DESIGN AND ANALYSIS OF A DRIVE SHAFT BY USING COMPOSITE MATERIALS

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ABSTRACT

Substituting composite structures for conventional metallic structures has many advantages because of higher specific stiffness and strength of composite materials. This work deals with the replacement of conventional two-piece steel drive shafts with a single-piece e-glass/ epoxy, high strength carbon/epoxy and high modulus carbon/epoxy composite drive shaft for an automotive application. Use of advanced composites has resulted in remarkable achievements in many fields including aviation, marine and automobile engineering, medicine, prosthetics and sports, in terms of improved fatigue and corrosion resistances, high specific strength and specific modulus and reduction in energy requirements owing to reduction in weight. The aim of this work is to replace the conventional steel driveshaft of automobiles with an appropriate composite driveshaft. The finite element analysis results are compared with already existed steel drive shaft values and it is observed that the single piece composite drive shaft is suitable for driveline application. The design optimization also showed significant potential improvement in the performance of drive shaft. In this present work an attempt has been to estimate the deflection, shear stresses by using ansys 14.5.

Keywords:-*Torque Transmission, Static analysis, Modal analysis, Buckling deformation, Ansys*.

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1. INTRODUCTION

The advanced composite materials such as Graphite, Carbon, Kevlar and Glass with suitable resins are widely used because of their high specific strength (strength/density) and high specific modulus (modulus/density). Advanced composite materials seem ideally suited for long, power driver shaft (propeller shaft) applications. Their elastic properties can be tailored to increase the torque they can carry as well as the rotational speed at which they operate. The drive shafts are used in automotive, aircraft and aerospace applications. The automotive industry is exploiting composite material technology for structural components construction in order to obtain the reduction of the weight without decrease in vehicle quality and reliability. It is known that energy conservation is one of the most important objectives in vehicle design and reduction of weight is one of the most effective measures to obtain this result. Actually, there is almost a direct proportionality between the weight of a vehicle and its fuel consumption, particularly in city driving.

1.1 Description of the Problem

The fundamental natural bending frequency for passenger cars, small trucks, and vans of the propeller shaft should be higher than 6,500 rpm to avoid whirling vibration and the torque transmission capability of the drive shaft should be larger than 3,500 Nm. The drive shaft outer diameter should not exceed 100 mm due to space limitations. Because in case of front engine rear wheel drive vehicles size of drive shaft increases chassis height and reduces floor space in passenger compartment. So here outer diameter of the shaft is taken as 90 mm with little compromise between strength of drive shaft and decrease chassis height.

Presently many SUVs and light commercial vehicle using front engine rear wheel drive system. The constraints for selecting propeller shaft dimensions are wheel base of vehicle, maximum torque transmission capacity and angular velocity of propeller shaft. The drive shaft of transmission system is to be designed optimally for following specified design.

In actual operating condition shaft is subjected to three types of loads, which are following torsional, vibrations and buckling. We are going to analyze the drive shaft for the torsional, modal and buckling analysis using steel and four different composite materials by varying the ply angle, no. of plies and ply thickness.

Almost all automobiles (at least those which correspond to design with rear wheel drive and front engine installation) have transmission shafts. The weight reduction of the drive shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal, if it can be achieved without increase in cost and decrease in quality and reliability.

1.2 Aim and Scope of the Work

This work deals with the replacement of a conventional steel drive shaft with E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composite drive shafts for an automobile application.

1.3 Optimum Design Using Genetic Algorithm

The design parameters are to be optimized for E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composite drive shafts of an automobile using Genetic Algorithm. The purpose of using Genetic Algorithm is to minimize the weight of the shaft, which is subjected to the constraints such as torque transmission, torsion buckling capacities and fundamental lateral natural frequency.

The design parameters to be optimized are,

- Ply thickness
- Number of plies required
- Stacking sequence of Laminate

1.4 Analysis

Modeling of the High Strength Carbon/Epoxy composite drive shaft using ANSYS.

Static, Modal and Buckling analysis are to be carried out on the finite element model of the High Strength Carbon/Epoxy composite drive shaft using ANSYS.

2. DESIGN OF STEEL DRIVE SHAFT

2.1 Specification of the Problem

The fundamental natural bending frequency for passenger cars, small trucks, and vans of the propeller shaft should be higher than 6,500 rpm to avoid whirling vibration and the torque transmission capability of the drive shaft should be larger than 3,500 Nm. The drive shaft outer diameter should not exceed 100 mm due to space limitations. Here outer diameter of the shaft is taken as 90 mm. The drive shaft of transmission system is to be designed optimally for following specified design requirements.

Table 2.1 Design requirements and specifications

S.NO	Specifications	Values
1.	Ultimate torque(T_{max})	3500 Nm
2.	Maximum speed of shaft(N_{max})	6500 rpm
3.	Length of shaft(L)	1250 mm

Steel (SM45C) used for automotive drive shaft applications. The material properties of the steel (SM45C) are given in Table 4.2. The steel drive shaft should satisfy three design specifications such as torque transmission capability, buckling torque capability and bending natural frequency.

2.2 Torque transmission capacity of the drive shaft

$T = SS\ \pi(d_o^4 - di^4)/16d_o$ …. (1) Taking factor of safety as 3.

$T = 2464.2$ Nm (where shear strength=123.3 MPa,d_0=90 mm,d_i=86.68 mm)

2.3 Mass of steel drive shaft

$$m= \varrho AL= \varrho\ (d_o^2 - d_i^2) \times L/4$$

Where
d_o = outer diameter (m)
di = inner diameter (m)
m = 8.58 Kg

3.DESIGN OF A COMPOSITE DRIVE SHAFT

3.1 Specification of the Problem

The specifications of the composite drive shaft of an automotive transmission are same as that of the steel drive shaft for optimal design.

3.2 Assumptions

1. The shaft rotates at a constant speed about its longitudinal axis.
2. The shaft has a uniform, circular cross section.
3. The shaft is perfectly balanced, i.e., at every cross section, the mass center coincides with the geometric center.
4. All damping and nonlinear effects are excluded.
5. The stress-strain relationship for composite material is linear & elastic; hence, Hooke's law is applicable for composite materials.
6. Acoustical fluid interactions are neglected, i.e., the shaft is assumed to be acting in a vacuum.
7. Since lamina is thin and no out-of-plane loads are applied, it is considered as under the plane stress.

3.3 Selection of Cross-Section

The drive shaft can be solid circular or hollow circular. Here hollow circular cross-section was chosen because:

- The hollow circular shafts are stronger in per kg weight than solid circular.
- The stress distribution in case of solid shaft is zero at the center and maximum at the outer surface while in hollow shaft stress variation is smaller. In solid shafts the material close to the center are not fully utilized.

3.4 Selection of Reinforcement Fiber

Fibers are available with widely differing properties. Review of the design and performance requirements usually dictate the fiber/fibers to be used.

Carbon/Graphite fibers: Its advantages include high specific strength and modulus, low coefficient of thermal expansion, and high fatigue strength. Graphite, when used alone has low impact resistance. Its drawbacks include high cost, low impact resistance, and high electrical conductivity.

Glass fibers: Its advantages include its low cost, high strength, high chemical resistance, and good insulating properties. The disadvantages are low elastic modulus, poor adhesion to polymers, low fatigue strength, and high density, which increase shaft size and weight. Also crack detection becomes difficult.

Kevlar fibers: Its advantages are low density, high tensile strength, low cost, and higher impact resistance. The disadvantages are very low compressive strength, marginal shear strength, and high water absorption. Kevlar is not recommended for use in torque carrying application because of its low strength in compression and shear. Here, both glass and carbon fibers are selected as potential materials for the design of shaft.

3.5 Selection of Resin System

The important considerations in selecting resin are cost, temperature capability, elongation to failure and resistance to impact (a function of modulus of elongation). The resins selected for most of the drive shafts are either epoxies or vinyl esters. Here, epoxy resin was selected due to its high strength, good wetting of fibers, lower curing shrinkage, and better dimensional stability.

3.6 Selection of Materials

Based on the advantages discussed earlier, the E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy materials are selected for composite drive shaft. The E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy materials used for composite drive shafts.

