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Design and Fatigue analysis of Steering Components

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ABSTRACT

The manual Rack And Pinion are widely used in the steering system due to its obviousness in design and manufacturing. In Rack and pinion the part that experiences maximum fluctuation in load on pinion, pinion shaft and steering intermediate shaft. In this paper the static analysis of pinion and pinion shaft with AISI 4130 is carried out and a comparison of fatigue analysis of pinion & pinion shaft with AISI 4130 & ASTM A36 is carried out. The intermediate shaft is analyzed with AISI 4130, ASTM A36, Al 4032 & Al 201.The primary modeling is carried out in SOLIDEDGE software and the analysis is carried out in CATIA V5. The objective of the study is to optimize the design and increase the life of steering components.

Keywords: Steering system, Pinion, Steering shaft, intermediate shaft, SOLIDEDGE, CATIA V5, Frequency , Stability

1. INTRODUCTION

The rack and pinion mechanism is commonly used to convert the rotary motion into linear motion. This mechanism contains a circular gear and teeth on a linear shaft. The circular gear is called pinion and the teeth on a linear shaft is called a rack. The rack and pinion steering mechanism are simple in construction and friendly to drive. The mechanism consists of a pinion at the end of the steering column that meshes with the rack.

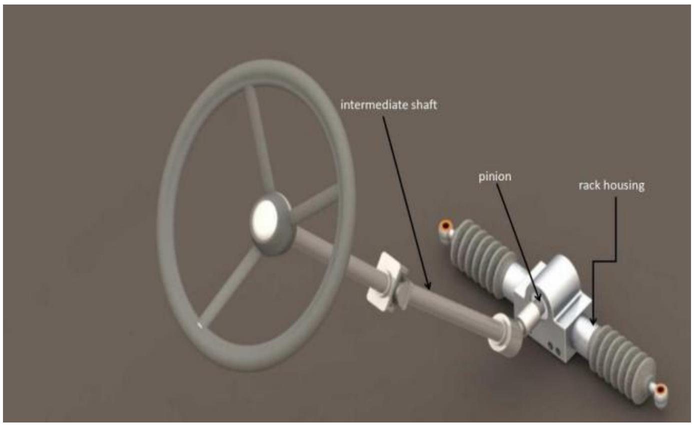

Fig-1: Steering Assembly

The pinion is fixed to the steering column at its end. As the pinion is in contact with the rack, the rotary motion given to pinion is converted to linear motion by the rack. To meet all the steering requirements the rack and pinion steering must be precise and direct under normal driving conditions. A manual rack and pinion gear suitable for a solar car. It is found that the simulation results are higher than the theoretical calculations. The error factor for stress and deformation is between 4% and 5% [1]. A sensitive rack and pinion system with zero play in pinion and rack gearbox. This study help in designing a pinion and rack assembly with an optimum performance of the steering system[2]. Optimized the intermediate steering shaft and have analyzed the fatigue characteristics under different frequencies. This study compared the existing intermediate shaft and optimized intermediate shaft[3]. The analysis results may differ from practical conditions due to steering linkages or steering arrangements [4]. The strength of gear tooth is a critical parameter to prevent failure and bending stress decreases with increasing face width. So optimum face width is calculated [5].

2. METHODOLOGY

- ✓ The literature reviews are studied completely and identified the problem statement.
- ✓ The design parameters are taken from references and design calculations are done.
- ✓ With the design calculations CAD model is designed using SOLIDEDGE software.
- ✓ The model is imported to CATIA V5 ans static analysis is carried out with a mesh size of 1mm.
- ✓ With reference to static analysis , fatigue analysis is conducted.

