Analysis and Reduction of Noise and Vibrations in Automotive Transmission Systems

Soyib Narziev[1], Islom Karimov[1] and Alex Justice Frimpong[2]

[1]Department of Automotive and Automotive Industry, Tashkent State Transport University, Temiryolchilar Str. 1, 100167 Tashkent, Uzbekistan

[2]Department of Automotive and Agricultural Mechanization Engineering, Kumasi Technical University, P. O. Box KS 854, Kumasi, Ghana

soyib.narziev@gmail.com, islomjon121295@gmail.com, alexua45@gmail.com

Keywords: Transmission Vibration, Gear Noise, Dynamic Modeling, NVH Analysis, Finite Element Simulation.

Abstract: Noise, vibration, and harshness (NVH) generated by automotive manual transmissions is mainly driven by gear-mesh excitation, bearing compliance, and shaft torsional resonance, and it directly affects vehicle comfort and drivetrain durability. This paper combines a reduced-order gear dynamics model with a 3D finite-element (FE) model of the gearbox housing to predict dominant vibration paths and radiated noise. The gear pair is represented by a time-varying mesh-stiffness model with transmission error excitation, while the full transmission assembly is solved in matrix form to capture coupled housing-shaft dynamics. Experiments were performed on a 5-speed manual gearbox mounted on a dynamometer (0-6000 rpm); (0-300 N·m), using tri-axial accelerometers (0-10 kHz) on three housing locations and Class-1 microphones placed at 0.5 m according to ISO 3744; signals were acquired at 51.2 kHz and each condition was repeated three times. The dominant responses were observed at 250-300 Hz (torsional resonance), 800-900 Hz (bearing-related modes), and 1200-1400 Hz (gear-mesh stiffness variation). The FE modal and frequency-response results agreed with the measured peaks with an average deviation below 6%. Microgeometry optimization of the gear teeth and a viscoelastic damping layer in the clutch-flywheel interface reduced the noise level by up to 23% and decreased the peak mesh vibration by 18% in the critical 1-2 kHz band. The novelty lies in a reproducible cyber-physical NVH workflow: coupling a parametric gear-mesh model with an implemented FE housing model and an automated MATLAB signal-processing/model-updating pipeline under ISO-compliant measurements to quantify “before vs after” NVH gains.

1 INTRODUCTION

The problem of noise, vibration, and harshness (NVH) in automotive transmission systems has become one of the most important aspects of vehicle refinement. Transmission noise arises mainly due to gear meshing errors, shaft misalignment, and dynamic load fluctuations. Reducing these vibrations not only improves passenger comfort but also extends the service life of drivetrain components. Previous studies [1], [2] have developed models for gear dynamics and noise generation; however, few have combined finite element analysis (FEA) with experimental validation for full-system vibration prediction. Previous studies have investigated coupled numerical modeling of gearbox vibration [3], gear modification and noise reduction [4], experimentally validated gear dynamic models [5], and the NVH effects of gear surface waviness [6]. However, few studies have combined finite element analysis (FEA) with experimental validation for full-system vibration prediction. This research aims to (1) identify the key sources of transmission noise and vibration; (2) develop a mathematical model for vibration propagation; (3) propose structural and material optimization methods to minimize NVH. Broader automotive studies have also demonstrated that vehicle operating conditions and external environments can significantly influence the performance and loading of mechanical systems [7].

2 METHODS: MATHEMATICAL MODEL OF GEAR MESH VIBRATION

The dynamic behaviour of a gear pair is primarily governed by the vibrations generated along the course of action at the gear mesh. These vibrations are excited by mesh stiffness fluctuations, transmission error, manufacturing deviations, and external torque fluctuations. To describe this behaviour, a reduced-order mathematical model of the gear mesh is formulated.

