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LATTICE STRUCTURES FOR MATERIAL EFFICIENCY: A COMPARATIVE FINITE ELEMENT ANALYSIS

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Abstract: *This study explores the use of lattice structures for optimizing material efficiency using a cubic part as an example. The main goal of the research is to evaluate the impact of these structures on material consumption, comparing them with the solid structure of the part, using static FEA (Finite Element Analysis) simulations. By performing comparative simulations, the mechanical performance and behavior of lattice structures under certain loading conditions are analyzed. Therefore, this study shows the significant potential of lattice structures in reducing the weight and material required, while maintaining structural integrity in some cases, and provides detailed understanding of the advantages and limitations of using these structures, ensuring a starting point for specialists who want to optimize mechanical parts in various application areas.*

Keywords: *lattice structures, Finite Element Analysis (FEA), material efficiency, additive manufacturing.*

1. INTRODUCTION

In the context of the advancement of additive manufacturing (AM) technologies and the growing need to optimize material consumption and reduce waste generated globally, lattice structures have become a promising solution for the lightweight design, material-efficient design or structural optimization for AM [1]. These structures are characterized by periodic and complex geometries that can significantly reduce the weight of parts while striving to maintain suitable mechanical properties for the environment in which the respective part will operate.

Due to their flexibility and versatility, lattice structures have begun to be used in a variety of fields, including the aerospace, medical and automotive industries, where the strength-to-weight ratio is essential.

Lattice structures are repetitive three-dimensional networks composed of cellular units that can have different geometries, thus, influencing the mechanical performance of the components in which they are integrated [2]. They can be classified according to the type of cell used and each type of network presents specific advantages regarding stress distribution, energy absorption and structural stiffness. In this study, six typologies of cubic unit cell lattice structures were investigated, each defined by a distinct cross-sectional geometry [3]: circular, hexagonal, square, triangular and elliptical:

- *Circular-profile cubic unit cell lattice* (Fig. 1,*a*), a structure formed by the intersection of two cylindrical elements along orthogonal axes, ensuring isotropic mechanical behavior and a balanced combination of strength and deformation capacity.
- *Hexagonal-profile cubic unit cell lattice* (Fig. 1,*b*), composed of hexagonal prismatic elements intersecting perpendicularly, providing enhanced stiffness and efficient load transfer under multi-directional stresses.
- *Square-profile cubic unit cell lattice* (Fig. 1,*c*), built from orthogonal beams with square cross-sections, characterized by uniform stress distribution and an improved stiffness-to-weight ratio.
- *Triangular-profile cubic unit cell lattice* (Fig. 1,*d*), consisting of intersecting prismatic geometry with triangular profiles, offering reduced material consumption and good energy absorption capability under dynamic loading.
- *Elliptical-profile cubic unit cell lattice* (Fig. 1, *e*), a geometry defined by smoothly intersecting elliptical struts, which promotes gradual stress transition, superior damping capacity, and high impact resistance.
- *Star-profile cubic unit cell lattice* (Fig. 1,*f*), a geometry generated by the orthogonal intersection of concave star-shaped surfaces, creating a smooth and continuous curvature.

