DESIGN AND DYNAMIC ANALYSIS OF70T DOUBLE GIRDER ELECTRICAL OVERHEAD CRANE

[1]APEKSHA.K.PATEL, [2] PROF. V.K.JANI,

[1]M.E.[CAD/CAM] Student, Department Of Mechanical Engineering, C.U.Shah College Of Engineering and Technology, Wadhavan, Gujarat

[2] Professors,Department Of Mechanical Engineering, C.U.Shah College Of Engineering and Technology, Wadhavan, Gujarat

Apeksha_prod09@yahoo.com, veera.jani@rediffmail.com

ABSTRACT: *Main Component of Overhead Crane is Girder Beam which transfers load to structural member. In Present Practice, industries overdesign girder beam which turns costly solution. So, our aim is to reduce weight of girder which has direct effect on cost of girder and also performance Optimization is done for fatigue (life) point of view. In this paper FE analysis of girder beam is carried out for the specific load condition i.e. turning operation. Here, we done a mathematical design calculation crane component, and thrust forces are used in FE analysis. Here, we used ANSYS WORK BENCH V12.1.Software for the FE analysis of the girder beam. Through this analysis we get the result in terms of stresses and deformation and this result are within the allowable limits.*

Keywords—70T double girder electrical overhead crane, dynamic analysis of girder.

1. INTRODUCTION

Crane and hoisting machine are used for lifting heavy loads and transferring them from one place to another.A crane is a lifting machine, generally equipped with a winder (also called a wire rope drum), wire ropes or chains and sheaves that can be used both to lift and lower materials and to move them horizontally.

It uses one or more simple machines to create mechanical advantage and thus move loads beyond the normal capability of a human. Cranes are commonly employed in the transport industry for the loading and unloading of freight, in the construction industry for the movement of materials and in the manufacturing industry for the assembling of heavy equipment.

Material handling is a vital component of any manufacturing and distribution system and the material handling industry is consequently active, dynamic, and competitive.

Main Component of Overhead Crane is Girder Beam which transfers load to structural member. In Present Practice, industries overdesign girder beam which turns costly solution. So, our aim is to reduce weight of girder which has direct effect on cost of girder and also performance

Optimization is done for fatigue (life) point of view.during the machining process results in chatter marks on the machined surface and thus creates a noisy environment. Higher cutting speeds can be facilitated only by structures which have high stiffness and good damping characteristics. The deformation of machine tool structures under cutting forces and structural loads are responsible for the poor quality of products and which in turn is also aggravated by the noise and vibration produced.

2. DESIGN CALCULATION OF CRANE COMPONENT

2.1 Basic Calculation of 70 ton EOT Crane

Total Lifting Capacity (W) = 70 ton
= 70 X 10000 N
= 700000 N

Lifting Height = 29.95 meter
= 29.95 X 1000
= 29950 mm

No. of rope parts (n_t) = 12

Efficiency of pulley (η_p) = 94%

Number of bends (n) = 11

From Design Data Book, for n =11,

$$\frac{D_{min}}{d} = 23 \tag{1}$$

Where,

Dmin = minimum diameter of drum or pulley

d = Diameter of rope

Load for this arrangement

$$P = \frac{700000}{\eta_p X no.of ropes} \tag{2}$$

$$= \frac{700000}{12 \; X \; 0.94}$$

p = 62056.74 N

2.2 Rope Design

Select Standard Rope size is 6 X 37

Where,

6 are the stands in the wire rope.

37 is the number of wire in each stand.

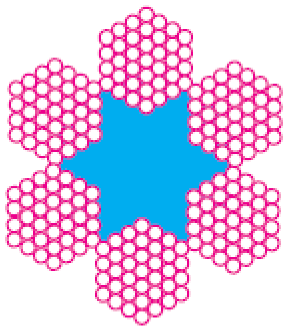


Figure 2.1 Wire rope

Now wire diameter (d_w) = 0.045d
Modulus of Elasticity of wire (E_f) =8 X 10^4 N/mm^2
Ultimate breaking stress for rope (σ_u) = 1500 N/mm^2
Factor of Safety n_f = 4

Area of Rope $A = \frac{P}{\frac{\sigma_u}{n_f} - \frac{d_w X E_f}{D_{min}}}$ (3)

$= \frac{62056.74}{\frac{1500}{4} - \frac{0.045dX8X10^4}{23\,Xd}}$

A = 284mm^2

But,
A = 0.4 X d^2=284 mm^2
So, d = 26.64 mm
From Design Data Book,
Take available standard d = 28 mm
Now approximate weight for rope = 1.48 kg/m
Braking Strength per rope = 206000 N
Required Breaking Strength per rope = P X n_f
= 206000 X 4
= 824000 N

