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DESIGN AND ANALYSIS OF 3- STAGE EPICYCLIC GEAR BOX FOR AIRCRAFT APPLICATION

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ABSTRACT :-Planetary Gear System is an assembly of gears where one or more planet gears revolve around a gear put at focus named Sun gear and a ring gear circles the entire framework. It is fit for accomplishing high decrease proportions in an extremely minimal bundle. Huge number of assembling mistakes can impact the anxieties actuated onto the stuff framework. In this paper, a nitty gritty investigation of the plan of a planetary gearbox is done from the start. In the wake of planning, this exploration uses the limited component technique to research the variety of stresses happening because of misalignments. Here, the anxieties analyzed are occurring due to misalignments resulting from the tolerances on center distance between gears and also from the tolerances on parts. In this paper, we have reviewed the research papers mainly dealing with the designing and analysis of a planetary gearbox.

***Key ward*:** Planetary Gearbox, Stress Analysis, FEM Analysis, Misalignments, Tolerances.

I. INTRODUCTION

PLANTRY gears are used to transmit power in a wide range of industrial applications. Planetary gear system had three body parts such as, the carrier, the ring gear, and the sun gear. The planet gears are associated by direction to the transporter and are all the while in network with the sun stuff and ring gear. The quantity of planet gears shifts relying upon the plan heap of the framework. Various planets offer freedom to have different stuff proportions.

All focal individuals kinematically pivot about a similar hub, albeit one focal part is ordinarily fixed. The planetary cog wheels pivot about fixed tomahawks of the transporter, which may likewise turn. In a star setup, the transporter doesn't turn, and all pinion wheels, including the planets, pivot about fixed tomahawks. From examined made unmistakably, planetary pinion wheels actually experience clamor and vibration issues. The vibrations created in cars prompts commotion and consequently further prompts proportion of vehicle quality organization. Planetary cog wheels are the primary commotion source in helicopter lodges. Estimated sound levels can exceed 100 dB. This causes problems with communication and creates a noise hazard for the pilot and passengers. Noise is also an issue in wind turbines, which contain one or more

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planetary gears, when they are located near populated areas. Life of planetary is mainly affected by dynamic loading and bearing loads. In aircraft engines, vibration can cause structural failure.

Several of these analysis tools are based on multimode dynamics codes and can model complete transmission systems, not just planetary gears. They can incorporate finite element models of the housing and carrier built using conventional finite element software. These methods historically have had only basic tooth mesh modeling, although more recently the focus has been on better mesh modeling. Multibody dynamics methods can provide quick answers that can be used to assess different designs, but they are less effective for root because failure analyses, because of simplifications of the gear mesh. There are also finite element/contact mechanics tools, for example, which can be used for high fidelity contact modeling and analysis.

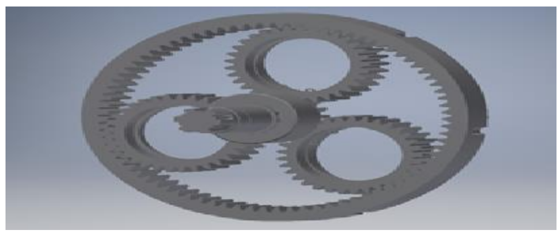

Fig -1.1 Epicyclic Gear Train

II. LITERATURE SURVEY

Ming Li, et.al (2017)[2]studied about Load sharing analysis and reliability prediction for planetary gear train of helicopter in 2017. There are many problems of unequal load sharing in planetary gear train. A model is established to predict the reliability of helicopter planetary gear train under the condition of partial load. The structure of gear system and partial load state are analyzed in detail, and the load histories of each gear are transformed into equivalent constant amplitude load spectrums to be as the load input variable for reliability model. At the same time, a tooth bending fatigue test is carried out with specific gears, and the statistical results of test data are as the strength input variable for reliability model

Yanzhong Wang, et.al (2017)[6]had carried out Investigation into the meshing friction heat generation and transient thermal characteristics of spiral bevel gears in 2017.Friction loss and scuffing failure are two primary research subjects in improving the performance of spiral bevel gears. Aimed at improving the thermal characteristics with machine-setting parameter adjustment, a coupled thermo-elastic 3D finite element model has been developed to analyze the frictional heat generation and transient thermal behavior of spiral bevel gears.

