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DESIGN AND ANALYSIS OF POWER TRAIN SYSTEM OF HEAVY TRUCK ENGINE

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ABSTRACT

A powertrain is a system of mechanical parts in a vehicle that takes the power, or output, of the engine and, through specific gear ratios, slows it and transmits it as torque. Through the driveshaft, the engine's torque is transmitted to the wheels of the vehicle, which, when applied to road, moves the body. Simply put, a powertrain is made up of a transmission system and a driveshaft.The mechanism that transmits the power developed by the engine of the automobile to the driving wheels is called the transmission system (or Power train).It provides a varied leverage between the engine and the drive wheels. It also provides the connection and disconnection of engine with rest of power train without shock and smoothly.The average person is most familiar with the powertrain of their car, which creates energy in the engine, which is transferred to the transmission. Main objective of this project is to design a powertrain system for a truck engine by manual calculation and computer aided designing.

KEYWORDS: Powertrain system, Transmission, Torque, Computer aided designing, Design Calculations.

I. INTRODUCTION

A powertrain is a system of mechanical parts in a vehicle that first transmits the power developed by the engine of the automobile to the driving wheels. The average person is most familiar with the powertrain of their car, which creates energy in the engine, which is transferred to the transmission. The transmission then takes the power, or *output*, of the engine and, through specific gear ratios, slows it and transmits it as *torque*. Through the driveshaft, the engine's torque is transmitted to the wheels of the car, which, when applied to road, moves the car. Simply put, a powertrain is made up of a transmission and a driveshaft.[2]

TATA Prima LX truck engine has been taken for powertrain design purpose.[7]

- Volume of the engine : 5883 cc
- Power : 230HP @ 2500 rpm (0r) 172 Kw @ 2500 rpm
- Torque : 850 Nm @ 1400-1700 rpm
- Gears : 6 speed gear box(5F + 1R)
- Clutch: Single plate dry friction clutch.

Main parts of powertrain system are:

- Clutch
- Gear box
- Propeller shaft
- Universal joint
- Differential

Layout of a powertrain system is shown below in Fig. 1. It consists of clutch, gearbox, propeller shaft, universal joint and differential unit.

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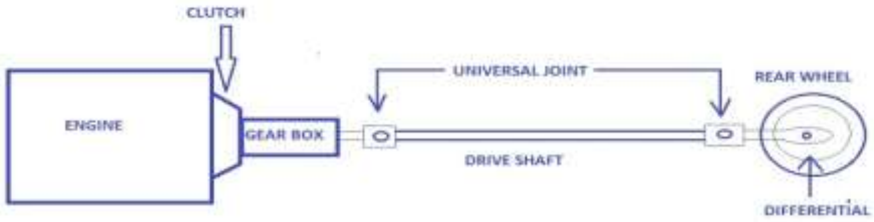

Fig. 1 Layout diagram of Powertrain System

II. DESIGN CALCULATIONS

Design Calculation for Single Plate Clutch
Clutch is designed based on two theories:

- Uniform pressure theory
- Uniform wear theory

Uniform Pressure Theory

Torque can be determined by using the formula below

$T = \mu \times W \times R$

- T= Torquetransmitted by engine.
- μ =Co-efficient of friction of material.
- Fa =Axial force =2× π×p×ri(r_o-r_i).
- R=Effective mean radius= 2/3 × (r_o^3- r_i^3 / r_o^2 – r_i^2)
- Pr = intensity of pressure = p×r_i= p×r_o(p wil be maximum where r is minimum= r_i)

