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ANALYTICAL MODELING AND ADVANCED EXPERIMENTAL ANALYSIS OF NATURAL FREQUENCIES IN PRINTED CIRCUIT BOARDS

BY

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Abstract. This paper presents a combined theoretical, numerical and experimental investigation of the vibrational behaviour of automotive printed circuit boards (PCBs), focusing on natural frequencies, mode shapes and the spatial distribution of the dynamic response. Two classical plate formulations, Kirchhoff–Love and Reissner–Mindlin theory, are implemented in MATLAB to predict modal parameters for two automotive electronic control unit (ECU) case studies (Audi A4 and Volkswagen Polo). Because analytical models cannot fully reproduce the behaviour of populated boards, experimental validation was performed using Laser Doppler Vibrometry, including full-field out-of-plane scanning, three-dimensional (SMART 3D-Scan) vibration analysis and single-point 3D sensing, in accordance with ISO 16750-3. The correlation between numerical and experimental results supports early-stage design assessment, optimized component placement and improved solder-joint reliability, providing a consistent methodology for evaluating PCB dynamic performance under realistic automotive operating conditions.

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Keywords: printed circuit boards, vibration analysis, plate theory, Laser Doppler Vibrometry, modal identification.

1. Introduction

Modern automotive electronic control units (ECUs) integrate increasingly dense printed circuit boards (PCBs) that must operate reliably under continuous mechanical vibration generated by the engine, the road surface and the powertrain. PCBs are complex multilayer structures with unevenly distributed mass and stiffness, since components such as microcontrollers, capacitors, connectors and transistors are mounted directly on the laminate. Material properties and mass distribution are therefore non-uniform across the board, and the boundary conditions imposed by the housing, connectors and fastening points significantly influence the dynamic response of the assembly (Steinberg, 2000).

When the excitation frequency approaches one of the board's natural frequencies, resonance amplifies the vibration amplitude and produces high local stresses, particularly in the solder joints and near heavy or poorly supported components, accelerating fatigue failure and compromising the reliability of the electronic system. Accurate prediction and experimental verification of the modal behaviour of PCBs is therefore an essential step in the design and validation of automotive-grade electronic assemblies (Rao, 2011).

Classical plate theories provide an efficient analytical basis for predicting the natural frequencies and mode shapes of PCBs at an early design stage, before a physical prototype exists. However, analytical models built on idealized boundary conditions and homogenized material properties cannot fully reproduce the behaviour of a populated PCB, where discrete components, connectors and local stiffening effects introduce significant deviations from theoretical predictions. Experimental validation using Laser Doppler Vibrometry (LDV) is therefore required, offering a mature, non-contact means of obtaining detailed, high-resolution vibration data across the whole board surface (Timoshenko and Woinowsky-Krieger, 1959; Reddy, 2007; Polytec GmbH, 2020).

This paper presents a combined theoretical, numerical and experimental study of the vibrational behaviour of two automotive ECU boards (Audi A4 and Volkswagen Polo). The theoretical framework, based on Kirchhoff–Love and Reissner–Mindlin plate theories, is implemented numerically in MATLAB to compute natural frequencies and mode shapes. The numerical predictions are compared against the qualitative and quantitative information obtained experimentally, through three complementary Laser Doppler Vibrometry techniques: full-field out-of-plane scanning, three-dimensional (SMART 3D-Scan) vibration analysis and single-point 3D measurement, performed in accordance with the vibration test requirements of ISO 16750-3 for road-vehicle electrical and electronic equipment. This work extends the preliminary

comparative results reported for the same two ECU architectures (Bumbea *et al.*, 2026; ISO 16750-3, 2023).

2. Theoretical modeling of the PCBs.

A PCB can be reasonably idealized as a thin elastic plate of uniform, or piecewise-uniform, thickness h for the purpose of predicting its global natural frequencies. Two classical formulations are commonly used for this purpose: the Kirchhoff–Love (thin-plate) theory and the Reissner–Mindlin (moderately thick, first-order shear-deformation) theory. The two models differ essentially in the kinematic assumptions used to describe the deformation of the normal to the mid-surface of the plate, as illustrated in Fig. 1, and are applicable to different thickness-to-span ratios h/L , as summarized in Table 1.