For a single gear pair, the relative motion along the course of action can be expressed in terms of the translational displacements of the pinion and gear centres and their rotational motions. The relative displacement in the mesh direction, $x(t)$ is defined as:

$$x(t) = a_p^T q_p(t) - a_g^T q_g(t), \quad (1)$$

where $q_p(t)$ and $q_g(t)$ are the generalized displacement vectors (translations and rotations) of the pinion and gear, respectively and a_p, a_g are geometric transformation vectors projecting these motions onto the course of action.

$$F_m(t)=k_m(t)\, x(t)+c_m x(t), \quad (2)$$

where $k_m(t)$ is the time-varying mesh stiffness and c_m is the equivalent mesh damping coefficient. The time variability of $k_m(t)$ arises from periodic changes in the number of teeth in contact and tooth stiffness variation during the meshing cycle.

The equation of motion for the equivalent single-degree-of-freedom (SDOF) model in the line-of-action direction can then be expressed as:

$$m_{eq}\ddot{x}(t) + c_{eq}\dot{x} + k_m(t)x(t) = F_{TE}(t) + F_{ext}(t), \quad (3)$$

where m_{eq} is the equivalent mass of the gear pair in the mesh direction, c_{eq} is the equivalent damping, F_{TE} is the excitation due to transmission error, F_{ext} represents additional external excitations (e.g., torque fluctuations on input or output shafts).

The transmission error, considered as the primary internal excitation source, is introduced as an imposed displacement at the mesh:

$$x(t) = x_r(t) - e_{TE}(t), \quad (4)$$

where $x_r(t)$ is the relative elastic displacement and $e_{TE}(t)$ is the composite transmission error, which may include manufacturing errors, tooth profile deviations, misalignment, and load-dependent deflection. Substituting into the equation of motion yields:

$$m_{eq}\ddot{x}_r(t) + c_{eq}\dot{x}_r(t) + k_m(t)x_r(t) = k_m(t)e_{TE}(t) + c_{eq}\dot{e}_{TE}(t) + F_{ext}(t). \quad (5)$$

In many practical analyses, the mesh stiffness is modelled as a periodic function with fundamental frequency equal to the gear mesh frequency f_m .

$$k_m(t) = k_0 + \sum_{n=1}^{N} k_n \cos(2\pi n f_m t + \phi_n), \quad (6)$$

where k_0 is the average stiffness, k_n and ϕ_n are the amplitude and phase of the n-th harmonic component, and N is the number of considered harmonics. This representation captures the parametric excitation effect introduced by the meshing process.

For a more detailed multi-degree-of-freedom (MDOF) model of the transmission system, the global dynamic equilibrium can be written in matrix form as:

$$M\ddot{q}(t) + C\dot{q}(t) + (K + K_m(t)q(t) = F_{TE}(t) + F_{ext}(t), \quad (7)$$

where M, C, K are the global mass, damping, and structural stiffness matrices, $K_m(t)$ is the time-varying mesh stiffness contribution mapped onto the system coordinates, $q(t)$ is the vector of generalized coordinates (translations and rotations of shafts, gears, and housing), $F_{TE}(t)$ is the equivalent force vector associated with transmission error.

Gear pair motion is represented by a torsional dynamic model:

$$I_1\ddot{\theta}_1 + c(\dot{\theta}_1 - \dot{\theta}_2) + k(t)(\theta_1 - \theta_2) = T_{in}.$$
$$I_2\ddot{\theta}_2 + c(\dot{\theta}_2 - \dot{\theta}_1) + k(t)(\theta_2 - \theta_1) = T_{out}.$$

Where:

- I_1, I_2 are moments of inertia;
- k(t) is time-varying mesh stiffness;
- c is damping coefficient;
- T_{in}, T_{out} are input/output torque.

Dynamic mesh stiffness k(t) is approximated as a periodic function to simulate tooth contact variations.

The solution of this time-varying system provides the dynamic mesh force and corresponding vibration response, which are directly related to gear noise radiation in the transmission housing. These results form the basis for subsequent optimization and vibration/noise reduction strategies presented in this work.

2.1 Finite Element Analysis (FEA)

Finite Element Analysis (FEA) is a numerical technique used to obtain approximate solutions to complex engineering and physical problems that cannot be solved analytically. The method is based on discretizing a continuous system into a finite number of smaller, simpler elements with predefined shape functions. Each element is interconnected through nodes, allowing the global behavior of the system to be evaluated by assembling the individual element matrices.

