



iJRASET

International Journal For Research in
Applied Science and Engineering Technology



INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 7 Issue: V Month of publication: May 2019

DOI: <https://doi.org/10.22214/ijraset.2019.5006>

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Torsional Behaviour and Whirling Speed Analysis of a Propeller Shaft

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Abstract: Cardan shaft is an essential mechanical component for power transmission in an Automobile. The regular Conventional steel shafts used should be robust, lightweight and should have a low critical speed to improve the overall performance of the vehicle. To experience the maximum power transmission and efficient performance of the vehicle, weight reduction of the drive shaft is one of the most important aspects need to be considered. Determination of the whirling speed of the shaft is another important feature is to be looked after, to get rid of the failure due to lateral vibrations. Shafts usually need to be maintained either below or above the value of critical speed to prevent such failure.

The present work is mainly focusing on the material optimization of the propeller shaft, includes the propeller shaft design for an automobile for a specific value of power transmission. This is sequenced with the design of a solid shaft first, then a hollow shaft is designed, for the same torque transmission. The hollow shaft designed is a good alternative to the solid shaft, whereby the material of the shaft is saved to a notable value. Further, the stress analysis has been carried out by analysis software ANSYS-19.2 to study the torsional behaviour of the shaft. The stress values obtained are favorable, by being within the limits of permissibility. The work is further extended to whirling speed analysis. If the vibrations at these whirling speeds found excessive will lead to the catastrophic failure of shaft. Finally, the Natural frequencies have been determined for the shaft at different modes, to prevent the failure.

Keywords: Propeller shaft, Material optimization, Torsional behaviour, Natural frequency, Vibration level, Critical speed or Whirling speed.

I. INTRODUCTION

A. Theory

A driveshaft, driven shaft, driving shaft, propeller shaft (prop shaft), or Cardan shaft is a mechanical component for transmitting torque and rotation, usually used to connect other components of a drive train, that cannot be connected directly because of distance or the need to allow for relative movement between them. As torque carriers, drive shafts are subject to torsion and shear stress, equivalent to the difference between the input torque and the load. They must, therefore, be strong enough to bear the stress, whilst avoiding too much additional weight as that would, in turn, increase their inertia. To allow for variations in the alignment and distance between the driving and driven components, drive shafts frequently incorporate one or more universal joints, jaw couplings, or rag joints, and sometimes a splined joint or prismatic joint. An automobile may use a longitudinal shaft to deliver power from an engine/transmission to the other end of the vehicle before it goes to the wheels. A pair of short drive shafts is commonly used to send power from a central differential, transmission, or transaxle to the wheels. In the front engine, rear wheel drive vehicles, a longer drive shaft is also required to send power the length of the vehicle. Two forms dominate. The torque tube with a single universal joint and the more common Hotchkiss drive with two or more joints. This system became known as Systeme Panhard after the automobile company Panhard et Levassor patented it.

Most of these vehicles have a clutch and gearbox (or transmission) mounted directly on the engine, with a drive shaft leading to a final drive in the rear axle. When the vehicle is stationary, the drive shaft does not rotate. Some vehicles (generally sports cars, most commonly Alfa Romeos or Porsche 924s), seeking improved weight balance between front and rear, use a rear-mounted transaxle. This places the clutch and transmission at the rear of the car and the drive shaft between them and the engine. In this case, the drive shaft rotates continuously with the engine, even when the car is stationary and out of gear.

A drive shaft connecting a rear differential to a rear wheel may be called a half-shaft. The name derives from the fact that two such shafts are required to form one rear axle. The drive shaft has served as an alternative to a chain-drive in bicycles for the past century, never becoming very popular. SAE 5120 is the steel that is chosen as the material for the shaft. This material consists of a carbon percentage between 0.17 – 0.22%. SAE 5120 is the widely used and most preferable material in the manufacturing of propeller shafts.

