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Design, Analysis, Optimization and Validation of Three Stage Helical Gearbox: Computational Approach

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ABSTRACT
This research present a comprehensive design, analysis, optimization, and validation of a triple stage helical reduction gearbox, addressing the limited exploration of three stage configurations compared to extensively studied single and double stage systems. Triple stage gearboxes offer enhanced reduction capabilities but pose challenges related to transmission vibrations, stress concentration and structural integrity. An existing gearbox model was examined to identify root causes of failure through detailed structural and vibrational analysis. Based on these findings, design modifications were introduced and optimized using advanced computational methods to minimize equivalent stresses and total deformation. The optimized model demonstrated significant improvements under defined loading conditions, with the curved top casing design (structural steel, 9 mm thickness) achieving an 80. 3% reduction in stress and a 97. 2% reduction in deformation compared to the baseline configuration. In contrast, the fully curved casing exhibited no substantial performance gain over the original design. Validation through numerical and mathematical approaches confirmed the reliability of optimized configuration. The results emphasize the potential of advanced design and optimization strategies to enhance efficiency, vibration control, and durability in triple stage gearboxes, contributing to innovation in industrial applications and gearbox technological development.

KEYWORDS: Triple stage helical gearbox, Structural and vibrational optimization, Stress and deformation analysis, Design validation, Gearbox reliability enhancement

INTRODUCTION
Gears are essential for mechanical power transmission in industries, enabling efficient energy transfer and motion control[1]. Among various types, helical gears are widely used due to their smooth operation, high load capacity and efficiency[3-5]. Enclosed in rigid casings, they ensure lubrication and structural integrity. Multistage helical transmission gearboxes especially triple stage configurations offer compact design, better load distribution and durability, despite slightly lower efficiency than single or double stage systems[4-9]. Optimizing gearbox design requires analyzing stress, deformation & dynamic loads. Numerical techniques are widely preferred for accuracy in enhancing performance addressing the growing demand for quieter and more efficient power transmission systems[10-13].

This study systematically reviews existing literature on triple stage helical gearboxes identifies research gaps and proposes future directions. A comprehensive review of research papers published between 2010 & 2025 was conducted to analyze technological advancements, research methodologies and existing challenges in gearbox optimization. Multiple databases were explored to ensure extensive coverage of relevant studies. Research on helical gearboxes has primarily focused on optimizing weight, volume and efficiency. Golabi *et al.*[14] optimized gearbox parameters using MATLAB, demonstrating that transmission stages, gear module, face width and shaft diameter significantly influence design. Later, Hofstetter *et al.*[15] extended this approach to multi objective optimization in xEV-axle drives improving performance and cost effectiveness. Afterwards, Maputi *et al.*[16] & Thompson *et al.*[17] employed analytical and evolutionary algorithms to refine gearbox design, achieving volume and center distance reductions by up to 55%.

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Further, investigations[18-22] has explored stress analysis, material selection and performs geometric optimization. Similarly, narayankutty[23] studied three stage planetary gearboxes identifying stress variations due to misalignment & tolerance deviations. Subsequent studies[24-27] focused on minimizing stress, deformation, weight and vibration by modifying casing profiles and selecting optimal materials such as high carbon steel, structural steel and fiber reinforced polymers (GFRP). Experimental findings indicated that GFRP based housings offer weight reduction and enhanced stress resistance compared to traditional cast iron designs. Furthermore, optimization of gear transmission ratios has also been a significant research area. Vu Ngoc Pi *et al.*[28] utilized computational models to determine optimal speed ratios for bevel helical gearboxes. Additional studies [29-32] examine gearbox height, cross sectional area and performance under varying load conditions. Findings indicated that triple stage gearboxes outperform double stage counterparts at higher speed ratios due to better stress distribution and reduced component sizes.

Moreover, cost optimization and manufacturing feasibility have also been explored by Dinh Ngoc Nguyen *et al.*[33] analyzed dominant design variables affecting gearbox costs, concluding that optimal intermediate gear ratios significantly reduce expenses. Besides, studies[34-40] highlighted the impact of helical tooth direction on bearing longevity, noise reduction and fatigue resistance confirming that modified gear orientations enhance durability. Additionally, advanced various computational methods including genetic algorithms and finite element analysis (FEA) have been instrumental in gearbox design[41,42]. Researchers have applied these techniques to optimize weight, size and thermal performance while ensuring structural integrity[43]. Over and above that, research studies[44-46] on material innovations suggest that composite materials and high strength alloys can improve gear efficiency without compromising durability. Also, the inter- disciplinary framework integrating mechanical, thermal and acoustic analyses[47] have demonstrated promising results in noise reduction and energy efficiency.

