

Article

Integration of Lattice Structures into the Boring Bars as a Passive Chatter Suppression Technique: Conceptualization, Modelling and Simulation

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Abstract: The present study concentrates on passive damping technology, in which the damping of vibrations is accomplished by the integration of lattice structures into the boring bar. To complete this process, several steps must be followed. First, the largest possible hollow space within the boring bar was determined, and the two main influencing factors—stiffness and natural frequency—were harmonized. A rigorous analysis of vibration reduction was conducted on the basis of a validated simulation model. This analysis involved six distinct lattice structures designed using ANSYS SpaceClaim 19.0. In light of the findings, a specialized, application-specific CAD simulation tool was developed, employing appropriate methodologies to circumvent the limitations of conventional CAD software. For the hollow integrated into the boring bar, ellipsoidal shapes were shown to be preferable to cylindrical ones due to their superior dynamic performance. The initial lattice structure, namely a cube lattice with side cross supports, exhibited an enhancement in damping of 55.58% in comparison with the reference model. Following this result, five additional modelling steps were performed, leading to an optimal outcome with a 67.79% reduction in vibrations. Moreover, the modifications made to the beam diameter of the lattice units yielded enhanced dynamic performance, as evidenced by a vibration suppression of 69.81%. The implementation of complex modelling steps, such as the integration of a hollow and the integration of lattice structures, could be successfully achieved through the development of a suitable and user-friendly simulation tool. The effectiveness of the simulation tool in enabling parameterized modelling for scalable lattice structures was demonstrated. This approach was found to be expeditious in terms of the time required for implementation. The potential exists for the extension of this simulation tool, with the objective of facilitating research projects with a view to optimization, i.e., a large number of research projects.

Keywords: chatter; vibration damping; lattice structures; modelling and simulation; CAD simulation tool



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1. Introduction

In the field of machining, the issue of unstable vibrations is of particular significance. These factors have been shown to significantly impair productivity, particularly during

the machining of superalloys, which are used in a wide range of applications (Figure 1). In comparison with milling and turning, drilling and boring are the most demanding machining techniques. Boring bars are subject to vibrations of a considerable magnitude due to their elongated, slender, and cantilever geometry and their inherently low structural damping [1]. These vibrations are commonly categorized as free, forced, and self-excited. Free vibration can be defined as the naturally dampened oscillatory motion of a part subsequent to the application of a sudden impact. Forced vibrations are a consequence of periodic and continuous forces. These can be addressed by applying predicted internal or perturbation forces to the system's transfer function. In other words, the solution of ordinary differential equations is employed in the time domain [2]. Depending on the magnitude of the cutting forces, the process of material removal can result in the destabilization of the material, which, in turn, can precipitate self-excited vibrations, also referred to as chatter [3]. Chatter has been shown to exert a detrimental effect on the quality of the workpieces and the tool life, as well as on the machine components [4]. The machining of titanium and nickel-based alloys is a complex process, often associated with high costs due to the limited capabilities of the cutting tools, which can only operate at relatively low speeds and have a limited tool life. Consequently, these materials have been found to generate a greater amount of chatter due to the necessity of higher cutting forces because of their hardness and strength.

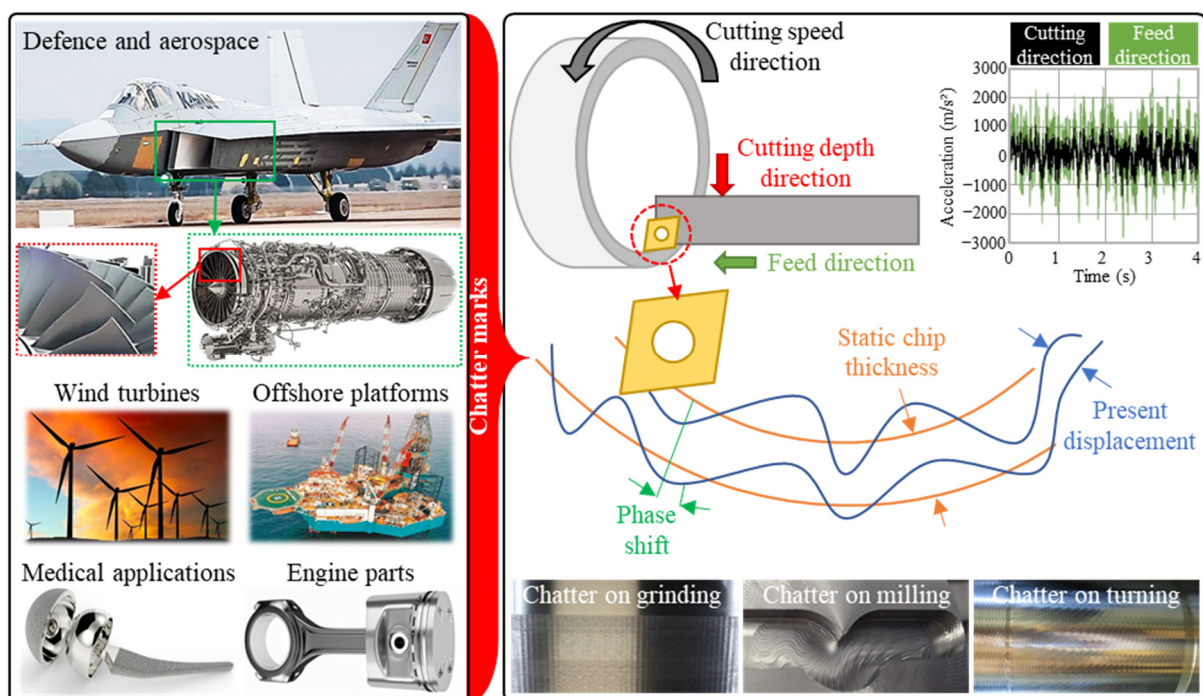


Figure 1. Application examples and mechanism of chatter. In accordance with the literature [5–12].

Self-excited vibrations depend on the motion parameters of the vibrating system and arise solely from the presence of a variable force. Furthermore, vibrations are generated within the system due to the presence of internal feedback loops between the motion of the machine tool housing elements (components of the mass-damper-spring system) and the operational process [13]. The phenomenon of self-excited vibrations is primarily attributable to the regenerative effect. The resultant cutting force generates vibrations in the tool, thereby creating a wavy surface on the workpiece. This surface is a reflection of the relative motion between the tool and the workpiece. These irregularities result in variations in chip thickness, which in turn affects the cutting force and causes further vibration. The issue of chatter has been the focus of academic research since the 1950s. Consequently, a

plethora of studies have been conducted that describe the methods by which chatter can be detected, avoided, reduced, or controlled, as well as modelled and predicted [14,15]. Various methodologies have been used to investigate the generation mechanisms of chatter and to identify and validate chatter control strategies [16,17]. In the field of machining, the identification of dynamic reactions is initiated through the utilization of sensors, with the subsequent determination of stable and unstable regions being achieved through chatter stability analysis. However, the prediction and suppression of chatter will continue to pose a significant challenge in the future, primarily due to the following factors: increasing material removal rate, limitations in the design procedure, low friction guiding systems, lightweight design, manufacture of flexible parts, process parameter selection, regeneration disturbance, process damping maximization, system stiffness enhancement, and system damping enhancement [18].

Vibration reduction can be achieved using various methods, including active, passive, and hybrid dampers [19]. Active dampers are equipped with sophisticated mechanisms that facilitate the instantaneous measurement of vibrations. These systems use complex algorithms to calculate the reactive motion and then apply counteracting forces to suppress vibrations. In the context of passive damping, the impact of material and design modifications on damping efficacy is examined. The employment of additive parts is identified as a simple and straightforward approach [20,21]. As illustrated in Table 1, the active and passive methods are characterized by a number of distinct features [22].

Table 1. Characteristics of active and passive damping methods [22].

Vibration Damping	Characteristics
Active Methods	Installation of additional devices, such as actuators
	Use of advanced and complex control algorithms
	Knowledge of vibrations eigen-frequencies
	Model-based strategy
Passive Methods	Based on viscoelastic materials, viscous fluids, magnetic, or passive piezoelectric, lightweight materials
	Vibration energy dissipation or redirection
	Cost effective
	Dampers are usually small size and easy to install

In the context of machining pre-drilled workpieces, the utilization of long and cantilever tool holders is imperative, particularly in the case of deep holes. These tool holders, known as boring bars, are composed of solid round steel, and their performance is influenced by two primary factors: stiffness and natural frequency [23]. While the cross-sectional area and moment of inertia remain uniform for a solid boring bar, these parameters are subject to alteration upon the introduction of a hollow within the tool holder during the modification process. It is evident that the length of the hollow is directly proportional to the decrease in static stiffness, while the diameter is inversely proportional to the increase in natural frequency. It is therefore of great importance that both factors are coordinated in order to optimize the hollow. Additionally, the position of the hollow significantly affects the natural frequency. The creation of the hollow in closer proximity to the cutting edge has been demonstrated to result in an increase in the natural frequency, owing to a decrease in the inertia of the region in close proximity to the aforementioned cutting edge. In the context of tool holder modification, it is anticipated that the hollow will be sufficiently dimensioned to accommodate the integration of lattice structures, which consist of a network of trusses comprising struts or plates that are interconnected with each

other. This integration has the potential to produce a structural response characterized by reduced vibrations [24]. Lattice structures have a wide range of applications, ranging from sheet-like parts with simple 2D patterns [25–28] to 3D solid parts filled with complex shapes such as Triply Periodic Minimal Surfaces (TPMS) [29–31]. The utilization of these designs engenders numerous advantages, including enhanced heat dissipation [32,33], vibration and impact damping [33,34], and weight reduction [29]. In addition to the aforementioned critical improvements, it has been demonstrated that exceptional mechanical properties [35] can be obtained, such as negative Poisson's ratio [36] at the same porosity level [37]. Kagome [38,39], cubic [26,30,38,40], octahedron [21,41], star, hexagon, and diamond [33,35,42] are the most commonly studied lattice structures due to their relatively simple and effective characteristics. For these lattice structures, dynamic performances and behaviors are determined by their density [40], dimension [27], and orientation [41,42].

Given the complexity of implementing lattice structures in this sector, rigorous modelling and analysis are crucial. To ensure the efficient use of resources, it is first necessary to undertake modelling, followed by analysis, and then production. It is imperative to recognize that the methodologies employed in the fabrication of lattice structures are, by their very nature, typically laborious and expensive. Therefore, modelling and analysis must be carried out with the utmost precision. The recent surge in interest in the design of lattice structures is largely attributable to the advent of additive manufacturing (AM). Consequently, a plethora of CAD (Computer-Aided Design) programs have been developed [43]. Commercial CAD software faces several difficulties in the generation of lattice structure models [24]. It is evident that most parametric CAD software tools do not fully utilize the potential for freedom of form and design. Instead, they are dependent on different criteria (for example, usability, reliability, availability, performance, support, and cost) that significantly influence the decision-making process [44]. It can be concluded that, despite the ease with which lattice structures can now be employed, no significant breakthroughs have yet been achieved in terms of design methods and CAD tools for general applications [45]. The conception of a lattice structure model in the CAD environment presents numerous challenges. It is not possible for product designers to automatically generate different types of configurations [24]. It has to consider issues of homogenization, which means that the individual lattice units must be much smaller than the design space in all directions, and of periodicity (the material inside the lattice unit must be such that it corresponds to the material in the adjoining lattice unit) [46]. The merits of lattice structures have recently come to the fore in production technologies. Yang et al. [47] aimed to increase damping by integrating six different lattice structures into the cutting tool during the milling operations. In this process, three distinct cell sizes and filling ratios were utilized for the fabrication of lattice structures. An investigation into the modal properties indicated that the design incorporating an FCC lattice exhibited superior performance, thus prompting the execution of experimental tests with this design. Following the conclusion of the experimental phase, a maximum enhancement of up to 100% in terms of stability limit was achieved. In a separate study, Yang et al. [48] similarly modified the milling tool with a lattice structure. However, damping particles were also incorporated into the lattice structure. It is evident that the combination of the damping capability of the particles with the lattice structure has resulted in an enhancement of the stability limit that exceeds 200% in comparison with the conventional cutting tool. The use of a lattice structure can be observed in turning as well as milling operations. Vogel et al. [21] modified a tool holder with a lattice structure, subsequently filling it with a variety of damping particles. Consequently, the lowest amplitude result was obtained by filling the lattice structure with 50% WC-ZrO₂. Even though the effect of filling particles is studied in detail, the existing literature reveals a necessity for further research, which must focus specifically on the

hollow geometry and lattice parameters of boring bars. It is therefore essential to present a methodology that demonstrates and explains the optimal modification procedure of boring bars for integrating lattice structures.