Properties of E-Glass/Epoxy, HS Carbon/Epoxy and HM Carbon/Epoxy

Properties	E GLASS	HM Carbon	HS Carbon
Young's Modulus X direction	5.e+010 Pa	1.90E+11	1.34e+011 Pa
Young's Modulus Y direction	1.2e+010 Pa	7.70E+09	7e+009 Pa
Young's Modulus Z direction	1.2e+010 Pa	7.70E+09	7e+009 Pa
Major Poisson's Ration XY	0.3	3.00E-01	0.3
Major Poisson's Ration YZ	0.3	3.00E-01	0.3
Major Poisson's Ration XZ	0.3	3.00E-01	0.3
Shear modulus XY	5.6e+009 Pa	4.20E+09	5.8e+009 Pa
Shear modulus YZ	5.6e+009 Pa	4.20E+09	5.8e+009 Pa
Shear modulus XZ	5.6e+009 Pa	4.20E+09	5.8e+009 Pa
Density	2000 Kg/m^3	1600 Kg/m^3	1600 Kg/m^3
Allowable Stress	400e+006 Pa	4.40E+08	600000000

Torque transmission capacity of the composite drive shafts

For **E glass epoxy** material, **T=15439.2 Nm**
(where shear strength=400 MPa, d_0=90 mm, d_i=83.2 mm)

For **HM carbon epoxy** material, **T=5519.04 Nm**
(where shear strength=440 MPa, d_0=90 mm, d_i=87.96 mm)

For **HS carbon epoxy** material, **T=7525.9 Nm**
(where shear strength=600 MPa, d_0=90 mm, d_i=87.96 mm)

4. FINITE ELEMENT ANALYSIS

4.1 Introduction

Finite Element Analysis (FEA) is a computer-based numerical technique for calculating the strength and behavior of engineering structures. It can be used to calculate deflection, stress, vibration, buckling behavior and many other phenomena. It also can be used to analyze either small or large-scale deflection under loading or applied displacement. It uses a numerical technique called the finite element method (FEM).

In finite element method, the actual continuum is represented by the finite elements. These elements are considered to be joined at specified joints called nodes or nodal points. As the actual variation of the field variable (like displacement, temperature and pressure or velocity) inside the continuum is not known, the variation of the field variable inside a finite element is approximated by a simple function. The approximating functions are also called as interpolation models and are defined in terms of field variable at the nodes. When the equilibrium equations for the whole continuum are known, the unknowns will be the nodal values of the field variable.

In this project finite element analysis was carried out using the FEA software ANSYS. The primary unknowns in this structural analysis are displacements and other quantities, such as strains, stresses, and reaction forces, are then derived from the nodal displacements.

4.2 Modeling Linear Layered Shells

SHELL99 may be used for layered applications of a structural shell model as shown in Fig 8.1. SHELL99 allows up to 250 layers. The element has six degrees of freedom at each node: translations in the nodal x, y and z directions and rotations about the nodal x, y and z-axes.

4.3 Meshing:

We have selected area mesh for the meshing with the element size of 10, which will provide us fine meshing. We have selected quadrilateral mesh element for accurate and uniform meshing of component. The meshing is the method in which the geometry is divided in small number of elements.

4.4 Static Analysis

Static analysis deals with the conditions of equilibrium of the bodies acted upon by forces. A static analysis can be either linear or non-linear. All types of non-linearities are allowed such as large deformations, plasticity, creep, stress stiffening, contact elements etc. this chapter focuses on static analysis. A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those carried by time varying loads. A static analysis is used to determine the displacements, stresses, strains and forces in structures or components caused by loads that do not induce significant inertia and damping effects. A static analysis can however include steady inertia loads such as gravity, spinning and time varying loads.

In static analysis loading and response conditions are assumed, that is the loads and the structure responses are assumed to vary slowly with respect to time.

The kinds of loading that can be applied in static analysis includes,

1. Externally applied forces, moments and pressures
2. Steady state inertial forces such as gravity and spinning
3. Imposed non-zero displacements

A static analysis result of structural displacements, stresses and strains and forces in structures for components caused by loads will give a clear idea about whether the structure or components will withstand for the applied maximum forces. If the stress values obtained in this analysis crosses the allowable values it will result in the failure of the structure in the static condition itself. To avoid such a failure, this analysis is necessary.

4.4.1 Boundary Conditions

The finite element model of HS Carbon/Epoxy shaft. One end is fixed and torque is applied at other end,

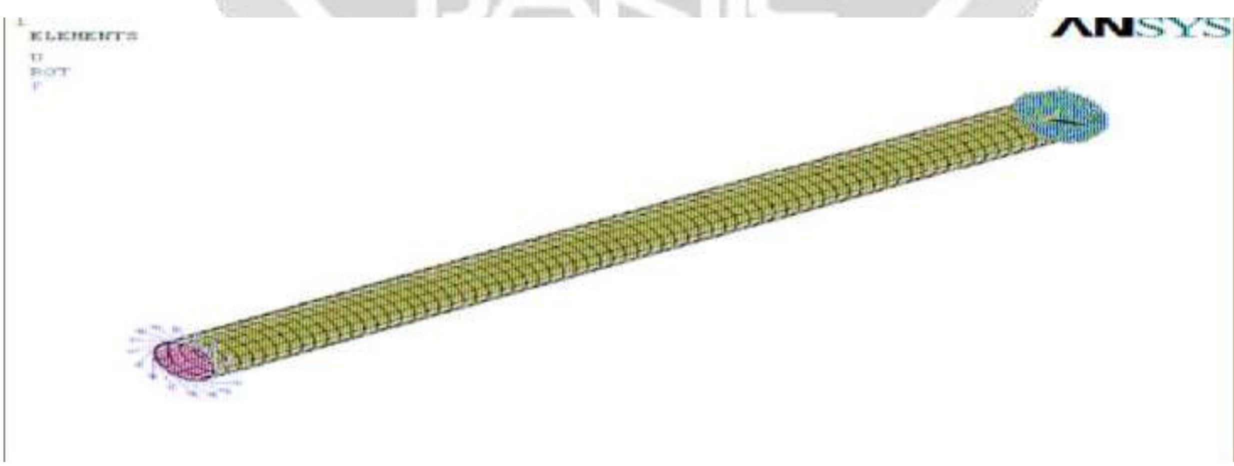
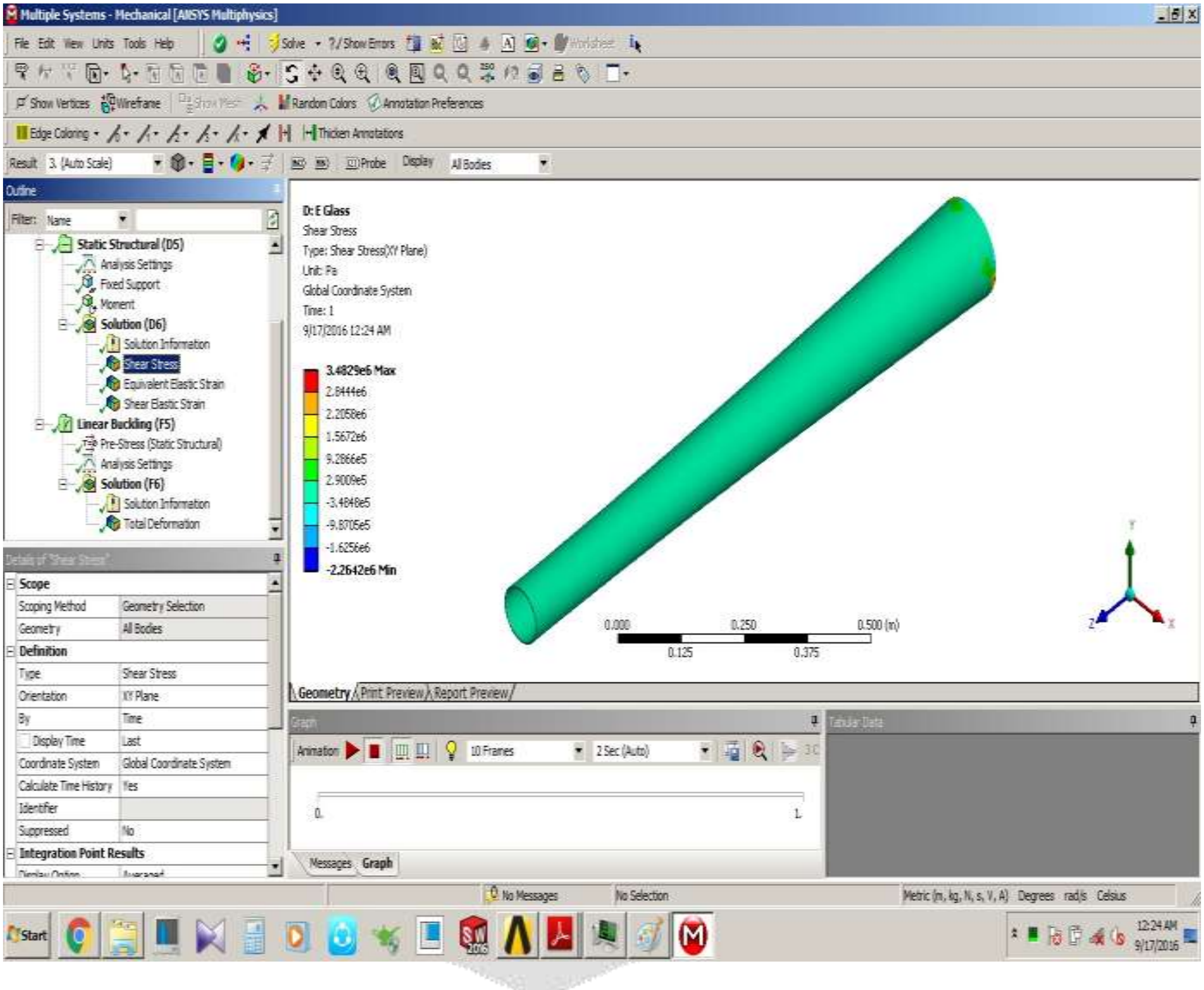