Centre distance layout of a single plate clutch is given in fig. 2 below

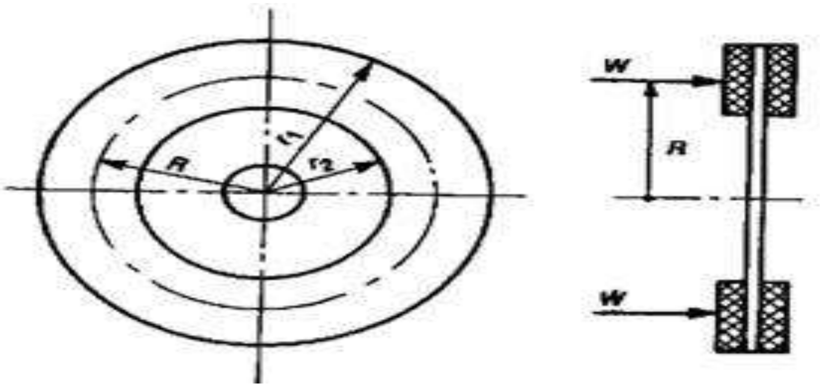

Fig 2 Centre Distance of Clutch Plate

Dimensions

By using design parameters we get the dimensions of single plate clutch

- The outer diameter of the clutch plate **d_o=230 mm**
- The inner diameter of the clutch plate is **$d_{i=}$200 mm.**

N / mm^2

- Intensity of pressure = **0.1**
- Co-efficient of friction , Asbestos material=**0.35**

Calculation of Axial force

Axial force can be determined by using the formula below

$$F_a = 2\pi p(r_o - r_i)$$
$$= 2 \times \pi \times 0.1 \times 10^6 \times 0.1(0.115 - 0.1)$$
$$F_a = 942.47N$$

Torque transmitted by Clutch

The formula of torque is given below

$$T = \mu WR = 0.35 \times 942.47 \times 0.107$$
$$T = 35.52Nm$$

Mean Radius of Friction Surface, R = 0.107m

Dimensions of the springs

Total Load on the springs, $= 1.25F_a = 1.25 \times 942.47 = 1178.08N$

Since there are 6 springs

Therefore maximum load on each spring is, $W_s = \frac{1178.0875}{6} = 196.34N$

We know that Wahls Stress factors, C=6

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = 1.2525$$

We also know that Maximum shear stress induced in the wire is 420Mpa

$$\tau = \frac{K \times 8W_s \times C}{\pi \times d^2}$$
$$420 = \frac{1.2525 \times 8 \times 196.34 \times 6}{\pi \times d^2}$$
$$1319.46d^2 = 1180396$$
$$d = 2.99mm$$

We shall take the Standard wire of size SWG8 having diameter,

\Mean diameter of the spring, $(d) = 3.251mm$

$$D = C.d = 6 \times 3.251 = 19.506mm$$

Let us assume that the spring has 4 active turns (n=4)

Therefore compression of the spring,

$$\delta = \frac{8 \times W_s \times C^3 \times n}{G.d}$$
$$G = 84 \times 10^3 \frac{N}{mm^2}$$
$$\delta = \frac{8 \times 196.34 \times 6^3 \times 4}{84 \times 10^3 \times 3.251}$$
$$\delta = 4.9695mm$$

Assuming squared and ground ends, total Number of turns $n' = n + 2 = 4 + 2 = 6$

We know that free length of the spring

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$$\frac{L_f}{n^{'}-1}=\frac{25.22}{6-1}=5.044mm$$

$$L_f=n^{'}d+\delta+0.15\delta=25.22mm$$

And pitch of the Coils,

Design of Six Speed Gear Box

I assumed the minimum and maximum speed of a six speed gear box is 500 r.p.m and 2500 r.p.m respectively.

Selection of Spindle Speeds

$$n=6; \quad N_{\min}=500r.p.m \qquad N_{\max}=2500r.p.m \qquad \phi^{n-1}=\frac{N_{\max}}{N_{\min}}$$

We Know that,

$$\phi^{6-1}=\frac{2500}{500} \qquad \phi=1.389\approx1.40$$

We find ϕ=1.40 is not a standard ratio, $1.12\times1.12\times1.12=1.404$

$\phi=1.12$

satisfies the requirement. Therefore the sipngle speeds from R20 series, skipping two speeds are given as,

500,710, 1000,1250,1800, 2500 r.p.m

Structural Formula: 3(1). 2(3)

