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IJIEMR Transactions, online available on 13 Jun 2023. Link :http://www.ijiemr.org/downloads.php?vol=Volume-12&issue=Issue 06

10.48047/IJIEMR/V12/ISSUE 06/28

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Volume 12, ISSUE 06, Pages: 199-206

Paper Authors **A. Bhargav, A. Sai Teja, P. Sridhar, G. Pavan, R. Manisai,**

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DESIGN AND ANALYSIS OF PLANETARY GEAR

A. Bhargav, A. Sai Teja, P. Sridhar, G. Pavan, R. Manisai,
Department of Mechanical Engineering, ACE Engineering College, Hyderabad, Telangana, India.

Abstract
Planetary gear systems are extensively used for the power transmission and are the most critical component. Thus, Planetary gear systems are widely used in various mechanical and automotive applications due to their unique advantages, such as ability to achieve high reduction ratios, compactness, high torque transmissional capabilities and efficient power distribution. This project focuses on design and analysis of planetary gear used in the CNC bending machines. The design process involves selecting suitable gear arrangements, determining gear sizes and ratios, and optimizing the system for specific applications. Here, both modelling and analysis are performed in Fusion 360 software. And stresses and deflection are calculated theoretically and then these analytical results are validated by Finite Element Analysis at static condition by considering different materials for gears like structural steel and Aluminium A365 and results are compared.

I. Introduction

A planetary gear system, also known as an epicyclic gear system, is a mechanical arrangement of gears that offers several advantages over traditional gear systems. It consists of a central gear, called the sun gear, surrounded by multiple outer gears, known as planet gears. These planet gears are typically mounted on a carrier, which allows them to rotate around the central gear. The planetary gear system derives its name from the motion of the planet gears, as they revolve around the sun gear while also rotating on their own axes. This configuration creates a unique combination of gear movements, resulting in a variety of applications and benefits. One key advantage of the planetary gear system is its ability to achieve different gear ratios and torque outputs. By manipulating the arrangement of the sun gear, planet gears, and the outer ring gear, engineers can design gear systems with various speed reduction or speed increase ratios. This flexibility makes planetary gears widely used in diverse industries, including automotive, aerospace, robotics, and industrial machinery. Another benefit of planetary gear systems is their compact size and high-power density. The planetary arrangement allows multiple gears to share the load, distributing the forces evenly and reducing the size and weight of the gear mechanism. This compactness makes planetary gear systems well-suited for applications where space is limited, such as in automotive transmissions.

II. Literature Review

Bernad presented a study on light weight design of planetary gears transmission. Higher number of applied planet gears results in a higher mesh load factor, as well as an increasing difficulty in assembling the planets with low numbers of teeth. Thus, the number of teeth for the central gears was increased in order to compensate for the advantage of a better power division for higher numbers of applied planets. Difference in the centre distances of two gear pairs was offset by applying addendum modifications for transmission concepts with high gear ratios.

Alexander Kapelevich reported analysis and design of differential epicyclical gear arrangements that provide extremely high gear ratios. But this was suitable for low torque application like positioning in robotics.

Ovidiu Bunga in their study on optimal mass minimization design of two stage coaxial speed reducer with Genetic Algorithm compared the traditional design of speed reducer with optimal design by GA. The study about the trade-off between the mass and the service life was presented & they concluded that the required service life has to be sacrificed with 75% for a 2.5 kg saving (roughly 7%).

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Prabhakar V Pawar in their research paper titled "Critical Review of Design of Planetary Gears and Gear Box" told that in automotive and industrial applications, there is a great need for powerful and lightweight gear boxes to enhance their power density while reducing their complex vibration and noise delivery. Planetary gear transmissions are recognized to have many benefits over traditional transmissions, such as a high-power density due to the division of power using several planetary gears.

Cheon-Jae Bahk presented a nonlinear dynamic model of a planetary gear with tooth profile modification (TPM). The minimal dynamic response was achieved at different combination of sun–planet and ring–planet mesh TPM. Different TPMs are required for minimizing gear vibration depending on the amount of mesh stiffness fluctuation and the mesh phase. Different TPM minimizes the vibration at different vibration modes.

Nenad Marjanovic in their study of practical approach to the optimization of gear trains with spur gears presented selection of optimal gear trains and selection of optimal position for shaft axes of gears trains with spur gears. The volume of the gear train with spur gears is reduced by 22.5%. Software GTO provides the required results in a very short time.

III. Methodology

As we know that the gear is one of the most critical components of the power transmission system, failure in the gear will affect the whole transmission system and thus it is necessary to optimize the gear for low load operation and its effective delivery of power transmission. The main acting loads on a gear pair are as the Tangential Load, the Effective Load, the Bending load or Beam Strength, and the Pitting or Wear Strength load. Module: It is the ratio of the pitch circle diameter (in milli meters) to the number of teeth. It is usually denoted by m, where m = D / T D=Pitch Circle Diameter, T= Number of Teeth The recommended series of modules in Indian Standard are 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40 and 50.

