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Study & Analysis of Wire Rolling Process in Drawing Operation to Modify Power Transmission System

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***Abstract*- The aim of this project worked is to analyze and redesign a machine to upgrade it to expand its manufacturing capabilities. With the Vinayaka Industry, Nagpur being an excellent metal wire producer, this machine required redesign to meet their quality standards Their current machine set-up could not properly produce material at its largest incoming material size of 16 mm. The technical team at the Vinayaka Industry had a general idea of how they thought the machine could be upgraded. Our project group worked together with the Vinayaka Industry to gather all the details of the necessary capabilities the machine must possess. From there, We have began designing upgrades to the machine including the larger bull drum. We have used calculations to ensure these changes did not compromise any of the machine's components. We calculated the maximum draw force needed to pull the 16mm diameter wire. Using the results, calculations were completed to find the maximum torque that will be put onto the gearbox and motor of the machine. These results were then compared to the published torque curve by the manufacturer for both the gearbox and motor. With all changes implemented, the machine worked as planned and allowed the machine to roll more than 16mm diameter of wire material without surface defects from improper setup.**

Index Terms- Torque, Force, Study and analysis.

INTRODUCTION

Vinayaka Group is a well-established in the manufacturing & traders of Stainless Steel Mild Steel, Carbon Steel, Alloy & Tool Steel, Die Steel & Bright Bars, Copper, Brass, Aluminium, Gun Metal, P.B. in various shapes & sizes. Vinayaka Group was incorporated in Nagpur year 1997. The Group has been scaling new heights in the market. Vinayaka Industry has one of the drawing units with them. In the existing system, they can draw & roll the wire of a maximum 16 mm diameter. But now they want to increase the drawing capacity which may process higher diameter wires. They had tested by taking 24 mm diameter wire but during operation, they found vibrations, jerks, surface deformation & improper rolling. Therefore the company is trying to improve the existing system by modifying the current transmission system with optimum cost. Vinayaka Industry, a wire manufacturing company located in Hingna MIDC Nagpur, is in need of improving one of their wire-drawing machines. The previous machine was not designed to process the desired size of the product they would like: the Wire encounters itself, which in turn breaks down the quality of the wire and causes surface deformation. Outlining the progress We have has made since beginning this project. In this final report, we will outline the machine's previous state, and compare it to its current state.

With our newly designed methods. The machine being evaluated for this project is a wire-drawing machine, which pulls intermediate sized wire through wire-drawing dies to reduce the cross-sectional area. Typically, the machine is largest capable of producing wire from incoming diameters of 16mm (smallest) to 24mm (largest). The dies used for this process are usually made up of either polycrystalline diamond or carbide. When operating the machine, the wire is fed through a die box that contains an abrasive material and a wire die. The abrasive material is applied at the face of the die where the wire enters, which reduces the size of the wire. After the die, the wire is transferred onto a bull drum. The bull drum is what drives the motion of the wire; they are large circular objects that are each attached to a gearbox and a motor that causes continual rotational speed.

LITERATURE REVIEW

From the literature review, we found that the machine being evaluated for this project is a wire-drawing machine, which pulls intermediately sized wires through wire-drawing dies to reduce the cross-sectional area. Typically, the machine is capable of producing wire from incoming diameters of less than or equal to 16mm. The dies used for this process are usually made up of either polycrystalline diamond or carbide. When operating the machine, the wire is fed through a die box that contains a lubricant and a wire die. The lubricant is applied at the face of the die where the wire enters, which reduces the size of the wire. After the die, the wire is transferred onto a bull drum. The bull drum is what drives the motion of the wire; they are a large circular object that is attached to a gearbox and a motor that cause continual rotational speed. We were able to obtain information regarding this motor from their website. The purpose of the motor is to use to drive the gearbox, which drives the speed rate of the wire-drawing machine. On average, the motor is set to 20 hp and 1440 RPM. An idler is a slightly smaller circular object that controls the alignment of the wire around the bull drum and prevents it from encountering each other. The wire is then transferred to the cutting machine where it cut at the required length

PROPOSED METHDOLOGY

From the literature review, We have has collected information for the machine's motor, gearbox, differential, bull drum and have put forth this information into useful data calculations. From the analysis of collected data on these four machine components, we will try to find out which of these factors are critical and need to be modified or replaced by our redesign of the machine. Using this information, we can calculate the torque of the motor, as well as the average force applied. After performing design calculation we will use Ansys software to verify that our design calculations are conforming with the obtained result of the simulation. Then we will suggest these changes to be made in the wire drawing machine to the Vinayaka industry and achieve the outputs of the experimentation that our project results in successful operation of the machine without failure.