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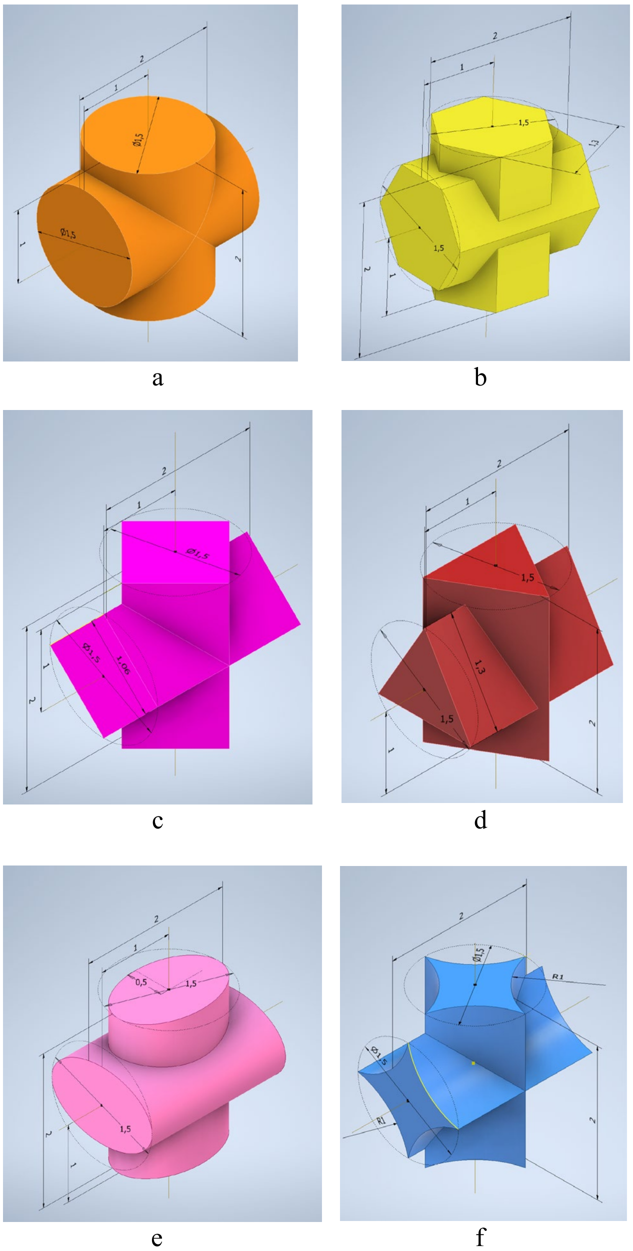

Fig. 1. Lattice cell structures used in this study of cubic unit cell lattice: *a* – Circular-profile; *b* – Hexagonal-profile; *c* – Square-profile; *d* – Triangular-profile; *e* – Elliptical-profile; *f* – Star-profile.

The designed cubic unit cell lattice typologies (circular, hexagonal, square, triangular, star and elliptical) were employed for material optimization and subsequently subjected to finite element analysis (FEA) to investigate their structural and mechanical response. For this purpose, each of these lattice configurations was integrated into the same reference part and compared with its solid variant. The FEA simulations were performed to evaluate the stress distribution, deformations and structural stiffness of each type of network.

Lattice structures make it possible to obtain lighter components with reduced material consumption while attempting in some cases not to significantly compromise mechanical performance. This has major implications in applications where weight reduction is important, such as aviation and the automotive industry, where every gram saved can contribute to lower fuel consumption and increased energy efficiency [4].

Comparing the mechanical performance of different lattice geometries provides a clear perspective on the advantages of each type of structure and helps identify the most suitable configurations for specific applications [5]. By using FEA, it is possible to determine which of these structures offers the best compromise between stiffness, strength and material efficiency [6]. This approach is important for designing innovative components that meet increasingly demanding industrial requirements.

Therefore, the analysis not only provides a systematic method for comparing lattice structures but also highlights the potential of these geometries in engineering innovation. The adoption of such structures in industrial production can reduce costs, improve mechanical performance and open new opportunities for future applications of AM [7].

2. MATERIALS AND METHODS

2.1. Model generation and lattice structures

The comparative analysis investigated six distinct volumetric lattice structures and one fully dense control component (the solid cube). All virtual models possessed uniform macro-dimensions (30 × 30 × 30 mm) and were subjected to identical loading scenarios to ensure comparability. The lattice structures analyzed integrate unit cells characterized by their internal profile geometry: circular, hexagonal, square, triangular and elliptical. An additional structure featuring a custom star pattern was also incorporated into the study.

