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RESEARCH ARTICLE OPEN ACCESS

Design Optimisation and Finite Element Analysis of Composite Drive Shaft

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Abstract:

The drive shaft transfers motion from the engine to the differential in automobiles. The power of rear-wheel drive is sent to the propeller shaft of a vehicle is also known as the drive shaft, and it connects the engine to the transmission and differential gear of the vehicle. Power transmission systems in vehicles are made up of many components that can fail at any time. Some of the most common reasons of failure are manufacturing faults, design flaws, maintenance flaws, raw material flaws, material processing flaws, and user-caused problems. The fracture of a universal joint yoke and a drive shaft from a vehicle power transmission system is the subject of this research. An attempt was made in this study to design an automobile driveshaft based on maximum torque, transmission capacity, and shear stress, material optimization with different materials such as steel alloy and composite materials such as carbon fiber reinforced plastic, life assessment, and weight reduction, then to test product performance, a virtual simulation utilising finite element analysis software is used.

Keywords **—Automobiles, Differential, Power Transmission, Fracture, Torque, Shear stress, Optimization, Simulation.**

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I. INTRODUCTION

A. *DRIVE SHAFT*

A driveshaft, also known as a driving shaft, is a mechanism that delivers engine power to the position where work is done. In automobiles, the drive shaft delivers engine torque to the drive axle, which unites the two wheels on opposite sides and rotates with them. The propeller shaft is another name for the driveshaft. Drive shafts are primarily torque carriers. drive shafts come in a variety of lengths based on their intended function. Shorter shafts are used when power must be sent from a central differential, transmission, or transaxle in front-engine, rear-drive cars, whereas longer shafts are used when power must be sent from a central differential, transmission, or transaxle in front-engine, rear-drive vehicles.

B. *TYPES OF DRIVE SHAFTS*

There are different types of drive shafts in Automobile Industry,

1. One-piece driveshaft.
2. Two-piece driveshaft.
3. Slip in Tube drive.

Drive shafts are utilised in a variety of industries as power transmission lines, including cooling towers, pumping sets, aircraft, trucks, and vehicles. The size of the shaft's cross section can be estimated in metallic shaft design by knowing the torque and the material's permissible shear stress.

The weight, low critical speed, and vibration characteristics of metallic drive shafts are seeming to be major drawbacks. When a steel drive shaft's length exceeds 2000 mm, it is split into two sections to enhance the fundamental natural frequency, which is inversely proportional to the square length and proportional to the square root of the particular frequency.A rear-wheel-drive vehicle's drive shaft delivers power from the engine to the differential gear. To eliminate whirling vibration, the drive shaft torque capability for HDT should be greater than 8000 Nm, and the basic bending natural frequency should be greater than 4000 rpm. Because the fundamental bending natural frequency of a one-piece drive shaft made of steel or aluminium is normally lower than 3800 rpm when the drive shaft length is around 2 m, the steel drive shaft is usually manufactured in two pieces to increase the fundamental bending natural frequency.

II. LITERATURE SURVEY

ANUPAM SINGHAL, R. K. MANDLOI [1]: "Failure Analysis of Automotive FWD Flexible Drive Shaft" has been published. According to this paper Drive shafts, are torque carriers. They are subjected to torsion and shear stress, which is proportional to the difference in input torque and load. They must therefore be strong enough to withstand the strain while avoiding adding too much weight, which would increase their inertia.

D.DINESH, F. ANAND RAJU [2]: "Optimum Design and Analysis of a Composite Drive Shaft for an Automobile Using Genetic Algorithm and Ansys" is the title of a paper produced by D. DINESH and F. ANAND RAJU. Because composite materials have higher particular stiffness and strength, substituting composite structures for conventional metallic structures has various advantages.

BIMLESH KUMAR SINHA and BHTRUD PANKAJ PRAKASH [3]: have published a journal titled "ANALYSIS OF DRIVE SHAFT." This document comprises the following: Composite materials can be designed to satisfy specific design requirements for strength and stiffness, and composite drive shafts are lighter than steel or aluminium drive shafts of comparable strength.

V. S. BHAJANTRI, S. C. BAJANTRI, A. M. SHINDOLKAR, and S. S. AMARAPURE[4]: "DESIGN AND ANALYSIS OF COMPOSITE DRIVE SHAFT." according to this article. Composite materials have been used extensively to improve the performance of a variety of structures. Composites provide a superior stiffness to mass ratio as well as a high strength to weight ratio when compared to traditional materials.

