Material optimization of composite drive shaft assembly in comparison with conventional steel drive shaft

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Abstract

Almost all automobiles (at least those which correspond to design with rear wheel drive and front engine installation) have transmission shafts. The weight reduction of the drive shaft can have a certain role in the general weight reduction of the vehicle and is a highly desirable goal. Substituting composite structures for conventional metallic structures has many advantages because of higher specific stiffness and strength of composite materials.

The advanced composite materials such as graphite, carbon, Kevlar and Glass with suitable resins are widely used because of their high specific strength and high specific modulus. Advanced composite materials seem ideally suited for long power driver shaft 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.

The present work includes, analysis done on drive shaft of Toyota qualis with different composite materials and concludes that the use of composite materials for drive shaft would induce less amount of stress which additionally reduces the weight of the vehicle.

CATIA V5R16 is the modeling package used to model the drive shaft assembly and ANSYS 11.0 is the analysis package used to carry out analysis.

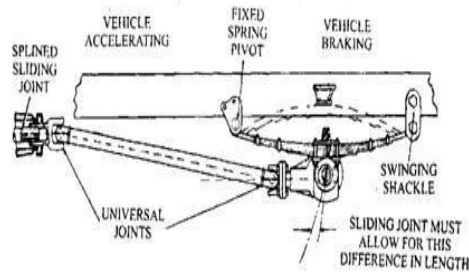## Purpose of the Drive Shaft (Or Propeller Shaft)

It must transmit torque from the transmission to the differential gear box.

The drive shaft must also be capable of rotating at the very fast speed required by the vehicle.

The drives shaft must also operate through constantly changing the angles between the transmission, the differential and the axels.

The length of the drive shaft must also be capable of changing while transmitting torque.

Propeller Shaft Arrangement in automobiles

Specifications

Dimensions	
Length	4425 mm
Width	1655 mm
Height	1880 mm
Kerb Weight	1505 Kg
Ground Clearance	178 mm
Wheel Base	2500 mm
Power	
Engine Type	4-cylinder In-line, 8-valve, OHC Belt Drive
Piston Displacement	2446 cc
Max. Power	75 PS @ 4200 rpm
Max. Torque	154 Kgm @ 2400 rpm
Transmission & Gear Box Type	5M/T
Suspension	
Front Double wishbone with Torsion Bar Rear Leaf Spring, Rigid	
Brakes	
Type - Superior Anti-fade Braking System with Load Sensing & proportioning valve which adjusts the braking performance as per axle loading.	
Front - Ventilated Disc	
Rear - Drum	
Supplementary Rear brake LSPV & BV	
Tyres	175R 14C
Fuel Tank Capacity	53 Liters

Problem description

We opted to do our work on material optimization on propeller shaft of Toyota Quails which transmits a maximum torque of 154N-m at 2400 rpm. We also extended the project by including more composite materials. The composites selected for this analysis are – carbon Epoxy, Glass Epoxy and E Glass Polyester Resin along with structural steel.

Demerits of conventional drive shaft

They have less specific modulus and strength

Increased weight

Conventional steel drive shafts are usually manufactured in two pieces to increase the fundamental bending natural frequency because the bending natural frequency of a shaft is inversely proportional to the square of beam length and proportional to the square root of specific modulus. Therefore the steel drive shaft is made in two sections connected by a support structure, bearings and U-joints and hence overall weight of assembly will be more.

Its corrosion resistance is less as compared with composite materials.

Steel drive shafts have less damping capacity.

Merits of Composite Drive Shaft

They have high specific modulus and strength

Reduced weight

The fundamental natural frequency of the carbon fiber composite drive shaft can be twice as high as that of the steel or

aluminum because the carbon fiber composite material has more than 4times the specific stiffness of , which makes it possible to manufacture the drive shaft of passenger cars in one piece. A one-piece composite shaft can be manufactures so as to satisfy the vibration requirements. This eliminates all the assembly, connecting the two piece steel shafts and thus minimizes the overall weight, vibrations and cost.

Due to weight reduction, fuel consumption will be reduced.

They have high damping capacity and hence they produce less vibration and noise.