Figure 4.2. Finite element model of HS Carbon/Epoxy shaft

4.5 ANSYS ANALYSIS RESULTS

4.5.1 E-GLASS EPOXY RESULTS

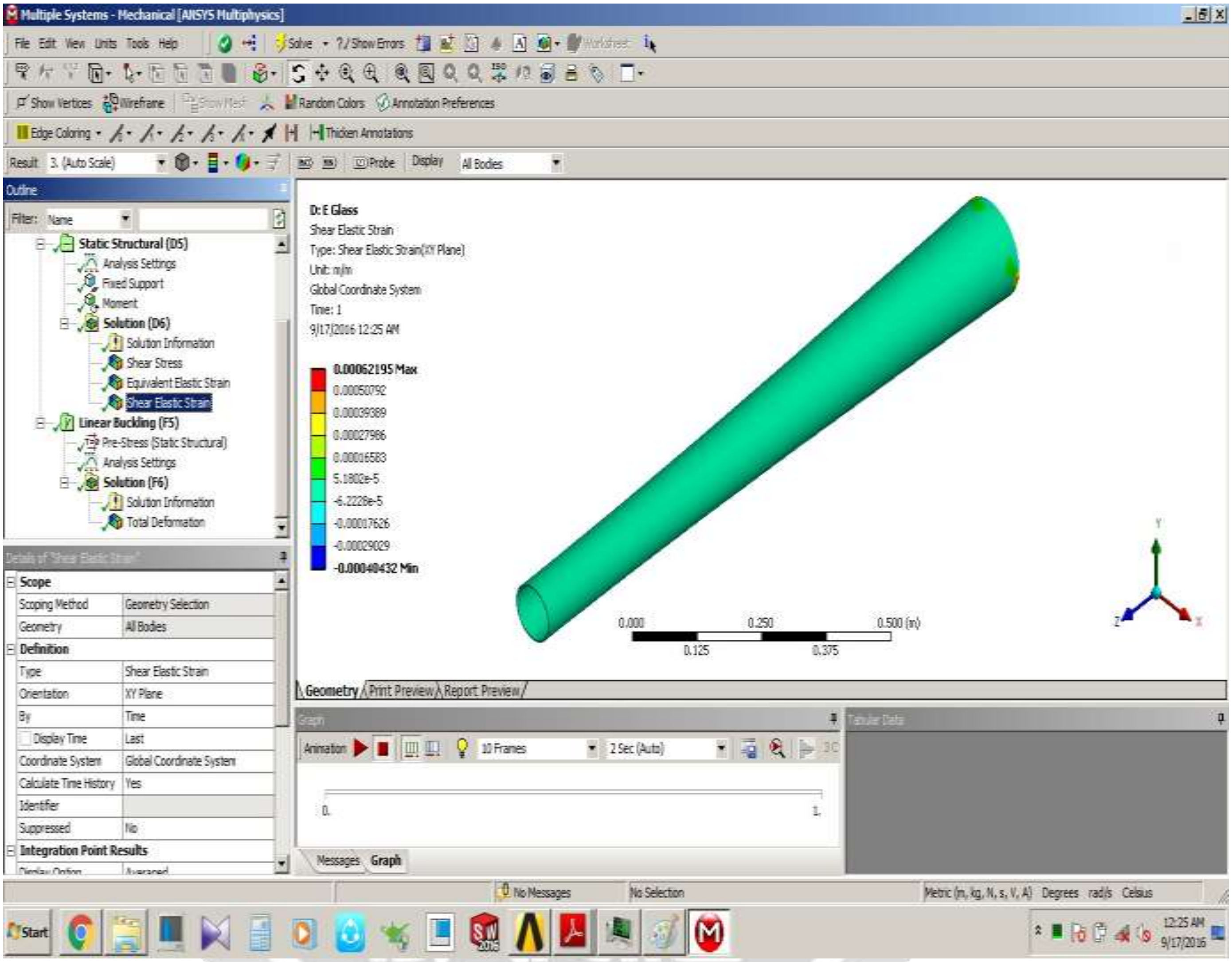
4.5.1.1 Shear stress results



Shear stress for e glass epoxy material is calculated by using ansys 14.5

- Maximum shear stress for e glass epoxy material is 3.4829e6 Pascal’s
- Minimum shear stress for e glass epoxy material is -2.2642e6 Pascal’s
- And the best value of shear stress for e glass epoxy material is **2.9009e5** Pascal’s

