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STRUCTURAL ANALYSIS OF A PROPELLER SHAFT UNDER MAXIMUM ROTATION FORCE

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Abstract: In this project work, the propeller shaft of a vehicle was chosen for a material replacement study, seeking alternative solutions to reduce weight while maintaining the vehicle's overall quality and reliability. The objective was to explore different materials that could be utilized in the construction of the propeller shaft, aiming to achieve weight reduction without compromising its performance. The project involved an in-depth analysis of various material options and their potential suitability for the propeller shaft application. The focus was on identifying alternative materials with favourable characteristics such as high strength, low weight, enhanced stiffness, and resistance to fatigue. By examining these materials, the project aimed to find substitutes that could effectively reduce the weight of the propeller shaft. Through the exploration of different material choices and thorough evaluation, the project sought to optimize the design of the propeller shaft, resulting in a lighter overall vehicle. This weight reduction would directly translate into improved fuel efficiency, especially in urban driving conditions where the impact of weight reduction on fuel consumption is particularly significant. In summary, the project centered on a material replacement analysis for the propeller shaft, with the goal of finding alternative materials that could offer weight reduction while ensuring the vehicle's quality and reliability.

Index Terms Catia Software, Propeller shaft, Solid Works 14.0, Hypermesh, Ansys.

I. INTRODUCTION

An automobile may use a longitudinal shaft to deliver power from an engine transmission to the other end of the vehicle before it goes to the wheels. A pair of short drive shafts is commonly used to send power from a central differential, transmission, or transaxle to the wheels. Drive shaft (Propeller shaft) is a mechanical part of transmission system which is used to transfer the power from engine to the wheel. The movement of vehicles can be provided by transferring the torque produced by engines to wheels after some modification. The transfer and modification system of vehicles is called as power transmission system and have different constructive features according to the vehicle’s driving type. Most automobiles today use rigid driveshaft to deliver power from a transmission to the wheels. A pair of short flexible driveshaft is commonly used in cars to send power from a differential to the wheels. In automobiles, axle shafts are used to connect wheel and differential at their ends for the purpose of transmitting power and rotational motion. In operation, axle shafts are generally subjected to torsional stress and bending stress due to self-weight or weights of components or possible misalignment between journal bearings.

II. LITERATURE REVIEW

The various works are carried out in stress analysis, Modal analysis of automobile chassis brackets. Among which few are categorized and discussed below.

Mohammad Reza Khoshravan, Amin Paykani, This paper presents a method for designing and analyzing composite propeller shafts, aiming to replace metallic drive shafts with two-piece composite alternatives. The design process focuses on two main aspects: designing the composite shaft itself and designing the couplings. Various parameters such as critical speed, static torque, and adhesive joints are studied, considering nonlinear isotropic behavior for adhesives of composite drive shafts. [1]

A.S.chavan, Mr.S.S.Chavan Shaft and coupling assembly produce as per the customer requirement. It includes pump, motor, shaft with coupling. After discussion, company came to conclusion that there is problem regarding torque and speed reduction. So company decided to use propeller shaft and gear box for this we are going to design propeller shaft. In our project shaft coupling assembly have to transmit torque 15000 Nm at 30 rpm. Innovative Patented Design and in-depth knowledge about sugar pumps Industry, has given us wide acceptance in sugar industry from India. [2]

D.L. Duquesnay, T.H. Topper. This investigation examines the hypothesis that variations in the crack opening stress level of short cracks can account for the observed variations in fatigue strength with mean stress under constant-amplitude cyclic loading. Support for the hypothesis is provided by the experimental data generated for a 2024-T351 aluminium alloy and a SAE

1045 steel. Based on the observed variations in crack opening stress with mean stress an effective stress range is postulated as a mean stress parameter. The effective stress range successfully reduces the fatigue data for all levels of mean stress to a unique fatigue-life curve for each material. [3]

K.**Solanki et al.** have studied the failure reason of AISI 304 stainless steel drive shaft. The main vibration reason for failure is low natural bending frequency. R.B. Ingle et al.[4]

