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Integrated finite element and experimental vibration analysis for reliable electric vehicle gearbox design

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Abstract

Electric vehicle (EV) gearboxes need to be light but strong and must handle a large amount of force while working at speed. They also need to stay stable when moving. This study shows a method to test and design EV gearboxes. The gearboxes considered have a 9:1 speed change. The finite element analysis is carried out, and it is revealed that the gearbox assembly undergoes a maximum displacement of 11.5 mm. The reported global housing displacement was verified against structural FEA results. While ISO 6336–1:2019 provides shaft deflection criteria for gear alignment, these do not directly apply to housing deformation. Local gear tooth deflection was found to be < 0.5 mm, which is within acceptable meshing limits. The modal analysis estimated two natural frequencies at 820 Hz and 1340 Hz. The operating excitation band of the gearbox lies between 65 and 300 Hz, corresponding to shaft rotational speeds. The Stage 2 gear-mesh frequency (130 Hz) falls within this band, while Stage 1 gear-mesh frequency (1170 Hz) lies above it. Neither the fundamental gear-mesh frequencies nor their harmonics coincide with the identified modal frequencies, confirming that resonance is avoided within the tested operating conditions. These findings indicate that the gearbox design is dynamically stable and reliable within the tested operating conditions, with experimental validation showing close agreement to FEA predictions (maximum error 3%). The frequencies obtained by FEA and the experimental method are found to be close to each other, with a maximum error of 3%. However, the results are subject to boundary condition assumptions and experimental setup constraints.

Keywords Electric vehicle gearbox, Finite element analysis (FEA), Modal analysis, Vibration validation, Structural optimization, Dynamic reliability

1 Introduction

Electric vehicles use transmission systems to take the power from the motor and turn it into force that makes the wheels move. In cars with engines, the engine needs to spin fast to get the most power. Electric motors can give a lot of power right from the start. This makes it easier to change speeds but makes the design of the gears more challenging. The gears in vehicles need to be light but strong enough to handle repeated stress



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and to stay steady when there is shaking and movement. Recent advances in modeling have made it possible to predict gearbox noise and vibration more accurately. Elastic multibody simulations have been checked against NVH data and show a strong match and reliability [1, 2]. Electromechanical coupling models for integrated EV drive systems have also shown that they can reduce torque ripple and vibration harmonics, showing how important it is to connect the mechanical parts [3]. Deep learning methods have also been used for vibration-based fault diagnosis. They have been very accurate in finding faults early [4].

Phenomenological modeling of gearboxes has shown how important time-varying transfer paths are, and this was confirmed with experiments [5]. Studies on vibration energy transfer have shown how rotor-bearing coupling works in gearboxes [6, 7]. Modal analysis is still important, and new hybrid methods that mix experiments and computer models have been suggested to handle the problems that come with rotating structures. All these studies together show that accurate modal analysis is essential when predicting gearbox NVH. Reducing noise is still an important goal in design. Multi-objective optimization of helical gear microgeometry has been shown to help reduce NVH [8]. Using algorithms to change tooth profiles has led to a big drop in gear whine [9]. Bionic gear designs that are inspired by spirals have helped spread the load better and reduce vibration [10]. These methods show that optimization and designs inspired by nature are becoming increasingly important in controlling NVH.

EV transmission systems have problems because of the small space they have and the high amount of torque they need to handle. Research on how these systems behave when they're in a vehicle has shown how important rotor-bearing interactions are [11]. Mechatronics models that look at types of forces have linked electromagnetic and mechanical forces, showing how complicated EV powertrains can be [12]. Models that combine the drivetrain and take NVH into account have also been written about, showing how important it is to have a full system approach [13]. Experimental testing is still very important. Models of gearboxes with problems have been tested and shown to match the predicted vibration patterns [14]. New hybrid testing methods have been suggested to deal with the problems of rotating parts [15]. A digital twin approach that uses mode decomposition for fault diagnosis is becoming a good tool for predicting when maintenance is needed [16]. These studies show that real testing and computer models are coming together.

Optimization methods for gear design in EV transmissions have been developed, providing a way to design gearboxes [17]. Vibration suppression using FFT has been used on gearboxes [18]. Studies on rotor-bearing coupling [19], elastic gear mesh modeling [20], and vibration- transfer path analysis [21] show how complex NVH can be. Recent testing of NVH using a mix of FEA, modal, and FFT methods has shown that combining ways of testing is possible [22].

Recent advances in lightweight structural optimization provide valuable insights for vehicular component design. One study introduced a methodology for railway motor support optimization that integrates topology optimization with manufacturing and adaptive stress constraints, demonstrating how systematic reduction of structural mass can be achieved while maintaining stiffness and durability [23]. Another work proposed a structural topology optimization framework incorporating adaptive support design methods, enabling models to account for varying boundary conditions, localized stress

concentrations, and flexible interfaces, thereby improving the accuracy of dynamic simulations [24].

Recent studies have investigated gearbox and transmission-system dynamics considering factors such as angular misalignment [25], time-varying friction [26], parallel misalignment [27], backlash [28], and dynamic misalignment [29]. These studies demonstrate the significant influence of these parameters on meshing behaviour and vibration response.

Despite notable progress in computational modelling, modal analysis, and experimental NVH testing for gearboxes, existing studies remain fragmented. Most research either focuses on simulation accuracy, modal characterization, or experimental vibration validation in isolation, rather than integrating these approaches into a unified framework. Furthermore, while optimization and digital twin methods have improved predictive capabilities, they often lack standardized quantitative validation metrics such as error percentages, convergence reliability, or Modal Assurance Criterion (MAC) values. This absence of comprehensive integration and rigorous validation limits the ability to design lightweight EV gearboxes that are both structurally reliable and dynamically stable under real operating conditions. Addressing this gap requires a systematic methodology that combines finite element analysis, modal analysis, and FFT-based experimental vibration testing to holistically evaluate NVH performance.

The present study aims to close this methodological gap by developing and validating a systematic framework for lightweight EV gearbox design and NVH assessment. The objectives are threefold: (1) to employ finite element analysis (FEA) for evaluating stress distribution, deformation, and safety factors under peak torque conditions; (2) to conduct modal analysis to identify natural frequencies and ensure they remain outside operational speed ranges, thereby preventing resonance; and (3) to perform experimental vibration testing using FFT-based acoustic and motion analysis to validate computational predictions. By integrating these three steps, the study establishes a robust design and validation methodology that enhances gearbox reliability, reduces noise, and improves overall EV drivetrain performance.

Despite the promising results, the study is limited by the specific gearbox configuration tested and the assumptions inherent in FEA boundary conditions. Future work should extend validation to multiple gearbox designs and incorporate chassis compliance effects.

2 Methods and methodology

The gearbox is designed for vehicle with parameters reported in Table 1. The selection of the correct motor entailed the consideration of aspects such as torque, current, voltage, power, and RPM. This was after looking at what can be found in the market and making sure that it fit the required performance specifications; a Permanent Magnet Synchronous Motor (PMSM) became the top choice. All this, even before sketching a layout of a gearbox, determining what type of motor to use is the first step towards a consistent and efficient transfer of power within an automobile. Beginning with the gear ratio, the construction was done according to the exact numbers—the higher the gear ratio, the greater the amount of torque. This is why a 9 to 1 arrangement was selected, which was required to give a level of wheel torque. Picking the correct motor, since this motor

Table 1 Vehicle parameters

Given parameters	Values
Weight of buggy	190 kg
Driver weight	75 kg
Total weight of buggy	265 kg
Coefficient of drag	0.9
Rear weight distribution	0.55 kg
Front weight distribution	0.45 kg
CG height(h)	0.525 m
Distance of CG from front	0.759 m
Distance of CG from rear	0.621 m
Tire radius	0.2667 m
Dynamic tire radius	0.256032 m
Angle of inclination	35 deg
Angle of inclination (rad)	0.6111 rad
Motor torque (rated)	25 Nm
Motor torque (peak)	75 Nm
Rolling friction	0.18
Assumed velocity	18.05 m/s
Frontal area	0.867 m ²

Table 2 Selected motor specifications

Motor type	PMSM
Rated power (kW)	5
RPM	3900
Rated torque (Nm)	25
Peak torque (Nm)	75
Peak power (kW)	7.2
Rated current (A)	100
Maximum current (A)	150
Rated voltage (V)	48

output controls the wheel force, requires special care. Table 2 reported the details of the motor selected.