Gear Material: The materials which are used for the gears depend upon the service factor and strength like wear or noise conditions etc. and they come in metallic and non-metallic form. For industrial purposes metallic gears are used, commercially can be obtained in steel, cast iron and Aluminium and bronze. Out of these, steel and Aluminium A365 materials have chosen analysis for best suitable one. These materials are chosen because it has high strength, toughness, can be accurately machined and be easily surface hardened. These properties are essential because in elements like gears, the surface is heavily stressed whereas the core stresses are comparatively less.

IV. Design and Analysis

4.1. Design Parameters

This work focuses on the planetary gear box used in the CNC bending machines where input torque and speed are generated by the servo-motor. Reduction ratio of 80-110 is required at the output shaft. The motor delivers a maximum power of 5.8 kW @3000 RPM. And design specifications are mentioned in the below table:

Gear Type	Symbol	Number of teeth	Pitch Diameter
sun	Zs	17	34
planet	Zp	61	122
ring	Zr	139	278

4.2. Determination of Reduction Ratio

The reduction ratio in the planetary gear train is determined as below: -

$$\text{Ratio} = \frac{\text{PCD of Ring Gear + PCD of Sun Gear)}}{\text{PCD of Sun Gear}}$$

Consider, Ts, Tp, Tr are Number of teeth of Sun Gear, Planet Gear and Ring Gear.

$$\text{Ratio} = \frac{T_r + T_s}{T_s}$$

$$i = \frac{17+139}{17}$$

$$i = 9.17$$

4.3. Calculations for Planetary System Speeds

When ring gear is fixed, Nr = 0

$$\text{Nc} = \frac{Ns \times Ts}{Tr + Ts} = \frac{3000\times17}{139+17}$$

$$\text{Nc} = 327 \text{ rpm}$$

Planet speed is given as,

$$Np = Nc\left(\frac{Tp+Ts}{Tp}\right) - \frac{Ns\times Ts}{Tp}$$

$$Np = 599.35\left(\frac{61+17}{61}\right) - \frac{3000\times17}{61}$$

$$Np = 418 \text{ rpm}$$

4.4. Transmission torque

The torque transmitted by the high-speed shaft is given by:

$$T = \frac{30\times P}{\pi\times N} \times i \times k_s$$

we know, motor power, P = 5.8 kW, rotational speed N = 3000 rpm and gear ratio i = 9.17/1.

Let, k_s is a service factor and

Assume,

1. Load Service: uniform
2. operating conditions at output: moderate shock
3. Duty hours of Gear box: t = 10hrs/day

$$k_s = k_a \times k_t$$

$$k_s = 1.25 \times 0.85$$

$$\therefore \; T = \frac{30\times5.8}{\pi\times3000} \times 9.17 \times 1.25 \times 0.85$$

$$T = 0.179\ KN - m = 179.87\ N - m$$

4.5. Pitch line velocity of a planet (v)

$$v = \frac{\pi D_p N_P}{60}$$

Where, D_p = Pitch diameter of planet

N_P = Speed of the planet

$$v = \frac{\pi\times0.122\times418}{60}$$

$$v = 2.67 \cong 2.7\ m/s$$

4.6. Calculation of Tangential load (Ft)

For Steel:
Ultimate tensile strength = 345 MPa
Yield strength = 207 MPa
Youngs modulus = 210000 MPa

For Aluminium A365 T6:
Ultimate tensile strength = 234 MPa
Yield strength = 165 MPa

Youngs modulus = 72400 MPa

Lewis equation,

$$F_t = \sigma_w P_c y_p b$$

Where,

σ_w is the permissible stress

P_c is the circular pitch

y_p is the form factor which is based on tooth profile (20° involute) and number of teeth.

b is the face width of the planet ($9.5m \leq b \leq 12.5m$)

and,

$\sigma_w = \sigma_d \times C_v$, where σ_d = allowable static stress and C_v = Coefficient of velocity

$\sigma_d = \frac{\sigma_y}{fos}$, where σ_y = Yield strength

For Structural steel, σ_y= 207 Mpa and considering fos = 3.2

$\sigma_d = \frac{207}{3.2} = 64.68\ N/mm^2$

For Aluminium A356 T6, σ_y= 165 Mpa and considering fos = 12

$\sigma_d = \frac{165}{12} = 13.75\ N/mm^2$

$C_v = \frac{3.05}{3.05+v}$, for the precision gears with v < 12 m/s

$C_v = \frac{3.05}{3.05+2.7} = 0.53$

$\therefore \sigma_w = 64.68 \times 0.53 = 34.28\ N/mm^2$ (steel)