Thereby, increasing relevance of multistage gearbox in industrial applications necessitates more comprehensive studies. So, existing research employs a mix of theoretical, analytical, optimization and validation methods with finite element analysis (FEA) being a widely used numerical approach. However, limited investigations have been conducted on modal, vibration & harmonic analyses to assess gearbox frequency behavior, lateral deformations & resonance conditions. Additionally, studies on fatigue stress evaluation and material optimization remain scarce. Thus, the primary objective of this research is to optimize the design of three stage helical gearboxes by minimizing the impact of transmission vibrations on equivalent stresses and total deformations. This will be achieved through modifications in casing profiles, adjustments in rib and cover thickness and appropriate selection of suitable materials after analyzing transmission housing vibrations caused by rotating gears through optimized gear and shaft designs, followed by formulating a mathematical model of the proposed design against the existing model using statistical optimization techniques, and validating the optimized configuration through numerical and mathematical comparisons to ensure improved performance and reliability.

RESEARCH METHODOLOGY

In this study, a **computer aided gearbox design optimization process** is introduced as shown in Fig. 1 to streamline early stage gearbox transmission development for industrial applications. The methodology involves a systematic approach to gearbox design, analysis, optimization & validation based on organization[48] specified **loading and boundary conditions**.

Initially, the transmission housing vibrations caused by rotating gears are analyzed to identify critical stress points and deformation areas. This analysis helps in understanding the structural weaknesses & optimizing design parameters to enhance gearbox performance. To achieve this, several design modifications were implemented like adjustments in casing profiles, variations in rib & cover thickness and the proper selection of suitable materials to minimize equivalent stresses and total deformations. Once the optimized gearbox mathematical model is developed, it undergoes validation against the existing model using statistical optimization and validation techniques for different unknown combinations of materials, thickness and profiles. This ensures improved reliability and efficiency while meeting industrial application requirements.

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Besides, the refined design approach contributes to the development of high performance, durable and cost effective gearbox solutions for real world applications.

