A COMPARATIVE STUDY OF WOODEN AND COMPOSITE CROSSARM OF AN ELECTRIC UTILITY POLE

by

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Abstract

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Currently, most of the crossarms existing in the nation are made of wood. However, wood has a set of negative features like shorter life span due to environmental cracking, damages by insects and birds. Secondly, an insufficient strength and stiffness which is compensated either by increasing the cross-section or by reducing the span between two poles. Also, it's reduced insulating property in case of high humidity of air which can be accounted to its hygroscopic behavior.

Replacement of wooden crossarms by composite crossarms is a known solution to many problems. However, the existing projects are far from optimal. Present investigation is done for both wooden and composite type of crossarms by Finite Element Analysis, taking into account the real geometry of the object.

Static analysis was done on popular species of wood such as Spotted Gum (Eucalyptus), Southern Pine (Loblolly) and Western Red Cedar. It was also carried on a hollow composite profile made of reinforced epoxy glass filled with polymeric foam for various configurations. All the simulations were made on ANSYS Workbench V17 software. Modeling of the crossarm assembly was carried on SOLIDWORKS. A set of chief characteristics like mass per unit length, relative stiffness, safety factor and cost were

compared between wooden and composite crossarms from the simulated results. Also, some parameters of the composite crossarm were varied to optimize its current design.

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Chapter 1

1.1 Introduction to crossarms

Crossarms are beams mounted on a utility pole which takes up load from the transmission wires and transfers it to the pole. Crossarms are made up of wood, reinforced concrete cement and composites. Majority of them are made of wood since it is inexpensive and easily available.

Figure 1.1.1 Crossarm

Figure 1.1.1 shows a composite crossarm carrying conductors over a certain distance.

The crossarms are of two different types. They are as shown as below

Figure 1.1.2(a)

Figure 1.1.2(b)

Figure 1.1.2(a) represents a tangent crossarm and Figure 1.1.2(b) represents a dead-end crossarm. A tangential crossarm have wires passing through the insulators on both sides, whereas in a dead-end cross arm the wires are connected to the insulators on one side only.

It should be noted that the insulators may be on top of the beam or in some cases at the bottom as well.

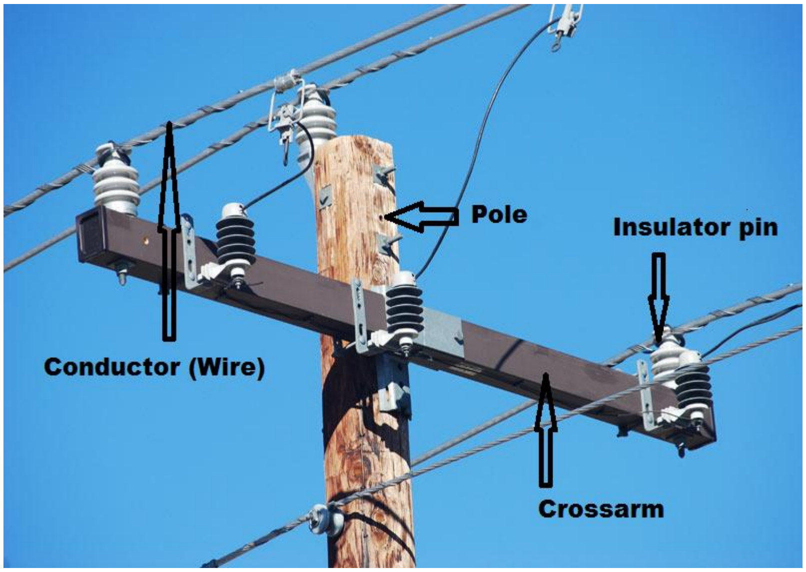


Figure 1.1.3

The different parts of the crossarm assembly which has been considered for analysis in this research are the conductor (wire), crossarm, pole and the insulator pin. This is as shown in figure 1.1.3.

1.2 Background

About 22 different varieties of woods were analyzed which were used to make crossarms around the world, out of which only 3 were used for analysis.15 different geometries,6 variations in fiber architecture, 4 variations in the thickness of the walls and numerous boundary conditions were changed during this entire research.

Over a 100 simulations were performed and the important results have been used for analysis.

1.3 Motivation

Electricity is now a basic need. Transporting them is as important as its generation. The electric poles play a vital role in the journey of electricity. It is on the crossarms that the conductors are supported. “Even the words you are reading now, has some amount of contribution from the crossarms”.

Majority of the crossarms produced today are made of wood. Composites have been used in its production too since late 1960’s. A comparative study of both in terms of deflection, costs and the safety factors can improve our understanding about the concept.

1.4 Objective

The main aim behind the following work is to compare and correlate results obtained from analysis on various types of wood, composites and existing cross-sections currently used for crossarm production.

An alternative design change has also been suggested for a composite crossarm which would not only reduce the mass, but also the cost of the crossarm.

1.5 Composites

Two or more constituents combined to form a composite material at a macroscopic level, such that they are not soluble in each other [1]. Fiber is the reinforcing constituent and matrix is the binding constituent in which the fiber is reinforced.

Each layer is called a lamina, multiple layers can be addressed as laminae and laminae stacked upon one another can be called as a laminate. Figure 1.5.1 shows a figure which would explain the concept.

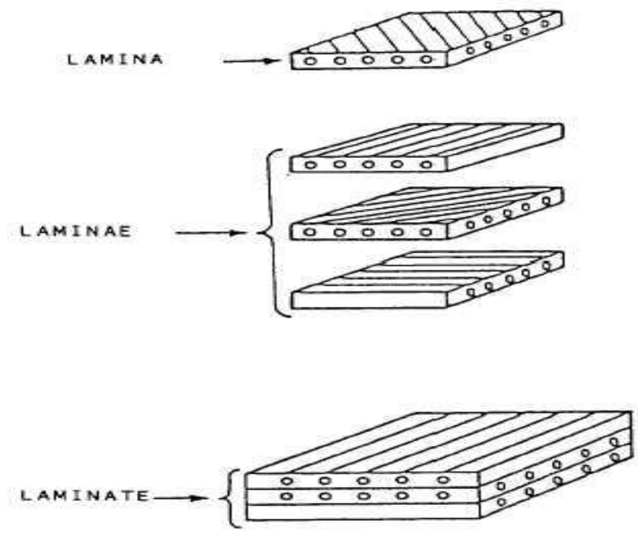


Figure 1.5.1

The laying up of lamina on top of one another is called a stack-up. The orientation of the fibres can be at different angles. It can be 0, ±30, ±45, ±60 & 90 degrees. In this research only 0 & 45 degree plies were used to increase the stiffness in the longitudinal direction and to reduce the effect of shear.

If the laminae are a mirror image about their midplane, then the laminate is termed as symmetrical. If the laminate has equal number of + and – ply angles except for 0 and 90 degrees, then the laminate can be termed as balanced [2].

Unsymmetrical laminates induce bending when axial load is applied and also induces warping when thermal load is applied. Unbalanced laminates induce twisting or shear when axial load is applied.

Advantages of Composites are [3],

- Composites are light weighted when compared to majority of metals.
- Composites have a high Strength to weight ratio.
- They have a high stiffness to weight ratio.
- Composites do not conduct electricity.

- Composites have a longer life and need little maintenance.
- Composites have a good dimensional stability as compared to wood, i.e. the latter could change depending upon hot, wet, cold or dry conditions.

However, there are some dis-advantages of composites which has to be reviewed too,

- Composites are expensive when compared to wood
- The manufacturing processes of composites are complex and hence labour intensive, which in turn would make it more expensive.
- Since the composites consist of multiple laminae, delamination might occur at weaker layers.
- Cracks and damages occur internally and hence its inspection techniques will be expensive.

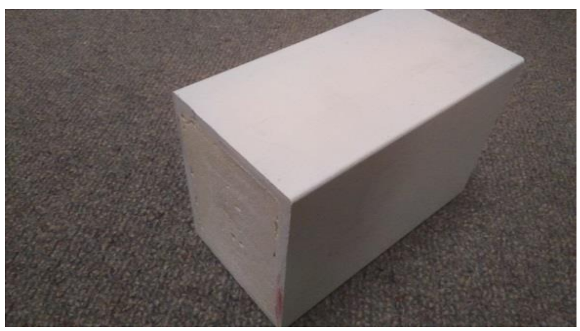


Figure 1.5.2

Figure 1.5.2 shows the existing model that has been analysed in the current research.

COMPANY NAME – PUPI [12]

MODEL – SERIES 2000 (3-5/8 X 4-5/8)"

WEIGHT – 19.95 KG

WALL THICKNESS – 8 mm

.

Chapter 2

Methodology

The existing geometries in the market were adopted for modeling the assembly on SOLIDWORKS. Assumptions were made about the porcelain pin and washer's dimensions. The material properties of the different types of wood were found from an existing data.

The geometry and material properties were entered to ANSYS V17 Workbench. Eccentric loading was applied on top of the pin and the central face of the beam was constrained in such a way that the six degrees of freedom were fixed.

Post-processing was then carried out. Normal stresses, Shear Stresses and deflections were analyzed at various regions of the model. The maximum stresses and the total deformations were taken into account. A simple box type failure criterion was used to analyze the failure.

The methodology followed for both the wood and the composites were the same. A comparison was made between the various outputs which were extracted.

Chapter 3

3.1 Geometry used for wood

This chapter gives the details of the geometries that have been used for research. The geometries of the beams are as per the dimensions used in the industries today.

Figure 3.1.1

Figure 3.1.1 shows a rendered image of the crossarm, modeled on SOLIDWORKS software. The assembly consists of a wooden beam, two porcelain pins and two washers. The various dimensions have been shown in the following images using the drafting tool on SOLIDWORKS.

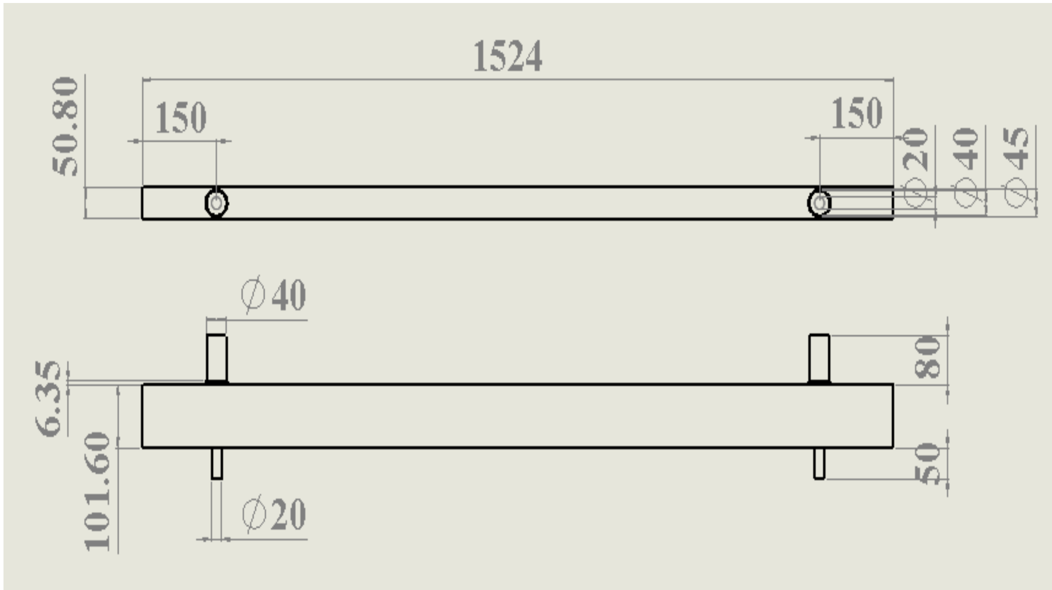


Figure 3.1.2

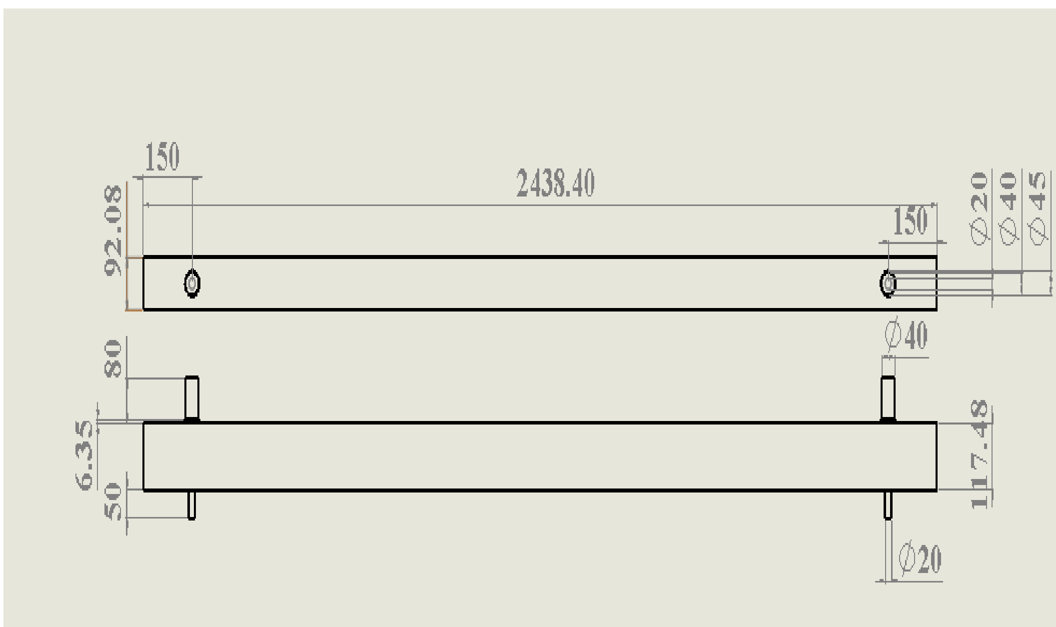


Figure 3.1.3

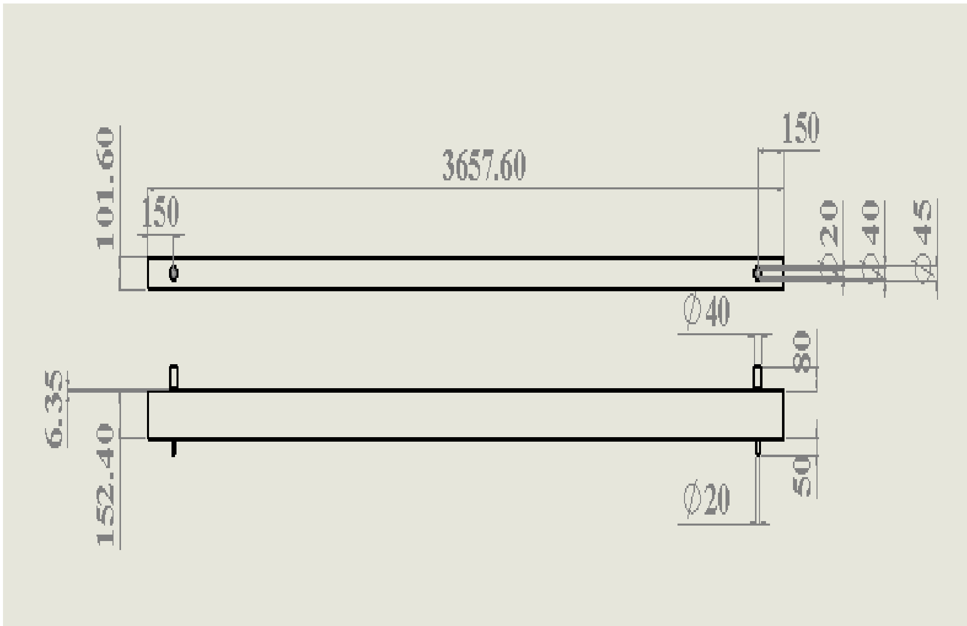


Figure 3.1.4

Note: - All the dimensions mentioned in the figures are in mm.