So, design is safe.
Rope Size = 6 X37-48

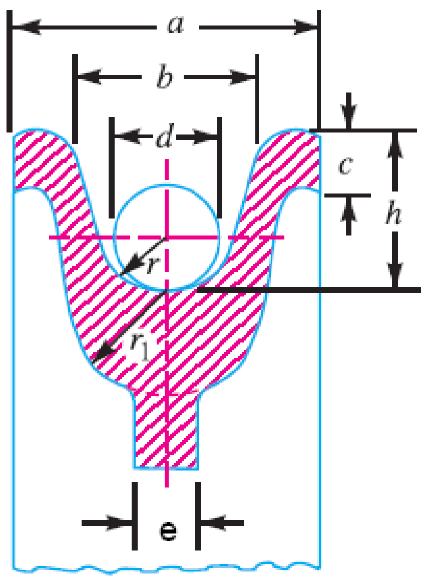


Figure 2.2 Pulley

$\frac{D_{min}}{d} = 23$
As d=28mm we get,
Minimum Diameter of Pulley = 23 X 28 = 644 mm
It is advisable to take diameter of pulley = 27d
So, D = 27 X 28
= 756 mm
Take D = 756 mm
Now other dimension for sheaves or pulley is as follows:
a = 2.7 X d = 2.7 X 28 = 75.6 mm
b = 2.1 X d = 2.1 X 28 = 58.8 mm
c = 0.4 X d = 0.4 X 28 = 11.2 mm
e = 0.75 X d = 0.75 X 28 = 21 mm
h = 1.6 X d = 1.6 X 28 = 44.8 mm
Diameter of Compensating Pulley,
D_1 = 0.6 X 756 = 453.6 mm
Take D_1 = 454 mm
Other dimensions are same as lower pulley.

2.3 Design of Drum

Diameter of drum = Diameter of Pulley
= 756 mm Number of turns on each side of drum

$z = \left(\frac{H_i}{\pi D}\right) + 2$ (4)

$= \frac{(29950\ X\ 2)}{3.14\ X\ 756} + 2$
= 27.23 = 28
Number of Turns = 28 turns

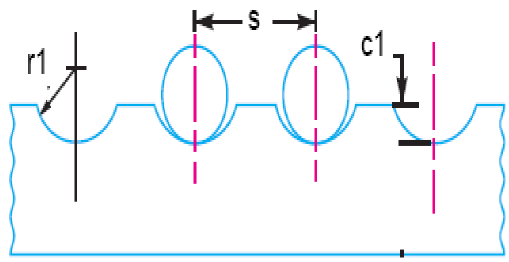


Figure 2.3 Drum

From Design Data Book,
S = 30 mm
r_1 = 16.5 mm
c_1 = 12.5 mm
l_1 = Free Space between each side = 150 mmFullLength of Drum

$L = \left[\left(\frac{2H_i}{\pi D}\right) + 12\right] XS + l1$ (5)

$= \left[\left(\frac{2\ X\ 29950}{3.14\ X\ 756}\right) + 12\right] X\ 30 + 150$
= 1267 mm

Take, L = 1270 mm
The Wall Thickness of drum (w) = 0.02 D + 10
= (0.02 X 756) + 10
= 25.12 mm
Wall Thickness (w) = 26 mm

Checking drum stresses,

Crushing Stress (σ_c) $= \frac{P}{wXs}$ (6)

$= \frac{62056.74}{26\ X\ 30}$
= 7 9.55 N/mm^2

Torque on drum = P X $\frac{D+d}{2}$ (7)

= 62056.74 X $\frac{756+28}{2}$

= 24326242.08 Nmm

2.4Hook Design

Design of hook for trapezoidal section:

Inner Diameter of hook(C) = $KX\sqrt{W}$ (8)

= 12X$\sqrt{\frac{700000}{1000}}$

= 317.49 mm

Take inner diameter of Hook(C) = 317.5 mm

From Design Data Book other parameters are as follows,

Depth (H) = 0.93 X C = 295.275

Base of the section (M) = 0.6X C = 190.5

Throat (J) = 0.75 X C = 238.125

Radius of Curvature of hook (E) = 1.25 X C

= 396.875

Radius of the base of the section(R) = 0.50 X C

= 158.75

The Overall Height of Hook Portion (A) = 2.75 X C

= 873.125

Radius of the corner (Z) = 0.12 X C

= 38.1

Design of Shank Portion:

W = $\frac{\pi * d_c^2 * \sigma_t}{4}$ (9)