R.B. Ingle et al. the research work highlights the composite material use in aerospace and aircraft industry in supercritical operating condition. The composite shaft made of carbon fibers in epoxy matrix, 16 layers with different stacking angles was used for analysis. The static and dynamic behaviour was investigated at high speeds (10000-65000) rpm. The vibration spectrum of composite shaft was observed in aerospace conical bearing. [5]

III. METHODOLOGY

- To review the existing literature on Propeller Shaft
- Based on application and reviewed literature design specifications were arrived.
- Geometrical modelling of Propeller Shaft will be created using Catia Software
- Finite element model was created using the Hypermesh Software
- FE analysis performed using ANSYS Software
- Design calculation has been carried out to evaluate the Stress criteria.

IV. PROBLEM IDENTIFICATION

Propeller shaft is used to transmit the power from the engine to the rear axle. Hollow propeller shaft is used to reduce weight and inertia losses. It is subjected to torsional shear stress. So they need to be strong enough to withstand this stress. Some of the cases, the propeller shaft contains higher torsional shear stress. Mostly it is depend on the road condition.
Considering the starting condition of the car as worst load case, the maximum torque is experience by the propeller shaft. As the first gear has high gear ratio increases the torque transmitted at the gear box and at less rpm. So the starting condition that is vehicle in first gear and just about to move will input the high torsional shear stress in the propeller shaft, hence this condition is taken to model and analyzed.

V. GEOMETRICAL MODELING

A geometrical model is created in with 1:1 scale in CATIA V5-R17 (figure no.). A geometrical model is created with universal joint yoke at one end and splines of slip joint at other end. This model as to be meshed for analysis, so considering time constrains, and to make geometry simple some simplification are made.
Simplifications with justification:

- All fillets are removed: fillets are difficult for meshing.
- Splines are removed: meshing is complicated in spline area.
- Yoke: yoke dimensions are simplified to avoid the complicated curved as it is difficult in meshing and also yoke is not area of interest. It reduces the time required for meshing.
- Welding area: welding area avoided in model, as welding is not uniform in area so it is difficult and critical to mesh.
- Flange is not considered for the modeling because of the time constrains.

A. Geometric Modelling of Propeller Shaft

The modelling is finished utilizing Catia with the assistance of standard measurements that we overcame the cautious survey. The Catia is one of the renowned demonstrating programming accessible in the market which empowers us not exclusively to do the displaying of the segments yet in addition the investigation of the same. Hence it is one of the ideal software for modeling and analysis problems. The following diagram is the model of the Propeller Shaft which is created using the Catia Software.

