JOURNAL OF ACOUSTICS AND VIBRATION RESEARCH (JAVR)
ISSN: XXXX-XXXX
VOL. 3, ISSUE 1, 2025, 1 – 13

Identification of the Modal Parameters of a Golf Club

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ABSTRACT

The dynamic behaviour of a golf is a critical role in its performance, user comfort, and structural durability during both the swing and impact phases. An Accurate understanding of its vibration characteristics, particularly the natural frequencies, mode shapes, and damping, is essential for optimizing its design and enhancing the user experience. However, most numerical models of golf clubs rely solely on finite element predictions with limited experimental validation, leading to significant discrepancies in mode shape and frequency correlation. The absence of experimentally validated data limits the reliability of such models in real-world performance evaluation. This study aims to identify the modal parameters of a golf club through both numerical and experimental approaches. A finite element model of a golf club is developed from a measured CAD model. The FE model created using an optimal element size of 2 mm is calibrated to match the Experimental Modal Analysis (EMA) results of the test club. EMA is performed on the golf club using an impact hammer and roving accelerometers under free boundary constraints. The comparison between the FE and EMA results shows substantial discrepancies, indicating the inability of the FE model to accurately reproduce the dynamic behaviour of the golf club. The results clearly demonstrate that accurate material properties and experimental validation are essential for reliable dynamic modelling. This study establishes a foundation for improving golf club design through integrated numerical–experimental approaches.

Article History

Received: 01/12/2024

Revised: 21/02/2025

Accepted: 30/04/2025

Published: 30/06/2025

Keywords: Golf Club; Experimental Modal Analysis; Finite Element Model; Modal Parameters

INTRODUCTION

The vibration behaviour of a golf club is a critical factor affecting performance, player comfort and structural durability throughout the swing and impact phases [1]. When the club accelerates and then strikes the ball, bending and torsional waves travel along the shaft and reflect at the head-shaft interface. These waves govern the tactile feedback perceived by a golfer and influence both accuracy and energy transfer to the ball. A clear understanding of modal parameters of the club, namely natural frequencies, mode shapes and damping ratios is therefore important for optimising design, minimising unwanted vibrations and extending service life.

Numerical approaches based on the FE method are widely used to predict the modal parameters of sports equipment, as they allow detailed stress and deformation fields can be obtained at low cost and without destructive testing. Studies on composite shafts, for example CFRP, and on forged steel clubheads have shown that FE analysis can capture first bending and torsional modes with reasonable accuracy, provided that material properties and boundary conditions are precisely defined [2], [3], [4]. Mesh convergence studies on beam-type structures further demonstrate that reliable modal predictions require adequate element refinement [5].

However, several authors have reported that purely numerical predictions deviate significantly from experimental observations when composite lay-ups, adhesive layers or complex head geometries are involved [6], and that causes the frequency errors to exceed 150% when material stiffness or damping is assumed rather than measured experimentally [7]. Recent constraint-based modal substructuring techniques enable faster simulation, they still depend on accurate stiffness inputs to prevent large prediction errors [8]. EMA, which uses impact hammer excitation and roving accelerometers, is therefore regarded as the benchmark for validating FE models and for showing the true vibration response under realistic boundary constraints [9].

Despite these advances, few studies combine high-resolution FE modelling with full EMA validation on commercially available clubs that use steel shafts and forged carbon-steel heads. These configurations remain widely used by both amateur and professional players for wedges and irons. Most published work has focused on graphite shaft drivers or isolated clubfaces without the shaft, leaving a clear knowledge gap for all steel assemblies [10]. For example, the author [11] showed that showed that neglecting hand-grip interaction in a steel-shafted tennis racket caused frequency predictions errors exceeding 30% while Habibi and Yazdi [12] demonstrated that grip strength variations can alter both damping and mode order during a swing. Similar validation challenges have been observed for badminton rackets, where simplified FE models tend to over-predict frequencies and damping ratios [9]. Inverse model-updating strategies, first developed for civil structures, have since been adapted for sports equipment to iteratively tune stiffness parameter and improve modal correlation [13]. However, due to limited validated data for steel-shafted clubs, manufacturers continue to rely heavily on iterative prototyping instead of simulation-driven optimisation.