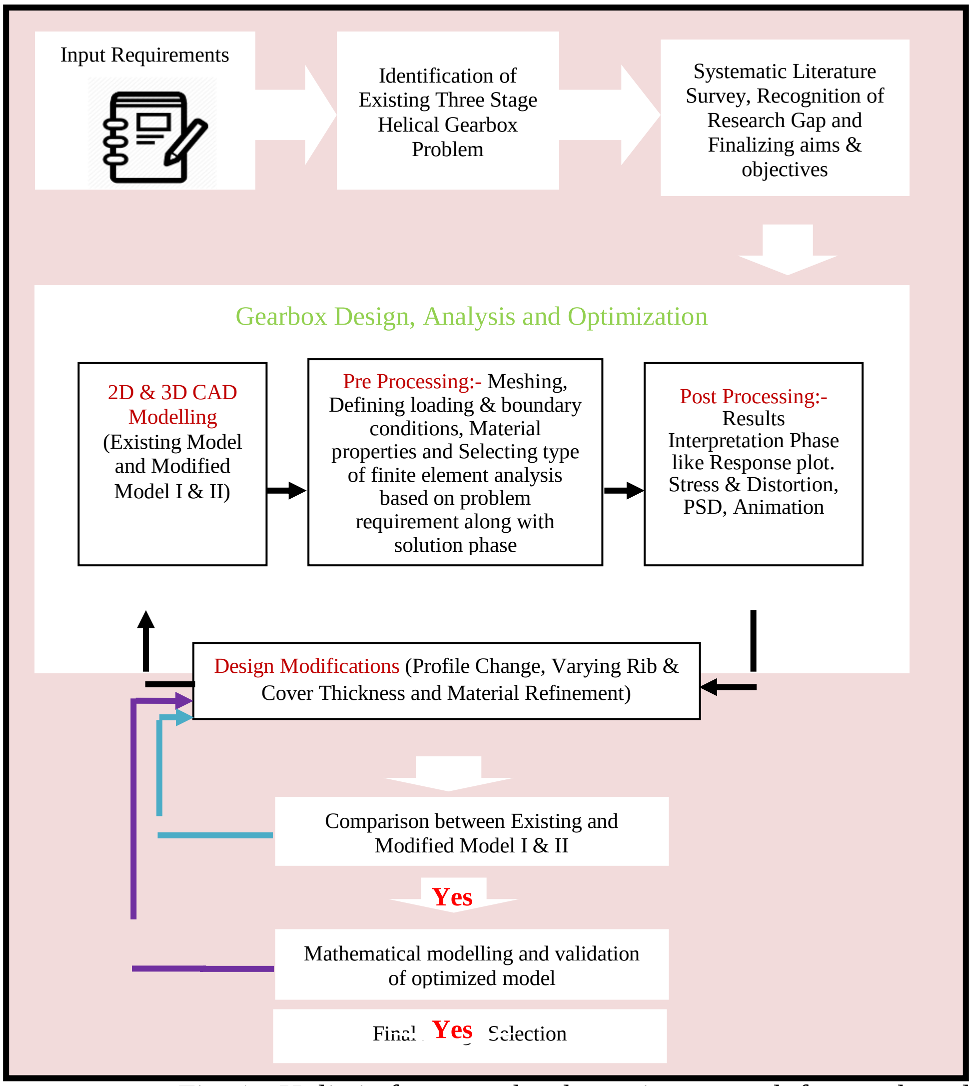

Fig. 1 - Holistic framework schematic approach for gearbox development process

2D & 3D CAD Modeling

Computer Aided Design (CAD) plays a crucial role in gearbox development by enabling precise creation, modification, analysis and optimization of designs. It incorporates essential parameters like geometry, dimensions, tolerances & manufacturing constraints to enhance design accuracy and efficiency. Based on the company's existing 3D CAD model of a three stage helical gearbox with a 5 mm casing thickness including shafts, gears and bearings, a redesigned model shown in Fig. 2 was developed using SolidWorks 2019. To minimize transmission vibrations and enhance structural integrity, the gearbox was optimized by varying the rib & cover thicknesses to 7 mm & 9 mm and changing casing profiles using suitable relevant materials as illustrated in Figs 3-4. Additionally, key industry provided parameters including module, helix angle, pressure angle, rake angle and gear teeth specifications as presented in Table 1 were incorporated for both the existing & modified designs.

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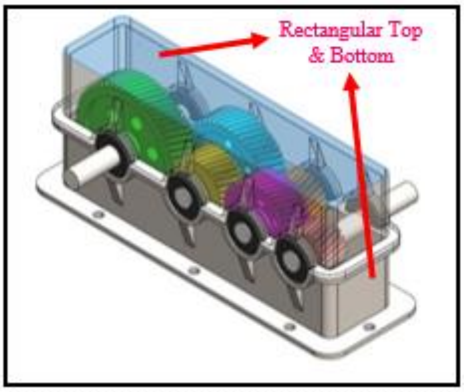

Fig. 2 - Existing 3D CAD model for three stage helical gearbox with rib and cover assembly cross sectional view

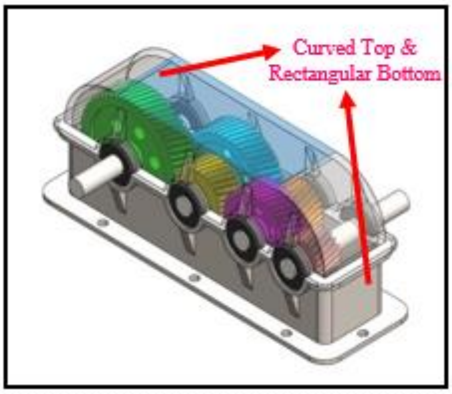

Fig. 3 - Modified 3D CAD model I for three stage helical gearbox with rib and cover assembly cross sectional view

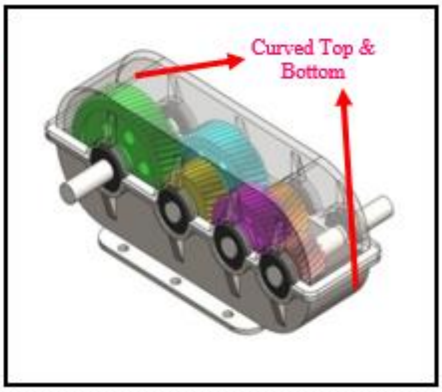

Fig. 4 - Modified 3D CAD model II for three stage helical gearbox with rib and cover assembly cross sectional view

Table 1 - Helical gears terminology for existing and refined model with varying rib and cover thickness (5, 7, 9 mm)

Sr. No.	Case	Profile Shape	Terminology	Gear I	Gear II	Gear III	Gear IV	Gear V	Gear VI
1	Existing Model	Rectangular Top & Rectangular Bottom Gear Box	Module	2	2	2	2	2	2
			Helix Angle	15°	15°	15°	15°	15°	15°
			Pressure Angle	20°	20°	20°	20°	20°	20°
			Rake Angle	6°	12°	7.2°	14.4°	9°	17.14°
			Teeth Number	60	30	50	25	40	20
2	Modified Model I	Curved Top Gear Box and Rectangular Bottom	Module	2	2	2	2	2	2
			Helix Angle	15°	15°	15°	15°	15°	15°
			Pressure Angle	20°	20°	20°	20°	20°	20°
			Rake Angle	6°	12°	7.2°	14.4°	9°	17.14°
			Teeth Number	60	30	50	25	40	20
3	Modified Model II	Curved Top and Curved Bottom Gear Box	Module	2	2	2	2	2	2
			Helix Angle	15°	15°	15°	15°	15°	15°
			Pressure Angle	20°	20°	20°	20°	20°	20°
			Rake Angle	6°	12°	7.2°	14.4°	9°	17.14°
			Teeth Number	60	30	50	25	40	20

Preprocessing using Ansys

The preprocessing stage included defining material properties, generating a finite element mesh and applying boundary & load constraints. Here, the meshing process covered both internal and external surfaces including gears, shafts, bearings and housing. Quadrilateral elements were first divided into triangular elements and then transformed into tetrahedral or hexahedral forms with a 5 mm mesh size. Laterly, existing S.S.A36 material was used having an elastic modulus (E) of 200 GPA, Poisson's ratio (μ) 0.26 and a density (ρ) of 7200 kg/m³. Moreover, the alternative materials were explored including AISI 1050 carbon steel (E = 207 GPA, μ = 0.29, ρ = 7850 kg/m³) and an Al-Ti-SiC metal matrix composite (E = 200 GPA, μ = 0.35, ρ = 5000 kg/m³) for performance enhancement.