B. Literature Review



Drive shafts used in many vehicles like cars, buses, trucks etc. are solid shafts as well as hollow shafts. The main reason for this was a significant saving in weight of drive shaft; the results showed that the hollow drive shaft has a mass of about 4.68 kg, while this amount for solid drive shaft is about 6.12 kg. Numerous studies have been carried out to investigate the optimal design and analysis of drive shafts. In the year 2011, **Amit Kumar Gupta [1]** conducted a static analysis of single piece and two-piece propeller shaft for automotive applications. The Torsional vibration is a type of severe twisting motion in improperly designed coating machines. In this investigation of torsional vibration characteristics of the shaft systems transmitting torque, has become an important part of the designer's responsibility. Satisfactory operation of the heavy duty transmission system may depend to a large extent upon the successful handling of the vibration problem. Propeller shaft of most of the vehicles is found to show no. of modes of failures. The propeller shaft is a rupture of tubes & excessive bending caused due to torsional shear stresses. In the year 2012, **N. Lenin Rakesh, V. Palanisamy and S. Jeevabharathi [2]** has conducted a stress analysis of a shaft using Ansys in this study the shaft is fixed at one end is selected and forces are given at particular points. The reactant forces act in opposite directions. Torque acts at two points in opposite directions. The reactant forces and bending moments are initially calculated. Based on these parameters, the maximum shear stress, normal stress is calculated. The same values are used then calculated by using ANSYS software. Finally, the theoretical and analytical results are compared and verified. In the year 2013, **P. Jayanaidu¹, M. Hibbatullah², Prof. P. Baskar³ [3]** has conducted an Analysis on a Drive Shaft for Automobile Applications. This study deals with the optimization of drive shaft using the ANSYS. Substitution of Titanium driveshafts over the conventional steel material for drive shaft has increased the advantages of design due to its high specific stiffness, strength and low weight. Drive shaft is the main component of the drive system of an automobile. Use of conventional steel for the manufacturing of drive shaft has many disadvantages such as low specific stiffness and strength. Many methods are available at present for the design optimization of structural systems. This paper discusses the past work done on drive shafts using ANSYS and design and modal analysis of shafts made of Titanium alloy (Ti-6Al-7Nb). In the year 2013, **Ravikant¹, Gopal Krishan², Mukesh Didwania³ [4]** has conducted a modal analysis on drive shaft using FEA, in this work finite element analysis of a drive shaft has been taken as a case study. In the present work, the modal analysis of a drive shaft has been carried out the inherent frequencies and vibration mode shapes with their respective deformation. The maximum stress point and dangerous areas are found by the deformation analysis of the drive shaft. The relationship between the frequency and the vibration modal is explained by the modal analysis of the drive shaft. In the year 2014, **¹Bhirud Pankaj Prakash, ²Bimlesh Kumar Sinha [5]** has conducted an analysis on the drive shaft, the objective of this work is to analyze a composite drive shaft for power transmission. 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 steel drive shafts with a Kevlar/epoxy or E glass polythene resin composite drive shaft for an automotive application. The intention of work is to minimize the weight of the drive shaft. In this present work, an attempt has been to estimate the deflection, stresses, and natural frequencies under subjected loads using FEA (Ansys). Further comparison carried out for both loads using FEA. Further comparison carried out for both optimized and stress intensity factor found for both Steel and composite drive shafts. In the year 2015, **Gara Bharat Kumar, M Manoj, P.Satish Reddy [6]** has done Design and Analysis of a Propeller Shaft in CAE tool and ANSYS, in this work the propeller shaft of a vehicle was chosen and analyzed by replacing it with different materials. In the year 2015, **Kiran A, Jagtap and P.M.Sonawane [7]** has done Design and Analysis of drive shaft for Heavy duty truck. In this work an attempt has been made to design a drive shaft for heavy duty truck on the basis of maximum torque transmitting capacity, maximum shear stress and critical speed requirement as design criteria followed by virtual simulation which uses FEA software for evaluating the product performance. In the year 2016, **Ganesh Jadhav, Rushikesh Mhaske, Pritam Chavanke, Amol Wabale, Chaudhari R. S. [8]** studied the failures and cracks developed due to the manufacturing defect like flaws, blow holes & indentation where stresses get concentrated. The maximum stress point and dangerous areas are found by the deformation produced during the modal analysis of Propeller Shaft. The relationship between the frequency and the vibration modal can be explained by the modal analysis of Propeller Shaft. The simulation results show that the natural frequency increases from first mode shape to tenth mode shape and also shows that as the crack depth goes on increasing, the natural frequency of crack shaft goes on decreasing for various mode shapes. The resonance vibration of a system can be avoided effectively by appropriate structure design. Based on the above works as resources the main objectives of the project are framed and has been organized as follows.

II. DESIGN OF AUTOMOTIVE PROPELLER SHAFT

A. Introduction

Design is the creation of a plan or convention for the construction of an object, system or measurable human interaction. The design has different connotations in different fields. In some cases, the direct construction of an object is also considered to use design thinking. Designing often necessitates considering the aesthetic, functional, economic and sociopolitical dimensions of both the design object and design process. It may involve considerable research, thought, modeling, interactive adjustment, and re-design.