$d_c^2 = \frac{W*4}{\pi * \sigma_t}$

= (700000 X 4)/ (3.14 X80)

d_c = 105.58 mm

Nominal Diameter (G1) = 105.58/0.84

= 125.69 mm

Checking of Hook Section

Inner Radius (r_i) = C/2 = 317.5/2 = 158.75mm

Outer Radius (r_o) = r_i + H = 158.75 + 295.275

= 454.025

R= r_i+ (H/3) = 158.75 + (454.025/3) = 310mm

Radius at neutral axis (r_n) = $\frac{\frac{b_1+b_2}{2}Xh}{\frac{b_1r_2-b_2r_1}{h}log_e\frac{r_2}{r_1}-(b_1-b_2)}$

$= \frac{\frac{MXH}{2}}{\frac{Mr_0}{H}log\frac{r_0}{r_i}-M}$

$\frac{\frac{190.5\ X\ 295.275}{2}}{\frac{190.5\ X\ 454.025}{295.275}log_e\frac{454.025}{158.75}-190.5}$

= 239.76 mm

Eccentricity (e) = R - r_n

= 310 – 239.76

= 70.243 mm

h_1 = r_n -r_i

= 239.76 – 158.75

h1 = 81.01 mm

h2 = r_0 - r_n

= 310 -239.76

= 70.24 mm

Area of Section = M X H/2= 190.5 X 295.275/2

Area of Section = 28124.94 mm2

Bending stress in hook (σ_b) = $\frac{W\ X\ R\ X\ h_1}{A\ X\ e\ X\ r_i}$

$= \frac{700000\ X\ 310\ X\ 81.01}{28124.94\ X\ 70.243\ X\ 158.75}$

= 56.052 N/mm2

Direct stress in hook (σ_t) = $\frac{W}{A} = \frac{700000}{28124.94}$

= 24.89 N/mm2

Total stress in hook = $\sigma_b + \sigma_t$

= 56.052 + 24.89

Total stress in hook = 80.94 N/mm2 [12] [13]

3. DYNAMIC ANALYSIS OF GIRDER FOR 70T E.O.T CRANE

3.1 Introduction

An approximation for the component deformation was introduced by means of a weighted sum of constant shape functions. When dealing with the task of deriving these functions the finite element method can be very effective. Here, a well-known and widely used concept known as component mode synthesis (CMS) can be used. One of the most common approaches (Craig and Brampton 1968) is based on the idea of using normal mode analysis techniques to calculate eigenvectors for use as shape functions, or shape vectors, respectively. While employing eigenvectors for approximation was already very widespread, Craig and Bampton among others (Hurty 1965) enhanced the method by taking into account additional types of vectors. In the following, Craig and Bampton's method is dealt with in more detail since it is also implemented in MSC.ADAMS/Flex. Here, the following types of vectors or modes are utilized:

1. Fixed boundary normal modes
2. Static correction modes

Fixed boundary normal modes are eigenvectors that result from a finite element normal mode analysis. They are connected with the boundary condition implying that all nodes of the finite element model are fixed at which forces and joints that is applied within the multi body system. In the following sections, these nodes are referred to as interface nodes. Static correction modes are deformation vectors that result from static load cases with which loads are applied to interface points. Typically, a unit load is applied to every nodal coordinate, whereas all other interface nodes are fixed. This leads to six static correction modes for each interface node. Figure 2 illustrates some mode shapes for a one-dimensional bar. The shapes (a) and (b) are fixed-boundary normal modes, shapes (c) and (d) are static correction modes resulting from a unit displacement (c) and a unit rotation (d), respectively. The use of static correction modes ensures a good approximation of the deformation when forces and moments are applied to interface points. The fixed boundary normal modes are important as soon as high frequency excitation is expected, i.e., if the loading may not be considered "quasi-static". Note: In the following, the flexible component is always assumed to be represented by a finite element model.

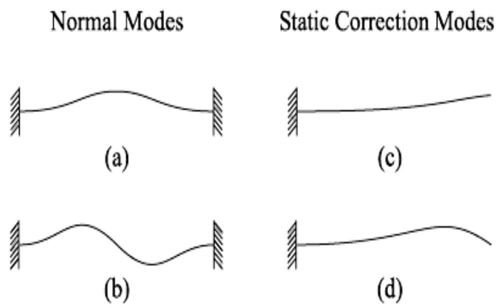


Figure 3.1 Mode shapes of one-dimensional bar

3.2 Benefits of Modal Analysis:-

1. Allows the design to avoid resonant vibrations or to vibrate at a specified frequency (speaker box, for example).
2. Gives engineers an idea of how the design will respond to different types of dynamic loads.
3. Helps in calculating solution controls (time steps, etc.) for other dynamic analyses.