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In addition, loading and boundary conditions were applied to the bottom type foot mounted gearbox casing, which was securely fixed using six bolts. The gearbox and its casing experienced both static and dynamic loads. The steady load on bearing holes caused by transmission gears and the driving shaft was divided into radial and axial forces. Alongside with it, dynamic loads from rotational moments and vibrations were transmitted through the driveshaft, joints and bearings as shown in Figs 5-7. These vibrations originated at the gear mesh are due to interactions at gear and shaft interfaces on each shaft. Besides, rotational moment and speed calculations were performed using various equations for a three stage helical gearbox driven with a 500 HP induction motor at 1400 rpm, followed through gear ratio computation based on the number of gear teeth and their diameters as represented mathematically in Eqs (1)–(8).

Gear Ratio 1 (m_{g_1}) = (Number of teeth on Gear 5) / (Number of teeth on Gear 6) = 2:1

Gear Ratio 2 (m_{g_2}) = (Number of teeth on Gear 3) / (Number of teeth on Gear 4) = 2:1

Gear Ratio 3 (m_{g_3}) = (Number of teeth on Gear 1) / (Number of teeth on Gear 2) = 2:1

Therefore,

$$\text{Speed of Shaft 1 } (S_1) = 1400 \text{ rpm} \quad \text{.........................(1)}$$

$$\text{Speed of Shaft 2 } (S_2) = mg_1 = (T_5 / T_6) = (N_6 / N_5) = (2 / 1) = (1400 / N_5) = 700 \text{ rpm} \quad \text{.........................(2)}$$

$$\text{Speed of Shaft 3 } (S_3) = mg_2 = (T_3 / T_4) = (N_4 / N_3) = (2 / 1) = (700 / N_3) = 350 \text{ rpm} \quad \text{.........................(3)}$$

$$\text{Speed of Shaft 4 } (S_4) = mg_3 = (T_1 / T_2) = (N_2 / N_1) = (2 / 1) = (350 / N_1) = 175 \text{ rpm} \quad \text{.........................(4)}$$

Additionally,

$$\text{Torque } (\tau) = P / \omega = (P \times 60) / (2 \times \pi \times N)$$

$$\tau_1 = (500 \times 30) / (3.14 \times 1400) = 3.412 \text{ Nm} \quad \text{........................(5)}$$

$$\tau_2 = (500 \times 30) / (3.14 \times 700) = 6.824 \text{ Nm} \quad \text{........................(6)}$$

$$\tau_3 = (500 \times 30) / (3.14 \times 350) = 13.64 \text{ Nm} \quad \text{........................(7)}$$

$$\tau_4 = (500 \times 30) / (3.14 \times 175) = 27.29 \text{ Nm} \quad \text{........................(8)}$$

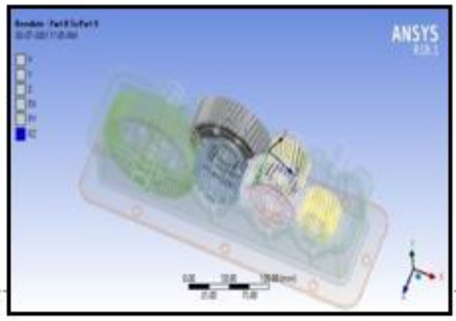

Fig. 5 - Joint interfaces applied to capture rotational moments in original & redesigned gearbox assemblies

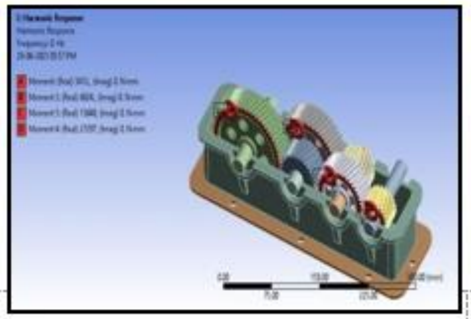

Fig. 6 - Driveshaft loading conditions applied to capture rotational effects in existing and modified gearboxes.

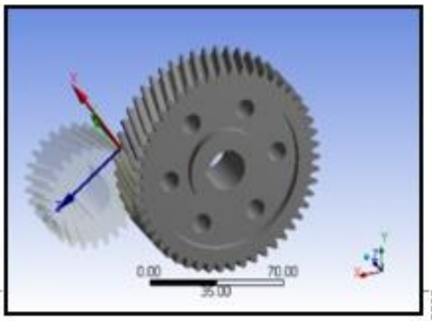

Fig. 7 - Visualization of joint applications on gears, shafts, and casing to represent rotational moment in both configurations.

Solutions and Postprocessing Phase

The analysis phase depends on the problem type, with harmonic analysis being more realistic for gearbox applications as it evaluates vibrational behavior under periodic loads. Alongside frequency domain analysis, it ensures accurate assessment of system performance, resonance identification and vibration reduction.

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Harmonic Analysis

Processing machinery experiences periodic (harmonic) loads, making structural components susceptible to reversed stresses. When external frequencies match a structure's natural frequency, resonance can cause excessive vibrations, strain or failure. To ensure reliability, engineers design components to mitigate these risks. Hence, harmonic analysis predicts a structure's steady state response to sinusoidal loads, verifying its resistance to resonance, fatigue and forced vibrations. It relies on modal analysis to identify natural frequencies and differs from transient analysis, which examines time dependent responses.