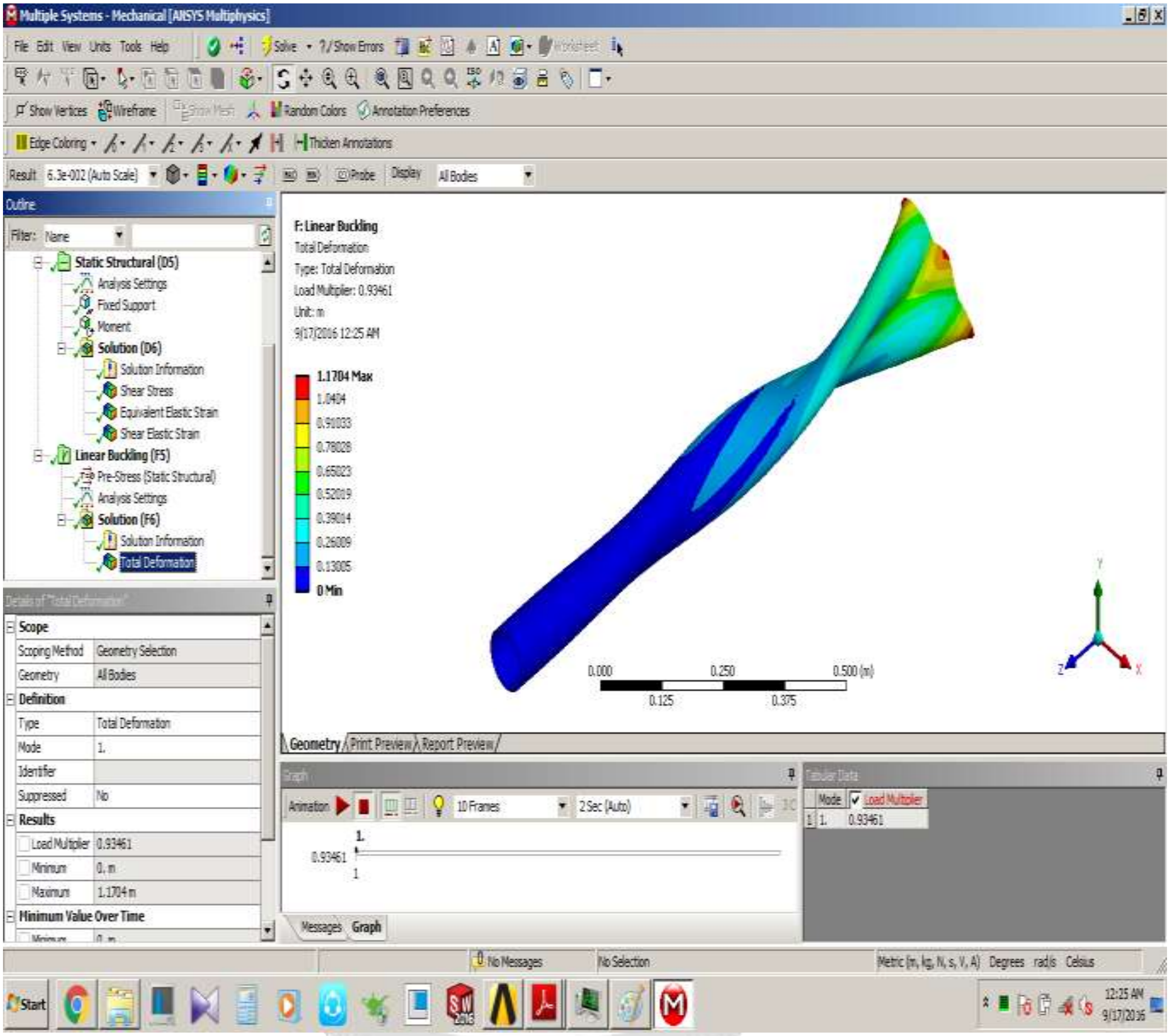
4.5.1.2 Shear strain result



Shear strain for e glass epoxy material is calculated by using ansys 14.5

- Maximum shear strain for e glass epoxy material is 0.0006219
- Minimum shear strain for e glass epoxy material is 0.000404
- And the best value of shear strain for e glass epoxy material is **0.000165**

4.5.1.3 Buckling results

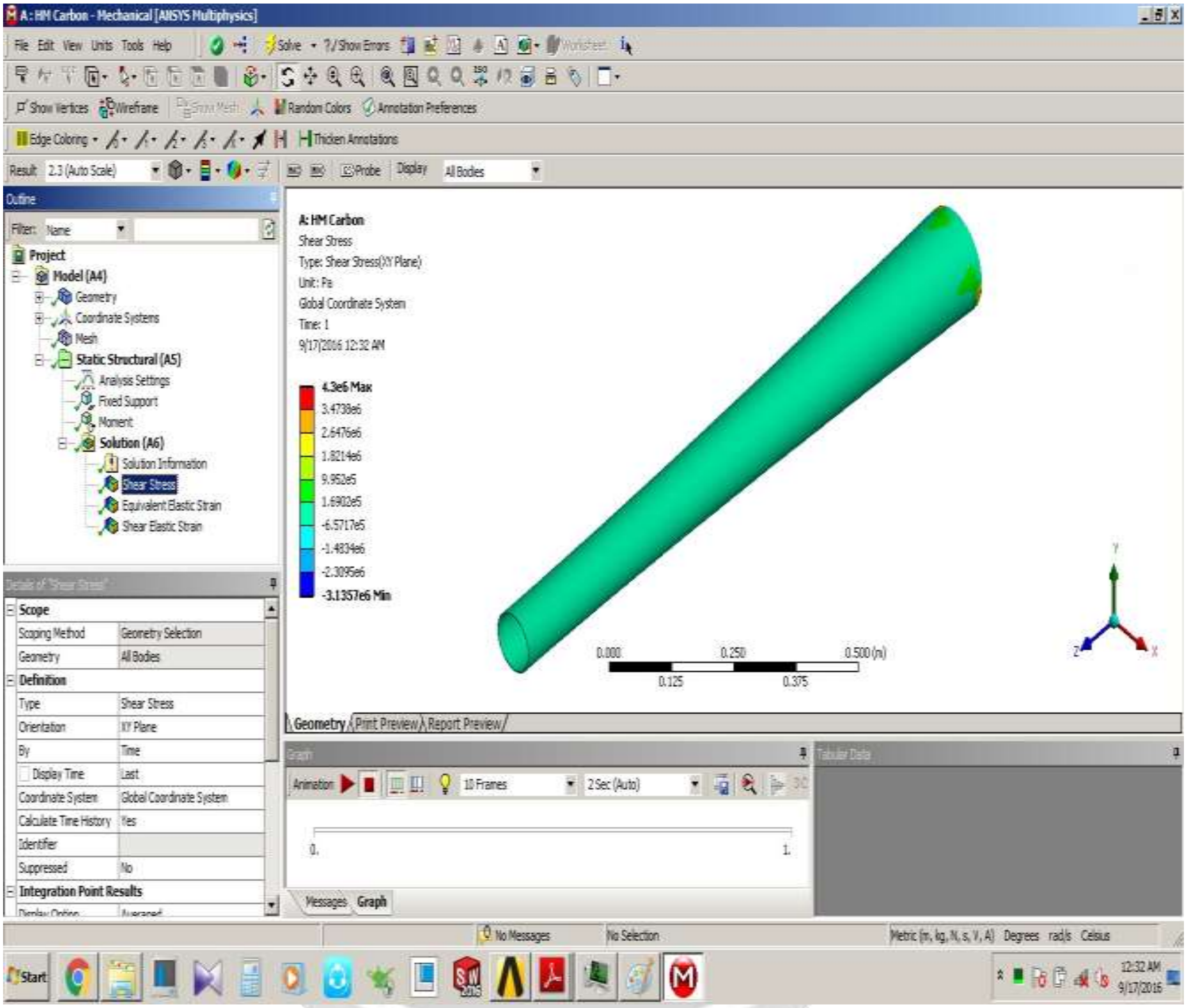


Buckling deformation for e glass epoxy material is calculated by using ansys 14.5

- Maximum Buckling deformation for e glass epoxy material is 1.170 m
- Minimum Buckling deformation for e glass epoxy material is 0 m
- And the best value of Buckling deformation for eglass epoxy material is **0.52019 m**