The topology with periodic lattice structures repeating in all directions has been shown to exhibit both great strength and high specific stiffness. To achieve integration of the periodic lattice structure, it is necessary to trim the structure outside or inside the design space. However, this may result in the lattice unit becoming non-conformal to the boundary surfaces of the design space. To generate a complete lattice structure, each CAD lattice unit must be designed to receive inputs and provide feedback to support the formation of the overall structure. There is still a considerable amount of research and development work to be carried out in the field of adapted commercial design solutions. Consequently, each innovation constitutes a substantial enhancement to the process [49]. In particular, the number of individual CAD lattice unit models is too high for the creation of lattice structures, so alternative approaches are needed [50]. Commercial CAD software often relies on parametric Non-Uniform Rational B-Spline (NURBS) systems, which are suitable for traditional manufacturing models but pose challenges for generating complex digital 3D lattice structures [46,51]. The manual design of lattice structures, where multiple lattice units need to be connected and integrated into a design space such as a hollow, reaches a level of complexity that cannot be solved by conventional CAD software [52].

In this paper, the initial step involved the formulation of a simulation model for the establishment of a design space (hollow) for a boring bar, and the subsequent development of a 3D lattice simulation model. The objective of this study was to utilize the design space to assess the vibrations of five distinct 3D lattice structures. The objective of the numerical analysis is to ascertain whether a reduction in undesirable vibrations can be achieved with this method (Figure 2).

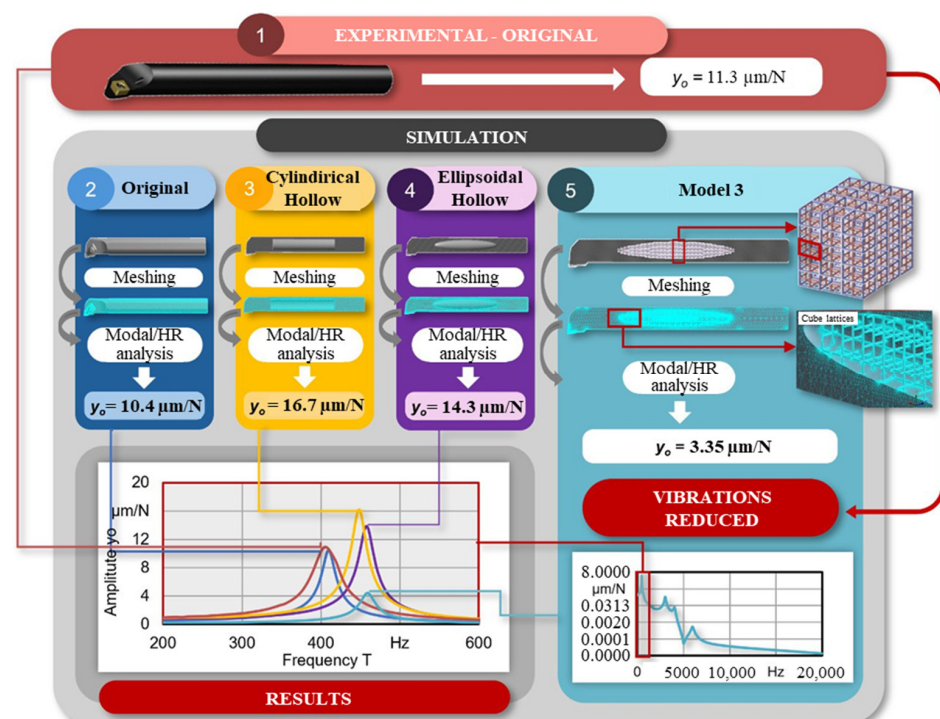


Figure 2. Schematic illustration of the first step: Determination of an optimal design space for a boring bar; testing of varying 3D lattice structures which was created by ANSYS SpaceClaim19.0 CAD software; evaluation of various modified Boring bar simulation models using modal and harmonic response analysis (HRA).

Based on the results and the possibility to reduce the vibrations with this method, the second step was the development of a parameterized CAD software tool (Figure 3). This tool enables the special application, i.e., for the analysis of the vibration reductions with different 3D lattice structures, and will be a great support for future research work. The primary focus of this paper is the practical numerical implementation and development of suitable numerical approaches. It does not concern itself with concrete vibration analyses or with a comparison of their simulated and experimental results. This research constitutes a fundamental basis in itself, thus paving the way for future studies in this domain. To summarize this paper:

- calculate a suitable design space for a mold with an existing boring bar base body
- make it available in a simulation model in order to
 - fill this design space with 3D lattice structures and
 - use the finite element method (FEM) simulation to determine whether and on what approximate scale the undesirable vibrations during machining could be reduced using this method.

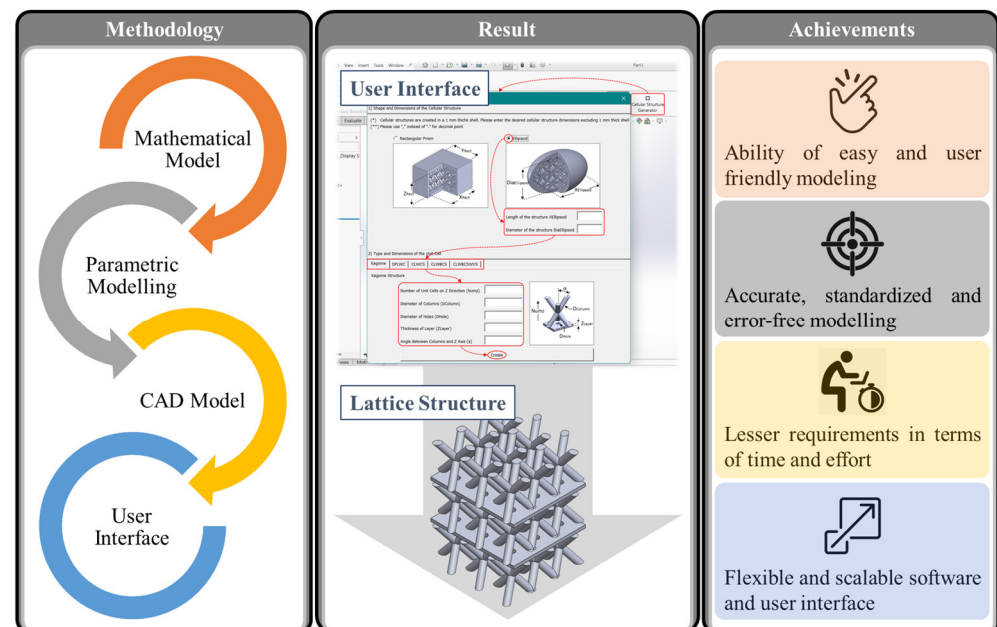


Figure 3. Step by step methodology (mathematical model, parametric modelling, CAD and GUI development), resulted lattice structure and achievements (user-friendliness, accurate modelling, efficient and scalable software).

2. Determination of an Optimal Design Space and Testing 3D Lattices Simulation Model

In this study, the basics and the numerical determination of an optimal design space for a boring bar and the testing of the 3D lattices simulation model are presented. The accuracy of the simulation model has been verified experimentally.

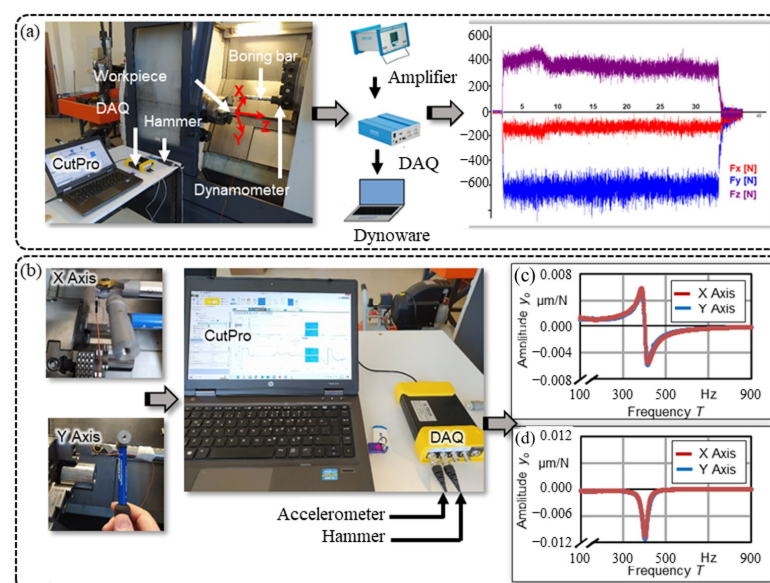
2.1. Experimental Setup for First Step

Boring tests were conducted to obtain the dynamic behavior of the original boring bar. Therefore, the simulative and experimental results of the original boring bar were compared and evaluated. The tests were carried out on a Spinner TC400 52 MC CNC Lathe (Faculty of Technology, University of Afyon Kocatepe/Turkey) using a boring bar made of chrome-vanadium alloyed spring steel (51CrV4) under wet conditions (a water-soluble coolant). The chemical composition and material properties of steel are given in Table 2.

Table 2. Chemical and mechanical properties of the boring bar material (51CrV4).

Physical	Tensile Strength	Yield Point	Elongation	Contraction	Impact Resistance	Elasticity Modulus	Density	Poisson's Ratio
	1000–1200 MPa	>550 MPa	>10%	>45%	>30 J	210×10^3 MPa	7.7 kg/dm ³	0.29
Chemical	Minimum	C %	Si %	Mn %	P %	S %	Cr %	V %
	Maximum	0.47	-	0.70	-	-	0.90	0.10
		0.55	0.40	1.10	0.025	0.025	1.20	0.25

Inconel 718, which is a nickel-based super alloy, was used in the tests as the work material. First, a pilot hole with a diameter of $d = 20$ mm was drilled into the workpiece. For this purpose, an indexable insert drill (ISO code: 870-2000-20LX1-5) and a cutting insert (ISO code: 870-2000-20-MM 2334) suitable for this drill were used. The pilot hole was then enlarged up to $d = 30$ mm. The reference drill rod (ISO code: E20S-SCLCR 09-R) with a diameter of $d = 25$ mm was used for hole enlargement. In the study, the cutting conditions recommended by the tool manufacturer were taken into consideration. In this framework, boring tests were performed at a constant cutting speed ($V_c = 75$ m/min), feed ($f = 0.1$ mm/rev), and depth of cut ($a_p = 2$ mm). Each of the tests was repeated three times for statistical validation. A new indexable insert was fixed on the tool holder at the beginning of each test to ensure comparable results. The cutting forces created during machining were measured using a Kistler 9119A dynamometer (Faculty of Technology, University of Afyon Kocatepe/Turkey) (Figure 4a). To measure the natural frequencies of the reference tool, an accelerometer was connected to the tooltip. The dynamic characteristics of the boring bar were measured using a hammer (Dytran brand, model number 5800B4). An accelerometer (Dytran brand, model number 322F1) was mounted at the position of interest to measure the response. A four-channel data acquisition system (Novian brand, model number S04) was used in combination with CutPRO's® Tap Testing measurement module [53]. Since the stability of machining operations is governed by the dynamics of the tool in radial and tangential directions [54,55], i.e., the X and Y directions in Figure 4b, the amplitude value of the tool holder was obtained by applying the hammer test in the X and Y directions. As can be seen from Figure 4b, the Real and Imaginary graphs obtained in the X and Y axes overlap with each other. This confirms that the boring bar exhibits similar dynamic characteristics in both axes.

**Figure 4.** (a) Experimental setup for force measurement, (b) Tap testing system, and (c) real and (d) imaginary parts for X and Y axis.

2.2. Modal Analysis and Harmonic Response Analysis with ANSYS

Simulative modal analysis using the FEM is an effective and reliable choice to determine the quality of the lattice structure and its dynamic properties. For the FEM, different solution functions are available in ANSYS such as the Reduced Method, the Subspace Method, Unsymmetrical Method, and Damped Method. There is also the Block Lanczos Method, which is suitable for undamped linear systems with symmetrical matrices [56].

