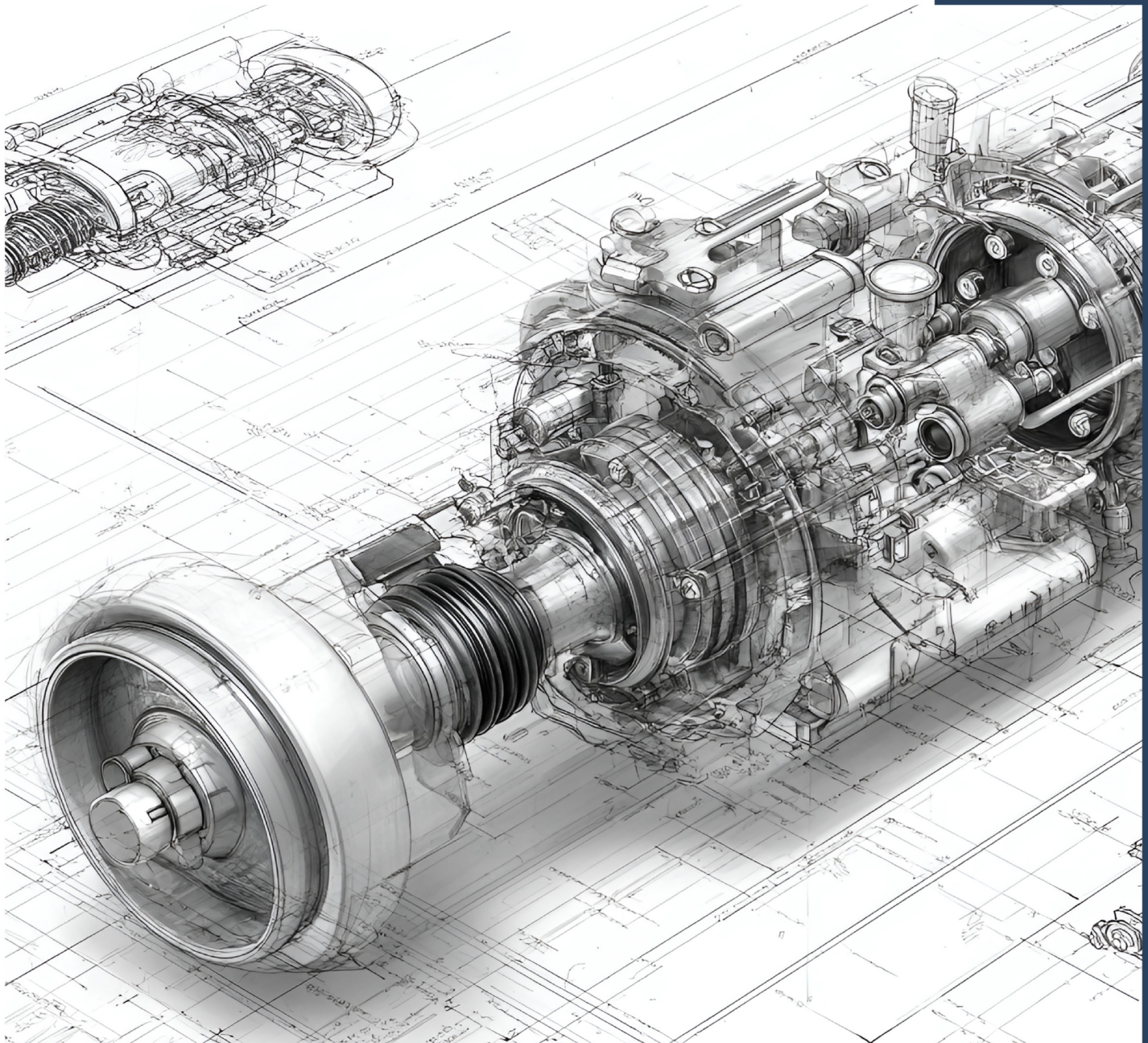


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Article

Parametric weight reduction design study of a small-diameter axial bearing for automotive tie rod applications

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Abstract: Reducing component mass in automotive steering systems is a key strategy for achieving energy consumption reduction, cost efficiency, and sustainability, provided that functional safety and durability are preserved. This study experimentally investigates the feasibility of reducing the inner tie rod ball diameter from Ø29 mm to Ø26 mm while maintaining compliance with functional and durability requirements. Unlike predominantly simulation-driven studies, an industrial, hands-on experimental methodology is employed under realistic production conditions. A total of 192 inner tie rod assemblies were manufactured, comprising a Ø29 mm reference configuration and multiple Ø26 mm variants produced by systematically varying pressing force, tempering method, and tempering temperature. All assemblies were first subjected to functional screening based on articulation torque, axial travel, and minimum axial stiffness. Only configurations satisfying all acceptance criteria were advanced to wear and setting durability tests. Each assembly was evaluated using repeated measurements, and results are reported as mean values with corresponding standard deviations. The results show that articulation torque is the most sensitive parameter in reduced-diameter designs and is strongly influenced by tempering strategy and press force. Combining ball tempering with an increased press force provided stable functional behavior before and after both wear and setting tests, satisfying all acceptance criteria. The validated Ø26 mm configuration enables an estimated 10% weight reduction, approximately 8% material-related cost savings, and associated energy consumption reduction without introducing additional manufacturing steps. The findings deliver production-ready guidance for lightweight inner tie rod design while meeting stringent automotive safety standards.

Keywords: energy efficiency; parametric design; tempering; automotive applications; articulation torque

1. Introduction

The evolution of automotive steering systems, from basic mechanical setups to complex, power-assisted designs like rack and pinion, has highlighted the critical role of tie rods in vehicle safety and performance. Tie rods, which transmit steering input from the steering rack to the wheels, consist of two parts (inner and outer) connected by a threaded joint, allowing for precise steering alignment and vehicle handling. Despite advancements in steering technology, tie rods remain fundamental to controlling vehicle movement. **Figure 1** illustrates the components of the inner tie rod assembly.

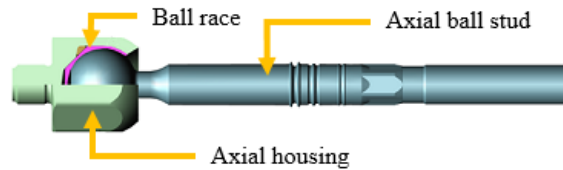


Figure 1. Inner tie rod components.

Tie rods, critical to a vehicle's steering mechanism, endure significant forces during operation, leading to wear, particularly in the plastic bearings within their ball joints, which facilitate horizontal and vertical movement. Ensuring durability is vital for safety, as tie rods are designed to fail before more expensive components like the steering gear or knuckle sustain damage.

Inner tie rods are manufactured from cold-forged steel and consist of three main components: a ball stud, housing, and plastic bearing. Manufacturing considerations include using high-strength materials like steel or heat-treated aluminum to balance strength, weight, and cost while meeting global standards such as AK-LH 14, 2014 [1]. Testing and design account for variable forces from different road conditions, ensuring performance across smooth and rough terrains.

Previous research on tie rod and ball joint assemblies has emphasized the critical role of joint design, bearing material selection, and parametric sizing in ensuring functional performance and durability. Comprehensive system-level descriptions of tie rod assemblies highlight the use of thermoplastic bearing materials such as polyoxymethylene (POM) and PEEK, as well as sealing strategies and design criteria governing load capacity and service life [2]. Parametric design approaches have further enabled durability-driven sizing of ball joints to achieve infinite fatigue life [3]. Fundamental joint behavior, including the influence of clearance, lubrication, and friction on dynamic response, has also been investigated [4]. Fatigue life prediction and experimental validation studies have underscored the sensitivity of ball joint performance to stress distribution [5]. More recently, it has been shown that reduced bearing race thickness significantly increases deformation and contact stress, reinforcing the importance of stiffness and contact mechanics in joint stability [6].

Several studies have examined failure mechanisms and reliability concerns in tie rods and ball joints under service loading. Fatigue has been identified as the dominant failure mode in tie rod ends and ball joints, often initiated at stress concentration regions and exacerbated by improper processing conditions [7,8]. Reliability-oriented methods such as failure mode, effects, and criticality analysis (FMECA) have been applied to steering tie-rod joints to assess safety implications of joint degradation [9]. Reliability-based testing strategies and statistical evaluation of tie-rod joint failures further emphasize the importance of durability validation under realistic operating conditions [10].

Numerous studies have explored the lightweighting of tie rods through material substitution and structural optimization. Significant weight reductions have been demonstrated by replacing steel with aluminum alloys and applying metamodel-based optimization while maintaining durability and buckling performance [11,12]. Topology optimization techniques have also been used to improve outer tie rod shaft designs under

buckling and fatigue constraints [13]. Other investigations have focused on geometry and material modifications to reduce mass while preserving structural integrity [14,15]. These studies primarily address global structural optimization rather than joint-level manufacturing effects.

Finite element-based studies have been widely used to evaluate the structural behavior of tie rods under operational loading. Stress distribution, deformation, vibration behavior, and natural frequencies of tie rod assemblies have been analyzed using detailed FE models [16–19]. While these works provide valuable insight into global structural performance, they generally do not address joint-level behavior or the influence of manufacturing parameters on functional performance.

Broader studies on automotive lightweighting and sustainability emphasize material efficiency, energy reduction, and environmental impact. Reviews of advanced lightweight materials and manufacturing strategies highlight approaches aimed at reducing vehicle mass, energy consumption, and emissions, providing contextual motivation for component-level weight reduction studies [20–22].

Other studies have addressed automotive lightweighting and performance improvement through system-level optimization, advanced materials, and component-specific design. Multiobjective optimization frameworks integrating CAD, CAE, and multibody dynamics have been shown to significantly improve suspension system performance while reducing mass and stress levels [23]. In parallel, material-focused studies highlight the role of advanced aluminum alloys and microstructure control in achieving improved strength-to-weight ratios and durability [24], driven by fuel economy and emission reduction targets [25]. At the component level, several works have investigated tie rod optimization through material substitution, geometric modification, and topology optimization to reduce weight while maintaining structural integrity [26–30]. Finite element-based analyses have further explored stress, vibration, fatigue, and shear failure behavior of tie rods under realistic loading conditions [31,32]. Manufacturing-oriented lightweight design strategies and forming technologies provide additional context for enabling mass reduction without compromising performance [33]. The wear intensity of the ball stud and bearing in the tie rod end was calculated in a study where real-world vehicle steering was modeled [34].

Although prior research has extensively explored tie rod lightweighting through material substitution, structural optimization, and numerical modeling of ball joint mechanics and fatigue, the combined effects of manufacturing parameters—specifically pressing force, tempering method, and tempering temperature—on the functional and durability performance of reduced-diameter ball joints remain insufficiently understood. In particular, hands-on experimental studies directly linking these process variables to articulation torque, axial displacement, and stiffness under both wear and thermo-mechanical loading are scarce. Addressing this gap, the present study conducts a systematic, industrial-scale experimental investigation under realistic production conditions to evaluate the feasibility of reducing the inner tie rod ball diameter. The results provide practical, process-level insights that support the development of lighter, cost-effective, and durable steering components

while maintaining compliance with automotive performance and safety standards, contributing to broader lightweighting and sustainability objectives.

2. Materials and methods

The materials used in the inner tie rod assembly are first introduced. Then, the study details its characteristics and production methods before arranging test samples to evaluate the key parameters of the tempering method, tempering temperature, and press force.

2.1. Materials

The inner tie rod assembly consists of three main components: the ball stud, housing, and ball race, each made from specific materials to meet performance requirements. The housing is made from C15 steel, in compliance with EN 10277-2 [35], and undergoes case hardening. It is externally manufactured through cold forging. The ball stud is produced from 30MnVS6 steel, a micro-alloyed material specified by DIN EN 10267. Manufactured via cold forging, this process eliminates the need for post-forging heat treatment, resulting in cost advantages. The ball race is made of POM, a plastic material commonly used in inner tie rod ball joints due to its exceptional durability, high crystallinity, and thermoplastic properties. POM offers excellent performance, including high strength, stiffness, and toughness, with minimal water absorption [36].

The initial tie rod diameter was set at $\varnothing 29$ mm, matching the reference design. The target ball size is $\varnothing 26$ mm. The key dimensions of the $\varnothing 29$ ball stud and its housing are illustrated in **Figures 2** and **3**, respectively. The production and testing of the inner tie rods were carried out in the labs of a private company in Türkiye.

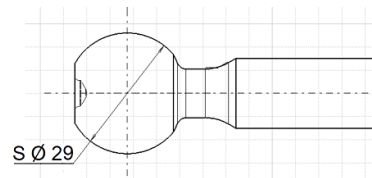


Figure 2. Inner tie rod ball stud.

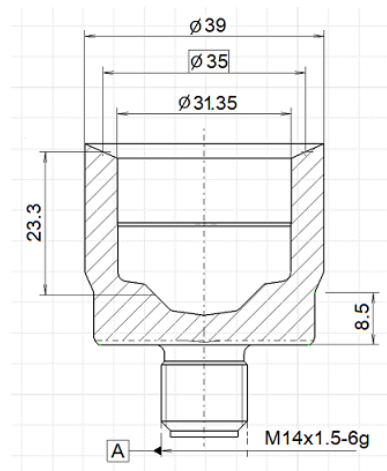


Figure 3. Housing of the inner tie rod.

2.2. Assembly procedure

Inner tie rod assemblies were produced following a controlled three-step assembly sequence, as illustrated in **Figure 4**. First, the POM ball race was inserted into the steel housing. This was followed by the insertion of the ball stud into the housing, after which the complete assembly was consolidated by axial pressing using a hydraulic press. To investigate the influence of assembly force on functional performance, three pressing force levels were employed: the minimum press force (MPF) of 120 kN, MPF + 10 kN, and MPF + 20 kN.

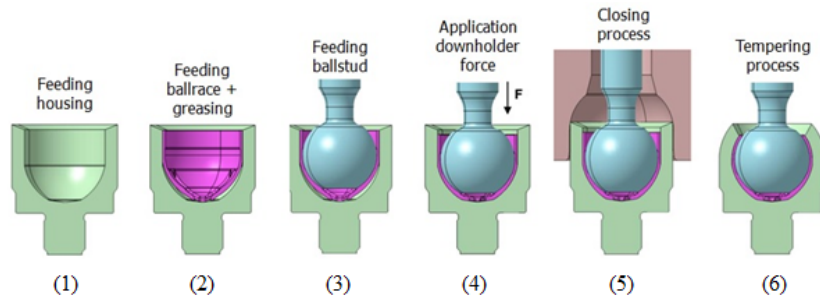


Figure 4. Steps of the assembly process.

2.3. Tempering procedure

Tempering is applied after assembly to stabilize ball–race contact and reduce articulation torque. Without tempering, ball joints experience excessive friction and high torque. To achieve optimal performance, two methods are used: oven tempering and ball tempering. The temperature and duration must be precisely controlled according to the inner tie rod’s functional requirements.

2.3.1. Oven tempering

In oven tempering, the fully assembled inner tie rod is heated to 145 °C or 155 °C using an electric oven rated at 160 °C and 55 kW. Assemblies are placed on open-frame carts to ensure uniform airflow and heat transfer. This process stabilizes ball–race contact but heats the entire assembly, including nonessential parts, which increases energy consumption without affecting the ball stud.

2.3.2. Ball tempering

Ball tempering focuses exclusively on heating the spherical ball stud to 145 °C for a controlled period before insertion. Unlike oven tempering, this method preheats only the ball between the second and third assembly steps, ensuring the required articulation torque values are achieved. By targeting the critical contact area directly, ball tempering eliminates the need for full-assembly oven heating and reduces overall energy consumption.

2.4. Test sample design

The experimental study was designed to evaluate whether the ball diameter of an inner tie rod assembly could be reduced without compromising functional performance or durability. In total, 192 inner tie rod assemblies were manufactured, comprising 12 reference samples with a Ø29 mm ball diameter and 180 samples

with a reduced Ø26 mm ball diameter. The Ø29 mm reference group was produced under customer-approved manufacturing conditions and served as the baseline against which all reduced-diameter configurations were assessed. The Ø26 mm design was subsequently investigated through a controlled parametric study in which the key manufacturing variables—pressing force, tempering method, and tempering temperature—were systematically varied to assess their individual and combined effects on joint performance.

For each parameter combination, a batch of 12 assemblies was produced. Each assembly was tested using three repeated measurements, and the reported results represent mean values with corresponding standard deviations, ensuring statistical consistency and reliability. The experimental program followed a sequential, industry-oriented screening and validation workflow. First, all assemblies underwent baseline functional testing, including axial travel (S_a), minimum axial stiffness ($C_{a,min}$), and articulation torque (M_t). These pre-test measurements were evaluated against the AK-LH-14 acceptance criteria to identify viable process windows. Parameter sets failing to meet any acceptance limit at this stage were excluded from further testing, as they were considered unsuitable for series production. Only configurations that fully satisfied all pre-test functional requirements were subsequently advanced to durability testing, consisting of wear and setting tests to assess long-term and thermo-mechanical performance. This stepwise screening strategy reflects standard industrial validation practice by prioritizing functional compliance prior to durability assessment, minimizing unnecessary testing of non-viable configurations, and enabling systematic identification of robust manufacturing conditions. The functional implications of the selected parameters and the reasons for excluding specific sets are discussed in detail in the Results and Discussion section.

2.5. Functional measurements

To ensure durability, safety, and performance, key functional properties of inner tie rods are measured both before and after standardized wear and setting tests, as mandated by the global AK-LH-14 standard. The acceptance criteria for these tests are detailed in Table 1.

Table 1. Acceptance criteria for functional measurements.

	Pre-testing	Post-testing
Axial (spring) travel (S_a)	max. 0.05 mm at ± 3 kN	max. 0.4 mm at ± 3 kN
Min. axial stiffness ($C_{a,min}$)	100 kN/mm	2 kN/mm
Articulation torque (M_t)	0.5–3.5 Nm	0.2–3.5 Nm

2.5.1. Axial (spring) travel (S_a)

The process evaluates axial movement between the joint housing and ball race by applying a force of ± 3 kN, where the resulting axial travel of the ball stud must not exceed a maximum allowable limit, measured using the apparatus shown in Figure 5.

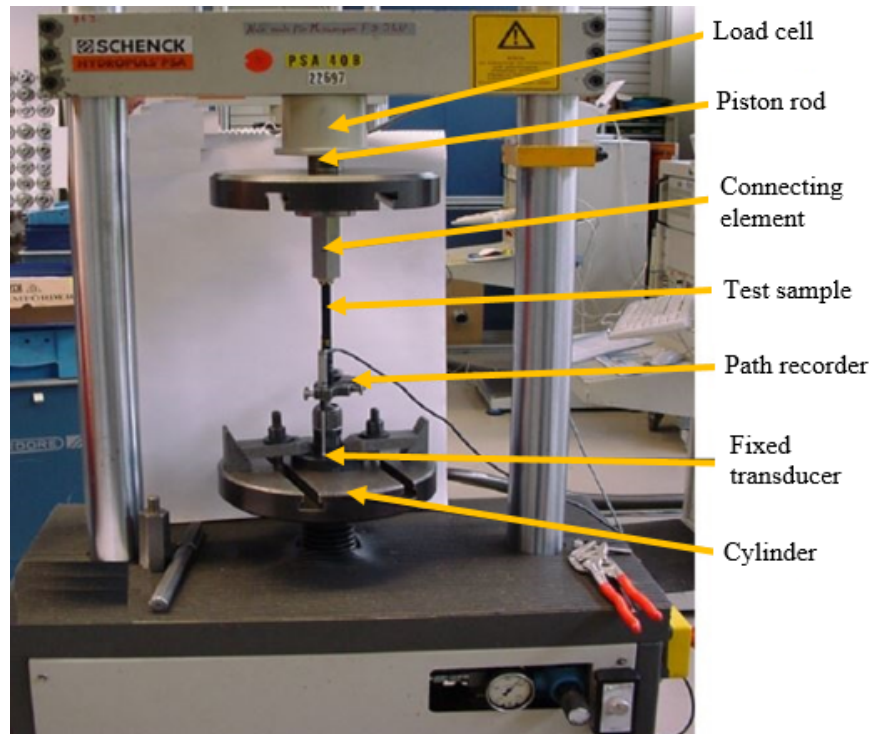


Figure 5. Axial travel recording apparatus.

2.5.2. Minimum axial stiffness ($C_{a,min}$)

Minimum axial stiffness is defined as the smallest tangent slope in the linear section of a joint's load-displacement curve, determined by calculating the gradient between two points on the flattest portion of the curve.

2.5.3. Articulation (tilting) torque (M_t)

Articulation torque, which measures the sliding friction within a joint after its initial breakaway movement, is conditioned by first articulating the ball stud five times to ensure uniform grease distribution, and is then measured over five cycles of $\pm 22^\circ$ articulation at $10^\circ/\text{s}$, as illustrated in **Figures 6 and 7**.

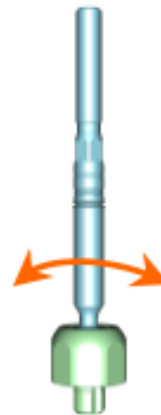


Figure 6. Articulation torque.

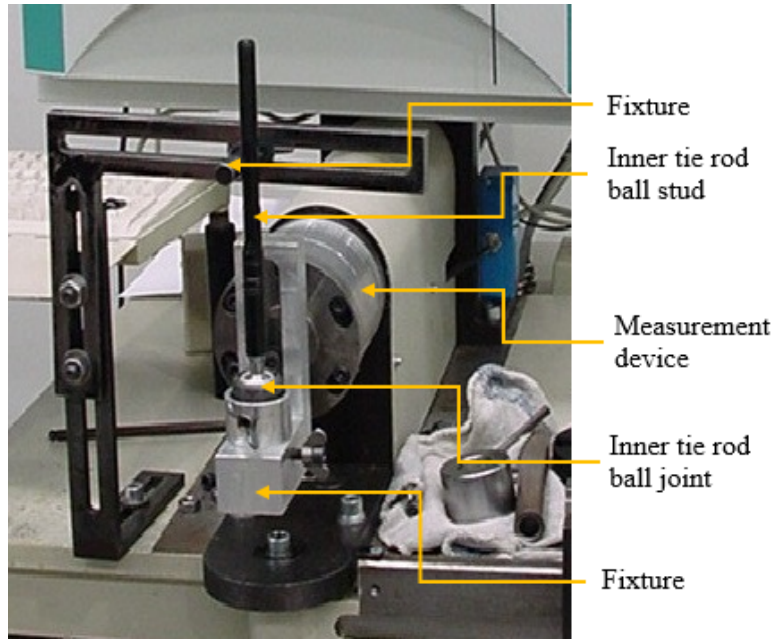


Figure 7. Articulation torque recording apparatus.

2.6. Setting test

The setting test evaluates the deformation behavior of inner tie rods under combined mechanical loading and elevated temperature. It focuses on assessing the ability of the plastic bearing to maintain structural integrity and alignment when subjected to maximum expected service loads. The maximum assist force is determined from strain-gauge measurements obtained on test tracks under extreme operating conditions [37]. During the test, a sinusoidal axial load of ± 23 kN is applied over three cycles while the ball joints are heated to 80°C , as illustrated in **Figure 8**.

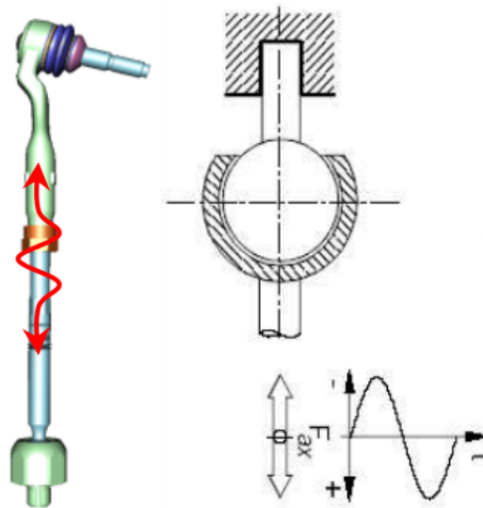


Figure 8. Setting test.

2.7. Wear test

The wear test assesses the durability and resistance to material degradation of inner tie rods under simulated operational conditions. A total of 62,500 wear cycles is applied, following the specific load phases detailed in **Table 2**. Throughout the entire test, the

component is subjected to an axial load, a constant articulation angle of 14°, and a controlled temperature of 80 °C. These elevated conditions are designed to accelerate wear, particularly in the POM material, thereby enabling the reliable prediction of the component's long-term performance and service life in the vehicle.

Table 2. Wear test cycle phases.

Test load [kN]	Frequency [Hz]	Cycle
±8885	2	1250
±7626	4	2500
±5724	4	2500
±4464	2	6250
±3276	1	25,000
±2468	2	25,000

3. Results and discussion

After the initial functional screening, selected configurations were subjected to durability testing to evaluate wear behavior and thermo-mechanical stability. For each selected configuration, specimens were divided equally between wear and setting tests to independently assess friction-related degradation and temperature-assisted deformation.

For the Ø26 mm ball design, pressing force was increased stepwise to define a robust manufacturing window. This incremental approach enabled the elimination of unstable parameter combinations and supported the selection of process conditions that met both functional and durability requirements, confirming the feasibility of the reduced ball diameter for series production.

Prior to durability testing, all assemblies were characterized by articulation torque and minimum axial stiffness measurements, which were repeated after wear and setting tests to verify compliance with performance criteria. Based on these evaluations, representative parameter sets were selected and relabeled as Batches 1–4, as shown in **Figure 9**, with the corresponding pressing force, tempering method, and tempering temperature summarized in **Table 3**.

As illustrated in **Figure 9**, Groups 1 and 2 employed oven tempering, while Group 3 used ball tempering. In Group 1, all parameter sets were eliminated due to excessive axial travel or articulation torque exceeding acceptance limits. Increasing the tempering temperature in Group 2 by +10 °C resulted in a stable configuration that met all functional requirements and was selected for durability testing (Batch 2). In Group 3, most parameter sets satisfied pre-test requirements; two representative sets were selected to capture the performance range (Batches 3 and 4).

Comparison of Batches 1 and 2 demonstrates that articulation torque is highly sensitive to tempering temperature for the Ø26 mm design. At the initial tempering temperature, insufficient softening of the POM ball race limited surface conformity to the steel ball, resulting in elevated torque due to increased contact pressure and friction. Increasing the tempering temperature by +10 °C improved the viscoelastic deformation of the POM material, reduced torque to acceptable levels, and compensated for process-related heat losses during handling. This temperature adjustment, therefore,

represents a practical and robust solution aligned with industrial production conditions.

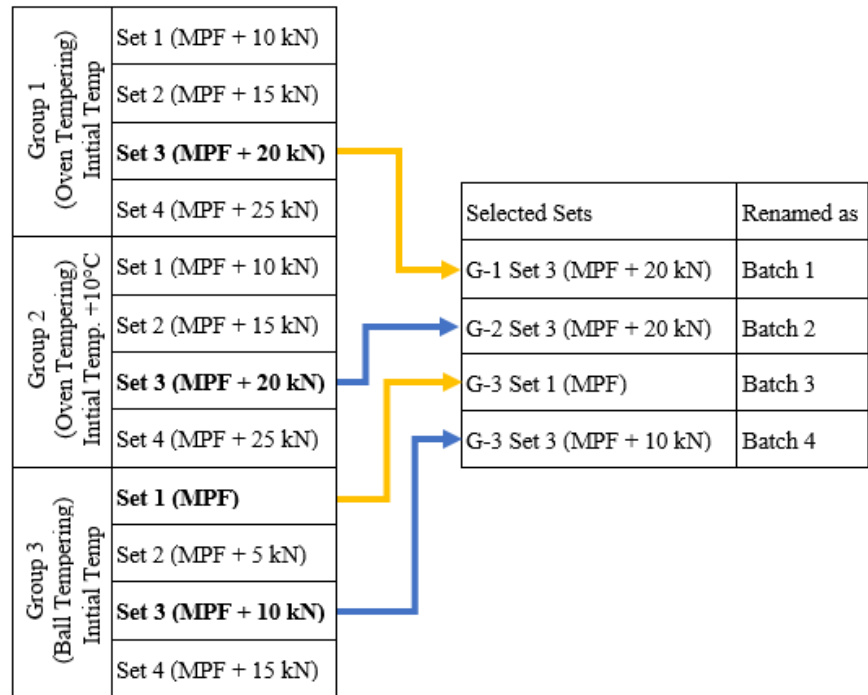


Figure 9. Test set arrangement.

Table 3. Inner tie rod sample sets.

Batch No.	Pressing force	Tempering method	Tempering temperature
1	MPF + 20 kN	Oven	Initial
2	MPF + 20 kN	Oven	Initial + 10 °C
3	MPF	Ball	Initial
4	MPF + 10 kN	Ball	Initial

This behavior indicates that the initial tempering temperature was insufficient to promote adequate viscoelastic softening of the POM ball race, limiting its ability to plastically conform to the steel ball during assembly and resulting in elevated articulation torque. Increasing the tempering temperature by +10 °C enhances material softening and surface conformity, thereby reducing localized contact pressure and friction. From an industrial perspective, this temperature increase also compensates for unavoidable heat losses during handling and transfer of components from the tempering furnace to subsequent assembly stages. Consequently, the selected +10 °C adjustment reflects both material behavior and realistic production conditions.

3.1. Test results of Ø29 ball size

The Ø29 mm reference design (Batch 0) was evaluated to establish a baseline for the reduced ball joint. All functional parameters (S_a , $C_{a,min}$, and M_t) satisfied the AK-LH-14 requirements. Wear tests (samples 29-0-1 to 29-0-6) and setting tests (samples 29-0-7 to 29-0-12) were conducted, with results summarized in **Figures 10** and **11**. Bars represent mean values with standard deviations, while blue and orange markers denote pre- and post-test measurements and acceptance limits, respectively.

Note that for axial travel (S_a), values must remain below the specified limit lines; for minimum axial stiffness ($C_{a,min}$), values must exceed the specified limit lines; and for articulation torque (M_t), values must fall between the defined upper and lower limit lines.

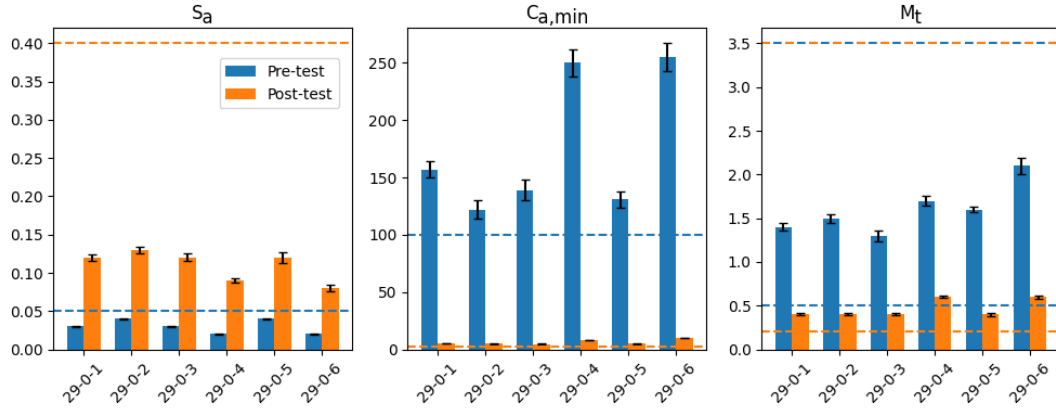


Figure 10. Functional measurements of Ø29 inner tie rod samples from Batch 0 before and after wear test.

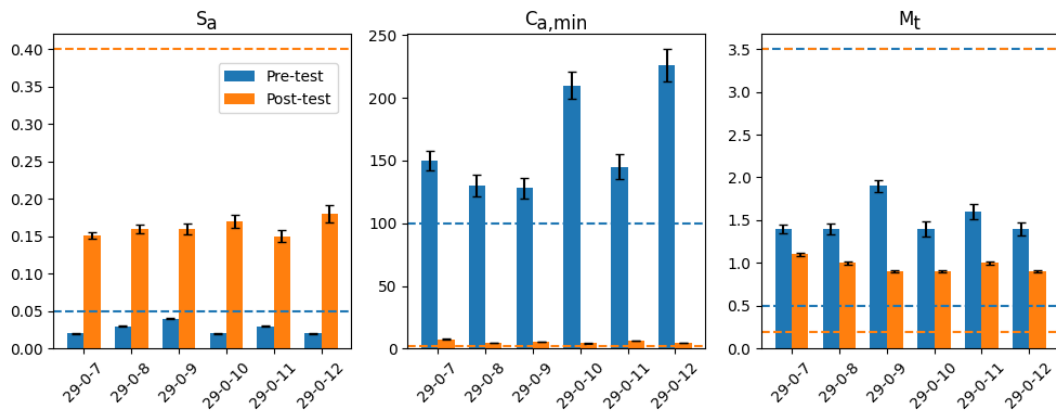


Figure 11. Functional measurements of Ø29 inner tie rod samples from Batch 0 before and after setting the test.

As shown in **Figure 10**, wear testing increased axial travel and reduced minimum axial stiffness and articulation torque due to plastic race wear and reduced friction; however, all pre- and post-test values remained within acceptance limits.

Figure 11 shows similar trends after the setting test, with increased axial travel and reduced articulation torque and stiffness relative to pre-test values due to thermo-mechanical deformation of the plastic bearing under elevated temperature and loading, as well as reduced friction from surface wear; nevertheless, all post-test values remained within the acceptance limits.

3.2. New small-diameter axial bearing design

3.2.1. Oven tempering test results

For Batch 1, the samples were produced using a pressing force of “MPF + 20 kN” and oven tempering at the “initial” temperature. As shown in **Figure 12**, samples

26-1-1, 26-1-5, and 26-1-6 failed to meet the articulation torque requirement prior to the setting test (marked by red X). Although these samples satisfied the acceptance criteria after testing, they were classified as unacceptable due to their pre-test noncompliance; therefore, wear testing was not performed. These results indicate that, under the initial tempering condition, the applied press force is either excessive or the tempering temperature is insufficient to ensure acceptable articulation torque before durability testing.

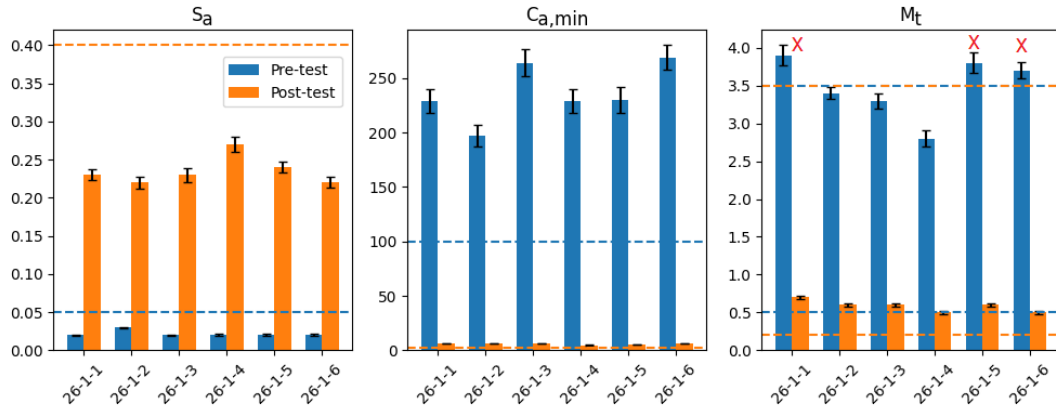


Figure 12. Functional measurements of Ø26 inner tie rod samples from Batch 1 before and after the setting test.

Batch 2 was produced using the same pressing force (MPF + 20 kN) as Batch 1, with the tempering temperature increased by +10 °C to achieve acceptable pre-test articulation torque. As shown in **Figures 13** and **14**, the elevated tempering temperature effectively reduced articulation torque compared to Batch 1, bringing the pre-test values closer to the specified limits. Although pre-test torque values for the wear test remained near the upper limit, they were consistently lower than those observed in Batch 1, confirming that the temperature increase provided the desired torque reduction without altering the press force. Post-wear test results, as well as pre- and post-test measurements from the setting test, remained within the acceptance criteria.

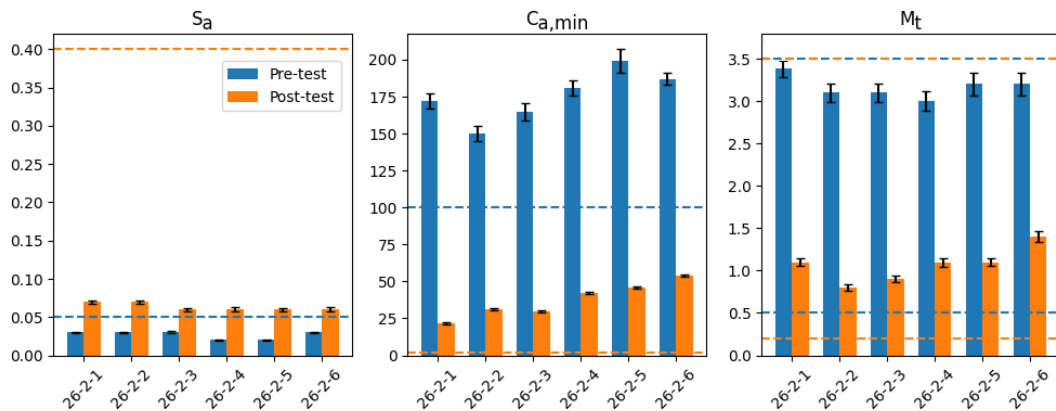


Figure 13. Functional measurements of Ø26 inner tie rod samples from Batch 2 before and after the wear test.

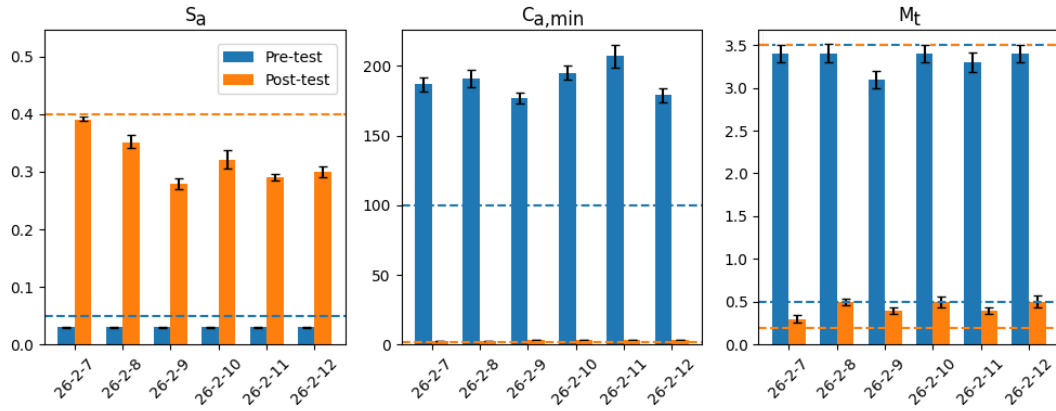


Figure 14. Functional measurements of Ø26 inner tie rod samples from Batch 2 before and after the setting test.

Accordingly, it is concluded that for Ø26 ball joint production, a pressing force of MPF + 20 kN combined with an oven tempering temperature of Initial + 10 °C satisfies the AK-LH-14 requirements and is suitable for series production.

3.2.2. Ball tempering test results

Batch 3 samples were produced using ball tempering with an MPF press force. As shown in **Figures 15** and **16**, both pre- and post-wear test measurements satisfied the specified acceptance criteria, and pre-test measurements for the setting test were within allowable limits. However, following the setting test, axial travel (Sa) exceeded the allowable limits and minimum axial stiffness (Ca,min) dropped below the acceptance threshold in five out of six samples, rendering the majority of the batch unacceptable.

The results indicate that although the reduced press force effectively lowers internal stresses and improves pre-test articulation torque (Mt), it provides insufficient resistance to thermo-mechanical loading during the setting test. Excessive deformation of the plastic ball race under elevated temperature and load leads to increased axial travel and reduced stiffness. Therefore, while ball tempering combined with MPF press force improves initial torque performance, relying solely on MPF press force is not sufficient to ensure dimensional stability and functional robustness for series production.

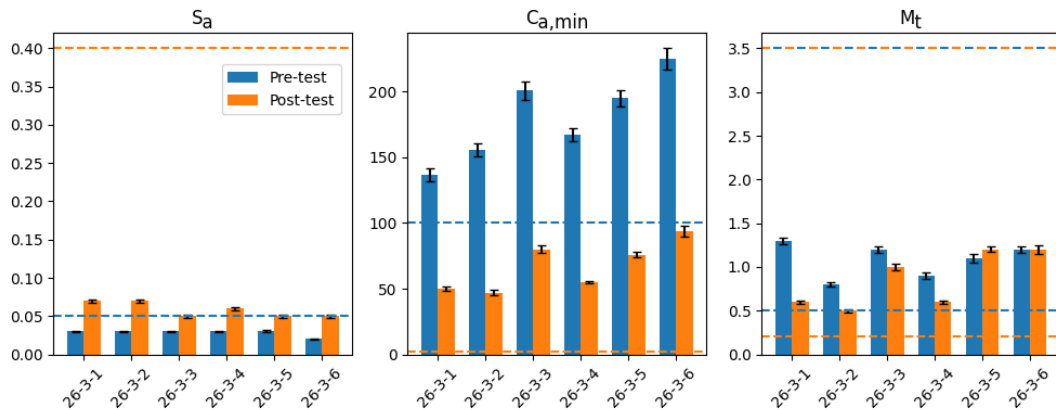


Figure 15. Functional measurements of Ø26 inner tie rod samples from Batch 3 before and after the wear test.

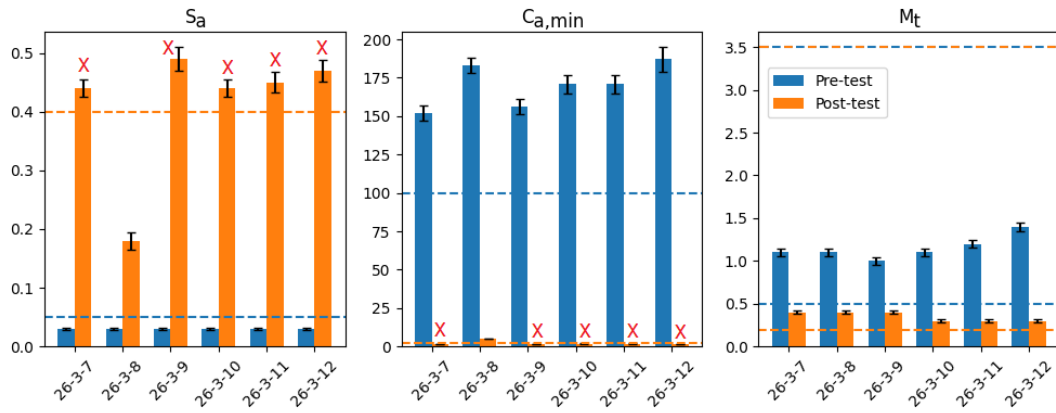


Figure 16. Functional measurements of Ø26 inner tie rod samples from Batch 3 before and after the setting test.

As shown in **Figures 17 and 18**, Batch 4 samples produced with an increased press force exhibited improved pre-test articulation torque, while all wear test measurements satisfied the specified acceptance criteria. In addition, both pre- and post-test results from the setting test remained within allowable limits, confirming compliant functional performance across all evaluated parameters.