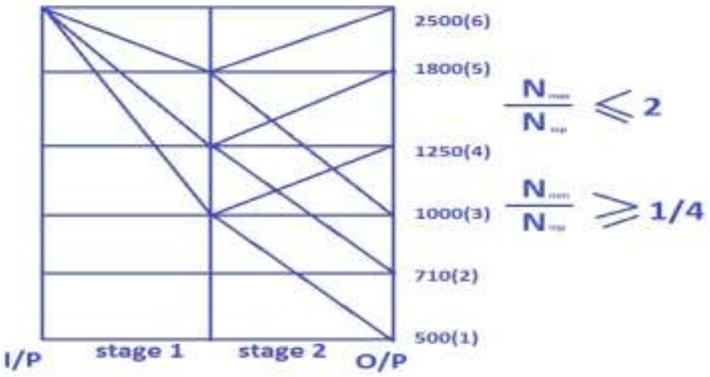

Fig 3 Ray diagram

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Kinematic Arrangement for 6-Speed gear box

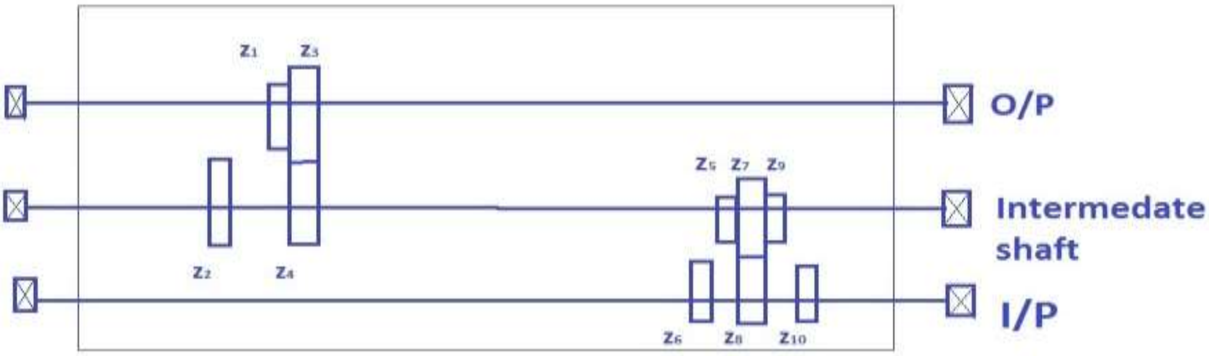

Fig 4 Kinematic Arrangement

Groups

$Z_1 + Z_2 = Z_3 + Z_4$; $Z_5 + Z_6 = Z_7 + Z_8 = Z_9 + Z_{10}$

Design of Spur gears

C45 steel material is selected (PSG DB, Page 8.5)

Design stress, $[\sigma_b] = 1350\ kgf/cm^2$, $[\sigma_c] = 5000\ kgf/cm^2$

E= 2. 15 × $10^6 kgf/cm^2$Ψ=0.3ψ_m =10

We assume Z_1=20 , kk_d=1.3

Centre distance between output shaft and input shaft

Nominal twisting moment transmitted, $[M_t] = 97420 \times \frac{kw}{n} \times 1.3$ = 21783 kgf-cm

(PSG DB, Page 13)$a \geq (i+1)\sqrt[3]{[\left[\frac{(0.74)}{[\sigma_c]}\right]^2 \frac{E\ [M_t]}{i\psi}]}$

I=1000/500=2 $a \geq 16cm\ or\ 160\ mm$

Module $m \geq 1.26\sqrt[3]{\left[\frac{[M_t]}{y\ [\sigma_b]\psi_m z}\right]} \geq 0.75cm$ y=0.38

From PSG DB Page 8.2, module=8 mm

Calculation of number of teeth

$$Z_1 = \frac{2a}{m(i+1)} = \frac{2 \times 160}{8(2+1)} = 13.3\ ; Z_2 = i \times Z_1 = 26.6 \cong 27$$

