DESIGN OF REAR AXLE SHAFT FOR LIGHT WEIGHT GREEN VEHICLE

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Declaration

This thesis work is my original work and has not been presented for a degree in any other university, and that all source of material is duly acknowledged.

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Nomenclature

T_{w-} Torque on the wheel

F_{l-} Lateral force during skidding

F_{y-} Vertical force due to the vehicle weight

F_{r-} Reaction force on the wheel

$T_{e\,max-}$ Maximum engine torque

i_{f-} Final drive ratio

K_{d-} Dynamic factor

m_{a-} Automobile mass accounted for the driving axle

w_{t-} Weight transferred

g- Gravitational force

h- Height of the center of gravity from the road surface datum

M – Moment due to the lateral force

d- The axle shaft diameter

b- The overhanging length

$\varphi -$ Coefficient of adhesion

R – Radius of turn of the road

V – Vehicle speed

$K_{dr\,-}$ Dynamic factor of the road

E – Modulus of elasticity

G – Modulus of rigidity

S_{ut-} Ultimate strength

ρ – Density

$\vartheta -$ Poison ration

$\gamma -$ Axial Poisson's ratio

V_{f-} Fiber volume fraction

V_{m-} Matrix volume fraction

E_{11-} longitudinal elastic modulus

E_{22-} Transverse elastic modulus

A – Area

$\alpha -$ Strain to failure (%)

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Abstract

Vehicle weight reduction is a known strategy to address growing concerns about greenhouse gas emissions and fuel use by passenger vehicles.

This paper intended to reduce the weight of rear axle shaft by replacing the existing material solid alloy steel with the material, hybrid of AISI4140 steel with carbon fiber/epoxy material. This thesis starts with by taking the measurement of commercially available rear axle shaft dimensions, which is 35mm diameter, 5.3064kg mass for single shaft and 0.66m length made of chrome molybdenum semi-floating steel shaft. Analyzing the equivalent stress, total deformation and mass of the shaft for making the values as a baseline for the light weight design. Obtaining the analysis for hollow shaft and hybrid shaft with 10, 12, 14, 16, 18, 20 (mm) diameter hole using ANSYS 15 software and comparing the result to the baseline shaft value determined. The values of stress to percentage reduction ratio can be selected as optimum solution for the thesis.

The weight of the shaft could be significantly reduced by removing the inside of the shaft and replacing it with a lower density material. Based on the study performed, the weight of the axle shaft could be reduced by more than 10 % without significant amount of stress raise. Therefore, 14mm cylindrical hole from AISI4140 alloy steel axle shaft replaced by 14mm carbon fiber epoxy shaft is the optimum choice. This can achieve 12.45% mass reduction which is 1.32kg of unnecessary mass can be reduced from the vehicle weight with 0.8% stress rise, the stress is kept between 248.49 MPa - 241.61 MPa.

Key words: Hybrid, Semi-floating, full-floating, three-quarter floating, weight reduction, light weight design, shaft, fiber, epoxy, steel.

CHAPTER ONE

1 Introduction

1.1 Axle

On cars and trucks, several senses of the word "axle" occur in casual usage, referring to the shaft itself, it's housing, or simply any transverse pair of wheels. An **axle** is a central shaft for a rotating wheel or gear. On wheeled vehicles, the axle may be fixed to the wheels, rotating with them, or fixed to the vehicle, with the wheels rotating around the axle[1]. Strictly speaking, a shaft which rotates with the wheel, being either bolted or splined in fixed relation to it, is called an "axle" or "axle shaft". However, in looser usage an entire assembly including the surrounding "axle housing" (typically a casting) is also called an "axle". An even broader (somewhat figurative) sense of the word refers to every pair of parallel wheels on opposite sides of the vehicle, regardless of their mechanical connection to each other and to the vehicle frame or body. Thus, transverse pairs of wheels in an independent suspension may be called "an axle" in some contexts. This very loose definition of "axle" is often used in assessing toll roads or vehicle taxes, and is taken as a rough proxy for the overall weight-bearing capacity of a vehicle, and its potential for causing wear or damage to roadway surfaces.

1.1.1. Vehicle axles

Axles are an integral component of most practical wheeled vehicles. In a live-axle suspension system, the axles serve to transmit driving torque to the wheel, as well as to maintain the position of the wheels relative to each other and to the vehicle body. The axles in this system must also bear the weight of the vehicle plus any cargo. A non-driving axle, such as the front beam axle in heavy duty trucks and some 2-wheel drive light trucks and vans, will have no shaft, and serves only as a suspension and steering component. Conversely, many front wheel drive cars have a solid rear beam axle.

In other types of suspension systems, the axles serve only to transmit driving torque to the wheels; the position and angle of the wheel hubs is an independent function of the suspension system. This is typical of the independent suspension found on most new cars and SUV's, and on the front of many light trucks. These systems still have a differential, but it will not have attached axle housing tubes. It may be attached to the vehicle frame or body, or integral in a transaxle. The axle shafts (usually velocity type) then transmit driving torque to the wheels. Like a full floating axle system, the drive shafts in a front wheel drive independent suspension system do not support any vehicle weight.

1.1.2. Structural futures

In split-axle designs, the wheel on each side is attached to a separate shaft. Modern passenger cars have split drive axles. In some designs, this allows independent suspension of the left and right wheels, and therefore a smoother ride. Even when the suspension is not independent, split axles permit the use of a differential, allowing the left and right drive wheels to be driven at different speeds as the automobile turns, improving traction and extending tire life.

A tandem axle is a group of two or more axles situated close together. Truck designs will use such a configuration to provide a greater weight capacity than a single axle. Semi-trailers usually have a tandem axle at the rear. Axles are typically made from SAE grade 41xx steel or SAE grade 10xx steel. SAE grade 41xx steel is commonly known as chrome-molybdenum steel (or "chrome-moly") while SAE grade 10xx steel is known as carbon steel. The primary differences between the two are that chrome-moly steel is significantly more resistant to bending or breaking, but is very difficult to weld with tools normally found outside a professional welding shop[2].

In rear wheel drive cars and trucks, the engine turns a driveshaft (also called a propeller shaft or tail shaft) which transmits rotational force to a drive axle at the rear of the vehicle. The drive axle may be a live axle, but modern rear wheel drive automobiles generally use a split axle with a differential. In this case, one half-axle or half-shaft connects the differential with the left rear wheel, a second half-shaft does the same with the right rear wheel; thus, the two half-axles and the differential constitute the rear axle.

Some simple vehicle designs, such as leisure go-karts, may have a single driven wheel where the drive axle is a split axle with only one of the two shafts driven by the engine, or else have both wheels connected to one shaft without a differential (kart racing). However, other go-karts have two rear drive wheels too.

1.1.3. Dead axle

A dead axle, also called lazy axle, is not part of the drive train but is instead free-rotating. The rear axle of a front-wheel drive car is usually a dead axle. Many trucks and trailers use dead axles for strictly load-bearing purposes. A dead axle located immediately in front of a drive axle is called a pusher axle. A tag axle is a dead axle situated behind a drive axle. Dead axles are also found on semi-trailers, farm equipment, and certain heavy construction machinery serving the same function. On some vehicles (such as motor coaches), the tag axle may be steerable. In some designs the wheels on a lazy axle only come into contact with ground when the load is significant, thus saving unnecessary tire wear.

Rear axle construction

In cases where the rear suspension is non-independent, the type of axle used is either a dead axle or a live axle. The former only has to support the weight of the vehicle, where the latter has to fulfill this task and, in addition, contain a gear and shaft mechanism to drive the road wheels[3].

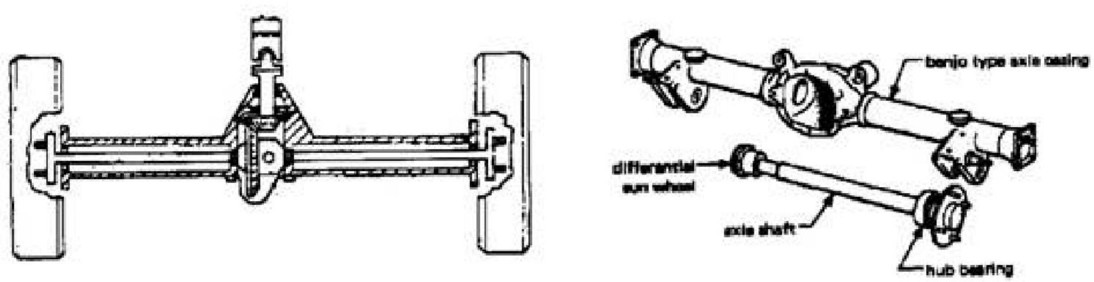


Figure 1: rear axle shaft construction[3]

1.1.4. Axle shafts

The axle shaft (half shaft) transmits the drive from the differential sun wheel to the rear hub. The arrangement of a simple rear axle can be seen in the figure, the road wheel attached to the end of the half shaft, which in turn is supported by bearing located in the axle casing. The diagram illustrates the forces acting on the rear axle assembly under an under different operating conditions.

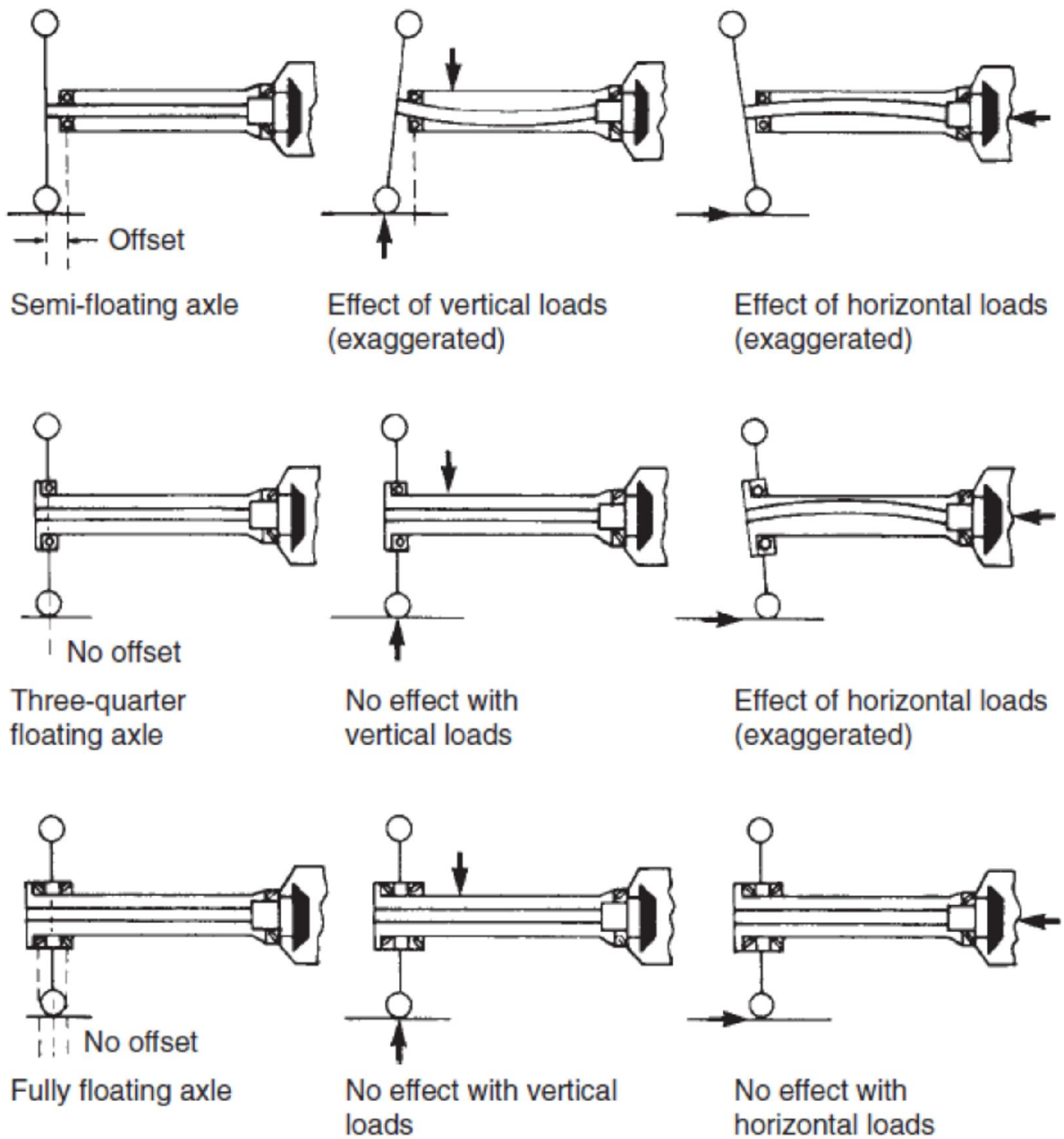


Figure 2: Force acting on axle shaft in different operating condition [4]

1.1.5. Operating conditions

The total weight of the rear of the vehicle may exert a bending action on the half shaft. Furthermore, there is a tendency for the overhanging section of the shaft to be subject to a shearing force. During cornering a side force acts upon the road wheel which imposes a bending load and an end thrust becomes a 'pull'. A side force also tends to bend the overhanging section of the half shaft. Finally, under driving conditions the half shaft has to transmit the driving torque which subjects the shaft to torsional stress[3].

Stresses

The various types may be compared by considering the stresses the shaft has to resist fig 2 shows a line sketch of a simple shaft which is subjected to[5]:

1- torsional stress due to driving and braking torque.
2- Shear stress due to the weight of the vehicle
3- Bending stress due to the weight of the vehicle
4- Tensile and compressive stress due to cornering forces.

1.1.6. Types of axles

Axle shafts are divided into three main groups depending on the stresses to which the shaft is subjected:

- Semi-floating
- Three-quarter floating
- Fully floating.

1.1.6.1 Semi-floating

Fig.3 shows a typical mounting of an axle shaft suitable for light cars. A single bearing at the hub end is fitted between the shaft and the casing, so the shaft will have to resist all the stresses previously mentioned. To reduce the risk of fracture at the hub end (this would allow the wheel to fall off), the shaft diameter is increased. Any increase must be gradual, since a sudden change in cross-sectional area would produce a stress-raiser and increase the risk of failure due to fatigue. (Fatigue may be defined as breakage due to continual alteration of the stress in the material)[3].

Although the final-drive oil level is considerably lower than the axle shaft, the large amount of 'splash' would cause the lubricant to work along the shaft and enter the brake drum. Sealing arrangements normally consists of an oil retainer fitted at the hub end (the lip of the seal is positioned towards the final drive). The half shaft in this assembly required to be able to withstand the torsion load involved in driving the road wheel, and bending loads in both the horizontal and vertical planes plus the percentage of car weight on the wheel.