CALCULATION

1) Drawing force calculation

i) For SAE1040 without considering friction

Drawing Stress (Ϭ2) = Ϭy ((1+B)/B)[1-Ai/Ao]B

Yield strength (Ϭy) = 415 Mpa

μ = 0.3

Angle of deformation (α) = 22.4160

Ao = π/4 * Do2= 201.06 mm2= 2.01*10-4m2

Ai = π/4 * Di2= 143013 mm2= 1.43*10-4 m2

B = μ Cot α = 0.30*Cot (22.416) = 0.727

Ϭ2= 415 * ((1+0.727)/0.727)* [1-143.13/201.06]0.727

= 215.814 MPa

Drawing force (F)= Ϭ2* Ai

= 215.814*143.13

= 30889.561 N

Torque required

T = F*r

=30889.561*0.409

=12633.830 N-m

ii) By considering Friction

ϬT = Ϭy + (Ϭ2 – Ϭy) e -(2μL)/R1

Radius of work piece at exit (R1) = 6.75 mm

Length of deformation cone (L) = 13.03 mm

ϬT= 4165 + (215-415) e -(2*0.3*13.03)/6.75

= 352.191 MPa

Force (F) = ϬT* A1

= 50409.126 N

Torque (T) = F*r

= 20617.332 N-m

Diameter (Do-DI) in mm	Area (Ao -Ai) in mm2	Stress without considering friction in mpa	Respective force in N	Stress considering friction in mpa	Respective force in N
14-11.5	153.938 -	245.232	25471.75	371.41	38577.82

	103.868				
16-13.5	201.06-143.13	215.81	30889.56	352.19	50409.13
18-15.5	254.47-188.69	192.65	36349.81	333.70	62962.22
20-17.5	314.16-240.53	173.97	41844.65	316.36	76094.4
22-19.5	380.132-298.65	158.56	47363.62	300	89594.1
24-21.5	452.289-363.05	145.71	52901.1	284.87	103422.42
26-23.5	530.93-433.74	134.76	58451.99	270.93	117513.83

Motor torque at 1455 rpm

$P = \frac{2*\pi*N*T}{60}$

$15*10^3 = \frac{2*\pi*1450*T}{60}$

T = 98.78 Nm

Maximum torque generate at 2.472 rpm

$15 * 10^3 = \frac{2*\pi*2.472*T}{60}$

T = 57944.76 N m

For 16 mm rod

F = 50409.13

for safety assuming 20 % extra

F = 50409.13*1.2

F = 60490.956 N

Required torque at output

T = F*r

T = 60490.96* 1/2

T = 30245.48 N m

$T_{req} < T_{output}$

For 24 mm rod

F= 103422.416

F = 103422.416*1.2

F = 124106.8992 N

Required torque

T= F*r

T=124106.416*0.2

T_r =51711.208 N m

$T_{req} < T_{output}$

From above, we can say that the motor is generating enough power to draw the 24 mm rod.

Safety calculation

1. Gear

1] Tooth Load (ft) = Pd/Vp

Design Power (Pd) = 15000 w

$Vp = \frac{\pi*Dg*Ng}{60*1000}$

= 0.095 m/s

Where,

Vp – Pitch line velocity

Dg – Diameter of gear

Ng – speed of gear

Ft = Pd/Vp

= 15000/0.075 = 200000 N

This force is transmitted by two gears so the load on each gear equals to

Ft=200000/2= 100000

for safe condition Fb ≥ Ft

Ft = Fb = 100000 N

Fb = So*Cv*b*y*m

Where,

Fb- Bending strength

So- Basic Strength

Velocity factor (Cv) = 3/3+Vp = 0.97

Face width of gear (b) = 110 mm

Modified Lewis factor (y) = 0.44

Module (m) =10

100000 = So*0.97*110*0.44*10

So = 213 Mpa

As So (213) < So of material of shaft i.e. En 9 (So=240) design is safe.