Ajay Narayankutty (2016) [11], carried out a review of design and analysis of a 3-stage planetary gearbox in 2016. Planetary Gear System is an assembly of gears where one or more planet gears revolve around a gear placed at centre named Sun gear and a ring gear encircles the whole system. It is capable of achieving high reduction ratios in a very compact package.

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Prabhakar Vitthal Pawar, P.R. Kulkarni (2015) [12], studied about Design of Two Stage Planetary Gear train for high reduction ratio in 2015. Planetary Gear Trains are extensively used for the power transmission and are the most critical component. Planetary gearboxes are used frequently to match the inertias, lower the motor speed, boost the torque, and at the same time provide a sturdy mechanical interface for pulleys, cams, drums and other mechanical components

III. OBJECTIVE

The main objectives of this paper is –

1. To enhance gear parameters.
2. To maximize service life of gear box.
3. Minimizing power losses by reduction in vibrations produced in the system.
4. To reduce wear of system
5. Optimize the 3 stage epicyclic gear box.
6. Tooth profile modification to avoid losses

IV. SPECIFICATIONS OF GEAR SYSTEMS

The specification of drawing of Epicyclic gear train used for aircraft application.

Table 1 Drawing Specification

Sr No	Description	Remarks
1	Reduction Ratio	3.6 to 100:1
2	Output Torque	10 Nm to 1,450 Nm
3	Input Power	0.18 Kw to 224 Kw
4	Mounting Type	Flange/ Foot
5	Prime Mover	Electric Motor / Engine
6	Input Type	Male Shaft

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7	Output Type	Male Shaft
8	Planet Dia	86 mm
9	Sun Dia	42 mm
10	Ring Dia	228 mm
11	Number of Plant Teeth	33
12	Number of Sun Teeth	15
13	Number of Ring Teeth	82
14	Face width	34

A. Material

For modeling of Epicyclic gears various materials are studied but best suitable material for heavy duty application and , operating conditions of higher mode of vibration, is case hardened steel named as AISI 8620. The selected material generally available as rolled to HB 255 max. Carburising and heat treatement develops a hard wear resistant case to HRC 60-63 and a tough strong core with a typical tensile strength range of 700-1100 MPa, in small to medium sized sections. Properties of AISI 8620 are as follows

Table 2 Chemical Properties of Selected Material

Sr. No.	Content	Percentage
01	Carbon	0.20%
02	Silicon	0.25%
03	Manganese	0.80%
04	Chromium	0.50%
05	Nickel	0.55%
06	Molybdenum	0.20%

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V. MODELING SIMULATION OF FEA

A. STATIC SIMULATION

1) STATIC STRUCTURE ANALYSIS

Structural analysis is critical because it can determine cause and predict failure, evaluating whether or not a specific structural design will be able to withstand the external and internal stresses and forces expected for the design. As mentioned above, Finite Element Analysis is to be performed for given design of Planetary Gear Box under design loading conditions.