The total deformations and equivalent von misses stresses for the existing and optimized models with varying rib and cover thicknesses (5, 7, and 9 mm), were analyzed using Structural Steel A36, AISI 1050 Carbon Steel and Al-Ti-SiC Metal Matrix composites materials. The few corresponding results, illustrated in Figs 8–9 and Table 2, provides insights into structural performance under harmonic loading conditions.

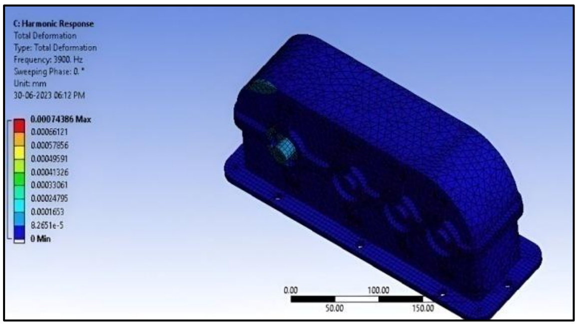

Fig. 8 - Total deformations from harmonic analysis on modified helical gearbox model I AISI 1050 H.C.S. - 7 mm

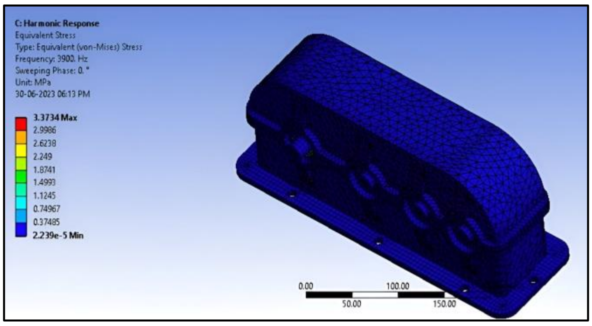

Fig. 9 - Equivalent stresses from harmonic analysis on modified helical gearbox model I AISI 1050 H.C.S. - 7 mm

Optimization and Validation of Modified Results

Optimization is essential for enhancing the efficiency and reliability of a three stage helical gearbox, as it minimizes energy losses, reduces weight, and improves durability, while validation ensures compliance with structural and functional requirements. Among various techniques, the design of experiments (DOE) is widely adopted due to its systematic methodology. Within DOE, factorial design analyzes parameter interactions, screening design identifies key factors, & mixture design optimizes material composition. Response surface methodology (RSM) refines predictive models, whereas the taguchi method effectively minimizes variations. Comparative analysis confirmed the taguchi technique's superiority in optimizing gearbox performance across multiple trials. Using the taguchi approach, harmonic analysis of all gearbox configurations assessed signal-to-noise (S/N) ratios for shape, thickness & material. Residual, main effects and interaction plots determined optimal parameter settings based on total deformation and von misses stress. Additionally; linear model analysis estimated S/N ratio coefficients, while regression analysis developed mathematical models as part of modeling phase, ensuring predictive accuracy. For validation, regression-based predictions of intermediate thickness levels were compared with finite element harmonic simulations, with the observed error margin (≈11–16.5%) confirming model robustness and reliability. This integrated framework of optimization and validation not only enhances gearbox performance but also ensures that the selected design is structurally reliable, dynamically stable, and industrially viable under fluctuating load conditions.

RESULTS & DISCUSSION

In this research study; the construction, accessibility and operation of a three stage helical gearbox were thoroughly examined to identify the root causes of failure. Subsequently, several design modifications & optimizations were implemented to reduce oscillation induced overall distortion, equivalent von misses stresses while enhancing overall strength. Finally, an optimized model was proposed and validated under corresponding loading conditions.

Table 2 presents the findings of harmonic analysis conducted for the existing & modified models which reveals

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significant insights into influence of material selection, thickness variation and geometric refinement on dynamic behavior. The reference configuration, constructed from Structural Steel A36 at 5 mm thickness, exhibited a maximum distortion of 0.0035 mm and equivalent von misses stress of 3.51 MPa. Enhancing this design by increasing thickness to 7 mm and 9 mm progressively reduced distortion by 31.4% and 71.4% and stress by up to 85.8%. These improvements are attributed to increased structural rigidity and flexural stiffness. When replaced with AISI 1050 high carbon steel, the 7 mm model achieved the lowest observed stress (0.994 MPa), a 71.7% decrease from the baseline owing to the alloy's higher modulus of elasticity & strength which reduce stress concentrations under dynamic excitation. Interestingly, the 9 mm AISI 1050 H.C.S. variant did not outperform the 7 mm case, indicating diminishing returns from thickness beyond optimal stiffness thresholds. The 5 mm Al–Ti–SiC metal matrix composites also offered improvements, lowering distortion and stress by 31.4% and 28.7% respectively, due to its high stiffness-to-weight ratio and inherent damping capacity. However, thicker composite variants exhibited instability likely driven by resonance amplification from mode coupling resulting in elevated stress levels and distortion.

