
Bending Stiffness of Continuous Fiber Reinforced 3D-Printed Honeycomb Sandwich Panel Cores

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ABSTRACT

This study investigates the bending stiffness of continuous fiber reinforced 3D-printed honeycomb sandwich panel cores, focusing on the combined influence of material properties, fiber reinforcement, and core geometry. At the material level, both quasi-static and dynamic experimental methods were applied to characterize the mechanical behavior of additively manufactured Onyx[®] and carbon fiber reinforced composites. A combination of these experimental methods was found to be suitable for obtaining a reliable estimate of the effective material properties. At the structural level, sandwich panels with varying honeycomb cell angles and fiber contents were tested under three-point bending. The results show that core geometry and filament orientation significantly influence flexural stiffness, in some cases more than relative density. In addition, the location of fiber reinforcement within the core was found to be critical, with reinforcement near the neutral axis contributing less effectively to bending stiffness. The flexural modulus could be determined consistently, whereas failure behavior showed greater variability due to local effects such as adhesion quality and manufacturing imperfections. Overall, the results highlight the importance of geometric design and fiber placement in optimizing the mechanical performance of 3D-printed sandwich structures.

KEYWORDS

Continuous fiber composites, Additive manufacturing,
Honeycomb structures, Sandwich panel cores, Flexural behavior,
Experimental validation, 3D-printed composites

Abbreviations: AM, Additive Manufacturing; FDM, Fused Deposition Modeling; FRAM, Fiber Reinforced Additive Manufacturing; CF, Carbon Fiber; DMA, Dynamic Mechanical Analysis; ToF, Time of Flight; FEA, Finite Element Analysis; ROM, Rule of Mixtures; HS, Hashin-Shtrikman; PAT, Parallel Axis Theorem

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1 | INTRODUCTION

Additive manufacturing technology is rising in popularity due to its ease of use, customization, and rapid prototyping possibilities. One major downside to standard FDM polymer 3D printing is the use of relatively weak materials, resulting in designs that cannot be used in high performance or structural applications where strength or stiffness are main requirements [1]. To address this limitation, reinforced filaments were introduced, in which small carbon, glass, or aramid fibers are embedded in a polymer matrix to create more durable parts. However, in order to create a material with superior mechanical properties in a particular direction, it is preferred to print with continuous fibers. This novice technique requires a completely different printing method, in which two separate nozzles extrude the polymer matrix and the continuous fiber distinctly to form one object.

One of the possible applications for this state-of-the-art 3D printing method is the construction and experimental analysis of sandwich panel cores. Sandwich panels are commonly applied in structures that demand a high stiffness-to-weight ratio. Other important properties of sandwich structures include excellent thermal insulation values and high impact resistance [2]. Recently, abundant research has been performed on this topic in order to find the optimal core dimensions and topology for specific applications. Aviation is a classical example of such an application where the properties of sandwich structures are particularly beneficial, with structural integrity, lightweight design and thermal protection being main design objectives. Currently, skins of sandwich panels for aerospace applications are mainly produced from carbon fiber, fiberglass or aluminium, while the cores are generally constructed with Nomex, aluminium or foam types.

Honeycomb structures are well known for their excellent material properties and low density, making them appropriate as core components in sandwich panel designs. Typical sandwich panels consist of two face sheets on either side of a core component. Often used sandwich cores can be classified into two types, being either stochastic or periodic [3]. Examples of stochastic cores include polymeric foams, consisting of a porous, cellular structure that has excellent thermal insulation and is lightweight. Foams can be either open-celled or close-celled, and can be manufactured using both polymeric or metallic materials.

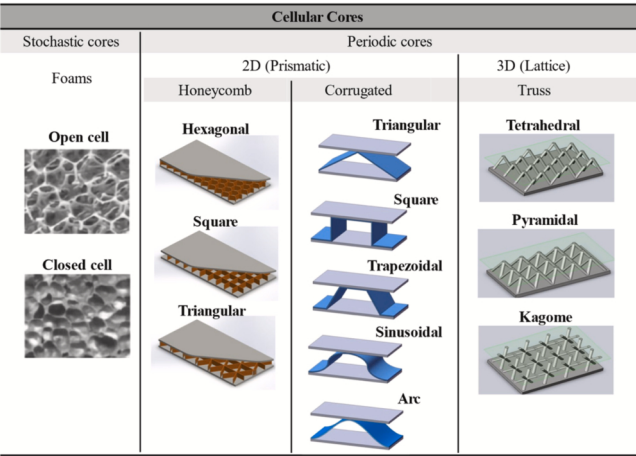


FIGURE 1 Overview of cellular core geometries: stochastic vs. periodic cores [3].

Periodic cores consist of different type of cellular solids, and are either 2- or 3-dimensionally shaped. Two-dimensional prismatic periodic cores consist of repeating internal structures that extend uniformly in one direction. These cores can be classified into two main types: in-plane honeycomb structures, which have a flat, cellular geometry, and corrugated structures, which are formed by periodically shaped profiles such as triangular, trapezoidal, or rectangular geometries. Three dimensional periodic lattice cores consist of repeating truss based structures, such as tetrahedral or pyramidal geometries. This study will primarily focus on honeycomb geometry as the core structure.

Continuous fiber additive manufacturing may not only be useful for the direct manufacturing of small scale honeycomb sandwich panel cores for real-world applications, it also enables quick prototyping of particular core properties such as cell topology, relative density, and core dimensions. The incorporation of additional fiber content improves the material's properties to the extent that its mechanical performance can be comparable to that of certain metals, such as aluminum [4].