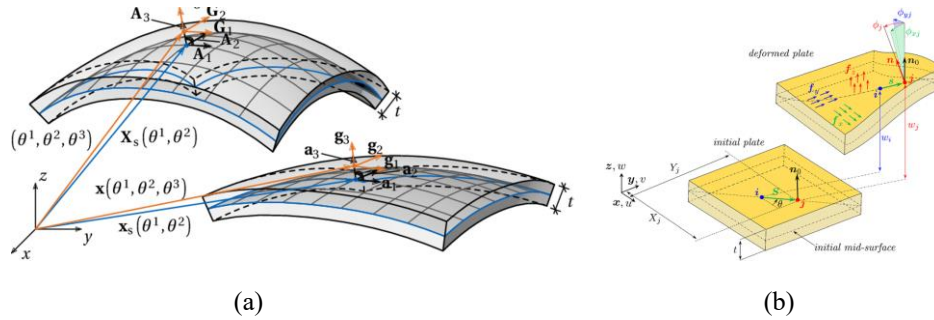


Fig. 1 – Kinematics of the plate normal: (a) Kirchhoff–Love theory; (b) Reissner–Mindlin theory.

2.1. Kirchhoff–Love plate theory

The Kirchhoff–Love (classical) plate theory assumes that straight lines normal to the mid-surface before deformation remain straight, inextensible and perpendicular to the mid-surface after deformation (Fig. 1a), which restricts its validity to very thin plates ($h/L \ll 1$). Consequently, transverse shear deformation and rotary inertia are neglected, and the plate is governed purely by bending stiffness. The free-vibration equation of motion for the transverse displacement $w(x,y,t)$ of a homogeneous, isotropic plate — known as the Kirchhoff equation — is given by Eq. (1),

$$D \nabla^4 w(x,y,t) + \rho h \frac{\partial^2 w(x,y,t)}{\partial t^2} = q(x,y,t) \quad (1)$$

where: $w(x,y,t)$ - transverse displacement of the plate at point (x,y) at time t [m], D is the flexural rigidity of the plate [Nm] and is known as:

$$D = \frac{Eh^3}{12(1-\nu^2)} \quad (2)$$

with E the Young's modulus [N/m^2], h the plate thickness [mm], ν the Poisson's ratio, ρ the mass density of the plate material [kg/m^3] and $q(x,y,t)$ the external distributed load [N/m^2]. Assuming harmonic motion and a simply-supported rectangular plate of in-plane dimensions $a \times b$, the natural angular frequencies ω_{mn} follow directly from Eq. (1) as Eq. (3),

$$\omega_{mn} = \pi^2 \sqrt{\frac{D}{\rho h}} \left[\left(\frac{m}{a} \right)^2 + \left(\frac{n}{b} \right)^2 \right] \quad (3)$$

where m and n are the number of half-waves in the x and y directions, respectively, each pair (m,n) defining a distinct vibration mode. The corresponding natural frequencies, expressed in Hz, are obtained from Eq. (4),

$$f_{mn} = \frac{\omega_{mn}}{2\pi} \quad (4)$$

Because shear deformation is neglected, this theory is accurate for thin PCBs (span-to-thickness ratio typically above 15–20), but the omitted shear compliance makes the plate appear stiffer than it really is, so the theory systematically overestimates the natural frequencies of thicker or heavily populated boards (Kirchhoff, 1850).