Meanwhile, diverse kinds of objects may be designed or processes of designing. Thus “design” may be a substantive referring to a categorical abstraction of a created thing or things. The design of the solid and hollow shaft is done by considering the dimensions that are theoretically calculated by using the material properties and power and torque specifications.

B. Material Properties

SAE5120 is the steel, which is chosen as the shaft material. As per SAE grade, the material is graded as SAE5120. This material consists of a carbon percentage between 0.17 – 0.22%. SAE5120 is widely used and most preferable material in the manufacturing of propeller shafts.

Sl.no:	Properties	SAE5120	Units
1.	Density	7850	Kg/m ³
2.	Young’s modulus	205	GPa
3.	Poisson’s ratio	0.29	
4.	Shear modulus	80	GPa
5.	Tensile strength	360	MPa

Table:- 2.2 Shows the properties of the Material.

C. Design of Solid Shaft and Hollow Shaft

1) *Theoretical Calculations (Solid Shaft)*: These calculations are based on the material (SAE 5120), the length and the torque required to be transmitted are taken from an automobile application. (A Commercial vehicle Tata Sumo Victa)

The Maximum torque required to be transmitted is, T = 223 N-m @ 1000rpm.

Length of a shaft is 970 mm.

The power transmitted is,

$$P = \frac{2\pi NT}{60} = 23.352 \text{ KW}$$

where N = Speed of the shaft in r.p.m

T = Torque transmitted by the shaft in N-m

Tensile Yield strength(S_{yt}) = 360 MPa (for material SAE 5120)

Factor of safety taken as,(FOS) = 5

Tensile Stress(σ_t) = (Tensile Yield Strength) / (Factor of safety) =72 MPa

Shear Stress(τ) = Half of the Tensile Stress =36 MPa(Permissible)

$$\text{Tensional Shear Stress } \tau = \frac{16T}{\pi D^3}$$

Where D = Diameter of Solid Shaft in mm is obtained as 31.597 mm

The nearest standard value for $D \approx 32 \text{ mm}$.

Safety Check for the rounded value of D as 32mm.

$$\tau = \frac{16T}{\pi D^3} = \frac{16 * (223 * 10^3)}{\pi * (32)^3} = 34.65 \text{ MPa} < \text{Permissible value (36 MPa)}$$

Therefore, the diameter of the solid shaft recommended is D=32mm is safe.

2) *Design of Hollow shaft (Theoretical Calculations)*: The hollow propeller shaft is designed for the same amount of torque (223 N-m @ 1000rpm)is to be transmitted. These calculations are based on the material (SAE 5120). The length is considered as it is as mentioned for the solid shaft.

$$\text{Shear Stress } \tau = \frac{16T}{\pi d_o^3 (1-k^4)}$$

Where, $k = \frac{d_i}{d_o} = 0.6$ (where k is the ratio of inner and outer diameters of a shaft)

d_o = Outer Diameter of the Shaft

d_i = Inner Diameter of the Shaft

The diameters of the hollow shaft,

$$d_o = \sqrt[3]{\frac{16 * T}{\pi * \tau * (1 - k^4)}} = \sqrt[3]{\frac{16 * 223 * 10^3}{\pi * 35 * (1 - (0.6)^4)}} = 33.094 \text{ mm}$$

The standard value $d_o \approx 35 \text{ mm}$

The inner diameter of the hollow shaft,

$$d_i = 0.6 * d_o = 0.6 * 33.094 = 19.8564 \text{ mm}$$

$$d_i = 0.6 * d_o = 0.6 * 35 = 21 \text{ mm}$$

The standard value $d_i \approx 21 \text{ mm}$

The standard value for the inner diameter is taken as $d_i \approx 21 \text{ mm}$

Safety Check for rounded values of inner and outer diameters as 21mm and 35mm respectively.

$$\tau = \frac{16T}{\pi d_o^3 (1 - k^4)}$$

$$\tau = 30.43 \text{ MPa} < \text{Permissible value (35.5 MPa)}$$

Therefore, the diameters of the hollow shaft

$d_o=35\text{mm}$, $d_i=21\text{mm}$ are safe

The material optimization is expressed in terms of percentage weight saving when a Hollow one replaces the solid shaft.

