

Design and Evaluation of a Single-Stage Planetary Gearbox for Lightweight Electric Vehicles

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Abstract - This study explores the design and analysis of a compact gearbox intended for a small electric vehicle, aiming to deliver a solution that meets specific performance and design requirements. It begins with a review of various gearbox types, followed by a comparative analysis to identify suitable candidates. Several conceptual designs were developed and evaluated based on how well they aligned with the defined criteria. The most promising concept was then analyzed in depth using SolidWorks, with key design parameters determined through analytical calculations. Multiple SolidWorks models were created, varying factors such as gear module, face width, and material selection. Simulation results were encouraging, and the factor of safety was within the targeted range. The study concludes by offering suggestions for future improvements and identifying additional considerations for advancing compact gearbox design.

Keywords: lightweight; gearbox; electric vehicle; CAD; efficiency; design requirements.

1. Introduction

Electric vehicles offer a major advantage: they produce no harmful emissions such as carbon dioxide or carbon monoxide, which are key contributors to air pollution. As highlighted in [1], the world is currently facing two pressing challenges, rising oil prices and increasing carbon emissions. Designing a gearbox for an electric vehicle presents several challenges. It requires careful consideration of factors such as efficient power transmission from the motor to the wheels, while also ensuring the gearbox is lightweight, durable, and compact enough to fit within the vehicle's limited space. A successful design depends on a solid understanding of gearbox fundamentals, including selecting suitable gear types and materials and knowing how these components interact in operation [1]. Kinematics [2], [3] and control considerations [4], [5] are essential in the conceptual design phase to ensure feasible motion, adequate workspace, and controllable dynamic behavior with suitable actuation and sensing strategies.

One of the main advantages of low-speed electric vehicles is their affordability and environmental benefits. These vehicles are becoming increasingly popular for urban and short-distance transportation. However, to operate efficiently, they require compact and efficient drivetrain systems. For low-speed EVs, it is essential that the drivetrain be reliable, lightweight, and energy-efficient. Traditional drivetrain systems are often bulky and complex, characteristics that make them unsuitable for low-speed applications, as they tend to consume more energy and reduce battery range. A lightweight drivetrain, on the other hand, can significantly enhance vehicle performance and improve energy efficiency. Moreover, a reliable design not only supports better long-term performance but also helps reduce maintenance requirements and extend the vehicle's service life.

In this study, spur and helical gears are analyzed and compared as common options, while the ultimate goal is to develop a planetary gear system, which is also discussed in detail. These three types represent widely used configurations and comparing them is essential to determine the most suitable choice for the intended application. Spur gears are the simplest type of gear, known for their straightforward design and high efficiency. They are easy to manufacture and perform well in terms of power transmission. However, they tend to generate significant noise at high speeds and are generally not ideal for heavy-load applications due to increased stress on the gear teeth [6]. Helical gears differ in that their teeth are cut at a helix

angle (Ψ), which enables smoother and quieter operation compared to spur gears. This angled design improves load capacity through continuous tooth engagement, allowing better force distribution. However, the helix angle introduces axial forces in addition to radial and tangential forces, making the design more complex. Despite this, helical gears are favored in many applications for their durability and ability to handle higher loads [6].

The planetary gear system is the primary focus of this study. Its compact design allows it to transmit high torque while saving space, a critical factor in small electric vehicles. A typical planetary setup includes a sun gear, multiple planet gears, a carrier, and a ring gear. This configuration distributes power efficiently and enables smooth transmission in a minimal footprint [7]. Despite these advantages, planetary gear systems also present certain challenges. Their design and analysis are more complex, leading to more costly manufacturing compared to simpler gear types. An unbalanced system can result in unwanted vibrations and noise, negatively impacting performance and ride comfort. Additionally, the closely packed components make lubrication and heat dissipation more difficult, which may accelerate wear over time. Designers must also balance efficiency with gear ratio flexibility, maximizing one can limit the other. Ongoing research aims to improve planetary gear performance. This includes exploring compound planetary systems (like 3K variants) that offer higher efficiency while maintaining compactness. Efforts are also being made to reduce vibration and noise through advanced modeling and experimental techniques.

