Design and Analysis of Carbon-Kevlar/Epoxy Hybrid Composite Propeller Shaft

Beka Dahine Hordofa

A Thesis Submitted to department of Mechanical Engineering

School of Mechanical, Chemical and Materials Engineering

Office of Graduate Studies

Adama Science and Technology University

December 2023

Adama, Ethiopia

Design and Analysis of Carbon-Kevlar/EpoxyHybrid Composite Propeller Shaft

Beka Dahine Hordofa

A/PE 21031/12

Advisor: Siraj Kedir Busse (PhD)

A Thesis Submitted to department of Mechanical Engineering

School of Mechanical, Chemical and Materials Engineering

Office of Graduate Studies

Adama Science and Technology University

December 2023

Adama, Ethiopia

DECLARATION

I hereby declare that in this thesis entitled "Design and Analysis of Composite carbon-kevlar/epoxy hybrid on propeller shaft" is my own work. That is, it has not been submitted for the award of any academic degree, diploma or certificate in any other university. All sources of materials that are used for this thesis has been duly acknowledged through citation.

Beka Dahine Hordofa	______________	______________
Name of student	Signature	Date

RECOMMENDATION

We, the advisors of this thesis, hereby certify that we have read the revised version of the thesis entitled "Design and Analysis of Composite carbon-kevlar/epoxy hybrid propeller shaft" prepared under our guidance by Beka Dahine Hordofa, in Partial Fulfillment of the Requirements for the degree of Master of Science in Automotive Engineering. Therefore, we recommended the submission of revised version of the thesis to the Department following the application procedures.

Dr. Siraj Kadir	______________	_____________
Major Advisor	Signature	Date

APPROVAL PAGE

We, the advisers of the thesis entitled “Design and Analysis of Composite carbon-kevlar/epoxy hybrid propeller shaft” and developed by Beka Dahine Hordofa, herby certified that the recommendation and suggestions made by the Board of Examiners are appropriately incorporated in to the final version of the thesis.

Dr. Siraj Kadir	____________________	____________________
Major Advisor	Signature	Date

We, the under signed members of the Board of Examiners of the thesis by Beka Dahine Hordofa, have read and evaluated the thesis entitled “Design and Analysis of Composite carbon-kevlar/epoxy hybrid propeller shaft” and examined the candidate during open defense. Therefore, this is to certify that the thesis has been accepted in partial fulfillment of the requirement of the Degree of Master of Science in Automotive Engineering.

____________________	____________________	____________________
Chair person	Signature	Date
____________________	____________________	____________________
Internal examiner	Signature	Date
____________________	____________________	____________________
External examiner	Signature	Date

Finally, approval and acceptance of the thesis is contingent upon submission of its final copy to the Office of Postgraduate Studies (OPGS) through the Department Graduate Council (DGC) and School Graduate Committee (SGC).

____________________	____________________	____________________
Head of the Department	Signature	Date
____________________	____________________	____________________
School Dean	Signature	Date
____________________	____________________	____________________
Office of Post Graduate Studies, Dean	Signature	Date

ACKNOWLEDGMENT

I would like to take this opportunity to express my profound gratitude and respect to all those who helped me through the duration of this thesis work. First, I would like to express my gratitude to God who gave me the strength, health, and chances, and making all things possible towards the success of this thesis. I am highly grateful to my advisor Dr. Siraj Kedir Busse for his friendly approach, valuable advice, guidance and continuous follow up from the beginning to the end of the research work. I have been benefited at large from their experience and expertise in the subject matter. I would like to thank all friends then my work colleagues for their support and understanding not just during the thesis lifetime but throughout my study under the postgraduate program. I would also like to acknowledge staff members of the Adama science and technology university school of Mechanical, Chemical and Materials engineering, for their valuable comments on my work. Furthermore, I want to show my appreciation to everyone who had an impact on my work.

Finally, I must express my very profound gratitude to my parents specifically my wife Safi Hayilu for providing me with unfailing support and continuous encouragement throughout my years of study and through the process of researching and writing this research works and my love son Uma Beka who enjoyably me when I want to enjoy.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

ACRONYMS AND SYMBOLS

d_o Outer diameter of hollow steel Shaft

d_i Inner diameter of hollow steel Shaft

t Thickness of the hollow steel shaft

v Poisson's ratio

l Length of shaft

E Young's modulus

M Micrometer

T_1 Input torque

T_2 Output torque

Ω Rotational angular velocity

ϕ Rotational angle of driven shaft

Θ Operating angle

K_s Stiffness constant

ρ Density

G Shear modulus

ΔT Total deformation

J Polar moment of inertia

τ Shear Stress

r_m Mean radius

f_{nb} Natural frequency

I_x Area moment of inertia

M Mass

Σ Stress

Hz Hertz

NVH Noise, vibration and harshness

FRP Fiber-reinforced plastic

MMC Metal matrix composite

CMM Ceramic Matrix Materials

PMC Polymer matrix composite

PRP Particle reinforced polymer

CVC onstant Velocity

U_{TS} Ultimate tensile strength

SBS Short beam shear

FOS Factor of safety

CATIA Computer-aided three-dimensional interactive application

FEA Finite Element Analysis

CAD Computer Aided Design

CAE Computer Aided Engineer

N_{cr} Critical Speed

3D Three-dimensional

I Moment of inertial

K_s Stiffness for Solid

K_h Stiffnessfor hollow

ABSTRACT

Propeller shaft connects gearbox to the final drive gears of the vehicle. Propeller shafts are traditionally made of steel and according to various researches, the majority of the propeller shafts suffer damage from corrosion, high fatigue, high bending, high impact, high vibration, high compressive loading and severe shock loading. Fortunately, Composite materials have a great damping capacity, they produce less vibration and noise while also being corrosion resistant. Therefore, in this thesis, the propeller shaft of Tata 207 DI Single Cabin Pick up was replaced with a two-piece propeller shaft carbon-Kevlar epoxy hybrid composite drive shaft. The main objective of this thesis is to design and analyses of propeller shaft using carbon-Kevlar epoxy hybrid composite. Modeling of the existing and the composite propeller shaft is carried out in CATIA V5 and its analysis is performed in ANSYS. Basic dimensions of the existing model are taken by direct measurement whereas the dimension of the composite propeller shaft is obtained from analytical design. The analysis which is done on ANSYS are static structural, transient structural, modal analysis, harmonic analysis and rigid dynamics. Three modified models were prepared in order to consider all the necessary conditions and to select the similar design two-piece mode propeller shaft for compare result from analysis. These are two-piece modified propeller shaft, one-piece modified propeller shaft without center bearing and one-piece modified propeller shaft with center bearing. According the result obtained from the analysis the two-piece modified propeller shaft selected. The result from the analysis shows that the weight of the two-piece modified propeller shaft is improved by 45.25% compared to the existing propeller shaft. It also has a lowest stress distribution, lowest deformation and better performance in terms of vibration compared to the existing and the other modified propeller shaft. It has the lowest deformation at higher frequency ranges, for instance the maximum deformation of the two-piece modified propeller shaft at 368.94Hz is 23.295mm which is less than the existing model.

Key Words: Carbon-Kevlar ,propeller shaft, Tata 207 DI Single Cabin Pickup, ANSYS

CHAPTER ONE
INTRODUCTION

1.1 Background

A propeller shaft is a mechanical component for transmitting torque and rotation that is commonly used to connect other drive train components that cannot be directly connected. The majority of engines or motors generate torque by rotating motion. Transmission propeller shafts account for 7-15 percent of sprung weight in an automotive transmission system, which is the weight of the propeller shaft without any change in torque bearing capacity or stiffness(Pallavi et al., 2015).

Drive shafts are traditionally made of steel (SM45C) and convey power from the gearbox to the vehicle's rear axle. The graphite, carbon, glass fiber, and aluminum driveshaft tubes were developed in response to industry demand for improved performance and efficiency in light trucks, vans, and high-performance cars. In the automotive and aerospace sectors, glass fiber/epoxy resin composites are often used. Glass fiber offers various advantages over other materials, including simplicity of fabrication, the capacity to construct very long strands, and high impact resistance. The primary reason for this is a significant weight decrease in the driveshaft. The composite driveshaft weighs around 2.7kg, whereas the steel driveshaft weighs approximately 10 kg(A & B. Kerur, 2021).

Most cars, vans, and small trucks have propeller shafts that are 1.25 to 2 meters long and made of two pieces of steel or aluminum with an additional center bearing support. These shafts should be able to transfer torques greater than 3000 Nm and operate at speeds ranging from 6000 to 7000 rpm. Vehicles with this type of propeller shaft are frequently used outlying areas with poor road topography and construction site work. According to the owners, when compared to other vehicles, the majority of the propeller shafts suffer damage from corrosion, high fatigue, high bending, high impact, high vibration, high compressive loading, and severe shock loading (H.B.H. Gubran and K. Gupta 2014).

To meet the vibration requirements, a one-piece composite driveshaft may be constructed. This removes any assembly by attaching the two-piece steel shaft, reducing total weight, vibrations, and cost. Fuel consumption will be lowered due to weight reduction. The car's complete driveline is made up of multiple rotating mass components. The rotational mass of the drive train is estimated to lose 17-22 percent of the power provided by the engine(Sweety P. Mhaske, 2016; Yusuf et al., 2015).

Composite materials have a great damping capacity, they produce less vibration and noise while also being corrosion resistant. More available power is communicated by a lighter spinning because weight. A composite driveshaft improves vibration dampening, cabin comfort, drive train wear, and tire traction. Composite materials also have a lower modulus of elasticity. As a result, when torque peaks occur in the driveline, the composite driveshaft can act as a shock absorber, lowering stress on a component of the drive train while still transferring torque. Drive shafts and propeller shafts are interchangeable terminology (Parshuram & Mangsetty, 2013).Every 10% weight reduction from the average new vehicle or light truck reduces fuel consumption by 7%. Noise, vibration, and harshness (NVH) have become more important aspects in vehicle design as a result of the pursuit of better refinement. Harshness is strongly connected to vehicle refinement and the quality and transitory nature of vibration and noise (Stoffels, 2017).

Hybrid composites are identified by the use of two or more distinct materials combined in a single matrix. Hybrid composites show some unique features that can be used to meet the exact requirements of the structure under consideration. Compared to monolithic composite hybrid composites have given technological advances because they provide more balanced properties. Some specific advantages of hybrid composites over monolithic composites include balanced strength and stiffness, balanced bending and membrane mechanical properties, balanced thermal distortion stability, reduced weight and/or cost, improved fatigue resistance, reduced notch sensitivity, improved fracture toughness and/or crack arresting properties, and improved impact resistance (Radulović, 2020).

Fiber hybrids take advantage of the best qualities of several fiber types while potentially lowering raw material prices. Carbon, basalt, glass, jute, kenaf, flax, hemp, kevlar, and other fiber composites are all available. Kevlar is one of the most advantageous composite materials among these accessible materials. Kevlar has several properties, including a high stiffness modulus, toughness, thermal stability, and, most significantly, strength. Because of their low weight and outstanding mechanical qualities, Kevlar fiber composites are becoming increasingly popular (Priyanka et al., 2019).

Weight reduction has been the primary emphasis of automotive manufacturers in the current development in order to save natural resources and save energy. Weight reduction in vehicles may be achieved by the use of better materials, design optimization, and improved manufacturing techniques (Chirinda & Matope, 2020; Czerwinski, 2021).

To cut back on unnecessary fuel use, component counts must be lowered and composite materials with enhanced corrosion resistance, stiffness, fatigue resistance, torsion strength, balancing, and shock loading interference must be used.

In this thesis research, the propeller shaft was replaced with a two-piece carbon-Kevlar epoxy hybrid composite drive shaft. The propeller shaft is built of a carbon-Kevlar epoxy hybrid composite material with two universal joints and a jaw coupling. The composite propeller shaft is analyzed to determine the efficiency of the composite material on the load carrying capacity and torque transmission of the propeller shaft. Because of the increased weight of the propeller shaft, the stainless steel was replaced with composite materials, which are much lighter than stainless steel. When compared to stainless steel, composite materials are less expensive(Kale & Secanell, 2018).

The dimensions of the car's existing propeller shaft, which will be used for further computation and analysis is taken by direct measurement. Theoretical design is carried out by employing fundamental material strength ideas. Developing a solid model: Using the CATIAV5 program, a three-dimensional solid model of the shaft is built on the computer. This 3D model is then sent to the ANSYS program for finite element analysis. Finite Element Method: Pre-processing, solution, and post-processing are the three basic processes.

The geometric domain of the problem, the element types to be used, the material properties of the elements, the geometric properties of the elements (length, area, and the like), the element connectivity (the mesh of the model), the physical constraints (boundary conditions), and the loadings are all defined in pre-processing (model definition). The computed findings are then utilized in the solution phase to determine things like deformation and element stress, which are carried out by commercial software. ANSYS software will be used for post-processing analysis on the finite element model of the carbon-kevlar/Epoxy composite drive shaft. In comparison to steel and composite shafts, the study discovered that the use of composite drive shafts decreased weight.

1.2 Statement of the Problem

A vehicle's propeller shaft is one of its most important parts. The two or three-piece propeller shaft assembly, which consists of three universal joints, a central support bearing, and a bracket, has caused the assembly's overall weight to grow, adding weight and complicating the mechanism (Chirinda & Matope, 2020; Czerwinski, 2021; V.Jose Ananth Vino, 2015).

Steel or aluminum has a lower rotational frequency than the shaft, increasing the amplitude of vibration of the drive shaft (Gubran & Gupta, 2014). Drive shafts are often constructed in two halves of high-quality steel (Steel SM45) to increase the basic bending natural frequency, which is inversely proportional to the square of beam length and proportionate to the square root of specific modulus. The two-piece steel drive shaft is composed of three universal joints, a central supporting bearing, and a bracket, all of which increase weight and reduce fuel efficiency (Gubran & Gupta, 2014).

Kevlar fibers and Kevlar fiber-reinforced polymer (KFRP) composites have potential applications in defense and impact-loaded scenarios due to their high strength and impact resistance (Priyanka et al., 2019). Carbon fibers have better strength and stiffness than other fibers but have limited extensibility and low damage tolerance, whereas Kevlar fibers offer a higher degree of toughness and damage tolerance, and their hybridization might be a good solution to carbon fiber limits. Kevlar and carbon composites can be used to create lightweight and effective body armor. Hybrid (carbon-Kevlar) composites exhibited better results than Kevlar composites and it was observed that hybrid fabric composites exhibited superior properties compared to Kevlar composites(Madarvoni & Sreekanth, 2022).

Tata 207 DI Single Cabin is designed for urban as well as rural use, the 207 DI is packed with safety features that make for better stability and ease of handling. With the highest ground clearance in its category, and larger tires for more traction, the 207 DI is the ideal multi-terrain, multi-application vehicle which marries high performance with low maintenance requirements .The design of the composite propeller shaft is done for Tata 207 DI Single Cabin Pickup which uses two-piece propeller shaft. According to the owners, when compared to other vehicles, the majority of this vehicle’s propeller shafts suffer damage from corrosion, high fatigue, high bending, high impact, high vibration, high compressive loading, and severe shock loading when we use rural service like construction site work, bad topography etc .

As a result, the goal of this study is to evaluate the fatigue, impact, and shock resistance of Tata 207 DI Single Cabin propeller shafts using fatigue analyses, vibration analyses, and dynamic analyses of a two-piece propeller shaft composite material made of carbon-kevlar epoxy hybrid composite, composed of 60% Carbon fibers for better strength and stiffness and 40% Kevlar fibers for higher degree of toughness and damage tolerance. which has a higher strength-to-weight ratio, high conformability, stiffness while absorbing energy, dimensional stability, high impact resistance, and fatigue resistance. To assess the strength and behavior of engineering structures, the finite element technique, a computer-based numerical approach, is utilized.

1.3 Objective

1.3.1 General Objective

The general objective of this thesis is to design and analyses of propeller shaft made of carbon-Kevlar epoxy hybrid composite.

1.3.2 Specific Objectives

The main objective of this thesis will be achieved through the following specific objectives:

- ✓ Model and analyze the existing propeller shaft under various conditions.
- ✓ Design and model the Carbon –Kevlar epoxy hybrid composite propeller shaft
- ✓ Perform static, transient and vibration analysis on the modified model
- ✓ Compare the performance of Carbon –Kevlar epoxy hybrid composite propeller shaft with the existing shaft.

1.4 Significance of the Study

The significance of this study is that it provides guidelines for designing propeller shafts from composite materials and analyzing their performance using ANSYS software. The study also highlights the advantages of composite materials over conventional steel propeller shafts and how they can reduce the weight of vehicles. The findings of this study can be useful for individuals, researchers, and composite propeller shaft manufacturers. It would provide of this study is that it provides guidelines for designing propeller shafts from hybrid composite materials and analyzing their performance using ANSYS software. The study also highlights the advantages of composite materials over conventional steel propeller shafts and how they can reduce the weight of vehicles. The findings of this study can be useful for individuals, researchers, and composite propeller shaft manufacturers. It would provide some guidelines guideline, how to design propeller shafts from hybrid composite materials and analyze their performance using ANSYS software. It also gives a better understanding of the advancement of composite materials over the conventional steel propeller shaft and reduces the weight of the vehicle.

1.5 Scope of the Thesis Work

The scope of the study in this research paper is design and analysis of propeller shaft from carbon-Kevlar epoxy hybrid composite material by using CATIAV5 and FEA ANSYS software. The study was conducted at Adama Science and Technology University, School of Mechanical, Chemical and Materials Engineering, department of Mechanical Engineering. The paper outlines six stages of the solution for the problem, which includes defining dimensions, material selection, theoretical analysis, creating a solid model, finite element analysis, and comparative study.

1.6 Limitation

Difficulties of obtaining proper materials from the market, increased cost and experimental investigation prepare composite materials has not been don due to unavailability of proper laboratory for conducting material tests, relevant materials have urged me to depend on only materials from internet. Also prototype of propeller shaft is not manufactured due to lack of budget.

1.7 Organization of the Thesis

This thesis is divided into six chapters. The first chapter includes information about the study's background, issue statement, goal, and significance. A review of the literature that is relevant to this thesis topic and has been read by many academics is given in Chapter two. Chapter three discusses the materials and methods utilized for the investigation and design. Chapter four covers , the design of a composite propeller shaft, modeling and analysis of the propeller shaft, and selection of cross-section. In Chapter five, results and discussion are presented. Chapter six covers conclusion and recommendation.

CHAPTER TWO
LITERATURE REVIEW

2.1 Introduction

Propeller shafts are used as power transmission pipes in a variety of various applications, including cooling towers, pumping systems, vehicles, and aircraft. By knowing the torque and the material's permitted shear stress, the size of the shaft's cross section can be determined when building a metallic shaft. When the area beneath the automobile cabin constrains the outer radius, the shaft's inner radius has a specific value because the geometric parameter (polar moment of inertia of the cross-sectional area divided by the outer radius) equals the torque divided by the permitted shear stress (Rangaswamy & Vijayarangan, 2005). Steel propeller shafts are typically made in two parts to raise the fundamental bending natural frequency, which is inversely proportional to the square of beam length and proportionate to the square root of specific modulus. The two-piece steel drive shaft is made up of three universal joints, a central supporting bearing, and a bracket, all of which add to the overall weight of the vehicle. Because the bending natural frequency is inversely related to the square of the span length, increasing the critical speed of the shaft under varied load circumstances requires either decreasing the span length or increasing the bending strength.

The weight, low critical speed, and vibration properties of the iron propeller shaft are its drawbacks. The use of standard metal propeller shafts causes performance limitations owing to a lower critical speed, weight, fatigue, and vibration, which has led to the development of composite propeller shafts, which have solved various automotive and industrial difficulties. Numerous solutions, including vibration, shock absorbers, vibration dampers, flywheels, harmonic dampers, numerous shafts with bearings, couplings, and heavy accompanying gear, have only patchy success in solving the issues ParshuramD,SunilMangsetty 2013.

Composite propeller shafts provide superior vibration dampening, improved passenger comfort, reduced wear on drive train components, and improved tire traction. One-piece torque tubes are also more efficient in terms of assembly, maintenance, and part complexity. It is well known that a vehicle's weight must be reduced in order to minimize fuel consumption and meet strict environmental laws. In order to reduce the weight of structural components while maintaining the quality, performance, and dependability of vehicles, the automotive industry is utilizing composite materials (ParshuramD,SunilMangsetty 2013).

A composite material is a macroscopic blend of two or more materials having an obvious interface. In composite materials, the fiber or particle phase is frequently stiffer and stronger than the continuous phase. Today, a variety of sectors, including aerospace, automotive, construction, and others, employ composite materials. People employ composite materials to make car door panels and bonnets for automotive purposes(Constantine D. Papaspyrides && Kiliaris, 2014).

A composite material is formed by combining a reinforcing material with a matrix. A composite's qualities outperform the properties of its separate components. Reinforcement is the primary load-bearing component and is responsible for the composite material's strength and stiffness. Fibers, particles, and flakes are examples of reinforcement forms. Matrix, on the other hand, maintains the orientation of the reinforcement and protects it against chemical and physical harm. It is also in charge of the uniform distribution of an applied load among the reinforcing elements. Composite materials are frequently utilized when traditional materials like metals, ceramics, and polymers are unable to precisely match the requirements of a certain application. One of the key advantages of composite materials is their versatility, which may be achieved by varying the type and ratios of the component materials, their orientations, process parameters, and other elements. Composites are appropriate materials for automotive and aerospace applications because they have excellent mechanical properties and are lightweight. Composites also have high toughness, thermal conductivity, fatigue resistance, and corrosion resistance(Buckner et al., 2016).

Advanced composite materials such as graphite, carbon, Kevlar, and glass are widely used due to their high specific strength (strength/density) and specific modulus (modulus/density). Applications for long power driver shafts (propeller shafts) appear to be best suited for modern composite materials. The torque-management design of propeller shafts causes torsion and shear stresses. The elastic properties of composite materials can be adjusted to improve the load bearing capacity (for example, torque) of propeller shafts using a range of fiber and ply stacking sequence designs(Ganeshan et al., 2015).

The use of fiber reinforced composite materials are increasing day by day due to their excellent mechanical, corrosion and wear resistance properties. They are widely used in different high-performance applications such as aerospace naval, automotive vehicle, construction etc. These composite materials are advantageous over conventional materials (iron, steel and aluminum alloys) because their light weight, good fatigue performance, design flexibility and low through life maintenance and processing costs. Composites are two unique types of materials that are combined to improve their respective qualities (Md. MilonHossain[1,*] , M A Khan [2] , R A Khan [3] , Md. Abu BakarSiddiquee[4] and Tauhidul Islam [5] 2015).One or more discontinuous phases are embedded in a continuous phase. The discontinuous phase is known as reinforcement and provides support and strength to the composite, whereas the continuous phase is known as matrix and holds the reinforcement together. In the previous decade, many high-performance fibers such as carbon, boron, graphite, and Kevlar have developed as a new generation of high-performance materials. Carbon fibers have better strength and stiffness than other fibers but have limited extensibility and low damage tolerance, whereas Kevlar fibers offer a higher degree of toughness and damage tolerance and its hybridization might be a good solution to carbon fiber limits. (MilonHossain[1,*] , M A Khan [2] , R A Khan [3] , Md. Abu BakarSiddiquee[4] and Tauhidul Islam [5] 2015)

The utilization of two or more unique materials together in a single matrix is referred to as hybrid composites. Hybrid composites have several unique characteristics that can be exploited to satisfy the specific needs of the construction under consideration. In comparison to monolithic composites, hybrid composites have provided technical improvements because to their better-balanced features. Balanced strength and stiffness, balanced bending and membrane mechanical properties, balanced thermal distortion stability, reduced weight and/or cost, improved fatigue resistance, reduced notch sensitivity, improved fracture toughness and/or crack arresting properties, and improved impact resistance are some of the advantages of hybrid composites over monolithic composites (Md. MilonHossain[1,*] , M A Khan [2] , R A Khan [3] , Md. Abu BakarSiddiquee[4] and Tauhidul Islam [5] 2015).

There are various types of hybrid composites identified as:

1. Interplay- where two or more constituent types of fiber are reinforced together either regular or random orientation
2. Sandwich hybrids or core-shell- in this type of composites one material is sandwiched between two layers of another
3. Interplay or laminated, where alternate layers of the two (or more) materials are stacked in a regular manner
4. Intimately mixed hybrids, in this type of composites reinforcing fibers are mixed randomly to ensure homogenous distribution
5. Other kinds or super-hybrid which are based on organic polymer and metal matrix.

Matrix materials play crucial role in fabricating composite materials. Load is transferred to reinforcement by matrix and provide the composites toughness, damage tolerance and impact resistance. Thermo set and thermoplastics are two distinct types of matrixes commonly used in polymeric composite materials. Thermo set matrices include PE, vinyl esters, epoxies, bismaleimides, cyan ate esters, polyamides, and phenolicetc (Md. MilonHossain[1,*] , M A Khan [2] , R A Khan [3] , Md. Abu BakarSiddiquee[4] and Tauhidul Islam [5] 2015).

On other hand Polyetheretherketone (PEEK), polyetherketoneketone (PEKK), polyphenylene sulfide(PPS), and PP are semi crystalline thermoplastics, while polyetherimide (PEI) is an amorphous thermoplastic matrix. Matrices influences the ultimate thermo-mechanical characteristics of the composite and also have a major influence on the interlaminar shear, and on the in-plane shear properties (Md. MilonHossain[1,*] , M A Khan [2] , R A Khan [3] , Md. Abu BakarSiddiquee[4] and Tauhidul Islam [5] 2015)

While thermoplastic matrices dominate the short fiber reinforced composite; the long and continuous fiber reinforced composites are dominated by thermo set matrices due to their suitability in impregnating the fiber reinforcement. Although they are used very widely but they show some shortcoming such as low storage temperature, complex and long curing process etc. result in defective manufacturing process. In this situation, thermoplastic matrices may be a good option but higher processing temperature isrequired. Hence viscosities make the processing difficult. In addition, cost of thermoplastic matrices is higher than thermo set. In this regard variation in matrices affects the properties and processing of composite materials directly.(Md. MilonHossain[1,*] , M A Khan [2] , R A Khan [3] , Md. Abu BakarSiddiquee[4] and Tauhidul Islam [5] 2015)

Drive shafts must be very strong and lightweight to improve the overall performance of the vehicle; as a result, the automotive industry is investigating composite materials to achieve weight reduction without noticeably lowering vehicle quality and dependability. To completely do away with the assembly linking two-piece steel drive shafts, it is feasible to produce a composite drive shaft in one piece. Other advantages of the composite drive shaft include less weight, quieter operation, and decreased vibration. Because hollow round shafts can support more specific weight than solid circular ones, they are frequently utilized(S V Gopals Krishna, B V Subrahmanyam, 2013).

While the stress distribution is zero at the center and highest at the outside surface for solid shafts, the stress variance is minimal for hollow circular shafts(Carloni, 2014). Hollow circular shafts are preferred to solid circular shafts because the material at the center is not fully used in solid shafts. Long power driver shaft (propeller shaft) applications seem to be best suited for advanced composite materials. It is possible to modify their elastic characteristics to improve the torque and rotational speed that they can support. It is well recognized that weight reduction is one of the most efficient ways to achieve energy saving, one of the key goals in vehicle design(Zheng et al., 2021).

A propeller shaft is made up of an assembly of one or more tube shafts connected by universal, constant velocity, or flexible joints. The number of tube sections and the joints are both influenced by the separation between the gearbox and the axle. In certain four-wheelers, the front wheels are pushed by a second propeller shaft, which is identical to rear wheel drive for the rear wheels. (ParshuramD,SunilMangsetty 2013)

In this case the second propeller shaft is replaced between a transfer gear box and the front axle. Hence, it can be observed that a drive shaft is one of the most important components, which is responsible for the actual movement of the vehicle once the motion is produced in the engine. The designing of such a critical component is usually stringent, as any fracture in this part could lead to as catastrophic failure of the vehicle when it is in motion (Parshuram D[1],Sunil Mangsetty[2] 2013).

2.2 Theoretical background of design analysis of propeller shaft

Muni Kishore et al.(2016) examined the characteristics of the propeller shaft to replace its material with composite material and suitability of material by evaluating and comparing stress distribution and deformation within the shaft to replace the steel drive shaft with a piece of E-glass/epoxy and E-carbon/epoxy using the properties of the materials. Commercial programs like CATIA and ANSYS are used for modeling and analysis. When weight savings, deformation, shear stress generated, and frequency are taken into consideration, it is clear that the E-glass/Epoxy composite among the three materials investigated has the most promising qualities to serve as a steel substitute. They only looked at the relationship between stress distribution and propeller shaft deformation, and they only used one type of composite material. They failed to take into account factors such as fatigue analysis, corrosion resistance, vibrate analysis, dynamic analysis, etc. and also used composite material E-glass/epoxy has low strength when we compare other epoxy and also carbon/epoxy has low impact resistance but in this thesis I use hybrid composite material carbon-kevlar epoxy which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to carbon and E-glass composite material propeller shafts.which was one of the gaps that this thesis study used to identify.

The design and analysis of a composite driveshaft for power transmission have been researched by Mohan et al. (2017). In order to achieve more efficiency, it also compares the convectional two-piece drive shafts with various composite materials including S-glass, E-glass, and carbon epoxy. Here, the goal is to improve mechanical characteristics including deflection, weight reduction, shaft natural frequencies, and stress under applied load. It claims that due to the composite material's better specific stiffness and strength, replacing convectional steel buildings with them offers various benefits. Additionally, it used the CATIA and ANSYS tools for design and analysis. This study claims that replacing the convectional driveshaft reduces the weight of the car, improving fuel efficiency. S-Glass epoxy composite material may be employed as an alternative propeller shaft material, it has been determined.

They employed a single composite material and exclusively studied deflection, weight reduction, and natural frequencies of the propeller shaft. They didn't analyze fatigue analysis ,vibration analysis ,dynamic analysis or corrosion resistance, The used composite material E-glass/epoxy has low rigid when we compare other epoxy, S-glass /epoxy has low rigid and also carbon/epoxy has low impact resistance but in this thesis hybrid composite material carbon-Kevlar epoxy is used which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to carbon and E-glass composite material propeller shafts. which is regarded a research gap in this thesis study and may be filled utilizing two epoxy hybrid composites made of carbon-Kevlar composite material .

S. Thennarasan et al., (2017) investigated the fundamental design requirements of torsion strength, torsion buckling, and bending natural frequency. It takes four examples by winding different layers of composite materials and conducting an experiment in which the steel plays a part in transmitting the needed torque, whilst the E-glass epoxy composite enhances the bending and natural frequency. According to the findings of this study, increasing the number of layers improves maximum static torsion by 66% for +45/-453s laminates over pure aluminum and reduces mass by 42% when compared to steel driveshaft. A one-piece hybrid composite complete driveshaft is best studied using finite element ANSYS software, and the findings show that employing composite materials may save weight by up to 30% when compared to structural steel. According to the article, E-glass/epoxy composite has better weight-saving, deformation, shear stress induced, and resonant frequency qualities than steel. Limited work is done on the basis of increased specific stiffness, fatigue analysis, corrosion resistance; vibration analysis, dynamic analysis utilizing two different epoxy hybrids, composite material made of Carbon-Kevlar epoxy hybrid composite which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to previously composite material propeller shafts don from single materials. and so on. they use E-glass/epoxy has Lowest Strength to Weight Ratio, Comparably Lower Strengths than Carbon Fiber, and Less Rigid than Carbon Fiber and Kevlar epoxy.

D. Ganesh and K.H. Munde (2017) explored how the fatigue life of the generated composite driveshaft is assessed and how design adjustment improves propeller shaft performance. The FEA study of a composite shaft with varied degrees of glass fiber orientation is investigated in this work. It entails modeling the shaft with CATIA software.

Hyper mesh will be used for meshing and boundary condition application and ANSYS will be used for fatigue analysis of the composite shaft. When compared to conventional material, the composite material did not break after 22100 cycles after applying a moderate stress. They only considered fatigue life and used a single composite material they use E-glass/epoxy has Lowest Strength to Weight Ratio, Comparably Lower Strengths than Carbon Fiber, and Less Rigid than Carbon Fiber and Kevlar epoxy. However, the focus of this thesis is on the enhanced specific stiffness, corrosion resistance, vibration analysis, dynamic analysis, and other features of two epoxy hybrid composites constructed from different epoxy which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to previously composite material propeller shafts don from single materials. Based on this knowledge gap, it is vital to design and test propeller shafts made of carbon-Kevlar epoxy hybrid composites rather than traditional propeller shafts.