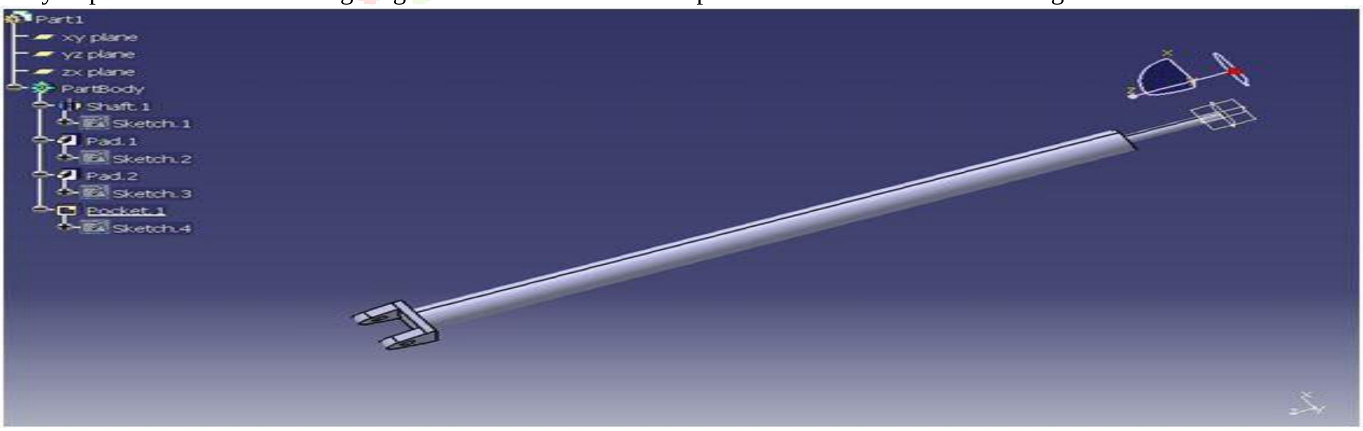


Fig 1:- Propeller Shaft Geometric Model

B. Design Assumptions of Mono Crankshaft

Propeller shaft transmit power coming out from the gear box to the differential and then the axle. It transmits the torque either positive or negative (in reverse gear). It uses to transmit the angular torque; it also permits the vertical movement of the wheels while transmitting the torque; the propeller shaft stands with axial thrust, bending moment; torsional shear stress.
Most of the time the propeller shaft transmits the power at some angle, it is not horizontal mounted, but some cars have propeller shaft in horizontal position rather than angled. Propeller shaft is connected by the universal joints to the transmission shaft and

differential. In small cars the shaft is in horizontal position it and generally connected with the flange the other end to connect the transmition shaft.

VI. FINITE ELEMENT MODELING

In this chapter a geometric model is converted into a Finite Element Model, selection of element type, assumption of loading and boundary condition are discussed as below:

The limited component is a scientific strategy for settling common and halfway differential conditions, due to its numerical technique it has a capacity to take care of complex issues which are spoken to as differential condition. These sorts of conditions happen normally in all fields of the physical science and application shrewd it is boundless as concern the arrangement of handy plan issues. Commonly FEA is described as a discretization technique.

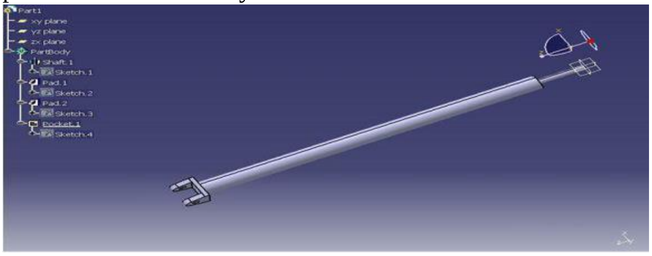


Fig 2:- Geometric Model of the Mono crankshaft

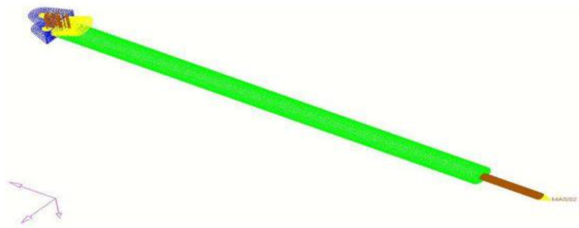


Fig 3:- Finite Element model of a mono crankshaft

Basically geometry have a various entities like point, lines, curves, areas, surfaces, volume & solids. But in the FEA we have only two entities i.e, nodes & elements. FEA entities are build with respect to the geometric entities & for the simulation only FEA entities are considered.

Finite element modeling of the Propeller Shaft is done in the Hypermesh 10.0 software. And the mesh model of the crankshaft is as shown in the figure 4.

Table 1. Element Quality Parameters for a 3D Solid Elements

Sl.No	Description	Required Quality	Achieved
		Parameter	Parameter
1	Warpage	<5	0.65
2	Aspect ratio	<5	4.98
3	Jacobian	>0.65	0.67
4	Min. Angle	>45	45.56
5	Max. Angle	<135	132.55
6	Total No. Elements	-	161840