Material selection plays a critical role in gear system reliability. Through-hardened steels such as AISI 8620 and 9310 are often used for gears due to their toughness and fatigue resistance after carburizing. High-carbon chromium steels like AISI 52100 are commonly used in bearings for their wear resistance under cyclic loading. For carriers and housings, lightweight yet strong materials such as cast iron or high-strength aluminum are preferred to balance durability and weight [8], [9].

Finite Element Analysis (FEA) has played a significant role in the design and evaluation of planetary gearboxes. In the early stages of development, simplified models were often used to quickly explore design concepts. As the design matured, more detailed 3D models were introduced to analyze complex gear meshing behavior, load distribution, and housing deformations. These simulations typically included contact stress analysis, modal analysis for dynamic response, and transient dynamic simulations to mimic realistic operating conditions. Techniques such as sub-modeling were used in scenarios where high-resolution results were needed in specific areas without excessively increasing computation time [10]. Fatigue analysis has long been a key focus in gearbox research. Initially, S-N curves combined with empirical corrections—such as the Goodman relation—were used to account for the effects of mean stresses. With the integration of FEA data, more advanced fatigue-life prediction methods were adopted, including multiaxial fatigue theories that better reflect real-world loading conditions. Some studies even introduced variable load spectra, moving beyond constant amplitude loading to more accurately estimate fatigue performance under realistic usage scenarios [10], [11].

One of the challenges encountered during this project was the limited availability of research targeting the specific gearbox configuration of interest. Many existing studies focused on one- or two-stage planetary gearboxes designed for lightweight, high-speed applications, often intended for a single passenger. These designs typically featured gear ratios ranging from 11.7:1 to 13.1:1, which is higher than the range targeted in this study. While a few references were found for similar applications, such as low-speed vehicles, they were designed for different load conditions, including one notable case developed for six passengers rather than four [12], [13]. During the brainstorming phase, the environments selected which the vehicle was to be used in are campus', hospitals, parks and even golf courses. Since the application was decided a gear ratio of 7-8 was seen as reasonable since the highest speeds in these applications is chosen to be 40 km/h. Since these applications may have people who use this transportation instead of walking with luggage, the vehicle was decided to handle a mass of 350kg. Finally, the gearbox shouldn't be heavy and take up much space inside the vehicle as there should be some flexibility around the gearbox for other parts of the electric vehicle.

The rest of the paper follows the order that will be mentioned, it will first start off with what the design is required and aims to design, followed up by how it was created in SolidWorks. After that the analysis on how the gear ratio was obtained will be explained, in addition to that, the internal forces acting on the gear will be shown how to calculate.

Immediately after that fatigue and structural analysis will be provided and explained, all which have been provided by SolidWorks. Later, the cost and manufacturing will be explained in detail, but the cost is also obtained from SolidWorks. To conclude this paper there will be a discussion and conclusion about the design and its results, coupled with the future work that can be added to the design.

2. Design Requirements and Methodology

Before proceeding with the design and simulation in SolidWorks, it is essential to analyze the key parameters and constraints for the small electric vehicle. These parameters serve as the foundation for deriving the conceptual design and narrowing down the design space based on engineering requirements.

In this study, the vehicle is assumed to have a maximum speed of 40 km/h (equivalent to 11.11 m/s), which it reaches in 8 seconds. The total load on the vehicle is 450 kg. To calculate the rolling resistance force, the rolling coefficient is taken as $C_r = 0.015$, along with gravitational acceleration $g = 9.81 \text{ m/s}^2$. A slope angle of $\theta = 5^\circ$ is also assumed to compute the slope resistance force.

The acceleration force F_a is calculated based on the target speed and acceleration time. By combining the rolling, slope, and acceleration forces, the required power P (in watts) is determined. Motor efficiency is also considered, and an efficiency of $\eta_{motor} = 92\%$ is assumed for more realistic estimation.