R. Srinivasa Moorthy et al., (2017) investigated the replacement of the traditionally used steel driveshaft of vehicles with a suitable composite driveshaft using different fiber orientations such as 00, 45, and 900 for the composite ply orientations, each with its own set of advantages. Torsion strength, bending natural frequency, and torsion buckling were designed and analyzed for their appropriateness after being made from composites of carbon/epoxy and Kevlar/epoxy by comparing them with the convectional drive shaft under the same design constraints, and they discovered that composite material takes precedence, so it is recommended for use. This study clearly shows a weight reduction percentage of 89.756% when built from carbon/epoxy driveshaft with nearly half the wall thickness of convectional steel driveshaft. They only investigated the torsion strength, bending natural frequency, and torsion buckling of a propeller shaft made of a single composite material carbon/epoxy has low impact resistance and l and also Kevlar/epoxy Low Compressive Strengths. However, not only is weight decreased in this thesis study, but also fatigue analysis, corrosion resistance, vibration analysis, dynamic analysis, and so on, by applying two epoxy hybrid composites constructed from different epoxy. According to the study gap researcher, it is necessary to design and test propeller shafts made of carbon-Kevlar epoxy hybrid composite rather than traditional propeller composite material made of Carbon-Kevlar epoxy hybrid composite which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to previously composite material propeller shafts don from single materials.

Esmael Adem et al. (2015)investigated the characterization of E-glass/Epoxy and E-glass-/Polyester Composite Material. The compressive, shear, and flexural properties of the E-glass/epoxy composite improve as the strain rate increases, but the tensile strength decreases. He just used one composite material they use E-glass/epoxy has Lowest Strength to Weight Ratio, Comparably Lower Strengths than Carbon Fiber, and Less Rigid than Carbon Fiber. However, based on this research gap, two epoxy hybrid composites derived from different epoxy composite material made of Carbon-Kevlar epoxy hybrid composite which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to previously composite material propeller shafts don from single materials have been used to create and evaluate propeller shafts.

Hussain and Vino (2015) explored the substitution of convectional driveshaft material with several alternative composite materials such as E-glass/epoxy, which resulted in a considerable weight reduction of 28% when compared to structural steel shaft. Considering the weight savings, deformation, shear stress created, and resonant frequencies, it is obvious that the E-Glass/Epoxy composite has the most promising properties for use as a steel alternative of the two materials studied. they use E-glass/epoxy has Lowest Strength to Weight Ratio, Comparably Lower Strengths than Carbon Fiber, and Less Rigid than Carbon Fiber and Kevlar. However, minimal research has been conducted with an emphasis on enhanced specific stiffness, fatigue analysis, corrosion resistance, vibration analysis, and dynamic analysis, among other things, using two separate epoxy hybrids composite material made of Carbon-Kevlar epoxy hybrid composite which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to previously composite material propeller shafts don from single materials.

Asmamaw (2012) investigated a one-piece composite shaft optimized for E-Glass/Epoxy composites using Finite Element Analysis Software, with the goal of minimizing shaft weight while meeting constraints such as torque transmission, critical buckling torque capacity, and bending natural frequency. He investigated changing the diameter and length and came to the conclusion that selecting the lowest outer radius can save a large amount of bulk.

In contrast, he simply used one composite material. he use E-glass/epoxy has Lowest Strength to Weight Ratio, Comparably Lower Strengths than Carbon Fiber, and Less Rigid than Carbon Fiber In the thesis research, however, a carbon-Kevlar epoxy hybrid composite substitutes a regular propeller shaft with a composite propeller shaft built of a different epoxy composite material made of Carbon-Kevlar epoxy hybrid composite which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to previously composite material propeller shafts don from single materials. This study focuses not only on comparing torque transmission, critical buckling torque capacity, and bending natural frequency propeller shafts, but also on greater specific stiffness, fatigue analysis, corrosion resistance, vibrating analysis, dynamic analysis, and other factors.

Lelisa (2019) researched the design and analysis of a composite driveshaft for power transmission in order to achieve maximum efficiency, weight reduction, greater comfort, noise reduction, and vehicle part complexity. This thesis is concerned with the replacement of a two-piece steel drive shaft with a composite material made of E-glass fiber/epoxy in one piece, as well as the experimental investigation of mechanical properties of composite material, such as tensile, bending, and compression strength tests, by preparing specimens of composite shaft according to ASTM using hand lay-up technique. He uses E-glass/epoxy has Lowest Strength to Weight Ratio, Comparably Lower Strengths than Carbon Fiber, and Less Rigid than Carbon Fiber. The limited work is done with a focus on enhanced specific stiffness, fatigue analysis, corrosion resistance, vibration analysis, and dynamic analysis, among other things, using two distinct epoxy hybrids composite material made of Carbon-Kevlar epoxy hybrid composite which have high strength-to-weight ratio, high conformability, stiffness while able to absorb energy, dimensional stability, have high impact resistance and have fatigue resistance when compare to previously composite material propeller shafts don from single materials

2.3 Design criteria for propeller shaft

Basic terms and mathematical relations which are used while designing a propeller shaft are discussed as follows.

- The shaft needs to withstand critical torsional buckling (T_{cr}) such that T_{cr}> T.
- The minimum bending natural frequency of the shaft (f_{nb} (min)) = 80 Hz.

2.3.2Moment Inertia of Composite Hollow Shaft

The general formula for calculating the moment of inertia of a hybrid carbon-Kevlar/epoxy resin composite propeller shaft is as follows(Nadaf & Raikar, 2017):

$$I = \frac{\pi}{64}\left[d_o^4 - d_i^4\right] \tag{2.1}$$

Where ***I***=Moment of inertia and $\boldsymbol{d_o}$=outer diameter of composite propeller shaft

2.3.3Polar Moment of Inertia Composite Propeller Shaft

General formula for calculating the polar moment of inertia of a hybrid carbon-Kevlar/epoxy resin The composite propeller shaft is as follows(Nadaf & Raikar, 2017):

$$J = 2I \tag{2.2}$$

Where ***I***=Moment of inertia

2.3.4Maximum Shear Strain Composite Propeller Shaft

The general formula for calculating the maximum shear strain of a hybrid carbon-Kevlar/epoxy resin composite propeller shaft is as follows(Nadaf & Raikar, 2017):

$$\phi = \frac{TL}{GJ} \tag{2.3}$$

Where ***T***= Ultimate Torque,***J***= Polar moment of inertia and***G***= Shear modulus

2.3.5Total Deformation Composite Propeller Shaft

The total deformation of a hybrid carbon-Kevlar/epoxy resin composite propeller shaft was calculated using general formula.

$$\delta = \phi \times l \tag{2.4}$$

Where ∅= Maximum shear strain

ℓ=length of shaft

2.3.6Maximum Shear Stress, Composite Propeller Shaft

The maximum shear stress of a hybrid carbon-Kevlar/epoxy resin composite propeller shaft was calculated using general formula(Salaisivabalan & Natarajan, 2016).

$$\tau = \frac{Td_o}{J} \tag{2.5}$$

Where:***T***= Ultimate torque, d_o=Outer diameter &***J***= Polar moment of inertia

2.3.7Torque Transmission Capacity of Propeller shaft(Nadaf & Raikar, 2017)

$$\frac{\tau}{r} = \frac{G\theta}{l} = \frac{T}{J} \tag{2.6}$$

For hollowshaft(S. Mohan & M. Vinoth, 2016)

$$T = \frac{\pi\tau}{16}(d_o^4 - d_i^4) \tag{2.7}$$

For hollow *s*haft (S. Mohan& M. Vinoth 2016)

$$J = \frac{\pi}{32}(d_o^4 - d_i^4) \tag{2.8}$$

The maximum torsional strength of the shaft is calculated by using the following equation from equation (3.6).

Where: G = Shear Modulus in **N/m²**.

ϴ = angle of twist in **radians**.

l = length of shaft in **m**.

Thus, the torsional strength of the shaft will be calculated by(Nadaf & Raikar, 2017),

$$\frac{\tau}{r} = \frac{G\theta}{l} = \frac{T}{J} \tag{2.9}$$

2.3.8Torsional Buckling Capacity of the propeller Shaft (T_{cr})

As the shaft is designed based on static torque capacity, it is needed to calculate the buckling torque to verify shaft is safe under buckling with current torque following equation gives us value of torsional buckling capacity of shaft (Timoshenko & Gere, 1963).

$$\frac{1}{\sqrt{1-\mu^2}} \times \frac{L^2 t}{(2r)^3} > 5.5$$

Where, r is mean radius

$$r = \frac{r_o + r_i}{2} \tag{2.10}$$

For long shaft, the critical stress is given by,

$$\tau_{cr} = \frac{E}{3\sqrt{2} \times (1-\mu^2)^{3/4}} \times \left(\frac{t}{r}\right)^{\frac{3}{2}} \tag{2.11}$$

The relation between the torsional Buckling Capacity and critical stress is given by,

$$T_{cr} = \tau_{cr} \times 2\pi r^2 t$$

2.3.9 Bending Natural Frequency

The shaft is considered as simply supported beam undergoing transverse vibration or can be idealized as a pinned-pinned beam. Natural frequency can be found using the Bernoulli's-Euler theories. It neglects both transverse shear deformation as well as rotary inertia effects. Natural frequency based on the Bernoulli-Euler beam theory is given by (Rao, 2002).

$$f_{nb} = \frac{\pi P^2}{2l^2}\sqrt{EI_x/m_1} \tag{2.12}$$

Where: f_{nb} = natural frequency based on Bernoulli Euler theory, ***HZ***

P =1, first natural frequency

r = mean radius of shaft

I_x =Area moment of inertia in x direction in ***m^4***

m_1 = mass per unit length in ***kg/m***

l = length of the shaft in ***m***

E =Young's modulus

The moment of inertia of hollow shaft is given by(Nadaf & Raikar, 2017)

$$I_X = \frac{\pi}{64}(d_o^4 - d_i^4) \tag{2.13}$$

The mass per unit length of the shaft is given by(Nadaf & Raikar, 2017)

$$m_1 = \rho(\pi/4)(d_o^2 - d_i^2) \tag{2.14}$$

Where: ρ =Density of Mechanical properties of carbon-kevlar epoxy

d_o=Outer diameter shaft

d_i=Inner diameter of shaft

Therefore, upon substitution of equation (3.9) and (3.10) values in equation (3.8) bending natural frequency can be obtained. Here, the fundamental bending natural frequency of steel shaft is greater than the minimum natural frequency of the shaft assumed that is 80Hz.

2.3.10 Hooke's Law

Generalized Hooke's law for orthotropic material is given by:

$$\sigma = [Q]\{\varepsilon\} \tag{2.15}$$

Where: [Q] = material stiffness matrix.

σ = stresses and

ε = strains in material direction

2.4 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 (Bhajantri et al., 2014)
- 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. Here is prove that a hollow shaft is stronger and stiffer than a solid shaft of same material, length and weight

Let,D: Diameter of solid shaft

D_1: Outer diameter of hollow shaft

D_2: Inner radius of hollow shaft

For solid shaft, torque resisted is:

Since material, weight and length are same, Weight of the solid shaft is the same as weight of the hollow shaft

$$\frac{\pi D_2 L\rho}{4} = \frac{(D_1^2 - D_2^2)L\rho}{4} \tag{2.16}$$

$$D^2 = D_1^2 - D_2^2$$

*For solid shaft, torque resisted**(T_S)** is,*

$$\mathrm{T_S} = \frac{\mathrm{J_S} \times \tau}{\mathrm{R}} = \frac{(\pi \mathrm{D}^4/32)\tau}{\mathrm{D}/2} = \frac{\tau\pi \mathrm{D}^3}{16} \tag{2.17}$$

For hollow shaft, torque resisted*(T_H)* is:

$$\mathrm{T_H} = \frac{\tau\pi(\mathrm{D}_1^4 - \mathrm{D}_2^4)}{16D_1} \tag{2.18}$$

Therefore, the ratio of hollow shaft torque resisted to solid shaft torque resistedwill be

$$\frac{\mathrm{T_H}}{\mathrm{T_S}} = \frac{\mathrm{D}_1^4 - \mathrm{D}_2^4}{\mathrm{D}^3 \times \mathrm{D}_1} \tag{2.19}$$

Substituting equ (3.13 & 3.14) in the above equation proves, hollow shafts are stronger and stiffer than solid shafts of same material, length and weight. Hence hollow shaft is used for the design purpose depending on its both stiffness and strength.

2.4.1 Torsional strength

Since the primary load on a drive shaft is torsion, the maximum shear stress (τ_{max}) at the outer diameter (d_o) and inner diameter (d_i) of the shaft is given by(Nadaf & Raikar, 2017);

$$\frac{\tau}{r} = \frac{G\theta}{l} \tag{2.20}$$

$$\frac{TJ}{\tau} = G \times \theta \times r \tag{2.21}$$

T is the maximum torque applied in N-m, $\boldsymbol{d_o}$ and $\boldsymbol{d_i}$ are outer and inner diameters of the shaft respectively in m. Factor of safety (FOS).

Thus, the wall thickness of the hollow steel shaft:

$$t = r_o - r_i \tag{2.22}$$

$$\frac{\tau_{max}}{FOS} = \frac{T}{2\pi r^2 t} \tag{2.23}$$

2.4.2 Torsional critical buckling (T_{cr}) for composite propeller shaft

When a hollow shaft is subjected to torsion for a certain amount of torsional load instability occurs.It is called torsional buckling load. Considering the hollow composite shaft as an isotropic cylindrical shell, the buckling torque is given by Equation

$$T_{cr} = (2\pi r^2 t)(0.272)\left(E_X E_y^3\right)^{0.25}(t/r)^{1.54} \tag{2.24}$$

Where: t = the overall wall thickness

r = the mean radius

E_x and E_y are the young's moduli in 'x' and 'y' directions respectively

And $T_{cr} > T$ condition satisfied

2.4.3 Bending natural frequency

The drive shaft is idealized as a pinned-pinned beam. The lowest natural frequency is calculated using the equation(Nadaf & Raikar, 2017)

$$f_{nb} = \frac{\pi}{2}\sqrt{gE_x I_x / WL^4} \tag{2.25}$$

$$f_{nb} = \frac{\pi}{2}\sqrt{E_x I_x / m_1 L^4} \tag{2.26}$$

Here, the moment of inertia of hollow shaft is given by,

$$I_X = \frac{\pi}{64}(d_o^4 - d_i^4) \tag{2.27}$$

The mass per unit length of the shaft is given by (Nadaf & Raikar, 2017)

$$m_1 = \rho(\pi/4)(d_o^2 - d_i^2) \tag{2.28}$$

Bending natural frequency of carbon-kevlar/Epoxy Propeller Shaft

Where: f_{nb} = the lowest natural frequency in hertz.

W/g= the mass per unit length,

I= the moment of inertia and

L = the length of the Propeller shaft

2.4.4 Specification of The Propeller Shaft Problem

To prevent whirling vibration, the propeller shaft's basic natural bending frequency should be higher than 6,500 rpm and the torque transmission capability of the drive shaft for automobiles, small trucks, and vans should be greater than 3,500 Nm. Due to space restrictions, the drive shaft's outside diameter shouldn't be larger than 100 mm. The transmission system's drive shaft has to be properly constructed to meet the parameters listed in Table 3.2 below(Nadaf & Raikar, 2017).

Table 2.1: Design requirements and specifications of existing model(Nadaf&Raikar, 2017)

Parameter	***Notation***	***Value***	***Unit***
Ultimate Torque	T_{max}	3500	Nm
Max. Speed of Shaft	N_{max}	6500	Rpm
Outer Diameter of a shaft	Do	90	Mm
Inner diameter of a shaft	Di	80	Mm
Length of shaft	L	1250	Mm

Because the drive shaft would be subjected to dynamic stress, natural frequency will be an important design issue; hence, dividing the drive shaft in two halves with an extra universal joint at its center would raise the driveshaft's fundamental natural frequency. Vibration is the most typical cause of drive shaft problems. The vibration problem can be solved by extending the shaft's diameter; however, this raises its strength beyond what is necessary for torque carrying while also increasing its inertia, which opposes the vehicle's acceleration and deceleration(Kumar et al., 2016).

Additionally important in automotive applications is weight reduction. Drive shafts made of composite materials will have a greater strength-to-weight ratio than metallic shafts. Additionally, composites have higher dimensional stability because to their enhanced corrosion resistance, substantial damping capability, and low coefficient of thermal expansion (Sivakandhan & Suresh Prabhu, 2012).

2.5 Finite Element Method (FEM)

Finite element analysis first developed in 1943 by R. courant; who used the Ritz method of numerical analysis and minimization of vibration calculated to approximate solution to vibration system shortly thereafter, a paper published in 1956 by turner, clough, Martine and top established a boarder definition of numerical analysis. The Finite Element Analysis (FEA) is a numerical method for solving problems of engineering and mathematical physics (Reddy et al., 2007). Useful for problems with complicated geometries, loadings, and material properties where analytic solutions cannot be obtained. In mathematics, the FEM is a numerical technique for finding approximate solutions to boundary value problems for partial differential equations. It uses subdivision of a whole problem domain into simpler parts, called finite elements, and variational methods from the calculus of variations to solve the problem by minimizing an associated error function (Reddy et al., 2007).

Finite element methods are numerical methods for approximating the solutions of mathematical models. Mathematical models are mathematical problems formulated so as to precisely state an idea of some aspect of physical reality (Hoe, 2015). A finite element method is characterized by a variational formulation, a discretization strategy, one or more solution algorithms and post-processing procedures, FEA uses a complex system of points called nodes which make a grid called a mesh.

This mesh is programmed to contain the material and structural properties which define how the structure will react to certain loading conditions. Examples of variational formulation are the Galerkin method, the discontinuous Galerkin method, mixed methods, etc. (Eddy & Angeelal, 2017).

- ✓ **Steps in Finite Element Implementation**

The solution of a general continuum problem by the finite element method always follows an orderly step-by-step process viz: Discretization of the Structure, Selection of proper interpolation or displacement model, Derivation of Element Stiffness Matrix and Load Vectors, Assemblage of Element Equations to Obtain the Overall Equilibrium Equations, Solution for the nodal displacement and Computation of Element Strains and Stresses for structural problems (Sharma et al., 2020).

2.5.1 Static Analysis

The representation of static system of equations, is expressed as (Kumar, 2021):

$$K\{D\} = \{F\} \tag{2.30}$$

There are multiple techniques to solve for global static displacement vector of all elements, $\{D\}$. The Gauss elimination or LU decomposition methods are commonly used for solving the above system of equations (Kumar, 2021).

For further analysis, the global displacement vector of element, {De}, is extracted from $\{D\}$, which is utilized to evaluate the local displacement vector of element, {de}, using the mathematical relation expressed below (Kumar, 2021).

$$\{d_e\} = T\{D_e\} \tag{2.31}$$

Finally, the static analysis provides the nodal stress, $\{\sigma_i\}$, calculated using $\{\epsilon_i\}$ from the expression given below.

$$\{\sigma_e\} = C\{\epsilon_e\} \tag{2.32}$$

2.5.2 Modal Analysis

Modal analysis is also known as normal mode analysis, eigenvalue analysis, or eigenvalue extraction. You can perform modal analysis in FEA to:

- ✓ Determine the natural frequency and mode shape the structure will represent.
- ✓ Check communication between systems
- ✓ Understand that BC has been applied correctly to the system.
- ✓ Predict the dynamic response of the system

After all, the natural frequency must not match the frequency of the applied periodic load, which will result in resonance, i.e., large deformation of the system or system failure. The natural frequency of a structure depends on the mass and boundary conditions associated with the structure, including its stiffness and its own weight (Abd Elsalam et al., 2017).

Modal analysis is a technique carried out using block lanczos method to obtain the modal parameters (Abd Elsalam et al., 2017):

1. ***Natural frequency***: the frequency at which the structure naturally tends to vibrate if subjected to a pulse.
2. ***Mode shape***: deformed shape of the structure will vibrate when excited at the specific natural frequency.

2.5.3 Harmonic Analysis

A harmonic analysis is used to determine the response of the structure under a steady-state sinusoidal (harmonic) loading at a given frequency from the modal analysis (Chittilla et al., 2013). Loads may be out of phase with one another, but the excitation is at a known frequency. For automobiles, the low frequency noise and vibration problems radiated into the car caused by the vibration of vehicle parts will not only seriously affect the ride comfort of the driver and passengers, but also have a great impact on the driving safety and service life of the vehicle (Hatti et al., 2021).

In order to obtain a more comfortable and safe driving and riding environment, and to improve the market competitiveness of vehicles, it is extremely important to control the low frequency noise radiated into the vehicle by the vibration of the automobile parts. The generation path of low-frequency noise radiated by automobile parts. The car is driving on rough roads, and the displacement excitation of the road is transmitted to the body vehicle parts through the tire–suspension system.

Excitation generates low-frequency vibration and radiates low-frequency structural noise into the car, causing serious discomfort to passengers and drivers (Liao et al., 2022).

A harmonic, or frequency-response, analysis considers loading at one frequency only. Loads may be out of phase with one another, but the excitation is at a known frequency. This procedure is not used for an arbitrary transient load (Wu et al., 2012).

To better understand a harmonic analysis, the general equation of motion is provided first (Aydin et al., 2021):

$$[M]\left\{\frac{d^2U}{dx^2}\right\} + [C]\left\{\frac{dU}{dx}\right\} + [K]\{U\} = \{F\}$$

Where, *[M]* is structural mass matrix, *[C]* is structural damping matrix, *[K]* is structural stiffness matrix, d^2U/dx^2 is nodal acceleration vector, dU/dx is nodal velocity vector, U is nodal displacement vector and F is applied load vector.

2.6 Research Gap

According to the findings of this study's literature review, almost all of the researchers used a single composite material. They did not uses hybrid composite which have balanced strength and stiffness, balanced bending and membrane mechanical properties, balanced thermal distortion stability, reduced weight and/or cost, improved fatigue resistance, reduced notch sensitivity, improved fracture toughness and/or crack arresting properties, and improved impact resistance when compare to single composite material (Radulović, 2020).

The goal of this study is to evaluate the fatigue, impact, and shock resistance of automobile propeller shafts using fatigue analyses, vibration analyses, and dynamic analyses of a two-piece of Tata 2027 DI Ex for use rural construction work or bad topography ,rough road etc. Propeller shaft composite material made of carbon-kevlar epoxy hybrid composite, which has a higher strength-to-weight ratio, high conformability, stiffness while absorbing energy, dimensional stability, high impact resistance, and fatigue resistance. Because hybrid composites are identified by the use of two or more distinct materials combined in a single matrix. Hybrid composites show some unique features that can be used to meet the exact requirements of the structure under consideration. Compared to monolithic composite hybrid composites have given technological advances because they provide more balanced properties.(Radulović, 2020).

To assess the strength and behavior of engineering structures, the finite element technique, a computer-based numerical approach, is utilized.

CHAPTER THREE

MATERIALS AND METHODOLOGY

3.1 Introduction

This chapter includes a list of the sources used in the study. The many phases of the techniques are also explained in this chapter. The most important section to look at when evaluating the overall quality of any research distribution product is the materials and methods section since it tells readers about the procedures, approaches, designs, and treatments employed in the study and makes it possible to replicate the findings. A detailed literature research was used to help choose the best working materials, learning strategies, and simulation software. The ideal selection of working materials was used in the design of this investigation.

3.1 Design for hybrid Carbon-Kevlar /epoxy resin composite propeller shaft

During the design of a propeller shaft the following assumptions were made in calculations:

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
6. The shaft is assumed to be acting in a vacuum.

3.2 Materials

The tools utilized in this investigation were a meter to measure the propeller shaft's original length, a caliper to measure its inner and outer diameter, a flash to capture soft cop pic material, and a digital camera. propeller shaft will be photographed in its entirety using a camera.

3.3 Methodology

- Computer Aided Three-Dimensional Interactive Application is referred to as CATIA. The program is much more than a simple CAD (Computer Aided Design) tool. This complete software set includes CAD, CAE (Computer-Aided Engineering), and CAM (Computer-Aided Manufacture). The propeller shaft diagram in this project was simulated using CATIA V5

ANSYS was the second piece of software utilized to assess the design outcomes. It did so by analyzing the analytical results of both steel and carbon-Kevlar epoxy composite propeller shafts.

This thesis work is focused with the design and analysis of a carbon-Kevlar/epoxy hybrid composite propeller shaft, which entails analyzing the techniques employed and the theories or ideas underlying them in order to create a strategy that is in line with the goals.The method solution for the problem is performed in the following six stages: defining dimensions, material selection, theoretical analysis, creating a solid model, finite element analysis, and comparative study.

- **Defining dimensions**: Taking the dimensions of the pre-existing propeller shaft of the car and using them for further calculation and analysis
- **Material selection**: selecting the feasible material for the propeller shaft of a car considering the required parameters These include conventional materials as well as different composite materials.
- **Theoretical Design**: Theoretical design is performed by using the basic concepts of strength of materials.
- **Creating a Solid Model**: A three-dimensional solid model of the shaft is created on the computer using CATIA 5 software. This 3D model is exported to ANSYS software for performing finite element analysis.
- **Finite Element Analysis:** There are three main steps, namely, pre-processing, solution, and post-processing.

In pre-processing (model definition), the geometric domain of the problem is defined, the element type(s) to be used, the material properties of the elements, the geometric properties of the elements (length, area, and the like), the element connectivity (the mesh of the model), the physical constraints (boundary conditions), and the loadings.In the solution phase, the computed results are then used to determine things like deformation and element stress, which are carried out commercially.Post-processing is to be carried out on the finite element model of the Carbon-Kevlar Epoxy composite drive shaft using ANSYS software.

Comparative study: weight reduced due to the use of composite drive shafts compared to steel shafts, also suggesting different possible materials depending upon the comparison of composites.

3.3.1 Data Collection

Data were gathered for this research from both primary and secondary data sources. Primary data Length of propeller shaft 125Mm,Outer diameter 90Mm and Inner diameter 80Mm direct measured from vehicle and secondary data book, Internet. The following strategies are employed in thesis research to meet the goals of the thesis:

- A relevant data was collected through literature review
- An actual data was collected by observing and measuring the specifically selected propeller shaft specification.
- Design and analyses of theoretical calculation of the steel Propeller shaft and hybrid Carbon-Kevlar epoxy propeller shaft is done.
- The 3D model ofthe existing and hybrid Carbon-Kevlar propeller shaft is done using CATIA.
- Finally, the analysis for the existing and the new propeller shaft is conducted byusing ANSYS

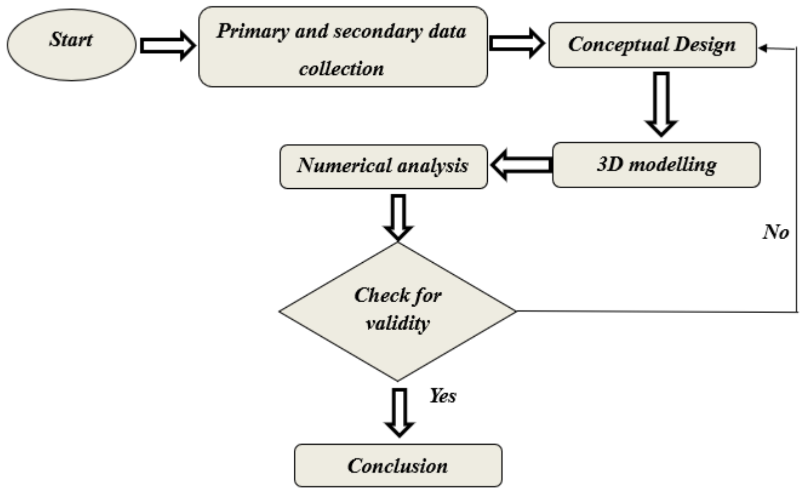


Figure 3.1:Block diagram of the methodology

3.4 Vehicle Specification

The design of the composite propeller shaft is done for Tata 207 DI Single Cabin Pickup which uses two-piece propeller shaft. According to the owners, when compared to other vehicles, the majority of this vehicle's propeller shafts suffer damage from corrosion, high fatigue, high bending, high impact, high vibration, high compressive loading, and severe shock loading.

Figure 3.2: Tata 207 DI Single Cabin Pickup

Tata 207 DI Single Cabin Pickup

Designed for urban as well as rural use, the 207 DI is packed with safety features that make for better stability and ease of handling. With the highest ground clearance in its category, and larger tires for more traction, the 207 DI is the ideal multi-terrain, multi-application vehicle which marries high performance with low maintenance requirements. Select a variant below for more details

Table 3.1: Specification of TATA 207 DI single cabin diesel

Driveline – Engine	
Model	TATA 207 DI
Type	Water-cooled, Direct injection, Diesel
Maximum Engine Output	65hp @ 2800 rpm
Maximum torque	180 Nm at 1500 – 2000 rpm
	2956cc
Performance	
Speed	100 kmph

Gradability	24%
Minimum turning circle dia. (mm)	15
GVW	2950
Capacity	

3.5 Material Properties of Carbon-Kevlar/Epoxy Hybrid Composite

Table 3.2:Material Properties of Carbon-Kevlar/Epoxy Hybrid Composite (Parshuram& Sunil, 2013)

Property	***Symbol***	***Units***	***Values***
Longitudinal Modulus	E11	GPa	123.6
Transverse Modulus	E22	GPa	7.4
Shear Modulus	G12	GPa	4.28
Poisson's Ratio	υ	----	0.328
Density	ρ	Kg/m^3	1470

CHAPTER FOUR

DESIGN, MODELING AND ANALYSIS OF THE PROPELLER SHAFT

4.1 Introduction

Modeling of the existing and the composite propeller shaft is carried out in CATIA V5 and its analysis is performed in ANSYS. Basic dimensions of the existing model are taken by direct measurement whereas the dimension of the composite propeller shaft is obtained from the analytical design which is included under this chapter. The analysis which is done on ANSYS are static structural, transient structural, modal analysis, harmonic analysis, rigid dynamics and random vibration. Each of the analysis systems are discussed in the result and discussion part.

4.2 Design of Hybrid Carbon-Kevlar /Epoxy Resin Composite Propeller Shaft

The following assumptions were made in calculations:

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. The shaft is assumed to be acting in a vacuum.

First, the fibers are selected to provide the best stiffness and strength beside cost consideration.

4.2.1 Moment inertia of composite hollow shaft

$$I = \frac{\pi}{64}\left[d_o^4 - d_i^4\right] \tag{4.1}$$

Where: I=Moment of inertia

d_o=outer diameter of propeller shaft

d_i=Inner diameter of propeller shaft

According to table 4.2 outer diameter and inner diameter of the shaft is taken as 90mm and 80mm respectively. So, substituting in the equation gives**,**

$$I = \frac{3.14}{64}[0.09^4 - 0.08^4] = 0.0000013m^4$$

The physical size of the hybrid carbon-Kevlar/epoxy resin composite propeller shaft is $0.0000013m^4$.

4.2.2 Polar Moment of inertia composite propeller shaft

$$J = 2I \tag{4.2}$$

$$J = 0.0000024\ m^4$$

4.2.3 Maximum shear strain composite propeller shaft

$$\phi = \frac{TL}{GJ} \tag{4.3}$$

Where: T= Ultimate Torque

J= Polar moment of inertia

G= Shear modulus

T=3500Mpa Ultimate Torque

G=4.28Gpa

Therefore, the maximum shear strain of the composite propeller shaft is 0.4259 m/m^4.

4.2.4 Total deformation composite propeller shaft

$$\delta = \emptyset \times l \tag{4.4}$$

Where: $\emptyset$= shear strain

ℓ= Length of shaft

ℓ= 1250mm

Therefore, the total deformation of the composite propeller shaft is 0.0156m.

4.2.5 Maximum shear stress, composite propeller shaft

$$\tau = \frac{Td_o}{J} \tag{4.5}$$

Where: T= Ultimate Torque

d_o=Outer diameter

J= Polar moment of inertia

Substituting in the equation,τ

$$\tau = \frac{3500 \times 0.09}{0.000008164m^4} = 131MPa$$

4.3 Design Criteria for Propeller Shaft

4.3.1 The maximum torsional strength

$$\frac{\mathrm{\tau}}{\mathrm{r}} = \frac{\mathrm{G\theta}}{\mathrm{l}} = \frac{\mathrm{T}}{\mathrm{J}} \tag{4.6}$$

$$\mathrm{J} = \frac{\pi}{32}[\mathrm{d_o^4} - \mathrm{d_i^4}] \tag{4.7}$$

$$\mathrm{J} = \frac{\pi}{32}[0.09^4 - 0.08^4] = 2.41 \times 10^{-6} m^4$$

And

$$r = \frac{r_0 + r_i}{2} \tag{4.8}$$

$$r = \frac{0.045 + 0.04}{2} = 0.0425m$$

The shear stress can be calculated as

$$\tau = \frac{G \times \theta \times r}{L} \tag{4.9}$$

Where: G = Shear Modulus in N/m^2

Ɵ = angle of twist in radians and L = length of shaft in m.