The present study addresses this gap by identifying the modal parameters of a Callaway Mack Daddy 2 wedge fitted with a True Temper Dynamic Gold S200 steel shaft using a combined numerical and experimental approach. A simplified CAD model of the club is created and meshed for FE analysis, while EMA is performed under free-free boundary conditions using an instrumented impact hammer and roving accelerometers. The resulting natural frequencies and mode shapes are compared using the MAC analysis to evaluate the quality of correlation between the FE and EMA results.

METHODOLOGY

Structural Modelling

The reference club for this study is a Callaway Mack Daddy 2 wedge fitted with a True Temper Dynamic Gold S200 steel shaft. The overall geometry was acquired in two stages. First, the club head outline, leading-edge radius and principal shaft diameters were measured using a CMM touch probe (repeatability: ± 0.02mm). Second, the features inaccessible to the probe, such as the trailing-edge relief, hosel undercut and shaft-head fillet were measured manually with a standard vernier calliper (resolution: 0.05 mm, accuracy: ~ ± 0.10 mm) and photographed for visual cross-checking. Although detailed manufacturer drawings were unavailable, the combined-measurement dataset provided sufficient fidelity to reproduce the overall shape and mass distribution necessary for modal analysis.

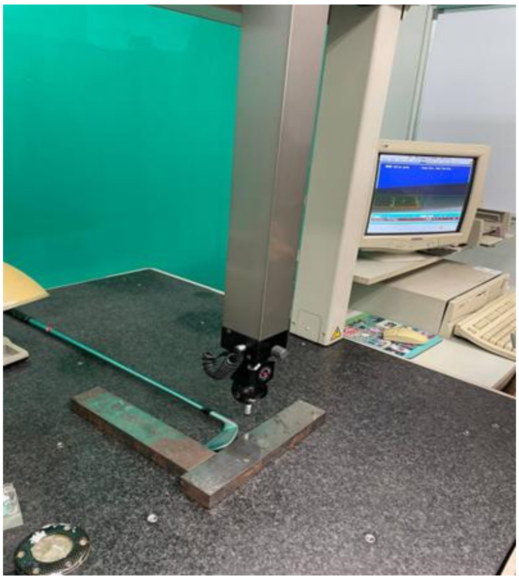


Figure 1: Measuring golf club using CMM probe

A three-dimensional solid model was created in CATIA V5, as shown in Figure 2 below. Cosmetic details such as the scorelines, laser engravings, and fillets smaller than 0.5 mm were suppressed to limit element count. The shaft was idealised as a smoothly tapered cylinder interpolated from the measured butt and tip diameters, while the hosel bore was modelled as a uniform circular hole. The moulded rubber grip was excluded because EMA was conducted on the bare shaft where its mass accounted for later when tuning density. Despite these simplifications, the overall model retains the correct total mass once the density is adjusted in the FE analysis stage.

Figure 2: CAD model of the golf club- the completed CAD model, exported in Parasolid format, forms the geometric basis for the subsequent FE and EMA investigations.