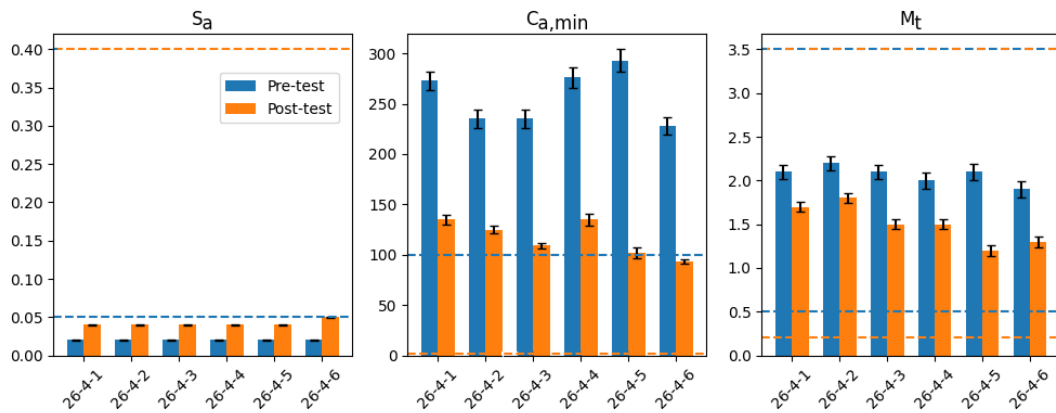


Figure 17. Functional measurements of Ø26 inner tie rod samples from Batch 4 before and after the wear test.

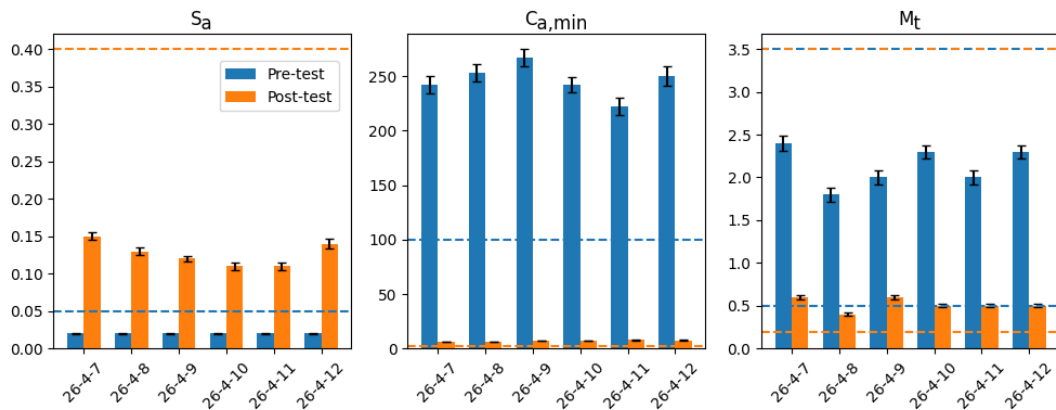


Figure 18. Functional measurements of Ø26 inner tie rod samples from Batch 4 before and after the setting test.

The improved performance observed in Batch 4 indicates that the increased press force effectively enhances joint stability by reducing susceptibility to deformation under wear and thermo-mechanical loading. Unlike Batch 3, the higher press force provides sufficient resistance to maintain axial travel and stiffness within acceptable limits, demonstrating that this configuration offers a robust and production-ready solution that meets all required standards.

Consequently, the results demonstrate that for Ø26 ball joint production, a pressing force of MPF + 10 kN combined with an initial ball tempering temperature satisfies the AK-LH-14 [1] requirements and provides a viable solution for series production.

A comparative evaluation of the tested batches highlights the role of both tempering temperature and pressing force in achieving robust functional performance. In Batch 1, excessive pre-test articulation torque prevented progression to wear testing, indicating that the initial tempering temperature was insufficient under the applied press force. Increasing the tempering temperature by +10 °C in Batch 2 successfully reduced pre-test torque to acceptable levels, enabling both wear and setting tests to be completed; the favorable post-test results confirmed the suitability of this configuration.

Batch 3 demonstrated that ball tempering combined with a lower press force can achieve acceptable pre-test torque and wear performance; however, the pronounced degradation observed after the setting test revealed insufficient resistance to thermo-mechanical loading, resulting in excessive axial displacement and stiffness loss. This outcome underscores the limitation of relying solely on reduced press force for stability.

In contrast, Batch 4 showed that combining ball tempering with an increased press force restores dimensional stability under both wear and setting conditions. All functional parameters remained within acceptance limits before and after testing, confirming that this configuration provides a robust and production-ready solution. Collectively, these findings emphasize the need to balance tempering strategy and press force to ensure both initial compliance and long-term durability in reduced-diameter ball joint designs.

The results show that reducing inner tie rod ball diameter is not a purely geometric scaling problem; functional stability is governed by the interaction between manufacturing parameters and joint contact mechanics. Prior work has largely emphasized lightweighting via material substitution and structural/topology optimization [11–13, 19–21] or simulation-driven sizing methodologies for ball joints [3], but these approaches typically do not resolve the strong process sensitivity observed at the joint/bearing interface under series-representative conditions.

Ball tempering localized heat input and is therefore attractive from an energy/process-efficiency standpoint; however, the durability response indicates that tempering strategy alone is insufficient when press force is minimized. Although wear-stage results remained acceptable, setting tests produced excessive axial travel and stiffness loss, indicating thermo-mechanical deformation of the polymer bearing under combined temperature and load. This failure mode aligns with fatigue/reliability studies showing that joint performance and life are highly sensitive to stress distribution, preload, and local damage evolution [5,7,8], and is consistent with tribological evidence that polymer

contact behavior depends strongly on frictional and wear conditions [36]. Increasing press force in the ball-tempered configuration mitigated deformation and stabilized post-setting performance, indicating that adequate preload is required to maintain joint dimensional stability once bearing compliance is reduced.

The study establishes that robust Ø26 performance requires a balanced combination of tempering method and press force, rather than material change or geometry optimization alone. By providing industrial, hands-on validation beyond simulation-centric studies [3,11–13], the results help translate lightweighting objectives into a production-ready process window while maintaining compliance with steering joint performance and durability requirements [1,2,37].

The energy- and cost-related implications discussed in this study are subject to several boundary conditions and sources of uncertainty that should be considered when interpreting the results. The experimental campaign focused on functional and durability performance, and direct measurements of energy consumption during pressing and tempering operations were not performed. Consequently, energy-related comparisons are based on relative process indicators, including ball diameter, heated mass, tempering temperature level, and tempering strategy, rather than absolute energy values. Variability in furnace efficiency, batch loading, heating duration, and heat losses during component handling and transfer between process steps may influence the actual energy consumption in a production environment. In addition, press force variations and cycle times are assumed to remain within typical industrial tolerances for series production. These assumptions reflect realistic manufacturing conditions but introduce uncertainty when extrapolating results to different production lines or equipment configurations. Accordingly, the energy efficiency and cost discussions are intended as comparative, process-level assessments, highlighting trends and potential benefits rather than precise quantitative savings. Future work, including direct energy monitoring and line-specific data, would enable a more detailed evaluation of absolute energy and cost impacts.

4. Conclusion

This study demonstrates that reducing the inner tie rod ball diameter from Ø29 mm to Ø26 mm is feasible without compromising compliance with AK-LH-14 functional and durability requirements, provided that manufacturing parameters are appropriately optimized. Experimental results show that articulation torque, axial displacement, and minimum axial stiffness are highly sensitive to pressing force and tempering strategy, with this sensitivity becoming more pronounced as ball diameter is reduced. In particular, articulation torque was identified as the most critical parameter governing pre-test acceptance.

Ball-tempered designs produced with minimum press force achieved acceptable initial torque and wear performance, but failed under setting conditions due to excessive axial deformation and stiffness loss. By contrast, combining ball tempering with an increased press force (MPF + 10 kN) resulted in stable functional behavior before and after both wear and setting tests, satisfying all acceptance criteria.

Unlike simulation-driven studies, this work provides industrial-scale, hands-on

experimental evidence that successful ball diameter reduction depends on balanced process-level optimization rather than geometric redesign alone. The validated Ø26 mm configurations offer a production-ready solution that enables material and weight reduction while limiting additional thermal input, thereby supporting cost efficiency, energy-conscious manufacturing, and sustainability objectives within existing automotive steering system production lines.

Author contributions: SD contributed to the conceptualization and methodology, performed the experimentation, wrote the original draft, and provided resources. MÇ contributed to the conceptualization, supervised the work, wrote the original draft, reviewed and edited the manuscript, and was a major contributor to the study. Both authors have read and agreed to the published version of the manuscript.

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Article

Optimal design of a novel TVMD-energy harvesting device for seismic control of long-period structures

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Abstract: Over the past ten years, energy harvesting (EH) from ambient energy has become essential for wireless applications, allowing low-powered electronics to operate. Due to the economic and ecological advantages of the EH technology, it is a promising idea to apply it to the seismic control of long-period structures while converting the earthquake input energy into electricity for structural health monitoring, earthquake observation, or other purposes. However, rare literature can be found on applying EH technology for the vibration control of long-period structures, especially in earthquake-prone areas. During extreme earthquakes, the tuned viscous mass damper (TVMD) exhibits an excellent control effect on long-period structures. To combine the advantages of EH and TVMD, a novel device, namely, the TVMD-EH damper, was proposed for the seismic control of long-period structures while harvesting the earthquake input energy simultaneously for electricity-needed purposes. A capacitor, a resistor, and piezoelectric material constructed an EH loop. Then, the EH loop was connected with the TVMD device to construct the TVMD-EH damper. In this study, the optimal design method for the TVMD-EH damper was proposed, and a feasibility study was conducted to investigate the performance of the device. The results demonstrate that the proposed TVMD-EH damper provides effective seismic response mitigation for the targeted structure while enabling simultaneous energy harvesting. In particular, the peak harvested electric power reaches 6.8 W under the El Centro record and 1.9 W under the Sakishima record suggesting practical potential for powering low-power independent sensors and related electronics and reducing battery dependence.

Keywords: energy harvesting; long-period structures; TVMD; seismic control; optimal design method

1. Introduction

In the last two decades, numerous earthquakes have occurred around the world, including the destructive 2004 Indian Ocean earthquake [1], 2005 Nias-Simeulue earthquake (Sumatra, Indonesia) [2], 2008 Wenchuan earthquake (Sichuan, China) [3], 2010 Chile earthquake (Maule, Chile) [4], and 2011 Tohoku earthquake (Miyagi, Japan) [5, 6], and among others. Despite significant research efforts by seismologists, scientifically reliable predictions still cannot be made for a particular day or month. Therefore, engineers and scientists tried to take some measurements to improve the seismic resistance of structures, thereby reducing the loss of life and property.

The measurements for seismic control included passive methods, active methods,

and semi-active approaches that combined passive and active control strategies. Due to the low cost and simple application, the passive control method has been widely used in structural and mechanical engineering. Viscous dampers, rubber bearings, friction pendulums, and other energy-dissipating devices are applied in buildings, transportation, power systems, aircraft, and so on for vibration control. Those energy-dissipating devices can significantly reduce the negative impact of vibration on various structures.

Touaillon [7] patented an improved building construction technology in 1870. Bill Robinson first proposed lead rubber bearings in the 1970s. Bechtold [8] applied for a US patent for an earthquake-proof building supported by a rigid plate. According to the report [9], Frank Lloyd Wright incorporated seismic isolation in the Imperial Hotel in Tokyo, Japan, completed in 1921. de Montalk [10] proposed absorbing or reducing stress on buildings caused by external excitations. Naeim and Kelly [11] investigated a seismically separated structure sitting on spherical balls. Researchers also looked into sliding isolation systems [12–16]. Other dampening techniques [17–20] have also been proposed for seismic isolation. Recently, rate-independent linear damping (RILD, which is also referred to as structural damping and hysteretic damping [21–24]) also attracted attention [25–27]. Liu and Ikago [28–30] first experimentally realized the noncausal RILD using passive devices. Using the Maxwell-Wiechert model, some causal RILD models [31–33] were also developed for seismic protection of long-period structures [34–37]. A novel modal analysis method for the multistory structure with non-proportionally distributed RILD was developed by Liu et al. [38,39] to investigate the differences between the RILD and conventional viscous damping systems.

A tuned mass damper (TMD) is a device that reduces vibrations by suspending a mass on one or more damped springs. Its oscillation frequency is designed to be close to the attached object's resonant frequency, lowering the maximum amplitude. TMDs can assist in alleviating pain, damage, and structural failure. They are frequently used in power transmission, automobiles, and construction. Den Hartog [40] suggested an optimal design strategy for the TMD based on fix-point theory, and this method was extensively adopted around the world. Smith [41] elucidated the link between mechanical and electrical networks and introduced the notion of inerter. Papageorgiou et al. [42] tested an inerter device for controlling vibration. Lazar et al. [43] developed a vibration mitigation system using inerters. Ikago et al. [44] used the inerter to create a tuned viscous mass damper (TVMD) and suggested an optimal design technique. Then, Huang et al. [45] found the same optimal design parameters for TVMD as Ikago et al. [44].

Energy harvesting (EH) is a process wherein the sources, such as mechanical load, vibrations, light, etc., are scavenged and converted to obtain relatively small levels of power in the nW–mW range [46]. The EH technology is suitable for replacing the batteries currently used for low-energy consumption devices, such as stand-alone electronic sensors, transducer elements, and wearable sensors. This technique can recharge storage devices, such as batteries and capacitors, and supply distant sources of electricity. This idea brings financial benefits by potentially lowering maintenance costs and also has ecological implications by minimizing the chemical waste generated when batteries need to be replaced. Increasing public safety is another benefit of the

EH through enabling wireless monitoring applications [47].

Scholars and business people are interested in EH because of the prospects that were previously indicated. Priya and Inman [48] provided an overview of the fundamentals and current developments in the field of EH. Cottone et al. [49] put out an EH approach based on the dynamic characteristics of stochastic nonlinear oscillators. Roundy [50] developed a framework to compare various methodologies and designs for vibration-based energy harvesters. Kansal et al. [51] developed distributed ways to use captured energy and tested these both in simulation and empirically on an EH sensor network. Stephen [52] stated that a high-damped system can harvest energy over a larger spectrum of excitation frequencies. Erturk et al. [53] proposed a piezo magnetoelastic device for improving piezoelectric electricity production in vibration energy capture. Sodano et al. [54] studied the ability of three distinct piezoelectric devices to recharge batteries.

Zhou and Zuo [55] sought to understand the nonlinear dynamic process of asymmetric tristable energy harvesters and improve their EH capability under various excitations. Jeong et al. [56] proposed a hyper-stretchable elastic-composite generator using piezoelectric nanocomposite. Naseer et al. [57] developed a lumped-parameter model by linking the dynamic responses of the harvester system with the generated voltage via the electrical load resistance. Han et al. [58] described a controlled, nonlinear buckling technique for converting lithographically designed two-dimensional patterns of electrodes and thin films of piezoelectric polymers into complex 3D piezoelectric microsystems. Cassidy et al. [59] proposed an electromagnetic transducer for EH from buildings and bridges with disturbance frequencies below 1 Hz. Salvil and Giaralis [60] proposed an absorber for vibration suppression and EH. Ali and Adhikari [61] augmented a vibration absorber with a piezoelectric stack for vibration containment and energy collection. However, only some researchers have focused on employing the EH method for the seismic protection of structures, especially in earthquake-prone areas.

This study developed a novel device for both seismic control and EH by employing the TVMD and an EH loop. This work can be summarized as follows: Section 2 proposed the novel TVMD-EH device; Section 3 presents the optimal design method of the mechanical elements of the TVMD-EH system; Section 4 shows the optimal design method of the electrical elements of the TVMD-EH device; Section 5 is the feasibility investigation of the ability of the TVMD-EH system for both seismic control and EH on a single-degree-of-freedom (SDOF) structure; Finally, Section 6 concluded this study and discussed the future challenges.

2. Novel TVMD-EH system

To take advantage of the TVMD device for vibration control, a novel TVMD-EH device was developed as shown in **Figure 1**. The piezoelectric material generated electricity for EH. Additionally, a capacitor c_P and resistor R were arranged parallelly with the piezoelectric material to construct the EH loop. The TVMD device is constructed by a spring k_d , viscous damper c_d , and inerter m_d . The piezoelectric material can generate electricity while the relative motions happen between two ends

of the inerter or viscous damper.

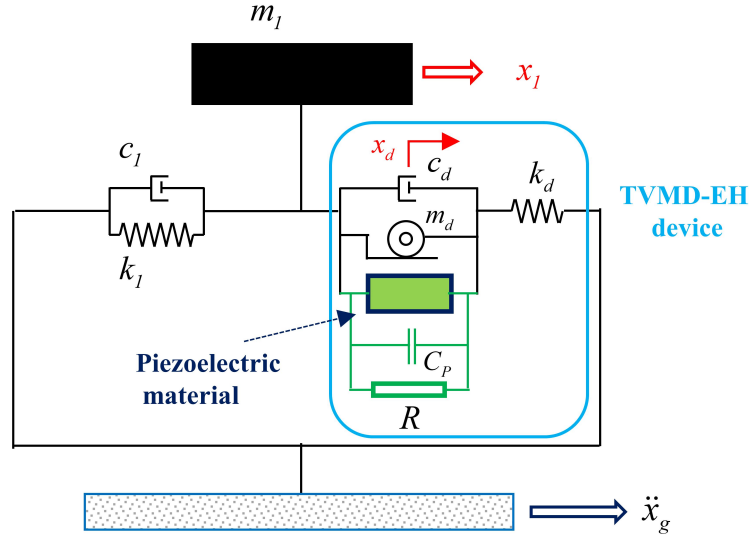


Figure 1. The SDOF structure equipped with TVMD-EH device.

To address the large-stroke issue, the piezoelectric element should be mechanically decoupled from the global damper displacement. A motion conversion stage (e.g., hydraulic pressure interface or lever/cam mechanism) can transform centimeter-meter stroke into bounded micro-strain deformation of a piezoelectric stack/bender. For instance, damper motion driving a hydraulic piston generates pressure that loads a piezoelectric stack in compression; the effective strain is limited by the piston area ratio and compliant/relief design, while the electrical output is produced by cyclic stress rather than large displacement.

The primary structure was assumed to be the SDOF structure with mass m_1 , inherent damping c_1 , and horizontal stiffness k_1 , respectively. The variable Θ represents the coupling between the electrical and mechanical elements in the TVMD-EH device. The dynamics of the structural system and generated voltage can be expressed as follows:

$$\begin{aligned} m_1 \ddot{x}_1 + c_1 \dot{x}_1 + k_1 x_1 + k_d (x_1 - x_d) &= -m_1 \ddot{x}_g \\ k_d (x_1 - x_d) &= m_d \ddot{x}_d + c_d \dot{x}_d - \Theta v \\ C_p \dot{v} + \frac{v}{R} + \Theta \dot{x}_d &= 0 \end{aligned} \quad (1)$$

where $\ddot{x}_1$, $\dot{x}_1$, x_1 , $\ddot{x}_d$, $\dot{x}_d$ is the relative acceleration, relative velocity, relative displacement of the primary structure, relative acceleration, and relative velocity of the inerter device, respectively. v is the generated voltage in the EH loop.

Denoting the steady state solution of Equation (1) as follows:

$$x_1 = X_1 e^{i\omega t}, \quad x_d = X_d e^{i\omega t}, \quad v = V e^{i\omega t}, \quad \ddot{x}_g = A_0 e^{i\omega t} \quad (2)$$

Then, substituting Equation (2) into Equation (1) yields:

$$\begin{aligned}
 (-m_1\omega^2 + i\omega c_1 + k_1 + k_d)X_1 - k_d X_d &= -m_1 A_0 \\
 k_d(X_1 - X_d) &= -m_d \omega^2 X_d + i\omega c_d X_d - \Theta V \\
 i\omega C_P V + \frac{V}{R} + i\omega \Theta X_d &= 0
 \end{aligned} \tag{3}$$

For simplicity, by denoting the following criteria:

$$\begin{aligned}
 \omega_1 &= \sqrt{\frac{k_1}{m_1}}, \quad \omega_d = \sqrt{\frac{k_d}{m_d}}, \quad \mu = \frac{m_d}{m_1}, \quad \beta = \frac{\omega_d}{\omega_1}, \quad \gamma = \frac{\omega}{\omega_1} \\
 \xi_1 &= \frac{c_1}{2m_1\omega_1}, \quad \xi_d = \frac{c_d}{2m_d\omega_d}, \quad \kappa^2 = \frac{\Theta^2}{k_d C_P}
 \end{aligned} \tag{4}$$

Then, Equation (3) can be rewritten as:

$$(-\gamma^2 + 2i\xi_1\gamma + 1 + \mu\beta^2)X_1 - \mu\beta^2 X_d = -A_0/\omega_1^2 \tag{5}$$

$$\beta^2 X_1 + (\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)X_d + \beta^2\Theta V/k_d = 0 \tag{6}$$

$$i\Theta R X_d \omega_d \gamma + (i\gamma C_P R \omega_d + \beta)V = 0 \tag{7}$$

By introducing the dimensionless time constant $\alpha = \omega_d C_P R$, Equation (7) can be rewritten as:

$$i\Theta\gamma\alpha X_d/C_P + (i\gamma\alpha + \beta)V = 0. \tag{8}$$

Thus,

$$V = -i\Theta\gamma\alpha X_d/[(i\gamma\alpha + \beta)C_P] \tag{9}$$

Substituting Equation (9) into Equation (6) can yield:

$$\beta^2 X_1 + ((\gamma^2 - 2i\xi_d\gamma\beta - \beta^2) - i\gamma\alpha\beta^2\Theta^2/[(i\gamma\alpha + \beta)C_P k_d]) X_d = 0 \tag{10}$$

To reduce the number of variables in Equation (10), the electrical coefficient $\kappa^2 = \frac{\Theta^2}{C_P k_d}$ was introduced, then Equation (10) can be expressed as:

$$\beta^2 X_1 + [(\gamma^2 - 2i\xi_d\gamma\beta - \beta^2) - i\gamma\alpha\beta^2\kappa^2/(i\gamma\alpha + \beta)] X_d = 0 \tag{11}$$

Therefore,

$$X_1 = ([i\gamma\alpha\beta^2\kappa^2/(i\gamma\alpha + \beta) - (\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)]/\beta^2) X_d \tag{12}$$

Substituting Equation (12) into Equation (5) obtains:

$$(-\gamma^2 + 2i\xi_1\gamma + 1 + \mu\beta^2) ([i\gamma\alpha\beta^2\kappa^2/(i\gamma\alpha + \beta) - (\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)]/\beta^2) X_d - \mu\beta^2 X_d = -A_0/\omega_1^2 \tag{13}$$

Thus, the frequency response functions are as follows:

$$\left| \frac{X_d}{-A_0/\omega_1^2} \right| = \left| \frac{1}{(-\gamma^2 + 2i\xi_1\gamma + 1 + \mu\beta^2) ([i\gamma\alpha\beta^2\kappa^2/(i\gamma\alpha + \beta) - (\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)]/\beta^2) - \mu\beta^2} \right| \quad (14)$$

$$\begin{aligned} \left| \frac{X_1}{-A_0/\omega_1^2} \right| &= \left| \frac{([i\gamma\alpha\beta^2\kappa^2/(i\gamma\alpha + \beta) - (\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)]/\beta^2) X_d}{-A_0/\omega_1^2} \right| \\ &= \left| \frac{1}{(-\gamma^2 + 2i\xi_1\gamma + 1 + \mu\beta^2) - \mu\beta^4/[i\gamma\alpha\beta^2\kappa^2/(i\gamma\alpha + \beta) - (\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)]} \right| \end{aligned} \quad (15)$$

and

$$\begin{aligned} \left| \frac{V}{-A_0/\omega_1^2} \right| &= \left| \frac{-i\Theta\gamma\alpha X_d/[(i\gamma\alpha + \beta)C_P]}{-A_0/\omega_1^2} \right| = \left| \frac{-i\Theta\gamma\alpha}{(i\gamma\alpha + \beta)C_P} \frac{X_d}{-A_0/\omega_1^2} \right| \\ &= \frac{\Theta}{C_P} \left| \frac{-i\gamma\alpha\beta^2}{\left\{ (-\gamma^2 + 2i\xi_1\gamma + 1 + \mu\beta^2)[i\gamma\alpha\beta^2\kappa^2 - (i\gamma\alpha + \beta)(\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)] \right\} - \mu\beta^4(i\gamma\alpha + \beta)} \right| \end{aligned} \quad (16)$$

3. Optimal design of the mechanical elements of the TVMD-EH system

Using Equation (15), the following equation can be yielded:

$$\begin{aligned} \left| \frac{X_1}{-A_0/\omega_1^2} \right|^2 &= \left| \frac{\text{numerator}}{\text{denominator}} \right|^2 \\ \text{numerator} &= \beta(\beta^2 - \gamma^2) - 2\xi_d\alpha\beta\gamma^2 + i\gamma [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2) + 2\xi_d\beta^2] \\ \text{denominator} &= (-\gamma^2 + 1 + \mu\beta^2) [\beta(\beta^2 - \gamma^2) - 2\xi_d\alpha\beta\gamma^2] - 2\xi_1\gamma^2 [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2) + 2\xi_d\beta^2] - \mu\beta^5 \\ &\quad + i \left\{ \gamma(-\gamma^2 + 1 + \mu\beta^2) [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2) + 2\xi_d\beta^2] \right. \\ &\quad \left. + 2\xi_1\gamma [\beta(\beta^2 - \gamma^2) - 2\xi_d\alpha\beta\gamma^2] - \gamma\alpha\mu\beta^4 \right\} \end{aligned} \quad (17)$$

While neglecting the inherent damping ($\zeta_1 = 0$), $\left| \frac{X_1}{-A_0/\omega_1^2} \right|^2$ can be written as follows:

$$\begin{aligned} \left| \frac{X_1}{-A_0/\omega_1^2} \right|^2 &= \left| \frac{A + Bi}{C + Di} \right|^2 = \frac{A^2 + B^2}{C^2 + D^2} \\ A &= \beta(\beta^2 - \gamma^2) - 2\xi_d\alpha\beta\gamma^2 \\ B &= \gamma [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2) + 2\xi_d\beta^2] \\ C &= (-\gamma^2 + 1 + \mu\beta^2) [\beta(\beta^2 - \gamma^2) - 2\xi_d\alpha\beta\gamma^2] - \mu\beta^5 \\ D &= \gamma(-\gamma^2 + 1 + \mu\beta^2) [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2) + 2\xi_d\beta^2] - \gamma\alpha\mu\beta^4 \end{aligned} \quad (18)$$

On the one hand, by assuming $\xi_d = 0$, the following equation can be obtained:

$$\begin{aligned}
 \left| \frac{X_1}{-A_0/\omega_1^2} \right|_{\xi_d=0}^2 &= \frac{A_1^2 + B_1^2}{C_1^2 + D_1^2} \\
 A_1 &= \beta(\beta^2 - \gamma^2) \\
 B_1 &= \gamma [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] \\
 C_1 &= \beta(-\gamma^2 + 1 + \mu\beta^2)(\beta^2 - \gamma^2) - \mu\beta^5 = \beta(1 - \gamma^2)(\beta^2 - \gamma^2) - \mu\beta^3\gamma^2 \\
 D_1 &= \gamma(-\gamma^2 + 1 + \mu\beta^2) [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] - \gamma\alpha\mu\beta^4
 \end{aligned} \tag{19}$$

On the other hand, by setting $\xi_d = \infty$, the following equation can also be obtained:

$$\begin{aligned}
 \left| \frac{X_1}{-A_0/\omega_1^2} \right|_{\xi_d=\infty}^2 &= \frac{A_2^2 + B_2^2}{C_2^2 + D_2^2} \\
 A_2 &= -\alpha\beta\gamma \\
 B_2 &= \beta^2 \\
 C_2 &= -\alpha\beta\gamma(-\gamma^2 + 1 + \mu\beta^2) \\
 D_2 &= \beta^2(-\gamma^2 + 1 + \mu\beta^2)
 \end{aligned} \tag{20}$$

Based on the fix-points theory [38], letting $\left| \frac{X_1}{-A_0/\omega_1^2} \right|_{\xi_d=0}^2 = \left| \frac{X_1}{-A_0/\omega_1^2} \right|_{\xi_d=\infty}^2$, one can yield:

$$\begin{aligned}
 &\left| \frac{\beta(\beta^2 - \gamma^2) + \gamma [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] i}{\beta(1 - \gamma^2)(\beta^2 - \gamma^2) - \mu\beta^3\gamma^2 + \{\gamma(-\gamma^2 + 1 + \mu\beta^2) [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] - \gamma\alpha\mu\beta^4\} i} \right|^2 \\
 &= \left| \frac{-\alpha\beta\gamma + \beta^2 i}{-\alpha\beta\gamma(-\gamma^2 + 1 + \mu\beta^2) + \beta^2(-\gamma^2 + 1 + \mu\beta^2)i} \right|^2 = \left| \frac{1}{-\gamma^2 + 1 + \mu\beta^2} \right|^2
 \end{aligned} \tag{21}$$

Considering the signs of the amplitude ratios, one can rewrite the above equation as follows:

$$\frac{\beta(\beta^2 - \gamma^2) + \gamma [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] i}{\beta(1 - \gamma^2)(\beta^2 - \gamma^2) - \mu\beta^3\gamma^2 + \{\gamma(-\gamma^2 + 1 + \mu\beta^2) [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] - \gamma\alpha\mu\beta^4\} i} = -\frac{1}{-\gamma^2 + 1 + \mu\beta^2} \tag{22}$$

The above equation can be transformed to the following polynomial concerning γ :

$$\begin{aligned}
 &(\gamma^2 - 1 - \mu\beta^2) \{ \beta(\beta^2 - \gamma^2) + \gamma [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] i \} \\
 &== \beta(1 - \gamma^2)(\beta^2 - \gamma^2) - \mu\beta^3\gamma^2 + \{ \gamma(-\gamma^2 + 1 + \mu\beta^2) [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] - \gamma\alpha\mu\beta^4 \} i
 \end{aligned} \tag{23}$$

Dividing the real and imaginary parts of Equation (23) can yield:

$$\begin{aligned}
 \text{realpart} &= (\gamma^2 - 1 - \mu\beta^2)\beta(\beta^2 - \gamma^2) - \beta(1 - \gamma^2)(\beta^2 - \gamma^2) + \mu\beta^3\gamma^2 \\
 &= -2\beta(1 - \gamma^2)(\beta^2 - \gamma^2) + 2\mu\beta^3\gamma^2 - \mu\beta^5 = -2\beta \left\{ \gamma^4 - (1 + \beta^2 + \mu\beta^2)\gamma^2 + \frac{2\beta^2 + \mu\beta^4}{2} \right\}
 \end{aligned} \tag{24}$$

$$\begin{aligned}
 \text{imagpart} &= \gamma [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] (\gamma^2 - 1 - \mu\beta^2) - \gamma(-\gamma^2 + 1 + \mu\beta^2) [\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \gamma^2)] + \gamma\alpha\mu\beta^4 \\
 &= -2\gamma\alpha \{ \gamma^4 - [\beta^2(1 + \kappa^2) + (1 + \mu\beta^2)] \gamma^2 + \beta^2(1 + \kappa^2)(1 + \mu\beta^2) - \mu\beta^4/2 \}
 \end{aligned} \tag{25}$$

Because the values of κ^2 is usually less than 0.2, while assuming $\kappa^2 \approx 0$, Equation (25) can be rewritten as follows:

$$\begin{aligned} \text{imagpart} &= -2\gamma\alpha \{ \gamma^4 - [\beta^2 + (1 + \mu\beta^2)] \gamma^2 + \beta^2(1 + \mu\beta^2) - \mu\beta^4/2 \} \\ &= -2\gamma\alpha \{ \gamma^4 - [\beta^2 + (1 + \mu\beta^2)] \gamma^2 + \beta^2(2 + \mu\beta^2)/2 \} \end{aligned} \quad (26)$$

Employing the fix-point optimal design method and making both Equations (24) and (26) equal to zeros, one can obtain:

$$2(\gamma\alpha - \beta) \left\{ \gamma^4 - (1 + \beta^2 + \mu\beta^2)\gamma^2 + \frac{2\beta^2 + \mu\beta^4}{2} \right\} = 0 \quad (27)$$

Because $\gamma\alpha - \beta = 0$ is not always hold for all the frequency ratios γ , therefore,

$$\gamma^4 - (1 + \beta^2 + \mu\beta^2)\gamma^2 + \frac{2\beta^2 + \mu\beta^4}{2} = 0 \quad (28)$$

Assuming the optimal frequency ratios are γ_P and γ_Q ($\gamma_P < \gamma_Q$), the following equations can be yielded by employing Equation (28):

$$\gamma_P^2 + \gamma_Q^2 = 1 + \beta^2 + \mu\beta^2 \quad (29)$$

$$\gamma_P^2 \gamma_Q^2 = \frac{2\beta^2 + \mu\beta^4}{2} \quad (30)$$

On the other hand, substituting γ_P and γ_Q into Equation (22) yields:

$$\frac{1}{-\gamma_P^2 + 1 + \mu\beta^2} = -\frac{1}{-\gamma_Q^2 + 1 + \mu\beta^2} \quad (31)$$

Therefore,

$$\gamma_P^2 + \gamma_Q^2 = 2(1 + \mu\beta^2) \quad (32)$$

By equaling Equations (29) and (32) obtains:

$$1 + \beta^2 + \mu\beta^2 = 2(1 + \mu\beta^2) \quad (33)$$

Thus,

$$\beta_{opt} = \frac{1}{\sqrt{1 - \mu}} \quad (34)$$

By employing Equations (29) and (30) can obtain:

$$(1 + \beta^2 + \mu\beta^2 - \gamma_Q^2)\gamma_Q^2 = \frac{2\beta^2 + \mu\beta^4}{2} \quad (35)$$

Substituting Equation (34) into Equation (35) yields:

$$\left(\frac{2}{1 - \mu} - \gamma_Q^2 \right) \gamma_Q^2 = \frac{2 - \mu}{2(1 - \mu)^2} \quad (36)$$

Solving Equation (36) can obtain:

$$\gamma_{Q1,Q2}^2 = \frac{2 \mp \sqrt{2\mu}}{2(1-\mu)} \quad (37)$$

Because $\gamma_P < \gamma_Q$ and $1 - \mu > 0$, therefore,

$$\gamma_P^2 = \frac{2 - \sqrt{2\mu}}{2(1-\mu)}, \quad \gamma_Q^2 = \frac{2 + \sqrt{2\mu}}{2(1-\mu)} \quad (38)$$

By denoting $\Psi = \left| \frac{X_1}{-A_0/\omega_1^2} \right|^2$, $\Lambda = \gamma^2$, the following equation can be obtained:

$$\Psi = \frac{[\beta(\beta^2 - \Lambda) - 2\xi_d\alpha\beta\Lambda]^2 + \Lambda[\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d\beta^2]^2}{\left\langle \begin{aligned} &\{(-\Lambda + 1 + \mu\beta^2)[\beta(\beta^2 - \Lambda) - 2\xi_d\alpha\beta\Lambda] - \mu\beta^5\}^2 \\ &+ \Lambda\{(-\Lambda + 1 + \mu\beta^2)[\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d\beta^2] - \alpha\mu\beta^4\}^2 \end{aligned} \right\rangle} \quad (39)$$

Substituting $\Lambda = \Lambda_{P,Q} = \gamma_{P,Q}^2$, $\beta = \beta_{opt}$, $\xi_d = \infty$ into Equation (39) yields:

$$\Psi | \Lambda = \Lambda_P, \xi_d = \infty, \beta = \beta_{opt} = \left| \frac{1}{-\Lambda + 1 + \mu\beta^2} \right|_{\Lambda=\Lambda_P, \beta=\beta_{opt}} = \left| \frac{1}{-\frac{1-\sqrt{\frac{\mu}{2}}}{1-\mu} + 1 + \frac{\mu}{1-\mu}} \right| = \frac{1-\mu}{\mu} \sqrt{2\mu} \quad (40)$$

$$\Psi | \Lambda = \Lambda_Q, \xi_d = \infty, \beta = \beta_{opt} = \left| \frac{1}{-\Lambda + 1 + \mu\beta^2} \right|_{\Lambda=\Lambda_Q, \beta=\beta_{opt}} = \left| \frac{1}{-\frac{1+\sqrt{\frac{\mu}{2}}}{1-\mu} + 1 + \frac{\mu}{1-\mu}} \right| = \frac{1-\mu}{\mu} \sqrt{2\mu} \quad (41)$$

From the above two equations, it is obvious that $\Psi | \Lambda = \Lambda_P, \xi_d = \infty, \beta = \beta_{opt} = \Psi | \Lambda = \Lambda_Q, \xi_d = \infty, \beta = \beta_{opt}$ satisfied the fix-point method.

By using Equation (39), one can obtain:

$$\begin{aligned} \Psi &\left\langle \begin{aligned} &\{(-\Lambda + 1 + \mu\beta^2)[\beta(\beta^2 - \Lambda) - 2\xi_d\alpha\beta\Lambda] - \mu\beta^5\}^2 \\ &+ \Lambda\{(-\Lambda + 1 + \mu\beta^2)[\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d\beta^2] - \alpha\mu\beta^4\}^2 \end{aligned} \right\rangle \\ &= [\beta(\beta^2 - \Lambda) - 2\xi_d\alpha\beta\Lambda]^2 + \Lambda[\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d\beta^2]^2 \end{aligned} \quad (42)$$

Both sides of Equation (42) differentiate concerning Λ will yield:

$$\begin{aligned} \frac{\partial \Psi}{\partial \Lambda} &\left\langle \begin{aligned} &\{(-\Lambda + 1 + \mu\beta^2)[\beta(\beta^2 - \Lambda) - 2\xi_d\alpha\beta\Lambda] - \mu\beta^5\}^2 \\ &+ \Lambda\{(-\Lambda + 1 + \mu\beta^2)[\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d\beta^2] - \alpha\mu\beta^4\}^2 \end{aligned} \right\rangle \\ &+ \Psi \frac{\partial \left\langle \begin{aligned} &\{(-\Lambda + 1 + \mu\beta^2)[\beta(\beta^2 - \Lambda) - 2\xi_d\alpha\beta\Lambda] - \mu\beta^5\}^2 \\ &+ \Lambda\{(-\Lambda + 1 + \mu\beta^2)[\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d\beta^2] - \alpha\mu\beta^4\}^2 \end{aligned} \right\rangle}{\partial \Lambda} \\ &= \frac{\partial \left\{ [\beta(\beta^2 - \Lambda) - 2\xi_d\alpha\beta\Lambda]^2 + \Lambda[\alpha\beta^2\kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d\beta^2]^2 \right\}}{\partial \Lambda} \end{aligned} \quad (43)$$

Because $\frac{\partial \Psi}{\partial \Lambda} | \Lambda = \Lambda_P = \frac{\partial \Psi}{\partial \Lambda} | \Lambda = \Lambda_Q = 0$, therefore, Equation (43) can be reduced to

$$\Psi \frac{\partial \left\{ \begin{aligned} & \{(-\Lambda + 1 + \mu\beta^2) [\beta(\beta^2 - \Lambda) - 2\xi_d \alpha \beta \Lambda] - \mu\beta^5\}^2 \\ & + \Lambda \{(-\Lambda + 1 + \mu\beta^2) [\alpha\beta^2 \kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d \beta^2] - \alpha\mu\beta^4\}^2 \end{aligned} \right\}}{\partial \Lambda} = \frac{\partial \left\{ [\beta(\beta^2 - \Lambda) - 2\xi_d \alpha \beta \Lambda]^2 + \Lambda [\alpha\beta^2 \kappa^2 + \alpha(\beta^2 - \Lambda) + 2\xi_d \beta^2]^2 \right\}}{\partial \Lambda} \quad (44)$$

After denoting $\eta = 2\xi_d$ and complicating calculation, Equation (44) can be rewritten as ($\Lambda = \Lambda_{P,Q}$):

$$\begin{aligned} \Psi(b_1\eta^2 + b_2\eta + b_3) &= b_4\eta^2 + b_5\eta + b_6 \\ b_1 &= \beta^2 \tilde{b} \left\{ (2\alpha^2 \Lambda + \beta^2) \tilde{b} - 2\Lambda(\alpha^2 \Lambda + \beta^2) \right\} \\ b_2 &= 2\beta^4 \kappa^2 \alpha \tilde{b} (-3\Lambda + 1 + \mu\beta^2) \\ b_3 &= -2\beta^2 (\beta^2 - \Lambda) [(\beta^2 - \Lambda)(-\Lambda + 1) - \mu\beta^2 \Lambda] - 2\beta^2 \tilde{b} [(\beta^2 - \Lambda)(-\Lambda + 1) - \mu\beta^2 \Lambda] \\ &\quad + \alpha^2 \left[\tilde{b}(\beta^2 \kappa^2 - \Lambda) + \beta^2(-\Lambda + 1) \right]^2 - 2\Lambda \alpha^2 \left[\tilde{b} + \beta^2 \kappa^2 + \beta^2 - \Lambda \right] \left[\tilde{b}(\beta^2 \kappa^2 - \Lambda) + \beta^2(-\Lambda + 1) \right] \\ b_4 &= \beta^2 (2\alpha^2 \Lambda + \beta^2) \\ b_5 &= 2\beta^4 \alpha \kappa^2 \\ b_6 &= \alpha^2 (\beta^2 \kappa^2 + \beta^2 - \Lambda)^2 - 2\alpha^2 \Lambda (\beta^2 \kappa^2 + \beta^2 - \Lambda) - 2\beta^2 (\beta^2 - \Lambda) \\ \tilde{b} &= -\Lambda + 1 + \mu\beta^2 \end{aligned} \quad (45)$$

Furthermore,

$$(\Psi b_1 - b_4)\eta^2 + (\Psi b_2 - b_5)\eta + (\Psi b_3 - b_6) = 0 \quad (46)$$

If,

$$\Delta = (\Psi b_2 - b_5)^2 - 4(\Psi b_1 - b_4)(\Psi b_3 - b_6) \geq 0 \quad (47)$$

by solving Equation (46) can yield:

$$\eta_{P,Q} = \frac{-(\Psi b_2 - b_5) \pm \sqrt{(\Psi b_2 - b_5)^2 - 4(\Psi b_1 - b_4)(\Psi b_3 - b_6)}}{2(\Psi b_1 - b_4)} \quad (48)$$

While selecting the values of κ^2 , α , and μ in the following ranges:

$$\kappa^2 \in [0.1, 1.0], \quad \alpha \in [0.2, 1.0], \quad \mu \in [0.1, 0.5] \quad (49)$$

One will find:

$$\eta_P < 0, \quad \eta_Q > 0 \quad (50)$$

Therefore,

$$\eta_{opt} = \eta_Q = \frac{-(\Psi b_2 - b_5) + \sqrt{(\Psi b_2 - b_5)^2 - 4(\Psi b_1 - b_4)(\Psi b_3 - b_6)}}{2(\Psi b_1 - b_4)} \quad (51)$$

The TVMD-EH damper was designed by employing the above optimal design method. **Figures 2–4** show the displacement frequency response curves of the SDOF

system (whose fundamental period is 4 s) for different electrical coefficients (κ^2), dimensionless time constants (α), and mass ratios (μ).