In general, the FEM attributes discretization to the differential equation systems, so that typical structural dynamic problems arise during the solution. Based on d'Alembert's principle, the generalized equation of motion of the discretization process of a continuous structure can be described by FEM:

$$M\ddot{q} + C\dot{q} + Kq = f(t) \quad (1)$$

where M is the mass matrix, $\ddot{q}$ the second derivative of the displacement q , $\dot{q}$ is the velocity, C the damping matrix, K is the stiffness matrix, and $f(t)$ is the vector of applied forces at time t . In ANSYS, programmed FEM codes exist for the solution of static displacement and static stresses, natural frequencies and mode forms, forced harmonic response amplitude and dynamic stress, transient dynamic behavior, and transient stresses, so that the first part of the solution for the free oscillation of the structure can be given with the following matrix equation:

$$M\ddot{q} + Kq = 0 \quad (2)$$

With this equation, the natural frequencies and the mode forms of linear undamped systems can be determined and correspond mathematically to the solution of an eigenvalue problem. With modal analysis, the dynamic properties of the model can be understood, and based on this, a harmonic response analysis can be performed to determine the behavior of the structure under a stationary sinusoidal (harmonic) load at a certain frequency:

$$f = (f_{max}e^{j\psi})e^{j\Omega t} \quad (3)$$

$$q = (q_{max}e^{j\theta})e^{j\Omega t} \quad (4)$$

where e^j is the sinusoidal motion with a phase shift and Ω is the excitation frequency at which the load is applied. There may be a force shift ψ due to different loads and a phase shift θ . The answer to harmonic response analysis is then solved with:

$$(-\Omega^2 M + j\Omega C + K)(x_1 + jx_2) = (F_1 + jF_2) \quad (5)$$

2.3. Development of a Simulation Model

In this section, a simulation model is developed for the passive damping modification of the boring bar, aiming to insert the largest possible and most suitable volume defined by an optimized diameter-to-length ratio of a hollow into the bar. During the simulations, a modal damping ratio of 2% was assumed throughout the structure to account for energy dissipation effects. For the finite element mesh, Solid186 tetrahedral elements (3D, second-order elements) were used, with a maximum element size of 2 mm to ensure sufficient accuracy in capturing the dynamic behavior. The material properties of the boring bar correspond to chrome-vanadium steel (51CrV4), as listed in Table 2. The boring bar was fixed at the clamping (tool holder) end as a boundary condition. An excitation force of 5 N was applied at the tool tip, acting separately in the X and Y directions to simulate operational vibrations. The frequency range for the analysis was set from 200 Hz to 800 Hz, with a step size of 1 Hz to capture the resonance peaks accurately. The Block Lanczos

Method was utilized as the solver approach for the eigenvalue extraction during the modal analysis stage.

The allowable bending stress of the material is an important criterion that must not be exceeded during the modification. Figure 5 gives an overview of the modification procedure for the cylindrical hollow space. The procedure involves two preliminary steps and they utilize the same bending stress equations. The initial step pertains to the analytical calculations of the original boring bar, whereas the subsequent step employs the identical approach to the simulated model. In a similar manner, the third and fourth steps also utilize the same control mechanism for a modified boring bar with a cylindrical hollow.

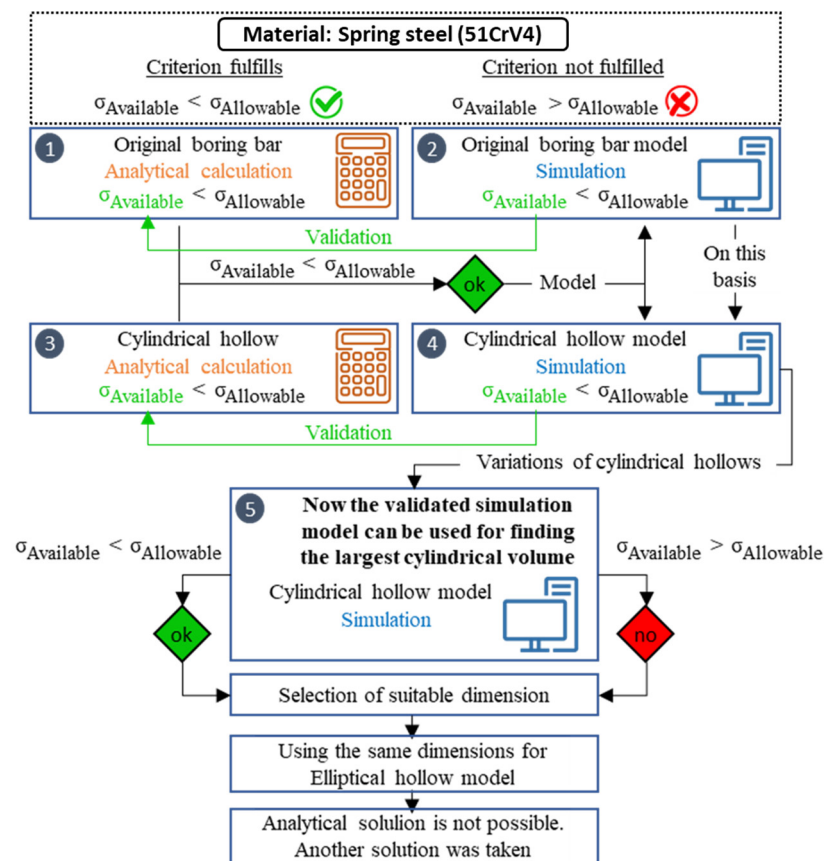


Figure 5. A procedure for the validation of the simulation model; control steps for original (step 1–2) and modified (step 3–4) boring bars followed by the finding of largest volume for elliptical hollow (step 5).

The allowable bending stress value for the boring bar material 51CrV4 is $\sigma_{b_{allowable}} = 183.3$ MPa. For reference tool 1, the existing bending stresses were calculated analytically, and the existing values of the bending stress were validated with the simulated reference model 2. On this basis, it was possible to simulate the cylindrical hollow model. To be able to validate the cylindrical hollow model as well, the existing bending stresses were calculated analogously in the same way as described above and were compared with the existing simulation values.

It is evident that, in light of the aforementioned validation, the subsequent determination and selection of additional cylindrical hollows, characterized by heterogeneity in terms of length and diameter, can be facilitated on the basis of this criterion. The appropriate dimension for the cylindrical hollow is also to be used for the ellipsoidal hollow. Due to the ellipsoidal geometry, the analytical calculation cannot be performed, so a different solution was used to validate the simulation model (Figure 6). For the analyses, the measured force data from the test were considered.

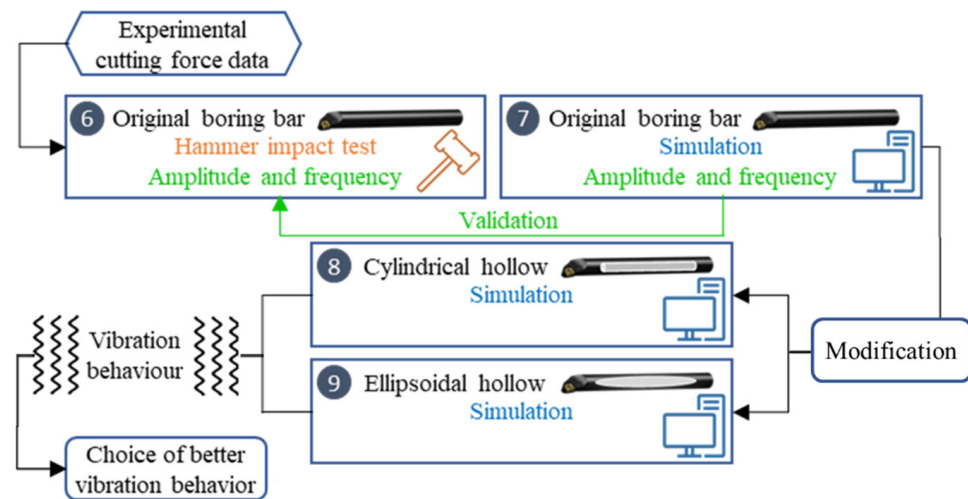


Figure 6. Procedure for the validation of the simulation model; validation of the simulated boring bar (step 7) by the experimental test (step 6) and implementing the modifications of cylindrical (step 8) and ellipsoidal (step 9) hollow geometries.

2.4. Determination of the Maximum Hollow Volume by FEM Simulation

Dynamic stiffness is defined as the ability to resist process-induced vibrational forces. This factor is strongly dependent on the tool overhang and the diameter of the boring bar [57]. This explains why the dynamic stiffness in boring is typically low and decreases as the L/d ratio increases, thereby amplifying vibrations [58]. As the hollow is modified, the internal cross-sectional area changes and so does the area moment of inertia, marked A-A and B-B in Figure 7. It is located to the left and right sides of where the hollow ends. The dimensions of the boring bar, acting forces on it, and the boundary conditions for the simulation are also shown in Figure 7. A cylindrical hollow space was selected in the geometry modelling area to perform the validation. As the drill starts to remove material, the feed and depth of cut create a comma-shaped chip profile. This real intervention situation was considered in the simulations.

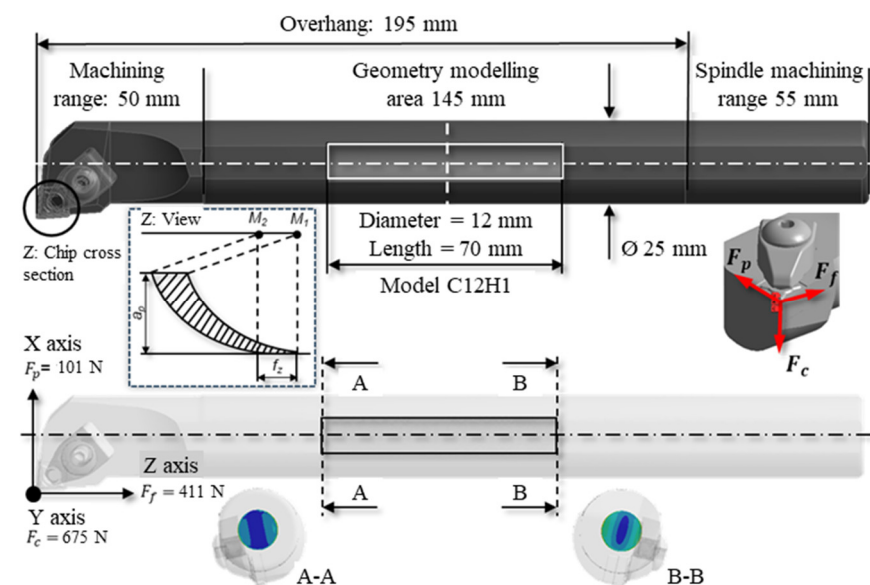


Figure 7. Dimensions of the boring bar; geometry modelling area with cylindrical hollow model C12H1, cross-sections (A-A and B-B), and the corresponding moments of inertia.

In the run-up to the modification, the existing bending stress on the:

- reference model (original boring bar) was calculated analytically with $\sigma_{(Available)} = 42.90$ MPa. This value is far below the permissible bending stress $\sigma_{(Allowable)} = 183.3$ MPa, which means that the criterion is fulfilled.
- The cylindrical hollow space was also calculated analytically. This value is also far below the allowable bending stress with $\sigma_{(Allowable)} = 45.30$ MPa.

Subsequently, the reference model (original boring bar) and the model of the cylindrical hollow space were generated and structured for the simulation. These are shown in Figure 8, although pragmatically the ellipsoidal hollow is also shown here, which will not be discussed until Section 2.6.

- The simulation of the reference model (original boring bar) resulted in a value of $\sigma_{(Available)} = 43.30$ MPa for the existing bending stress. Validated with the analytical value, a very good agreement was achieved with only a marginal deviation of 0.92%.
- Based on the validated reference model, the simulation of the cylindrical hollow was performed and an existing bending stress of $\sigma_{(Available)} = 47.76$ MPa was achieved. Validated with the analytical value, a very good agreement was also obtained by a small deviation of 5.15%.

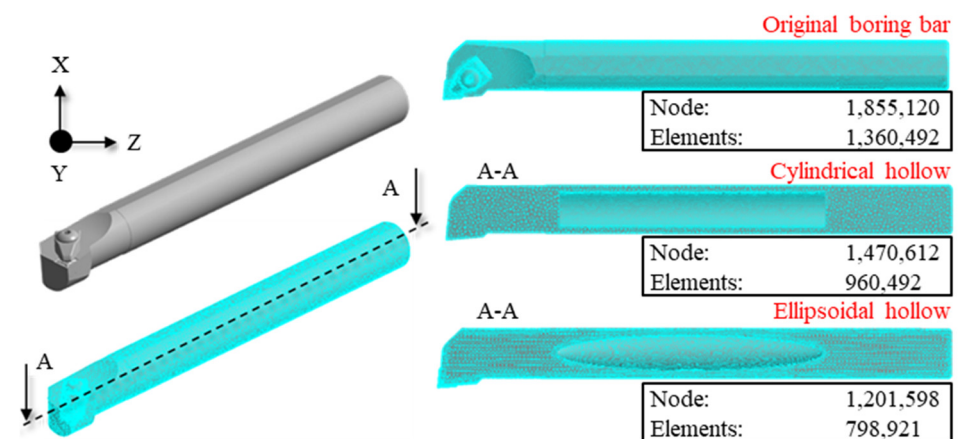


Figure 8. Structuring of the boring bar: original boring bar model, cylindrical hollow model, and ellipsoidal hollow model.