$Z_1 + Z_2 = Z_3 + Z_4 = 40$; $Z_3 + iZ_4$ =40; $\frac{Z_4}{Z_3} = i = 1.25$

$$Z_3 = 18\ ;\ Z_4 = 22$$

Calculation of gear Diameter

$$d_1 = mZ_1 = 104mm\ ; d_2 = 216mm\ ;\ d_3 = 144mm;\ d_4 = 176mm$$

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Centre Distance between Intermediate shaft and Input shaft

$[M_t] = 97420 \times \frac{kw}{n} \times 1.3$ =12101.7 kgf-cm

$$a \geq 28.5\ cm\ or\ 285mm$$

$Z_5 = \frac{2a}{m(i+1)} = 25\ ;\ \ Z_6 = 45$

$Z_7 + Z_8 = 70\ ; Z_7 + iZ_8 = 70\ i = \frac{Z_8}{Z_7} = 1.38$

$$Z_7 = 29\ ;\ Z_8 = 41$$

$$Z_9 + Z_{10} = 70\ ;\ Z_9 + iZ_{10} = 70\ ;\ Z_9 = 28;\ \ Z_{10} = 42 i = \frac{Z_{10}}{Z_9} = 1.44$$

$d_5 = 200mm$; d_6=360mm; $d_7 = 232mm$; $d_8 = 328mm$; d_9=224mm; $d_{10} = 336mm$

Calculation of Revised Centre distance

Centre distance b/w output and intermediate shaft, $a_1 = \left(\frac{d_1 + d_2}{2}\right) = 160mm$

Centre distance b/w intermediate and input shaft, $a_2 = \left(\frac{d_5 + d_6}{2}\right) = 280mm$

Calculation of face width

$b = \psi \times m = 80$

b= 80 mm

Calculation of Length of shaft

Length of the Shaft, L= (30+10+4b+20+7b+7b+10+30)mm, L=100+18b=1540mm

Design of shafts

Design of output shaft

1) To find Maximum bending moment (M); $M = \frac{F_n \times L}{4}$

the lowest speed is 500 r.p.m is obtained by meshing gears 1 and 2

$$T_1 = \frac{P\times 60}{2\pi N} = \frac{172\times 10^3 \times 60}{2\times\pi\times 500} = 3286.64\ N-m$$

Fn = Normal load on gear = $\frac{F_t}{\cos\alpha} = \frac{3286.4}{\cos 20^\circ} = 3497.3N$

Maximum B.M, M $= \frac{3497.3\times 1540}{4} = 1.346\times 10^{64}\ N-mm$

To find the Equivalent torque (Teq)

The formula is given below

$$T_{eq} = \sqrt{M^2 + T^2} = \sqrt{(13.46\times 10^6)^2 + (3287\times 10^3)^2} \qquad T_{eq} = 13.8\times 10^6\ N-mm$$

Diameter of the Spindle

From PSG book table 9.5 $\tau = 30\frac{N}{mm^2}$

$$d_s = \left[\frac{16\times T_{eq}}{\pi[\tau]}\right]^{1/3} = \left[\frac{16\times 13.8\times 10^6}{\pi\times 30}\right]^{1/3} = 132.8mm \cong 133mm$$

Rounded off values of the diameter, using R20 series is 135mm

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Selection of Bearing

According to the spindle diameter, Deep Groove Ball Bearing SKF6026 28 has been selected.

From PSG data book Series 60, Page 4.12

Design of Universal Coupling

In designing a universal coupling, the shaft diameter and the pin diameter is obtained and discussed below,

d = diameter of shaft

dp = Diameter of pin

τ1= Allowable shear stress for the material of the shaft and pin respectively.