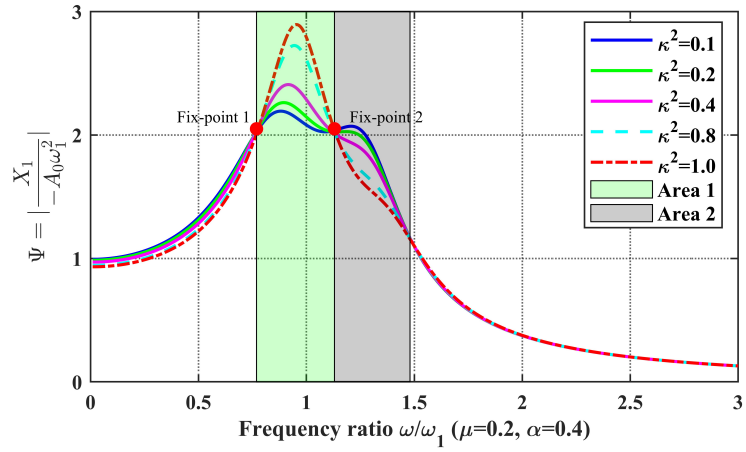


Figure 2. Displacement frequency response of the structure with proposed TVMD-EH device for different electrical coefficients.

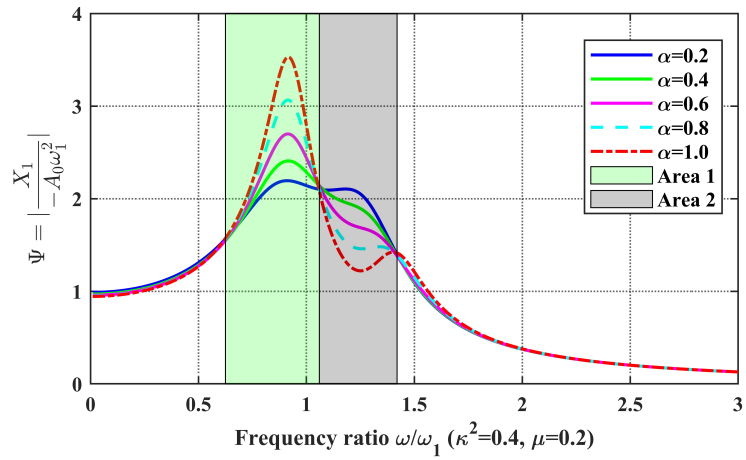


Figure 3. Displacement frequency response of the structure with proposed TVMD-EH device for different dimensionless time constants.

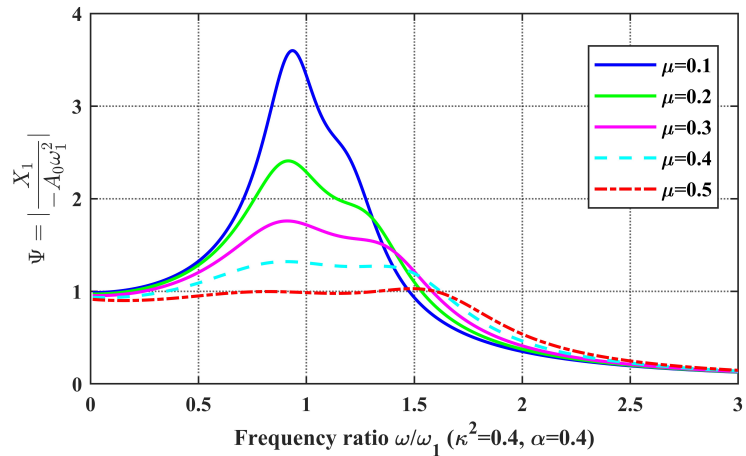


Figure 4. Displacement frequency response of the structure with proposed TVMD-EH device for different tuned mass ratios.

Figures 2 and 3 demonstrate that the fix-points exist for the SDOF system equipped with the proposed TVMD-EH device, which is consistent with the fact that the fix-points method was employed for the optimal design of the mechanical devices. A larger electrical coefficient (κ^2) and dimensionless time constant (α) will compromise the displacement control effect in the frequency band between two fixed points (Area 1, which is shown in light green) while the excitation frequency is near the fundamental frequency of the SDOF system. However, the greater electrical coefficients (κ^2) and dimensionless time constants (α) can enable the structure to experience smaller displacements in the frequency range as shown in light-dark color (Area 2). (In **Figure 2**, two fixed points are identified and marked. Although a third “intersection” may appear visually, it is not a genuine fixed point: the two curves approach each other asymptotically and seem to cross due to finite plotting resolution, but they do not intersect exactly and the fixed-point condition is not satisfied.)

Figure 4 shows the displacement frequency response of the structure with the proposed TVMD-EH device for different mass ratios. The results show that a larger tuned mass ratio (in the range of $[0.1, 0.5]$) of the damper can be better for mitigating displacement responses. However, it is notable that there is no fixed point while the tuned mass ratio is the variable.

Figure 5 is the face curve of the optimal loss factors (η_{opt}) of the TVMD-EH device for various κ^2 and α while the mass ratio and inherent damping ratio are set to 0.2 and 0.05, respectively. The increase of electrical coefficients and dimensionless time constants will decrease the optimal loss factors, which underlying the fact that the existence of the energy harvesting component will increase the overall damping of the TVMD-EH device. However, it is worth noting that the optimal loss factor is rapidly approaching zero while κ^2 and α are closing to 0.5 and 2.0, respectively. This will make it maybe impossible to obtain a reasonable optimal loss factor for high electrical coefficients and dimensionless time constants.

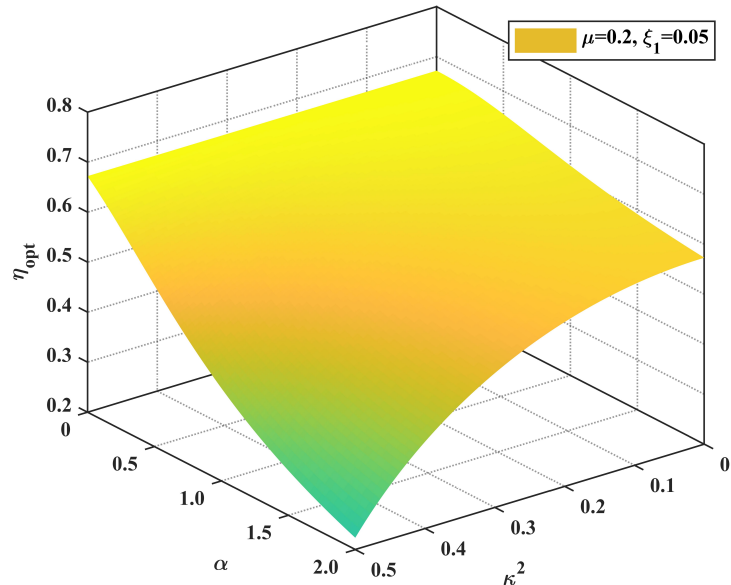


Figure 5. Optimal loss factors (η_{opt}) of the proposed TVMD-EH device for various electrical coefficients and dimensionless time constants.

4. Optimal design of the electrical elements of the TVMD-EH system

Figure 6 shows the normalized displacement frequency responses of the primary structure for various values of different dimensionless time constants and electrical coefficients against frequency ratio $\gamma (= \omega/\omega_1)$. Two subplots share a similar qualitative nature and both of them have hump-shaped cross sections. With a close look at **Figure 6a**, there exist two peaks when the dimensionless time constants are in the range of $[0, 2.0]$ and it is possible to obtain an optimal dimensionless time constant α_{opt} at which the curve has a minimum. Regarding **Figure 6b**, there also exists two peaks at low values of κ^2 . However, those two peaks dissolve into one as the increase of κ^2 and this also implies that a larger electrical coefficient will bring a negative impact on displacement control.

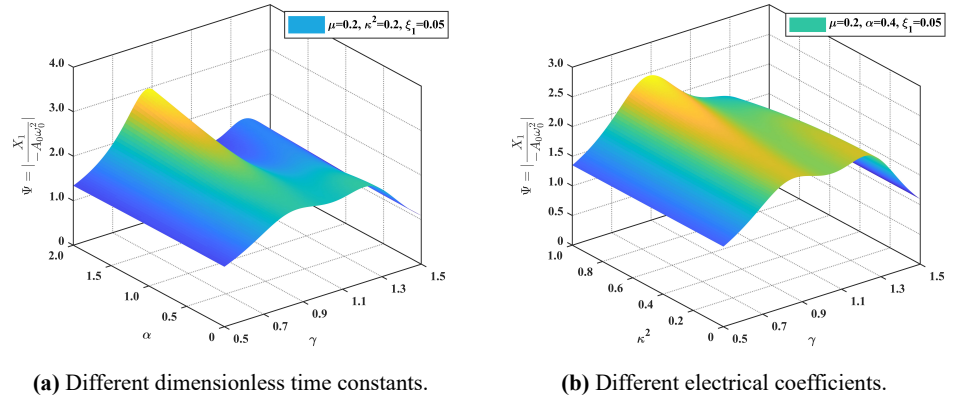


Figure 6. Normalized displacement frequency responses of the primary structure for various values of different dimensionless time constants and electrical coefficients.

The power harvested from the TVMD-EH device is [62,63]:

$$P = \frac{|V|^2}{R} = \frac{|V|^2}{\alpha} C_P \omega_d \quad (52)$$

therefore,

$$\frac{P}{(-A_0/\omega_1^2)^2} = \frac{\frac{|V|^2}{\alpha} C_P \omega_d}{(-A_0/\omega_1^2)^2} = \frac{C_P \omega_d |V|^2}{\alpha (-A_0/\omega_1^2)^2} \quad (53)$$

By recalling,

$$\left| \frac{V}{-A_0/\omega_1^2} \right| = \left| \frac{\frac{-i\Theta\gamma\alpha\beta^2}{C_P}}{\left\{ \begin{aligned} &(-\gamma^2 + 2i\xi_1\gamma + 1 + \mu\beta^2)[i\gamma\alpha\beta^2\kappa^2 - (\gamma^2 - 2i\xi_d\gamma\beta - \beta^2)(i\gamma\alpha + \beta)] \\ &- \mu\beta^4(i\gamma\alpha + \beta) \end{aligned} \right\}} \right| \quad (54)$$

thus,

$$\begin{aligned}
 \frac{P}{(-A_0/\omega_1^2)^2} &= \frac{C_P \omega_d |V|^2}{\alpha (-A_0/\omega_1^2)^2} \\
 &= \frac{\frac{C_P \omega_d}{\alpha} \left(\frac{\Theta \gamma \alpha \beta^2}{C_P} \right)^2}{\left| (-\gamma^2 + 2i\xi_1 \gamma + 1 + \mu\beta^2)[i\gamma\alpha\beta^2\kappa^2 - (\gamma^2 - 2i\xi_d \gamma \beta - \beta^2)(i\gamma\alpha + \beta)] - \mu\beta^4(i\gamma\alpha + \beta) \right|^2} \\
 &= \frac{\alpha \omega_d \gamma^2 \Theta^2 \beta^4 / C_P}{\left| (-\gamma^2 + 2i\xi_1 \gamma + 1 + \mu\beta^2)[i\gamma\alpha\beta^2\kappa^2 - (\gamma^2 - 2i\xi_d \gamma \beta - \beta^2)(i\gamma\alpha + \beta)] - \mu\beta^4(i\gamma\alpha + \beta) \right|^2}
 \end{aligned} \tag{55}$$

Because $\Theta^2 = C_P k_d \kappa^2$, therefore,

$$\begin{aligned}
 \frac{P}{(-A_0/\omega_1^2)^2} &= \frac{C_P \omega_d |V|^2}{\alpha (-A_0/\omega_1^2)^2} \\
 &= \frac{\alpha \omega_d k_d \gamma^2 \beta^4 \kappa^2}{\left| (-\gamma^2 + 2i\xi_1 \gamma + 1 + \mu\beta^2)[i\gamma\alpha\beta^2\kappa^2 - (\gamma^2 - 2i\xi_d \gamma \beta - \beta^2)(i\gamma\alpha + \beta)] - \mu\beta^4(i\gamma\alpha + \beta) \right|^2}
 \end{aligned} \tag{56}$$

By substituting,

$$\omega_d = \beta \omega_1, \quad k_d = m_d \omega_d^2 = \mu m_1 \beta^2 \omega_1^2, \quad \omega_d k_d = \mu m_1 \beta^3 \omega_1^3 \tag{57}$$

into Equation (56) obtains:

$$\frac{P}{(-A_0/\omega_1^2)^2} = \frac{\alpha \mu m_1 \omega_1^3 \gamma^2 \beta^7 \kappa^2}{\left| (-\gamma^2 + 2i\xi_1 \gamma + 1 + \mu\beta^2)[i\gamma\alpha\beta^2\kappa^2 - (\gamma^2 - 2i\xi_d \gamma \beta - \beta^2)(i\gamma\alpha + \beta)] - \mu\beta^4(i\gamma\alpha + \beta) \right|^2} \tag{58}$$

Letting $\beta = \beta_{opt} = \frac{1}{\sqrt{1-\mu}}$ yields:

$$\frac{P}{(-A_0/\omega_1^2)^2} \Big|_{\beta = \beta_{opt}} = \frac{\frac{\alpha \mu m_1 \omega_1^3 \gamma^2 \kappa^2}{(1-\mu)^3 \sqrt{1-\mu}}}{\left| (-\gamma^2 + 2i\xi_1 \gamma + 1 + \mu\beta_{opt}^2)[i\gamma\alpha\beta_{opt}^2\kappa^2 - (\gamma^2 - 2i\xi_d \gamma \beta_{opt} - \beta_{opt}^2)(i\gamma\alpha + \beta_{opt})] - \mu\beta_{opt}^4(i\gamma\alpha + \beta_{opt}) \right|^2} \tag{59}$$

Furthermore,

$$\begin{aligned}
 \frac{P}{(-A_0/\omega_1^2)^2} \frac{1}{m_1 \omega_1^3} &= \frac{P}{m_1 A_0^2 / \omega_1} \\
 &= \frac{\frac{\alpha \mu \gamma^2 \kappa^2}{(1-\mu)^3 \sqrt{1-\mu}}}{\left| (-\gamma^2 + 2i\xi_1 \gamma + 1 + \mu\beta_{opt}^2)[i\gamma\alpha\beta_{opt}^2\kappa^2 - (\gamma^2 - 2i\xi_d \gamma \beta_{opt} - \beta_{opt}^2)(i\gamma\alpha + \beta_{opt})] - \mu\beta_{opt}^4(i\gamma\alpha + \beta_{opt}) \right|^2}
 \end{aligned} \tag{60}$$

Finally, by setting $\xi_d = \xi_d^{opt} = \frac{1}{2} \eta_{opt}$ can obtain:

$$\frac{P}{m_1 A_0^2 / \omega_1} \bigg|_{\xi_d = \xi_d^{opt}} = \frac{\frac{\alpha \mu \gamma^2 \kappa^2}{(1-\mu)^3 \sqrt{1-\mu}}}{\left| (-\gamma^2 + 2i\xi_1 \gamma + 1 + \mu\beta_{opt}^2)[i\gamma\alpha\beta_{opt}^2\kappa^2 - (\gamma^2 - 2i\xi_d^{opt}\gamma\beta_{opt} - \beta_{opt}^2)(i\gamma\alpha + \beta_{opt})] \right|^2 - \mu\beta_{opt}^4(i\gamma\alpha + \beta_{opt})} \quad (61)$$

Figure 7 shows the normalized power frequency responses of the primary structure for various values of different dimensionless time constants and electrical coefficients. **Figure 7a** depicts that the frequency response curve is unimodal near the fundamental frequency of the SDOF system while the dimensionless time constant falls in the range of $[0, 2.0]$. However, **Figure 7b** exhibits a bimodal characteristic while the electrical coefficient is small, and the frequency response curve changes to an unimodal shape while the electrical coefficient increases. The above observation underlies the possibility of finding the optimal electrical parameters of the EH element to achieve broadband energy harvesting at low values of the electrical coefficient.

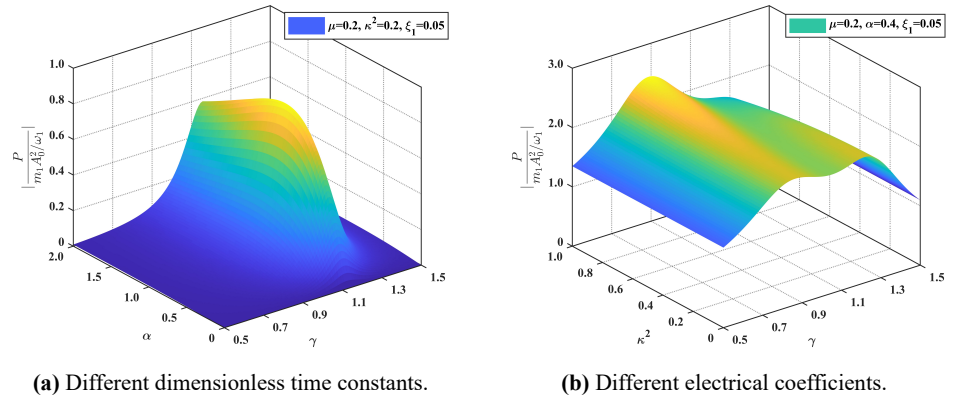


Figure 7. Normalized power frequency responses of the primary structure for various values of different dimensionless time constants and electrical coefficients.

Figure 8 shows the maximum normalized displacement frequency responses against dimensionless time constants and electrical coefficients. The responses first decrease with the increase of dimensionless time constants and electrical coefficients and then again increase with these two parameters. The peak normalized displacement frequency responses have the minimums while both the α and κ^2 have small values. In **Figure 8**, the marked line shows the minimum peak normalized displacement frequency responses, and this line can provide a linkage between α and κ^2 .

This study determines the optimal values of dimensionless time constant and electrical coefficient based on achieving the best displacement control for the protected structure. **Figure 9** is the polynomial fit (two-order) of dimensionless time constant (α) and electrical coefficient (κ^2) corresponding to minimum peak normalized displacement frequency responses. It is worth noting that the fitting results will be different for various tuned mass ratios (μ) and inherent damping ratios (ξ_1). Afterward, the optimal parameter pairs can be obtained by employing the curve-fitting relationship for which the structure exhibits minimum peak normalized displacement.

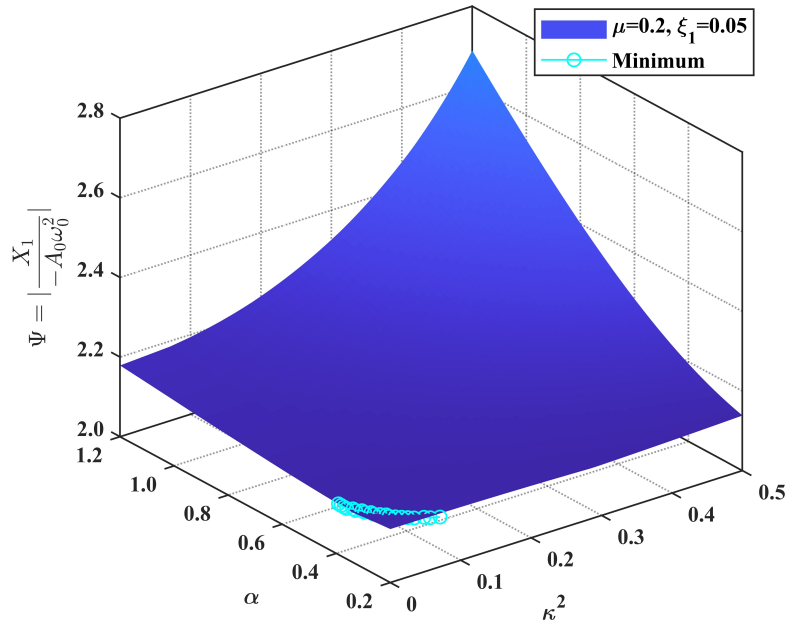


Figure 8. Peak normalized displacement frequency responses.

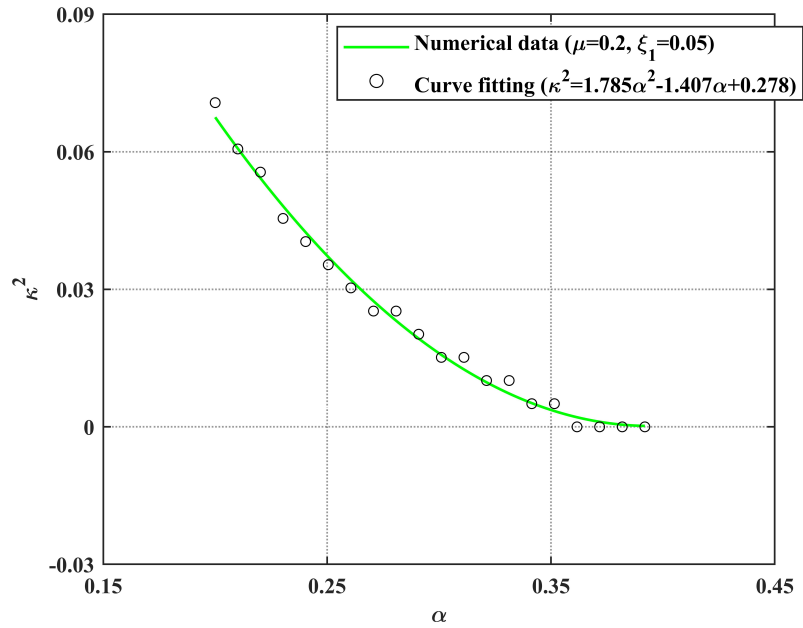


Figure 9. Polynomial fit of dimensionless time constants and electrical coefficients.

Figures 10 and 11 show the normalized displacement and power frequency responses for various dimensionless time constants, respectively. The related electrical coefficients are decided using the polynomial fit curve shown in **Figure 9**. The dimensionless time constant (α) is 0.36, for which the peak normalized displacement is minimum. However, its value is 0.20, while the normalized power is maximum. Therefore, these two figures provide a tradeoff for designers to choose the optimal dimensionless time constant.

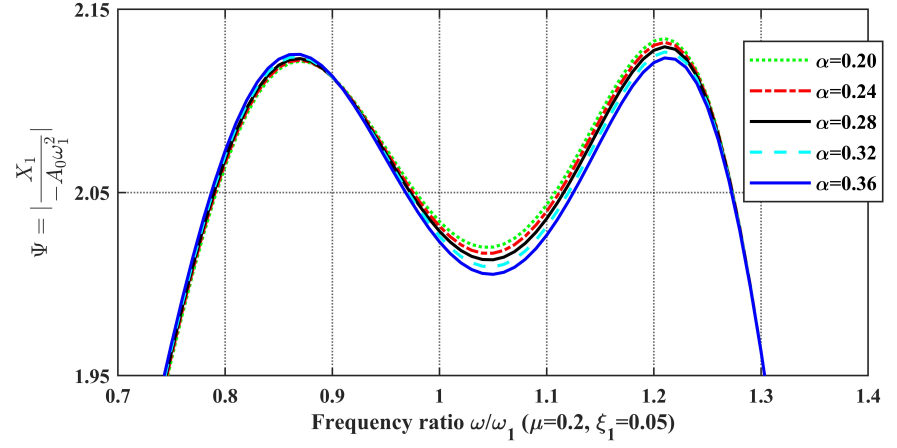


Figure 10. Normalized displacement frequency responses of the primary structure for various dimensionless time constants corresponding to mass ratio $\mu = 0.2$ and $\zeta_1 = 0.05$.

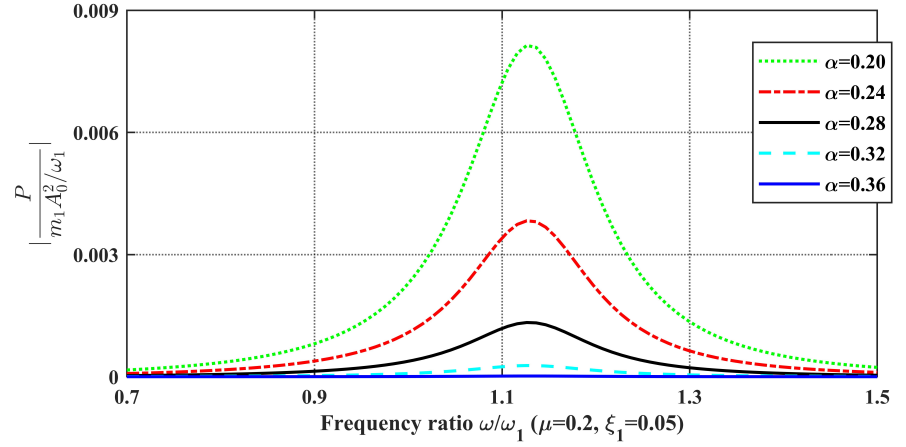


Figure 11. Normalized power frequency responses for various dimensionless time constants corresponding to mass ratio $\mu = 0.2$ and $\zeta_1 = 0.05$.

As depicted in the two figures, the differences among the normalized displacement frequency responses are less apparent than those of the normalized power frequency responses. Thus, one can choose $\alpha = 0.20$ which can almost achieve a similar reduction of displacement frequency responses and exhibit the best energy harvesting performance.

Figure 12 compares normalized displacement frequency responses between the structures equipped with the proposed TVMD-EH device and the conventional optimal-designed TVMD device [40, 44]. The results show that the TVMD-EH device exhibits better control of displacement frequency responses than a conventional optimal-designed TVMD device without energy harvesting capability. This also shows that while the TVMD device has enabled energy harvesting ability, the damping requirement will increase due to a larger optimal damping ratio and the existence of the EH loop.

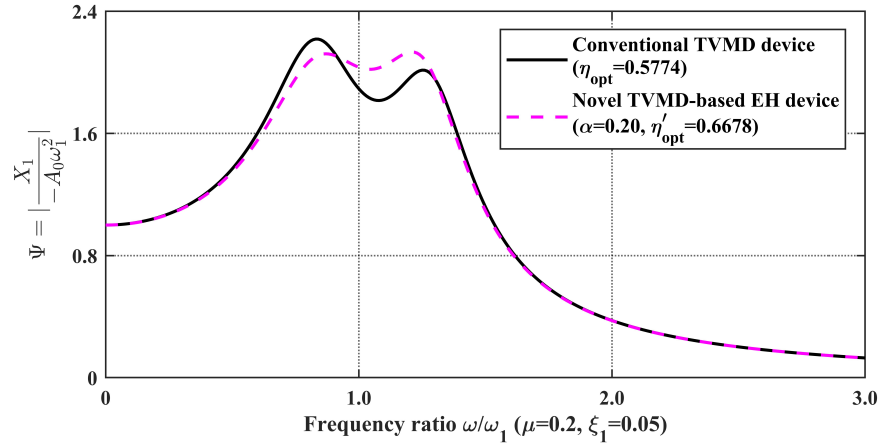


Figure 12. Comparison of normalized displacement frequency responses between the structures equipped with the proposed TVMD-EH device and conventional optimal-designed TVMD device corresponding to mass ratio $\mu = 0.2$ and $\zeta_1 = 0.05$.

5. Feasibility investigation of the TVMD-EH system

5.1. Parameters of damping systems

An SDOF system was employed for a feasibility study to investigate the performance of the proposed TVMD-EH device for both seismic control and energy harvesting purposes. By recalling Equation (1), the state-space equation of the SDOF system equipped with the proposed TVMD-EH device is as follows:

$$\begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_d & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_d \\ \ddot{v} \end{bmatrix} + \begin{bmatrix} c_1 & 0 & 0 \\ 0 & c_d & 0 \\ 0 & \Theta & C_P \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_d \\ \dot{v} \end{bmatrix} + \begin{bmatrix} k_1 + k_d & -k_d & 0 \\ -k_d & k_d & -\Theta \\ 0 & 0 & 1/R \end{bmatrix} \begin{bmatrix} x_1 \\ x_d \\ v \end{bmatrix} = - \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_d & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix} \ddot{x}_g \quad (62)$$

An SDOF system whose mass, horizontal stiffness, and inherent damping ratio are 12,600 t, 31,089 kN/m, and 0.05, respectively, serves as the numerical building model (its fundamental period is 4 s). Therefore, the parameters of the TVMD-EH device can be obtained by employing the proposed optimal design method, as shown in **Table 1**.

Table 1. Parameters of the TVMD-EH and conventional TVMD devices.

Items	TVMD-EH	Conventional TVMD
β^{opt}		1.118
η^{opt}	0.6678	0.5774
c_d (kNs/m)	2955	2555
m_d (t)		2520
k_d (kN/m)		7772
ω_d (rad/s)		1.7562
μ		0.2
α	0.2	-
κ^2	0.0676	-
$C_P R$ (s/rad)	0.1139	-
Θ^2 / C_P (kN/m)	525.4	-

In this study, the reported “mass ratio” should be understood as an equivalent inertial ratio realized by an inerter-based TVMD, rather than an actual requirement

to add a large physical secondary mass. Owing to the inerter's mass amplification effect, a compact mechanical device can provide a large apparent inertia, so the effective inertia in the model can be much larger than the device's physical mass. Consequently, adopting a relatively high equivalent inertial ratio is practically feasible, and the energy-harvesting performance can be maintained when the physical auxiliary mass is reduced, as long as the required inertance (effective inertia) is preserved.

Two earthquake records (namely, El Centro and Sakishima records) were selected for conducting numerical analyses. The Sakishima record was recorded in Osaka during the 2011 Great East Japan Earthquake. The time histories and power spectral densities are shown in **Figures 13** and **14**, respectively.

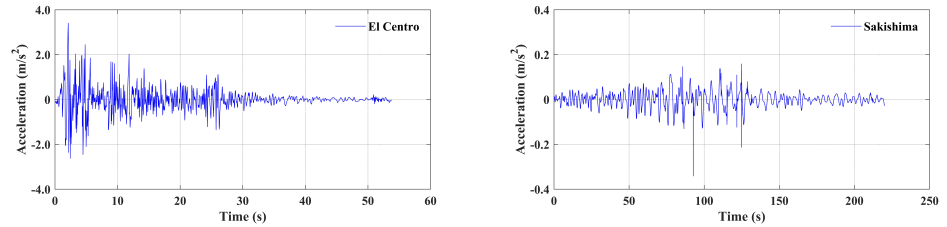


Figure 13. Time histories of the El Centro and Sakishima records.

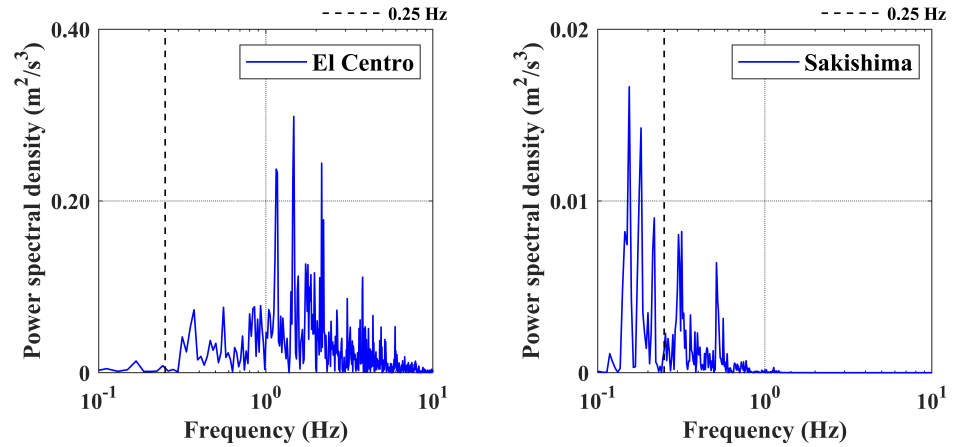


Figure 14. Power spectral densities of the El Centro and Sakishima records.

5.2. Responses of the TVMD-EH and TVMD systems

Figures 15 and **16** show the displacement and absolute acceleration of the SDOF system equipped with the TVMD-EH and TVMD devices when subjected to the El Centro and Sakishima records. The two damping devices exhibit a similar control effect on displacement and absolute acceleration. Generally, an increase in the damping ratio can achieve better control of displacement responses; however, an increase in acceleration will often accompany a larger damping ratio. The figures show no significant difference in the maximum displacements because the electrical coefficient is small. Although a larger optimal damping ratio of the TVMD part of the TVMD-EH device is selected, the absolute acceleration for the TVMD-EH system exhibits no significant disadvantage compared with its counterpart. Therefore, the TVMD-EH device, which includes an energy harvesting component, does not compromise the acceleration responses, while a larger damping ratio was selected.

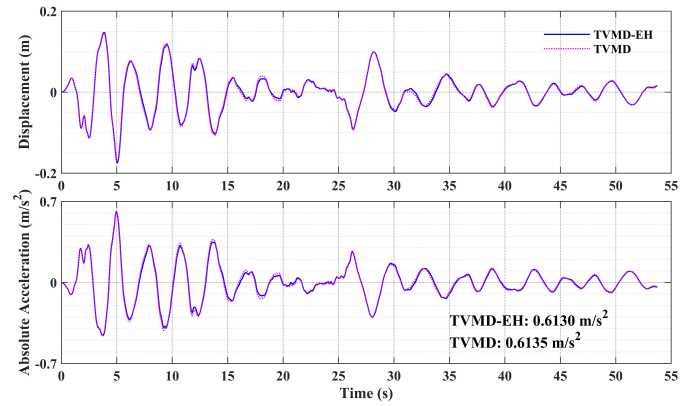


Figure 15. Displacement and absolute acceleration of the TVMD-EH and TVMD systems under the El Centro record.

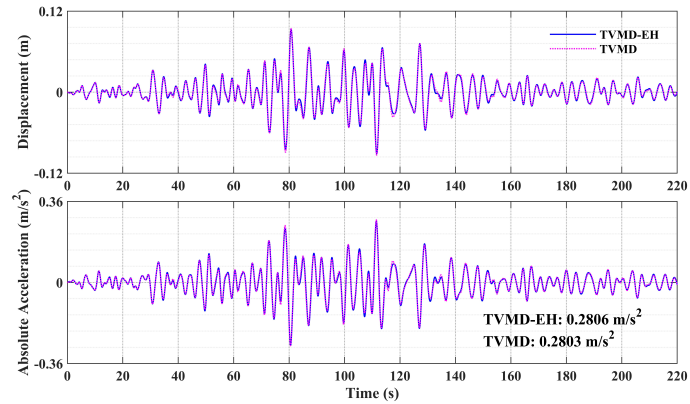


Figure 16. Displacement and absolute acceleration of the TVMD-EH and TVMD systems under the Sakishima record.

Figures 17 and 18 show the displacement and relative velocity of the viscous dampers of the TVMD-EH and TVMD devices. The viscous damper's displacement and relative velocity in the conventional TVMD device are more significant than those of the TVMD-EH device. This is caused by the fact that the damping coefficient of a viscous damper in the TVMD device (2555 kNs/m) is lower than that of the TVMD-EH device (2955 kNs/m).

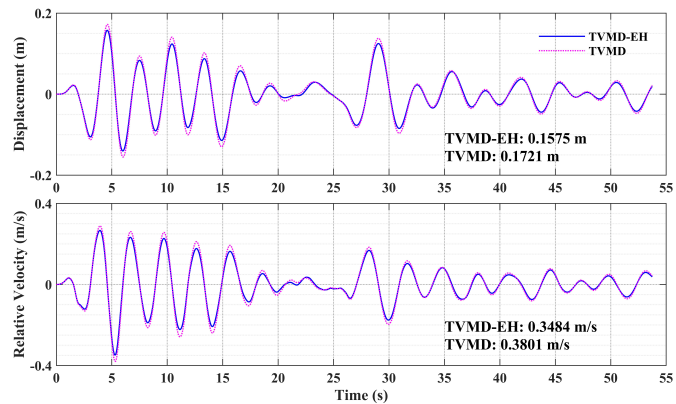


Figure 17. Displacement and relative velocity of the viscous dampers of the TVMD-EH and TVMD devices under the El Centro record.

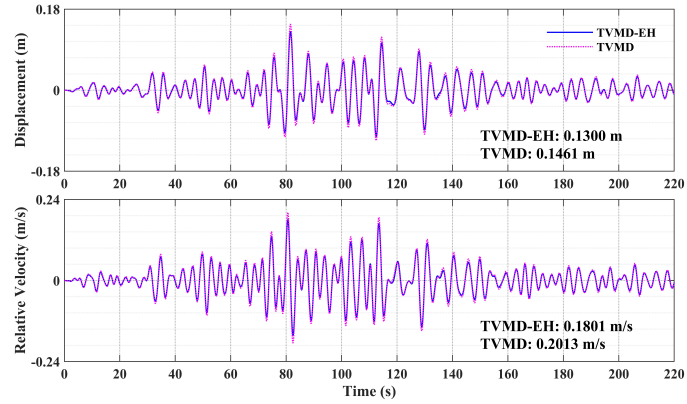


Figure 18. Displacement and relative velocity of the viscous dampers of the TVMD-EH and TVMD devices under the Sakishima record.

Figure 19 depicts the hysteresis curves of the TVMD-EH and TVMD devices. The curves for these two damping devices are similar in the cases of two selected records. However, the damping force of TVMD is slightly greater than that of the TVMD-EH device.

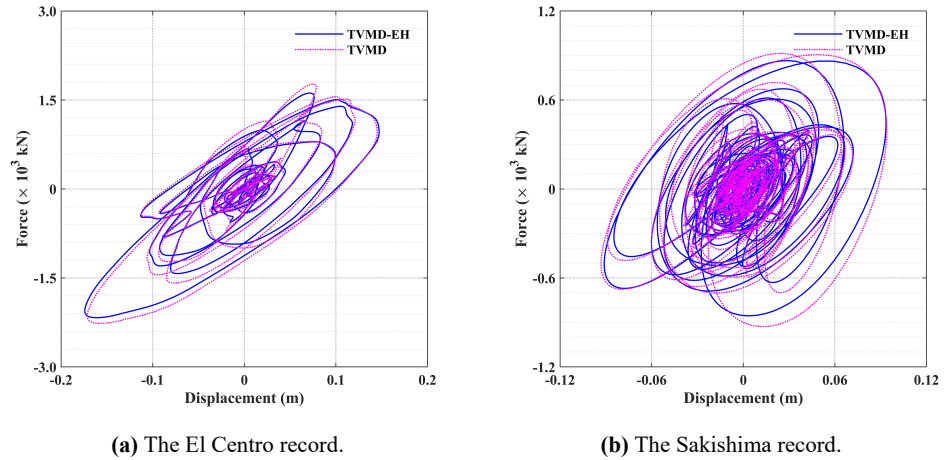


Figure 19. Hysteresis curve of the TVMD-EH and TVMD systems.

5.3. Energy harvesting of the TVMD-EH device

Figures 20 and 21 show the generated voltage, electric power, and harvested energy of the TVMD-EH system. Considerable voltage and electric power were generated in the device when the system was subjected to the El Centro and Sakishima records. The voltage, power, and harvested energy during the El Centro record are significantly larger than those during the Sakishima record because the amplitude of the Sakishima record is much less than those of the El Centro record.

The electric power for the two records is up to 6.8 and 1.9 W, respectively, which is sufficient to replace the batteries currently used in independent electronic sensors, transducer elements, and electronics [64–66]. Because earthquakes frequently happen in countries in earthquake zones, for example, Japan, the proposed TVMD-EH system is a promising approach for protecting buildings or significant mechanical devices in remote areas and converting some earthquake input energy into electricity.

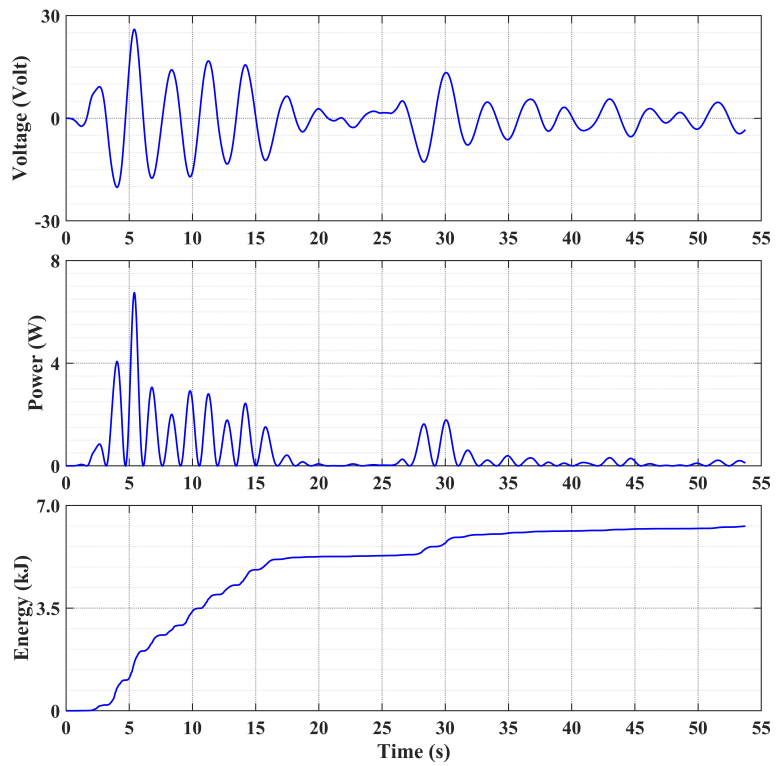


Figure 20. Generated voltage, electric power, and harvested energy of the TVMD-EH system under the El Centro record.

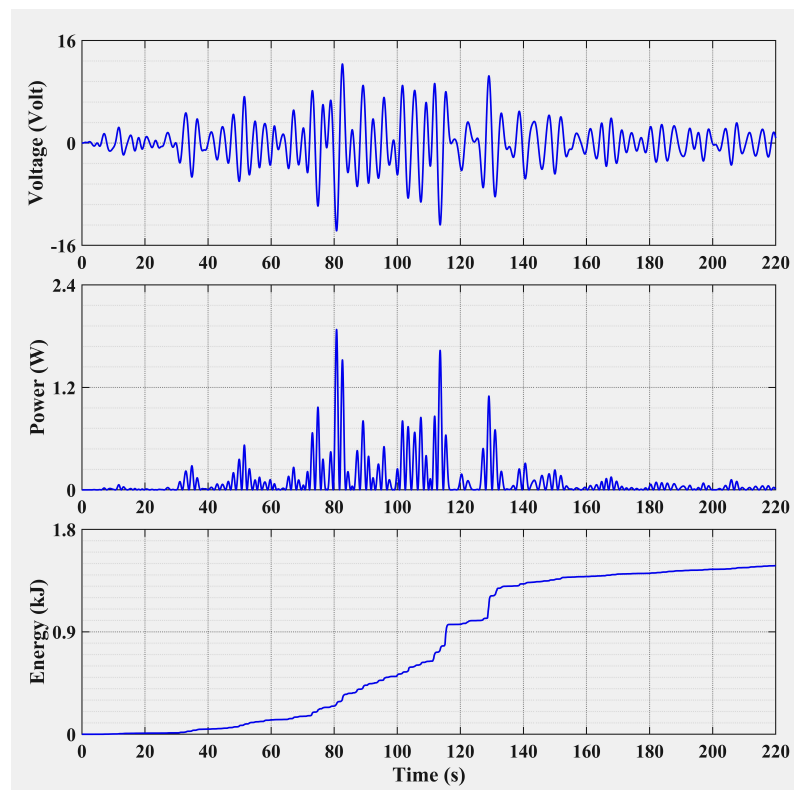


Figure 21. Generated voltage, electric power, and harvested energy of the TVMD-EH system under the Sakishima record.

5.4. Energy balance and energy distribution under earthquake excitation

To objectively evaluate the role of energy harvesting (EH) in the proposed seismic control system, we quantify the energy flow during the whole earthquake duration based on an energy balance formulation for a SDOF structure. The governing equation is written as:

$$m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 + f_{TVMD-EH}(t) = -m_1\ddot{x}_g \quad (63)$$

where $f_{TVMD-EH}(t)$ represents the additional force associated with the TVMD-EH device.