2.2. Reissner–Mindlin plate theory

The Reissner–Mindlin theory relaxes the normality assumption of the Kirchhoff–Love theory: normals to the mid-surface are allowed to rotate independently of the slope of the mid-surface, so that transverse shear deformation is included (Fig. 1b), extending validity to thin-to-moderately-thick plates ($h/L \leq 0.1$). The independent kinematic variables are the transverse displacement w and the two independent rotations θ_x and θ_y of the normal. For a rectangular plate, the natural angular frequencies are expressed in closed form as Eq. (5)–(7) and Eq. (8),

$$c_{mn}^2 = \left(\frac{m\pi}{a} \right)^2 + \left(\frac{n\pi}{b} \right)^2 \quad (5)$$

$$I = \frac{h^3}{12} \quad (6)$$

$$G = \frac{E}{2(1+\nu)} \quad (7)$$

$$\omega_{mn} = \sqrt{\frac{Dc_{mn}^4 + \kappa G h c_{mn}^2}{\rho h + \rho I c_{mn}^2}} \quad (8)$$

where G is the shear modulus [Pa], κ is the shear correction factor ($\kappa \approx 5/6$ for a homogeneous rectangular cross-section) and $I = h^3/12$ is the rotary-inertia term per unit width. In Eq. (8), the term Dc_{mn}^4 represents the bending contribution, $\kappa G h c_{mn}^2$ the shear contribution and $\rho I c_{mn}^2$ the rotary-inertia contribution to the dynamic stiffness. The corresponding natural frequencies in Hz are obtained from Eq. (9),

$$f_{mn} = \frac{\omega_{mn}}{2\pi} \quad (9)$$

For thin PCBs, the shear and rotary-inertia terms in Eq. (8) become negligible and the Reissner–Mindlin solution converges toward the Kirchhoff–Love result; for thick or heavily populated boards, however, the Reissner–Mindlin model consistently predicts lower and more realistic natural frequencies than the Kirchhoff–Love model. Because it accounts for transverse shear deformation and rotary inertia, the Reissner–Mindlin theory provides a more accurate description of moderately thick or multilayer PCBs, and captures higher-order modes and localized component effects more reliably than the Kirchhoff–Love model (Mindlin, 1951; Reissner, 1945).

The main assumptions of the two theories, and their practical implications for PCB modal analysis, are summarized comparatively in Table 1.

Table 1
Comparison between Kirchhoff–Love and Reissner–Mindlin plate theories for PCB modal analysis

Criterion	Kirchhoff–Love (KL)	Reissner–Mindlin (RM)
Thickness-to-span ratio	$h/L \ll 1$	$h/L \leq 0.1$
Typical PCB applicability	Thin PCBs, 1–1.6 mm, 1–4 layers	Thick / multilayer PCBs, 6–14 layers, automotive-grade
Physical interpretation	Thickness effects negligible	Thickness effects significant
Shear deformation	Neglected	Included ($\kappa \approx 5/6$)
Rotary inertia	Neglected	Included
Governing stiffness behaviour	Pure bending stiffness dominates	Coupled bending + shear stiffness
Effect on natural frequencies	Tends to overestimate f_n , especially for thicker PCBs	More accurate, systematically lower prediction of f_n

Criterion	Kirchhoff–Love (KL)	Reissner–Mindlin (RM)
Sensitivity to multilayer structure	Low	High (captures layer effects)
Applicability in PCB modelling	Simplified analytical estimation	Engineering-grade modelling
Experimental correlation	Moderate	High

3. Numerical Implementation in MATLAB

The two plate formulations described in Section 2 were implemented in MATLAB to compute the natural frequencies of two real automotive ECU boards used as case studies: an Audi A4 engine-control PCB and a Volkswagen Polo engine-control PCB. Both boards belong to the same general architecture — a rigid laminate populated with a microcontroller unit (MCU), flash memory, a QFP driver circuit, MOSFETs, signal drivers, capacitors and input/output connectors — but differ in their overall plan dimensions, layer stack-up and component layout, which is reflected in their computed dynamic behaviour.

For each board, the plate geometry ($a \times b$), thickness h and equivalent homogenized material properties (E , ν , ρ) were introduced, and Eqs (3) and (5)–(8) were solved numerically for boundary conditions representative of the actual mounting configuration. The first six natural frequencies were extracted for both plate theories and both ECU geometries and are summarized in Table 2.