2] Pinion

1] Tooth Load, Ft = Pd/Vp

$Vp = \frac{\pi*Dg*Ng}{60*1000}$

$Vp = \frac{\pi*178*8.654}{60*1000} = 0.08$ m/s

Ft = 15000/0.08 = 187500 N

This load is transmitted by two gears Ft = 187500/2 = 93750

let Ft = Fb

93750 = So*0.97*110*0.341*10

So = 237.66 Mpa

As So (237.66) < So of material of shaft i.e En 9 (So=240) design is safe.

3] Differential

1] Tooth Load, Ft = Pd/Vp

$Vp = \frac{\pi*Dg*Ng}{60*1000}$

$Vp = \frac{\pi*460*8.654}{60*1000} = 0.20$

Ft = 75000 N

75000 = So*0.93*102*0.44* 7

So = 244.7 Mpa

As So (244.7) < So of material of shaft i.e SAE 1045 (So=245) design is safe.

Design calculation

4] Shaft

1] Force on Shaft due to gear 1
Force = ρ*g * volume
Volume = $\pi/4* d^2*w = 0.02463\ m^3$
$\rho = 8055\ Kg/m^3$
F = 0.02463*8055*9.81 = 1950 N
2] Force on Shaft due to bull drum
Force = g* ρ* volume
$\rho = 8055\ Kg/m^3$
Volume = $0.026\ m^3$
Force = 0.026*9.81*8055
= 2055 N
3] Tangential Force, Ft =T/ Rg
Ft =104294.82 N
Where,
T = Torque = 57944.76 N-m
Rg = Radius of gear = 0.5 m
4] Radial Force, Fr =Ft*tanφ
Fr =3796.21N

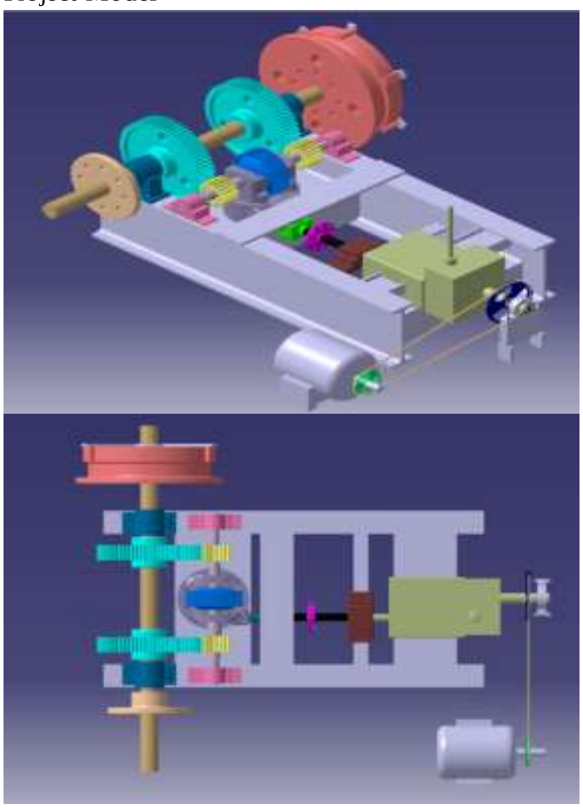
5] Reactions on gear
RA + RB = 214545 N
∈Ma = 0
106245*0.15 + 106245*0.77 + 2055*0.12 = Rb *0.92
RB = 108925.43 N
RA =105619.56 N
Bending Moment on shaft,
In Vertical plane
A=E=0
B = 575.4 Nm
D = - 15455.16 Nm
C = 2055*1.05 + 106245 *0.62 – 108925.43*0.77
= -15842.93 Nm
In Horizontal plane
$R_A + R_B$ = 179342.83 N
= 179343 N
∈Ma = 0
37960.021*0.15 + 37960.21*0.77 + 103422.41*1.2 = R_B* 0.92
R_B = 172859
R_A = 648398
Bending moment on shaft,
A = E = 0
B = 28958.27 Nm
C = 103422.416*1.05 + 179943*0.77 + 37360.21* 0.62
= - 5965.249
Resultant moment
A= 0
B = 16928.74 Nm
C = 24138.40 Nm
D = 28963.71 Nm
$\tau = \frac{16*103}{\pi*d2}\sqrt{Kb*M+Kt*T}^2$
Where,
τ =Shear stress
d= diameter
Kb=moment constant
Kt=torque constant
T=torque
M= Bending moment
$232.5 = \frac{16*103}{\pi*d2}\sqrt{1.75*289632 2+1.5*57944}^2$
d = 119.88 mm
= 120 mm