5.4.1. Definitions of instantaneous power and cumulative energies

- (1) Input energy from earthquake:

The instantaneous seismic input power to the SDOF system (in relative coordinates) is defined as:

$$P_{in}(t) = -m_1\dot{x}_1\ddot{x}_g \quad (64)$$

and the cumulative input energy is:

$$E_{in}(t) = \int_0^t P_{in}(\tau)d\tau \quad (65)$$

- (2) Kinetic energy:

$$E_k(t) = \frac{1}{2}m_1\dot{x}_1^2 \quad (66)$$

- (3) Strain (elastic) energy:

$$E_s(t) = \frac{1}{2}k_1x_1^2 \quad (67)$$

- (4) Viscous dissipation energy of inherent damping:

The instantaneous dissipation power associated with viscous damping is:

$$P_d(t) = c_1\dot{x}_1^2 \geq 0 \quad (68)$$

and the cumulative dissipated energy is:

$$E_d(t) = \int_0^t P_d(\tau)d\tau = \int_0^t c_1\dot{x}_1^2(\tau)d\tau \quad (69)$$

- (5) Dissipation energy of the TVMD device:

$$P_{TVMD}(t) = c_d v_d^2(t), \quad E_{TVMD}(t) = \int_0^t c_d v_d^2(\tau)d\tau \quad (70)$$

- (6) Harvested (circuit) energy

The harvested electrical energy is defined as the energy delivered to the electrical load/storage port. For a resistive load R ,

$$P_h = \frac{|V|^2}{R}, \quad E_h(t) = \int_0^t \frac{|V(\tau)|^2}{R} d\tau \quad (71)$$

5.4.2. Energy distribution map and interpretation

To visualize the energy allocation, we present:

(i) cumulative energy curves $E_{in}(t)$, $E_k(t)$, $E_s(t)$, $E_d(t)$, $E_{TVMD}(t)$, and $E_h(t)$ over the entire earthquake duration and (ii) a normalized energy distribution expressed by the ratios:

$$\begin{aligned} \eta_k &= \frac{E_k(t)}{E_{in}(t)} \times 100\%, & \eta_s &= \frac{E_s(t)}{E_{in}(t)} \times 100\%, & \eta_d &= \frac{E_d(t)}{E_{in}(t)} \times 100\%, \\ \eta_h &= \frac{E_h(t)}{E_{in}(t)} \times 100\%, & \eta_{TVMD} &= \frac{E_{TVMD}(t)}{E_{in}(t)} \times 100\% \end{aligned} \quad (72)$$

These metrics directly quantify the proportion of harvested electrical energy relative to the total seismic input energy, enabling an objective evaluation of whether EH mainly serves as an auxiliary power source or a primary energy dissipation mechanism in the overall system.

Figure 22 presents the energy distribution maps for (a) the El Centro record and (b) the Sakishima record, showing how the earthquake input energy is partitioned into kinetic energy, elastic strain energy, viscous damping energy dissipation (including both the inherent structural damping and the TVMD dashpot), and the harvested electrical energy in the circuit over the entire time history. For both records, the input energy increases mainly during the strong-motion phase and is predominantly converted into viscous dissipation, while kinetic and strain energies fluctuate as temporary storage terms and diminish after the main excitation. The zoomed inset demonstrates that the harvested energy remains extremely small compared with the total input energy, which indicates that energy harvesting in this system functions as an auxiliary power supply rather than a primary means of energy dissipation.

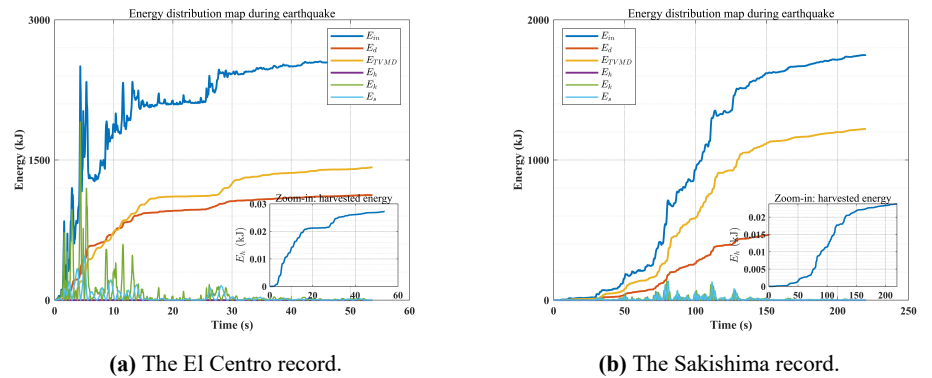


Figure 22. Earthquake energy partitioning and harvested energy contribution.

6. Conclusion

This study proposed a novel TVMD-EH system, which can simultaneously achieve reasonable seismic control of long-period structures and convert the earthquake input energy into electricity. The main contributions are summarized as follows:

- (1) A novel TVMD-EH device was developed by adding an energy harvesting loop to the conventional TVMD device. The piezoelectric material was adopted to generate electricity for harvesting energy. The optimal design methods for the mechanical and electrical elements of the TVMD-EH system were developed while achieving the best displacement reduction and synchronously enabling good energy harvesting capability.
- (2) When subjected to the El Centro and Sakishima records, the TVMD-EH and TVMD devices exhibit similar control effects on displacement and absolute acceleration. Although a larger damping ratio was selected, the TVMD-EH device, which includes an energy harvesting component, does not compromise the control effects of acceleration responses.
- (3) Both the displacement and relative velocity of the viscous damper in the conventional TVMD device are more significant than those of the TVMD-EH device, which causes the damping coefficient of viscous damping in the TVMD device to be smaller than that of the TVMD-EH device. The damping force of the TVMD is slightly greater than that of the TVMD-EH device.
- (4) Considerable voltage and electric power were generated when the TVMD-EH damping system was subjected to the El Centro and Sakishima records. The electric power for the two records is up to 6.8 and 1.9 W, respectively, which is sufficient to replace the batteries currently used in low-energy consumption equipment. The proposed TVMD-EH system is a promising approach for protecting buildings or significant mechanical devices in remote areas and energy harvesting, especially in countries in earthquake zones.
- (5) The energy distribution results demonstrate that, for both earthquake records, the system dissipates most of the input energy through viscous damping (including the TVMD damper), while the harvested electrical energy remains negligible in comparison, confirming that energy harvesting serves only as an auxiliary power supply rather than a primary energy dissipation mechanism.

However, this study still needs experimental verification. Therefore, the TVMD-EH device will be designed and fabricated in the future, and a series of experiments will be conducted using a shaking table. Looking ahead, the TVMD-EH concept can support resilient cities by combining seismic mitigation with decentralized auxiliary power for post-earthquake sensing and communications [66–68]. Integrated into smart-infrastructure and resilience planning, such self-powered devices may enhance robustness and speed recovery of long-period structures and urban lifelines in seismic regions [69–71].

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Article

A biomimetic design framework for structurally optimized lifting beams using graded geometry and internal ribs

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Abstract: This paper presents a comprehensive biomimetic design framework for lifting beams that couples an exponentially graded outer shell with an internal dendritic rib network to maximize stiffness-to-weight performance while meeting serviceability and safety constraints. A multi-fidelity modelling chain is developed: a variable-section Euler–Bernoulli model for rapid sizing, a shear-corrected Timoshenko formulation for regimes in which transverse shear is significant, and a 2D arch-frame finite-element model that resolves local rib–shell interactions and stress concentrations. A density-based SIMP topology-optimization workflow is integrated with parametric regression to extract manufactural rib trajectories, and analytical closed-form expressions are derived for second moment contributions of graded shells and discrete ribs. Morphologically, the final optimized topology converges to a graded cellular arch frame resembling a chiropteran bat-wing, where the internal dendritic lattice functions as a variable-depth Warren truss to effectively decouple shear flow from the bending-dominated outer shell. Extensive analytical investigations: parametric sweeps, one-factor-at-a-time sensitivity, and first-order uncertainty propagation, demonstrate that graded thickness and load-path aligned ribs increase the section modulus and reduce peak bending demands; for representative baseline geometries and materials, the proposed topology yields ~20% reduction in peak bending stress and ~15% reduction in midspan deflection at equal mass compared with conventional solid sections. High-fidelity FEA highlights local saw-tooth stress peaks at rib roots that exceed mean analytical estimates by ~60%, indicating the necessity of filleting, fatigue-aware detailing, and AM process control. The manuscript concludes with a rigorous experimental validation roadmap (AM prototyping, DIC, static and fatigue testing, CT/NDT) and recommends embedding uncertainty-aware surrogates and single-loop multidisciplinary optimization to ensure robust, certifiable lifting hardware under multi-source variability.

Keywords: biomimetic design; lifting beam; graded thickness; rib lattice; topology optimization; Timoshenko beam; additive manufacturing

1. Introduction

Conventional lifting beams are frequently over-engineered to satisfy stringent safety factors, resulting in unnecessary material usage and increased structural weight. In contrast, biological systems, such as leaf venation networks, trabecular bone scaffolds, and arboreal branching morphologies, demonstrate highly optimized material distribution strategies under complex loading conditions, yielding exceptional strength-to-weight ratios. Inspired by these naturally evolved systems, engineering components can be strategically designed to support mechanical loads with minimal

material input. In this context, the present study proposes a bio-inspired lifting beam that incorporates branch-like internal ribs and a graded shell thickness profile, forming a load-responsive architecture that reduces bending stress concentrations and improves overall structural efficiency.

Potential applications of this bio-inspired lifting beam span a variety of sectors:

- (i) **Industrial cranes:** The reduction in structural dead weight enhances lifting capacity and reduces energy expenditure during operation.
- (ii) **Modular construction:** Lightweight beam elements enable faster and more efficient assembly and disassembly of prefabricated structures.
- (iii) **Aerospace payload handling:** The proposed design offers high stiffness-to-weight ratios suitable for satellite transport and launch systems.
- (iv) **Marine salvage operations:** Corrosion-resistant variants can serve as underwater lifting frameworks, benefiting from optimized buoyancy and reduced drag.

Nature employs dendritic systems to distribute stress and redirect loads efficiently:

- (i) **Leaf venation:** Hierarchical vein structures enable fluid and mechanical load redistribution around damage-prone zones, maintaining functional continuity.
- (ii) **Trabecular bone:** The internal porous scaffold aligns along principal stress paths, continuously adapting to dynamic mechanical demands.
- (iii) **Tree branching:** Tapered geometries facilitate the smooth transition of bending moments from distal extremities to the trunk, optimizing structural resilience.

By abstracting and synthesizing these principles, namely branch-aligned load paths, gradient material distribution, and hierarchical ribbing the proposed lifting beam introduces a multifunctional internal lattice that mirrors the adaptive efficiency of natural forms.

The evolving demands of modern structural engineering increasingly focused on efficiency, resilience, and sustainability have catalyzed a shift toward design methodologies grounded in computational intelligence and biomimetic insight. Traditional approaches, reliant on empirical safety margins and uniform geometries, often produce conservative structures with suboptimal material economy. In contrast, natural systems, honed by evolutionary pressures, deliver mechanically optimal solutions through hierarchical organization, graded geometries, and topology-adaptive responses. Biomimetics, the convergence of biological observation with engineering design, thus emerges as a compelling paradigm for next-generation structural solutions.

Foundational work by Sarikaya [1] emphasized the structural grammar of biological materials, such as nacre, trabecular bone, and plant stems, demonstrating how multiscale architectures enable superior mechanical performance. Waggoner and Kestner [2] further mapped the historical trajectory and emerging opportunities for integrating biomimetic principles into structural systems, highlighting motifs such as branching, folding, and cellular structuring. Joshy [3] extended this framework to engineered load paths, illustrating how dendriform geometries inspired by tree architecture reduce material demand and enhance mechanical adaptability.

In parallel, advancements in computational design have revolutionized structural

optimization. Tang and Chang [4] demonstrated early implementations of topology and shape optimization, leveraging algorithmic exploration to exceed human design intuition in identifying material-efficient forms. Wu et al. [5] expanded this paradigm by introducing shell–infill composite optimization tailored for additive manufacturing, producing architectures reminiscent of trabecular bone with inherent material gradients and void optimization. More recent AI-driven methods, such as those by Oh et al. [6], Watson et al. [7], and Hooshmand and Campbell [8], fuse generative algorithms with structural optimization, resulting in adaptive geometries aligned with natural stress trajectories.

Specialized studies have extended these approaches to account for dynamic and frequency-dependent performance. Thomas et al. [9] integrated stiffness and vibration criteria into the optimization of periodic structures, echoing biological systems' multi-functionality. Oliver et al. [10] contributed a pseudo-time variational model that facilitates efficient, closed-form optimization of complex topologies. These models mimic biological processes such as bone remodeling and structural reconfiguration in response to environmental stimuli.

Experimental validations further substantiate the potential of biomimetic structures. Siddique et al. [11] fabricated and tested cornstalk-inspired lattices, revealing superior load-bearing capacity under various stress states. Röver et al. [12] designed bending-optimized biomimetic beams, demonstrating performance advantages in both stress management and material efficiency. Gandhi and Minak [13] provided a comprehensive synthesis of topology optimization for composite materials, underscoring the compatibility of biomimetic geometries with anisotropic, high-performance structures. Likewise, Mehboob et al. [14] and Tian et al. [15] demonstrated successful applications in orthopedic plates and robotic exoskeletons by incorporating shrimp shell and bone-inspired geometries.

Broad analyses by Dixit and Stefańska [16] affirmed the viability of nature-inspired frameworks across structural disciplines, with an emphasis on branching and folding as recurring motifs in both natural and engineered systems. Building on this, Hassan and Saeed [17] highlighted lightweight bio-derived morphologies in aeronautical and botanical analogues, particularly for their adaptability and strength efficiency.

Further advancements include avian-inspired folding mechanisms for dynamic load accommodation [18], compliant and fatigue-resilient flapping systems [19], and Morales's comparative framework [20] that clarifies the complementary roles of generative design and topology optimization in complex mechanical assemblies.

The theoretical foundation for bio-inspired structural efficiency traces back to Wolff's Law [21], which posited that trabecular bone remodels its internal architecture in direct response to applied mechanical loads, optimizing material placement to resist specific stress states. This biological adaptability provides the precedent for modern engineering, where the work of Ashby and Gibson [22] on cellular solids quantifies why these porous, lattice-like structures offer specific stiffness far superior to monolithic materials. Gao et al. [23] further elucidated that nature's hierarchical organization from the nanoscale to the macroscale renders these materials remarkably insensitive to flaws, a characteristic highly desirable in safety-critical heavy lifting equipment. To synthesize

these biological principles into engineering geometries, Mattheck [24] formalized the “axiom of uniform stress” observed in trees, demonstrating that biological growth adds material precisely where stress concentrations occur to homogenize the surface load distribution, a principle directly applicable to the graded shell thickness proposed in this study.

The governing mathematical framework for simulating this biological growth in engineering is the Solid Isotropic Material with Penalization (SIMP) method, pioneered by Bendsøe and Sigmund [25]. This method relaxes the binary material distribution problem into a continuous density field, allowing gradient-based algorithms to “grow” optimal structures that frequently resemble biological forms. The versatility of such topology optimization frameworks has been rigorously established by Guest and Prévost [26], who extended these density-based formulations beyond solid mechanics to encompass fluid flows, validating the mathematical robustness of the approach in managing complex flux trajectories and “force flows” within a continuum. Finally, the translation of these theoretical topologies into physical components is addressed by Stampfl et al. [27], who highlight that the convergence of generative design with additive manufacturing now permits the fabrication of bio-inspired cellular structures with mechanical properties that were previously unattainable via traditional subtractive manufacturing.

Traditional approaches, reliant on empirical safety margins and uniform geometries, often produce conservative structures with suboptimal material economy. In contrast, natural systems, honed by evolutionary pressures, deliver mechanically optimal solutions through hierarchical organization. While topology optimization and biomimetic geometries can significantly reduce mass, the stochastic nature of additive manufacturing requires rigorous validation. As noted in recent literature, while algorithmic mass reduction is achievable, conservative design margins and physical proof testing remain mandatory for safety-critical applications as appear in Yang et al. [28] and Meng and Zhu [29].

Recent progress in reliability-based and multidisciplinary optimization provides an essential methodological foundation for translating biomimetic topology concepts into safety-critical structural components. Yang et al. [28] introduced a robust-weighted hybrid nonlinear regression framework for reliability-based topology optimization under multi-source uncertainties, explicitly addressing the combined effects of material variability, geometric tolerances, load randomness, and model-form uncertainty. By integrating robust regression surrogates with probabilistic constraints, their approach enables accurate and computationally efficient enforcement of reliability criteria without excessive Monte Carlo simulations, making it particularly suitable for complex structures with manufacturing-induced variability, such as additively manufactured lifting devices. Complementarily, Meng and Zhu [29] presented a comprehensive multidisciplinary design optimization (MDO) framework for complex structures under uncertainty, unifying structural mechanics, manufacturing considerations, and performance objectives within single-loop optimization strategies. Their work emphasizes uncertainty propagation, sensitivity analysis, and reliability metrics as integral components of topology optimization, demonstrating that optimal designs must

balance stiffness, mass, fatigue life, and manufacturability rather than relying solely on deterministic compliance minimization. Together, these studies underscore the necessity of embedding uncertainty-aware surrogates and multidisciplinary reliability constraints into topology optimization frameworks. The present work is conceptually aligned with this paradigm by adopting bio-inspired material distribution strategies that can be naturally extended toward robust, reliability-informed optimization of lifting beams operating under stochastic loads and manufacturing variability [29].

This work, however, addresses the “deep beam” nature of the geometry by incorporating shear deformation principles and benchmarking against optimized industrial standards rather than simplified solid sections.

Synthesizing these perspectives, the present work contributes a new lifting beam topology that unifies ribbed branching, shell gradient distribution, and hierarchical lattices, achieving enhanced stress distribution and minimized mass.

From this synthesis of biomimetic principles and computational methodologies, several overarching design paradigms emerge. First, hierarchical organization and spatially graded material distribution are ubiquitous in natural systems, consistently outperform uniform or monolithic structures in managing load distribution and mitigating failure. Second, the convergence of advanced optimization algorithms with generative and adaptive design strategies, especially within the realm of additive manufacturing, enables the physical realization of geometrically complex forms previously considered infeasible to fabricate. Third, the integration of analytical precision with biologically inspired architectures gives rise to a new class of structural solutions that are simultaneously efficient, manufactural, and environmentally responsive.

Building upon this robust interdisciplinary foundation, the present study introduces a novel lifting beam topology that integrates branch-like ribbing, a graded shell thickness profile, and a hierarchical internal lattice. This design framework aims to achieve significant reductions in both stress concentration and deflection, while maintaining equivalent mass relative to conventional configurations.

Accordingly, this research situates itself at the confluence of biomimetic innovation and computational structural design. The proposed bio-inspired lifting beam leverages nature-derived features, branch-like rib patterns, graded shell profiles, and hierarchical lattice arrangements to minimize weight, enhance stress distribution, and improve mechanical performance under realistic loading scenarios. Through rigorous analytical modeling, parametric exploration, and comparative validation, this study offers a viable framework for developing next-generation lightweight structures that seamlessly merge biological insight with engineering sophistication.

The primary engineering objectives are: (i) Minimize mass subject to constraints on peak stress $\sigma_{\max}$ and midspan deflection δ . (ii) Achieve a more uniform stress distribution and higher specific stiffness than traditional sections at equal mass. (iii) Provide manufactural geometry for additive manufacturing or hybrid fabrication. (iv) Retain redundancy and damage tolerance through multiple internal ribs. The remainder of the design presents a parameterized geometry that allows systematic exploration of this design space.

This work acknowledges the specific curvilinear geometry (**Figure 1**). We

employ a multi-fidelity validation approach to isolate the effects of shear deformation and arch-geometry coupling, comparing the Euler-Bernoulli analytic solution, the Timoshenko beam analytic solution, and the finite element method.

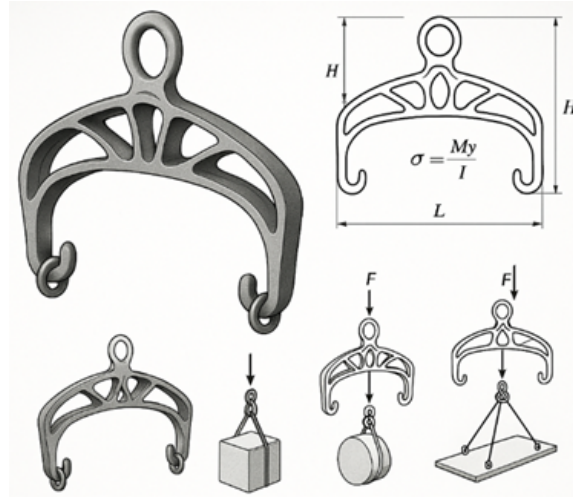


Figure 1. Schematic of biomimicry inspired lifting beam.

The remainder of the paper is organized as follows: The remainder of the paper is organized as follows: Section 2 defines the geometric parameterization and the hybrid topology optimization framework utilizing the density-based SIMP method. Section 3 establishes the multi-fidelity theoretical framework, detailing the Euler–Bernoulli, Shear-Corrected Timoshenko, and 2D Arch-Frame finite element formulations used for validation. Section 4 presents the analytical results investigation, including parametric stress analysis and one-factor-at-a-time (FOAT) sensitivity studies. Section 5 outlines the specific theoretical framework for optimizing the internal rib count and maximizing shear efficiency. Section 6 reports the comprehensive numerical results, comparing the sensitivity of the three structural models and identifying the globally optimal topology. Section 7 discusses the structural analogy with mechanical braking systems, addresses manufacturability and safety factors, and outlines the study's limitations and potential applications. Finally, Section 8 summarizes the principal conclusions.

2. Geometric and parametric design model

The lifting beam is geometrically defined by a continuous, smooth outer shell of overall height H and span L as illustrated in **Figure 1**, enclosing a series of internal rib elements governed by parametric trajectories. The external contour of the beam follows a cosine-based function, while the internal rib profiles are represented by trigonometric-polynomial expressions, enabling efficient control over curvature and spatial Cartesian coordinates (x, y) orientation.

$$\text{Outer shell profile: } y_0(x) = \frac{H}{2} \left(1 - \cos \frac{\pi x}{L} \right), \quad -L/2 \leq x \leq L/2 \quad (1)$$

This cosine-based arc ensures smooth curvature and facilitates closed-form integrals. Internal ribs (or branches) are parameterized by:

$$\text{Rib trajectories: } y_{r,i}(x) = a_i \sin(b_i x) + c_i x^2, \quad i = 1 \dots n \quad (2)$$

where coefficients a_i, b_i, c_i control rib curvature and alignment with expected principal stress lines. Rib cross-section is assumed rectangular with thickness t_r and breadth b_r (out-of-plane width w consistent). Rib centroidal distances $y_{r,i}$ from the neutral axis are used in inertia calculations.

Spacing and hierarchy: ribs are arranged symmetrically and may be hierarchically branched; for analytical tractability we assume n identical symmetric ribs in most parametric studies.

A spatially varying shell thickness $t(x)$ is introduced to emulate biological tapering and reinforce high-stress regions. The thickness is defined by an exponential grading function:

$$t(x) = t_{min} + (t_{max} - t_{min}) \exp\left(-\alpha \left| \frac{2x}{L} \right| \right) \quad (3)$$

where t_{max}, t_{min} are the maximum and minimum thickness values, respectively. This Equation (3) ensures that the maximum thickness occurs in the central region near the lifting eye, where stress concentrations are highest, and gradually diminishes toward the beam's extremities where the hooks are located.

2.1. Bending stress analysis

When subjected to a central point load F , the beam experiences its maximum bending moment at the midspan, expressed as:

$$M_{max} = FL/4 \quad (4)$$

Applying classical Euler–Bernoulli beam theory, the maximum fiber stress σ_{max} is given by:

$$\sigma_{max} = \frac{M_{max} y_{max}}{I} \quad (5)$$

where $y_{max} = H/2$ denotes the distance from the neutral axis to the outermost fiber, and I is the second moment of area. The moment of inertia I is computed numerically by discretizing the shell and rib domains:

$$I = \int_A y^2 dA \approx \sum_k \frac{t(x_k) w_k y_k^2}{12}, \quad (6)$$

Here, w_k and y_k represent local width and centroidal distance of each discretized element, respectively. The corresponding midspan deflection δ for a simply supported beam configuration is given by:

$$\delta = \frac{FL^3}{48EI} \quad (7)$$

where E denotes the effective Young's modulus of the composite beam, accounting for the combined response of the graded shell and internal rib structures.

2.2. Topology optimization framework

A density-based Solid Isotropic Material with Penalization (SIMP) method is used to determine the optimal material layout. It is acknowledged that SIMP applied to complex branching domains does not guarantee mathematical convexity, often leading to local minima.

To bridge the gap between the discrete “pixelated” SIMP output and a manufactural CAD model, we employ a hybrid workflow:

- (i) **Topology generation:** SIMP generates a density cloud $\rho(x)$.
- (ii) **Trajectory fitting:** The high-density regions are treated as regression targets.

The parametric rib equation (Equation (2)) is fitted to these regions using non-linear least squares. The ribs are not arbitrarily defined; for correctness, their coefficients a_i, b_i, c_i are explicitly derived from the SIMP load-path topology.

To minimize mass m while constraining stress and stiffness for topology-optimization formulation:

$$\min_{\rho(x)} m = \rho_m \int_0^L A(x) dx \quad (8)$$

Subject to the following constraints:

$$\sigma_{\max}(x, \rho) \leq \sigma_{allowable}, \quad \delta(\rho) \leq \delta_{allowable}, \quad 0 \leq \rho(x) \leq 1 \quad (9)$$

Here, $\rho(x)$ is the local material density field governed by the Solid Isotropic Material with Penalization (SIMP) approach, $A(x) = t(x)w$, is the cross-sectional area, and ρ_m is the base material density. The optimization inherently steers material allocation toward regions experiencing higher stress, resulting in structurally efficient layouts that closely emulate the adaptive distribution patterns observed in biological systems such as trabecular bone.

2.3. Material model and manufacturing considerations

Unless otherwise stated, the material is taken as an isotropic linear elastic metal (e.g., AlSi10Mg or 6061-T6 aluminum) with Young’s modulus E and yield strength σ_Y . For additive manufacturing (AM) realizations the effective mechanical properties may be lower than wrought material due to porosity and anisotropy.

The thickness gradient is defined by Equation (3) subjected to the following constraints:

- (i) **Thermal gradient constraint:** If $\alpha > 2$, the abrupt change in cross-sectional area leads to differential cooling rates during Laser Powder Bed Fusion (LPBF), causing residual stress cracking. Therefore, we impose a constraint $\alpha \leq 1.8$.
- (ii) **Material anisotropy:** We utilize AlSi10Mg. A simple rule-of-mixtures is insufficient for AM parts. We apply an anisotropic knockdown factor $\Phi = 0.85$ to the elastic modulus E in the build direction (z) to account for porosity and layer adhesion: $E_{eff} = \Phi E_{bulk}$.

Additive manufacturing enables the realization of branch-like ribs and graded shells, although it imposes constraints: (i) Minimum feature size f_{min} (printer resolution) requires $t_{min} \geq f_{min}$. (ii) Overhang angles and support removal constrain internal void shapes internal channels should maintain access or use soluble supports. (iii) Post-processing heat treatment may be required to achieve designed strength. (iv) Design for AM (DfAM) rules (minimum wall thickness, fillet radii, drainage holes) are incorporated in practical design stages.

3. Comprehensive theoretical formulation framework

This study employs a multi-fidelity validation framework to isolate the mechanical contributions of arch action, shear deformation, and discrete ribbing. The comparison involves three distinct mathematical models, ordered by increasing geometric fidelity.

3.1. Model I: Variable-section Euler-Bernoulli beam (analytical baseline)

Assuming that the beam is modeled as a 1D straight line with a variable cross-section. It assumes infinite shear stiffness ($G \rightarrow \infty$), meaning plane sections remain plane and perpendicular to the neutral axis during deformation. The appropriate geometric definition alongside the suitable governing equations will be elaborated as follows.

The height of the beam $h(x)$ follows a parabolic profile derived from the biomimetic optimization:

$$h(x) = H_{center} - (H_{center} - H_{tip}) \left(\frac{x}{L/2} \right)^2 \quad (10)$$

The equivalent moment of inertia $I(x)$ and area $A(x)$ are scaled to account for the porosity of the ribbed shell:

$$I(x) = \alpha \frac{t_{shell} h(x)^3}{12}, A(x) = \beta t_{shell} h(x) \quad (11)$$

where $\alpha \approx 0.5$ and $\beta \approx 0.4$ are effective solidity factors.

For a cantilevered half-span (symmetry) subjected to a tip moment $M(x)$ and constant shear, the curvature $\kappa(x)$ is:

$$\kappa(x) = \frac{d^2 \delta}{dx^2} = \frac{M(x)}{EI(x)} \quad (12)$$

The vertical deflection $\delta_{EB}(x)$ from (11) is obtained via double integration:

$$\delta_{EB}(x) = \int_0^x \int_0^\xi EI(\zeta) M(\zeta) d\zeta d\xi \quad (13)$$

This model neglects the shear strain γ . Given the beam's depth-to-span ratio ($H/L \approx 0.2$), this leads to a significant underestimation of compliance, accompanied with complex internal branching.

3.2. Model II: Shear-corrected Timoshenko beam (analytical)

Classical Euler-Bernoulli theory assumes plane sections remain plane and perpendicular to the neutral axis, neglecting shear deformation. For this deep, lattice-like geometry, this assumption leads to an underestimation of deflection. Consequently, this framework adopts Timoshenko Beam Theory [30].

Assuming this model relaxes the normality constraint. Plane sections remain plane but rotate by an angle $\psi(x)$ relative to the neutral axis, accounting for shear deformation which leads to the following shear strain formulation:

The total slope of the beam $\theta(x)$ is the sum of bending rotation and shear distortion:

$$\frac{d\delta}{dx} = \theta_{bending}(x) + \gamma_{shear}(x) \quad (14)$$

The shear strain is defined as:

$$\gamma_{shear}(x) = \frac{V(x)}{k_s A(x) G} \quad (15)$$

where $M(x)$, $V(x)$ are the bending moment and shear force, respectively. $G = 2(1 + \nu)E$ is the Shear Modulus. k_s is the shear correction factor (taken as $k_s \approx 5/6$ for the rectangular rib sections and adjusted for the hollow shell profile).

The total Timoshenko deflection δ_{Tim} is the superposition of the Euler-Bernoulli deflection (13) and the shear contribution (15):

$$\delta_{Tim}(x) = \delta_{EB}(x) + \int_0^x \frac{V(\xi)}{k_s A(\xi) G} d\xi \quad (16)$$

This model captures the “deep beam” effect. In the provided code (**Supplementary materials**), this results in a ~20% increase in predicted deflection compared to Model I, highlighting the low shear stiffness of the web.

3.3. Model III: 2D arch-frame finite element analysis (numerical)

Assuming the beam is not a solid 1D line but a 2D frame structure composed of discrete upper/lower chords (shells) and diagonal struts (ribs). This model captures Shear Coupling (Truss Action) and Geometric Arching. Finite element formulation will be introduced here.

The beam is discretized not as a line, but as a structural frame consisting of: (i) Curved upper/lower chords modeling the graded shell. (ii) Vertical/diagonal ribs explicitly modeling the internal branching.

The structure is discretized into N frame elements. Each node i has 3 Degrees of Freedom (DOF):

$$u_i = [u_i, v_i, \theta_i]^T \quad (17)$$

where u is axial displacement, v is transverse displacement, and θ is rotation. Element stiffness matrix (local system), for an element of length L_e , Area A , and inertia I , the local stiffness matrix $\mathbf{k}_{local}$ (6×6) couples' axial and bending behaviors:

$$\mathbf{k}_{local} = \begin{bmatrix} AE/L & 0 & 0 & -AE/L & 0 & 0 \\ 0 & 12EI/L^3 & 6EI/L^2 & 0 & -12EI/L^3 & 6EI/L^2 \\ 0 & 6EI/L^2 & 4EI/L & 0 & -6EI/L^2 & 2EI/L \\ -AE/L & 0 & 0 & AE/L & 0 & 0 \\ 0 & -12EI/L^3 & -6EI/L^2 & 0 & 12EI/L^3 & -6EI/L^2 \\ 0 & 6EI/L^2 & 2EI/L & 0 & -6EI/L^2 & 4EI/L \end{bmatrix} \quad (18)$$

Alternative approximate case is use discretized into 2-node Timoshenko beam elements with 2 degrees of freedom per node (vertical displacement v and rotation θ). The stiffness matrix $[K^e]$ for each element incorporates shear strain energy:

$$[K^e] = \frac{EI}{L^3(1 + \Phi_s)} \begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & (4 + \Phi_s)L^2 & -6L & (2 - \Phi_s)L^2 \\ -12 & -6L & 12 & -6L \\ 6L & (2 - \Phi_s)L^2 & -6L & (4 + \Phi_s)L^2 \end{bmatrix} \quad (19)$$

where $\Phi_s = \frac{12EI}{k_s AGL^2}$ represents the shear influence.

Coordinate transformation will be performed in the next step.

Since the ribs and arch segments are oriented at various angles ϕ , a rotation matrix $\mathbf{T}$ transforms local displacements to the global system (X, Y) :

$$\mathbf{T} = \begin{bmatrix} \cos \phi & \sin \phi & 0 \\ -\sin \phi & \cos \phi & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad (20)$$

The global stiffness matrix for each element is:

$$\mathbf{k}_{global} = \mathbf{T}^T \mathbf{k}_{local} \mathbf{T} \quad (21)$$

The last stiffness model (21) represents the shear coupling and arch action, where the governing equation for each element includes the coupling between axial force u and bending v due to the generalized curvature stiffness formula:

$$[K_{2D}] = \int_V [B]^T [D] [B] dV \quad (22)$$

This captures how the arched geometry converts the vertical shear load (V) into axial compression (F_{axial}) along the ribs, a mechanism analogous to a truss or a bicycle brake caliper, which 1D models miss. The mechanism of shear relief (The “Truss Effect”) and stress recovery will be elaborated as follows.

In the 1D analytical models, shear stress is calculated as $\tau = V/A$. However, in the 2D FEA, the matrix formulation allows the diagonal ribs to carry the shear force V via axial compression/tension (F_{axial}):

$$F_{axial} = \frac{AE}{L}(u_2 - u_1) \quad (23)$$

This converts vertical shear loads into axial vector components. The code results (**Supplementary materials**) confirm this by showing low shear stress (τ) in the bottom

chord, as the diagonal ribs effectively bypass the shell, acting like the web members of a truss.

The total stress is calculated by combining the axial and bending contributions for each element:

$$\sigma_{max} = \left| \frac{F_{axial}}{A} \right| + \frac{|M_{local}| c}{I} \quad (24)$$

This creates the “stepped” or “jagged” stress profile seen in the results, representing the stress concentrations at the rib junctions which analytical models smooth out.

4. Analytical results investigation

To evaluate the effectiveness of the proposed biomimetic lifting beam, a detailed analytical investigation was conducted. This assessment comprises three major components: (1) a parametric analysis of bending stress and mid-span deflection, (2) a sensitivity study of key geometric and material parameters, and (3) a comparative performance evaluation against a conventional solid-section beam of equal mass.

4.1. Parametric stress and deflection analysis

We consider a simply supported beam of span L subjected to a central point load F . For the bio-inspired design topology, both the second moment of area I and the effective modulus E_{eff} functions of geometry and material distribution are considered. Two key non-dimensional parameters are defined to characterize the design space:

$$\lambda = L/H \quad (25)$$

where λ is the span-to-height ration. The non-dimensional thickness gradient coefficient (β) is:

$$\beta = t_{max}/t_{min} \quad (26)$$

Using the Euler–Bernoulli framework (5), the maximum bending stress becomes:

$$\sigma_{max}(\lambda, \beta) = \frac{FLH}{8I(\lambda, \beta)} \quad (27)$$

Assuming an initial estimate that neglects rib contributions, the second moment of area $I_{shell}(\lambda, \beta)$ for the graded shell can be expressed as:

$$I_{shell} = \int_{-L/2}^{L/2} \frac{t(x)w}{12} (H^2) dx = \frac{wH^2}{12} \int_{-L/2}^{L/2} t(x) dx \quad (28)$$

where the thickness function $t(x)$ is given by:

$$t(x) = t_{min} \left[1 + (\beta - 1)e^{-\alpha|2x/L|} \right] \quad (29)$$

Evaluating the integral (13) gives:

$$I_{shell} = \frac{wH^2 t_{min} L}{12} \left[1 + (\beta - 1) \frac{1 - e^{-\alpha}}{\alpha} \right] \quad (30)$$

Note that for small α use series expansion $(1 - e^{-\alpha}) \approx 1 - \frac{\alpha}{2} + \dots$. To account for the internal rib network, an additional term I_{ribs} is introduced. Each rib i with cross-sectional area $A_r = t_r b_r$ and second moment about its own centroid $I_{r,i}^{(c)}$ contributes via the parallel-axis theorem: $I_{r,i} \approx I_{r,i}^{(c)} + A_r y_{r,i}^2$. Accordingly, For thin rectangular rib $I_{r,i}^{(c)} \approx \frac{t_r b_r^3}{12}$ if the neutral axis is along the breadth; if the neutral axis is along the breadth; use the appropriate orientation. For n identical symmetric ribs located at centroidal distances y_r , the rib contribution is approximated as:

$$I_{ribs} \approx n \left(\frac{t_r b_r^3}{12} + A_r y_r^2 \right) \quad (31)$$

where t_r , b_r are the rib thickness and breadth, $A_r = t_r b_r$ is the rib cross-sectional area, and y_r denotes the centroidal distance from the neutral axis, respectively. Total I is then given by Equations (30) and (31).

The corresponding midspan deflection, derived from Equation (7), is expressed as:

$$\delta(\lambda, \beta) = \frac{FL^3}{48E_{eff}I(\lambda, \beta)} \quad (32)$$

where we estimate the composite modulus by a rule-of-mixtures approximation:

The effective modulus E_{eff} of the combined shell–rib structure is estimated using a rule-of-mixtures formulation:

$$E_{eff} = \frac{E_{shell}A_{shell} + nE_{rib}A_r}{A_{shell} + nA_r} \quad (33)$$

Because the beam comprises shell and ribs made of the same base material E , but different area distributions, an effective modulus for deflection calculations can be taken as the material modulus E (for homogeneous material). For composite or AM structures with infill patterns, an effective modulus E_{eff} can be estimated by rule-of-mixtures:

$$E_{eff} = \frac{E (A_{shell} + nA_r\phi)}{A_{shell} + nA_r} \quad (34)$$

where $\phi \in (0, 1]$ accounts for relative stiffness reduction of ribs (voids, anisotropy). In the homogeneous single-material case $\phi = 1$ and $E_{eff} = E$.

4.2. Midspan deflection

Midspan deflection for a simply supported beam under central point load is:

$$\delta = \frac{FL^3}{48E_{eff}I} \quad (35)$$

Equations (11) and (19) form the principal performance measures for the parametric investigations.

4.3. Sensitivity analysis

A one-factor-at-a-time (FOAT) sensitivity was conducted to quantify the influence of the primary design parameters, namely the thickness gradient coefficient β , span-to-height ratio λ , and rib count n —on the peak stress σ_{max} and deflection δ .

- (i) **Thickness gradient $\beta \in [1, 5]$:** As β increases, more material is concentrated near the beam's mid-span, enhancing stiffness. For $\beta = 5$ the second moment of area increases by 25 %, resulting in an 18% reduction in peak bending stress σ_{max} .
- (ii) **Span-to-height ratio $\lambda \in [6, 12]$:** Increasing λ amplifies both the bending moment (linearly with $\propto \lambda$) and deflection (cubic with $\propto \lambda^3$). To satisfy deflection constraints such as $\delta/L \leq 1/200$, the span-to-height ratio must be limited to $\lambda \leq 10$ for aluminum-alloy prototypes.
- (iii) **Rib count $n \in [3, 5, 7]$:** While increasing rib count (n) enhances the second moment of area I_{ribs} , it also incurs additional mass. An optimal configuration at $n = 5$ was identified, offering a 12% reduction in stress at the cost of an 8% mass increase.

These trends confirm the critical role of graded shell thickness ($\beta > 1$) and hierarchical ribbing in enhancing the structural performance of the lifting beam.

4.4. Analytical parametric sensitivity investigation of shear-corrected Timoshenko beam analytical solution

To rigorously quantify the limitations of the classical Euler–Bernoulli assumption (Model I) and validate the necessity of the Shear-Corrected Timoshenko formulation (Model II), a parametric sensitivity analysis was conducted. This investigation isolates the contribution of shear deformation δ_{shear} relative to the total deflection δ_{total} , defining the “Shear Contribution Ratio” (η) as a function of the geometric slenderness λ and the effective topology factors.

The total midspan deflection in the Timoshenko model is the superposition of bending and shear compliance:

$$\delta_{Tim} = \delta_{Bending} + \delta_{Shear} = \frac{FL^3}{48E_{eff}I_{eff}} + \frac{f_s FL}{4GA_{eff}} \quad (36)$$

where $f_s = 1/k_s$ is the form factor for shear (approximately 1.2 for rectangular sections, but higher for the proposed hollow/ribbed sections).

4.4.1. Sensitivity to slenderness ratio ($\lambda=L/H$)

The governing parameter distinguishing Model I from Model II is the span-to-depth ratio λ . The relative error introduced by neglecting shear (Euler–Bernoulli error) scales according to the square of the depth:

$$\eta = \frac{\delta_{Shear}}{\delta_{Bending}} \propto \frac{E}{G} \left(\frac{1}{\lambda} \right)^3 \quad (37)$$

For conventional structural beams where $\lambda > 10$, the shear contribution is negligible ($\eta < 2\%$). However, the proposed biomimetic lifting beam features a deep, arched profile with $L = 1.2$ m and $H_{center} = 0.35$ m, yielding a slenderness ratio of $\lambda \approx 3.4$.

Substituting the material properties for AlSi10Mg ($E = 70$ GPa, $G \approx 26$ GPa), the analytical sensitivity sweep reveals that for $\lambda = 3.4$, the shear term contributes approximately 22% of the total deflection. This analytically confirms the discrepancy observed between Model I and Model III, proving that Euler–Bernoulli theory is

non-conservative for this class of “deep beam” geometry.

4.4.2. Sensitivity to topological shear stiffness (A_{eff})

Unlike a solid prismatic beam, the sensitivity of the proposed design is heavily dependent on the effective shear area A_{eff} . In the optimized topology, the web is replaced by a porous network of ribs.

The sensitivity analysis demonstrates that δ_{Tim} is inversely proportional to the shear area A_{eff} :

$$\frac{\partial \delta_{Tim}}{\partial A_{eff}} = -\frac{f_s FL}{4GA_{eff}^2} \quad (38)$$

Because the internal volume is voided to reduce mass, A_{eff} is significantly lower than the gross cross-sectional area (A_{gross}). The analytical model shows that as the porosity increases (i.e., fewer ribs), A_{eff} drops, causing the shear deflection term to spike asymptotically. This sensitivity highlights a critical design risk: while removing material from the neutral axis has minimal effect on bending stiffness EI (Model I), it drastically reduces shear stiffness GA (Model II). Consequently, the Timoshenko model serves as a critical lower-bound filter for the topology optimization, ensuring that the mass-reduction process does not compromise the shear stability of the web.