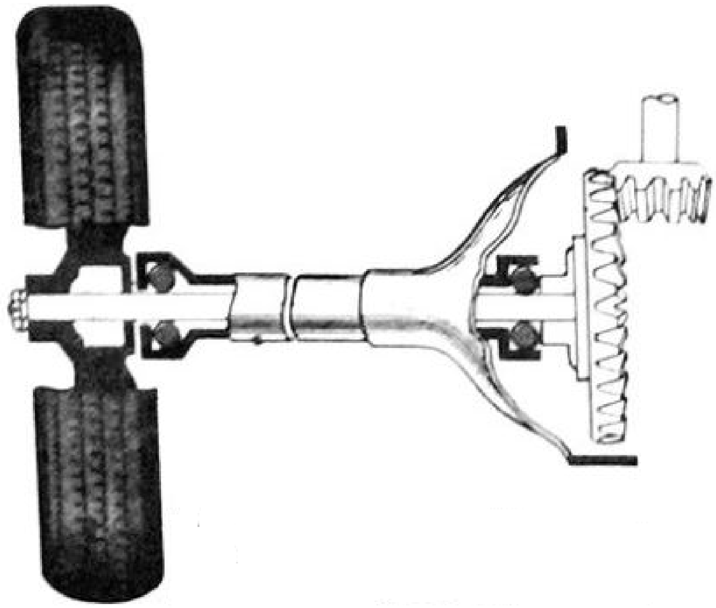


Figure 3: Semi-floating axle shaft structure[5]

1.1.6.2 Three-quarter floating

Having defined the semi-and the fully floating shaft, any alternative between the two may be regarded as a three-quarter floating shaft. Fig.4 shows a construction which has a single bearing mounted between the hub and the casing. The main shear stress on the shaft is relieved but all other stresses still have to be resisted. The half shaft must withstand bending loads due to side thrust when cornering and, of course, at the same time transmit driving torque[3].

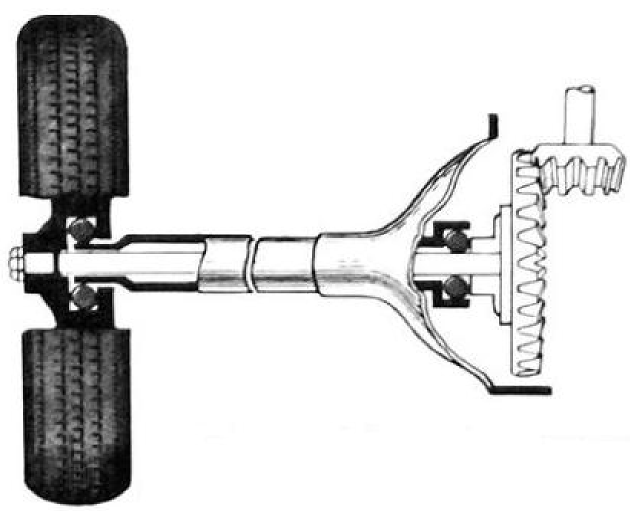


Figure 4:Three-quarter floating axle shaft structure [5]

1.1.6.3 Fully floating

This is generally fitted on commercial vehicles where torque and axle loads are greater.

The construction shown in Fig.5 consists of and independently mounted hub which rotates on two bearings widely spaced on the axle casing. This arrangement relieves the shaft of all stresses except torsional, so the construction is very strong. Studs connecting the shaft to the hub transmit the drive and when the nuts on theses studs are removed, the shaft may be withdrawn without jacking up the vehicle. The shaft is to transmit only the driving torque to the rear wheel[3].

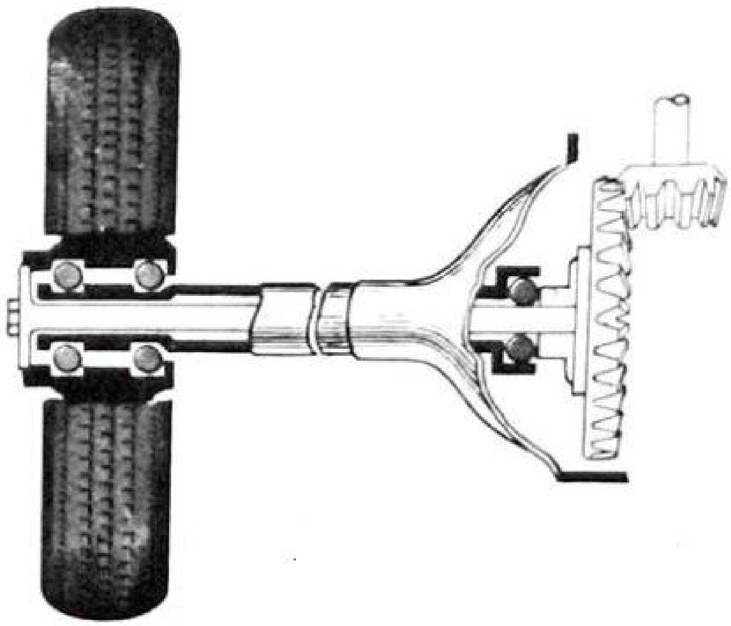


Figure 5: Full-floating axle shaft structure [5]

1.1.7. Axle Shaft Seal

A round seal located at the point where an axle shaft goes into the vehicle's differential or transmission. It prevents fluid from spilling out of the gearbox as the axle shaft rotates. In some vehicles, the axle shaft seal also helps to keep the axle shaft in proper alignment. These are natural wear items and should be replaced any time they begin leaking, or when a shaft is removed for other reasons. Also known as axle gaskets.

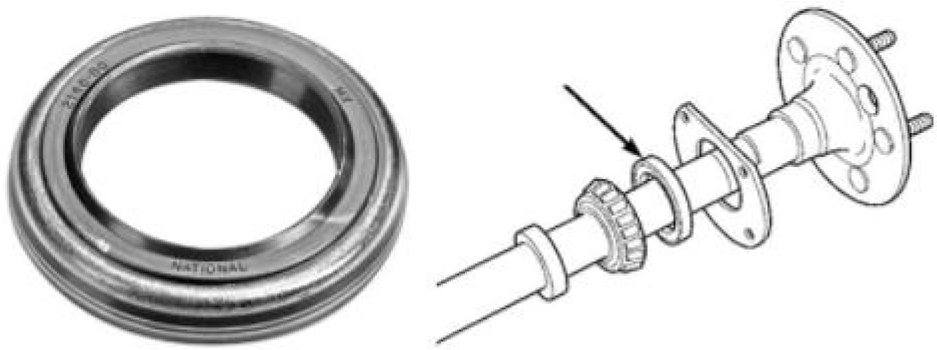


Figure 6: axle shaft seal

1.1.8. Composite material

The advanced composite materials such as graphite, carbon, Kevlar and Glass with Suitable resins are widely used because of their high specific strength (strength/density) and high specific modulus (modulus/density). Advanced composite materials seem ideally suited for long, power driver shaft (propeller shaft) applications. Their elastic properties can be tailored to increase the torque they can carry as well as the rotational speed at which they operate. The drive shafts are used in automotive, aircraft and aerospace applications. The automotive industry is exploiting composite material technology for structural components construction in order to obtain the reduction of the weight without decrease in vehicle quality and reliability. It is known that energy conservation is one of the most important objectives in vehicle design and reduction of weight is one of the most effective measures to obtain this result. Actually, there is almost a direct proportionality between the weight of a vehicle and its fuel consumption, particularly in city driving. Composites consist of two or more materials or material phases that are combined to produce a material that has superior properties to those of its individual constituents. The constituents are combined at a macroscopic level and or not soluble in each other. The main difference between composites, where as in alloys, constituent materials are soluble in each other and form a new material which has different properties from their constituents.

1.1.8.1 Classification of Composite Materials

Composite materials can be classified as

- Polymer matrix composites
- Metal matrix composites

- Ceramic Matrix

Technologically, the most important composites are those in which the dispersed phase is in the form of a fiber. The Design of fiber-reinforced composites is based on the high strength is the ratio between strength and density. Specific modulus is the ratio between strength and density. Specific modulus is the ratio between modulus and density. Fiber length has a great influence on the mechanical characteristics of a material [6]. The fibers can be either long or short. Long continuous fibers are easy to orient and process, while short fibers cannot be controlled fully for proper orientation. Long fibers provide many benefits over short fibers. These include impact resistant, low shrinkage, improved surface finish and dimensional stability. However short fiber provide low cost are easy to work with and have fast cycle time fabrication procedures. The principal fibers in commercial use are various types of glass, carbon, graphite, Kevlar. All these fibers can be incorporated into a matrix either in continuous lengths or in discontinuous lengths as shown in the Fig. The matrix material may be a plastic or rubber polymer, metal or ceramic. Laminate is obtained by stacking a number of thin layers of fibers and matrix consolidating them to the desired thickness. Fiber orientation in each layer can be controlled to generate a wide range of physical and mechanical properties for the composite laminate [7].

1.1.8.2 Properties of Composite Materials

The physical properties of composite materials are generally not isotropic (independent of direction of applied force or load) in nature, but rather are typically orthotropic (depends on the direction of the applied force or load). For instance, the stiffness of a composite panel will often depend upon the orientation of the applied forces and/or moments. Panel stiffness is also dependent on the design of the panel. In contrast, isotropic materials (for example, aluminum or steel), in standard wrought forms, typically have the same stiffness regardless of the directional orientation of the applied forces and/or moments. While, composite materials exhibit different properties in different directions. The relationship between forces/moments and strains/curvatures for an isotropic material can be described with the following material properties: Young's Modulus, the Shear Modulus and the Poisson's ratio, in relatively simple mathematical relationships. For the anisotropic material, it requires the mathematics of a second order tensor and up to 21 material property constants. For the special case of orthogonal isotropy, there are three different material property constants for each of Young's Modulus, Shear Modulus and Poisson's ratio a total of 9 constants to describe the relationship between forces/moments and strains/curvatures[7].

1.1.8.3 Advantages of Composites Over the Conventional Materials

- High strength to weight ratio
- High stiffness to weight ratio
- High impact resistance
- Better fatigue resistance
- Improved corrosion resistance
- Good thermal conductivity
- Low coefficient of thermal expansion. As a result, composite structures may exhibit a better dimensional stability over a wide temperature range.
- High damping capacity.

1.1.8.4 Limitations of composites

- Mechanical characterization of a composite structure is more complex than that of metallic structure
- The design of fiber reinforced structure is difficult compared to a metallic structure, mainly due to the difference in properties in directions
- The fabrication cost of composites is high
- Rework and repairing are difficult
- They do not have a high combination of strength and fracture toughness as compared to metals
- They do not necessarily give higher performance in all properties used for material Selection.

1.1.8.5 Applications of composites

The common applications of composites are extending day by day. Nowadays they are used in medical applications too. The other fields of applications are,

- Automotive: Drive shafts, clutch plates, engine blocks, push rods, frames, Valve guides, automotive racing brakes, filament–wound fuel tanks, fiber Glass/Epoxy leaf springs for heavy trucks and trailers, rocker arm covers, suspension arms and bearings for steering system, bumpers, body panels and door

- Aircraft: Drive shafts, rudders, elevators, bearings, landing gear doors, panels and floorings of airplanes etc.
- Space: payload bay doors, remote manipulator arm, high gain antenna, antenna ribs and struts etc.
- Marine: Propeller vanes, fans & blowers, gear cases, valves &strainers, condenser shells.
- Chemical Industries: Composite vessels for liquid natural gas for alternative fuel vehicle, racked bottles for fire service, mountain climbing, underground storage tanks, ducts and stacks etc.
- Electrical & Electronics: Structures for overhead transmission lines for railways, Power line insulators, Lighting poles, Fiber optics tensile members etc.[7]

1.1.9. Carbon fiber

The earliest commercial use of carbon fibers is often attributed to Thomas Edison's carbonization of cotton and bamboo fibers for incandescent lamp filaments. However, practical commercial use of carbon fibers for reinforcement applications began in the late 1950s with the pursuit of improved ablative materials for rockets. Union Carbide marketed a carbonized rayon based fabric in the early 1960s . DuPont's work with "black Orlon" in the late 1950s showed that acrylics could be thermally stabilized, while Shindo in Japan and Watt et al. in the United Kingdom demonstrated that, by using tension through the carbonization process, high mechanical properties could be realized. Activity increased rapidly during the 1960s and 1970s to improve the performance/price ratio of carbon fibers. Much of this effort focused on evaluation of various precursors, since carbon fiber can be made from almost anything that yields a quality char upon pyrolysis. Donnet and Bansal present a good overview of various researchers' efforts to evaluate different precursors, including PAN (polyacrylonitrile), pitch, rayon, phenol, lignin, imides, amides, vinyl polymers, and various naturally occurring cellulosic materials.

Overall carbon fiber demand grew to approximately 1000 metric tons by 1980, fueled primarily by the aerospace industry, with the sporting goods industry taking some excess capacity and off-specification fiber. Polyacrylonitrile based carbon fiber usage had exceeded all other precursors at that time. This was a surprise to some, since the anticipation in the late 1970s had been that the significantly lower raw material price and higher char yield of pitch would result in the winning combination. However, higher processing costs are required to make a spin able pitch, so better overall properties for PAN fibers resulted in their dominance. Rayon was relegated to third place, despite having a lower raw material cost, because inferior properties and a low char yield (20 to 25%) after carbonization made for a higher overall cost. Properties can be improved by stress graphitization at high temperatures, but this increases cost further, making the fiber even less desirable. Rayon is still used today for insulating and ablative applications but not for structural applications. By the mid-1990s, a new cost-effective, PAN based carbon fiber made from a modified textile precursor was being aggressively promoted by companies like Zoltek and Fortafil for commercial applications. In 1995, one manufacturer announced the goal of reaching a price level of $5/ lb ($11/kg) by the year 2000, which brought a lot of attention to and greatly accelerated application development (Ref 6). An overall trend of improved performance/price ratio for both pitch and PAN fiber manufacturers has sustained this growth. Carbon fiber demand has grown to an estimated 16 _ 106 kg (35 _ 106 lb) per year. Usage in 1997 was estimated at 30% aerospace, 30% sporting goods, and 30% commercial/ industrial applications, with the industrial applications poised for the greatest growth[7].