2] For Shaft 2 ,
1) Reaction on gear, In Vertical
$R_A + R_E$= 447740 N
∈Ma = 0
$R_E = \frac{31480+12880+1616}{0.92}$
R_E= 223870 N
Bending Moment on shaft,
$M_A = M_E$ =0
M_D = 33580.65 N-m
M_C =37920.96 N-m
M_B = 33581.27 N-m
2) In Horizontal
$R_A + R_E$= 225585
∈Ma = 0
$R_E = \frac{11480+33550+58770}{0.92}$
R_E= 112795 N
Bending Moment on shaft,
$M_A = M_E$ =0
M_D = 16920 N-m
M_C = 28225 N-m
M_B = 16920 N-m
Total,
M_D = 37603 N-m
M_C = 47273 N-m
M_B = 37605 N-m
$D^3 = \frac{16*103}{\pi*232.5}*\sqrt{1.05*(16775)2+1.75*(47273)2}$

= 112.97 mm
= 115 mm
M = C
V*g*ρ = V*g*ρ
$\pi/4*d^2*L = \pi/4*d^2*L$

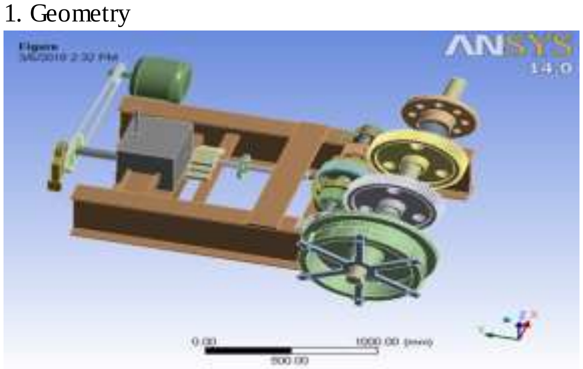
L = 1246.07 mm
Weight of wire (W) = V*g
Weight of wire = 10.47 Kg
Force = g*W
= 102.72 N
Speed,
74/15 = N2 / 8.65
N2 = 42.62
Torque = Force / Speed
103.85/42.62 = 2.43
Torque = 2.43 N-m

