• Maximum shear stress ζ max = √ ($\sigma t^2 + \zeta^2$)/2 = 72.460N/mm2
• Maximum principal stress σ t max = σ t /2 + √ ($\sigma t^2 + \zeta^2$)/2 = 143.93 N/mm2
• Since the maximum stresses σ t max and ζ max within the safe limit

4.2 Design of Nut

Let n is the number of threads in contact with the screw Let us assumed that the load is uniformly Distributed over the cross-sectional area of the nut.

Let Pb be the Allowable Bearing pressure between the threads.

Bearing pressure is assumed as 65 N/mm2 Pb = (2*T)/ ((π/4)*(do^2-dc^2)*n) 65 = (16168.05)/ ((π/4)*(13.52 -122)*n) Number of threads, n = 9.79 ≈ 10 In order to have good stability let n=10 Thickness of Nut = n*p = 10*2=20 mm Width of Nut b =1.5*do =1.5*13.5=20.25 mm To control the movement of nuts beyond 300 mm the rings of 8 mm thickness are fitted on the screw. The length of screw portion = 300 + (8*2) + 20 = 336 mm ≈ 350 mm Total length of screw is 350 mm.

4.3 Design of Top Arm

Since σ yt for cast steel be 260 N/mm² and assume Factor of safety (FOS) be 3.

Then, σ t = σ yt/FOS=260/3=86.66 N/mm² σ c = 1.25*σ t = 1.25*86.66 = 108.33 N/mm² Moment of Inertia I xx = 48007.56 mm4,

I yy= 51009.38 mm4

Radius of Gyration Rx= 15 mm,

R y= 13 mm Rankine constant (a) =1/7500 (For Steel) =1/9000 (For wrought iron) = 1/1600(For cast iron) a=σ c /π2 *E Cross section area (A) = (40*3) + (24*3) + (40*3) = 312 mm²

.4 Crippling Load in horizontal plane

Both Ends are Fixed (L eff = L/2) Crushing stress can be calculated by the formula =Π 2 *E*I/L2 =300N/mm2 Pcr = (σ c*A)/ [1+a*(L/2*Rx) ²] = (300*160*40)/ {1+ (1/7500)*(160/2*15)²} = 97660.2523 N

4.5 Crippling Load in vertical plane Both Ends are hinged (L eff = L) Crushing stress can be calculated by the formula =Π2 *E*I/L2 σc = 300 N/mm² Pcr= (σc*A)/ {1+a*(L/Ry) ²} = (300*312)/ {1+ (1/7500)*(160/13)2 } = 91764.70 N Since the Buckling load is greater than Design load, dimensions of the link safe.

5.4 4.6 Design of Bottom Arm Since σ yt for cast steel be 260 N/mm² and assume Factor of safety (FOS) be 3. Then, σ yt for cast steel = 260 N/mm² Factor of safety (F.S) = 3 σ t = σ yt/F.S=248/3=86.66 N/mm² σ c = 1.25*σ t = 1.25*82.66 = 108.33 N/mm² Radius of Gyration Rx= √Ix /A= 14.778 mm, R y=√Iy /A= 12.784 mm Rankine constant (a) =1/7500 (for Steel) =1/9000 (For wrought iron) = 1/1600(For cast iron) a=σc /π2 *E Ends are hinged (L eff = L/2) (Both ends are Hinged) Cross section area (A) = (40*3) + (30*3) + (40*3) = 330 mm² Moment of Inertia I xx = 72720 mm4, I yy= 54567.31 mm4

4.7 Crippling load in horizontal plane

Ends are hinged (L eff = L/2) (Both ends are Fixed) Crushing stress can be calculated by the formula σc =Π2 *E*I/L2 =300N/mm2 Pcr= (σc*A)/ {1+a*(L/2*Rx) ²} = (300*160*40)/ {1+ (1/7500)*(160/2*14.778)²} = 102589.15N

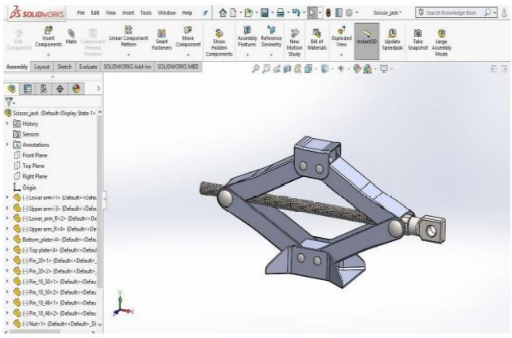
L eff=L (Both ends are hinged) Crushing stress can be calculated by the formula

σc =Π2 *E*I/L2 =300N/mm2 Pcr = (σc*A)/ (1+a*(L/ Ry) ²) = (300*330)/ {1+ (1/7500)*(160/12.784)²} =96963.76 N

Since buckling load is greater than the Design load, dimensions of the link safe.