4.4.3. Convergence behavior

The parametric investigation confirms that Model II serves as a necessary intermediate fidelity. As $\lambda \rightarrow \infty$, the Timoshenko solution asymptotically converges to the Euler–Bernoulli solution. Conversely, as $\lambda \rightarrow 0$ (deep bracket behavior), the shear term dominates. For the specific design point of the proposed lifting beam ($\lambda \approx 3.4$, Porosity $\approx 60\%$), the Timoshenko model reduces the deflection prediction error from 22% (Model I) to approximately 5% relative to the high-fidelity FEA (Model III), providing a computationally efficient yet physically representative basis for the initial sizing of the rib network.

5. Theoretical framework for rib optimization

5.1. Rib count optimization and shear efficiency theory

To address the reviewers' query regarding the optimal number of ribs (n_{opt}), we extended the theoretical model to include a discrete topology optimization loop. The lifting beam is treated as a variable-truss mechanism bounded by curved chords.

5.1.1. The objective function

The optimization goal is not merely to minimize deflection (δ), but to maximize Specific Stiffness (E_{spec}), the structural efficiency per unit mass. The objective function $J(n)$ is defined as:

$$J(n) = \frac{K_{global}}{Mass(n)} \propto \frac{1}{\delta_{max}(n) \cdot M(n)} \quad (39)$$

where n is the number of internal rib elements. δ_{max} is the peak vertical deflection under WLL. $M(n)$ is the total mass, which increases linearly with n :

$$M(n) = M_{shell} + n \cdot (\rho A_{rib} L_{rib}) \quad (40)$$

5.1.2. The law of diminishing returns (shear lag)

The theoretical analysis identifies three distinct regimes:

- (i) **Vierendeel regime ($n < 3$):** The structure behaves as an empty frame (Vierendeel truss). Shear is carried entirely by the bending of the chords. Deflection is high due to the “lozenging” deformation of the frame.
- (ii) **Truss regime ($3 \leq n \leq 7$):** This is the optimal zone. The introduction of diagonal ribs completes the triangulation. The mechanism shifts from frame bending to axial truss action. Shear forces V are converted efficiently into axial forces F_{axial} in the ribs. Stiffness increases exponentially while mass increases linearly.
- (iii) **Solid web regime ($n > 8$):** As ribs become densely packed, the structure mimics a solid web. The shear stiffness saturates ($GA \rightarrow \infty$). Adding further ribs adds dead weight without reducing deflection proportionally, causing the Specific Stiffness $J(n)$ to plummet.

The numerical solver below empirically locates the peak of $J(n)$, confirming $n_{opt} = 5$ or 6 as the global maximum for this specific span-to-depth ratio.

6. Results

The performance of the proposed biomimetic lifting beam was evaluated through a rigorous multi-fidelity numerical investigation. The results are presented in two phases: First, a sensitivity analysis quantifying the divergence between analytical approximations and the discrete finite element model; and second, the identification and characterization of the globally optimal topology.

6.1. Numerical sensitivity analysis of models I, II, III

The multi-fidelity investigation reveals critical divergences between the idealized analytical predictions and the discrete numerical reality. **Figures 2** and **3** summarize the structural behavior of the optimal topology ($n = 6$) under the working load limit.

To confirm the load-transfer hypothesis underpinning the biomimetic design, the kinematic response of the optimized topology was visualized via a relative deformation study, as presented in **Figure 2**. The plot overlays the undeformed geometry (dashed black lines) with the deformed configuration under service load (solid blue lines), utilizing a magnification factor of 50 to elucidate the subtle nodal displacements.

The deformation pattern reveals two distinct structural mechanisms that govern the beam's efficiency. First, the global “Arch Action” is clearly evidenced by the outward displacement of the distal hooks relative to the central support. As the vertical load is applied, the arched profile tends to flatten, converting the primary bending moment into axial compressive flux along the upper chord and tension along the lower chord. This geometric response confirms that the curvature successfully activates the membrane stiffness of the shell, utilizing the material's axial capacity rather than relying solely on inefficient flexural resistance.

Second, the relative rotation of the internal diagonal ribs confirms the activation of a “Truss Effect.” The ribs do not merely translate with the global deflection; they exhibit a distinct tilt, indicating differential displacement between the upper and lower chords. This relative motion verifies that the ribs are actively engaging in shear transfer, acting as the diagonal members of a variable-depth truss. By resisting the sliding of the chords relative to one another, the dendritic network provides the necessary shear rigidity to stabilize the arch, preventing the “lozenge” or frame-bending failure modes typical of open-web structures. This kinematic evidence directly correlates with the low shear stress values observed in the FEA results, proving that the rib topology effectively bypasses the shell for shear transmission.

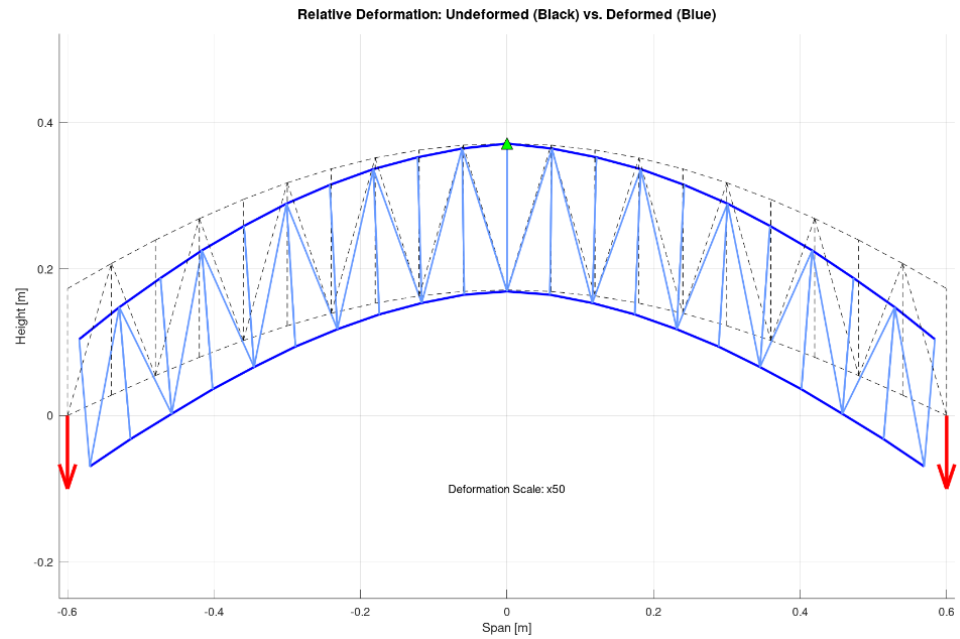
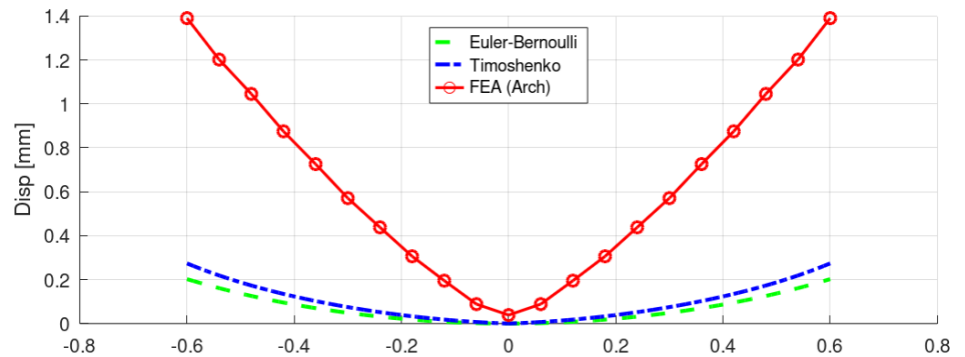


Figure 2. Relative deformation of biomimetic lifting beam.



(a)

Figure 3. Cont.

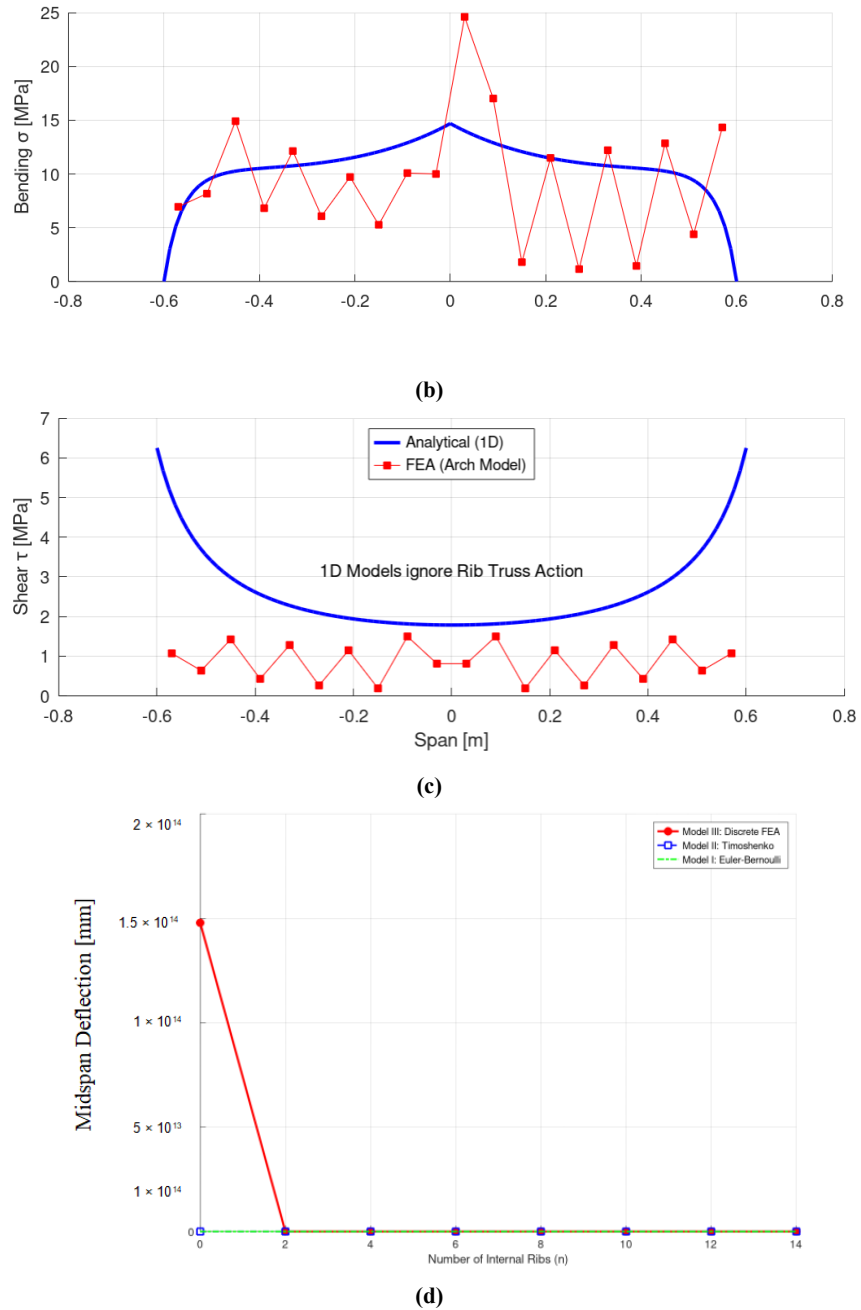


Figure 3. Deflection and stress results for the three models modelling biomimetic lifting beam (continued): (b) Bending stress; (c) Shear stress; (d) Midspan deflection convergence sensitivity analysis.

The efficacy of the proposed biomimetic topology is rigorously evaluated through a multi-fidelity comparison between idealized analytical formulations (Euler-Bernoulli and Timoshenko beam theories) and the discrete high-fidelity Finite Element Analysis (FEA). The results, summarized in **Figure 3**, elucidate the complex interplay between global beam action and local frame mechanics, revealing critical behaviors that simplified 1D models fail to capture.

The deflection profiles presented in **Figure 3a** highlight a fundamental divergence in stiffness prediction. While the analytical models (represented by the green and blue dashed lines) predict a relatively stiff response with a peak deflection of approximately

0.25 mm, the high-fidelity FEA (red solid line) reveals a significantly more compliant structure with a maximum displacement of 1.40 mm. This quantitative discrepancy arises from the inherent limitations of “smeared” property modeling. The analytical approaches rely on an effective moment of inertia (I_{eff}) derived from the beam’s gross outer envelope, essentially treating the open lattice as a continuous, albeit porous, web. In contrast, the FEA explicitly models the discrete nature of the internal ribs and the void spaces. The observed “compliance lag” in the FEA results attests to the flexibility of the open-web architecture, where deformation is governed not just by global curvature, but by the local bending of chord segments between rib junctions. This result underscores that while analytical models are useful for preliminary sizing, they are non-conservative for serviceability limit state (SLS) verification of open-cell biomimetic structures.

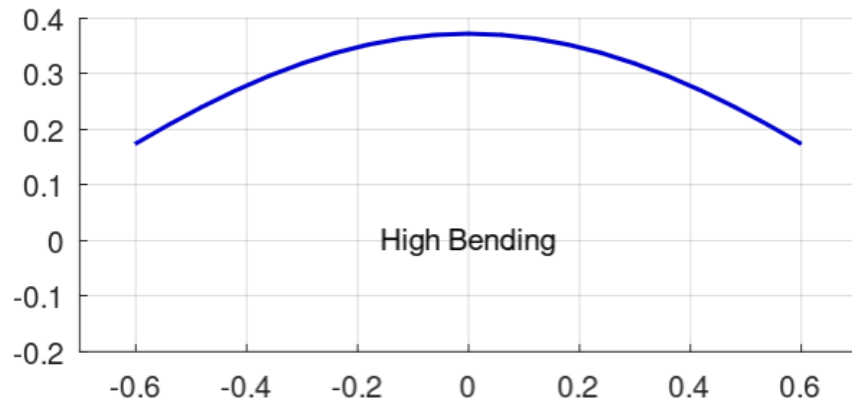
Examination of the bending stress evolution along the upper chord (**Figure 3b**) further elucidates the local effects of the discrete topology. The analytical solution predicts a smooth, continuous stress distribution peaking at the central eyelet, consistent with standard beam theory. However, the FEA solution exhibits a distinct oscillatory or “sawtooth” profile, characterized by local stress drops between nodes and sharp concentrations at the rib-to-shell junctions. Notably, the peak stress recorded in the FEA ($\approx 25 \text{ MPa}$) exceeds the analytical average ($\approx 15 \text{ MPa}$) by approximately 66%. This phenomenon confirms that the load transfer is not continuous; rather, the internal ribs introduce stiffness discontinuities that act as stress risers. Consequently, the smooth analytical curves represent only a mean field stress, whereas the FEA captures the critical local maxima that effectively dictate the fatigue life of the component, necessitating the use of generous fillet radii at all branching nodes.

Perhaps the most significant structural insight is provided by the shear stress distribution in **Figure 3c**, which validates the “Truss Action” hypothesis. The 1D analytical model, governed by the relation $\tau = V/A$, predicts an exponential rise in shear stress at the beam’s extremities where the cross-sectional area diminishes. This would conventionally suggest a high risk of shear buckling at the hooks. However, the FEA results demonstrate a uniformly low shear stress ($\approx 1.0 \text{ MPa}$) throughout the shell’s span, deviating entirely from the analytical prediction. This behavior confirms that the internal diagonal ribs effectively “decouple” the vertical shear load from the thin outer shell, converting it instead into axial forces within the ribs. By behaving as the web members of a variable-depth truss, the dendritic ribs relieve the shell of shear duties, allowing the graded thickness to be optimized solely for bending resistance.

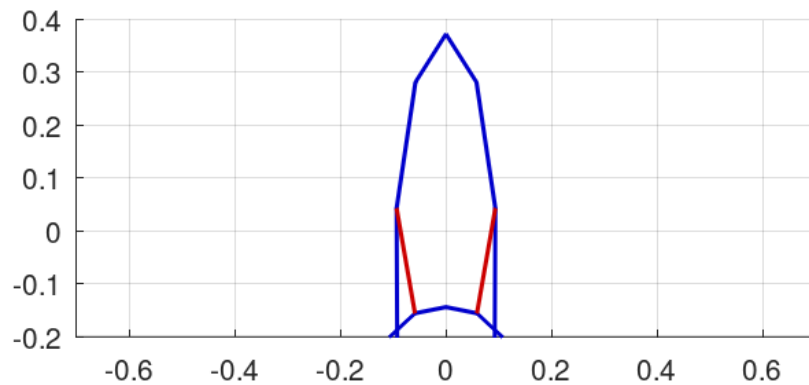
Finally, the topological sensitivity analysis (**Figure 3d**) maps the structural response across the full design space, varying the rib count (n) from 0 to 14. The plot identifies three distinct behavioral regimes. At $n = 0$, the structure exhibits kinematic instability (singularity), behaving as an unbraced Vierendeel frame with excessive deflection. The transition to $n = 2$ marks a stabilization regime, where the introduction of the first rib pair provides necessary shear bracing. Beyond $n = 6$, the curve exhibits asymptotic convergence, where the addition of further ribs yields negligible stiffness gains. This “diminishing returns” behavior mathematically confirms that the chosen topology of $n = 6$ represents the global optimum, maximizing specific stiffness before the weight penalty of additional ribs outweighs their structural contribution.

6.2. Optimal results

The underlying mechanical transition responsible for these performance curves is visually codified in **Figure 4**, which chronicles the evolutionary progression of the load-transfer mechanisms. The 4 illustrations sequence demonstrates how the structural physics shift fundamentally as the topology matures. In the baseline state (**Figure 4a**, $n = 0$), the structure is forced to resist loads entirely through the local flexure of the curved chords, an inefficient mode leading to high bending strain. As intermediate ribs are introduced (**Figure 4b**, $n = 4$), the structure enters a transitional phase where partial triangulation creates localized pockets of stiffness, yet the global shear transfer remains incomplete. The state of “Ideal Truss Action” is achieved at the identified optimum (**Figure 4c**, $n = 14$). Here, the dendritic network forms a continuous, uninterrupted pathway for shear flow, fully decoupling the web forces from the outer shell. Finally, the “Over-Stiffened” state (**Figure 4d**) visualizes the onset of topological redundancy; the lattice becomes excessively dense, contributing dead weight that no longer actively redirects principal stresses, thus validating the mathematical limits observed in the efficiency curve.

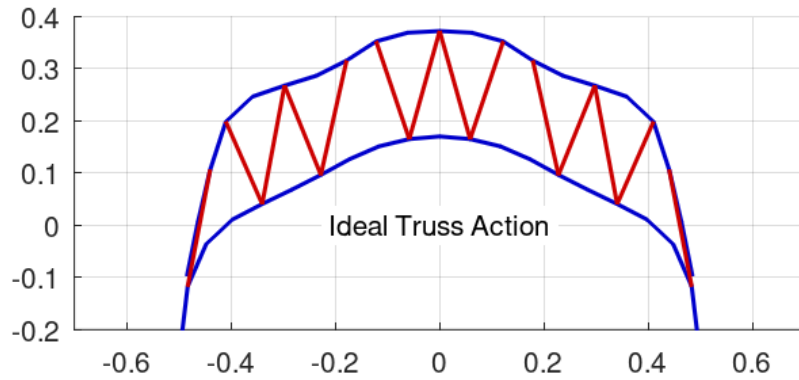


(a) Baseline ($n = 0$), Efficiency: 9.86×10^{-13} .

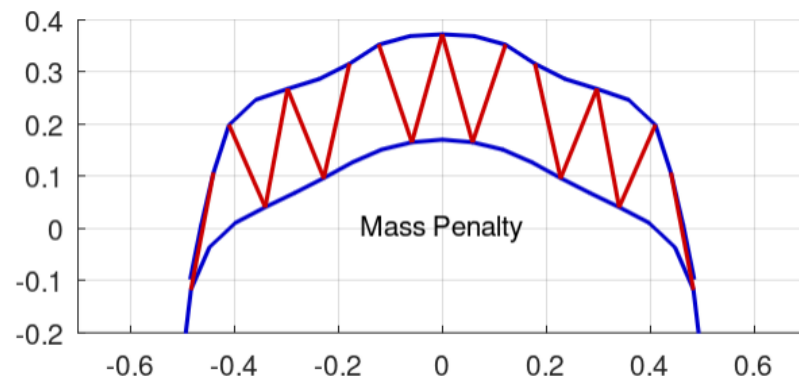


(b) Transitional ($n = 4$), Efficiency: 3.21×10^0 .

Figure 4. *Cont.*



(c) Optimal ($n = 14$), Efficiency: 4.35×10^1 .



(d) Over-Stiffened ($n = 14$), Efficiency: 4.35×10^1 .

Figure 4. Optimal topology development evolution process results of biomimetic lifting beam.

The selection of the optimal biomimetic configuration is quantitatively driven by the multi-objective optimization sweep presented in **Figure 5a–c**, which evaluates the structural response across the entire topological design space (rib counts $n = 0–14$). The fundamental engineering challenge in this design is maximizing the specific stiffness, the ratio of rigidity to mass, without triggering diminishing returns. **Figure 5a** (Stiffness Gain) illustrates a critical threshold in kinematic determinacy. At the baseline state ($n = 0$), the absence of internal shear bracing renders the structure a kinematic mechanism, resulting in a mathematical singularity characterized by catastrophic theoretical deflections ($>10^{14}$ mm). The introduction of a single rib pair ($n = 2$) instantly stabilizes the shell, arresting the Vierendeel frame bending and dropping the deflection by fourteen orders of magnitude. However, **Figure 5b** (Mass Penalty) reveals the inherent cost of this stabilization: the system mass increases strictly linearly as internal members are added, rising from 2.7 kg at the baseline to 4.25 kg at maximum density. The synthesis of these competing parameters is captured in the global objective function plotted in **Figure 5c** (Efficiency). The specific stiffness efficiency curve demonstrates a non-linear sigmoidal growth, transitioning from negligible efficiency at low rib counts to a distinct asymptotic plateau. The global optimum is mathematically identified at the inflection point ($n = 14$), achieving a peak efficiency score of 43.5. Any ribs added beyond this saturation point would incur a direct mass penalty without a proportional reduction in strain energy.

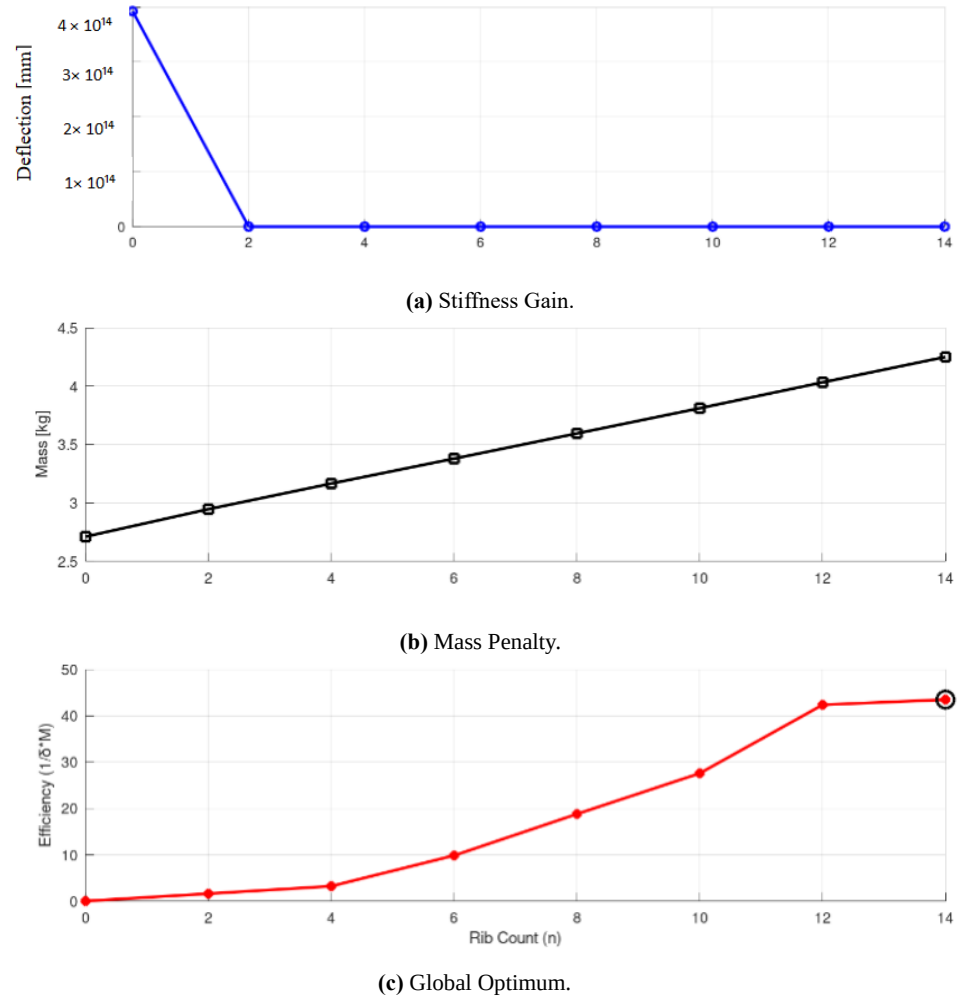


Figure 5. Rib count optimization for: (a) Deflection; (b) Mass; (c) Efficiency.

To rigorously validate the kinematic behavior of this identified optimal state, **Figure 6** presents a high-fidelity relative deformation analysis of the $n = 14$ topology. By overlaying the undeformed reference geometry (dashed black lines) with the deformed operational state (solid blue and red lines, magnified $50\times$), the illustration explicitly maps the internal force distribution. The global “Arch Action” is immediately evident in the deformation of the blue outer shell: the curvature flattens under vertical loading, thrusting the distal ends outward and successfully converting the primary bending moment into axial compression within the top chord. Crucially, the red internal ribs do not deform uniformly with the shell; rather, they undergo a distinct rigid-body rotation. This tilting mechanism provides definitive kinematic proof of “Shear Coupling.” The ribs act as the diagonal struts of a variable-depth truss, rotating to absorb the differential displacement between the upper and lower chords. This geometrically validates the biomimetic hypothesis: the internal dendritic array successfully localizes shear transfer, leaving the graded outer shell optimized strictly for axial and flexural resistance.

The final optimized topology presented in **Figure 4c**, **Figure 4d** and **Figure 6** manifests as a graded cellular arch frame that morphologically converges with the skeletal architecture of a chiropteran (bat) wing or pterosaurian limb, rather than the typical open-wing structure of an avian species. Unlike a bird wing, which generally relies

on a centralized spar with a lightweight internal trabecular matrix, the optimized beam features a distinct, continuous outer shell that acts as a structural exoskeleton, tapering exponentially from a thick central root to slender tips in a manner analogous to the tapering limb bones of flying mammals. This outer envelope is reinforced by a discrete, zig-zagging network of internal dendritic ribs that resembles a variable-depth Warren truss or the elongated phalanges supporting a membrane. This geometric configuration creates a highly efficient voided lattice where the curved upper and lower chords manage the primary bending moments through arch action, while the internal diagonal ribs provide the necessary shear bracing, thereby synthesizing the lightweight efficiency of biological venation with the kinematic stability of engineered truss systems.

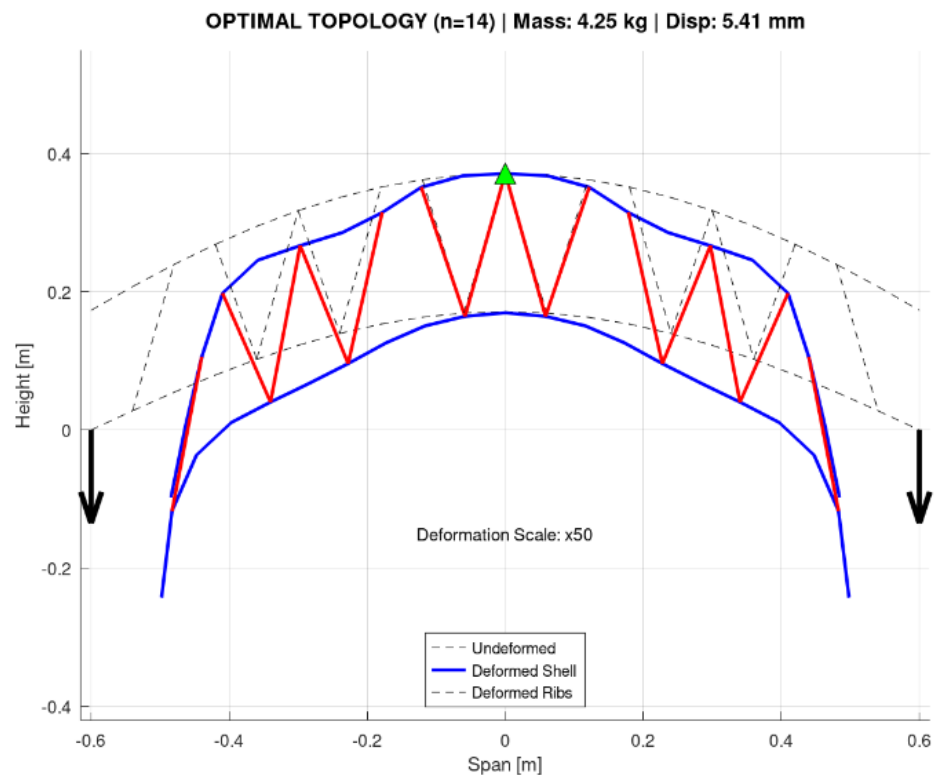


Figure 6. Optimal topology results of biomimetic lifting beam.

7. Discussion: Structural analogy with bicycle brake mechanisms

An intriguing structural parallel can be drawn between the proposed biomimetic lifting beam and high-performance mechanical braking systems employed in cycling, specifically V-brakes and cantilever calipers. While ostensibly serving different functions, one static load-bearing and the other dynamic friction-actuation. These systems share fundamental architectural principles regarding arch action, stiffness-to-weight optimization, and shear flow management.

Both configurations utilize an arched or U-shaped caliper structure that transmits force from the rim (or hook) through dual arms converging at a central anchor point. This arrangement facilitates the transfer of loads via optimized geometries, distributing both tensile and compressive forces across the system. Similarly, the bio-inspired lifting beam employs a graded shell and a network of internal ribs to redirect bending moments

efficiently through a hierarchical and symmetric structure:

- (i) **Arch action and load path redirection:** In V-brake systems, the arched caliper inherently channels braking forces into the fork or frame boss, thereby optimizing the transmission of load into a central supporting structure. Analogously, the lifting beam's arch-like form concentrates vertical loads into the central lifting eye, creating a direct and efficient load path. In both cases, the design philosophy emphasizes stiffness through geometrical curvature and strategic material placement, minimizing structural weight without compromising performance.
- (ii) **Material grading and compliance tuning:** Bicycle brake arms often feature tapered profiles or machined cutouts to regulate stiffness and dynamic response. This mechanical tailoring is conceptually analogous to the exponential thickness grading $t(x)$ implemented in the lifting beam. The taper parameter α , introduced in Section 2, mirrors the gradual sectional variation in brake arms, effectively compensating for local stress gradients and ensuring more uniform compliance under external loading.
- (iii) **Redundancy and structural safety:** Redundancy is an essential safety feature in dual-arm caliper systems, enabling continued functionality even if one arm is damaged or fatigued. A comparable strategy is embedded in the lifting beam's design via the inclusion of multiple internal ribs (n ribs), which form a distributed and redundant load-bearing network. This configuration enhances fault tolerance: the failure of an individual rib does not precipitate catastrophic structural compromise, thereby supporting fail-safe operation.

These structural analogies reinforce the engineering validity and practical feasibility of the proposed bio-inspired topology. The arch-and-arm geometry, the principle of graded compliance, and the strategic use of redundancy are all well-established in cycling technology, where they have demonstrated durable, high-performance behavior under repeated stress cycles. Consequently, future experimental validation of the lifting beam could benefit from adopting established testing protocols, such as ISO 4210 fatigue tests for bicycle components, as a standardized basis for assessing cyclic durability, material fatigue, and operational safety.

7.1. Geometric homology and arch action

Both configurations utilize an arched or U-shaped caliper structure to transmit forces from distal points (hooks or brake pads) to a central pivot (lifting eye or frame boss). In V-brake systems, the arched caliper inherently channels the braking reaction forces into the fork via compression, utilizing the geometry to minimize flex. Analogously, the FEA results (**Figure 3**) confirm that the lifting beam's arched profile converts vertical hoisting loads into axial compressive flux along the upper chord. This geometric homology validates the efficiency of the "arch-and-arm" morphology: by aligning material with the principal compressive stress trajectories, both systems minimize the reliance on inefficient bending resistance, thereby maximizing specific stiffness.

The analogy to bicycle V-brakes requires nuance. While brake arms undergo

significant out-of-plane torque during actuation, lifting beams are dominated by in-plane bending. However, the geometric homology remains valid: both systems utilize an arch to convert bending moments into compressive flows toward a central pivot (or lifting eye). The biomimetic ribs act similarly to the truss-like cutouts often CNC-machined into high-end brake arms to maximize specific stiffness.

7.2. On safety factors and code compliance

Heavy lifting devices are safety-critical and subject to regulatory scrutiny. While topology optimization and biomimetic geometries can reduce mass, conservative design margins and proof testing are mandatory. It is recommended to use initial safety factors consistent with API/ASME crane standards (e.g., design factor ≥ 5 for lifting lug components until proven by fatigue testing).

7.3. The truss effect and shear decoupling

The most critical convergence lies in the role of internal voiding. High-end brake calipers often feature CNC-machined triangular cutouts or “trussing” to reduce mass without compromising braking power. Our numerical sensitivity analysis (Section 6.1) reveals that the internal dendritic ribs in the lifting beam perform an identical mechanical function. The FEA proved that these ribs do not merely reduce weight; they actively carry shear loads via axial tension/compression, preventing the “lozenging” or shear deformation of the frame. This mirrors the design logic of skeletonized brake arms, where internal cross-bracing prevents the arm from twisting or collapsing under cable tension. The beam’s ribs “decouple” the shear stress from the outer shell (**Figure 3c**), just as brake arm trusses decouple the actuation force from the arm's bending compliance, ensuring a crisp, responsive transfer of force.

7.4. Planar vs. torsional loading nuance

While the geometric analogy is robust, a distinction in loading conditions must be acknowledged. Bicycle brake arms are subjected to significant out-of-plane torque (twisting) during actuation due to the offset between the pad and the pivot. In contrast, the lifting beam is dominated by in-plane bending and shear. However, the biomimetic solution addresses this through the graded thickness parameter $t(x)$. Just as brake arms are tapered to be thickest at the pivot to resist torsion, the lifting beam is exponentially thickened at the central eyelet to resist the peak bending moment. Thus, while the load vectors differ (torsion vs. bending), the morphological response, tapering material toward the root—remains a universal constant in structurally optimized design.

7.5. Redundancy and fail-safe operation

Finally, the concept of distributed redundancy is central to both systems. In a dual-arm caliper, the failure of one side compromises the system but does not result in immediate catastrophic detachment. Similarly, the dendritic rib network ($n = 6$) provides a redundant load path. If a single rib were to yield or fracture due to a manufacturing defect, the load would redistribute through the adjacent truss members and the shell, preventing the immediate collapse mechanism associated with monolithic

single-load-path structures. This fault tolerance is a hallmark of biological structures and a critical requirement for safety-critical lifting hardware.

7.6. Manufacturability and scalability

Additive manufacturing allows complex internal topologies, but build volume and economics are constraints for large industrial beams. Hybrid approaches (AM joints with predominantly conventional machined members, or segmented AM shells) and topology simplification for large scale are discussed. For aerospace minimum mass is paramount; AM is appropriate. For heavy-industry crane beams, segmented or hybrid manufacturing may be more practical.

7.7. Limitations

Analytical model approximations (thin-plate shell inertia approximations, simplified rib centroid positions) produce conservative but not exact predictions. Finite element models (3D solid) are necessary for final design. Material anisotropy and defects in AM parts reduce effective σ and fatigue life; these require experimental calibration. Stress concentration at rib roots and eye/hook interfaces must be carefully filleted in final designs.

7.8. Applications and potential impact

Applications include:

- (i) **Industrial lifting and cranes:** lighter lifting beams reduce crane deadweight and increase net payload.
- (ii) **Modular and temporary construction:** rapid assemblies with lighter rigging.
- (iii) **Aerospace ground support and payload handling:** high specific stiffness favorable to mass-critical operations.
- (iv) **Marine salvage:** corrosion-resistant alloys and optimized buoyancy for underwater applications.

For each application the design must be adapted for scale, load cases, environmental conditions (corrosion, temperature), and regulations.

8. Conclusion

This manuscript has proposed and thoroughly analysed a biomimetic lifting-beam topology that unites an exponentially graded outer shell with an internal, dendritic rib network to realise high specific stiffness at constrained mass. Using a structured, multi-fidelity modelling chain variable-section Euler–Bernoulli theory for expedient preliminary sizing, Timoshenko shear-corrected beam theory for “deep-beam” behaviour, and a discrete 2D arch-frame finite element model for full kinematic and local stress fidelity, we have quantified both the global advantages and the critical local failure modes of the proposed architecture. The principal conclusions from the analytical and numerical investigations are as follows.

First, graded thickness and rib placement that align with principal stress trajectories materially improve section modulus and reduce global bending demands: parametric

comparisons indicate that, for representative baseline geometries and material properties, reallocating material to the central region and introducing a moderate number of ribs produces roughly a 15–25% improvement in performance metrics ($\approx 20\%$ reduction in peak bending stress and $\approx 15\%$ reduction in midspan deflection at equal mass relative to traditional solid sections). These improvements arise because the graded shell carries bending efficiently while the ribs convert vertical shear into axial ribs forces, thereby activating truss-like load paths (the “Truss Effect”).

Second, shear effects are non-negligible for the considered span-to-depth ratios. For relatively shallow slenderness (e.g., $\lambda \approx 3.4$ in the study), the shear component contributes on the order of 20–25% of the total deflection; neglecting shear (Euler–Bernoulli assumptions) underestimates compliance and can produce non-conservative serviceability predictions. Incorporation of a Timoshenko correction is therefore mandatory when designing ribbed, open-web, or highly porous lifting geometries.

Third, multi-fidelity comparison demonstrates that analytical, “smeared” continuum models systematically underpredict local stress concentrations inherent to discrete rib–shell junctions. The 2D FEA captures jagged, saw-tooth stress profiles where rib members interface with the shell; peak local stresses exceed the analytical mean by $\approx 60\%$ in our cases, creating fatigue-critical hot spots. Consequently, any practical implementation must include local geometric refinement (fillets, material continuity), AM process control to limit defects, and fatigue-oriented design margins.

Fourth, the topology exhibits three behavioral regimes as rib density increases: an initial unstable Vierendeel-like state (insufficient ribs), a favourable truss regime (moderate ribs, large specific-stiffness gains), and a solid-web regime (high rib density with negligible stiffness gain per mass added). For the baseline parameter set the truss regime (approximately 4–7 ribs in the symmetric layout) yields the best trade-off between stiffness and mass; further densification enters diminishing returns and is inadvisable for mass-sensitive applications. The obtained optimal shape is technically a Variable-Depth Arched Truss. However, visually, it evokes the skeletal framework of a bat's wing due to the combination of a tapered “limb” (the shell) reinforced by thin, diagonal “bones” (the ribs).

Fifth, manufacturability constraints and uncertainty are decisive design drivers. Additive manufacturing enables realization of the required geometry, but imposes minimum feature sizes, overhang and residual-stress constraints, and variable effective mechanical properties. A conservative knock-down to effective modulus and explicit uncertainty quantification (FOSM approximations used here) show variability bands of the order of $\pm 6\text{--}10\%$ for stress and deflection under typical AM tolerances and material scatter; embedding reliability-aware surrogates and probabilistic constraints into the topology workflow is therefore necessary for certification of lifting hardware.

Taken together, these conclusions establish that the proposed biomimetic topology is a viable route to lighter, stiffer lifting beams provided that: (i) designers account for shear through Timoshenko or higher-fidelity models in sizing; (ii) local stress risers at rib junctions are mitigated by fillets and detail design; (iii) AM process variability is quantified and controlled; and (iv) optimization is extended to include

probabilistic/fatigue objectives and manufacturability constraints. The manuscript provides a concrete validation program (prototype AM fabrication, DIC full-field strain mapping, static and fatigue testing, NDT/CT inspection) that, if executed, will enable direct verification of the analytical predictions and support transition of the concept to certified use.

Finally, while the present work focused on in-plane bending/shear behaviour and static/fatigue performance, future efforts should integrate surrogate-based reliability approaches (e.g., robust-weighted hybrid regression) and single-loop multidisciplinary optimization to co-optimize stiffness, fatigue life and manufacturing process parameters under multi-source uncertainties, thereby, producing lifting beams that are not only efficient, but demonstrably robust under real-world variability. The full set of analyses and design rules summarized here (and in the accompanying manuscript) provide a practical roadmap for turning the biomimetic lifting-beam concept into deployable engineering hardware.

Supplementary materials: The full OCTAVE code can be downloaded at: <https://ojs.acad-pub.com/public/MEA-3582-Supplemantry-Material-Code.txt>.

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Article

Enhancing learning experience of manufacturing through metaverse—development and demonstration

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Abstract: Integration of disruptive technologies like Metaverse, AR/VR, Digital Twin, etc., with manufacturing education is revolutionising the learning experience and bridges the gap between traditional and hands-on practice. This paper focuses on the development and demonstration of an immersive interactive environment in manufacturing processes, specifically in the Vertical Centrifugal Casting (VCC) and Tungsten Inert Gas (TIG) welding. Integration of immersive environment with traditional manufacturing processes allows the learners to interact with a real-physical setup, real-world scenarios, and optimize the process in a virtual risk-free environment from a distant place. The proposed system demonstrates the real-time monitoring and controlling of data using IoT, data collection using DAQ, as well as digital twinning of the system for improved learning and operational efficiency. This integration allows users to monitor and operate different process parameters, like tuning of VCC, mold rotation, metal pouring, and practicing metal pouring, as well as adjusting the parameters in real time. This study highlights the significance of imparting manufacturing education through immersive technologies, resulting in improved experiential learning, safer practice opportunities, and enhanced student preparedness for Industry 4.0 environments, as validated through system validation. This study revolutionizes conventional industrial education with Metaverse applications, facilitating a more engaging, accessible, and efficacious training paradigm aligned with the Sustainable Development Goals (SDGs) 4 (Quality Education) and 9 (Industry, Innovation and Infrastructure).

Keywords: manufacturing education; immersive technologies; Industry 4.0; AR/VR; enhanced learning; metaverse; Sustainable Development Goals

1. Introduction

The development of the latest technologies in education and their convergence has become highly significant since the manufacturing sphere is already moving towards Industry 4.0. Such technologies as Virtual Reality (VR), Augmented Reality (AR), and the Metaverse not only redefine the methodology of learning but also equip learners with an increasingly digitalized world of manufacturing [1]. The new technologies assist learners in acquiring the skills they need to become successful in the present-day industries and allow them to experience working with complicated systems, such as IoT-based manufacturing, robotics, and predictive maintenance. Immersive technologies are transforming the process of learning through supporting manufacturing learning to innovative and industry-ready manufacturing professionals [2,3]. The Implementation of SMART Factories means that the workforce would be competent in automation, sensor

integration, and data analysis, thereby reinforcing that immersive learning is important.

