

USING FINITE ELEMENT ANALYSIS FOR DESIGN OPTIMIZATION OF TWIN BEAM CRANE ARM

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ABSTRACT: Gantry Crane is major industrial tool for objects loading/unloading job. Very minor concentration is given to the optimal design of crane arms in recent era. In an Overhead Crane the major component is its Twin beam which transfers load to structural part. Currently, industries overdesign the beam due to ease of design and fabrication which results in costly product. Use of latest design optimization tools is necessary due to the increase in the prices of the goods. This research demonstrates design optimization of Twin beam crane arm. A straightforward and innovative method is adopted to use nonlinear optimization code for design of twin beam and then evaluated the outcome with the FE simulation. The values of allowable bending stress, Maximum shear stress and deformation are treated as constraints to suggested limits of required model, considering the mass of the arm as goal. Optimal arm designed in this way is proficient performance in accordance with formulation method and found as economical as possible.

Key Words: Optimization of Crane, Finite Element Analysis, Twin beam

INTRODUCTION

Cranes and hoisting machines 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. Major part of overhead crane is its beam which exchanges load to structural part. In present practice, crane manufacturers overdesign twin pillar crane arm which results expensive equipment. Thus, author's point is to decrease amount of material used in the beam which has immediate impact on expense of overall mass of the structure, the initial cost for the structural building and electrical power consumption for the crane.

In the research, the aim is to use the computer aided engineering tools during the design phase so as to make the design as perfect and flawless as possible. The research methodology is based on Finite Element Analysis method. For this purpose, a Twin Beam Crane Arm is modelled in SOLIDWORKS and its Finite Element Analysis is executed in the same software package.

LITERATURE SURVEY

The increasing cost of structural material and energy consumption is a worldwide issue thus ideal utilization of the both can't be ignored. Overhead crane, which is an equivalent word for material handling, uses structural steel for its beam for its operation. For overhead heavy duty cranes, Light arm saves material cost as well as trim down power consumption during its operation. For configuration and optimization of twin beam crane arm, the common methodology is refined through direction stipulated in the predominating codes and benchmarks. Profile standardization of crane arm cross section was studied by Gibczynska [1]. While keeping the mass and cost as constraints, the ideal configuration of straightforward symmetrical welded box pillar Jarmai K [2] joined curving anxiety, shear force and clasp imperatives. Parametric and numerical work on load carrying beam at single point was carried out by T.H.Hallak [3]. Narayan [4], in his two successive papers analyzed quality limit of networks with rectangular slits. He also emphasized on expectation of anxiety in case of rectangular sections.

Niezgodzin'skia [6] measured clasping issue of trusses in Twin bar crane arms because of welding of the supporting plates.

The study on bar sections by G.j. Hancock's [7] is very supportive for the examination and locking and quality of dainty walled structures.

For the improvement of cool structured I-bar and open segments, the research and results of Magnuki [10, 11] is considerable. Heo et al's work is useful for the current design work for Twin bar crane arm regarding flimsy walled shut bar segment outline [12]

The imagination at the back of this exploration work is to show computational improvement and Finite Element Analysis plan advancement instruments in the territory of overhead crane scaffold outline which is an exceedingly intricate issue in the feeling of its dynamic nature of operation. This postulation proposes a strategy to focus streamlined outline parameters of bar crane arm for the Twin shaft crane arm against negligible weight destination, using numerical techniques (Grg2) which is accessible solver in the MS Excel program [14]. An overhead crane can be a solitary bar setup and also a twin shaft sort for which the Twin bar is fundamentally subjected to two (or more) focused vertical wheel stacks. As per the application and use of an overhead heavy crane, it is subjected to moving load not withstanding different other loads. Consolidated curving and shear stresses, greatest avoidance and locking variables suggested in CMAA-70 standard are followed for design. The design functions are lowest part rib, web and top sizes and dimension along. Whole design has been carried out in computational way and in addition Numerical model Separated from the compass of the pillar [14].