The total force and the wheel radius ($r_{wheel} = 0.25 \text{ m}$) are then used to calculate the torque at the wheel, τ_{wheel} . Using the velocity ratio $VR = 7.06$, the input torque τ_{input} is obtained.

It is worth noting that aerodynamic drag is neglected in this analysis, as the body of the vehicle is not modeled in the current scope.

$$F_r = C_r * m * g \quad (1a)$$

$$F_g = m * g * \sin(\theta) \quad (1b)$$

$$F_a = m * a \quad (1c)$$

$$F_{total} = F_r + F_g + F_a \quad (1d)$$

$$P = F_{total} * v \quad (1e)$$

$$P_{motor} = \frac{P}{\eta_{motor}} \quad (1f)$$

$$\tau_{wheel} = F_{total} * r_{wheel} \quad (1g)$$

$$\tau_{input} = \frac{\tau_{wheel}}{VR} \quad (1h)$$

The following results were obtained from the preliminary calculations: the vehicle acceleration is approximately 1.389 m/s^2 , the rolling resistance force is about 66.2 N , and the grade resistance force is estimated at 765.4 N . The acceleration force is around 625.1 N , resulting in a total tractive force of $1,456.7 \text{ N}$. The required power is calculated to be approximately 12.8 kW , and when motor efficiency is taken into account, the total required input power increases to about 15.1 kW .

These values indicate that the gearbox must be capable of handling an input power of around 15.1 kW . The torque at the wheels is calculated to be 288 Nm , and the corresponding input torque is 40.85 Nm , based on the velocity ratio. Given these requirements, a suitable motor would be the REX 30 electric motor, which offers a peak power of 25 kW , peak torque of 70 Nm , and a maximum speed of 6000 rpm . The motor weighs approximately 5.25 kg and features a hybrid cooling system (fluid/air). It has a diameter of 216 mm and delivers a continuous power output ranging between 15 and 17 kW [14], [15].

The initial step in the gear design process involved evaluating two conceptual options: a helical gear system and a planetary gear system. Given the target gear ratio of $7.05:1$, a helical configuration would require a two-stage setup, making it bulkier and less space-efficient. As a result, the planetary gear system was deemed more suitable for this application due to its compactness and ability to achieve the desired ratio in a single stage. Two planetary configurations were considered for this project. The first is a compound planetary setup consisting of one sun gear, three planet gears, and a ring gear. The

second configuration includes two sun gears and two planet gears, without a ring gear. In the first setup, power input can be applied through the sun gear, planet carrier, or ring gear, depending on which component is fixed. Similarly, in the second setup, the input and fixed components can vary among the two sun gears and the carrier.

Given the project’s requirements for a compact and lightweight drivetrain, the compound planetary gear was selected. It achieves the desired 7.05:1 ratio within a single stage, minimizing the gearbox footprint.

As part of the gear design process, selecting an appropriate face width is critical. A common rule of thumb is to set the face width at ten times the gear module. Alternatively, the minimum face width is often defined as 2.5 times the ring gear’s tooth height. In this study, the tooth height of the ring gear was measured using SolidWorks tools to guide the selection of a suitable face width [16], [17].

In SolidWorks, three design iterations were carried out. The first two involved variations in geometric properties, while the third focused solely on changing material selections to achieve satisfactory performance. In the initial design, the gear module was set to 1.5, with all gears having a thickness of 15 mm, except for the carrier, which was designed with a thickness of 10 mm. These gear components were selected from the SolidWorks Toolbox. As for the shafts, the input and planetary shafts each have a length of 35 mm and a diameter of 5 mm, while the output shaft is slightly shorter at 30 mm with a diameter of 10 mm. The key for the sun gear measures 4 mm in length, has a radius of 2.5 mm, and extends 3.85 mm from the shaft center to the tip. The key for the carrier is 5 mm long, with the same 2.5 mm radius, and a distance of 2.5 mm from the center to the shaft tip. Both the sun and planet gears feature a chamfer of 0.2 mm at their internal diameters, while the outer edge of the shafts is chamfered by 0.5 mm. Fillets were added where appropriate to enhance the finish and reduce stress concentrations.