Now

$$\mathrm{\tau} = \frac{80 \times 10^9 \times 5\pi/180 \times 0.0425}{1.250} = 237.24 \times 10^6 N/m^2$$

And

$$\frac{\tau}{r} = \frac{T}{J} \tag{4.10}$$

$$T = \frac{237.24 \times 10^6 N/m^2 \times 2.41 \times 10^{-6} m^4}{0.0425m} = 13425.9Nm$$

4.3.2 Torsional Buckling Capacity of the Propeller Shaft (T_{cr})

If

$$\frac{1}{\sqrt{1-\mu^2}} \times \frac{L^2 t}{(2r)^3} > 5.5 \tag{4.11}$$

Where, r is mean radius

$$r = \frac{r_o + r_i}{2} = 0.0425m$$

And the thickness is

$$t = r_o - r_i = 0.005mm$$

Let substitute the values in the equation (4),

$$\frac{1}{\sqrt{1-(0.3)^2}} \times \frac{(1.25)^2(0.005)}{(2 \times 0.0425)^3} > 5.5$$

$$13.65 > 5.5 \;\; so\ it\ is\ long\ shaft$$

Therefore, it is called as long shaft otherwise it is called as short or medium shaft. For long shaft, the critical stress is given by,

$$\mathrm{\tau_{cr}} = \frac{\mathrm{E}}{3\sqrt{2} \times (1-\mu^2)^{3/4}} \times \left(\frac{\mathrm{t}}{\mathrm{r}}\right)^{\frac{3}{2}} \tag{4.12}$$

$$\mathrm{\tau_{cr}} = 2140 \times 10^6 \mathrm{N/m^2}$$

The relation between the torsional Buckling Capacity and critical stress is given by,

$$\mathrm{T_{cr}} = \mathrm{\tau_{cr}} \times 2\pi \mathrm{r^2 t} \tag{4.13}$$

$$\mathrm{T_{cr}} = 2140 \times 10^6 (2\pi)(0.0425)^2(0.005)$$

$$\mathrm{T_{cr}} = 119{,}935.77\mathrm{Nm}$$

Hence T_{cr}>T, the condition is satisfied. Therefore, **the design is safe.**

4.3.3 Torsional strength

$$\frac{\tau}{\mathrm{r}} = \frac{\mathrm{G\theta}}{\mathrm{l}} \tag{4.14}$$

$$\mathrm{T\,J}/\tau = \mathrm{G} \times \theta \times \mathrm{r} \tag{4.15}$$

$$\tau = \frac{370\mathrm{Mpa} \times 5\pi/180 \times 0.0425}{1.250}$$

$$\tau = 10.97 \times 10^6 \mathrm{N/m^2}$$

T is the maximum torque applied in Nm. d_o and d_i are outer and inner diameters of the shaft respectively in m. Factor of safety (FOS) of 6,

r =0.0425m

Thus, the wall thickness of the hollow steel shaft:

$$t = r_o - r_i \tag{4.16}$$

$$t = 0.045m - 0.04m = 0.005m = 5mm$$

And it is known that the maximum shear stress is 10.97Mpa.

$$\frac{\tau_{max}}{FOS} = \frac{T}{2\pi r^2 t} \tag{4.17}$$

τ_{max} = 10.97Mpa and for a factor of safety of 6

$$r^2 t = 4.44005 \times 10^{-6} m$$

Thus, t ≥ 1.06×10^{-3}m.Since the thickness of each ply is 0.13 mm, the number of plies used is

$$n = \frac{1.106 \times 10^{-3} m}{0.13 \times 10^{-3} m} \approx 8$$

4.3.4 Torsional critical buckling (T_{cr}) for composite propeller shaft

$$T_{cr} = (2\pi r^2 t)(0.272)(E_X E_y^3)^{0.25}(t/r)^{1.54} \tag{4.18}$$

$$T_{cr} = (2 \times 3.14 \times 0.042^2 \times 0.005)(0.272)(123600 \times 7400^3)^{0.25}(0.005/0.0425)^{1.54}$$

$$T_{cr} = 34228.12 Nm$$

Where: t = the overall wall thickness, r = the mean radius, and E_x and E_y are the young's modulus in 'x' and 'y' directions respectively. Since T_{cr}> T, the condition is satisfied.

4.3.5 Bending natural frequency

$$f_{nb} = \frac{\pi}{2}\sqrt{g E_x I_x / W L^4} \tag{4.19}$$

$$f_{nb} = \frac{\pi}{2}\sqrt{E_x I_x / m_1 L^4} \tag{4.20}$$

The moment of inertia,

$$\mathrm{I_X} = \frac{\pi}{64}(\mathrm{d_o^4} - \mathrm{d_i^4}) \tag{4.21}$$

$$\mathrm{I_X} = \frac{3.14}{64}(0.09^4 - 0.08^4) = 1.2 \times 10^{-6} m^4$$

The mass per unit length of the shaft is given by;

$$m_1 = \rho(\pi/4)(d_o^2 - d_i^2) \tag{4.22}$$

$$m_1 = (1470)(0.785)(0.09^4 - 0.08^4) = 1.96 kg/m$$

Thus, bending natural frequency will be

$$f_{nb} = \frac{\pi}{2}\sqrt{123.6 \times 10^9 \times 1.2 \times 10^{-6} / 1.96 \times (1.25)^4} = 177 Hz$$

Therefore, bending natural frequency of carbon-kevlar/Epoxy Propeller Shaft is 177Hz.

Where: f_{nb} = the lowest natural frequency in hertz.

W/g = the mass per unit length,

I= the moment of inertia and

L = the length of the Propeller shaft

Upon substitution, f_{nb}=177 Hz (> 80 Hz).Thus, the designed carbon-kevlar/epoxy composite propeller shaft meets all the requirements.

4.3.6 Mass of hybrid carbon-kevlar composite

$$m = m_1 \times L_m \tag{4.23}$$

4.3.7 Critical Speed of Shaft

The critical speed of the shaft (N_{cr}) and natural frequency (f_{nb}) are related by using equation

$$N_{cr} = 60f_{nb}$$

$$N_{cr} = 60 \times 177$$

$$N_{cr} = 10620\text{rpm}$$

Therefore, the critical speed of the shaft is 10620 rpm which is more than the maximum speed of the transmission system.

4.3.8 Fatigue Analysis of Composite Drive Shaft

- ✓ **Design of shaft against fatigue loading**

To find bending moment in the shaft, it is assumed that only the force due to self-weight of the shaft is acting on the shaft. It is acting at the centered the shaft.

$$P = mg \tag{4.24}$$

Central distance of propeller shaft is 1250mm and the maximum bending moment is 7500Nmm.

4.4 Modeling of the Propeller Shaft

There are three basic steps in formulating finite element analysis. These are pre-processing, solution and post processing. The pre-processing step includes: define the geometric domain of the problem, the material properties of the element, the geometric properties of the elements (Length, area, and the like), the element connectivity (mesh the model), the physical constraints (boundary condition) and the loading. The next step is solution. The computed results are then used to determine like deformation and element stress which carried out by commercial software. The final step is posting processing, the analysis and evaluation of the result is conducted in this step. Specific procedures of pre and post processing are different up on the program (M. Mahesh Babu and Dr.N. Ramesh (2018).

4.4.1 CAD Modeling of the Existing Model

In order to do the numerical analysis, the necessary dimension for the existing propeller shaft is taken from direct measurement. Then, using the dimension the existing model is prepared with full scale. The geometry is created using CATIA V5 and the entire analysis and the post-processing is performed in ANSYS Workbench. The detailed process involved in geometry creation, mesh generation and the strategy used for optimization are explained in the subsequent sections.

The existing propeller shafts is a two pieces of steel shaft. It has six main parts these are front yoke, rear yoke, slip yoke, three universal joints, two shafts and a center bearing. All of these parts are modeled and assembled to be used for the analysis.

- **Front and Rear Yoke**

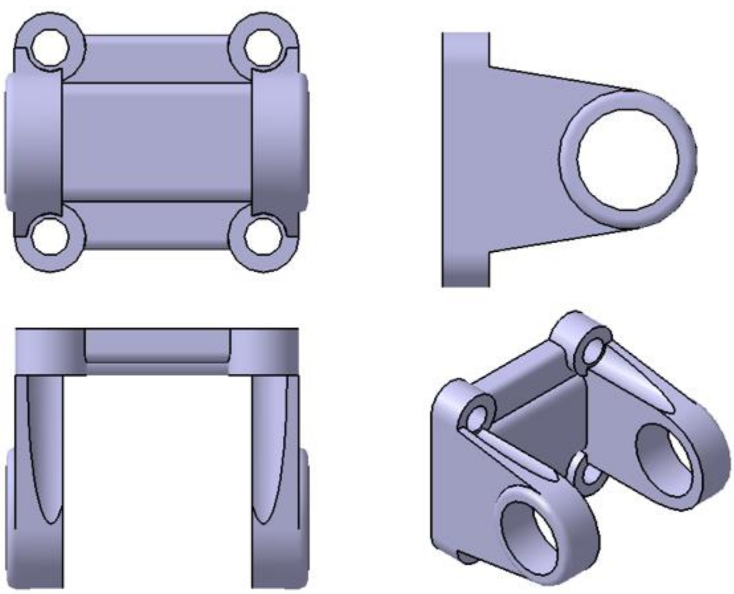


Figure 4.1:Different view of the yoke modeled using CATIA

- **Slip Yoke**

It is a Splined output shaft which allow for changes in driveline length by sliding in and out. The outer diameter is machined smooth, providing a bearing surface for the bushing and oil seal.

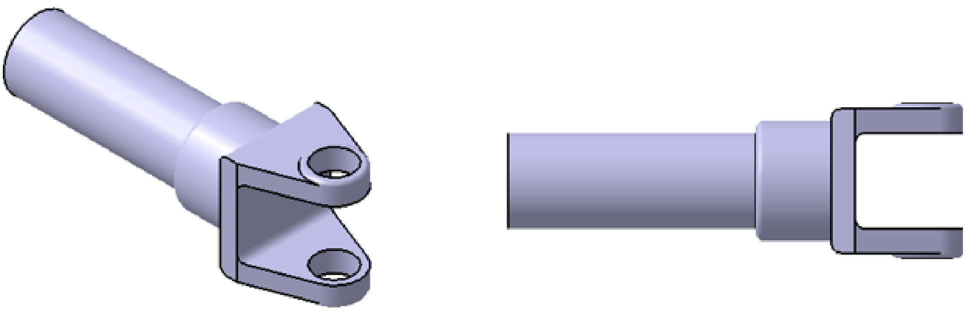
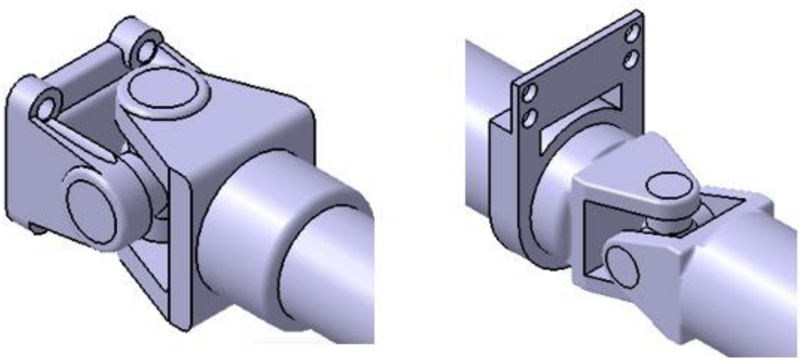


Figure 4.2:Different view of the yoke modeled using CATIA

- **Universal Joints**

It provides swivel connection capable of transferring a turning force between shafts at an angle to one another. It is made of two Y-shaped yokes, connected by a cross. Bearings on each end of the cross allow the yokes to swing into various angles while turning. Since the existing propeller shaft is a two-piece shaft three universal joints are used. Figure 4.3 (a & b) shows the universal joint used at the front and at the middle. The rear universal joint is the same as the front one.



(a) **(b)**

Figure 4.3:***(a)*** Universal joint at the front and ***(b)*** the intermediate universal joint

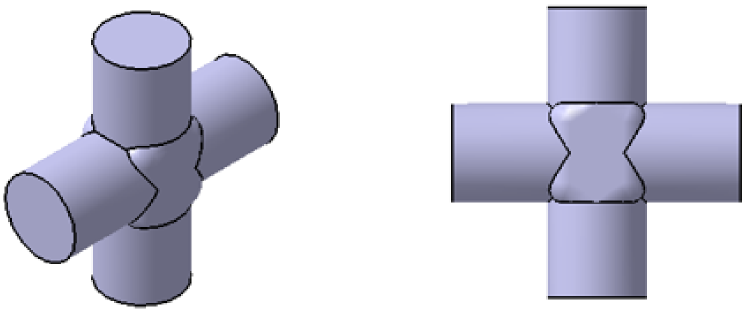


Figure 4.4:Cross or spider created using CATIA

- **Drive shafts**

It is a hollow steel tube with permanent yokes welded on each end. The existing propeller shaft is a two-piece shaft.

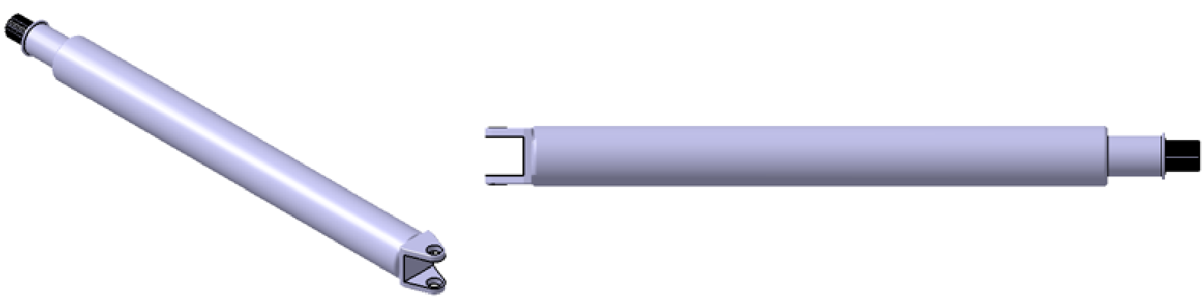


Figure 4.5:The front drive shaft created using CATIA

Figure shows the front drive shaft with a splined shaft at its front. The splined shaft integrated with the slip yoke to allow changes in driveline length by sliding in and out.

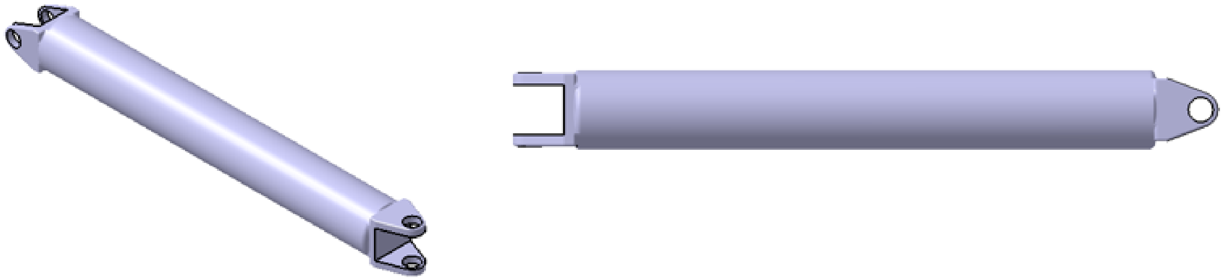


Figure 4.6:The middle drive shaft created using CATIA

- **Center Bearing**

Since the existing model is a two-piece shaft center bearing is needed to support the drive shaft.

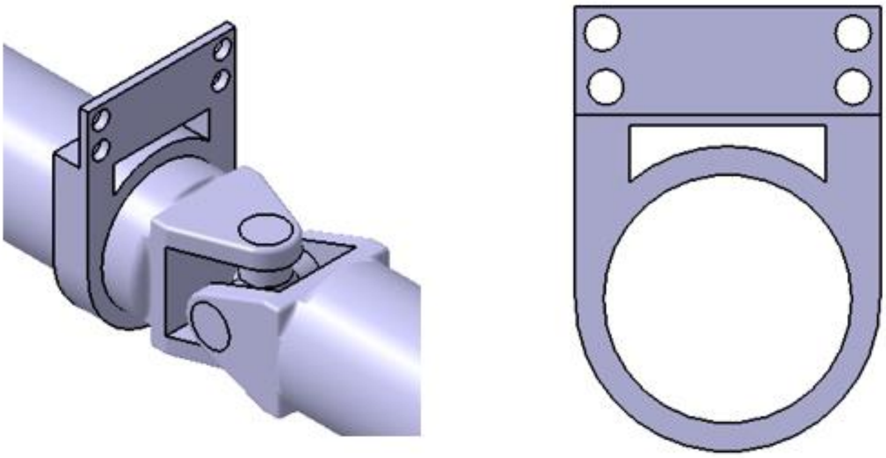


Figure 4.7:The middle drive shaft created using CATIA

Figure 4.8 shows the assembled regenerative shock absorber. the regenerative shock absorber is designed based on the load distributions and the results obtained from the coil spring analysis.

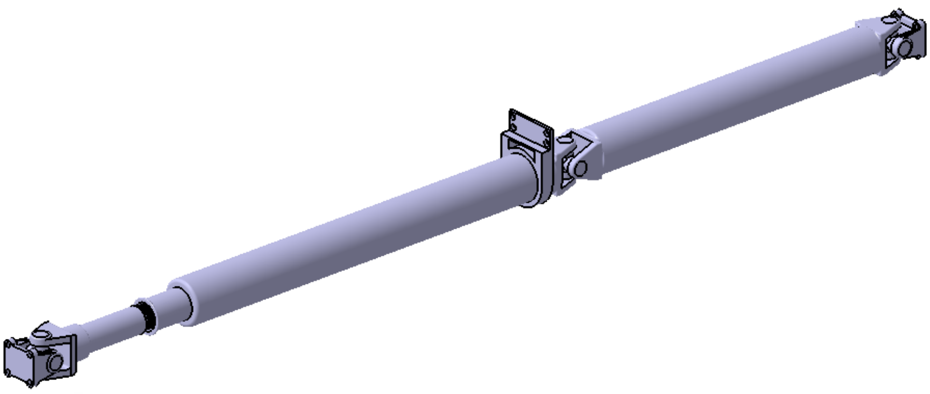


Figure 4.8:The existing drive shaft assembly created using CATIA

4.4.2 CAD Modeling of the Modified Model

Three modified models were prepared in order to consider all the necessary conditions and to select the better model. All the necessary dimension for the modified propeller shaft is taken from the results obtained in the design calculation. Then, using the dimension the model is prepared with full scale. Like that of the existing model, the modified geometry is created using CATIA V5.The three modified models are:

- ✓ Two-piece modified propeller shaft
- ✓ One-piece modified propeller shaft without center bearing
- ✓ One-piece modified propeller shaft with center bearing

Case 1: Two-piece modified propeller shaft

The two-piece modified propeller shaft looks like the existing model but they are different in terms of their thickness. There dimensions are given in Appendix.

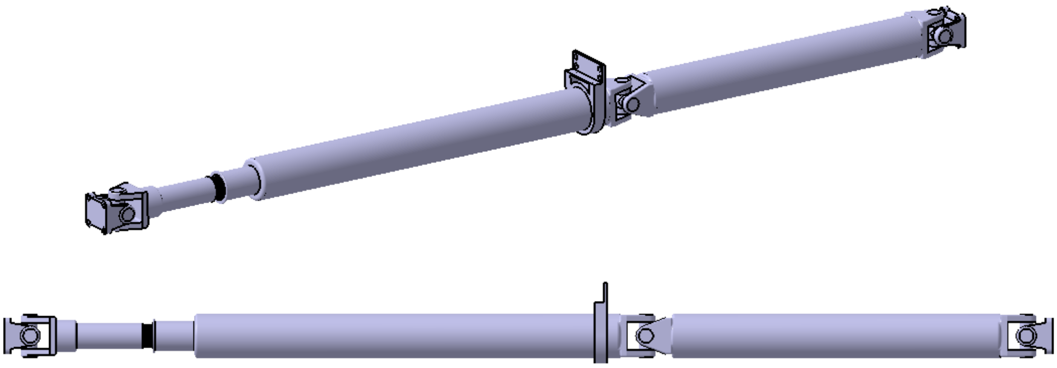


Figure 4.9:Two-piece modified propeller shaft

Case 2: One-piece propeller shaft without center bearing

As it is discussed in the literature review one piece propeller shafts are better in terms of vibration and weight. Therefore, that is why a single propeller shaft is used to compare its performance with the existing model.

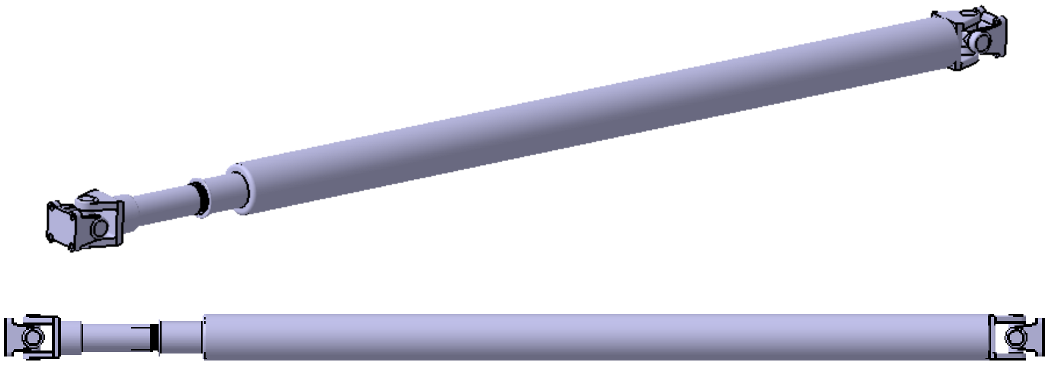


Figure 4.10: One-piece modified propeller shaft without center bearing

Case 3: One-piece propeller shaft with center bearing

In this case a center bearing is integrated on the one-piece shaft to analyze its effect on the performance of the propeller shaft.

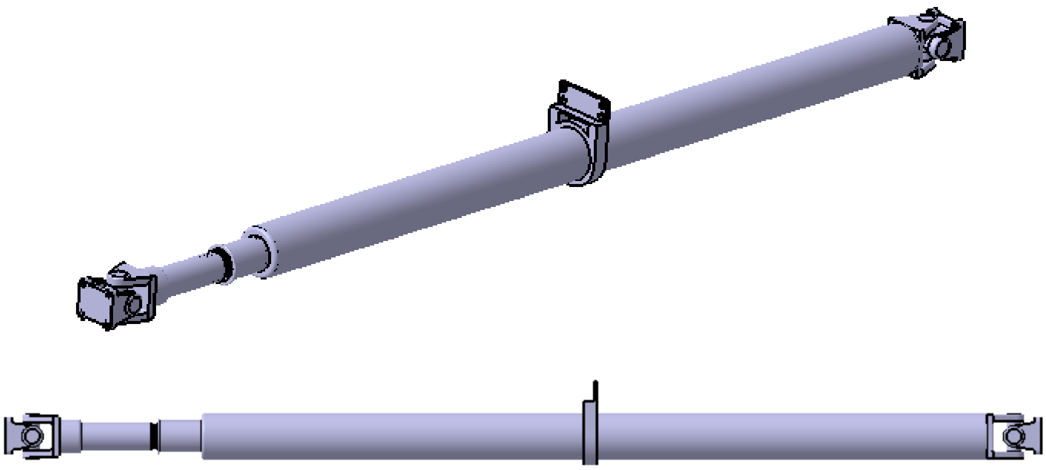


Figure 4.11: One-piece modified propeller shaft with center bearing

4.5 Analysis of the Propeller Shaft

4.5.1 Geometry Creation

The geometry of the existing and the modified propeller shafts are imported to ANSYS workbench in IGES format. Figure 4.12-4.16 shows the geometries after imported to ANSYS workbench in IGES format.

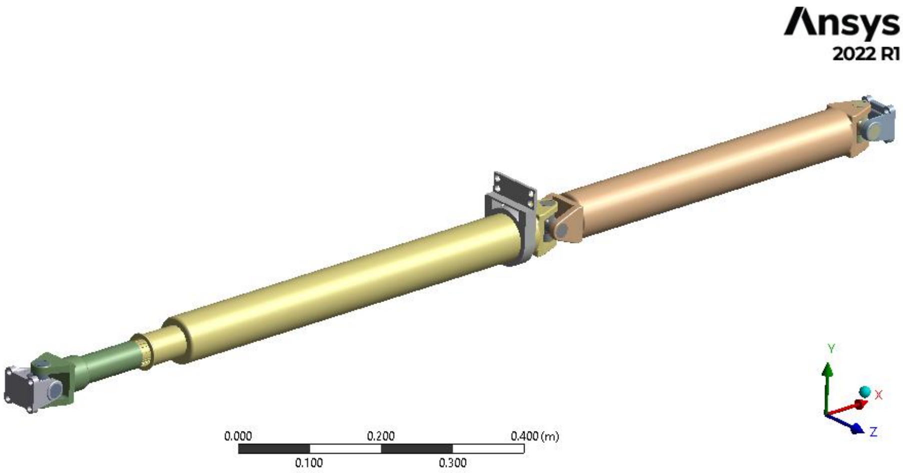


Figure 4.12: The Geometry of the existing shaft after imported to ANSYS

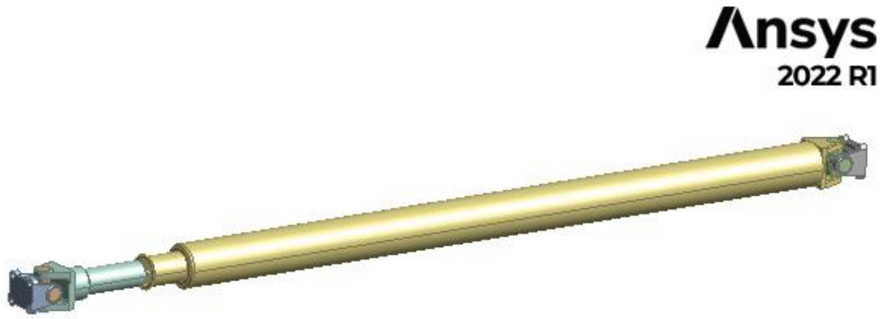


Figure 4.13: The Geometry of the one-piece modified shaft after imported to ANSYS

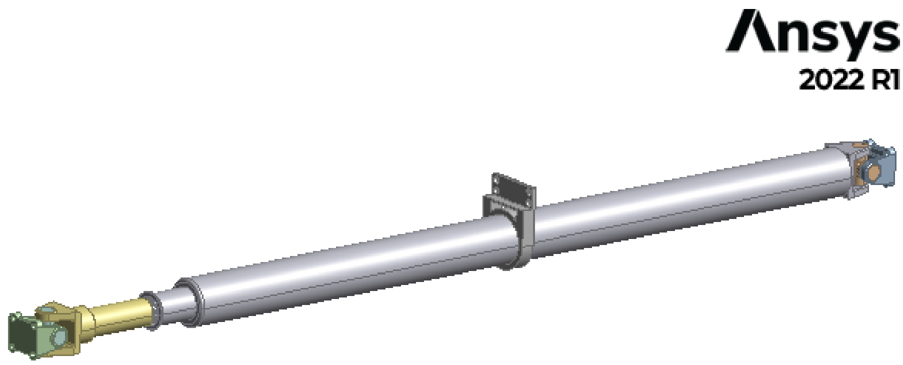


Figure 4.14: Geometry of the one-piece modified shaft with center bearing imported to ANSYS

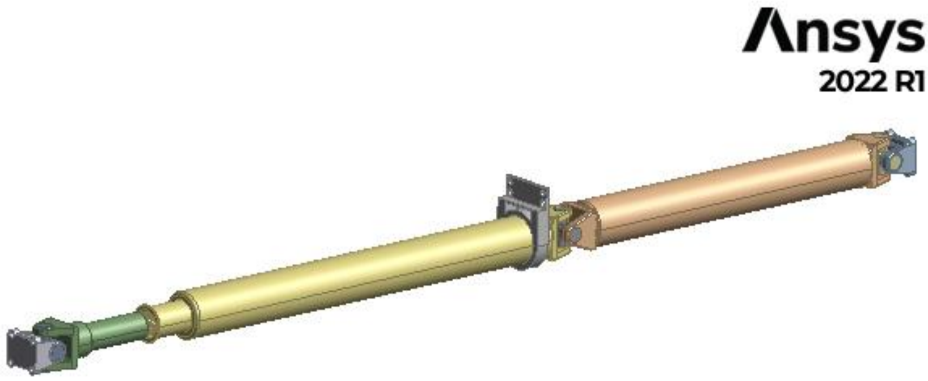


Figure 4.15: The Geometry of the two-piece modified shaft after imported to ANSYS

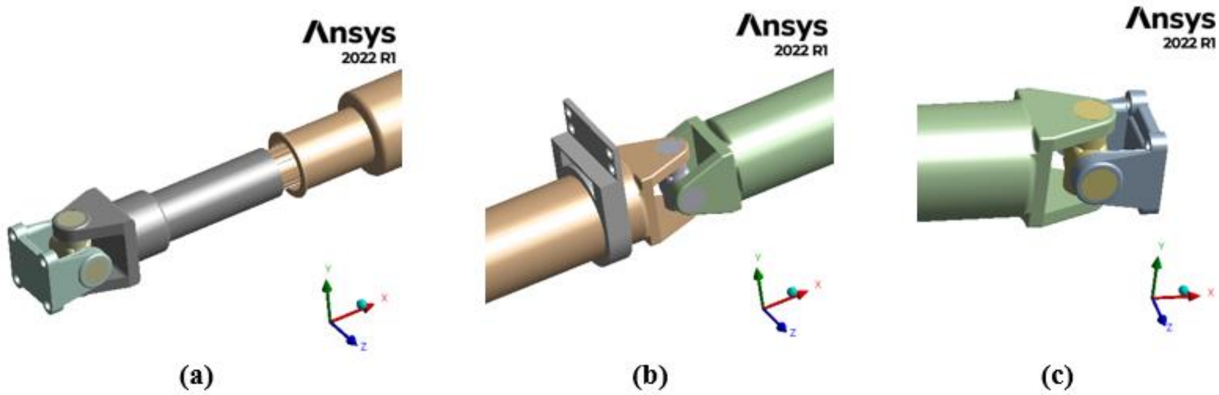


Figure 4.16:***(a)*** Slip yoke and front flange, ***(b)*** Center bearing and middle universal joint and ***(c)*** rear universal joint

4.5.2 Mesh Generation

Mesh generation is the method in which the geometry is divided in small number of elements. A structure consists of infinite number of particles or points hence they must be divided in to some finite number of parts. In meshing components divided into finite numbers. Dividing helps to carry out calculations on the meshed part. The component by is divided by nodes and elements. The components are going to mesh using 3D elements.

The results of the analysis can be affected by the mesh on the geometry. Therefore, it is important that the surface mesh resolves all relevant details of the geometry and satisfies the requirements of the physical models used in the simulation (Lanfrit, 2005).In order to obtain a good result, a fine mesh of unstructured triangles was put on the geometry of the propeller shaft. Figure 4.17 (a-d) shows the generated mesh for the existing and the modified model.