2 | LITERATURE REVIEW

2.1 | Overview of fiber reinforced additive manufacturing technologies

Additive manufacturing, also known as 3D printing, is a process that creates three-dimensional objects by depositing material layer by layer, based on CAD models created by the user. The technology has gained significant attention in recent years due to its many advantages, including rapid prototyping, customization options, and reduced waste production [5][6]. Additionally, 3D printing has become more accessible to the general public, as both machines and materials have become more affordable and user-friendly.

In addition to the technology becoming more accessible to the average consumer, significant advancements have also been made in machines designed for industrial purposes. Commercial adoption of additive manufacturing technologies first arose in the late 1980s and early 1990s, with stereolithography (SL) technology coming the first to market. Stereolithography is a process in which thin layers of liquid polymer are solidified using ultraviolet (UV) light from a laser. The most commonly used additive manufacturing technology used nowadays, fused deposition modeling (FDM) or fused filament fabrication (FFF), was first commercialized in 1991. In FDM technology, thermoplastic materials are extruded in the printhead of the machine to form parts in a layer by layer method [6]. Additive manufacturing technologies can be divided into seven different categories, as shown in Table 1.

In an industrial setting, AM technology must compete with traditional manufacturing methods like compression molding, CNC milling or casting. Although these production techniques are often faster, more cost effective, and better suited for high-volume production, AM offers significant advantages for low-volume production and customized products. One major disadvantage for traditional FDM 3D printing is the relatively low mechanical performance of standard thermoplastic filaments [1]. Over the last decade, fiber reinforced additive manufacturing techniques were introduced, that combine the flexibility of AM with the mechanical advantages of composite materials. In fiber reinforced additive manufacturing (FRAM), the matrix of the composite is often a traditional AM or engineering polymer such as PLA, Nylon or Polycarbonate. The fiber material can be distinguished into 3 different material groups: polymer based, such as kevlar® or carbon fiber; metal or ceramics base, like silicon carbide or aluminium oxide; or consisting of natural fibers like hemp or flax [8]. Although natural fibers are found to have a strong potential, it was found that synthetic fiber reinforcement is still the most effective [9]. Composite materials with synthetic fibers as reinforcing agents were found to have a tensile strength that is more than 100% greater than that of its matrix counterpart when printed separately [10].

The 7 Categories of AM Technologies	
Binder Jetting	Binder Jetting
Material Extrusion	Liquid Deposition Modelling Fused Deposition Modelling
Powder Bed Fusion	Selective Laser Melting Selective Laser Sintering Multi Jet Fusion Electron Beam Melting
Vat Polymerization	Stereo-Lithography Continuous Digital Light Processing Daylight Polymer Printing Digital Light Processing
Direct Energy Deposition	Electron Beam AM Laser Engineering Net
Material Jetting	Nanoparticle Jetting Material Jetting Drop on Demand
Sheet Lamination	Laminated Object Manufacturing

TABLE 1 The 7 categories of AM Technologies as discussed by [7], with corresponding examples of specific AM technologies in the right column.

The FRAM printing process can be divided into 2 different categories, based on the fiber that is involved. The most common fiber reinforcement in AM is that where short, chopped fibers are included in the matrix filament material. The printing process for this method is relatively straightforward, with one spool of filament that is being printed through a single nozzle. Due to the high temperature sensitivity of engineering filaments, printers designed for short fiber reinforced additive manufacturing usually have an enclosed printing area. Furthermore, the nozzle of these machines are typically made of hardened steel types (instead of softer stainless steel material in widely available commercial 3D printers), to avoid excessive wear caused by the short abrasive fiber particles that are incorporated in the filament. The printing process for this method is relatively straightforward, with one spool of filament that is being printed through a single nozzle. Due to the high temperature sensitivity of engineering filaments, printers designed for short fiber reinforced additive manufacturing usually have an enclosed printing area. Furthermore, the nozzle of these machines are typically made of hardened steel types (instead of softer stainless steel material in widely available commercial 3D printers), to avoid excessive wear caused by the short abrasive fiber particles that are incorporated in the filament.

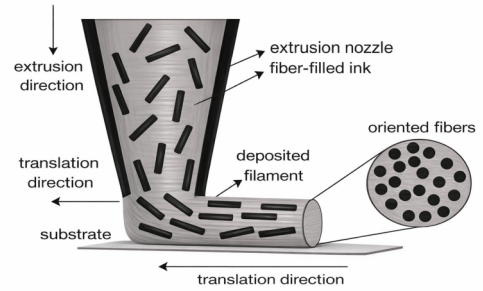


FIGURE 2 The short fiber 3D printing process showcased in a schematic. Adapted from [5].

The second type of FRAM is continuous fiber additive manufacturing, where long strands of fiber are printed through a secondary nozzle to form a single part. Here, fiber is also contained on a separate spool, and is pre-impregnated with a thermoplastic binder, since proper impregnation is difficult to achieve during the printing process itself. Furthermore, also the bonding between the fiber strands and the polymer matrix is enhanced in this manner. The slicing software precisely controls the fiber deposition path, as the brittle impregnated strands cannot tolerate sharp curvatures [11]. For this reason, continuous fibers are typically placed only within the infill region of a component. The outer walls are printed using the matrix material alone, which encloses the reinforcement and defines the final geometry and surface finish.