1.1.9.1 Manufacture of Carbon Fibers

Precursor sources used, in order of volume, are PAN, pitch, and rayon. Although the specific processing details for each recursor is different, all follow a basic sequence involving spinning, stabilization, carbonization, and application of a finish or sizing to facilitate handling, as shown in Fig.7. Discontinuous carbon fiber whiskers are also now produced in a batch process from

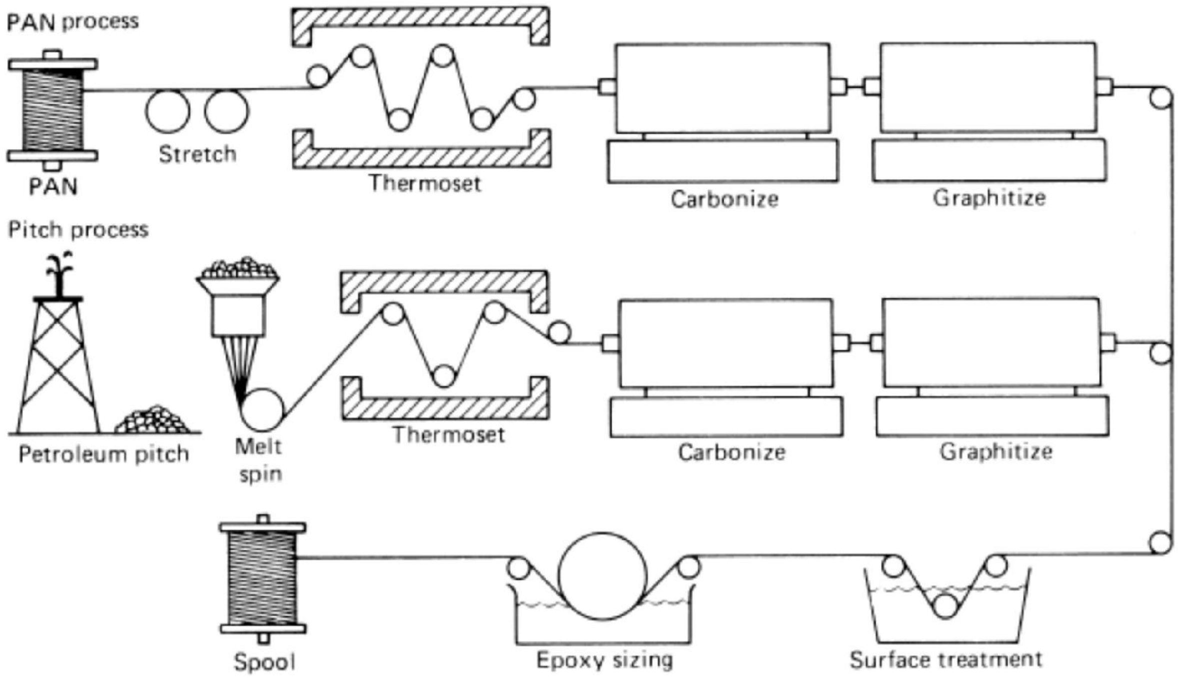


Figure 7:The processing sequence for polyacrylonitrile (PAN) and mesophase-pitch-based precursor fibers shows the similarities for the two processes. [7].

Fig. 1 The processing sequence for polyacrylonitrile (PAN) and mesophase-pitch-based precursor fibers shows the similarities for the two processes.

1.2. Statement of the problem

The rear axle shaft is designed to transmit torque load from the engine to the rear wheels and also can carry the weight of vehicle in semi-floating arrangement. The material used in commercially available shaft design are high strength alloy steels which has higher density material, result for rising the mass of vehicles. Providing light weight design technique is necessary contribution for the effort to maximize performance of the vehicle. The main constraint for light weight design is stress rise when trying to minimize the mass of rear axle shaft. In rough word the strength of the shaft will be affected. Therefore, the major challenge is reducing mass of a rear axle shaft without compromising its strength. The axle shaft should with stand the forces due to the weight of shaft, moment transferred from the differential and the moment force during the skidding of the car. Therefore, the thesis should be more conservative on the strength of the shaft while light weight design technique is implemented.

1.3. Objective

1.3.1 General objective

The objective of this thesis is to design light weight rear axle shaft without significant amount of stress raise by using hybrid of composite/steel material providing low density core.

1.3.2 Specific objectives

The specific objectives of this thesis are to:

- Determine the loads on the rear axle shaft through mathematical analysis of the load transfer.
- Carry out the analysis of the axle shaft using finite element analysis software ANSYS.
- Obtain the potential weight savings that can be achieved by innovative lightweight design technique.
    - Hybrid design of axle shaft with low density core
- Compare the results of the light weight design axle shaft with the commercial rear axle shaft available.

1.4. Methodology

- **Literature Study -**Study the literature on vehicle axle shaft for light weight green car to determine the current level of technology and to identify existing axle systems.
- **Identify Compatible Applications** Investigate possible/probable applications of light weight rear axle shaft systems.
- **Investigate Compatible materials for the rear axle shaft** Compile a shortlist of the most promising material.
- **Simulate the Most Promising Concepts** Investigate the shortlisted material in detail.
- **Preferred material Selection** selecting the material with a better strength weight ratio.
- **Demonstration Model Detail Design** modeling the rear axle shaft with different hole dimensions using solid work software.
- **Design** calculating the torque on rear axle shaft and the force due to the vehicle weight and moment due to the lateral load during skidding carried by the axle shaft and other parameter.

  Analyses of rear axle shaft models using ANSYS software and compare the result and select the optimum solution for the problem.
- **Final Report –**writing final summery and recommendation by comparing the results, comparing the weight for the new design and commercially available rear axle shaft.

1.5. Scope and Limitations of the Study

This thesis work focuses on providing a way for reducing the weight of the rear axle shaft by modifying the axle shaft and optimize the weight. The light weight techniques which will be examined includes making the shafts hollow or using a Hybrid design which utilizes low-density core. The thesis will disregard dynamic loads and fatigue analysis in favor of static analysis of the maximum loading condition.

CHAPTER TWO

2. Literature Review

2.1. The weight reduction on vehicles

Over the decades, harmful vehicular emissions have shown a negative impact on the environment and human health. The increasing air pollution from the transportation sector has led many government agencies to lay strict regulations on the automobile manufacturers to curb the harmful emissions under permissible limits.

One such example on fig.8 is the European agency (EU), which has set mandatory emission reduction targets for automakers in Europe. According to EU rules, the fleet average by cars to be achieved by 2021 must be 95 grams of CO2/ kilometer, which works out to a fuel consumption rate of around 4.1 liter/100 km of petrol or 3.6 liter/100 km of diesel [8].

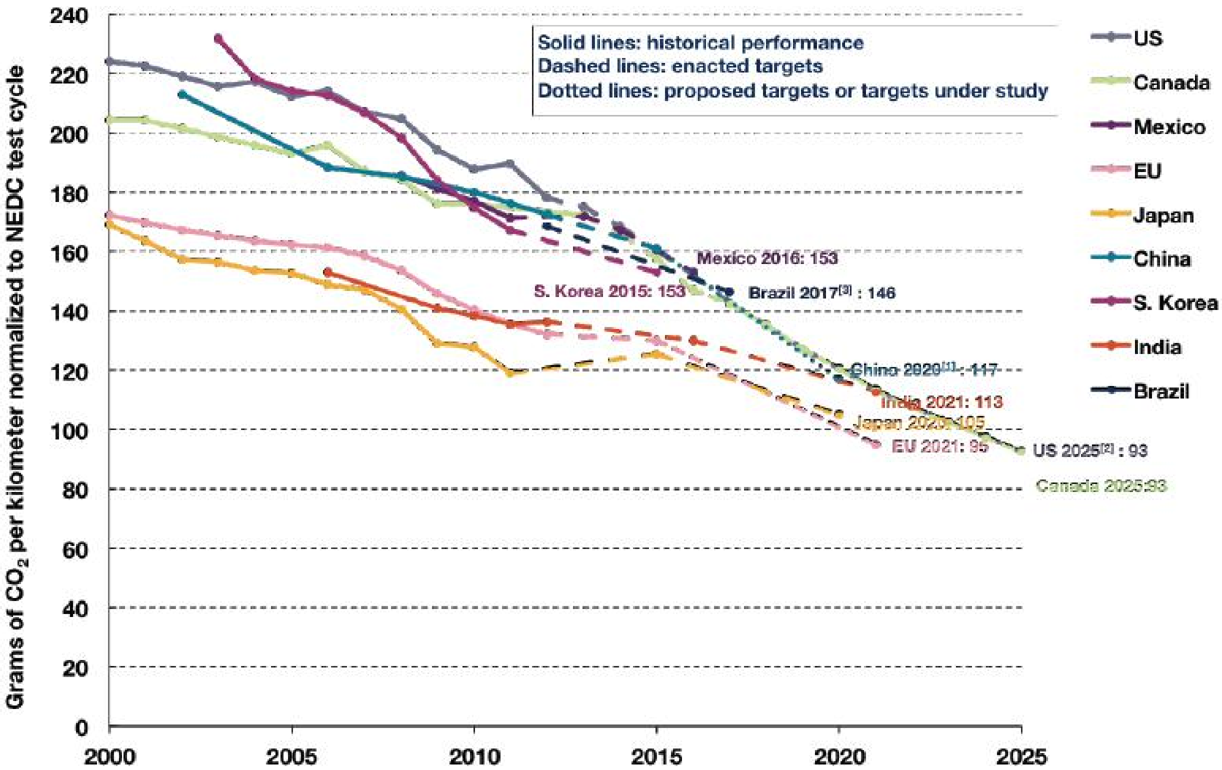


Figure 8: Country's target for fuel efficiency [8]

Weight reduction is a need for automotive industry. Vehicle weight and size reduction is one known strategy to improve fuel economy in vehicles and presents an opportunity to reduce fuel use from the transportation sector. By reducing the mass of the vehicle, the inertial forces that the engine has to overcome when accelerating are less, and the work or energy required to move the vehicle is thus lowered. Losing weight is the easiest way to increase fuel economy; according to thematic report, reducing a car's weight by just 10% can boost its mileage by almost 6 to 8 percent. [9].A general rule of thumb is that for every 10% reduction in vehicle weight, the fuel consumption of vehicles is reduced by 5-7% [10].

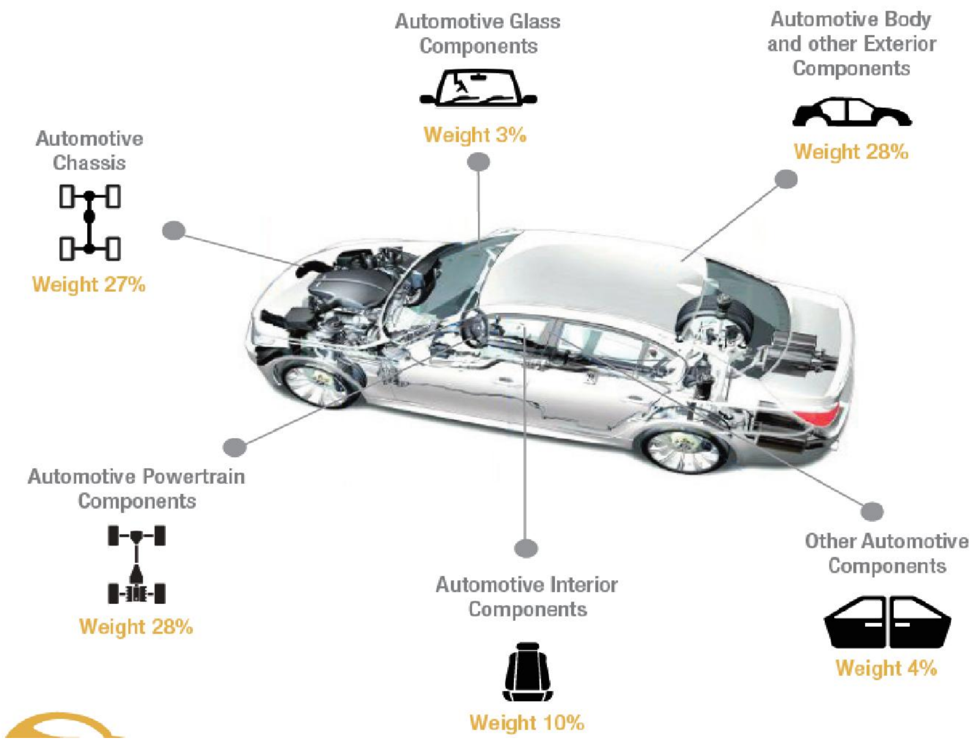


Figure 9: Automotive component weight statistics

From the above figure 8 we can understand that how much attention is given to improve fuel economy not only due to the cost, it is much greater danger faced this world due to the environment factor. This paper started for the sake of contributing for environmental protection by improving the fuel economy with reducing the weight of one power transmission component which is rear axle shaft. The power transmission and automotive body and other exterior components each hold around 28 percent of the vehicle weight which is collectively 56 percent of the vehicle mass[8]. Therefore, the power transmission system needs be given attention to achieve fuel economy.

2.2. The relationship between the mass and fuel consumption of the vehicle

It is well known that vehicle weight reduction has the potential to reduce fuel consumption. By reducing the mass of the vehicle, the inertial forces that the engine has to overcome as the vehicle is accelerated are reduced, so the work required to move the vehicle is thus lessened. To understand the physical impact of vehicle weight on fuel consumption, we examine the key parameters that contribute to a vehicle's fuel consumption from the following relation[11].

$$FC = \frac{\int b_e . P dt}{\int v dt} = \frac{\int b_e . \left(\frac{F_t . v}{\eta}\right) dt}{\int v dt}$$

Where, FC = Vehicle's fuel consumption [L/km], be = Engine's specific fuel consumption [L/kWh], P = Engine power output [kW], t = Time [s or hr.], v = Instantaneous vehicle speed [m/s or km/hr.], Ft = Tractive force [kN], η = Drivetrain efficiency

For a vehicle's given speed-time trace, or drive cycle, assuming that the engine's specific fuel consumption and efficiency as a function of load and speed are known, the key parameter that affects the amount of energy output needed from the engine is the amount of tractive or resistive forces the vehicle has to overcome. The tractive force is the sum of tire rolling resistance, acceleration or braking resistance, aerodynamic drag and climbing resistance. For an accelerating vehicle on a level road (with zero climbing resistance), as shown in Figure 10 , the total tractive force is: $F_t = F_{ROLL} + F_{ACC} + F_{DRAG}$.