3. NOMECULATURE

F	-	Force Applied
N_p	-	Pinion Speed
Ψ	-	Helix Angle
Φ	-	Pressure Angle
n_p	-	Number of teeth on pinion
m_n	-	Module
h_a	-	Addendum
h_d	-	Dedendum
D_p	-	Pitch diameter
P_d	-	Diametric pitch
t	-	Tooth Thickness
h_t	-	Whole Depth
C	-	Clearance
D_0	-	Outer diameter
D_r	-	Root diameter
H	-	Height of Pitch line
M_t	-	Moment
V	-	Pitch Line Velocity
S_{all}	-	Allowable Stress
S_0	-	Endurance Stress
S_{ind}	-	Actual Induced Stress
b	-	Face Width

F_t - Transmitted Load

F_d - Dynamic Load

F_0 - Limiting Endurance Load

F_w - Limiting Wear Load

S_u - Ultimate Strength

S_y - Yield Strength

4. MATERIAL SPECIFICATIONS

Properties	AISI 4130	ASTM A36	Al 4032 (T_6)	Al 201 (T_6)
Density(Kg/m^3)	7850	7850	2680	2800
Modulus of elasticity (GPa)	205	200	82	71
Tensile Strength (MPa)	670	400	370	485
Yield Strength (MPa)	435	250	317	435
Poisions ratio	0.29	0.3	0.33	0.31

Table-1: Material Properties

5. PARAMETERS CONSIDERED FOR DESIGNING OF STEERING SYSTEM

To design and analysis certain parameters are required.

Design Parameter	Specifications
F	2610N
N_p	8-15 rpm
Ψ	23^0
Φ	20^0
n_p	7
m_n	2

Table-2: Design Parameters

6. CALUCULATIONS

The calculations are made with the calculations from [6] & [7] as follows.

1. Calculation of pitch diameter,

$$D_p = n_p \times m_n$$

2. Calculation of torque

$$M_t = F_t \times D_p$$

3. The pitch line velocity can be calculated as

$$V = \frac{\prod \times N_p \times D_p}{60}$$

$$n_f = \frac{n_p}{\cos^3 \psi}$$

4. Allowable stress can be calculated as

$$S_{all} = S_0 \times \left(\frac{3}{3 + V} \right)$$

5. Calculation of endurance stress

$$S_0 = \frac{S_u}{3}$$

6. The calculation of actual induced stress can be done by using lewis equation as

$$S_{ind} = \frac{2\,M_t}{m^3 K \prod^2 y_p n_p \cos \psi}$$

7. With the above strength the strength is checked by using S_{all} and S_{ind}

For a good design $S_{all} > S_{ind}$

If not the module should be increased untill the condition

Satisfies.

8. The face width can be calculated as

$$b_{\min} = k_{red} \times \prod \times m_n$$

$$b_{\max} = k \times \prod \times m_n$$

$$k_{red} = k_{\max} \times \frac{S_{ind}}{S_{all}}$$

Thus the design is checked from a strength point of view. Before going to analysis it is necessary to check in Dynamics' point of view. To undergo Dymanics calculation the following equations are used:

1. The load transmitted can be given by

$$F_t = \frac{2M_t}{D_p}$$

2. Calculation of Dynamic load is done by using the following equation

$$F_d = F_t + \frac{21V(bC\cos^2 \psi + F_t)\cos \psi}{21V + \sqrt{(bC\cos^2 \psi + F_t)}}$$

3. Calculation of limiting endurance load is given by

$$F_0 = S_0 b y_p \cos \psi$$

4. limiting wear load can be calculated as

$$F_w = \frac{D_p \times b \times K \times Q}{\cos^2 \psi}$$

$$K = \frac{S_{es}^2 \times \sin \phi_n}{1.4} \times \left[\frac{2}{E}\right]$$

The Dynamic check should satisfy the condition as F0, $F_w > F_d$

If not keep on increasing the module and calculate until the condition to be satisfied. After the gear tooth dimensions are calculated.