Project Model

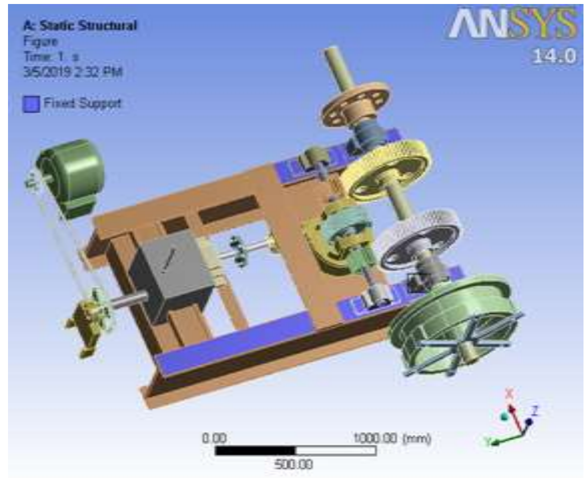


Software Analysis

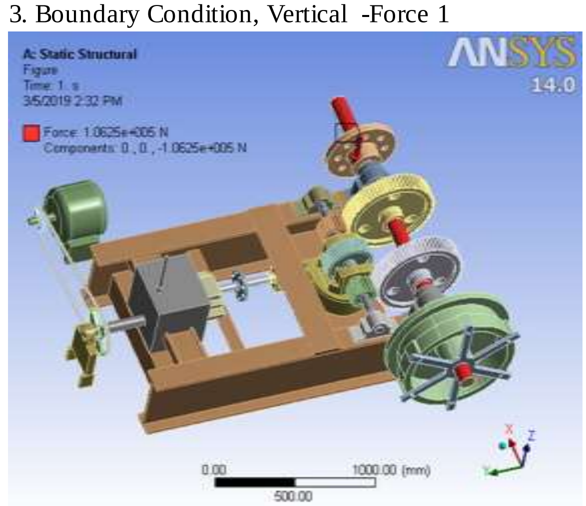
1. Geometry



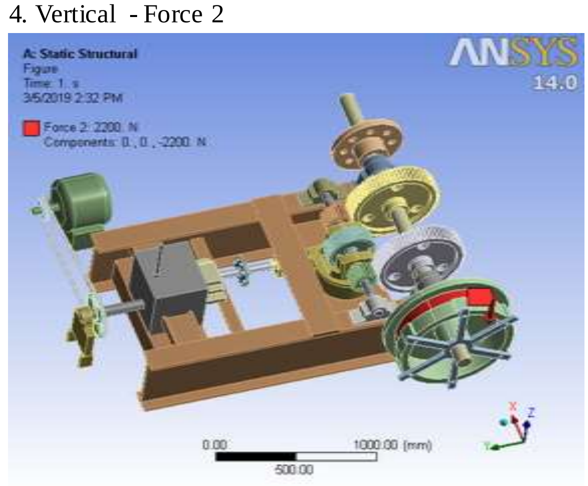
2. Fixed Support



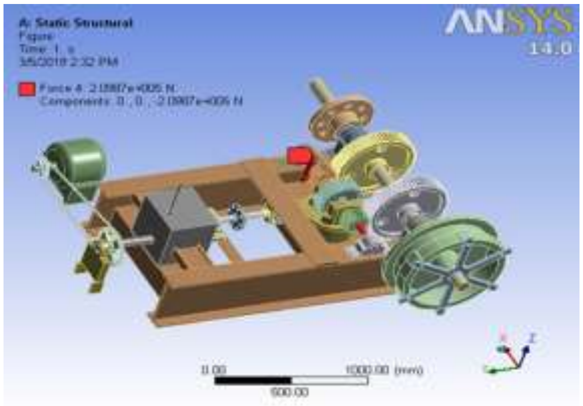
3. Boundary Condition, Vertical -Force 1