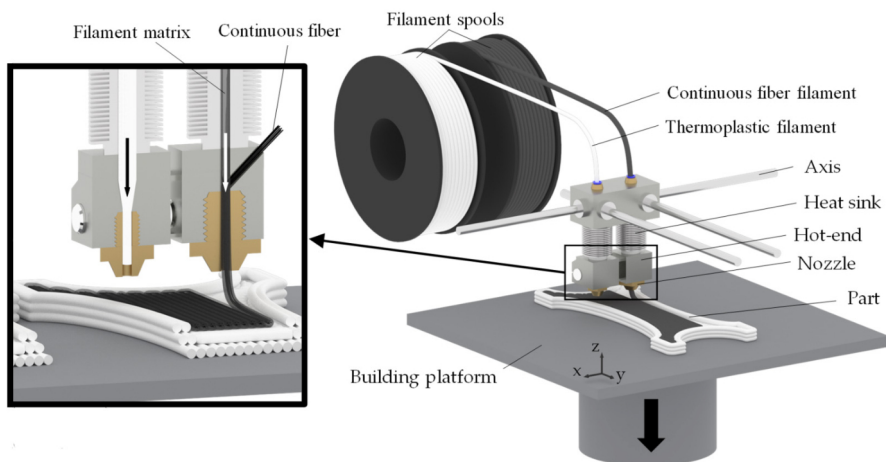


FIGURE 3 The continuous fiber printing process consists of two separate nozzles and two filament spools. White filament lines represent the filament matrix, black lines show continuous carbon fiber strands. Adapted from [12].

The 3D printer that will be used for this research is the Markforged Mark Two™. This machine is the successor of the Mark One from Markforged, which was put to market in January 2014 as the first commercially available continuous fiber 3D printer. The printer offers a wide range of possible matrix and fiber filaments to be used, including Precise PLA, Nylon and Onyx® as base polymers, and carbon fiber, aramid fiber and glass fiber as continuous fiber materials [13]. The materials applied in this research are Onyx®, a micro carbon fiber filled nylon that is approximately 1.4 times stiffer than ABS filament, and continuous carbon fiber, which offers the highest strength-to-weight ratio among the available Markforged reinforcement fibers [13].

2.2 | Mechanical behavior of fiber reinforced composites

The mechanical properties composites can be determined by using the Rule of Mixtures (ROM), as shown in Equation (1). When loaded parallel to the fiber direction, the tensile strength and tensile modulus of the composite is expressed as:

$$\sigma_c = \sigma_f V_f + \sigma_m^* (1 - V_f) \quad (1)$$

$$E_c = E_f V_f + E_m (1 - V_f) \quad (2)$$

In this formulation, σ denotes tensile strength, E the Young's modulus and V_f represents the fiber volume fraction. The subscripts c , f , and m refer to composite, fiber, and matrix, respectively. The parameter σ_m^* corresponds to the matrix stress at the strain level at which fiber failure occurs [14]. In continuous fiber-reinforced composites, fiber often behaves more brittle than the surrounding matrix, which typically results in initial failure in the fiber. After fiber fracture, the applied load is transferred to the matrix material [15]. The Rule of Mixtures, also known as the Voigt model, assumes that fibers are perfectly aligned and both the fiber and matrix experience the same strain. The Voigt model gives the upper bound of the elastic modulus of the composite, assuming the Poisson's ratios of the two phase materials are equal [16]. The corresponding lower bound is given by the inverse ROM, better known as Reuss model:

$$\frac{1}{E_c} = \frac{V_f}{E_f} + \frac{V_m}{E_m} \quad (3)$$

Reuss model provides the lower bounds of the elastic properties of the composite. In contrast to the Voigt model, it assumes uniform stress across both the matrix and fiber component.

The Voigt-Reuss model is known for providing relatively wide bounds, that may satisfy for quick engineering estimates, but not fulfill for a grounded scientific prediction of the elastic properties of a composite. In 1963, Hashin and Shtrikman came up with a new method to predict these constants, by using the variational energy principle. These new bounds, denoted as the Hashin-Shtrikman (HS) bounds, do not assume iso-strain or iso-stress across the matrix and fiber of the composite and incorporate phase interactions. With that, these bounds are stricter, and most often more realistic than the Voigt-Reuss model [17]. The HS lower bounds are given by:

$$G_c = G_m + \frac{V_f}{\frac{1}{G_f - G_m} + \frac{6(K_m + 2G_m)(1 - V_f)}{5(3K_m + 4G_m)G_m}} \quad (4)$$

$$K_c = K_m + \frac{V_f (K_f - K_m)(3K_m + 4G_m)}{(3K_m + 4G_m) + 3(K_f - K_m)(1 - V_f)} \quad (5)$$

In this relationship, G represents the shear modulus, and K the bulk modulus of the material. By interchanging

subscripts f and m , one obtains the upper HS bound. The corresponding Young's modulus can be found by substituting G and K from 6 with the values G_c and K_c found from equation 4 and 5.