The computer-aided design (CAD) models, including the complex internal lattice geometries, were generated and analyzed using Autodesk Inventor Version: 2025. The different internal topology resulted in different mass reductions across the virtual samples (e.g., the circular-profile cubic unit cell lattice having a physical mass of 18.6149 g and the elliptical-profile cubic unit cell lattice having 14.2926 g), contrasted with the solid cube reference model at 28.62 g (Table 1).

Table 1

Comparative mass analysis of cubic unit cell lattice models and solid reference structure

CAD Model	Structure Type	Total Mass (g)
Circular-profile cubic unit cell lattice	Lattice	18.6149
Hexagonal-profile cubic unit cell lattice	Lattice	19.0283
Square-profile cubic unit cell lattice	Lattice	16.9631
Triangular-profile cubic unit cell lattice	Lattice	14.3168
Elliptical-profile cubic unit cell lattice	Lattice	14.2926
Star-profile cubic unit cell lattice	Lattice	13.8426
Solid cube	Solid	28.62

Table 2

ABS material properties (Autodesk Inventor Material Library)

Property	Value	SI
Mass Density	1.06	g/cm^3
Young's Modulus (E)	2.24	GPa
Poisson's Ratio (ν)	0.36	ul
Yield Strength (σ_Y)	20	MPa
Ultimate Tensile Strength (σ_{UTS})	29.6	MPa
Shear Modulus	811.594	MPa

2.2. Material specifications

All components were modeled using isotropic material properties corresponding to ABS (Acrylonitrile Butadiene Styrene) plastic (*Autodesk Inventor Material Library*). The mechanical properties used for the finite element simulation are summarized in Table 2, representing common reference values for this material.

2.3. Finite element analysis

The mechanical behavior of the structures was assessed using static analysis. A linear elastic model was used, evaluating stresses and deformations under sustained load conditions (1000 N and 3000 N).

Given the built-in complex geometry of lattice structures, a refined mesh was defined across all models to accurately capture stress gradients, particularly at the nodes and struts. Important meshing parameters maintained consistently across all simulations:

- Average element size: 0.1 (as a fraction of the model size);
- Minimum element size: 0.2 (as a fraction of the average size);
- Grading factor: 1.5;
- Maximum turn angle: 60°
- The solver enabled the creation of curved mesh elements, which is typically necessary for accurate representation of complex surfaces using high-order solid elements (e.g., tetrahedral elements, common in volumetric studies).

To simulate the mechanical constraints representative of a simple component under compressive loading, identical boundary conditions were applied to all six structural variants:

1. Fixed constraint: One bottom face was assigned as a fixed constraint, restricting all translational (X, Y, Z) and rotational degrees of freedom.
2. Applied load: Two primary compressive load cases were investigated, acting uniformly upon the opposing top face of the cube, directed along the negative Y-axis:

- *Case* 1: magnitude of 1000 N (uniformly distributed force).
- *Case* 2 (high load): magnitude of 3000 N (uniformly distributed force).

The output analyzed included Von Mises Stress, Displacement (deformation) and Equivalent Strain.

2.4. Comparative analysis approach

The methodology used a comparative design approach wherein the mechanical responses of the six lattice geometries were directly benchmarked against the fully dense cube, solid model. Since the macro-geometry and boundary conditions (loads and constraints) remained invariant across all models, differences observed in stress magnitude, stiffness (inferred from displacement) and material efficiency (mass) were attributed solely to the topology of the internal lattice structure. The safety factor calculations adhered to the yield strength criterion of the ABS plastic (σ_Y=20 MPa).

3. RESULTS

The FEA investigated the mechanical response of six different lattice topologies and one solid reference model under static compressive loading conditions of 1000 N and 3000 N. All structures were composed of ABS Plastic, which possesses a nominal yield strength of 20 MPa.