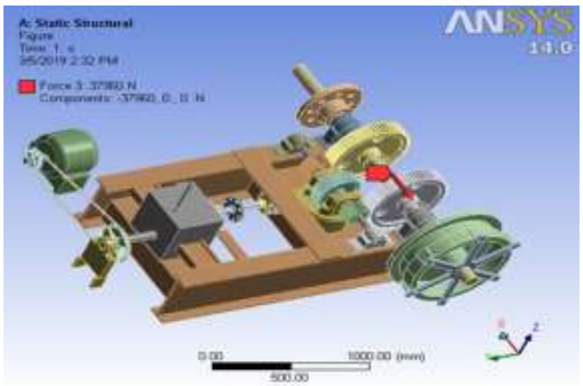
4. Vertical - Force 2



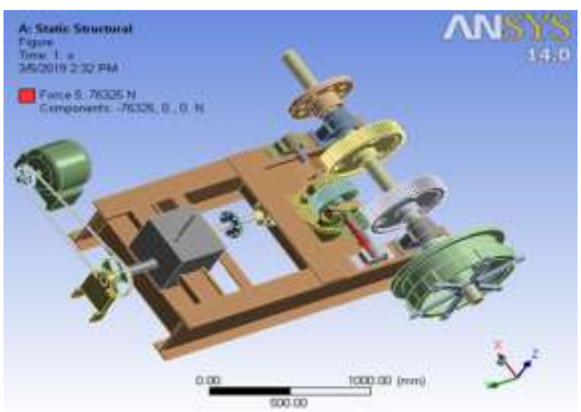
5. Vertical Force 3



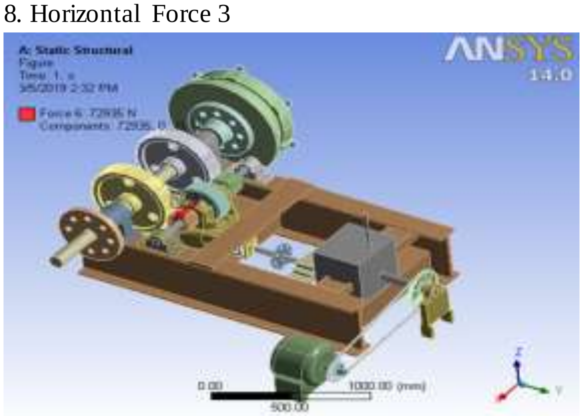
6. Horizontal Force 1



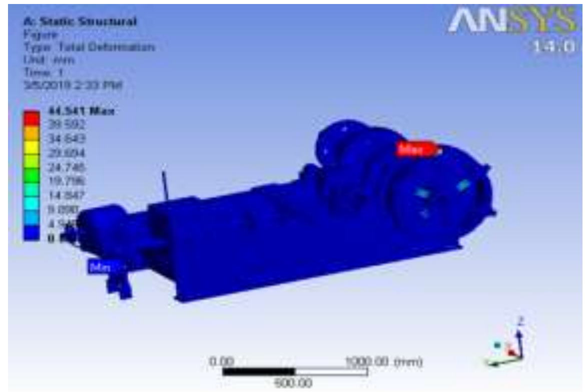
7. Horizontal Force 2



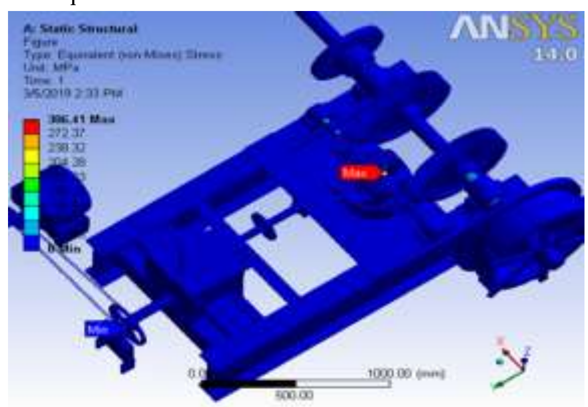
8. Horizontal Force 3



9. Total Deformation

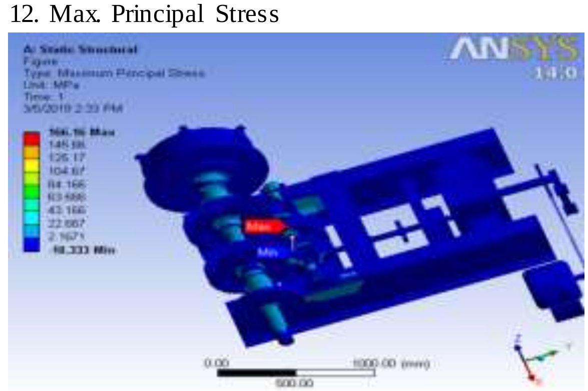


10. Equivalent Stress



11. Shear Stress

12. Max. Principal Stress

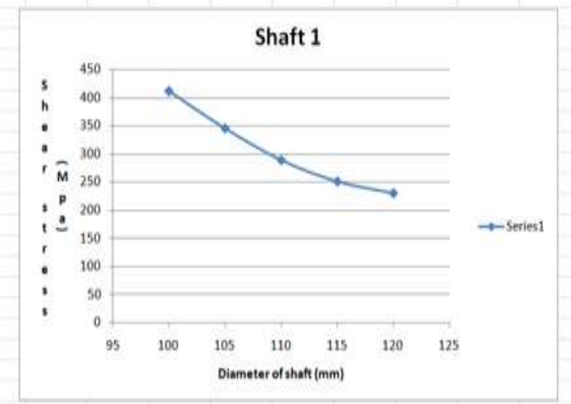


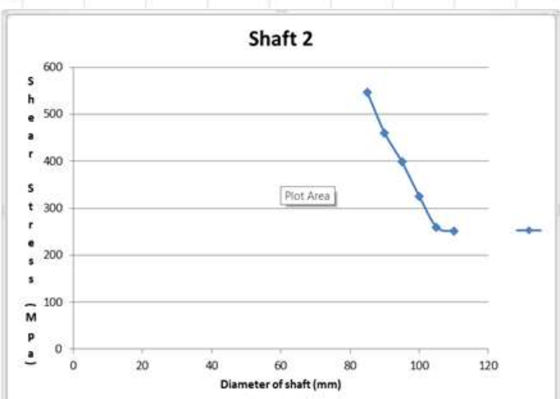
RESULT

Based on the calculations and using those values for the software analysis the following results are obtained as shown in table.

PARAMETERS	EXISTING SYSTEM	PROPOSED SYSTEM
DIAMETER OF MAIN SHAFT	100 MM	120 MM
DIAMETER OF 2ND SHAFT	85MM	110MM
TORQUE	30245.48 N-M	51711.208 N-M
WIRE DIAMETER	16-13.5 MM	24-21.5 MM

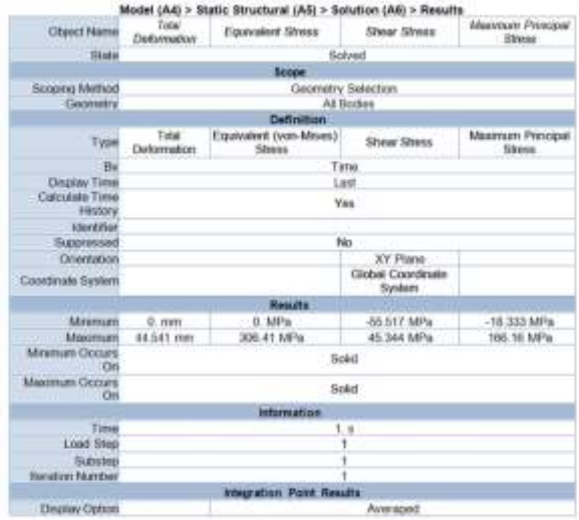
Graphical representation





From the above calculation we have seen that the current setup of wire drawing machine is generating enough power to draw 24 mm diameter rod or bar but if we draw the greater diameter bar some components may get damage. So to reduce the stresses we are increasing the diameter of shaft and providing an extra support. And From the analysis we are cross checking our work. The graph shows the relation between the diameter and stresses produced in shaft. we see that as diameter increases the stresses induced in shaft decreases.