SPECIFICATIONS OF CRANE

In the execution of analysis, data that has been gathered from the industry as a base and the data input to the analysis. The data includes geometry and design as explained below with impact loading factor of 1.25 (ASME B30.20 Safety Standard for below-the Hook Lifting Devices).

Type = Gantry Crane

Load = 35 ton = 343 350 N

Loading Factor = 1.25

Column Height = 16.01 m = 16010 mm

Span = 23m = 23000 mm

CLASSIFICATION OF THE GANTRY CRANE

Out of many standards and regulations that regulate various aspects of crane equipment, one of them is the classification according to the intensity of work in which the number of cycles designed into its parameters. Gantry-cranes included in the category of "heavy" are known to have:

Designed cycles (N): 2,000,000 cycles

Designed cycles per day (n): 350-500 cycle

In optimization of the arm in this study, author used the British Standard 2573: Rules for the design of cranes as standard. Crane classified as U7 in the list of classes as in Table 1 is selected for design.

Table 1: Classification of Cranes

Class of Utilization	Maximum Operating cycles	Remarks
U1	3.2×10^4	Infrequent use
U2	6.3×10^4	
U3	1.25×10^5	
U4	2.5×10^5	Fairly frequent use
U5	5×10^5	Frequent use
U6	1×10^6	Very frequent use
U7	2×10^6	Continuous use
U8	4×10^6	
U9	$> 4 \times 10^6$	

SPECIFICATION OF MATERIALS AND CONSTRUCTIONS

In addition to the data for the design aspect and class, this study is also based on the technical specifications and construction materials which are vital parameters in the analysis of the strength of the structure as listed below. Material selected for the arm is ASTM A36, whose properties are listed below.

Modulus of Elasticity ranges between 200000-210000 MPa (N / mm²).

Poisson's ratio of 0.26.

Tensile Yield Strength 250 MPa (N/mm²)

Compressive Yield Strength 250 MPa (N/mm²)

Ultimate Strength 460 MPa (N/mm²)

Steel Density 7850 Kg/m³

(Source: Steel Construction Manual, ASME)

THE DIMENSIONS OF THE ARM

The girder is designed using rectangular hollow profile as shown in following Figure with the specifications listed in Table 2, 3a, 3b and the Figure 1 is the demonstration of the dimensions of arm.

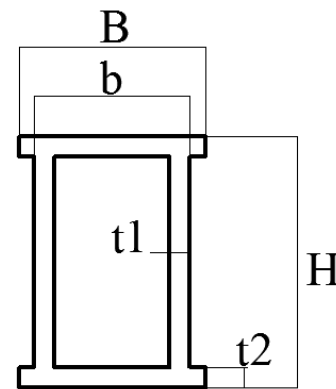


Figure 1: Design Optimization Process

Table 2: Girder Dimensions

Size				Thickness	
H x B		b	y	t1	t2
mm	mm	mm	mm	mm	mm
1550	650	600	775	8	16

Table 3a: Girder Specifications

Sectional Area	Mass/unit	
A	w	w
cm ²	Kg/m	kN/m
465.67	400.78	3.93

Table 3b: Girder Specifications

Moment of Inertia		Radius of Gyration	
I _{xx}	I _{yy}	r _{xx}	r _{yy}
cm ⁴	cm ⁴	cm	cm
1650535	300341	64.72	25.39

CALCULATIONS AND SIMULATION

Design calculations of the arm on the basis of engineering mechanics theories were carried out. The analysis based on the finite element method for more comprehensive results is then compared with theoretical results. Four Static loading cases are studied as tabulated in Table 4.

Table 4: Four Test Cases

Load case	Load (Ton)
1	5
2	15
3	17.5
4	35

MODELING OF THE ARM

The modeling of the prevailing design of the twin beam crane arm is carried out in the modeling software SOLIDWORKS. Afterwards, the Finite Element Analysis of the modelled part was carried out in the same software by assigning values for the constraints, design variables and goals of different parameters of the model. The Figure 2 shows the 3D solid model created in SOLIDWORKS and the Figure 3 shows the step of material assignment to the model for analysis.