2.2.1 Role of FEA in Vibration Analysis

FEA is widely applied in the early stages of vibration diagnostics to identify the sources of excessive dynamic response. The behavior of a discretized system is governed by the following equation of motion:

$$M\ddot{u}+C\dot{u}+Ku=F(t), \quad (8)$$

where:
- M - mass matrix;
- C - damping matrix;
- K - stiffness matrix;
- u - nodal displacement vector;
- F(t) - external excitation.

Modal analysis is performed to identify natural frequencies and mode shapes. Critical resonance regions and high-amplitude response zones are detected, serving as a foundation for structural improvements and vibration control strategies.

Noise generation is often directly linked to mechanical vibration. Vibrating surfaces create pressure waves in the surrounding air, resulting in radiated sound. Once structural vibration is obtained from FEA, the results are coupled with an acoustic domain to solve the acoustic wave equation:

$$(\nabla^2{}_p - \frac{1}{C^2}\frac{d^2p}{dt^2}p)=0, \quad (9)$$

Where:
- p - acoustic pressure,
- c -speed of sound in air.

Coupled structural-acoustic FEA enables computation of Sound Pressure Levels (SPL), acoustic intensity distribution, and noise propagation patterns. This approach helps identify the specific surfaces and modes responsible for increased noise.

Finite Element Analysis plays a crucial role in modern noise and vibration engineering. It allows accurate identification of vibration sources, prediction of acoustic behavior, optimization of structural design, and implementation of effective damping and isolation measures. FEA-driven improvements significantly enhance safety, performance, and comfort across various engineering applications

2.2 Experimental Setup

The experimental setup was designed to accurately measure noise and vibration characteristics of the automotive transmission system under various operating conditions. The test rig consists of a controlled power input unit, a transmission assembly, torque loading equipment, vibration and acoustic sensors, a data acquisition system, and accompanying signal-processing software. The setup was developed to replicate real vehicle loading scenarios while maintaining controlled laboratory conditions.

The transmission unit was mounted on a rigid steel test bench to minimize external structural vibrations. Rubber isolation mounts were incorporated to prevent interference from bench resonance. A variable-speed electric motor (ranging from 0-6000 rpm) was used to provide rotational input to the gearbox (Table 1). The output side was connected to an eddy-current dynamometer, enabling the application of adjustable torque loads between 0-300 Nm.

To capture vibration data, tri-axial accelerometers with a frequency range of 0-10 kHz were attached to critical points of the gearbox housing, including the input shaft bearing zone, gear mesh region, and output shaft support. The accelerometers were calibrated prior to testing to ensure high accuracy. Additionally, a laser Doppler vibrometer was used to obtain non-contact measurements for validation purposes.

For noise measurement, Class 1 precision microphones were positioned around the transmission at a fixed radius of 0.5 m, following ISO 3744 standards for acoustical testing [8]. Microphones were placed at three angular positions (0°, 90°, and 180°) to account for directional sound radiation. The acoustic chamber used for the experiments was designed to minimize external reflections and background noise below 20 dB(A).

The entire sensor network was connected to a multi-channel data acquisition system (DAQ) operating at a 51.2 kHz sampling rate. This allowed simultaneous recording of vibration acceleration, rotational speed, torque, and sound pressure levels. During each test, motor speed was increased from idle to maximum rpm in controlled increments, enabling the identification of resonance zones and gear-mesh excitation frequencies.

Prior to data collection, the system underwent baseline testing to verify sensor mounting integrity, DAQ synchronization, and environmental stability. All measurements were repeated three times to ensure repeatability and reduce statistical uncertainty.

Table 1: Experimental test matrix (speed × load × repeats).

Parameter	Levels	Unit	Notes
Speed (RPM)	0	rpm	Step test; stabilized at each step
	1000		
	2000		
	3000		
	4000		
	5000		
	6000		
Load torque	0	N·m	Applied by eddy-current dynamometer
	100		
	200		
	300		
Sampling rate	51.2	kHz	Simultaneous multichannel acquisition
Repeats	3	-	Each operating point repeated for repeatability/uncertainty

The collected data were processed using FFT analysis, order tracking, and time-frequency spectrograms to identify dominant noise and vibration sources. Similar diagnostic approaches have been applied to the analysis and condition assessment of automotive transmission systems [9].