3.1. Mechanical efficiency and performance comparison

The mechanical performance, assessed primarily through maximum Von Mises stress (σ_{vM}) as shown in Fig. 2 and Fig. 3, and minimum Safety Factor (SF_{min}), varied significantly across the lattice geometries. Under the 1000 N load case, the circular-profile cubic unit cell lattice demonstrated the highest mechanical efficiency relative to its material usage (mass: 18.6149 g). This topology resulted in the lowest maximum Von Mises stress (~6.08 MPa) and the highest minimum safety factor (~3.29) among all tested lattice structures.

Alternatively, the triangular-profile cubic unit cell lattice and star-profile cubic unit cell lattice topologies exhibited the least structural efficiency for a given volume, registering the highest maximum stresses and lowest safety factors. At 1000 N, the triangular-profile cubic unit cell lattice model experienced a maximum Von Mises stress of ~21.67 MPa and a minimum safety factor of ~0.93. Similarly, the star-profile cubic unit cell lattice model registered a maximum stress of ~21.34 MPa and a minimum safety factor of ~0.94. Table 3 summarizes the maximum Von Mises stress values obtained for all lattice configurations and the solid model under 1000 N and 3000 N loading conditions.

Table 3

Maximum Von Mises stress comparison of lattice and solid structures at two different loads

CAD Model	Maximum Von Mises Stress [MPa] (1000 N load)	Maximum Von Mises Stress [MPa] (3000 N load)
Circular-profile cubic unit cell lattice	6.076	18.227
Hexagonal-profile cubic unit cell lattice	6.90151	20.7045
Square-profile cubic unit cell lattice	8.75392	26.2618
Triangular-profile cubic unit cell lattice	21.6694	65.0083
Elliptical-profile cubic unit cell lattice	9.993	29.974
Star-profile cubic unit cell lattice	21.3391	64.0173
Solid cube	2.77455	8.324

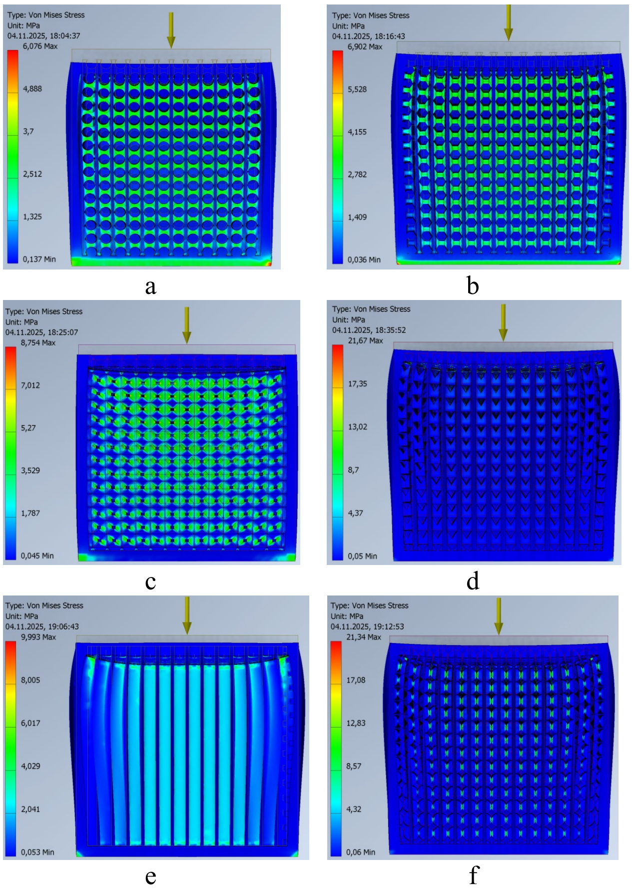

Fig. 2. Maximum Von Mises stress results of cubic unit cell lattice (1000 N load): *a* – Circular-profile; *b* – Hexagonal-profile; *c* – Square-profile; *d* – Triangular-profile; *e* – Elliptical-profile; *f* – Star-profile.