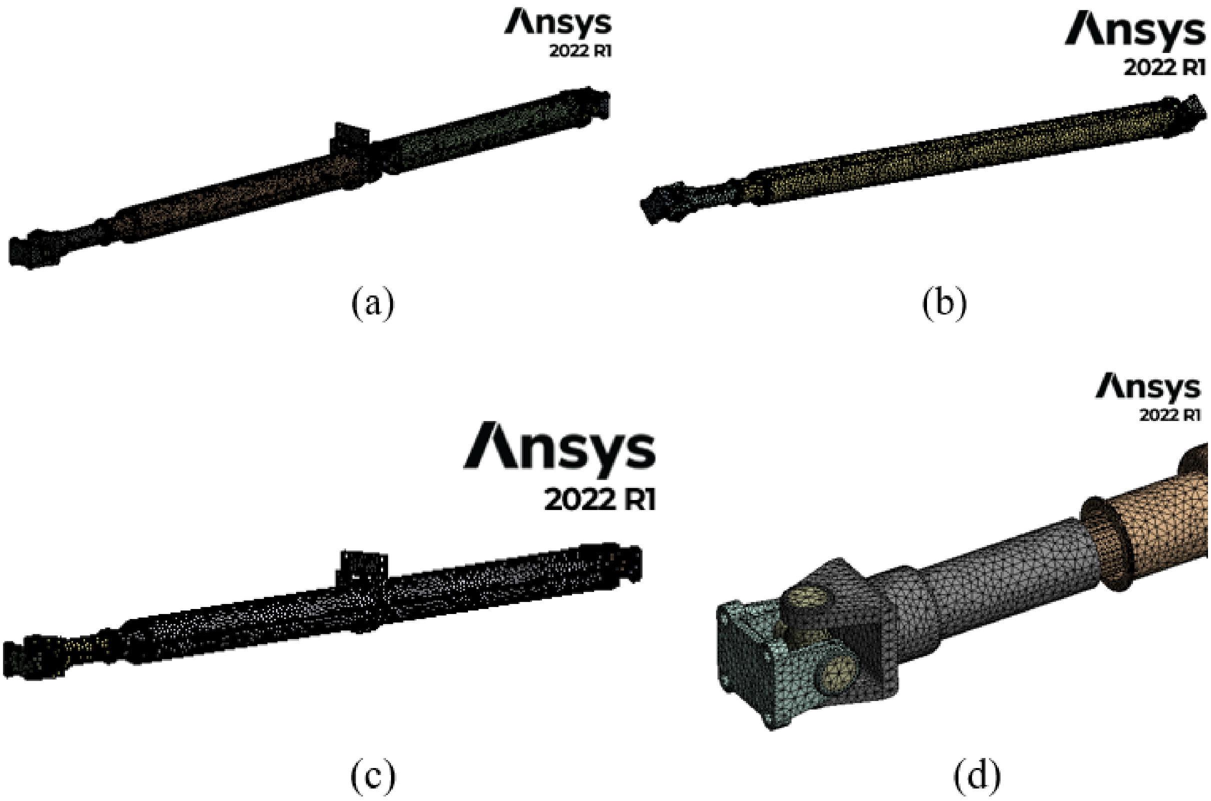


Figure 4.17:**(a)**The mesh generated on the existing assembly, **(b)**The mesh generated on one-piece modified assembly, **(c)**The mesh generated on the one-piece modified assembly with center bearing, **(d)** Closer look at the mesh generated on the assembly

The other thing that must be considered here is quality of the mesh. As it is mentioned in Fluent User's Guide the quality of the mesh plays a significant role in the accuracy and stability of the numerical computation. According to Fluent User's Guide Minimum orthogonal quality is an important indicator of mesh quality. Values for orthogonal quality can vary between 0 and 1 with lower values indicating poorer quality cells. The figures given below shows quality of the mesh for each case:

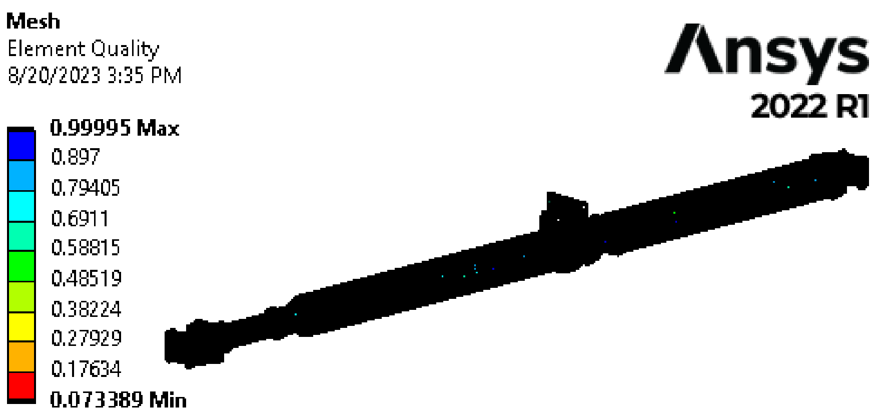


Figure 4.18: The quality of the generated mesh for the existing assembly

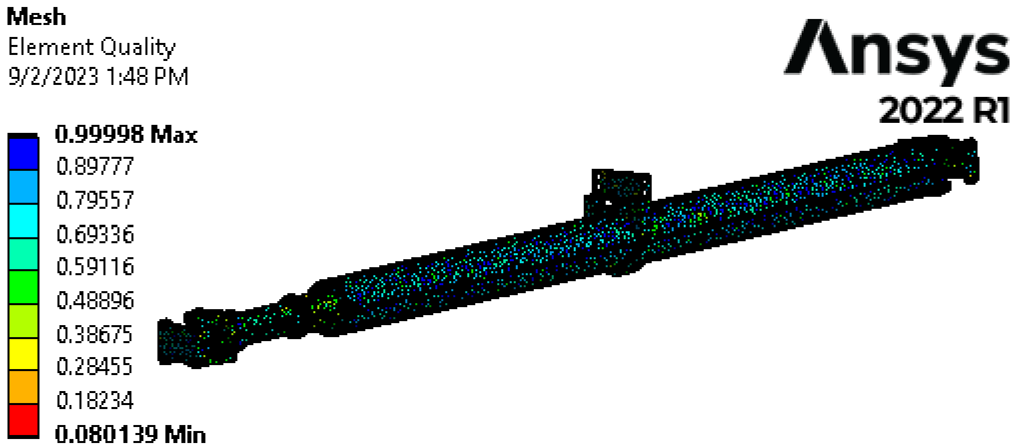


Figure 4.19: Quality of mesh for one-piecemodified shaft with center bearing

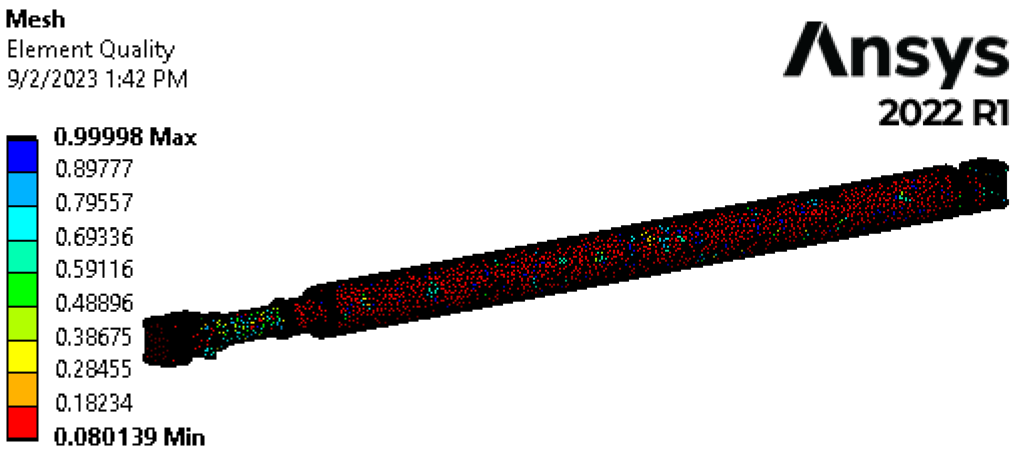


Figure 4.20: Quality of mesh for one-piece modified shaft without center bearing

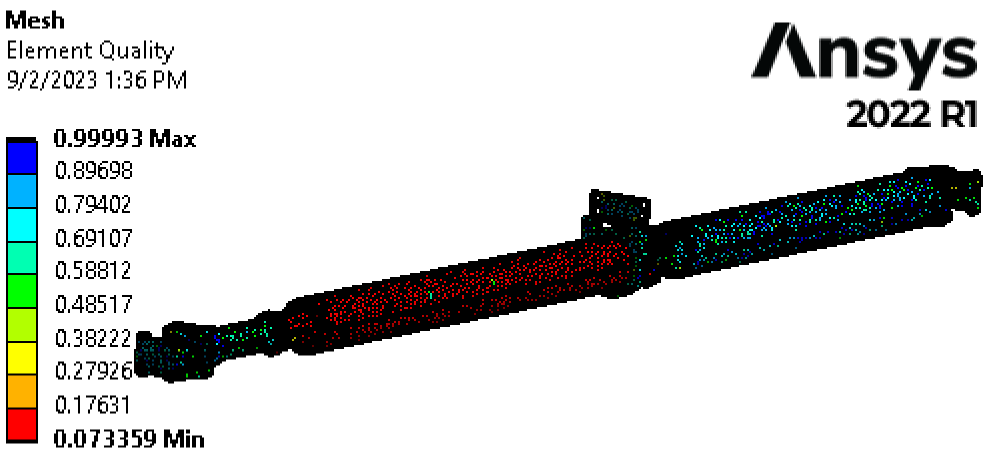


Figure 4.21: Quality of mesh for two-piece modified shaft

In general, the minimum orthogonal quality should not be below 0.01 with the average value significantly larger. As it is shown in Figure 4.18-4.21, the minimum quality of the mesh is 0.073389, 0.080139,0.080139 and 0.073359 respectively which is greater than 0.01. Therefore, the quality of the mesh is as per the standard given in Fluent manual. Therefore, it is possible to proceed to the next step.

4.5.3 Grid Independence Test

The choice of mesh size is always a critical thing in most of numerical analysis; hence it isone of the most influencing factors which determine the computational time and accuracy(Ali, 2017). Therefore, it is important to choose an optimum mesh size. A typical way of choosing mesh size is predicting the sensitivity on results (Wagner, 2018). In this case, the effect of mesh size on total deformation and on the value of maximum stress is studied and the mesh size is selected appropriately. The results of grid independence test are summarized in the tables given below. In order to do the grid independence test angular velocity is applied as a load. The magnitude of the load on the existing model, on the two-piece modified model, on the one-piece modified model without center bearing and, on the one-piece, modified model with center bearing is 314 rad/s, 105 rad/sec and 523 rad/sec respectively.

Table 4.1: Grid independence test of the existing model assembly

Mesh	Nodes	Elements	Maximum deformation (mm)	Maximum stress (MPa)
1	51013	27436	1.4498	56.306
2	59792	30622	1.1065	55.096
3	66706	35811	1.1360	55.668
4	87450	47097	1.2042	54.907

Table 4.2: Grid independence test of the two-piece modified model assembly

Mesh	Nodes	Elements	Maximum deformation (mm)	Maximum stress (MPa)
1	50790	27269	1.4726	10.213
2	54903	29621	1.3275	11.113
3	56827	30647	1.5011	9.940
4	66590	35720	1.4193	10.292

Table 4.3: Grid independence test of one-piece modified model without center bearing assembly

Mesh	Nodes	Elements	Maximum deformation (mm)	Maximum stress (MPa)
1	31637	16832	0.20098	1.71774
2	38730	20733	0.21478	1.71542
3	40663	22073	0.28081	1.94205
4	54347	29548	0.29471	1.92092

Table 4.4: Grid independence test of one-piece modified model with center bearing assembly

Mesh	Nodes	Elements	Maximum deformation (mm)	Maximum stress (MPa)
1	32866	17334	1.1583	515.04
2	40038	21223	1.5354	506.49
3	41890	22512	1.6625	520.46
4	55715	30048	1.6169	541.60

As it is seen from Table 4.1-4.4, the maximum deformation deviation between the values obtained in mesh 3 and mesh 4 is 6.0%, 5.4%, 3.5% and 2.7% respectively. Additionally, the maximum stress deviation is 1.36%, 3.54%, 1.08% and 4.06% respectively. Therefore, since the variation is very less mesh 4 is applied for all the cases.

4.5.4 Boundary Condition

Since the propeller shaft has a rotational motion, a proper joint has to be applied for the analysis. The applied joints are similar for all cases. As it is shown on the CAD modeling three universal joints are used for both two-piece existing and modified shafts. Whereas only two universal joints are used for the one-piece modified shaft. There are three basic joints applied on finite element model of the propeller shaft to get the right dynamics property. These are revolute joint, fixed joint and translational joint. Basically, the boundary condition is applied to the front yoke, rear yoke, universal joints, slip yoke and center bearing. Figure 4.22-4.28 shows all the joints applied on the existing model. The same procedure is followed to set the joints of all the modified models as well.

- ***Front and Rear Yoke***

Both the front and the rear yoke are subjected to moment and as a result they rotate. Therefore, in order to do the analysis and get a correct result, it is needed to set a revolute joint at both ends. Figure 4.22 – 4.23 shows the revolute joint applied at the front and rear yoke respectively.

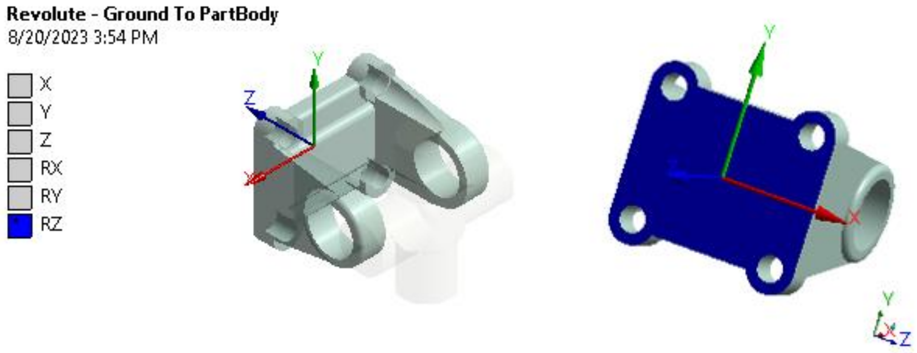


Figure 4.22: Revolute joint at the front yoke

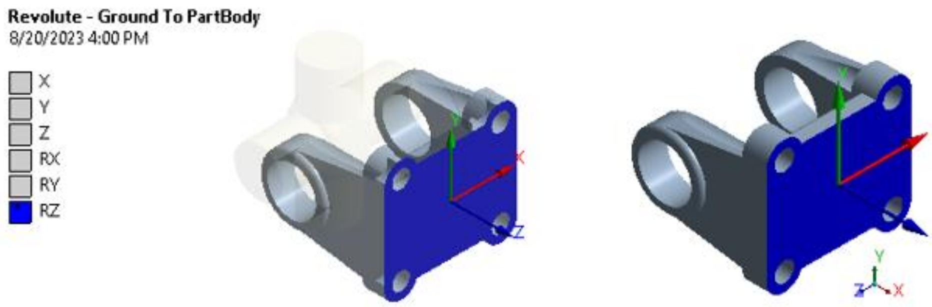


Figure 4.23: Revolute joint at the rear yoke

As it is shown in Figure 4.22-4.23 both the yokes can rotate along the Z-axis. Therefore, the analysis can be done at various conditions.

- ***Universal Joints***

On the existing propeller shaft three universal joints are used and the boundary condition applied to all is similar.

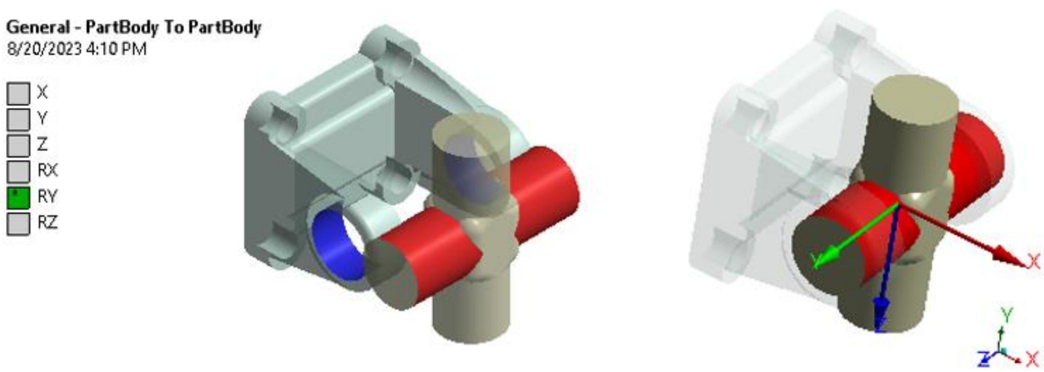


Figure 4.24: Revolute joint between the yoke and the cross

Since it provides swivel connection capable of transferring a turning force between shafts at an angle to one another, a revolute joint is applied to all. Figure 4.24-4.26 shows the revolute joint applied on the universal joint.

As it is shown in Figure 4.25 the cross and the front yoke have a relative rotation along the lateral direction which is Y-axis because of the revolute joint applied between them.

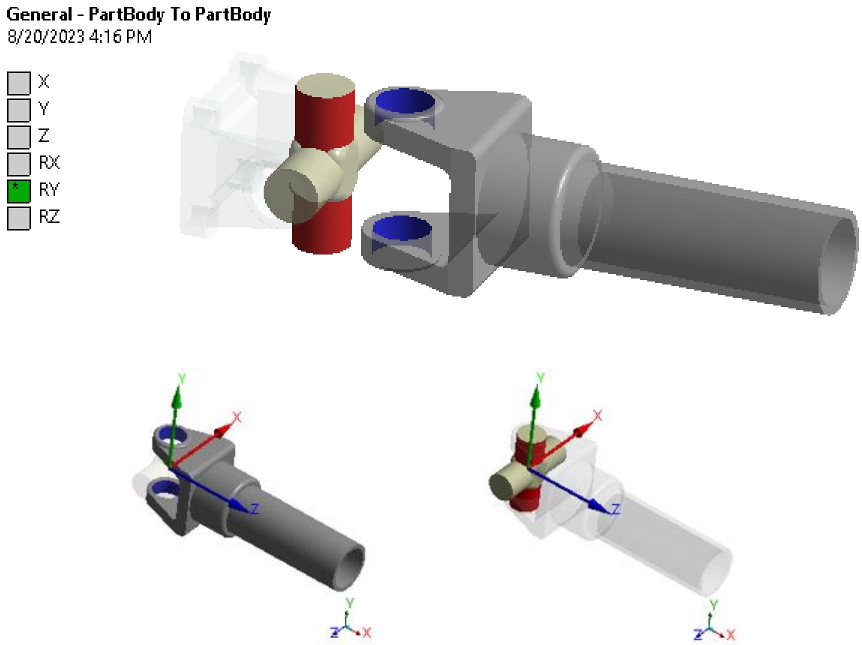


Figure 4.25: Revolute joint between the slip yoke and the cross

Figure 4.25 shows that there is a relative rotation along the vertical axis between the cross and the slip yoke.

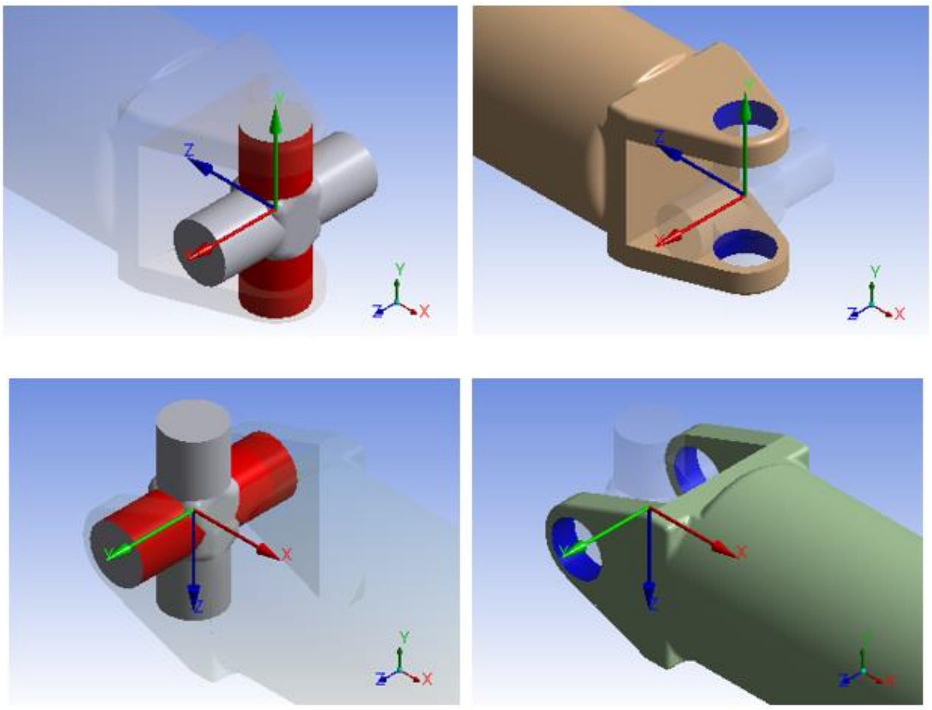


Figure 4.26: Revolute joint between the two propeller shafts

➢ *Slip Yoke*

As it is being explained before the slip yoke is a Splined to allow for changes in driveline length by sliding in and out. Therefore, to maintain this a translational joint is applied between the slip yoke and the drive shaft.

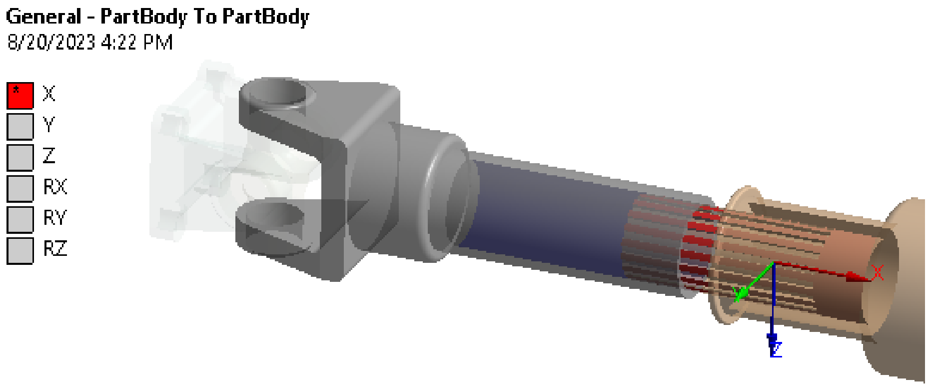
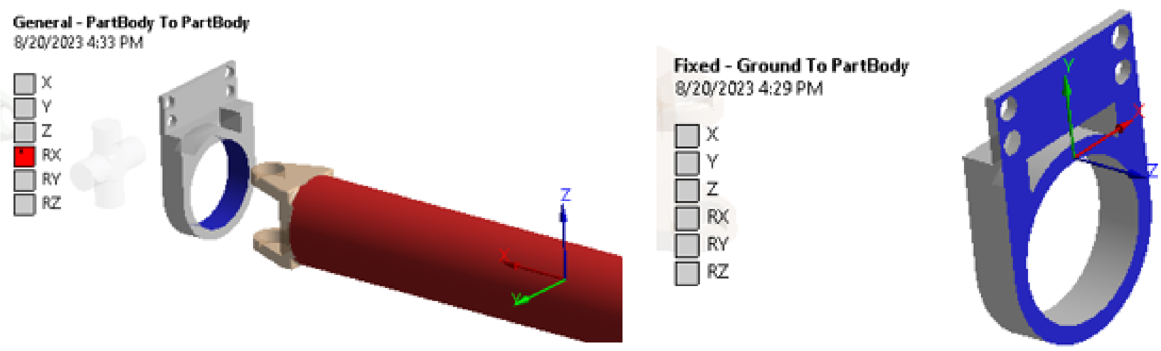


Figure 4.27:Translational joint between the slip yoke and the drive shaft

As it is shown in Figure 4.27, a translational joint is applied between the splined section. As is seen from the diagram, ***X*** is checked and it implies that there is a translational motion along the X-axis.

➢ *Center Bearing*

The center bearing is used to support the drive shaft. Two basic joints are applied at the center bearing, these are revolute joint and fixed joint. As it shown in Figure 4.28 (a), revolute joint is applied between the drive shaft and the center bearing because the drive shaft will be rotating in the center bearing.



(a) (b)

Figure 4.28:(a) Revolute joint between the drive shaft and the center bearing, (b) Fixed joint between the center bearing and the vehicle's frame

The center bearing is fastened with the vehicles frame to support the drive shaft. Therefore, as it is shown in Figure 4.28 (b) a fixed joint is applied between the center bearing ang the vehicle's frame.

The maximum torque and angular velocity of the engine are taken as a load applied on all cases of the propeller shaft. As it is given on the vehicle's specification the maximum torque of the engine is 180Nm at 1500 rpm to 2000 rpm which is taken for the analysis on both the existing and modified models of the shaft. Additionally, the idle speed and the maximum speed of the engine is taken for the analysis. The simulation is considered at three different angular velocity of 1000 rpm, 3000 rpm and 5000 rpm.

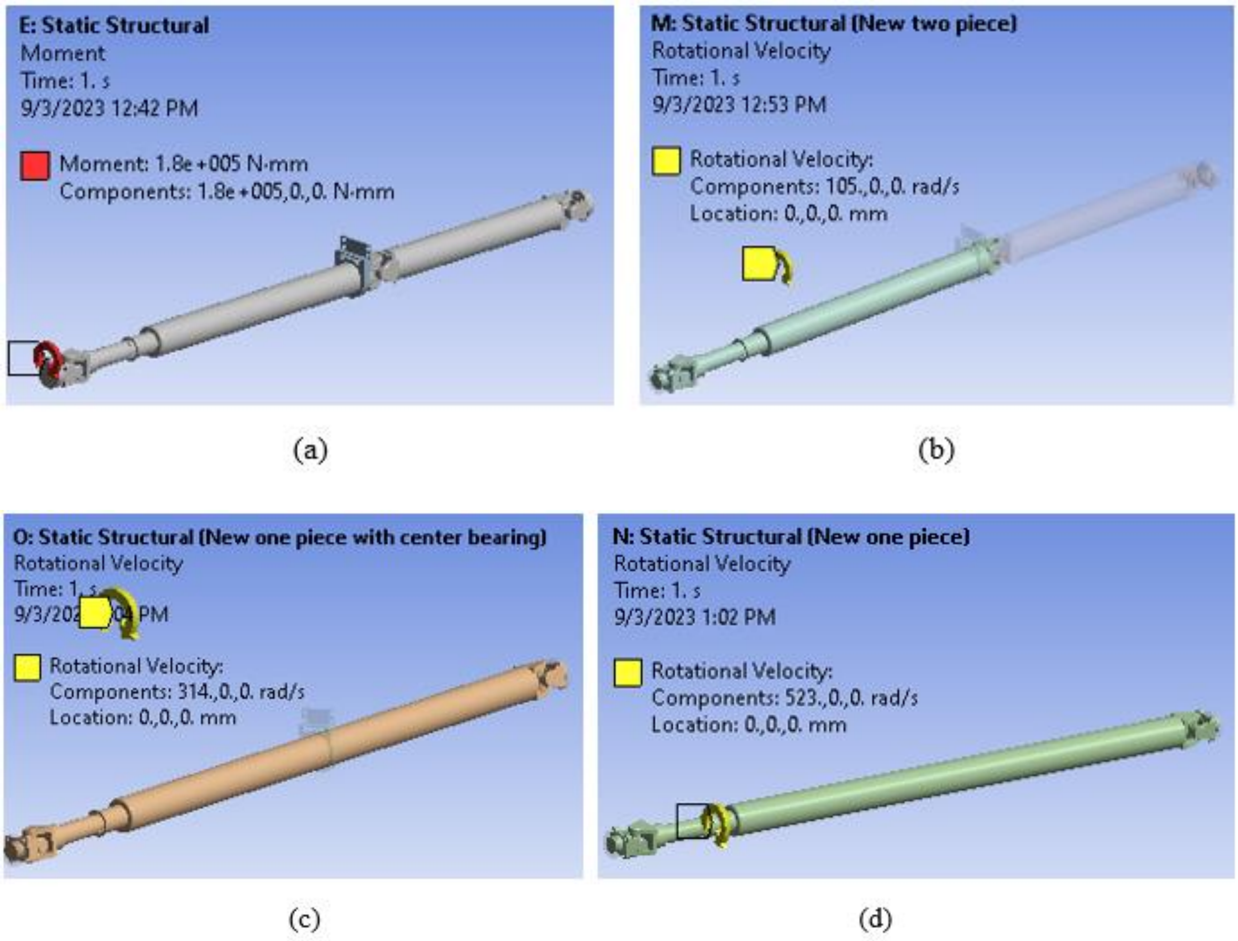


Figure 4.29: ***(a)***Torque applied on the existing shaft, ***(b)***Angular velocity applied on the two-piece modified shaft, ***(c)***Angular velocity applied on the modified one-piece shaft with center bearing ***(d)***Angular velocity applied on the modified one-piece shaft without center bearing.

Since all of the models have the same loading condition, Figure 4.29 shows only a single condition applied on each model but all of the four-conditions are applied on each model for the analysis.

CHAPTER FIVE
RESULT AND DISCUSSION

5.1 Introduction

The FEA was performed by simulating the existing and the modified assembly under actual loading conditions. First, the Static analysis is conducted on the 3D model of the propeller shaft to evaluate deformation and stress responses. Then, Modal analysis evaluates the characteristics of vibration such as natural resonance frequencies and corresponding mode shapes. The stress frequency response is generated after performing the harmonic analysis on the shaft. Dynamics and performance of the propeller shaft is analyzed over practical frequency range of ***0 Hz*** to ***1000 Hz***. The results led to identification of the better model from the given cases.

5.2 Rigid Dynamics of the Existing and Modified Model

Dynamics problem can be solved by implicit and explicit methods to obtain the response of structures under dynamic loads. In some cases, when deformation of material is not the main concern, but the rigid motion is the response which is needed, rigid dynamics is the right path. This applies when:

- ✓ Objects are very stiff.
- ✓ Deformation of material points is not important.
- ✓ You are dealing with a large-scale, complex system
- ✓ Different rigid parts are linked by joints/contacts.

Unlike dynamics for flexible bodies which looks for deformation at each material point, rigid body dynamics looks for the rigid motion of the system. Especially, it solves for relative displacement between different parts.

Therefore, in the case of the existing and the modified propeller shafts different parts are linked by three main joints. These are revolute joint, fixed joint and translational joint. In all cases universal joint is used to provide swivel connection capable of transferring a turning force. Additionally, splined slip yoke which allow for changes in driveline length by sliding in and out is used. With all these integrated parts and joints, the power needed to drive the vehicle should be delivered to the wheels. In order to achieve this, the propeller shaft should have a smooth rotation.

Therefore, rigid dynamics is used to solve the relative motion between the shaft and the joints. The propeller shaft should transmit the torque from the engine to the differential. As it is discussed under chapter 4 all the boundary conditions were applied and checked whether all the joints are linked with the shaft properly or not. The first step in this analysis is setting all the required the joints. Figure 5.1-5.4 shows the connection applied to the existing and modified model.