Table 2
Computed natural frequencies of the first six modes (Kirchhoff–Love vs. Reissner–Mindlin)

Mode (m,n)	Audi A4 – KL [Hz]	Audi A4 – RM [Hz]	VW Polo – KL [Hz]	VW Polo – RM [Hz]
(1,1)	302.2	229.4	287.0	173.2
(2,1)	611.4	328.9	580.6	320.3
(1,2)	899.5	393.7	854.2	442.2
(3,1)	1126.8	448.0	1070.1	524.1
(2,2)	1208.8	458.8	1147.9	555.6
(3,2)	1724.1	550.5	1637.4	721.4

As shown in Table 2, the fundamental mode (1,1) of both boards corresponds to a single, symmetric dome-shaped deflection, while the higher-

order modes (2,1), (1,2), (3,1), (2,2) and (3,2) display an increasing number of nodal lines and more complex, mixed-sign amplitude patterns. Under the Kirchhoff–Love formulation, the two ECU geometries show closely spaced fundamental frequencies (302.2 Hz for the Audi A4 board vs. 287.0 Hz for the VW Polo board), whereas under the Reissner–Mindlin formulation the gap between the two boards widens considerably (229.4 Hz vs. 173.2 Hz), reflecting the greater sensitivity of the more dispersed VW Polo layout to shear compliance and rotary inertia.

For both boards, the Reissner–Mindlin predictions are systematically and markedly lower than the Kirchhoff–Love predictions, and the gap widens with mode order — for example, for the Audi A4 board, mode (3,2) is predicted at 1724.1 Hz by the Kirchhoff–Love model but only 550.5 Hz by the Reissner–Mindlin model. This confirms that neglecting transverse shear deformation and rotary inertia causes the Kirchhoff–Love model to overestimate the effective stiffness of the plate, an effect that becomes increasingly significant at higher mode orders, where the modal wavelength approaches the plate thickness. This is why the choice of plate theory has a first-order effect on the predicted dynamic behaviour of a populated PCB, and why the Reissner–Mindlin formulation is recommended as the more conservative reference for automotive multilayer ECUs — while the numerical predictions of either theory must ultimately be confirmed experimentally.

4. Experimental Methodology — Vibration Measurement

The main objective of the experimental campaign is the accurate identification of the natural frequencies and the corresponding mode shapes of the tested ECU boards under vibration conditions representative of the automotive environment, as defined by ISO 16750-3, together with a direct visualization of the spatial distribution of the dynamic response. Because the mechanical response of a populated PCB is not uniformly distributed, but concentrated in regions of maximum modal amplitude and structural flexibility, identifying these localized high-amplitude regions is critical for assessing resonance behaviour and identifying potential structural weak points under real operating conditions (ISO 16750-3, 2023).

Laser Doppler Vibrometry (LDV) was selected as the primary experimental technique because it allows non-contact, full-field vibration measurement with high spatial and frequency resolution, without adding mass or stiffness to the tested structure — an important requirement given the small size and low mass of PCB assemblies. Three complementary LDV-based measurement configurations were used in this study: full-area out-of-plane scanning, three-dimensional (SMART 3D-Scan) vibration analysis, and 3D single-point measurement (SMART 3D-Fiber), summarized in Table 3 and detailed in the following subsections (Polytec GmbH, 2020).

Table 3
Overview of the experimental vibration-measurement methods

Method	Measured quantities	Typical application
Full-area out-of-plane measurement	Out-of-plane (z) velocity, dense point grid	Global modal analysis, FEM/analytical validation
3D vibration analysis (SMART 3D-Scan)	X, Y, Z velocity at every grid point	Complete 3D modal analysis, coupled in-/out-of-plane dynamics
3D single-point measurement (SMART 3D-Fiber)	X, Y, Z velocity at a single point	ECU testing, long-term monitoring of critical locations

4.1. Full-area out-of-plane measurement

The objective of this measurement is to obtain a complete full-field characterization of the out-of-plane vibration behaviour of the PCB under controlled excitation. The principle is based on scanning Laser Doppler Vibrometry, in which the vibration velocity is measured sequentially at a dense grid of points across the board surface, enabling reconstruction of the operating deflection shapes (ODS) in the out-of-plane (z) direction. The PCB is mounted on an electrodynamic shaker, which generates a controlled and repeatable excitation representative of real operating conditions, while a scanning LDV head is positioned above the specimen on a rigid support frame, aligned perpendicular to the board surface (Fig. 2).