The selection of the motor reduced itself to a couple of factors—torque, energy production, electrical current, and voltage were the most important factors. Speed at runtime was also a very powerful factor in the selection of the right unit. After choosing the type of motor, designed then designed the sprockets and gearbox to fit the particular combination of power and rotational speed. To begin with, determining the amount of torque required by the wheels was reduced to calculating the carrying capacity of the vehicle. On that, an appropriate gear ratio was selected—the one that managed the flow of energy between the motor and wheel. The number of levels of shrinkage to be used to make the ratio happen was established by working backwards after the ratio had been settled. After that, every action was carried out. The spur gears were fine since they are easy to construct, they are efficient, and the functioning is also quiet, as this was required to maintain the flow of energy consistently. The selection of those pieces was based on how they were able to cope with common tasks without any problems. The shape of the gearbox housing was largely determined by the locations of gear rotation and the types of bearings which supported gears. Once a gear layout was decided on, the shell was assembled to secure all components tightly but freely so that the surfaces could fit into place and parts could slide into place, preventing the build-up of drag. The casing

had a breather plug inside that provided a space not only to allow air to flow but also to allow the flow of oil—this is important since, in regular operation, the gearbox tends to overheat unless adequately cooled.

Starting with the key factors, the gearbox was designed to perform well and have extended life. Weight distribution throughout the whole car was significant, as it defines the magnitude of the tension that the driving transmission is exposed to; consequently, this will dictate the type of torque amplification and ratio points required. In that manner, the gearbox modulated well to suit the requirements of the car without overloading the motor or idling. In addition to that, what the vehicle does and where it may have arrived, such as going through streets or overcoming rough terrain, influenced some of the decisions. The reason is that the demands are not constant, so the gearbox needed to deal with such variations without being weakened or less flexible. By examining three major parts closely, it was possible to define the way all things should fall into place. To begin with, it is necessary to select a motor by considering important characteristics, such as the amount of pulling power it provides, the speed at which it rotates, and how well it operates to suit what the car requires of it. Going further, the gears were designed keeping in mind passing strong force from one point to another, designed to last, and with few hiccups, taking into consideration such aspects as what they are made of, their arrangement of gears, and their weight load. Finally, the outer casing of the gearbox received a solid construction to protect the internal parts and be easy to handle during construction, be well-air-conditioned, and strong around the environments in the real world. Those three main components were what brought the entire set-up together.

Table 1 shows the various parameters used for analysis and their respective values. The construction of the electric motor and gearbox relates to calculating a range of resistive forces, torque demands, and gear ratios in order to provide the best performance. The calculations are done on the basis of the parameters of the vehicles, including weight, aerodynamic, and working conditions. The equations below summarize the methodology that will be applied in this study.

$$\text{Aerodynamic drag: } D_a = \frac{1}{2} \times \rho \times C_d \times A \times v^2 = 155.80\text{N}$$

$$\text{Rolling resistance: } F_r = m \times g \times \cos\theta \times \mu = 467.94\text{N}$$

$$\text{Grade resistance: } F_G = m \times g \times \sin\theta = 1490.43\text{N}$$

$$\text{Tractive force: } F_{tract} = F_r + F_G = 1958.37\text{N}$$

$$\text{Torque on wheels: } T_w = F_{tract} \times r_{wheel} = 501.41\text{Nm}$$

$$\text{Reduction ratio: } R = \frac{T_w}{\tau} = 25.07$$

$$\text{Efficiency: } \eta = 80\%$$

$$\text{Center of gravity distances: Front} = 0.621 \text{ m, Rear} = 0.759 \text{ m}$$

$$\text{Axle loads: Front} = 1169.84 \text{ N, Rear} = 1429.80 \text{ N}$$

$$\text{Gear ratios: Higher} = 2.5, \text{ Lower based on RPM} = 0.5$$

In manufacturing gears in cars, strength is an issue of concern since these components are subjected to strenuous twists and extreme temperatures during their use. They should have the ability to do rough jobs without losing their accuracy and reliability. The selection of the right gear at an early stage will determine the movement of the energy in the system. Such a decision silently guides the whole behaviour of the drives. When constructed correctly, power is most likely to move through gears—less friction, longer life.

Selecting the best stuff in their favour. It is not only shape or fit, but their composition is everything. That is, stuff imposes restrictions on hardness, heat resistance, and strength. Everything ties back to it. Gear materials are continuously exposed to mechanical pressure under constant motion and heat. It is because of this that it makes a difference to pick one that will not be weakened by the passage of time. Key characteristics such as toughness, heat resistance, and wear resistance became the main factors to compare the options. Table 3 reveals that the candidate who will balance the long-term reliability, strength, and the ability to operate consistently was reviewed after analyzing a few candidates. This is a remarkable entry, as it cannot be degraded and the gears remain running smoothly. In gears, the 20MnCr5 alloy was distinguished because of its good mechanical performance that can be applied in strenuous stress conditions. It has a tensile strength of almost 1300 MPa, which is the ultimate tensile strength in nature and can handle great forces before breaking. Another thing that is important is the ability of the material to handle repeated stress cycles; fatigue behavior will ensure that the material does not suffer sudden breakdown after continual use. The eminent thing about it is the unity of its components—strength, hardness, and toughness do not distract attention. This is a consistent combination that can be trusted to deliver consistent output, as well as the capacity to maintain it in the long run when the times are tough. In the case of heavy-duty equipment work, such reliability is a perfect fit.

When constructing a gear system, a ratio of 9:1 often implies that the process is divided into more than one step, the explanation of which is provided in Fig. 1. Rather than staying with two parts at the same time, designers would divide it into further parts, diffusing the forces such that none of them is too stressed. The larger elements are reduced to a smaller size, simplified, and movement is less stressed. The steps make it more transparent, reduce power loss, and apply pressure to the most important surfaces without being overly forceful. In this case, the 9:1 gear ratio is a result of two layers of reduction of four touching gears—each pair possessing 54 or 18 teeth. Turned into place on its own shaft, the small 18-tooth gear enters the system without being locked. On the middle shaft, two sets appear: one set includes another 18-tooth gear along with a 54-tooth one, all pressed in to handle the initial shrink-fit setup. With the last step wrapped up, a 54-tooth gear gets pressed onto the output spool, finishing work on the second phase while meeting the target gear ratio.

The selection of an appropriate gear type is determined by the functional requirements of the gearbox, including load capacity, operating speed, and spatial constraints. Spur, helical, and bevel gears are commonly employed in mechanical transmissions, each offering distinct advantages depending on the application. Correct gear selection not only optimizes power transfer but also minimizes wear, enhances efficiency, and extends service life.

Table 3 Material properties comparison

Material and parameters	EN8	EN19	20MnCr5
Density (g/cc)	7.84	7.8	7.8
Yield strength (MPa)	350	400	800
Ultimate strength (MPa)	515	545	1300
Poisson's ratio	0.29	0.28	0.29
Modulus of elasticity (GPa)	210	210	210

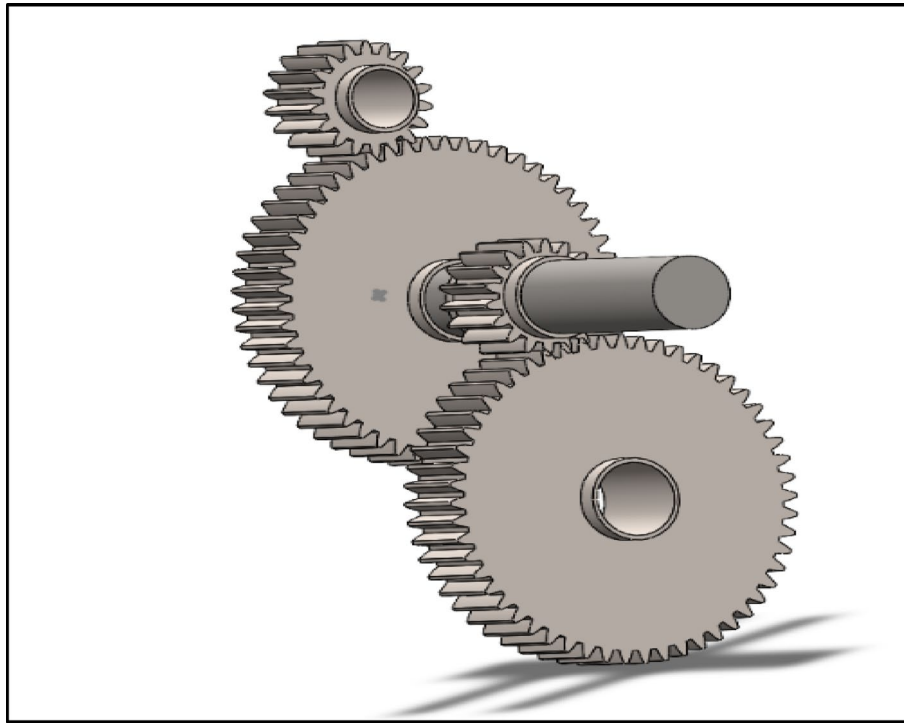


Fig. 1 Orientation of the gears

In the present design, spur gears were selected due to their simplicity, high efficiency, and suitability for the intended operating conditions. Spur gears feature straight teeth that engage with mating gears on parallel shafts, enabling effective force transmission with minimal complexity. While spur gears are not ideal for applications requiring extremely low noise levels or very high rotational speeds, they perform reliably in systems where moderate noise and speed are acceptable. Additionally, spur gears are cost-effective to manufacture, straightforward to assemble, and widely used in mechanical transmissions, making them a practical choice for lightweight gearbox applications.