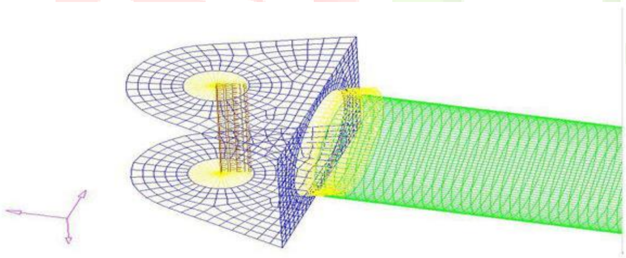


Fig 4:- Meshing of Yoke and Shaft Area with pipe element

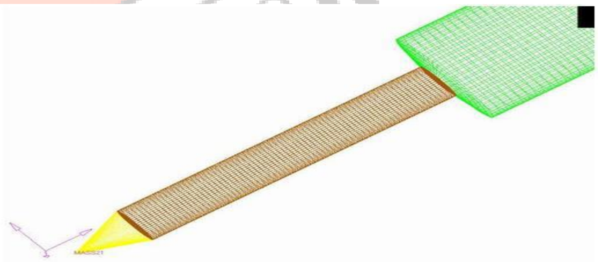


Fig 5:- Meshing of Spline Shaft and mass element

VII. MATERIALS PROPERTIES

The material properties used for the Sheet metal and its mechanical properties is as shown below

Table 2. Materials properties

SL No.	Description	
1.	Components	Crankshaft
2	Material	Steel
3.	Young's Modulus	2.1e5Mpa
4.	Poisson's Ratio	0.3
5.	Yield Strength	250Mpa

VIII. DEFINING THE PHYSICAL CONSTRAINT (BOUNDARY CONDITIONS)

The propeller shaft has flange at one side and yoke at other, so this shaft remains horizontal while working; the propeller shaft is subjected to following boundary condition while working as shown in fig.4 from figure.5 Z is the longitudinal direction of the shaft, while Y is the vertical and X is the transverse direction. Torque coming out of the engine is transmitted through gear box to flange. At this end the flange is fixed to the flange of transmission box shaft, because of that one degree of freedom is remains free, which is at flange rotating about its own axis (Z axis).

Defining the Loading conditions and Calculations

For this assignment the starting case 1 is consider for loading in FE model that is the vehicle is at the starting from the rest condition For this case consider the rated torque which is less than the maximum torque. That is 135 Nm. So 120 Nm torque is supposed to be applied when vehicle is in the first gear and it is about to move. This is the torque coming out of the engine shaft, and then there is the gear ratio for the first gear is 3.736:1, there for actual torque transmitted by the gear box to the propeller shaft = 110 x 3.376= 410.96 Nm = 410960 N-mm. Momentum experienced by the shaft at its centre is 410960 N.mm, so torque applied at the splines face to get that momentum is = 410960 / 11.25 = 36529.7 N.mm, as shown in fig. () torque applied at point P 39861.3 N.mm gives momentum of 448440 Nm.

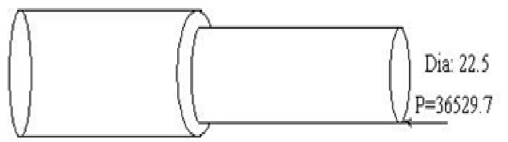


Fig 6:- Shaft with boundary conditions

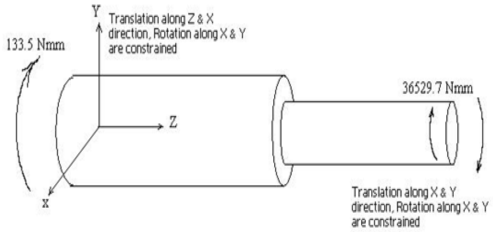


Fig 7:- Torque applied at splines on the shaft

Consider the vehicle is moving from the rest; engine torque has to overcome the inertia force of the vehicle that is, it has to overcome rolling resistance. This force acts as the torque in the opposite direction of the torque coming from the engine shaft at the universal joint at the final drive side. The vehicle has C.G. position at 55:45 of wheel base. That is load at rear wheel is 55% of the gross vehicle weight.

Gross vehicle weight is 1650 Kg that is (1650 x 9.81) N = 16186.5 N. The rolling resistance is (WR x μ x cosθ), condition the vehicle is standing on straight θ is zero, μ is 0.015, [4].