Extruded cuts were applied to the sun and planet gears to accommodate spacing and assembly requirements. For the sun gear, two internal diameters were cut: one at 7 mm and another at 17 mm to create necessary clearance. The planet gears had cuts at 13 mm and 48 mm, respectively. Bearings were also added from the SolidWorks Toolbox. For the output shaft, the selected bearing had dimensions of 1710 mm overall size, 10 mm bore diameter, 15 mm outer diameter, and 3 mm thickness, and included a cage with up to 20 balls. The other bearings used in the design were sized at 175 mm, with a 5 mm bore, 8 mm outer diameter, and 2 mm thickness, each containing up to 18 balls with a cage.

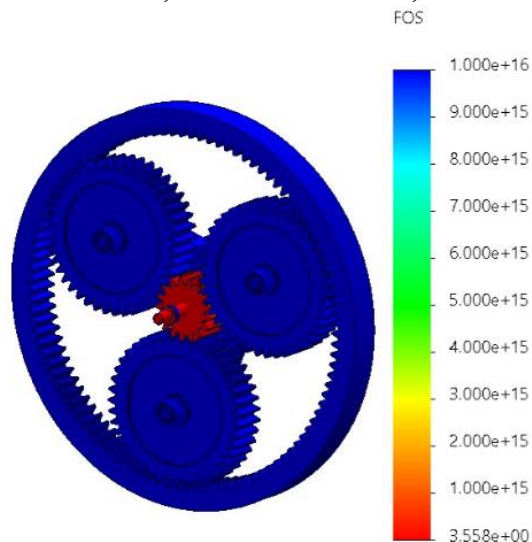


Fig. 1: Design number 1 with the Alloy Steel. Achieved factor of safety: 3.558.

Figure 1, shows the first iteration, using Alloy Steel and resulting in a Factor of safety of 3.558. The second design iteration, shown in Figure 2, retained most of the properties from the first model, with a few key modifications aimed at reducing weight and overall size. In this version, the gear thickness was reduced to 10 mm, and the gear module

decreased to 1. The extruded cuts were also adjusted: for the sun gear, the diameters were set to 7 mm and 12 mm, while for the planet gears, the cuts measured 15 mm and 35 mm. As a result of these changes, the overall length of the gearbox was reduced to 80 mm.

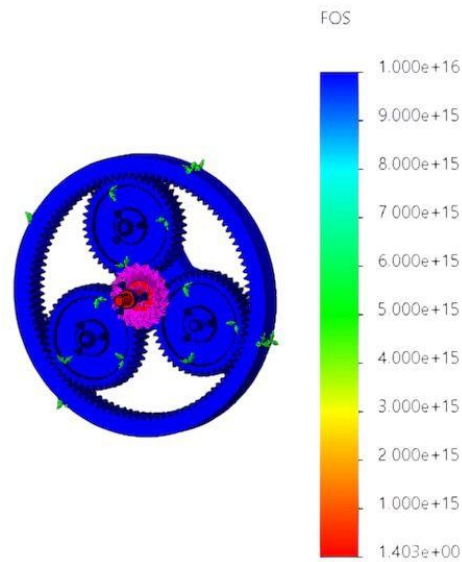


Fig. 2: Second design: low factor of safety.

Even after using the same materials in both designs there is a clear difference in which is safer, therefore in the third iteration Aluminum 7075-T6 (ss) helped achieving a satisfactory factor of safety of 2.913, as shown in Figure 3.

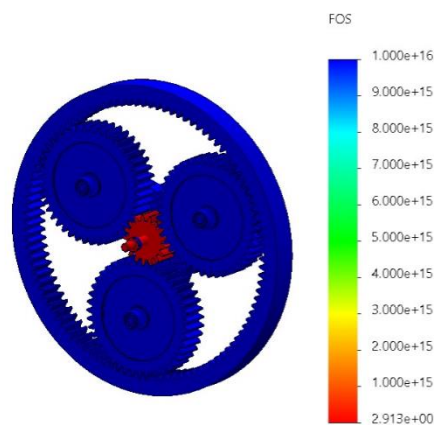


Fig. 3: Factor of Safety = 2.913 for Aluminum 7075- T6 (ss).