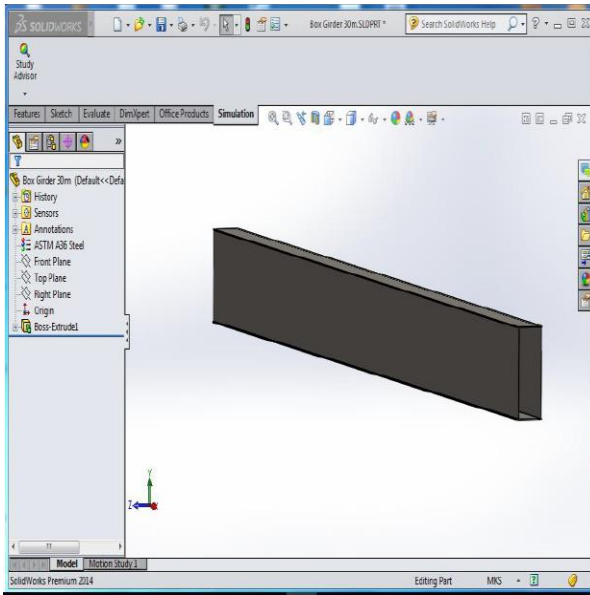


Figure 2: Model of the arm

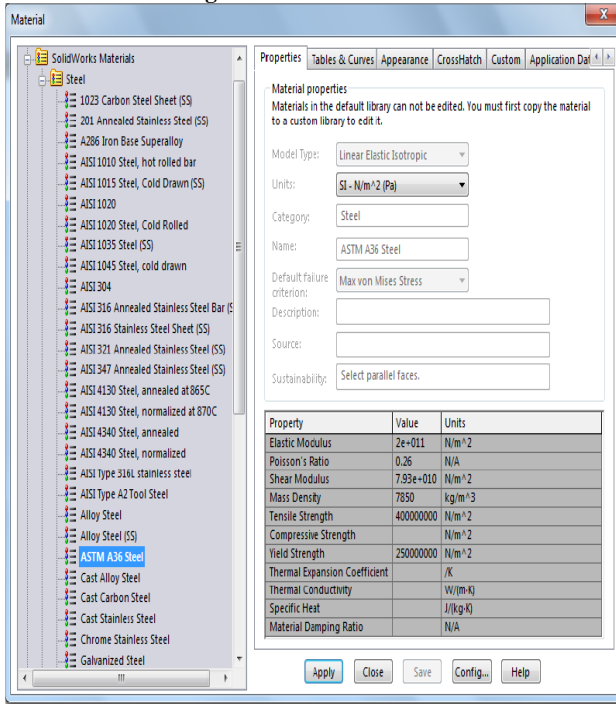


Figure 3: Material Specifications

MESHING (DISCRETIZATION)

Meshing process is done with parameters in accordance with the contents of Table 5 using tetrahedron cell type with resulted in a non-structured smooth mesh, as shown in Figure 4.7.

Table 5: Meshing Parameters

Properties	Values
Relevance center	Fine
Element size	250 mm
Smoothing	Medium
Transition	Fast
Minimum angle length	6 mm
Transition ratio	0.272
Growth rate	1.2
Nodes	333099
Elements	176593

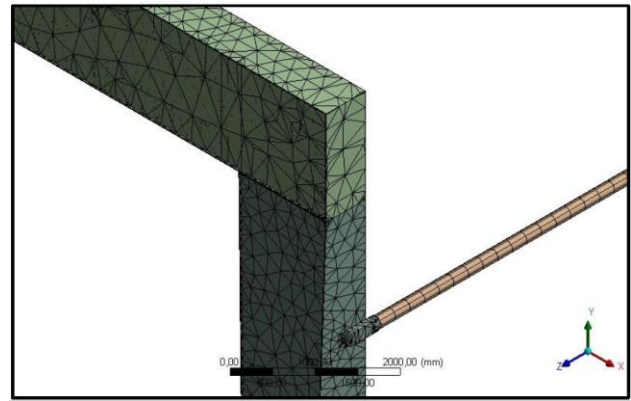


Figure 4: Mesh Generated for Analysis

BOUNDARY AND LOAD CONDITIONS

Before the simulation is run, note that the boundary conditions in the form of construction fixtures are defined. Following Figure 5a, 5b shows the load and fixtures applied on the arm in the software.