In Modified Model I **(Curved Top & Rectangular Bottom - CTRB)**, where the top casing was geometrically curved, the 5 mm steel model immediately yielded notable gains i.e. distortion and stress dropped by 65.7% and 56.4%, respectively. Increasing the thickness further (7 mm and 9 mm) led to peak distortion reductions of 82.9% and 97.1% and stress reductions of 70.2% and 80.3% correspondingly. These enhancements result from greater sectional stiffness and reduced vibrational compliance introduced by the curvature. Conversely, while the 5 mm AISI 1050 H.C.S. variant showed stress amplification (37.0%) due to concentrated stiffness, increasing its thickness to 9 mm successfully reversed this trend, reducing stress by 74.3% and distortion by 97.1%. The composite variant at 5 mm also outperformed the baseline, though thicker configurations showed inconsistent stress behavior reaffirming sensitivity to resonance interactions. Among all configurations, the 9 mm structural steel model in Modified Model I was the most effective, achieving the highest reductions in both distortions (97.1%) and stress (80.3%). Similarly, for Modified Model II **(Curved Top & Curved Bottom - CTCB)**, involving full curvature on both top and bottom, performance was less consistent. Although the 9 mm steel variant reduced distortion by 28.6%, it produced a greater percentage stress increase likely due to curvature induced modal amplification. The 7 mm steel model performed even worse, with significant increases in both parameters. On the other hand, AISI 1050 H.C.S. at 9 mm showed moderate improvement in distortion (28.6%) but retained high stress, reflecting a trade-off between increased stiffness and dynamic instability in curved geometries. The composite version displayed poor stress performance at all thicknesses, with only marginal distortion gains at 9 mm, confirming incompatibilities between the curved double profile and composite's vibrational characteristics.

Overall, increasing casing and rib thickness across all models generally enhanced distortion control with observed stress reductions in the range of 30%–65%, consistent with findings reported by Smita *et al.*[4] and Huang *et al.*[27] which affirm that increasing structural thickness enhances rigidity and mitigates stress concentrations under dynamic excitation. The present results further highlight that optimized gearbox profile design; strategic material selection and structural reinforcement significantly influence harmonic response behavior. However, the interplay between geometric refinement and material properties is inherently complex. While curvature in casing geometry can improve vibrational resistance through enhanced stiffness distribution and reduced compliance, it may also introduce localized stress amplification due to modal coupling phenomena particularly when resonance frequencies align unfavourably with excitation modes. This underscores the necessity for a harmonized design strategy that integrates geometric contouring, material tailoring and modal tuning to achieve superior dynamic stability and reduced harmonic distortion in gearbox assemblies subjected to fluctuating loads.

Further, a comparative analysis of the shortlisted models summarized in Table 3 was conducted to evaluate the vibrational behavior & structural integrity of the existing and modified designs under various material and thickness configurations. The baseline configuration **existing model with rectangular top and bottom at 5 mm thickness** served as the reference for evaluating the performance of all other configurations. Among the existing design variations, replacing the material with Metal Matrix Composites (5 mm) resulted in a

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31.7% reduction in distortion (0.0024256 mm) and a 28.7% reduction in stress (2.5015 MPa). Increasing the thickness to 7 mm with AISI 1050 High Carbon Steel yielded a further improvement, lowering distortion by 43.7% (0.0020 mm) and increasing stress to 9.9441 MPa, indicating improved stiffness but higher stress concentration. A 9 mm thickness using the same material showed the most notable reduction within the existing model group, achieving 71.8% distortion reduction (0.0010 mm) and 41.5% stress reduction (4.9708 MPa) compared to the baseline.

Transitioning to **Modified Design I** demonstrated a significantly improved harmonic response. The 5 mm Structural Steel A36 variant achieved **65.6% distortion reduction** (0.00122 mm) and a **56.4% reduction in stress** (1.5295 MPa). Increasing to 7 mm and 9 mm thicknesses led to further improvements, with distortion reduced to **0.0006 mm (83.1%)** and **0.0001 mm (97.2%)**, respectively. The corresponding stress values dropped to **2.8917 MPa** (17.6% reduction) and **0.692 MPa**, representing a **remarkable 80.3% decrease** compared to the original existing model. These reductions are consistent with ansys based simulations conducted by Patel *et al.*[3] & Saritas *et al.*[9], which indicate that optimized structural designs and increased thickness significantly enhance rigidity and mitigate dynamic stress concentrations.