3. Kinematic and Dynamic Analysis

Equations (2a) and (2b) are used to determine the minimum number of teeth for the pinion and the corresponding number of teeth for the larger gear relative to the pinion to avoid interference [10].

$$N_P = \frac{2k}{(1+2m)\sin^2\phi} \left(m + \sqrt{m^2 + (1+2m)\sin^2\phi} \right) \quad (2a)$$

$$N_G = \frac{N_P^2 \sin^2\phi - 4k^2}{4k - 2N_P \sin^2\phi} \quad (2b)$$

To find the diameter of the gear after selecting the number of teeth for any gear with the only exception of the ring which will be calculated, and the module, Equation (3) is used.

$$m = \frac{d}{N} \quad (3)$$

$$N_C + 2N_P = N_R \quad (4)$$

In turn, the train value is calculated by using Equations (5a), (5b), and (5c).

$$e = (-1)^q * \frac{N_{driver}}{N_{driven}} \quad (5a)$$

$$VR = \frac{n_{out}}{n_{in}} \quad (5b)$$

$$e = \frac{n_R - n_{arm}}{n_C - n_{arm}} \quad (5c)$$

Various tooth count combinations were tested to achieve the desired gear ratio of 7.05:1. The following number of teeth configurations were evaluated: [sun: 20, planet: 40], [sun: 20, planet: 42], [sun: 16, planet: 42], [sun: 18, planet: 42], [sun: 18, planet: 40], [sun: 17, planet: 42], and [sun: 17, planet: 43]. It was found that the combination of a 17-tooth sun gear and a 43-tooth planet gear yielded a gear ratio of approximately 7.06:1, closely matching the target.

With this gear ratio, and assuming the motor provides an input speed of 3000 rpm, the output speed after reduction is approximately 424.93 rpm. Given that the motor produces a maximum torque of 70 N.m, the resulting maximum torque at the wheels, based on the gear ratio, is calculated to be 494.2 N.m.

To analyze internal forces such as radial and tangential forces acting on the gear components, further calculations are required as explained in Equations (6a), (6b), and (6c). These will be performed under the assumption that forces are evenly distributed across all meshing gears. Figure 4 illustrates the gear configuration, where the sun gear (yellow) is designated as gear 2, the planet gear (blue) as gear 3, and the ring gear (red) as gear 4.

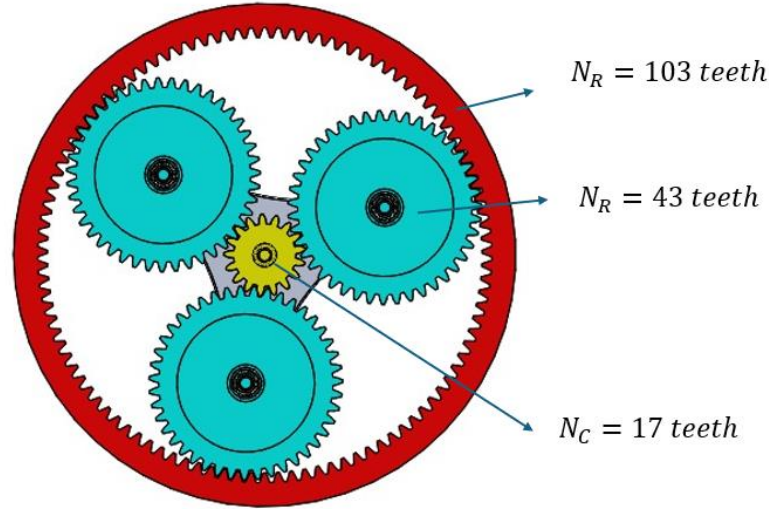


Fig. 4: Planetary gearbox configuration achieving a speed reduction ratio of 7.06:1

$$F_{23}^T = \frac{60,000 * H}{\pi * d * n_s} \quad (6a)$$

$$F_{23}^R = F_{23}^T \tan(\theta) \quad (6b)$$

$$F_{23} = \sqrt{F_{23}^T{}^2 + F_{23}^R{}^2} \quad (6c)$$

Where F_{23}^T is tangential component of the gear contact force in (N),

H is the power in (kW)

d is the gear pitch diameter in (mm)

n_s is the sun gear speed in (rev/min)