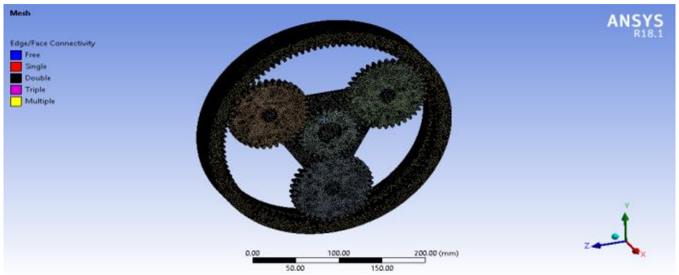

Fig 5.1. Meshing of Geometry

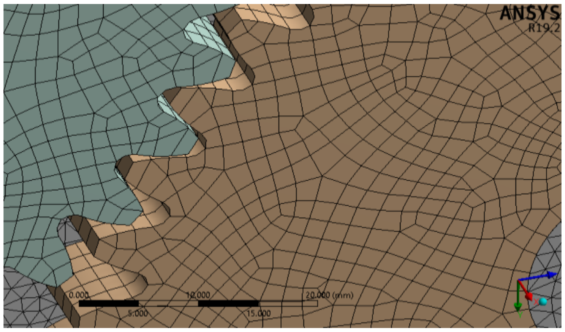

Fig.5.2 Geometry Meshing Properties

1. Ring Gear is fix constraint with all Six Degree of Freedom.
2. Sun Gear is fixed constraint with five degree of freedom except the rotational in X direction DOF.
3. Planetary Gear is fixed constraint with X linear direction which is normal to the rotational Plane.
4. Moment / Torque 10.2N.m apply on the Sun Gear.
5. All the contact assumes bonded for max stress concentration conditions.

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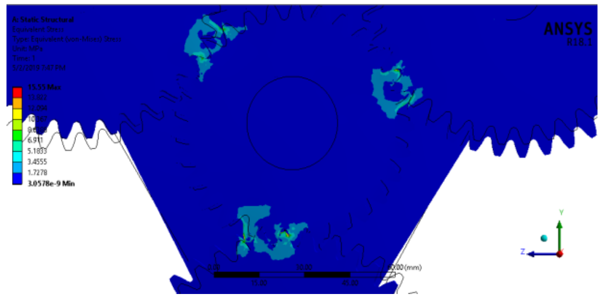

Fig 5.3 Von-misses Stress on Assembly

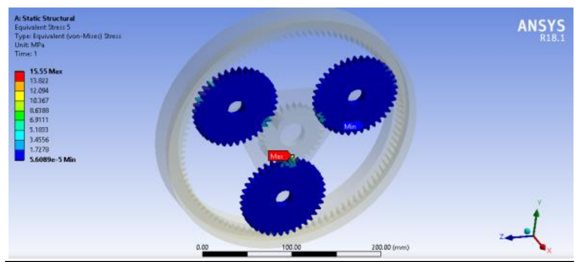

Fig.5.4 Von-Misses stress on Assembly Component – PLANETRY GEAR

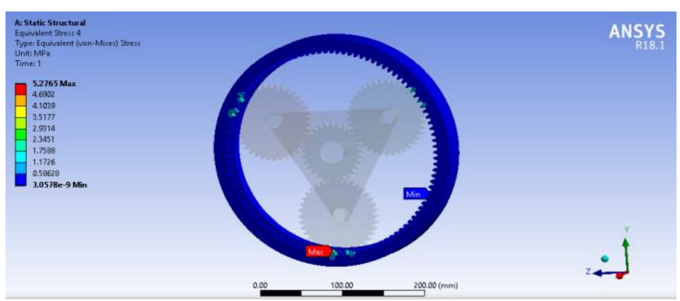

Fig.5.5 Von-Misses stress on Assembly Component – RING GEAR

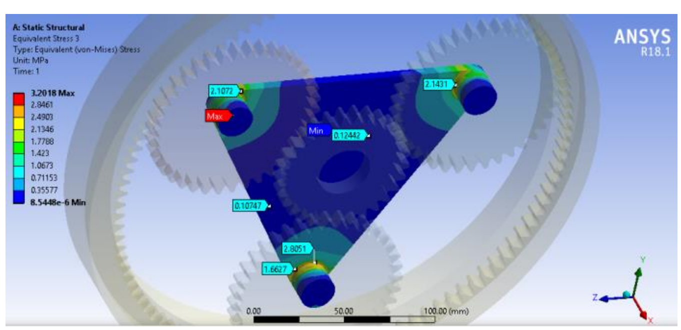

Fig.5.6 Von-Misses stress on Assembly Component – LINK Back

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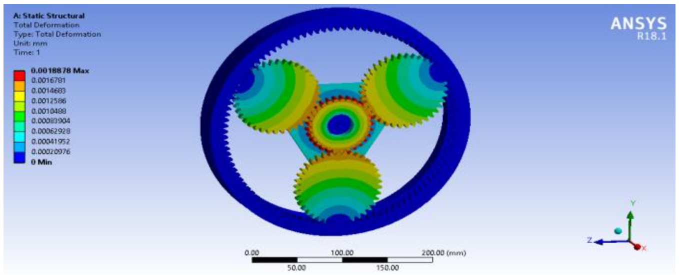