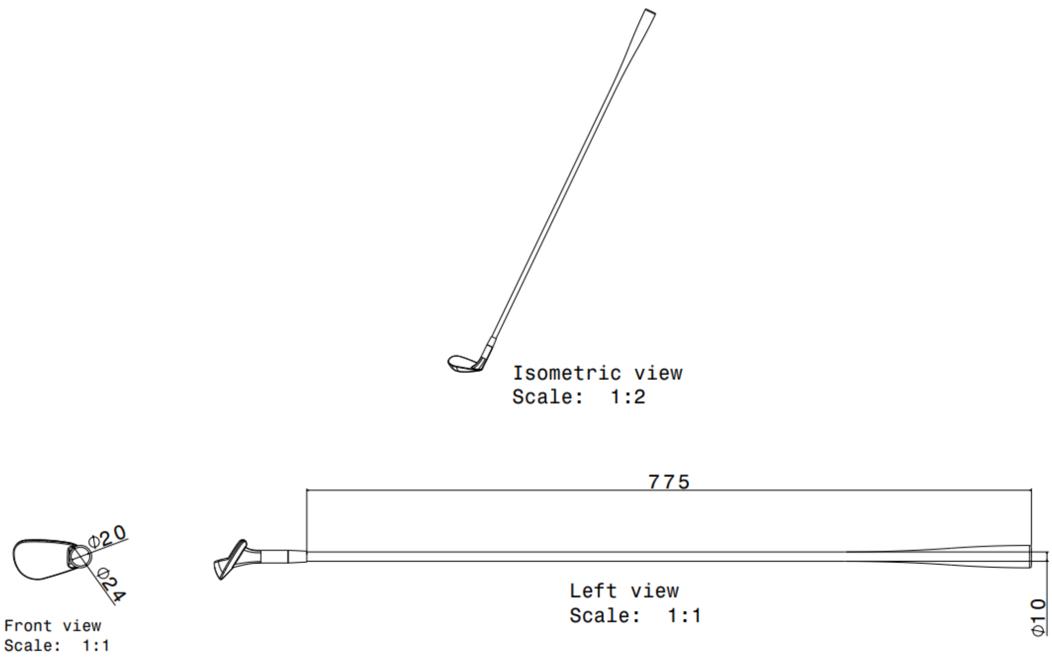


Figure 3: Drawing of the golf club

Finite Element Modelling of the golf club

The 3D CAD model of the golf club was imported into MSC Patran for finite element preprocessing. Meshing was carried out using volume tetra meshing with R-trias element shapes selected for its ability to conform to curved and tapered surfaces. This element type, commonly used in structural vibration analysis, enables improved accuracy in capturing local stiffness distribution while maintaining a balanced element count. A uniform global mesh density was used across both the clubhead and shaft regions to maintain consistency in element quality and avoid artificial stiffness transitions. The mesh was generated to balance computational cost with sufficient detail to capture the overall vibration characteristics of the structure.

Material properties were defined as shown in the table below assuming uniform steel throughout the model. Young's modulus was set to E = 20000 MPa, Poisson's ratio to $\nu = 0.30$ and the density was assigned as $\rho = 1.413 \times 10^{9}$ kg/mm^3. This density value was specifically tuned to ensure that the total mass of the FE model matches the measured physical mass of the actual golf club (0.469 kg), thereby compensating for the geometric simplifications applied during CAD modelling.

Table 1: Assumed material properties of composite material

Modulus of Elasticity, E	**Poisson's Ratio, ν**	**Density, ρ**
20000MPa	0.3	1.413×10^{-9} kg/mm^3

Free-free boundary constraints were applied by leaving all six degrees of freedom unconstrained, allowing the model to represent the suspended setup used in EMA. Normal mode extraction was performed in MSC Nastran using the Lanczos eigenvalue solver. The first 10 natural frequencies and corresponding mode shapes were computed and stored for comparison with experimental results via the MAC analysis.

Figure 4: FE using Normal Modes Analysis flow chart

Mesh Convergence Study

A mesh convergence study was conducted to determine the appropriate element size that balances computational efficiency with modal accuracy. Five different element sizes were tested, namely, 8 mm, 6 mm, 4 mm, 2 mm and 1 mm. For each mesh density, a normal mode analysis was performed in MSC Nastran using the Lanczos eigenvalue solver under free-free boundary constraints.