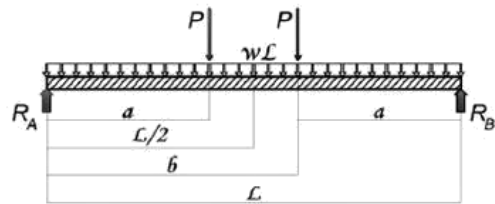


Figure 5a: Load Application Points

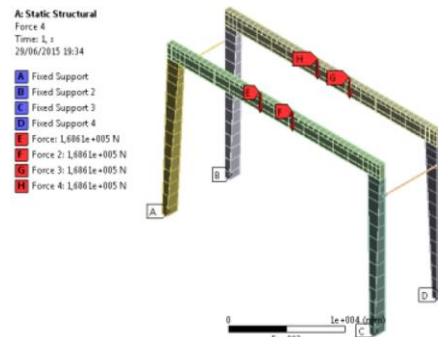


Figure 5b: Load Application Points

The amount of force that has been calculated as tabulated in Table 6

Table 6: Amount of Force for Each Case

Load case	Load (Ton)	Force (kN)
1	5	306.562
2	15	429.187
3	17.5	459.843
4	35	674.437

BENDING MOMENT

Calculation of bending moment followed by calculation of stress and deflection, with equations referring to the ANSINDS (National Design Specification) for Steel and wood Construction applied to free body diagram shown in Fig 4.8 are:

Given:

- L= 23470mm = 23.47 m
- L / 2= 11735mm = 11.74 m
- a= 9785mm = 9.79 m
- b= 13685mm = 13.69m

Where,

- L = effective length of girder (m)
- a = Distance of point of load (m)
- b = Distance of point of load (m)

SECTION WISE BENDING MOMENT CALCULATION

Bending moment is calculated at three sections shown in the following Figure 6

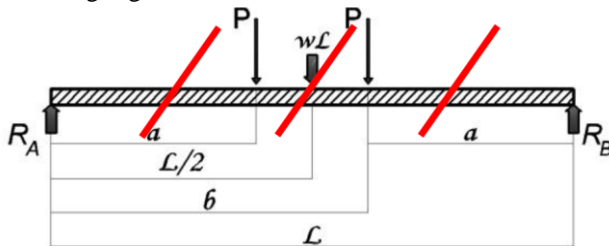


Figure 6: Bending Moment Calculation Sections

The value of calculated bending moment is given in the Table 7

Table 7: Results of Calculation of Bending Moment at Each Section

Load case	Section	Distance(x)	Bending moment (KNm)
5 ton	Section 1	0 < x ≤ 9.79	1013.200
	Section 2	0 < x ≤ 13.69	1013.324
	Section3	0 < x ≤ 23.47	0.457
15 ton	Section1	0 < x ≤ 9.79	1313.219
	Section2	0 < x ≤ 13.69	1313.295
	Section3	0 < x ≤ 23.47	0.457
17.5 ton	Section1	0 < x ≤ 9.79	1388.212
	Section2	0 < x ≤ 13.69	1388.288
	Section3	0 < x ≤ 23.47	0.457
35 ton	Section 1	0 < x ≤ 9.79	1913.162
	Section 2	0 < x ≤ 13.69	1913.238
	Section 3	0 < x ≤ 23.47	0.458

NORMAL STRESS FOR EACH LOAD CASE

1. The work load of 5 tons

As per the bending moment of the results listed in Table 4.7, then the normal stress is calculated as follows.

$$\begin{aligned} \sigma_{max} &= My/I \\ &= 1013324x (775/1650535) \\ &= 47.58 \text{ MPa} \end{aligned}$$

2. The work load of 15 tons

$$\begin{aligned} \sigma_{max} &= My/I \\ &= 1313295445 x (775/1650535) \\ &= 61.66 \text{ MPa} \end{aligned}$$