In contrast, Modified Design II displayed mixed results. While distortion remained within acceptable limits (e.g., 0.0025 mm at 9 mm thickness, a 29.5% reduction), the stress values surged significantly, particularly in the 7 mm AISI 1050 H.C.S. configuration, where stress peaked at 137.45 MPa, representing a tremendous percentage increase from baseline. This substantial increase indicates potential stress concentration issues that may compromise structural safety despite improved vibrational isolation in specific cases. However, the observed trend may also be influenced by the limited number of load cases considered in mode shape investigations or material-specific variations. In summary, Modified Design I consistently outperformed the baseline and other configurations, achieving up to 97% distortion reduction and 80% stress reduction, making it a superior alternative for minimizing harmonic response and enhancing structural efficiency. Moreover, harmonic analysis conducted for Modified Design I at frequencies beyond resonance confirms the model's superior vibration resistance as evidenced by its ability to minimize reverberation. Conversely, although Modified Design II introduces geometrical innovations, its susceptibility to elevated stress levels under certain material thickness combinations necessitate further optimization before practical implementation. Therefore, the findings presented in Table 3 substantiate that upgraded gearbox I fabricated using structural steel exhibits notable improvements in mitigating equivalent von misses stresses and total deformation.

Later, finite element analysis (FEA) results were optimized to ensure simulation accuracy and reliability. The optimized design was assessed by comparing computational predictions with statistical data obtained from taguchi and regression analyses. The correlation analysis presented in Table 4 demonstrates a strong agreement between the optimized parameters and the predicted FEA performance, confirming the robustness of the taguchi method in parameter optimization. This integration of computational and statistical techniques enhances the validation process, ensuring a highly reliable gearbox design with improved efficiency and durability for industrial applications. Moreover, the main effects plot for signal-to-noise (S/N) ratios in Fig. 10 illustrates the influence of shape, thickness, and material based on the "smaller-the-better" criterion, where higher (less negative) S/N values indicate improved performance in minimizing stress and deformation. Among the shape variations, CTRB exhibits the highest S/N ratio, signifying optimal performance, whereas CTCB records the lowest, indicating the least favorable outcome; RTRB lies intermediate between the two. Thickness also plays a critical role, with 7 mm showing the poorest response, while both 5 mm and 9 mm demonstrate significantly better performance. In terms of material influence, S.S.A36 outperforms the others with the highest S/N ratio, followed by AISI 1050 H.C.S., while MMC exhibits the least desirable outcomes. These results suggest that CTRB shape, 5 or 9 mm thickness, and A36 material represent the most effective combination for reducing mechanical stress and deformation.

Likewise, the interaction plot in Fig. 11 reveals dynamic interactions between shape and thickness, CTRB consistently exhibits the highest S/N ratios across all thickness levels, marking it as the optimal shape, especially at 9 mm thickness, while CTCB at 7 mm shows a sharp deterioration, denoting the worst

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configuration. When evaluating shape-material interaction, CTRB paired with AISI 1050 H.C.S. outperforms all other combinations, whereas CTCB with Al–Ti–SiC M.M.C. yields the poorest response. Thickness-shape trends reiterate CTRB's superiority, particularly at 5 and 9 mm, in contrast to CTCB's under performance at 7 mm. Similarly, thickness-material interactions reveal that 9 mm thickness coupled with AISI 1050 H.C.S. or Al–Ti–SiC M.M.C. achieves improved results, while 7 mm & Al–Ti–SiC M.M.C. results in a significant drop in S/N ratio. The material-shape panel further affirms CTRB & AISI 1050 H.C.S. as the best performing pair, whereas CTCB & Al–Ti–SiC M.M.C./S.S.A36 is undesirable. Finally, the material-thickness analysis shows that AISI 1050 H.C.S. maintains consistent and superior performance across all thickness levels, while Al–Ti–SiC M.M.C. exhibits vulnerability at 7 mm, & S.S.A36 remains moderately effective. These interaction effects emphasize the robustness of CTRB shape, 9 mm thickness and AISI 1050 H.C.S. material as a high performing combination. Finally, residual plot analysis as shown in Fig. 12 confirms the model's validity by demonstrating normality, randomness and reliability ensuring the absence of systematic bias and enhancing model accuracy. These collective findings reinforce the robustness of the statistical approach, confirming that the taguchi based model satisfies ANOVA assumptions and offers a reliable framework for optimizing structural response parameters.

Furthermore, the ***TD_Max*** and ***ES_Max*** mathematical models were developed using regression analysis in Minitab, with optimal models selected based on **Figs 13–14** and **Table 4**. A Taguchi based linear model including main effects, interaction, and residual plots was employed to evaluate factor influence and process optimization. Key statistical metrics such as R^2, p-values, and residual diagnostics ensured model robustness. Consequently, this approach enhances predictive accuracy, reduces experimental efforts, and supports efficient quality control and forecasting in engineering applications.

In summary, the taguchi based signal-to-noise (SN) ratio analysis under the "smaller-is-better" criterion identifies 5 mm and 9 mm thickness levels as statistically superior, with 7 mm performing least optimally. Complementing this, the FEA validation provides a more robust confirmation by capturing the actual mechanical response under dynamic loading, wherein Modified Design I (Curved Top & Rectangular Bottom) constructed from Structural Steel A36 at 9 mm thickness consistently exhibited the lowest total deformation (0.0001 mm) and equivalent von misses stresses (0.692 MPa). This superior performance among all design–material–thickness combinations are attributed to the optimized geometric contour, which ensures uniform load dispersion and the enhanced sectional stiffness at higher thickness, which minimizes oscillation-induced stress amplification. Collectively, these findings validate 9 mm thickness as the most reliable configuration, demonstrating strong agreement between taguchi optimization and FEA predictions. Therefore, for practical implementation in industrial gearbox applications subjected to fluctuating loads, 9 mm Structural Steel A36 variant of Modified Design I is recommended as the optimal solution offering reduced harmonic distortion, enhanced dynamic stability, and long-term mechanical durability.