$$E = \frac{9KG}{3K + G} \quad (6)$$

It is important to note that although the HS bounds give the tightest possible theoretical bounds for elastic moduli of two phase composites, it assumes that both phases are isotropic from nature. For continuous fiber 3D printed components, the matrix can be treated as quasi-isotropic at the material scale. However, the continuous fibers are inherently anisotropic, or more specifically, transversely isotropic (TI). Therefore, it is not justified to apply the HS bounds in the particular case of continuous fiber AM components. In 1965, Hashin published another study on the elastic behavior of fiber reinforced materials with an arbitrary transverse phase geometry [18]. In this paper, Hashin proved that the HS bounds are indeed the most accurate bounds that can be determined based on phase moduli and volume fractions, when no additional geometrical information on the composite is given. Furthermore, this article presents an extension of the original HS bounds by introducing a new set of bounds for 3 of the 5 effective moduli for transversely isotropic fiber composites as shown below:

$$k_c^{(-)} = k_m + \frac{f}{\frac{1}{k_f - k_m} + \frac{1-f}{k_m + G_m}} \quad (7)$$

$$m_c^{(-)} = G_m + \frac{f}{\frac{1}{G_f - G_m} + \frac{(1-f)(k_m + 2G_m)}{2G_m(k_m + G_m)}} \quad (8)$$

$$\mu_c^{(-)} = G_m + \frac{f}{\frac{1}{G_f - G_m} + \frac{1-f}{2G_m}} \quad (9)$$

For $k_f > k_m$ and $G_f > G_m$, which is generally the case for fiber reinforced composites. For transversely isotropic fiber composites, k_c is the plane-strain bulk modulus in the transverse plane, m_c is the transverse shear modulus and μ_c is the axial shear modulus. The relationships of the remaining two moduli E and ν were further clarified by Hill in 1964 [19].

However, this set of bounds is not yet suitable for determining the moduli of the continuous fiber AM components. Although the study is appropriate from overall transversely isotropic composites, it still assumes that the individual phases are linear elastic and isotropic. As mentioned earlier, this cannot be stated for the carbon fiber strands that show much greater longitudinal stiffness compared to their stiffness in the transverse direction. For predicting longitudinal properties of the composite, the error using this method is expected to be small, but nevertheless, it is not theoretically correct. In 2015, W. Parnell and Calvo-Jurado attempted to clarify this particular situation where the phase material is transversely isotropic in nature [20]. In this paper, two limiting cases of transversely isotropic composites are described, with the inclusion shape becoming either extremely flat or extremely long. In the article, this is denoted as $\delta = \epsilon \rightarrow 0$ or $\delta = \epsilon \rightarrow \infty$, where the former refers to a layered medium and the latter to a long fiber reinforced medium. The bounds for the set of constants k^B , ℓ^B , n^B , m^B , p^B that resemble the basis of the Hill tensor for transverse isotropy in the case of $\delta = \epsilon \rightarrow \infty$ are given below:

$$k^B = \left(\sum_{r=1}^n \frac{k_r \phi_r}{k_r + m_c} \right) \left(\sum_{r=1}^n \frac{\phi_r}{k_r + m_c} \right)^{-1}, \quad (10)$$

$$\ell^B = \left(\sum_{r=1}^n \frac{\ell_r \phi_r}{k_r + m_c} \right) \left(\sum_{r=1}^n \frac{\phi_r}{k_r + m_c} \right)^{-1} \quad (11)$$

$$n^B = \sum_{r=1}^n \phi_r \left(n_r - \frac{(\ell^B - \ell_r)(\ell_c - \ell_r)}{k_r + m_c} \right) \quad (12)$$

$$m^B = \left(\sum_{r=1}^n \frac{m_r \phi_r}{2m_c m_r + k_c(m_r + m_c)} \right) \left(\sum_{r=1}^n \frac{\phi_r}{2m_c m_r + k_c(m_r + m_c)} \right)^{-1} \quad (13)$$

$$p^B = \left(\sum_{r=1}^n \frac{p_r \phi_r}{p_r + p_c} \right) \left(\sum_{r=1}^n \frac{\phi_r}{p_r + p_c} \right)^{-1} \quad (14)$$

In the aforementioned equations, k is the in-plane bulk modulus, ℓ is the coupling modulus, n is the longitudinal modulus, m is the in-plane shear modulus and p is the longitudinal shear modulus. In the case of an isotropic material, just two engineering constants are required to determine these 5 Hill moduli. ϕ_r represents the volume fraction of phase r and subscript c refers to the comparison material. To obtain the lower bounds of the effective moduli, the comparison material must be chosen such that it possesses the smallest phase moduli. Conversely, the upper bounds are obtained by selecting the comparison material with the largest phase moduli. These moduli can be related directly to the 5 engineering constants for transversely isotropic materials.

2.3 | Mechanical characterization of 3D-printed composites

In this section, the currently available experimental methods to acquire various mechanical properties of AM composites are discussed. For the continuation of this study, it is essential to gather more detailed information on the mechanical properties of the used phase materials, such that the corresponding composite behavior can be predicted by using methods such as presented in 2.2. The experimental methods that will be discussed can be categorized into being either quasi-static or dynamic.

2.3.1 Quasi-static methods

Experimental methods can be defined as quasi-static when any loads or displacements are slow enough that dynamic effects can be neglected. The system shall remain in equilibrium or near equilibrium during the test. Results from quasi-static test can be considered time-independent and lead to fundamental mechanical properties of the tested material. Examples of quasi-static tests are presented in Table 2.

2.3.2 Dynamic methods

Experimental methods are found to be dynamic when displacements or loads are happening at a rate which is high enough that wave propagation, inertial effects or other time dependent responses are significant and cannot be ignored. Testing methods from this category are used to acquire time dependent or viscoelastic properties of the material. For instance, Bucur (2016) describes dynamic resonance methods for determining elastic constants, where specimens are excited and their resonance frequencies are used to calculate Young's and shear moduli [21]. Examples of dynamic experimental methods are presented in Table 3.