Fig. 5.7 Deformation on Assembly Conveyor

2) Factor of Safety on Assembly Conveyor

As shown in fig 6.7 the FOS is 15 which is max factor of safety for the system because of the min stress generated in the system.

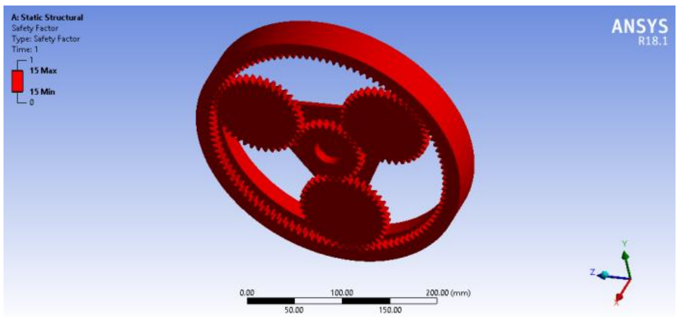
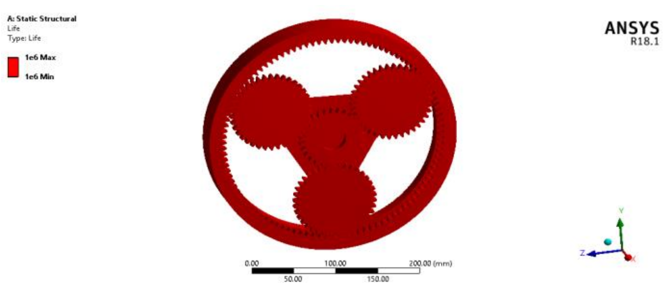


Fig. 5.8 Factor of Safety on Assembly

3) Fatigue Life Assembly



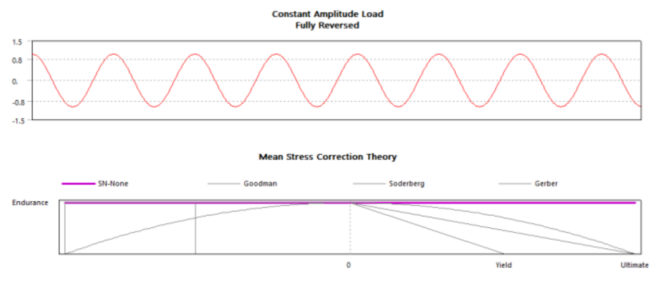


Fig. 5.9 Fatigue Life Assembly

As per above availability cycle vs loading history the curve we get from FEA simulation which is constant at 1e+6 which is max life cycle (Infinity cycle).

So the system works with infinity cycle without the system failure.

4) HARMONIC RESPONSE

Max acceleration we got at frequency 450Hz frequency it means the system vibrate maximum when the system reach at the 450Hz frequency in harmonic load but the vibration is to small so it’s not effect in the system stability