- So the rolling resistance = (16186.5 x 0.015) = 242.79 N.
- So rolling resistance for rear wheel is 2.4279 x 0.55 = 133.5 N

This can be mentioned as 133.5 N-mm torque. This force acts in the opposite direction of the torque coming out of the engine. Finally with boundary conditions shaft is like as shown in fig.

IX. STRESS ANALYSIS & RESULT REVIEW OF A PROPELLER SHAFT

Stress analysis is carried out based on the Static or steady state condition, where the solution is independent of time. Inertial forces are either ignored or neglected and so there is no requirement to calculate actual time derivatives.

Before analysis the basic assumptions are made as shown below:

- All deformations and strains are little.
- Structural disfigurements are relative to the heaps connected.
- All materials act in a direct versatile manner. Henceforth, the material twists along the straight line part of the pressure strain bend. Very confined pressure fixations are typically allowed insofar as gross yielding does not occur.
- Loads are on the whole static. This implies the heaps are connected to the structure in a moderate or enduring style and in a way that influences them to time autonomous
- No limit condition differs with time or utilization of load.

To perform a stress analysis of a crankshaft ANSYS 10 software is used, meshed model of Mono-Crankshaft with steel as a material property as shown in the table, is imported from the HYPERMESH 8 software.

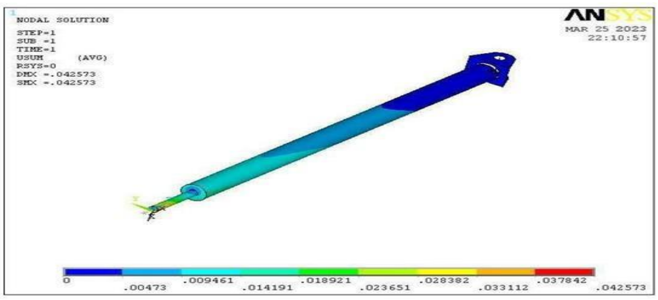


Fig 8:- Displacement propeller shaft

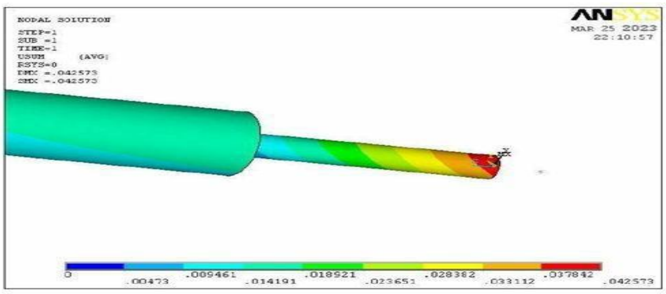


Fig 9:- Closer view of displacement propeller shaft

The graphical representation of the above figure 8 shows the displacement variation on several location of the Propeller Shaft. The maximum displacement observed due to surface load is 0.0425 mm

From the above figure it is clear that maximum displacement is in the solid shaft and the displacement is occurring in the plane inclined almost at 40-450 to the plane XY. Displacement vector sum is the sum of the displacement sum in X, Y, and Z direction. And the maximum displacement in the shaft is 0.0425 mm.