θ is the pressure angle in ($^\circ$)

F_{23} Resultant gear force

4. Structural Analysis

After completing the planetary gearbox design in SolidWorks, the simulation results for von Mises stress and factor of safety were highly satisfactory and well within expectations. The stress values remained below the material's yield strength of 505 MPa, indicating a structurally sound design. Additionally, both the displacement and strain levels were minimal, further confirming the reliability and stability of the gearbox under the given loading conditions.

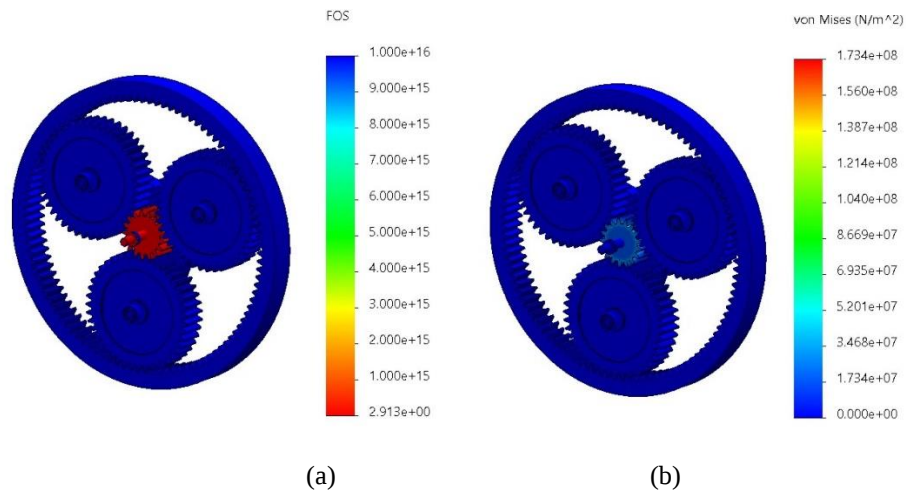


Fig. 5: Aluminum 7075- T6 (ss) with 1.5 gear module: (a) Factor of safety. (b) Von-mises stress.

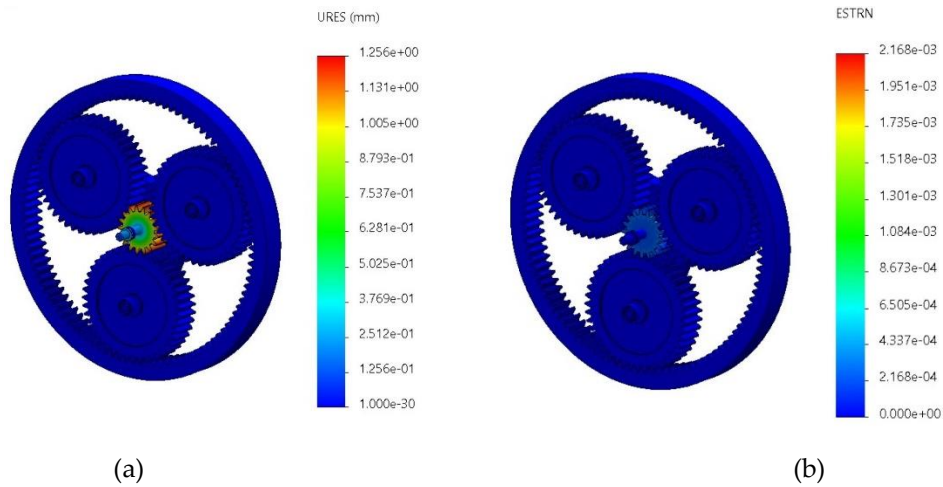


Fig. 6: Aluminum 7075- T6 (ss) with 1.5 gear module: (a) Displacement results in mm. (b) Strain results.