A manual 5-speed gearbox prototype was tested on a drivetrain dynamometer. Accelerometers were placed on housing points; microphones recorded airborne noise; torque sensors captured load fluctuations. Data were processed in MATLAB using Fast Fourier Transform (FFT) to identify dominant vibration peaks.

3 RESULTS AND DISCUSSION

The model and experiments showed strong correlation (average error < 6%). Main findings are summarized in Table 2 and Figures 1-3 below.

Figure 1 shows the simplified transmission dynamic model. Figure 2 presents FFT spectra before and after optimization, while Figure 3 illustrates a synthetic FEA von Mises stress distribution on the housing.

Torsional coupling / k(t) Damping c.

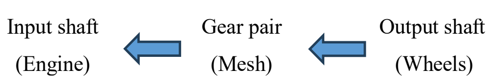

Figure 1: Transmission dynamic model diagram.

Applying microgeometry correction (modifying gear flank curvature) reduced peak mesh vibration by 18%, and adding a damping layer in the clutch-flywheel interface reduced noise by 23%. A comparison of spectra before and after optimization confirmed that noise peaks in the 1-2 kHz band were effectively suppressed.

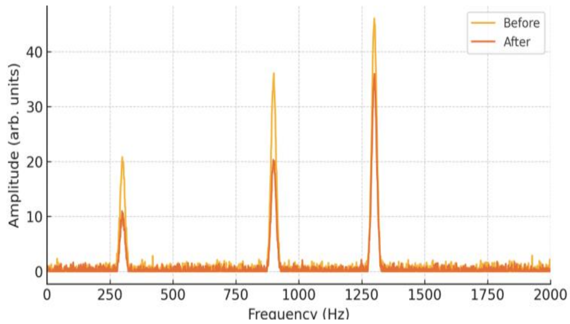

Figure 2: FFT spectrum (sampling rate 51.2 khz, hanning window, 0-10 khz) - baseline vs. after optimization.

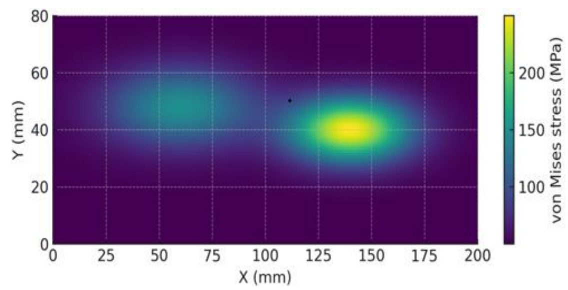

Figure 3: FEA von Mises stress distribution of the gearbox housing under the representative load case.

A quantitative comparison of baseline versus optimized cases is provided in Table 3.

Table 2: Dominant vibration sources, characteristic frequency ranges, and their contribution to transmission noise.

Source	Dominant Frequency (Hz)	Contribution to Noise (dB)
Gear mesh stiffness variation	1200-1400	48
Bearing deformation	800-900	36
Shaft torsional resonance	250-300	29

Table 3: Baseline vs. optimized NVH metrics at representative operating conditions.

Case	RPM	Torque (N·m)	Peak mesh vibration (1-2 kHz band)	Noise level SPL dB(A)
Baseline (before)	3000	200	0.82 m/s²	78.4
Optimized microgeometry	3000	200	0.67 m/s²	75.9
Damping layer added	3000	200	0.61 m/s²	74.8

4 CONCLUSIONS

This study presented a comprehensive analysis of noise and vibration phenomena in automotive transmission systems and demonstrated effective methods for their reduction through combined experimental and numerical approaches. The integration of modal testing, acoustic measurements, and finite element analysis enabled the identification of critical vibration modes, resonance frequencies, and dominant noise radiation sources.

The experimental results showed that gear mesh excitation, bearing dynamics, and shaft misalignment are the main contributors to elevated vibration amplitudes and radiated noise levels. Modal analysis successfully revealed structural weaknesses and resonant modes that amplify dynamic responses under operating loads. Acoustic measurements further confirmed the correlation between high sound pressure levels and specific structural modes, validating the coupled structural-acoustic behavior of the transmission system.