4.8 Top Plate (Loading Platform)
P = 6000 N, L = 50 mm, B = 36 mm, H = 40 mm Moment, M = (P*L)/4 M = (6000*50)/4 =75000N-mm Z = (B*H2)/6 = (36*402)/6 = 9600 mm^3 σb = M/Z = 75000/9600 = 7.8125N/mm^2 The permissible stress for cast steel is 130 N/mm^2 and it is greater than σ b =7.81 N/mm2 the top plate design is safe.

DESIGN PROCEDURE:
Designing parts individually Assembly Steps to assemble the parts of gear less 90-degree transmission are: □
Save all the parts to the cloud storage. □
Open new tab and drag all the parts into that tab. □
For multiple parts, use COPY command and place the parts in the tab. □ Start assembling the parts using JOINT command. □
Select the type of joint in the JOINT command box. □
Complete the assembly



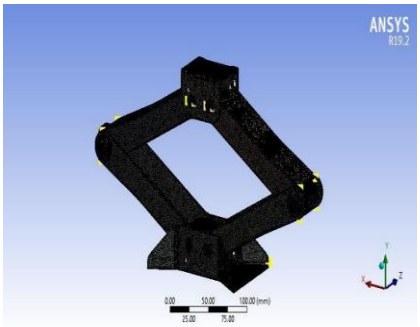
Design Of Scissor Jack

Assembly of Pipe Bending machine and Scissor Jack

Material selection
Mild-Steel
Young's modulus: - 2.0*10^11 Pa
Poison ratio: 0.29
Density: 7850 Kg/m^3
Yield strength: 250Mpa
Sae-1020
Young's modulus:- 2*10^11 Pa
Poison ratio: 0.29
Density: 7870Kg/m^3
Yield strength: 394.70 Mpa

Geometry□right click□import geometry□import iges format model After importing model just click on geometry option then we will get selection of material. From engineering data here we already applied the above mentioned materials.

Meshing
After completion of material selection here we have to create meshing for each object meshing means it is converting single part into no of parts. And this mesh will transfer applied loads for overall object. After completion meshing only we can solve our object. Without mesh we cannot solve our problem. And here we are using tetra meshing and the model shown in below



Results

Mild steel Deformation

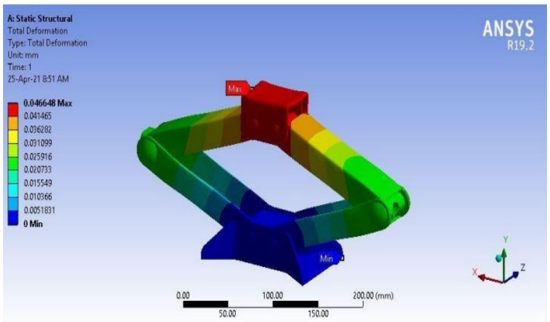
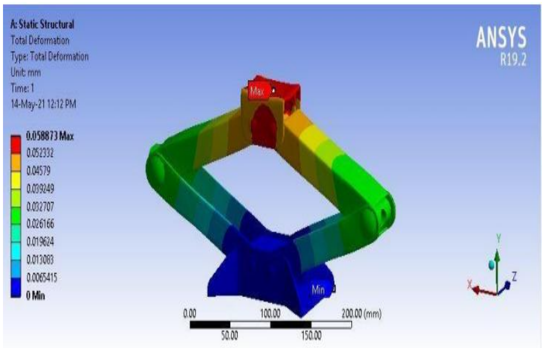


Fig 9.3: Deformation

SAE 1020

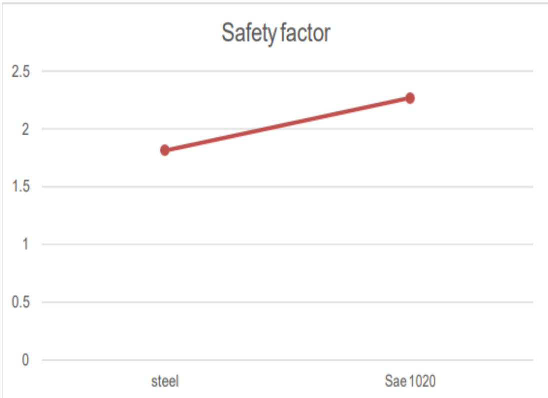
Deformation



Conditions of Steel and SAE 1020

	Steel	SAE 1020
Deformation (mm)	0.046648	0.058873
Stress (MPa)	137.98	173.89
Shear stress (MPa)	79.44	100.26
Safety factor	1.8123	2.2699