4.5.2 HM-CARBON RESULTS

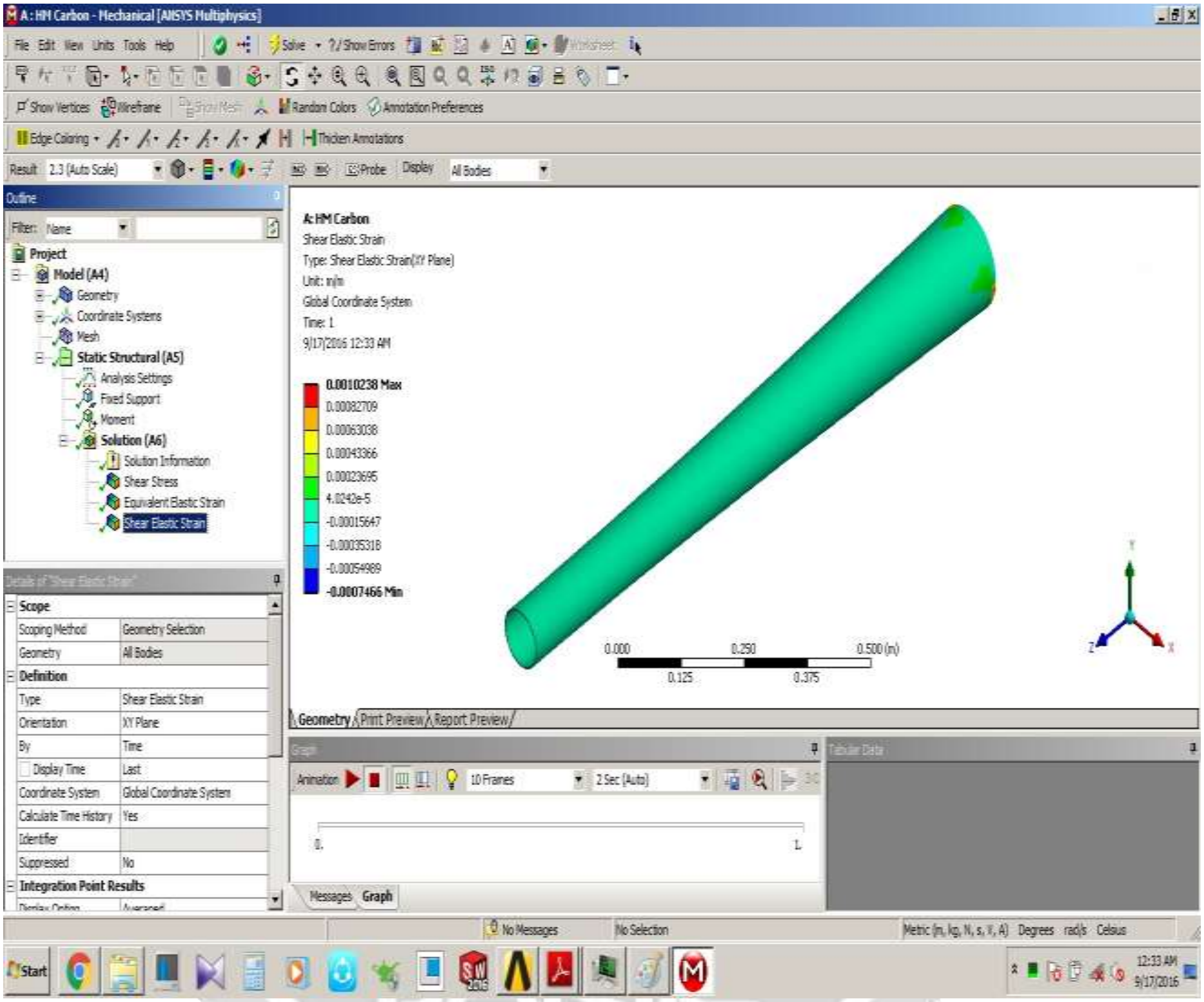
4.5.2 .1 Shear stress results



Shear stress for HM-carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum shear stress for HM-carbon/Epoxy material is 4.3e6 Pascal's
- Minimum shear stress for HM-carbon/Epoxy material is -3.1357e6 Pascal's
- The best value of shear stress for HM-carbon/Epoxy material is **1.6902e5** Pascal's

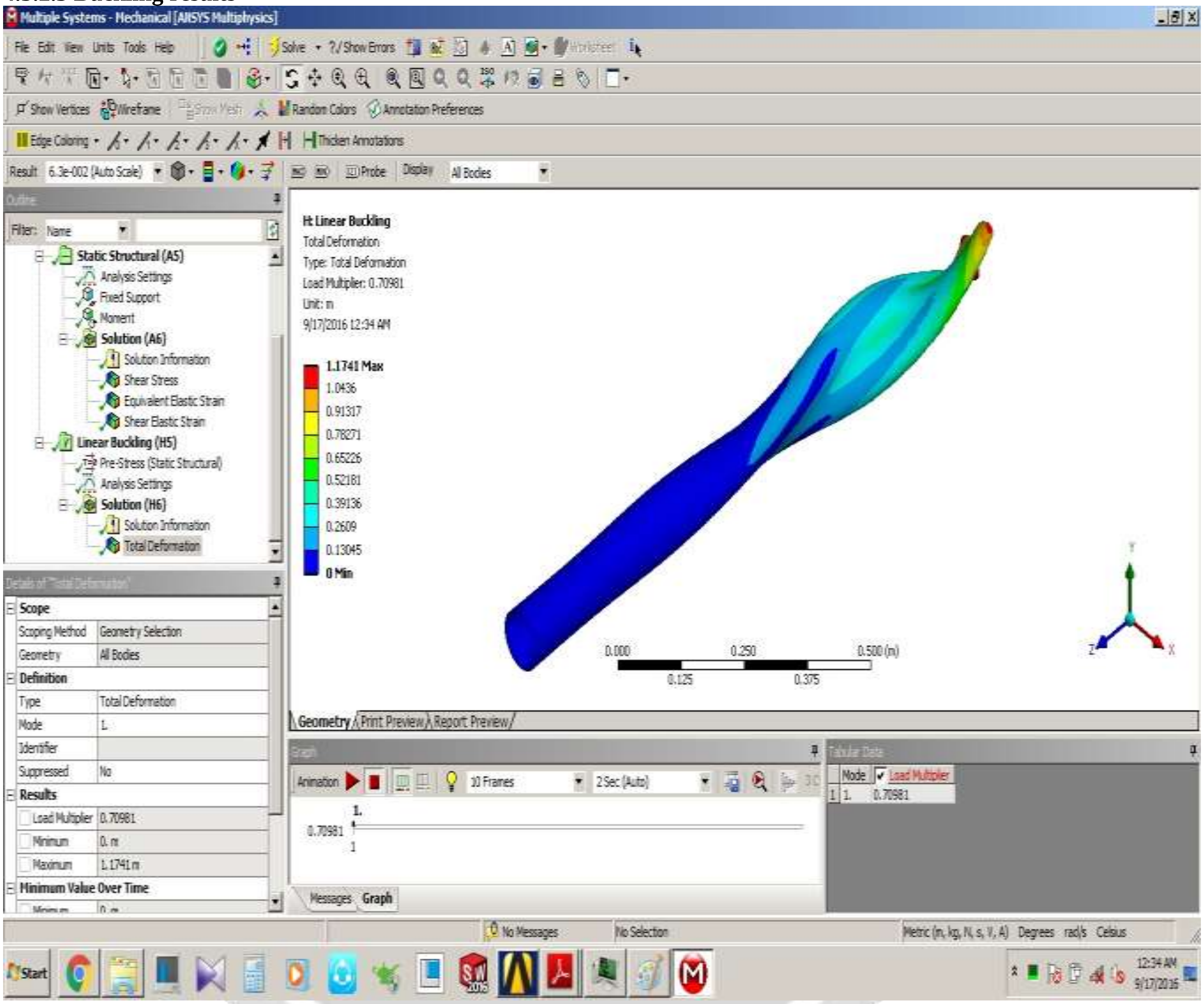
4.5.2.2 Shear strain result



Shear strain for HM-carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum shear strain for HM-carbon/Epoxy material is 0.0010238
- Minimum shear strain for HM-carbon/Epoxy material is 0.000404
- The best value of shear strain for HM-carbon/Epoxy material is **0.0002365**