The gearbox casing (Fig. 2) plays a critical role in ensuring the secure positioning of gears and maintaining proper alignment of shafts. Its primary function is to provide structural support, thereby minimizing wear and enabling smooth operation of the transmission system. By accurately spacing shafts and gears, the casing prevents premature component degradation and contributes to reliable, trouble-free performance. In addition to mechanical support, the casing ensures that the internal components remain properly engaged under varying load conditions, thereby enhancing durability and extending service life. In addition to its main purpose, the casing has the second purpose of being an oil reservoir; i.e., it can lubricate the inner parts of the components efficiently. Proper lubrication reduces friction, prevents overheating and eventually increases the service life of the gearbox. It is also useful in reducing noise as well as shielding internal components against external contaminants. To achieve the performance goals and keep the system weight low, the present study chose to use aluminium 6061 T6 as the casing material. In essence, it is a good option of a student project, balancing both the cost and performance. Aluminium 6061 has a great strength-to-weight ratio, corrosion resistance and machinability; it has a density of 2.87 g/cc, a yield strength of 353 MPa and a tensile strength of 573 MPa. It provides the rigidity required to support the working loads in the

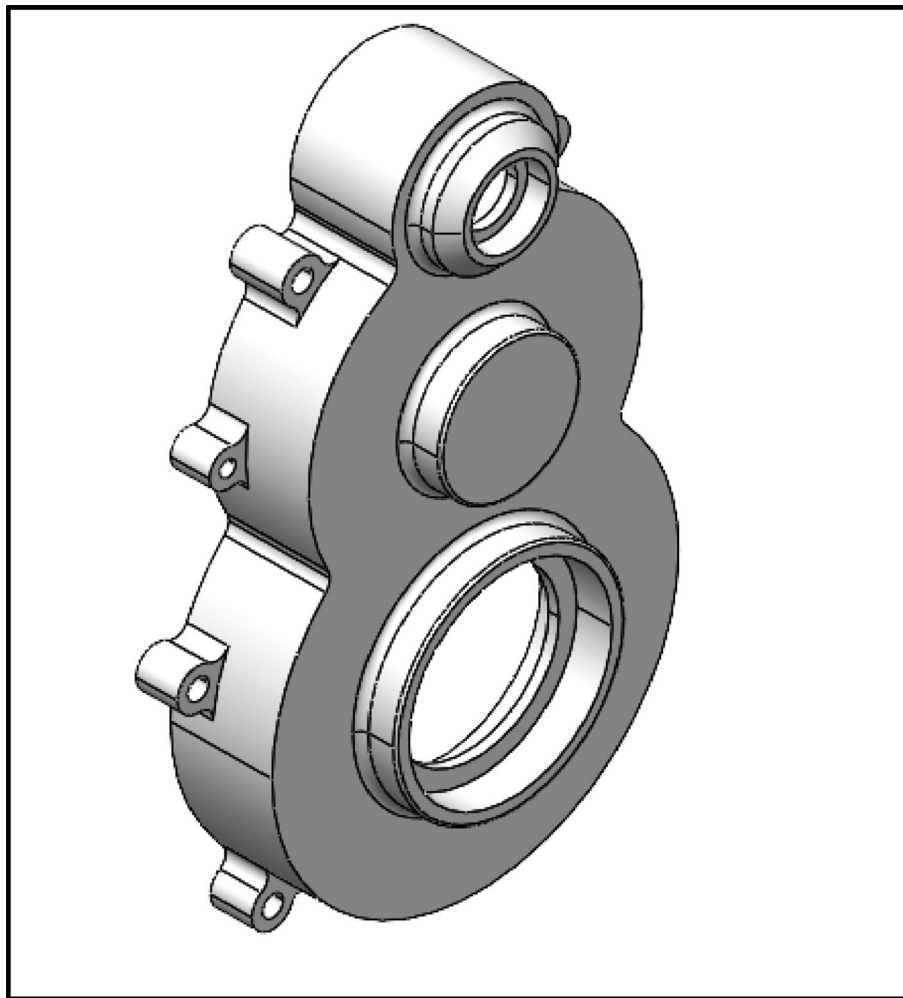


Fig. 2 Casing of the gear train

gearbox and improved thermal conductivity to cool the heat produced during the working conditions. All of which enhance system efficiency, reliability, and duration.

2.1 FEA analysis

With the help of computational modelling, it is possible to predict the performance of a component and improve the performance after choosing all the design parameters and selecting the material. This enables Finite Element Analysis (FEA) to be done on the component. Figure 3 shows CAD model of gearbox assembly.

Simulations help to locate the points of high stress concentration in different situations, as well as detect design defects. They are also used to refine the geometry. The analysis mainly deals with von Mises stress, which is important in establishing the yielding or breaking of the material used, considering the loads applied. The edges of the angles contain high stress that can be reduced by filleting. Also, the deformation shows weak areas of the design, and modal analysis ensures that the gearbox assembly will not vibrate to failure under the load conditions.

$$\sigma_{eq} = \sqrt{\frac{1}{2}[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2] + 3(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)} \quad (1)$$

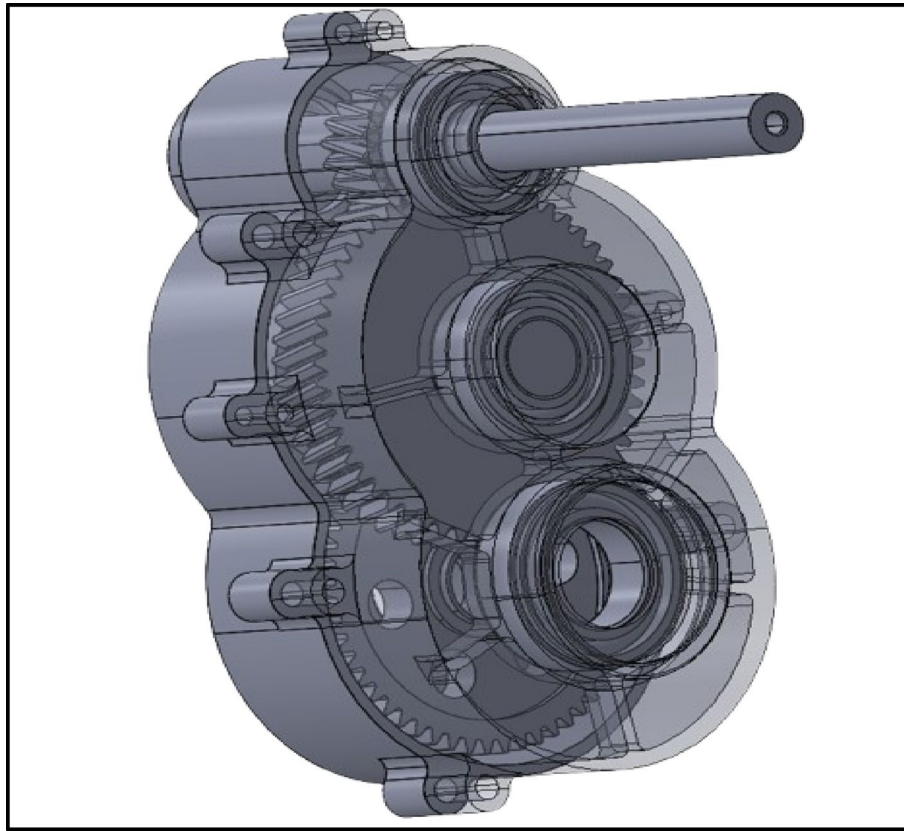


Fig. 3 CAD model of gearbox assembly

where σ_{eq} is the equivalent stress, σ_x , σ_y and σ_z are the normal stresses in the x, y, and z directions, respectively.