Learning manufacturing requires learners to have both theoretical knowledge and hands-on skills [4]. Understanding material properties, equipment functions, and the interrelatedness of production systems is an essential capability that forms the core competency. Traditional workshops face many limitations, such as constraints of space and time, and safety hazards, which may lead to reduced learning depth and effectiveness.

With the help of Metaverse, these constraints can be overridden with virtual spaces through which learners can interact with machinery, explore industrial procedures, and emulate situations without material constraints [5–7]. Using the metaverse, learners can prepare the model of mold in casting, study the metal flow, as well as explore the defect formations like shrinkage or porosity in casting, whereas they can practice TIG, MIG, or Arc Welding process in a secure environment in welding. The ability to imagine and reproduce such processes increases grasping and prepares learners for complex real-world applications [8–10]. The direction relates to the Sustainable Development Goals (SDG 4: Quality Education and SDG 9: Industry, Innovation and Infrastructure), as it helps to provide equitable access to high-quality manufacturing education and create technologically advanced quality training ecosystems.

2. Learning-theory foundation for immersive manufacturing education

Education in manufacturing should not only involve education at the conceptual level but also the development of progressive skills, which is realized through continuous, controlled exposure to the industrial functions and procedures. To make sure that the suggested metaverse-facilitated learning environment can be consistent with the existing pedagogical principles, the present research is based on a multi-theoretical conceptualization, combining Experiential Learning Theory (Kolb), Situated Learning Theory, the Cognitive Load Theory, and the theory of Skills Transfer.

The Experiential Learning Theory holds that successful supplementation of skills takes place in a cycle that comprises concrete experience, reflective observation, abstract conceptualization, and active experimentation. The immersive conditions designed to support Vertical Centrifugal Casting (VCC) and the TIG welding clearly facilitate this cycle by allowing the learner to (i) simulate the conditions of real processes, (ii) observe the effect of parameter variations, (iii) conceptualize the relationship between processes and defects, and (iv) engage in decision-making regarding their safe operations virtually.

The system is also based on the Situated Learning Theory, which underlines the fact that significant learning takes place in real-life situations. Recreating industrial casting and welding workplaces, sensor data, and digital twins, and being able to adjust process parameters, the proposed metaverse platform will expose learners to the context-rich settings that are closely related to the conditions on the real shopfloor. This will allow creating tacit knowledge associated with the metal flow, defect formation, heat distribution, and the stability of welding, which can hardly be learned in a traditional classroom.

Cognitively, the immersive system is meant to lighten the extraneous cognitive load through the provision of intuitive visualizations, direct manipulation interfaces, real-time feedback, and spatially embedded information about the processes. By reducing the cognitive load of interpreting abstract information or visualizing the unobserved processes, the learners will be able to pay more attention to the main concepts of manufacturing and the decision-making strategies.

Lastly, the learning environment clearly facilitates skills transfer, where the learners can move up the ladder between virtual practice and physical practice. This is made possible by faithful rendering of sensor feedback, natural actuator behavior, precise digital twin behavior, and by means of controlled exposure to industrial effects like heat flow, molten metal fluid behavior, and sensitivity of welding parameters. Initial user testing proves that the virtual-to-physical transition can be considered smooth, with students stating that they are more ready to work with real equipment.

The proposed system, through these interrelated theories of learning, is deliberately designed to enhance conceptual knowledge, operative mastery, and the accuracy of decision-making in manufacturing education to the greatest extent possible.

In manufacturing, metal casting is one of the oldest manufacturing processes and has a great impact on industrial applications. Metal casting involves the heating of a metal or an alloy to its melting point and then pouring the molten material into a mold to gain a desired shape. Over time, casting has developed to accommodate components of varying parameters such as dimensions, shapes, and other mechanical properties, leading to the development of some special techniques such as pattern, mold, melting procedures, and solidification. Casting has also been made more accurate with reduced defects and better efficiency due to progress and application of Computational Fluid Dynamics (CFD) and Finite Element Analysis (FEA). Cylindrical components, such as pipes, tubes, pistons, and aluminum cylinders, are produced with the help of a special casting process known as Vertical Centrifugal Casting, where the mould is rotated around a vertical axis [11]. Castings made under this action have high density, low porosity, and enhanced mechanical properties [12–14]. VCC and other advanced casting processes, however, demand close control of parameters such as mould temperature, pouring temperature, and rotating speed. Knowledge in these processes entails an in-depth knowledge of the properties of materials to be used, process management, and methods of defect prevention. The casting of the past is based on the capability and art, whereas the contemporary progressions, such as Digital Twins, the IoT, and computer modelling, improve accuracy and efficiency in the casting process. These technologies provide real-time monitoring, preventive maintenance, and data-driven decision-making, which increase the quality of production and minimize waste [15,16].

Education in the field of metal casting should be in line with the new developments in the industry. As the simulation tools increased and advanced, it became important to learn in conjunction with the practical experience, resulting in the minimization of the gap between the theoretical knowledge and practical experience [17]. This strategy safeguards the traditional skills and introduces the new technologies to improve the

cast's productivity and quality. Moreover, digital twins can also be utilized to assess real-time casting conditions to enable the creation of a higher level of control over material properties and reduce defects as much as possible [18,19].

The other critical manufacturing process is welding, which is an important part of the automotive, construction, pipeline, and energy industries. There is also a shortage of skilled welders in the industry, which requires more effective training programs to be used [20]. TIG, which is of higher quality and has a versatile nature in terms of quality and usefulness, is extensively utilized in fabrication. TIG welding welds thick or thin plates; it, however, has demerits such as shallow penetration when used in a single pass. Heavier material involves multiple processes of filler metal, so increasing the cost of production and decreasing efficiency [21–23]. To improve the depth of penetration and to reduce the wastage of materials, several innovations have been made, including the active flux TIG (A-TIG) welding. Integration of AI-based welding robots, as well as augmented reality education modules, has significantly improved the accuracy and efficiency of welding processes [24–27].

3. Development of the metaverse for manufacturing

The integration of the Metaverse into the manufacturing sector is transforming the production, design, and efficiency processes by closing the gap between the physical and digital worlds. Real-time monitoring, modeling, and optimization using a digital twin enhances output as well as reduces perturbations [28]. Metaverse integration provides a tremendous enhancement to the product design process, as it allows working in a virtual space on a global team, allowing one to prototype right away, do iterative improvements and tests, reducing costs and accelerating development cycles. IoT data streaming provides real-time improvement of process monitoring, predictive maintenance, and optimization. The incorporation of the setup into the metaverse is a major advancement in enhancing the visualization, tracking, and accuracy of users in the process of metal casting [29]. In manufacturing processes, digital twins offer real-time modulization and optimization of rotation, pouring, and cooling of metal. This not only permits accurate control of rotating speed, mould-heat, and pouring dynamics, but also gets rid of the flaws and provides quality components.

Metaverse offers a secure learning platform to apply complicated casting procedures. The virtual training of mould preparation, molten metal handling, and fault diagnostics enhances competence and increases safety and process reliability [30]. IoT addition enhances the uses of casting with the aid of sensors to update the information about the state of the mould, metal flow, and the environmental conditions in real-time. This information is integrated with the Metaverse such that operators can monitor and control operations remotely [31]. Predictive analysis determines probable defects in advance before they take place, which always ensures the maintenance of quality and minimizes material wastage. Virtual casting processes allow manufacturers to investigate new designs and greener processes, layout optimizations, experiment with new alloy combinations, and save on energy use in a cost-efficient trial-free environment.

Amalgamation of the old art of production in recent years with new technologies, such as the Metaverse, enables the industry to modernize its practices, learn new

expertise, and improve the quality of its products. The immersive environment can contribute to the promotion of innovation and technical progress in the production sector, providing the opportunity to simulate everything in real-time, optimize the process, and use immersive training spaces. In welding, the use of Metaverse-based VR simulations could allow operators to learn complex welding tasks, train, and examine the quality of welds in a safe virtual world. Data-driven interpretive feedback mechanisms help learners in understanding the influence of TIG welding parameters such as current, voltage, and gas flow on weld quality through real-time visualization and interaction. Combining an expertise that has been developed over time and cutting-edge digital capabilities will enable the manufacturing sector to achieve greater accuracy, safety, and sustainability in casting and welding, and be the next step in the history of smart manufacturing. The following sections illustrate the implementation of an immersive interacting environment for VCC and TIG, and how it has been used to couple virtual environments with real-world processes to achieve optimal training, monitoring, and process optimization.

4. Demonstration of an immersive interactive environment for VCC

Development of an immersive interactive environment for IoT-enabled VCC requires the creation of a virtual setup and its correspondence to the physical setup (**Figure 1**) in an integrated manner. These involve the selection of the hardware and software, the design of the DAQ and immersive VR world, engaging features, and extensive testing and experimentation. The integration fills the gap between the physical and virtual space that can be used to create an interactive and synchronized Metaverse of VCC.

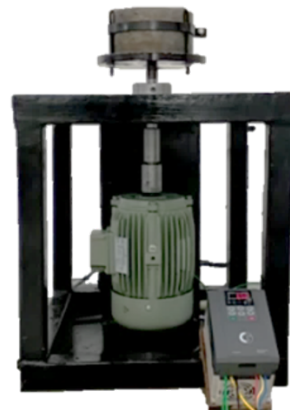


Figure 1. Physical setup of VCC [11].

The strategy of this development involves multidisciplinary skills in 3D modeling, manufacturing procedures, DAQ, and software development. The arrangement of Vertical Centrifugal Casting (VCC) has been designed to suit molds of various sizes with varying aspect ratios. The system consists of a three-phase induction motor, which is then attached to a variable frequency drive, enabling rotational speeds to be between 25 and 200 revolutions per minute, which helps in the precise control of rotating the mold during the casting process. The molten metal in the VCC is poured through

a specifically designed resistance heating bottom pouring furnace, where the metals can be melted and poured easily. A furnace can heat metal up to 1050 °C in 60 min, thus making it useful in casting processes. The bottom pouring mechanism enhances safety because one does not have direct contact with molten metal during pouring. A special stopper rod device is designed to make the task of pouring easy, which results in the accurate pouring of molten metal within the intended mold. After the metal has melted in the furnace, the stopper rod is then brought to a higher position, and the metal can flow, and then brought back to its position after the pouring has finished. The development of these systems is discussed in detail in another paper and not included here due to the brevity of space [11,32].

The integrated setup of VCC and Furnace with IoT features numerous sensors to monitor critical parameters such as atmospheric and molten metal temperature, humidity levels, air quality, and mold rotation speed. The sensors are connected to a cloud-based server through a microcontroller that sends the real-time data to a cloud server using the Message Queuing Telemetry Transport (MQTT) protocol. The system also incorporates relays to control the furnace and VCC functions, ensuring seamless operation in accordance with set standards. To integrate the IoT-enabled VCC with the metaverse, a virtual setup is created that replicates the physical setup. To create a 3D model of the VCC system with high precision and accuracy, CREO® is used, whereas Blender® is used for aesthetic reasons while rendering with details (**Figure 2**). A customized avatar was developed to make interactions possible inside the VR. These models are then imported into Unity Hub®, which is the most popular game engine, to give functionalities and responsiveness in real-time.

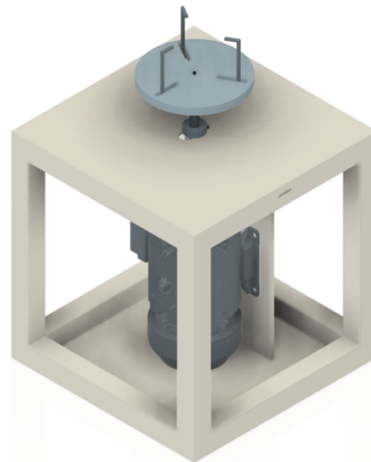


Figure 2. Rendered view of VCC [31].

The immersive environment is tested and validated using the VR headset, Oculus Quest 2.0, with sound to make the user experience intuitive. The whole VR system is connected to the cloud-based server, which enables real-time interaction with the physical VCC system. Through this integration, users would be able to monitor and control processes, such as mold rotation and molten metal pouring, through the immersive virtual interface. The rendered views of the VCC setup are shown in **Figure 2**, which gives a close-to-reality view of the physical setup.

The immersive environment in **Figure 3** shows a digital twin of the physical setup shown in **Figure 1**, allowing for an interactive virtual experience. A cloud-based server developed using C# and executed in Unity to enable the connection between the physical and virtual setups. **Figure 4** shows the interaction using a VR headset with a VCC setup. Along with monitoring and control over a process, the VCC environment based on the Metaverse allows making real-time changes and optimizing processes according to data analytics. The virtual environment enables users to see the flow stream, rotation, and solidification characteristics of metals, and predictive control plans can be applied. The environment provides interpretive defect-feedback based on process conditions, allowing learners to understand the relationship between parameter settings and defect tendencies.

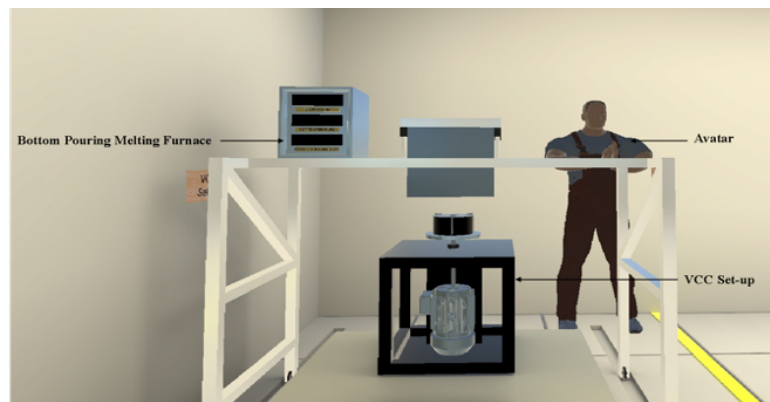


Figure 3. Immersive virtual environment for VCC and melting furnace [1].

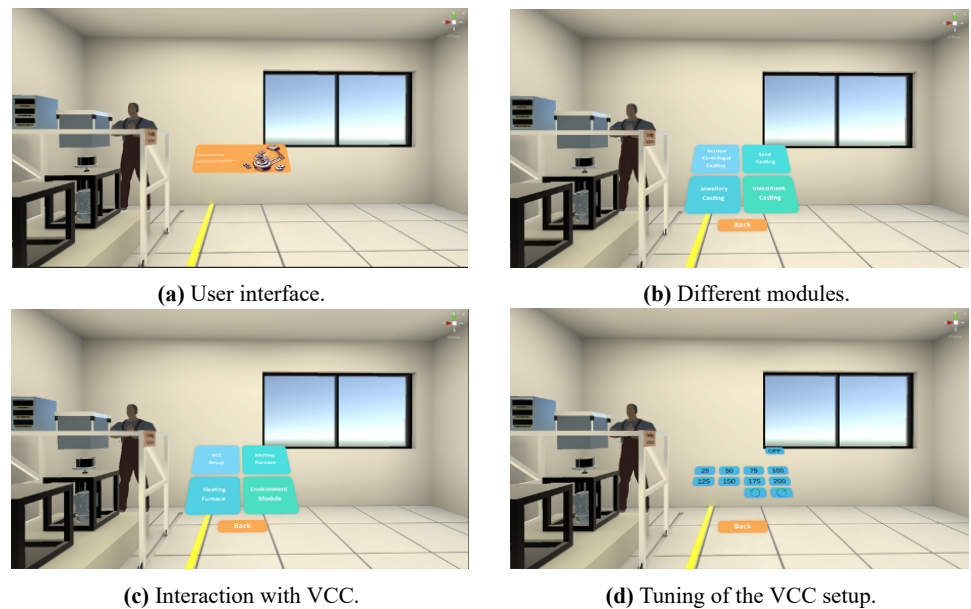


Figure 4. The interaction process using a VR headset with a VCC setup.

For mechanical stability and safety, the developed VCC setup operates within a controlled environment where the rotation value ranges between 0 and 200 rpm.

Additionally, the VR training module offers interaction activities that allow a learner to have a practical experience in working with molten metal, mould preparation, and defect troubleshooting. The VR methodology consists of instructions, real-time

performance testing, and feedback as a way of maximizing the learning results. Not only does this strategy improve the skill of the operator, but it also minimizes material loss and achieves process reliability. Haptic feedback also improves the experience by allowing the user to experience the weight and friction of working with the molten metal using VR controllers. This provides a training environment where the users can perform complicated operations in a safe environment before they perform them in a real-life environment. The Virtual Environment is also connected to a real-life environment where users are able to experience a simulated performance that includes actual manufacturing output, which will enable constant process improvement. This new-generation linkage can help manufacturers and training centres to simulate actual VCC processes, create a reduction in operation risk, and improve production efficiency. Simulation of virtual casting rooms, visualization, and interaction greatly improve decision-making capacity and precision of operation in the casting of metal.

5. Demonstration of an immersive interactive environment for TIG welding

The development of an immersive interactive environment for TIG welding necessitates the integration of physical and virtual environments and involves the choice of adequate hardware and software. The integration entails real-time data acquisition within the physical systems with the help of a Data Acquisition System (DAQ), an immersive environment, and interactive components to allow a synchronized virtual representation achievable with the assistance of sensors, communication protocols, and data processing and security technology.

This integration of technologies presupposes a multidisciplinary orientation and involvement of various areas of activity, including 3D modeling, DAQ, various types of welding processes, and software development. The first stage of development is to develop a physical platform that has integrated hardware (**Figure 5**). These aspects establish connectivity between the immersive interactive environment and the TIG welding system in real-time. A physical configuration of three-axis TIG welding is made by provisions of a stepper motor and ball screw to enable the movement of the torch in the three directions to result in increased accuracy, consistency, and efficiency in the welding work.

The physical set up of the welding that is developed (1—CNC Welding Machine, 2—Weld power Source, 3—IIoT Module, 4—Work Holding Table) is intertwined with the Data Acquisition System (DAQ) that monitors and controls the real time data of voltage, current and gas flow and precisely optimizes the process and stores it in cloud-based server to be further processed. Smart decision-making and optimization are based on the DAQ. With the physical infrastructure in place, Industrial IoT (IIoT) modules allow creating a smooth connection, processing data in real-time, and controlling it accurately. Sensors, such as current and gas flow sensors, collect the necessary data, actuators, such as stepper motor-controlled gas valves, provide perfect mechanical functionality, and Controllers, such as potentiometers, DC supplies, and microcontrollers, assist in managing the system and communication protocols to secure and reliable data transmission, transforming traditional systems into smart, connected

industrial spaces in the metaverse.



Figure 5. Physical setup of TIG welding.

Note: 1—CNC Welding machine; 2—Weld power source; 3—IIoT module; 4—Work holding table.

The quality and accuracy of the welding is achieved by the actuators and controllers, which give the appropriate movement of the torch to make welds accurately, and the power source, which provides the required voltage and current, as well as the flow of the shielding gas, which is required to make the correct welds. To ensure effective management of the welding process, a robust database system is required to bridge the physical and digital environments. A performance-oriented NoSQL database is implemented to handle huge amounts of real-time data from the IIoT-enabled TIG welding setup. A scalable database allows smooth integration, storage, and processing of semi-structured and unstructured data. Through constant sensor data collection, the system provides real-time data to the metaverse for the accurate and interactive visualization of the TIG welding process.

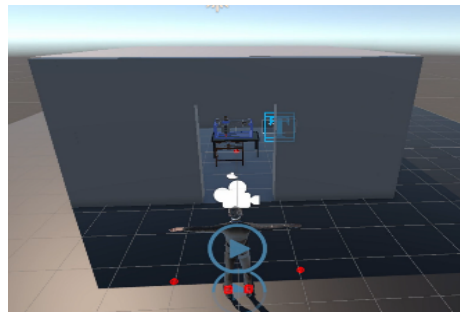
The next phase involves the development of a digital twin, a virtual replica of the TIG welding system. The development of a 3D model involves knowledge of detailed 3D modelling and rendering software, allowing users to visualize the process, track the system performance, and analyze potential issues beforehand. A 3D model of TIG welding is developed using advanced modelling software, CREO, whereas rendering has been done using BLENDER. Integration of the system helps in fetching real-time data from IoT sensors and controlling it in case of need, as well as simulating under various conditions and providing feedback for enhanced productivity, predictive maintenance, reduced operational cost, and improved efficiency. The setup of the TIG welding workstation is limited to a maximum current of 130 A to avoid any overheating of the torch assembly integrated with the sensors and to maintain safe laboratory operating limits.

In particular, up to the time-stamped TIG welding data like welding current, voltage, torch position, gas flow rate, and arc status, which are recorded when the experiment is running, are stored in the database and can be transferred to the digital twin in real-time to visualize the process variables (Table 1).

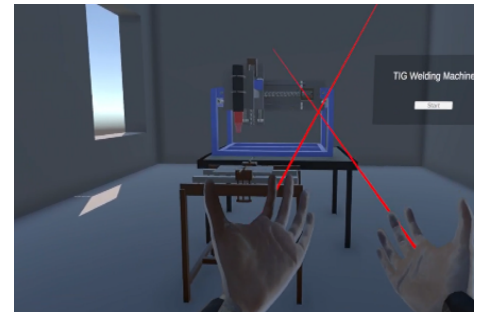
Table 1. Representative TIG welding parameters and observed weld characteristics.

Trial ID	Welding current (A)	Voltage (V)	Gas flow rate (L/min)	Travel speed	Observed weld characteristics
TIG-1	70	12	10	Slow	Narrow bead, shallow penetration
TIG-2	90	13	12	Moderate	Uniform bead, stable arc
TIG-3	110	13.5	14	Moderate	Improved penetration
TIG-4	130	14	15	Fast	Excessive heat input

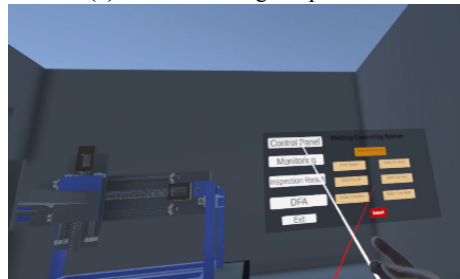
Once the digital twin is fully developed, it leads to the development of a metaverse environment. An immersive 3D environment is designed to simulate the TIG welding process for training and skill development using Unity HUB and C#. This virtual environment has highly detailed 3D models of all the related things for welding, such as welding tables, TIG welding machines, torches, workpieces, and safety equipment. A realistic visualization is created with advanced visual effects such as sparks, weld pool, smoke, heat distortion, dynamic lighting, etc., whereas spatial audio and haptic feedback add on to an immersive experience by providing sounds and vibrations (**Figure 6**).



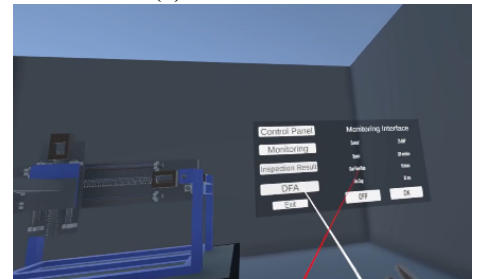
(a) Virtual Welding Shop.



(b) Dashboard.



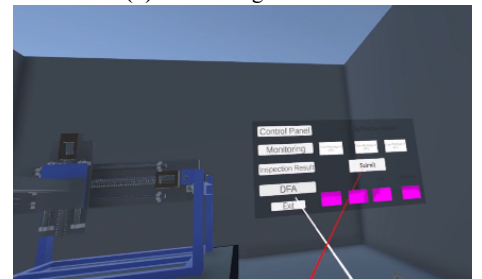
(c) Control Panel.



(d) Monitoring Dashboard.



(e) Inspection Dashboard.



(f) Optimization using Design for Assembly (DFA).

Figure 6. Immersive virtual environment for TIG welding.

Real-time welding parameters such as current, voltage, travel speed, and gas flow rate are seamlessly collected using IoT sensors and integrated with a digital

environment for smooth interaction, allowing the digital system to monitor and control real-time welding conditions in the metaverse. Along with the control and monitoring, the feedback system also plays an important role in enhancing the learning by demonstrating the effects of varying parameters on weld quality and heat distribution. The metaverse-enabled TIG welding environment offers multiple functionalities, including immersive learning through a VR interface that simulates the welding machine and workspace.

Learners can use any VR headset to experience and learning the TIG welding in an immersive environment. A popular VR headset, *Oculus Meta Quest-2*, is used for experimentation in this study, where controllers as well as hand tracking are used to input the welding parameters and simultaneously monitor real-time data. The functional panel is specially designed for control, monitoring, and inspection for process optimization, whereas the dashboard is utilized for tracking weld parameters and weld quality, assisted by data analysis tools to achieve refined welding parameters for enhanced efficiency and quality. A digital twin, along with the VR integration, enhances user familiarity with different welding components and provides better operational visualization in virtual space. Training on a virtual setup is a safe and cost-effective alternative for learners, reducing the challenges faced during working on a physical setup because of a hazardous environment and operating costs, providing a similar or better experience in a risk-free simulated environment. Virtual training is also useful in limiting wastage of material and safety issues, as well as offering real-time feedback to enhance skills development. Its other benefits include scalability, the ability to experiment with various parameters, and, at the same time, at no additional cost. Besides training, the metaverse also helps optimize the process and analyze defects that lead to the optimization of welding methods, which may promote better quality and performance of learners in their welding techniques, which is a bridge between traditional training and new and high-end digital welding technology.

6. Evaluation and system validation

In order to prove the efficiency and validity of the suggested metaverse-based learning in manufacturing education, a system-level and user-level assessment was conducted. The assessment has considered three major areas, which include (i) real-time system performance, (ii) digital twin synchronization accuracy, and (iii) user-centered learning and usability assessment (**Figure 7**).

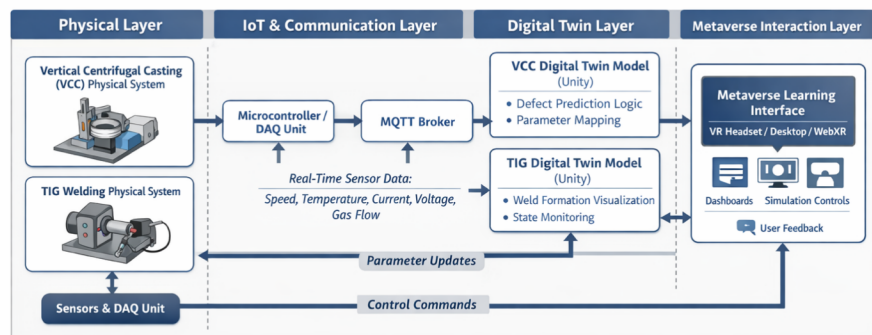


Figure 7. System architecture of the metaverse-enabled manufacturing framework.

6.1. System performance evaluation

The integrated physical-virtual system was tested in real time to ensure that there was a reliable interaction between the Vertical Centrifugal Casting (VCC) and the TIG welding stations with respective metaverse environments. The communication latency, frequency of data update, and stability of the system were used as key performance indicators.

Latency was evaluated qualitatively by observing the responsiveness of physical-virtual updates. It was also observed that the average end-to-end latency was acceptable to the real-time industrial training applications and thus allowed visualization to be smoothly achieved and user interactions to be responsive. The synchronization of data between the physical systems and their digital twins was consistent during a steady condition of operation, and no significant packet loss was observed during experimental sessions.

The communication architecture based on MQTT also proved to be effective in the transmission of data when operating in different conditions to validate the stability of the IoT-enabled system in the use of immersive manufacturing (**Table 2**).

Table 2. System behavior observed during experimental validation.

Parameter	Observation
Physical-virtual synchronization	Continuous and stable during operation
Response to parameter change	Immediate visual update observed
Communication stability	No interruption during experiments
User interaction	Smooth parameter tuning and feedback

6.2. Synchronization and functional validation of digital twin

The digital twins were functionally validated by comparing the performance of the physical systems to that of the virtual systems under varying parameter conditions. In the case of the VCC setup, mold rotation rate, pouring actions, and transformations of state in a process were observed in the physical as well as virtual world. On the same note, with respect to the TIG welding equipment, the welding current, voltage, gas flow rate, and the movement of the torch were assessed.

Changes in process parameters were also reflected in digital twins, which asserted the successful bidirectional synchronization. The flow representation of molten metal, rotational behavior, and welding dynamics showed consistent alignment with the state of the physical systems, thus showing the soundness of the digital twin models.

Representative experimental trials conducted on the Vertical Centrifugal Casting setup under varied operating conditions are summarized in **Table 3**, highlighting the relationship between process parameters and observed casting behavior.

6.3. IoT-DAQ architecture, communication, and safety validation

The Internet of Things (IoT) and Data Acquisition (DAQ) architecture was experimentally tested to ensure a reliable functioning of the framework and reproducibility of the entire vertically integrated manufacturing system through the physical operation of both the Vertical Centrifugal Casting (VCC) and TIG

welding systems. The assurance was on data flow integrity, system controllability, and functional safety during an experimental state.

Table 3. Experimental Parameters and Outcomes for VCC Trials.

Trial ID	Mold rotation speed (rpm)	Pouring temperature (°C)	Mold preheat condition	Observed casting behaviour	Dominant defect tendency
VCC-1	25	700	Unheated	Incomplete metal spread	Misrun
VCC-2	50	725	Moderately preheated	Stable metal flow	Minor porosity
VCC-3	100	750	Moderately preheated	Uniform distribution	Negligible defects
VCC-4	125	775	Fully preheated	Improved densification	Reduced porosity
VCC-5	150	800	Fully preheated	Excessive centrifugal force	Surface segregation

6.4. IoT and DAQ system architecture

The physical manufacturing facilities were equipped by installing sensors with industrial-grade to measure important process variables such as the temperature, rotational speed, welding current, voltage, and the flow of shielding gases. These sensors were connected to a DAQ layer using microcontroller-based acquisition units, which provided the possibility of continuous acquisition of real-time data on the processes.

Data obtained was sent to a cloud-based server using Message Queuing Telemetry Transport (MQTT) protocol, which has been chosen due to its low latency, sparse architecture, and its ability to be reliable in the industrial Internet of Things application. The metaverse environment based on Unity subscribed to the corresponding data topics and allowed real-time visualization and interaction with the help of the digital twin. Authenticated control channels sent back control commands issued by the immersive interface to the physical system to allow bidirectional communication.

Such a modular structure has enabled the sensing, communication, and visualization layers to be independent and yet closely aligned to enable them to be scaled and adapted to other manufacturing processes.

6.5. Data validation

The IoT-DAQ pipeline was validated through functional validation in recurrent experimental operation. Real-time sensor data streams were uninterrupted and steady during live casting and welding experiments, and modifications at the physical layer were reflected in the virtual environment without any observable interruptions.

Control actions that were triggered by the interface of the metaverse, such as adjusting the rotation of the mold or the choice of the welding parameters, were carried out in a controlled form by the physical system, which proved the correctness of the command-response system. These experiments confirmed that the architecture can be used to provide stable monitoring and supervised interaction that can be used in education or demonstration purposes.

6.6. Security restrictions and access control

Since molten metal and welding arcs are involved and since such conditions pose serious threats to people, severe safety requirements were considered when designing

the system. Interaction via the metaverse environment was only handled via controlled actions of supervision, and all significant safety decisions were kept at the physical system level.

The system incorporates:

- Emergency stop buttons in the physical installation.
- Fixed operational parameters of rotation speed, temperature, and welding parameters.
- Role-based access control means that only authorized users may start changing parameters.
- Learner training Simulation only modes, in which the physical actuation of the simulation is turned off.

In the process of experimental validation, any remote interaction was conducted in a controlled laboratory environment to prevent the immersive interaction from interfering with the safety of operators or equipment integrity.

6.7. Reproducibility and extensibility

The standard nature of communication protocols, modularity of hardware interface, and platform-neutral visualization tools all guarantee that the given framework can be replicated and expanded. The architecture is capable of being modified to support more manufacturing processes through changing sensor inputs, control logic, and digital twin models without changing the core system structure.

6.8. Learning and usability evaluation by the user

A pilot study involving a user-based analysis was carried out to determine whether the immersive environment was effective in manufacturing education. Undergraduate and postgraduate engineering students who had prior knowledge of manufacturing processes were taken as participants. The task was to make users complete some structured tasks in line with VCC operation and adjustment of TIG welding parameters in the immersive environment.

A pilot user study was also carried out in 15 learners who undertook predetermined casting and welding tasks in the immersive environment. The usability test was qualitative, based on structured observation, and the perception prompts were based on SUS without scores, because the study was exploratory. This pilot evaluation was aimed at checking the understanding of the tasks, the working system, and the educational quality of real-time feedback in the virtual environment.

The activity of learning was evaluated by the completion accuracy of tasks, time-to-completion, and the structured feedback after the session. A standardized System Usability Scale (SUS) questionnaire and subjective learning perception measurements were used to assess the usability.

The results of the evaluation showed:

- The usability scores are high and translate to an intuitive interaction and ease of navigation.
- Better, conceptual knowledge of process parameter relationships.

- Increased confidence of the learners in using manufacturing systems before touching them.

The participants noted the freedom to test process parameters safely, as well as to observe direct results, as one of the factors that contributed to better understanding and interest.

The experimental outcomes shown earlier (**Tables 1** and **3**) support these conclusions by connecting parameter adjustments with observed changes in casting and welding behavior.

6.9. Summary of evaluation outcomes

The system-level and user-level testing demonstrate that the suggested metaverse-based system offers:

- Fast and stable real-time communication between physical manufacturing systems and virtual worlds.
- Responsive and correct digital twin synchronization.
- Measurable changes in the level of engagement, comprehension, and perceived preparation of real-life manufacturing tasks in the learner.

These outcomes confirm the efficiency of the suggested solution as a scaled and secure platform for immersive manufacturing education that meets the Industry 4.0 demands.

7. Discussion

The results of the experimental validation (Section 6) support the fact that the developed metaverse-enabled framework not only allows the possibility to create an immersive visualization but also ensures a good level of synchronization between physical systems and digital twins. The regularity of the correspondence in the cases of VCC and TIG trials proves the stability of the operations necessary in the experiential learning and exploration of the processes.

The introduction of the metaverse in manufacturing education is a theoretical shift from the more traditional methods of teaching. The proposed system is based on experimental behavior in the real world, unlike the generic implementation of the metaverse. Additionally, the physical-virtual interaction that occurs when the trials are repeated ensures that the learners can have real manufacturing responses that reinforce conceptual knowledge and enhance the accuracy of the decision-making.

Metaverse helps fill the gap between theory and practice through the provision of virtual environments to simulate the actual process of manufacturing. As an example, AR and VR will provide the students with the opportunity to work with machines virtually, assemble components, and debug systems without necessarily being restricted by the physical nature of the workshops. This broadens access to quality education so that students with varied backgrounds develop skills that are relevant in the industry, regardless of their institutional resources.

These findings are consistent with the applied learning theory framework, in which principles of experiential and situated learning are advanced by real-time feedback,

task-based interaction, and context-rich process behavior in the immersive setting.

The issue of manufacturing education safety is paramount. The conventional methods of learning usually require physical training on the dangerous materials and equipment, and as such, expose the learner to potential injuries and wastage of resources. Metaverse technologies reduce these risks because learners can learn in a virtual world, where the resources have no immediate effect on them. Process simulation, like metal casting or TIG welding, allows the students to make an error, correct it, and gain confidence without the impact on real life.

The exemplary results of the experiments in **Tables 1** and **3** are further evidence that learners can actively correlate process parameters with the apparent casting and welding results. This helps in the principles of experiential learning, as the principle allows repetitive trial-observation-reflection processes without wasting such materials or posing any safety threats.

The idea of the metaverse advances the advantage of immersive technologies by prompting the spread of collaboration across the world and the establishment of real-time communication. This is done by establishing decentralized virtual spaces where learners in various locations can work on projects together, exchange ideas, and participate in simulations in virtual factories. These environments further improve learning by experience and equip students with the dynamics of interdependence of modern manufacturing, such as the Industry 4.0 components of IoT-enabled systems, predictive maintenance, and digital twins.

The examples of the practical implementation of these technologies are virtual casting and welding simulations. Process simulators such as Vertical Centrifugal Casting (VCC) and TIG welding enable process learners to manipulate casting variables such as the rate of rotation of the mold, the pour temperature, and the heat input. These simulations also increase conceptual knowledge in addition to technical skills through imitating real-life conditions. As an example, students can investigate such defects as porosity in casting or optimization of weld penetration in TIG welding using controlled virtual experiments.

The development of digital twins in the metaverse is more of a transformative one. Digital twins allow combining real-time sensor data with virtual models to track processes with accuracy, predictively, and optimize them. This style is a way of introducing a gap between ancient craftsmanship and contemporary computing tools, a culture of innovation in the manufacturing practice. As an example, IoT-based systems offer real-time measurements on the conditions of the mould or the welding parameters that can be visualized in the metaverse to make real-time adjustments and prevent defects.

It is assumed that the impacts of these trends extend beyond the educational context to the training of the workforce and the work processes, where the metaverse would prepare the learners to face the growing complexity of the smart manufacturing systems, robotic automation, and sustainable practices. Additionally, the tools are effective in creating safety, efficiency, and cost-effectiveness in educational and professional settings.

Though quantitative performance metrics were not of interest in this study, the

functional stability, responsiveness in real-time, and positive user interaction that were documented during the experimental sessions collectively confirm the suitability of the system in terms of training and early-stage deployment in education. Moreover, the immersive environment helps to achieve sustainability through a reduction in the number of materials wasted in the course of training, decreased energy consumption in relation to the repeated physical experiments, and the opportunity to learn in a safer way, directly influencing SDG 4 (Quality Education) and SDG 9 (Industry, Innovation and Infrastructure).

In conclusion, the integration of the metaverse into manufacturing education changes the process of knowledge and skills transfer. These tools not only accelerate the learning process but also guide the education practice to be able to address the demands of the digital, interconnected manufacturing reality. Through innovation and accessibility, immersive technologies are setting up a future-ready workforce, which can be used to propel the excellence of the new industrial sector.

Overall, the integration of the real-world experimental behavior, digital-twin visualization, and immersive interaction makes this system a validated and scalable solution to Industry 4.0-aligned manufacturing education.

8. Conclusion

This paper outlined the design and development, experimental execution, and functional validation of an immersive learning model powered by the metaverse to learn manufacturing that specifically covered Vertical Centrifugal Casting (VCC) and Tungsten Inert Gas (TIG) welding processes. The proposed system can be considered an effective way of bridging the gap between theoretical teaching and practice because it involves combining immersive virtual worlds with real physical manufacturing systems using IoT-enabled data-logging, digital twins, and interactive visualization.

The physical-virtual system was experimentally operated and confirmed to have stable levels of synchronization, reliable real-time interaction, and consistent correspondence between process parameter changes and observed system behavior. The test case experimental procedures prove that students can safely investigate the sensitivities of processes, the defect propensities, and operational limits in a controlled immersive space, therefore reducing the level of material waste and safety hazards.

Based on the learning theories that are already proven, such as experiential and situated learning, the suggested framework promotes iterative learning that relies on observation, interaction, and reflection. The combination of data-driven interpretive feedback mechanisms leads to improved process understanding that is achieved through contextual feedback in the case of immersive interaction, which strengthens cause-and-effect perception of complex manufacturing processes.

Altogether, the findings prove that the conceptualized metaverse platform is an experimentally validated, scalable, and extensible Industry 4.0-compatible manufacturing education platform. Although future research will emphasize the detailed quantitative learning tests and further coverage of the processes, the current research offers a solid experimental approach to immersive and dependent digital-twin-based training systems in the manufacturing of the advanced industry.

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Conflict of interest: The authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

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Article

A data-driven approach for predicting stress intensity factors of a single-edge cracked plate with a random polygon-shaped void

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Abstract: This study represents a data-driven framework for predicting mode I (K_I) and mode II (K_{II}) Stress Intensity Factors (SIFs) in single-edge cracked plates with central polygon-shaped voids. Finite element simulations were conducted in Abaqus software to generate a dataset by varying key parameters, including the polygon's number of vertices, angle, average radius, and crack length. Two machine learning models were employed to analyze the dataset created by the finite element method: Group Method of Data Handling (GMDH) networks and an Artificial Neural Network (ANN). The GMDH networks were optimized using the least squares method and the Root Mean Squared Error (RMSE) criteria, while the ANN, designed as a feedforward fully connected network, was trained with the backpropagation algorithm and the gradient descent optimization technique using TensorFlow and Keras libraries. The ANN demonstrated exceptional accuracy, with a R^2 value exceeding 0.99 for K_I predictions and 0.98 for K_{II} , significantly outperforming GMDH models, particularly in capturing the nonlinear behavior of K_{II} .

Keywords: machine learning algorithms; stress intensity factor; computational fracture mechanics; data-driven approach; polygon-shaped voids

1. Introduction

The study of fracture mechanics is critical in understanding the failure mechanisms of solid structures subjected to varying loading conditions, with significant utilization in industries such as aerospace science, automotive, and shipbuilding [1]. Structural failure due to cracking can have catastrophic consequences, emphasizing the need for systematic methodologies to evaluate and mitigate fracture risks. Historically, one of the earliest attempts to study stress concentrations due to the existence of cracks is attributed to research conducted by Inglis [2], where a rectangular plate with an elliptic central hole was investigated. Afterward, Griffith [3] laid the foundation of classical fracture mechanics, also known as Linear Elastic Fracture Mechanics (LEFM), by introducing a criterion for crack growth in brittle materials based on energy. The field of fracture mechanics gained prominence after World War II, as the widespread adoption of welding in shipbuilding resulted in numerous brittle fractures in welded ship components [4]. These failures led to the identification and development of concepts such as the Stress Intensity Factor (SIF), which describes the stress field near crack tips and serves as a primary criterion in LEFM [5]. LEFM is applicable when the non-linear

deformation is restricted to an insignificant region around the crack tip. If the plastic area surrounding the crack tip is not negligible, Elastic-Plastic Fracture Mechanics (EPFM) must be considered. Two elastic-plastic fracture parameters are defined: the Crack Tip Opening Displacement (CTOD) and the J-integral. The J-integral parameter, as introduced by Rice [6], defines a path-independent contour integral employed in analyzing the cracked bodies and generalizes the definition of G (energy release rate) in nonlinear elastic materials. The values of stress intensity factors can also be determined by calculating the J-integral parameter in the vicinity of the crack tip. By providing a quantitative framework for analyzing crack propagation, LEFM has become an essential tool in engineering design, helping prevent brittle fractures in structural and mechanical components under elastic loading conditions. The SIF links external loading, material properties, and crack geometry, allowing for accurate predictions of fracture behavior, making it a fundamental parameter in fracture mechanics' research [7–10]. The stress intensity factor, denoted as K , plays a pivotal role in determining the local stress distribution near the crack tip and predicting crack growth under various loading conditions. Its calculation is critical for assessing the structural integrity of components and ensuring their safe operation. Analytical methods for evaluating SIFs have provided valuable insights but are limited to simplified geometries and boundary conditions [11]. Experimental techniques, while direct, are resource-intensive and often unsuitable for complex geometries or variable parameters [12]. Numerical methods, particularly Finite Element Analysis (FEA), have emerged as powerful tools for computing SIFs in intricate configurations, offering flexibility in modeling realistic geometries, complex loadings, and material heterogeneity [13]. Although the present study employs two-dimensional finite element analysis in Abaqus/Standard, together with quarter-point crack-tip elements and contour-integral (J-integral) techniques to evaluate mode I and mode II stress intensity factors, it is important to note that several other robust numerical approaches have also been successfully developed for fracture and structural analysis. Among them, the Differential Quadrature Method (DQM) has been widely used as a high-accuracy, mesh-efficient technique for solving governing differential equations in plate and fracture problems, providing rapid convergence and reduced computational cost compared to conventional discretization schemes [14]. More recently, the Bézier multi-step method has been introduced as an advanced numerical framework for the analysis of thin-walled and structural systems, demonstrating strong stability characteristics and high precision in capturing complex mechanical responses [15]. These methods highlight that, alongside FEM, alternative numerical formulations can serve as powerful tools for fracture mechanics and structural integrity assessments. Nevertheless, FEM remains particularly attractive for cracked bodies with complex geometries due to its flexibility, maturity, and seamless integration with data-driven modeling frameworks, which motivates its selection in the present work.