Moving on to the gear deflection analysis, a tangential force was applied to evaluate the deformation of each gear within the gearbox. SolidWorks was used to simulate and assess the deflection, providing results for von Mises stress, strain, and factor of safety. Since sun gear is subjected to the higher, the analysis in this case focuses solely on that component to assess its structural response under the load.

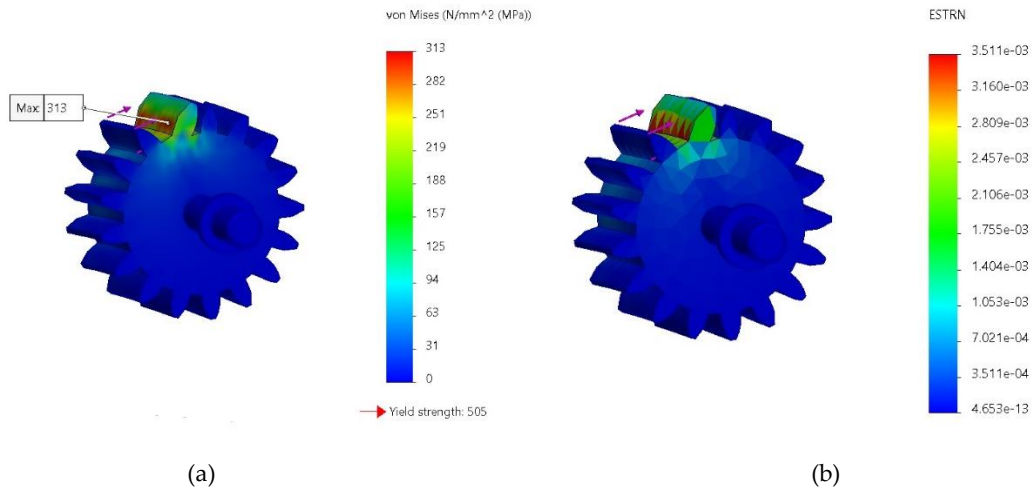


Figure 7. Simulation of the sun gear using torque from the motor. (a) Von-mises stress. (b) Strain.

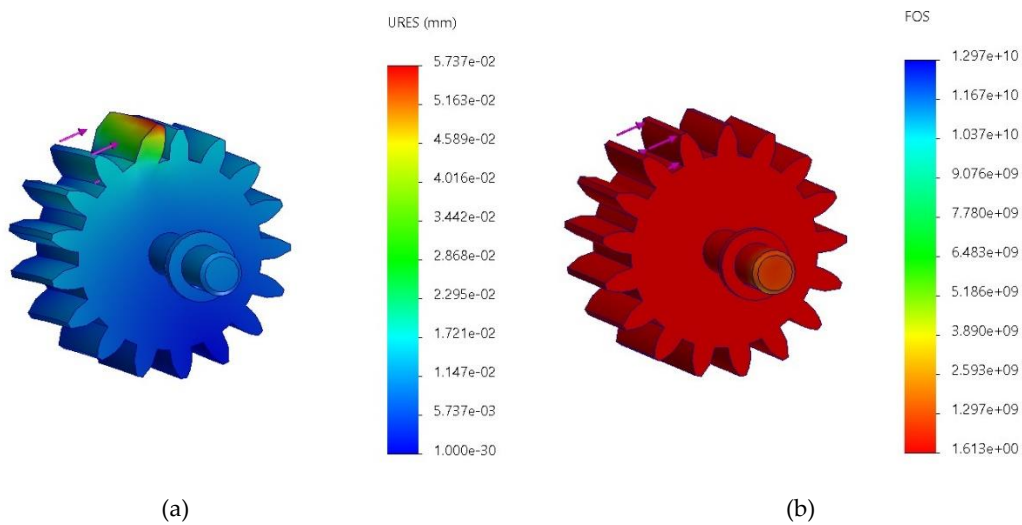


Figure 8. Simulation of sun gear using torque from the motor. (a) The displacement. (b) the factor of safety.