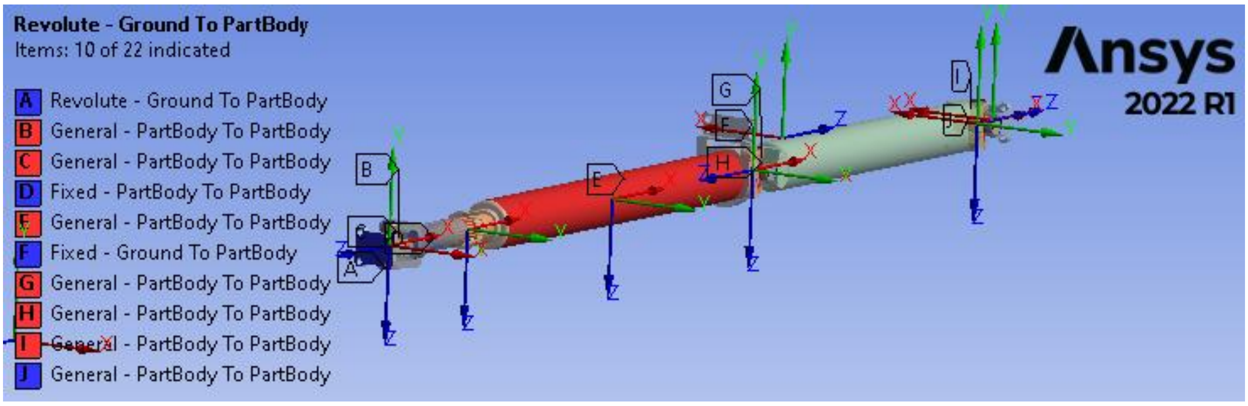


Figure 5.1:Jointconnection of the existing model

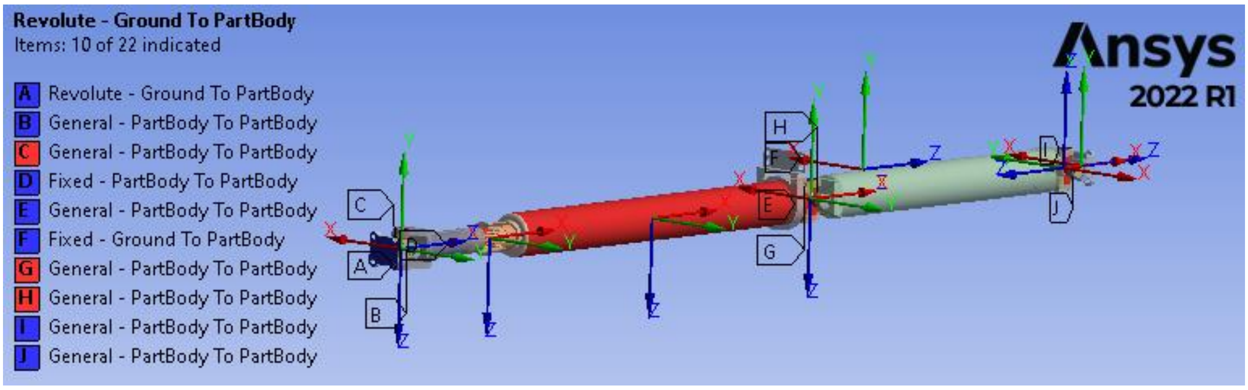


Figure 5.2:Jointconnection of the two-piece modified model

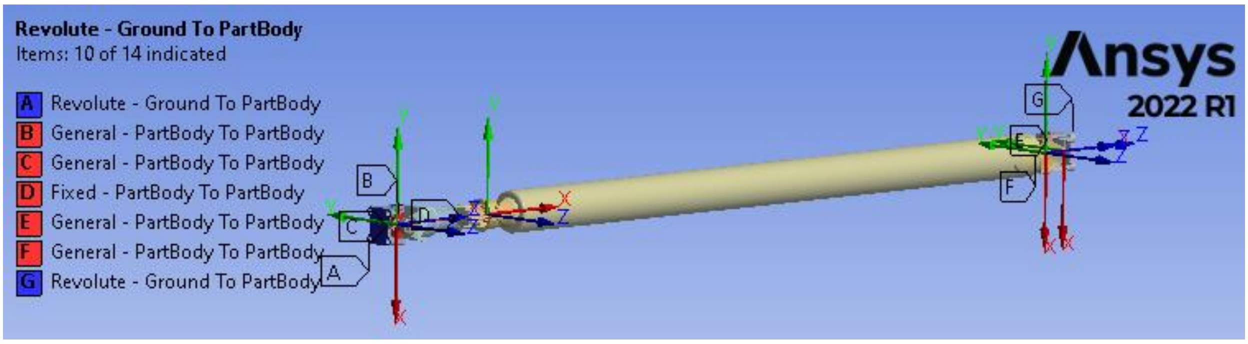


Figure 5.3:Jointconnection of the one-piece modified model without center bearing

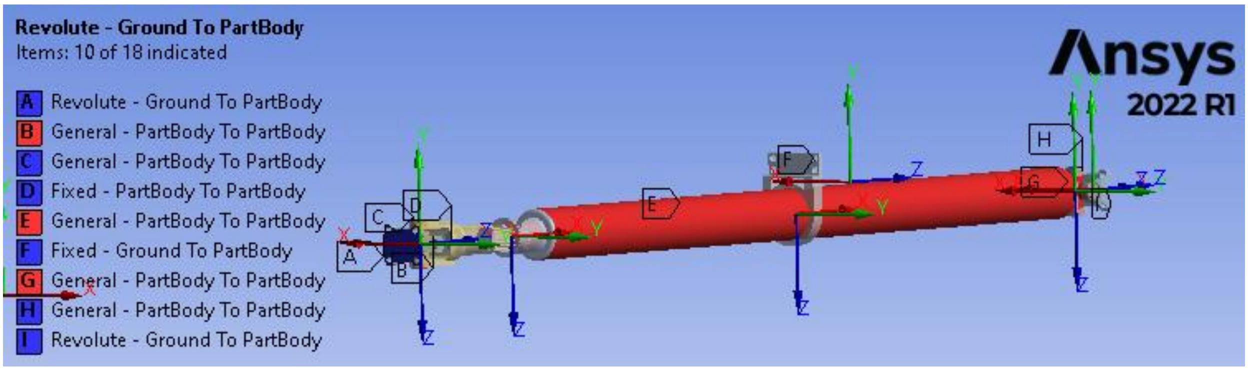


Figure 5.4:Jointconnection of the one-piece modified model with center bearing

The connections shown in Figure 5.1-5.4 are very essential to create the respective rotational motion on all the models of the propeller shaft which results an acceptable result. After this, the input load is applied at the front flange which is connected to the gearbox and then the output load at the outer flange is analyzed. If the connections are correct the input load have to be equal with the output load.

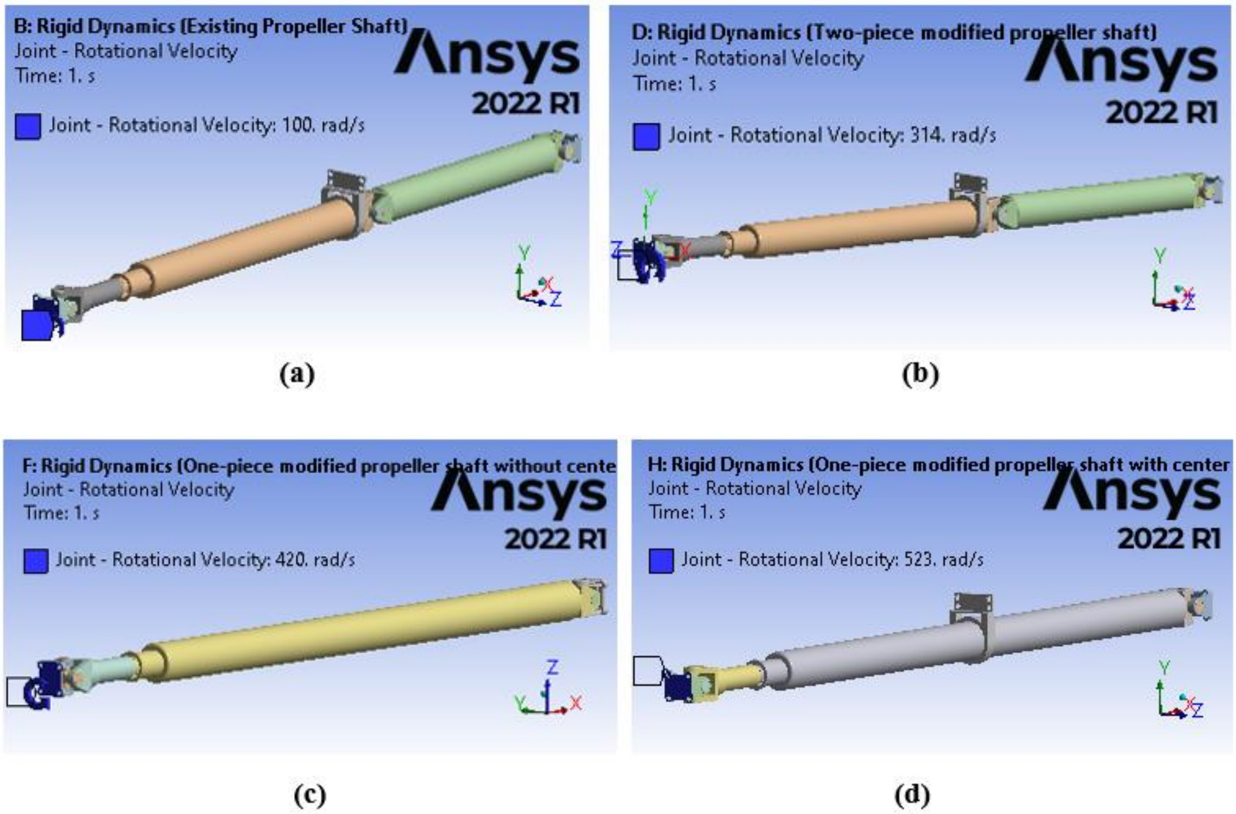


Figure 5.5: Angular velocity applied at the front flange of ***(a)***the existing model ***(b)*** two-piece modified model ***(c)***one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

As it is shown on Figure 5.5 (a-d) angular velocity is applied at the front flange of each model. The angular velocity applied on the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 100rad/sec, 314rad/sec, 420rad/sec and 523rad/sec respectively. As it seen from the diagram different angular velocity values are used but since the values are given to check if the joints are linked right or not, it has no effect on the analysis.

If the connections at the joints are right the input and the output values have to be similar. To check this, the output value is taken for the rear flange. As it is known since the rear flange is connected to the differential, taking the output from this part of the propeller shaft is logical.

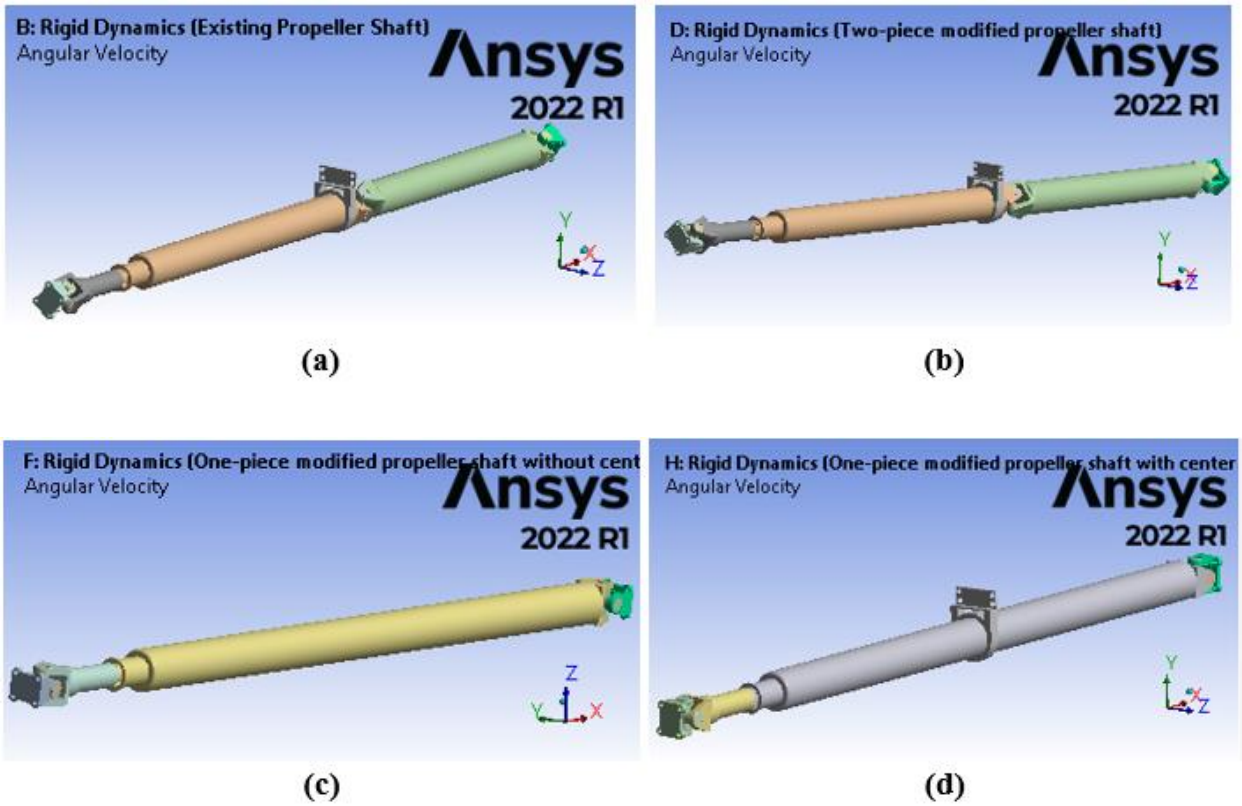


Figure 5.6: Output angular velocity applied at the rear flange of ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

Figure 5.6 (a-d) shows that the rear flange of each model is selected to see the angular velocity delivered to the differential. The output results are shown in Figure 5.7 (a-d).

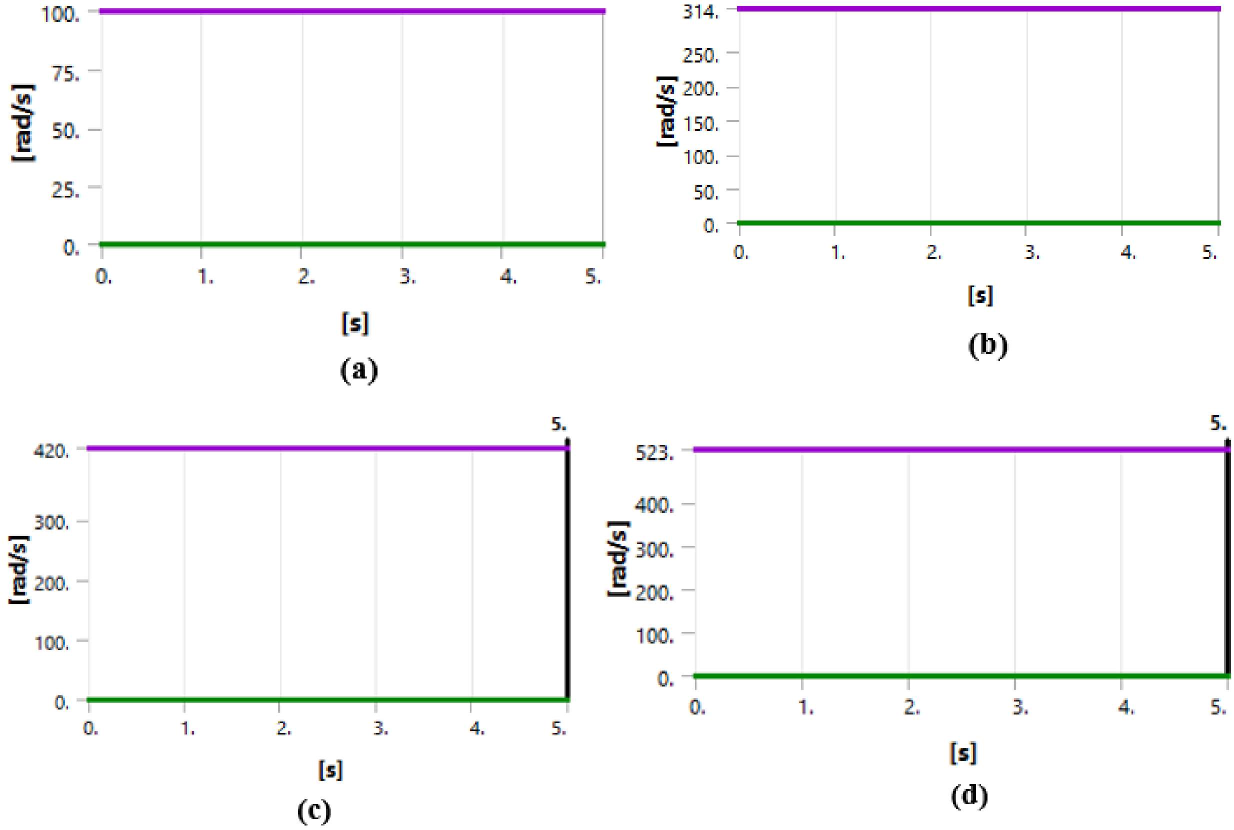


Figure 5.7: Output angular velocity at the rear flange of ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

As it is seen from Figure 5.7 (a-d) the output angular velocity is the same as the input value at any time. This implies that the applied boundary conditions at each model is correct and the results obtained from the analysis is acceptable. Therefore, it is now possible to do the numerical analysis.

5.3 Static Structural Analysis

The static analysis is conducted to see how the existing and the modified propeller shaft responds when subjected to a static loading. The results obtained from this analysis is used to compare and select the better design from the given modified alternatives. From the static analysis three results were taken, these are total deformation, equivalent (Von-Misses) stress and elastic strain. The analysis is carried out at three different angular velocity of the propeller shaft which are 1500rpm, 3000 rpm and 5000 rpm.

These values are taken by considering the angular velocity at which the propeller shaft is subjected to maximum torque which is 180Nm at 1500 rpm and by considering the maximum angular velocity. Therefore, based on the applied load, the results obtained from the analysis are given as follows.

5.3.1 Total Deformation

Deformation refers to the change in size or shape of an object. In ANSYS Workbench there are two types of deformation these are total deformation and directional deformation. Both deformations are used to find displacement through stress. Directional deformation acts in directions like X, Y and Z. In the case of total deformation, it is the square root of the total of the square of X, Y & Z direction (Balwant, 2020).

$$\textbf{Total deformation} = X^2 + Y^2 + Z^2$$

Figure 5.8-5.13 shows the total deformation of the existing and the modified propeller shaft at angular velocity of 1500rpm, 3000rpm and 5000rpm. The existing propeller shaft is made of steel, whereas all of the modified propeller shafts are made of Carbon-Kevlar/Epoxy Hybrid Composite.

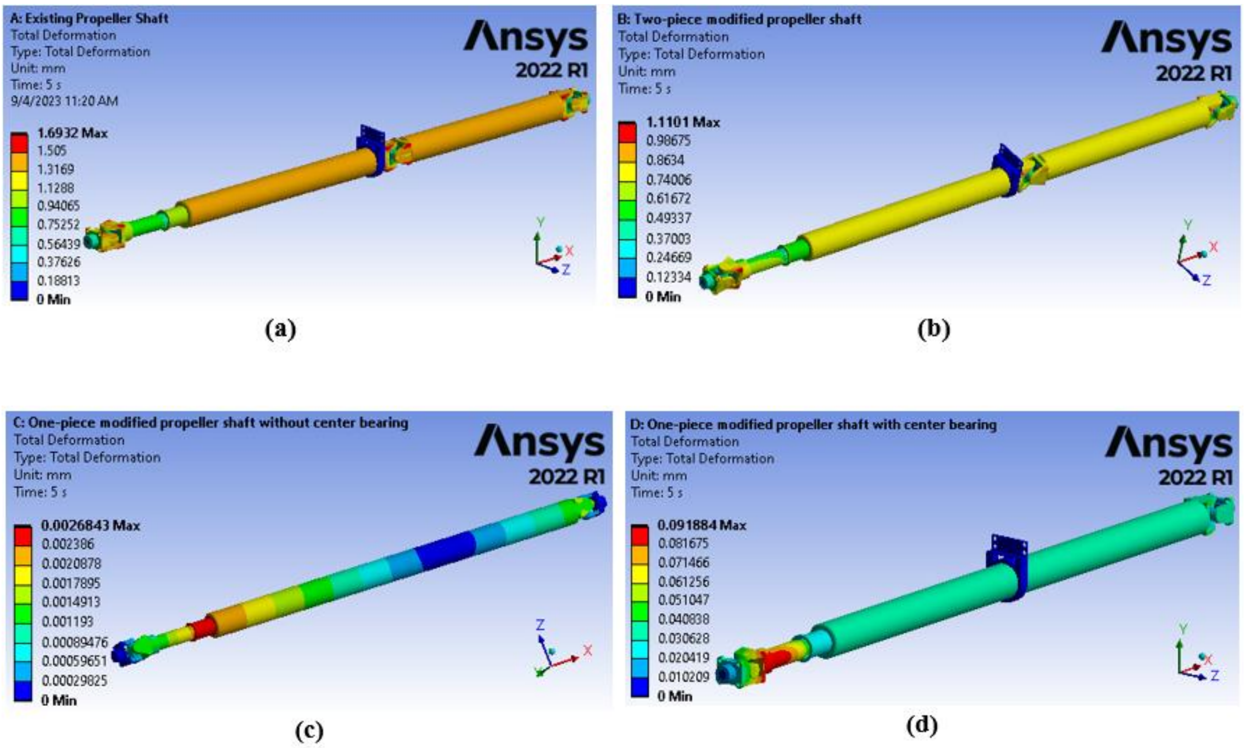
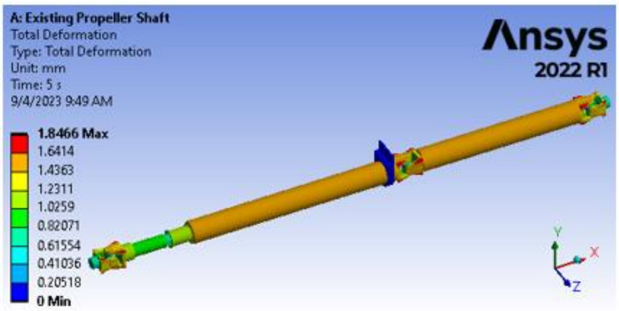
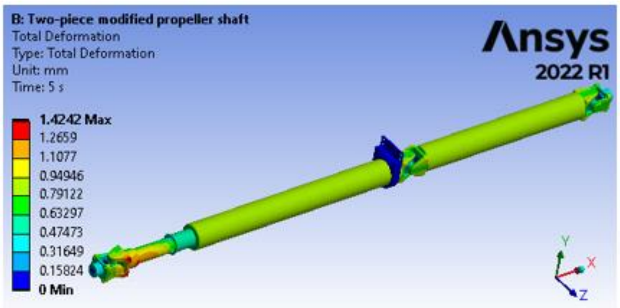


Figure 5.8: Total deformation at 1500rpm for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

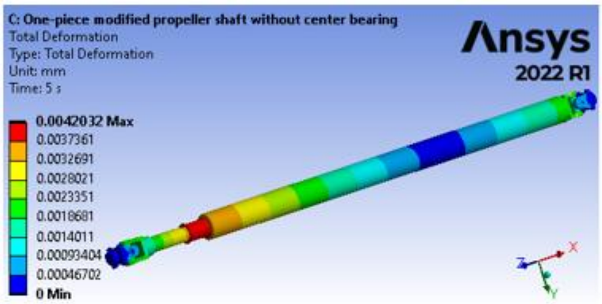
As it is seen from Figure 5.8 (a-d), the results of the total deformation of the existing and the modified propeller shafts are calculated at 1500rpm. From the results it can be seen that total deformation of the existing propeller shaft is maximum at the same load, which is about 1.6932mm. Additionally, from the given alternatives of the modified propeller shafts total deformation is minimum on the modified one-piece propeller shaft without center bearing two-piece modified model with a value of 1.1101mm.



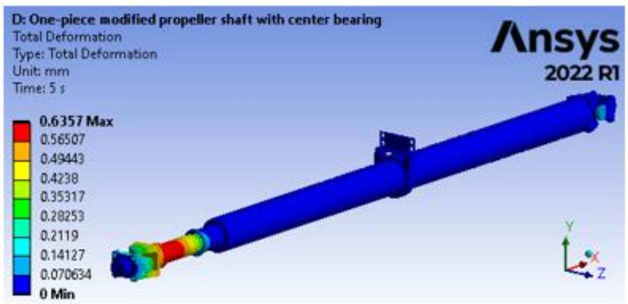
(a)



(b)



(c)



(d)

Figure 5.9: Total deformation at 3000rpm ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

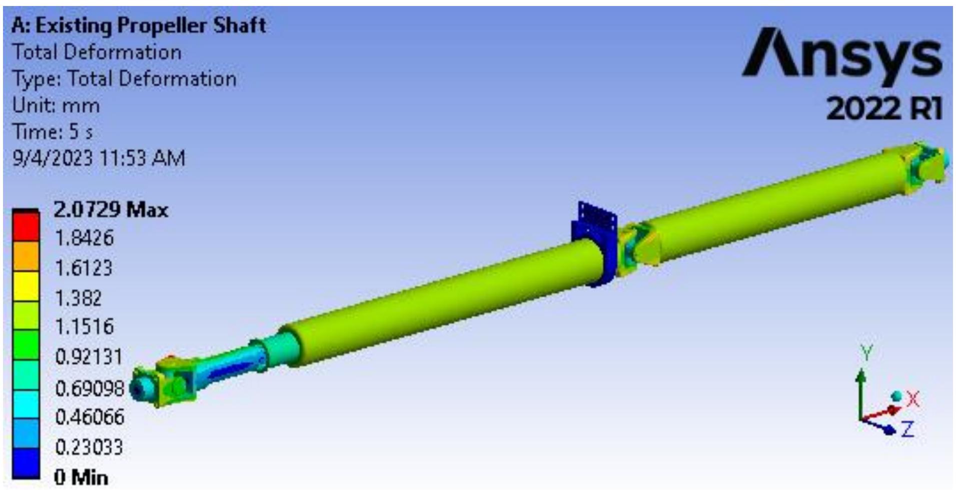


Figure 5.10:Total deformation of the existing propeller shaft at 5000rpm

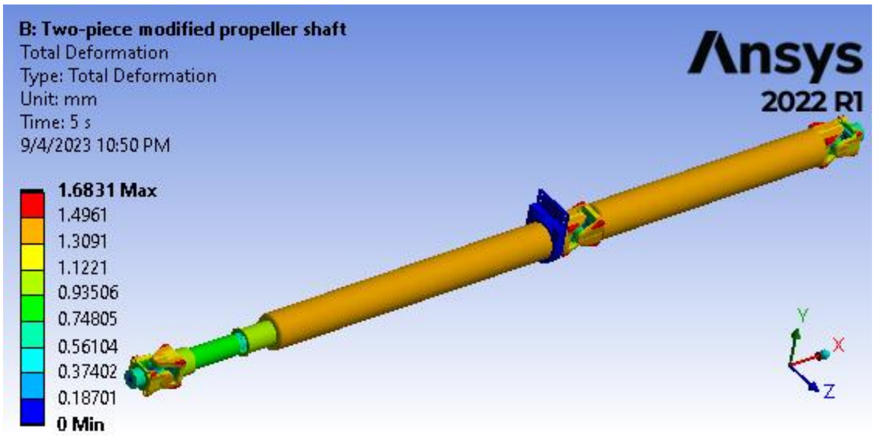


Figure 5.11:Total deformation of the modified two-piece propeller shaft at 5000rpm

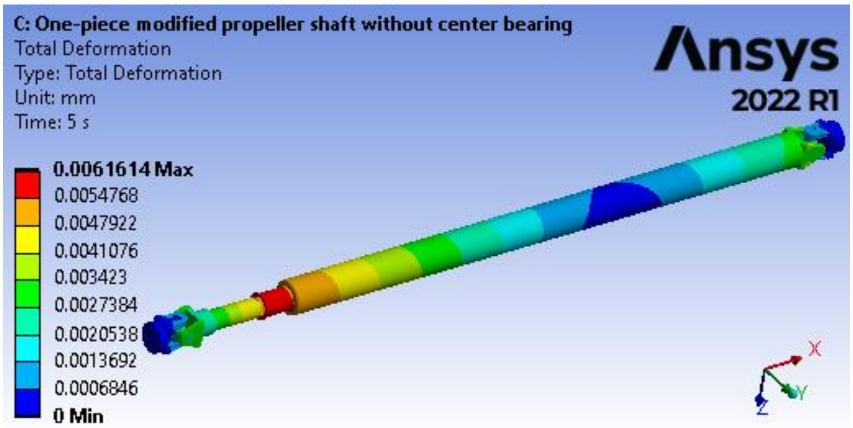


Figure 5.12:Total deformation of the modified one-piece propeller shaft at 5000rpm

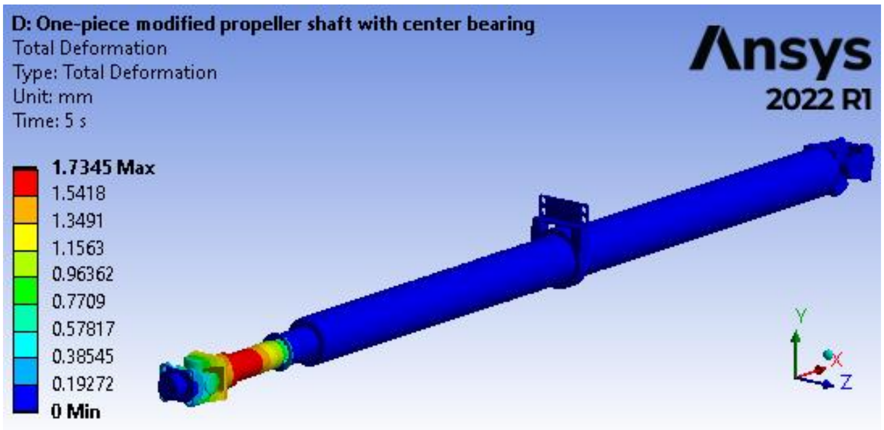


Figure 5.13: Total deformation of the modified one-piece propeller shaft with center bearing at 5000rpm

As it is seen from Figure 5.9-5.13, in all applied load the total deformation of the existing propeller shaft is maximum at the same load, which is about. Additionally, from the given alternatives of the modified propeller shafts total deformation is minimum on the modified two-piece modified model.

5.3.2 Equivalent (Von-Mises) Stress

Von Mises stress is a value used to determine if a given material will yield or fracture. The yield point is the point on a stress-strain curve that indicates the limit of elastic behavior and the beginning of plastic behavior. A material is said to start yielding when the von Mises stress reaches a value known as yield strength. In this case the maximum applied load and different deflection points were considered to see the equivalent stress of the existing and the modified propeller shaft. According to Figure 5.16, the maximum equivalent stress on the existing propeller shaft is 797.31MPa at 5000rpm.