3. The work load of 17.5 tons

$$\begin{aligned} \sigma_{max} &= My/I \\ &= 1388288296 x (775/1650535) \\ &= 65.19 \text{ MPa} \end{aligned}$$

4. The work load of 35 tons

$$\begin{aligned} \sigma_{max} &= My/I \\ &= 1913238257 x (775/1650535) \\ &= 89.83 \text{ MPa} \end{aligned}$$

Where:

- σ_{max} = Maximum normal stress (MPa)
- M = bending moment (Nm)
- y = distance from neutral axis (mm)
- I = Moment of inertia xx (cm4)

Simulation results for Normal stress

The simulation results show the normal stress induced in the member. Following figures are the screenshots taken from the simulation package software after application of the point loads on the member.



Figure 7a: Normal stress results of simulation for workload of 35 tons

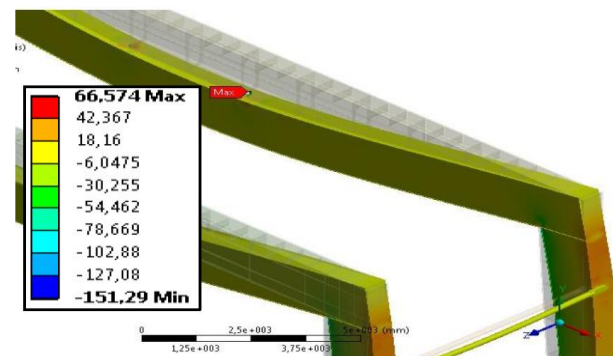


Figure 7b: Max. Normal stress results of simulation for workload of 35 tons

Table 8 Comparison of calculated & Simulated Normal Stresses

Normal Stress Y			
Load	Value From Calculation	Value From Simulation	Margin
	Maximum σ	Maximum σ	
(ton)	(MPa)	(MPa)	(%)
5	47	30	36.17
15	61	42	31.14
17.5	65	45	30.76
35	89	67	24.71

The results in Table 8 shows that there is a difference of up to 22 MPa on the comparison of calculated and simulated values of normal stresses arising in the structure of box girder.

DESIGN OPTIMIZATION ON THE BASIS OF VON MISES STRESS

The simulation results obtained in accordance with the von Mises stress at the maximum loading condition are expected to be below the maximum stress (allowable stress) as per AISC regulations under Allowable Stress Design (ASD) criteria. It is known that the tensile strength of the construction material is 250 MPa. The resultant maximum working stress limit is determined as follows

$$Ra \leq Rn / \Omega$$

$$Ra \leq 250 / 1.67$$

$$Ra \leq 150$$

Where:

- Ra = Working or Design stress Limit,
- Rn = Maximum allowable stress of the material
- Ω = safety factor