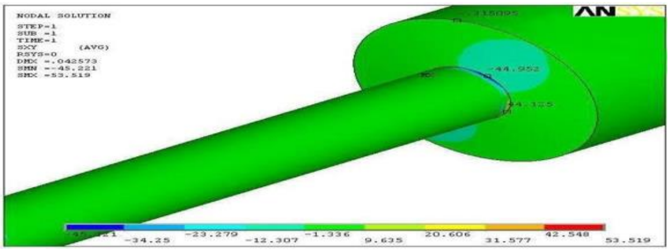


Fig 10:- Shear stress in XY plane of steel propeller shaft

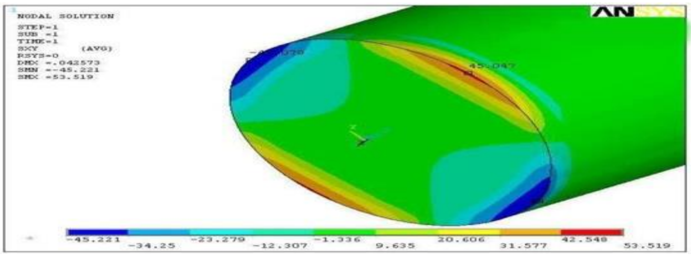


Fig 11:- Shear stresses in the XY plane of shaft end

perimeter of the solid shaft at the splines end. So the area in red colour in fig. Is nearer to the torque applied, also diameter against the torque is less, hence showing high displacement. From the fig.we can conclude that shaft is in the alternate compressive and tensile stress as go on circular surface also of equal magnitude. Figure shows the values of the shear stress 44.952 N/mm2 in compressive nature and 44.125 N/mm2 tensile in nature. As we are interested in the shear stress as moves away from the radial direction, shear stress in XY plane is the point of interest.Actually the spline could be the area of interest as the more changes in area, so that could be the area of high stress. Splines have less area against the torque so that should be highly stressed area. As shown in the fig. Nature of shear compressive and tensile stresses is with higher tensile value (53 N/mm2) at the spline end of the solid shaft, but here stresses are higher at the outer surface of the shaft. And goes on reduces as radius goes less.

X. DESIGN CALCULATION

Under maximum load so first calculate the shear yield point, with this value we can ensure that the values are with the permissible range or it is more than limit.

Shear yield point = 0.58 × $Ş_{yp}$ = 0.58× tensile yield strength

Tensile yield strength of the material steel is **284 Mpa.**

Shear yield point = 0.58 × 284 = **164.89 N/mm^{2}.**

Ultimate shear strength =0.5 x Ultimate tensile strength of the material

Ultimate shear strength = 0.5 x 385 = **192.5 Mpa.**

For calculating the shear stresses in the shaft. We have to calculate polar moment of inertia for the solid and hollow shaft.

J_{solid} = p/32×d^4 = p/32×22.5^4 = 25161.12mm; ……..[3]

J_{hallow} = p/32 x (D^4-d^4) = p/32× (64^4-60^4) =374754.13mm; ……..[3]

For calculate the angular deflection θ.

$\Theta = \frac{T\,L}{J\,G} = \frac{T}{G}(l_{solid}/J_{hollow} + l_{hollow} / J_{hollow})$

T= Maximum torque = 160 N-mm; …... [3]

G = Modulus of rigidity = 80 GPa

$\Theta_1 = \frac{160000}{80081.63}(152/25161.12 + 1090/374754.13) = 0.0179$ rad;

The inertia toque also applied on the reverse direction so that to get the actual deflection we have to take deduct this value from the opposite torque is also applied, so deflection due to that torque has to be calculated and then deduct that from that above deflection. θ1

$\Theta_2 = \frac{T\,L}{J\,G} = \frac{133.5}{374754.13}\,\frac{1090}{80081.63} = 2.66 \times 10^{-6}$ rad;

The stresses due to the maximum torque T in the solid shaft

$D = 1.72(\frac{Tmax}{\sigma max})^{1/3}$………[4];

For this Tmax is 160000 N-mm and D is 22.5 mm, therefore σmax 71.41 N/mm is the maximum shear stress in the solid shaft.

Maximum torque T in the hollow shaft can calculate by:

$T =$ p/16 f_x (D^4-d^4 / D)…….[4];

T = 160000 N-mm,

D is outer diameter of the shaft= 64 mm

d is inner diameter of the shaft = 60 mm.

So the fs = 13.66 N/mm.

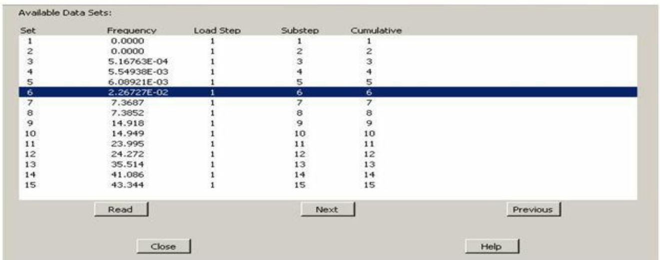
XI. MODAL ANALYSIS OF PROPELLER SHAFT

It is the technique used to determine a structure's vibration characteristics of a structure or machine components, mainly:

- Modal analysis is used to find the natural frequencies a structure
- The frequencies are calculated in increasing order of frequency magnitude. Users can define number of frequencies desired or a range of frequency magnitudes.
- Two things are important - mode shape and frequency. The actual values of displacement are not physically meaningful, only the shape of the deformation is important.
- The information modal analysis gives is extremely valuable - it can help to make design decisions without further vibration analysis required.