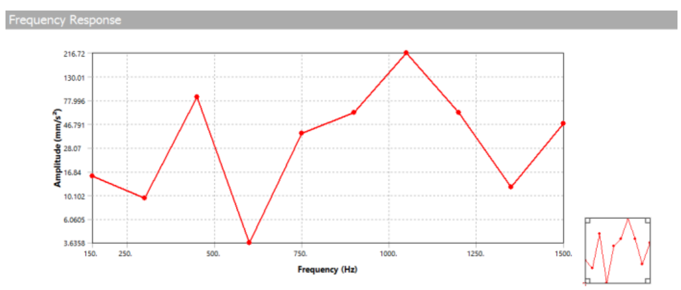


Fig.5.8 Amplitude acceleration vs frequency

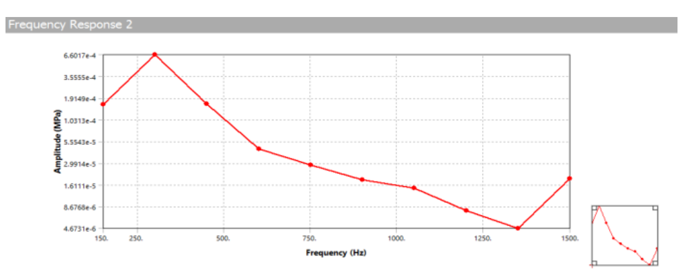


Fig 5.10 Amplitude vs frequency

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Max stress value we got at frequency 300 Hz frequency it means the system vibrate maximum when the system reach at the 300 Hz frequency in harmonic load but the stress generate is too small so it's not effect in the system stability and system is safe under the harmonic load.

Max von-misses stress we got at frequency 300 Hz frequency

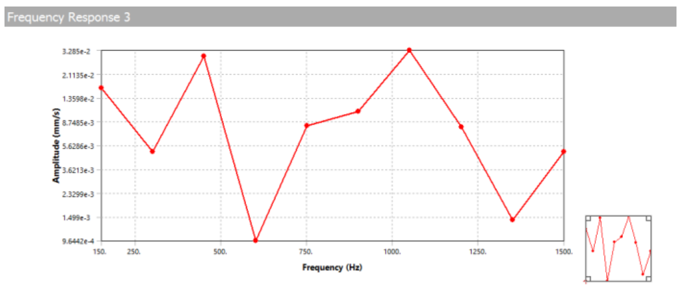

Fig. 5.11 Amplitude velocity vs frequency

Fracture and Delimitation Pitting Effect on gear teeth

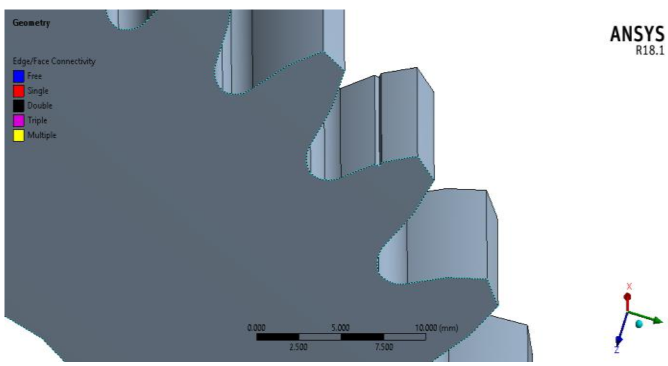

Fig.5.12 A Fracture Sun Gear Modeling (Initial Crack)