III. DEFINITION OF THE PROBLEM

To eliminate whirling vibration, the drive shaft's torque transmission capability should be more than 7000 Nm (Tmax) and its basic natural bending frequency should be greater than 4000 rpm (Nmax). The outside diameter of the shaft is 115 mm, and the length L of the driving shaft is 2000 mm. The parameters of an automotive transmission's drive shaft are the same as those of a steel drive shaft for optimal design.

A. Objective

1. The one-piece drive shaft was designed using CATIA software.
2. Ansys software was used to import the designed one-piece drive shaft.
3. Ansys software was used to do a finite element analysis on a one-piece drive shaft.
4. For torsional loads, the performance of the drive shaft is analysed using Ansys software using standard steel material as well as composite E-glass/Epoxy and Carbon/Epoxy materials.
5. A static, model study of the drive shaft is performed using Ansys software for several materials (steel, E-glass/Epoxy, and HM Carbon/Epoxy) to compute weight, deflections, and stresses.
6. The analytical results are compared, and the optimum material is recommended based on the weight to strength ratio.

B. Approach

1. FE approach for study the stress / strain and deformations
2. Using analytical equations, mathematical modelling is developed to verify the stresses and deformation
3. Linear static analysis and bilinear analysis of drive shaft
4. Dynamic analysis, modal and harmonic analysis of drive shaft
5. High fatigue life evaluation of drive shaft

IV. ANALYTICAL APPROACH

A. Assumptions

1. The shaft revolves along its longitudinal axis at a constant speed.
2. The cross section of the shaft is uniform and round.
3. The shaft is completely balanced, which means that the mass centre corresponds with the geometric centre at every cross section.
4. No damping or nonlinear effects are allowed.
5. Because the stress-strain relationship for steel is linear and elastic, Hooke's law applies to the material.
6. Acoustical fluid interactions are ignored, implying that the shaft operates in a vacuum.

B. Cross Section Of Drive shaft

The hollow round cross-section was chosen in this case becauseHollow circular shafts are stronger per kilogram weight than solid circular shafts and in the case of a solid shaft, the stress distribution is zero in the centre and maximal at the outside surface, but the stress variation is smaller in the case of a hollow shaft.

C. Design Requirements and Specification

Steel (SM45C) is now utilised to manufacture vehicle drive shafts. The steel (SM45C) material qualities are listed in the table below.

TABLE I
CRITICAL VALUES CONSIDERED FOR THE MATERIAL

SL NO	Nomenclature	Notation	Unit	Value
1	Maximum torque	Tmax	Nm	7500
2	Max speed of the shaft	Nmax	Rpm	3000
3	Length of the shaft	L	mm	2000
4	Dia of the shaft	D	mm	120

TABLE II
MECHANICAL PROPERTIES OF THE STEEL(SM45C)

Mechanical properties	Symbol	Units	Steel
Young's Modulus	E	MPa	210000
Shear Modulus	G	MPa	80000
Poisson's Ratio	υ	0.3	Poisson's Ratio
Density	ρ	Kg/m^3	7600
Yield Strength	Sy	N/mm^2	550
Shear Strength	Ss	N/mm^2	710

D. Capacity for Torque Transmission

Starting Torque(Nm) = Gear Ratio x Max.engine torque

This Max.Torque is used to calculate the maximum shear stress, which should be less than the material shear strength for material safety.

Moment Of Inertia, $I=(\pi/64)\,(Do^4-Di^4)$ mm^4

Polar Moment Of Inertia, $J=(\pi/32)\,(Do^4-Di^4)$ mm^4

Section Modulus, $Z=(\pi 16)\,(Do4-Di4Do4)$ mm4

Torsional Shear Stress $T_{Shear}=Starting\ Torque\ \times D2J$ $Nmm2$

E. Critical Speed Calculation

Critical speed $N_{cr}=(30/\pi)\sqrt{(g/\delta_{max})}\ rpm$

Where,
Max Deflection, $\delta_{max}=(5Mg(\cos\theta)L^3 / 384\times E\times J)\,mm$

by Rayleigh reitz method.

M = mass of the tube.