Table 2: Natural frequencies in different element sizes

Element size (mm)	1st mode (Hz)	2nd mode (Hz)	3rd mode (Hz)	4th mode (Hz)	5th mode (Hz)
1	32.42	35.70	108.13	119.33	225.04
2	50.50	50.54	153.88	153.95	321.19
4	50.54	50.57	154.07	154.14	321.63
6	50.55	50.59	154.09	154.15	321.66
8	50.59	50.61	154.26	154.33	322.26

The first five natural frequencies were extracted for each element size and plotted to evaluate convergence behaviour. As shown in Figure 5 below, the predicted frequencies exhibited a decreasing trend with mesh refinement. It was observed that the results began to stabilize at the 2mm element size, with frequency shifts between successive refinements falling below 1%. This behaviour is consistent with findings from other modal convergence studies involving tetrahedral elements in structural dynamics simulations, including mechanical and aerospace system [14].

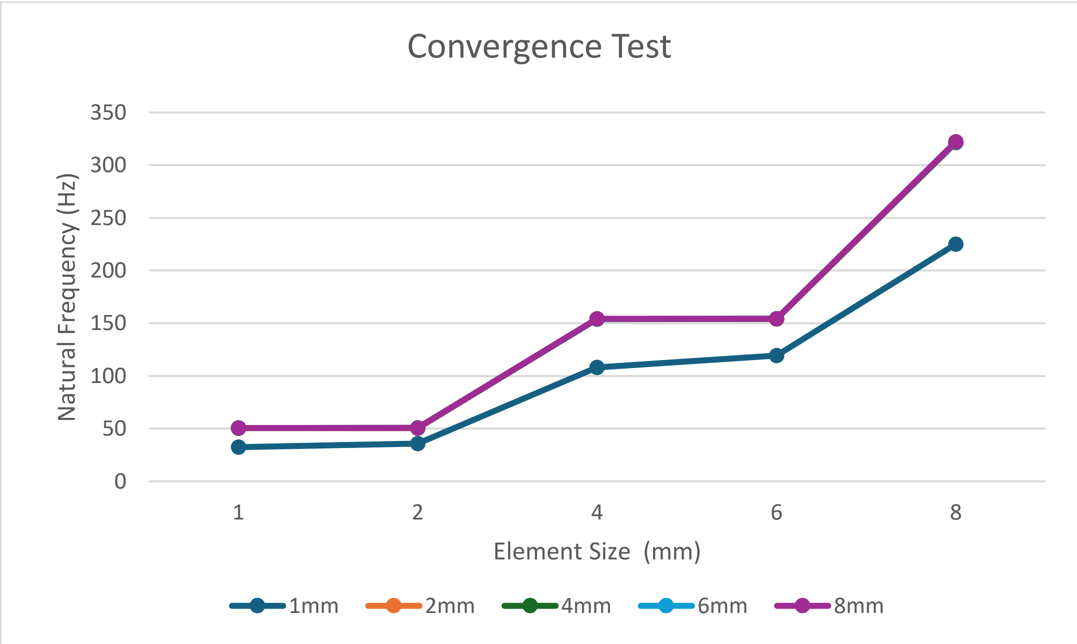


Figure 5: Convergence test (Natural frequency vs Element size)

At an element size of 2 mm, the frequency results began to stabilize, with changes falling below 1%. This threshold is considered acceptable for modal convergence in vibrational analysis, especially for tetrahedral elements, which exhibit higher mesh sensitivity is greater in complex geometries such as clubheads. This result aligns with established modal convergence trends reported in structural simulation literature for defence and aerospace applications, where convergence is critical for validating mode shape correlation [15].

Based on this trend, the 2 mm mesh was selected for the final finite element model. This configuration produced a high-resolution mesh consisting of approximately 27401 nodes and 102882 elements coming from both shaft and clubhead. While the finer mesh increases computational effort, it ensures that the predicted modal parameters are reliable enough for subsequent comparison with Experimental Modal Analysis (EMA) results. The selected mesh was then used throughout the project for mode shape extraction and MAC analysis.

Experimental Modal Analysis of the golf club

Experimental Modal Analysis (EMA) was conducted to identify the actual vibration characteristics of the golf club and to validate the finite element model. The test setup was designed to replicate free boundary constraints so that the experimental data could be directly compared with the unconstrained FE model.