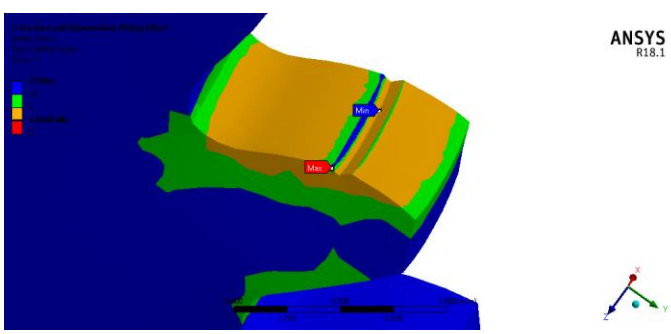

Fig. 5.13 Fracture crack growth Factor of safety due to pitting

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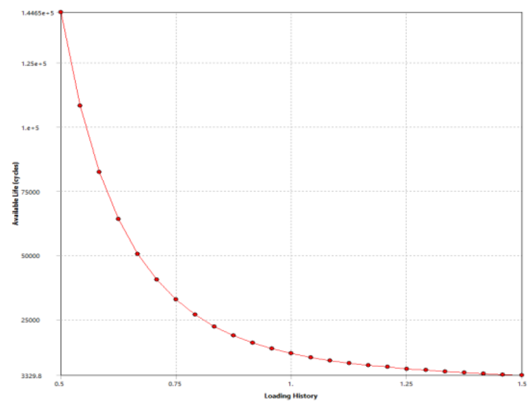
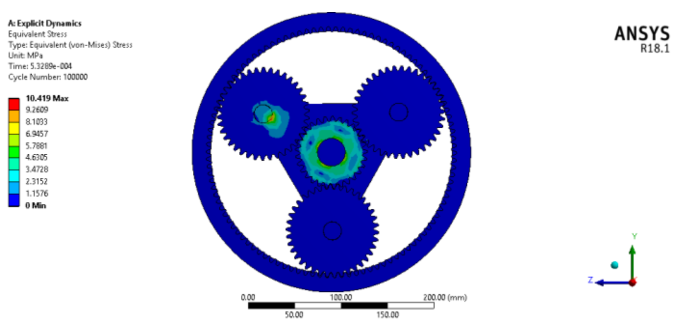
Graph 1E Fracture crack growth fatigue Factor due to pitting



5) EXPLICIT DYNAMIS SIMULATION ON GEARBOX



|Fig. 5.14 Von-Misses stress for the assembly

Figure 5.1 to 5.5 shows generated stresses on component under maximum allowable stress, for static simulation which is 15.55MPa which is within maximum allowable stress limit forgiven material (SS Yield strength at 25°C is 250Mpa) so design is safe to construct.

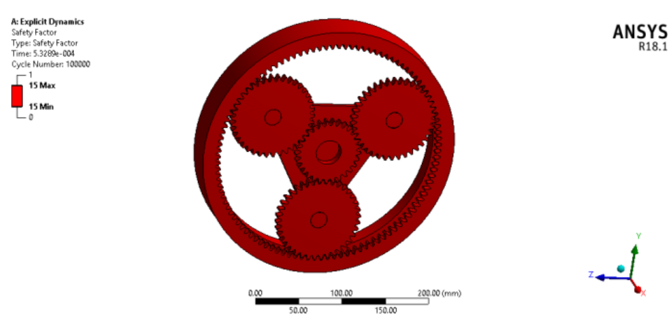


Fig.5.15 Factor of safety

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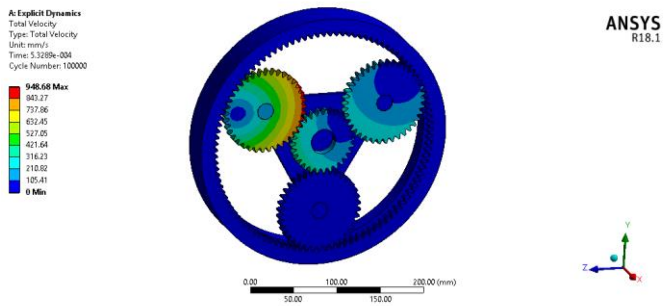
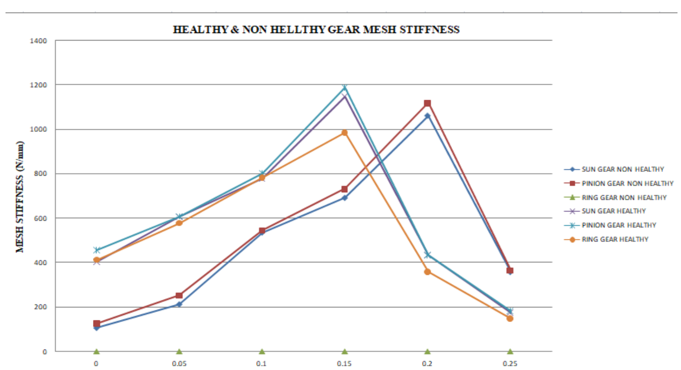


Fig .5.16 Total Velocity

For above graph 5.14 there is a comparison for all the Healthy and Non healthy gear Mesh stiffness with the small rotation of 0.25 degree of Rotation. Conclusion from the graph there is a always the mesh stiffness for the healthy gear is more as compare to non-healthy gear.