ANALYSIS RESULT


Model (A4) > Static Structural (A5) > Solution (A6) > Results

Object Name	Total Deformation	Equivalent Stress	Shear Stress	Maximum Principal Stress
State	Solved			
Scope				
Scoping Method	Geometry Selection			
Geometry	All Bodies			
Definition				
Type	Total Deformation	Equivalent (von-Mises) Stress	Shear Stress	Maximum Principal Stress
By	Time			
Display Time	Last			
Calculate Time History	Yes			
Identifier				
Suppressed	No			
Orientation			XY Plane	
Coordinate System			Global Coordinate System	
Results				
Minimum	0. mm	0. MPa	-55.517 MPa	-18.333 MPa
Maximum	44.541 mm	306.41 MPa	45.344 MPa	166.16 MPa
Minimum Occurs On	Solid			
Maximum Occurs On	Solid			
Information				
Time	1. s			
Load Step	1			
Substep	1			
Iteration Number	1			
Integration Point Results				
Display Option		Averaged		

CONCLUSIONS

From the project we can conclude that it is not feasible to draw wire of 24 mm in the present setup of drawing machine but after changing the diameter of shaft from 100mm to 120mm also the internal diameter of gears, bearings and pulleys mounted on shaft changes, providing support to it from bull drum side with no changes in power consumption. Hence our proposed design can be implemented for drawing machine of "Vinayaka Industries".

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