In order to determine dimensions the following are used:

1. Addendum

$$h_a = 0.8 m_n$$

2. Pitch Diameter

$$D_p = n_p \times m_n$$

3. Diametral pitch

$$P_d = \frac{n_p}{D_p}$$

4. Tooth thickness

$$t = \frac{1.5708}{P_d}$$

5. Whole depth

$$h_t = 1.8 m_n$$

6. Clearance

$$C = 0.2 m_n$$

7. Outer diameter

$$D_0 = D_p + 2 h_a$$

8. Dedendum

$$h_d = 1 m_n$$

9. Root Diameter

$$D_R = D_p - 2 h_d$$

In general the intermediate shafts are with circular cross-section. So ,

1) Maximum power used is given by

$$P = \frac{2\pi T N_{\max}}{60}$$

2) Maximum torque acts on the intermediate shaft

$$T_{\max} = \frac{60P}{2\pi N_{\min}}$$

3) Minimum torque acts on intermediate shaft

$$T_{\min} = \frac{60P}{2\pi N_{\max}}$$

4) Mean Torque acting can be calculated as

$$T_a = \frac{T_{\max} - T_{\min}}{2}$$

5) Maximum shear stress is given by

$$\tau_{\max} = \frac{S_u}{FS}$$

The FS= Factor of safety = 15 (consider)

6) The inner diameter can be calculated as

$$\tau_{max} = \frac{16T_{max}}{\pi(D_0^3 - D_i^3)}$$

7) The mean shear stress developed is given by

$$\tau_m = \frac{16T_m}{\pi(D_0^3 - D_i^3)}$$

8) The alternating stress can be calculated as

$$\tau_a = \frac{16T_a}{\pi(D_0^3 - D_i^3)}$$

Dimension	Result
D_p	14mm
M_t	37.8N-m
V	0.0109955m/s
b	37.6mm
h_a	1.6mm
P_d	0.5 mm^{-1}
t	3.1416mm
h_t	3.6mm
C	0.4mm
D_0	17.2mm
h_d	2mm
D_R	10mm

Table-2: result data of tooth dimensions

Parameter	AISI 4130	ASTM A36	Al 4032 (T_6)	Al 201 (T_6)
τ_{max}	44.67 MPa	26.67 MPa	25.26 MPa	32.33 MPa
D_i	18 mm	16mm	16.63 mm	17.2 mm
τ_m	42.79 Mpa	15.879 MPa	19.415 MPa	24.78 MPa
τ_a	13.017 MPa	4.838 MPa	5.915 MPa	7.55 MPa

Table-3: calculation results of intermediate shaft

8. STATIC ANALYSIS OF PINION

Static analysis is a process of applying static load on the designed CAD model. The pinion is designed using the calculation results in SOLIDEDGE software. Then it is applied a force of 2610N on teeth.

I. THE DEFORMATION, STRESS(VON-MISES), AND FACTOR OF SAFETY ARE CALCULATED.

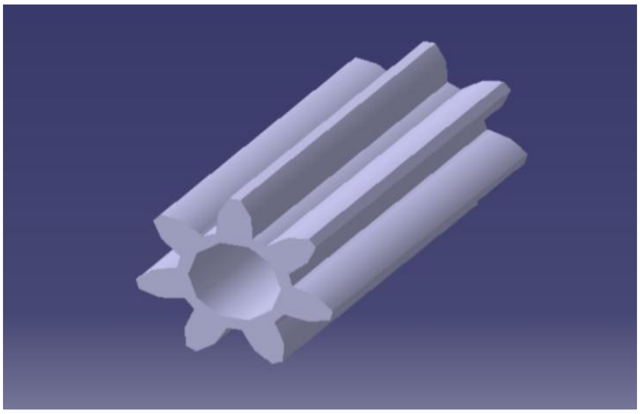

Fig-2: CAD model of pinion

The designed CAD model is applied with AISI 4130 material . and the analysis parameters are :