Figure	Dimension in inches	Dimension in mm
2-2	2 X 4 X 60	50.8 X 101.6 X 1524
2-3	3.625 X 4.625 X96	92.075 X 117.475 X 2438.4
2-4	4 X 6 X 144	101.6 X 152.4 X 3657.8

Table 3.1.1

The above table shows the dimensions of the main part, i.e. the wooden beam in inches and in millimeters. Dimensions of porcelain insulator pin & the washer are kept fixed except for the length of the pin which changes with the thickness of the beam. This can be made inferred from the figure wherein the length of the pin below the beam remains constant.

3.2 Geometry used for Composites Crossarms

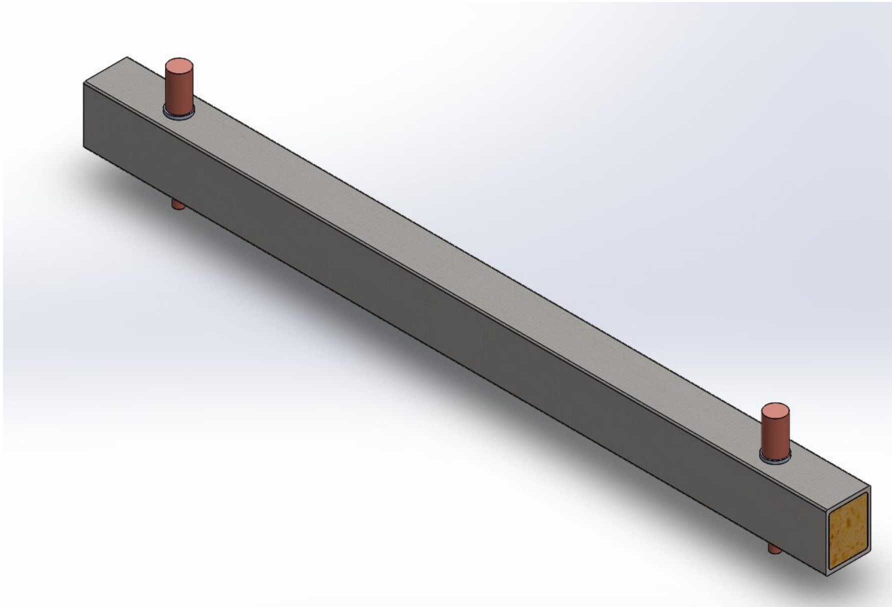


Figure 3.2.1

Figure 3.2.1 shows a rendered image of the crossarm, modeled on SOLIDWORKS software. Incase of composite crossarms, stacking procedure was carried on ansys for the hollow thin section. Hence, only a surface model of the part was created on ANSYS. The foam, washer and the porcelain pin's were modeled separately.

There were two types of stacking carried on this research. One was termed as "square stack", wherein all the surfaces of the hollow part was considered as one part and stacked . The other type was termed as "Plate stack", wherein the four sides of the hollow part were considered as four different surfaces and stacking was done separately for each side. The edges were modeled separately and assembled on ANSYS.There were small differences in results of calculations based on two different stacking procedures. However, these differences can be considered as negligable taking into account precision of our knowledge

The main reason for the change in geometry was that, it helped to analyze each ply separately on each face.

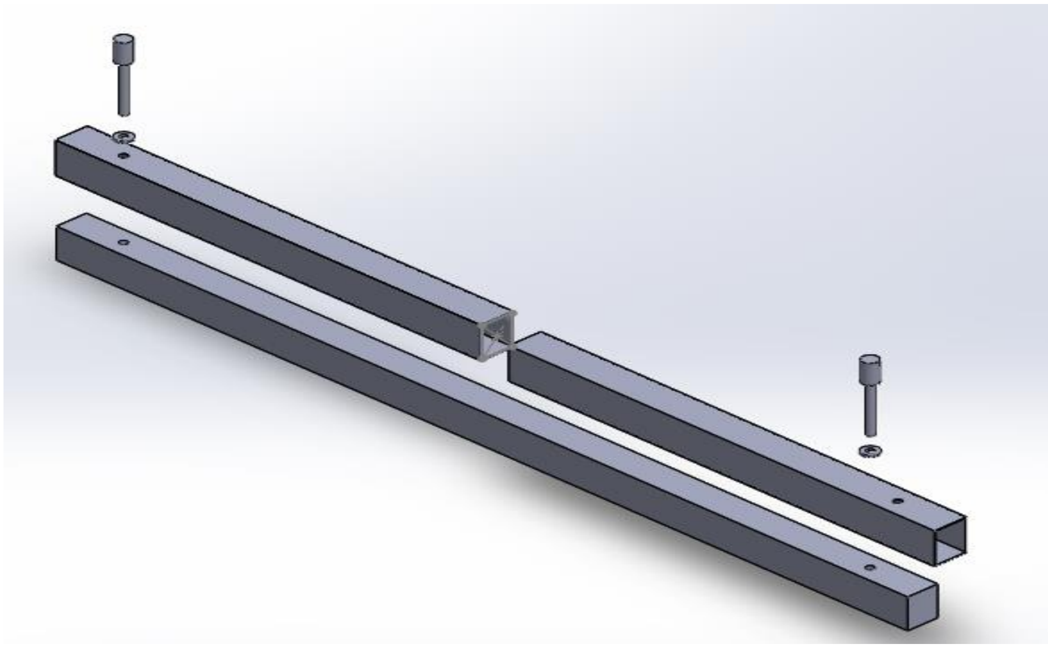


Figure 3.2.2

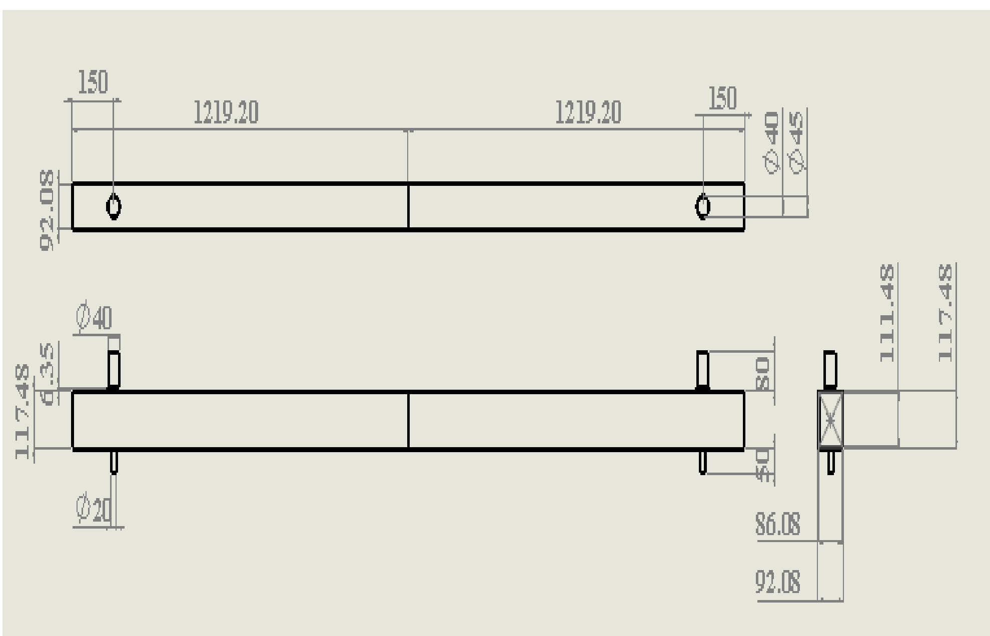


Figure 3.2.3

Figure 3.2.2 and 3.2.3 represents a square stacked geometry of (3.625 X 4.625 X 96) inches, with a 3 mm thick wall of the hollow part.

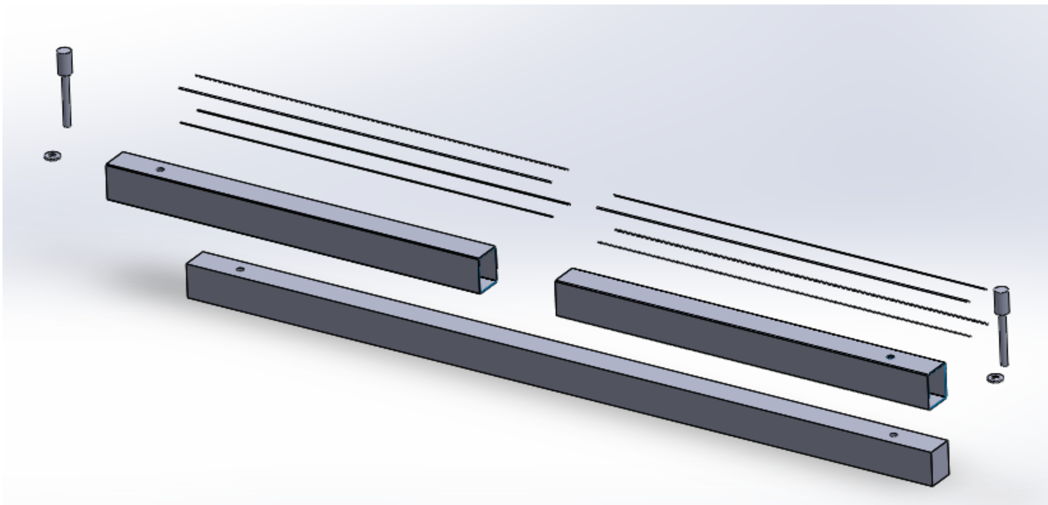


Figure 3.2.4

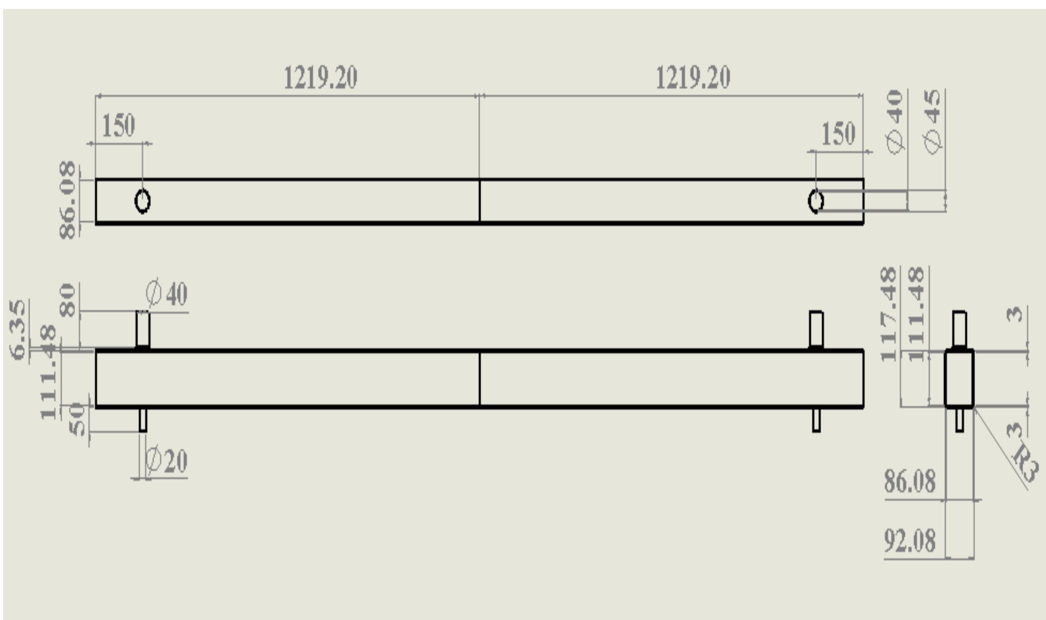


Figure 3.2.5

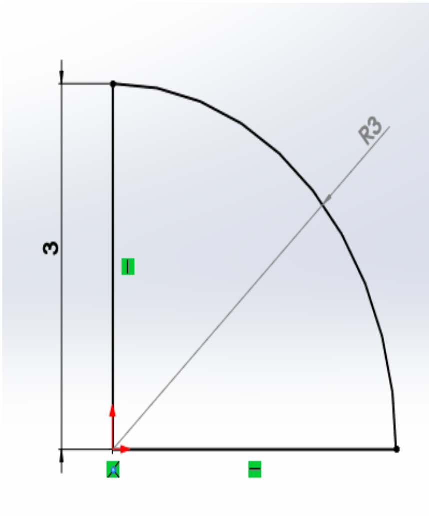


Figure 3.2.6

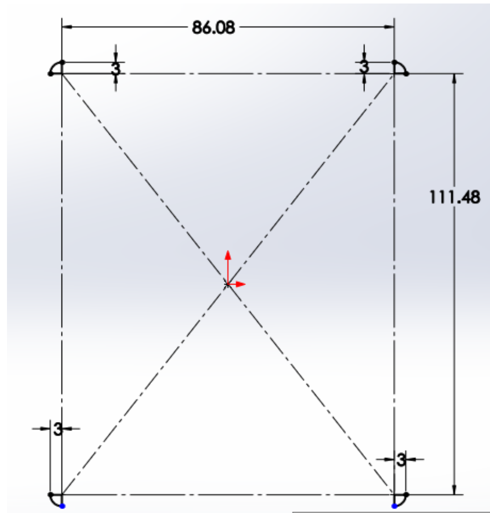


Figure 3.2.7

Figure 3.2.4 shows the various components of the plate stack model with a 3mm thick wall and dimension of (3.625 X 4.625 X 96) inches. It shows the extra outer edges present. Figure3.2.5 shows the dimensions of the various parts of the assembly. Figure 3.2.6 shows the dimensions of the edges and figure 3.2.7) shows the distance coordinates at which the edges are modelled.

(92.08 x 117.48 x 2438.4)mm						
Thickness of the Outer wall (mm)	Dimensions of the foam(mm)			EDGES both left and right(mm)		
	Length	height	Width	Length	Sides in contact with the walls	Radius on the third side
3	2438.4	111.48	86.08	1219.2	3	3
4	2438.4	109.48	84.08	1219.2	4	4
6	2438.4	105.48	80.08	1219.2	6	6
8	2438.4	101.48	76.08	1219.2	8	8

Table 3.2.1

(101.6 x 152.4 x 2438.4)mm						
Thickness of the Outer wall (mm)	Dimensions of the foam(mm)			EDGES both left and right(mm)		
	Length	height	Width	Length	Sides in contact with the walls	Radius on the third side
3	2438.4	146.4	95.6	1219.2	3	3
4	2438.4	144.4	93.6	1219.2	4	4

Table 3.2.2

Table 3.2.1 and Table 3.2.2 represent various dimensions of the foam and edges for different dimensions mentioned in the table.

It should be noted that the outer dimension of the foam is equal to the inner dimension of the hollow part. Hence, a surface model of the hollow part is created with the same dimensions as that of the foam.