3.2. Comparison to solid reference model

All lattice structures displayed markedly higher peak stresses and displacements compared to the solid cube reference model. The total displacements recorded for each lattice configuration and the solid reference model at 1000 N and 3000 N load cases are summarized in Table 4.

Table 4

Maximum displacement comparison of lattice and solid structures at two different loads

CAD Model	Displacement [mm] (1000 N load)	Displacement [mm] (3000 N load)
Circular-profile cubic unit cell lattice	0.02857	0.08571
Hexagonal-profile cubic unit cell lattice	0.03105	0.09314
Square-profile cubic unit cell lattice	0.03942	0.1183
Triangular-profile cubic unit cell lattice	0.06092	0.1828
Elliptical-profile cubic unit cell lattice	0.04911	0.1473
Star-profile cubic unit cell lattice	0.06277	0.1883
Solid cube	0.01487	0.0446

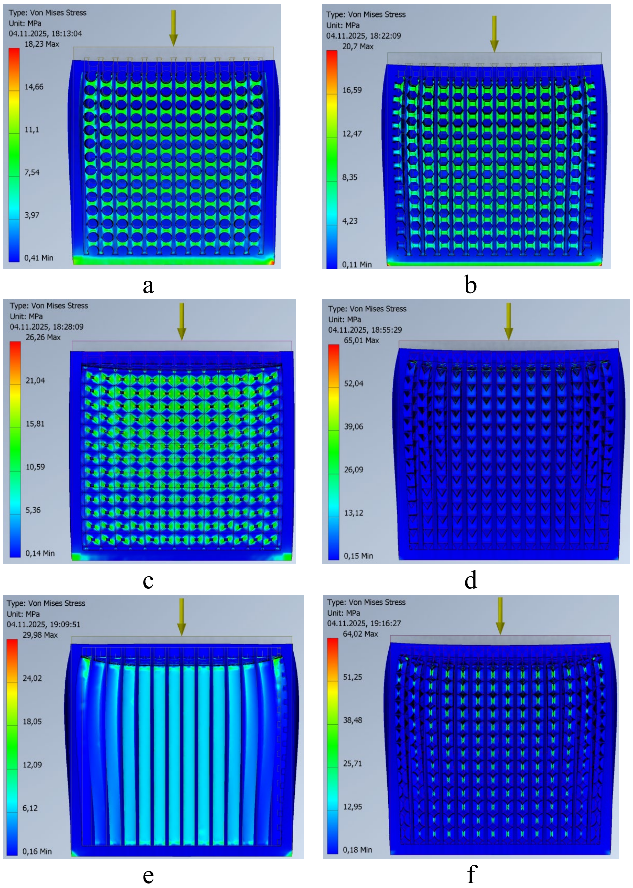

Fig. 3. Maximum Von Mises stress results of cubic unit cell lattice (3000 N load): *a* – Circular-profile; *b* – Hexagonal-profile; *c* – Square-profile; *d* – Triangular-profile; *e* – Elliptical-profile; *f* – Star-profile.

The solid reference model (mass: 28.62 g) demonstrated superior stiffness and integrity, registering a maximum Von Mises stress of only ~2.77 MPa and a peak displacement of ~0.015 mm under the 1000 N load. This performance resulted in a resilient minimum safety factor of 7.25. The maximum Equivalent Strain results for the lattice and solid structures at both loading levels are summarized in Table 5.