A. Modal Analysis results

Modal analysis is used to find out the natural frequencies and mode shapes of a structure. These are very important parameters in the design of structure for dynamic loading conditions. In the ANSYS, modal analysis is a linear analysis. Modal analysis is done by the "Block Lancoz method". For this analysis meshed model is imported from Hypermesh software, and analysis is done. For the modal analysis material properties like Modulus of elasticity (2.1e5) and density (7.86e-6)of SM45C are necessary. Also we have to make a decision how many numbers of mode shapes to be extracted. Natural frequency of the shaft is calculated when there is no load and no constrains, this analysis is called free-free analysis. In fixed-fixed analysis constrains are applied without load. In this type of analysis no constrains are applied to the propeller shaft and first 15 modes are extracted within frequency range of 0-25 Hz. As maximum rpm of propeller shaft would be 5500 for this selected car, the frequency of maximum rotation is 14.59 Hz. For higher frequency shaft will not give that much response, so next higher frequencies are not the point of interest. Extracted natural modal frequencies are shown in table .



Available Data Sets:

Set	Frequency	Load Step	Substep	Cumulative
1	0.0000	1	1	1
2	0.0000	1	2	2
3	5.16763E-04	1	3	3
4	5.54938E-03	1	4	4
5	6.08921E-03	1	5	5
6	2.26727E-02	1	6	6
7	7.3687	1	7	7
8	7.3852	1	8	8
9	14.918	1	9	9
10	14.949	1	10	10
11	23.995	1	11	11
12	24.272	1	12	12
13	35.514	1	13	13
14	41.086	1	14	14
15	43.344	1	15	15

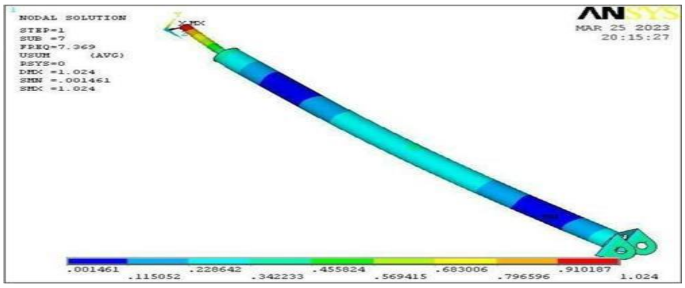


Fig 12:- Mode shape of fundamental natural frequency

Fig 13:- Natural frequencies in free-free conditions

We get the first two modes are exactly zero and next four are almost zero, because in free-free conditions first six modes have the natural frequency zero. The next one is 7.3687 Hz, and it is called as fundamental natural frequency. Mode shape for this frequency is shown in fig.12. We can observed that there are no fixed nodes on the hollow shaft, there for there is no zero displacement value is showing in fig. The ninth mode of vibration is 14.918 Hz and it is relative to the maximum rpm of the rotation of the shaft that is 5500 is shown in fig