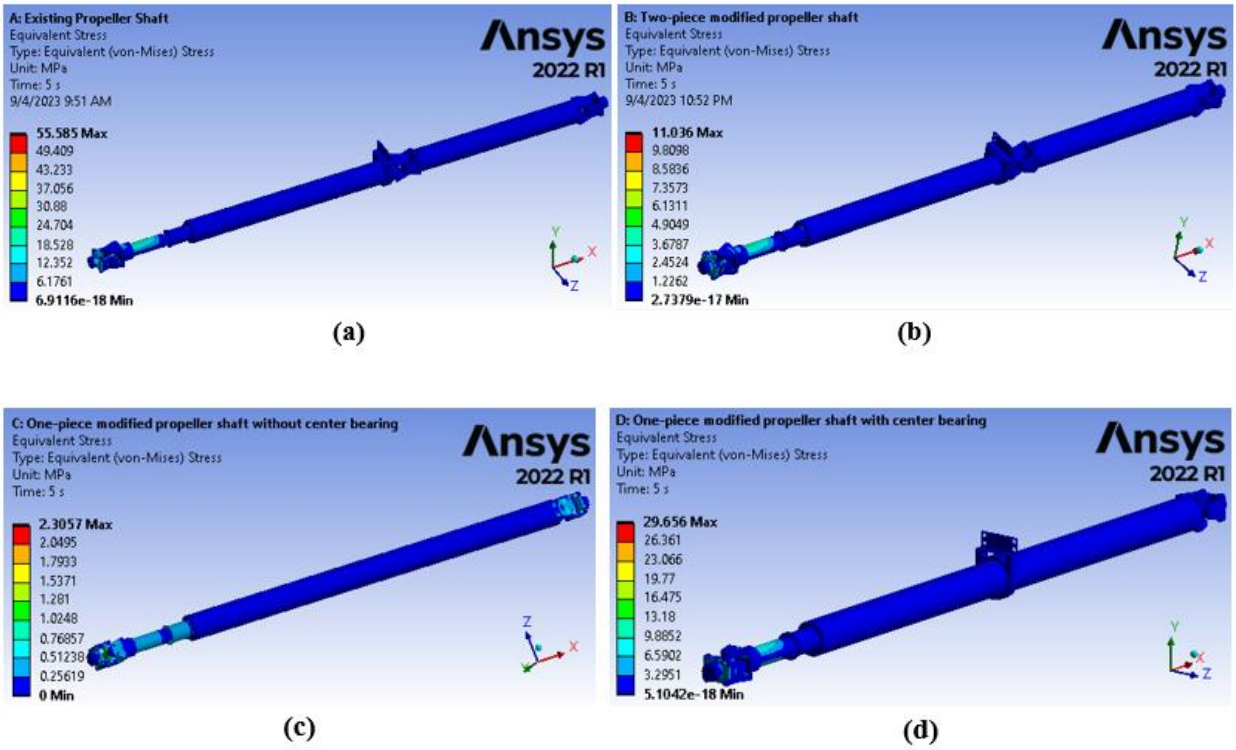
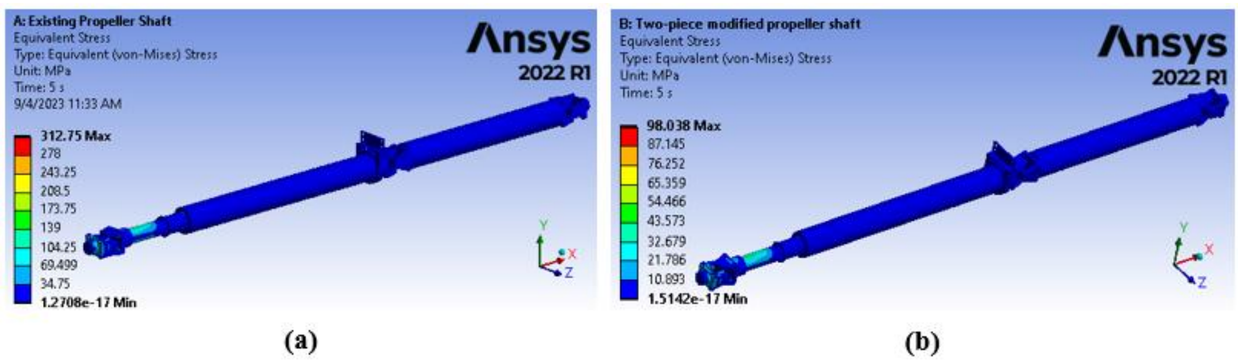
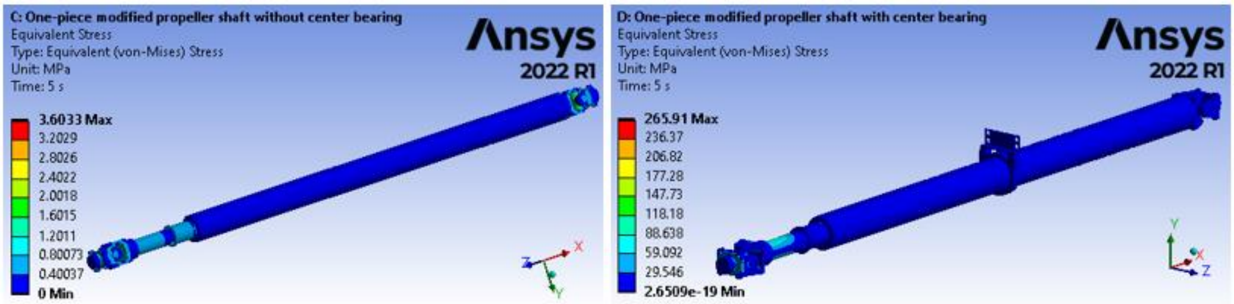


Figure 5.14: Equivalent (Von-Mises) Stress at 1500rpm for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing





(c) (d)

Figure 5.15:Equivalent (Von-Mises) Stress at 3000rpm for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

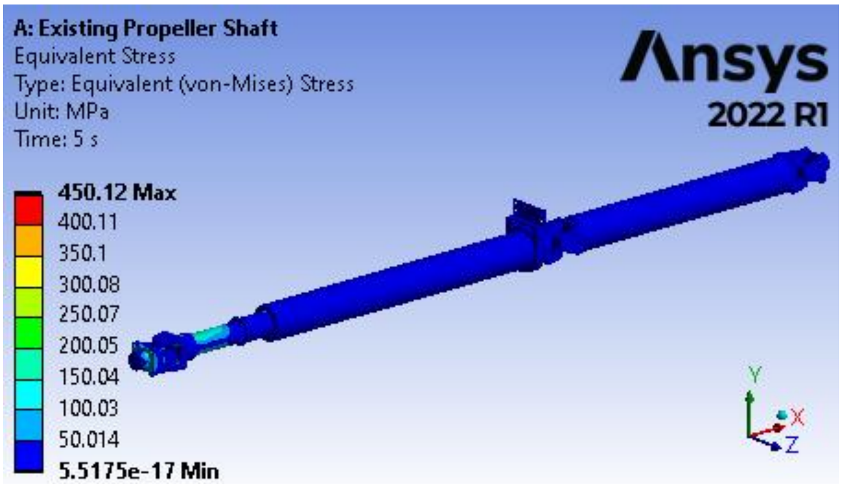


Figure 5.16:Von-Mises Stress of the existing propeller shaft at 5000rpm

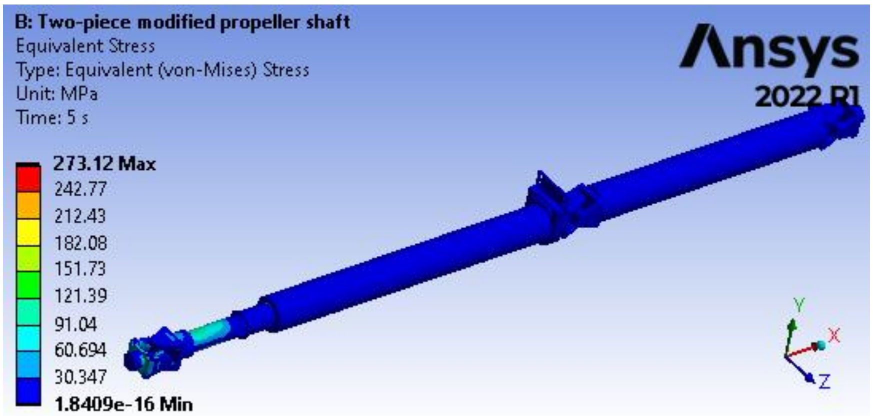


Figure 5.17:Von-Mises Stress of the modified two-piece propeller shaft at 5000rpm

As it is seen from Figure 5.14-17, the analysis of the existing and the modified propeller shafts is conducted at three different angular velocity which are 1500rpm, 3000rpm and 5000rpm. From the results it can be seen that the equivalent stress of the existing propeller shaft is maximum in all applied loads compared to the modified models.

The maximum value is obtained at 5000 rpm which is 797.31Mpa. Whereas, from the given alternatives of the modified propeller shafts equivalent stress is minimum on the modified one-piece propeller shaft without center bearing with a value of (11.036MPa, 98.038MPa and 273.12MPa at 1500rpm, 3000rpm and 5000rpm respectively.

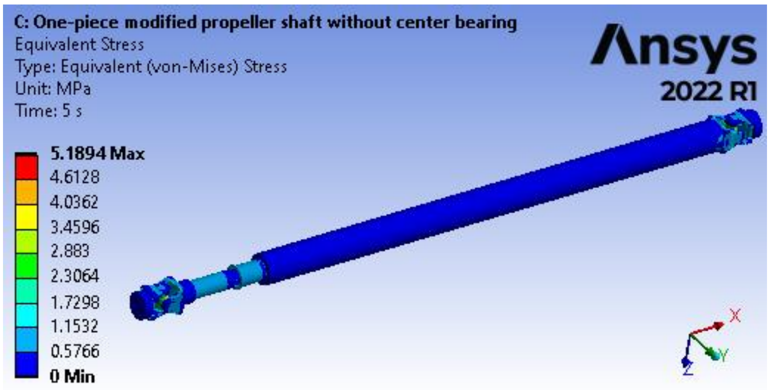


Figure 5.18:Von-Mises Stress of the modified one-piece propeller shaft at 5000rpm

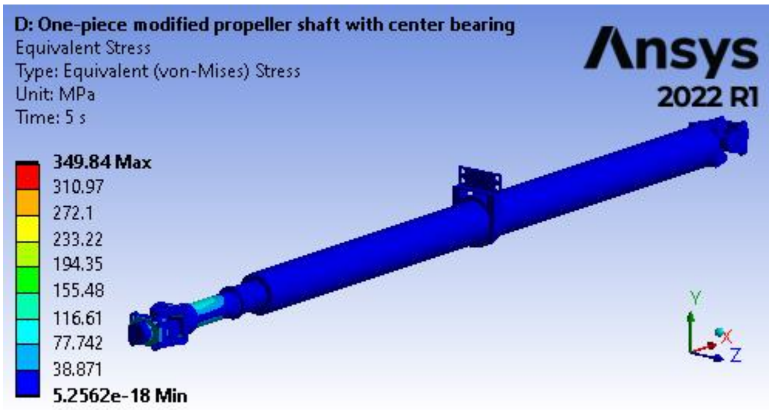


Figure 5.19:Von-Mises Stress of the modified one-piece propeller shaft with center bearing at 5000rpm

In engineering design, the equivalent (Von Mises) stress is computed from equations based on distortion theory. These criteria used to predict or check whether the design will withstand a given load within a safe working condition or not. As it seen from Figure 5.14 – 5.19 the equivalent stress keeps increasing as the value of angular velocity gets higher. The maximum equivalent stress is obtained at an angular velocity of 5000 rpm. At this respective point the maximum equivalent stress for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 450.12MPa, 273.12Mpa, 5.1894Mpa and 349.84Mpa respectively. In all cases the maximum stress is below the yield strength of the material which is 519.6MPa for alloy steel of the existing model and 755.4MPa for Carbon-Kevlar/Epoxy Hybrid Composite of the modified models.

5.3.3 Equivalent Elastic Strain

In engineering, the deformation of a structure is related to the concept of strain. When an external load is applied to an engineering assembly, its components may experience a change in shape, quantified by "strain". The strain is useful in determining the amount of elongation or distortion a structure may experience under various loading conditions.

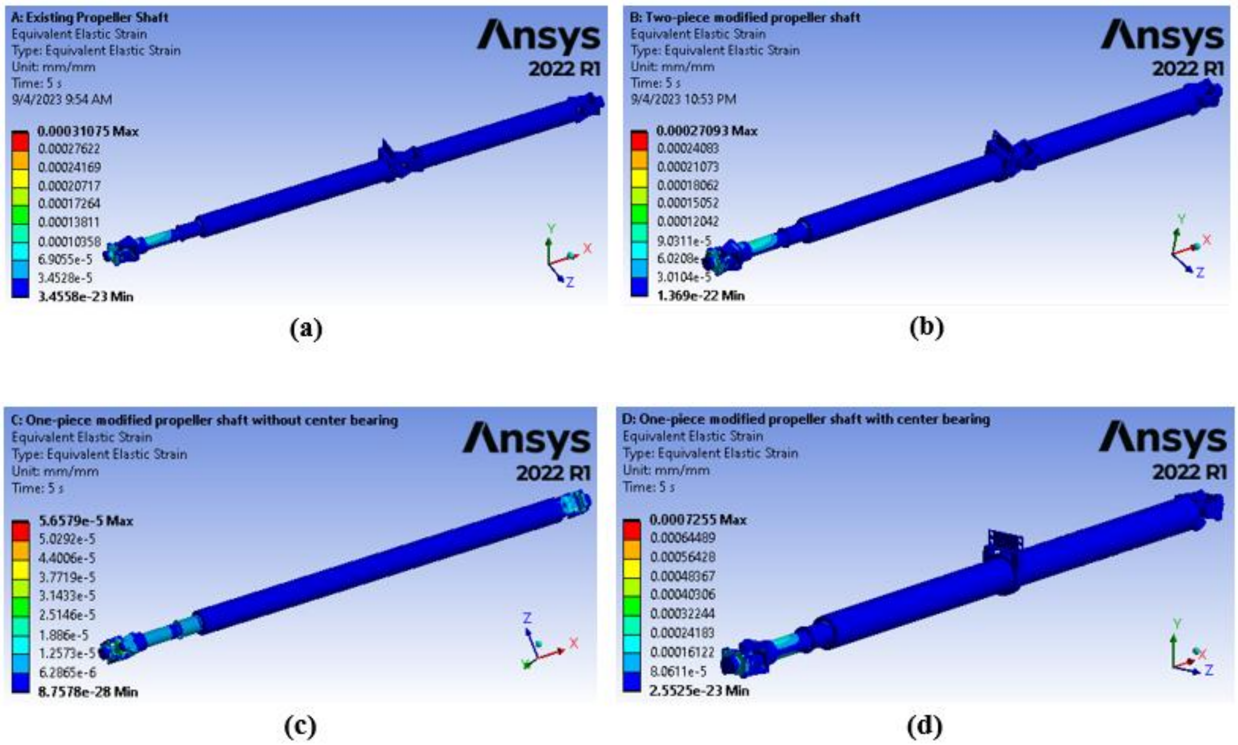
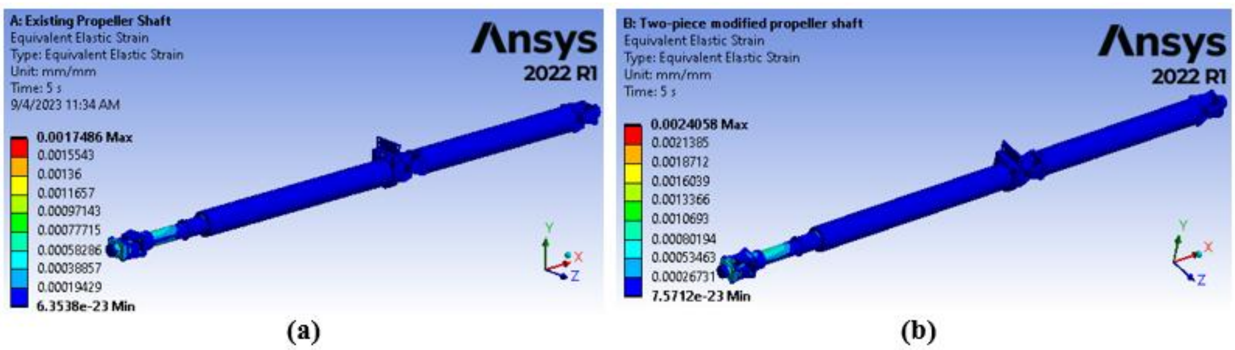
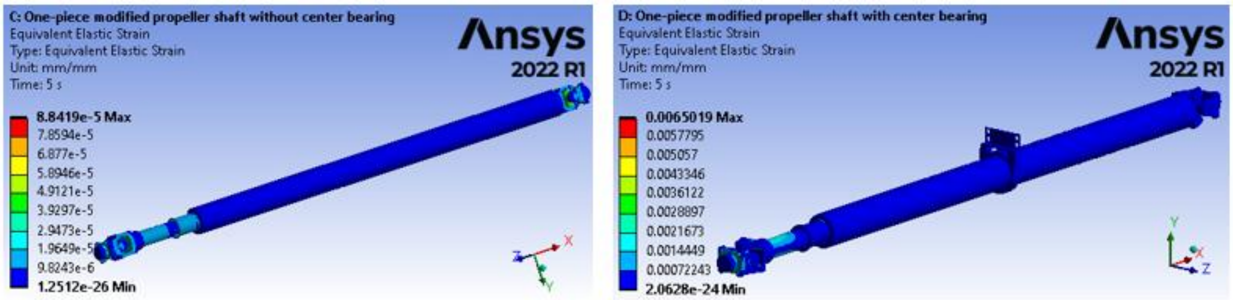


Figure 5.20: Equivalent Elastic Strain at 1500rpm for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing





(c) (d)

Figure 5.21: Equivalent Elastic Strain at 3000rpm for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

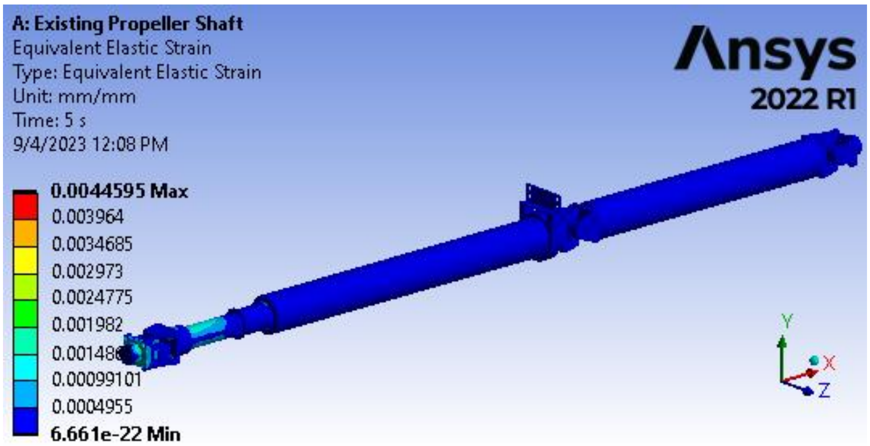


Figure 5.22:Equivalent elastic stress of the existing propeller shaft at 5000rpm

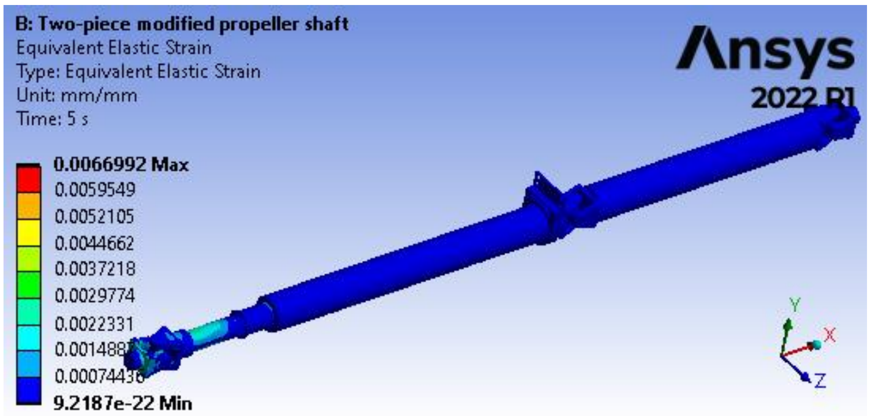


Figure 5.23:Equivalent elastic stress of the modified two-piece propeller shaft at 5000rpm

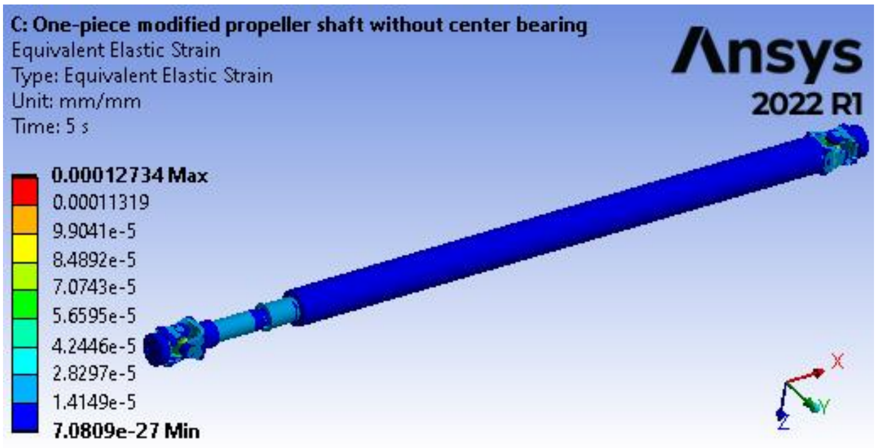


Figure 5.24:Equivalent elastic stress of the modified one-piece propeller shaft at 5000rpm

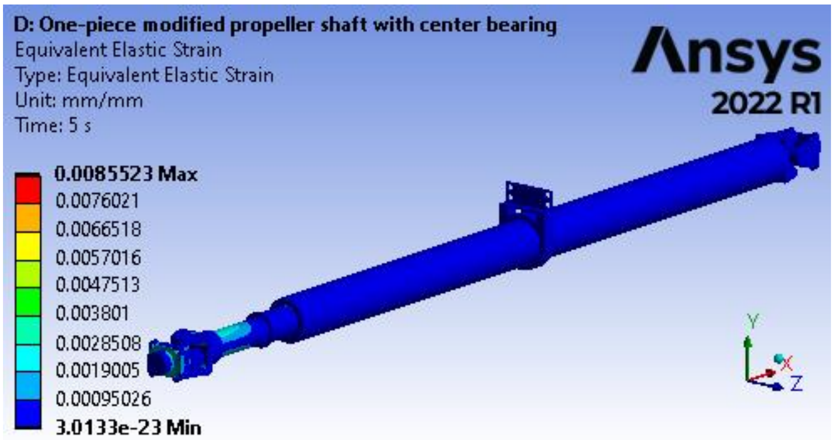
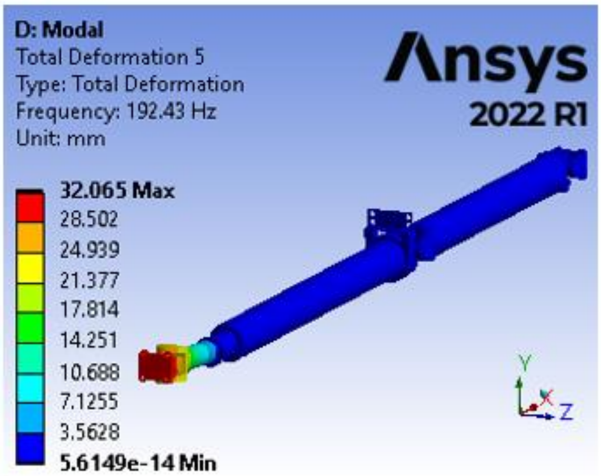


Figure 5.25:Equivalent elastic stress of the modified one-piece propeller shaft with center bearing at 5000rpm

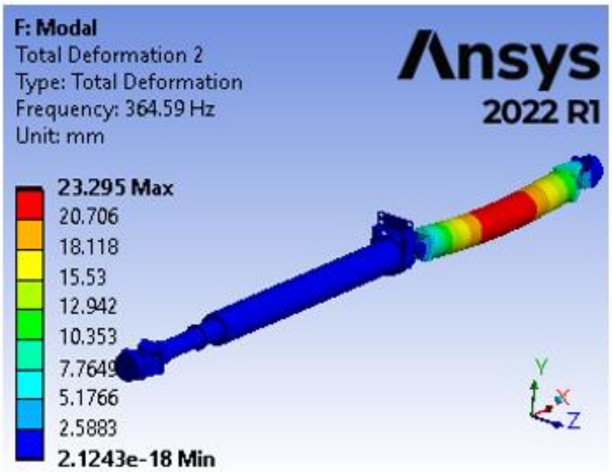
The strain is useful in determining the amount of elongation or distortion a structure may experience under various loading conditions. Figure 5.20 – 5.25 shows elastic strain of the models at 1500rpm, 3000rpm and 5000rpm. As it seen from the results, the equivalent elastic strain keeps increasing as the value of angular velocity gets higher. The maximum equivalent elastic strain is obtained at an angular velocity of 5000 rpm. At the maximum angular velocity, the maximum equivalent elastic strain for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 0.0044595 (0.44%), 0.0066992 (0.66%), 0.00012734 (0.013%) and 0.0085523 (0.855%) respectively. These values can also be expressed in terms of percentage and the percentage shows how much the respective model is elongated compared to the model before a load is applied to it. The least percentage obtained at single-piece modified propeller shaft without the center bearing.

5.4 Modal Analysis

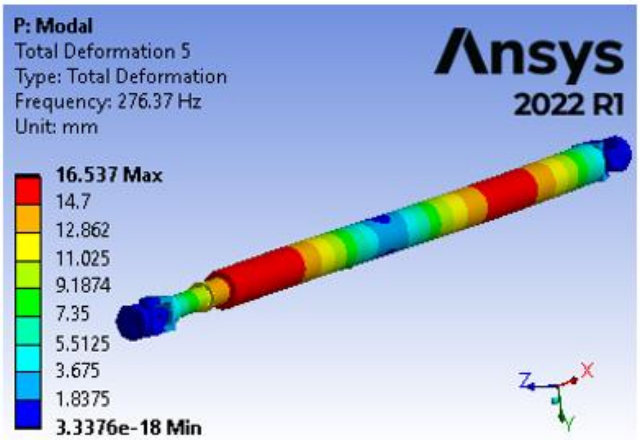
Modal analysis provides insights to the dynamic characteristics of the existing and modified propeller shafts. It is a linear analysis that does not utilize any excitations or loads. Mode frequencies depend on stiffness and mass. It provides information on how the design may respond to different types of loading and it is the basis of other dynamic analysis such as harmonic response and random vibration. The modal analysis is performed to determine the natural frequency and mode shape of the existing and the modified propeller shaft. In order to compare the model's similar frequency range is used in the analysis. The modal analysis is done for the frequency range of 0 – 1000Hz.



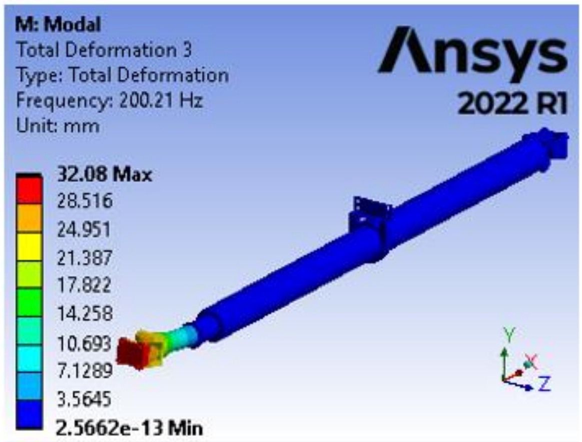
(a)



(b)



(c)



(d)

Figure 5.26: Modal analysis at 150 – 400Hz for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

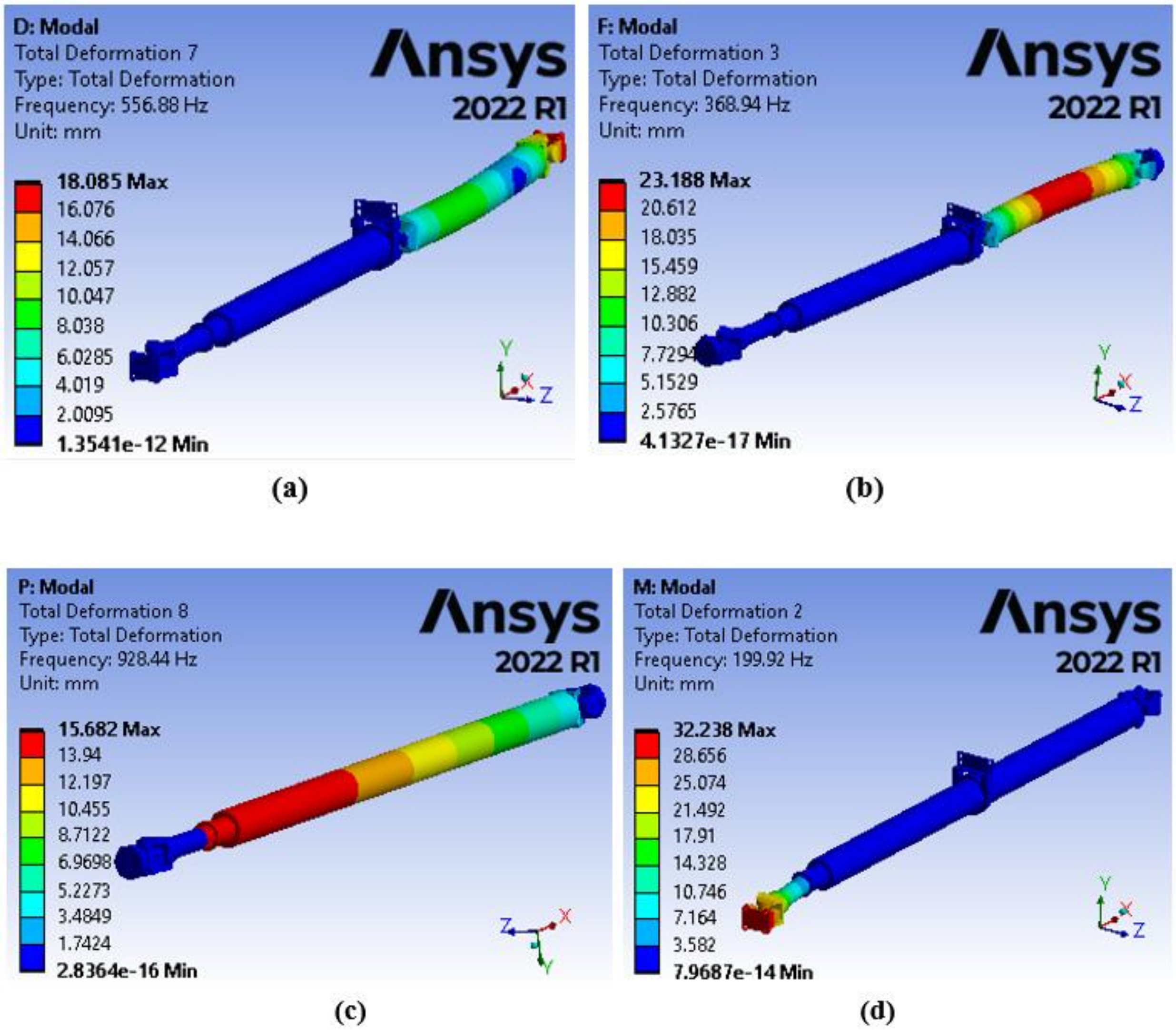


Figure 5.27: Modal analysis at 200 – 1000Hz for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

The analysis is also used to analyses participation factor and effective mass which gives an idea how the design will respond to different types of dynamic loading. Participation factor measures the amount of mass moving in each direction for each mode. A high value of a participation factor in a direction indicates that the mode will likely be excited by forces in that direction. The effective mass is a measure of how much mass is associated with each node.

As it is seen from Figure 5.27(a) the maximum deformation of two-piece modified model364.59Hzis 23.295mm .

which is less than the existing model and other modified models even at lower frequency. For instance, as it is shown in figure 5.26(a) the maximum deformation of the existing shaft at 192.43 is 32mm which is higher than the modified model at 364.59Hz.

5.5 Transient Analysis

Transient analysis is used to determine the response of the propeller shafts to arbitrarily time-varying loads. To conduct the transient analysis time varying load is needed. Therefore, to do this various driving condition of the vehicle need to be taken in to consideration. The analysis is carried out for five seconds. The conditions considered in each second were between the angular velocity in which the torque becomes maximum and the maximum angular velocity of the engine which is 5000rpm. From the dynamic analysis three results were taken for comparison, these are total deformation, equivalent (Von-Misses) stress and elastic strain.

5.5.1 Total Deformation

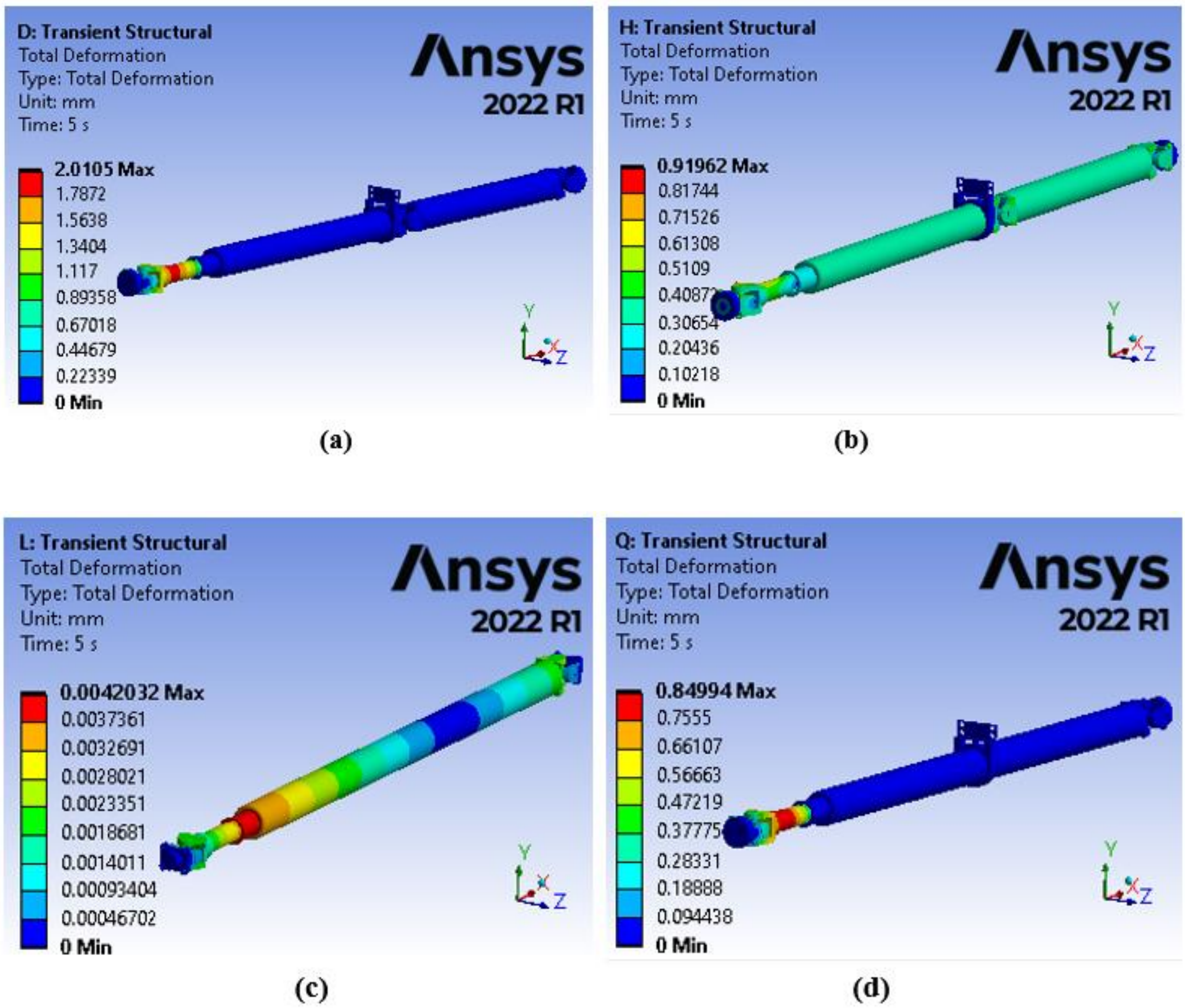


Figure 5.28: Total deformation of transient analysis for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

As it is shown in Figure 5.28 (a - d), the maximum deformation from the dynamic analysis for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 2.0105mm, 0.91962mm, 0.0042032mm and 0.84994mm respectively. Therefore, it can be seen that the total deformation is less on single-piece modified propeller shaft without center bearing two-piece modified propeller shaft compared to the existing and the other modified models.