5. Manufacturing Considerations

The production process for the planetary gearbox components must be precise, efficient, and cost-effective. Two primary manufacturing methods were considered: die casting and precision machining using Computer Numerical Control (CNC) machines. Die casting is well-suited for mass production due to its ability to produce complex shapes with high dimensional accuracy and minimal material waste, making it a cost-effective option on a scale. However, it requires a significant upfront investment in mold fabrication, which makes it less suitable for low to medium production volumes [18].

When it comes to achieving high precision, CNC machining is a highly effective manufacturing method. It enables the production of components with excellent dimensional accuracy and tight tolerances. CNC machining is especially suitable for manufacturing critical features such as gear tooth profiles, internal splines, and bore diameters that demand a high level of precision.

While CNC machining can be costly and time-consuming for large-scale production, it offers significant advantages during prototyping and low-volume manufacturing. It allows greater design flexibility and facilitates quick design changes with minimal tooling adjustments, making it ideal for iterative development and customization [19].

CNC manufacturing proves to be both feasible and advantageous under the design constraints of the planetary gearbox. This gearbox demands precise gear meshing and alignment, which requires tight manufacturing tolerances, something CNC machining is well-equipped to achieve. One of the key benefits of CNC technology is its ability to produce complex parts directly from CAD models, eliminating the need for specialized molds. This significantly reduces lead time and lowers initial manufacturing costs. Additionally, for small to medium production runs, such as in electric vehicle prototyping or specialized applications, CNC machining offers a cost-effective solution. It ensures high part quality, enables rapid iteration during the design validation phase, and supports the production of custom components with minimal additional investment. Given these advantages, CNC machining emerges as the most suitable choice for manufacturing the components of the planetary gearbox [20].

6. Discussion

This project focused on the design of a compact and robust high-speed gearbox for a small electric vehicle. Electric vehicles offer significant environmental benefits, primarily due to their zero emissions of harmful gases such as carbon monoxide, carbon dioxide, and nitrogen oxides. The gearbox designs were modeled, tested, and refined using SolidWorks, with the planetary gear system selected for its high torque capacity, compact size, and ability to achieve the required 7.05:1 gear ratio in a single stage. Three different design iterations were developed, varying in material and component thickness. Among the materials considered, Aluminum 7075-T6 emerged as the most suitable choice due to its favorable strength-to-weight ratio and acceptable safety factor. Simulation results confirmed the design's ability to meet torque and speed requirements, validating its feasibility for the intended application. Overall, the proposed gearbox contributes to improving electric vehicle efficiency while maintaining system compactness, durability, and performance reliability.

7. Conclusion and Future Work

In conclusion, the objective of this project was to propose gearbox configuration for small electric vehicles capable of carrying four passengers with a total load of 450 kg. A key goal was to create a design that stands out from conventional gearboxes currently available on the market. While meeting these specific conditions posed several challenges, thorough research, particularly in understanding various gearbox types and their functions, made it possible to arrive at a viable solution. The planetary gearbox was chosen due to its compact size and high-power transmission capability. Critical factors such as performance, manufacturing complexity, noise characteristics, and overall suitability for electric vehicle applications were carefully considered. SolidWorks CAD software played a crucial role in developing and presenting the design. Detailed calculations supported the engineering decisions, including torque, power, force analysis, and factor of safety, all of which were vital to ensuring the reliability of the system.

A major next step in this project involves manufacturing a physical prototype of the finalized gearbox. This would enable experimental validation of the theoretical and simulation-based findings. Real-world testing would provide insight into key performance characteristics such as torque output, efficiency, thermal behavior, noise, and vibration. Moreover, prototyping would reveal practical challenges related to manufacturing and assembly. Identifying and addressing such issues early on would be essential for refining the design and moving toward a production-ready solution.

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