Mesh size	1mm
No. Of nodes	51552
No. Of elements	30296
Mesh type	tetrahedron

Table-4: analysis parameters

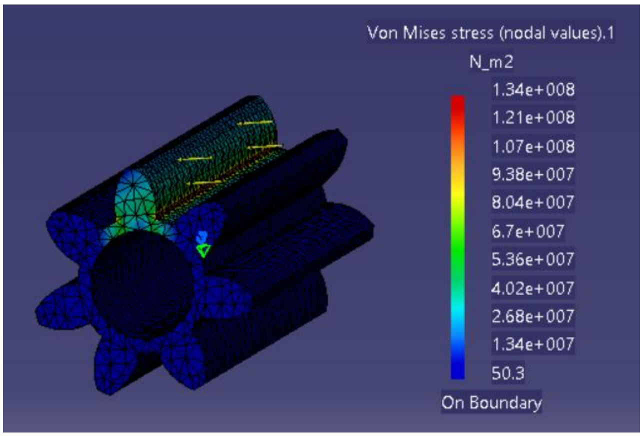

Fig-3: Von Mises stress in pinion

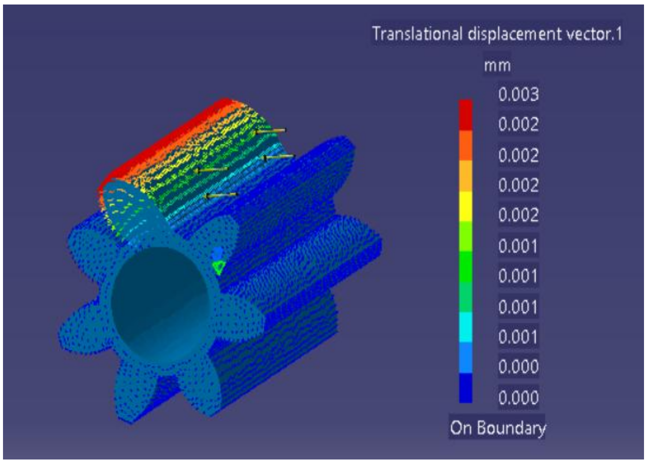

Fig-4: Displacement in pinion

Max. Stress = $1.34e+008 N/m^2$
Max. Displacement = 0.003mm
Factor of Safety = 5

9. STATIC ANALYSIS OF PINION SHAFT

THE STEERING SHAFT IS THE CONNECTION BETWEEN THE PINION AND INTERMEDIATE SHAFT. THE DESIGNED CAD MODEL FROM THE CALCULATION RESULTS IS APPLIED WITH AISI 4130 MATERIAL AND A STATIC STUDY IS CONDUCTED.
The shaft of 8mm diameter and 150mm is designed as

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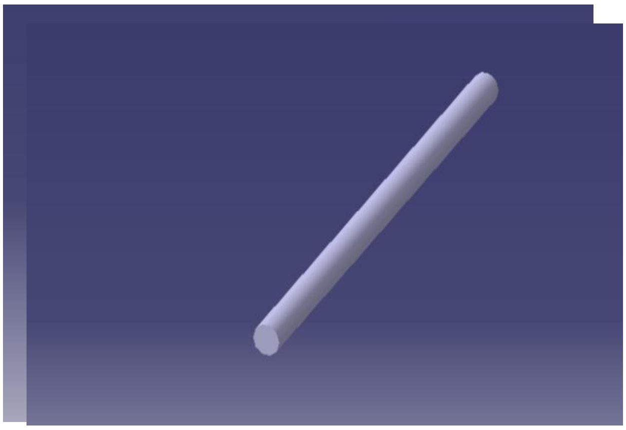

Fig-5: CAD model of pinion shaft

The analysis parameter:

Fig-6: Von Mises stress in pinion shaft

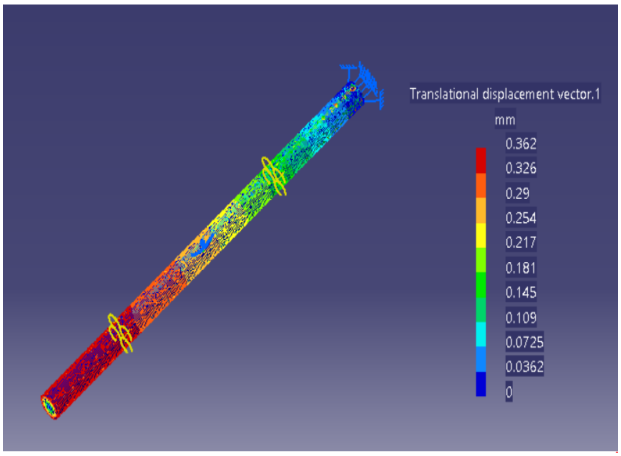

Fig-7: Displacement in pinion shaft

Max. Stress = 6.59e+008 N/m^2
Max. Displacement = 0.362 mm
Factor of safety = 1.1

The static analysis of pinion and steering shaft is carried out. Now the study further proceeds with the comparative of frequency analysis of pinion with AISI 4130 and ASTM A36, also frequency analysis of steering shaft with the same materials.

10. FATIGUE ANALYSIS OF PINION AND PINION SHAFT:

THE FATIGUE ANALYSIS OF A COMPONENT GIVES THE STABILITY OF A COMPONENT CONCERNING CYCLIC LOADS. HERE THE PINION AND PINION SHAFT ARE APPLIED WITH FREQUENCY AND THE RESULTS OF TWO MATERIALS ARE AS FOLLOWS:

Material	Steering Pinion	Steering Shaft
AISI 4130	1.23E+005 Hz	2.54E+002 Hz
ASTM A36	7.89E+004 Hz	2.55E+002 Hz

Table-5: Fatigue results

1. Calculation of AISI 4130 pinion

A) The CPM is further converted to CPD as follows

CPD = CPM×60×24
= 7380000×60×24
= 1.06272×10^{10}

B) Life calculation in days

Generally life is calculated for 10^6 cycles.So,

Life in days = 10^6/CPD
= 9.4098×10^{-5}

C) Let us consider the average speed of vehicle is 50Km/h
Life in Km = $9.4098\times10^{-5}\times50\times24$
= 0.112917795 Km

2. Calculation of ASTM A36 Pinion
A) The CPM is further converted to CPD as follows
CPD = CPM×60×24
= 4734000×60×24
= 6.8169×10^9

B) Life calculation in days
Generally life is calculated for 10^6 cycles.So,
Life in days = 10^6/CPD
= 1.46692954×10^{-4}

C) Let us consider the average speed of vehicle is 50Km/h
Life in Km = $1.466929\times10^{-4}\times50\times24$
= 0.17603 Km

3. Calculation of AISI 4130 steering shaft
A) The CPM is further converted to CPD as follows
CPD = CPM×60×24
= 15240×60×24
= 21945600

B) Life calculation in days
Generally life is calculated for 10^6 cycles.So,
Life in days = 10^6/CPD
= 0.04556722

C) Let us consider the average speed of vehicle is 50Km/h
Life in Km = 0.04556722×50×24
= 54.680 Km

4. Calculation of ASTM A36 steering shaft
A) The CPM is further converted to CPD as follows
CPD = CPM×60×24
= 15300×60×24
= 22032000

B) Life calculation in days
Generally life is calculated for 10^6 cycles.So,
Life in days = 10^6/CPD
= 0.045388

C) Let us consider the average speed of vehicle is 50Km/h
Life in Km = 0.30127×50×24
= 54.466 Km

11. ANALYSIS OF INTERMEDIATE SHAFT

The static analysis is conducted to know the stress induced and deformation. The moment of 6N-m is applied at one end and other end is clamped at bolted position. The tetrahedral mesh of size 1mm is used to evaluate results of all the CAD models designed. The intermediate shaft is analyzed with AISI 4130, ASTM A36, Al 4032 and Al 201.