5.5.2 Equivalent (Von-Mises) Stress

Von Mises stress is a value used to determine if a given material will yield or fracture. Figure 5.29 (a-d) shows equivalent Stress of transient analysis for the existing and the modified propeller shafts.

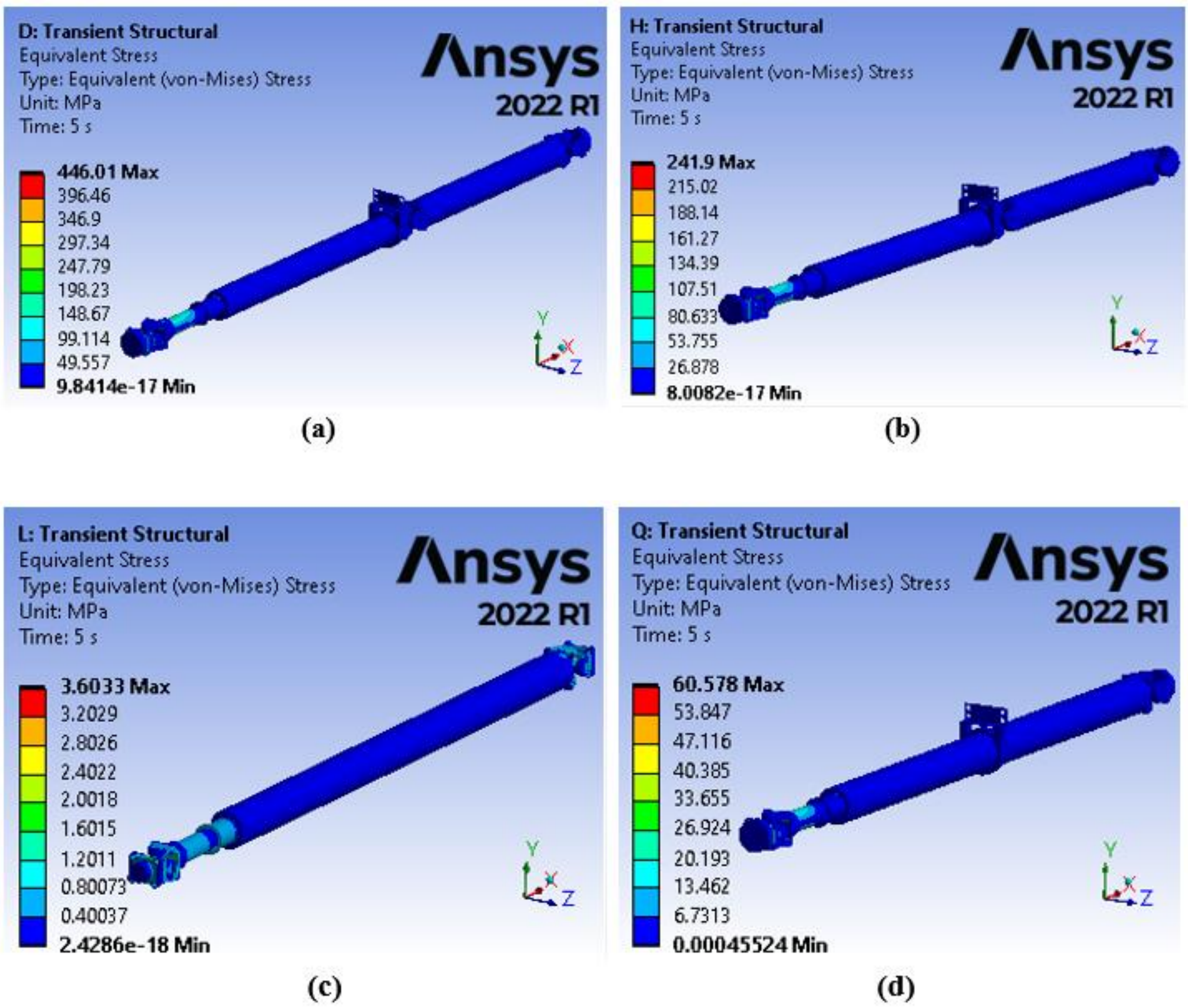


Figure 5.29:Equivalent Stress of transient analysis for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

From Figure 5.29 (a - d), it is seen that the maximum equivalent stress from the dynamic analysis for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 446.01MPa, 241.9MPa, 3.66MPa and 60MPa respectively. Therefore, it can be seen that the maximum equivalent stress is less on single-piece modified propeller shaft without center bearing two-piece modified propeller shaft compared to the existing and the other modified models. In all cases the maximum stress is below the yield strength of the material which is 519.6MPa for alloy steel of the existing model and 755.4MPa for Carbon-Kevlar/Epoxy Hybrid Composite of the modified models.

5.5.3 Equivalent Elastic Strain

Figure 5.30 (a-d) shows equivalent elastic strain of transient analysis for the existing and the modified propeller shafts.

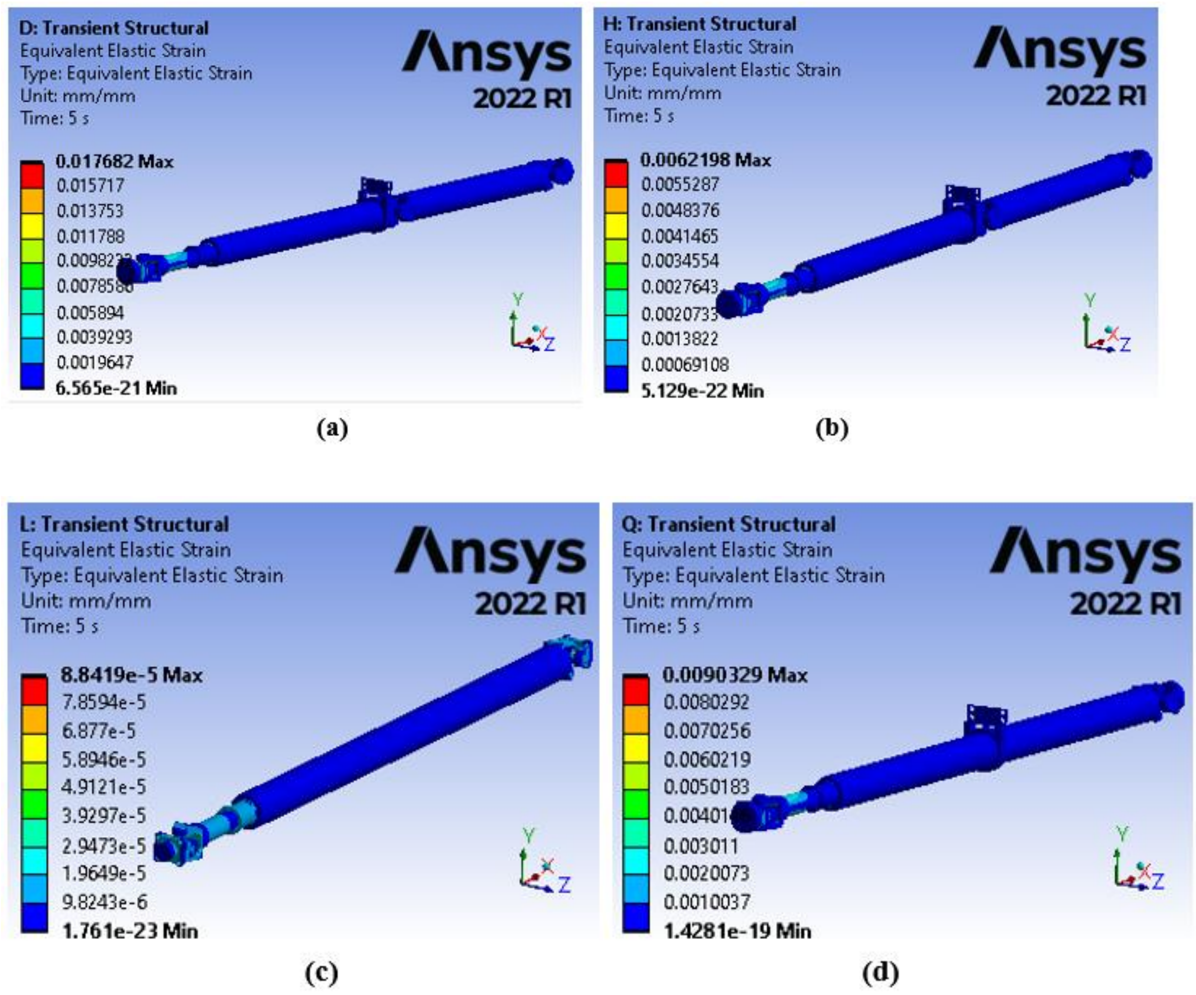


Figure 5.30: Equivalent elastic strain of transient analysis for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

Figure 5.30(a-d) shows elastic strain of all the models. As it seen from the results, the minimum equivalent elastic strain is obtained at the single-piece modified propeller shaft without the center bearing. The maximum equivalent elastic strain for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 0.0017682 (0.177%), 0.0062198 (0.622%), 0.0000884 (0.0088%) and 0.0090329 (0.9033%) respectively. These values expressed in terms of percentage shows how much the respective model is elongated compared to the model before a load is applied to it.

5.6 Harmonic Analysis

Harmonic analysis is used to determine the response of the models under a steady-state sinusoidal (harmonic) loading at a given frequency from the modal analysis (Chittilla et al., 2013). Loads may be out of phase with one another, but the excitation is at a known frequency.

Harmonic analysis can be used to verify whether or not a machine design will successfully overcome resonance, fatigue, and other harmful effects of forced vibrations. Different from transient dynamic analysis harmonic analysis doesn't solve the time history response of the structure. It treats the structure dynamic behavior in the frequency domain. Sinusoidal response of harmonic analysis only applied to direct results like directional displacement, normal stress or strain and shear stress or strain. For derived results like total deformation, principal stress or strain and equivalent stress or strain the results ca be periodic. The frequency response data obtained from harmonic analysis is conducted over frequency range of 0Hz to 1000Hz.

5.6.1 Applied Load

To conduct harmonic analysis, it is crucial to do the modal analysis first. Therefore, this analysis is linked with the modal analysis that is why the frequency range is 0 Hz to 1000 Hz. According to Figure 5.31(a-d) the maximum torque is applied in all models as a load.

(a)

(b)

Figure 5.31: Applied load for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

Unlike modal analysis, to conduct harmonic analysis it is necessary to apply a load on the models. Therefore, other than the frequency range, the maximum torque is considered as loading condition. Figure 5.31(a-d) shows the applied load on the existing and modified models. The reason for the maximum torque to be selected for the loading condition is that it is the critical point in terms of design.

5.6.2 Harmonic Frequency Response

The frequency response data obtained from the harmonic analysis shows the amplitude of the model over the frequency range which is 0Hz – 1000Hz. Figure 5.32-5.35 shows the harmonic frequency response of the existing and the modified propeller shaft. The results are necessary to know the frequency value which causes a higher amplitude and then the deformation as well as the stress distribution over each model will be analyzed.

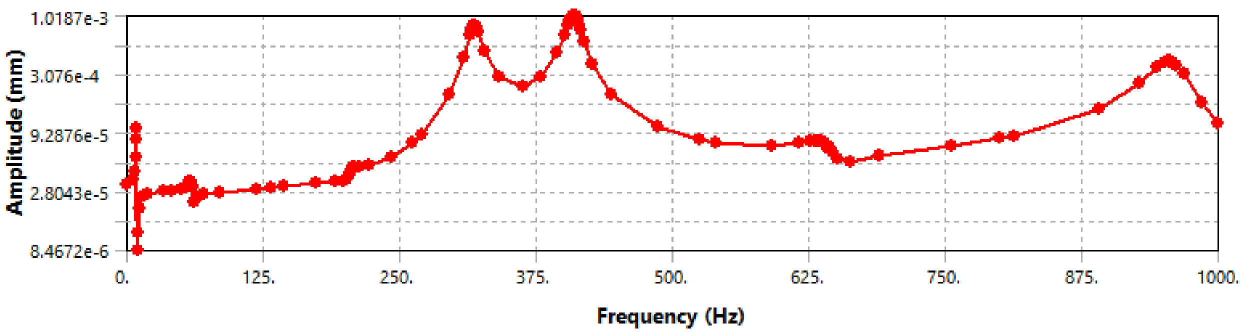


Figure 5.32: Harmonic response of the existing propeller shaft

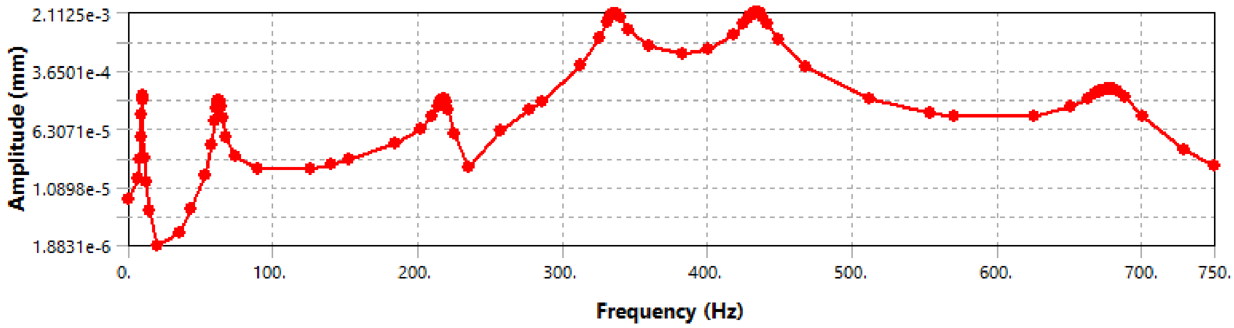


Figure 5.33: Harmonic response of the two-piece modified propeller shaft

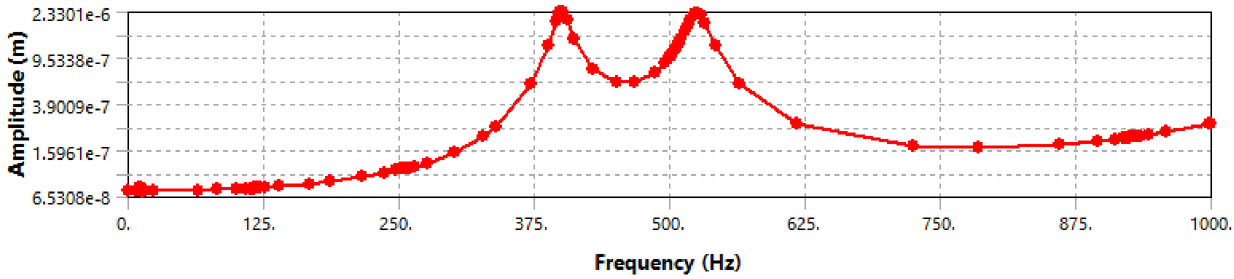


Figure 5.34: Harmonic response of the single-piece modified propeller shaft without center bearing

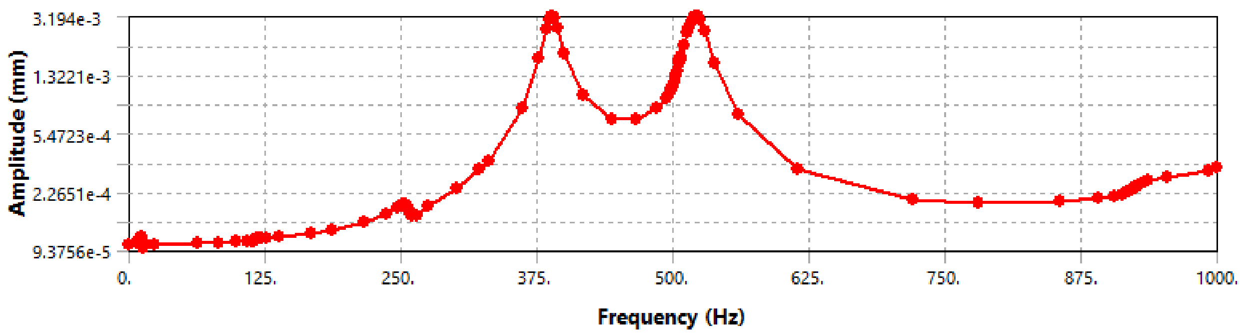
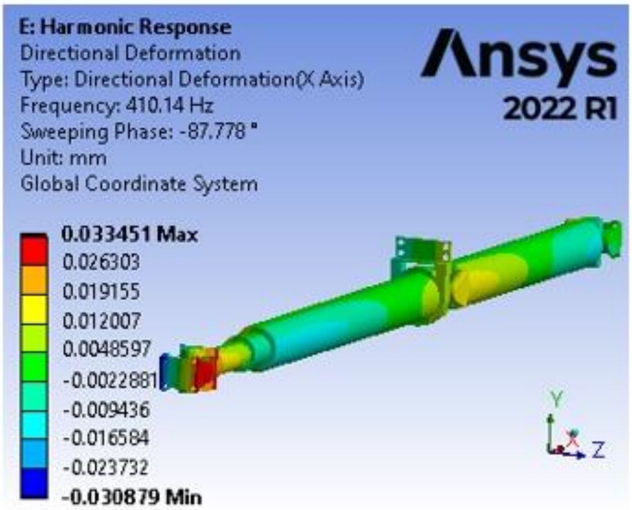


Figure 5.35: Harmonic response of the single-piece modified propeller shaft with center bearing

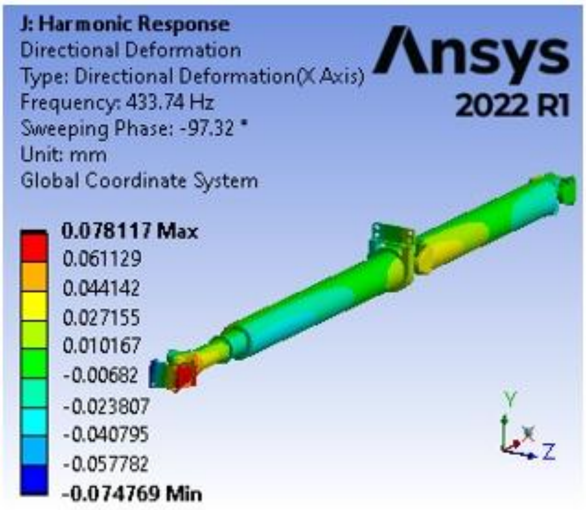
Figure 5.32-5.35 shows the harmonic frequency response of the existing and the modified propeller shaft. The frequency in which maximum amplitude is obtained for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 410.14Hz, 433.74Hz, 400Hz and 521.2Hz respectively. These values are useful to know the maximum deformation and the stress distribution over the models. As is seen from the results, the frequency value of the modified models is closer to the existing one.

5.6.3 Directional Deformation

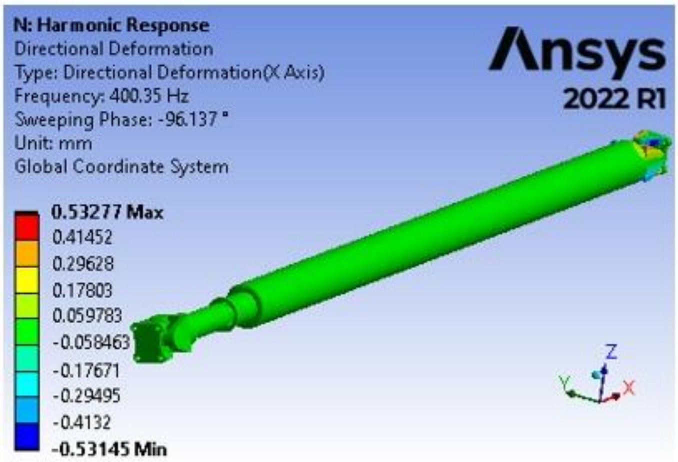
Figure 5.36(a-d) shows the directional deformation of the existing and the modified models at the frequency value in which the amplitude high.



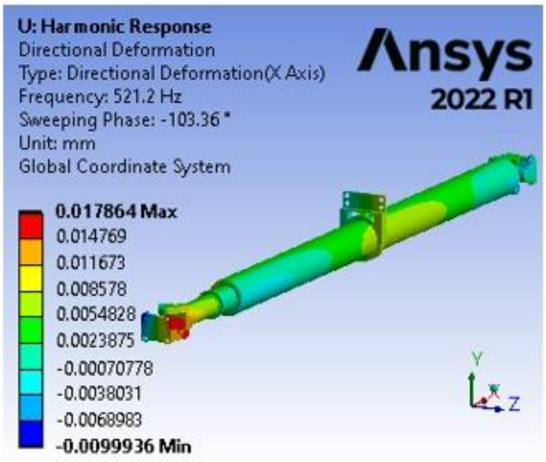
(a)



(b)



(c)



(d)

Figure 5.36: Directional deformation from the harmonic analysis for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

The directional deformation for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 0.033451mm, 0.078117mm, 0.53277mm and 0.017864mm respectively. As is seen from the results, the directional deformation of the modified models is closer to the existing one.

5.6.4 Equivalent Stress

As it is seen from Figure 5.37 (a - d), the maximum equivalent stress from harmonic analysis for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 50.33MPa, 48.154MPa, 77.772MPa and 28.72MPa respectively.

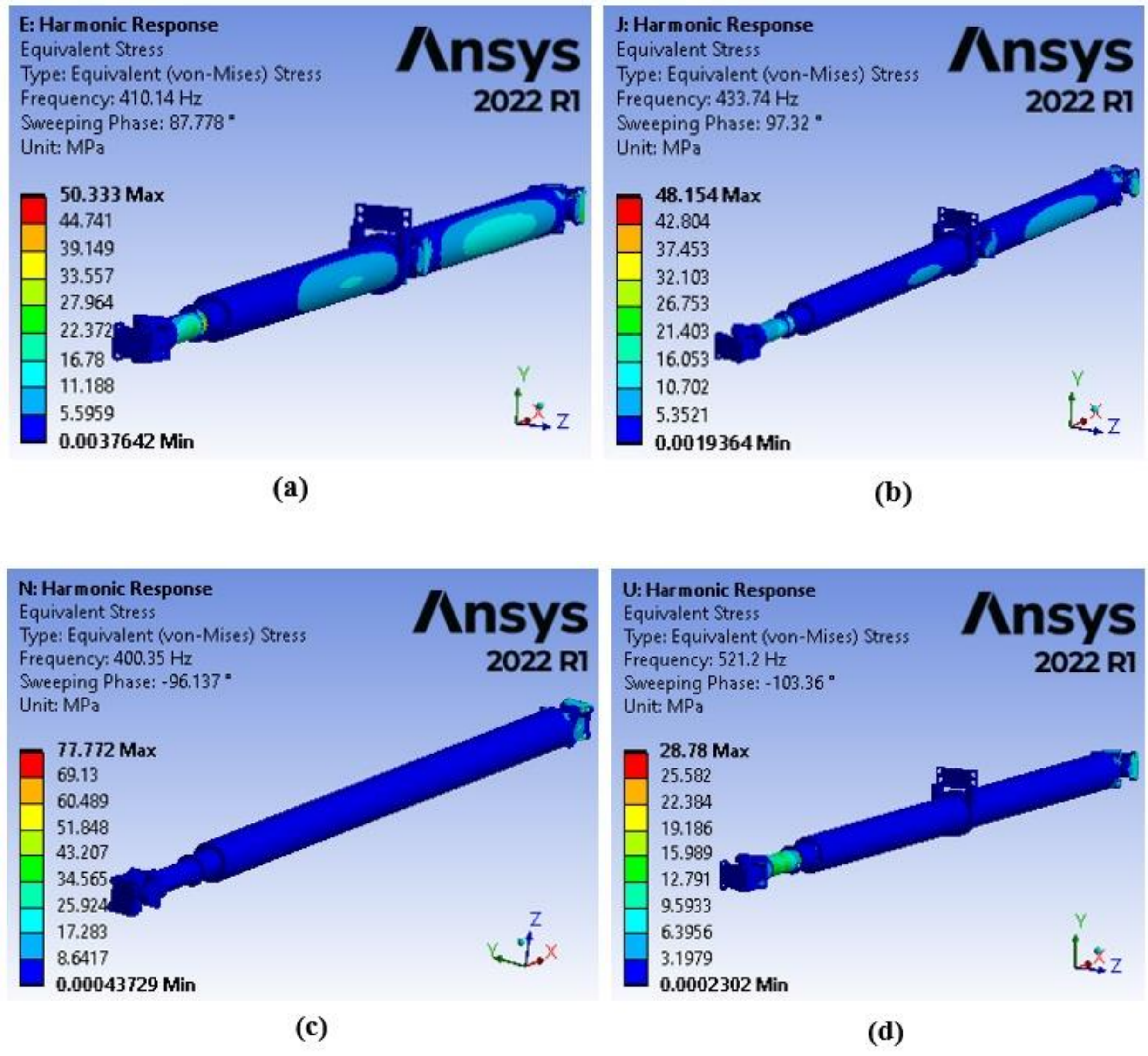


Figure 5.37: Equivalent stress from the harmonic analysis for ***(a)*** the existing model ***(b)*** two-piece modified model ***(c)*** one-piece modified model without center bearing ***(d)*** one-piece modified model with center bearing

In all cases the maximum stress is below the yield strength of the material which is 519.6MPa for alloy steel of the existing model and 755.4MPa for Carbon-Kevlar/Epoxy Hybrid Composite of the modified models. Therefore, the modified models are safe from the design aspect.

5.7 Comparison Between the Existing and the Modified Propeller Shaft

5.7.1 Comparison in terms of Weight

The existing propeller shaft is made of alloy steel whose density is 7850Kg/m^3, whereas the modified propeller shafts are made of Carbon-Kevlar/Epoxy Hybrid Composite whose density is 1570Kg/m^3. Since density and mass are directly proportional, the overall mass of the modified propeller shaft will be less than the existing propeller shaft. The mass of each model can be known from ANSYS.

Table 5.1: Comparison in terms of weight

Models	**Weight (Kg)**	**Material**	**Density (Kg/m^3)**
Existing propeller shaft	*16.2*	*Alloy steel*	*7850Kg/m^3*
Two-piece modified propeller shaft	*8.8692*	*Carbon-Kevlar/Epoxy Hybrid Composite*	*1570Kg/m^3*
Single-piece modified propeller shaft without center bearing	*8.0079*		
Single-piece modified propeller shaft with center bearing	*8.559*		

As it seen from Table 5.1 the weight of the modified propeller shafts is reduced in half compared to the existing models. The difference in weight between the modified models is because of their slight design difference. Since the two-piece modified propeller shaft has three universal joints and a center bearing, its weight is higher than the other modified models. The weight of the two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is reduced by 45.25%, 50.57% and 47.16% respectively compared to the existing propeller shaft. There is a slight weight difference between the modified models because of some additional parts. The weight of the two-piece modified propeller shaft is higher by 9.7% and 3.5% compared to one-piece modified model without center bearing and one-piece modified model with center bearing respectively. Therefore, one of the objectives is achieved. The weight reduction is helpful in terms of fuel consumption and performance.

5.7.2 Comparison in terms of Deformation

The maximum deformation obtained at the static analysis, transient analysis and harmonic analysis is considered to compare the existing propeller shaft with the modified propeller shaft.

Table 5.2: Comparison in terms of deformation

Models	***Maximum Deformation (mm)***				
	Static analysis			***Transient analysis***	***Harmonic analysis***
	At 1500rpm	***At 3000rpm***	***At 5000rpm***		
Existing propeller shaft	1.6932	1.8466	2.0729	2.0105	0.03345
Two-piece modified propeller shaft	1.1101	1.4242	1.6831	0.91962	0.07812
Single-piece modified propeller shaft without center bearing	0.0026	0.0042	0.00616	0.0042	0.53277
Single-piece modified propeller shaft with center bearing	0.0918	0.6357	1.7345	0.84994	0.01786

In the case of harmonic analysis, the frequency in which maximum amplitude is obtained is used to find the deformation. This frequency value for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 410.14Hz, 433.74Hz, 400Hz and 521.2Hz respectively. These values are useful to know the maximum deformation and the stress distribution over the models.

As it is seen from Table 5.2 the maximum deformation of the modified model is minimum compared to the existing propeller shaft. From all the modified models the Single-piece modified propeller shaft without center bearing has lower deformation in all cases.

5.7.3 Comparison in terms of Stress Distribution

Stress distribution is used to determine if a given material will yield or fracture. The yield point is the point on a stress-strain curve that indicates the limit of elastic behavior and the beginning of plastic behavior. A material is said to start yielding when the stress reaches a value known as yield strength. The maximum stress distribution obtained at the static analysis, transient analysis and harmonic analysis is considered to compare the existing propeller shaft with the modified propeller shaft.

Table 5.3: Comparison in terms of stress distribution

Models	***Maximum stress (MPa)***				
	Static analysis			***Transient analysis***	***Harmonic analysis***
	At 1500rpm	***At 3000rpm***	***At 5000rpm***		
Existing propeller shaft	55.565	312.75	450.12	446.01	50.333
Two-piece modified propeller shaft	11.036	98.03	273.12	241.9	48.333
Single-piece modified propeller shaft without center bearing	2.3057	3.6033	5.1894	3.6033	77.772
Single-piece modified propeller shaft with center bearing	29.656	265.91	349.84	60.578	28.78

As it seen from Table 5.3 the maximum stress of the modified propeller shaft is minimum compared to the existing propeller shaft. Especially Single-piece modified propeller shaft without center bearing has lower stress value in all cases which make it preferable from the other modified models. In all cases the maximum stress is below the yield strength of the material which is 519.6MPa for alloy steel of the existing model and 755.4MPa for Carbon-Kevlar/Epoxy Hybrid Composite of the modified models.

5.8 Summary

Generally, this thesis presents design and numerical analysis of a carbon-Kevlar/epoxy hybrid composite propeller shaft for Tata 207 DI Single Cabin Pickup. Three modified models were prepared in order to consider all the necessary conditions and to select the better model. These are two-piece modified propeller shaft, one-piece modified propeller shaft without center bearing and one-piece modified propeller shaft with center bearing. All the necessary dimension for the modified propeller shaft is taken from the results obtained in the design calculation. Then, using the dimension the model is prepared with full scale. The analysis which is done on ANSYS are static structural, transient structural, modal analysis, harmonic analysis and rigid dynamics.

- ✓ ***Rigid dynamics*** is used to solve the relative motion between the shaft and the joints. The propeller shaft should transmit the torque from the engine to the differential.
- ✓ ***Static analysis*** is conducted to see how the existing and the modified propeller shaft responds when subjected to a static loading.
- ✓ ***Modal analysis*** provides insights to the dynamic characteristics of the existing and modified propeller shafts. It is a linear analysis that does not utilize any excitations or loads.
- ✓ ***Harmonic analysis*** is used to determine the response of the models under a steady-state sinusoidal (harmonic) loading at a given frequency from the modal analysis.
- ✓ ***Transient analysis*** is used to determine the response of the propeller shafts to arbitrarily time-varying loads.

According the result obtained from the above analysis the Single-piece modified propeller shaft without center bearing is selected from the other modified models. These is because:

- ✓ The weight of two-piece modified model without center bearing is reduced by 45,25% compared to the existing propeller shaft. It is also the highest compared to other modified models.
- ✓ It has the lowest stress and deformation value in all analysis.
- ✓ It has a better performance in terms of vibration. It has the lowest deformation at higher frequency ranges, for instance the maximum deformation of the propeller shaft at two-piece modified model 364.59Hz is 23.295mm .which is less than the existing model and other modified models even at lower frequency.
- ✓ It has a simple design.

CHAPTER SIX
CONCLUSION AND RECOMMENDATION

6.1 Conclusion

In this thesis design and numerical analysis of a carbon-Kevlar/epoxy hybrid composite propeller shaft for Tata 207 DI Single Cabin Pickup is conducted. Modeling of the existing and the modified composite propeller shaft is carried out in CATIA V5 and its analysis is performed in ANSYS. Basic dimensions of the existing model are taken by direct measurement whereas the dimension of the composite propeller shaft is obtained from the analytical design. The analysis which is done on ANSYS are static structural, transient structural, modal analysis, harmonic analysis and rigid dynamics. Three modified models were prepared in order to consider all the necessary conditions and to select the similar model propeller shaft for compare. According the result obtained from the analysis the two-piece modified propeller shaft is selected from the other modified models for compare similarity design propeller shaft.