Fig. 2 – Scanning LDV experimental setup: vibrometer head and electrodynamic shaker.

The measurement grid is defined interactively in the SMART Lab software, directly on the live camera image of the PCB surface, allowing different mesh densities to be applied locally: a coarser grid over the full board and a denser grid near components or critical regions requiring higher spatial resolution (Fig. 3). The software also supports multilayer scan configurations, enabling both global board measurements and detailed component-level analysis within the same project. For the present measurements, the acquisition bandwidth was set to a maximum frequency of 100 kHz, sufficient to capture both the global board resonances and higher-order, component-level modes.

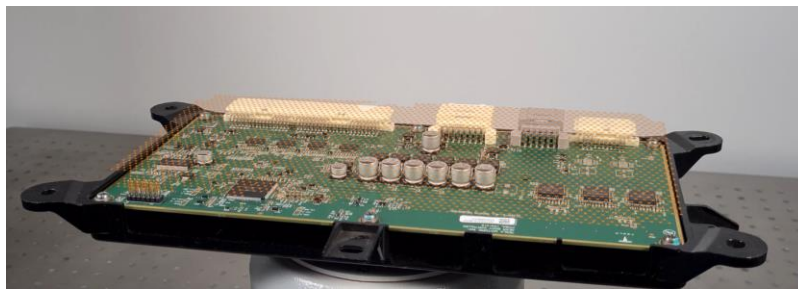


Fig. 3 – Measurement grid defined on the ECU PCB surface in SMART Lab.

After scanning, SMART Lab provides several visualization modes, including interpolated colour maps, contour plots and animated operating deflection shapes, which allow direct observation of the global mode shapes and resonance behaviour across the PCB surface (Fig. 4). The highest dynamic response is typically observed near connectors, board edges and component-dense areas, and component-level analysis further reveals that capacitors and integrated-circuit packages may exhibit distinct local vibration responses, including vertical displacement and lateral tilting.

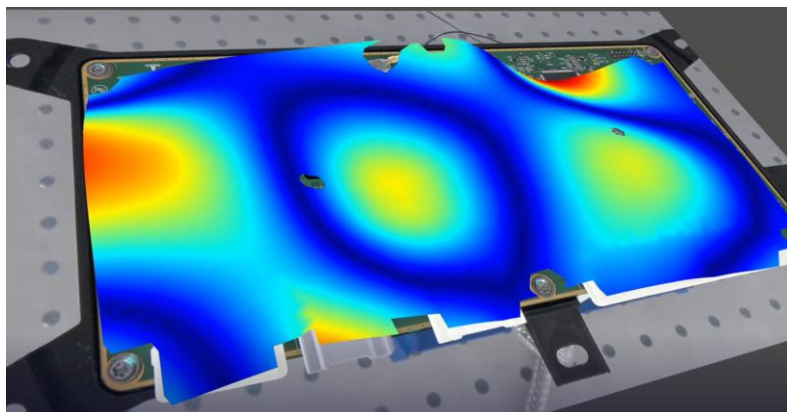


Fig. 4 – Full-field operating deflection shape (ODS) obtained by scanning LDV.