4.5.2.3 Buckling results

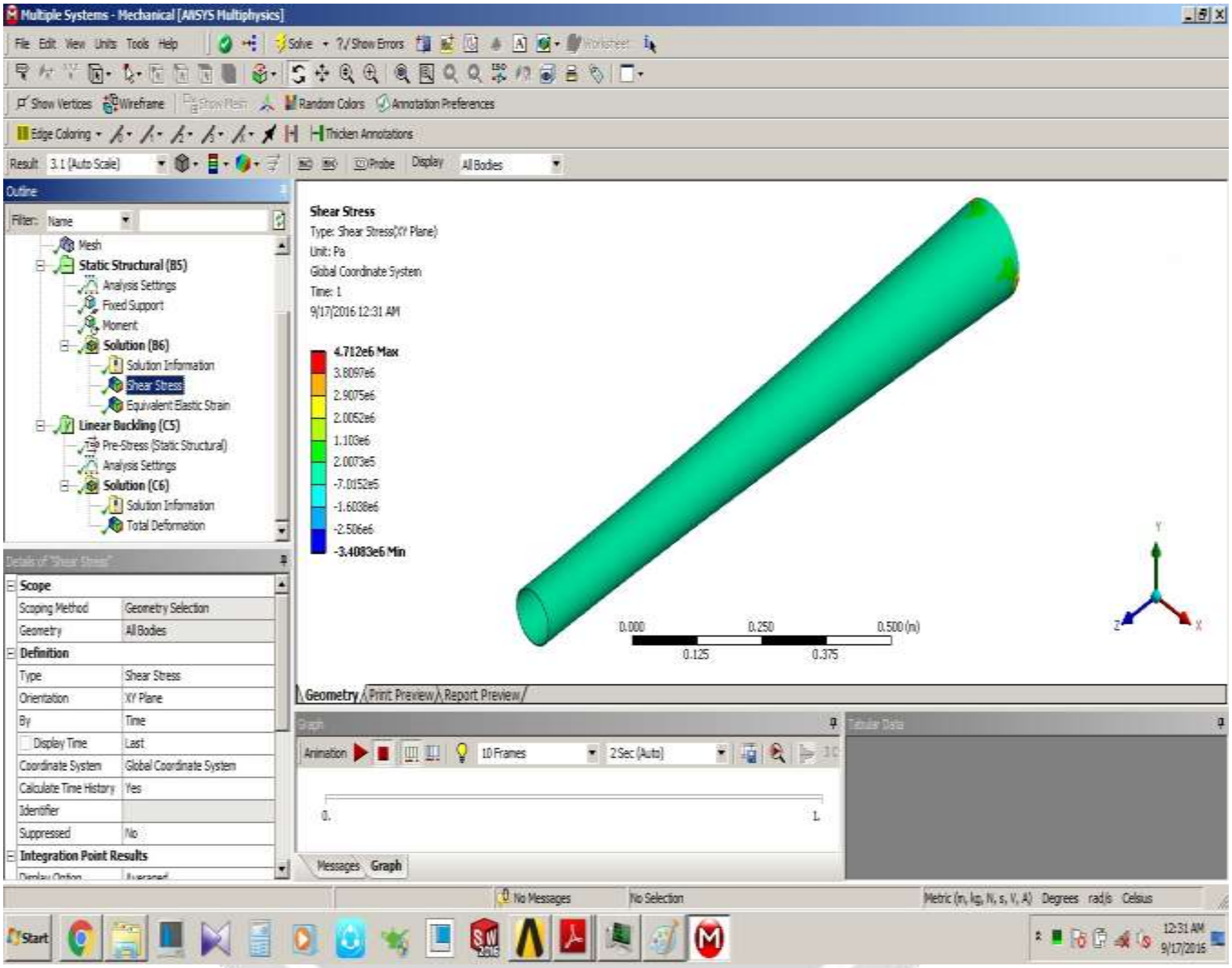


Buckling deformation for HM-carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum Buckling deformation for HM-carbon/Epoxy material is 1.1741 m
- Minimum Buckling deformation for HM-carbon/Epoxy material is 0 m
- The best value of Buckling deformation for HM-carbon/Epoxy material is **0.52181** m

4.5.3 HS –CARBON RESULTS

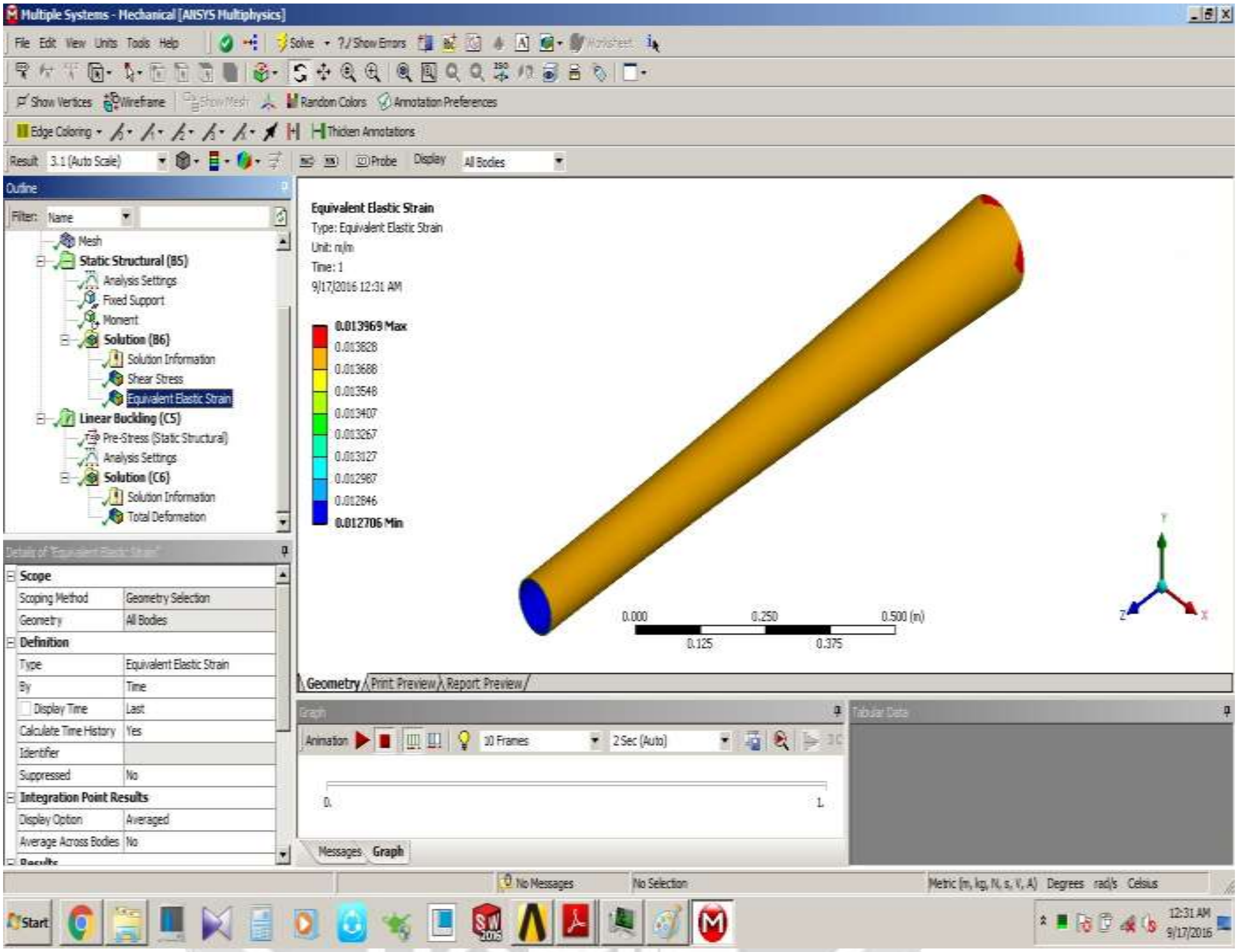
4.5.3.1 Shear stress results



Shear stress for HS –carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum shear stress for HS –carbon/Epoxy material is 4.712e6 Pascal’s
- Minimum shear stress for HS –carbon/Epoxy material is -3.4083e6 Pascal’s
- The best value of shear stress for HS –carbon/Epoxy material is **2.0073e5** Pascal’s

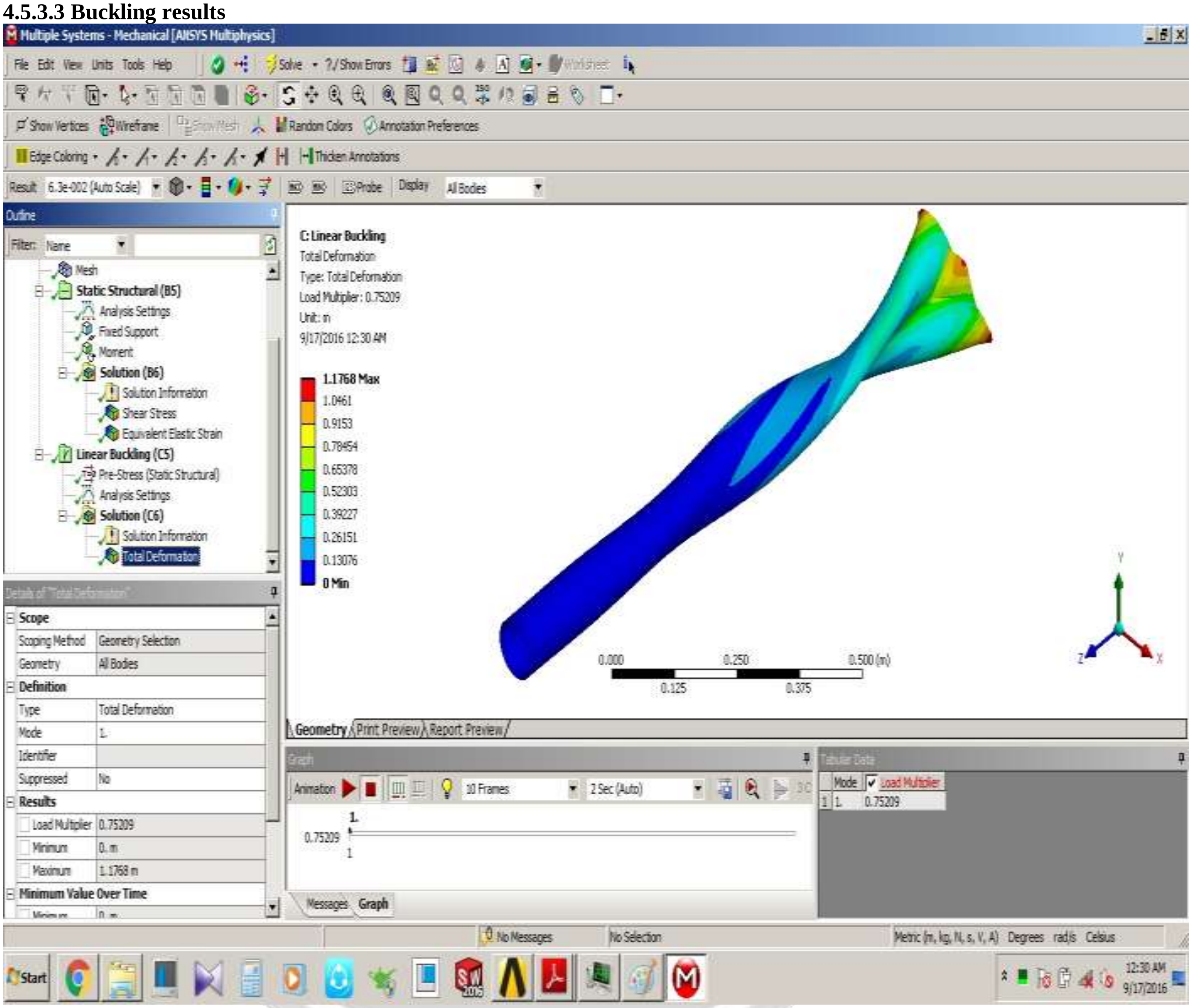
4.5.3.2 Shear strain result



Shear strain for HS –carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum shear strain for HS –carbon/Epoxy material is 0.01396
- Minimum shear strain for HS –carbon/Epoxy material is 0.12706
- And the best value of shear strain for HS –carbon/Epoxy material is **0.013267**