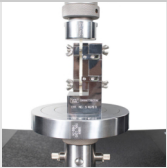
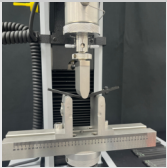
Quasi-static Experimental Mechanical Characterization Methods				
Method	Tensile test	Compression test	In-plane shear test	Bending test
Description	A tensile test is used to determine the mechanical response of a material under uniaxial tension. During the test, a specimen is subjected to a controlled increasing load until failure.	A compression test measures how a material deforms and withstands load when a compressive force is applied until a specified deformation or failure occurs.	An in-plane shear test determines the shear modulus and shear strength of a material by applying a load that induces shear stresses within the plane of the specimen.	A bending test evaluates how a material behaves when subjected to a transverse load, by measuring its response as the specimen is bent until a defined limit or failure occurs.
Obtained quantities	Young's modulus (E), Tensile strength (σ_{UTS}), Elongation at break (ϵ_b)	Young's modulus (E_c), Compressive strength (σ_c), Strain at failure (ϵ_c)	Shear modulus (G_{12}), Shear strength ($\tau_{u,12}$), Shear strain at failure (γ_{12})	Flexural modulus (E_f), Flexural strength (σ_f), Maximum strain (ϵ_f)
Image				

TABLE 2 Overview of quasi-static experimental methods [22] [23] [24].

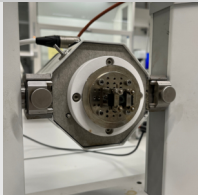
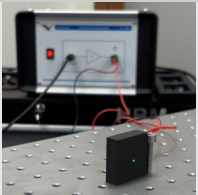
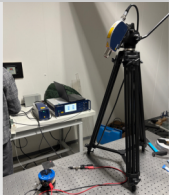
Dynamic Experimental Mechanical Characterization Methods			
Method	Dynamic Mechanical Analysis (DMA)	Time of Flight (ToF)	Modal Analysis
Description	Dynamic Mechanical Analysis evaluates the viscoelastic behavior of a material by applying an oscillating load and measuring the response as a function of temperature or frequency.	The Time of Flight method determines material properties by measuring the propagation time of an ultrasonic wave through the specimen.	Modal analysis identifies the dynamic characteristics of a structure by measuring its vibration response to an excitation.
Obtained quantities	Storage modulus (E'), Loss modulus (E''), Damping factor ($\tan \delta$)	Wave velocity (v), Elastic modulus (E), Attenuation coefficient (α)	Natural frequencies (f_n), Mode shapes, Damping ratios (ζ)
Image			

TABLE 3 Overview of dynamic experimental methods [25][26][21].

2.4 | Sandwich panel theory

Sandwich beams can be defined as a specific type of layered composite structures, typically consisting of two strong face sheet layers bonded to a lightweight core material positioned between them [27]. The core of a sandwich panel can be made from a wide range of materials and structural configurations, such as honeycomb cores (as illustrated in Figure 5), polymeric foam cores, corrugated structures, and truss-based cores (see Figure 1). In general, the face sheets of sandwich structures are made from a stiff material and are relatively thin compared to the core. The core itself is typically lightweight and more compliant, providing thickness and stability without significantly increasing weight. This study focuses on sandwich panels where the core also significantly contributes to the overall stiffness.

Sandwich panels are increasingly applied in engineering applications where lightweight design combined with high flexural strength and stiffness is required. Industries that make extensive use of sandwich structures include the automotive, aerospace, and marine sectors. In the aerospace sector, face sheets are typically made from graphite-epoxy or carbon-epoxy laminates due to their high stiffness-to-weight ratio. For aircrafts, the core of the sandwich panels are often constructed using aluminium or Nomex® material in a honeycomb structure. In other sectors, such as marine or civil engineering, where cost considerations become more dominant compared to weight reduction, glass-vinyl and glass-epoxy face sheets are more commonly used. The core of sandwich panels in marine technology can be created from balsa wood, while open- and closed cell foams are frequently used in civil engineering applications [27]. The way that sandwich structures are designed is determined by the type of strain the is generally applied during its application. For instance, under flatwise compression, the sandwich core largely governs the energy absorption. In edgewise compression, both the core and the face sheets contribute and are of roughly equal importance. Under bending loads, the face sheets carry most of the tensile and compressive stresses, while the core primarily maintains the separation between the faces and transfers shear [28].

The overall bending stiffness of a sandwich structure consisting of one core component surrounded by two face sheets on opposite sides can be determined using the Parallel Axis Theorem (PAT). If the sandwich beam has a symmetrical cross section, its bending stiffness is given as:

$$E_x I = E_x^c I_c + 2E_x^f I_f, \quad (15)$$

where E_x is the flexural modulus of the material and I its moment of inertia with respect to the neutral axis, which located in the center of the sandwich structure's core [29]. The subscripts c and f denote either the core or face sheet components, respectively. The moment of inertia of the core I_c is given by the standard equation for the moment of inertia of a rectangular shaped section about the centroidal horizontal axis and that of a single face sheet is given by applying the PAT to this relationship:

$$I_c = \frac{bt_c^3}{12}, \quad (16)$$

$$I_f = \frac{bt_f^3}{12} + \frac{bt_f d^2}{4}, \quad (17)$$

where b and t resemble the width and thickness of structure, and $d = t_c + t_f$ for symmetric sandwich panels (see

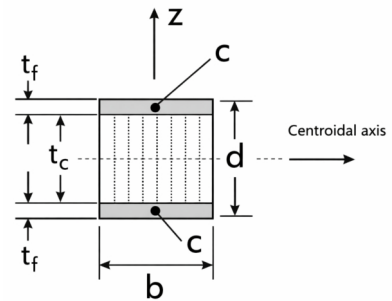


FIGURE 4 The cross section of a symmetrical sandwich panel. Adapted from [27].