3.3 Steps of Dynamic Analysis

Model Analysis

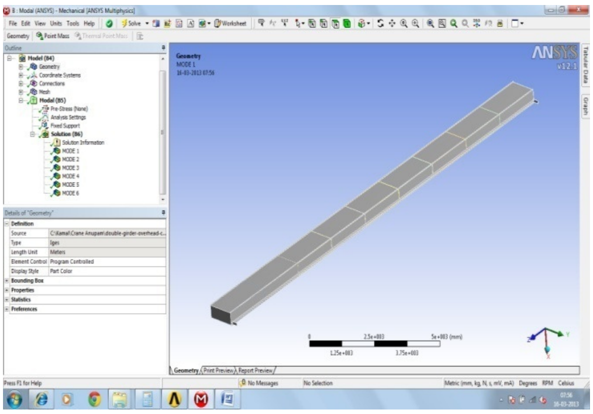


Figure 3.2 Geometry of girder using dynamic analysis

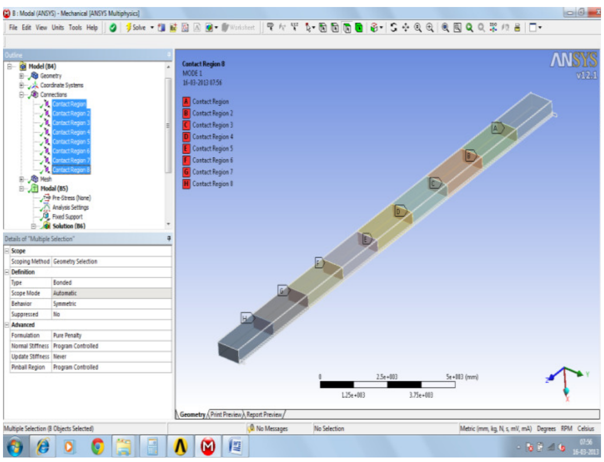


Figure 3.3Connection between parts

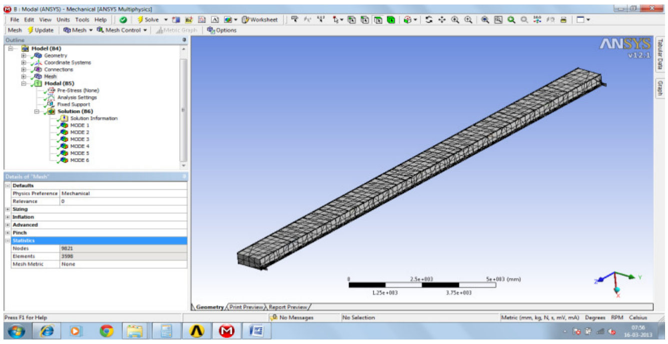


Figure 3.4 Mesh Model Of Girder Beam

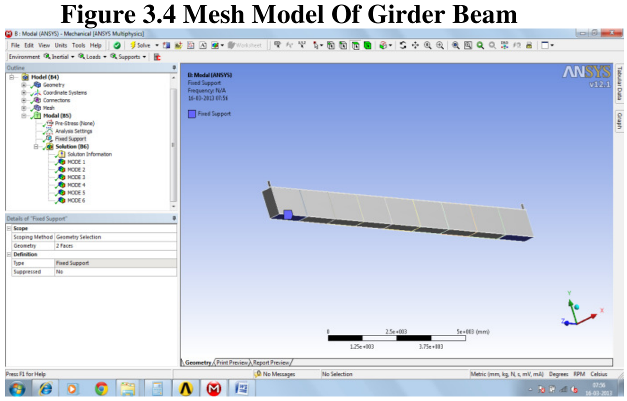


Figure 3.5 Application of Fixed Support

Results of Analysis

Mode – 1

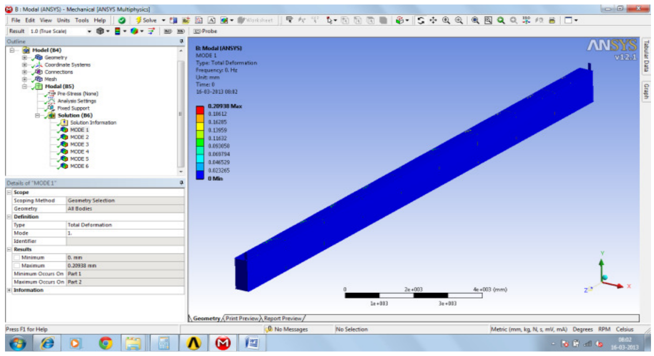


Figure 3.6 Total deformation of mode 1

Mode-2

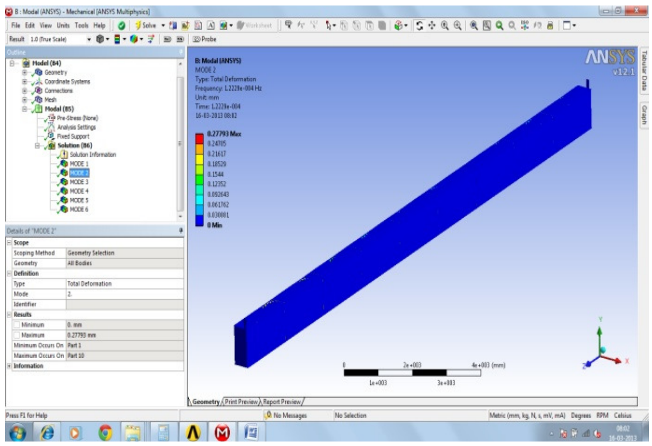


Figure 3.7 Total deformation of mode 2

Mode-3

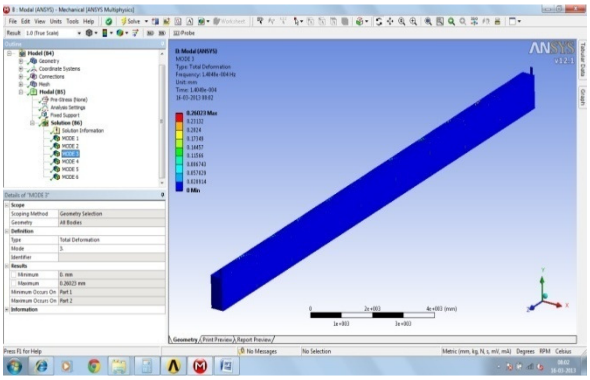


Figure 3.8 Total deformation of mode

Mode-4

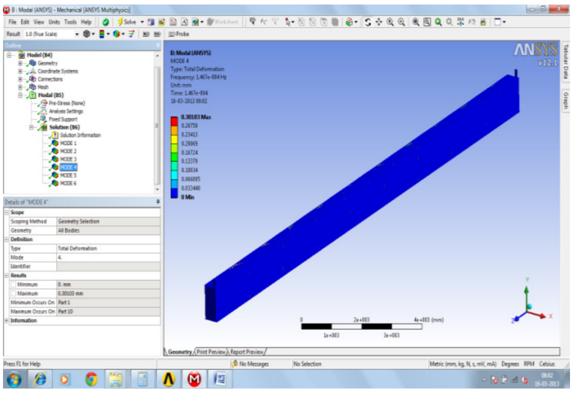


Figure 3.9 Total deformation of mode 4

Mode-5

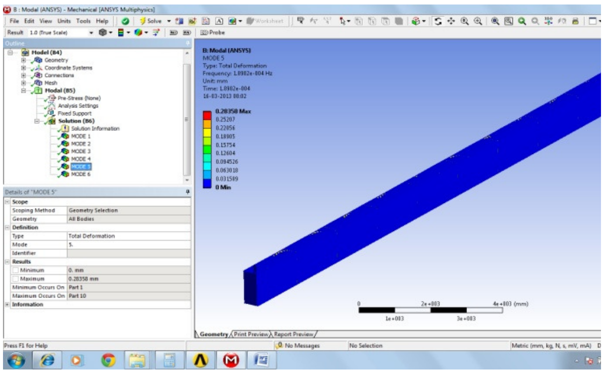


Figure 3.10 Total deformation of mode 5