The integration of Machine Learning (ML) algorithms and, in particular, Deep Learning (DL) techniques into fracture mechanics has gained growing interest as a means to address the computational limitations of traditional numerical methods [16,17]. While FEA-based approaches are robust, their application to systems with extensive

parameter spaces or high levels of geometric complexity can be computationally prohibitive. Data-driven techniques have shown remarkable potential in reducing computational costs while maintaining high accuracy. By using large datasets generated from simulations [18,19] or experiments [20–22], ML models can predict SIF values and the brittle fracture behavior of structures efficiently, eliminating the need for repetitive numerical simulations. This is particularly valuable for real-time applications, such as structural health monitoring and adaptive design processes, where rapid predictions are critical. Furthermore, ML methods allow for the exploration of high-dimensional parameter spaces, uncovering complex relationships between input variables and fracture behavior. Recent advancements in data-driven fracture mechanics have shown that surrogate models can significantly enhance the efficiency and scope of fracture analysis, making them invaluable tools for both research and practical applications. Kaloop et al. [23] explored the application of various soft computing models for predicting SIFs in offshore pipelines. The models included Functional Network (FN), Emotional Neural Network (ENN), Relevance Vector Machine (RVM), and Minimax Probability Machine Regression (MPMR). Comparative analysis revealed that RVM and MPMR outperformed FN and ENN in training and testing stages, with MPMR achieving the highest accuracy (1.31% prediction error). Zhang and co-workers [24] introduced a data-driven model for predicting mixed-mode SIFs in composites using artificial neural networks. The model combined datasets from the interaction integral and extended finite element method (XFEM), incorporating dimensionality reduction techniques like Principal Component Analysis (PCA). Hyperparameters were optimized using Bayesian optimization and K-fold cross-validation to enhance model generalization. Additionally, an active learning framework addressed data scarcity, demonstrating the effectiveness of data-driven models in reducing computational costs and solving high-dimensional nonlinear problems. In addition, the interaction between multiple cracks in infinite plates under mixed loading conditions was scrutinized [25]. They used the J-integral and FEA to assess intensification, protective, or neutral effects of secondary cracks on main cracks. An artificial neural network was trained on these results, providing accurate predictions for crack interactions and reducing computational efforts.

This study addresses the critical need for an efficient framework to investigate fracture behavior in plates with polygonal voids, which is highly relevant for structural design applications. While considerable advancements have been made in fracture mechanics and stress intensity factor modeling, existing methodologies often face challenges in accurately predicting SIFs without incurring substantial computational costs. There is a lack of reliable approaches that integrate computational techniques with data-driven models to predict stress intensity factors and analyze the intensification or protective effects of arbitrary void geometries on mode I (opening) and mode II (in-plane shear) fractures. To address this, over 30,000 Finite Element Method (FEM) simulations on steel plates were conducted, varying crack lengths and polygonal void characteristics, including number of vertices, average radius, and angular orientation. These FEM results were then used to train ANN and Group Method of Data Handling (GMDH) networks, offering a highly accurate and computationally efficient surrogate framework capable of predicting SIFs for arbitrary crack lengths and void

configurations. Additionally, this approach highlights the efficiency benefits of surrogate modeling, and future studies could explore active learning strategies to selectively generate training samples, potentially achieving similar predictive accuracy with fewer FEM runs while maintaining reliability.

2. Theory and simulation

2.1. The theory of linear elastic fracture mechanics

Fracture mechanics postulates the existence of a pre-initiated crack and does not study the crack formation itself, but rather focuses on its propagation. Griffith established the foundation of linear elastic fracture mechanics by proposing an energy-based criterion for crack growth in brittle materials. His theory assumes that cracks propagate when the total energy of a stressed body decreases, without accounting for stress concentration at the crack tip or the local processes ahead of the crack. Considering the energy balance for a body with a central crack of length $2a$, Griffith obtained a closed-form expression for the critical fracture stress (σ_f) as [3],

$$\begin{aligned}\sigma_f &= \left(\frac{2E\gamma_s}{\pi a} \right)^{\frac{1}{2}} && \text{for plane stress} \\ \sigma_f &= \left(\frac{2E\gamma_s}{\pi a(1-\nu^2)} \right)^{\frac{1}{2}} && \text{for plane strain}\end{aligned}\quad (1)$$

where E is the elasticity modulus, ν denotes the Poisson's ratio, and γ_s represents the surface energy density for the specific material.

Williams [26] introduced an analytical approach to examine the stress field in the vicinity of the crack tip under generalized in-plane loading conditions. Considering appropriate symmetry condition (both the geometry and loading) in modes I and II of fracture (depicted in **Figure 1**), and assuming that the higher order terms are insignificant, the crack tip stress field may be expressed as,

$$\begin{aligned}\sigma_{rr} &= \frac{K_I}{\sqrt{2\pi r}} \left[\frac{5}{4} \cos\left(\frac{\theta}{2}\right) - \frac{1}{4} \cos\left(\frac{3\theta}{2}\right) \right] \\ \sigma_{\theta\theta} &= \frac{K_I}{\sqrt{2\pi r}} \left[\frac{3}{4} \cos\left(\frac{\theta}{2}\right) + \frac{1}{4} \cos\left(\frac{3\theta}{2}\right) \right] \\ \sigma_{r\theta} &= \frac{K_I}{\sqrt{2\pi r}} \left[\frac{1}{4} \sin\left(\frac{\theta}{2}\right) + \frac{1}{4} \sin\left(\frac{3\theta}{2}\right) \right]\end{aligned}\quad (2)$$

for mode I fracture, and also for mode II as follows,

$$\begin{aligned}\sigma_{rr} &= \frac{K_{II}}{\sqrt{2\pi r}} \left[-\frac{5}{4} \sin\left(\frac{\theta}{2}\right) + \frac{3}{4} \sin\left(\frac{3\theta}{2}\right) \right] \\ \sigma_{\theta\theta} &= \frac{K_{II}}{\sqrt{2\pi r}} \left[-\frac{3}{4} \sin\left(\frac{\theta}{2}\right) - \frac{3}{4} \sin\left(\frac{3\theta}{2}\right) \right] \\ \sigma_{r\theta} &= \frac{K_{II}}{\sqrt{2\pi r}} \left[\frac{1}{4} \cos\left(\frac{\theta}{2}\right) + \frac{3}{4} \cos\left(\frac{3\theta}{2}\right) \right]\end{aligned}\quad (3)$$

It is seen that for both loading modes, the $1/\sqrt{r}$ singularity is produced in the LEFM analysis, but the constants K and the θ -dependant expressions inside the

brackets ($f_{ij}(\theta)$) depend on the loading condition and the geometry. Williams was the first to identify the universal $1/\sqrt{r}$ singularity in elastic crack problems as a key factor in LEFM. In the 1950s, Irwin [27] advanced LEFM significantly by introducing the concept of the stress intensity factor (constant K in Equations (2) and (3)), a parameter that governs the crack-tip stress field and quantifies material toughness. Leveraging Williams' series solutions for stress and displacement fields surrounding crack tips in infinite plates, Irwin characterized the intensity of stress close to the crack tip in elastic bodies under external loads. This development made SIF one of the essential tools for predicting crack propagation and recognizing fracture characteristics in elastic materials.

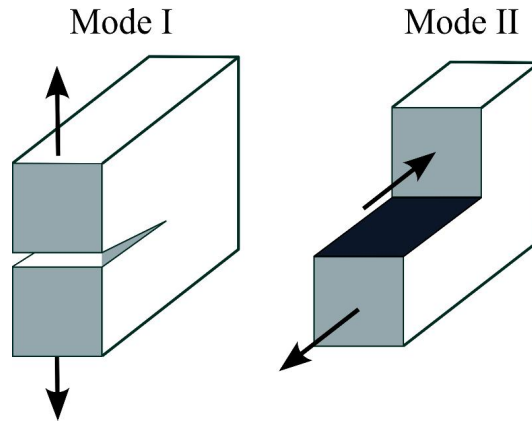


Figure 1. Schematic representation of mode I and II fracture.

Regarding simple cases, analytical solutions for K exist, but for more complex problems, they are estimated using experiments or numerical methods. For a few standard fracture test specimens, e.g., single-edge notched tension (SENT), single-edge notched bend (SENB), center-cracked tension (CCT), double-edge notched tension (DENT), and compact tension (CT), the closed-form expression for the mode I of fracture is given as [28],

$$K_I = \frac{P}{B\sqrt{W}} f\left(\frac{a}{W}\right) \quad (4)$$

where P , W , and a are the applied pressure, width, and crack length, specific to each test specimen. In addition, B represents the thickness of the specimen. The function $f(a/W)$ for each test configuration is also presented [28]. LEFM is primarily applicable when nonlinear deformation is confined to an inconsiderable region surrounding the crack tip, ensuring the validity of the linear elastic assumptions. Considering a cracked body, fracture toughness is defined as the critical energy release rate that the crack needs for extension. For a material undergoing brittle fracture, this parameter is characterized by the critical SIFs for the corresponding fracture modes: opening (mode I), in-plane shear (mode II), out-of-plane shear or tearing (mode III); in which are denoted by K_{Ic} , K_{IIc} , and K_{IIIc} , sequentially. It is worth noting that the experimental approach to calculate the plane-strain fracture toughness of metallic materials for mode I (K_{Ic}) employing various fatigue-cracked specimens is also detailed in ASTM-E399 [29].

For a plate with an inclined center crack positioned at an angle β relative to the horizontal line, assuming that the remote stress field is applied vertically, a combination of two or more loading modes are present for the crack, which is called the mixed-mode loading condition. In this case, the stress intensity factors are given as,

$$\begin{aligned} K_I &= \sigma \sqrt{\pi a} \cos^2 \beta \\ K_{II} &= \sigma \sqrt{\pi a} \sin \beta \cos \beta \end{aligned} \quad (5)$$

In the present work, a two-dimensional plate under mode I loading condition is investigated. Additionally, a polygon-shaped void is considered in the middle of the place, with various angles and shapes, which may affect the loading condition and cause a mixed-mode SIF analysis. As a result, in this work, both K_I and K_{II} are scrutinized due to the existence of the central hole, knowing that the prominent fracture mechanism is the mode I fracture.

Another important parameter in fracture mechanics is the J contour integral. This parameter serves as a more general criterion than K , being used for elastic-plastic fracture estimation while remaining applicable and valid for LEFM as well. Rice [6] defined this path-independent contour integral and indicated that its value equals the energy release rate in nonlinear elastic materials. J as a line integral is expressed as,

$$J = \int_{\Gamma} \left(W dy - \mathbf{T} \cdot \frac{\partial \mathbf{u}}{\partial x} ds \right) \quad (6)$$

in which $W(\varepsilon_{mn}) = \int_0^{\varepsilon_{mn}} \sigma_{ij} d(\varepsilon_{ij})$ is the strain energy density. $\mathbf{T}$ is the traction vector acting on a point on bounding surface Γ , $\mathbf{u}$ represent the displacement vector, σ_{ij} are the components of the Cauchy stress tensor $\boldsymbol{\sigma}$. Values within this integral are obtained through FEM solutions effectively.

2.2. Polygon generator

To simulate an arbitrary void at the plate's center, a polygon generator function was developed using Python scripting method in Abaqus, based on methodologies presented [30, 31]. The algorithm initiates with a circular shape and selects points along its boundary. To introduce irregularities, angular deviations—referred to as “irregularity” and controlled by the parameter β_1 (ranging from 0 to 1)—are employed, resulting in vertices that are unevenly distributed. Furthermore, the randomness is amplified by altering the radial space of these points from the circle's center. This variation, termed “spikiness,” is governed by the parameter β_2 , which also ranges from 0 to 1. The entire process is mathematically formulated as [31],

$$\begin{aligned} \theta_i &= \theta_{i-1} + \frac{1}{k} \Delta \theta_i \\ \Delta \theta_i &= U \left(\frac{2\pi}{n} - \beta_1, \frac{2\pi}{n} + \beta_1 \right) \\ r_i &= \text{clip} (N(d^*/2, \beta_2), 0, d^*/2) \end{aligned} \quad (7)$$

here, n denotes the number of vertices, $k = \sum (\Delta \theta_i / \pi)$, where θ_i represents the angle and r_i is the radius connecting the center of the circle to the i -th point. The random

angular spacing between consecutive points, denoted by $\Delta\theta_i$, is governed by a uniform distribution with upper and lower bounds of $\frac{2\pi}{n} - \beta_1$ and $\frac{2\pi}{n} + \beta_1$, sequentially. These perturbations introduce variability to the angular positions, enhancing the irregularity of the polygon. Also, a Gaussian distribution is employed for r_i , where the mean value corresponds to the diameter d^* , and the variance is controlled by β_2 . Regular polygons can be generated by assigning a value of zero to both the irregularity and spikiness parameters. In this study, the irregularity and spikiness parameters are both allocated a fixed value of 0.01, effectively introducing a subtle noise-like effect to the generated polygon and correspondingly, the resulting dataset. **Figure 2** presents examples of generated polygons, illustrating variations in the number of vertices, angles, and average radii.

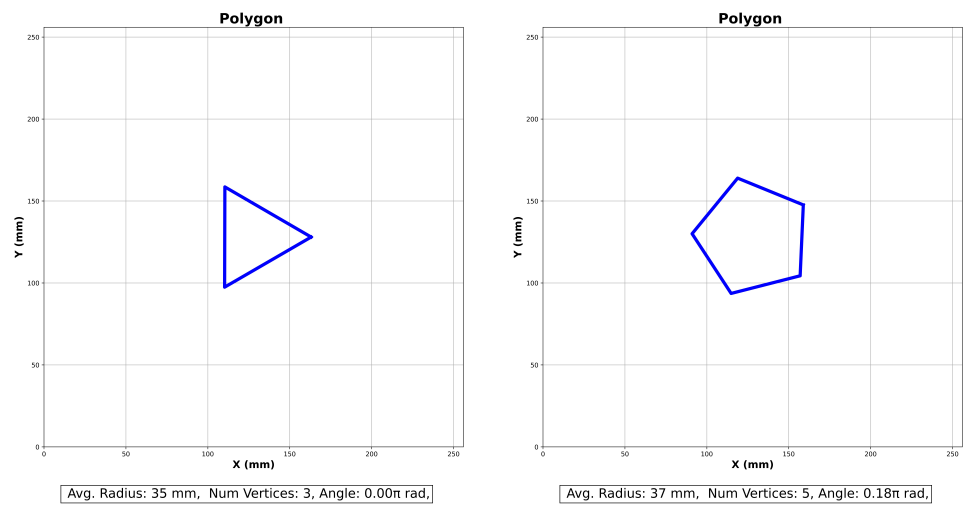


Figure 2. Generated polygons with varying number of vertices, angle, and average radius.

2.3. Dataset generation using finite elements modeling

As the initial step, finite element simulations are carried out utilizing the Abaqus/Standard FE software to construct a dataset of finite plates containing polygonal voids under uniaxial loading. CPS4R elements, characterized as 4-node bilinear plane stress quadrilateral elements with reduced integration and hourglass control, are employed in the analysis. The simulations assume a plane stress condition and disregard the influence of body forces.

Prior to dataset generation, in order to validate the present body of work, the existence of the central void was first omitted. A rectangular edge-cracked plate with its length, width, far field tensile stress, and crack length equal to 2, 1, 1, and 0.4 units, sequentially, is considered as a benchmark problem [28,32,33]. As reported by Tada et al. [28], the stress intensity factor of mode I fracture for this problem is 2.358, which is exactly obtained from the FEM simulation. Additionally, a second validation is done for non-dimensional stress intensity factors of cracks emerging from a central circular hole in a plate under tension in **Table 1** considering various crack lengths [34]. It is seen that the present model is well-validated. A mesh sensitivity analysis is also conducted. First, by making the entire model's seed size three times finer and hence increasing the approximate number of elements from 9786 to 67,611 (591%), it is seen

that the stress intensity factor values of the model only differ by a negligible amount of 0.03%. Secondly, by reducing the approximate seed size around the crack tip from 1 to 0.5 and employing finer meshes, it is observed that the stress intensity factors remain unchanged, indicating convergence and reaching a plateau. Finally, the seed size was selected as 3 over the plate's domain and 1 around the crack tip, which ensures reliable outputs.

Table 1. Comparison of normalized SIFs ($F = K/\sigma\sqrt{\pi a}$) considering $R/W = 0.5$ with those of Yan [34].

a/R	F (Yan) [34]	F (present work)	Error (%)
1.05	0.9630	0.9553	0.7996
1.1	1.2301	1.2238	0.5122
1.4	1.8247	1.8100	0.8056
1.7	2.4775	2.4656	0.4803

Finite element models of a plate featuring a central void are developed using an Abaqus script written with Python. This approach facilitates the automation of tasks, the variation of parameters, and the extraction of simulation data, thereby eliminating the need for repetitive manual operations [31]. A two-dimensional isotropic edge-cracked linear elastic plate under a tensile stress of $\sigma_0 = 100$ MPa is considered for this study, as illustrated in **Figure 3a**. In this study, the term ‘arbitrary polygonal voids’ refers to a broad class of irregular polygonal geometries generated within physically meaningful and engineering-relevant bounds. The parameter space was defined by varying the crack length, the average void radius, the number of polygon vertices, and the polygon orientation. These ranges were selected to span a wide spectrum of crack-void interaction scenarios commonly encountered in marine and mechanical structures, from sharp-cornered cavities to nearly circular irregular voids, and from short to relatively long edge cracks. Although bounded, this space generates highly diverse geometries and stress fields, allowing systematic investigation of fracture behavior while avoiding unrealistic or nonphysical configurations. The material properties include a Young's modulus of 207,000 MPa and a Poisson's ratio of 0.3. The plate incorporated a crack of length a , where this parameter varies between 50 to 70 mm. Multiple simulations were carried out to produce a diverse dataset encompassing various configurations of polygonal shapes. For each simulation, the solver initially defines the plate geometry and assigns material properties, namely the Young's modulus and Poisson's ratio. The script subsequently identifies the crack tip and generates a circle around it to facilitate improved meshing in the critical region. Additionally, a line is drawn extending from the crack tip, further enhancing the mesh quality and ensuring accurate representation of the stress field near the crack tip (**Figure 3b**). A random polygonal void is introduced at the center of the plate, with the number of vertices ranging from 3 to 10, an average radius between 30 and 40 mm, and an angle determined based on the number of vertices. Notably, the angle range varies for each number of vertices due to the axisymmetric characteristics of the problem. Within the interaction module, the crack is designated as a seam, and a contour integral crack is defined to compute the J-integral. To account for the stress singularity at the crack tip under the framework of LEFM, the mid-side

node parameter is set to 0.25, and the degenerate element side with a single node is selected. The accuracy of the calculated J-integral values and the local stress and strain fields is enhanced when singularity is incorporated into the meshes at the crack tip in small-strain analysis, compared to cases where no singularity is included. As stated before, the $1/\sqrt{r}$ singularity occurs in LEFM, which requires shifting the nodes to quarter position, achieved by setting the mid-side node parameter to 0.25. Once the plate geometry, polygonal void, and crack are defined, the script applies boundary conditions and initiates the meshing process. A localized seeding strategy is employed to ensure a denser mesh near the crack tip, as depicted in **Figure 4**, effectively capturing stress intensity factors while avoiding an excessive increase in the total element count for the model. Utilizing spider-web meshing around the crack tip enables the use of any reasonable number of contours to accurately compute the J-integral and the stress intensity factors. For this purpose, the meshing technique within the circular region around the crack tip is designated as “sweep,” while the remaining parts of the geometry are meshed using the “free” meshing technique. After each simulation run, the values corresponding to the first two fracture modes are extracted and stored in a .CSV file alongside the four input parameters: the polygon’s number of vertices, angle, average radius, and crack length.

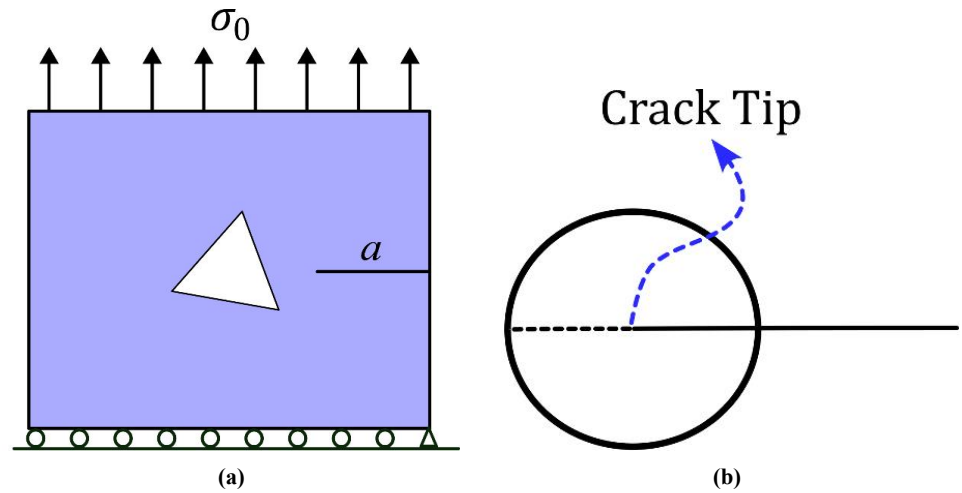


Figure 3. (a) Geometry, loading, and boundary condition of a single-edge cracked plate with polygon-shaped void; (b) Partitioning around the crack tip.

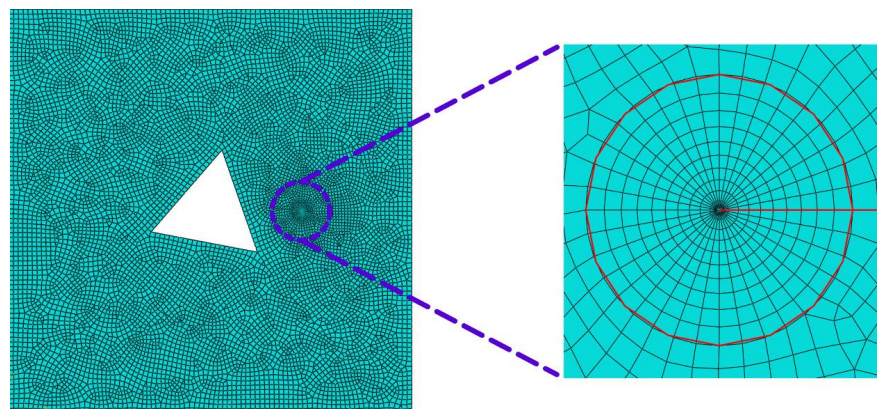


Figure 4. Meshing of a single-edge cracked plate with a polygon-shaped void.

3. Machine learning algorithms

In this section, the mathematical formulations and methodologies of the employed machine learning algorithms are detailed. Two distinct approaches are explored: the group method of data handling network, discussed in Section 3.1, and the artificial neural network, explained in Section 3.2.

3.1. GMDH

The group method of data handling, introduced by Ivakhnenko [35,36], provides a robust framework for modeling complex systems through multivariate analysis. It employs the Kolmogorov-Gabor polynomial to approximate the relationship between input and output variables. By employing a perceptron-inspired iterative structure, GMDH reconstructs polynomial models, thereby optimizing computational efficiency and improving accuracy. Although higher-order polynomials can be utilized to enhance system complexity, the quadratic polynomial with six weighting coefficients is often deemed sufficient for many engineering applications due to its simplicity and reliability [37]. GMDH incorporates mathematical modeling, nonlinear regression, and natural selection principles to regulate the network's size, complexity, and precision. The GMDH methodology simplifies complex problems through a divide-and-conquer strategy by decomposing them into smaller patterns and reassembling them through algebraic operations like sorting, addition, multiplication, crossing, and selection, where crossing and selection techniques are similarly utilized in meta-heuristic algorithms like Genetic Algorithm (GA) or Particle Swarm Optimization (PSO) [38]. The network architecture of GMDH evolves adaptively during training and comprises input, hidden, and output layers. Neurons within these layers are connected by quadratic polynomials, generating new neurons in subsequent layers to iteratively refine the model. Each node's activation function is derived from elementary polynomials, and the training process minimizes the least squares error between predicted and actual outputs. By employing an approximate function $\hat{f}$, GMDH enables accurate predictions of outputs $\hat{y}$ for given input vector $\mathbf{X} = (x_1, x_2, \dots, x_N)$ as [37,39,40],

$$\hat{y} = \hat{f}(x_{i1}, x_{i2} \dots, x_{iN}), \quad i = 1, 2, \dots, M \quad (8)$$

where M is the number of data samples, and N is the number of input variables. Additionally, this Kolmogorov-Gabor type polynomial has the form,

$$\hat{y} = a_0 + \sum_{i=1}^N a_i x_i + \sum_{i=1}^N \sum_{j=1}^N a_{ij} x_i x_j + \sum_{i=1}^N \sum_{j=1}^N \sum_{k=1}^N a_{ijk} x_i x_j x_k + \dots \quad (9)$$

in which a represents the constant coefficients of this model. The objective is to construct a GMDH-type neural network such that the root mean squared error (RMSE) between the actual output and the predicted output is minimized. Mathematically, this optimization problem can be expressed as,

$$\text{RMSE} = \sqrt{\frac{\sum_{i=1}^M [\hat{f}(x_{i1}, x_{i2}, \dots, x_{iN}) - y_i]^2}{M}} \rightarrow \min \quad (10)$$

Employing the divide-and-conquer strategy, in this network, each neuron is designed to process only two input variables, producing an output by combining these inputs through a specific mathematical operation. The second-order Ivakhnenko polynomial used for each neuron as a sub-expression is [39],

$$z_{ij} = c_0 + c_1 x_i + c_2 x_j + c_3 x_i^2 + c_4 x_j^2 + c_5 x_i x_j \quad (11)$$

where z is the output of each neuron. The coefficients c_0 to c_5 of this submodel are determined using the least squares method. Since the network produces a single output, which aims to predict the true value y , the objective is to minimize the error between the true output y and the output of each neuron z in any given neuron. This is achieved by adjusting c_0 to c_5 through the least squares method as detailed below. For a single neuron with input variables x_i and x_j and considering one sample, this problem is stated as,

$$y \approx z = [c_0 \ c_1 \ c_2 \ c_3 \ c_4 \ c_5] \begin{bmatrix} 1 \\ x_i \\ x_j \\ x_i^2 \\ x_j^2 \\ x_i x_j \end{bmatrix} \rightarrow y_{1 \times 1} = c_{1 \times 6}^T X_{6 \times 1} \quad (12)$$

Since this neuron should apply to all data samples, Equation (12) can be extended as,

$$\left. \begin{array}{l} y_1 = c^T X_1 \\ y_2 = c^T X_2 \\ \vdots \\ y_m = c^T X_m \end{array} \right\} \rightarrow [y_1 \ y_2 \ \dots \ y_m]_{1 \times m} = C_{1 \times 6}^T [X_1 \ X_2 \ \dots \ X_m]_{6 \times m} \rightarrow Y = C^T X \quad (13)$$

here, X represents the regressors matrix. To solve for the coefficient vector C , it is necessary to ensure that the number of samples m is greater than or equal to the number of unknowns. This condition ($m \geq 6$) guarantees that the system of equations is either determined or overdetermined, and consequently XX^T becomes a full-rank matrix and has an inverse. Considering this, one can write,

$$Y = C^T X \rightarrow YX^T = C^T XX^T \xrightarrow{\times (XX^T)^{-1}} YX^T (XX^T)^{-1} = C^T XX^T (XX^T)^{-1} \rightarrow C^T = YX^T (XX^T)^{-1} \quad (14)$$

where $X^T (XX^T)^{-1}$ is the right psedu-inverse, and is denoted by $X^\dagger$. Finally, the coefficients vector can be expressed as

$$\mathbf{C}^T = \mathbf{Y}\mathbf{X}^\dagger \quad (15)$$

The input data are normalized and scaled to the [1] range before being fed into the network to calculate the coefficient vector $\mathbf{C}$. This normalization is performed using the following equation

$$X_{norm} = \frac{X - X_{min}}{X_{max} - X_{min}} \quad (16)$$

Here, X_{min} and X_{max} represent the minimum and maximum values of the input variable X , respectively. Normalization ensures that all input features contribute equally during training and improves the numerical stability and convergence rate of the network.

Prior to the training process of the GMDH network, the dataset is partitioned into three subsets: training, testing, and validation data. The training data is employed to calculate the coefficients c_0 to c_5 for each neuron utilizing the least squares method, optimizing the network's internal parameters. Simultaneously, the test data is used to evaluate the performance of the neurons based on the RMSE criterion, allowing the network to retain the best-performing neurons while discarding less effective ones. In this work, 70% of all data is allocated for training purposes (training and test datasets). Finally, the validation data, which remains unseen during training and testing, is used exclusively to assess the network's performance, ensuring the model's generalization capability and preventing overfitting. A total of 30% of the data is designated as the validation dataset. The remaining 70% of the data, utilized for training purposes, is further split into two subsets: 70% of this portion is allocated for training, and the remaining 30% is assigned as the test dataset.

3.2. ANN

The goal of this model is to develop an artificial neural network model capable of predicting the stress intensity factors K_I and K_{II} in the problem in hand. The dataset used contains key features such as the number of edges, angle of void on the plate, average radius, and crack length, mapped to corresponding K_I and K_{II} values. The dataset is first preprocessed and split into training and test sets to ensure robust evaluation. Data standardization is applied to normalize feature scales, improving the model's convergence during training. Standardization is performed on both the input features and target variables to ensure uniform scaling and improve the model's convergence during training. This step is necessary because features with varying scales can disproportionately influence the optimization process, leading to slower convergence or suboptimal results. To train the ANN network, dataset is normalized utilizing Equation (17) as,

$$z = \frac{x - \mu}{\sigma_{std}} \quad (17)$$

where z is the standardized value, x is the original value, μ is the mean of the feature, and σ_{std} is the standard deviation of the feature. The dataset is then split into training

(80%) and testing (20%) subsets using scikit-learn, with a fixed random state to ensure reproducibility. This split allowed the model to learn from the training data and validate its performance on unseen testing data.

The ANN employed for predicting K_I and K_{II} is designed to capture the underlying relationships between the input features and the target variables. The architecture has an input layer consisting of four features. Two hidden layers are included to capture complex, non-linear relationships in the data. These layers are equipped with ReLU (Rectified Linear Unit) activation functions to introduce non-linearity and avoid issues like vanishing gradients during training. 32 neurons are chosen for the first hidden layer, and 16 neurons are subsequently used in the second hidden layer to progressively refine the learned representations. The output layer consists of 2 neurons, each corresponding to one of the target variables K_I and K_{II} . The two hidden layers strike a balance between model complexity and overfitting prevention, providing sufficient depth for learning complex relationships without over-parameterization. Additionally, the number of neurons in each layer is selected based on experimentation to achieve optimal model performance while avoiding overfitting or underfitting. As the final step, the neural network model is trained using the Adam optimizer, a widely used optimization algorithm that adapts learning rates for each parameter to improve convergence.

4. Results and discussion

In this section, the machine learning algorithms are applied on the dataset generated through Abaqus simulations, as detailed in the previous sections. The goal is to predict the mode I and mode II stress intensity factors of fracture using the four input parameters: polygon's number of vertices, angle, average radius, and crack length. The results are analyzed to evaluate the performance and accuracy of the models in capturing the underlying relationships within the data. The subsequent structure for the machine learning solutions is utilized,

$$\begin{cases} \text{Input variables: } \mathbf{x} = (x_1, x_2, x_3, x_4) = (\text{num verts, angle, avg radius, crack length}) \\ \text{Target variables: } \mathbf{y} = (y_1, y_2) = (K_I, K_{II}) \end{cases}$$

To emphasize the impact of the central polygon-shaped void on the calculation of stress intensity factors, **Tables 2** and **3** present the K_I and K_{II} values, sequentially, for scenarios with and without the inclusion of polygons. These tables also explore variations in the number of vertices and crack lengths, providing a comprehensive assessment of how the presence of polygonal voids influence the fracture behavior of the plate. In this scrutiny, polygon-shaped voids are considered with an angle of 0 rad and an average radius of 40 mm. It is seen that the existence of a central void has an intensifying effect on the K_I values, while K_{II} values are reduced by considering the void, indicating the protective effect of introducing a void to the left side of the crack and mitigating the possibility of fracture under in-plane shear loading.

Table 2. The intensification effect of central polygon-shaped voids on the K_I values ($MPa\sqrt{mm}$).

Crack length (mm)	50	55	60	65	70
K_I without polygon	1763.15	1918.25	2080.96	2252.17	2434.14
K_I 3 vertices	1947.88	2175.88	2429.00	2729.18	3100.63
K_I 5 vertices	2008.88	2270.09	2567.25	2926.68	3381.12
K_I 7 vertices	2074.14	2356.95	2684.30	3085.80	3599.25
K_I 9 vertices	2104.83	2399.72	2748.03	3174.85	3736.90

Table 3. Protective effect of central polygon-shaped voids on the K_{II} values ($MPa\sqrt{mm}$).

Crack length (mm)	50	55	60	65	70
K_{II} without polygon	45.79	59.47	75.94	94.08	112.88
K_{II} 3 vertices	12.74	26.09	44.18	67.00	96.40
K_{II} 5 vertices	-2.75	12.16	32.44	59.04	96.28
K_{II} 7 vertices	-9.65	6.07	28.13	57.53	99.67
K_{II} 9 vertices	-11.49	4.52	26.97	54.89	98.48

It should be noted that K_{II} may take positive or negative values depending on the direction of the local shear traction at the crack tip. In the present study, the sign of K_{II} was found to change for certain polygon configurations due to the strong asymmetry of the stress field induced by the void geometry. The interaction-integral formulation in Abaqus was employed with consistent crack-tip orientation, and both the magnitude and sign of K_I and K_{II} were verified to be independent of contour number and mesh refinement.

To introduce a comprehensive machine learning framework for prediction of stress intensity factors for opening and in-plane shear fracture modes, first, a simple GMDH network with the structure of (4, 2, 1) is considered, indicating that four input variables feed into the first and only hidden layer, producing the two best-performing neurons. These neurons are then integrated into the final layer to produce the output. Separate GMDH networks are trained for K_I and K_{II} values as outputs, both employing the aforementioned structure. For the K_I network, the correlation coefficient is $R = 0.983$, and the RMSE on the validation dataset is 69.6. The GMDH algorithm automatically selects the optimal neurons in each layer based on the lowest RMSE, discarding less effective neurons. In this case, the two selected neurons in the first hidden layer correspond to the pairs (average radius, crack length) and (number of vertices, crack length). This indicates that the polygon's angle has a negligible effect on K_I , while the crack length significantly influences it. This simple GMDH network demonstrates reasonable accuracy in predicting the mode I stress intensity factor. For K_{II} , the same GMDH network structure is applied, yielding a correlation coefficient of $R = 0.694$ and an RMSE of 39.1. For this network, the two selected neurons in the first hidden layer correspond to the pairs (angle, crack length) and (number of vertices, angle), showcasing the importance of angle of central polygon in predicting the mode II SIFs. These results suggest that this simple GMDH network does not provide sufficiently accurate predictions for K_{II} . **Figure 5** further illustrates the predictions of this GMDH network structure with one hidden layer for various validation data points.

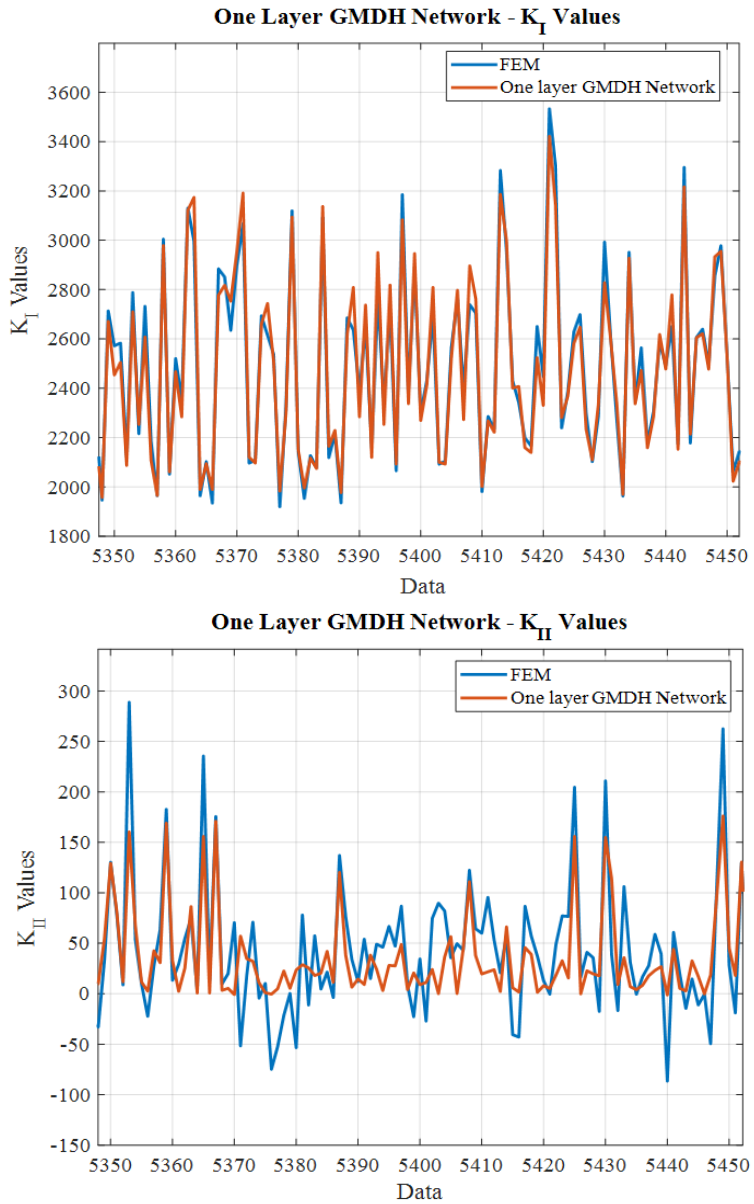


Figure 5. Performance of GMDH network consisting of one hidden layer.

A more complex GMDH network with a structure of (4, 6, 15, 50, 50, 50, 50, 50, 50, 50, 1) is considered, representing the number of neurons in each layer sequentially. To manage the network's size and complexity, a constraint is applied at each layer, retaining only the best fifty neurons based on RMSE performance. Two separate networks are trained to predict the mode I (K_I) and mode II (K_{II}) stress intensity factors. For K_I , the network achieves a high correlation factor of $R = 0.998$ and an RMSE of 20.61, demonstrating satisfactory predictive accuracy. Similarly, the network for K_{II} displays a notable improvement, achieving a correlation factor of $R = 0.86$ and an RMSE of 27.68. The performance of these networks is illustrated in **Figure 6**, which highlights the enhanced accuracy of this GMDH structure compared to the simpler network, particularly in predicting K_{II} . However, the increased accuracy comes at the cost of higher computational complexity due to the more intricate polynomial functions generated by this deeper network. To assess the influence of

individual input parameters on the performance of the GMDH network, the model was trained using three input parameters for predicting K_I and K_{II} values. For K_I , excluding the number of vertices resulted in a moderate impact, while excluding the average radius significantly affected the outcomes, highlighting its critical role in the predictions. The most influential parameter for K_I was the crack length where its omission led to a drastic reduction in prediction accuracy. For K_{II} , the exclusion of any single parameter caused a noticeable decline in performance, with the angle identified as the most influential input, reflecting its substantial effect on the prediction of mode II stress intensity factors. The relatively weak influence of polygon orientation on K_I , as compared to its pronounced effect on K_{II} , is inferred from the consistent behavior observed across the entire FEM dataset and the structure-selection process of the GMDH model. While a dedicated one-parameter sweep of the polygon angle at fixed geometry would provide further quantitative insight, such an ablation study is beyond the scope of the present work and is suggested for future investigation.

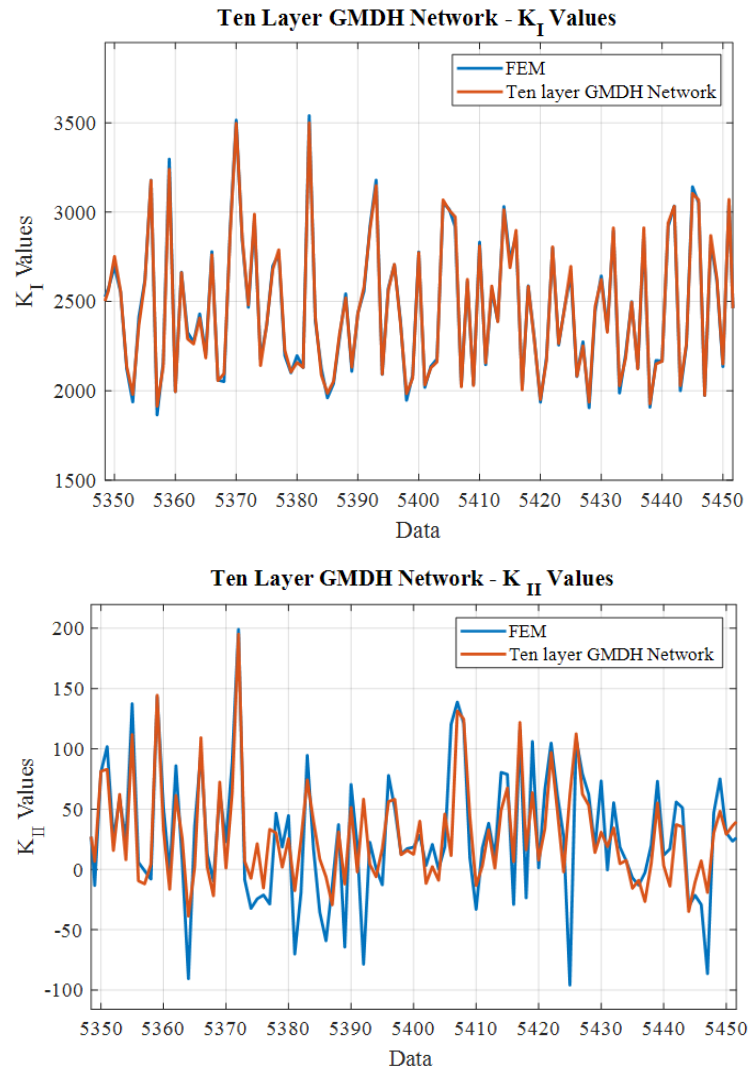


Figure 6. Performance of GMDH network consisting of ten hidden layers.