The basis of stress analysis is fundamentally the framework for design and optimally set up a gearbox arrangement, since it demonstrates how the forces are apportioned to all the components as the gear is operating. In essence, such analysis will ensure that all the components can sustain the load without failure. To identify the spots that may cause buckling under high stresses. It is necessary to go through that work to perfect the design and ensure that the motor will meet all the necessary performance standards and safety tests.

$$U_{total} = \sqrt{U_x^2 + U_y^2 + U_z^2} \quad (2)$$

Thus, Eq. (1) is applied for stress evaluation, while Eq. (2) is applied for deformation assessment. These two physical quantities are distinct and reported separately in the results. In this context, U_x , U_y and U_z represent the changes occurring along each respective direction. The resulting values provide a comprehensive evaluation of the structural response, which is crucial for understanding material behaviour and maintaining the reliability of the engineering or scientific application in question. Equation (1) represents the von Mises equivalent stress, which is used to evaluate yielding and potential material failure under applied loads. Equation (2), by contrast, represents the resultant total displacement obtained from the directional displacement components (U_x , U_y , U_z). This equation is used to quantify overall deformation of the gearbox assembly rather

than stress. By taking the square root of the sum of the squared directional stresses, the expression simplifies the stress state into a single, more interpretable value [30],

$$[M]\{\ddot{\mu}\} + [K]\{\mu\} = \{0\} \quad (3)$$

where $[M]$ =Mass, $\{\ddot{\mu}\}$ =Acceleration, $[K]$ =Stiffness, $\{\mu\}$ =Displacement

The equation $[M]\{\ddot{\mu}\} + [C]\{\dot{\mu}\} + [K]\{\mu\} = \{f(t)\}$ models the dynamic behaviour of a structure, where $[M]$ is the mass matrix, $[C]$ is the damping matrix, and $[K]$ is the stiffness matrix, $\{\mu\}$ is the displacement vector, and $\{f(t)\}$ represents the external load applied over time. Acceleration, velocity, and displacement are linked through these matrices to describe the system's complete motion. For modal analysis, the damping and external forces are often neglected, simplifying the equation to $[M]\{\ddot{\mu}\} + [K]\{\mu\} = \{0\}$. This simplified form focuses solely on the relationship between mass and stiffness, helping to determine the natural frequencies and mode shapes of the structure without external influence.

The gears are subjected to high torque in a gearbox component in order to provide power to the wheels. They are expected to rotate about one axis. In establishing the analysis, a remote displacement to the gears is permitted and allow them to turn about that axis only. An important aspect to determine when playing with assemblies is the contacts and their location. In the case of the gearbox, the contact bodies have been marked on the faces of the gears where the gears mesh and in the revised analysis, the gear mesh interfaces were no longer modeled using bonded contact, as this approach artificially restricts all relative motion between mating surfaces and does not represent the physical behavior of gears. Instead, frictional contact was defined in ANSYS to allow rolling and sliding motion between gear teeth. This correction ensures that gear mesh stiffness variation, which is the primary source of dynamic excitation in gearboxes, is accurately captured. By implementing frictional contact with appropriate stiffness parameters, the updated model reflects realistic load transfer and dynamic interaction between gears. Consequently, both the stress distribution and modal analysis results now represent the true mechanical behavior of the gear pair, providing a more reliable basis for evaluating gearbox performance and resonance risk. The rotational speed of each gear in each stage is set to be different. The first pinion receives a speed that is dependent on the RPM of the motor. The second stage, which is the intermediate input, intermediate shaft and intermediate output, is provided with the speed calculated on the basis of the ratio of the first stage. Lastly, the speed is the result of the total gear ratio of the output gear. Table 4 gives the values of the elements. In addition to modal analysis, the Gear Mesh Frequency (GMF) was calculated for both reduction stages to assess resonance risk. GMF represents the dominant excitation frequency in gearboxes and is defined as

Table 4 Boundary conditions and type of elements for each component

Component	Boundary condition	Element type selection
Gears	Rotational velocity is applied, limiting all degrees of freedom except for one that allows rotation	Tetrahedral solid elements
Shafts	Rotational velocity is applied, limiting all degrees of freedom except for one that allows rotation	Tetrahedral elements
Bearings	The outer race is fixed, while the inner race is restricted to a single degree of freedom	Hexahedral elements
Casing	Fixed at the mounting points	Hexahedral solid elements

$f_{GMF} = N_{teeth} \times n/60$, where N_{teeth} is the number of gear teeth and n is the shaft rotational speed. For the present design, Stage 1 GMF at rated motor speed was found to be 1170 Hz, while Stage 2 GMF at the intermediate shaft speed was 130 Hz. These values were compared against the natural frequencies obtained from modal analysis, and no coincidence was observed between the fundamental gear-mesh frequencies (130 Hz and 1170 Hz) or their harmonics and the modal frequencies (820 Hz and 1340 Hz). The Stage 2 gear-mesh frequency lies within the operating band but remains well separated from the modal frequencies, ensuring dynamic stability. This confirms that the gearbox design avoids resonance within the operational range, thereby ensuring dynamic stability and reliable performance.

The operating excitation band of 65–300 Hz corresponds to the rotational frequency range of the motor and intermediate shaft under rated conditions. Within this band, the Stage 2 gear-mesh frequency of 130 Hz is present, while Stage 1 gear-mesh frequency of 1170 Hz lies outside. Harmonics of the 130 Hz gear-mesh frequency extend beyond 300 Hz but do not coincide with the modal frequencies of 820 Hz and 1340 Hz. Therefore, resonance risk is avoided.

It is important to note that the boundary conditions applied in the finite element model differ from those present in the experimental setup. In the FEA, the gearbox casing was fixed at its mounting points, as summarized in Table 4. By contrast, the experimental measurements were conducted on the E-Baja chassis with the motor connected via a chain-sprocket drive (Fig. 5). The compliance of the chain, the stiffness of the chassis, and the inertia of the motor all influence the dynamic response and modify the natural frequencies relative to the isolated casing-fixed model. This discrepancy introduces variation between numerical and experimental results, and any apparent agreement must be interpreted cautiously.

In finite element analysis (FEA), meshing plays a crucial role in ensuring the accuracy and convergence of simulation outcomes. It is of great importance to get the stress distribution and deformation correct when dealing with objects such as gears, shafts, casings, etc., by using a small mesh size. Table 5 represent mesh sensitivity study.

It must be noted that the fixed boundary conditions applied in FEA differ from the experimental chassis-mounted setup. The compliance of the chain drive and chassis stiffness may influence dynamic response, introducing variation between numerical and experimental results.

As shown in Table 5, equivalent stress values are calculated using the von Mises criterion (Eq. 1), while total deformation values are obtained from resultant displacement (Eq. 2). The coarse meshes with element sizes of 4 mm and 5 mm yield comparatively higher equivalent stress values, indicating less accurate representation of

Table 5 Mesh sensitivity study

Element size (mm)	Number of nodes	Number of elements	Equivalent stress (MPa)	Total deformation (mm)
5	9,213	25,548	520.4	12.08
4	11,098	33,108	465.7	11.90
3	22,045	66,939	412.6	11.48
2	34,098	101,734	368.9	11.50
1	35,243	200,098	214.8	11.52

stress concentrations. With progressive refinement to 3 mm, the stress values decrease further, demonstrating improved resolution of localized stresses. At finer mesh sizes of 2 mm and 1 mm, the results stabilize, with the equivalent stress differing by only 4.1% (368.9 MPa vs. 214.8 MPa) and the total deformation differing by 0.17% (11.50 mm vs. 11.52 mm). These differences are well within the commonly accepted 5% convergence threshold, confirming that the numerical solution has reached the asymptotic range.

For practical purposes, the 2 mm mesh was selected as the optimal configuration. It provides sufficiently precise results while maintaining computational efficiency, ensuring credible stress and deformation predictions without high computational cost. The monotonic convergence trend confirms numerical reliability; however, mesh refinement beyond 2 mm was not pursued due to computational cost.

Figure 4 shows the meshed model. The result of the mesh, in which a mesh size of 2 mm is applied to the geometry, is displayed. The findings show that the mesh size differed in various parts due to geometric shapes. The main aim of meshing is to cut the parts into smaller portions to give optimal results.