3) *Design Comparison between the Solid Shaft and Hollow Shaft:* Material optimization (material reduction) is the main objective of these designs, when they are compared between each other, a mass of the hollow shaft is much lesser than that of the solid shaft. The details are as follows.

Mass of the solid automotive propeller shaft is found as 6.12 kg.

Mass of the hollow automotive propeller shaft is found as 4.68 kg.

The total mass of 1.44 kg is being reduced from solid shaft to hollow shaft.

Hence, the percentage reduction of mass of the shaft material is found to be 23.5%.

4) *Model Analysis of the Solid Shaft:* The solid shaft has been model analysis by using the software ANSYS-19.2. Length of the shaft as 970 mm and the diameter as 32 mm is considered.

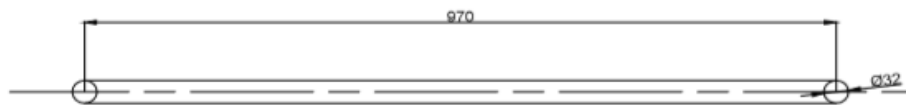


Fig 2.5 Schematic diagram of the solid shaft

Above figure shows the model of solid automotive propeller shaft modeled by using software Catia V5. The material (SAE 5120) is the most commonly used material by many of the automobile industries. The modal Analysis is carried out by ANSYS, maximum stress values are identified and found within the permissible range. Results as follows.

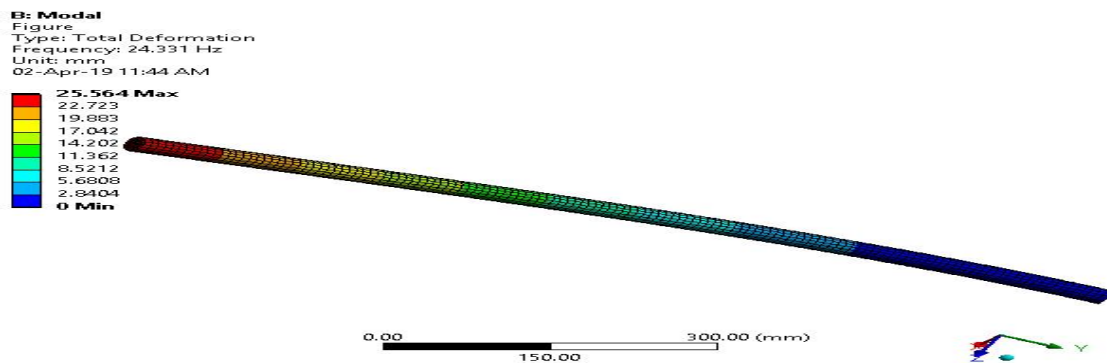


Fig 2.5.0.1: Shows the Total deformation of a solid shaft with a critical frequency of 24.331 Hz at Node 1.

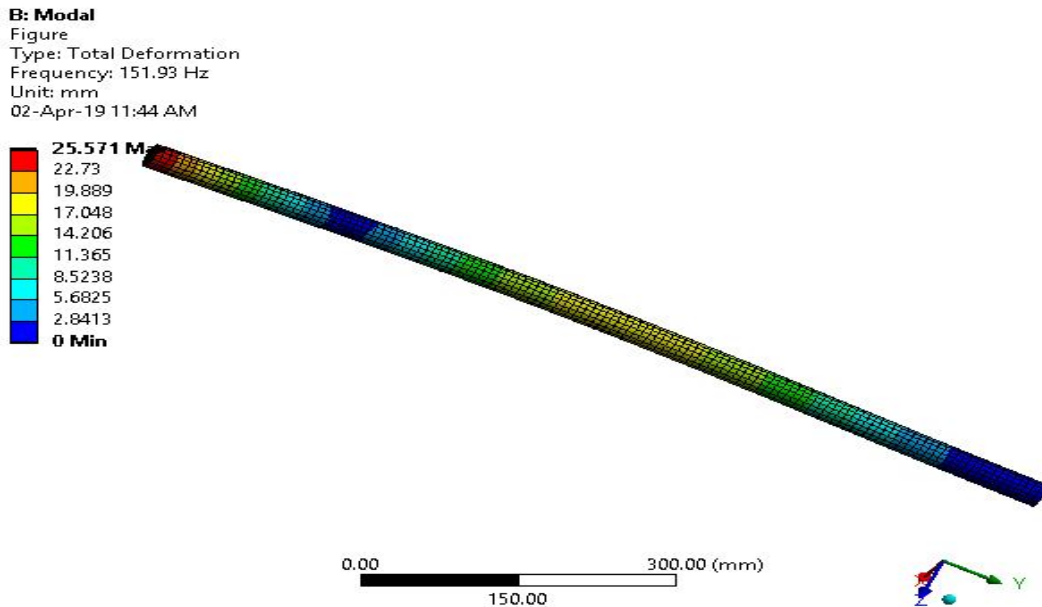


Fig 2.5.0.2: Shows the Total deformation of a solid shaft with a critical frequency of 151.93 Hz at Node 2.