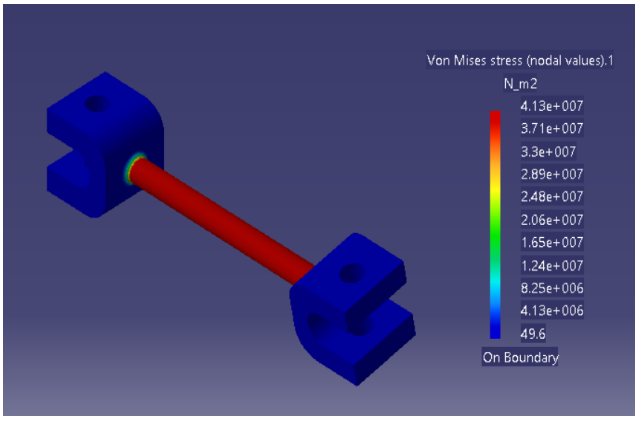

Fig-8: Stress induced in Intermediate shaft made of AISI 4130

Mass	=	2.445 Kg
Max. Stress	=	4.13e+007 N/m^2
Max. Deformation	=	0.258 mm
Factor of Safety	=	16.22

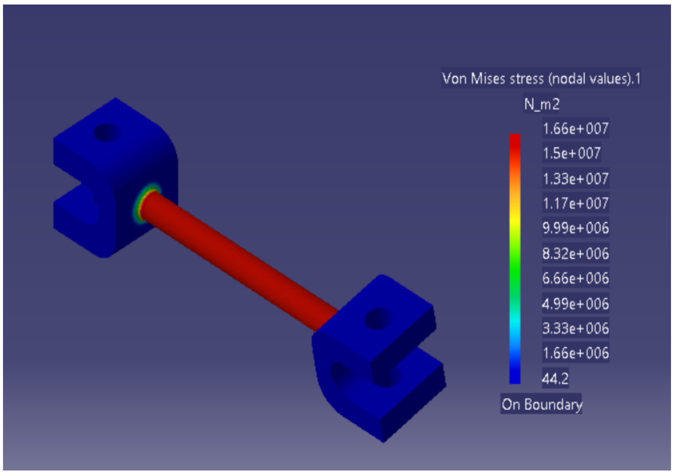

Fig-8: Stress induced in Intermediate shaft made of ASTM A36

Mass	=	2.526 Kg
Max. Stress	=	1.66e+007 N/m^2
Max. Deformation	=	0.102 mm
Factor of Safety	=	24.10

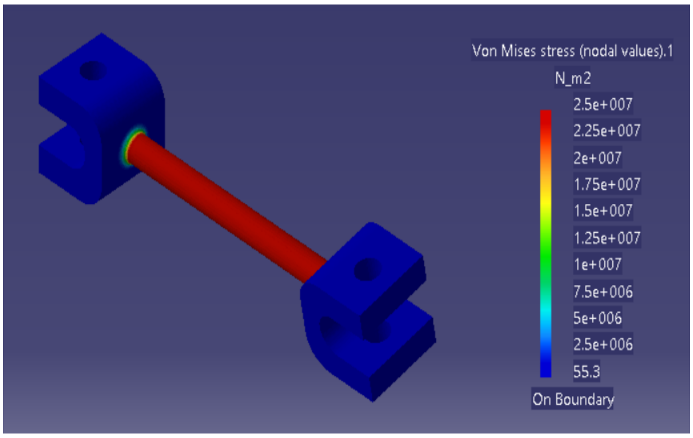

Fig-9: Stress induced in Intermediate shaft made of Al 4032

Mass = 0.854 Kg
Max. Stress = 2e+007 N/m^2
Max. Deformation = 0.317 mm
Factor of Safety = 18.5