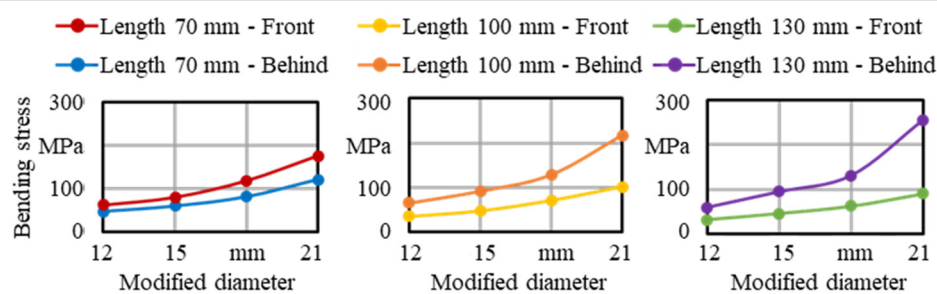
For the cylindrical hollow, different lengths and diameters were selected and all configurations were simulated. The results are listed and graphically illustrated in Table 3. In the region of the moment of inertia B-B, the values of the bending stresses are higher than in area A-A for all results. For the models C21H2 and C21H3, the allowable bending stress is already exceeded and the model C21H1 is only just below it. However, since the length here is only $L = 70$ mm and the aim is to achieve the largest possible hollow volume, these were no longer considered in subsequent analyses. The models with diameters of $d = 12$ mm and 15 mm are also no longer included in the analyses for the same reason. It is evident that the model C18H3 has been deemed eligible for progression to subsequent phases of the investigative process, which will involve a range of analytical procedures.

Table 3. Insertion of a cylindrical hollow into the boring bar and simulation of the available bending stress.

Model	Modified Diameter	Length	Bending Stress in Section A-A	Bending Stress in Section B-B
C12H1	12 mm	70 mm	47.768 MPa	63.119 MPa
C12H2	12 mm	100 mm	36.219 MPa	65.659 MPa
C12H3	12 mm	130 mm	32.275 MPa	59.626 MPa

Table 3. Cont.

Model	Modified Diameter	Length	Bending Stress in Section A-A	Bending Stress in Section B-B
C15H1	15 mm	70 mm	60.100 MPa	81.006 MPa
C15H2	15 mm	100 mm	48.598 MPa	92.165 MPa
C15H3	15 mm	130 mm	46.153 MPa	94.645 MPa
C18H1	18 mm	70 mm	81.493 MPa	118.690 MPa
C18H2	18 mm	100 mm	71.630 MPa	128.910 MPa
C18H3	18 mm	130 mm	62.477 MPa	131.050 MPa
C21H1	21 mm	70 mm	121.370 MPa	175.840 MPa
C21H2	21 mm	100 mm	102.110 MPa	217.450 MPa
C21H3	21 mm	130 mm	90.315 MPa	255.730 MPa



2.5. Modal Analysis and Harmonic Response Analysis of the Modified Hollow

For the modification of the ellipsoidal hollow, the same dimensions are used as for the C18H3 model, i.e., $d = 18$ mm and $L = 130$ mm. A harmonic response analysis (HRA) is performed with the ANSYS software, and the resulting amplitude and frequency are compared. Subsequently, the HRA is also carried out with the two modified hollows. Therefore, it will be possible to decide which geometry (elliptical or cylindrical) has the lower vibration amplitude.

The basis of HRA is modal analysis, which is used for determining the natural frequencies and natural oscillations. The vibration modes are investigated without excitation or loading (non-linear changes or influences). Based on the reference model (original boring bar), the two modified hollows are taken into account in the idealization and discretization using FEM. For this purpose, the clamping length of the boring bars and the material properties were determined. Obtained results are given in Figure 9. The modal analysis resulted in six rigid body eigenmodes for each boring bar. It can be seen that the respective frequencies of the reference model (original boring bar) are higher than those of the modified boring bars. The frequencies of the cylindrical hollow are lower than those of the ellipsoidal hollow.

The HRA was initiated with the reference model (original boring bar), utilizing the findings from the modal analysis and the experimental determination of loads. It can be seen that the largest deflection, i.e., oscillation, occurs both in the experiment and in the simulation around an amplitude of $y_0 = 11.3 \mu\text{m}/\text{N}$ and a frequency of $T = 416.94$ Hz for the first mode. With this very good agreement, the reference model (original boring bar) also shows that the validity requirements are met and can be used predictively (Figure 10).

The HRA was performed in the same way with the two modified boring bars, i.e., C18H3 and E18H3 (Figure 11). The cutting force values used in the analysis were taken from Figure 7. As expected, the stiffness of the two boring bars in which a hollow form was inserted was reduced, which is why the deflection is higher than in the experiment and in the reference model (original boring bar). Interestingly, the E18H3 exhibited a

significantly lower vibration amplitude compared to the C18H3. The development of the simulation model has not only allowed the determination of a suitable hollow volume with a weighting according to the two variables, diameter and length, but also a more suitable hollow shape.

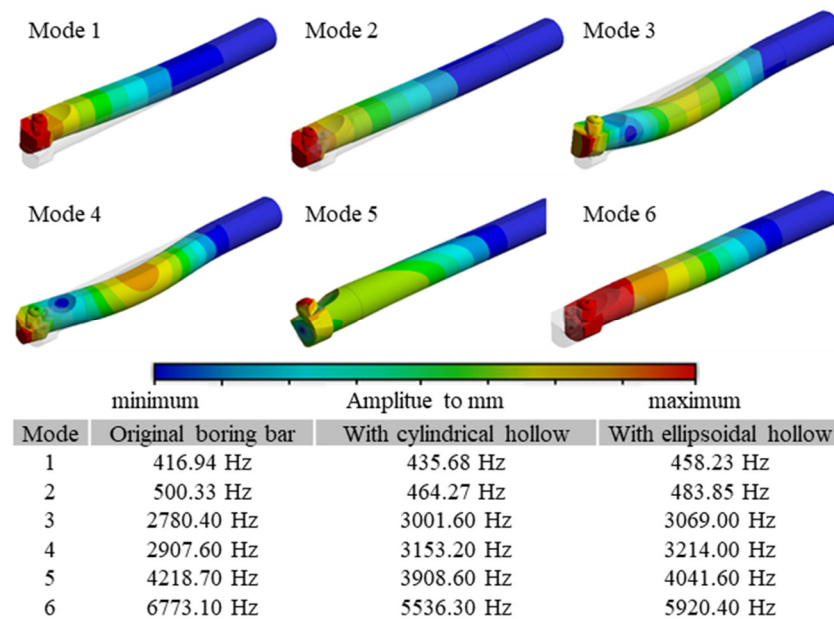


Figure 9. Solutions of the modal analysis: natural frequencies of three models for six eigenmodes.

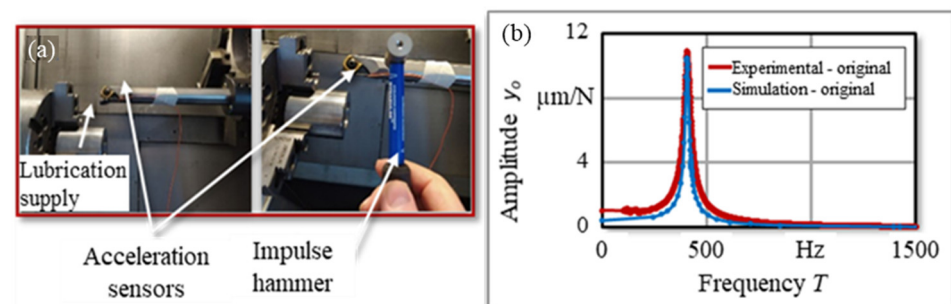


Figure 10. (a) Experimental validation and (b) comparison of the simulation results with the experimental data.

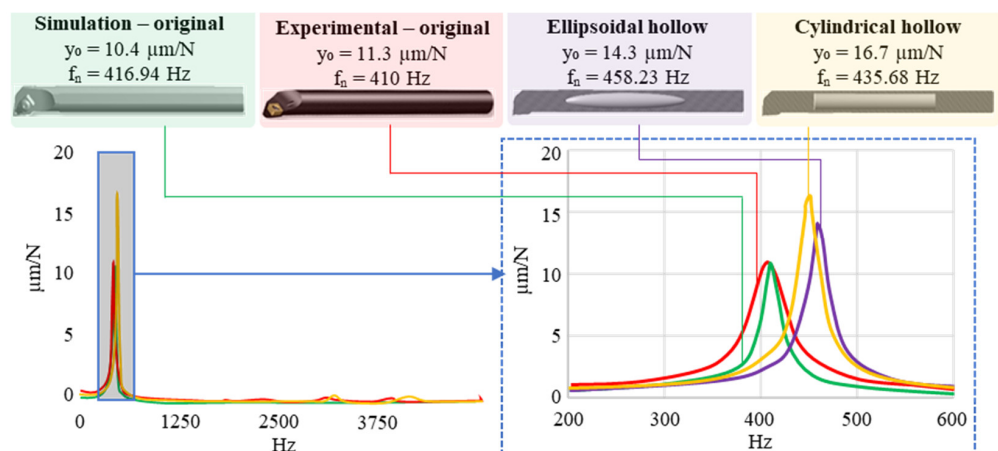


Figure 11. Comparison of the results: validation of the simulation model and the dynamic performance of hollow models (E18H3 and C18H3).

2.6. FEM Reference Model Damping Analysis with Lattice Cube

Several studies have highlighted the influence of lattice structures on the vibration behavior of products [59–62]. Therefore, this study models the effect of lattice structures on chatter—a critical issue in boring—for the first time. In this context, a single type of lattice structure of CLWSCS (Cube lattice with side cross support), which is believed to have a vibration-damping effect, was preferred. The modelled boring bar E18H3 was filled with a lattice structure consisting of individual cube lattices with side cross supports (Figure 12). As mentioned before, since the modelling phase is extremely difficult, this issue is discussed separately, and the solution-related studies are explained in Section 3. The 3D lattice structure has several advantageous properties compared to the reference model (original boring bar), such as good energy absorption properties, specific stiffness, lower weight/density, and multifunctionality. These properties are further enhanced by the addition of cross-support. The filled model was subjected to Harmonic Response Analysis (HRA) using the same methodology described earlier.

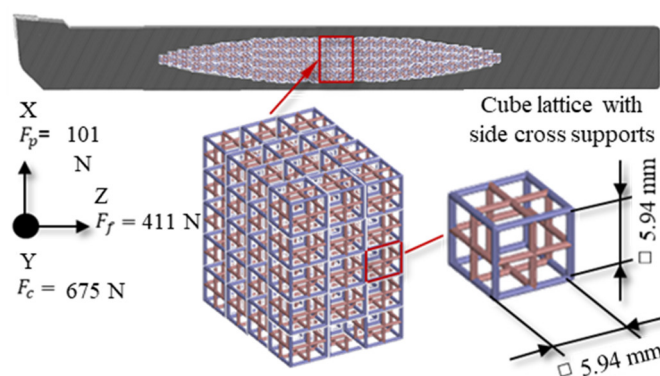


Figure 12. Ellipsoidal hollow filled with cube lattices.

The structuring of the cube lattice structure, which is located in the ellipsoidal hollow of the boring bar, was very challenging. The physical properties of the vibration behavior to be analyzed are strongly dependent on the degree of structuring. The structures are tetrahedral-independent structures, which means that the element size is independent of the local geometry (Figure 13). The structure of the three-dimensional cube lattices could not be solved individually but had to be solved in conjunction with the neighboring cubes at all nodes and the edge junctions of the hollow.

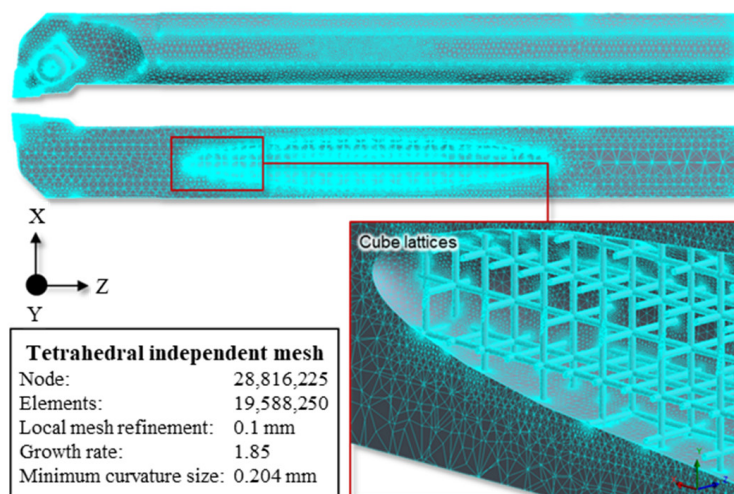


Figure 13. Ellipsoidal hollow with cube lattices.