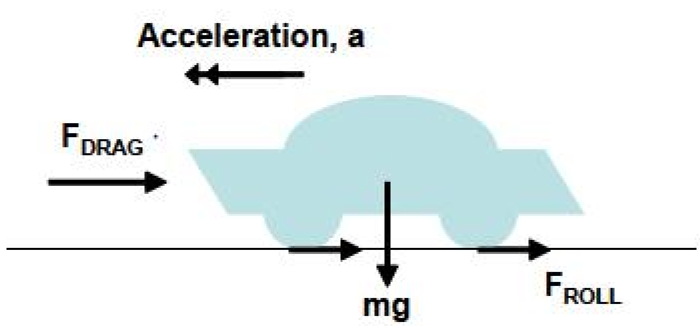


Figure 10:Forces on an accelerating vehicle on a level road [10]

The National Research Council reports that a 5-10% reduction in the coefficient of drag (CD) can reduce fuel consumption by 1-2%. A 10% weight reduction is expected to improve it by 6-7%[12]. vehicle size reduction improves fuel consumption mainly by reducing its mass. Figure 11 shows how the vehicle mass is related to tractive force;

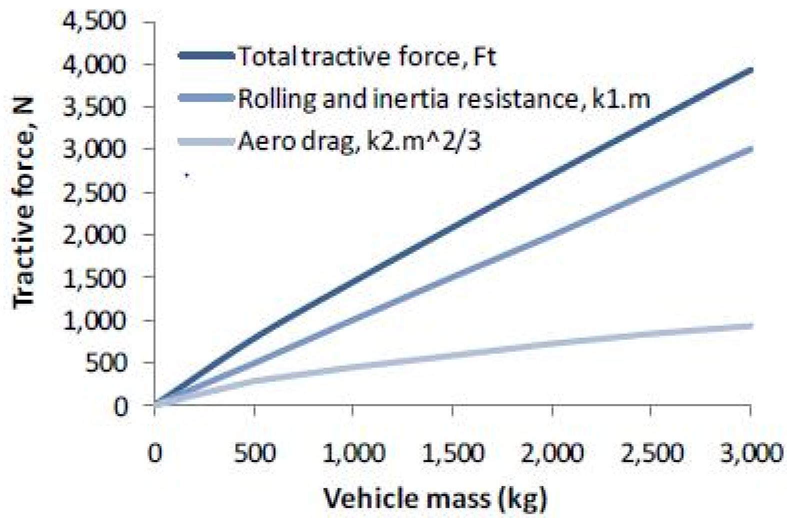


Figure 11: The relationship between vehicle mass and tractive force [13]

There are several studies that attempt to quantify the vehicle fuel consumption reduction benefit associated with light weighting. For vehicles using conventional gasoline-fueled internal combustion engines (ICE), a summary of the results from literature is shown in Table 1. The reported improvement in fuel consumption varies widely from 1.9-8.2% for every 10% reduction in vehicle weight. The average of these numbers, which has no inherent validity, gives 4.9%. Factors that affect this relationship include the size and type of vehicle, the drive cycle used to evaluate the vehicle (e.g. city, highway, or combined) and whether or not the powertrains were resized to maintain vehicle performance (i.e., secondary weight savings included or excluded)[10].

Table 1: Mass-fuel consumption relationship for gasoline vehicles from literature

Size/type	Fuel consumption reduction per 10% mass reduction (%)	Includes secondary weight savings?	Drive cycle	source
Small car	2.6	No	NEDC	[14]
Small car	3.5	No	43/57 City/hwy	[15]
Small car	4.7	No	55/45 City/hwy	[16]
Small car	5.3	Yes	43/57 City/hwy	[15]
Small car	6.8	Yes	NEDC	[14]
Midsize car	1.9	No	NEDC	[14]
Midsize car	2.5	No	43/57 City/hwy	[15]
Midsize car	4.1	No	55/45 City/hwy	[16]
Midsize car	5.6	Yes	43/57 City/hwy	[15]
Midsize car	7.0	Yes	55/45 City/hwy	[16]
Midsize car	6 to 8	(not specified)	(not specified)	[7]
Midsize car	8.0	Yes	55/45 City/hwy	[17]
Midsize car	8.2	Yes	NEDC	[14]
Small SUV	3.1	No	43/57 City/hwy	[15]
Small SUV	5.2	Yes	43/57 City/hwy	[15]
Small SUV	7.9	Yes	55/45 City/hwy	[17]
Midsize SUV	2.4	No	NEDC	[14]
Midsize SUV	7.4	Yes	NEDC	[14]
Large SUV	2.5	No	43/57 City/hwy	[15]
Large SUV	5.2	Yes	43/57 City/hwy	[15]
Large pickup	2.7	No	43/57 City/hwy	[15]
Large pickup	3.7	Yes	43/57 City/hwy	[15]

Note: NEDC is the New European Driving Cycle, which is supposed to represent the typical usage of a car in Europe. 43/57 and 55/45 city/hwy drive cycles are used by the U.S. EPA that combine city and highway drive cycles. These cycles are based on the standardized Federal Test Procedure (FTP) and Highway Fuel Economy Driving Schedule (HWFET). They are calculated using an updated weighting of 43%/57%, or the previous 55%/45% city/highway weighting.

2.3. Efforts to reduce the weight of axle shaft

A variety of weight reduction strategies are adopted by different automakers to minimize weight in automobiles. Weight reduction can be achieved by a combination of lightweight material substitution, redesigning the vehicle to minimize weigh and downsizing the new vehicle fleet by shifting sales away from larger and heavier vehicles. Material substitution involves replacing heavier iron and steel used in vehicles with weight-saving materials like aluminum, magnesium, high-strength steel, and plastics and polymer composites. Vehicles can be redesigned to optimize the size of the engine and other components as vehicle weight decreases, or to improve the vehicle's packaging and reduce exterior vehicle dimensions while maintaining the same passenger and cargo space. Finally, downsizing can provide further weight reduction by shifting sales away from larger and heavier to smaller and lighter vehicle categories[10].

To be specific to power transmission component which is axle shaft despite many research gap existed some literatures tried to find the best way to achieve weight reduction. The research done for full floating axle shaft for heavy duty trucks only torque moment considered for light weight design. significantly reducing the weight of the part by removing the parasitic material from the center of the shaft. For the purposes of these simulations, the shaft was hollow by removing a cylindrical section of material from the center of the axle shaft. This resulted in a weight savings, but came at the cost of increasing the stress in the shaft. This rise in stress will have to be mitigated by either changing materials to one which is stronger, or by increasing the diameter of the shaft. Fortunately, the stress in the shaft is proportional to $1/r^3$ while the weight in the shaft is only proportional to r^2, meaning that the stress in the shaft will decrease faster than the weight will rise as a result of the added material [18].so that by allowing small increase on shaft diameter to compensate the stress rise. The researchers tried to achieve some amount of mass reduction. Approach is used the same geometry as the hollow axle shaft simulations above, except the hollow region has been filled with a lighter material, in this case Aluminum, while the outer material remains steel.

2.4. Hybrid material design of shaft

Hybrid design of shaft is more effective way of reducing weight of shaft. This method usually used in the manufacturing of drive shaft by combining two materials together. such as, the conventional drive shafts consist of three universal joints, a center support bearing and bracket due to which weight of the shaft increases and affects fuel efficiency. The conventional drive shafts are made of stainless steel whereas the composite drive shaft is made of carbon or glass fiber epoxy resin material which is light in weight. The composite drive shaft has a higher specific rigidity, a higher specific strength, a higher resonance frequency and a higher vibration damping capability compared to the conventional metal drive shafts[19]. The composite drive shaft is produced using the fiber-reinforced composite material eliminates the need for the universal joint, so that the drive shaft can be even lighter and generates less noise and resist the working moment. The hybrid composite drive shafts are also manufactured with the help of aluminum composite material in which the composite material was co-cured inside aluminum tube[20]. There are different methods of manufacturing the automotive composite driveshaft. These methods of the manufacturing are autoclave molding, vacuum bag pressure molding, filament winding etc.[21].

Design and Analysis of Composite Drive Shaft using ANSYS and Genetic Algorithm". This study deals with the review of optimization of drive shaft using the Genetic Algorithm and ANSYS. Here the replacement of the conventional steel is done by the composite materials of glass fiber of carbon fiber and optimization is done for further selection of most effective material Genetic algorithm technique is used. Substitution of composite material over the conventional steel material for drive shaft has increasing the advantages of design due to its high specific stiffness and strength[22].Use of composite material for replacement of steel in conventional two wheeler axle", In this paper, the aim is to manufacture the composite wheel axle and compare the results with conventional steel axle under different mechanical testing with evaluating of different mechanical properties such as tensile strength, bending strength, impact strength, fatigue strength by using appropriate experimental technique. The replacement of composite materials has resulted inconsiderable amount of weight reduction about 64% when compared to conventional mild steel shaft[23].

2.5. Gap identified from literature review

From the literatures the researcher found out the importance of reducing the weight of vehicle on the environmental protection and took this motivation for further study. The larger mass from the vehicle body is the power transmission which is around 28 percent of the weight of vehicle. Knowing this fact helps to decide which area of the vehicle should be given an attention to. so that one of power transmission element is selected which is the rear axle shaft. Some idea of reducing weight of axle shaft is tried like substituting material, by removing cylindrical mass from the solid shafts then compensating the stress rise with increasing the diameter. Also, Its is understandable that, hybrid structure which is substituting a part of the shaft with less density material is tried on drive shaft and on full-floating axle shaft.

CHAPTER THREE

3. DESIGN OF REAR AXLE SHAFT

3.1Physical Modeling

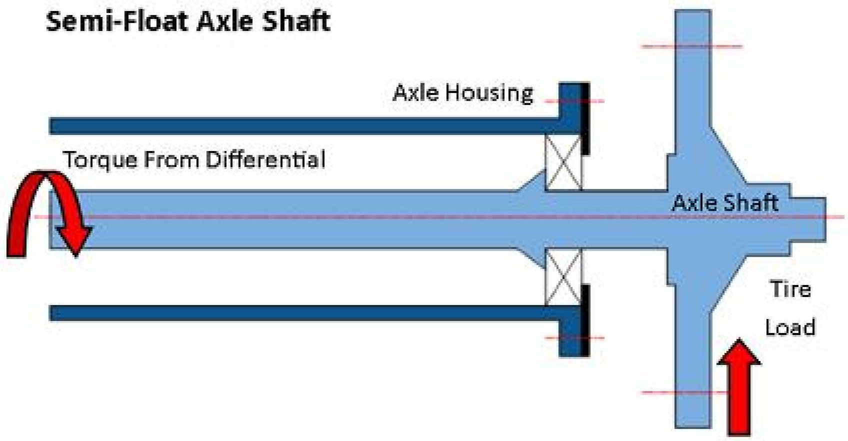


Figure 12: semi-floating axle shaft [18]

Assumptions

1. About longitudinal axis, the shaft rotates at a constant speed.

2. The shaft has a circular and the uniform cross section along the length.

3. The shaft is such that at every cross section, the mass center coincides with the geometric center due to which the shaft is perfectly balanced.

4. All the non-linear and the damping effects are excluded [24].

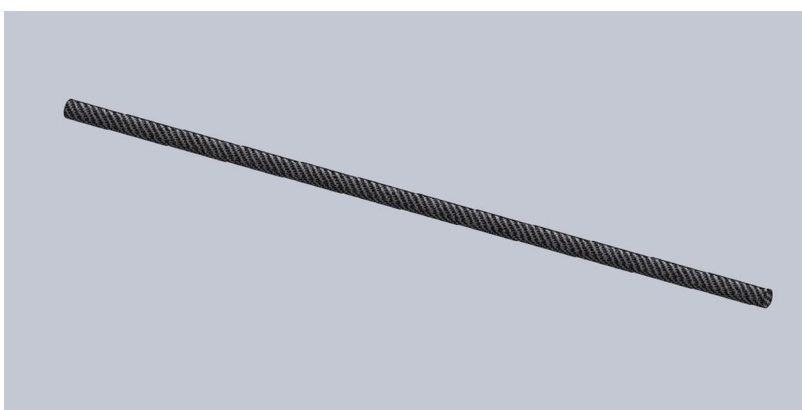


Figure 13: Inner shaft

Figure 14:Outer hollow shaft

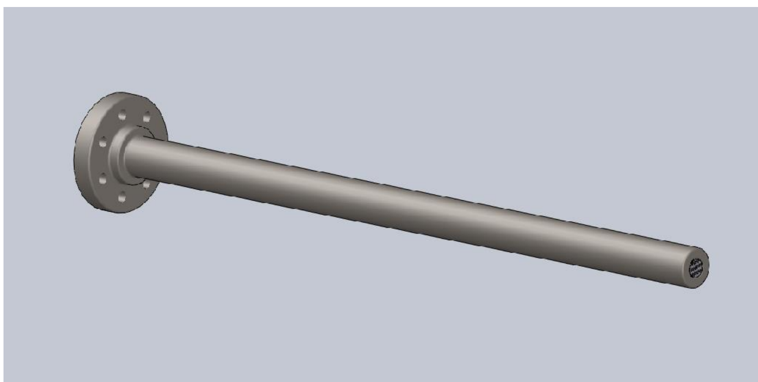


Figure 15:Assembled rear axle shaft

The above figures show the conceptual idea of this paper in order to reduce the weight of the rear axle shaft. This method shown from the figures the solid steel shaft is hollowed by removing cylindrical mass of the steel and replacing with composite material which is low density material. The figure 13 show the low-density composite and figure 14 show the hollow axle shaft also the figure 15 give the assembled axle shaft. The idea is reducing the mass of the rear axle shaft without significant raise of stress on the body.

3.3. Material selection

3.3.1. Selection of Reinforcement Fiber for the axle shaft

The different types of fibers have different properties. The fibers like carbon fiber, glass fiber and Kevlar fiber are commonly used for the composite shaft manufacturing. The selection of fiber for design of shaft depends on the physical properties and performance requirements.

Table 2:Mechanical Properties of Fibers [21].

Material	E (GPa)	G (GPa)	γ	Sut (MPa)	α (%)	ρ (Kg/m3)
Carbon fiber HM	385	20	0.23	3630	0.4	2170
E-Glass fiber	72	27.7	0.3	3450	4.7	2580
S-Glass fiber	87	33.5	0.3	4710	5.6	2460
Kevlar 49 fiber	124	5	0.3	3850	2.8	1440

Here, **carbon fiber** is selected due to its strength, modulus of rigidity and other mechanical properties.

3.3.2. Selection of the resin

The mixture or formulation of the polymer and polymer precursor material with various additives or chemically reactive components is known as the Resin. The processing, fabrication and the ultimate properties of composite material will affect the chemical composition and physical properties of the resin. The handling ability and processing ability of the composites may be affected by the variation in the composition, physical state or morphology of a resin and presence of impurities or contaminants in a resin. Also, it affects the long term durability of the composite material and the properties of lamina/laminate.

Table 3:Mechanical properties of resins[21]

Material	E (GPa)	G (GPa)	γ	Sut (MPa)	α (%)	ρ (Kg/m3)
Epoxy	3.1	1.2	0.3	70	4.0	1200
Polyester	3.5	1.4	0.3	70	5.0	1100
Resin RTM 6	2.8	1.0	0.34	75	3.4	1140
Resin RTM120	2.6	0.9	0.35	77	-	1200

The main factors considered in the selecting resin are mechanical properties and also resistance to impact (a function of modulus of elongation), elongation to failure, temperature capability and the important one is cost of the resin. The commonly used resins for the composite shaft are either epoxies or the vinyl esters. Here, **epoxy resin** is selected due to its strength, good wetting of fibers and lower curing shrinkage.