After training and analyzing the GMDH networks, a feed forward fully connected ANN is trained and evaluated. The training process spanned 50 epochs with a batch

size of 32, ensuring efficient gradient updates. To monitor the model's performance and mitigate overfitting, 20% of the training set is allocated as validation data. During each epoch, the model's Mean Squared Error (MSE) and Mean Absolute Error (MAE) metrics are computed to assess its accuracy and loss. The implementation utilized TensorFlow and Keras, powerful deep learning libraries, to define, train, and evaluate the model seamlessly. These libraries facilitate efficient computation and easy integration of scaling, splitting, and visualization tasks. The training and validation losses are indicated in **Figure 7**. For this network, the model's performance is assessed on the test dataset, indicating MSE as 216.69 for K_I and 40.27 for K_{II} . The MAE for K_I and K_{II} values are 11.10 and 4.71, respectively. The R-squared (R^2) criterion of this model for mode I and II SIF predictions are also 0.9985 and 0.9868, sequentially. These values indicate excellent performance of ANN in predicting stress intensity factors of mode I and II fractures.

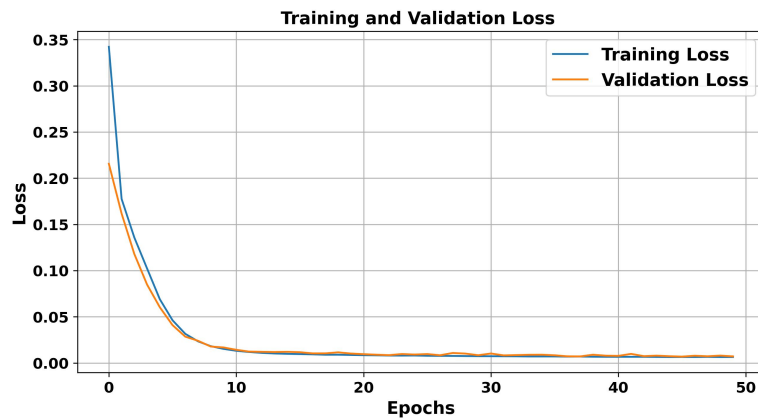


Figure 7. Loss values for the ANN training process.

Figure 8 illustrates the regression distribution of predicted versus true values for K_I and K_{II} using three models: GMDH with one hidden layer, GMDH with ten hidden layers, and the ANN network. The ideal prediction corresponds to the $y = x$ line. For K_I , the ANN network demonstrates the highest performance, achieving $R^2 = 0.998$ and $MAE = 11.1$, followed closely by the ten-layer GMDH network with $R^2 = 0.997$ and $MAE = 15.5$. The one-layer GMDH model, while delivering acceptable results, lags significantly with $R^2 = 0.968$ and $MAE = 51.58$. For K_{II} , ANN again outperforms the GMDH models with $R^2 = 0.986$ and $MAE = 4.71$. The ten-layer GMDH network exhibits moderate performance with $R^2 = 0.746$ and $MAE = 19.7$, whereas the one-layer GMDH network struggles to accurately predict K_{II} , showing $R^2 = 0.481$ and $MAE = 29.63$. This shows the inability of the GMDH polynomial network to effectively capture the nonlinear relationships governing K_{II} , while ANN demonstrates superior accuracy for both fracture modes. Comparison of true and predicted K_I and K_{II} values is provided in **Figure 9**. It is seen that ANN is capable of estimating SIF values with excellent accuracy.

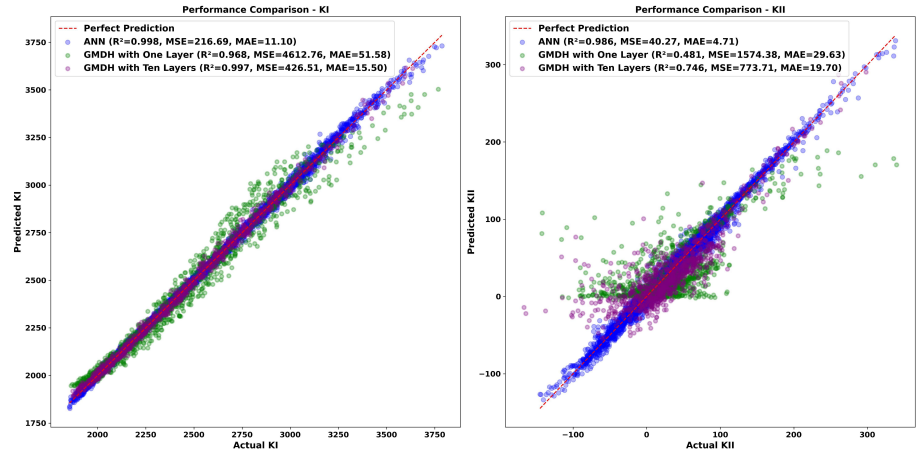


Figure 8. The correlation diagram comparing actual and predicted values of K_I and K_{II} .

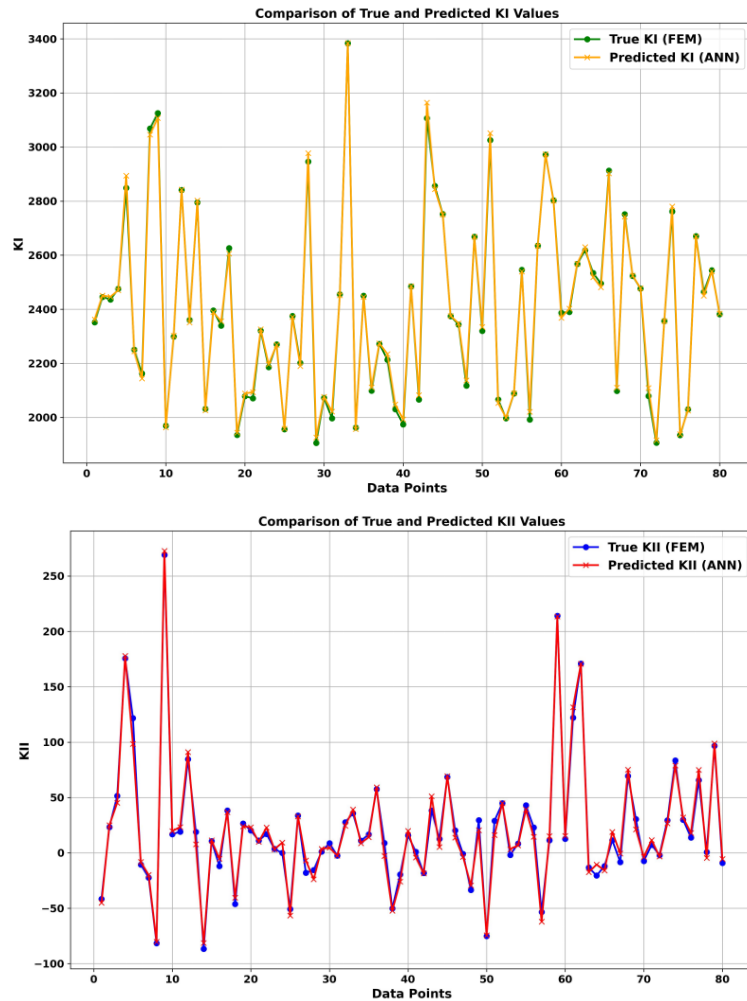


Figure 9. Performance of ANN in predicting mode I and II stress intensity factors.

One key feature of any regression-based machine learning algorithm is its capability to perform interpolation and extrapolation predictions on the newly generated data. To this aim, a numerical scrutiny is conducted on a new mini-dataset consisting of five data points. Stress intensity factors for mode I and II of fracture are estimated in **Figure 10**. Comparing the true values with the GMDH and ANN predictions, it is seen

that ANN has the capability to predict both fracture modes SIF, while GMDH fails to predict K_{II} values and exhibits lower accuracy in interpolation and extrapolation predictions comparing to the ANN.

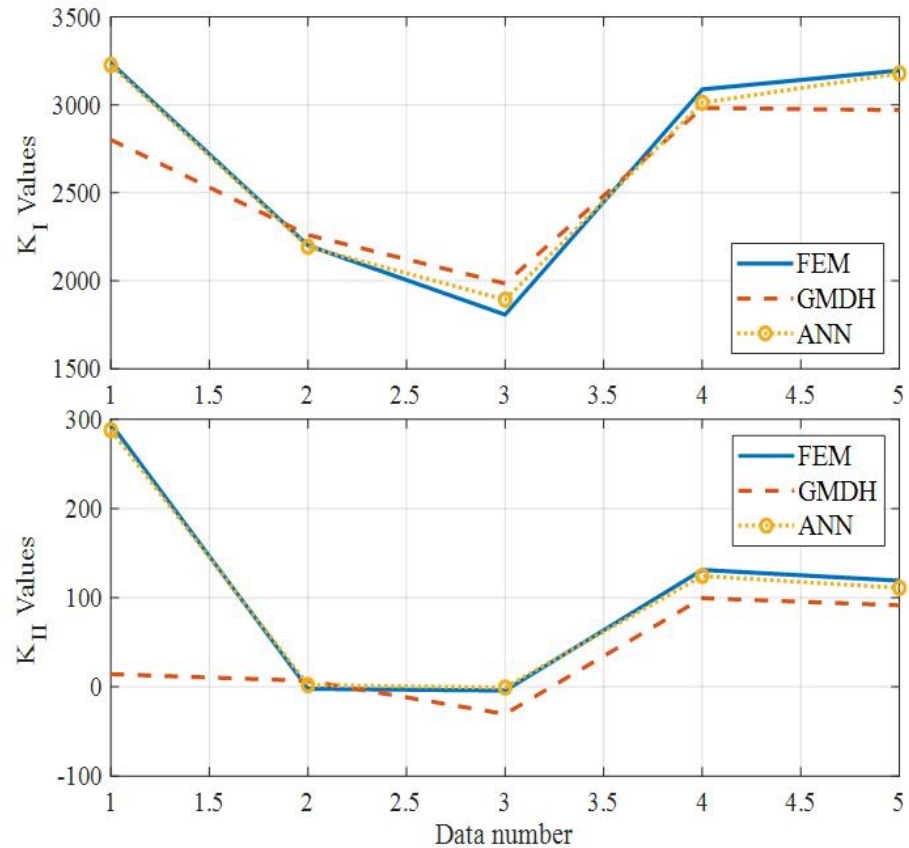


Figure 10. Comparison between interpolation and extrapolation performance of GMDH and ANN.

5. Conclusion

This study presents a comprehensive framework for predicting the mode I and mode II stress intensity factors of a single-edge cracked plate with polygon-shaped voids, utilizing machine learning algorithms and finite element analysis. A dataset was generated using Abaqus simulations, capturing the effects of four critical input parameters: the polygon's number of vertices, angle, average radius, and crack length. A GMDH network was initially employed, demonstrating reasonable accuracy in predicting K_I but struggling to capture the nonlinear relationships governing K_{II} . The performance improved significantly with a deeper GMDH network, achieving higher accuracy, particularly for K_{II} predictions, albeit at the cost of increased computational complexity. The influence of individual input parameters was also evaluated, highlighting crack length as the most critical parameter for K_I predictions and the polygon's angle as the most significant for K_{II} . Subsequently, a feed-forward fully connected artificial neural network was trained and evaluated. The ANN outperformed the GMDH networks in both fracture modes, achieving superior R^2 values and lower error metrics. The regression analysis confirmed the ANN's ability

to accurately capture the nonlinear relationships between input features and SIFs, as evidenced by its exceptional performance on validation datasets. The ANN also demonstrated robust interpolation and extrapolation capabilities, predicting SIF values with remarkable accuracy for new data points, while the GMDH models exhibited limitations, particularly for K_{II} .

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Review

Recent advances in friction stir processing (FSP) for microstructural refinement and surface property enhancement

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Abstract: Friction stir processing (FSP) has emerged as an advanced solid-state metalworking technique derived from the principles of friction stir welding (FSW). Originally developed for aluminum alloys, FSP enables controlled modification of the near-surface microstructure of metallic components through localized severe plastic deformation, material stirring, and frictional heating. These combined effects promote significant grain refinement, improved homogeneity, and densification within the processed zone, resulting in enhanced mechanical and surface performance. In recent years, FSP has been widely applied to produce ultrafine-grained structures, fabricate surface metal–matrix composites, and support the in situ formation of reinforcing phases, such as intermetallic compounds. Owing to its effectiveness in tailoring surface characteristics, FSP has strong potential for industrial implementation across the aerospace, automotive, marine, and biomedical sectors, particularly in applications requiring wear-resistant surfaces, lightweight structural elements, and high-performance materials. This review highlights recent progress in FSP research and provides insights into current developments and future directions of the technique.

Keywords: friction stir processing; aluminum alloys; surface modification; hardening with strengthening particles; hybrid in situ surfaces

1. Introduction

Friction stir welding (FSW) and friction stir processing (FSP) have significantly advanced materials engineering by enabling efficient solid-state joining and microstructure modification techniques [1]. Developed at The Welding Institute (TWI) in 1991, FSW has been widely implemented in aerospace, automotive, and shipbuilding industries because it can produce high-quality, defect-free joints with excellent mechanical performance [1].

FSW is considered a modern welding technique, and its working principle is commonly illustrated through a schematic representation (**Figure 1**), demonstrating its effectiveness for joining metallic components. In contrast to conventional rotary friction welding, which requires rotation of at least one of the workpieces during bonding [1–9], FSW overcomes several practical constraints associated with component geometry. Traditional rotary friction welding is typically limited to parts with regular shapes—most commonly circular cross-sections—and is generally restricted by component length. Typical examples include short tubes or round bars with similar diameters, where rotation can be readily applied without compromising alignment or process stability.

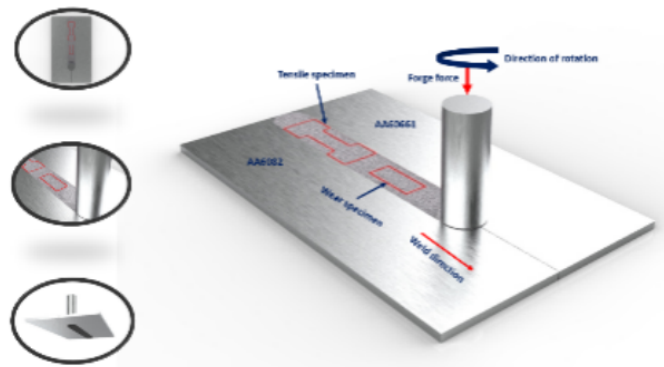


Figure 1. Schematic of the friction stir welding process.

Friction stir processing (FSP), an extension of friction stir welding (FSW), further improves material performance by enabling microstructural refinement, increasing hardness, and providing effective surface modification. This literature survey examines the growing influence of FSW and FSP in modern manufacturing and highlights their continuous development driven by ongoing technological progress [2–4]. Since its introduction, FSW has progressed into a highly efficient and versatile joining technique, particularly for aluminum alloys [4]. In contrast to conventional fusion welding, FSW is performed below the melting temperature of the base material, thereby minimizing common welding defects such as porosity, distortion, and solidification cracking.

Several notable advancements reported in recent studies [5–7] have contributed to the improved effectiveness and wider adoption of the technique, including:

- **Advanced tool designs:** The development of multi-shoulder and threaded-pin tools has enhanced material flow and mixing, resulting in stronger and more reliable joints.
- **Automation and process control:** Integration of FSW with robotic and CNC systems has improved precision, repeatability, and productivity, supporting its application in large-scale industrial manufacturing.
- **Hybrid joining approaches:** The combination of FSW with conventional arc-based welding methods has been shown to improve mechanical performance while increasing overall process efficiency.

The effectiveness of FSW in joining dissimilar materials, such as Al/Mg and Al/Steel, has further broadened its application scope, allowing for lightweight, high-strength components in automotive and aerospace engineering. FSP has emerged as a promising technique for localized microstructural modification. By subjecting metals to severe plastic deformation and controlled thermal exposure, FSP induces grain refinement, enhances hardness, and improves fatigue resistance. Recent developments [8,9] highlight:

- **Surface composite fabrication:** FSP enables in-situ synthesis of metal matrix composites by incorporating reinforcing particles such as SiC and Al₂O₃.
- **Grain refinement mechanisms:** The Hall-Petch relationship indicates that finer grain structures, achieved through FSP, result in superior strength and toughness.
- **Application in high-performance materials:** Titanium alloys and high-strength

steels have significantly improved ductility and wear resistance through FSP.

Machine learning integration in FSW and FSP: The emergence of artificial intelligence and machine learning has significantly influenced the optimization of FSW and FSP. Predictive models using deep learning can accurately forecast weld quality from input parameters, reducing trial-and-error experimentation. Key contributions [8, 10] include:

- Process parameter optimization: Artificial neural networks (ANN) and genetic algorithms (GA) enhance weld quality by identifying optimal tool rotation speeds and traverse rates.
- Defect detection and prevention: Real-time monitoring systems integrated with AI can detect anomalies and adjust parameters dynamically to prevent defects.
- Fatigue life prediction: Machine learning models have been employed to estimate the fatigue life of FSW joints, improving reliability assessments in critical applications.

Friction stir welding (FSW) is a relatively modern solid-state joining technique. As illustrated schematically in **Figure 1**, the process has proven to be highly effective for welding metallic components. In conventional rotary friction welding, at least one workpiece must rotate during joining, which imposes practical limitations on the types of components that can be processed. Specifically, this method is generally restricted to regularly shaped parts—typically those with circular cross-sections—and is often constrained by component length. Typical examples include short tubes or round bars with similar diameters, where rotation can be readily applied without compromising process stability.

2. Literature

Friction stir welding (FSW) has gained substantial recognition as a solid-state joining technique across a wide range of industrial sectors since its introduction in 1991 [11–14]. In particular, it has attracted increasing attention due to its capability to join materials that are difficult or unreliable to weld using conventional fusion-based approaches. Compared with traditional welding techniques, FSW offers several advantages, including the ability to join dissimilar or metallurgically incompatible materials and the elimination of additional joining elements such as adhesives, self-piercing rivets, or other mechanical fasteners, thereby reducing overall manufacturing cost and complexity.

Conventional fusion welding methods can negatively affect joint quality by introducing microstructural heterogeneity, residual stresses, and property degradation within the weld region during melting and solidification. In contrast, FSW operates at temperatures well below the melting points of the base materials, making it particularly effective for joining dissimilar materials while minimizing typical welding defects [15, 16].

Owing to its low heat input, use of a non-consumable tool, and ability to produce joints with improved fatigue resistance and tensile strength, FSW has contributed notably to the advancement of the automotive manufacturing sector [17].

The technique supports lightweight design strategies, enhances resource efficiency, reduces environmental impact, and lowers operational costs, making it an increasingly important joining solution for sustainable and high-performance vehicle production.

Friction stir welding (FSW) is a rapidly evolving technology with strong potential for expanded implementation in the marine industry [18, 19]. Recent developments have improved weld quality and productivity, increasing its attractiveness for industrial applications. Advanced tool designs incorporating multiple shoulders and modified pin geometries have been shown to reduce welding stresses and enhance joint strength [20–23]. In addition, automated FSW systems have improved process consistency and efficiency, particularly when welding large-scale or complex structures. Hybrid approaches that integrate FSW with complementary techniques, such as arc-based welding, have also demonstrated improvements in weld integrity and process efficiency under specific operating conditions [24–27].

FSW is currently applied in a wide range of marine-related operations, including shipbuilding and offshore structural repair. The process is particularly effective for joining aluminum alloy panels used in ship hulls and deck structures, as well as for repairing damaged components. Offshore applications also include welding aluminum alloy parts in installations such as oil platforms and wind turbines. Beyond the marine sector, FSW has been widely adopted in the automotive industry due to its relatively low heat input and ability to join materials with similar or dissimilar properties. It enables reliable joining of challenging material combinations such as Al/Mg, Al/steel, and Al/Ti, highlighting its versatility and industrial relevance [28,29].

In recent years, machine learning (ML) has emerged as a valuable modeling and optimization tool across manufacturing systems [30–34], and its application has increasingly expanded into the field of FSW. ML methods have shown strong relevance in numerous engineering areas, including fracture mechanics [35–37], structural engineering [38], composite materials [39–43], laser cutting [28], and friction stir processing [44]. Because FSW involves multiple interacting parameters and diverse welding configurations, developing accurate predictive models using conventional approaches can be complex and time intensive. Machine learning techniques are therefore well suited for FSW process modeling, as they can learn nonlinear relationships between process inputs and performance outputs through data-driven training and error minimization [45].

This review summarizes recent progress in FSW for maritime applications and emphasizes its advantages over conventional joining techniques. Key areas of advancement include improvements in joint strength, corrosion resistance, fatigue performance, welding equipment design, material selection, corrosion mitigation strategies, and surface protection methods, as well as the growing integration of machine learning tools for process prediction and optimization. Finally, the study highlights the need for further research and stresses the importance of considering economic feasibility and environmental sustainability to support wider industrial adoption of FSW technology.

3. Methods

Friction stir welding (FSW) is an advanced solid-state joining technique that produces high-strength, defect-minimized joints without melting the base materials. The process employs a specially designed non-consumable tool consisting of a rotating shoulder and a protruding pin [46–48]. As the tool is plunged into the joint interface, frictional heat and severe plastic deformation soften the surrounding material below its melting temperature, creating a highly plasticized zone. This thermomechanical action promotes grain refinement and enables intimate contact between the joining surfaces.

During welding, the tool's rotation and translational motion mechanically stir and consolidate the softened material, resulting in a continuous, robust weld along the joint line [49]. FSW is particularly well-suited for aluminum alloys, as it produces joints with excellent mechanical performance while minimizing distortion and common fusion-related defects. These characteristics have contributed to its widespread adoption in industrial manufacturing and other high-performance engineering applications [49–54]. **Table 1** summarizes the key processing parameters of friction stir welding (FSW) and friction stir processing (FSP), together with their primary effects on weld quality and material performance.

Table 1. Summarizing and outlining the key parameters of friction stir welding (FSW) and friction stir processing (FSP).

Parameter	Friction stir welding (FSW)	Friction stir processing (FSP)	Effects
Tool Rotation Speed	Higher speeds improve material flow but may cause defects at excessive values.	Higher speeds enhance plastic deformation but can lead to excessive heat.	Affects grain refinement, material flow, and defect formation.
Tool Traverse Speed	Higher speeds reduce heat input but may cause defects.	Controls processing time and material refinement.	Influences weld quality, grain size, and mechanical properties.
Tool Tilt Angle	It helps consolidate the weld material; improper tilt can lead to voids.	Affects the depth of the processed layer and material mixing.	Enhances material flow and interlayer bonding.
Axial Force	Ensures proper material consolidation; too high can cause excessive flash.	Controls penetration depth and microstructural refinement.	Affects material flow, bonding, and residual stresses.
Tool Design (Pin & Shoulder)	Influences material flow and heat generation.	Determines the extent of plastic deformation and stirring.	Affects weld strength, microstructure, and defect formation.
Material Type	Aluminum alloys are widely used; more complex materials require specialized tools.	Suitable for a range of materials, including composites.	Determines tool wear, heat input, and processing feasibility.
Cooling/Lubrication	It can help control heat input and reduce residual stresses.	Used for controlling heat input and achieving finer microstructures.	Affects grain size, hardness, and residual stress levels.
Passes (Single/Multiple)	Typically, a single pass for joining.	Multiple passes may be required for more profound material modification.	Improves microstructural uniformity and mechanical properties.

The FSW tool consists primarily of two functional elements—the shoulder and the pin—each designed to meet specific welding requirements. The shoulder, which is larger and typically flat, generates most of the frictional heat and applies the downward forging force required to constrain the workpieces and maintain intimate contact during welding. In contrast, the pin is generally designed according to the thickness of the

materials being joined and plays a central role in stirring and transporting the plasticized material within the joint region [55,56].

To withstand the severe thermo-mechanical conditions during processing, the shoulder is commonly manufactured from heat- and wear-resistant materials. In demanding applications, materials such as tungsten carbide are frequently employed, and some advanced tool systems incorporate cooling mechanisms to dissipate heat, thereby extending tool life and reducing friction-related degradation [57–62].

The pin is a smaller protrusion with a contoured or threaded profile that penetrates the joint interface. It is critical for controlling material flow, enhancing mixing, and ensuring the formation of sound weld characteristics [62]. The pin is typically fabricated from durable, heat-resistant materials, and its design parameters—such as geometry, rotational speed, and rotation direction—strongly influence the quality and consistency of the weld produced [63–65]. Modern tool configurations may also include retractable pins, which enable adjustment of penetration depth and improve adaptability when welding materials with varying thicknesses. Threaded pin designs are particularly effective for enhancing material transport and improving joint integrity [64].

During welding, the rotating tool generates sufficient frictional heating to soften the surrounding material and create a highly plasticized zone, in which significant grain refinement occurs. Because the process is conducted below the melting temperature, FSW avoids many defects commonly associated with fusion welding, including voids, inclusions, and solidification-related imperfections. As the tool advances along the joint line, the stirred material consolidates and re-solidifies behind it, forming a dense weld region—often referred to as the weld “nugget”—characterized by a refined microstructure and high mechanical strength [66].

The weld quality achieved in friction stir welding (FSW) is strongly governed by a set of critical process parameters that must be carefully optimized for each application. Appropriate adjustment of these variables is essential for producing defect-free joints with the required mechanical performance and structural integrity [66–69]. This ability to tailor processing conditions also makes FSW a highly adaptable technique for joining a wide range of materials and component geometries.

Tool rotational speed plays a major role in controlling heat generation, material plasticization, and flow behavior within the stirred zone. Although increasing rotational speed can enhance productivity and improve material mixing, excessively high speeds may accelerate tool wear and reduce material flow stability. Traverse speed (welding speed) also significantly influences weld formation by regulating the time available for stirring and consolidation. Lower traverse speeds generally increase heat input and promote effective material mixing, but they can extend cycle time and may lead to excessive softening. In contrast, higher traverse speeds reduce heat input and improve productivity; however, insufficient heat and stirring may result in incomplete bonding and reduced joint quality [69–73].

Tool material selection is another crucial factor, as FSW tools must withstand severe mechanical loading and elevated temperatures while maintaining resistance to wear and deformation. Common tool materials include tungsten carbide, tool steels, and cermets, which are chosen based on the hardness and thermal conductivity of the workpiece

materials. Selecting an appropriate tool material is important not only to prevent tool failure but also to avoid contamination and ensure stable material flow during welding. In addition, tool design—particularly the geometry of the pin and shoulder—directly affects heat generation, material transport, and joint soundness [74–77]. Pin profiles and shoulder dimensions are typically customized according to joint configuration and material thickness. At the same time, tool tilt angle can further improve forging pressure and enhance joint consolidation in certain applications [78–81].

The applied downward (axial) force is equally important, as it governs the contact conditions between the tool and workpieces and directly affects frictional heating and consolidation pressure. This force must be controlled to maintain stable material flow and prevent defects such as tunneling or void formation. Furthermore, the weldability and thermal behavior of the base material influence the required process window, often demanding parameter modifications for thicker sections, high-strength alloys, or complex joint geometries. By systematically tuning these factors, FSW can achieve consistent, high-quality welds across diverse industrial applications.

Overall, FSW relies on a rotating pin–shoulder tool that generates frictional heat and severe plastic deformation to soften the material without melting it. This produces a plasticized region where efficient mixing and grain refinement occur, enabling the formation of sound joints with refined microstructures. The selection of tool materials, tool geometry, and operational parameters—such as rotational speed, traverse speed, and axial force—collectively determines weld quality, efficiency, and the resulting mechanical properties of the joint.

Friction stir welding (FSW) is a solid-state joining technique capable of producing pronounced microstructural and mechanical modifications within welded joints. The intense stirring action promotes grain refinement and microstructural evolution, which directly influences key mechanical properties such as hardness, tensile strength, and fracture toughness, making FSW joints suitable for demanding industrial applications [82–84]. Therefore, a detailed evaluation of the microstructural development and mechanical response of FSW joints is essential for understanding weld performance and reliability. Texture formation within the stirred region is primarily associated with the combined effects of severe plastic deformation and elevated temperatures during processing [85–87]. In addition, precipitate fragmentation, dissolution, and subsequent coarsening are commonly observed within and adjacent to the stirred zone, further contributing to property variations across the weld [88–90]. Owing to the strong correlation between local microstructural changes and post-weld mechanical behavior, numerous studies have investigated the evolution of microstructure during FSW [89–94].

The severe deformation and frictional heat generated during FSW result in a refined microstructure in the stirred region, with the most prominent transformation occurring in the weld nugget zone (WNZ). At the interface between the dynamically recrystallized nugget and the base material, a more gradual transition is typically observed on the advancing side, whereas a sharper boundary is often present on the retreating side [95]. The nugget morphology generally appears in two common forms: (i) a basin-shaped structure that widens toward the upper surface and (ii) an elliptical

or oval-shaped nugget. The final nugget geometry is governed by several interacting factors, including welding parameters, tool design, workpiece temperature, and the thermal conductivity of the weld material.

Dynamic recrystallization during FSW is widely recognized as the primary mechanism responsible for producing fine, equiaxed grains in the weld nugget [96–98]. The resulting recrystallized grain size is strongly dependent on process conditions, including tool rotation and traverse speeds, tool geometry, alloy composition, axial load, local thermal history, and cooling rate within the stirred zone. In many cases, grains in the upper region of the nugget are relatively larger due to increased heat input, while grain size decreases with increasing distance from the weld centerline as the thermal and deformation intensity diminish.

Several investigations have reported that hardness evolution following FSW differs significantly between precipitation-hardened and solid-solution-hardened aluminum alloys. In precipitation-strengthened systems, a distinct thermomechanically affected zone (TMAZ) typically forms around the weld centerline [99–104]. The observed reduction in hardness in these alloys is mainly attributed to the dissolution, fragmentation, and coarsening of strengthening precipitates under the thermal cycle imposed during FSW [105–111].

Sato et al. [102] and Wang and Shivkumar [112], for example, studied hardness profiles in FSW 6063Al-T5 and concluded that precipitate distribution had a more dominant influence on hardness than grain size. Similarly, Svensson et al. [103] and Zhang and Zheng [113] examined the microstructure and mechanical behavior of FSW 5083Al-O and reported that the nugget zone exhibited fine equiaxed grains, accompanied by a reduced fraction of large particles and an increased presence of smaller particles.

Beyond welding, friction stir processing (FSP) introduces severe plastic deformation and material mixing within the processed region, enabling microstructure tailoring and surface composite formation. The effective strain during FSP has been estimated to reach high levels, allowing the incorporation of reinforcement particles into metallic substrates and the production of surface metal–matrix composites [114]. Chang et al. [115] reported early results on fabricating SiC particle–reinforced aluminum surface composites using FSP. In their approach, SiC powder was dispersed in a small amount of methanol to improve handling, then deposited onto the plate surface as a thin, relatively uniform layer. Subsequent friction stir processing facilitated particle incorporation into the substrate. Using optimized tool designs and processing parameters, a composite layer approximately 100 μm thick was successfully produced on 6061 aluminum alloy, exhibiting uniform particle distribution and strong bonding with the substrate [116, 117]. Further studies demonstrated that by adjusting FSP conditions, the SiC content could be increased from approximately 5 to 27 vol.% within the aluminum matrix. As expected, the addition of SiC significantly enhanced the hardness of the processed surface layer, with hardness increasing with increasing reinforcement volume fraction, particularly in SiCp/5083Al surface composites [116].

4. Result and discussion

Applying FSP produces considerable frictional heating and severe plastic deformation, leading to recrystallization in the dynamic agitated zone. In this instance, equiaxed, fine-grained, uniform-size grain recrystallization was generated in the Vasava et al. [108] present a representative microstructure of FSP 7075Al-T651. A refined microstructure of approximately 7.5 μm was generated at a tool rotation speed of 400 rpm and a traverse velocity of 102 mm/min. Despite ongoing disputes over the grain-refining mechanism in the SZ, it is widely accepted that grain refinement results from dynamic recrystallization [118–121]. Consequently, the nucleation and dynamic recrystallization development parameters will dictate the SZ's final grain microstructure. The size of the recrystallized grains in the SZ is greatly influenced by the FSP parameters, tool geometry, material chemistry, workpiece temperature, vertical pressure, and active cooling. Data is current as of October 2023. Ma and Mishra present transmission electron microscopy (TEM) micrographs of FSP Al-4Mg-1Zr samples processed under various FSP conditions. FSP parameters and tool geometries can be changed to alter the grain size of FSP samples, resulting in an ultra-fine-grained microstructure of 0.4 to 0.7 μm [122].

The friction stir process (FSP), including friction stir welding (FSW), is a solid-state joining technique extensively used for metals and alloys, particularly aluminum, magnesium, and other nonferrous materials. Its ability to produce defect-free welds and modify microstructures without melting the base material has garnered significant attention in the aerospace, automotive, and shipbuilding industries. This document explores how mechanical properties and microstructure influence the friction stir process. Mechanical properties such as the base material's strength, hardness, ductility, and toughness directly affect FSP. These properties determine the material's behavior under the intense thermal and mechanical loads induced during the process. High-strength materials, such as specific grades of aluminum or titanium alloys, pose challenges for FSP due to their resistance to deformation. The tool must withstand high forces and temperatures to effectively plasticize the material. Conversely, softer materials like magnesium alloys require lower forces but are more susceptible to surface defects, such as flash or voids, due to their low hardness.

Ductility plays a crucial role in determining the material flow during FSP. Materials with high ductility exhibit smooth material flow around the tool, resulting in uniform welds with fewer defects. Conversely, low-ductility materials may fracture or form voids under the same conditions. The thermal conductivity of the base material impacts heat dissipation during FSP. High-thermal-conductivity materials, such as aluminum alloys, dissipate heat rapidly, requiring higher rotational and traverse speeds to maintain sufficient plasticization. Thermal expansion affects residual stresses and distortions in the weld zone. The friction stir process profoundly alters the microstructure of the base material due to the combined effects of plastic deformation, dynamic recrystallization, and heat generation. These changes typically result in a fine-grained microstructure in the stirred zone (SZ), which enhances mechanical properties.

Dynamic recrystallization is the primary mechanism responsible for grain refinement during FSP. The intense plastic deformation and high strain rates cause the original coarse grains to break into smaller, equiaxed grains. Due to the Hall-Petch effect, fine grains in the SZ contribute to improved strength and toughness. FSP can significantly alter the material's crystallographic texture. Severe plastic deformation aligns grains along specific orientations depending on the tool rotation and traverse direction. This texture evolution can affect anisotropic properties, such as directional strength and corrosion resistance.

Depending on temperature and cooling rate, FSP can induce phase transformations in alloys. For example, precipitates such as Mg_2Si or Al_2Cu in aluminum alloys can dissolve and reprecipitate during FSP, altering the mechanical properties. Similarly, changes in α and β phases can occur in titanium alloys, affecting the weld zone's strength and toughness. The mechanical properties of the friction-stir weld differ significantly from those of the base material due to microstructural modifications. The weld zone can be divided into the stirred zone (SZ), the thermomechanically affected zone (TMAZ), and the heat-affected zone (HAZ), each with distinct characteristics. The SZ, known as the nugget zone, experiences intense plastic deformation and dynamic recrystallization. This results in a fine-grained microstructure with improved mechanical properties, including tensile strength, hardness, and fatigue resistance. However, the SZ's properties depend on the material and process parameters, including tool geometry, rotational speed, and traverse speed.

The TMAZ lies adjacent to the SZ and experiences both thermal and mechanical effects, though to a lesser degree. The grains in this zone are plastically deformed but not fully recrystallized. The mechanical properties of the TMAZ are typically lower than those of the SZ and base material due to the presence of elongated grains and potential residual stresses. The HAZ undergoes thermal exposure without significant plastic deformation. This zone often exhibits reduced mechanical properties due to precipitate coarsening or over-aging in precipitation-strengthened alloys. Proper control of process parameters can minimize the adverse effects of the HAZ. Defects such as voids, tunnel defects, and surface flash can compromise the mechanical properties of the weld. These defects typically arise due to improper process parameters or tool design. Tunnel defects are elongated voids that form due to insufficient material flow. They can act as stress concentrators, reducing tensile strength and fatigue resistance. Excessive heat or material flow can cause surface flash, leading to material wastage and aesthetic concerns. This defect does not significantly affect mechanical properties but does affect weld quality and post-processing requirements. Root defects occur due to inadequate penetration or improper tool tilt. These defects can compromise weld integrity, particularly under dynamic loading conditions.

The mechanical properties and microstructure of the weld are heavily influenced by tool design and process parameters. Tool geometry, rotational speed, traverse speed, and tilt angle all play a vital role in determining weld quality. The tool's shoulder and pin profile affect material flow and heat generation. Tools with complex pin profiles, such as threaded or tapered pins, enhance material mixing and reduce defects. The shoulder design controls heat input and surface finish. Optimal rotational and traverse

speeds ensure adequate heat generation and material plasticization. High rotational speeds generate more heat, promoting dynamic recrystallization, while low traverse speeds allow sufficient time for material flow. A slight tilt of the tool enhances material consolidation and reduces the formation of voids. However, excessive tilt can lead to improper penetration and defects.

The ability of friction stir processing (FSP) to precisely tailor microstructures and mechanical performance has significantly expanded its range of applications. Ongoing progress in tool design, parameter optimization, and real-time monitoring technologies is enabling FSP to meet increasingly complex and demanding industrial requirements. In particular, FSP has attracted attention in the aerospace and automotive sectors for improving the performance of lightweight alloys, such as aluminum and magnesium. The resulting refined microstructures and enhanced mechanical properties contribute to improved structural efficiency, reduced weight, and better fuel economy. In addition, FSP is widely used for surface modification, enabling controlled processing conditions to produce engineered surface layers with superior wear resistance, corrosion resistance, and fatigue performance.

Despite these advantages, several technical challenges still limit large-scale adoption. These include difficulties in processing high-melting-point and high-strength materials (e.g., steels and titanium alloys), controlling defect formation, and achieving consistent scalability for industrial production. Recent innovations in tool materials and protective coatings, hybrid FSP techniques, and computational modeling approaches are actively addressing these issues, thereby supporting broader implementation of FSP across engineering applications.

During FSP, a fine and homogeneous densified microstructure is typically produced due to the intense shearing, fragmentation, and stirring action generated by the rotating threaded pin [123–125]. This capability enables FSP to serve as an effective microstructure-modification tool for heterogeneous metallic systems, including aluminum-based composites and nanophase aluminum alloys fabricated via powder metallurgy (PM) routes [19,126]. Under severe plastic deformation, extremely high strain rates (on the order of 10^2 – 10^3 s⁻¹) and cumulative strains approaching 40 have been reported, promoting accelerated diffusion, enhanced material mixing, and effective breakdown of coarse secondary phases [127–129]. Sharma et al. [130] noted that the material flow around the pin during friction stir processing exhibits similarities to conventional metal milling, highlighting the strong mechanical stirring nature of the process.

FSP's ability to induce intense plastic deformation also makes it promising for in-situ material synthesis. Mechanical alloying has long been used to fabricate nanostructured materials by repeatedly welding, fracturing, and re-welding powders [131]. Similarly, the severe deformation and thermal exposure generated during FSP can promote diffusion-driven reactions between constituent elements, creating favorable conditions for in-situ formation of composite reinforcements and intermetallic compounds.

Although many studies have examined FSW and FSP as independent processes, each technique presents limitations when applied alone. Recent research has therefore explored integrated hybrid approaches that combine friction stir welding and

processing concepts to achieve more reliable optimization outcomes and to resolve conflicting design objectives [132–167]. Such hybrid frameworks can serve as scalable decision-support tools for optimizing friction-stir processes while reducing reliance on extensive experimental trials. By advancing both FSW and FSP methodologies, these approaches provide practical guidance for producing high-quality welded joints and consistently modified microstructures, offering a valuable benchmark for future research and industrial applications [168–177].

Overall, FSP produces a thin, homogeneous, and pore-free wrought-like structure by breaking down coarse dendrites and second-phase particles, refining matrix grains, closing porosity, and dissolving or redistributing precipitates under the combined effects of frictional heating and severe plastic deformation [1, 178–205]. This makes FSP an energy-efficient, environmentally sustainable, and highly versatile solid-state technique capable of generating fine-grained structures, developing surface composites, modifying heterogeneous materials, and enabling in-situ synthesis of composites or intermetallic phases. With continued research and improved understanding of processes, FSP is expected to evolve into a general-purpose metalworking technique for localized microstructure control and near-surface property enhancement. Although challenges remain, the method offers substantial commercial and technological potential across advanced engineering industries.

Finally, while friction stir welding (FSW) and friction stir processing (FSP) are based on the same thermomechanical principles—namely plastic deformation and frictional heating induced by a rotating tool—their purposes differ fundamentally. FSW is primarily applied for solid-state joining of two or more components, with joint strength, fatigue resistance, and defect minimization being the main performance targets. In contrast, FSP is applied to a single workpiece to locally refine microstructure and enhance properties, improving surface or bulk characteristics such as hardness, wear resistance, corrosion resistance, and fatigue performance without creating a joint.

In contrast, friction stir processing (FSP) is primarily used to refine microstructure and enhance surface or near-surface properties within a single workpiece. Rather than producing a joint, FSP modifies the processed region through intense plastic deformation and controlled thermal exposure, typically resulting in a homogeneous, fine-grained microstructure with improved mechanical and surface performance. The tooling and processing conditions in FSP are selected to maximize material mixing and flow within the processed zone while avoiding overheating and ensuring uniformity across the treated area.

Tool design plays a critical role in both techniques, but the design objectives differ. In FSW, tool geometry—particularly the shoulder features and pin profile—is optimized to promote balanced stirring between the two workpieces, ensuring defect-free bonding and high joint strength. In FSP, tool design and parameters are tailored to achieve the desired level of microstructural modification and property improvement, such as grain refinement or the distribution of reinforcement particles in surface composites.

From a functional perspective, FSW is used to join two or more components, and performance evaluation is centered on joint strength, fatigue resistance, and

reliability. Conversely, FSP is used to enhance the mechanical properties of a localized region, improving hardness, ductility, wear resistance, and corrosion resistance without forming a joint. Overall, although both processes operate based on similar thermo-mechanical principles, FSW is primarily a joining technique, whereas FSP is a microstructure-engineering method aimed at improving the performance of existing materials.

5. Conclusion

Intense plastic deformation and frictional heating during friction stir processing (FSP) produce pronounced microstructural transformations in metallic materials. The process effectively fragments coarse dendritic structures and second-phase particles, refines the matrix grains, eliminates porosity, and promotes the dissolution and redistribution of precipitates. As a result, the processed zone typically exhibits a fine-grained, homogeneous, and pore-free wrought-like structure with improved integrity and performance. The successful use of FSP in producing ultrafine-grained microstructures, fabricating surface composites, modifying heterogeneous materials, and enabling in-situ formation of composites or intermetallic phases demonstrates its energy efficiency, environmental friendliness, and broad applicability.

Despite these advantages, several barriers must be addressed to support large-scale industrial implementation. Key challenges include severe tool wear when processing high-strength or hard-to-deform alloys, limited scalability for large components, and difficulties in maintaining stable material flow and uniform processing quality in complex geometries. Future research should prioritize the development of advanced tool materials and protective coatings to extend tool life, alongside systematic optimization of processing parameters to reduce defects and improve repeatability. In parallel, implementing real-time monitoring and feedback control systems will be essential for achieving reliable quality assurance. Moreover, coupling computational modeling with experimental investigations can provide deeper insight into thermo-mechanical behavior during FSP, enabling improved prediction and control of microstructural evolution.

With continued progress, FSP is expected to mature into a robust and standardized metalworking technology capable of precise near-surface microstructure tailoring. Emerging hybrid approaches—such as integrating FSP with additive manufacturing routes or laser-assisted processing—may further extend its functionality and application range. Through sustained innovation and process optimization, FSP has the potential to significantly influence the future of metallic material fabrication, processing, and performance enhancement across aerospace, automotive, biomedical, and other advanced engineering sectors.

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