4.2. Three-dimensional vibration analysis (SMART 3D-Scan) and single-point measurement (SMART 3D-Fiber)

While the full-area scanning technique described above captures only the out-of-plane (z) component of motion, many PCB failure mechanisms—particularly those associated with solder-joint fatigue, lead deformation, and connector reliability—are also significantly influenced by in-plane (x, y) vibrations. To overcome this limitation, the SMART 3D-Scan system employs three synchronized SMART Scan+ laser vibrometer heads positioned at different viewing angles around the specimen (Fig. 5a). This multi-head configuration enables the simultaneous measurement of vibration velocity in the X , Y , and Z directions at every scanned location, providing a complete three-dimensional characterization of the structural response.

A complementary technique, used when full-field scanning is not required but a complete description of the local dynamic response at a specific critical location is still needed, is the single-point three-dimensional measurement. The SMART 3D-Fiber system (Fig. 5b) performs a true 3D vibration measurement at a single point by using three laser beams that converge on the same surface location, allowing simultaneous acquisition of the vibration velocity components in the X , Y and Z directions. This enables a complete description of the local dynamic response—including both out-of-plane and in-plane motion components not captured by conventional single-axis accelerometers—and is particularly suitable for long-term ECU monitoring at a fixed critical location, complementing the full-field modal analysis described above.

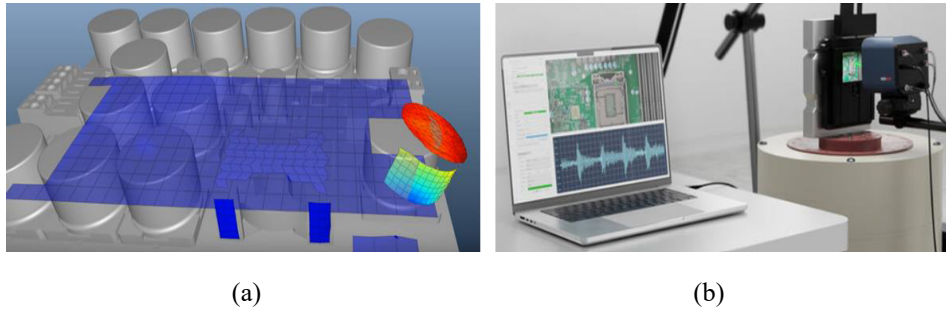


Fig. 5 – (a) SMART 3D-Scan multi-sensor configuration; (b) SMART 3D-Fiber single-point measurement setup.

Table 4 summarizes how each of the three experimental configurations relates to the theoretical framework of Section 2, indicating which aspect of the Kirchhoff–Love or Reissner–Mindlin kinematics it is best suited to validate.

Table 4
Correspondence between the experimental LDV configurations and the analytical plate models

Experimental method	Motion components captured	Plate-theory aspect validated	Primary use in this study
Full-area out-of-plane scanning	Out-of-plane velocity (z)	Global bending mode shapes, common to KL and RM	Global modal identification and analytical/FEM correlation
3D vibration analysis (SMART 3D-Scan)	X, Y, Z velocity, every grid point	Coupled bending–shear behaviour (RM)	Full 3D operating deflection shapes
3D single-point (SMART 3D-Fiber)	X, Y, Z velocity, single point	Local rotation and shear effects near components	Long-term monitoring of critical, high-amplitude locations

5. Results and Discussion

The combination of the analytical–numerical framework (Section 3) and the experimental methodology (Section 4) provides a consistent basis for evaluating the dynamic behaviour of automotive ECU boards from complementary perspectives. The numerical results (Table 2) confirm that the two classical plate theories, although based on the same underlying geometry and material assumptions, produce systematically different predictions — with the Kirchhoff–Love model overestimating the natural frequencies relative to the Reissner–Mindlin model, an effect that becomes more pronounced at higher mode orders — which highlights the importance of selecting a plate model appropriate to the thickness ratio and construction of the board under investigation. The experimental results, obtained through full-field scanning, three-dimensional vibration analysis and single-point measurement, confirm that the vibration response of a populated ECU board is far from uniform: the highest dynamic amplitudes were consistently observed near connectors, board edges and component-dense regions, in agreement with the localized, mixed-sign amplitude patterns predicted numerically for the higher-order modes.