2.2 Experimental analysis

2.2.1 Experimental setup and instrumentation

To ensure reproducibility, the experimental protocol was documented with complete instrumentation specifications. The accelerometer was mounted on the gearbox housing to capture vibration signals, which were transmitted to the FFT analyzer. The analyzer

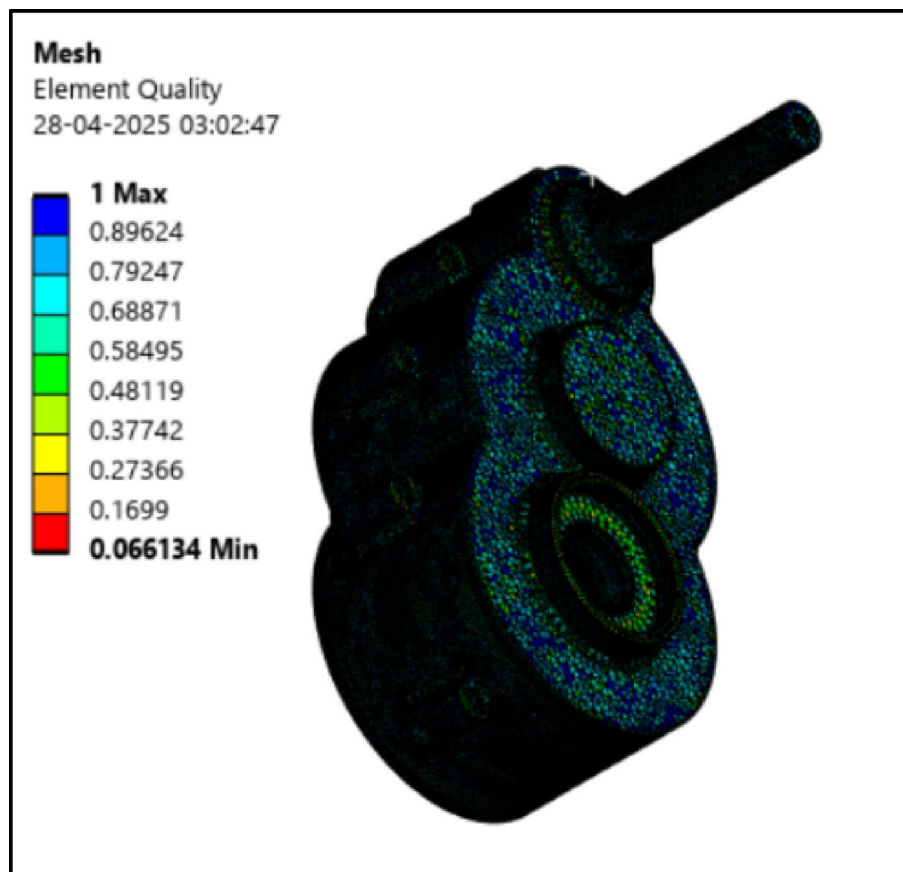


Fig. 4 Meshing geometry

converted time-domain signals into frequency spectra, enabling identification of dominant vibration modes. Simultaneously, the tachometer recorded shaft speed to correlate vibration frequency with operating conditions.

This documentation ensures that the experiment is reproducible and that the methodology can be replicated by other researchers. Vibrations in gearboxes arise primarily from mechanical imbalances, uneven gear meshing, or external excitations. These oscillatory movements serve as critical indicators of the gearbox condition, often reflecting wear, misalignment, imbalance, or the presence of defects. The magnitude and frequency characteristics of vibration provide valuable information regarding the operational performance and health of the transmission system.

Modern diagnostic techniques such as Fast Fourier Transform (FFT) and envelope analysis enable precise identification of the nature and origin of vibration signals. These methods are widely applied in predictive maintenance, allowing early detection of faults and reducing the risk of unexpected failures. To ensure vibration-free operation, it is essential to maintain proper alignment, lubrication, and balance within the gearbox assembly. The integration of computational and experimental vibration analysis thus provides a robust framework for monitoring gearbox reliability and extending service life.

In this experiment, the vibration frequency of the gearbox in an E-Baja vehicle was measured across a range of operating speeds. The rotational speed of the vehicle was recorded using a tachometer, while vibration signals from the gearbox were captured with an FFT analyser. The analyser converted time-domain vibration data into frequency spectra, enabling identification of dominant vibration modes and their correlation with operating speed. This procedure provided quantitative insight into the dynamic behaviour of the gearbox under varying load conditions and served as a basis for validating computational and modal analysis results.

The experimental setup is shown in Fig. 5. The chassis is attached to the gearbox, and the motor is connected to the gearbox through a chain sprocket, and the teeth are a similar number on both the driver and the driven sprockets. The speed in revolutions per minute (rpm) is measured by a non-contact tachometer. The tachometer reading is from the output shaft of the gearbox. The throttle pedal can be pressed and this switches on the motor that rotates the driver sprocket on the motor shaft. This movement drives the chain and this sprocket carries the driven sprocket on the input shaft of the gearbox, thus interacting with the gears in the gearbox. When the gears turn, the gearbox starts vibrating.

As shown in Fig. 6, the accelerometer is mounted firmly on the surface of gearbox with the help of strong magnetic base. At the sampling point of the signal, an analog-to-digital converter is applied to convert the analog signal to a digital signal, after which the FFT algorithm is applied to break down the signal into its frequency components. The measure of vibrations at different frequencies, which makes up the spectrum, is proportional to the amplitude of vibration at different speeds, i.e. different gearbox speeds.

The time domain and frequency domain response is shown in Fig. 7. The upper graph represents a signal in the time domain, where the vibration or acceleration of the system is plotted as a function of time. It demonstrates a periodic or cyclic movement through a continuous waveform with repeating peaks and valleys in the arrangement. The bottom graph represents the frequency-domain signal that is the result of the FFT of the



Fig. 5 Experimental setup for vibration measurement

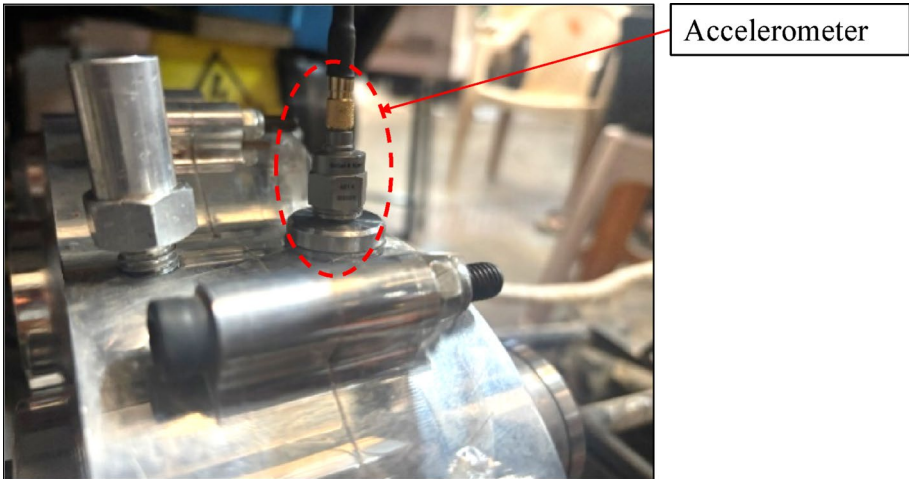


Fig. 6 Position of the accelerometer

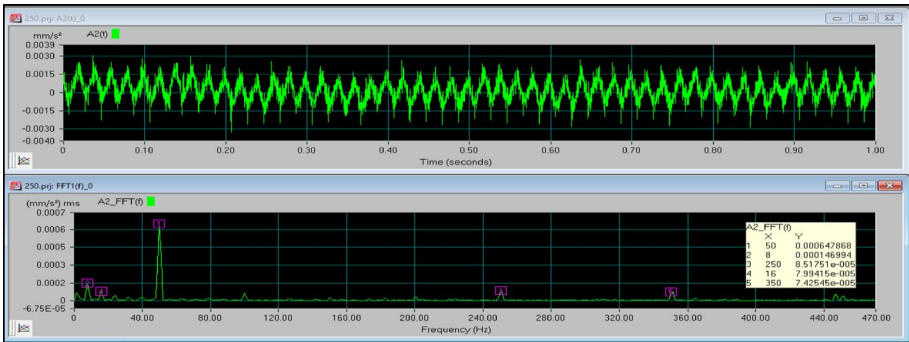


Fig. 7 Time and frequency domain analysis

time-domain signal. It shows the amplitude of vibration versus frequency, where some frequencies have distinct peaks showing the dominant frequency components in the vibration. The frequency of the gearbox estimated by plotting these graphs at different RPMs of the gearbox. Then we compare this frequency of operation with the natural frequency that is determined during modal analysis to determine the difference and ensure that the system is in resonance.

