

Design and Optimization of Planetary Gearbox for a Formula Student Vehicle

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ABSTRACT

This paper presents the method of designing an optimized light weight, cost effective planetary gearbox for a Formula Student vehicle. The gearbox has a high speed functioning capability, in addition to the compact size and light weight. The iterative optimization procedure used provides a technique for selecting the best possible configuration of the gearbox. Conventional gearboxes used for this purpose are generally two step reduction gearboxes, which are bulkier in terms of weight and volume. Also, a review of the existing market reveals that the planetary gearboxes manufactured in India are not capable of handling high speeds, thus rendering them futile for racing applications.

The target reduction ratio for the gearbox is a fixed parameter. The method involves design and optimization of the gear-train with the calculated ratio. A detailed algorithm has been used, which involves developing a mathematically modelled code for deciding gear characteristics, physical modelling using CAD, and structural testing of the same through simulation, with repeated iterations with a target to reduce both weight and volume. The program used has been modelled on the theoretical data acquired through perusal of available literature. The selection of bearings have been made after calculating the forces acting on the bearings, followed by market research to select bearings with the requisite load capability. The shafts for the gears have also been designed according to force calculations, followed by structural analysis. The casing for the gears is constructed around the gear-train for efficient packaging and validated through static structural simulation. A preliminary thermal analysis has been completed to verify the design according to thermal considerations. The result of the same has been presented.

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INTRODUCTION

Planetary Gearboxes (or Epicyclic gearboxes) have been widely used as one of the better ways to transmit power for high torque applications. With more efficiency than that of normal series gearing, their use is becoming more and more prevalent in the automotive field, especially in the case of Formula Student electric vehicles. Low spatial requirement and a high power-to-weight ratio make these gearboxes superior, especially when compared to double-step single reduction gearboxes. The planetary arrangement also keeps the motors concentric to the driveshaft, thus providing more efficient packaging of components. This concentricity can also be used to produce wheel hub-mounted motors, especially in Formula Student vehicles which are powered at all four wheels.

However, one major disadvantage of the planetary arrangement is its incapability to function at higher speeds due to high noise and wear. Additionally, the planetary arrangement can only be used for geartrains with a reduction ratio of at least 3:1 or greater [1].

DESIGN CHALLENGE

The system that has been designed consists of a single stage conventional planetary gearbox ($N_R > N_S$) which can sustain the peak torque and RPM rating of the motor (54 N-m and 6000 RPM respectively) [2] with which it is to be coupled. This paper covers all the aspects of designing the gearbox, including selection of the gears, shafts, bearings and the outer casing. The gearbox ratio is fixed at to give a target reduction in between 6.2:1 and 6.5:1.

Efficiency Analysis of Planetary Configurations

This paper considers three possible configurations for the geartrain to be used in the design, since these configurations are capable of providing the requisite reduction ratios for speed reduction. The available choices are summarized in [Table \(1\)](#) shown below, with their respective values of efficiency [3].

Table 1. Configuration Specifications.

Cfg.	Driver	Follower	Fixed	Red. Ratio	Efficiency
1	S	C	R	$1+N_R/N_S$	$(m_{GSR}\eta_{SR}-1)/(m_{GSR}-1)$
2	S	R	C	N_R/N_S	η_{SR}
3	R	C	S	$1+N_S/N_R$	$(m_{GSR}\eta_{SR})/(m_{GSR}-1)$

Here, R, S and C represent Ring, Sun and Carrier respectively. The efficiency of the configuration with fixed carrier is taken to be η_{SR} , which is a positive quantity between zero and one. Since the ring gear is always larger in dimension than the sun gear, the value of m_{GSR} is always greater than one.

From simple inequality and attribution of the above mentioned limits to the variables η_{SR} and m_{GSR} , it can be seen that the efficiencies of the three configurations lie in the order $3>1>2$, where the integers refer to the first column shown in Table (1).

From Table (1), configuration 1, i.e., the configuration with fixed ring gear, has a maximum reduction ratio, since $N_R>N_S$. Thus, selection of this configuration is justified insofar as the paper aims at optimization of the planetary gearbox in terms of weight and volume occupied. The selected configuration can provide the requisite reduction ratio while maintaining the smallest size, since the ratio N_R/N_S can be smaller than in other configurations. The selection of configuration 3 can be easily ruled out although it is the most efficient since it cannot provide the requisite reduction ratio.

In addition to lower efficiency and higher weight and volume occupied as shown above, considering the fact that the manufacturing of a shaft attached to a rotating annulus may not be economically viable, especially in order to guarantee concentricity and eliminate eccentricity in the output shaft, configuration 2 can be eliminated.

The input is provided at the sun gear, with the output at the planet carrier and the ring gear remaining stationary. The requirements are summarized in Table (2) shown below.

Table 2. Design Specifications.

Rated Torque (Peak)	54 N-m
Rated RPM (Peak)	6000 RPM
Time of Operation	~30 min (continuous)
Reduction Ratio (Target)	6.2:1-6.5:1
Input	Sun Gear
Output	Planet Carrier
Maximum/Average Power of Motor	30 kW/18kW

CALCULATIONS

The three main components which have to be designed are the gears, the bearings and the shafts. In this section, the mathematical equations involved in the design procedure of gears, bearings and shafts will be looked at.

Gears

Gears can be of multiple types. The most commonly used gear types for parallel shaft gearboxes are spur and helical gears. For the gearbox, spur gears have been used in order to avoid the axial loads generated in a helical mesh, which in turn affects the bearing and casing design. All the gears present in the single stage planetary arrangement have similar methods of design, barring a few alterations which have been mentioned below.