Torque transmitted by the shafts

$$T = \frac{\pi}{16} \times \tau \times d^3$$

Since the pin is in double shear, therefore the torque transmitted

$$T = 2 \times \frac{\pi}{4} (d_p)^2 \tau_1 \times d$$

Assuming the torque transmitted by the engine is

$$T = 14 \times 10^6 \, N - mm$$

The allowable shear stresses for the shaft and pin may be taken as 60 MPa and 30 MPa respectively**.**

Diameter of Shaft $T = \frac{\pi}{16} \times \tau \times d^3$ $\quad d^3 = 1.188 \times 10^6$

$$14 \times 10^6 = \frac{\pi}{16} \times 60 \times d^3 \quad d = 105.9mm \cong 106mm$$

Diameter of Pin $T = 2 \times \frac{\pi}{4} \times (d_p)^2 \times \tau_1 \times d$

$$14 \times 10^6 = 1335.30 \times (d_p{}^2) \quad d_p = 52.95mm \cong 53mm$$

Design of Propeller Shaft

- The assumptions are shaft rotates at a constant speed about its longitudinal axis.
- The shaft has a uniform, circular cross section.
- The shaft is perfectly balanced, i.e., at every cross section, the mass center coincides with the geometric center.
- All damping and nonlinear effects are excluded.

Taking a simple drive shaft or propeller shaft is designed using following assumed data,

Maximum Torque (T) = 3500 N-m

Length of the shaft L = 1250 mm

Inclination angle (θ) = 2 Deg

Density = 7600 $\frac{Kg}{m^3}$

Yield Stress in shear = 370 Mpa

Rotational speed (N) = 6500 rpm

Young's Modulus = 38×10^4 Mpa

To start with, take the diameter of driving shaft is 50 mm

The assumed diameter is then will be used as input for the torsional shear stress

Calculation of polar moment of inertia (I)

$$I = \frac{\pi d^4}{32} = \frac{\pi \times 50^4}{32} = 6135923 mm^4$$

Where, d = diameter of driving shaft

Calculate the Maximum torsional shear stress, $S_{\max}$

$$S_{\max} = \frac{Td}{2I} = \frac{3500 \times 10^3 \times 50}{2 \times 6135923} = 142.2 Mpa$$

We know Yield strength for driveshaft material is 370 Mpa (C45)

So, we can conclude the propeller shaft is safe for the transmitted torque in the shaft.

Calculate Maximum static deflection of the drive shaft (δ)

$$\delta = \frac{5.m.g.\cos\theta.L^3}{384.E.I}$$

$$= \frac{5 \times 18.65 \times 9.8 \times COS\,2 \times (1250^3)}{384 \times (38 \times 10^4) \times 6135923}$$

$$= 0.02009 mm$$

Where, m = mass of the propeller shaft = 18.65 kg

g = acceleration due to gravity = 9.81 m/s2

θ = Inclination angle = 2 deg

L = length of the propeller shaft = 1250 mm

E = Young's modulus = $38 \times 10^4 Mpa$

Calculate the Critical speed of the Shaft (N_c)

$$N_c = \frac{30\sqrt{g}}{\pi\sqrt{\delta}} = \frac{30\sqrt{9.8}}{\pi\sqrt{0.00002009}}$$

$$= \frac{30}{\pi} \times 701.87 = 6696 rpm.$$

Therefore it is seen that speed of the drive shaft is 6500 r.p.m, which is lower than the calculated critical speed of shaft(6696rpm). Design is safe.

Design of Differential

By using PSG Data book, the following formulas are listed below

Formula Used

- No of Teeth = Z
- Module, m_t = 5mm
- Cone Distance, R = $0.5m_t\sqrt{(z_1^2 + z_2^2)}$
- Outer Pitch Diameter, d = $m_t .z$
- Tip Diameter $d_{a1} = m_t (z+2 \cos\delta)$
- Face width, b = $10.m_t$
- Pitch angle, $\tan\delta = i$
- Tip angle, $\delta_a = \delta + \theta_a$
- Root angle, $\delta_f = \delta - \theta_a$
- Addendum, $h_a = m_t$
- Dedendum, $h_f = 1.88m_t$
- Addendum angle, $\tan \theta_a = (m_t f_0)/R$
- Dedendum angle, $\tan \theta_f = (m_t(f_0+c))/R$

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- Height factor = 1
- Clearance = 0.2 mm