Figure 14 shows the respective frequencies and the six eigenmodes of the modified boring bar. Compared to the modes of the reference model in Figure 9, the modes of the modified model show a more balanced behavior. The vibration characteristics, especially in modes 3 and 4, are very well balanced compared to the reference model. Overall, it can be seen that the vibrations are very well absorbed and damped by the lattice structure throughout the modified tool.

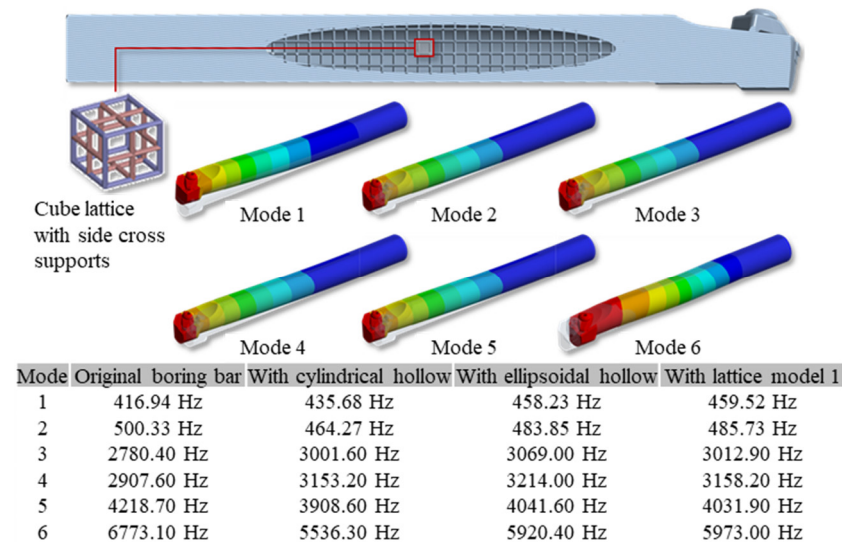


Figure 14. Modal analysis of the modified boring bar.

Figure 15 shows the final results of the HRA. While the amplitude in the E18H3 model was still higher than in the reference model, it has rapidly decreased by 67.69% due to the filling of the lattice structure by lattice model when it is compared to the E18H3 model. The frequency range between the model E18H3 model and the lattice model 1 is almost identical, and the vibration process takes place at 416.94 Hz on the one hand and 459.52 Hz on the other. Compared to the reference model (original boring bar), the amplitude decreases from $y_0 = 10.4 \mu\text{m}/\text{N}$ to $y_0 = 4.62 \mu\text{m}/\text{N}$ with the lattice structure (cube lattice with side cross support). As a result, the vibrations of the boring bar are significantly reduced by 55.58% compared to the reference model (original boring bar).

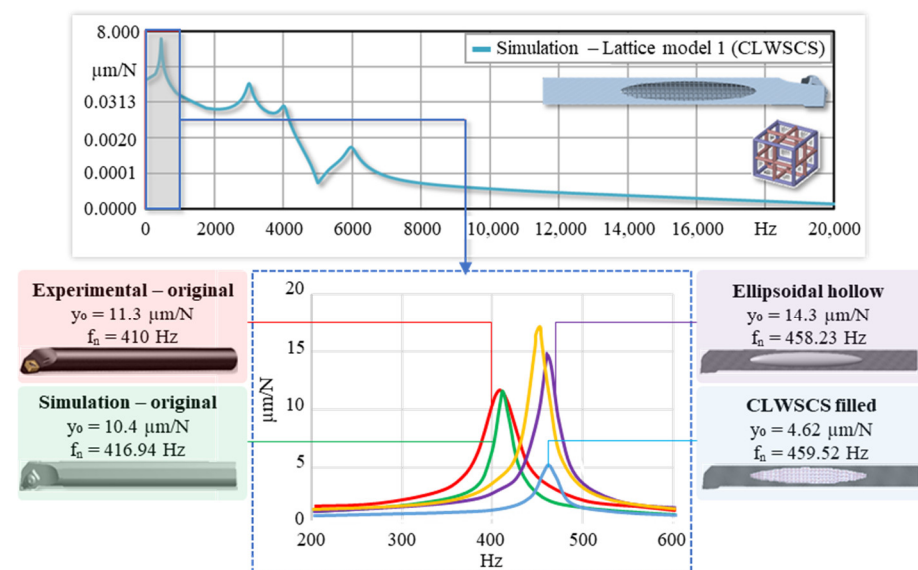


Figure 15. Comparison of the results: comparison of model C18H3 and E18H3 and the effect of lattice structure.

2.7. Testing of Different 3D Lattice Structures

Following this evaluation and the new knowledge gained, further numerical analyses were carried out. Five 3D lattice structures were used for filling the ellipsoidal hollow, which are discussed below (Figure 16).

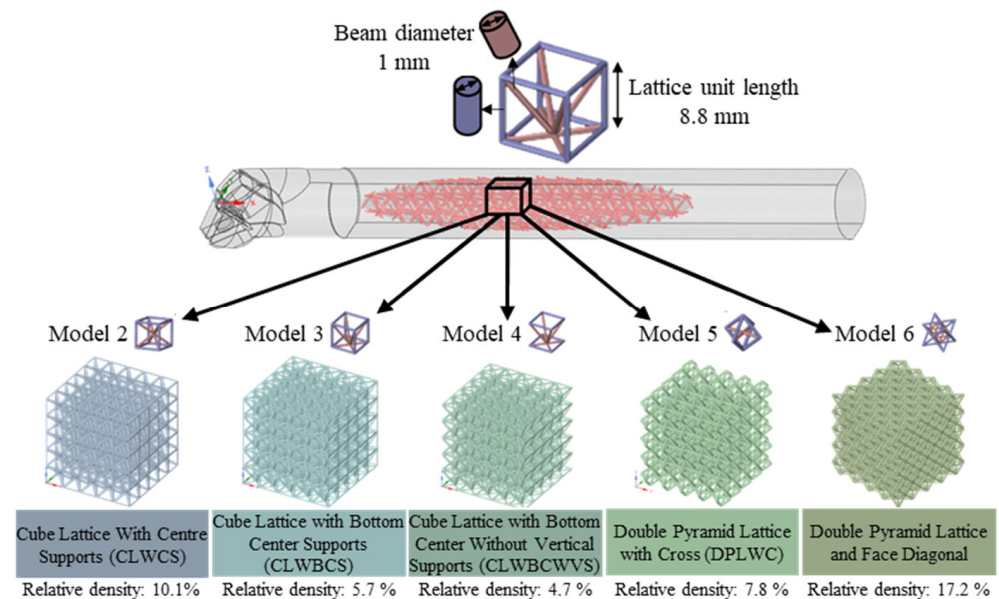


Figure 16. Different 3D Lattices structures for analysis of vibration reduction.

The common feature of these structures is that the rod elements in the 3D lattice structures have a junction point. This junction point is located either at the center or at the base of the center of the 3D lattice structures. During the modelling phase, the beam diameters and lattice unit heights have been kept constant as 1 mm and 8.8 mm, respectively, for all lattice types. However, when the beam diameter and 3D lattice structure height are equalized, the area density of the lattice structure changes and cannot be controlled. Therefore, the relative density for each 3D lattice structure is different.

Figure 17a presents the vibration behavior of all models, including the reference model (original boring bar), experimental results, and the five newly designed 3D lattice structure models. In addition, the vibration behavior of the two hollows (E18H3 (ellipsoidal) and C18H3 (cylindrical)) was also taken into account. All newly simulated models clearly show a remarkably lower vibration behavior. These have been enlarged in the lower image section for detailed analysis. Compared to the already simulated models, the vibration process begins much later in models 2–6, namely at an average of 470 HZ, with an equally significant reduction in amplitude. Model 3 shows an amplitude of $y_0 = 3.35 \mu\text{m}/\text{N}$. Compared to model 1, model 3 shows a further vibration reduction of 28.57% and, compared to the reference model, an improvement of 67.79% was achieved.

For further analysis, the beam diameter of Model 3 was varied from 1 mm by 0.25 mm in two directions (Figure 18). On the one hand, the beam diameter was reduced to 0.75 mm, and on the other hand, it was increased to 1.25 mm. The results show that the reduction leads to lower stability and thus to a higher vibration behavior than with a beam diameter of 1 mm. In contrast, increasing the beam diameter to 1.25 mm shows an improvement of 6.27% compared to 1 mm.

Due to the relative density, this increases from 4.7% (1 mm beam diameter) to 5.9 mm with a beam diameter of 1.25 mm. To maintain these properties (stability and energy absorption) and, above all, the lightweight structure, a further increase in the beam diameter does not appear to make sense.

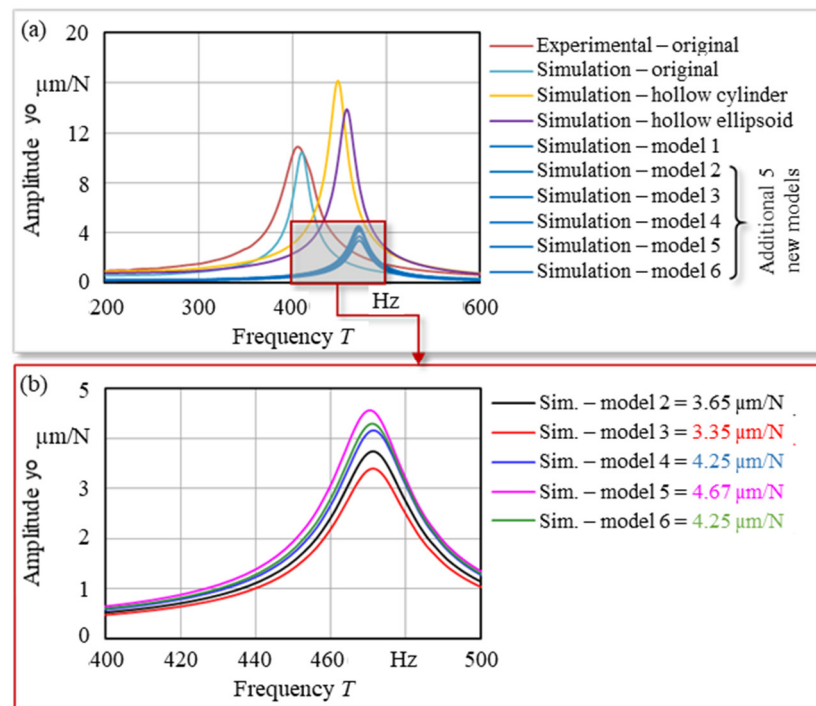


Figure 17. (a) Comparison of the results and (b) the zoomed view.

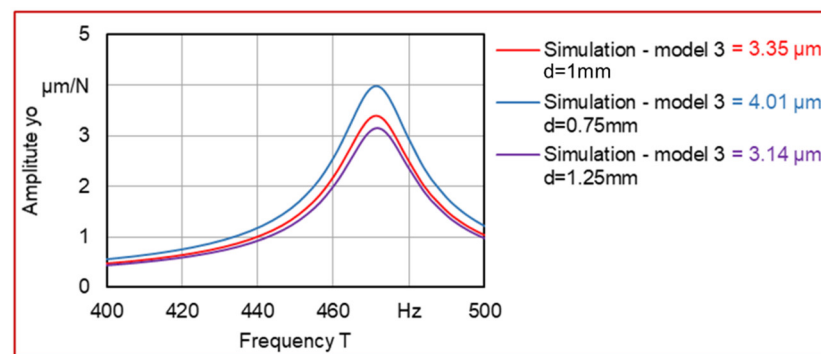


Figure 18. Comparison of two varying beam diameters in model 3.

2.8. Computation Time for the Simulations

The computation time for the FEM simulations depends on the complexity of the algorithms, such as the stiffness and force matrices, as well as the deformation and stress fields. The formation takes place via the nodes and elements as well as the boundary conditions. At each simulation time step, the structure is updated, and the partial differential equations are solved iteratively in three-dimensional space. For each finite element, a set of equations is solved. Neighboring elements have common nodes, and these shift with each deformation. The solver loosens the system of equations until the displacements are available for each node, and this requires large computing resources (Table 4). Compared to the reference model (original boring bar), the number of nodes and elements is understandably reduced for the modified boring bars with hollow space. However, once the ellipsoidal hollow space is filled with 3D cube lattice structures, the number of nodes increases significantly.

Table 4. Computational resources for simulation.