Results of von Mises stress simulation are shown in Figure 8a, 8b, 8c and 8d

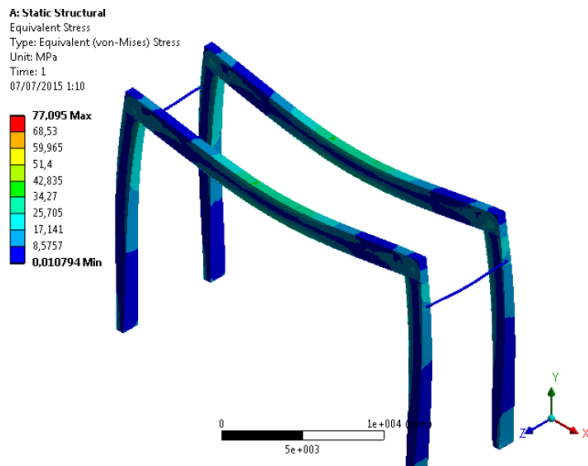


Figure 8a: Von Mises Stress Simulation Results for the Load of 5 Tons

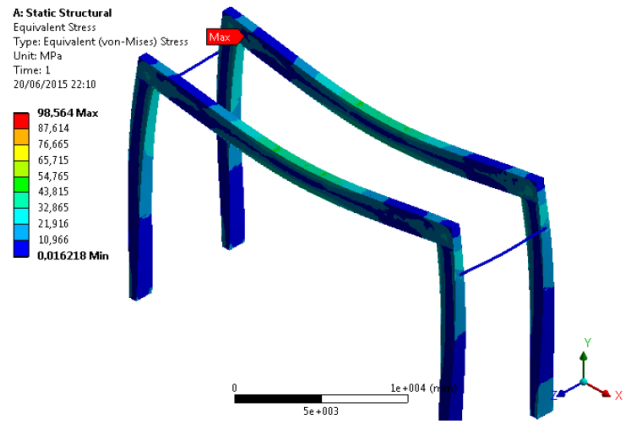


Figure 8b: Von Mises Stress Simulation Results for the Load of 15 Tons

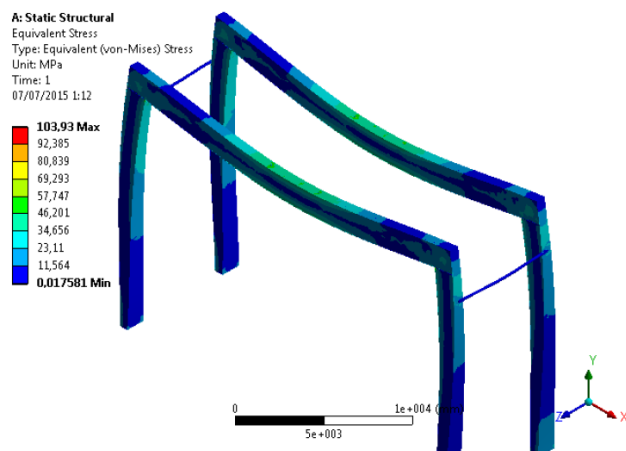


Figure 8c: Von Mises Stress Simulation Results for the Load of 17.5 Tons

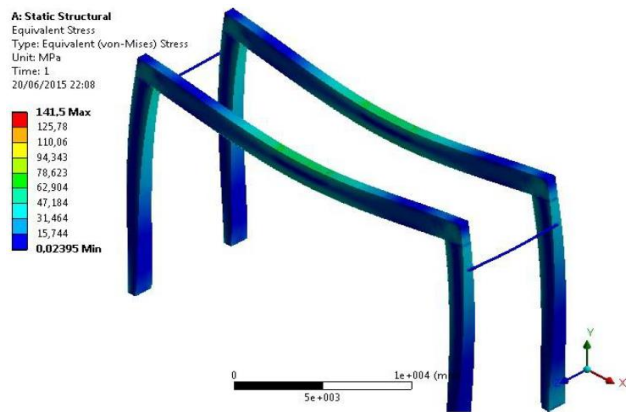


Figure 8d: Von Mises Stress Simulation Results for the Load of 35 Tons

Table 9: Comparison of the Results of Simulation with Von Mises Stress

Equivalent Stress		
Load (TW)	Maximum σ	Allowable σ
(ton)	(MPa)	
5	77.10	150
15	98.56	
17.5	103.93	
35	141.50	

SAFETY FACTOR

Beam construction safety factor based on yield strength and von Mises stress under maximum loading condition is calculated as:

$$SF = \text{yield strength/Maximum stress} = 250\text{MPa}/141.5\text{MPa}=1.8$$

The above results exceeded the safety factor of the Allowable Stress Design (ASD) that concluded construction safety.

DEFLECTION OF THE BEAM

Deflection of the girder for each load case is calculated as:

1. Work load of 5 tons

$$\text{Deflection } (\delta) = P(3L2-4a2)a/24EI = -0.01202\text{m} = 12.02 \text{ mm (-Y direction)}$$

2. Workload of 15 tons

$$\text{Deflection } (\delta) = P(3L2-4a2)a/24EI = -0.01356 \text{ m} = -13.56\text{mm} = 13.56 \text{ mm (-Y direction)}$$

3. The work load of 17.5 tons

$$\text{Deflection } (\delta) = P(3L2-4a2)a/24EI = -0.01803 \text{ m} = -18.03 \text{ mm} = 18.03 \text{ mm (-Y direction)}$$