V. FINITE ELEMENT APPROACH

Finite element Method (FEM) is a numerical approach for obtaining evaluated responses for point of confinement respect difficulties for divided differential conditions based on the discretisation of section into FEA (elements).

A. *Procedure Steps In FEM*

- Modelling
- Continuum description
- Model selection
- Derivation of element stiffness matrix
- Assemblage of element equations to obtain equilibrium equations
- Enforcing the boundary condition
- Solution of system equation to find nodal values of displacement
- Computation of element strains and stresses

B. *Modal Analysis*

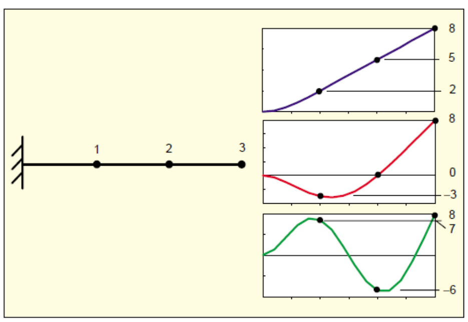

Fig. 1 Modular Analysis

An analysis of measured information is a method that breaks down the observed recurrence response functions to find a theoretical model that most closely fits the dynamic behavior of the structure under test.

VI. VIEW OF DRIVE SHAFT

A. *3D View of Drive Shaft*

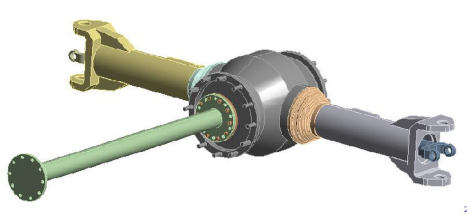

Fig. 2 3D View of Drive Shaft

B. *Isometric View of Drive Shaft*

Fig. 3 Isometric View of Drive Shaft

C. *Meshing 3D Model*

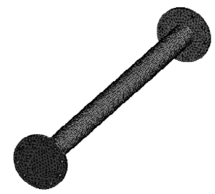

Fig. 4 Meshing of Drive Shaft in 3D

The hex dominant approach was used to mesh the 3D model of the flat edge specimen, using 22000 element numbers and 31540 nodes.

D. *Applied Boundary Conditions*

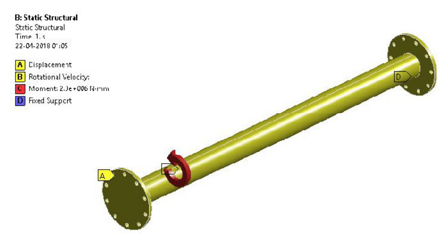

Fig. 5 Static Structural Analysis Applying Boundary Conditions

Point D of the part is referred as flange and is constraint shown in fig 6.4 and a moment of 2.3e006 N/mm is applied at the point C i.e., shaft the displacement of the entire section due to the moment is measure at the point A.

E. *Equivalent Stress Analysis*

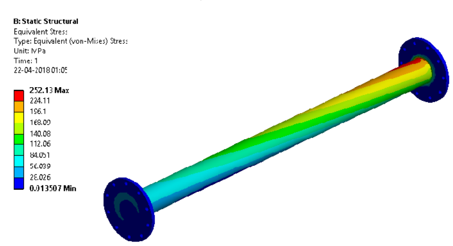

Fig. 6 The Von Misses Yield Criterion

An equivalent stress analysis shows 252.13Mpa is observed at the section D which is represented colour 'red' and minimum of 0.013 Mpa is developed at the section A

F. Deformation In The Shaft

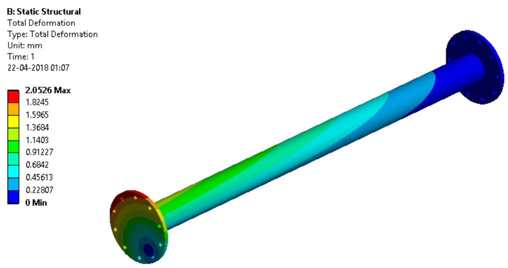

Fig. 7 Deformation in the shaft

When a moment of 2.3e006 is applied at the point B or free end of the total deformation of 2.05mm is observed at the free end away from the fixed support.