4.5.3.3 Buckling results



Buckling deformation for HS –carbon/Epoxy material is calculated by using ansys 14.5

- Maximum Buckling deformation for HS –carbon/Epoxy material is 1.176 m
- Minimum Buckling deformation for HS –carbon/Epoxy material is 0 m
- The best value of Buckling deformation for HS –carbon/Epoxy material is **0.5303**m

5. RESULTS AND DISCUSSIONS

5.1 RESULTS

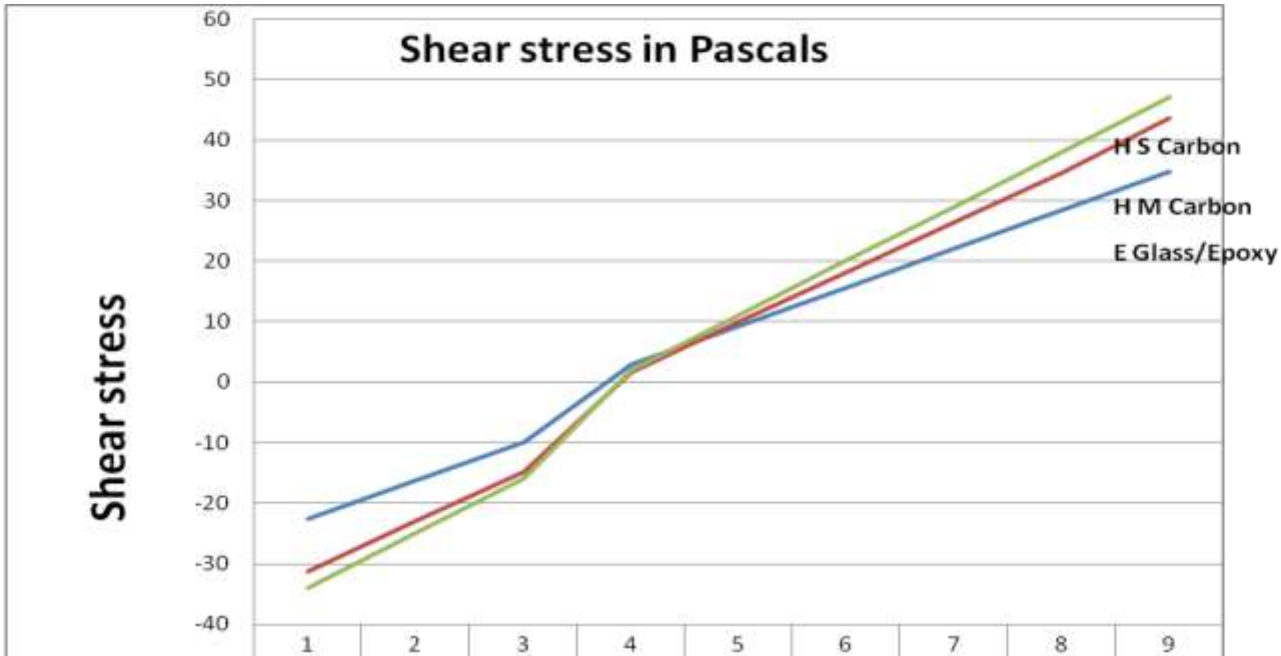
A one-piece composite drive shaft for rear wheel drive automobile was designed optimally by using genetic Algorithm for E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composites with the objective of minimization of weight of the shaft which is subjected to the constraints such as torque transmission.

5.1.1 Optimization

Parameters	Steel	E-Glass/Epoxy	HS Carbon/Epoxy	HM Carbon/Epoxy
d_o (mm)	90	90	90	90
L (mm)	1250	1250	1250	1250
t_k (mm)	3.318	0.4	0.12	0.12
Optimum no. of Layers	1	17	17	17
t (mm)	3.318	6.8	2.04	2.04
Stacking sequence	-	[46/-64/-15/-13/ _ 39/-84/-28/20/- 27]s	[-56/-51/74/-82/ _ 67/70/13/-44/- 75]S	[-65/25/68-63/ _ 36/-40/-39/74/- 39]S
Weight (kg)	8.604	4.443	1.1273	1.1274
Weight saving (%)	-	48.36	86.90	86.90

Table No 7.1

5.2 Shear stress



	1	2	3	4	5	6	7	8	9
E Glass/Epoxy	-22.642	-16.256	-9.8705	2.9009	9.2866	15.672	22.058	28.444	34.829
H M Carbon	-31.357	-23.095	-14.834	1.6902	9.952	18.214	26.476	34.738	43.738
H S Carbon	-34.083	-25.06	-16.038	2.0073	11.03	20.052	29.075	38.097	47.12

Shear stress for e glass epoxy material is calculated by using ansys 14.5

- Maximum shear stress for e glass epoxy material is 34.829e5 Pascal’s
- Minimum shear stress for e glass epoxy material is -22.642e5 Pascal’s
- And the Best value of shear stress for e glass epoxy material is **2.9009e5** Pascal’s

Shear stress for HM-carbon material is calculated by using ansys 14.5

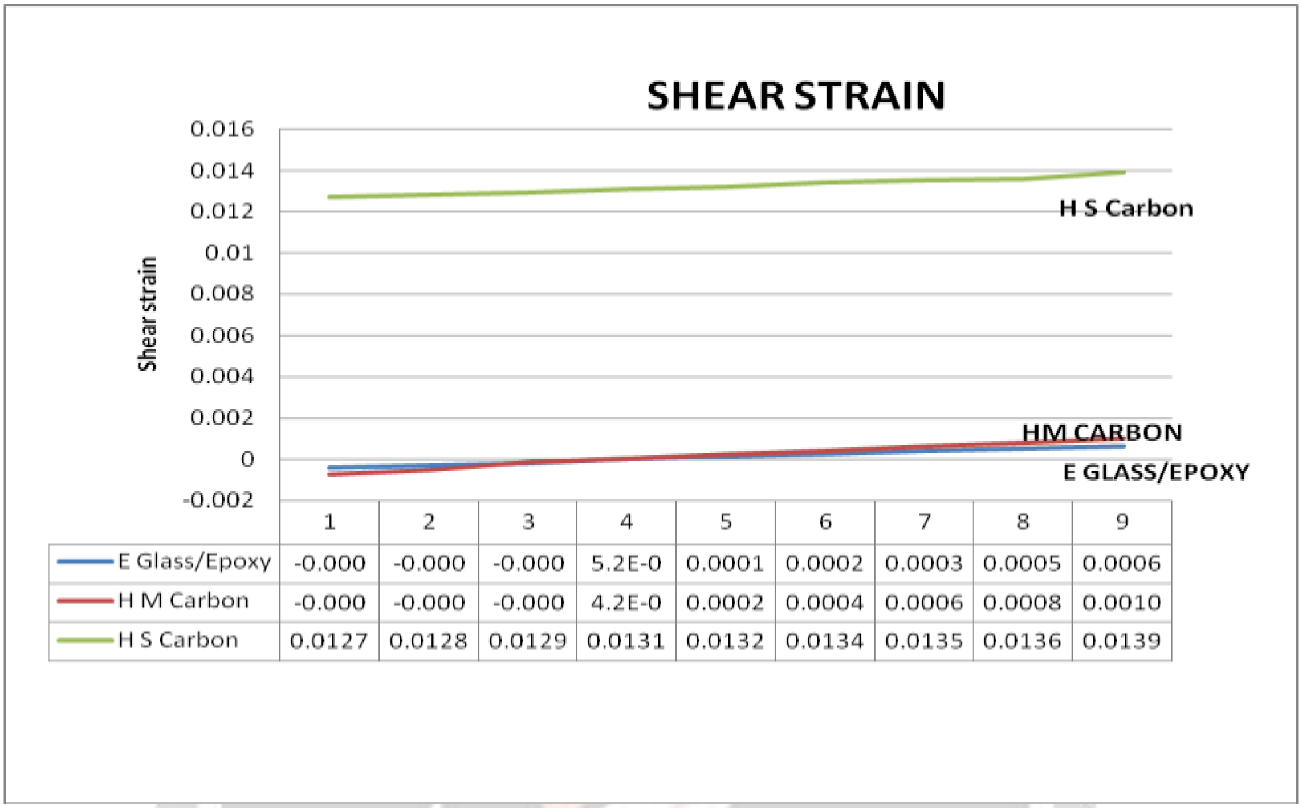
- Maximum shear stress for HM-carbon/Epoxy material is 43.e5 Pascal’s
- Minimum shear stress for HM-carbon/Epoxy material is -31.357e5 Pascal’s
- The Best value of shear stress for HM-carbon/Epoxy material is **1.6902e5** Pascal’s