Mode-6

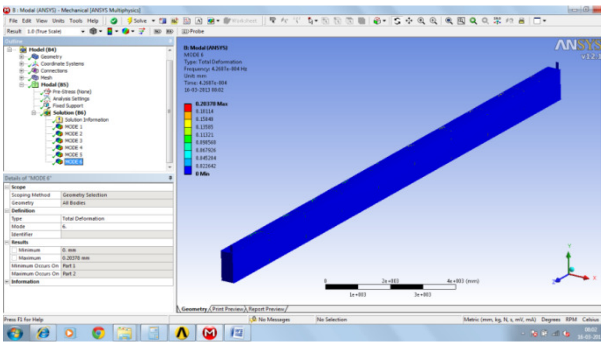


Figure 3.11 Total deformation of mode 6

4. Transient Analysis Of Overhead Crane Component

4.1 Introduction

Transient structural analysis provides users with the ability to determine the dynamic response of the system under any type of time-varying loads. Unlike rigid dynamic analyses, bodies can be either rigid or flexible. For flexible bodies, nonlinear materials can be included, and stresses and strains can be output.Transient structural analysis is also known as time-history analysis or transient structural analysis.

Transient structural analyses are needed to evaluate the response of deformable bodies when inertial effects become significant.If inertial and damping effects can be ignored, consider performing a linear or nonlinear static analysis instead.