G. Total Weight Of The Material Steel

TABLE III
TOTAL WEIGHT OF THE MATERIAL

Details of "Front Transfer Rod"

Properties	
Volume	4.6707e+006 mm³
Mass	12.938 kg
Centroid X	-2643.5 mm
Centroid Y	-381.54 mm
Centroid Z	-1007.1 mm
Moment of Inertia Ip1	11747 kg·mm²
Moment of Inertia Ip2	1.5418e+006 kg·mm²
Moment of Inertia Ip3	1.5418e+006 kg·mm²

The table shows the overall mass of the steel member is 12.938 kgs, volume of 4.6707e+006 of the material centroid about X, Y, Z. Axial moment of inertia along XYZ direction.

H. Load Applied And Meshed

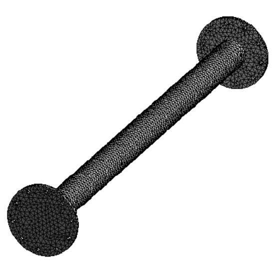

Fig. 8 Meshing of The Steel Drive Shaft

I. Equivalent Stress

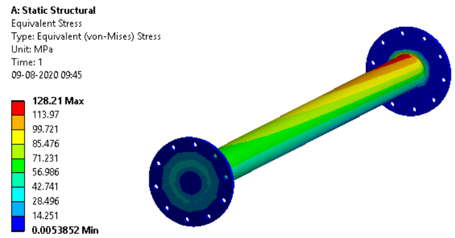

Fig. 9Maximum Equivalent Stress of Drive Shaft.

The maximum equivalent stress developed at point C of about 128.21Mpa indicated with red colour and the minimum equivalent stress developed is 0.0053 Mpa indicated with blue colour.

J. Maximum Principal Stress

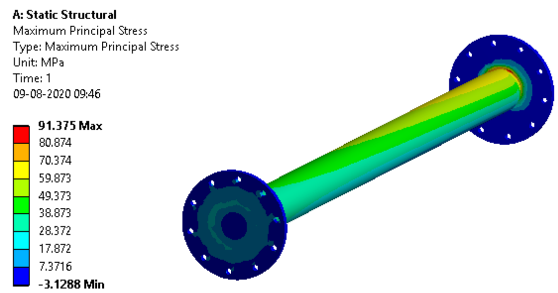

Fig. 10Maximum Principal Stress

The member will under developed will varies from -3.1288Mpa to 91.375 Mpa of maximum principle stress at the point C.

K. Minimum Principal Stress

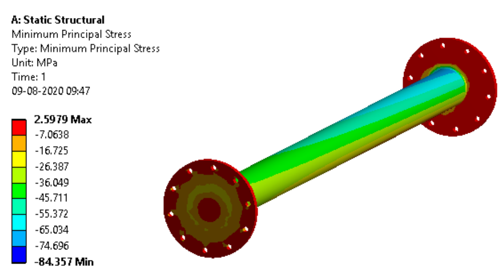

Fig. 11Minimum Principal Stress

The member will develop minimum principle stress at point C locates between the range of -84.357 Mpa to 2.597Mpa.

L. Total Deformation

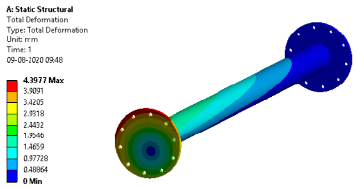

Fig. 12Total Deformation

The maximum equivalent stress established at point C and the total deformation observed is 4.39mm.

M. Total Weigh Of The Material CFRP Shaft

TABLE IV
WEIGHT OF THECFRPMATERIAL

Properties	
Volume	$4.6707e+006\ mm^3$
Mass	7.0061 kg
Centroid X	-2643.5 mm
Centroid Y	-381.54 mm
Centroid Z	-1007.1 mm
Moment of Inertia ...	$6361.\ kg{\cdot}mm^2$
Moment of Inertia ...	$8.349e+005\ kg{\cdot}mm^2$
Moment of Inertia ...	$8.349e+005\ kg{\cdot}mm^2$

The overall mass of the shaft with CFRP is 7.0061 kgs, volume of 4.6707e+006 of the material centroid about X, Y, Z.

N. Equivalent Stress

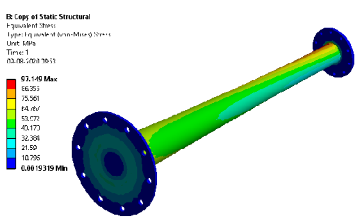

Fig. 13Equivalent Stress of Drive Shaft.