3.3.3. Axel shaft material

A tough, hard material must be used to withstand the various stresses, resist spline wear and provide good resistance to fatigue. Medium carbon alloy steel containing such elements as nickel, chromium and molybdenum is the usual choice. In this design the usual material for axle shaft, **AISI4140** selected. **AISI4140** is a chrome-molybdenum steel and features improved hardness, greater elasticity and resistance to corrosion [25].

Table 4: The following table outlines the mechanical and physical properties of AISI 4140 alloy steel[26]

Properties	**Metric**					
Tensile strength	1150 MPa					
Yield strength	835 MPa					
Bulk modulus (typical for steel)	140 GPa					
Shear modulus (typical for steel)	80 GPa					
Elastic modulus	210 GPa					
Poisson's ratio	0.30					
Hardness, Brinell	352					
Density	7.85 g/cm^3					
Coefficient of Thermal Expansion	1.2e-5					
Melting point	1416°C					
Chemical composition average values %						
C	Si	Mn	Cr	Mo	S	P
0.4	0.2	0.8	1	0.2	0.025	0.025

Factor of Safety

The important factor while designing a structure is factor of safety and it must be taken into account while designing. As the composites are orthotropic in nature and the fractures occur in composite material, the factor of safety for the composite shaft is taken as 6 [21].

Specifically, the minimum safety factors for the oil quenched part are 4.0 for the full float axle shaft and 3.78 for the semi-float shaft [25].

3.4. Specification for input Design data

The change in vehicle specification data can change the physical dimension of the rear axle shaft but does not affect the output of this thesis. Because, when the vehicle dimension, torque and weight increase the diameter become bigger. Therefore, if we apply the same hybrid design technique on the shaft, the composite and the steel size can change but still we can achieve significant amount of mass reduction. In this paper the specification data used shown below.

TOYOTA HILUX PICK UP DOUBLECAB SSR [33]

Model code = KB-LN108

Engine Model = 2L-TE

Displacement (cc) = 2446cc

Gross Vehicle Weight = 1730 kg

Maximum payload = 737 kg

The wheelbase length = 2855 mm

Overall length = 4690 mm

Overall height = 1760 mm

Overall width = 1690 mm

Max. RPM of the engine = 3800 rpm

Max. Torque = 240.3 Nm

Tire size radius 15" = 0.38 m

Gear ratio: - 1^{st} Gear = 3.592

2^{nd} Gear = 2.246

3^{rd} Gear = 1.415

4^{th} Gear = 1.000

5^{th} Gear = 0.821

Reverse = 3.657

Final Drive Gear Ratio = 4.625

Minimum turning radius = 6.1 m

Axle shaft type = Semi-floating

Diameter = 1.3785" (35mm), 23 splines

3.5. Preparing and elastic properties of a composite

Fiber and matrix volume fraction (V_f and V_m)

$$V_f = \frac{w_f}{\rho_f} \quad 3.1$$

$$V_m = \frac{w_m}{\rho_m} \quad 3.2$$

$$w_f = \frac{w_f}{w_m + w_f} \quad 3.3$$

$$w_m = \frac{w_m}{w_m + w_f} \quad 3.4$$

The stiffness matrix, strain and strength of composite lamina can be calculated using analytical methods using the following formula on [27]

Longitudinal modulus of composite

$$E_{11} = E_f V_f + E_m V_m \quad 3.5$$

Transverse modulus

$$E_{22} = \frac{E_f E_m}{E_f V_m + E_m V_f} \quad 3.6$$

Major poison's ratio $v_{12} = v_f V_f + v_m V_m$ 3.7

Minor poison's ratio $v_{12} = \frac{E_{22}}{E_{11}} * V_{12}$ 3.8

To calculateG_{12}, we need to know the value of G_f andG_m. Assuming isotropic relationships, we estimate.

$$G_f = \frac{E_f}{2(1+V_f)} \quad 3.9$$

$$G_m = \frac{E_m}{2(1+V_m)} \quad 3.10$$

In a plane shear modulus $G_{12} = G_{21} = \frac{G_f G_m}{G_f V_m + G_m V_f}$ 3.11

To calculate the elastic properties of an angle ply Lamina in which continuous fibers are aligned at an angle θ with the positive x direction.

$$\frac{1}{E_{xx}} = \frac{\cos^4\theta}{E_{11}} + \frac{\sin^4\theta}{E_{22}} + {}^1\!/_4 \left(\frac{1}{G_{12}} - \frac{2v_{12}}{E_{11}}\right) \sin^2 2\theta \quad 3.12$$

$$\frac{1}{E_{yy}} = \frac{\sin^4\theta}{E_{11}} + \frac{\cos^4\theta}{E_{22}} + {}^{1}\!/_{4}\,(\frac{1}{G_{12}} - \frac{2v_{12}}{E_{11}})\sin^2 2\theta \qquad 3.13$$

$$\frac{1}{G_{xy}} = \frac{1}{E_{11}} + \frac{2v_{12}}{E_{11}} + \frac{1}{E_{22}} - (\frac{1}{E_{11}} + \frac{2v_{12}}{E_{11}} + \frac{1}{E_{22}} - \frac{1}{G_{12}})\cos^2 2\theta \qquad 3.14$$

$$G_{xy} = G_{xz} \qquad 3.15$$

$$\frac{1}{G_{yz}} = \frac{E_{22}}{2(1+v_2)} \qquad 3.16$$

$$V_{xy} = E_{xx}\left[\frac{v_{12}}{E_{11}} - \frac{1}{4}(\frac{1}{E_{11}} + \frac{2v_{12}}{E_{11}} + \frac{1}{E_{22}} - \frac{1}{G_{12}})\sin^2 2\theta\right] \qquad 3.17$$

$$V_{yx} = \frac{E_{yy}}{E_{xx}} \times V_{xy} \qquad 3.18$$

$$V_{yz} = \frac{v_{12}(1-v_{21})}{(1-v_{12})} \qquad 3.19$$

$$v_{23} = V_{yz} \qquad 3.20$$

The value of θ i.e. stacking sequence angle will be taken so that the torsional buckling strength will be more than that of the maximum torque applied. Here, the stacking sequence angle is taken as 45° as it resists torsion. The stacking sequence is as follows$[\pm45/\pm45/\pm45/\pm45/45]$ $(_{2s})$[21]. The selection is more dependent on the torsion is because of the obvious reason which is the failure on axle shaft mostly come from the torsional moment induced on the end of the axle shaft.

Table 5: Composite laminate ratio [21]

	Carbon fiber	Epoxy resin
Mass (g)	0.4	0.6
Density (${}^{g}\!/_{cm^3}$)	2.17	1.2
Volume (cm^3)	$\frac{0.4}{2.17} = 0.184$	$\frac{0.6}{1.2} = 0.5$

Volume of 1g of composite is $(0.184 + 0.5) = 0.684cm^2$ now, we calculate.

Fiber volume fraction $(V_f) = \frac{0.184}{0.684} = 0.269\ or\ 26.9\%$ 3.21

Matrix volume fraction $(V_m) = \frac{0.5}{0.684} = 0.73\ or\ 73\%$ 3.22

$E_{11} = 385 \times 0.269 + 3.1 \times 0.73 = \underline{105.828\ GPa}$ 3.23

$E_{22} = \frac{385\times3.1}{385\times0.73+3.1\times0.269} = \underline{11.277\ GPa}$ 3.24

Major poison's ratio

$v_{12} = 0.23 * 0.269 + 0.3 * 0.73 = \underline{0.28}$ 3.25

Minor poison's ratio

$v_{12} = \frac{11.277}{105.828} * 0.28 = \underline{0.0298}$ 3.26

$G_f = \frac{3.1}{2(1+0.269)} = \underline{1.221\ GPa}$ 3.27

$G_m = \frac{385}{2(1+0.73)} = \underline{111.271\ GPa}$ 3.28

$$\frac{1}{E_{xx}} = \frac{\cos^4 45}{105.828} + \frac{\sin^4 45}{11.277} + {}^1\!/_4\left(\frac{1}{4.407} - \frac{2*0.28}{105.828}\right)\sin^2 2(45) = 0.0799$$

$E_{xx} = \underline{12.51\ GPa}$ 3.29

$$\frac{1}{E_{yy}} = \frac{\sin^4 45}{105.828} + \frac{\cos^4 45}{11.277} + {}^1\!/_4\left(\frac{1}{4.407} - \frac{2*0.28}{105.828}\right)\sin^2 2(45)$$

$E_{yy} = \underline{12.51\ GPa}$ 3.30

$$\frac{1}{G_{xy}} = \frac{1}{105.828} + \frac{2*(0.28)}{105.828} + \frac{1}{11.277} - \left(\frac{1}{105.828} + \frac{2*0.28}{105.828} + \frac{1}{11.277} - \frac{1}{4.407}\right)\cos^2 2(45)$$
$$= 0.2268$$

$G_{xy} = \underline{4.407\ GPa}$ 3.31

$G_{xy} = G_{xz}$ 3.32

$$\frac{1}{G_{yz}} = \frac{11.277}{2(1+0.28)} = 4.405$$

$G_{yz} = \underline{0.227\ GPa}$ 3.33

$$V_{xy} = 12.51\left[\frac{0.28}{105.828} - \frac{1}{4}\left(\frac{1}{105.828} + \frac{2*0.28}{105.828} + \frac{1}{11.277} - \frac{1}{4.407}\right)\sin^2 2(45)\right]$$

$$V_{xy} = \underline{0.466} \quad 3.34$$

$$V_{yx} = \frac{0.0799}{12.51} \times 0.466$$

$$V_{yx} = 0.002975 \quad 3.35$$

$$V_{yz} = \frac{0.28(1 - 0.0298)}{(1 - 0.28)}$$

$$V_{yz} = \underline{0.377} \quad 3.36$$

$$v_{23} = V_{yz} \quad 3.37$$

$$\rho_c = \rho_f V_f + \rho_m V_m$$

$$\rho_c = 2170 * 0.269 + 1200 * 0.73$$

$$\rho_c = \mathbf{1459.73}\ {kg}/{m^3}$$

Table 6:Forces acting on the rear axle shaft with different arrangement

Type of hub	Hub-bearing arrangement	Stresses which act on shaft	Application
Semi-floating	Single bearing between shaft and casing	1. Torsional 2. Bending 3. Tension and compression	cars
Three-quarter floating	Single bearing between hub and casing	1. Torsional 2. Slight bending	cars
Fully floating	Two bearings widely spaced between hub and casing	1. Torsional	Commercial vehicles

For the light and medium weight vehicles semi- floating rear axle is used.

Semi- floating axle hub

The road-wheel is attached to the axle hub, which is an extension of the axle half –shaft. A single bearing inside the tubular axle-casing supports the outer end of the shaft. The inner end of the shaft is splined and supported by the final- drive unit, which itself is mounted on bearings within the axle casing. The semi- floating axle along with its overhanging hub is subjected to the driving torque as well as to both vertical and horizontal load. The vertical load produces a shearing force, and the distance between the wheel and the suspension-spring seat on the axle causes a bending moment, the reaction of which is shared between the axle bearing and the final drive-unit bearings. The horizontal load due to tilting of the vehicle, cornering centrifugal force, or side wind gives rise to both side-thrust and a bending moment. This bending moment may add to the vertical bending moment or may oppose it, depending on the direction of application of the side-force. A semi-floating axle suitable for small and medium size cars.

The axle half shaft and flanged hub are forged from a single piece of nickel chrome steel. The hub end of the shaft is provided with a larger diameter than the rest of its length, which resists the vertical and horizontal loads. The outer face of the flanged hub is shouldered so that it centralizes accurately the brake drum. The flange is provided with evenly spaced holes around it for wheel studs.

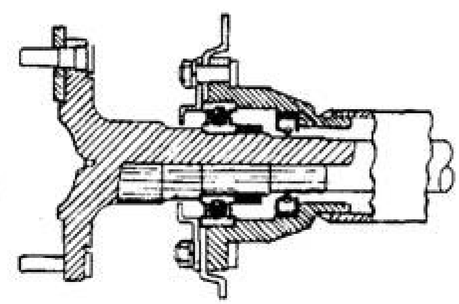


Figure 16: Semi floating ball-race-bearing axle.

A pre-greased and sealed deep grooved ball- race bearing is pressed over and along the shaft up to its shoulder. The bearing is sandwiched on one side by the axle-casing and on the other by the brake back-plate and the retaining plate by four nuts and bolts. To prevent excess oil leakage to the end of the axle-casing, a radial-lip oil-seal is pressed into a recess in the casing. Oil level of the final-drive is considerably lower than the axle shaft. However, the large amount of splash may cause the lubricant to spread along the shaft and enter the brake drum. An oil retainer is fitted at the hub end and the lip of the seal is positioned towards the final drive in the sealing arrangement.

A semi-floating axle showed in the fig 17 below uses a taper-roller bearing, which is suitable for larger and higher- performance cars because of its greater load-carrying capacity. A separate hub is wedged on to a keyed and tapered half-shaft and a castellated nut holds it is position. The taper-roller-bearing inner cone fits with a light force inside the mouth of the casing. The exact position of the bearing in the casing is provided by shims packed between the casing flange and the brake back-plate. Increasing the thickness of the shims on one side and decreasing it on the other shifts both half-shafts further to one side relative to the axle casing. On either road-wheel the outward thrust is absorbed by the adjacent hub bearing, while inward thrust is transmitted to the opposite bearing through the axle half- shafts and a slotted axle-shaft spacer. Therefore, each hub bearing takes thrust in one direction only.