Pinion Gear Calculation

Pinion Gear – It is the small Gear in the differential unit

- No of Teeth = 16 teeth
- Module, m_t = 5mm
- Cone Distance, R = 126.491mm
- Outer Pitch Diameter, d = 80mm
- Tip Diameter d_{a1} = 89.486mm
- Face width, b = 40mm
- Pitch angle, δ = 18.44°
- Tip angle, δ_a = 20.7°
- Root angle, δ_f = 15.719°
- Addendum, h_a = 5mm
- Dedendum, h_f = 9.4mm
- Addendum angle, θ_a = 2.263°
- Dedendum angle, θ_f = 2.715°
- Height factor = 1
- Clearance = 0.2 mm

Ring Gear Calculation

Ring Gear – It is largest size gear in the differential unit

- No of Teeth = 48 teeth
- Module, m_t = 5mm
- Cone Distance, R = 126.491mm
- Outer Pitch Diameter, d = 240mm
- Tip Diameter d_{a1} = 243.163mm
- Face width, b = 40mm
- Pitch angle, δ = 71.56°
- Tip angle, δ_a = 73.823°
- Root angle, δ_f = 68.845°
- Addendum, h_a = 5mm
- Dedendum, h_f = 9.4mm
- Addendum angle, θ_a = 2.263°
- Dedendum angle, θ_f = 2.715°
- Height factor = 1
- Clearance = 0.2 mm

Identical Bevel Gear (miter gear) Calculation

Miter Gear - A type of bevel gear used in pairs with intersecting shafts at 90° angles. Both the driving gear and driven gear in a miter gear pair have the same diameter, same number of teeth, and a mechanical advantage of 1.

- No of Teeth = 18 teeth
- Module, m_t = 5mm
- Cone Distance, R = 63.639mm
- Outer Pitch Diameter, d = 90mm
- Tip Diameter d_{a1} = 97.071mm
- Face width, b = 20mm
- Pitch angle, δ = 45°

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- Tip angle, $\delta_a = 49.492°$
- Root angle, $\delta_f = 39.614°$
- Addendum, h_a = 5mm
- Dedendum, h_f = 9.4mm
- Addendum angle, $\theta_a = 4.492°$
- Dedendum angle, $\theta_f = 5.386°$
- Height factor = 1
- Clearance = 0.2 mm

III. MODELING

Modeling of Universal Coupling

Modeling of Universal coupling is done by using CATIA V5 software and shown in fig.5 below

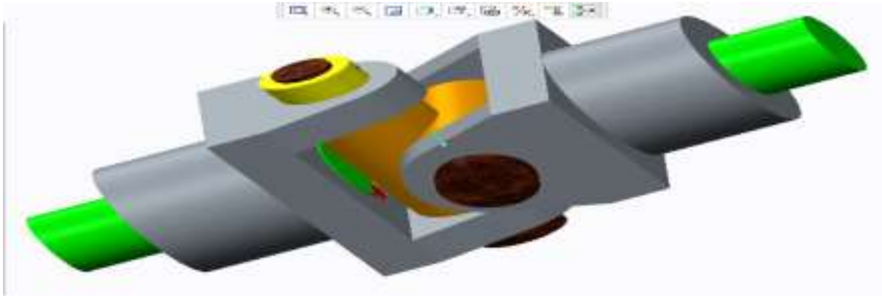

Fig 5 Universal Coupling Assembled diagram

Modeling of Propeller Shaft

Modeling of propeller shaft is done using CATIA V5 software and assembled with Universal coupling. The assembled model is shown in fig. 6 below

Fig 6. CATIA Model of Propeller Shaft

IV. ANALYSIS

Analysis of Propeller Shaft Using ANSYS 12.1

Analysis of the Propeller Shaft is done using ANSYS 12.1 software. First the model is meshed properly and then it is analyzed. The meshed model is shown below in fig. 7