They have good corrosion resistance

Greater torque capacity than steel and aluminum shaft

Longer fatigue life than steel and aluminum shaft

Manufacturing Of Composite Drive Shaft

Filament winding process is used in the fabrication of composite drive shafts. It is automated processes for creating parts of simple geometry wherein continuous resin impregnated fibers are wound over a rotating male tool called mandrel. Fig 1 depicts a filament winding process where a continuous fiber roving passes through a shuttle, which rotates and the roving is wrapped around a revolving or stationary mandrel. After completion of the winding, the filament wound structure is cured at room temperature or in an oven. The mandrel is removed after the curing. The mandrel, which determines accurate internal geometry for the component, is generally the only major tool. Low cost mandrel materials such as cardboard or wood can be used for winding low cost routine parts. For critical parts requiring close tolerances, expensive mandrels designed for

long term use may be required. For high temperature cure 315°C (600°F), graphite mandrels with low thermal expansion may be advantageous. However, attention should be paid for potential difficulties for mandrel removal.

In this technique the winding angle, fiber tension, and resin content can be varied. Filament winding is relatively inexpensive, repetitive and accurate in fiber placement
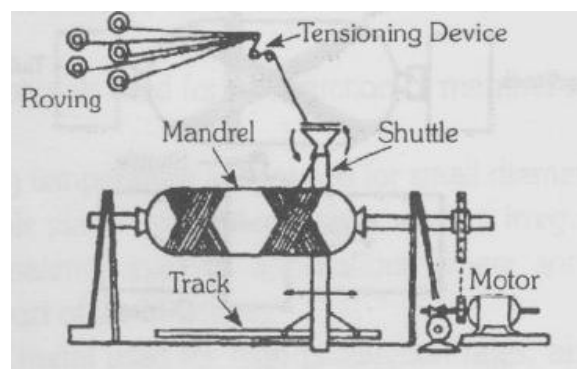

Figure 1 filament winding process

Modeling of Universal Joint

The modeling steps of universal joint in CTAIA V5 are briefly discussed below.

First an arbitrary plane is selected and sketcher workbench in sketcher toolbar is invoked.

Then the sketch shown in the figure 2.1 is drawn with appropriate dimensions and corners are filleted with radius of 5mm by invoking corner command in operation toolbar.

Then the sketch is padded for 8mm by invoking pad command in sketch based features after exiting from the sketcher workbench.

Next the view of the sketch is changed to side view and the sketch shown if the figure 2.2 is drawn by projecting the required edges (by invoking project 3D command in operations toolbar). And the sketch is padded by 22.5mm on both sides by invoking pad command in sketch based features toolbar.
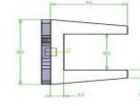
Then the top and bottom surfaces are drafted by 5degrees each side by invoking draft command in dress-up features toolbar. This is shown in figure 2.3
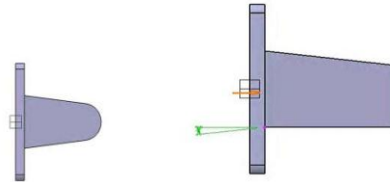
Then the edge is chamfered by invoking tritangent fillet in dress-up toolbar. And the whole operations are mirrored to the other side by invoking mirror command in transformation features toolbar.
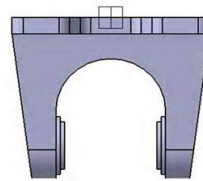
Again tritangent fillet in dress-up toolbar is invoked to obtain the contour between the yokes as shown in the figure 2.4

The circular projections of diameters 28mm and 23mm are obtained by invoking particular surface as working plane and doing pad operation of about 2mm each as shown in the figure 2.5
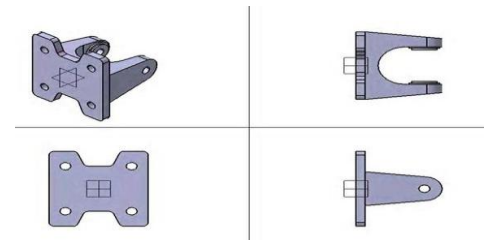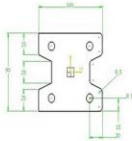
A circle of 10mm diameter is pocketed on the yoke by drawing a circle of 10mm diameter on particular face and by invoking pocket command in sketch based features toolbar.

Thus the universal joint yokes are designed and the complete figure is as shown in the figure 2.6.