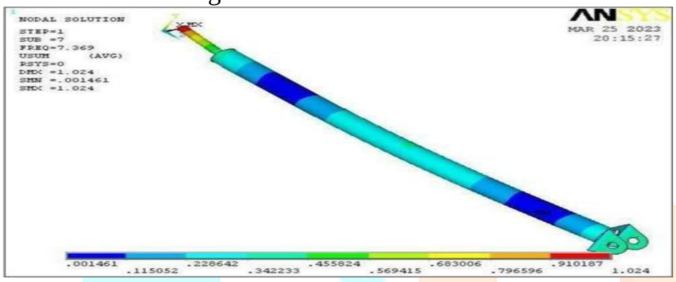


Fig 14:- Ninth mode of vibration of the shaft at 14.918 Hz

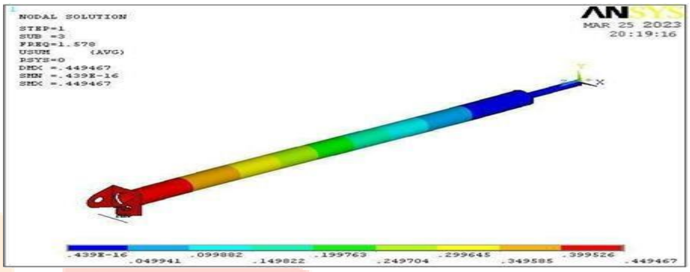


Fig 15:- The third mode of vibration of shaft at 1.578 Hz

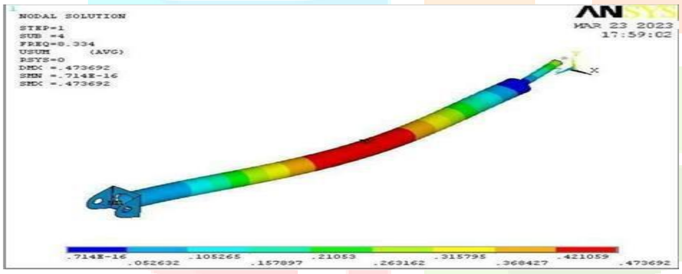


Fig 16:- Shear stress in X-Y plane shaft.

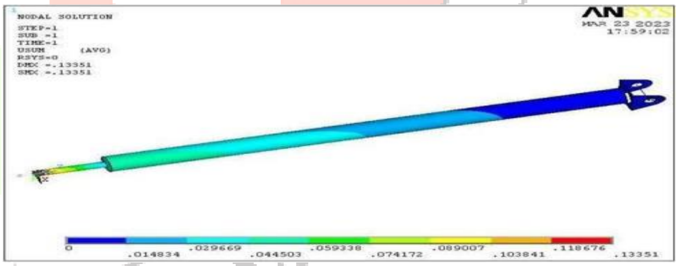


Fig 17:- Fourth mode of at 8.334 Hz

Shear yield point for aluminum is:

- Figure tells about the shear stress above 22.03 N/mm2 is measuring on solid shaft.
- For the same dimensions aluminum with tensile strength 40 MPa and shear modulus 27 GPa. Cannot able to uphold this load.
- Though the aluminum is failing on this load we can also use the aluminum alloy with the higher grade of tensile strength and shear with geometrical modifications. But there are certain drawbacks of using aluminum as a material for propeller shaft, Manufacturing with aluminum is difficult and costly Availability of aluminium is less than steel
- Welding is the big problem with aluminum
  0.55 x Tensile yield strength = 0.55 x 40.04 =22.03 N/mm^2 ……………..[2]

Comparing the values shear stresses variation in aluminum and steel, values of shear stress both in compressive and tensile is almost same. But deflection in aluminum shaft is larger than steel material shaft.

XII. CONCLUSION

- In this thesis an existing vehicle propeller shaft modeling is carried out in CATIA. And the iges file of the model is transfer to hypermesh and the meshing of the model is done and analysis of the model carried out in the Ansys.
- The maximum displacement observed due to surface load is 0.0425 mm From the analysis we can conclude that shaft is in the alternate compressive and tensile stress are go on circular surface also of equal magnitude and the tensile shear stress observed is 44.125 N/mm2 and the compressive shear stress observed is 44.952 M/mm2.Allowable yield strength of the material is 284 Mpa and the allowable ultimate shear strength is 192.5 Mpa.
- Hence the maximum torque on the solid shaft is 160000 N-mm, therefore the obtained stress due to maximum torque on the solid shaft is 71.41 N/mm2 and the obtained stress due to maximum torque on the hollow shaft is 13.66 N/mm2.Therefore the shear stress obtained on the solid shaft is 71.41 N/mm2 which is under the shear yield point of 164.89 N/mm2 having FOS>2. Hence the design is considered to be safe.

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