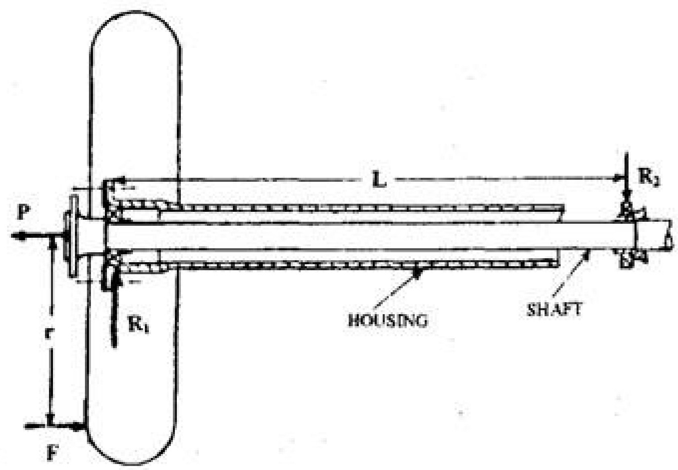
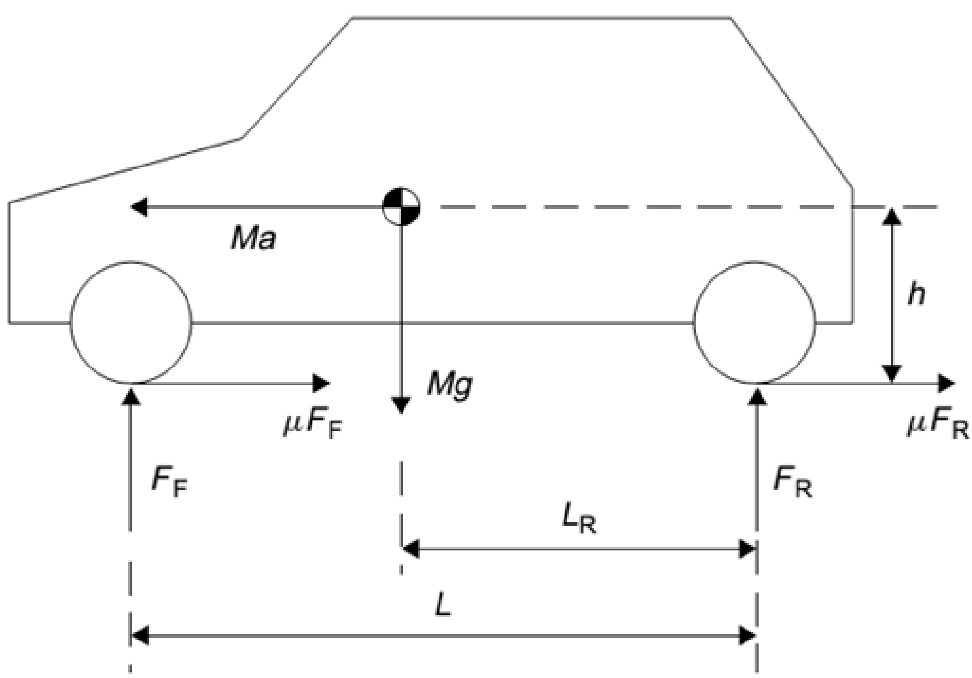


Figure 17: bearing loads due to side thrust on semi-floating axle.



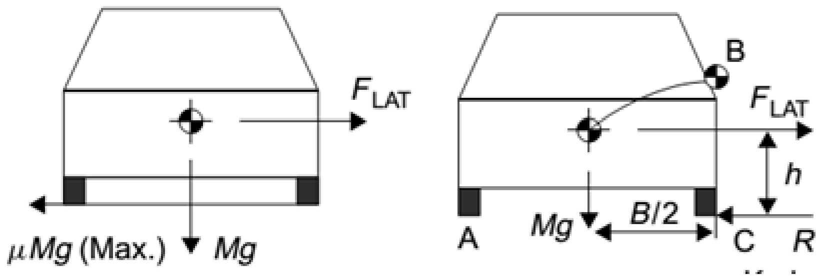


Figure 18:The forces acting on a moving wheel [28]

The following forces act on a moving wheel:

- The torque due to the traction or braking force (T_w and T_b)
- The traction or braking force (F_w and F_b)
- The lateral force $F_y = F_{LAT}$ when the vehicle makes a turn or skid
- The normal reaction $F_R = R_W$

Simultaneous appearance of maximum longitudinal and transverse forces at the wheel road contact is not possible, for joint action is restricted by the adhesion force [28].

$$R_w \varphi = \sqrt{F_w^2 + F_y^2} \qquad 3.38$$

3.6Loading condition

The loading conditions of axle shafts and beams reduce to the following three cases:

3.6.1 Rectilinear motion

The longitudinal force (F_w or F_b) attain its maximum value equal to $R_w\ \varphi$, Maximum torque is

$$T_w = \frac{1}{2} T_{emax} i_f k_d (1 + k_l) \qquad 3.39$$

$$T_w = \frac{1}{2}(240.3\ Nm)(4.625)(1.75)(1 + 0.8)$$

$$T_w = \mathbf{1750.43Nm} \qquad 3.40$$

Driving torque in each wheel $= \frac{T_w}{2} = \mathbf{875.215\ Nm}$

Where:

T_w = wheel torque

$T_{e\,max}$ = maximum engine torque

i_f = final drive ratio

k_d = dynamic factor (k_d = 1.75)

k_l = the coefficient of differential locking

m_a = automobile mass accounted for the driving axle

g = 9.81 m/s^2

h= (the height of the center of gravity from a road surface datum for Modern car is typically 0.51m)[4]

wt = transferred weight

$$wt = w_a\,\varphi\frac{h}{L} = m_a g(\varphi\frac{h}{L}) \quad [29] \qquad 3.41$$

$$wt = (1233.5)(9.81)((0.8)\frac{0.51m}{2.855m})$$

$$wt = \mathbf{1729.267N} \qquad 3.42$$

$$R_w = \frac{1}{2}m_a g \pm wt \qquad 3.43$$

$$R_w = \frac{1}{2}(1233.5 * 9.81) - 1729.27N$$

$$R_w = 4321.05N \qquad 3.44$$

$$R_w = \mathbf{7779.59N} \qquad 3.45$$

Where, φ = coefficient of adhesion (0.8) [4]

* In this case, lateral force, $F_y = 0$

3.6.2 Skidding of automobile

In this case a lateral force and normal reaction are acting on the wheel. Assume that the longitudinal force $F_w = 0$. The largest lateral force- centrifugal force- whose value is limited by the wheel-road grip equals.

$$F_y = F_{yi} + F_{yo} = \frac{m_a v^2}{R*(3.6)^2} = (R_{wi} + R_{wo})\varphi \quad [29] \qquad 3.46$$

$$F_y = (R_{wi} + R_{wo})\varphi \qquad 3.47$$

$$F_y = (2398.65N + 9701.99N)\varphi$$

$$F_y = \mathbf{12100.64N} \qquad 3.48$$

The vertical reactions and lateral forces of the inner and outer wheels are [30]

$$wt = w_a\varphi\frac{h}{t} = m_a g(\varphi\frac{h}{t}) \qquad 3.49$$

$$wt = 1233.5kg * 9.81(1 * 0.51m/1.69m)$$

$$wt = \mathbf{3651.67N} \qquad 3.50$$

$R_{wi,0} = \frac{1}{2} m_a g \pm wt$ 3.51

$$R_{wi,0} = \frac{1}{2}(1233.5kg * 9.81) - 2770.1N$$

$$R_{wi,0} = 2398.65N$$

$$R_{wi,0} = \mathbf{12100.64N}$$ 3.52

$$F_{yi,o} = R_{wi,o}\varphi$$

Where,

v = vehicle speed (180 km/h)

R = radius of turn of the road (R=6.1m)

t = wheel track width (1690mm)

φ = coefficient of road adhesion during sidewise skidding =1.0 [4]

$\pm$ = plus sign is used for the axle shaft of the wheel which is inner relative to the skidding direction, and the negative sign, the outer wheel.

3.6.3 Driving wheels overcome irregularities

Here, only the vertical force is accounted for

$F_w = (\frac{m_a g}{2}) k_{dr}$ 3.53

$$F_w = \left(\frac{1233.5 * 9.81}{2}\right) * 1.75$$

$F_w = \mathbf{10588.06N}$ 3.54

Where, k_{dr} = is the dynamic factor of road; [28]

for cars, k_{dr} = 1.75; for trucks, k_{dr}= 2.50

The bending moment on the rolling center can be found;

$M_i = F_l \times r_w - F_w \times l_b$ 3.55

$$M_i = 2398.65 \times 0.381 - 10588.06 \times 0.03$$

$$M_i = \mathbf{596.2N.m}$$

Where F_l= lateral force during skidding of the automobile

F_w= vertical force due to the vehicle weight

-The axle shaft dimensions are determined for the most dangerous case of loading. For s semi-floating axle, the dangerous cross section lies in the bearing installation zone. The equivalent stress due to bending and torsion, when the driving wheels overcome an irregularity, the bending stress is

$$\sigma_b = m_a g k_{dr} b \frac{m_a g k_{dr} b}{0.1 d^3} \qquad 3.56$$

$$\sigma_b = 1233.5 * 9.81 * 0.8 * 0.026 \frac{1233.5 * 9.81 * 0.026}{0.1(0.035)}$$

$$\sigma_b = \mathbf{22.62\ MPa} \qquad 3.57$$

Where

d = the axle shaft diameter (35mm)

b = the overhanging length (26mm)

Bending stress during rectilinear motion

$$\sigma_b = \frac{\sqrt{R_w^2 b^2 + F_w^2 b^2 + T_w^2}}{0.1 d^3}$$

$$\sigma_b = \sqrt{\frac{7779.59^2 * 0.026^2 + 2398.65^2 * 0.026^2 + 1750.43^2}{0.1 * 0.035^2}}$$

$$\sigma_b = \mathbf{25.38\ MPa}$$

The mass of the shaft can be calculated

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

$$m = \rho \frac{\pi}{4} (d^2) l \text{ for solid shaft}$$

$$m = \rho \frac{\pi}{4} (d_o^2 - d_i^2) l \text{ for hollow shaft}$$

3.7 Joining method of the shafts

The design and manufacture of joints have become a very important research area because they are often the weakest part in engineering structures. Generally, there are three kinds of joining methods: adhesive joining, welding and mechanical joining with bolts or rivets. The adhesively bonded joint uses an adhesive interlayer between the adherents. The adhesively bonded joint can distribute the required load over a larger area than the mechanical joint, requires no holes, and adds very little weight. However, the adhesively bonded joint requires careful surface preparation of the adherent, is affected by service environments and is difficult to disassemble for inspection and repair [33]. The welding is created by the coalescence of two metal substrates, which is achieved by a combination of temperature, pressure, and metallurgical condition. However, for the hybrid structures, the composite materials may be degraded due to the temperature rise of metal materials during welding process. the mechanical joint with bolts and rivets is created by fastening the substrates with bolts or rivets. Since it requires holes for bolts and rivets, stress concentration, fatigue and galvanic corrosion problems are vulnerable to occur at the holes. In this work, press fitting method for fastening the steel/composite hybrid shaft whose inner surface has many small teeth was employed. to improve reliability and to reduce manufacturing cost, rather than other joining methods such as adhesive joint, mechanical joint and welding. The teeth on the inner surface of the steel shaft produced and also grooves on the surface of the carbon fiber epoxy shaft is obtained, then in the press fitting process which, produce mechanical interlocking between the steel teeth and the engraved grooves on the surfaces of the steel tube and composite shaft. Since the teeth on the inner surface of the steel shaft can be easily formed by broaching or die pressing in mass production, it may reduce manufacturing time and cost through the elimination of several joining processes. By this method, much reliable press fit joint between the steel shaft and carbon fiber shaft with 1.5 times higher torque capacity than that of an adhesive joint was realized [31].

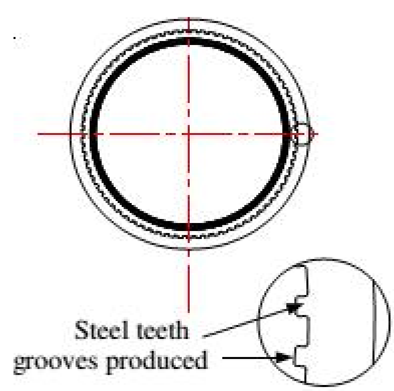


Figure 19 shows the teeth joining between the shafts

The press fit joining method between the hybrid steel/composite shaft and the steel shaft with protrusions was developed to increase reliability and to reduce manufacturing cost compared to other joining methods. To improve the torque capability of the press fitted joint, the protrusions in the axial direction were generated on the inner surface of the steel [31].

CHAPTER FOUR

4. RESULT AND DISCUSSION

4.1 Introduction

From load determined in chapter 3, it is possible to create the finite model of the rear axle shaft. The finite element model first used to examine the distribution of stresses in the conventional, solid axle shaft. These stresses are then used as a baseline to define the required performance of a light weight axle shaft design. Two methods for reducing the weight of the shaft are then investigated and evaluated in regards to the baseline study. finally, the possible weight savings that could be achieved by adopting the lightweight techniques investigated.

The analysis of the rear axle shaft is done using ANSYS software version 15. The first we should do is, determining the size of the rear axle shaft dimensions from commercial store for using it as a baseline. In this analysis we used the axle shaft with diameter 35mm, length 669mm shaft size for commercial vehicles.

Only the material's elastic behaviors were simulated, as plastic deformation of the part would be considered failure. The torque load was applied as a moment boundary condition on the area of the shaft into which splines would be cut, and the shaft was held in place via fixed displacement boundary conditions in to which the tires are mounted on the flange. The force due to the vehicle weight will be applied on the bearing installation area, also the moment due to the lateral force considered. The analysis will be done considering maximum loading condition.

4.1.1Static analysis

A static analysis can be either linear or non-linear. All types of non -linearity's are allowed such as large deformations, plasticity, creep, stress stiffening, contact elements etc. this chapter focuses on static analysis. A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those carried by time varying loads. A static analysis is used to determine the displacements, stresses, strains and forces in structures or components caused by loads that do not induce significant inertia and damping effects. A static analysis can however include steady inertia loads such as gravity, spinning and time varying loads. If these values exceed above the allowable values then component is going to fail. Hence static analysis is necessary.

4.1.2 Boundary condition

The boundary condition for the torsional and bending analysis of rear axle shaft are given as the one end is constrained with zero displacement in the both linear and rotational. At the other end of shaft torque is applied and vertical load due to vehicle weight also introduce on the bearing installation zone.

4.2 ANSYS Result for Rear Axle Solid Shaft

From the above tables it is possible to show that the mass, equivalent stress, maximum shear stress and total deformation for solid rear axle shaft. The mass of the solid shaft is 5.3064 kg, the equivalent stress 248.49 MPa, 143.4 MPa, total deformation 0.00080059 m obtained for semi-floating axle shaft structure which is the shaft needs to withstand the moment due to torque transfer from the differential, the vertical force due to the vehicle weight, and the moment due to the lateral force created during skidding of the automobile [Appx B-1].

4.3. ANSYS Result for Hybrid of Alloy Steel with Carbon Fiber Epoxy Composite Axle Shaft with Different Size

4.3.1. For 10mm hole size shaft

The 10mm diameter of mass is reduced from the solid shaft made of AISI4140 steel and the hole is replaced by 10mm diameter made up of carbon fiber epoxy material which has very small density compared steel shaft. From the geometry and solution tables shown below the mass reduced to 4.9663kg and the equivalent stress increased to 267.89 MPa, maximum shear stress 154.64 MPa total deformation 0.00081926 m. the mass is reduced by 6.4 %, 7.24 % increase in stress, 2.27 % increase in total deformation [Appx. B].

4.3.2. For 12mm hole size shaft

The 12mm diameter of mass is reduced from the solid shaft made of AISI4140 steel and the hole is replaced by 12mm diameter made up of carbon fiber epoxy material which has very small density compared steel shaft. From the geometry and solution tables shown below the mass reduced to 4.8192 kg and the equivalent stress increased to 267.44 MPa, maximum shear stress 154.34 MPa total deformation 0.0008238 m. the mass is reduced by 9.2 %, 7.08 % increase in stress, 2.82 % increase in total deformation [Appx. C].