Key findings are summarized as follows:

- ✓ The weight of the two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is improved by 45.25%, 50.57% and 47.16% respectively compared to the existing propeller shaft.
- ✓ The modified models have better performance in terms of vibration., two-piece modified propeller shaft has the lowest deformation at higher frequency ranges, for instance it has a maximum deformation of two-piece modified model 364.59Hz is 23.295mm .
- ✓ From the static analysis the maximum equivalent stress is minimum on the modified one-piece propeller shaft without center bearing with a value of for modified two-piece propeller shaft(11.036MPa, 98.038MPa and 273.12MPa) at 1500rpm, 3000rpm and 5000rpm respectively. It is below the yield strength of the material which is 519.6MPa.
- ✓ At the maximum angular velocity, the maximum equivalent elastic strain for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 0.0044595 (0.44%), 0.0066992 (0.66%), 0.00012734 (0.013%) and 0.0085523 (0.855%) respectively.
- ✓ The maximum deformation from the dynamic analysis for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 2.0105mm, 0.91962mm, 0.0042032mm and 0.84994mm respectively.

- ✓ The frequency in which maximum amplitude is obtained for the existing model, two-piece modified model, one-piece modified model without center bearing and one-piece modified model with center bearing is 410.14Hz, 433.74Hz, 400Hz and 521.2Hz respectively.

6.2 Recommendation

Based on the results obtained from finite element analysis, carbon-Kevlar/epoxy hybrid composite is a promising material for propeller shaft since it has better mechanical and physical property in terms of total deformation, equivalent von mess stress, vibration and weight. When However, prior to the implementation of the new composite material, cost effectiveness analysis and experimental investigation are needed, in order to utilize the benefits associated with the material.

6.3 Recommendation for Future Work

The following recommendations can be taken into consideration for future work:

- ✓ In this thesis, the finite element analysis is done only on ANSYS workbench, it is recommended to validate the numerical analysis by using experimental analysis.
- ✓ The usage of high-performance computers is recommended for future work in order to get accurate results from numerical analysis.

REFERENCE

A, S. and, & B. Kerur, S. (2021). Static Analysis of Hybrid Composite Propeller Shaft. *International Advanced Research Journal in Science, Engineering and Technology*, *8*(11), 114–121. https://doi.org/10.17148/iarjset.2021.81117

Buckner, C. A., Lafrenie, R. M., Dénommée, J. A., Caswell, J. M., Want, D. A., Gan, G. G., Leong, Y. C., Bee, P. C., Chin, E., Teh, A. K. H., Picco, S., Villegas, L., Tonelli, F., Merlo, M., Rigau, J., Diaz, D., Masuelli, M., Korrapati, S., Kurra, P., … Mathijssen, R. H. J. (2016). Textile Reinforced Structural Composites for Advanced Applications. *Intech*, *11*(tourism), 13.https://www.intechopen.com/books/advanced-biometric-technologies/liveness-detection-in-biometrics

Carloni, C. (2014). Analyzing bond characteristics between composites and quasi-brittle substrates in the repair of bridges and other concrete structures. In *ELSEVIER Ltd*. Woodhead Publishing Limited. https://doi.org/10.1533/9780857097019.1.61

Chirinda, G. P., & Matope, S. (2020). The lighter the better: Weight reduction in the automotive industry and its impact on fuel consumption and climate change. *Proceedings of the International Conference on Industrial Engineering and Operations Management*, *59*(Hirsch 2011), 520–533.

Choure, A. (2017). *Design Review of Shaft for Strength and Rigidity Considerations*. *5*(02), 1740–1742.

Constantine D. Papaspyrides& Kiliaris, P. (2014). Polymer Green Flame Retardants. In *ELSEVIER*.

Czerwinski, F. (2021). Current trends in automotive lightweighting strategies and materials. *Materials*, *14*(21). https://doi.org/10.3390/ma14216631

Ganeshan, P., Raja, K., Lingadurai, K., & Kaliappan, M. (2015). Finite element analysis of alternate composite material for an automobile drive shaft. *International Journal of Applied Engineering Research*, *10*(49), 447–452.

Jadhav, S. S., Jadhav, S. S., Kavitake, A. S., & Lawand, A. A. (2019). A Theoretical Study of Critical Speed of Shaft Carrying a Single Rotor. *IOSR Journal of Engineering (IOSR JEN)*, 14–19.

Kale, V., & Secanell, M. (2018). A comparative study between optimal metal and composite rotors for flywheel energy storage systems. *Energy Reports*, *4*, 576–585. https://doi.org/10.1016/j.egyr.2018.09.003

Karthikeyan, P., Gobinath, R., Ajith Kumar, L., & Xavier Jenish, D. (2017). Design and Analysis of Drive Shaft using Kevlar/Epoxy and Glass/Epoxy as a Composite Material. *IOP Conference Series: Materials Science and Engineering*, *197*(1). https://doi.org/10.1088/1757-899X/197/1/012048

kishore, M., Keerthi, J., & kumar, V. (2016). Design and Analysis of Drive Shaft of an Automobile. *International Journal of Engineering Trends and Technology*, *38*(6), 291–296. https://doi.org/10.14445/22315381/ijett-v38p253

Kumar, A., Jain, R., & Patil, P. P. (2016). Dynamic Analysis of Heavy Vehicle Medium Duty Drive Shaft Using Conventional and Composite Material. *IOP Conference Series: Materials Science and Engineering*, *149*(1). https://doi.org/10.1088/1757-899X/149/1/012156

Madarvoni, S., & Sreekanth, R. P. S. (2022). Mechanical Characterization of Graphene—Hexagonal Boron Nitride-Based Kevlar–Carbon Hybrid Fabric Nanocomposites. *Polymers*, *14*(13). https://doi.org/10.3390/polym14132559

Nadaf, A. R., & Raikar, V. A. (2017). Design & Analysis of Composite Shaft of Passenger Vehicle. *International Research Journal of Engineering and Technology*, *4*(8). https://irjet.net/archives/V4/i8/IRJET-V4I828.pdf

Pallavi, M., Swaroop Raj, T. J., Prasad, A. S., & Madhavi, M. (2015). Experimental Investigations on Static and Dynamic Parameters of Steel and Composite Propeller Shafts for a Light Passenger Vehicle. *IOSR Journal of Mechanical and Civil Engineering*, *12*(4), 2278–1684. https://doi.org/10.9790/1684-12460107

Parshuram, D., & Mangsetty, S. (2013). Design and Analysis of Composite Hybrid Drive Shaft for Automotives. *The International Journal of Engineering And Science (IJES)*, *2*(01), 160–171.

Priyanka, P., Dixit, A., & Mali, H. S. (2019). High strength Kevlar fiber reinforced advanced textile composites. *Iranian Polymer Journal (English Edition)*, *28*(7), 621–638. https://doi.org/10.1007/s13726-019-00721-7

Radulović, J. (2020). Hybrid filament-wound materials: Tensile characteristics of (aramide fiber/glass fiber)-epoxy resins composite and (carbon fibers/glass fiber)-epoxy resins composites. *Scientific Technical Review*, *70*(1), 36–46. https://doi.org/10.5937/str2001036r

Ramchandra D Patil, D. D. M. P. (2019). Design & Analysis of Composite Drive Shaft for Automobile. *International Journal of Novel Research and Development*, *4*(1), 13–20. www.ijnrd.org

Rangaswamy, T. & Vijayarangan, S. (2005). Optimal sizing and stacking sequence ofcompositedrive shafts. *Materials Science*, *11*(2), 133–139.

S V Gopals Krishna, B V Subrahmanyam, and R. S. (2013). Finite Element Analysis and optimization of Automotive Composite. *Finite Element Analysis and Optimizati[1] J. Thomsen, "Finite Element Analysis and Optimization of Automotive Composite Drive Shaft," in Finite Element Analysis and Optimization of Automotive Composite Drive Shaf, Vol. 5, No. 7, 1990, Pp. 346–351.on of Au*, *5*(7), 346–351.

Salaisivabalan, T., & Natarajan, R. (2016). Design and Analysis of Propeller Shaft of an Automobile Using Composite Materials. *International Journal of Innovative Research in Science, Engineering and Technology*, *5*(5), 7916–7923. doi.org/10.15680

Sivakandhan, C., & Suresh Prabhu, P. (2012). Composite drive shaft is a good strength and weight saving to compare conventional materials design and analysis of e glass/epoxy composite drive shaft for automotive applications. *European Journal of Scientific Research*, *76*(4), 595–600.

Stoffels, H. (2017). Balancing driveability, NVH, and fuel consumption on automotive powertrains using integrated simulation techniques. *Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-Body Dynamics*, *231*(3), 556–567. https://doi.org/10.1177/1464419317715844

Sweety P. Mhaske, N. P. D. (2016). Failure Analysis and Design Modification of Drive Axle for ATV Car. *International Journal for Research in Applied Science & Engineering Technology (IJRASET)*, *4*(X), 560–569. www.jetir.org431

V.Jose Ananth Vino, and J. H. H. (2015). Design and Analysis of Propeller. *International Journal of Innovative Research in Science, Engineering and Technology*, *4*(8), 7311–7319. https://doi.org/10.1088/1757-899X/455/1/012018

Yusuf, A. A., Umar, S., & Shasu, U. (2015). Use of Polymer Matrix Composites for Conventional Steel Drive Shafts: A Study. *ELK Asian Pacific Journals*, *Special Issue*, 179–186.https://www.researchgate.net/profile/Abdulfatah_Yusuf/publication/314174381_use_of_polymer_matrix_composites_for_conventional_steel_drive_shafts_a_study/links/58b84cd7aca27261e51cd7c9/use-of-polymer-matrix-composites-for-conventional-steel-drive-shafts-a-

Zheng, J. H., Lin, J., Allwood, J. M., & Dean, T. (2021). A universal mass-based index defining energy efficiency of different modes of passenger transport. *International Journal of Lightweight Materials and Manufacture*, *4*(4), 423–433. https://doi.org/10.1016/j.ijlmm.2021.06.004

M.pallavi, T. joel swaroop Raj, A. syam Prasad and M. madhavi (2015). Experimental Investigations on Static and Dynamic Parameters of Steel and Composite Propeller Shafts for a Light Passenger Vehicle.*Journal of Mechanical and Civil Engineering*

Bhirud Pankaj P. & Bimlesh Kumar S. (2014).Analysis of Drive Shaft.*International Journal of Mechanical and Production Engineering*, ISSN: 2320-2092, Vol 2.

Deepti kushwaha & Gaurav saxena (2014).Optimal design and analysis of composite drive shaft for a light commercial vehicle.*International Journal of Advance Engineering and Research Development (IJAERD)*.

Robert S. Salzar (1998). Design considerations for rotating laminated metal-matrix composite Shafts.*CUNY Graduate School of Civil Engineering*, The City College of New York, NY 10031, USA.

G.V. Mahajan and Prof. V.S. Aher (2012).A Review over Current Development and Automotive Application.*International Journal of Scientific and Research Publication*, India. 9

Muni kishore, JaligamKeerthi & Vinay kumar (2016).Design and Analysis of Drive Shaft of an Automobile.IJETT, Volume 38.

Mohan.S. R, Balamuralikrishna.L, GowthamRam.G, KarthikSugumar.K, and Magesh.M (2017). Design and finite element analysis of automotive propeller shaft using composite materials.SSRG *International Journal of Mechanical Engineering.*

S. Thennarasan, Bertram nirmalhilip, A. Prabhu&T. ArulNicholas(2017). Modeling and analysis of propeller shaft with universal joints.*International Journal of Emerging Technologies in Engineering Research* (IJETER), Volume 5.

Ganesh D. shined and K.H. Munde (2017). Fatigue Analysis of Epoxy Composite Material Reinforcement on Propeller Shaft, *IJREST*.

R. Srinivasa Moorthy, Yonas Mitiku &K. Sridhar(2017).Design of Automobile Driveshaft using Carbon/Epoxy and Kevlar/Epoxy Composites.*American Journal of Engineering Research* (AJER).

Esmael Adem,Mukesh Didwania, Gurala Muralidhar R & Ermias G. Koricho (2015).Experimental Analysis of E-Glass /Epoxy & E-Glass /polyester Composites for Auto Body Panel.*American International Journal of Research in Science*.

V.JoseAnanth Vino & J.Hameed Hussain (2015) Design and Analysis of Propeller Shaft.*International Journal of Innovative Research in Science*, Engineering and Technology, Vol.4, India.

Abd Elsalam, A., Gohary, M. A., & El-Gamal, H. A. (2017), “Modal analysis on tire with respect to different parameters”, Alexandria Engineering Journal, 56(4), 345–357.https://doi.org/10.1016/j.aej.2016.09.022

Aydin, Y. N., Korkut, T., Ozaydin, O., Armakan, E., Sari, G., and Goren, A. (2021), “Numerical and Experimental Modal Analysis of Wheels and Parametric Design of Lightweight EV Wheel”, 23(68), 689–699, https://doi.org/10.21205/deufmd.2021236829

Chittilla, K., Yeola, Y., Tiwari, A., & Rajamanickam, R. (2013), ″Effect of excitation methods on experimental modal analysis of passenger car tire″, SAE Technical Papers, https://doi.org/10.4271/2013-01-2854

Eddy & Angeelal, D. M. (2017), ″Static Analysis of Plate Made of Carbon /Epoxy Composite Material by using FEA″, 06(04), 795‒798.

Hatti, P. S., Somanakatti, A. B., & Rakshith, M. (2021), “Vibration and dynamic characteristics of an automobile wheel: Prediction and verification”, AIP Conference Proceedings, 2341(May). https://doi.org/10.1063/5.0050177

Hoe, Y. S. (2015), ″Finite element method″, 188‒195. https://en.wikipedia.org/w/index.php?title=Finite_element_method&oldid=891797554

Liao, Y., Huang, H., Chang, G., Luo, D., Xu, C., Wu, Y., & Tang, J. (2022), “Research on Low Frequency Noise Control of Automobiles Based on Acoustic Metamaterial”

Reddy, P. R., Ramalakshmi, P., Kumar, K. N., & Head, P. &. (2007), “Static Analysis of Plate Made of Carbon/Epoxy Composite Material by Using ANSYS”, International Journal of Innovative Research in Science, Engineering and Technology (An ISO Certified Organization), 3297(8). https://doi.org/10.15680/IJIRSET.2015.0410053

Sharma, A., Ali, M., Bansal, R. A. M., Khan, A. M., & Khan, A. (2020), “Weight Reduction of Wheel Rim Using ‘Peek’ Composites”, 20(2), 191–201.

Asmamaw Gebresilassie (2012). Design and analysis of Composite Drive Shaft for Rear-Wheel-Drive Engine, *International Journal of Scientific & Engineering Research*. Volume 3, Issue 5, India.

Akula Jagadeesh Kumar (2007).Optimum Design and Analysis of a Composite Drive Shaft for an Automobile. Sweden.

R.P. KumarRomp Charla (2012).Design and Optimization of Driveshaft with Composite Materials.*International Journal of Engineering Research & Technology*.Vol. 1.

Anirudha P. Jangam and Vishavjit L. Bhanavase (2015).Design and Analysis of Drive Shaft in Automotive Using Composite Material.ARDIJEET.

Gyadari Ramesh and G. Chandra Mohan Reddy (2015).Structural and Modal Analysis of Composite Material Shaft with Damping Material.*International Journal for Innovative Research in Science &Technology (IJIRST)*, India.

Vinodh Kumar S, Sampath V and Baskar P.(2015). Analysis of Propeller Shaft for Composite Materials: School of Mechanical and Building Sciences, VIT University, Vellore, India.

Satendra Kumar (2017).Study of design parameters of propeller shaft of heavy-dutytrucks.*National institute of technology*.Jamshedpur.

G.Devendhar Rao,K.Srinivasa Reddy, P.Raghavendra Rao& P. Madusudana (2017). Mechanical Properties of E-Glass Fiber Reinforced Epoxy Composites with SnO2 and PTFE. *International Journal of Emerging Research in Management & Technology*.

M. S. EL-Wazery, M. I. EL-Elamy& S. H. Zulfikar (2017). Mechanical Properties of Glass Fiber Reinforced Polyester Composites.*International Journal of Applied Science and Engineering*, Egypt.

Vishal S.& Laukik B. (2016).Experimental investigation of mechanical properties of glass fiber/epoxy composites with variable volume fraction.*IARJSET*.

J Narsaiah, K Venkatesh & P Avinash (2017). Design & Analysis of Composite Drive Shaft Made of Different Materials for Application in Automobiles.*International Journal for Scientific Research & Development* (IJSRD) Vol. 5.

VirendraV.M. and Anup B. Patel (2015).Design, Analysis and Optimization in Automobile Drive Shaft, *International Journal for Innovative Research in Science &Technology (IJIRST)*. 26.

Timoshenko SP and Gere JM (1963).Theory of Elastic Stability.New York, McGraw-Hill, pp.500-509.

V. S. Bhajantri, S. C. Bajantri, A. M. Shindolkar & S. S. Amarapure (2014).Design and Analysis of Composite Drive Shaft, *International Journal of Research in Engineering and Technology*.

Lelisa Gezu(2019).Design and analyze a composite driveshaft for power transmission to get maximum efficiency, weight reduction, increasing comfort, decrease noise and part complexity of the vehicle.

B. James Prasad Rao, D. V Srikanthb (2016). Design and analysis of automotive composite propeller shaft using FEA. Proceeding 3673-3679.

M. Mahesh Babu & N. Ramesh (2018).FAE simulation of composite propeller shaftof automobile.*International journal of current engineering and scientific research* (IJCESR).

Milon Hossain, M. A. Khan, R. A. Khan, Abu Bakar Siddiquee &Tauhidul Islam(2015). carbon/kevlar reinforced hybrid composite: impact of matrix variation

Mohammed Hisham, Mohammed Fahaduddin, Mohammed Azhar Khan, Ashok B. C,Prashant Kumar Shrivastava (2019). A Research on Kevlar and Hybrid Kevlar Composites; A Report

APPENDIX

Appendix: ANSYS Results

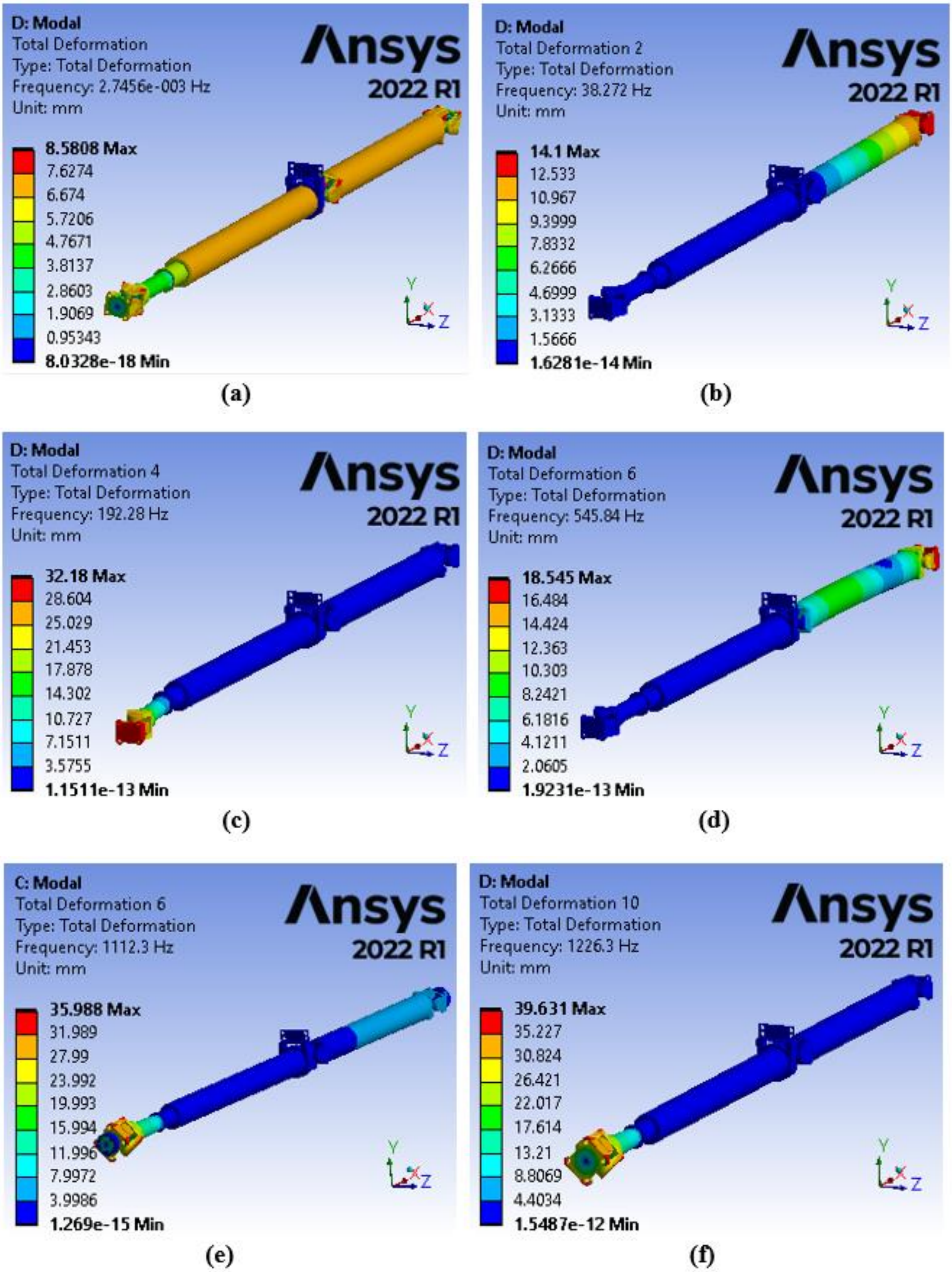


Figure A.1: Total deformation of the existing propeller shaft at different mode shapes

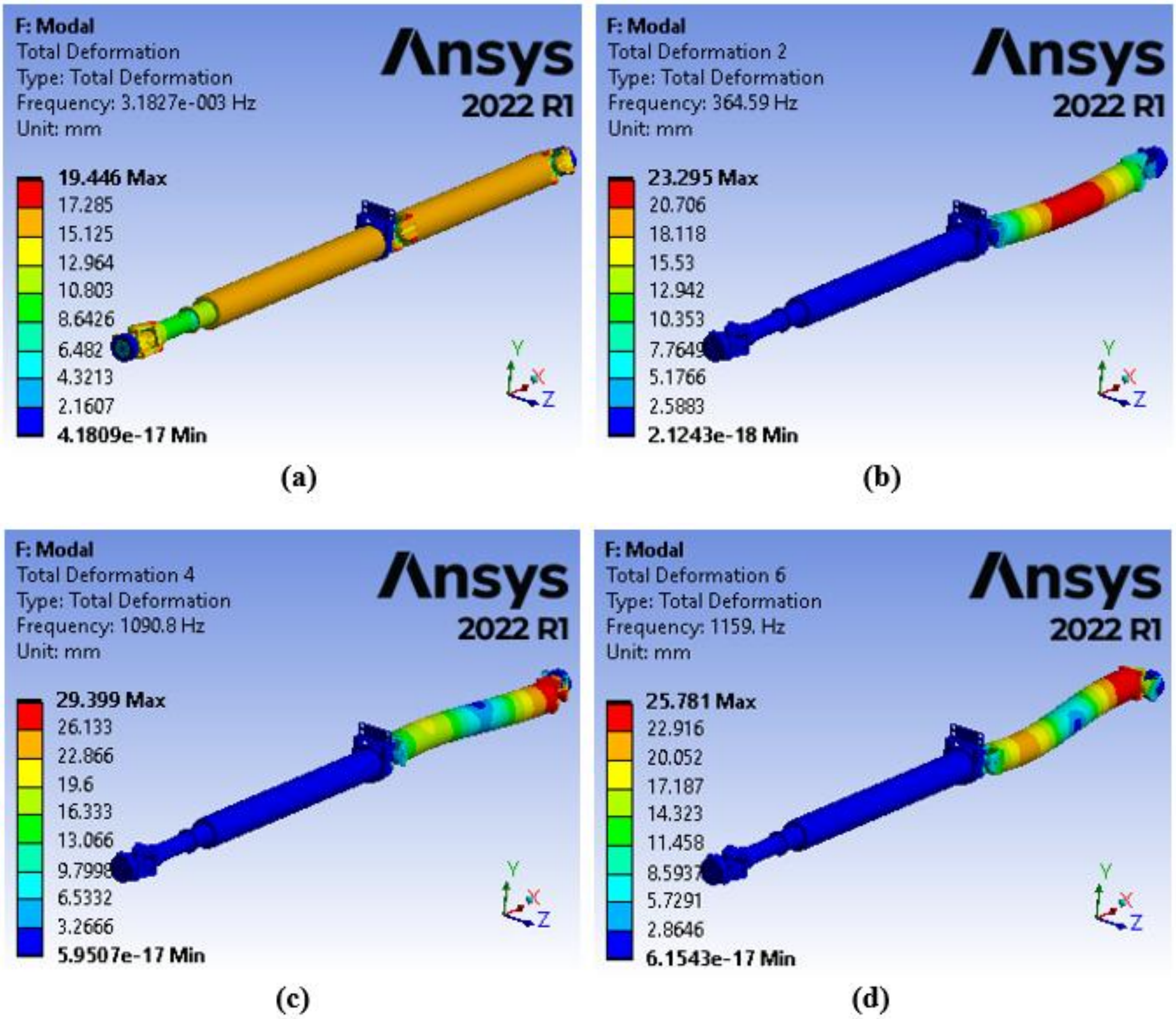
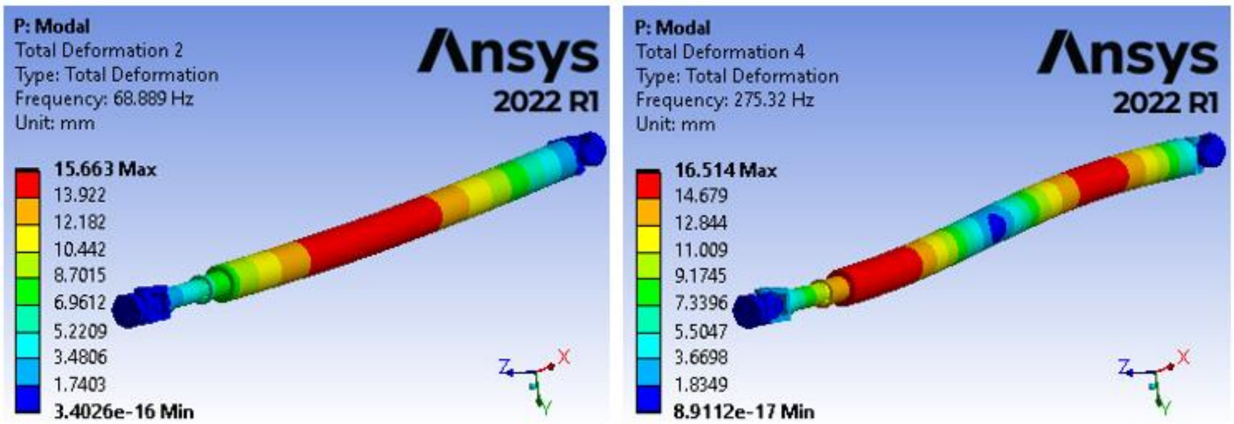
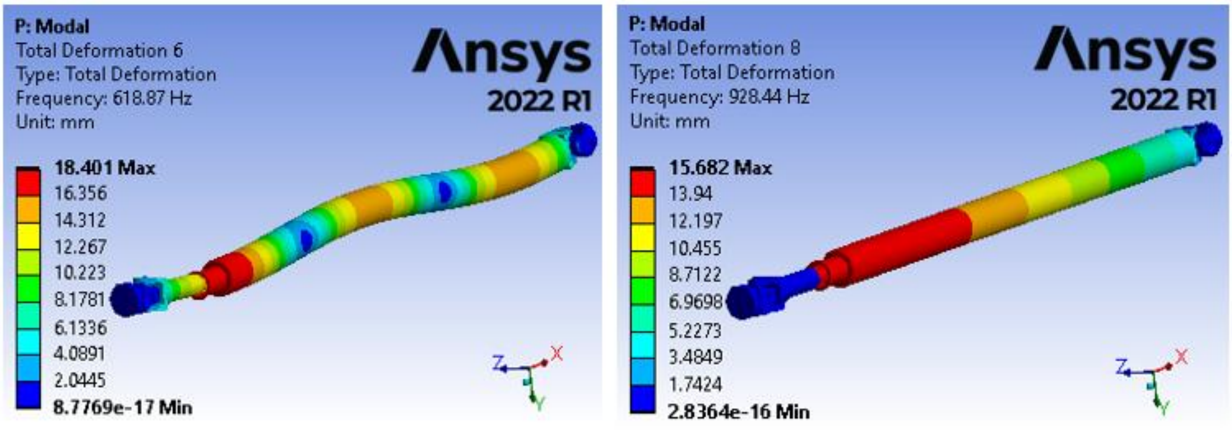


Figure A.2: Total deformation of the Two-piece modified propeller shaft at different mode shapes

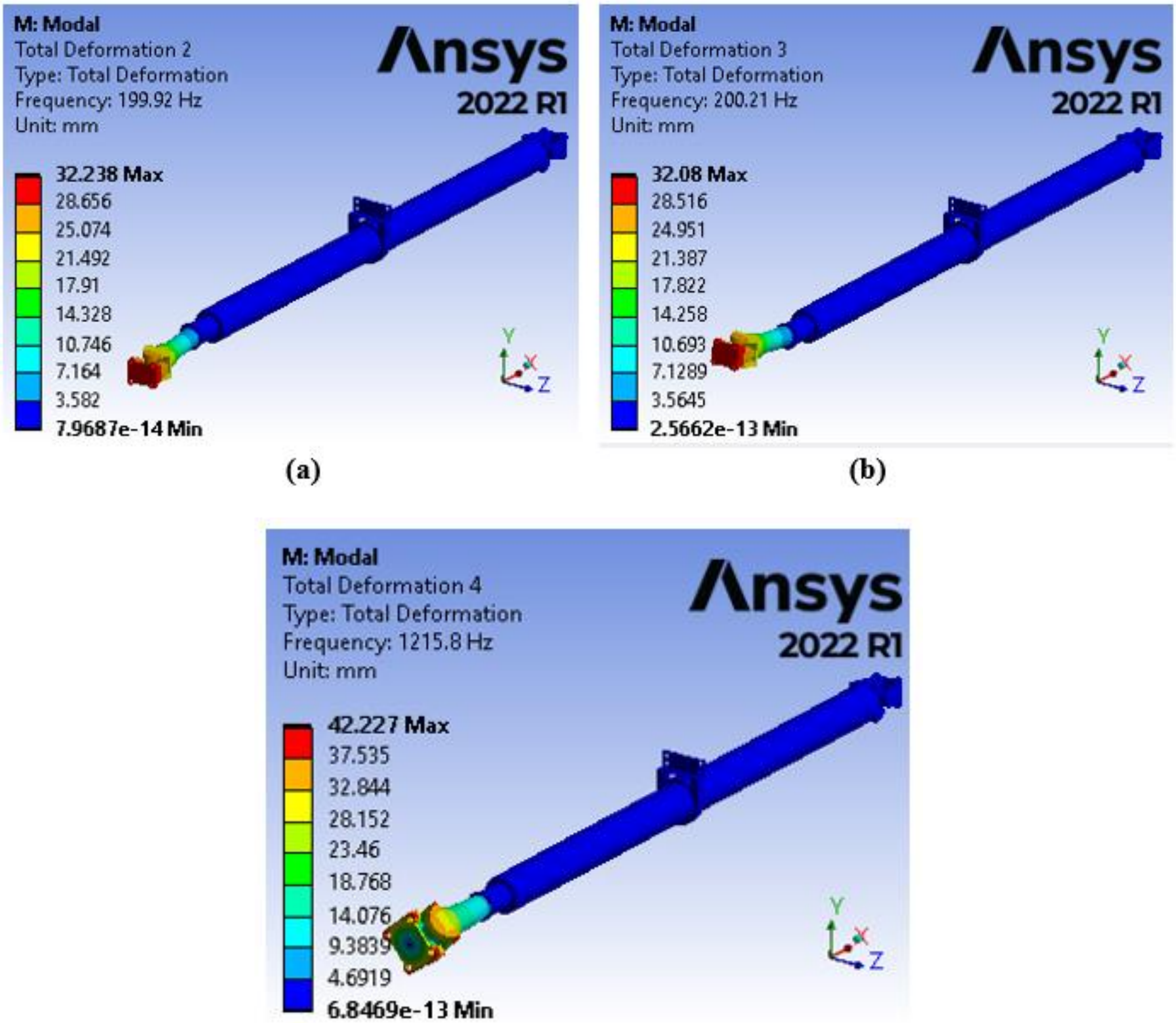


(a) (b)



(c) (d)

Figure A.3: Total deformation of the single-piece modified propeller shaft without center bearing at different mode shapes



(c)

Figure A.4: Total deformation of the single-piece modified propeller shaft with center bearing at different mode shapes