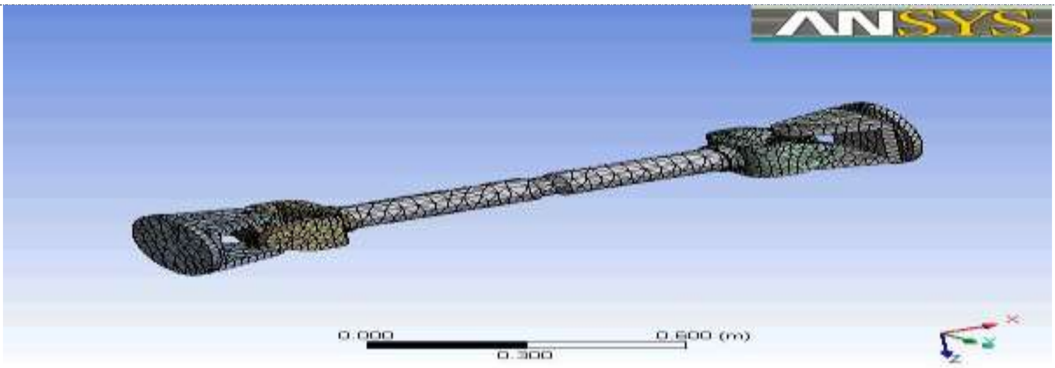

Fig 7. Meshed Model

Next 370 N/m^2 load is applied and total deformation is observed .Maximum and minimum deformation is also determined. The total deformation diagram is given below in fig. 8

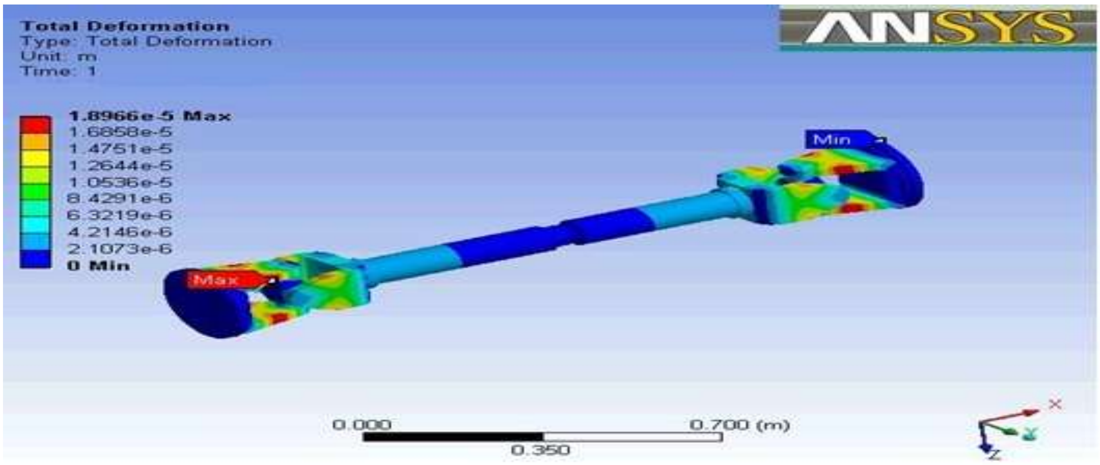

Fig 8. Total deformation in Propeller shaft

Below in fig.9 the equivalent stress model (von-misses) is given, where we can determine the maximum and minimum stress induced for the loading condition.