4.3.3. For 14mm hole size shaft

The 14mm diameter of mass is reduced from the solid shaft made of AISI4140 steel and the hole is replaced by 14mm diameter made up of carbon fiber epoxy material which has very small density compared steel shaft. From the geometry and solution tables shown below the mass reduced to 4.6454 kg and the equivalent stress increased to 241.61 MPa, maximum shear stress 139.49 MPa total deformation 0.00078146 m. the mass is reduced by 12.45 %, -2.76 % increase in stress, -2.38 % increase in total deformation [Appx. D]. From equivalent stress and total deformation result the negative sign indicate that the hybrid shaft stress value is decreased when we compared to the standard solid axle shaft. This result intended to decrease the mass without affecting the strength, so this shows better design value in 14mm diameter core.

4.3.4. For 16mm hole size shaft

The 16 mm diameter of mass is reduced from the solid shaft made of AISI4140 steel and the hole is replaced by 16 mm diameter made up of carbon fiber epoxy material which has very small density compared steel shaft. From the geometry and solution tables shown below the mass reduced to 4.4449 kg and the equivalent stress increased to 257.23 MPa, maximum shear stress 148.46 MPa total deformation 0.00083031 m. the mass is reduced by 16.23 %, 3.4 % increase in stress, 3.58 % increase in total deformation [Appx. E].

4.3.5. For 18mm hole size shaft

The 18 mm diameter of mass is reduced from the solid shaft made of AISI4140 steel and the hole is replaced by 18 mm diameter made up of carbon fiber epoxy material which has very small density compared steel shaft. From the geometry and solution tables shown below the mass reduced to 4.2176 kg and the equivalent stress increased to 279.98 MPa, maximum shear stress 161.59 MPa total deformation 0.00083881 m. the mass is reduced by 20.5 %, 11.25 % increase in stress, 4.56 % increase in total deformation [Appx. F].

4.3.6. For 20mm hole size shaft

The 20 mm diameter of mass is reduced from the solid shaft made of AISI4140 steel and the hole is replaced by 20 mm diameter made up of carbon fiber epoxy material which has very small density compared steel shaft. From the geometry and solution tables shown below the mass reduced to 3.9636 kg and the equivalent stress increased to 310.73 MPa, maximum shear stress 179.38 MPa total deformation 0.00082318 m. the mass is reduced by 25.3 %, 20.03 % increase in stress, 2.74 % increase in total deformation [Appx. G].

Table 7: Total Result Summery for Solid and Hybrid axle shaft

	Solid(D)	Hybrid axle shaft with different hole size (mm)					
	35mm	10	12	14	16	18	20
Mass of the shafts (kg)	5.3064	4.9663	4.8192	4.6454	4.4449	4.2176	3.9636
Equivalent stress, σ_{eq}(MPa)	248.49	267.89	267.44	241.61	257.23	279.98	310.73
Max shear stress, τ_{max}(MPa)	143.4	154.64	154.34	139.49	148.46	161.59	179.38
Total deflections (m)	0.00080059	0.00081926	0.0008238	0.00078146	0.00083031	0.00083881	0.00082318
σ_{eq} stress % increase		7.24	7.08	-2.76	3.4	11.25	20.03
Total deformation % increase		2.27	2.82	-2.38	3.58	4.56	2.74
Mass % reduction		6.4	9.2	12.45	16.23	20.5	25.3

From the above total summery table 9, it shows the values for the baseline shaft which is the 35mm solid axle shaft. the dimension of the baseline shaft is taken from direct measurement then analysis was done using the ANSYS software, the result used as guideline for further design of light weight rear axle shafts.

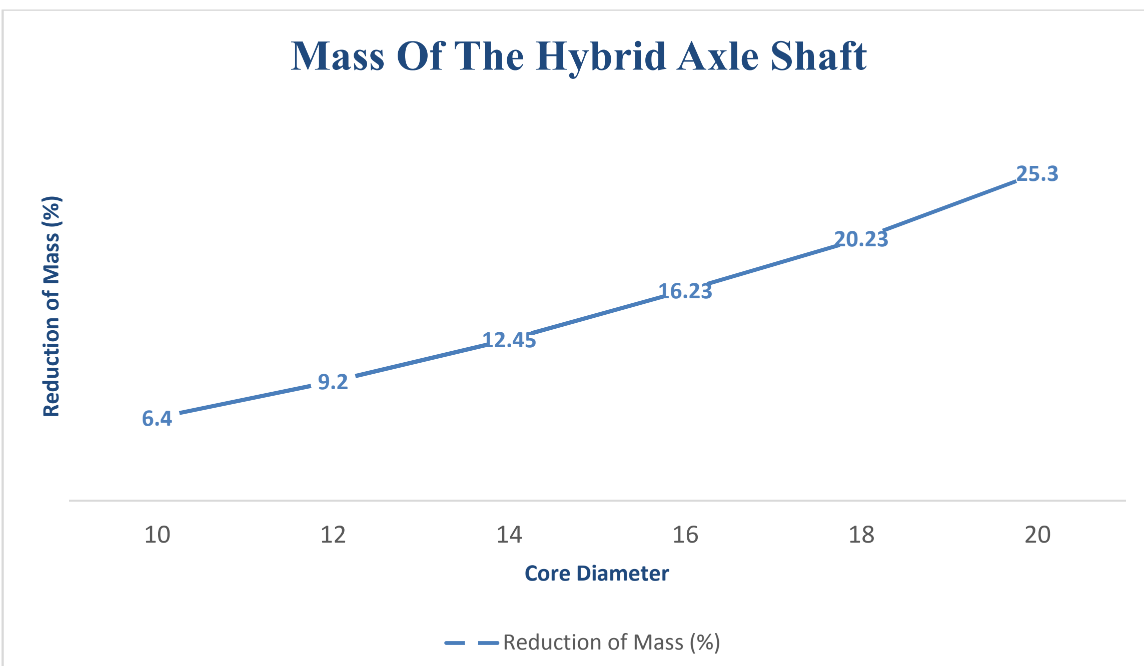


Figure 20 Mass of low density core axle shaft

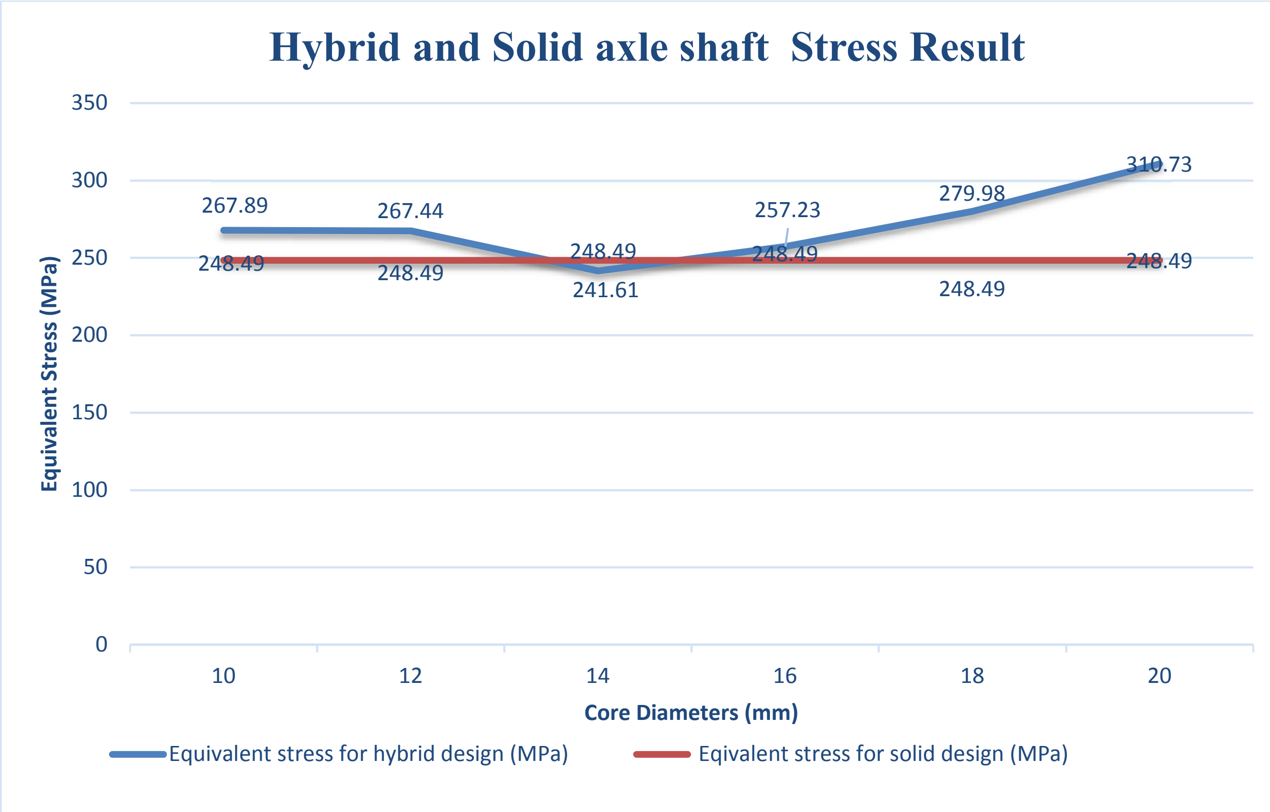


Figure 21 Result summery graph for hybrid design

The result shows to get significant amount of weight reduction; light weight design methods are used. Which is changing solid shaft shown in this design is using hybrid shaft with low density core. To explain more the shaft made from carbon fiber/epoxy material is fitted to the alloy steel hollow shaft. From the analysis results summarized with the above graph the horizontal axis show the core diameter of low density material (carbon fiber epoxy) and the vertical axis indicate the maximum stress level while the forces applied. This stress raise should be minimum as much as possible when we reduce the mass of the shaft. Therefore, to maintain the stress values within acceptable range and achieve significant amount of reduction of mass, hybrid shaft is used by replacing the mass reduced from the solid steel shaft with light weight composite material core. To obtain optimized hybrid axle shaft with low density core diameter analysis has been done with different size. The equivalent stress of hybrid design represented by blue line on the graph show on 14mm core diameter the value is dropped to 241.61 so The optimum solution will be selected by considering the significant mass reduction with minimum stress value. Which is the equivalent (Von-Mises) stress of the hybrid shaft is closer to the baseline stress value of solid shaft.

CHAPTER FIVE

5. CONCLUSION AND RECOMMENDATION

5.1 Conclusion

The result from summery table 7 which shows the percentage stress values and how much the mass of the shaft can be reduced with different low-density core diameters. Determining the potential weight savings that can be achieved by switching from a solid steel axle shaft to a Hybrid rear axle shaft obtained from the analysis. In low density core design other promising solution was found that can achieve more than result shown above of reduction in mass. But in semi-floating axle shaft, should withstand the weight of the vehicle, the torque from the differential, and the lateral force, so we should be conservative on stress raise. Therefore, 35mm diameter AISI4140 alloy steel rear axle shaft provided 14mm carbon fiber epoxy core rear axle shaft is the optimum choice. This can achieve 12.45 percentage mass reduction which is 1.32kg of unnecessary mass can be reduced from the vehicle axle shaft mass. Comparing the result of 35mm standard axle shaft made of steel with hybrid axle shaft designed with the same diameter, the mass of the shaft is reduced from 5.3063 to 4.6454 and the equivalent stress also reduced from 248.49 MPa to 241.61 MPa. Therefore, the hybrid design shown from the analysis, provided the axle shaft with lower mass without compromising the strength

5.2 Recommendation

- In this thesis I understand that there is research gap needs to be filled, on weight reduction of rear axle shaft. Most research works in this area started to get attention a few years back and are still progressing. So, in the future more mass reduction on axle shaft can be achieved. Specially on heavy duty trucks having more than four rear axle shafts, if we achieve significant amount of mass reduction on one axle shaft, we can reduce more weight from the heavy-duty vehicles. This will reduce pay load capacity or improve fuel efficiency.
- This paper is limited to static structural analysis of rear axle shaft, as future work we can work the dynamic analysis of the axle shaft for further understanding of the system.

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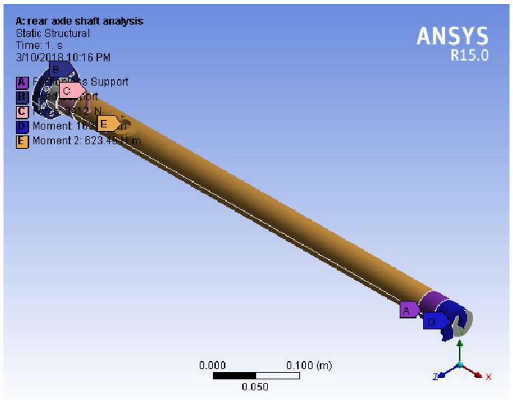
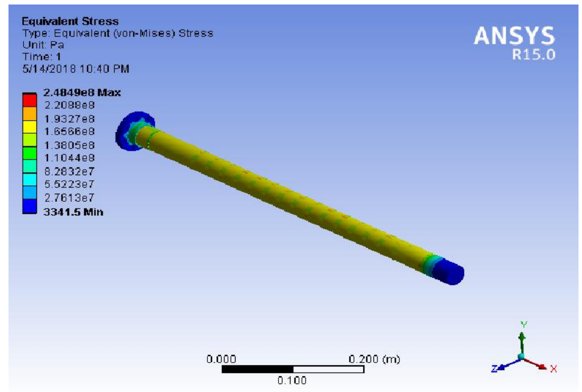
Appendix

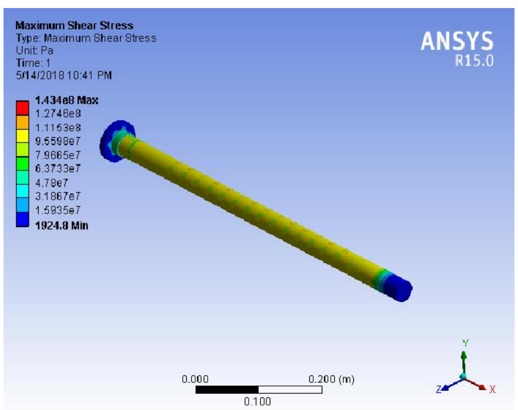
Appendix- A-Result for solid shaft



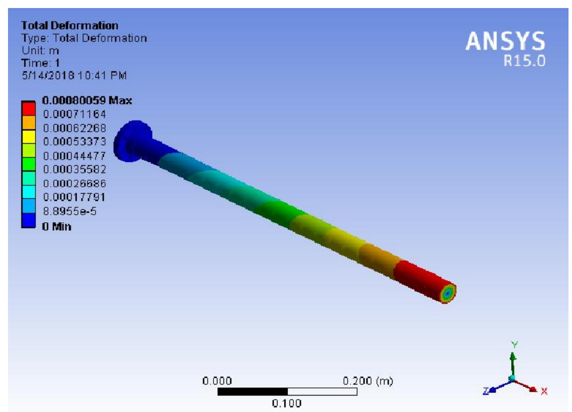
1-Load set up of the semi-floating axle shaft