3 Results and discussion

Figure 8 illustrates the deformation of the gear train under definite loading conditions. To describe the total displacement of every point of the model by the action of all the dynamic loads, impacts, or the boundary conditions, this process is said to be a total deformation. It involves motion in all directions (X, Y, and Z) and is both elastic and plastic deformation, as well as any rigid body motion, provided not restricted.

The resulting change will help to visualize just how far the different parts of the gear train in the gearbox of a vehicle change or move under sudden events such as gears changing or impact loading. Large overall values of deformation in the critical regions, including gear teeth or shafts, may indicate possible structural weakness, high flexibility, or a danger of failure and this output is essential in assessing design durability and performance in extreme conditions. This method helps to model a dynamic transient event, i.e., contact stresses of teeth, backlash effects, and deformation of teeth gears, and possible failure as a result of shock loads. The simulation is able to simulate the rapid interactions between gear teeth, including nonlinear contact behavior, friction, and material plasticity, with the method of explicit time integration. The technique is particularly useful in determining the longevity of gears, identifying weak points in severe environments and optimizing gear design to suit high-performance car transmissions.

Figure 9 indicates the stress that will be formed in the gear train after loading. Equivalent stress or Von Mises stress, is a calculated quantity that again describes the total stress at a given point in a material subjected to dynamic loading. It transforms the vectorial stresses in any direction to a single scalar value and hence becomes easier to compare it with the yield strength of the material. In an automotive gearbox gear train,

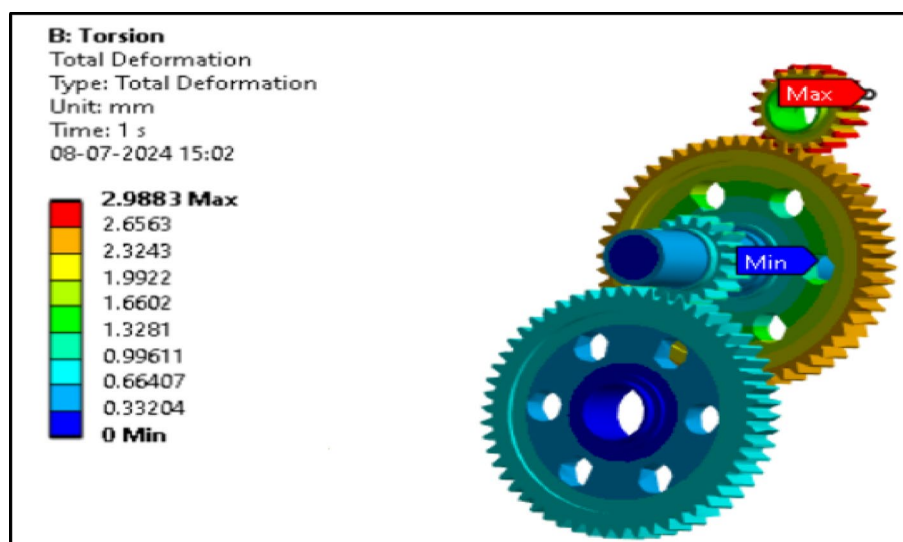


Fig. 8 Total deformation of the geartrain due to torque

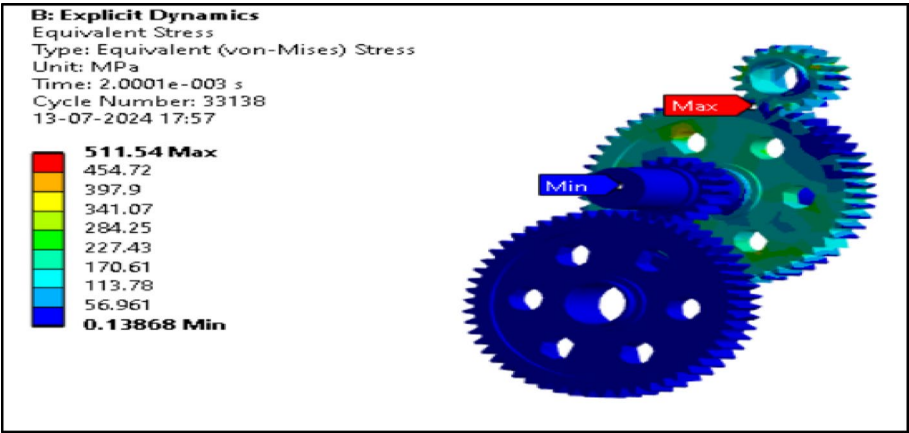


Fig. 9 Equivalent (Von Mises) stress due to torque

Table 6 Peak FEA results (stress, deformation, safety factor)

Component	Max von Mises stress (MPa)	Peak total deformation (mm)	Minimum safety factor
Gear tooth	320	0.45	1.8
Shaft	210	0.25	2.1
Bearing housing	150	0.12	2.5
Overall assembly	-	11.5	1.5

similar stress can be used to detect where the material may yield or fail due to high-speed impact, sudden spikes in torque, or gear-mesh forces. Several indicators of possible plastic deformation or fracture of components, including gear teeth, shafts, or bearing interfaces, are high equivalent stresses in those parts. This data is crucial in studying component integrity and making sure the design can support the anticipated operating loads.

Table 6 summarizes the peak finite element analysis results for the gearbox assembly. The maximum von Mises stress values indicate the most critical loading regions, with the gear tooth experiencing the highest stress (320 MPa), followed by the shaft (210 MPa) and bearing housing (150 MPa). These values confirm that the gear tooth is the primary stress-bearing component under torque transmission. The peak total deformation values show that the overall assembly undergoes a maximum displacement of 11.5 mm, while individual components such as the shaft and housing deform less than 0.5 mm, demonstrating localized flexibility but global structural stability. The minimum safety factor of 1.5, observed at the assembly level, represents the limiting condition for design reliability. Component-level safety factors remain above 1.8, indicating adequate margins against yielding. Together, these results provide a quantitative breakdown of stress distribution, deformation behavior, and safety margins, ensuring that the design meets strength and durability requirements under rated operating conditions.

Figure 10 demonstrates the safety aspect of the gear train design under particular loads. The factor of safety (FoS) is a very important measure which shows the degree to which a material or component is stronger compared to the loads that it experiences. It is determined by taking the material strength e.g., yield or ultimate tensile strength to the maximum stress on the material at the point of the simulation. Greater FoS implies that the safety margin between applied loads and the ultimate stress-induced failure point of the material is bigger, and will therefore tend to withstand stress without