Chapter 4

4.1 Mechanical properties of wood

Sl No.	Wood Type	Type of wood Hardwood/Softwood	Young's Modulus in the longitudinal direction (Gpa)	Density (Kg/m^3)	Modulus of Rupture(Mpa)
1	Southern Pine - loblolly	Softwood	12.3	570	88.3
2	Southern Pine - Shortleaf	Softwood	12.1	570	90.3
3	Southern Pine - Longleaf	Softwood	13.7	650	100
4	Southern Pine -Slash	Softwood	13.7	655	112.4
5	Douglas -FIR - Coast	Softwood	13.4	510	86.2
6	Douglas -FIR - Interior west	Softwood	12.6	-	85
7	Douglas -FIR - Interior north	Softwood	12.3	-	87
8	Douglas -FIR - Interior South	Softwood	10.3	-	90
9	Western red cedar	Softwood	7.66	370	51.7
10	Lodgepolepine	Softwood	9.24	465	64.8
11	Jackpine	Softwood	9.31	500	68.3
12	Scots Pine	Softwood	10.08	550	83.3
13	Radiata Pine	Softwood	10.06	515	79.2
14	Birch, yellow	Hardwood	13.9	690	114.5
15	Norway Spruce	Softwood	9.7	405	63
16	Eucalyptus - Spotted Gum	Hardwood	26.14	1060	141.8
17	Eucalyptus - Tallwood	Hardwood	21.08	1090	121.8
18	Oak - Red	Hardwood	12.14	700	99.2
19	Oak -White	Hardwood	12.15	755	102.3
20	Yellow poplar	Hardwood	10.9	455	69.7
21	Maple	Hardwood	12.62	705	109
22	Bamboo	Grass	18	850	168.6

Table 4.1.1 [4][5][6]

Mechanical Properties	Southern Pine - Longleaf	Western red cedar	Eucalyptus - Spotted Gum
Density (kg/m^3)	650	370	1060
Modulus of rupture(Mpa)	100	51.7	141.8
EL(Gpa)	13.7	7.66	26.15
ET(Gpa)	0.75	0.42	1.50
ER(Gpa)	1.40	0.62	2.41
GLR(Gpa)	0.97	0.67	1.74
GLT(Gpa)	0.82	0.66	1.53
GRT(Gpa)	0.16	0.04	0.84
Vrt	0.38	0.48	0.66
Vlt	0.37	0.30	0.55
Vlr	0.33	0.38	0.49

Table 4.1.2[5][6]

Table 4.1.1 shows the different types of wood that has been used around the world and its important mechanical properties. Table 4.1.2 shows the properties of wood that has been used for analysis in this research. The moisture content for all the three types of wood was considered to be 12%. Eucalyptus (Spotted Gum) has the highest mechanical properties in terms of Young's modulus, density and modulus of rupture while Western red cedar has the least. This was the reason for their selection. Southern pine wood was chosen as their properties lies in between Eucalyptus and Western red cedar and also, most of the crossarms present currently are made from it.

4.2 Properties of Composites

Mechanical Properties	Epoxy-E Glass UD (Hollow Part)	Epoxy - S Glass (Hollow Part)	Epoxy-E glass (Washer)
Density(Kg/m^3)	2000	2000	1900
Ex(Gpa)	45	50	25
Ey(Gpa)	10	8	25
Ez(Gpa)	10	8	9
Vxy	0.3	0.3	0.3
Vyz	0.4	0.4	0.2
Vxz	0.3	0.3	0.2
Gxy(Gpa)	5	5	9.6
Gyz(Gpa)	3.85	3.85	3
Gxz(Gpa)	5	5	3

Table 4.2.1[7][8]

Table 4.2.1 shows the properties of the different composites that have been used for this research. About 61% fibre concentration per volume was assumed for the composites constituting the hollow part.

4.3 Properties of porcelain and foam

It should be noted that the porcelain is of a higher grade whose applications could be found in electrical appliances. A high density closed cell rigid polyurethane foam is used which not only provides strength for the hollow surface but also does not let the water to seep in and affect the hollow surface.

Properties	Rigid Polyurethane foam	Insulating Porcelain
Density(kg/m^3)	77	2400
Young's Modulus(Mpa)	26.2	110000
Poisson's Ratio	0.37	0.17

Table 4.3.1 [9] [10] [11]

Table 4.3.1 shows the properties of the foam and porcelain used in this research

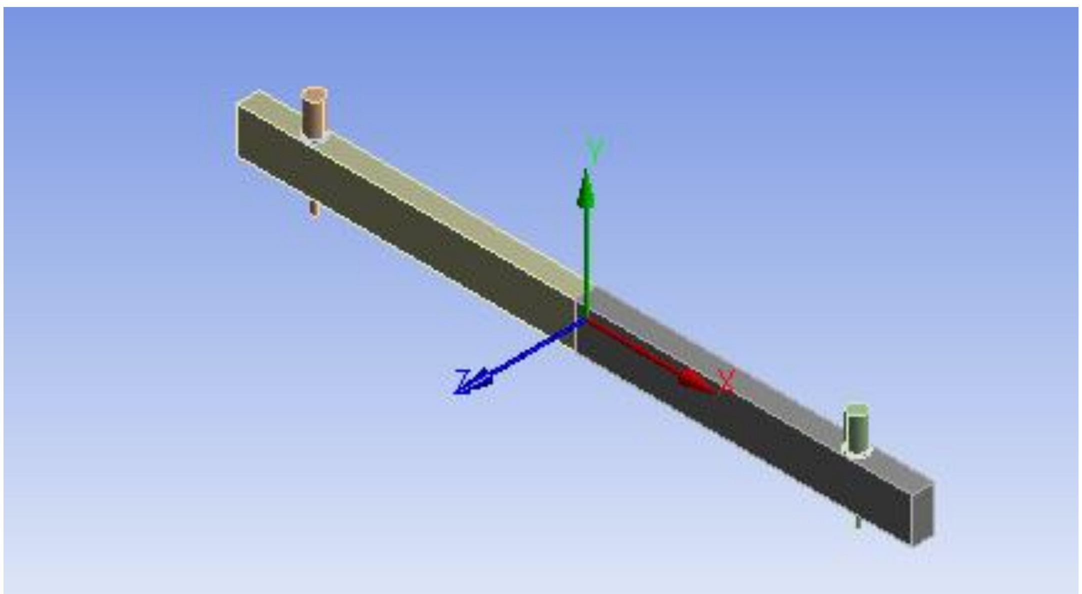


Figure 4.3.1

Figure 4.3.1 shows the co-ordinate axis considered for the material properties. Variation in this would result in the change of the results. X acts in the longitudinal direction, Y acts in the radial direction and Z acts in the transverse direction.

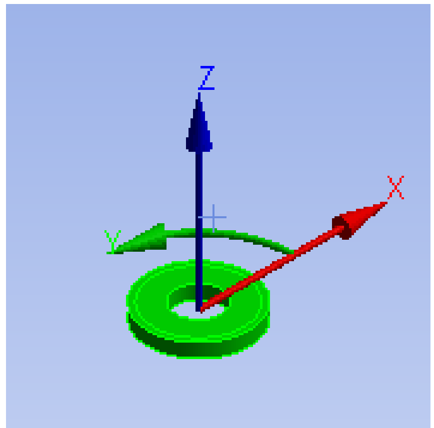


Figure 4.3.2

A cylindrical co-ordinate system was used for the washer. The properties were assigned based on the co-ordinate system.

Chapter-5

Boundary Conditions

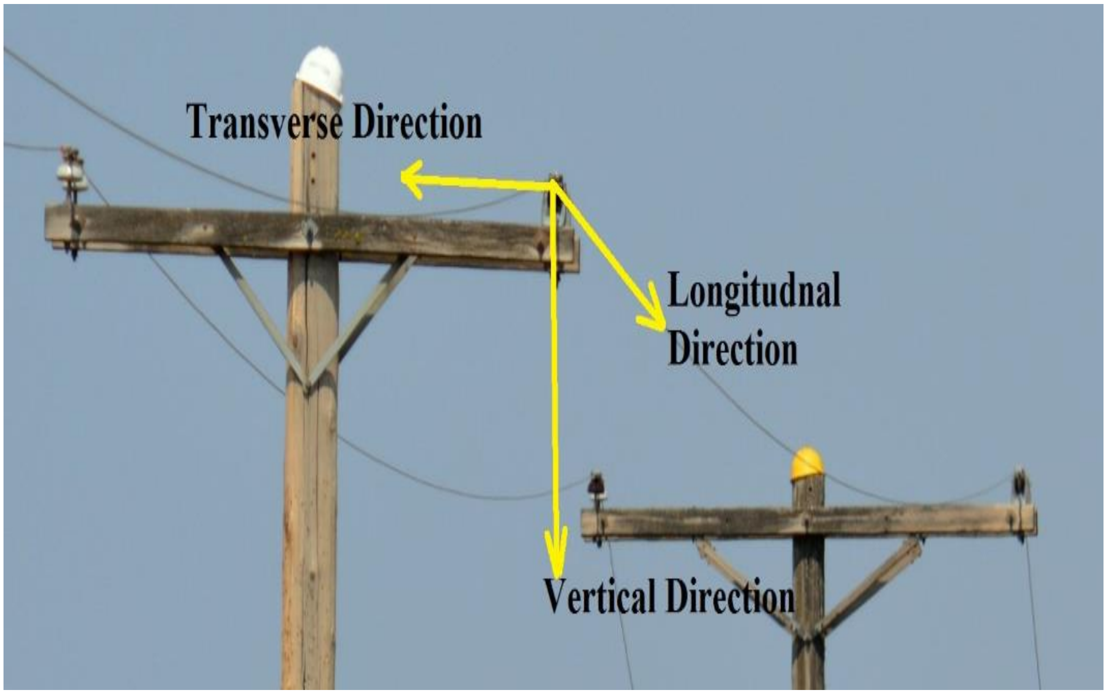


Figure 5.1.1

There were three different types of loads that were assumed during this research. The longitudinal, transverse and vertical loads. The longitudinal loads were due to the wires running along both sides of the insulator. This load can be neglected since equal loads on opposite sides cancel each other. However, 50 N is considered due to any torque difference. The vertical loads were defined by the NESC standards [13] for 1-inch ice around 7/16 high strength steel with a span of 120 feet between two poles. The transverse load was taken about 1500N.

Calculations:

Vertical load of cable with 1inch ice on it = 2.18 lbf/ft

Load for 120 feet = 120*2.18 = 261.6lbf = 1164 N

For design purpose we take it as 1200 N

From the manual [13] transversal load is 1.25 times the vertical load.

Thus, transversal load = 1500N.

Final Loads	
Type of loads	Loads acting (N)
Longitudinal	1500
Vertical	1200
Transverse	±50

Table 5.1.1

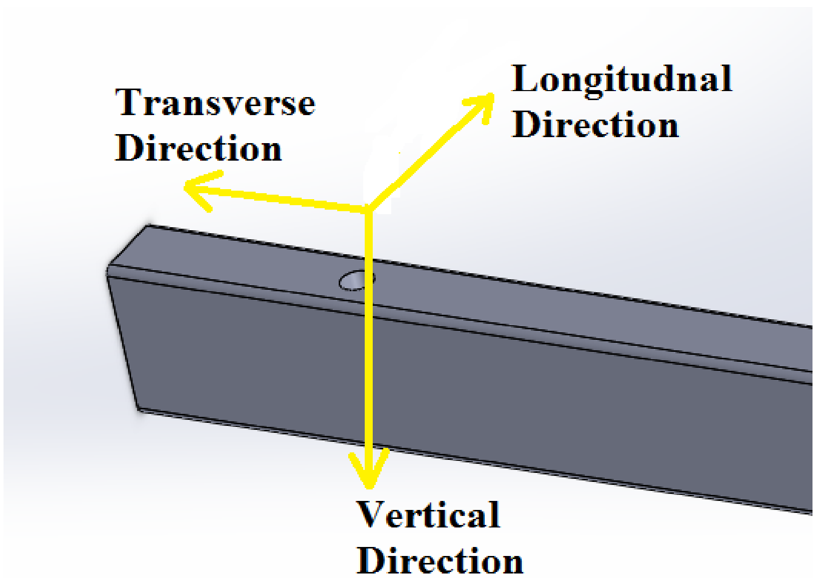


Figure 5.1.2

Figure 5.1.2 shows the three different directions in which the eccentric loading is applied in all the three different directions for a beam.

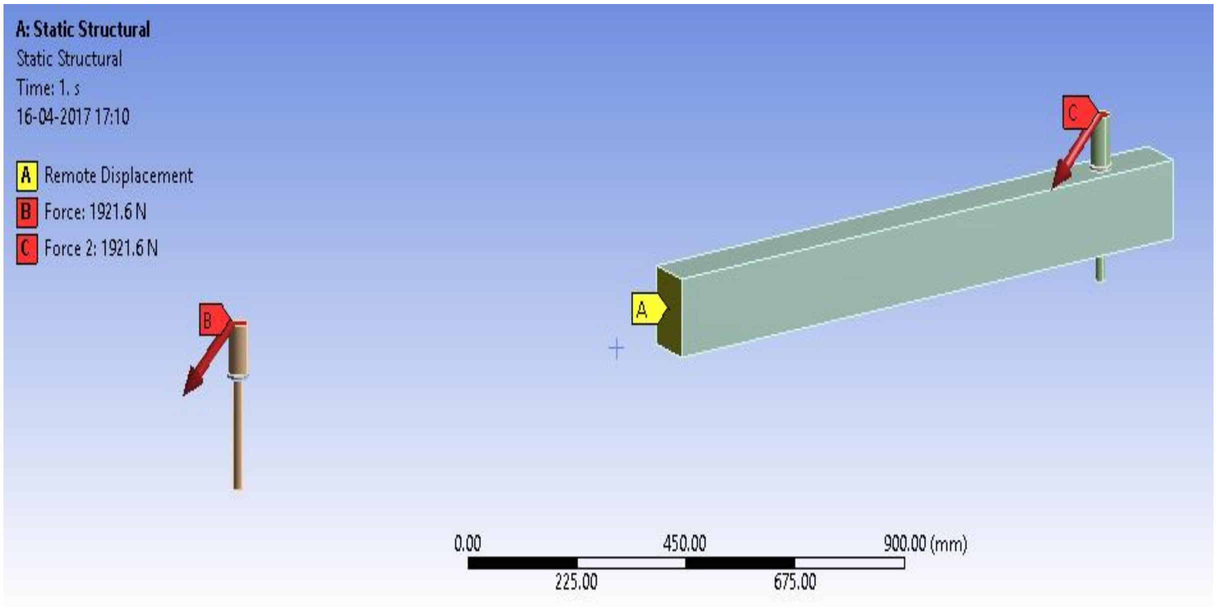


Figure 5.1.3

Figure 5.1.3 shows the loading on top of the pin. The resultant force due to all the three forces is 1921.6 Newton. Remote displacement is applied to the central face of the beam fixing all the six degrees of freedom to 0. This does not let the beam to shift from its original position.

The boundary conditions remain almost same for both composite and wooden surface except in case of composites the remote displacement is given to each face of the central face separately in order to increase the accuracy of results on ANSYS.

Chapter-6

Meshing

For wood,

Mesh element size was taken as 5mm for configurations of (2 X 4 X 60) inches & (3.625 X 4.625 X 96) inches. 7 mm element size was taken for (4 X 6 X 144) inches. A Body method of Hex dominant meshing with fine smoothing, fast transition and fine span angle was used**.**

Dimension (mm)	Number of nodes	Number of elements
50.8 x 101.6 x 1524	316495	75039
92.075 x 117.475 x2438.4	956876	227558
101.6 x 152.4 x 3657.6	774074	181314

Table 6.1.1

Table 6.1.1shows the number of nodes and elements generated due to meshing.