Figure 5). Substituting the found values for I_c and I_f into equation 20 yields:

$$E_x I = \frac{E_x^c b t_c^3}{12} + \frac{E_x^f b t_f^3}{6} + \frac{E_x^f b t_f d^2}{2} \quad (18)$$

When simplified, and written only in terms of t_c and t_f , we obtain:

$$E_x I = E_x^c \left(\frac{b t_c^3}{12} \right) + E_x^f b t_f \left(\frac{t_f^2}{6} + \frac{(t_c + t_f)^2}{2} \right) \quad (19)$$

Engineers often refer to $E_x I / b$ simply as the bending stiffness, although it formally represents the bending stiffness per unit width. This is given as:

$$E_x I / b = E_x^c \left(\frac{t_c^3}{12} \right) + E_x^f t_f \left(\frac{t_f^2}{6} + \frac{(t_c + t_f)^2}{2} \right) \quad (20)$$

In classical sandwich beams, the $t_f \ll t_c$ and $E_f \gg E_c$, which indicates that the soft, low density core primarily serves to maintain separation between the stiff, thin face sheets. In his book structural and failure mechanics of sandwich composites, Carlsson shows that when this property holds to the extent that $t_c / h_f \geq 5.35$, then the

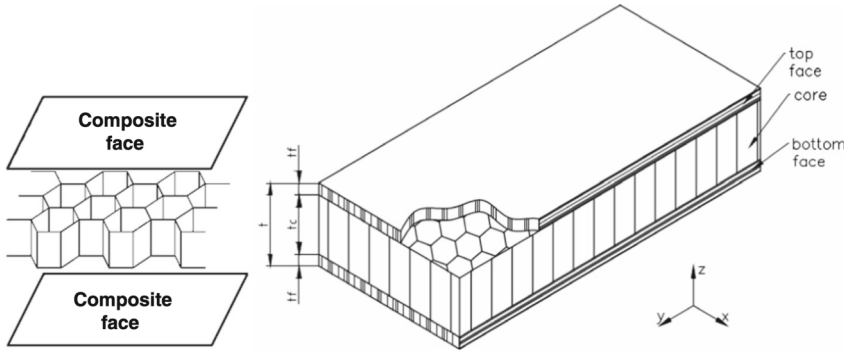


FIGURE 5 A schematic showing a symmetric sandwich panel consisting of a honeycomb core and two composite face sheets [30].

The efficiency of sandwich structures (i.e. high bending stiffness with low weight) comes from large face separation. However, this same geometrical property increases sensitivity to instabilities. Therefore, the resistance of the structure to global and local buckling is another important factor that structural designers must take into account.

For a simply supported beam, Euler's formula for buckling load is given as:

$$F_{c,Euler} = \frac{\pi^2 E I}{L^2} \quad (21)$$

At first glance, this implies that the critical buckling load proportionally increases with an increase in bending stiffness of the panel. However, this formula assumes a homogeneous beam that undergoes pure bending deformation. In

the case of a sandwich panel, also transverse shear deformation of the relatively soft core material occurs. This significantly reduces the critical buckling load:

$$F_{c,sandwich} = \frac{F_{c,Euler}}{1 + \frac{F_{c,Euler}}{t_c G_c}}, \quad (22)$$

where t_c denotes the core thickness and G_c is the transverse shear modulus of the core material [31]. Due to a lightweight core, also local buckling effects can take place. Analytical research on this topic resulted in the classical wrinkling stress formula as found by Hoff and Mautner in 1945 [32], which is presented below:

$$\sigma_c = 0.91 \sqrt[3]{E_{skin} E_{core} G_{core}} \quad (23)$$

where E represents the Young's Modulus of the referred material, and G the shear modulus of the core. The factor 0.91 comes from an analytical analysis by Hoff and Mautner. Although experimental results sometimes show significant deviations from the equation and alternative formulations have been proposed, this version remains the most widely used due to its simplicity.

As mentioned before, structural designers may opt for a sandwich construction if the weight of the structure is of high importance. The weight of a sandwich panel is given as :

$$W = 2t_f b \rho_f + t_c b \rho_c \quad (24)$$

The earlier mentioned Equation 20 consists of three main parts that contribute to the overall bending stiffness of the sandwich structure. The first part $\frac{E_c t_c^3}{12}$ relates to the bending stiffness of the core. In many sandwich panels, the ratio between the modulus of the core and face sheets is to such low extent (e.g. $E_f/E_c = 1000$), that this term is not of significant contribution to the total bending stiffness of the panel. In their book Structural and Failure Mechanics of Sandwich Composites, Carlsson and Kardomateas stated that when the ratio between the thickness of the core and the thickness of the face sheets satisfies the property $h_c/h_f \geq 5.35$, the second term $\frac{E_x t_f^3}{6}$ can also be neglected, as the faces can then be considered as 'thin', and t_f is relatively small [29]. What remains is the most simplistic expression for bending stiffness of sandwich panels with stiff, thin face sheets and soft, low weight cores (per unit width):

$$D_x = \frac{E_x t_f d^2}{2} \quad (25)$$

2.5 | Honeycomb core structures

Honeycomb structures are widely used as the core material in sandwich panels due to their exceptional stiffness-to-weight ratio. The cellular geometry originated from nature, as the name suggests, where honeybees have developed the ability to construct beehives made of repeating hexagonal cells. These cells form a highly organized structure that efficiently stores honey and pollen [33]. In 2001, Hales showed that the hexagonal structure is the most efficient in terms of optimizing internal space compared to the minimum usage of material [34]. This characteristic is particularly important for sandwich panel cores, where achieving a large separation between the face sheets while using as little material as possible is a key design objective.