	Node	Elements	Data Size	Simulation Time
High-End Workstation Features	HP Z800 2× CPU Intel Xeon W5580, 3.50 GHz, 8 MB L3, 1333 MHz Memory, 130W32 144 GB (4 × 12 GB; 6 × 16 GB) DDR3-1333 ECC Registered RAM 2-CPU NVIDIA Quadro FX 5800 4 GB PCIe Graphics Card 1110 W 89% (SILVER) Efficient wide-ranging, active Power Factor Correction			
FEM- MODAL/HRA reference model (full)	1855.120	1360.492	47.54 GB	4 h 22 min
FEM- MODAL/HRA cylindrical hollow model	1470.612	960.958	51.55 GB	5 h 38 min
FEM- MODAL/HRA ellipsoidal hollow model	1201.598	798.921	54.29 GB	12 h 59 min
FEM static/mechanical analysis 13 models (see Table 3)	-	-	120.18 GB	32 d 10 h 46 min
FEM-MODAL/HRA ellipsoidal hollow with simulation model 1	28,816.225	19,167.250	277.15 GB	10 d 21 h 27 min
FEM-MODAL/HRA ellipsoidal hollow with simulation model 2	27,972.315	18,245.017	263.27 GB	10 d 10 h 13 min
FEM-MODAL/HRA ellipsoidal hollow with simulation model 3	28,012.064	18,644.344	265.05 GB	11 d 7 h 38 min
FEM-MODAL/HRA ellipsoidal hollow with simulation model 4	29,106.924	19,909.789	281.98 GB	11 d 13 h 3 min
FEM-MODAL/HRA ellipsoidal hollow with simulation model 5	29,181.181	19,960.379	282.23 GB	11 d 33 h 42 min
FEM-MODAL/HRA ellipsoidal hollow with simulation model 6	28,776.861	19,227.129	278.33 GB	11 d 47 h 7 min
FEM-MODAL/HRA ellipsoidal hollow with simulation model 3 d = 0.75 mm	27,004.154	17,991.879	259.85 GB	10 d 1 h 41 min
FEM- MODAL/HRA ellipsoidal hollow with simulation model 3 d = 1.25 mm	29,914.361	20,044.485	291.96 GB	12 d 5 h 52 min

The analyzed models had a very high number of nodes and elements and had to be solved numerically in their entire filling lattice structure. The lattices, which were already very small in size, required a correspondingly very fine structuring for good resolution. At the nodes, the numerical calculation generates a linear system of equations for the characteristic quantities of the structure. Solving the system of equations for the unknowns represents the problem. The calculation of the forces and deformations of the structure triggers a discrete displacement and forces at the nodes. Since an evaluation only takes place at discrete points, intermediate values must be suitably interpolated. The independent displacement possibilities of a node are called degrees of freedom, i.e., in the three-dimensional calculation, a node has three translational degrees of freedom and three rotational degrees of freedom in x, y, and z directions. The total degree of freedom of an FE model is the sum of all node degrees of freedom and determines the computational complexity.

3. Development of a Specific Parametric CAD Software Tool

As previously stated, the process of manually creating lattice structures is notably laborious and time-consuming. Furthermore, it is important to note that designs created during this process are susceptible to inaccuracies in production, which can be attributed

to human error. It is evident that, in instances where the lattice structure is produced on a single occasion, the modelling process becomes unfeasible in scenarios where a multitude of designs are utilized during the study. Consequently, there is a necessity for a tool that can address and eliminate these difficulties. The following sections are devoted to an explanation of the fundamentals and the evolution of a particular parametric CAD software tool.

3.1. Methodology

The lattice unit density [37], dimensions [35], and the orientation [63,64] of the constitutive supports determines the behavior of the lattice structure. For example, supports parallel to the applied force increase the stiffness of the component, while structures perpendicular to this force can provide flexibility. The parameters are adjusted and analyzed using each individual model, which requires very large quantities of models. For this reason, a specific parametric CAD software tool is being developed. The CAD Software SolidWorks 2020 and its Application Programming Interface (API), which contains multiple functions that can be called Visual Basic for Applications (VBA), was used. With this method, five different lattice units were modelled (Table 5).

Table 5. Lattice unit types used for the specific parametric CAD software tool.

#	Type of Lattice Unit	Model	#	Type of Lattice Unit	Model
1	Kagome		4	CLWBCWVS (Cube Lattice with Bottom Centre Without Vertical Supports)	
2	CLWCS (Cube Lattice with Center Supports)		5	DPLWC (Double Pyramid Lattice with Cross)	
3	CLWBCS (Cube Lattice with Bottom Center Supports)				

As the first step of the modelling process, the selected lattice units were defined with all their properties and mathematical models were created accordingly. Then, parametric modelling software was created to be compatible with mathematical models. Finally, the user interface was developed to increase ease of use.

3.2. Mathematical Model of 3D Lattice Structures

The modelling strategy aimed to fill an empty shell volume with a lattice structure. This design space can be selected in the specific parametric CAD software tool as an ellipsoidal or rectangular prism. At the stage of lattice unit multiplication, the lattice structure is allowed to extend beyond the desired volume to avoid surface defects and topological errors. As shown in Table 5, lattice units have different geometric and topological properties. Therefore, the number and orientation of lattice structure must be determined according to the lattice unit and shell parameters (Figure 19).

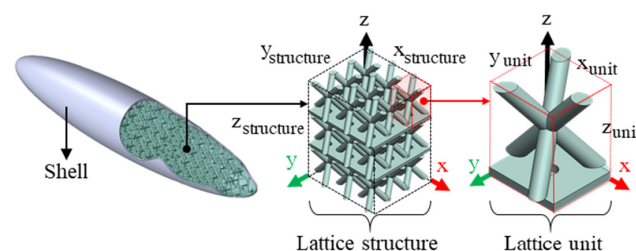


Figure 19. Lattice structure and lattice unit with their outer dimensions in cartesian coordinate system.

In developing the mathematical models, rather than developing separate formulae for each lattice unit, an attempt was made to develop general models that could be applied to all lattice unit types. Therefore, similar steps were taken to determine and calculate lattice units and lattice structure parameters.

The aim was to develop general models that can be applied to all lattice units. Therefore, steps were taken to determine and calculate lattice units and lattice structure parameters. Several parameters have been selected as user-defined inputs and are explained in the following sections. The first is the number of lattice units on the z-axis (Num_z). The value of Num_z is used to calculate the length of the lattice unit in the z-axis (z_{unit}) and the number of lattice units in the x- and y-axes (x_{unit} and y_{unit}). The z_{unit} parameter can be calculated using the following equation:

$$z_{unit} = \frac{z_{structure} - ((Num_z - 1) * z_{layer})}{Num_z} \quad (6)$$

where $Z_{structure}$ is the height of the volume to be filled with lattice units and z_{layer} is the separation layer thickness of the Kagome unit. The value of z_{layer} must be set to zero for other lattice unit types. The separation layer is placed as a unifying interlayer under the supports with the diameter of d_{column} . This layer is defined by the user, as the z_{layer} changes the height of the supports in the z-axis at a fixed z_{unit} (Figure 20).

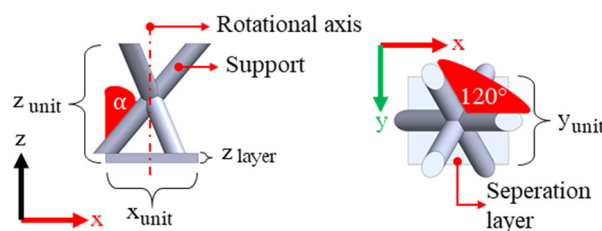


Figure 20. Kagome lattice unit and its features.

Although the height of the lattice unit on the z-axis has been specified, the dimensions on the x- and y-axes need to be determined. There is a relationship between outer dimensions for each type of lattice unit. However, while determining these dimensions, it was necessary to consider the Kagome lattice unit separately from other lattice units. As shown in Figure 20, the Kagome lattice unit consists of three supports that intersect at the center at an angle of 120° around the axis of rotation. When this model is viewed parallel to the z axis, a hexagonal pattern with 60° angles is observed. Thus, the properties of the equilateral hexagon can be used when establishing the relationship between x_{unit} and y_{unit} . Furthermore, these supports are tilted from the z-axis by an angle α . As another user-defined variable, α was supplied to the relevant equations. As a result, the relationship between the outer dimension of the Kagome lattice unit was defined by considering the previously explained α and z_{unit} values as follows:

$$x_{unit} = z_{unit} * \tan(\alpha) \quad (7)$$

$$y_{unit} = x_{unit} * \tan(60^\circ) \quad (8)$$

On the other hand, the values of z_{unit} , y_{unit} , and x_{unit} are the same for all lattice units except Kagome because the geometric shapes of the lattice units are cubic. Cubic lattice units with similar characteristics do not have a separating layer between the lattice unit layers. Considering these characteristics of cubic lattice units, the outer dimensions were redefined by updating Equations (7) and (8) as follows:

$$x_{unit} = y_{unit} = z_{unit} \quad (9)$$

After defining the Kagome structure, the characteristics of the cubic lattice unit types are presented in the following sections. The first cubic lattice unit, CLWCS, consists of supports and a cubic frame. These supports extend from the top corner to the bottom corner in the form of volume diagonals within the cubic frame and are arranged symmetrically around the axis of rotation at 45° (Figure 21).

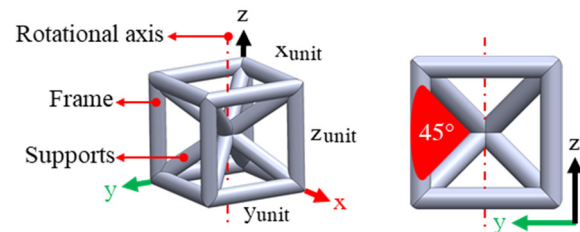


Figure 21. CLWS lattice unit and its features.

Similar to the previous lattice unit, the CLWBCS has a cubic frame. However, the lattice unit contains two different types of supports: main and cross supports. Main supports start from the top corners of the cubic frame and intersect in the center of the xy plane. On the other hand, two diagonal supports on the xy plane represent the cross-support (Figure 22).

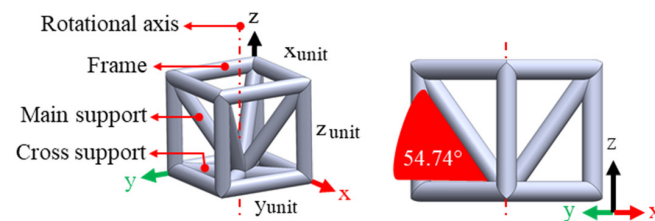


Figure 22. CLWBCS lattice unit and its features.

The structure of CLWBCWVS is almost identical to that of CLWBCS. The only difference between these two structures is the absence of vertical frame elements (Figure 23).

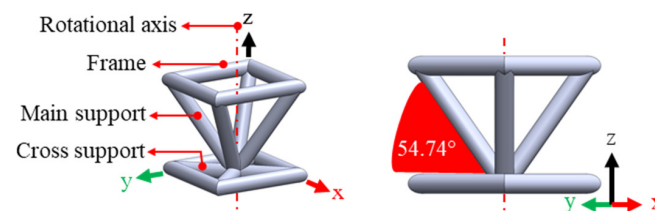


Figure 23. CLWBCWVS lattice unit and its features.

As the last lattice unit type, the DPLWC resembles two pyramids joined from the base. For this lattice unit, instead of a cubic frame, there are two sets of main supports and two cross supports. The main supports are positioned at a 45° angle to the z -axis, while making a rotation of 90° around the axis of rotation. In addition, two transverse (cross) supports are placed on the symmetry plane (Figure 24).

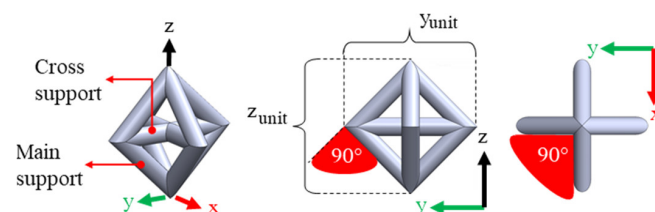


Figure 24. DPLWC lattice unit and its features.

3.3. Lattice Structure Generation

The unique characteristics of each lattice unit were taken into account, and their individual layers and supports were modeled separately. These features were combined to form a complete 3D lattice structure. Since a lattice unit is not capable of any practical use, it must be multiplied in each direction of the Cartesian coordinate system to construct a versatile lattice structure. Two critical improvements were implemented to ensure accurate and seamless connections between lattice units. The first improvement is the merging of lattice unit boundaries during multiplication. In this step, the volumes of the lattice structures were properly merged to prevent the formation of zero-thickness volumes between lattice units (Figure 25).

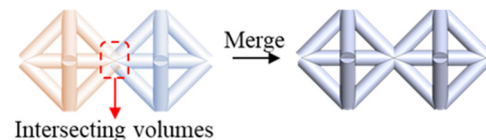


Figure 25. Merging of lattice units.