The main equations required for gear design are the two AGMA stress equations for bending stress and contact stress respectively. The two equations are as follows [4]:

$$\sigma = W_t K_o K_v K_s \left(1/bm_t \right) \left(K_H K_B / Y_J \right) \quad (1)$$

$$\sigma_c = Z_E \sqrt{W_t K_o K_v K_s \left(K_H / D_p b \right) Z_R / Z_I} \quad (2)$$

The value of tangential load acting on the gears can be calculated from the value of rated Torque and Speed rating of the gearbox, while the other factors are as mentioned in [4], which uses AGMA standards. Here, the Geometry Factor for Bending Strength (Y_J) is approximated in Eqn. (3) as [5]:

$$Y_J = \pi \left(0.154 - \frac{0.912}{n} \right) \quad (3)$$

This equation gives a good estimate of the values presented in the AGMA standards used for gear manufacture.

The selected material for the gears is carburized and case hardened EN 353 grade alloy steel due to its economic feasibility and ready availability with manufacturers.

The maximum stress within the material should be less than the allowable stress values for bending and contact stresses. These are defined as [4]

$$\sigma_{all} = \left(S_t / S_F \right) \left(Y_N / Y_\theta Y_Z \right) \quad (4)$$

$$\sigma_{c,all} = \left(\frac{S_c}{S_H} \right) \left(\frac{Z_N Z_W}{Y_\theta Y_Z} \right) \quad (5)$$

The values of S_t (Allowable Stress Number for Bending) and S_c (Allowable Stress Number for Contact) are derived from the AGMA standards for case hardened carburized steel alloy. The Safety Factor values are taken as per requirement. Since the motor used will never generate torque and RPM values which are more than the peak ratings, the Factor of Safety can be kept at unit value. However, the value has been chosen to be 1.2 as a precautionary measure.

The above equations are useful for the design of the Sun Gear in the planetary arrangement. For the design of the planet gears, the following changes have to be made:

1. The planets face bi-directional loading. Hence, the value of S_t is taken to be 0.7 times the actual value [4]. This value is recommended by AGMA, and lies in between the Goodman and Gerber failure loci
2. The planet gears should be designed for at least $[1+0.9(\eta_1)]$ times the actual load it is supposed to sustain in the absence of a load sharing mechanism [6]
3. The gear ratio to be used for calculation does not directly take the value of (N_p/N_s) . The actual value to be used is $(N_p/N_s)(\omega_i - \omega_o)/\omega_i$ [3]

The different gears are shown in Fig. (1), Fig. (2) and Fig. (3).

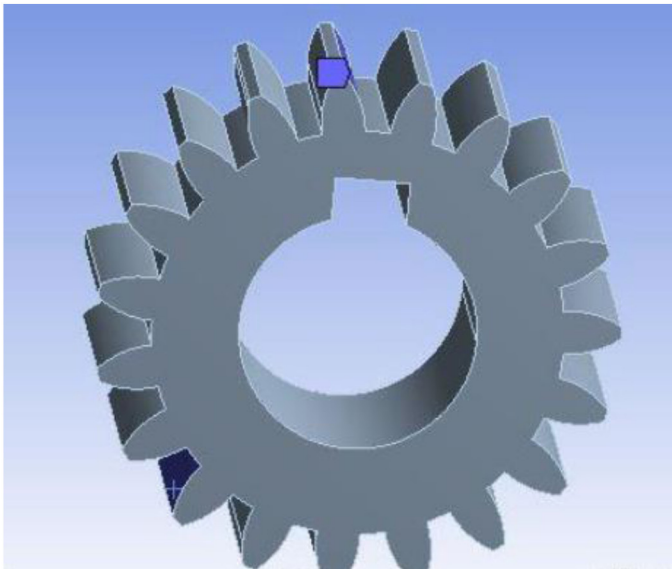


Figure 1. Sun gear.

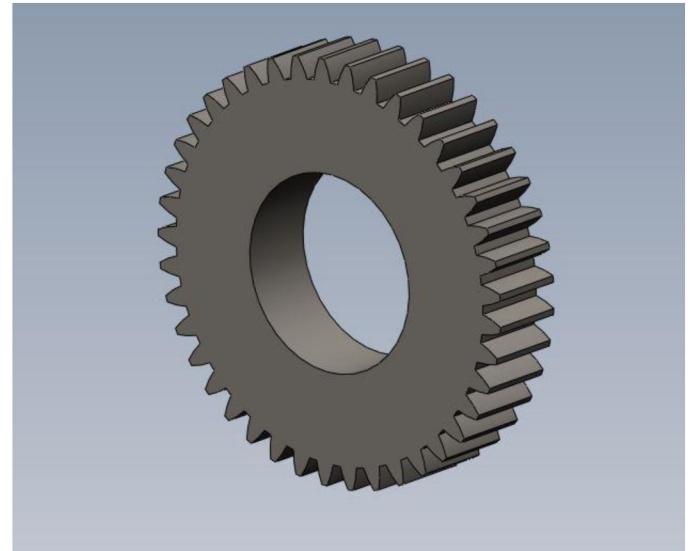


Figure 2. Planet gear.



Figure 3. Ring gear.

Floating Arrangement of Planetary Gears

For equal load sharing in the planet gears, there are 3 methods as follows:

1. Completely rigid system with zero tolerances
2. System with flexibility (floating)
3. With mechanical means to adjust planets to provide load equalisation

Out of these three options, the second option is the cheapest to implement, and is ensured in the following ways: i) The Sun Gear shaft is kept flexible so that it can adjust its position according to requirement, and ii) $(N_R + N_S)/\eta$ should be an integer [6].