An equivalent stress analysis reveals a maximum of 97.149 Mpa at section D, which is represented by the colour red. And of the minimum value is 0.0019Mpa represents by blue colour.

O. Maximum Principal Stress

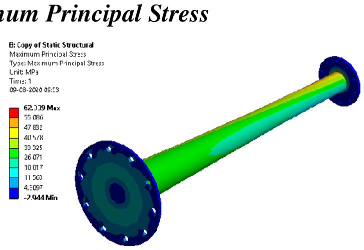

Fig. 14Maximum Principal Stress of Drive Shaft.

When the complete structure is subjected to moment of twist at the point A it is evident that the member will under developed 62.339 Mpa of maximum principle stress at the point C and is graduated from the value of -2.944Mpa.

P. Minimum Principal Stress

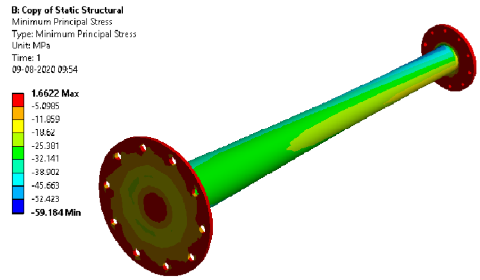

Fig. 15Max Minimum Principal Stress of Drive Shaft.

The member begins to develop 1.662 Mpa incremented from the value of -59.184Mpa. minimum principle stress at point C

Q. Total Deformation

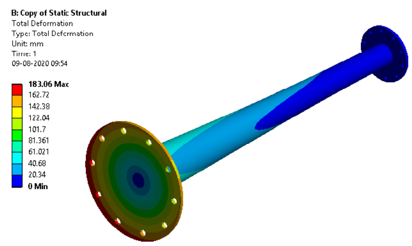

Fig. 16Max Total Deformation is 183mm of Drive Shaft.

VII. MODAL ANALYSIS FOR CFRP DRIVE SHAFT

TABLE V
INITIAL SIX MODES AND CORRESPONDING NATURAL FREQUENCY

Trial no	Modes	Frequency
1.	1.	93.136 Hz
2.	2.	133.51 Hz
3.	3.	138.8 Hz
4.	4.	340.55 Hz.
5.	5.	400.05 Hz
6.	6.	403.22 Hz

When the member is subjected to modular analysis it is evident in the table showing the frequency at different mode the deflection in the member is as shown in the below illustrations.