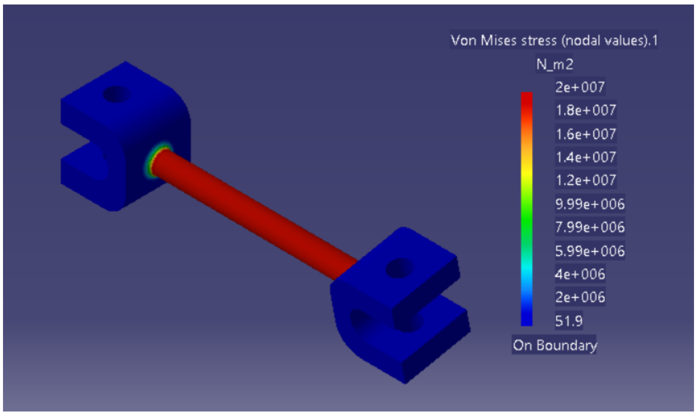

Fig-10: Stress induced in Intermediate shaft made of Al 201

Mass = 0.884 Kg
Max. Stress = 2.5e+007 N/m^2
Max. Deformation = 0.452 mm
Factor of Safety = 19.4

12. FATIGUE ANALYSIS OF STEERING INTERMEDIATE SHAFT

With reference to the static case solution the life of intermediate shaft is determined. The result in the form of Cycles per Second (CPS) is converted to Cycles Per Minute (CPM) and the final result is calculated in kilometers.

Material	AISI 4130	ASTM A36	Al 4032	Al 201
Life in CPS	38.417	60.749	59.909	48.804
Life in CPD	2305.02	3644.94	3594.54	2928.24

Table-6: Fatigue Results

1. The life calculation of intermediate shaft with AISI 4130 is as follows:

A) The CPM is further converted to CPD as follows

CPD = CPM×60×24
= 2305.02×60×24
= 3319228.8

B) Life calculation in days

Generally life is calculated for 10^6 cycles.So,

Life in days = 10^6/CPD
= 0.30127

C) Let us consider the average speed of vehicle is 50Km/h

Life in Km = 0.30127×50×24
= 361.524 Km

2. The life calculation of intermediate shaft with ASTM A36 is as follows:

A) The CPM is further converted to CPD as follows

CPD = CPM×60×24
= 3644.94×60×24
= 5248713.6

B) Life calculation in days

Generally life is calculated for 10^6 cycles.So,

Life in days = 10^6/CPD

= 0.19052

C) Let us consider the average speed of vehicle is 50Km/h

Life in Km = 0.19052×50×24

= 228.62 Km

3. The life calculation of intermediate shaft with Al 4032 is as follows:

A) The CPM is further converted to CPD as follows

CPD = CPM×60×24

= 3594.54×60×24

= 5176137.6

B) Life calculation in days

Generally life is calculated for 10^6 cycles.So,

Life in days = 10^6/CPD

= 0.1932

C) Let us consider the average speed of vehicle is 50Km/h

Life in Km = 0.1932×50×24

= 231.83 Km

4. The life calculation of intermediate shaft with Al 201 is as follows:

A)The CPM is further converted to CPD as follows

CPD = CPM×60×24

= 2928.24×60×24

= 4216665.6

B) Life calculation in days

Generally life is calculated for 10^6 cycles.So,

Life in days = 10^6/CPD

= 0.2371

C) Let us consider the average speed of vehicle is 50Km/h

Life in Km = 0.30127×50×24

= 284.58

13. RESULT AND DISCUSSION

The Static analysis has shown that AISI 4130 has shown the deformation of 0.003mm and stress of 1.34e+008N/m2. the fatigue analysis of pinion with AISI 4130 can withstand continuous fatigue loads up to 0.112917795 Km and pinion with ASTM A36 can withstand continuous fatigue loads up to 0.17603 Km. The fatigue analysis of pinion shaft with AISI 4130 can withstand continuous cyclic loads up to 54.680 Km and pinion shaft with ASTM A36 can withstand cyclic loads up to 54.466 Km. From the results, it is preferable to use ASTM A36 for pinion and AISI 4130 for pinion shaft.On considering the static and fatigue analysis, the intermediate shaft made of AISI 4130 has shown the maximum life among the considerations. The life of Intermediate shaft made of AISI 4130, ASTM A36, Al 4032 and Al 201 are 361.524Km, 228.62 Km, 231.83 Km and 284.58 Km respectively. When the objective is to increase life AISI 4130 is selected. But when the objective is to reduce weight with optimum life Al 201 is selected.

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