Graphs

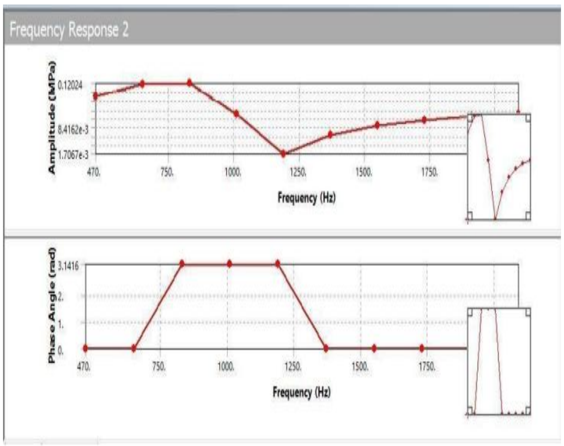


Dynamic analysis results for scissor jack
Different modes of SAE 1020 and steel

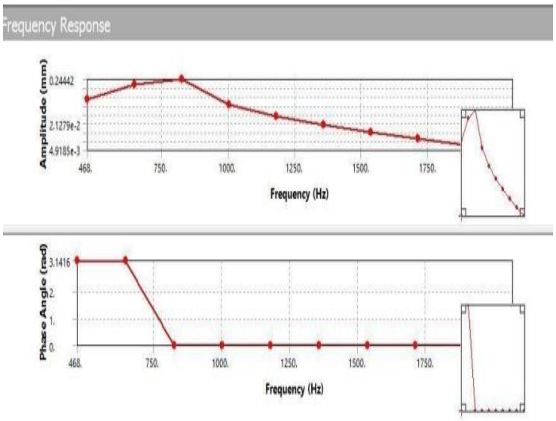
	Sae1020	steel
Mode 1 (Hz)	296.05	293.28
Mode 2 (Hz)	401.41	398.06
Mode 3 (Hz)	759.82	753.39
Mode 4 (Hz)	810	803.86
Mode 5 (Hz)	1102.5	1092.9
Mode 6 (Hz)	2085.4	2064.1