Fig 2.1 front view sketch of universal joint fig 2.2 side view

Fig2.4 Filleting edges surfaces   **Fig2.3 drafting top & bottom**

Fig2.5 creating circular projections

Fig2.6 various views of universal joint

Modeling Of Spider (Center Block)

The modeling steps of centre block in CTAIA V5 are briefly discussed below.

First an arbitrary plane is selected and sketcher workbench in sketcher toolbar is invoked.

Then a circle of 10mm diameter is drawn using drawing tools and sketcher is exited.

Now the circle is padded about 36mm on both sides using pad command in sketch based features toolbar.

Again the same plane is selected and entered into the sketcher to draw a circle of 23mm diameter. This circle is also padded about 19mm in both sides by invoking pad command in the sketch based features toolbar.

Above three operations are repeated in any other plane which is perpendicular to the previously selected plane.

Now the plane which is selected initially is selected and sketcher is invoked.

A sketch as shown in the figure 2.7 is drawn. And this sketch is pocketed towards the centre block up to 12mm by invoking the pocket command in sketch based features toolbar.

The edges are filleted about 5mm by invoking edge fillet command in dress-up features toolbar.

The above two steps are mirrored by invoking mirror command in transformation features toolbar.

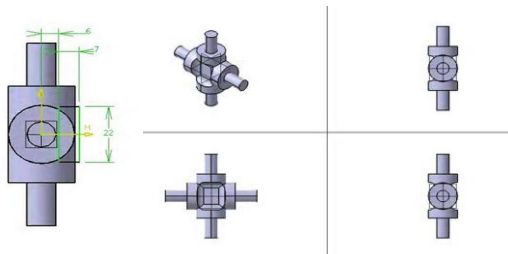Thus the centre block is modeled and is shown in the figure 2.8

Fig 2.7Side view of spider fig 2.8 Various views of spider

Modeling of Propeller Shaft

The modeling steps of propeller shaft in CTAIA V5 are briefly discussed below.

First an arbitrary plane is selected and sketcher workbench in sketcher toolbar is invoked.

A circle of diameter 75mm is drawn and is padded by invoking pad command in sketch based features toolbar.

The side view of the shaft is made as the working plane and the sketch shown in the figure 2.9 is drawn by projecting required edges using project 3D command in operation toolbar.

This sketch is padded about 22.5mm on both sides using pad command in sketch based features toolbar.

Now tritangent fillet is applied on both the yokes as shown in the figure 2.10

The same tritangent fillet is made use to obtain the contour between the yokes. Ant the circular projections are obtained by invoking particular plane as working plane, drawing circles of diameters 28mm and 23mm and then finally padding about 2mm each by invoking pad command in sketch based features toolbar. This is shown in the figure 2.11

A circle of 10mm diameter is pocketed on the yoke by drawing a circle of 10mm diameter on particular face and by invoking pocket command in sketch based features toolbar.

The whole operations are carried out on the other side of the shaft.

Finally a side of the shaft is selected as the working plane and the sketch shown in the figure 2.12 is drawn by making use of project 3d command in operation toolbar.

This sketch is grooved for 180 degrees about the bottom edge by invoking the groove

command in sketch based features toolbar. This forms an inner groove in the propeller shaft.

This completes the modeling of the propeller shaft and is shown in the figure 2.13

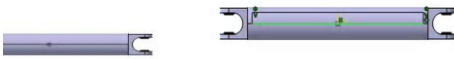Fig 2.9 Sketch for side view for shaft fig 2.10 filleting of edges

Fig 2.11 Creating circular projections Fig 2.12Front view of drive shaft

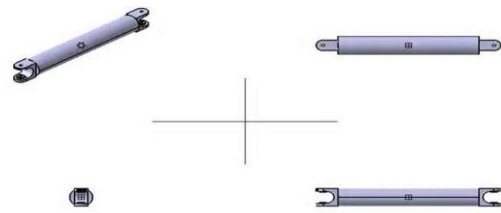Fig 2.13 various views of drive shaft

Modeling of Slip Yoke

The modeling steps of slip yoke in CTAIA V5 are briefly discussed below.

First an arbitrary plane is selected and sketcher workbench in sketcher toolbar is invoked.
The sketch shown in the figure 2.14 is drawn as per the dimensions and this sketch is padded about 22.5mm in both the sides by invoking pad command in the sketch based features toolbar.
The front and back faces of the yokes are drafted about 7 degrees towards the inner side by invoking the draft command in the dress-up features toolbar. This is shown in the figure 2.15.