Table 5

Maximum Equivalent Strain comparison of lattice and solid structures at two different loads

CAD Model	Equivalent Strain [ul] (1000 N load)	Equivalent Strain [ul] (3000 N load)
Circular-profile cubic unit cell lattice	0.002513	0.007538
Hexagonal-profile cubic unit cell lattice	0.002952	0.008856
Square-profile cubic unit cell lattice	0.003645	0.01093
Triangular-profile cubic unit cell lattice	0.009256	0.02777
Elliptical-profile cubic unit cell lattice	0.004176	0.01253
Star-profile cubic unit cell lattice	0.008868	0.02661
Solid cube	0.001197	0.00359

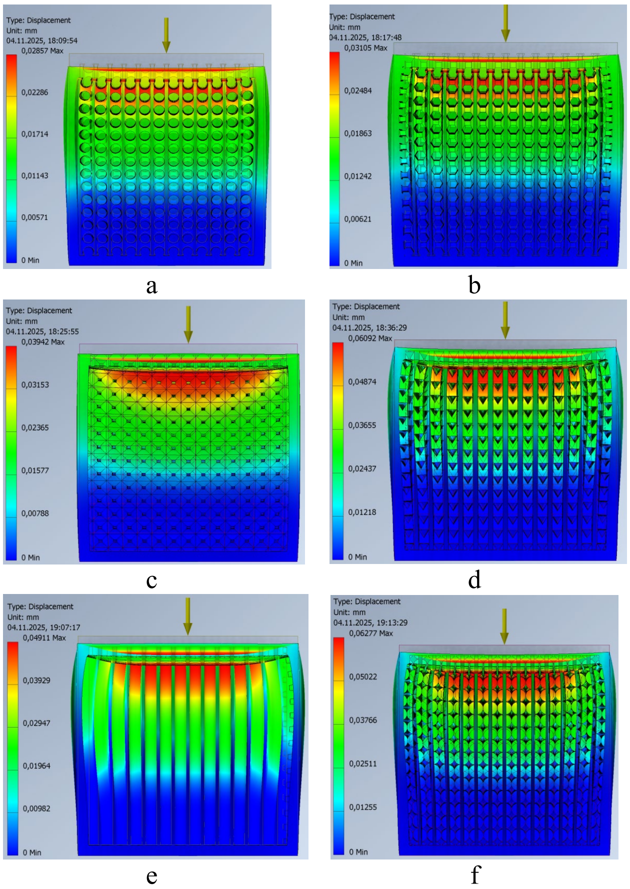

Fig. 4. Maximum displacement of cubic unit cell lattice (1000 N load): *a* – Circular-profile; *b* – Hexagonal-profile; *c* – Square-profile; *d* – Triangular-profile; *e* – Elliptical-profile; *f* – Star-profile.

On the other hand, even the best-performing lattice, the circular-profile cubic unit cell lattice type, showed stresses approximately 2.2 times higher (~6.08 MPa) and displacements almost double (~0.029 mm) compared to the solid cube under the same load. The least efficient lattices (star and triangular profiles cubic unit cell lattices) experienced peak stress nearly 8 times greater than the solid cube (~21.67 MPa for triangular profile vs ~2.77 MPa for the solid cube model).

Under the severe 3000 N load, the trend amplified: the solid cube maintained a minimum safety factor of 2.4, whereas the triangular and star profile lattices failed or operated well below safe conventional limits, dropping to minimum safety factors of ~0.308 and ~0.312.

3.3. Stiffness and material efficiency trade-off

The study highlights a direct, inverse relationship between material efficiency (mass reduction) and structural stiffness (resistance to displacement). The lattice models achieved substantial material reduction, with the star and triangular-profile cubic unit cell lattice models being the lightest (13.8426 g and 14.3168 g), representing approximately 51–50% mass saving compared to the solid cube (28.62 g).

However, the structures that yielded the greatest mass reduction (star, triangular and elliptical profiles) consistently suffered the greatest loss in stiffness, exhibiting the highest maximum total displacement (ranging from ~0.049 mm to ~0.063 mm at 1000 N).