If the loading is purely sinusoidal and the response is linear, a harmonic response analysis is more efficient If the bodies can be assumed to be rigid and the kinematics of the system is of interest, rigid dynamic analysis is more cost-effectiveIn all

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K(x)]\{x\}=\{F(t)\}$$

other cases, transient structural analyses should be used, as it is the most general type of dynamic analysis In a transient structural analysis, Workbench Mechanical solves the general equation of motion:

4.2 Some points of interest:

Applied loads and joint conditions may be a function of time and space. As seen above, inertial and damping effects are now included. Hence, the user should include density and damping in the model.Nonlinear effects, such as geometric, material, and/or contact nonlinearities, are included by updating the stiffness matrix.

Transient structural analysis encompasses static structural analysis and rigid dynamic analysis, and it allows for all types of Connections, Loads, and Supports.

4.3Step of Transient Analysis

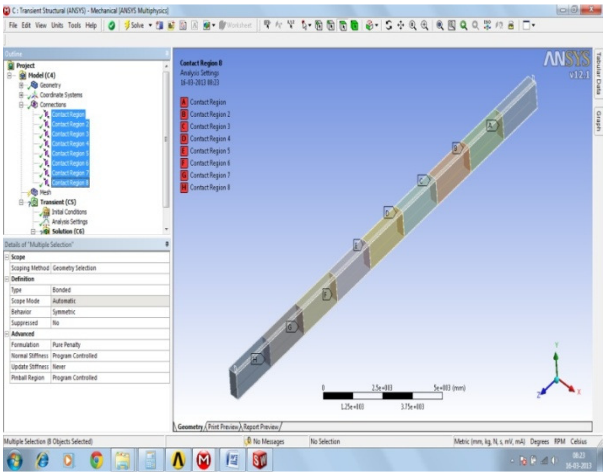


Figure 4.1 Connections between Parts

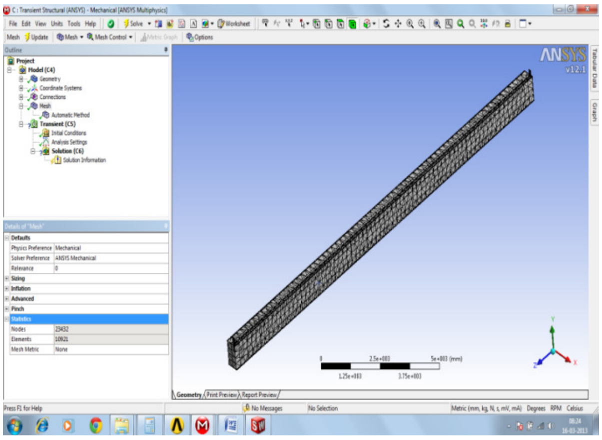


Figure 4.2 Mesh Model of Girder Beam

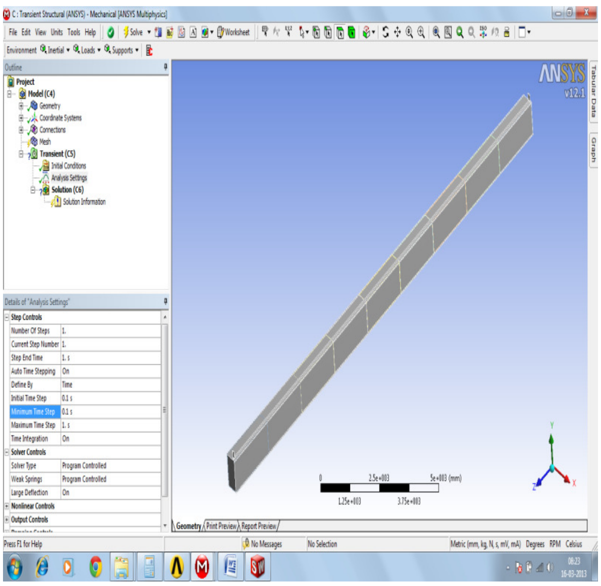


Figure 4.3 Geometry of Girder Using Transient Analysis

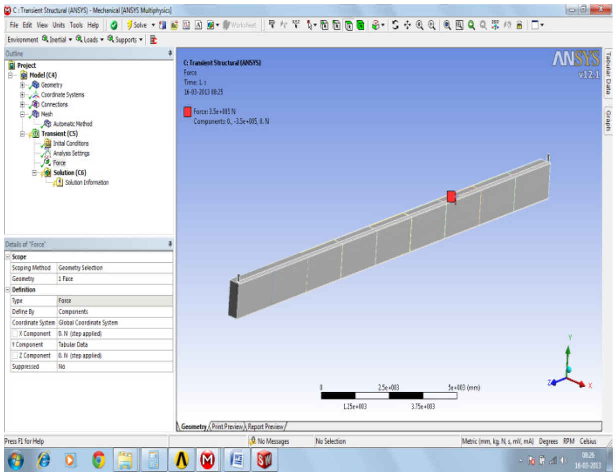


Figure 4.4 Application of Excitation Load

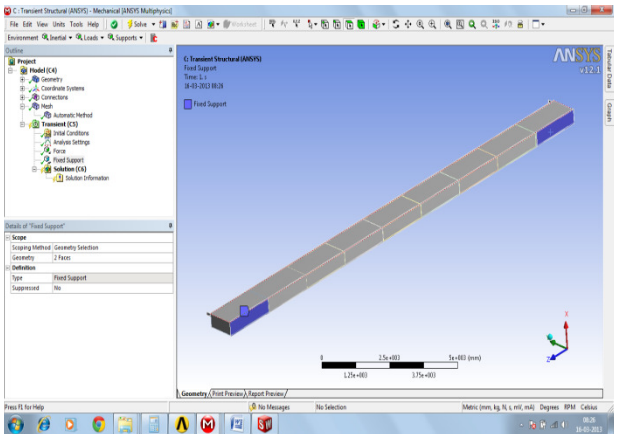


Figure 4.5 Application of Fixed Support

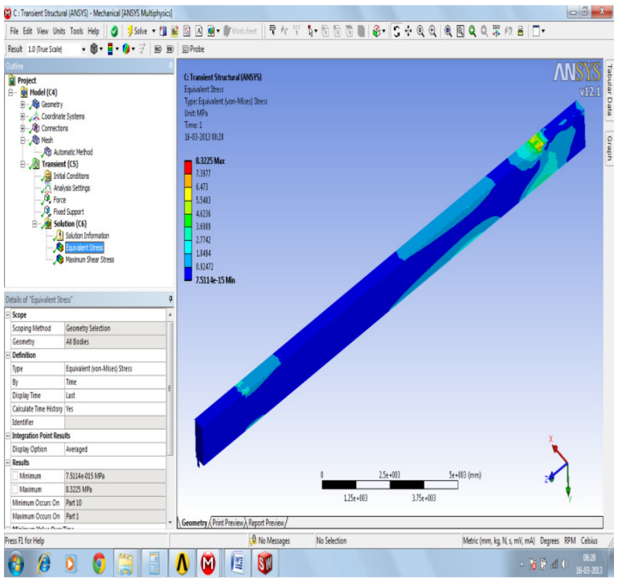
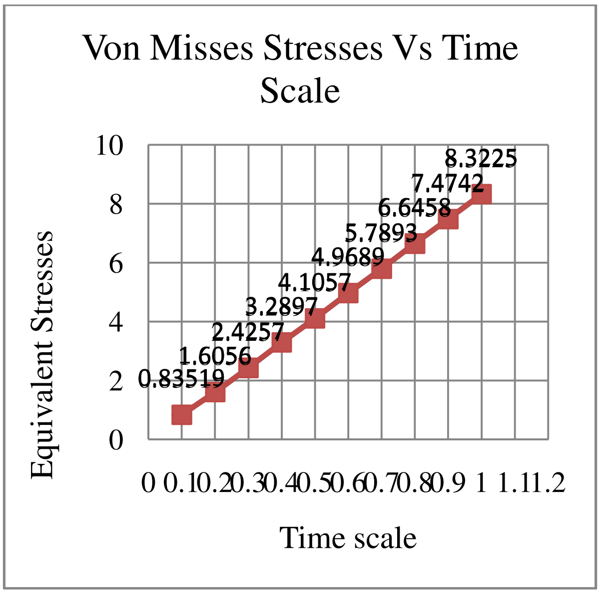
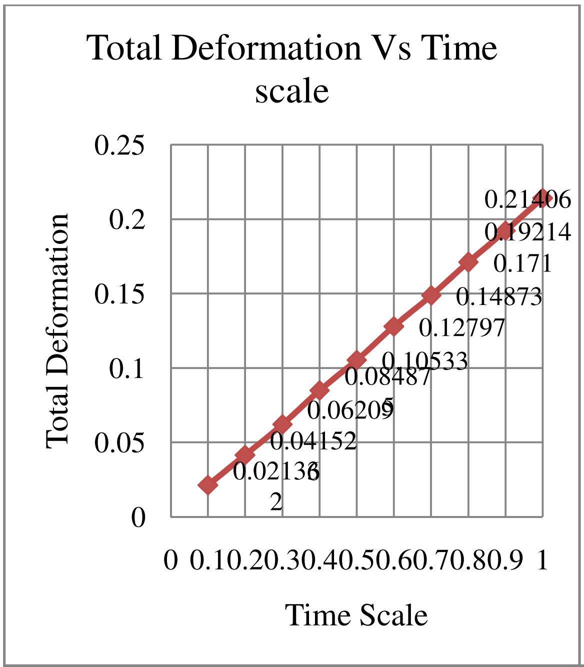
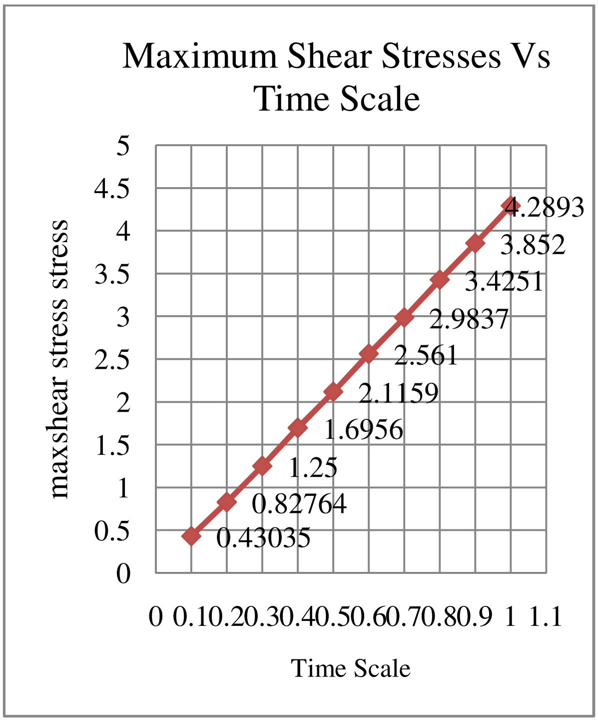
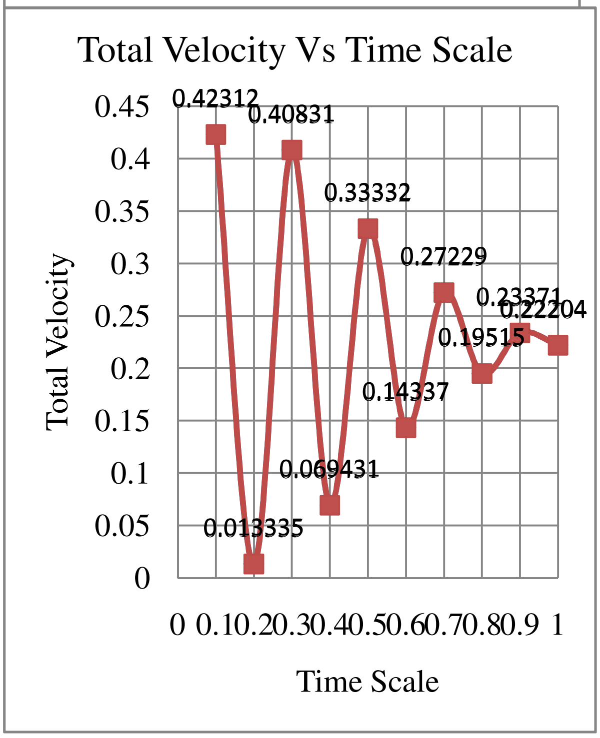