Because of their favorable mechanical and functional properties, such as high fracture toughness, strong impact resistance, efficient heat dissipation, and the ability to damp vibrations, honeycomb structures have attracted significant

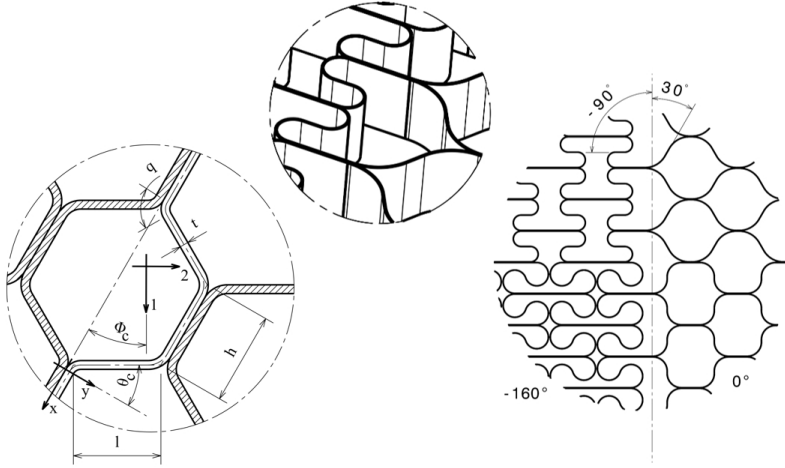


FIGURE 6 The geometry of a honeycomb unit cell, described by a fixed set of parameters. Structures may deviate from the classical hexagonal design when the cell angle is changed. [36]

attention in engineering. These characteristics have led to their widespread adoption in applications including automotive components, medical implants, and aerospace structures [33]. Regarding the latter, recent research was conducted by Vladimír Hostinský, who introduced an analytical model to predict the homogenized mechanical properties of honeycomb core structures. In his work, Hostinský compares the results of his analytical model with finite element analysis (FEA), looking at several parameters of the cell geometry, specifically focusing on the cell angle θ_c as seen in Figure 6.

The Young's Modulus of the honeycomb material when loaded in axial direction can be related to several parameters, including the cell angle, according to the following relationship:

$$E_y = E_s \left(\frac{t}{l} \right)^3 \frac{\cos \theta}{(\sin \theta + h/l) \sin^2 \theta} \quad (26)$$

Here, E_y is the effective Young's Modulus of the honeycomb in y direction (see Figure 5), E_s is the Young's Modulus of the solid base material, t is the width, l is the length of the inclined cell walls, h is the height of the vertical walls and θ is the angle of the inclined walls relative to the horizontal [35].

2.6 | Research gap and positioning of present study

2.6.1 Research gap

Despite the extensive body of literature on fiber reinforced composites, sandwich panel theory, and honeycomb core structures, several important gaps remain when these domains are combined in the context of continuous fiber additive manufacturing.

At the material level, existing theories such as the Rule of Mixtures and Hashin–Shtrikman bounds provide estimates

for the effective stiffness of fiber reinforced composites. However, these models assume idealized conditions, including perfect bonding and homogeneous phase distribution. As discussed in Section 2.2, these assumptions are not fully valid for continuous fiber 3D printed materials, where anisotropy and imperfect bonding between the fiber and the matrix play a significant role. Consequently, the direct applicability of classical composite theory to additively manufactured continuous fiber materials remains limited.

At the structural level, sandwich panel theory offers well established analytical expressions for predicting bending stiffness, where the stiffness contribution of the core is typically considered secondary compared to the face sheets. However, these formulations rely on simplified representations of the core, often treating it as a homogeneous material characterized by effective properties. In reality, honeycomb cores exhibit mechanical behavior that is geometry dependent, as shown by the analytical relations that link Young's modulus to parameters such as cell angle, wall thickness, and cell dimensions.

The current literature lacks an integrated understanding of how (1) fiber volume fraction influences the effective stiffness of continuous fiber 3D-printed composites, (2) honeycomb geometry and fiber reinforcement determine core behavior, and (3) these combined effects translate into the bending stiffness of sandwich panel structures.

This research aims to address these gaps by establishing a direct link between material composition, geometric design, and structural performance in continuous fiber reinforced 3D-printed honeycomb sandwich panels. First, the research investigates the relationship between continuous fiber content and the effective Young's modulus of additively manufactured composites, thereby evaluating the applicability of existing composite theories to this novel manufacturing process. In order to reach this goal, the study evaluates different experimental methods for determining the mechanical properties of continuous fiber 3D-printed materials. And lastly, it is examined how variations in fiber content and cell geometry influence the bending stiffness of the manufactured sandwich panels, bridging the gap between material level properties and structural response.

2.6.2 Research questions

Based on the identified gaps, the following research questions are formulated:

Main Research Question

How do continuous carbon fiber reinforcement and honeycomb core configuration influence the bending stiffness of additively manufactured sandwich panels?