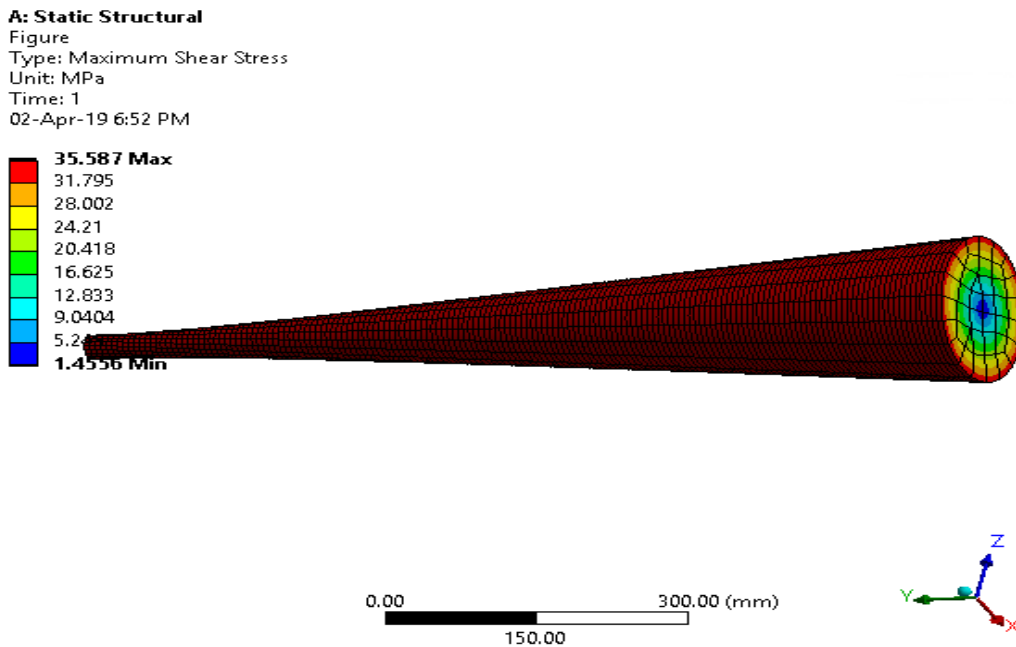


Fig 2.5.0.3: Shows the Maximum shear stress 35.587 Mpa of a solid shaft which is less than the permissible shear stress of 36 Mpa.

Parameters	Node 1	Node 2
Stress value	25.564 Mpa	25.571 Mpa
Critical f_n	24.331 Hz	151.93 Hz

Table: 2.5 Results observed from above Simulation.

From the simulation, observed that stress are 25.564 MPa, 25.571 MPa with respective node 1 and node 2. So There is a slight variation in stresses with a huge difference of critical speed from 1459.86 Rpm to 9115.8 Rpm.

5) Theoretical Calculations for Critical speed: (Solid shaft):

The lowest Critical Speed is $f_n = 0.565 \sqrt{\frac{g * E * I}{W * L^4}}$

Formula for Critical Speed $f_n = \frac{\pi}{2} \left(n - \frac{1}{2}\right)^2 \sqrt{\frac{g * E * I}{W * L^4}}$

Where $n = 2, 3, 4 \dots \dots \dots$

f_n = Natural Frequency (Hz)

I = Moment of inertia of shaft (mm⁴)

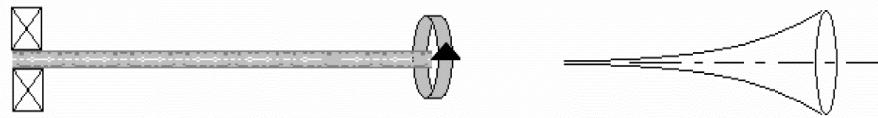
E = Young's Modulus (GPA)

W = weight of the shaft

L = length of the shaft (m)

g = Acceleration due to Gravity (m/s²)