Then the edges are filleted using tritangent fillet in dress-up features toolbar and this operation is mirrored to the other yoke by invoking mirror command in transformation toolbar. This is shown in figure 2.16

Now the sharp corners of the yoke edges are filleted about 15mm by invoking the edge fillet command in dress-up features toolbar.
The circular projections of diameters 28mm and 23mm are obtained by invoking particular surface as working plane and doing pad operation of about 2mm each as shown in the figure 2.17
A circle of 10mm diameter is pocketed on the yoke by drawing a circle of 10mm diameter on particular face and by invoking pocket command in sketch based features toolbar.
The back of the slip yoke is selected as the working plane and circles of diameters 24mm and 34 mm are drawn to pad about 105mm by invoking pad command in the sketch based features toolbar.
This completes the modeling of slip yoke and is shown in the figure 2.18
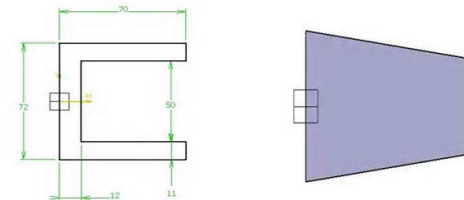
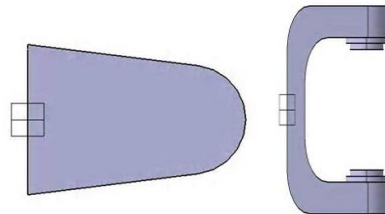Fig 2.14 Sketch for side view of slip yoke   fig 2.15 drafting surfaces

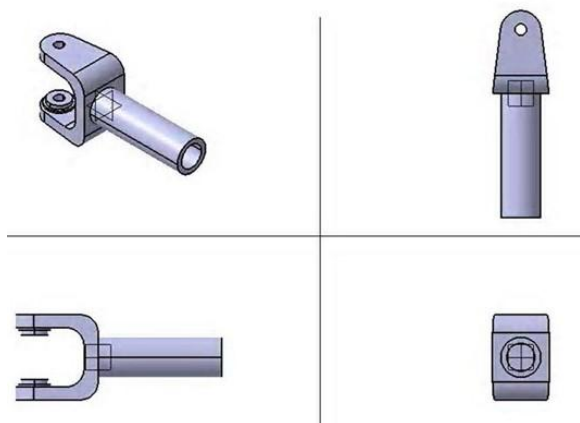Fig 2.16 Filleting edges   fig 2.17 creating circular projections

Fig 2.18 Various views of slip yoke

## Materials used in analysis and their properties

The materials and their properties that were used in this analysis are listed below.

Structural Steel	
Young's Modulus	2.07e+011 Pa
Poisson's Ratio	0.3
Density	7600. kg/m³
Allowable stress	370e+006 pa
E Glass	
Young's Modulus X direction	5.e+010 Pa
Young's Modulus Y direction	1.2e+010 Pa
Young's Modulus Z direction	1.2e+010 Pa
Major Poisson's Ratio XY	0.3
Major Poisson's Ratio YZ	0.3
Major Poisson's Ratio XZ	0.3
Shear Modulus XY	5.6e+009 Pa
Shear Modulus YZ	5.6e+009 Pa

Shear Modulus XZ	5.6e+009 Pa
Density	2000. kg/m³
Allowable stress	400e+006 Pa
E Carbon	
Young's Modulus X direction	1.9e+011 Pa
Young's Modulus Y direction	7.7e+009 Pa
Young's Modulus Z direction	7.7e+009 Pa
Major Poisson's Ratio XY	0.3
Major Poisson's Ratio YZ	0.3
Major Poisson's Ratio XZ	0.3
Shear Modulus XY	4.2e+009 Pa
Shear Modulus YZ	4.2e+009 Pa
Shear Modulus XZ	4.2e+009 Pa
Density	1600. kg/m³
Allowable Stress	440e+006 Pa
E Glass Polyester Resin	
Young's Modulus X direction	3.4e+010 Pa
Young's Modulus Y direction	6.53e+009 Pa
Young's Modulus Z direction	6.53e+009 Pa
Major Poisson's Ratio XY	0.217
Major Poisson's Ratio YZ	0.366
Major Poisson's Ratio XZ	0.217
Shear Modulus XY	2.433e+009 Pa
Shear Modulus YZ	1.698e+009 Pa
Shear Modulus XZ	2.433e+009 Pa
Density	2100. kg/m³
Allowable Stress	420e+006 Pa