A. *Illustration 1*

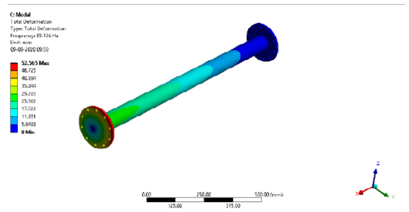

Fig. 17First Mode And Natural Frequency is 93.136 Hz

B. *Illustration 2*

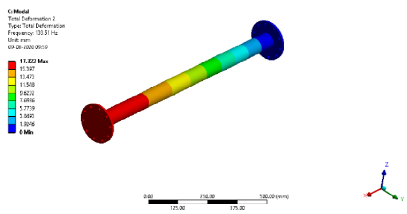

Fig. 18Second Mode And Natural Frequency is 133.51 Hz

C. *Illustration 3*

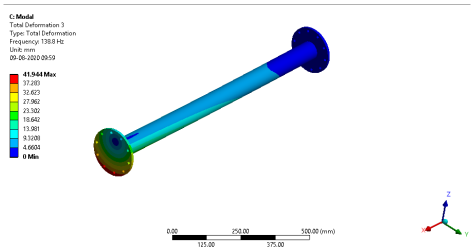

Fig. 19Third Mode And Natural Frequency is 138.8 Hz

D. *Illustration 4*

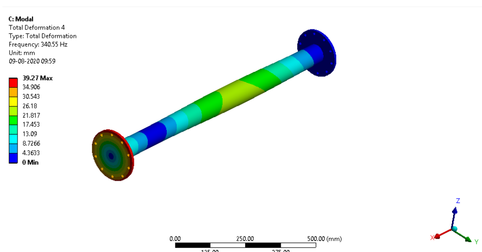

Fig. 20Fourth Mode And Natural Frequency is 340.55 Hz

E. *Illustration 5*

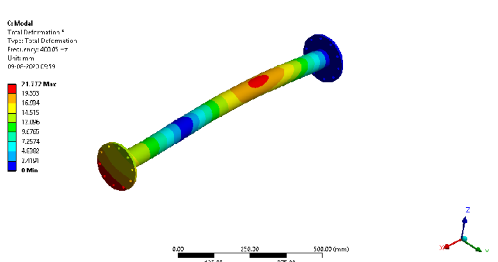

Fig. 21Fifth Mode And Natural Frequency is 400.05 Hz

F. *Illustration 6*

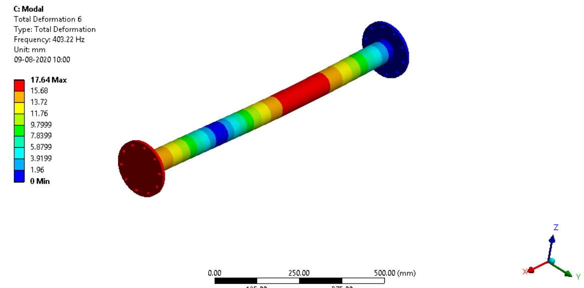

Fig. 22Sixth Mode And Natural Frequency is 403.22 Hz

G. *Validation Of Life Estimation With Analytical And FEM Results*

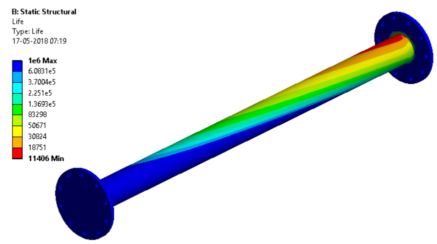

Fig. 23Estimation Life of Drive Shaft

In materials science, fatigue is the wear and tear of a material induced by repeated loads. When a material is subjected to cyclic stress, it suffers from both broad-minded and limited structural damage.The fatigue life, N_f, is defined by the American Society for Testing and Materials as the number of stress cycles of a specific type that a specimen can withstand before failure of a stated sort occurs.Engineers have determined the fatigue life of a material using one of three methods: the stress-life technique, the strain-life approach, or the linear-elastic fracture mechanics method.

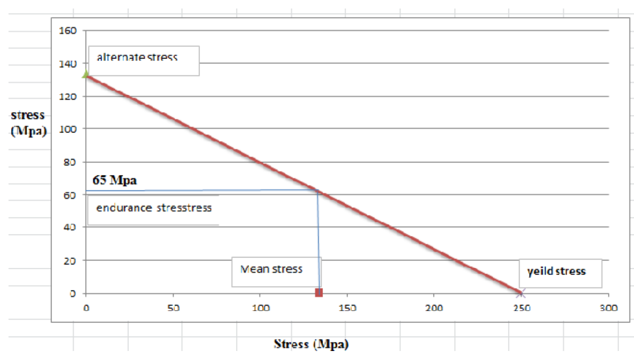

Fig. 24Goodman's Diagram

Mean Stress can be calculated from,

$\sigma_{mean}=\sigma_{von}/2$

$\sigma_{mean}=\sigma_1+\sigma_{22}= 268.31+0.00282=270\ Mpa$,

Alternatestress

$\sigma_{alt}=\sigma_1-\sigma_{22}=268.31-0.00282$
$=134.153 Mpa$

Life calculation

$N_f=\{[\sigma_{ult}-\sigma_{ult}(1/fos-\sigma_e/\sigma_a)]/\sigma_a\}^{1/0.08}=1.12*e6$

VIII. CONCLUSION

A custom-made approach for the structural integrity of the shaft in a trucks power Transmission system for various materials has been developed.

For the analysis, we used steel and CFRP, a composite material and determined the equivalent stress, maximum principal stress, minimum principal stress, and total deformation.

Based on the examination of different materials, we can infer that aluminium is better and lighter than steel, and similarly, CFRP is lighter and better than aluminium, resulting in superior performance in CFRP material.

As a result, CFRP material is superior to steel and aluminium because it is less in weight, creates less stress, and enhances overall efficiency and performance.

However, in our project, we did not account for the fact that CFRP is more expensive than aluminium and steel. Other than the cost disadvantage, there are no other characteristics to consider while picking CFRP material.

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