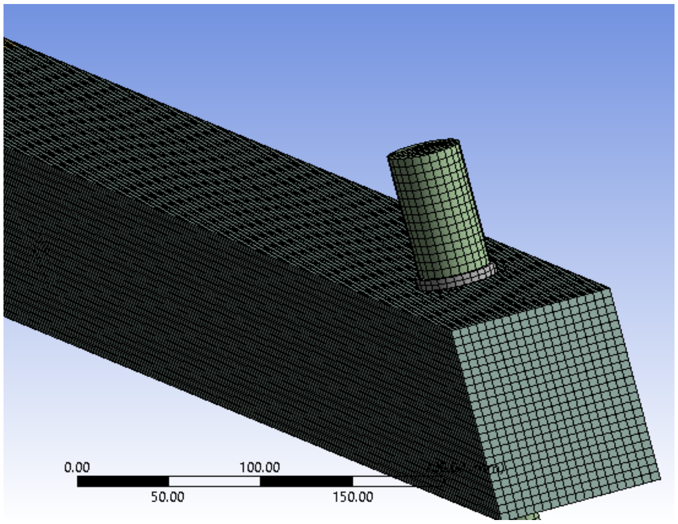


Figure 6.1.1

For Composites,

Mesh element size of 8mm was used for all configurations. Fine smoothing, fast transition and fine span angle was used**.**

Dimension of the entire beam(mm)	Thickness of the wall(mm)	Number of elements	Number of Nodes
92.075 x 117.475 x 2438.4	3	315871	602617
	4	308667	583711
	6	291993	545706
	8	282126	527868
101.6x 152.4 x 2438.4	3	390515	748791
	4	384878	736216

Table 6.1.2

Table 6.1.2 shows the number of elements and nodes generated due to meshing.

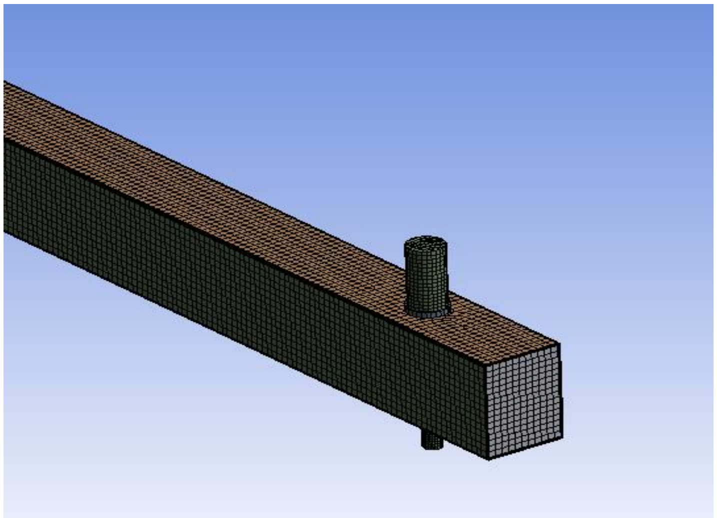


Figure 6.1.2

Chapter 7

Simulation

In this chapter the various simulations carried out in this research have been explained. Simulations were carried on ANSYS Workbench 17. Static structural, ACP – Pre and ACP –post were the modules which were used on ANSYS.

Simulation on ANSYS involves three main steps,

1. Pre-processing – In this stage all the inputs such as the geometry, material properties, boundary conditions and meshing were defined. Any minute error in the inputs given will alter the results obtained. Care was taken and inputs were checked repeatedly to avoid any errors.
2. Solution – In this stage the problem is solved.
3. Post-processing – In this stage the results obtained can be viewed. In this research normal stress, total deformation, and shear stress has been extracted.

Various cases were simulated for wood and composites separately.

For wood,

Case 1 – In this case (3.625 X 4.625 X 96) inch configuration made of Eucalyptus wood has been analysed with the boundary condition that has been mentioned in chapter – 5. The stresses and deformation acting at all the faces and for the entire body has been analysed in this case.

Case 2 – In this case Southern Pine – Longleaf wood with three different types of configurations i.e., (2 X 4 X 60), (3.625 X 4.625 X 96) and (4 X 6 X 144) inch beams with the same loads and boundary conditions have been analysed.

Case 3 – In this case Eucalyptus – Spotted gum, Southern pine –Longleaf and Western Red Cedar wood s have been analysed for (3.625 X 4.625 X 96) inch configuration.

For composites,

Case 1 – Different types of stacking processes have been analysed.

Case 2 – In this case the thickness of the hollow structure of the beam has been altered along with the percentage of 45 degree plies in the stack-up.

Case 3 – In this case S-2 glass has been analysed and compared with E-glass for 25 % of 45-degree ply angle.

Case 4 – In this case (4 X 6 X 96) inch composite beam has been analysed and compared with (3.625 X 4.625 X 96) inch composite beam for 25 % of 45-degree ply

Case 5 – In this case a dead end assembly crossarm has been analysed and compared with tangential crossarm for 25 % of 45-degree ply.

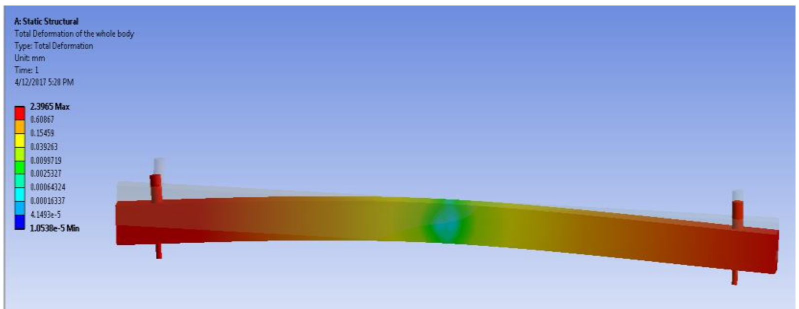
Chapter – 8

Results

8.1 Results for wood

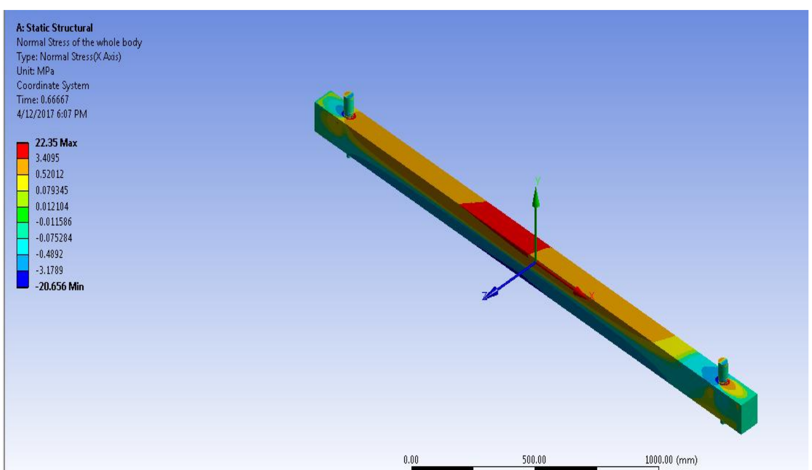
8.1.1 Case 1 – Whole Assembly of (3.625 X 4.625 X 96) inch Eucalyptus wood

In this case the whole assembly and each face of the beam were analysed. The results were as following,