Weight calculations:

Volume of universal joint	$= 0.0001726 m^3$
Volume of spiders	$= 0.00005166\ m^3$
Volume of shaft	$= 0.002\ m^3$
Volume of slip yoke	$= 0.0001504\ m^3$
Volume of total assembly	$= 0.00255\ m^3$

Weight of assembly = volume×density×9.81(n)

For steel

Weight of assembly = 0.00255×7600×9.81 = 177.13n

For e-glass:

Weight of assembly = 0.00255×2000×9.81

= 46.62n

For e-carbon

Weight of assembly 0.00255×1600×9.81

= 37.3n

For e-glass polyester

Weight of assembly = 0.00255×2100×9.81

= 48.932n

Meshing Element = tetrahedral Element length = 0.02m
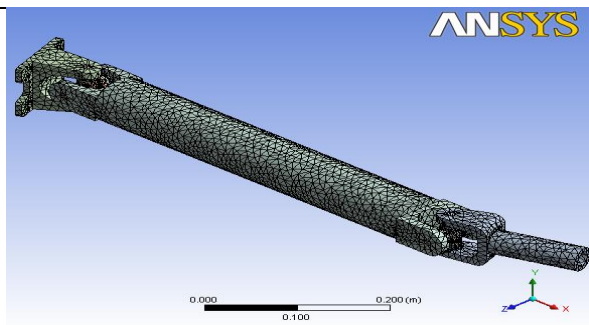
Results

Property/material	Structural Steel	E-glass	E-carbon	E-glass poly.resin
WEIGHT(N)	177.2	46.6	37.3	49
%REDUCTION IN WEIGHT	-	74%	79%	72.4%
MAXIMUM PRINCIPAL STRESS(Pa)	1.28e008	9.74e07	1.11e08	8.11e007
%REDUCTION OF MAXIMUM PRINCIPAL STRESS	-	24%	13.3%	37%
MAXIMUM SHEAR STRESS(Pa)	8.34e007	8.16e07	7.34e07	7.14e007
%REDUCTION OF MAXIMUM SHEAR STRESS	-	2.2%	11.9%	14.4%
DEFORMATION (m) — MAX	0.00123	0.00155	0.00165	0.00345
DEFORMATION (m) — %INCREASE IN	-	20.6%	24.5%	64.4%

	DEFORMA TION				
EQUIVALE NT STRESS OR VON- MISES STRESS (Pa)	MAX	1.6e00 8	1.56e 008	1.41e 008	1.3e00 8
	ALLOWAB LE STRESS	3.7e0 09	4e00 9	4.4e0 09	4.2e0 09

Conclusion:

By conducting analysis on three different composite materials

E-GLASS POLYSTER has 19%reduction in Von-Mises stress and 72.4%reduction in weight than Structural Steel. But it has 64% increase in deformation than Structural Steel.

E-CARBON has 12%reduction in Von-Mises stress and 79%reduction in weight than Structural Steel. But it has 24.5% increase in deformation than Structural Steel.

E-GLASS POLYSTER has 2.5%reduction in Von-Mises stress and 74%reduction in weight than Structural Steel. But it has 20.6% increases in deformation than Structural Steel.

By the obtained results it can be conclude that the stresses induced in all the materials are within their allowable limits. And it can also be observed that the materials which develop less von-mises stress exhibit a little more deformation. Though E-Glass Polyester Resin induces 19% less stresses compared to structural steel, considering the changes in both deformation and stress and weight (which is least - 1600 kg/m^3 among all the above materials), it can be concluded that E-CARBON can be used instead of conventional material like structural steel. So that the weight and stresses induced in the drive shaft can be considerably decreased.

References

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Details of ANSYS

http://en.wikipedia.org/wiki/ANSYS

Specification of Toyota quails

http://www.cybersteering.com/cbmain/utilcars/qualis_gs.html

Composite materials
http://en.wikipedia.org/wiki/Composite_material

CATIA V5R16

http://en.wikipedia.org/wiki/CATIA