The second improvement is to create a thin shell around the lattice unit multiplication. This shell has a crucial function as a unifying cover. There may be parts detached from the main lattice unit after the trimming process. Therefore, a 1 mm thick shell is placed around the lattice unit to prevent the loss of these parts and to ensure that they can be used along with the main body (Figure 26). In addition, this shell provides convenience in both merging and positioning the parts while placing the lattice structure into the part where it will be used.

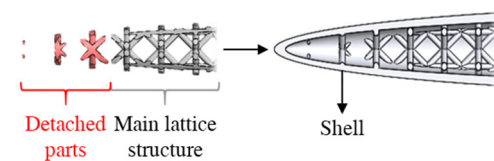


Figure 26. The lattice unit parts detached from the lattice structure.

Using Equations (6)–(9), the values of x_{unit} , y_{unit} , and z_{unit} of all lattice units were calculated. Since the goal is to construct a lattice structure based on the external dimensions of a single lattice unit, the number of units to be replicated along each axis must be determined. While the lattice unit dimensions on the x and y axes are calculated in accordance with the input parameters, the lattice structure dimensions are freely defined by the user. As a result, the calculated lattice unit dimensions and the desired lattice structure size may not always match. Additional lattice structure columns are used to avoid any space between the generated lattice unit array and the lattice structure volume. For this reason, the number of lattice units that can fill the structure volume has been calculated using the modulo operation in mathematics (Figure 27).

The number of lattice units multiplied in the x and y directions is calculated using Equations (10) and (11). After creating a lattice structure with extra features, its unwanted parts are trimmed to obtain a continuous and consistent lattice structure.

$$Num_x = \frac{x_{mesh} - (x_{structure} \bmod x_{unit})}{x_{unit}} + 1 \quad (10)$$

$$Num_y = \frac{y_{structure} - (y_{structure} \bmod y_{unit})}{y_{unit}} + 1 \quad (11)$$

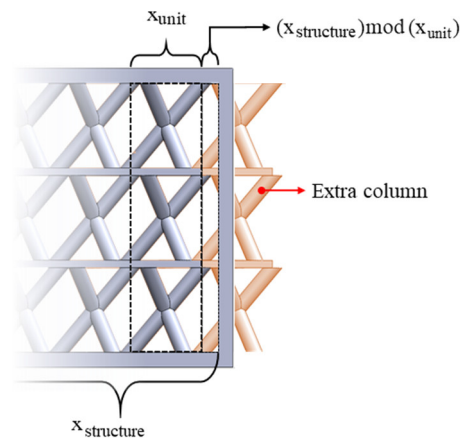


Figure 27. Lattice unit layout approach.

3.4. Modelling in the Specific Parametric CAD Software Tool

The mathematical equations were used to model the lattice units and their structures within the parametric CAD software tool environment. In this step, user-defined inputs and relationships between all parameters were taken into account to obtain a flexible and scalable model. This model considered not only the different lattice units, but also the lattice structures at different levels. This allows the selected volume to be filled with the preferred lattice unit type and size. However, at this stage, it is necessary to intervene in the software code used for modelling, which can lead to various errors. A user interface has been developed to overcome these problems. This has resulted in the development of an effective and visual parametric modelling method, the details of which are explained in the following sections.

3.5. Computer Aided Parametric Modelling

A technique similar to microservice architecture is used throughout the development of the parametric CAD software tool. This approach used small parts of code to perform specific tasks throughout the modelling operations. These code parts were divided into five groups:

- lattice unit generation,
- multiplication,
- shell generation,
- trimming, and
- scaling.

Initially, the software generates a selected lattice unit type using the parameters entered by the user. The Kagome lattice unit requires six user-defined inputs ($x_{structure}$, Num_z , d_{column} , d_{hole} , Z_{layer} , and α) to define its geometric properties, while three parameters ($x_{structure}$, Num_z , and d_{column}) are sufficient for other cubic lattice units. The subsequent code section replicates the lattice units along the x, y, and z axes. To determine the number of lattice units to be used in the multiplication process, it is necessary to calculate Num_x and Num_y in addition to the Num_z value previously entered by the user. Using values derived from the mathematical equations in Sections 3.2 and 3.3, the lattice units are arrayed to form the raw lattice structure. To facilitate subsequent operations, the resulting structure (raw lattice structure) is placed at the origin of the cartesian coordinate system. Following the arraying process, the lattice structure may exceed the target dimensions. It is therefore necessary to trim the raw lattice structure into a finer state. However, in order to avoid surface defects and to maintain the integrity of the structure, the structure is enveloped by a shell according to the selected structure volume type: rectangular prism or

ellipsoid. User-defined values are used to create the shell. The trimming process removes unnecessary parts to achieve the desired structure type. However, after the trimming process, the lattice structure has the correct shape but the wrong size. This problem is caused by an incompatibility between the units of two development environments. A unit of a value entered as millimeters via the API is converted to meters in SolidWorks. The result is a solid model that is much larger than the desired size. This problem is overcome by scaling to 1/1000 as a final operation. As a result, a consistent and flexible code structure was created by combining discrete blocks of code that perform specific operations. Figure 28 shows the flowchart of the software developed. This flowchart was created by considering the BPMN model, where:

- Start Event (●) indicates the start of the process;
- The exclusive gateway (⬢) directs sequence flows to only one of the outgoing branches. When merging flows, the exclusive gateway waits for an incoming branch to complete before continuing the flow;
- The parallel gateway (⬢) directs the process path down multiple outbound sequence flows to be followed in parallel. When merging flows, the parallel gateway acts as a synchronization mechanism, waiting for all branches to complete before continuing the flow; and
- End event (⦿) indicates where a path will end.

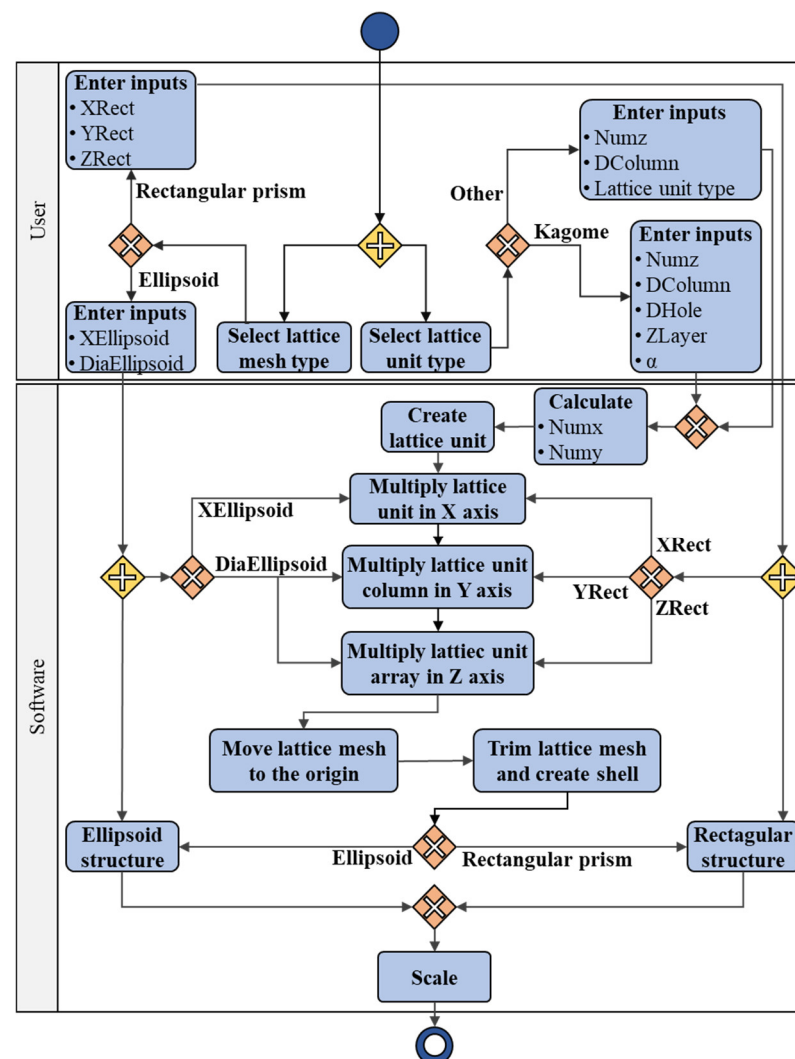


Figure 28. Flowchart of the parametric CAD software tool functions.

3.6. The User Interface

While the parametric and automatic modelling capability for the desired lattice structures was achieved, a user-friendly interface was needed to facilitate the developed parametric CAD software tool for the user. A user interface was developed with the principle of minimum user interaction, and a button was placed on the SolidWorks toolbar for easy access to the user interface (Figure 29).

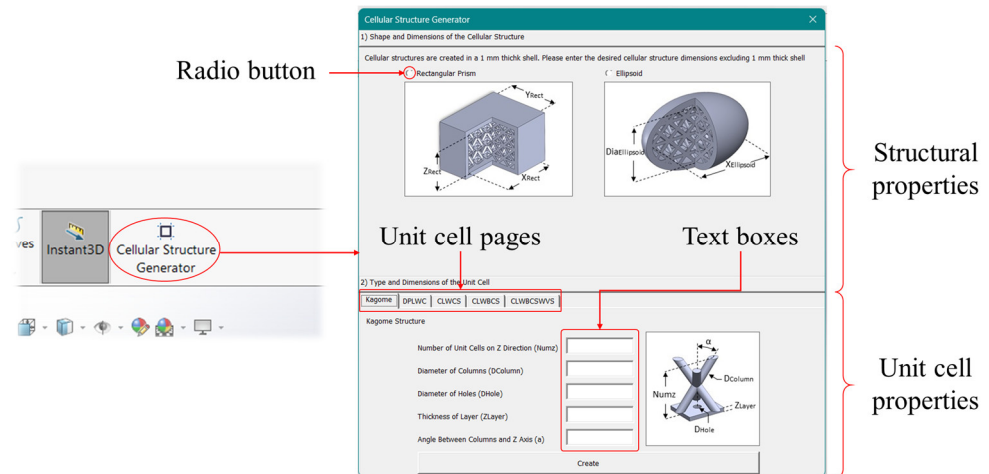


Figure 29. User interface for the parametric modelling method.

The interface consists of two main sections. To make the necessary selections and complete the data input, the required buttons and text boxes have been embedded in the relevant sections of the interface. In the upper section, where the structural properties can be defined, there are two shell type options, specified as rectangular prism and ellipsoid. For guidance, the user is initially presented with two options for selecting the shell structure using radio buttons placed above the relevant visual. Once the selection has been made by clicking on the radio buttons, the input parameters of the relevant shell type volume become visible and allow data to be entered (Figure 30).

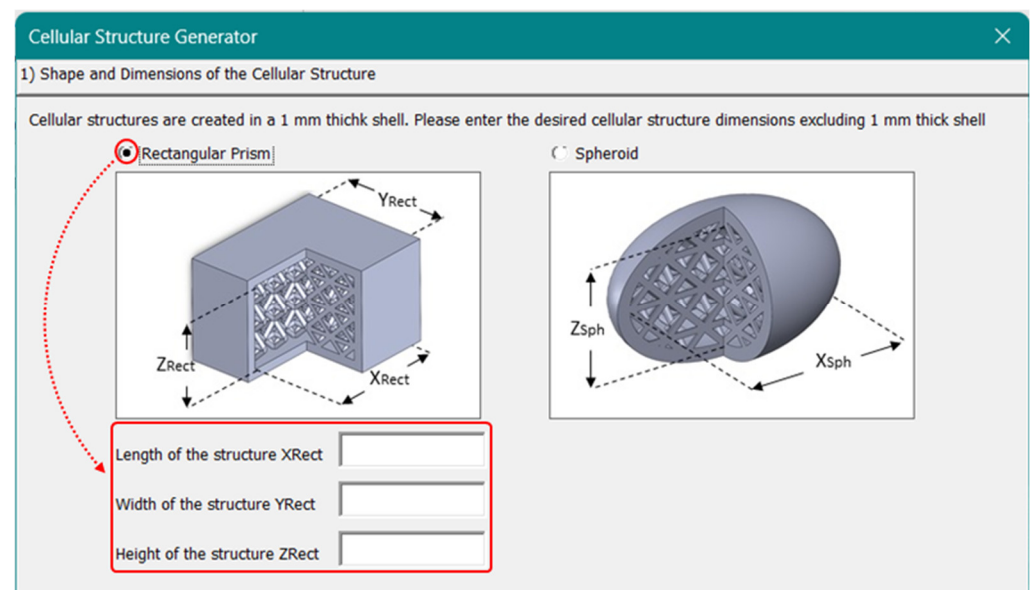


Figure 30. Selection of the shell type and size.