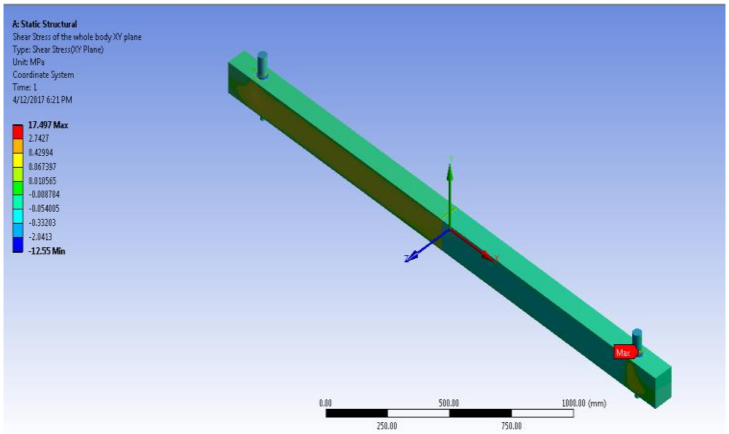
Maximum Deflection (mm)	2.40

Figure 8.1.1.1



Maximum Normal Stress(Mpa)	22.35

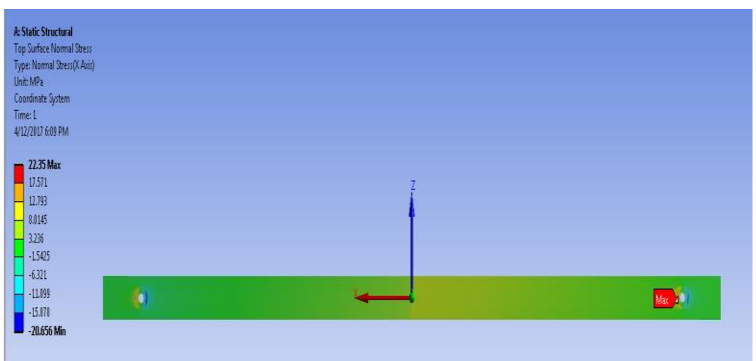
Figure 8.1.1.2



Maximum Shear Stress (Mpa)	17.50

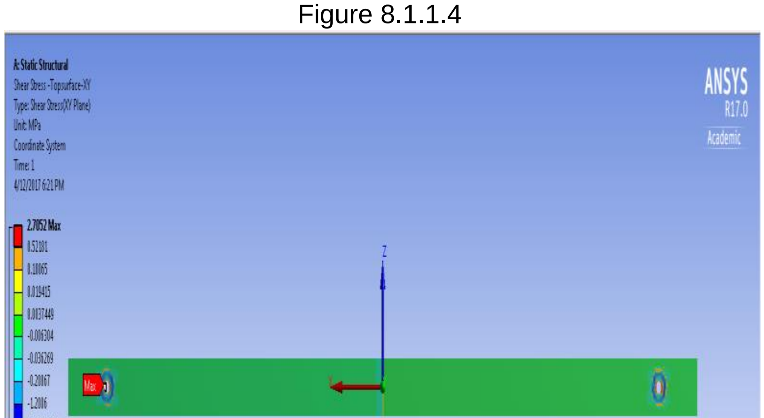
Figure 8.1.1.3

Top Surface -



Maximum Normal Stress(Mpa)	22.35

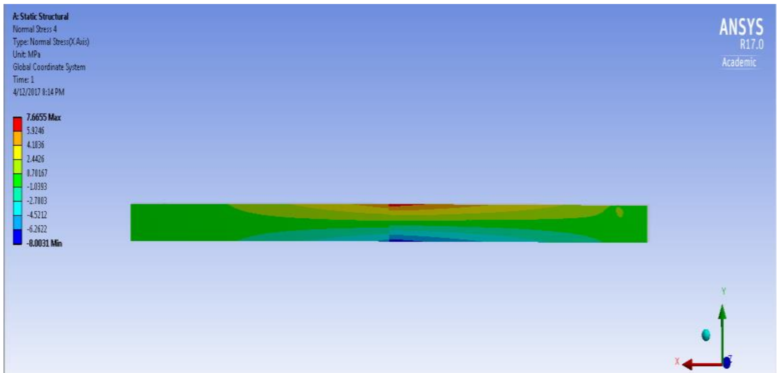
Figure 8.1.1.4



Maximum Shear Stress (Mpa)	2.71

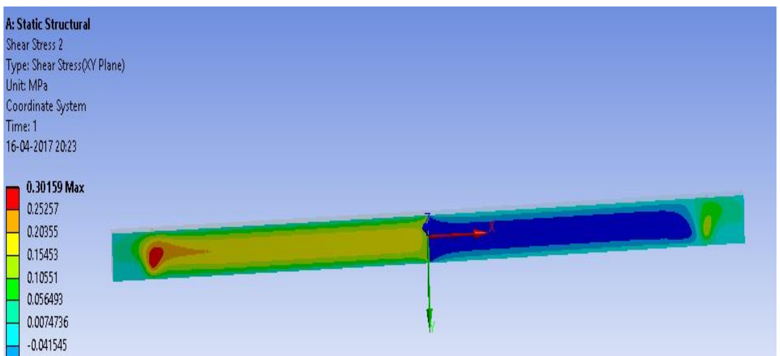
Figure 8.1.1.5

Back Surface –



Maximum Normal Stress (Mpa)	7.67

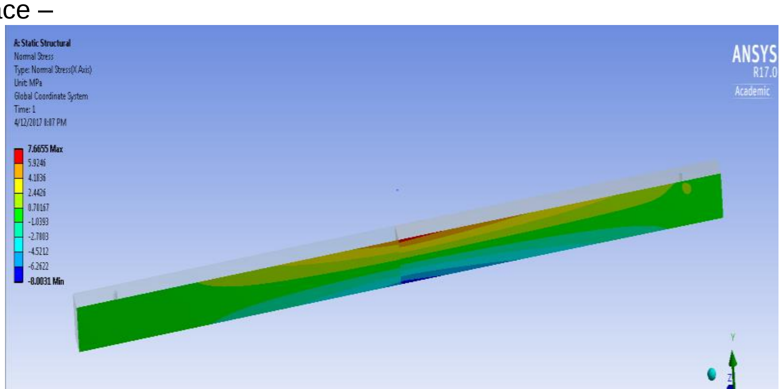
Figure 8.1.1.6



Maximum Shear Stress (Mpa)	0.30

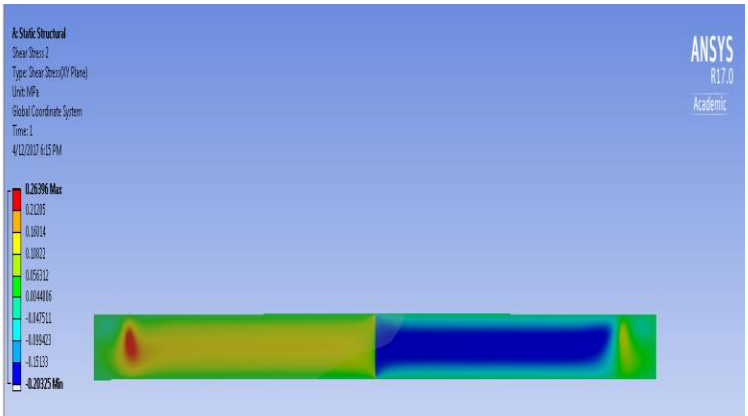
Figure 8.1.1.7

Front Surface –



Maximum Normal Stress (Mpa)	7.67

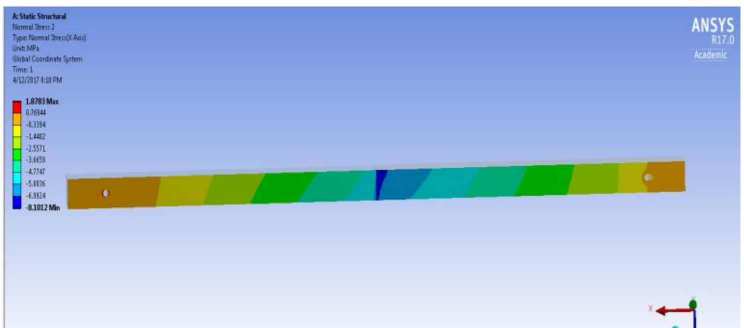
Figure 8.1.1.8



Maximum Shear Stress (Mpa)	0.26

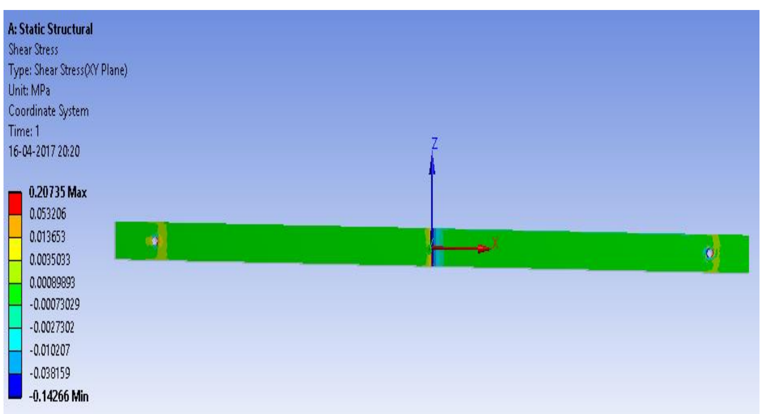
Figure 8.1.1.9

Bottom Surface –



Maximum Normal Stress (Mpa)	1.88

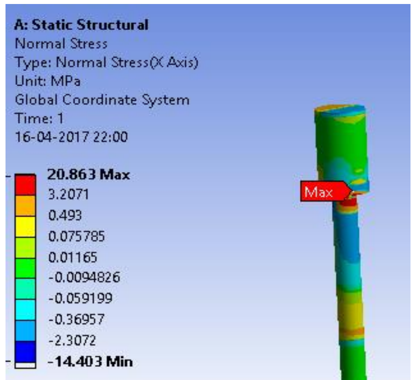
Figure 8.1.1.10



Maximum Shear Stress (Mpa)	0.21

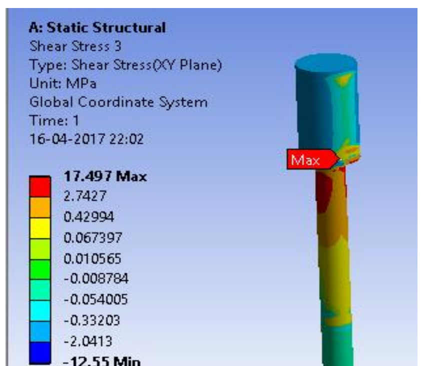
Figure 8.1.1.11

Porcelain pin -



Maximum Normal Stress (Mpa)	20.86

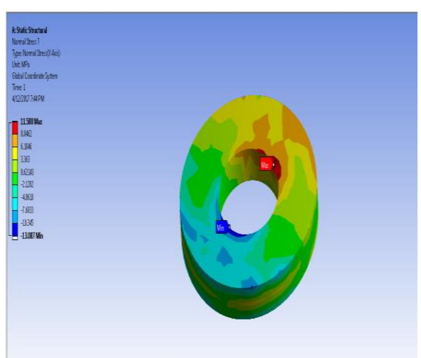
Figure 8.1.1.12



Maximum Shear Stress (Mpa)	17.50

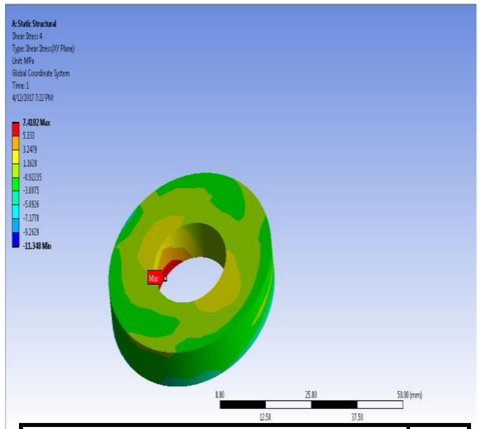
Figure 8.1.1.13

Washer



Maximum Normal Stress (Mpa)	11.59

Figure 8.1.1.14



Maximum Shear Stress (Mpa)	7.42

Figure 8.1.1.15

It can be observed that the stress concentration is maximum at the centre and the hole in case of the beam

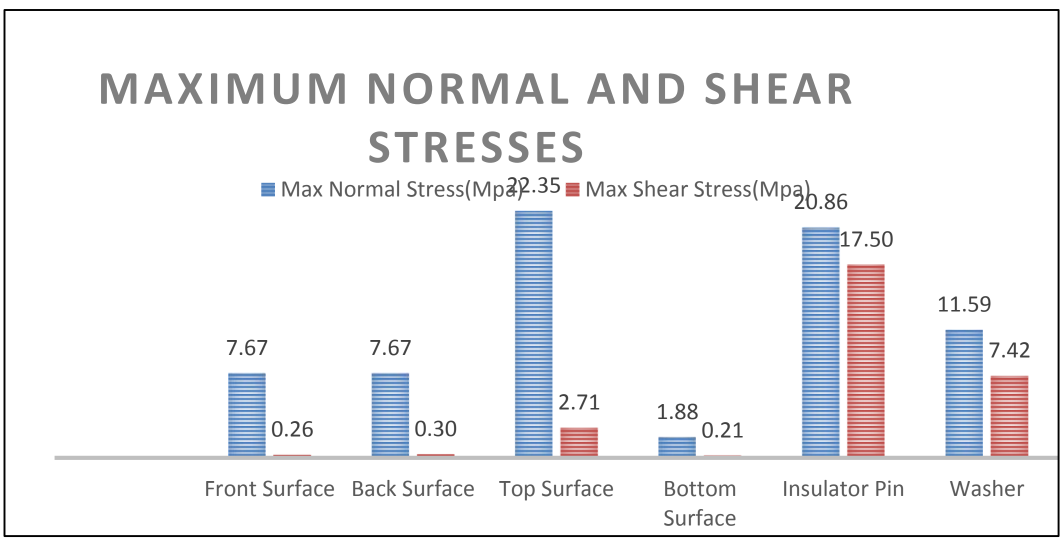


Figure 8.1.1.16

Maximum Normal and Shear stresses occur at the top surface for the specified load and boundary conditions. Hence, only the top surface stresses are analysed in the further conditions.

8.1.2 Case 2 – With Southern Pine-Longleaf as wood and different Configurations

Configuration (inches)	Max Shear Stress (Mpa)	Max Deflection(mm)	Max Normal Stress (Mpa)
2 X 4 X 60	2.98	3.22	23.34
3.625 X 4.625 X 96	2.72	4.56	18.99
4 X 6 X 144	3.27	6.40	23.43

Table 8.1.2.1

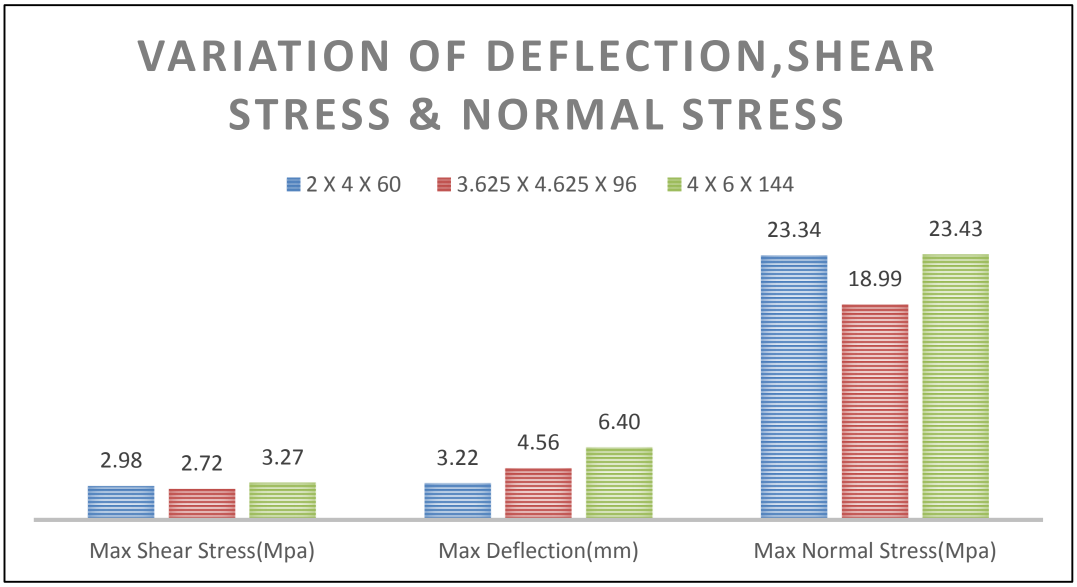


Figure 8.1.2.1

It can be observed from the graphical representation that deflection is maximum for (4 X 6 X 144) inch configuration and least for (2 X 4 x 60) inch configuration, while that of (3.625 x 4.625 X 96) inch lies in between the two.

Maximum Normal and Shear stresses are maximum at (4 X 6 X 144) inch configuration, least at (3.625 X 4.625 X 96) inch configuration and that of (2 X 4 X 60) lies in between the two.

8.1.3 Case 3- With (3.625 X 4.625 X 96) inch configuration for different types of wood

Type of Wood	Max Shear Stress(Mpa)	Max Deflection(mm)	Max Normal Stress(Mpa)
Eucalyptus	2.71	2.40	22.35
Southern Pine	2.72	4.56	18.99
Western Red Cedar	2.69	8.09	19.33

Table 8.1.3.1

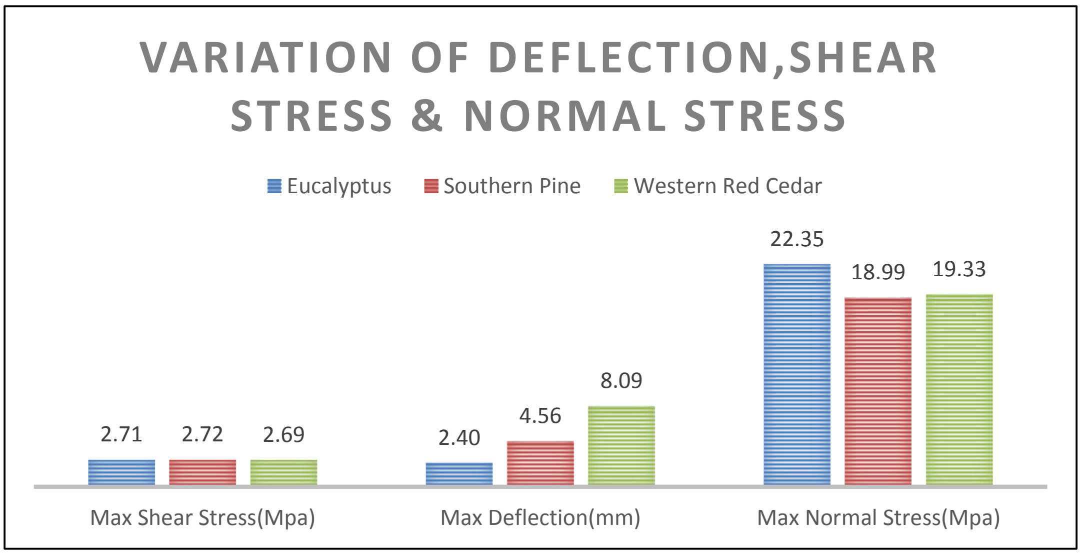


Figure 8.1.3.1

It can be observed from the graphical representation that the deflection is maximum for Western red cedar, least for Eucalyptus and that of Southern Pine wood lies in between the two.

Shear stress is maximum for Southern Pine wood, least for Eucalyptus and that of Western red cedar lies in between the two.

Normal stress is maximum for Eucalyptus, least for Southern pine wood and that of Western red cedar lies in between the two.

8.1.4 Analytical calculations for wood:

The following steps were followed, [13]

1. The moment of Inertia was found using I = $\frac{bd^3}{12}$

2. Considering half of the crossarm, deflection of the cantilever beam was calculated at the pin using the equation, W = $\frac{P_w L^3}{3EI} + \frac{ML^2}{2EI}$

3. The total deflection at the beam end was then calculated W_e= W + L * $\frac{P_w L+M}{EI}$ * x in mm

4. Finally stress is calculated by using $\sigma = \frac{M}{I} * y_{max} + \frac{P_{wind}}{A}$ in Pascal.

5. Comparisons are made with ANSYS results. Normal stresses were considered away from the stress concentration regions and the central line and the edge as shown in the figure.

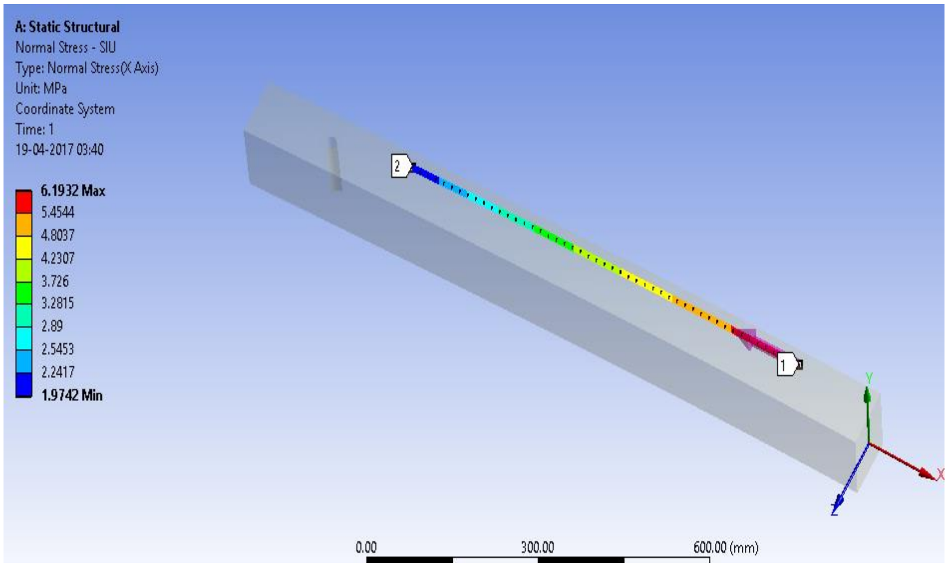


Figure 8.1.4.1

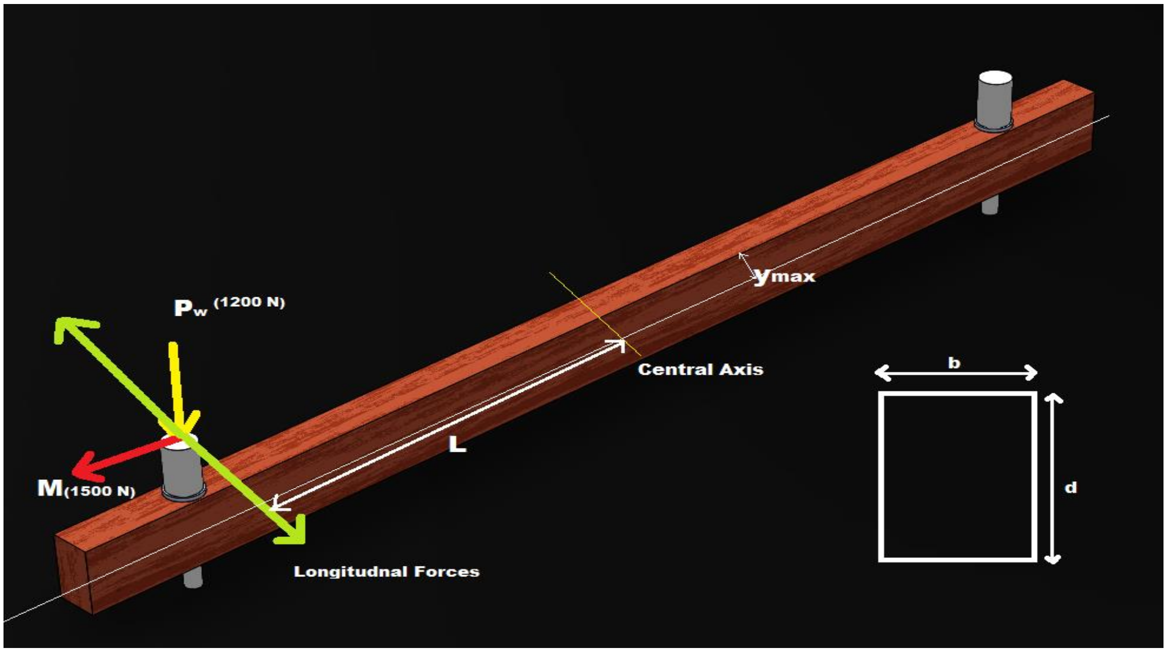


Figure 8.1.4.2

Geometry(inches)	Type of wood	Analytical values		ANSYS values		ERROR %	
		Deflection (mm)	Normal Stresses (Mpa)	Deflection (mm)	Normal Stresses (Mpa)	Deflection (mm)	Normal Stress (Mpa)
3.625 x 4.625 x 96	Eucalyptus	2.608	7.189	2.3965	6.4722	8%	10%
3.625 x 4.625 x 96	Southern Pine	4.978	7.189	4.5614	6.4726	8%	10%
3.625 x 4.625 x 96	Western Red Cedar	8.9	7.189	8.0929	6.4679	9%	10%
2 x 4 x 60	Southern Pine	3.517	10.94	3.23	9.1485	8%	16%
4 x 6 x 144	Southern Pine	6.792	5.814	6.3985	5.5098	6%	5%

Table 8.1.4.1

8.1.5 Inference for wood,

1. Case 1 – Highest stresses were developed on top part of the beam around the hole. Stresses developed at the insulator pin and the washers were also high, hence care should be taken while selecting them.
2. Case 2 – Beam with the largest size had largest deformation and large stresses acting on it.
3. Case 3 – Beam made of eucalyptus had highest strength and the least deflection.