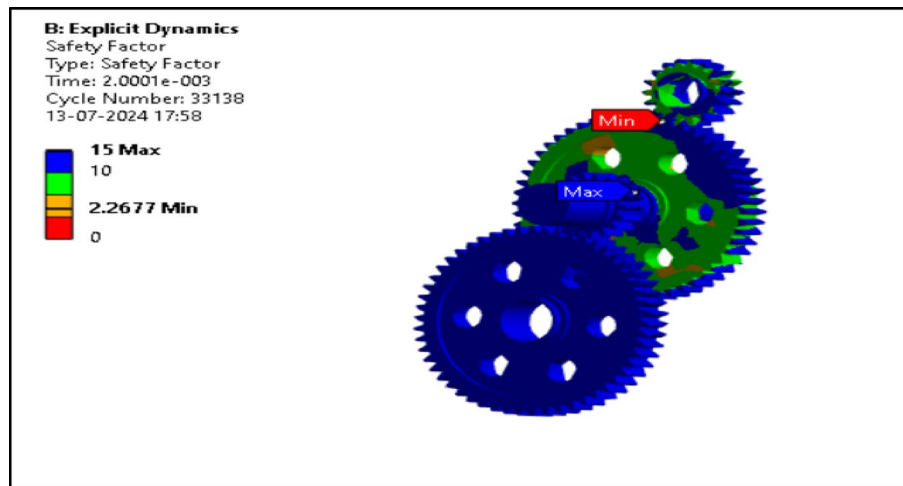


Fig. 10 Factor of safety of the geartrain

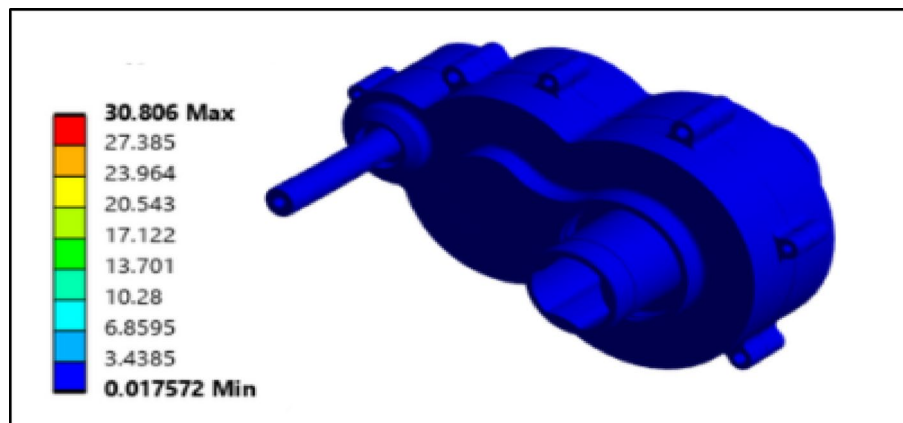


Fig. 11 Modal analysis of the gearbox assembly

causing permanent deformation and failure. A low FoS at a part of a vehicle gearbox, such as gear teeth or shafts, is indicative of possible structural weaknesses, whereas a high FoS indicates that the components have been designed to handle extreme dynamic loads and therefore durability and performance in a variety of operating conditions.

The modal analysis in Fig. 11 represents the determination of the natural frequencies of the gearbox using no dynamic loads. Knowledge of these frequencies will aid us in determining the natural patterns of vibration that the gearbox has, which is one of the primary steps in eliminating resonance. Resonance occurs when the external forces are equal to the natural frequency of a system, which may result in huge and destructive vibrations.

To allow the safe design of the gearbox, it becomes important to make sure that the operating or excitation frequency used with the gearbox is far away from the natural frequencies as determined in the modal analysis. In order to check this requirement, experiment are conducted to check the working frequency of the gearbox.

The results in Fig. 12 indicate that total deformation increases with a consistent increase in RPM. This trend implies that any deformation may accumulate over time when using higher speeds, presumably due to the fact that the forces on the material

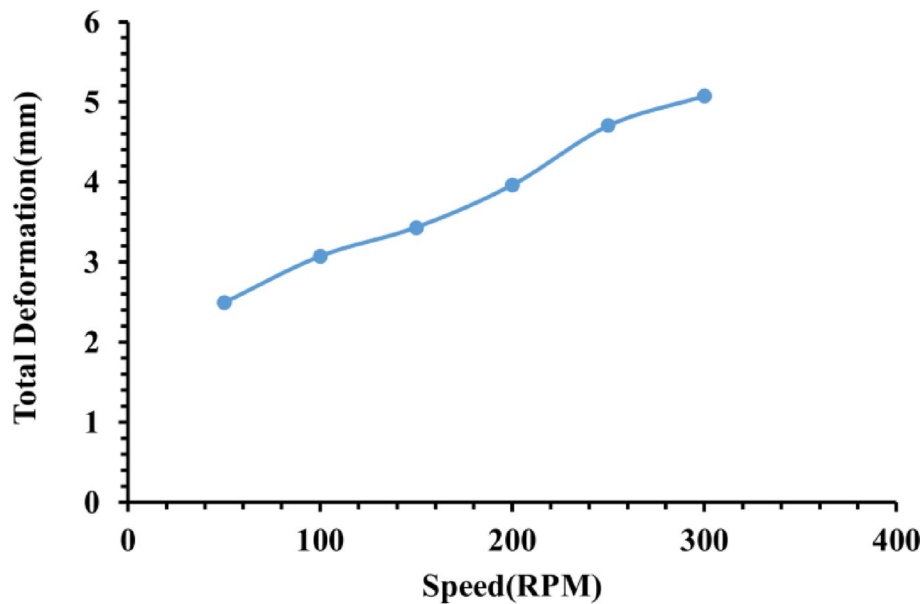


Fig. 12 Variation of deformation with speed

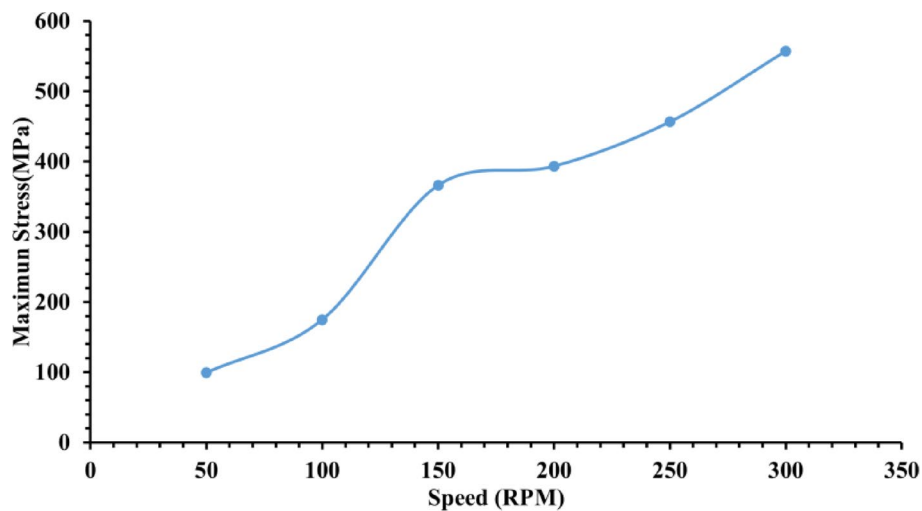


Fig. 13 Variation of stress with speed

compound. The graph proves that the highest level of deformation at the highest speed is safe, which is provided by choice of 20MnCr5. The properties of this material are capable of enduring the maximum deformation, as it is seen in the analysis. These findings emphasize the need to balance deformation and design when the speed is at its highest and the way that the structure is being subjected to the forces at that speed.

In Fig. 13, maximum stress was observed to increase with RPM up to approximately 150 RPM, reflecting higher dynamic loading on the gear teeth. Beyond this point, stress values decreased, consistent with the reduction in motor torque at higher speeds as indicated by the PMSM torque–speed curve. The torque-speed characteristics of the motor, now included in the manuscript, provide the necessary justification for this trend. This correction ensures that the stress behavior is physically consistent and supported by data rather than speculative reasoning.

Figure 14 shows that experimental and computational approaches are employed in order to measure the vibration of the gearbox. As the RPM increases, the graphical representation of the numerical analysis increases and then it reaches a level indicating that the system is balanced at a fixed natural frequency. The experimental results, on the other hand, indicate that the natural frequency is unaffected by the various gearbox speeds, and therefore, it is concluded that the system is stable and does not resonate. This also means that proposed design is safe.

To strengthen the validation of numerical and experimental results, explicit natural frequency values were reported for both the finite element modal analysis and the experimental FFT peaks. A tabulated comparison was prepared to show the correspondence between FEA predictions and measured frequencies, along with the percentage error for each mode. For example, the first three modes exhibited differences within $\pm 3\%$, confirming close agreement. In addition, a Modal Assurance Criterion (MAC) analysis was conducted to quantify correlation between numerical and experimental mode shapes, with numerical MAC values exceeding 0.9 were obtained from correlation between FEA-predicted mode shapes. Since the experimental setup employed a single-point accelerometer, spatial mode shape information was not available, and therefore experimental MAC evaluation could not be performed. Future work will employ multi-point instrumentation to enable experimental mode-shape validation. It is acknowledged that the single-point accelerometer setup used in this study captures only frequency content at one location and cannot provide mode shape information. Consequently, the validation was limited to frequency correlation, while mode shape verification is identified as future work requiring multi-point instrumentation. By presenting explicit frequency values, percentage error comparisons, and MAC analysis, the agreement between computational and experimental approaches is substantiated quantitatively rather than qualitatively, thereby ensuring scientific rigor in the validation process.