This qualitative correlation between numerical mode shapes and experimentally observed operating deflection shapes supports the general validity of the plate-theory-based numerical models, while also confirming that discrete components, connectors and non-ideal boundary conditions — not represented in the idealized analytical model — introduce local stiffness and mass variations that measurably shift the actual resonance frequencies relative to the analytical

predictions. From a practical standpoint, these findings support several recommendations: (i) natural frequencies identified through the combined methodology should be compared against the excitation spectrum expected in the vehicle, defined according to ISO 16750-3, to avoid resonance in the operating range; (ii) components located in regions of high modal amplitude, particularly near connectors and board edges, should be given special attention during solder-joint design; and (iii) because it consistently yields lower, more conservative frequency estimates and captures the coupled bending–shear behaviour confirmed experimentally by the SMART 3D-Scan measurements (Table 4), the Reissner–Mindlin formulation should be preferred whenever board thickness, multiple laminate layers, or the requirement for higher-order modal accuracy make shear-deformation and rotary-inertia effects significant.

6. Conclusions

This paper presented a combined theoretical, numerical and experimental investigation of the vibrational behaviour of two automotive ECU printed circuit boards. The combination of Kirchhoff–Love and Reissner–Mindlin plate theories provided a rigorous theoretical basis for modelling PCB vibrational behaviour across thin and moderately thick structures, and their implementation within a MATLAB-based computational framework allowed the systematic computation of natural frequencies for two real board geometries.

The numerical results showed that the two theories produce substantially different predictions, with the Reissner–Mindlin model systematically lower than the Kirchhoff–Love model and the gap widening at higher mode orders, confirming that an appropriate choice of plate theory is essential for reliable early-stage design assessment. Because numerical models alone cannot fully capture the behaviour of real, populated PCB assemblies — owing to the presence of discrete components, connectors and non-ideal boundary conditions — experimental validation using Laser Doppler Vibrometry was performed, combining full-field scanning, three-dimensional vibration analysis and single-point measurement to obtain complementary global and local descriptions of the dynamic response.

The integrated theoretical–numerical–experimental framework established in this study represents a robust and consistent methodology for the validation of PCB dynamic models, supporting engineering decisions such as optimized component placement, improved solder-joint reliability and early identification of resonance-related risks, before physical prototyping or production implementation. Future work will focus on extending the experimental correlation to a broader set of automotive ECU geometries, and on incorporating the measured, non-uniform mass and stiffness distribution of mounted components directly into the numerical plate models.

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MODELARE ANALITICĂ ȘI ANALIZĂ EXPERIMENTALĂ AVANSATĂ A FRECVENȚELOR PROPRII ALE PLĂCILOR DE CIRCUIT IMPRIMAT

(Rezumat)

Lucrarea prezintă o investigație teoretică, numerică și experimentală combinată a comportamentului vibratoriu al plăcilor de circuit imprimat (PCB) din echipamente electronice auto, cu accent pe frecvențele proprii, formele proprii de vibrație și distribuția spațială a răspunsului dinamic. Două formulări clasice de placă, teoria Kirchhoff–Love și teoria Reissner–Mindlin, sunt implementate în MATLAB pentru a prezice parametrii modali pentru două studii de caz de unități de control electronic (ECU) auto (Audi A4 și Volkswagen Polo). Deoarece modelele analitice nu pot reproduce integral

comportamentul plăcilor populate cu componente, validarea experimentală a fost realizată prin vibrometrie laser Doppler, incluzând scanare completă în afara planului, analiză vibrațională tridimensională (SMART 3D-Scan) și măsurători 3D într-un singur punct, în conformitate cu standardul ISO 16750-3. Corelația dintre rezultatele numerice și cele experimentale susține evaluarea timpurie în proiectare, optimizarea amplasării componentelor și îmbunătățirea fiabilității îmbinărilor prin lipire, oferind o metodologie consistentă pentru evaluarea performanțelor dinamice ale PCB-urilor în condiții reale de exploatare auto.