From the normal stress and shear stress obtained from case 2 and case 3, we formulate the factor of safety using the formula,

Factor of Safety = Modulus of Rupture / working (normal stress)

Dimension (inches)	Factor of safety
2 X 4 X 60	4.29
3.625 X 4.625 X 96	5.26
4 X 6 X 144	4.3

Table 8.1.5.1

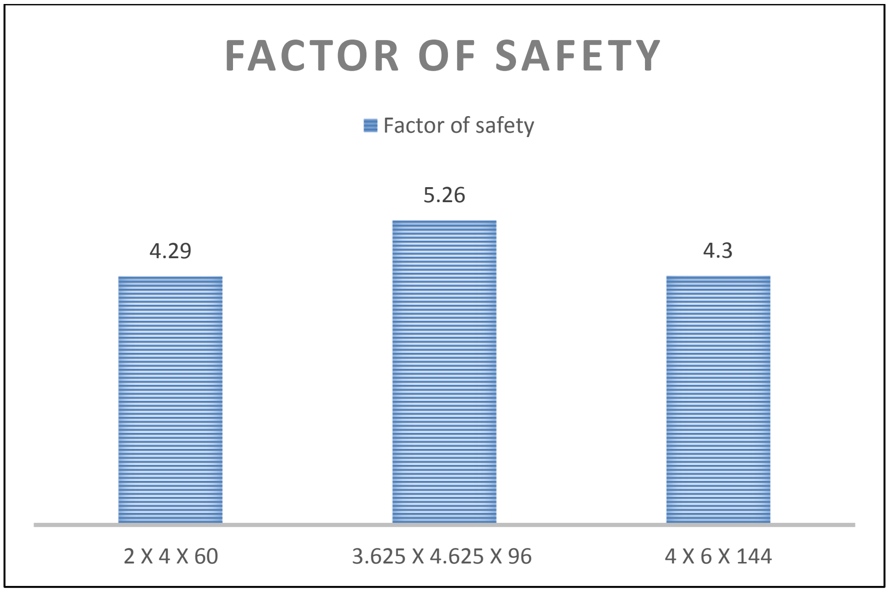


Figure 8.1.5.1

Wood	Factor of safety
Western Red Cedar	2.67
Southern Pine	5.26
Eucalyptus	6.34

Table 8.1.5.2

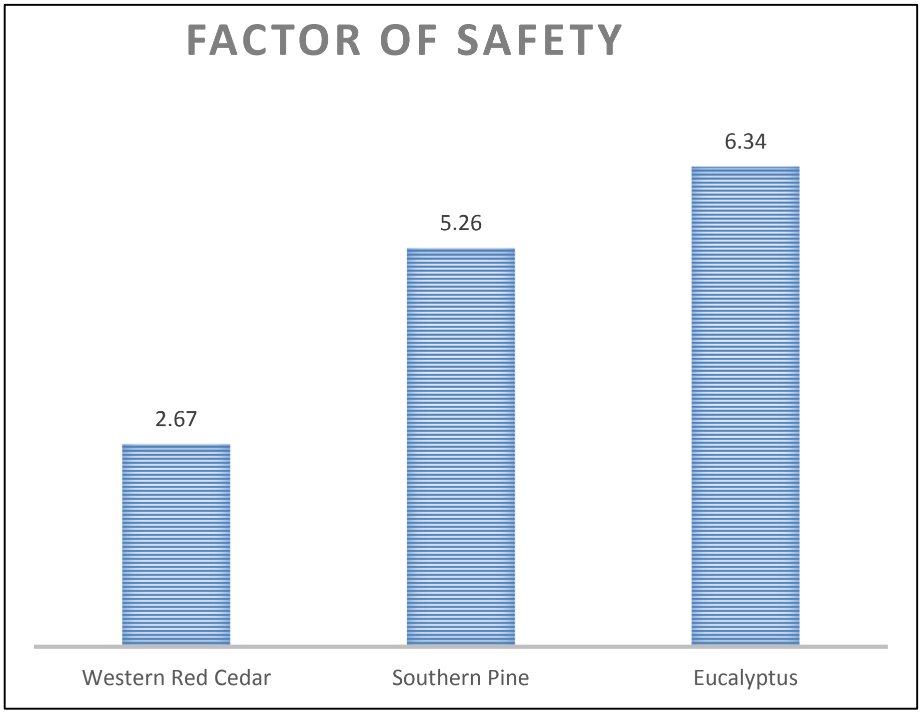


Figure 8.1.5.2

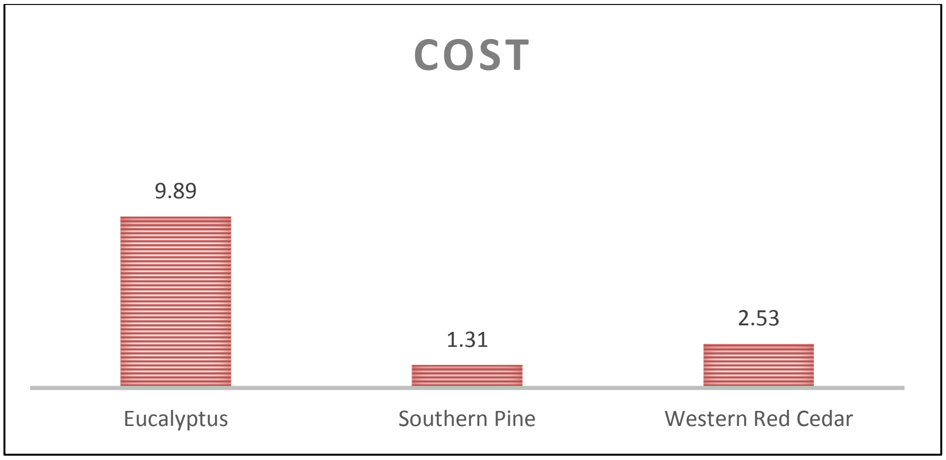


Figure 8.1.5.3

Figure 8.1.5.3 shows the cost of the woods for (3.625 X 4.625 X 12) inch configuration. It should be noted that these prices were linearized for this particular volume.

In case 2, factor of safety for the beam with configurations (3.625 X 4.625 X 96) inch is the highest. In case 3, factor of safety of both Eucalyptus and Southern Pine wood are quite close. Keeping in mind about the cost of the material**,** Southern Pine Wood with (3.625 X 4.625 X 96) inch would be the best option.

8.2 Results for composites

8.2.1 Case 1 – Varying ply thickness and percentage of 45 degree plies

Ply number	0% 45deg	12.5% 45deg	25% 45 deg	50% 45 deg
1	0	0	0	0
2	0	0	0	0
3	0	0	0	-45
4	0	0	-45	45
5	0	0	45	0
6	0	0	0	-45
7	0	0	0	45
8	0	-45	0	0
9	0	45	0	0
10	0	0	0	45
11	0	0	0	-45
12	0	0	45	0
13	0	0	-45	45
14	0	0	0	-45
15	0	0	0	0
16	0	0	0	0

Table 8.2.1.1

Sl No.	Number of plies	Wall thickness (mm)	Lamina thickness(mm)
1	16	8	0.5
2	16	6	0.375
3	16	4	0.25
4	16	3	0.1875

Table 8.2.1.2

Table 8.2.1.1 shows the stack up pattern of various plies and table 8.2.1.2 shows the varying thickness of the lamina as the wall thickness varies. Table8.2.1.3 is as shown below. It tabulates all the maximum normal and shear stresses & its corresponding factor of safety

Ply angle and thickness		LEFT		RIGHT		FOS - Normal Stress	FOS - Shear stress
		Maximum Shear Stress (Mpa)	Maximum Normal Stress(Mpa)	Maximum Shear Stress(Mpa)	Maximum Normal Stress(Mpa)		
3mm	0% 45 deg	6.35	43.04	5.03	32.22	48.83	6.69
	12.5%45deg	6.16	45.30	5.06	33.74	40.60	16.69
	25%45deg	6.19	48.96	5.13	35.73	32.19	26.37
	50%45deg	6.02	57.74	5.25	42.03	18.20	47.15
4mm	0% 45 deg	4.96	34.90	5.75	28.61	60.22	7.39
	12.5%45deg	5.57	35.10	4.99	27.89	52.39	18.47
	25%45deg	5.62	37.60	5.03	28.32	41.91	29.03
	50%45deg	5.57	44.09	5.14	32.51	23.83	50.97
6mm	0% 45 deg	2.97	32.23	7.86	29.54	65.21	5.41
	12.5%45deg	3.11	32.79	7.85	29.08	56.08	13.11
	25%45deg	3.31	34.35	7.90	29.43	45.88	20.65
	50%45deg	3.56	36.63	7.85	29.21	28.68	36.16
8mm	0% 45 deg	3.36	26.93	2.16	26.18	78.03	12.64
	12.5%45deg	3.36	27.35	7.97	25.96	67.23	12.91
	25%45deg	3.34	28.30	7.36	26.16	55.70	22.18
	50%45deg	3.32	29.87	7.58	26.27	35.18	37.46

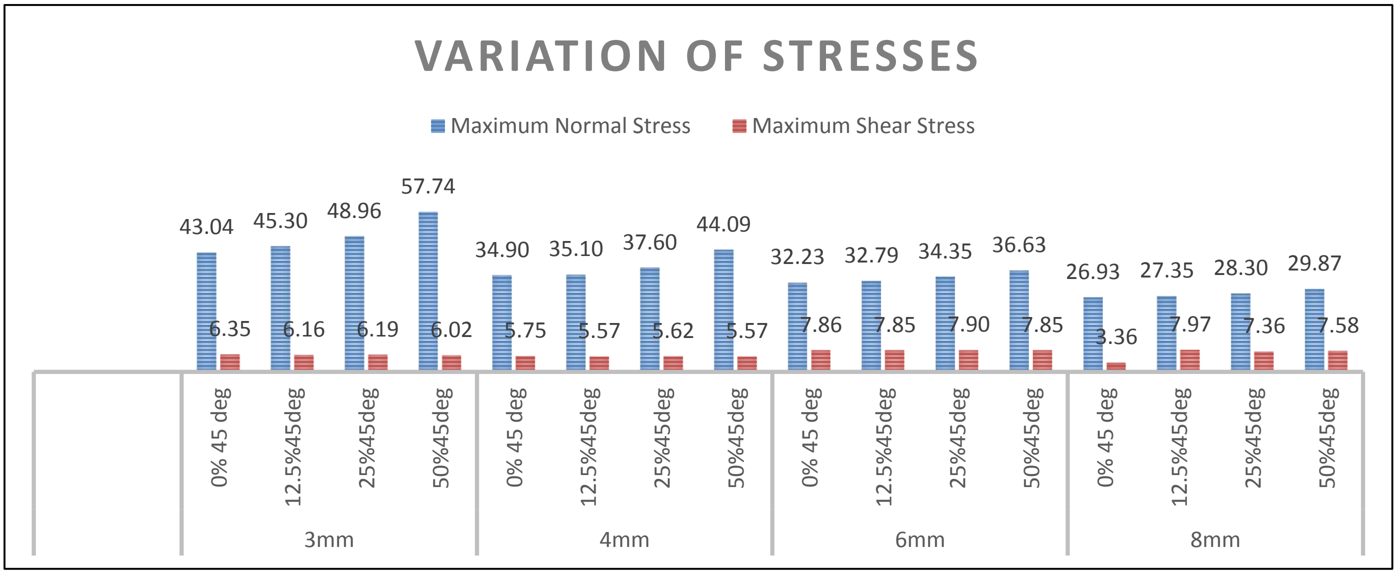


Figure 8.2.1.4

Figure 8.2.1.4 shows the maximum shear and normal stresses for different thicknesses & percentages of 45-degree plies in the stack up. It can be observed that normal stresses are maximum in case of 3mm thick wall. Shear stress is maximum for 8mm thick wall.

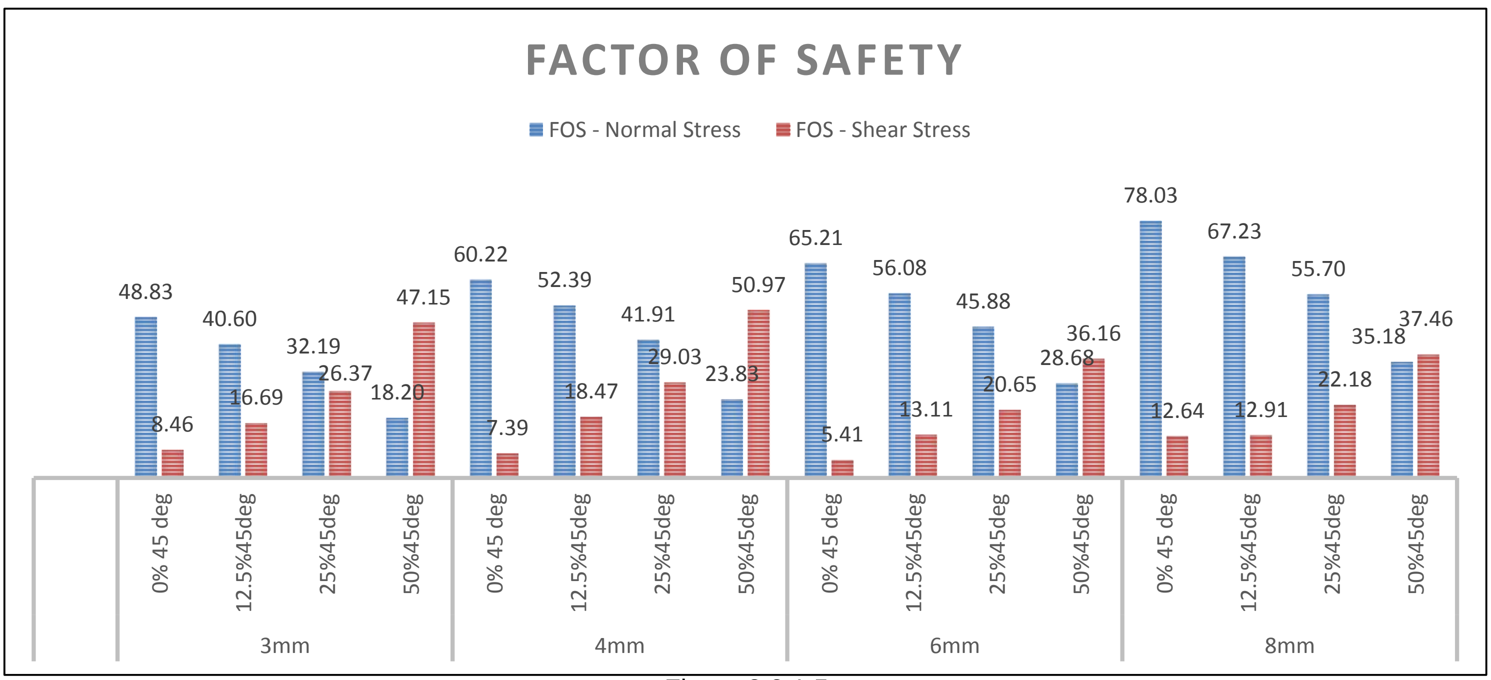


Figure 8.2.1.5

Figure 8.2.1.5 shows the factor of safety for different thickness of the hollow wall and for different ply stack-up.