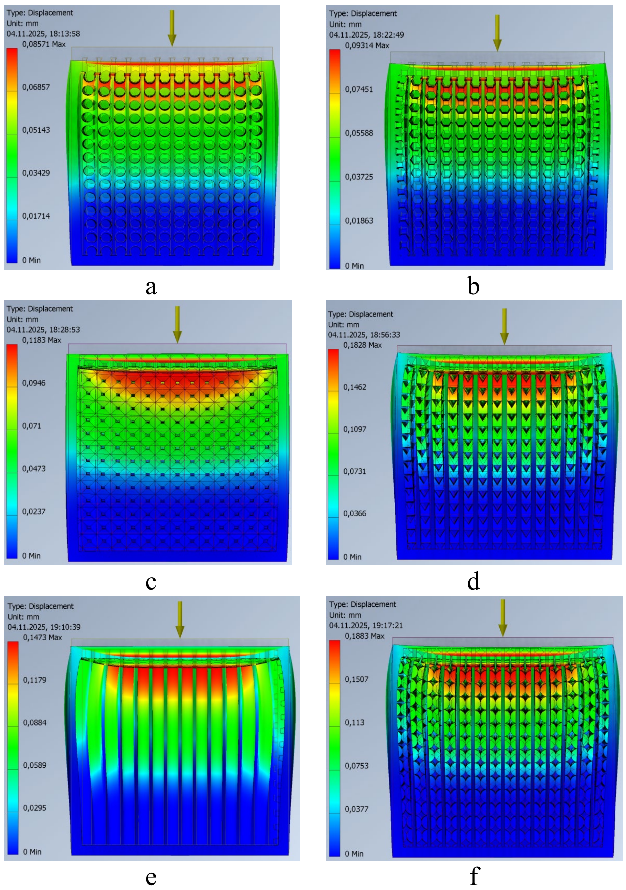

Fig. 5. Maximum displacement of cubic unit cell lattice (3000 N load): *a* – Circular-profile; *b* – Hexagonal-profile; *c* – Square-profile; *d* – Triangular-profile; *e* – Elliptical-profile; *f* – Star-profile.

Fig. 4 and Fig. 5 show the displacement distributions obtained from the FEA simulations for all lattice configurations under the two applied loading conditions. Fig. 4 corresponds to the 1000 N load case, whereas Fig. 5 presents the results for the 3000 N load. The displacement fields reveal how the internal lattice topology influences the overall deformation behavior, with visibly higher displacements occurring under increased loading and in the less dense structures.

In contrast, the lattices that retained slightly more material, such as the circular-profile cubic unit cell lattice (18.6149 g) and hexagonal-profile cubic unit cell lattice (19.0283 g), maintained superior stiffness properties and safety margins (*SF* 3.29 and 2.89, at 1000 N), demonstrating a more effective trade-off balance.

The minimum safety factor (SF_{min}) demonstrated a strong negative correlation with mass reduction across the lattice structures. The two lightest CAD models (with star and triangular profiles) exhibited SF_{min} values below 1.0 at the 1000 N load case (~0.94 and ~0.93), indicating a localized failure condition exceeding the material's yield strength of 20 MPa.

Figures 6 and 7 present the Equivalent Strain distributions for all analyzed lattice configurations under the two applied loading conditions (1000 N and 3000 N applied load).

The circular-profile cubic unit cell lattice, which reduced mass by approximately 35% relative to the solid reference, achieved a minimum safety factor (~3.29) that