$\therefore \sigma_w = 13.75 \times 0.53 = 7.28\ N/mm^2$ (Aluminium A356 T6)

$y_p = 0.154 - \frac{0.912}{61} = 0.139$

Take face width, $b = 10\ m = 10 \times 2 = 20\ mm$ and Circular pitch, $P_c = \pi \times 2 = 6.28$

Therefore,

Tangential load,

For Structural steel,

$$F_t = 34.28 \times 6.28 \times 0.139 \times 20$$
$$F_t = 598.47\ N$$

For Aluminium A356 T6,

$$F_t = 7.28 \times 6.28 \times 0.139 \times 20$$
$$F_t = 127.09\ N$$

4.7. Calculation of beam strength (F_s)

Considering a gear tooth as a cantilever beam, its strength can be calculated under the tangential load.

$F_s = \sigma_d P_c y_p b$

$F_s = 64.68 \times 6.28 \times 0.139 \times 20$

$F_s = 1129.20\ N$

4.8. Calculation of dynamic load (F_d)

It is the sum of tangential load (F_t) and dynamic induced load (F_i).

According to Bucking hams equation,

Dynamic load,

$$F_d = F_t + F_i$$

$F_i = \frac{21v(bC+F_t)}{21v+\sqrt{bC+F_t}}$, where C is the deformation factor

and $C = \frac{K \times e}{\frac{1}{E_p}+\frac{1}{E_s}}$

where, K is a factor depending upon form of the teeth and e = machining error

K = 0.111, for 20° full depth involute teeth

$e = 25 \times 10^{-3} = 0.025$

$$C = \frac{0.111 \times 0.025}{\frac{1}{210 \times 10^3} + \frac{1}{210 \times 10^3}}$$

$C = 291.37\ N/mm^2$

$$F_i = \frac{21 \times 2.7(20 \times 291.37 + 598.47)}{21 \times 2.7 + \sqrt{20 \times 291.37 + 598.47}}$$

$F_i = 2662.15\ N$

Therefore,

$F_d = 598.47 + 2662.15$

$F_d = 3260.62\ N$

4.9. Check for endurance strength (F_{en})

$F_{en} = \sigma_{en} bYm$

Where,

$\sigma_{es} = 2.75\ BHN - 70 = 2.75 \times 200 - 70 = 480\ MPa$ and

b = 20 mm ; $Y = \pi \times y = \pi \times 0.139 = 0.43$

$F_{en} = 480 \times 20 \times 0.43 \times 2$

$F_{en} = 8256\ N$

Since $F_{en} > F_d$ (8256N > 3260.62 N), the design will be satisfactory from the point of wear and durability.

4.10. Theoretical Stresses and deflection

Bending Stress:

Lewis Bending Stress,

$$\sigma_b = \frac{F_t}{b \times m \times y}$$

Therefore,

For Structural steel,

$$\sigma_b = \frac{598.47}{2 \times 20 \times 0.139} = 107.63\ N/mm^2$$

For Aluminium A356 T6,

$$\sigma_b = \frac{127.09}{2 \times 20 \times 0.139} = 22.85\ N/mm^2$$

Deflection:

Deflection = $\frac{A}{B}$

A = $F_t \times L^3$

Where, L = H = Tooth Height

= 2.25 × Module

= 4.5 mm

Therefore,

A = $598.47 \times 4.5^3 = 54535.57$

A = $127.09 \times 4.5^3 = 11531.07$

and, B = $3 \times E \times I$

I = $\frac{a \times b^3}{12}$

Where, a = face width = 20 mm

b = tooth width = $\frac{\pi \times m}{2}$ = 3.14 mm

hence,

I = $\frac{20 \times 3.14^3}{12}$ = 51.59 mm^4

Therefore,

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$B = 3 \times 2.1 \times 10^5 \times 51.59 = 3250170$

$B = 3 \times 0.724 \times 10^5 \times 51.59 = 11205348$

∴ For structural steel,

$$\text{Deflection} = \frac{54535.57}{3250170} = 0.017 \text{ mm}$$

For aluminium A365 T6,

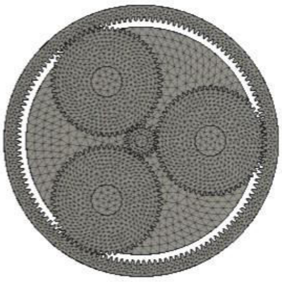
$$\text{Deflection} = \frac{11531.07}{11205348} = 0.001 \text{ mm}$$

4.11. Simulation

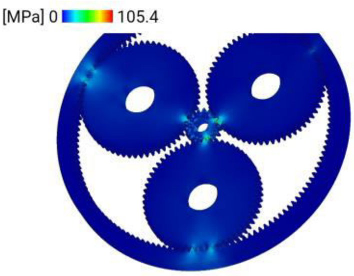
Meshing:

Number of nodes = 128110

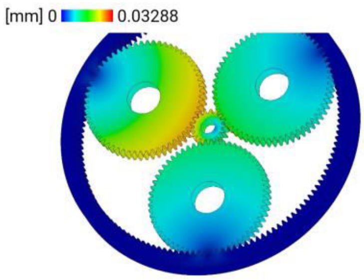
Number of elements = 75139



Structural steel:

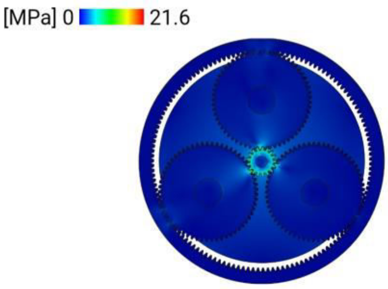


Stress induced

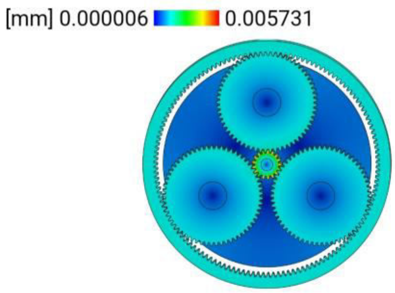


Total deflection

Aluminium A356 T6:



Stress induced



Total deflection

V. Result

The analytical calculations for bending stresses and deflection have been done by Lewis's equation. Below tables shows the bending stresses and total deflections for structural steel and Aluminium A365 materials. The results showed that both the theoretical values and experimental values were almost nearer. Hence, we can say that the design is validated.

Structural steel:

	Theoretical	Experimental
Stress, MPa	107.63	105.4
Deflection, mm	0.017	0.032

Aluminium A365 T6:

	Theoretical	Experimental
Stress, MPa	22.85	21.6
Deflection, mm	0.001	0.005

VI. Conclusion

The main objective of this work was to redesign the planetary gear box which will be suitable for machine tool application by increasing reduction ratio and performing the structural analysis of planetary gear considering materials like structural steel and Aluminium A365 T6. Based on the study the results obtained have been analysed and summarized below:

- The design parameters like Number of teeth, module, number of planets, face widths, tooth profile modification, material are very important in deciding the load capacity in bending and wear, life and cost of gears.
- It is seen that higher reduction ratio results in greater torque for smaller input.
- The Fem based static analysis shows that the total deflection and stress induced in the material were less for the Aluminium A365 T6 as compared to Structural steel. Hence, we may conclude that the Aluminium A365 T6 is best suitable material for planetary gear.
- This study will help to understand more the behaviour of the planetary gear and gives information for the manufacturer to improve the strength of the planetary gear. It can help to reduce the cost and time required for the research and development of new product.

References

1. Bernd-Robert Höhn, Karsten Stahl and Philipp Gwinner―Light-Weight Design for Planetary Gear Transmissions‖ GEAR TECHNOLOGY, Sept. 2013, PP: 96-103
2. Lehao Chang, Geng Liu, Liyan WuDr, ―A robust model for determining the mesh stiffness of cylindrical gears‖ Mechanism and Machine Theory 87 (March-2015) PP: 93–114.
3. Dr. Alexander Kapelevich ―High Gear Ratio Epicyclic Drives Analysis‖ GEAR TECHNOLOGY, June 2014 PP: 61-67
4. Ovidiu Buiga, Lucian Tudose ―Optimal mass minimization design of a two-stage coaxial helical speed reducer with Genetic Algorithms‖ Advances in Engineering Software 68 (2014) PP: 25–32
5. Syed Ibrahim Dilawer, Md. Abdul Raheem Junaidi, Dr.S.Nawazish Mehdi ―Design, Load Analysis and Optimization of Compound Epicyclic Gear Trains‖ American Journal of Engineering Research ISSN 2320-0936 Vol.-02, Issue-10, 2013, PP: 146-153
6. Cheon-Jae Bahk, Robert G. Parker ―Analytical investigation of tooth profile modification effects on planetary gear dynamics‖ Mechanism and Machine Theory 70 (2013) PP: 298–319
7. Nenad Marjanovic, Biserka Isailovic, Vesna Marjanovic, Zoran Milojevic, Mirko Blagojevic, Milorad Bojic, ―A practical approach to the optimization of gear trains with spur gears‖ Mechanism and Machine Theory 53 (2012) PP:1–16

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8. Jelena Stefanović-Marinović and Miloš Milovančević ―Load Allocation and Equilibrium for Planetary Gear Reducers of Earth-Pressure-Balance Shield Machine‖ Journal of Mechanics Engineering and Automation 2 (2012) PP: 365-373