The golf club was suspended vertically using thin nylon cords attached near the toe region of the clubface (Figure 6). This suspension method allowed the club to hang freely and minimized external constraint forces, effectively simulating free-free boundary conditions. While not supported at multiple points, the use of a soft, flexible cord and its attachment at a non-critical structural location helped reduce boundary influence during testing: a practice consistent with standard procedure in experimental modal analysis of irregular-shaped component.

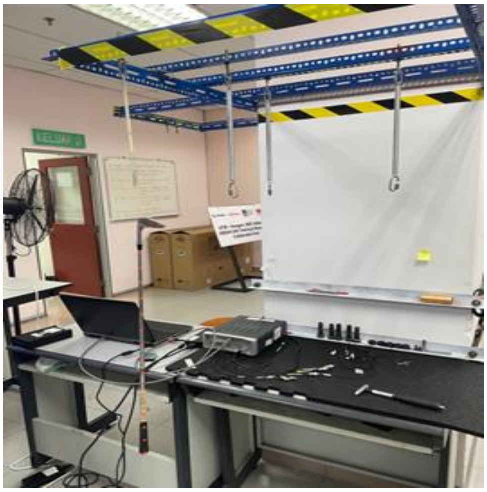


Figure 6: Experimental setup of the golf club

A PCB Piezotronics Model 086C03 instrumented impact hammer, fitted with a hard plastic tip to provide broadband input up to 5000 Hz, was used to excite the structure at a single fixed location positioned at the back edge of the clubhead surface (Figure 6). This point was selected to ensure effective excitation of both bending and torsional modes due to its position near a structurally active region. The vibration response was measured using a PCB 356A12 tri-axial accelerometer, which was roved across 24 locations (measuring nodes) on the golf club. Although only one excitation point was used, this fixed-excitation, roving-response strategy provided sufficient data for identifying the modal parameters of the golf club. This approach is commonly used in modal testing of asymmetric components where spatial coverage through response measurements is prioritized over multiple impact points [16].

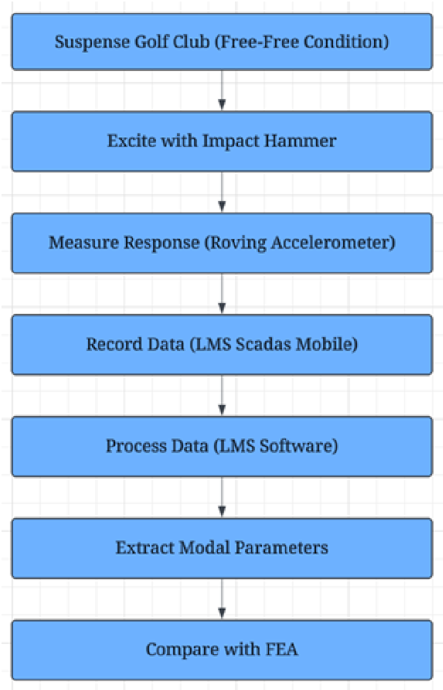


Figure 7: Modal Analysis through EMA flow chart

RESULTS AND DISCUSSION

The FE and EMA results are presented in this section. The comparisons include natural frequencies extracted from both methods, along with the calculated percentage errors. Additionally, the correlation between FE and EMA mode shapes is presented using the MAC Analysis.

Percentage error between FEA and EMA

Table 3: Comparison between the FE and EMA natural frequencies (element size = 2)

Mode	FE (Hz)	EMA (Hz)	Error (%)
1	50.50273	46.81	7.89
2	50.53985	55.23	8.49
3	153.8777	138.51	11.10
4	153.9543	166.99	7.81
5	321.1895	278.93	15.15
		Total	53.26