It can be observed that the safety of factor for 8 mm, 6mm and 4mm are pretty high, which is not required. As we know that if the safety of factor is higher than required then the material is considered to be wasted. Thus, 3 mm thick wall can be considered to be sufficient for our application.

Thickness (mm)	Deflections(mm)			
	0%-45deg	12.5%-45deg	25%-45deg	50%-45deg
3mm	6.97	7.44	8.04	9.83
4mm	5.39	5.76	6.21	7.49
6mm	3.83	4.08	4.39	5.25
8mm	3.05	3.25	3.49	4.15

Table 8.2.1.4

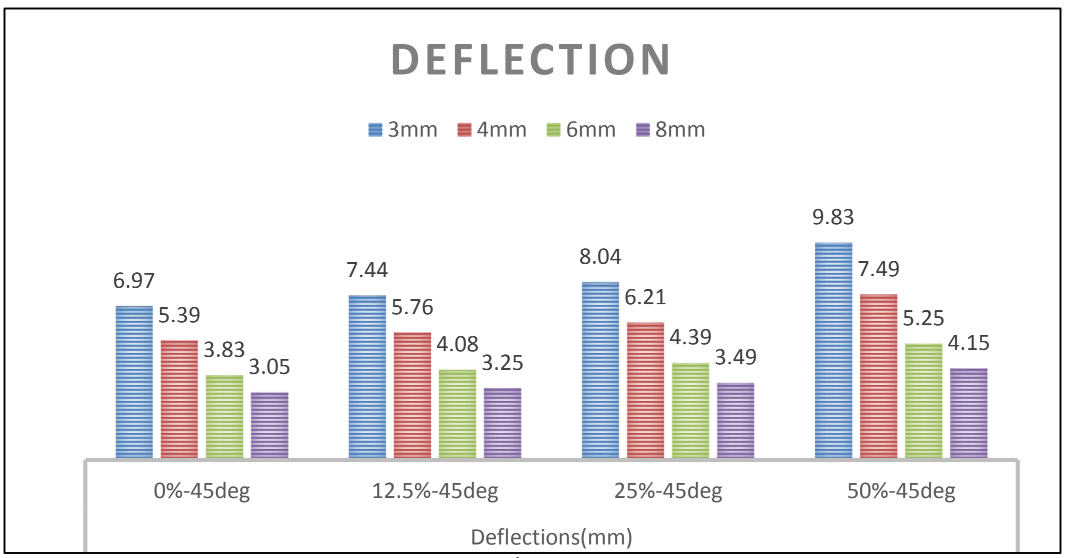


Figure 8.2.1.7

Figure 8.2.1.7 shows the figure showing the deflections of various cases. We can observe that deflection is maximum for 3mm wall thickness and it increases linearly as the thickness of the wall increases. The deflection in case of 3mm thick wall and 25%-45-

degree ply stack-up is 8.04 mm. This deflection is acceptable since it is in almost the same deflection range as that of wood.

It should be also noted that we are considering safety factor for both normal and shear stress. Hence, we should select a stack-up such that both shear and normal factor of safeties are high. Thus, 3mm thick wall with 25%-45-degree ply stack up will be the best option out of the rest with respect to deflection and the factor of safeties.

8.2.2 Case 2 - Dead End Assembly loading conditions

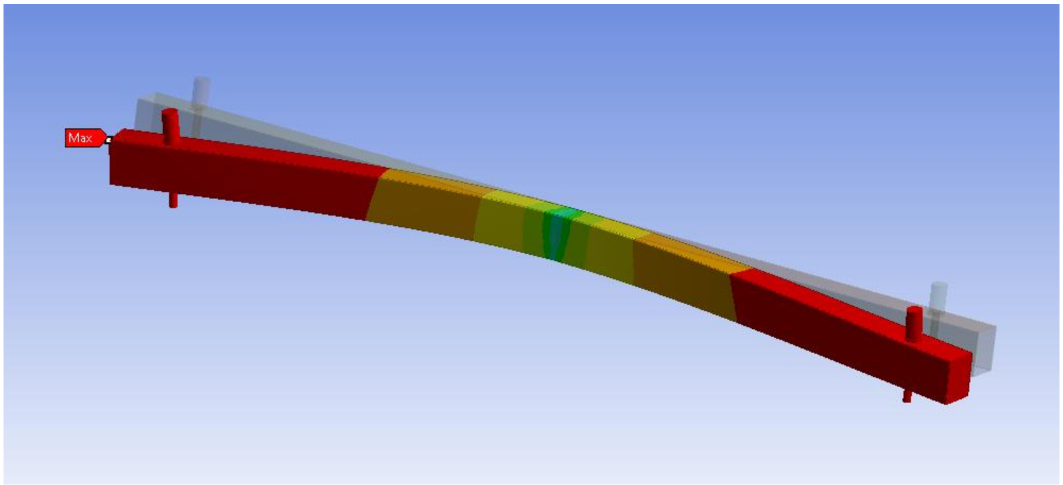


Figure 8-2.2.1

Figure 8.2.2.1 shows the dead end assembly crossarm. Refer figure 1-2(c) for a real image. A pulling force is acting from the conductors in one direction only. Thus an additional 1500 N is applied in the positive Z direction and the transverse forces are cut-off.

Location	Maximum Normal Stress (Mpa)	Maximum Shear Stress (Mpa)
Top part	90.08	31.26
Front part	22.49	11.58
Bottom Part	39.96	18.53
Back Part	98.33	12.24

Table 8.2.2.1

Table 8.2.2.1 and figure 8.2.2.1 shows the normal and shear stress acting at different parts of the hollow structure. It can be observed that the maximum normal stress is acting at the back part of the cross-arm and maximum shear stress is acting at the top surface of the crossarm.

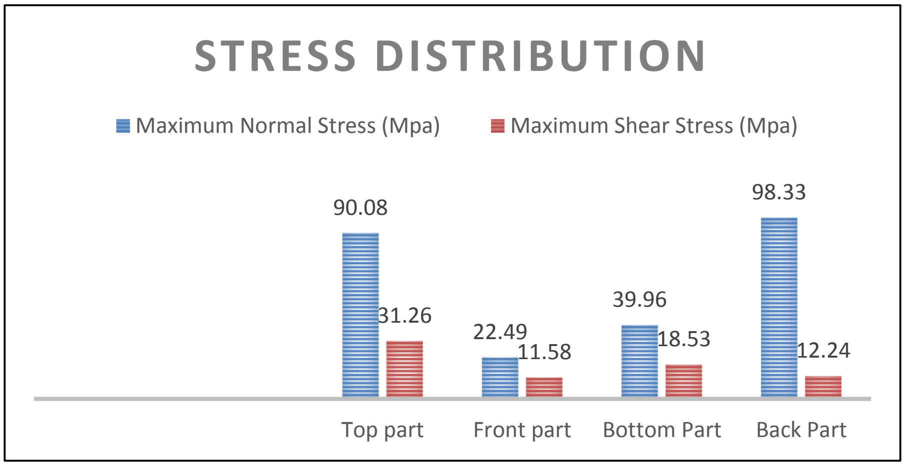


Figure 8.2.2.2

Properties	Dead End Assembly	Tangential Beam Condition
Deflection	13.84	8.04
Maximum Shear Stress	23.59	5.14
Maximum Normal Stress	98.33	48.96
Factor of Safety (Normal Stress)	8.88	42.93
Factor of Safety (Shear Stress)	6.92	26.37

Table 8.2.2.2

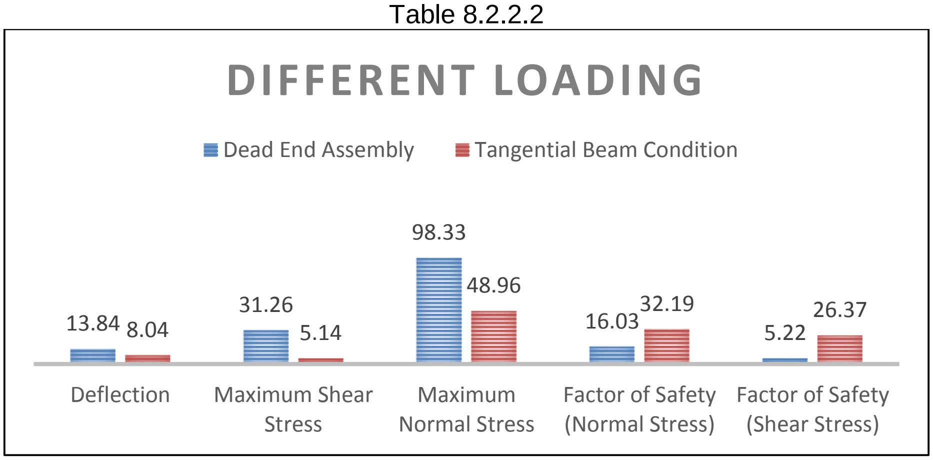


Figure 8.2.2.3

The factor of safety calculated for the above loading condition proves it to be safe.

A comparison has been shown in table 8.2.2.2 and figure 8.2.2.3 in terms of deflection, normal stress, shear stress, factor of safety for both normal & shear stress has been shown for different loading conditions. It can be observed that deflection, normal & shear stresses developed for dead end assembly crossarms are higher.

8.2.3 Case 3 - Using S-2 Glass

S-2 glass has higher strength as compared to E-glass. Hence it is preferred for structural applications.

Properties	S-2 Glass	E-Glass
Deflection(mm)	7.38	8.04
Maximum Normal Stress(Mpa)	49.79	48.96
Maximum Shear Stress	6.30	6.18
Cost of the current beam ($/ft.)	14.58	4.59
Factor Of Safety (Normal Stress)	44.93	32.19
Factor Of Safety (Shear Stress)	25.89	26.4

Table 8.2.3.1

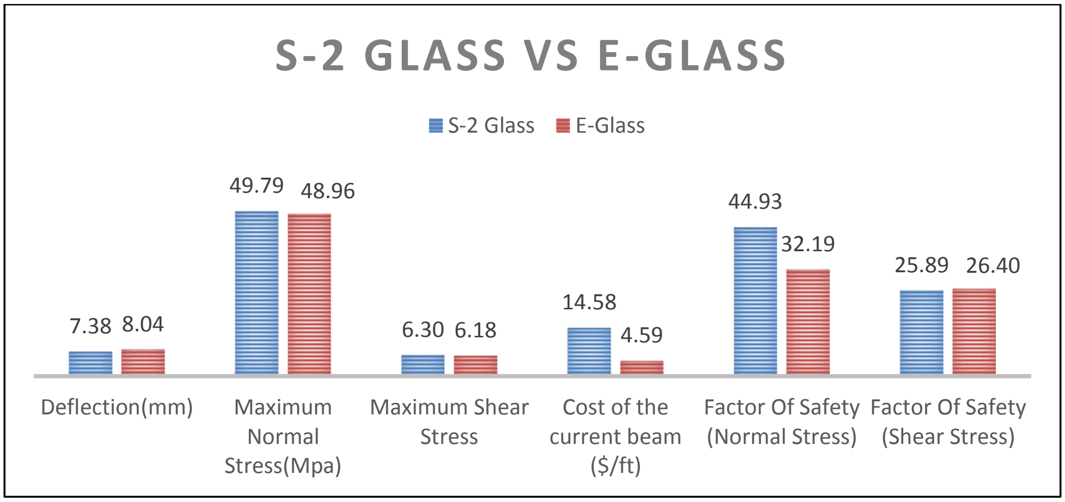


Figure 8.2.3.1

It can be observed that the deflection for S-2 glass is 8% lesser than that of E-glass. The factor of safety for S-2 glass in case of normal stress is about 28% more than that of E-glass. The factor of safety for E-glass in case of shear stress is about 1% more than that of S-glass. The cost of S-2 glass is about 68% greater than that of E-glass.

Since E-glass has already enough safety factor and difference in deflection being too less as compared to S-2 glass, from the cost perspective, E-glass is preferred over S-glass.

8.2.4 Case 4 - Changing the dimension to (4 X 6 X 96) inch

Properties	3.625 x 4.625 x 96	4 x 6 x 96
Deflection	7.44	4.29
Maximum Normal Stress	74.11	46.12
Maximum Shear Stress	5.13	4.15
Cost of the beams($/ft.)	4.59	6.14
Factor of safety (Normal Stress)	32.19	34.17
Factor of Safety (Shear Stress)	26.37	39.33

Table 8.2.4.1

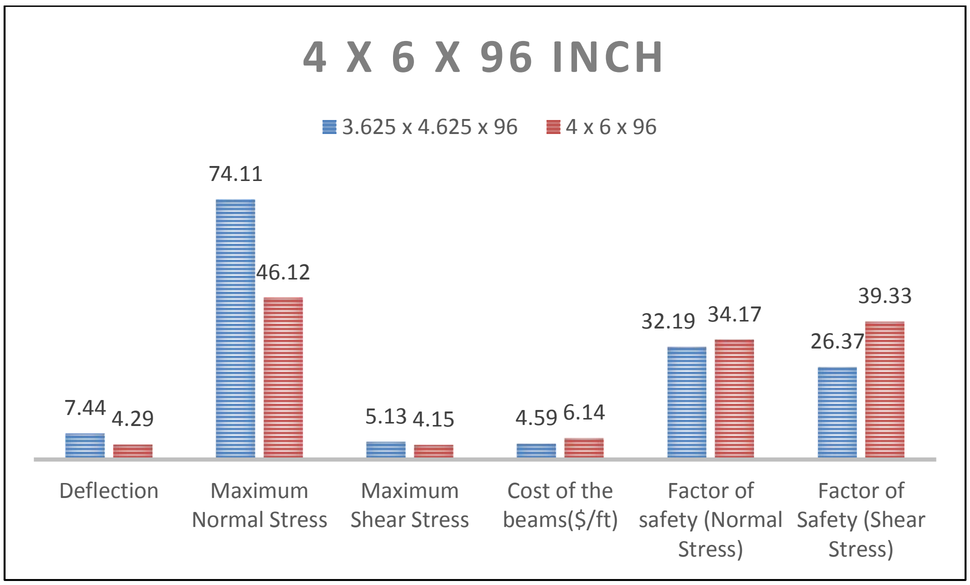


Figure 8.2.4.1

It can be observed that changing the dimension and keeping the length constant, the deflection reduces considerably by 42%. The factor of safety increases by 5 % in case of normal stress and by 33% in case of shear stress.

The design gets better with increase in the dimensions. This will result in the increase of the material cost by 25 %.

8.2.5 Analytical Calculations for composites crossarms

The following steps were followed, [13]

1.The moment of Inertia was found using I = $\frac{\boldsymbol{bd^3-(b-2t)*(d-2t)^3}}{\boldsymbol{12}}$

2.Considering half of the crossarm, deflection of the cantilever beam was calculated at the pin using the equation

W = $\frac{P_w L^3}{3EI} + \frac{ML^2}{2EI}$

3.The total deflection at the beam end was then calculated W_e= W + L * $\frac{P_w L+M}{EI}$ * x in mm

4.Finally stress is calculated by using $\sigma = \frac{M}{I}$* $y_{max} + \frac{P_{wind}}{A}$ in Pascal.

5.Comparisons are made with ANSYS result. Normal stresses were considered away from the stress concentration regions & the central line and the edge as shown in the figure.