Noted that the lowest speed almost corresponding to $n = 1$



At $n = 1$

$$f_n = 0.565 \sqrt{\frac{g * E * I}{W * L^4}}$$

$$E = 205 * 10^9$$

$$W = \rho * A * g$$

$$= 7850 * \frac{\pi}{4} (0.032)^2 * 9.81$$

$$= 61.933 \text{ N/m}$$

$$I = \frac{\pi}{4} r^4, \text{ for circular cross section}$$

$$f_n = 0.565 \sqrt{\frac{9.81 * 205 * 10^9 * 5.147 * 10^{-8}}{61.933 * 0.97^4}}$$

$$= 24.548 \text{ Hz}$$

At $n = 2$

$$f_n = \frac{\pi}{2} \left(n - \frac{1}{2}\right)^2 \sqrt{\frac{g * E * I}{W * L^4}}$$

$$= 153.56 \text{ Hz.}$$

D. Model Analysis of the Hollow Shaft:

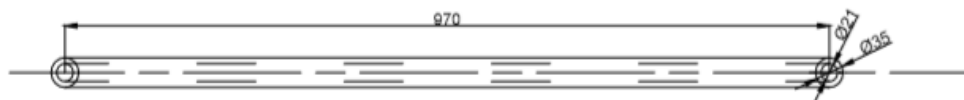


Fig 2.6 Schematic diagram of the hollow shaft

Above figure shows the model of Hollow propeller shaft modeled by using software Catia V5. The material used in the shaft is SAE 5120 is the most commonly used material by many of the automobile industries. The modal Analysis is carried out by ANSYS; maximum stress values are identified and found within the permissible range. Results as follows.

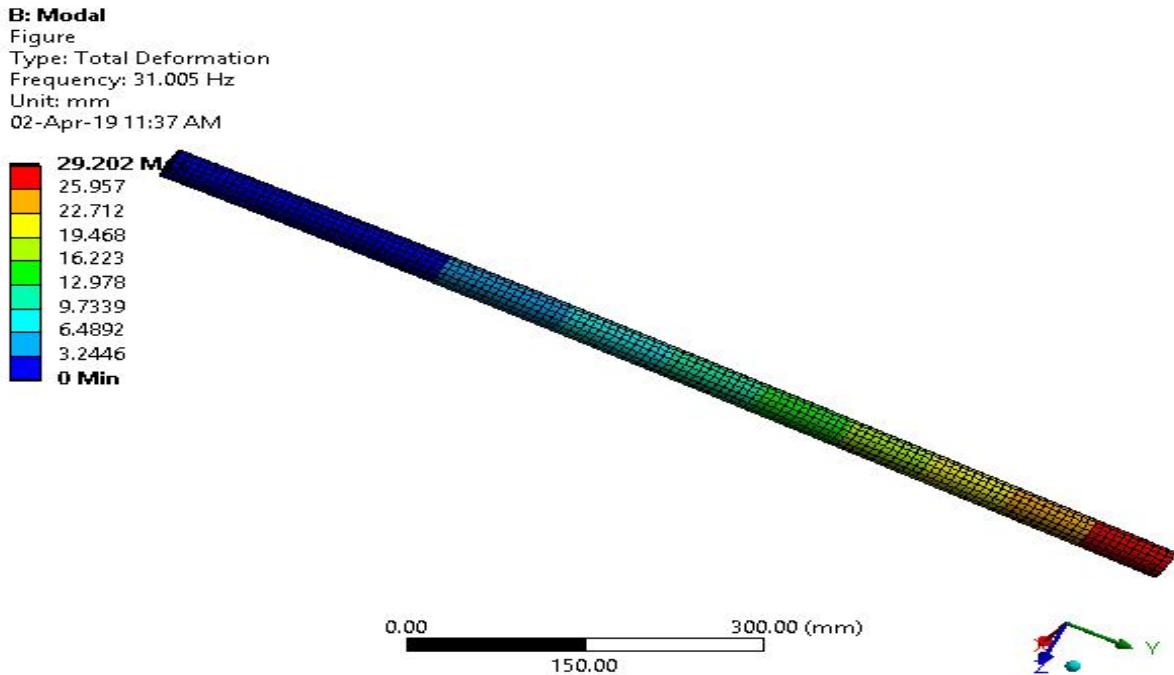


Fig 2.6.0.1: Shows the Total deformation of a hollow shaft with a critical frequency of 31.005 Hz at Node 1.