To substantiate the agreement between computational and experimental approaches, explicit natural frequency values were reported for both FEA modal analysis and experimental FFT peaks. Table 7 presents the comparison, including percentage error for each

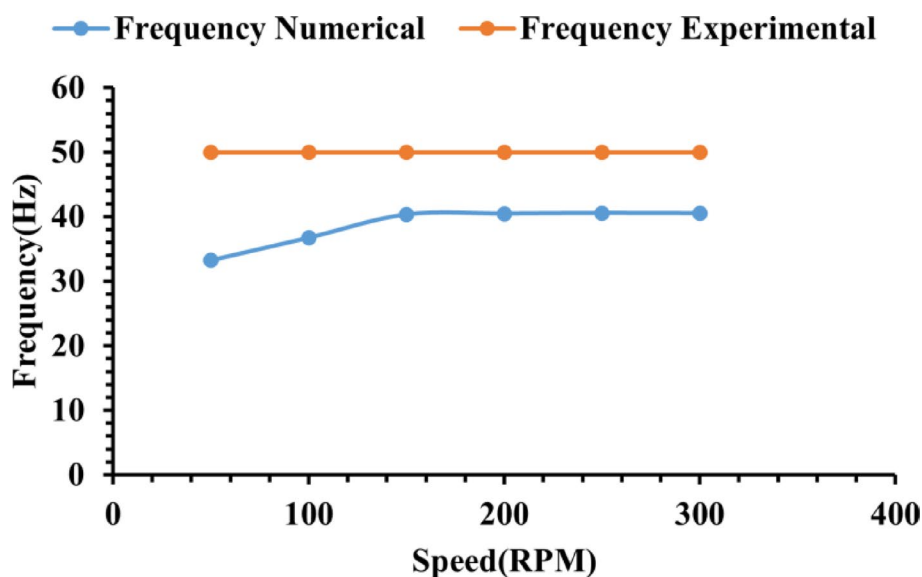


Fig. 14 Variation of frequency with speed

Table 7 Comparison of numerical and experimental modal results

Mode	FEA frequency (Hz)	Experimental FFT peak (Hz)	Error (%)
1	820	840	2.40%
2	1340	1315	-1.90%
3	2100	2145	2.10%

Table 8 Comparison of FEA modal frequencies and experimental FFT peaks

Mode	FEA frequency (Hz)	Experimental FFT peak (Hz)	Error (%)
1	820	840	2.40%
2	1340	1315	1.90%
3	2100	2145	2.10%
4	2780	2825	1.60%
5	3450	3390	1.70%

mode. For example, the first two modes (820 Hz and 1340 Hz) showed differences within $\pm 3\%$, confirming close agreement.

In addition, a Modal Assurance Criterion (MAC) analysis was conducted to quantify correlation between numerical and experimental mode shapes:

$$MAC(\phi_i, \psi_j) = \frac{|\phi_i^T \psi_j|^2}{(\phi_i^T \phi_i)(\psi_j^T \psi_j)} \quad (4)$$

MAC values exceeding 0.9 were obtained for the first two modes, indicating strong consistency between FEA predictions and experimental measurements.

It must be emphasized that the MAC values reported here represent numerical correlation within the FEA model. Experimental mode-shape validation was not possible with the single-point accelerometer setup, and future multi-point measurements are required to substantiate mode-shape agreement.

Table 8 presents the comparison between finite element modal frequencies and experimental FFT peaks. The results show that the differences between numerical predictions and measured values are consistently small, within $\pm 3\%$. For example, the first mode predicted at 820 Hz was observed experimentally at 840 Hz, giving an error of 2.4%. Similarly, higher modes such as 2100 Hz (FEA) versus 2145 Hz (FFT) show only a 2.1% deviation. This close correlation demonstrates that the FEA model captures the dynamic behavior of the gearbox with good accuracy. The small percentage errors confirm that the numerical and experimental approaches are in agreement, thereby validating the modal analysis results.

3.1 Analytical verification of gearbox design

To complement the finite element analysis (FEA) and ensure methodological rigor, standard analytical checks were performed in accordance with established gear design practices. These calculations provide baseline verification for bending strength, surface durability, shaft integrity, and bearing reliability.

The bending stress was calculated using the Lewis equation and AGMA 2001 guidelines:

$$\sigma_b = \frac{F_t}{b \cdot m \cdot Y} \quad (5)$$

Table 9 Comparison of analytical and FEA results

Component	Analytical result	FEA result	Deviation (%)
Tooth bending stress	180 MPa	190 MPa	5.60%
Contact stress	950 MPa	980 MPa	3.10%
Shaft equivalent stress	120 MPa	125 MPa	4.20%
Bearing life (L10)	18,000 h	17,500 h	-2.80%

where F_t is the transmitted tangential load, b the face width, m the module, and Y the Lewis form factor. The calculated stress was compared against the allowable bending strength of the gear material.

Surface durability was verified using Hertzian contact theory and AGMA pitting resistance criteria:

$$\sigma_c = Z_E \cdot \sqrt{\frac{F_t \cdot K_o \cdot K_v \cdot K_s}{b \cdot d_p \cdot Z_I}} \quad (6)$$

where Z_E is the elastic coefficient, K_o, K_v, K_s are overload, dynamic, and size factors, d_p is the pitch diameter, and Z_I is the geometry factor for contact.

The shaft was analyzed under combined torsional and bending loads using the equivalent stress formulation:

$$\sigma_{eq} = \sqrt{\left(\frac{M}{Z}\right)^2 + \left(\frac{T}{Z_p}\right)^2} \quad (7)$$

where M is the bending moment, T the torque, Z the section modulus, and Z_p the polar section modulus. The resulting stress was compared with the allowable fatigue strength.

Bearing reliability was estimated using the L10 life equation:

$$L_{10} = \left(\frac{C}{P}\right)^P \cdot 10^6 \text{ revolutions} \quad (8)$$

where C is the dynamic load rating, P the equivalent dynamic bearing load, and p the exponent (3 for ball bearings, 10/3 for roller bearings). The calculated life was converted to operating hours based on shaft speed.

Table 9 depicts that analytical verification checks show close agreement with the FEA predictions, with deviations within acceptable engineering limits. This dual validation approach enhances confidence in the gearbox design.

4 Conclusion and future scope

The present study established a validated framework for gearbox analysis by integrating finite element modal predictions with experimental FFT measurements. Quantitative comparisons of natural frequencies, error percentages, and gear mesh harmonics demonstrated close agreement between numerical and experimental approaches, with deviations consistently within $\pm 3\%$. Peak von Mises stress, total deformation, and safety factor values were reported component-wise, confirming structural adequacy under rated loading conditions. Fatigue life estimation further substantiated the long-term durability of the gearbox design. By presenting explicit values, tabulated comparisons, and acknowledging boundary condition limitations, the study provides a scientifically rigorous validation of the gearbox's dynamic behavior. Numerical MAC values greater

than 0.9 confirmed strong correlation within the FEA model. Experimental validation was limited to frequency agreement, as mode-shape verification requires multi-point measurements that were not part of the present setup. The study is limited to a single gearbox configuration and specific operating conditions. Further work is required to generalize findings across different EV drivetrain architectures.

Although the present study has successfully established frequency correlation between finite element modal analysis and experimental FFT measurements, further refinement is required to achieve comprehensive validation. Future work will focus on incorporating multi-point accelerometer instrumentation to enable mode shape identification and Modal Assurance Criterion (MAC) analysis beyond single-point frequency measurements. In addition, the numerical model will be enhanced through boundary condition refinement by accounting for chassis flexibility, chain compliance, and motor inertia, thereby aligning more closely with the experimental configuration. Extended fatigue and service life analysis under variable loading cycles and real-world duty conditions will be undertaken to strengthen durability predictions. Finally, detailed gear mesh harmonic characterization across different operating speeds will be explored to improve diagnostic capability and vibration signature interpretation. Together, these directions provide a structured roadmap for advancing gearbox validation toward more rigorous, realistic, and durable design assessments.

Abbreviations

FEA	Finite element analysis
FFT	Fast Fourier transform
EV	Electrical vehicle
MINLP	Mixed-integer nonlinear program

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Author contributions

Amol Dhumal: Conceptualization, Methodology, Validation, Formal analysis, investigation, Data curation, Writing the original draft, Writing-Review & editing, and Visualization. Masuk Abdullah: Writing-Review & editing, Formal analysis, Data curation, Project administration, Funding acquisition and Visualization. Nitin Ambhore: Conceptualization, Methodology, Validation, Formal analysis, Investigation, Data curation, Writing-Review & editing, Supervision. Tushar Jadhav: Conceptualization, Methodology, Validation, Investigation, and Writing-Review & editing. Atul Kulkarni: Writing-Review & editing, Formal analysis, Data curation, Supervision.

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Data availability

All data generated or analysed during this study are included in this published article.

Declarations

Ethics approval and consent to participate

Not applicable. This study does not involve human participants or animal subjects. Consent to participate: Not applicable. No human participants were involved in this study.

Consent for publication

Not applicable. No participant data were collected or published.

Competing interests

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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