Graph 2 Mesh Stiffness Vs Rotation graph for Non Healthy & Healthy Sun, Planetary, Ring Gear-



NUMERICAL SOLUTION-

Number of Teeth Sun Gear Z1 : 29
Number of Teeth Planetary Gear Z2 : 38
Number of Teeth Ring Gear Z3 : 107

A) Determination of Reduction Ratio for each Stage-

$$\text{Ratio} = \frac{\text{PCD of Ring Gear} + \text{PCD of Sun Gear}}{\text{PCD of Sun Gear}} \quad \ldots\ldots (2)$$

$$\text{Ratio} = \frac{z1 + z3}{z1} \qquad (3)$$

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PCD of Ring Gear: 72 mm
PCD of Sun Gear: 100 mm
PCD of Ring Gear: 264 mm

B) Planetary Gear Design

To find the center distance, pcd and width of gears

a. Determine Z_2 so that $i = Z_2 / Z_1$
b. Determine the Centre Distance a‟.
c. If a‟ >a (minimum C.D. already calculated), take a‟ as the final center distance (Don't change to Standard value)
d. If a‟<a, increase Z1 & Z2 (or) module and again calculate a‟ so that a‟>a.
e. Calculate d1& d2, the pitch diameters of pinion and wheel respectively (P 8.22, Table26)
f. Calculate b (width of the gear wheel) using the values of ψ and ψ_m and take the bigger value.

MATERIAL PROPERTIES

Aluminum As A Material For Gears	
Density (Kg/M3)	7850
Young Modulus (Gpa)	200
Poisson Ration	0.33
Yield Strength (Mpa)	250
Ultimate Strength (Mpa)	460

A) Checking The Design

$$\sigma_c = 0.74 \frac{i+1}{a} \sqrt{\frac{i+1}{ib} E [M_t]} \quad \leq [\sigma_c]$$

Compressive strength = 12.36 Mpa which is less than the Gear strength. The design is safe against contact compressive stress.

B) Checking For Plastic Deformation And Brittle Crushing

a) Under surface contact stress

$$\sigma_{c\max} = \sigma_c \sqrt{\frac{[M_t]_{\max}}{M_t}} \quad \leq [\sigma_c]_{\max}$$

For pinion and Sun gear $[\sigma_c]_{max} = 3.1\ \sigma_y$ (for steel) (<350BHN)

= 3.1 *150

= 465 Mpa.

b) Under bending

$$\sigma_{b\max} = \sigma_b \frac{(M_t)_{\max}}{M_t} \qquad \leq [\sigma_b]_{\max}$$

For Pinion and sun gear $[\sigma_b]_{max}$ = 0.8 σ_y (for steels, < 380 BHN)

= 0.8 X 250

= 200Mpa

$\therefore$ The design is safe against plastic deformation and brittle crushing.

c) Checking for contact compressive strength

$$\sigma_c = 0.74 \frac{i+1}{a} \sqrt{\frac{i+1}{ib} E [M_t]} \qquad \leq [\sigma_c]$$

Compressive strength = 12.36 Mpa which is less than the Gear strength.

The design is safe against contact compressive stress.

d) Checking for the bending strength of gear tooth

$$\sigma_b = \frac{i \pm 1}{amby} [M_t] \leq [\sigma_b]$$

Bending strength for gear tooth = 8.89 Mpa.

The design is safe and the teeth are sufficiently strong against bending stress.

V. CONCLUSION

By conducting the writing study it was tracked down that two phase epicyclic stuff train groups a few vibrations issues and furthermore influences on mechanical effectiveness, which is stay away from by the utilization of 3 phase epicyclic stuff train rather than 2 phase epicyclic gear train for transmitting power. Above both analyses shows the same result of using 3 stage epicyclic gear train as a solution.

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