Ultimately, building on the regression based models, intermediate thickness values of 5.5, 6, 6.5, 7.5, 8, and 8.5 mm were evaluated for the optimal combination of Modified Design I with Structural Steel A36. These values were deliberately selected within the validated 5–9 mm design space since statistical tools such as regression and taguchi analysis remain reliable only under interpolation, whereas extrapolation may introduce large errors. Importantly, nonlinear structural responses outside this interval such as warping in very thin sections (<5 mm) or disproportionate stiffness and weight effects in thicker sections (>9 mm) reinforce the rationale for restricting validation within the chosen limits. A prediction summary for minimum total deformation and equivalent von misses stress was generated (Table 5) providing Fit, SE Fit, 95% CI, and 95% PI values for intermediate thickness levels (5.5, 6, 6.5, 7.5, 8, and 8.5 mm) of the optimal Modified Design I configuration. To access predictive accuracy, harmonic response analyses were performed in ANSYS for these interpolated thicknesses. The comparative results (Fig. 15 and Table 6) indicated deviations of 11–16.5% for total deformation and 12.5–16.5% for von misses stresses relative to the statistical predictions. Overall, these findings confirm that the regression model offers dependable predictive capability within defined design window, ensuring robust validation of structural response parameters for gear box.

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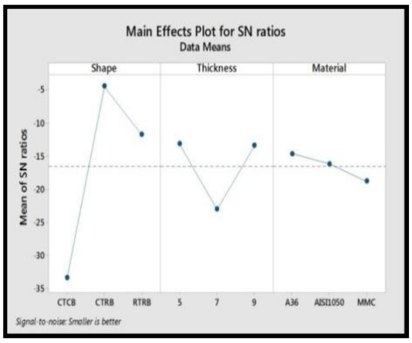

Fig. 10 - Main effects plot for SN ratios highlighting the individual influence of shape, thickness and material on the response graph

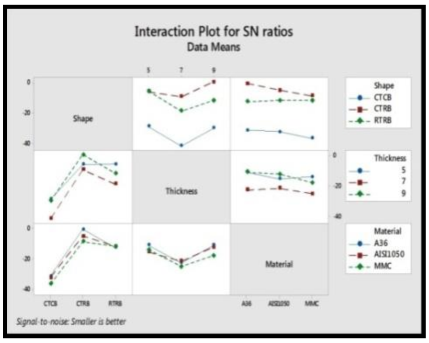

Fig. 11 - Interaction effects plot for SN ratio highlighting the individual influence of shape, thickness and material on the response graph

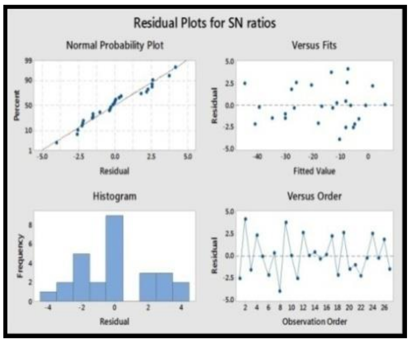

Fig. 12 - Residual plots for SN ratios diagnostics using normal probability plot, histogram, residuals vs fitted values and observation order graphs

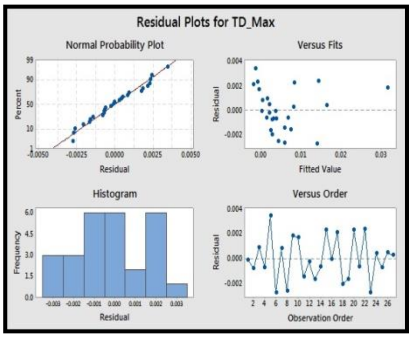

Fig. 13 - Residual plots for TD_Max diagnostics using normal probability plot, histogram, residuals vs fitted values and observation order graphs

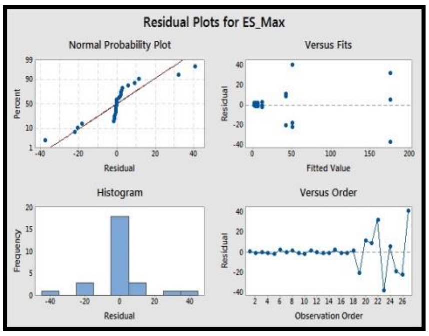

Fig. 14 - Residual plots for ES_Max diagnostics using normal probability plot, histogram, residuals vs fitted values and observation order graphs

Note: For simplicity, the variables were abbreviated throughout all tables and figures as follows: Thickness (T), Shape (S), Material (M), Recoded Shape (RS), Recoded Material (RM), Structural Steel A36 (S.S.A36), AISI 1050 High Carbon Steel (AISI 1050 H.C.S.) and Al-Ti-SiC Metal Matrix Composites (Al-Ti-SiC M.M.C.)

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