2-Equivalent (Von-Mises) Stress for solid axle shaft

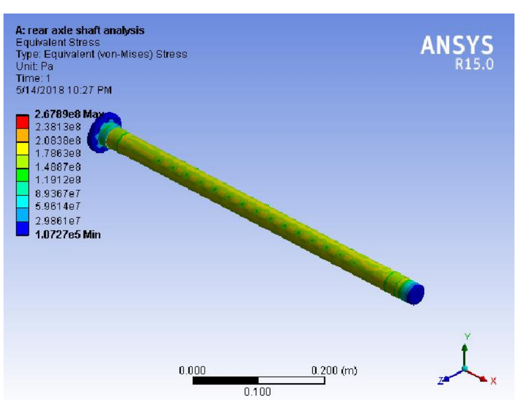


3-Maximum shear stress value for solid axle shaft

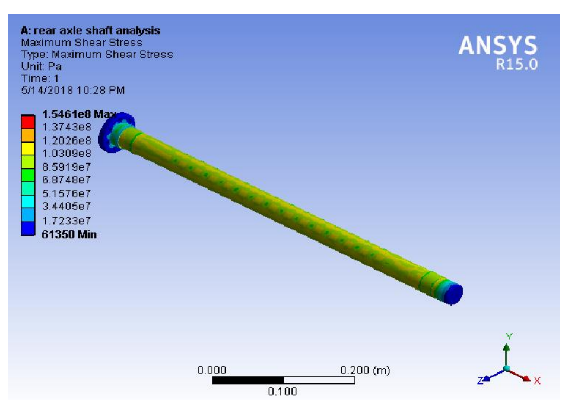


4-Total deformation for solid axle shaft

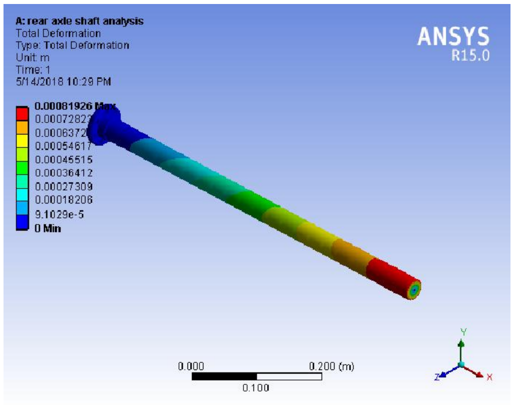
Appendix-B- Result for 10mm hybrid rear axle shaft



1-Stress for 10mm inside Dia. hybrid axle shaft

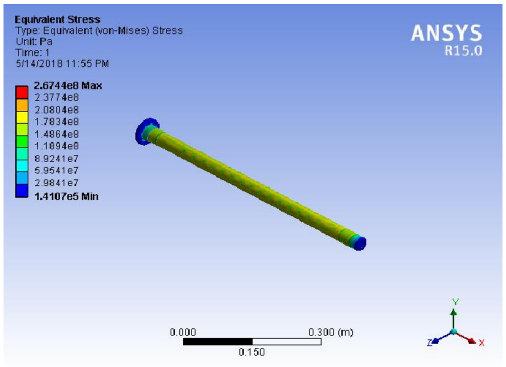


2-Maximum shear stress for 10mm inside Dia. hybrid

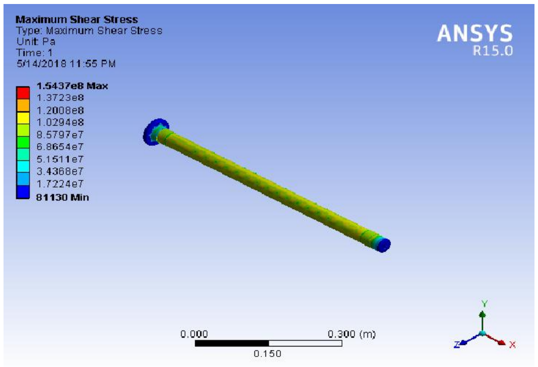


3. Total deformation for solid axle shaft

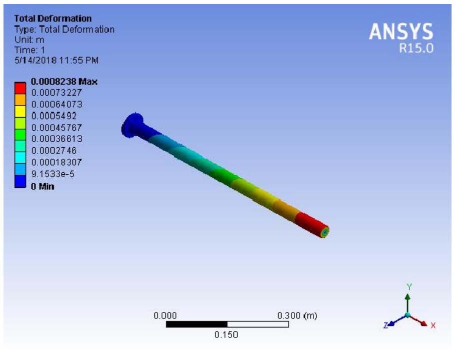
Appendix- C- Result for 12mm hybrid rear axle shaft



1-Equivalent stress for 12mm inside diameter

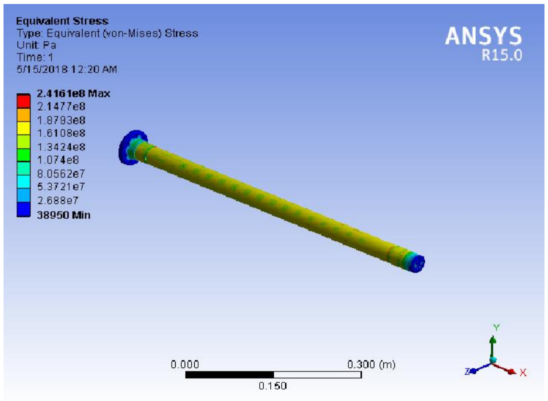


2- Maximum shear stress for 12mm inside diameter

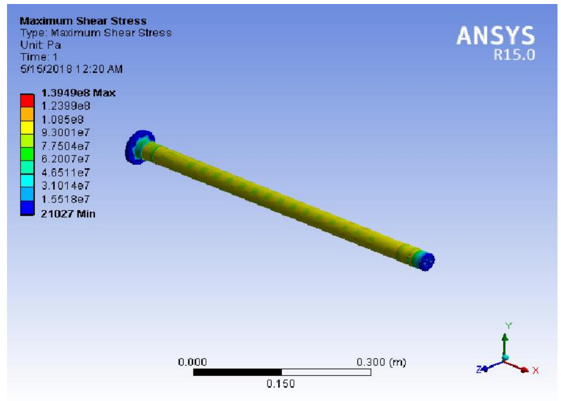


3. Total deformation for hybrid axle shaft

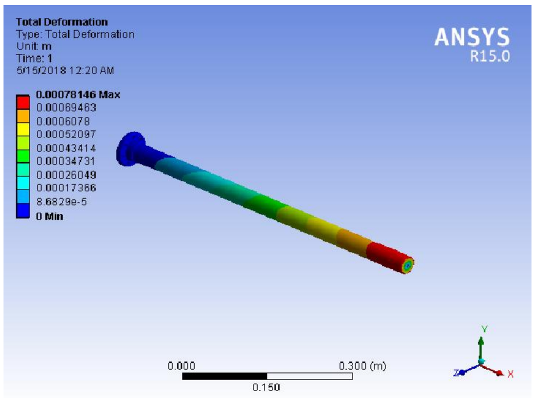
Appendix- D-_Result for 14mm hybrid rear axle shaft



1-Equivalent stress for 14mm inside diameter

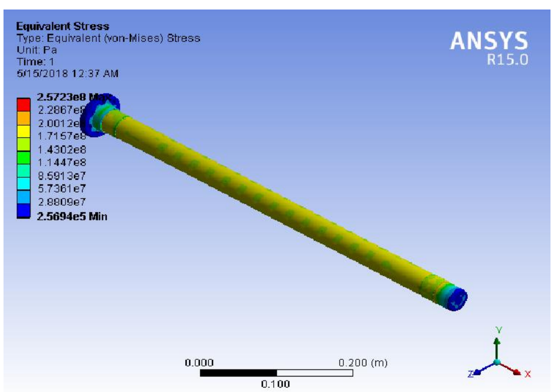


2- Maximum shear stress for 14mm inside diameter

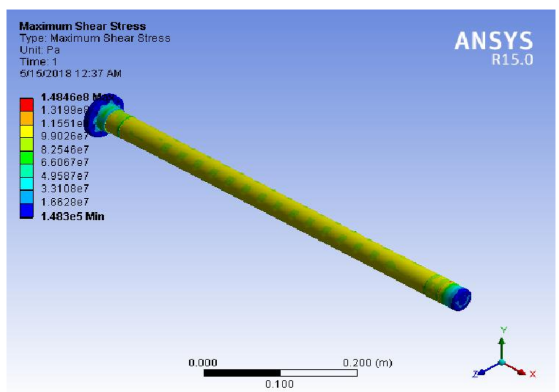


3-Total deformation for 14mm inside diameter hybrid axle shaft

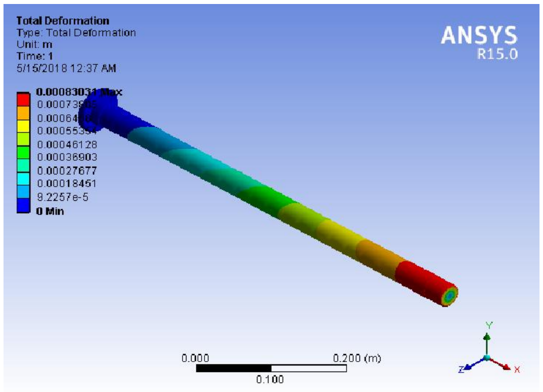
Appendix-E-_Result for 16mm hybrid rear axle shaft



1-Equivalent stress for 16mm inside diameter

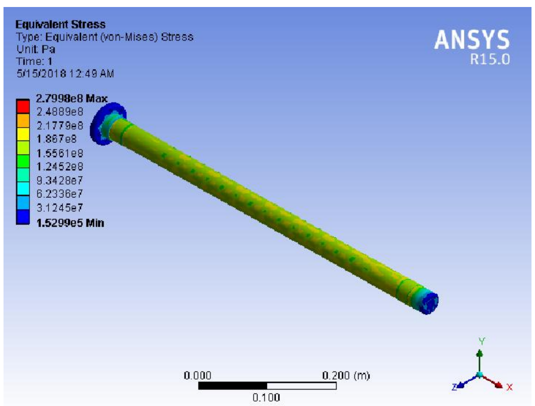


2- Maximum shear stress for 16mm inside diameter

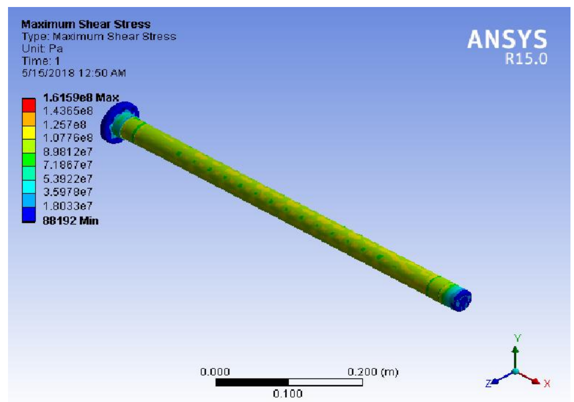


3- Total deformation for hybrid axle shaft

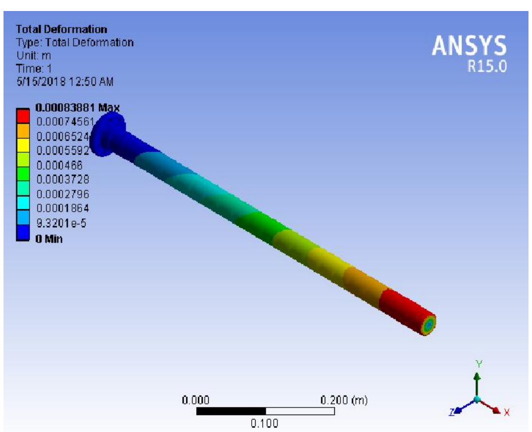
Appendix- F- Result for 18mm hybrid rear axle shaft



1-Equivalent stress for 18mm inside diameter

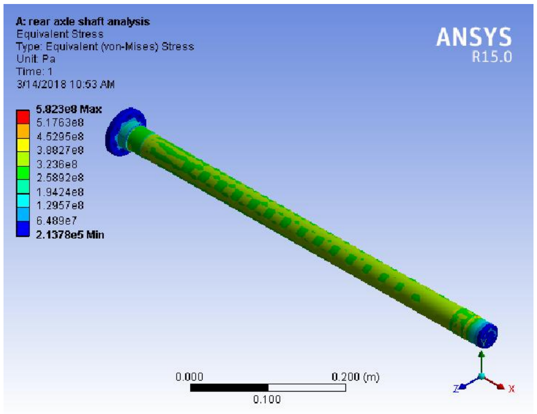


2- Maximum shear stress for 18mm inside diameter

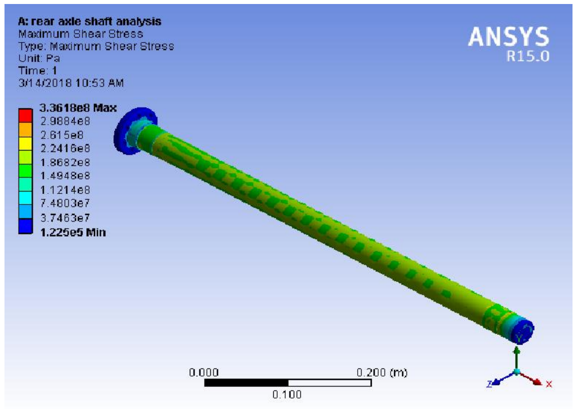


3-Total deformation for hybrid axle shaft

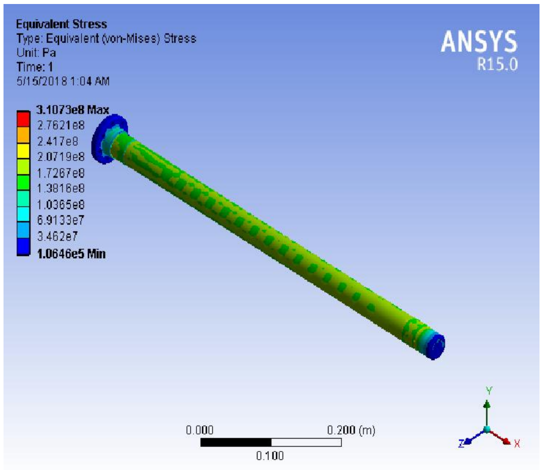
Appendix- G- Result for 20mm hybrid rear axle shaft



1-Equivalent stress for 18mm inside diameter



2- Maximum shear stress for 18mm inside diameter



3-Total deformation for hybrid shaft