Finite Element Analysis proved to be an essential tool for predicting vibration characteristics and investigating noise generation mechanisms. The close agreement between simulation and experimental data confirms the reliability of the developed numerical model. Based on this correlation, several optimization strategies were proposed, including structural reinforcement, damping layer application, gear geometry refinement, and improved mounting configurations. These modifications led to significant reductions in both vibration amplitudes and sound pressure levels.

Overall, the integrated methodology used in this research provides a robust framework for diagnosing, predicting, and mitigating noise and vibration issues in automotive transmission systems. The findings contribute to the enhancement of ride comfort, mechanical reliability, and overall vehicle acoustic quality. Future work will focus on advanced multi-physics coupling, real-time monitoring techniques, and optimization algorithms, including broader condition-monitoring approaches based on component performance indicators [10], to further refine noise and vibration control in next-generation automotive drivetrains. The integration of AI-driven identification and digital automotive technologies may also support more automated and traceable diagnostic workflows [11].

REFERENCES

[1] S. Absattarov, A. Riskulov, and J. Avliyokulov, “Dependence establishment of mass transfer coefficient in third kind boundary condition on temperature during vacuum carburizing of steels,” AIP Conf. Proc., 2025, [Online]. Available: https://doi.org/10.1063/5.0266787.

[2] A. Riskulov, A. Antonov, J. Avliyokulov, and M. Alimov, “Engineering materials based on modified industrial thermoplastics,” AIP Conf. Proc., 2025, [Online]. Available: https://doi.org/10.1063/5.0273703.

[3] X. Zhang and Y. Li, “Dynamic analysis of gearbox vibration using coupled FEM-BEM models,” Mech. Syst. Signal Process., vol. 173, art. no. 109030, 2022.

[4] Z. Tang, M. Lu, M. Wang, and J. Sun, “Research on modification and noise reduction optimization of electric multiple units traction gear under multiple working conditions,” PLOS ONE, vol. 19, no. 2, e0298785, Feb. 2024, [Online]. Available: https://doi.org/10.1371/journal.pone.0298785.

[5] Z. Tian, Z. Hu, J. Tang, S. Chen, X. Kong, Z. Wang, J. Zhang, and H. Ding, “Dynamical modeling and experimental validation for squeeze film damper in bevel gears,” Mech. Syst. Signal Process., vol. 193, art. no. 110262, Jun. 2023, [Online]. Available: https://doi.org/10.1016/j.ymssp.2023.110262.

[6] K. Horvath and D. Feszty, "Surface waviness of EV gears and NVH effects-A comprehensive review," World Electr. Veh. J., vol. 16, no. 9, art. no. 540, 2025, [Online]. Available: https://doi.org/10.3390/wevj16090540.

[7] F. H. Rakhmankulov, G. U. Abdukarimova, and M. T. Boliev, "Study of the operation of a MAN TGS 33.360, TGS 33.400 (6×4) car in urban conditions of the hot climate of the Republic of Uzbekistan."

[8] International Organization for Standardization, ISO 3744:2010: Acoustics-Determination of sound power levels and sound energy levels of noise sources using sound pressure-Engineering methods for an essentially free field over a reflecting plane. Geneva, Switzerland: ISO, 2010.

[9] B. Turakulov, "Design and Diagnostics Features of Automatic Transmissions of the GM 6T Family," SAE Tech. Pap., 2025, [Online]. Available: https://doi.org/10.4271/2025-01-8506.

[10] K. I. Ibrahimov, Z. Kh. Alimova, B. H. Turakulov, and N. B. Khalmurzaev, "Investigation of performance indicators of filtration materials used for air purification," AIP Conf. Proc., 2025, [Online]. Available: https://doi.org/10.1063/5.0269660.

[11] B. M. Nuritdinovich and O. M. Uktamjonovich, "AI-driven VIN verification and RFID integration for error-proofing in automotive manufacturing," Vibroengineering Procedia, 2025, [Online]. Available: https://doi.org/10.21595/vp.2025.24957.