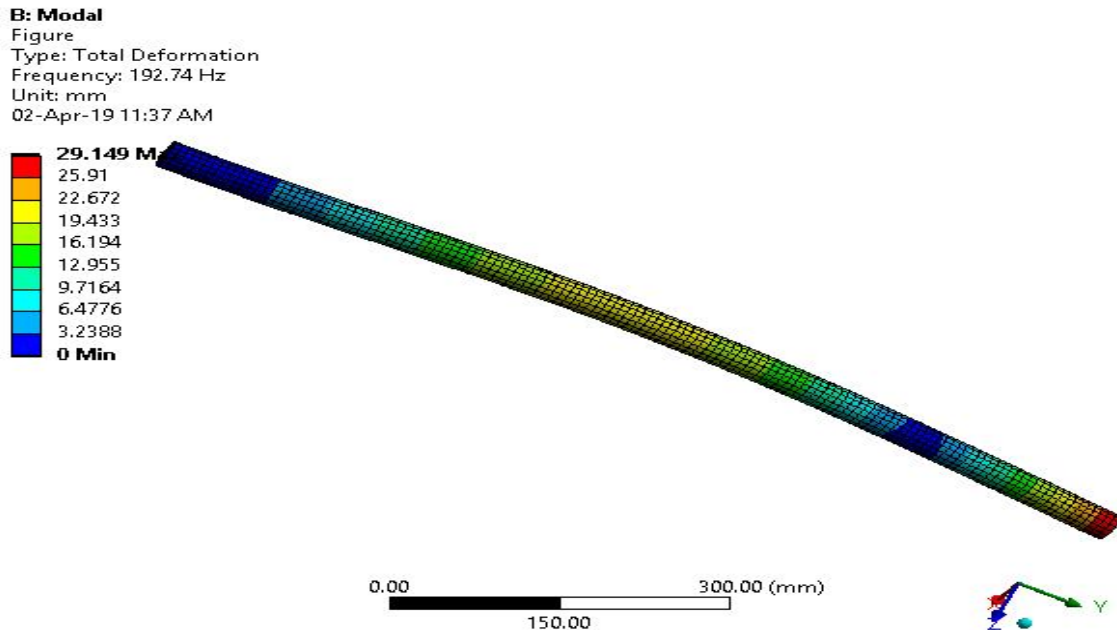


Fig 2.6.0.2: Shows the Total deformation of a hollow shaft with a critical frequency of 192.74 Hz at Node 2.

A: Static Structural

Figure
 Type: Maximum Shear Stress
 Unit: MPa
 Time: 1
 02-Apr-19 5:44 PM

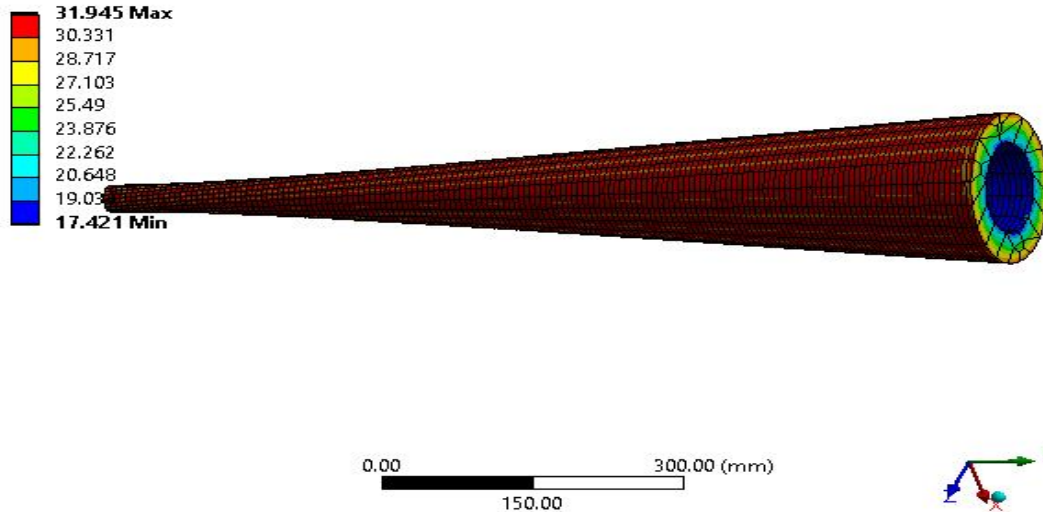


Fig 2.6.0.3: Shows the Maximum shear stress 31.945 MPa of a hollow shaft, which is less than the permissible shear stress of 36 MPa.