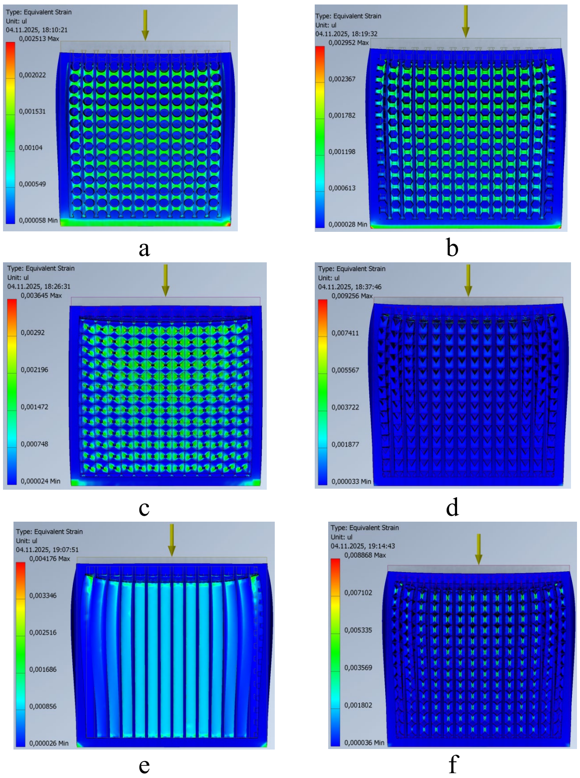

Fig. 6. Maximum Equivalent strain of cubic unit cell lattice (1000 N load): *a* – Circular-profile; *b* – Hexagonal-profile; *c* – Square-profile; *d* – Triangular-profile; *e* – Elliptical-profile; *f* – Star-profile.

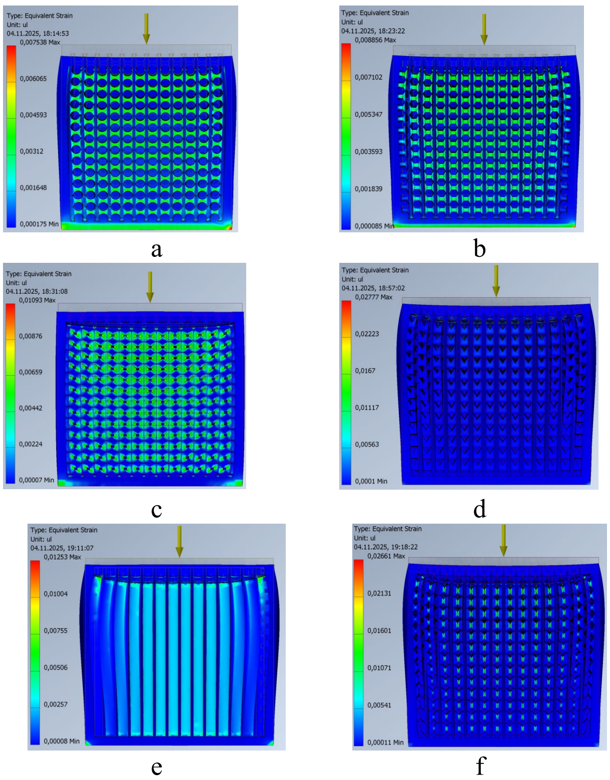

Fig. 7. Maximum Equivalent strain of cubic unit cell lattice (3000 N load): *a* – Circular-profile; *b* – Hexagonal-profile; *c* – Square-profile; *d* – Triangular-profile; *e* – Elliptical-profile; *f* – Star-profile.

was significantly higher than those structures reducing mass by approximately 50% (star, triangular, elliptical profiles, where SF_{min} ranged from ~0.93 to ~2 at 1000 N). This quantitative evidence establishes that maximizing material reduction severely compromises the structural safety edges under equivalent compressive loading.

4. CONCLUSIONS

This study employed comparative FEA to rigorously assess the mechanical behavior and material efficiency of six distinct lattice structures against a fully dense solid reference component. The findings provide important design implications for AM.

While AM enables the fabrication of these complex geometries to achieve substantial weight reduction, designers must recognize that maximizing material removal often severely compromises structural safety and stiffness.

Implementing lattice structures represents a viable pathway for sustainable mechanical engineering by significantly reducing material consumption and waste. However, successful implementation requires precise topology optimization, as evidenced by the superior performance of the circular-profile cubic unit cell lattice over the lighter star and triangular profiles, to ensure that weight saving does not translate into critical component failure under expected loads. Thus, reliable FEA validation is indispensable for determining the optimal lattice configuration that satisfies both material efficiency and requisite structural performance thresholds.

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