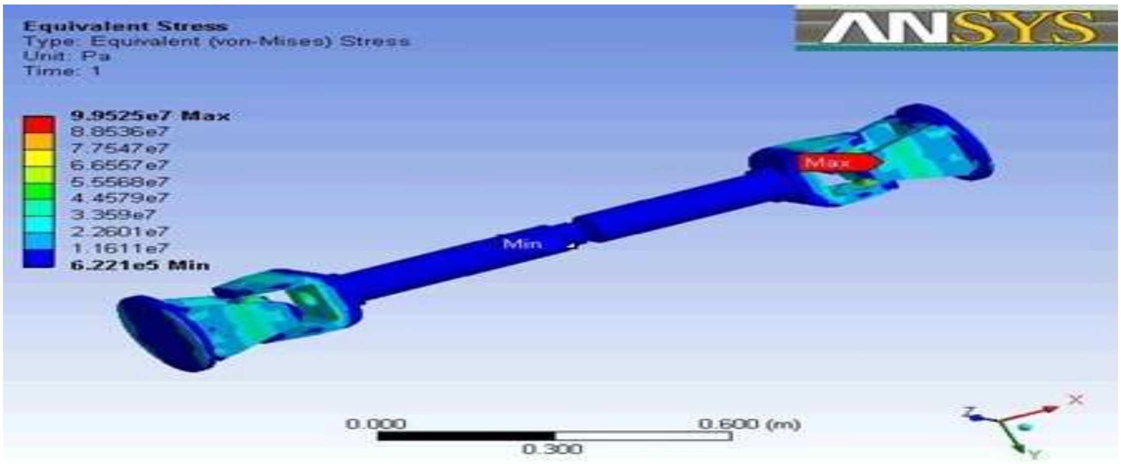

Fig 9. Equivalent Stress model

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V. RESULTS

Results of Single Plate Clutch

- Axial force $F_a = 942.47N$
- Torque transmitted by Clutch $T = 35.52Nm$
- Total Load on the springs is 1178.08 N
- Standard wire of size SWG8 having diameter, $(d) = 3.251mm$
- Compression of the spring, $\delta = \dfrac{8 \times 196.34 \times 6^3 \times 4}{84 \times 10^3 \times 3.251}$

  $\delta = 4.9695mm$
- Free length of the spring, $L_f = n'd + \delta + 0.15\delta = 25.22mm$
- Pitch of the Coils5.044 mm

Results of Six Speed Gear box

- Centre distance b/w output and intermediate shaft is 160 mm
- Centre distance b/w intermediate and input shaft is 280 mm
- Face width, b=80 mm
- Length of shaft, L= 1540 mm
- Diameter of spindle, d=135mm
- According to the spindle diameter, Deep Groove Ball Bearing SKF6026 28 has been selected

Results of Universal Coupling

- Diameter of shaft, D=106 mm
- Diameter of pin, d=53mm

Results of Propeller Shaft

- Polar moment of Inertia, $I = \dfrac{\pi d^4}{32} = \dfrac{\pi \times 50^4}{32} = 6135923mm^4$
- Maximum Torsional Shear Stress, $S_{s\max} = \dfrac{Td}{2I} = \dfrac{3500 \times 10^3 \times 50}{2 \times 6135923} = 142.2Mpa$
- Maximum static deflection of the drive shaft (δ)= 0.02009 mm
- Critical speed of the Shaft (N_c)= 6696 r.m.p

Results of Differential

All the results of Differential unit are given in the table no. 1 below

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Table No.1: Results of Differential Unit

Components	*Pinion Gear*	*Ring Gear*	*Milter Gear*
No. of teeth	16	48	18
Module (mm)	5	5	5
Cone distance(mm)	126.49	126.49	63.639
Outer pitch diameter (mm)	80	240	90
Tip diameter(mm)	89.486	243.13	97.071
Face width (mm)	40	40	20
Pitch angle (degree)	18.44	71.56	45
Tip angle (degree)	20.7	73.823	49.492
Root angle(degree)	15.719	68.845	39.614
Addendum (mm)	5	5	5
Dedendum (mm)	9.4	9.4	9.4
Addendum angle(degree)	2.263	2.263	4.492
Dedendum angle (degree)	2.715	2.715	5.386
Height factor	1	1	1
Clearance	0.2	0.2	0.2

VI. CONCLUSION

The results of this report have illustrated the entire design methodology into the powertrain system assembly. The efforts taken to account for all the necessary design and analysis considerations have provided a solid starting point into the fundamentals of powertrain system design. The opportunity to participate in the global collaboration project has provided tremendous insight into the nature of cooperation essential to the successful completion of a multi-faceted project.
The design procedure includes the following design calculations

- Design of Clutch
- Design of 6 speed gear box
- Design of Universal coupling
- Design of Propeller shaft
- Design of Differential unit

Using CATIA V5 modelling software, the propeller shaft design is done.Analysis of Total Deformation and Equivalent Stress is also done by using ANSYS 12.1

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