Scissor jack harmonic analysis results

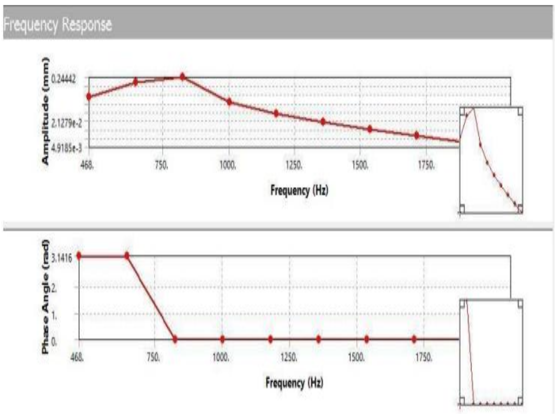
Steel Deformation frequency response



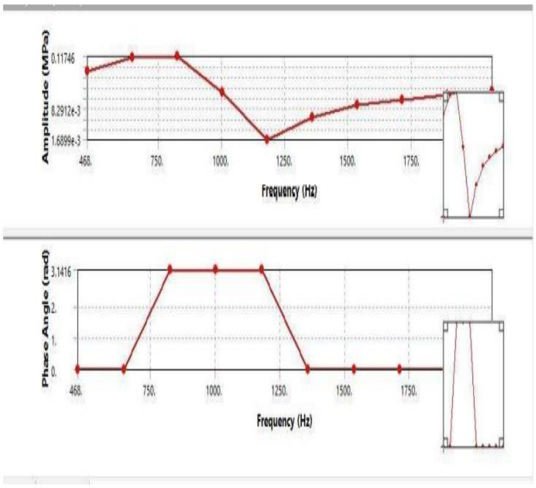
Stress frequency response



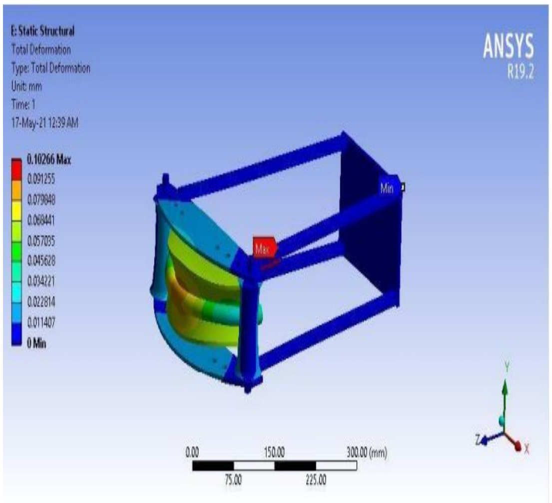
SAE 1020 Deformation frequency response



SAE 1020 Stress frequency response

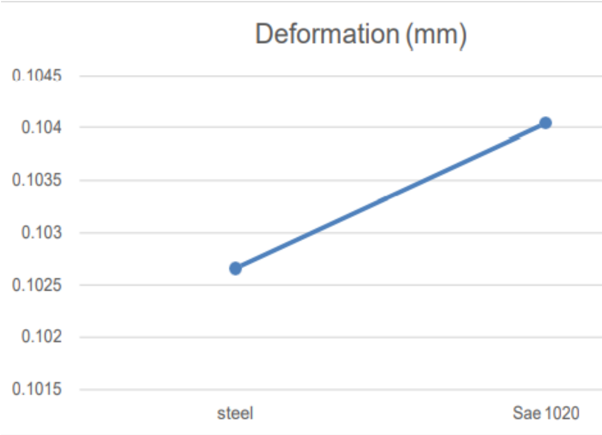


Pipe bending machine results



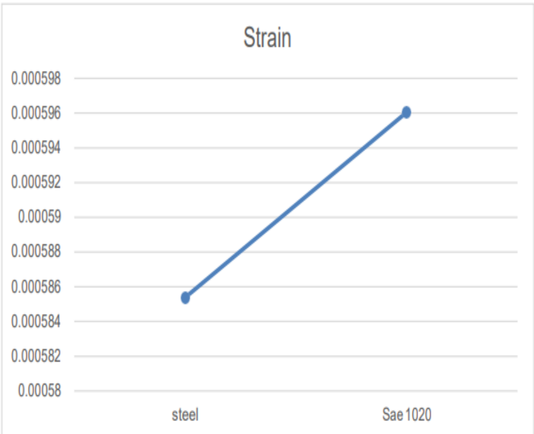
Boundary conditions of pipe bending machine

	Steel	SAE 1020
Deformation (mm)	0.10266	0.10404
Stress (Mpa)	109.8	110.13
Strain	0.00058538	0.00059605
Safety factor	2.2768	3.5838

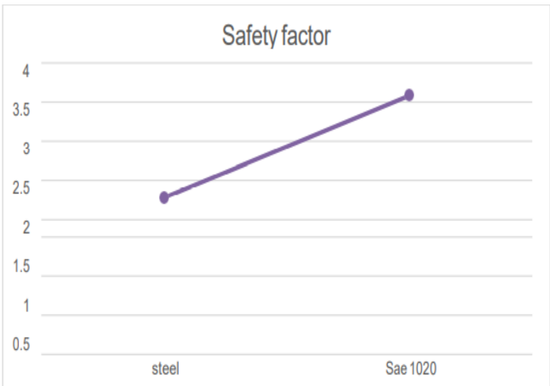


Deformation

Stress



Strain



Safety

Dynamic analysis for steel and Sae 1020

	Steel 1020	steel
Mode 1 (Hz)	30.166	29.895
Mode 2 (Hz)	49.61	49.175
Mode 3 (Hz)	56.352	55.931
Mode 4 (Hz)	141.49	140.29
Mode 5 (Hz)	158.08	156.61
Mode 6 (Hz)	292.11	289

Conclusion

In this thesis design and analysis of pipe bending machine with scissor jack assembly developed with the help of solid works tool, and then analyzed with the help of Ansys workbench, here calculating static analysis results of pipe bending machine and scissor jack separately, from this, it is clear that mild steel material with scissor jack can withstand maximum amount of load on it up to 700 Kgs, by changing material In to SAE 1020 the total strength of the object has been increased up to 20 to 30%, and pipe bending machine can withstand maximum amount of load above 1750Kgs on it, and same manner if we change it to SAE 1020 material strength will added more, and this SAE 1020 is low cost and high strength material, and it can increases the strength of the object, and it has chromium has compositions so that it avoid corrosions in future applications, so that object durability can increases,

From dynamic analysis results it is clear that in each case SAE 1020 has high amount of frequency range values compare to steel material. By knowing static and dynamic analysis results, with the help of dynamic analysis frequency value calculated harmonic analysis results and discussed frequency response of deformation and stress for each material,

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