As mentioned earlier, the lattice structure is placed inside a 1 mm thick shell to avoid topological and geometric errors. The shell type and its parameters, defined by the user,

determine the volume type and dimensions to be filled by the lattice units. The shell thickness is excluded from these defined dimensions. Once the shell type has been defined, the lattice unit parameters can be defined using the second section of the interface. This section allows the user to select the desired lattice unit type using predefined lattice unit pages. A tab has been prepared for each lattice unit type and its data input components have been placed in the appropriate tabs. When a lattice unit tab is selected, the user inputs the required parameters in the designated text boxes and clicks the “Create” button to automatically generate the final lattice structure. The appropriate text boxes have been constructed separately for each lattice unit type in the relevant tabs (Figure 31).

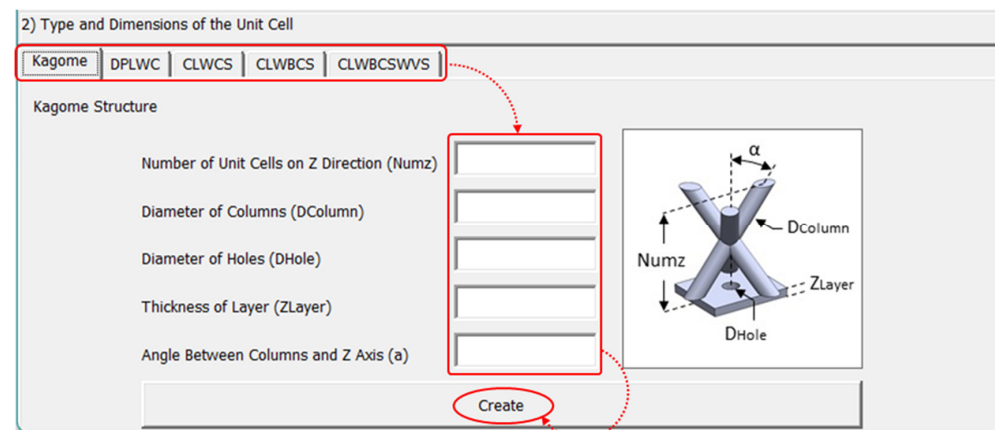


Figure 31. Selection of lattice unit type and size.

4. Discussion

This study, which sought to passively suppress the chatter, was conducted in two significant stages. In the initial stage, the hammer impact test was conducted using the original boring bar to ensure the reliability of the simulation. A comparison of the obtained results with the simulations revealed an approximate 8% variation in amplitude. This negligible discrepancy led to the conclusion that the simulation model exhibited adequate precision. Consequently, the simulation model, employed as the fundamental framework for this study, has been validated. To utilize the lattice structure for the purpose of enhancing the dynamic performance of the boring bar, it is first necessary to create a hollow geometry within the boring bar. The behavior and strength of the boring bar against both dynamic and static loads are affected by this space. This is due to the fact that the change in the internal cross-section and its length also changes the surface moments of inertia. Consequently, this results in a decrease in static stiffness and an increase in natural frequency. It is therefore essential to first ascertain the geometries and subsequently define the dimensional limits of these geometries. In this study, cylinder and ellipsoid shapes were utilized as the two hollow geometries of choice, a decision that was made due to the high strength they provide. These geometries, which are radially symmetrical about the boring bar axis, can be defined with two parameters: width and diameter. Consequently, a total of 12 designs were created for each geometry by utilizing 4 distinct diameter values (12, 15, 18, and 20 mm) and 3 different length values (70, 100, and 130 mm). Subsequently, an examination was conducted of the predicted bending stresses that would occur in the designs. It was determined that the design dimensions with a diameter of 18 mm and a length of 130 mm, which offer the most volume among the acceptable designs (has a maximum bending stress below the allowable maximum = 183.3 MPa), were to be selected. An investigation was conducted to examine the damping and frequency values of the ellipsoid and cylindrical hollow geometries created using specific dimensions. The results demonstrated significant differences in both damping and frequency values. The frequency value of the first mode,

which was approximately 410 Hz for the original boring bar, increased to over 430 Hz with the incorporation of the cylindrical hollow. In the case of the ellipsoid hollow, this value reached around 450 Hz, representing an approximate 10% increase. A very different result was obtained for the amplitude values. When the cylindrical hollow was placed inside the boring bar, the vibration amplitude increased from approximately $10.4 \mu\text{m}/\text{N}$ to $16.7 \mu\text{m}/\text{N}$. However, the results were slightly improved by the use of the ellipsoid hollow as opposed to the cylindrical hollow, with the ellipsoid hollow performing approximately 14% better than the cylindrical hollow and reducing the amplitude value to $14.3 \mu\text{m}/\text{N}$. Although the results seem to have deteriorated, this is quite expected. A hollow created at the free end of the boring bar causes the mass, and therefore the inertia, to decrease. This increase in natural frequency is accompanied by a decrease in the damping ability and the strength of the boring bar due to the removal of material, resulting in an increase in amplitude value. However, this situation can be enhanced by incorporating a lattice structure within the hollow. The ellipsoid hollow, selected due to its superior dynamic performance, was filled with a cube lattice that featured side cross supports. The lattice unit height and column diameter were selected as 5.94 mm and 1 mm, respectively. The collective structure of the lattice units provided significant dynamic stiffness to the boring bar by affecting its mass and, consequently, its natural frequency to a minimum extent. The result was an amplitude value of $4.62 \mu\text{m}/\text{N}$, representing a 55.58% improvement in amplitude value compared to the original boring bar. The findings of this study led to the modelling of five additional lattice structures, with the ellipsoidal hollow being filled. Of the total six lattice structures, model 3, with a 1.25 mm beam diameter, exhibited the most optimal results, with a 69.81% reduction in vibrations compared to the original boring bar. This outcome demonstrates that the configuration of the columns and supports that form the lattice units exerts a significant influence on the dynamic performance. The proposed approach provides a flexible, scalable foundation for future research, including experimental validations and optimization studies on lattice-integrated tool holders. The integration of novel hollow geometries and lattice types expands the applications of lattice structures beyond conventional uses, such as in boring bars, to encompass various tool holders (Figure 32) and diverse components in different domains.

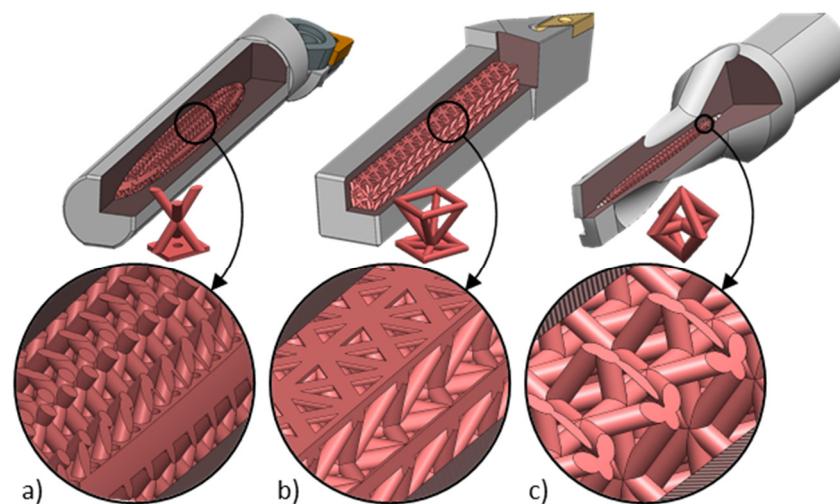


Figure 32. Potential use of cellular structures in the manufacturing industry: (a) boring bar with Kagome lattice unit structure; (b) tool holder with CLWBCWVS lattice unit structure; and (c) indexable insert drill with DPLWC lattice unit structure.

In the second stage of the study, a modelling tool was utilized throughout the research process, rather than being a result that emerged at the conclusion of the study. In the creation phase of the tool, lattice units with all their parameters were initially considered,

and mathematical models were subsequently created. Utilizing the mathematical models, a macro was coded over the SolidWorks API to perform the modelling process. The objective was to minimize the number of steps required to create the desired model. Various precautions were taken to prevent topology errors, and the modelling process was performed automatically without any issues. Although the modelling process was automated at this stage, an interface was required where parameters could be selected for long-term use. The final part of the modelling tool was the design of the user interface. The tool's benefits extend beyond the creation of the desired lattice structure. The integration of automatic modelling techniques offers significant advantages in design processes, particularly in reducing the time spent on manual modelling. The minimization of modelling steps has been demonstrated to result in expedited outcomes, thereby enhancing the efficiency of the overall modelling process. This automation also helps to prevent user errors that often arise during repetitive manual tasks, ensuring the production of standardized models. Furthermore, the measures implemented within the system address critical issues such as zero-thickness volumes, insufficient lattice units, and topology errors, which can otherwise complicate the design process. A significant advantage of the developed interface is that it functions as a protective barrier between the user and the source code, thereby preventing unintended errors. Furthermore, the system enables users to select parameters with ease, thereby ensuring the creation of lattice structures that are fully compliant with hollow geometry requirements. Finally, the integration of various hollow geometries and lattice unit types into the parametric modelling tool enhances flexibility and enables the design of more complex, customized structures. It is evident that these advancements have a significant impact on the modelling process, enhancing its efficiency and precision. Furthermore, they empower the user with enhanced control, contributing to an optimized experience.

5. Conclusions

While this study primarily focuses on the development and validation of a simulation framework and modelling tool, it is intended to serve as a basis for future fundamental investigations into the optimization of lattice structures for vibration suppression in machining applications. To this end, a new methodology for integrating lattice structures into boring bars for passive chatter suppression was proposed, simulated, and evaluated. The key outcomes of the research can be summarized as follows:

- A validated FEM-based simulation model was developed, accurately predicting the dynamic behavior of the boring bar.
- In comparison with the cylindrical hollow, the optimal design was determined as ellipsoidal hollow shape by considering natural frequency and decrease in vibration amplitude.
- Six different lattice structures were designed, modelled, and analyzed, with the best-performing design achieving up to a 69.81% reduction in vibration amplitude.
- A dedicated parametric CAD simulation tool was developed to enable rapid and error-free generation of complex lattice structures.

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Nomenclature

Abbreviations

2D	Two Dimensional	FE	Finite Element
3D	Three Dimensional	FEM	Finite Element Method
AM	Additive Manufacturing	GUI	Graphical User Interface
API	Application Programming Interface	HRA	Harmonic Response Analysis
BPMN	Business Process Model and Notation	ISO	International Organization for Standardization
C	Carbon	Mn	Manganese
CAD	Computer Aided Design	NURBS	Non-uniform Rational Basis Spline
CLWBCS	Cube Lattice with Bottom Centre Supports	P	Phosphorus
CLWBCWVS	Cube Lattice with Bottom Centre Without Vertical Supports	S	Sulphide
CLWCS	Cube Lattice with Centre Supports	Si	Silicon
CNC	Computer Numeric Control	TPMS	Triply Periodic Minimal Surfaces
Cr	Chromium	V	Vanadium
DAQ	Data acquisition	VBA	Visual Basic for Applications
DPLWC	Double Pyramid Lattice with Cross		

Latin Letters

A	Cross sectional area	Num_y	Number of lattice units in the y -axis direction
a_p	Depth of cut (mm)	Num_z	Number of lattice units in the z -axis direction
C	Damping matrix	V_c	Cutting speed (m/min)
d	Diameter (mm)	$x_{structure}$	Length of the lattice structure in x -axis direction (mm)
d_{column}	Diameter of the columns (mm)	x_{unit}	Length of the lattice unit in x -axis direction (mm)
d_{hole}	Diameter of the holes (mm)	y_0	Vibration amplitude ($\mu\text{m}/\text{N}$)
E	Modulus of elasticity	$y_{structure}$	Length of the lattice structure in y -axis direction (mm)
e^j	Sinusoidal motion	y_{unit}	Length of the lattice unit in y -axis direction (mm)
F	Force (N)	z_{layer}	Thickness of the separation layer between the lattice rows (mm)
f	Feed (mm/rev)	$z_{structure}$	Length of the lattice structure in z -axis direction (mm)
K	Stiffness matrix	z_{unit}	Length of the lattice unit in z -axis direction (mm)
L	Length (mm)	T	Vibration frequency (Hz)
L/d	Slenderness ratio	t	Time (s)
Num_x	Number of lattice units in the x -axis direction		

Greek Letters

α	Angle between a support and vertical axis for kagome lattice ($^\circ$)	σ_b	Bending stress (MPa)
θ	Phase shift	ψ	Force shift
ρ	Density (kg/m^3)	Ω	Excitation frequency
σ	Stress (MPa)		

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