Shear stress for HS –carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum shear stress for HS –carbon/Epoxy material is 47.12e5 Pascal’s
- Minimum shear stress for HS –carbon/Epoxy material is -34.083e6 Pascal’s
- The Best value of shear stress for HS –carbon/Epoxy material is **2.0073e5** Pascal’s

Finally between these three composite materials **HM-Carbon**/Epoxy material having less shear stress(**1.6902e5 Pascal’s**) comparing with e glass epoxy and HS –carbon/Epoxy materials.

5.3 shear strain



Shear strain for e glass epoxy material is calculated by using ansys 14.5

- Maximum shear strain for e glass epoxy material is 0.0006219
- Minimum shear strain for e glass epoxy material is 0.000404
- And the Best value of shear strain for e glass epoxy material is **0.000165**

Shear strain for HM-carbon/Epoxy **material is calculated by using ansys 14.5**

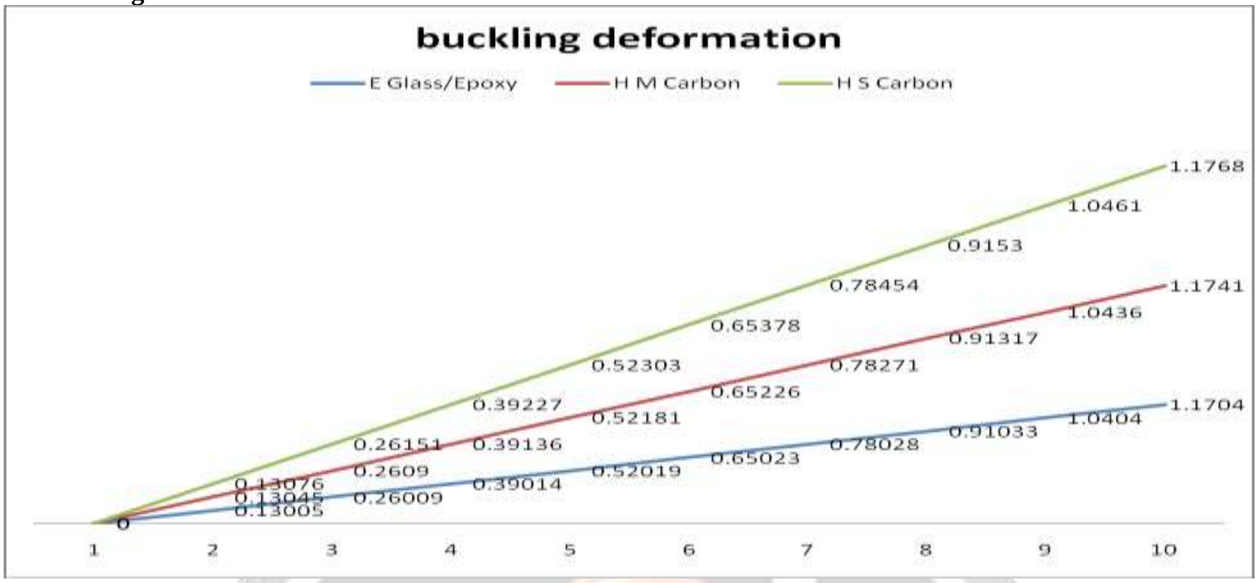
- Maximum shear strain for HM-carbon/Epoxy material is 0.0010238
- Minimum shear strain for HM-carbon/Epoxy material is 0.000404
- And the Best value of shear strain for HM-carbon/Epoxy material is 0.0002365

Shear strain for HS –carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum shear strain for HS –carbon/Epoxy material is 0.01396
- Minimum shear strain for HS –carbon/Epoxy material is 0.12706
- And the Best value of shear strain for HS –carbon/Epoxy material is 0.013267

Finally between these three composite materials **e glass epoxy** material having less shear strain (**0.000165**) comparing with HM-carbon/Epoxy and HS –carbon/Epoxy materials.

5.4 Buckling deformation



Buckling deformation for e glass epoxy material is calculated by using ansys 14.5

- Maximum Buckling deformation for e glass epoxy material is 1.170 m
- Minimum Buckling deformation for e glass epoxy material is 0 m
- The Best value of Buckling deformation for e glass epoxy material is 0.52019 m

Buckling deformation for HM-carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum Buckling deformation for HM-carbon/Epoxy material is 1.1741 m
- Minimum Buckling deformation for HM-carbon/Epoxy material is 0 m
- Best value of Buckling deformation for HM-carbon/Epoxy material is 0.52181 m

Buckling deformation for HS –carbon/Epoxy **material is calculated by using ansys 14.5**

- Maximum Buckling deformation for HS –carbon/Epoxy material is 1.176 m
- Minimum Buckling deformation for HS –carbon/Epoxy material is 0 m
- Average value of Buckling deformation for HS –carbon/Epoxy material is 0.5303m.

Finally between these three composite materials **E Glass/Epoxy** material having less Buckling deformation (**0.52019** m) comparing with HM-carbon/Epoxy and HS–carbon/Epoxy materials.

Material	Torque transmission (Nm)	Deformation(m)	Shearb stress (Pascal's)	Mass of drive shafts (kg)
Steel	2464.2	1.2407	4.34E+05	8.604
E-Glass/Epoxy	15439.5	0.52019	2.90E+05	4.443
HS Carbon/Epoxy	5519.04	0.5303	2.01E+05	1.1274
HM Carbon/Epoxy	7525.9	0.52181	1.69E+05	**1.1273**

6. CONCLUSIONS

The stress distribution and the maximum deformation in the shaft are the functions of the stacking of material. The optimum stacking of material layers can be used as the effective tool to reduce weight and stress acting on the drive shaft. The design of drive shaft is critical as it is subjected to combined loads. The designer has two options for designing the drive shaft whether to select solid or hollow shaft. The solid shaft gives a maximum value of torque transmission but at same time due to increase in weight of shaft the 1st mode frequency decreases. Also shaft outer surface facing most of the stress coming on to it and the inner material layer experienced less stress, hence the inner layers increasing the weight of shaft and not utilized for stress distribution properly, that's why the hollow drive shaft is best option.

The following conclusions are drawn from the present work.

1. The E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composite drive shafts have been designed to replace the steel drive shaft of an automobile.
2. A one-piece composite drive shaft for rear wheel drive automobile has been designed optimally by using Genetic Algorithm for E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy composites with the objective of minimization of weight of the shaft which was subjected to the constraints such as torque transmission, torsional buckling capacities.
3. The weight savings of the E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy shafts were equal to 48.36%, 86.90% and 86.90% of the weight of steel shaft respectively.
4. The Torque transmission of E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy shafts were equal to **15439.5,** 7525.9 and 5519.04 Nm respectively.
5. The Buckling Deformation of E-Glass/ Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy shafts were equal to **0.52019,** 0.5303 and **0.52181** m respectively.
6. The shear stress of E-Glass/Epoxy, High Strength Carbon/Epoxy and High Modulus Carbon/Epoxy shafts were equal to 2.9009e5**,** 2.0073e5 and **1.6902e5** Pascal's respectively

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