Parameters	Node 1	Node 2
Stress value	29.202 MPa	29.149 MPa
Critical f_n	31.005 Hz	192.74 Hz

Table:- 2.6 Results observed from above Simulation.

From the simulation, observed that stress are 29.202 MPa, 29.149 MPa with respective node 1 and node 2. So There is a slight variation in stresses with a huge difference of critical speed from 1860.3 Rpm to 11564.4 Rpm

1) *Theoretical Calculations for Critical speed (Hollow shaft):* Since it is a cantilever position

$$\begin{aligned}
 &\text{At } n = 1 \\
 f_n &= 0.565 \sqrt{\frac{g * E * I}{W * L^4}} \\
 E &= 205 * 10^9 \\
 W &= \rho * A * g \\
 &= 7850 * \frac{\pi}{4} (0.035^2 - 0.021^2) * 9.81 \\
 &= 47.418 \text{ N/m} \\
 I &= \frac{\pi}{4} (r_o^4 - r_i^4), \text{ for circular cross section (Hollow shaft)} \\
 I &= 6.412 * 10^{-8} \text{ m}^4 \\
 f_n &= 0.565 \sqrt{\frac{9.81 * 205 * 10^9 * 6.412 * 10^{-8}}{47.418 * 0.97^4}} \\
 &= 31.314 \text{ Hz}
 \end{aligned}$$



$$\begin{aligned}
 & \text{At } n = 2 \\
 f_n &= \frac{\pi}{2} \left(n - \frac{1}{2} \right)^2 \sqrt{\frac{g * E * I}{W * L^4}} \\
 &= 195.882 \text{ Hz.}
 \end{aligned}$$

E. Comparison of Results among Solid and Hollow Shafts for Material SAE 5120:

Solid Shaft values					
Theoretical τ_{max} (MPa)	Analytical τ_{max} (MPa)	$F_n T_{theo}$ (Hz)		$F_n A_{nyl}$ (Hz)	
		N=1	N=2	N=1	N=2
34.65	35.587	24.548	153.56	24.33	151.93
Hollow Shaft values					
Theoretical τ_{max} (MPa)	Analytical τ_{max} (MPa)	$F_n T_{theo}$ (Hz)		$F_n A_{nyl}$ (Hz)	
		N=1	N=2	N=1	N=2
30.43	31.945	31.314	195.88	31.00	192.74

Stress validation is the main objective of this analysis. when they are compared between each other, maximum shear stress acting on the hollow shaft is much lesser than that of the solid shaft. The details are mentioned as follows

- Maximum shear stress acting on the solid automotive propeller shaft is 35.587 MPa.
- Maximum shear stress acting on the hollow automotive propeller shaft is 31.945 MPa.

F. Statement

The maximum shear stress acting on both the shafts is lesser than the permissible stress. Hence both the designs are considered to be safe. Even though both the shafts are considered to be safe hollow shaft is more preferable, since the maximum value of shear stress acting on it is less than that of the solid shaft, and the percentage weight of the shaft material saved, while preferring hollow over the solid is found to be 23.5%

III. CONCLUSION

Automobiles with lightweight have become the present trend for most of the Automotive companies, to meet the demands and requirements of the customers. Many changes have been introduced and implemented to satisfy the customers, without compromising at the performance and efficiency of the vehicles. Weight reduction of various components in an automobile is being a challenging topic for most of the researchers for the past a few decades, which will directly affect the performance and fuel consumption of the vehicle. This is justified by many alternatives, such as material replacements, weight reduction of existing components, without reducing the efficiency of the vehicle. In this context, the present work is focused on the design of solid propeller shaft for a specific vehicle model, and a hollow shaft is designed, that saved 23.5% of the weight of shaft material, replaces it. Stress analysis has been carried out for both the shafts (solid & hollow) to check the working stress values, which is satisfied by being within the limits of permissibility. This also includes the determination of the permissible value of whirling speed, that could prevent the failures due to lateral deflections. The Ansys-19.2 simulation package is used for validation of stresses, where stress obtained are found to be safe.

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Nanda Kishor Reddy Yakkaluru- Graduated from Department of Mechanical Engineering, GST, GITAM Deemed to be University, Bangalore. Presently engaged in the project work of Designing of shafts and Its Material Optimization”.



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