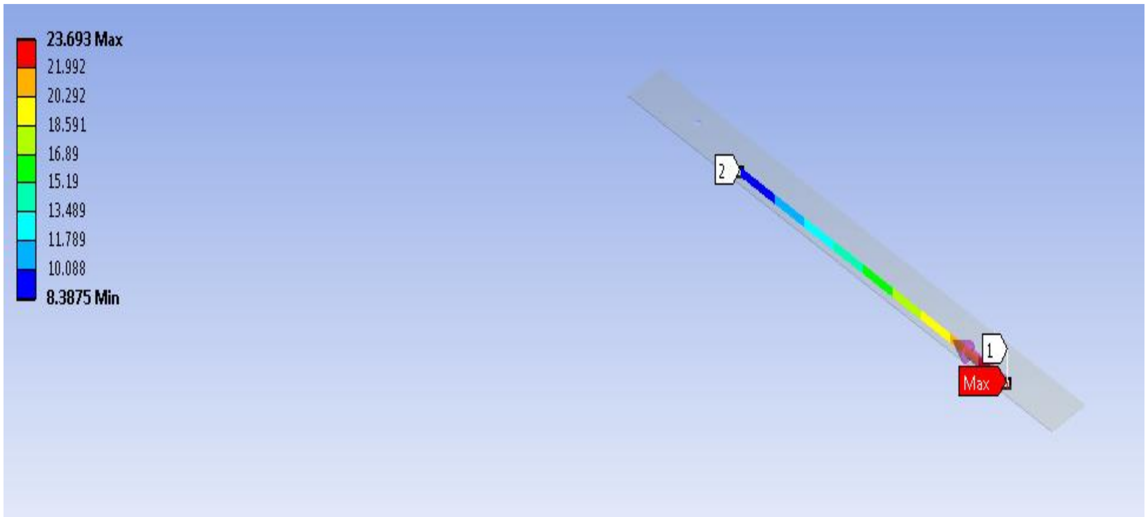


Figure 8.2.5.1

Figure 8-2.5.1 shows the line along which stresses has been taken for comparison from ANSYS. It is done to eliminate the maximum stress concentration which occurs at the central support and the stress concentration at the holes near its ends.

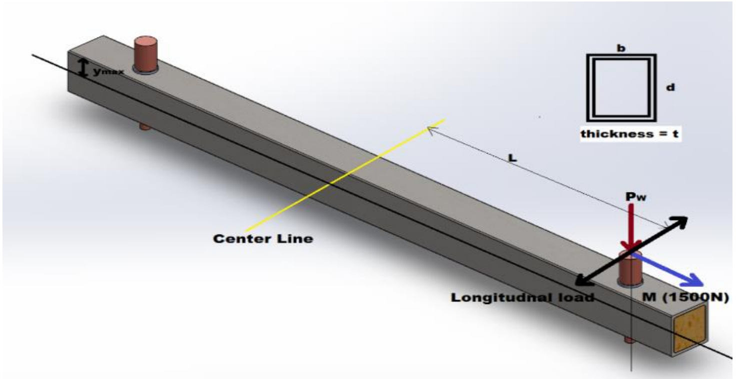


Figure 8.2.5.2

Dimension (inches)	Thickness of the wall (mm)	Analytical values		ANSYS values		ERROR %	
		Deflection (mm)	Normal Stresses (Mpa)	Deflection (mm)	Normal Stresses (Mpa)	Deflection (mm)	Normal Stress (Mpa)
3.625 x 4.625 x 96	3	7.53	35.18	7.44	34.95	1%	1%
3.625 x 4.625 x 96	4	5.81	27.15	5.76	27.03	1%	0%
3.625 x 4.625 x 96	6	4.09	19.16	4.08	19.02	0%	1%
3.625 x 4.625 x 96	8	3.24	15.22	3.25	15.13	0%	1%
4 x 6 x 96	3	3.87	23.35	3.99	23.69	3%	1%

Table 8.2.5.1

It can be observed in this case that the percentage of error is quite negligible and hence the analytical and ANSYS results matches.

8.2.6 Inferences for composites

- It can be inferred from case 1 that the crossarm with greater thickness have a higher factor of safety which is not required. The factor of safety for 3 mm wall thickness are good enough to resist failure. By varying the fibre architecture i.e., by changing the percentage of 45-degree ply stack-up, we observed the differences in the factor of safety for both normal and shear stresses. Since the 25 % 45-degree ply stack-up has higher factor of safety in case of both shear and normal stresses for a 3mm thick wall, it is the best option.
- In case of different load conditions and different configuration the structure was safe.
- In case of comparison with S-2 glass for small benefits in deflection the cost is too high which is tough to market.

Chapter – 9

Comparison

9.1 Comparison of wood and composite crossarm based on deflection

Material	Maximum deflection (mm)
Eucalyptus	2.40
Southern Pine	4.56
Western Red Cedar	8.09
E-glass(3mm - 0% 45 deg)	6.97
E-glass(3mm - 12.5% 45 deg)	7.44
E-glass(3mm - 25% 45 deg)	8.04
E-glass(3mm - 50% 45 deg)	9.83
S2-glass(3mm-25%)	7.98

Table 9.1.1

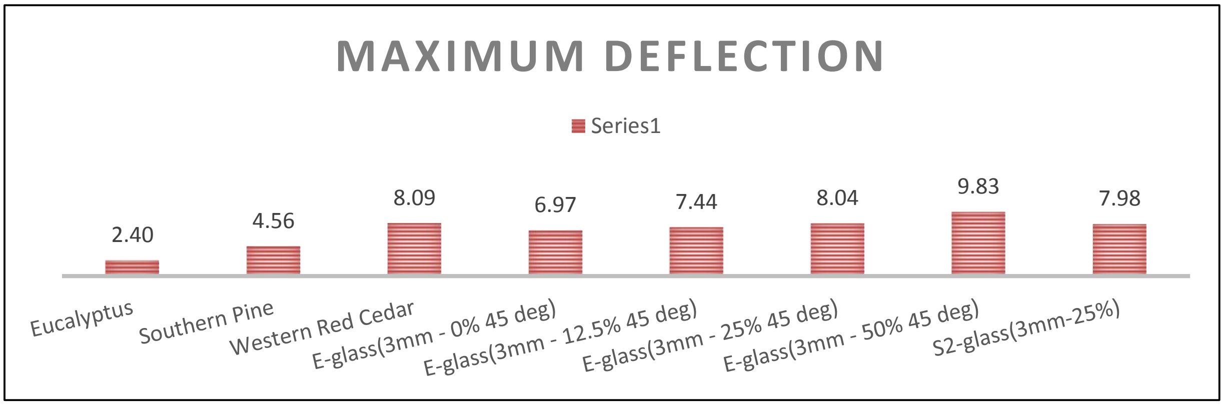


Figure 9.1.1

It can be observed that the deflection in case of the proposed E-glass 25% 45-degree ply stack-up is very close to the deflection of Western Red Cedar. It is about 43% lesser than that of Southern Pine and about 70% less than that of eucalyptus. It should be observed that in case of wood higher the mechanical properties, lesser is the deflection. In case of composites, as the percentage of 45-degree plies increases in the stack-up, the deflection also increases. This can be accounted for the reduction in the 0-degree plies which is responsible for maintaining the stiffness.

9.2 Comparison of wood and composite crossarm based on mass

MATERIAL	MASS(kg)
Eucalyptus	27.88
Southern Pine	17.10
Western Red Cedar	9.73
E-glass (8mm)	15.28
E-glass (6mm)	12.43
E-glass (4mm)	9.27
E-glass (3mm)	7.57

Table 9.2.1

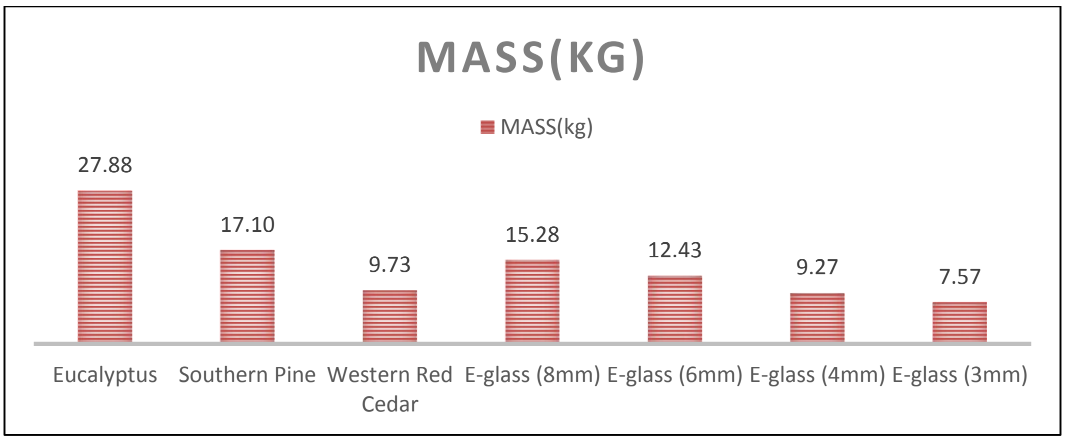


Figure 9.2.1

Mass of the crossarm was compared keeping the volume constant which is (3.625 X 4.625 X 96) inches. It can be observed that the mass of Eucalyptus wood is the highest. The proposed model of 3mm weighs 50% lesser than the current existing model of 8mm.When we compare the most commonly used wood i.e., Southern pine wood with the proposed model, Southern pine weighs more than 2.25 times as that of the proposed model.

9.3 Comparison of wood and composite crossarm based on factor of safety

MATERIAL	FACTOR OF SAFETY
Eucalyptus	6.34
Southern Pine	5.26
Western Red Cedar	2.67
E-glass (Shear)	26.37
E-glass (Normal)	32.19

Table 9.3.1

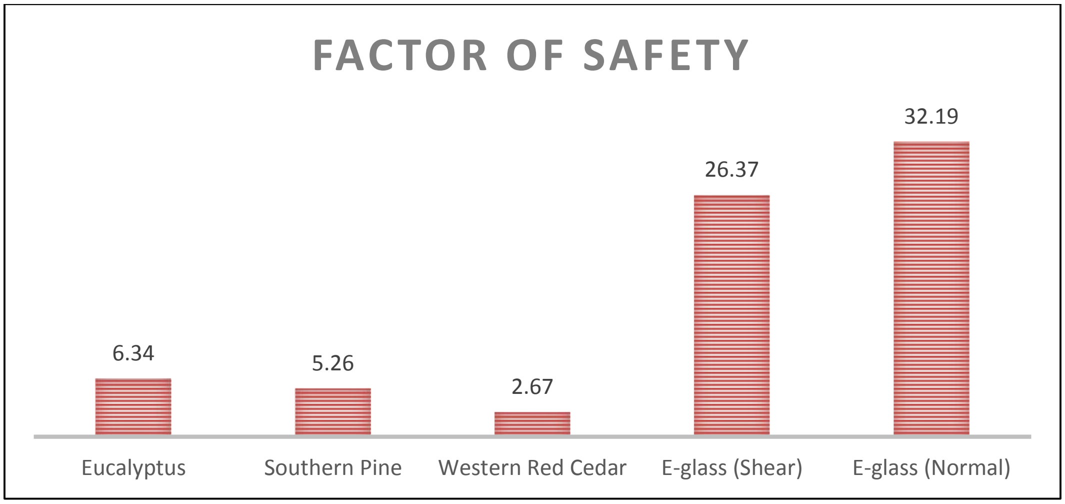


Figure 9.3.1

The factor of safety for wood is much less as compared to that of composites. As an illustration if we compare the strongest wood out of the three different types of wood that has been used in this research, we find that the factor of safety of E-glass is about 5 times as that of Eucalyptus. Analytically, this means that life cycle of the proposed model is 5 times as that of Eucalyptus.

9.4 Comparison of wood and composite crossarm based on Cost

MATERIAL	COST($/ft.)
Eucalyptus	9.89
Southern Pine	1.31
Western Red Cedar	2.53
E-glass (8mm) ,Foam	7.74
E-glass (6mm) ,Foam	6.52
E-glass (4mm) ,Foam	5.25
E-glass (3mm) ,Foam	4.59
S2-glass(3mm),Foam	14.58

Table 9.4.1

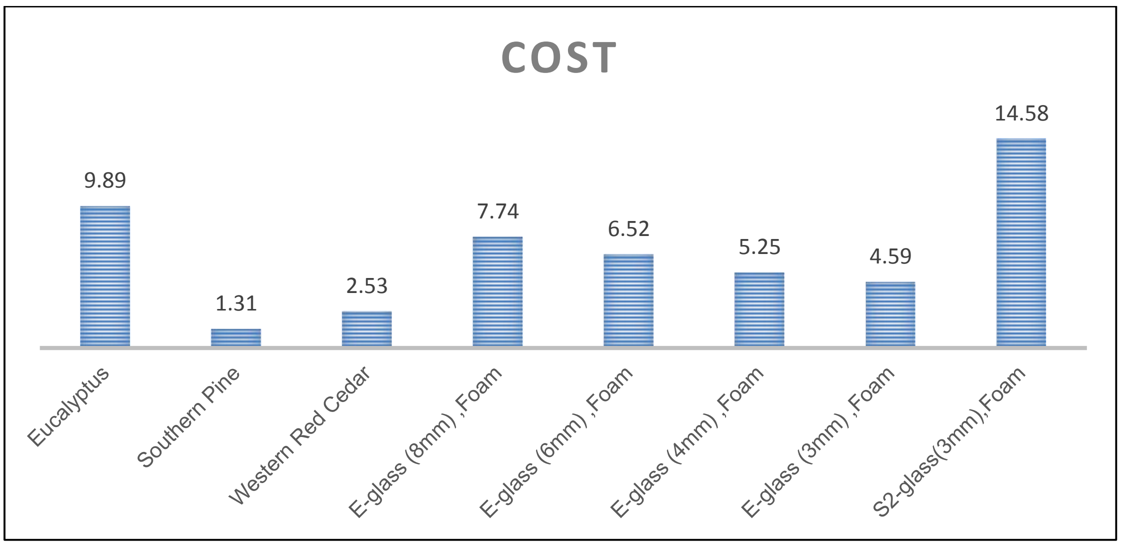


Figure 9.4.1

It should be noted that the cost here refers to the material cost only. Cost of the material for a single volume i.e., (3.625 X 4.625 X 12) inches has been calculated here.

Cost of proposed model is about 3.5 times more than that of wood. When it is compared to the existing model, it costs about 41 % lesser.

Chapter -10

Conclusion

- The proposed design weighs 50% lesser than the current design. Thus, reducing the material cost by 41%.
- Composite has bigger advantages than wood- structurally that is, it has a higher factor of safety which can withstand loads in severe weather conditions and gusty winds.
- Cost of composite is slightly higher than wood, however this can be decreased by reducing the thickness and by optimising fibre architecture.
- Composite structures have significantly lower mass which would reduce the material handling and serviceability costs.
- Lifetime prediction for composite crossarms is high, which would approximate to about 2 to 4 times greater than that of wood.

Future Work

- Dynamic analysis of the entire system can be made with the cables and its effect on the pole
- Analysis based on hybrid composites can be done

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Biographical Information

Amith Venkatesh was born in Bangalore, Karnataka, India in January1992. He received his Bachelor of Engineering in Mechanical Engineering from Visvesvaraya Technological University, India in 2014. He represented his school and country at the SAE Baja competition held at Yeungnam University, South Korea in July 2014. He worked for Omkar CNC Center as a Floor Manager from July 2014 to July 2015. He then worked for Bosch Private limited from July 2015 to December 2015. He enrolled into Master of Science in Mechanical and Aerospace Engineering program at the University of Texas at Arlington in Spring 2016. He started working on topics related to composite materials from summer 2016 under Dr. Andrey Beyle. He is proficient in ANSYS Workbench ACP Pre-post, Static Structural and SOLIDWORKS. He graduated in May 2017.