Figure 4.6 Von-Misses of Fixed Support


Von Misses Stresses Vs Time Scale
Equivalent Stresses
Time scale
0.83519
1.6056
2.4257
3.2897
4.1057
4.9689
5.7893
6.6458
7.4742
8.3225


Total Deformation Vs Time scale
Total Deformation
Time Scale
0.21406
0.19214
0.171
0.14873
0.12797
0.10533
0.08487
0.06209
0.04152


Maximum Shear Stresses Vs Time Scale
maxshear stress stress
Time Scale
4.2893
3.852
3.4251
2.9837
2.561
2.1159
1.6956
1.25
0.82764
0.43035


Total Velocity Vs Time Scale
Total Velocity
Time Scale
0.42312
0.40831
0.33332
0.27229
0.23371
0.22204
0.19515
0.14337
0.069431
0.013335

5. CONCLUSION

parameter	result	allowable value
max.von misses stress	8.3225 mpa	240 mpa
max.shear stress	4.2893 mpa	130 mpa
max.deformation	Result	Allowable Limit
mode-1	0.20938 mm	0.5 mm
mode-2	0.27713 mm	
mode-3	0.26023 mm	
mode-4	0.30103 mm	
mode-5	0.28358 mm	
mode-6	0.20378 mm	

As shown from analysis that the maximum stress is 8.3225 mpa which is within limit and also prove that over design of girder ,so the further scope of work is optimization of design and weight of girder for the cost point of view.

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