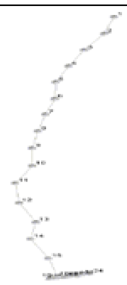
The comparison of natural frequencies between FE and EMA demonstrated significant discrepancies. As presented in Table 3 the frequencies predicted by FEA were consistently higher than those obtained through EMA, with errors ranging from 7.89% to 15.15%. Such deviations indicated that the stiffness of the golf club was overestimated in the FE model. One of the primary causes of this overestimation lies in the assumptions about material properties. Due to the lack of detailed manufacturer specifications for the Callaway Mack Daddy 2 wedge and the True Temper Dynamic Gold S200 shaft, the club was idealised as a uniform steel component. Standard values were applied for Young's modulus, Poisson's ratio and density, with the latter tuned to ensure that the total FE model mass matched the physical mass of 0.469 kg. However, the actual structure may incorporate proprietary alloy compositions or a combination of materials that significantly differ in stiffness and damping characteristics compared to the assumed material properties.

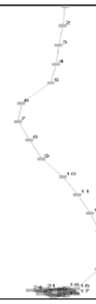
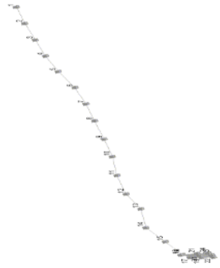
Another contributing factor to the frequency discrepancy is the simplified geometry used in the CAD model. Certain details such as surface curvature transitions, fillets, groves and fine clubface features were excluded to streamline the meshing process. These seemingly minor geometric details can have a significant impact on local flexibility and inertial properties, particularly in a structure as dynamically sensitive as a golf club [17].

Furthermore, the connection between the shaft and the clubhead was modelled as a perfectly bonded interface. This joint may include adhesive bonding or mechanical fits that introduce additional compliance. These simplifications, although necessary to some extent for model manageability, can affect the stiffness distribution and reduce the accuracy of the frequency predictions [18].

Mode shape correlation

Table 4: Mode shape correlation between FE and EMA

Mode		FE (Hz)	EMA (Hz)
1			
2			

3			
4			
5			

The mode shape correlation between FE and EMA for Modes 1 to 5 is shown in Table 4, based on the FE model with an element size of 2 mm. This mesh was selected after convergence test has been done, and the results showed that this element size produced the lowest percentage error in natural frequencies compared to the other element sizes. A visual comparison of the mode shapes shows that both the FE and EMA results display bending behaviour in these first five modes. More importantly, the direction and pattern of deformation are consistent across both domains for each respective mode. This directional agreement supports the decision to pair the modes directly based on their sequence. Although numerical discrepancies exist in terms of the magnitude of natural frequency, the FE model captured the overall bending deformation trends observed experimentally. The visual similarity between mode shapes indicates that the simplified FE model was sufficient to represent the global dynamic response of the golf club within the bending frequency range.

CONCLUSIONS

This project aimed to identify and validate the modal parameters of a golf club through both numerical and experimental approaches. All three main objectives were successfully addressed. The achieved objective includes an analytical FE model of the golf club was successfully developed using MSC Nastran. Secondly, the modal parameters including natural frequencies and mode shapes were extracted from FE and EMA. Lastly, a correlation study between both sets of results was conducted.

The FE consistently overestimated natural frequencies, with errors ranging from 7.89% to 15.15% compared to experimental values. These discrepancies were primarily due to simplified material assumptions, idealised geometry and the rigid representation of the shaft and clubhead interface.

In conclusion, this study successfully established a combined FE and EMA workflow for modal parameters identification, while also highlighting key limitations in modelling accuracy and test setup. These findings underscore the importance of accurate material characterisation, joint interface modelling and experimental excitation strategies to achieve reliable structural validation. Future work should incorporate more detailed material data, realistic boundary modelling and broader excitation coverage to improve the accuracy of validation between FE and EMA data.

ACKNOWLEDGEMENTS

This study was conducted at the Structural Dynamics Analysis and Validation (SDAV) Laboratory, Faculty of Mechanical Engineering, Universiti Teknologi MARA (UiTM). The authors would like to express their gratitude to the SDAV team for their technical support and access to laboratory facilities.

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