4. Workload 35 tons

$$\text{Deflection } (\delta) = P(3L2-4a2)a/24EI = -0.02644 \text{ m} = -26.44\text{mm} = 26.44 \text{ mm (-Y direction)}$$

Where:

- δ = deflection (m)
- P = Force for respective load (KN)
- L = length of girder (m)
- a = distance of point of load (m)
- E = modulus of elasticity (Pa)
- I = Moment of inertia x-x (m4)

Simulation results of the Deflection of Beam

Referring to the BS-5950 Structural Use of steelwork in building regulations for structural steel buildings and bridge cranes the value of the maximum deflection on the girder structure is worth 1/600 of the total length of span. $\delta \text{ max} = L/600 = 23/600 = 39.1\text{mm}$

Where:

- $\delta \text{ max}$ = maximum deflection (mm)
- L = length of girder (mm)

Simulation results and the response of deflection are shown in Figure 9a and Figure 9b.

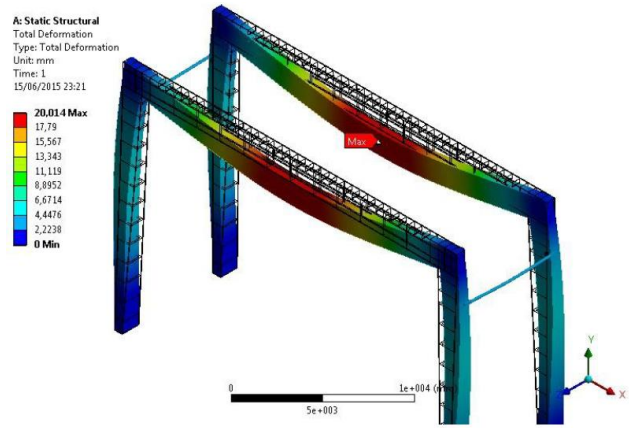


Figure 9a: Deflection Simulation Results for 35 Tons Work Load

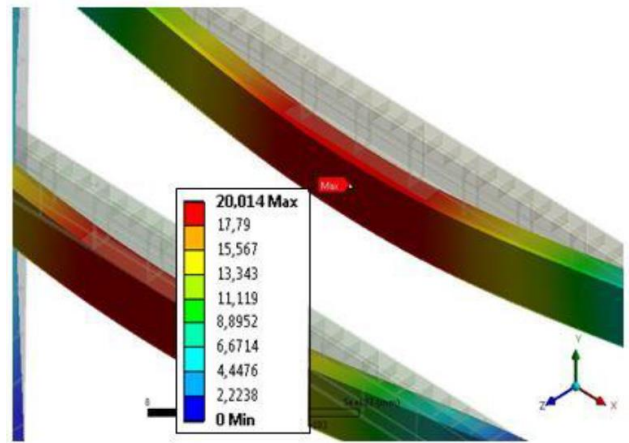


Figure 9b: The Simulation Results of Deflection for Workload of 35 Ton

Table 10: Deflection Comparison

Load	Total Beam Deflection		Maximum Allowable δ
	Value From Calculation	Value From Simulation	
(ton)	(mm)	(mm)	(mm)
5	12.02	11.70	39.11
15	16.82	13.81	
17.5	18.03	14.58	
35	26.44	19.30	

From Table 10 and Figure 9b it is evident that the construction deflection does not pass the maximum limit, thus it can be concluded still safe construction

CONCLUSIONS

The Model created in SOLIDWORKS needs only parameter, loads and spans initialization. The computational worksheet has not been considered for fatigue and weld design optimization for the arm which has been deliberately ignored to avoid complications. These cases can be incorporated into the spreadsheet in the future.

The main significance of this research result is in crane manufacturing Industry in reducing the cost of the product by 10-35% of the total cost.

The further scope of work is optimization of design by targeting the material of the arm as objective function for further cost effectiveness. Optimal designs can be standardized by Finite Element Methods and hence commercialized for cost effectiveness.

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