Sub-Questions

1. What is the relationship between continuous fiber content and the effective Young's modulus of continuous fiber reinforced 3D-printed materials?
2. How does continuous fiber content influence the bending stiffness of continuous fiber reinforced 3D-printed specimens?
3. Which experimental methods are suitable for reliably determining the mechanical properties of continuous fiber reinforced 3D-printed materials?
4. How does variation in continuous fiber content within the sandwich panel core affect the global bending stiffness of the complete sandwich structure?
5. What is the influence of honeycomb cell angle on the bending stiffness of 3D-printed honeycomb sandwich panels?

3 | MATERIALS & METHODS

3.1 | Materials

As introduced in Section 2.1, Onyx[®] is selected as the matrix material for this study. This material consists of a carbon fiber reinforced nylon, which offers reliable printability with the Markforged Mark Two 3D printer and improved mechanical properties compared to conventional thermoplastic filaments, primarily due to the presence of chopped fiber inclusions. The continuous fiber reinforcement selected for this study is carbon fiber, as it offers the highest mechanical performance among the fiber materials available from Markforged. In addition, previous projects have demonstrated good compatibility with Onyx as the matrix material.

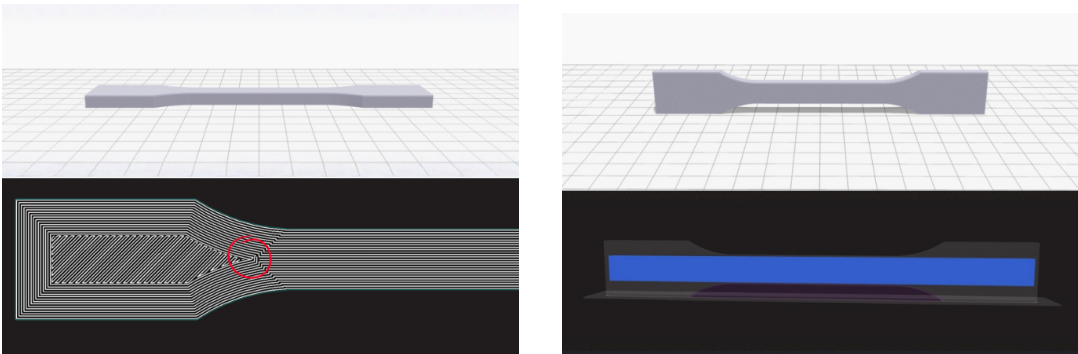
Property	Onyx	Carbon Fiber
Tensile Modulus (GPa)	2.4	60
Tensile Strength (MPa)	37	800
Tensile Strain at Break (%)	25	1.5
Flexural Strength (MPa)	71	540
Flexural Modulus (GPa)	3.0	51
Heat Deflection Temp (°C)	145	105
Density (g/cm ³)	1.2	1.4

TABLE 4 Material properties of Onyx[®] and carbon fiber, as presented in the datasheet by Markforged [13].

3.2 | Specimen design

3.2.1 Material level: sample design

The specimen that is used in the tensile tests is dogbone shaped and follows the ISO 527-2 Type 2B norm [37]. It is constructed using different fiber volume fractions. Although Markforged's slicer software, Eiger, gives some customization options in terms of print orientation, infill pattern and layer height, a comprehensive module for designing the fiber placement and orientation lacks. The parameters available in Eiger for controlling fiber placement include the number of layers containing fiber reinforcement, the geometric region in which fibers are placed, and the orientation of fiber strands. In this study, variation in fiber volume fraction is primarily controlled by two parameters: (1) adjusting the number of Onyx[®] wall layers, and (2) modifying the selection of layers in which continuous fibers are deposited.



(a) When printing in the 'flat' orientation, a stress concentration occurs where layer lines meet. (b) When printed in 'upright' orientation. Blue lines represent carbon fiber layers.

FIGURE 7 Comparison of the possible ISO 527-2 Type 2B print orientations.

In first instance, the specimen was placed on the printbed in a flat orientation to acquire maximum adhesion of the first layer to the build plate. However, as observed during tensile testing, samples printed using this configuration consistently fractured at the same location, namely at the end of the gauge length where the specimen widens again. The cause of this behavior was identified from the cross-sectional view of the part in Eiger. It was observed that this printing orientation caused layer lines from different directions to meet at a central point. This can lead to small voids, creating a local defect in the structure and resulting in a stress concentration at that location. To prevent this issue, the part was reoriented in an upright position, which required a support structure underneath the gauge length area, and a brim to ensure good bed adhesion. The part was printed in 4 variations: as pure matrix material, or with either 1, 2 or 3 Onyx[®] wall layers and carbon fiber strands as infill. The fiber volume fraction could not be directly obtained from the expected material usage in Eiger, as the wider ends of the specimen would lower the overall fiber fraction. Instead, only the relative amount of fibers within the gauge length should be taken into account. Therefore, the actual fiber ratio was calculated by using the following equation:

$$V_f = \frac{0.9}{4} \sum_{k=1}^3 \frac{l_{f,k}}{l_t} k \quad (27)$$

Where $l_{f,k}$ is the number of fiber layers with k strands, l_t is the total number of layers, 0,9 mm is the width of one fiber strand, and 4 mm is the total thickness of the part. The fiber ratios of the samples were determined to be 5,2%, 41,5% and 46,7%, respectively. The results of the tensile test for these samples are presented in Figure 18.

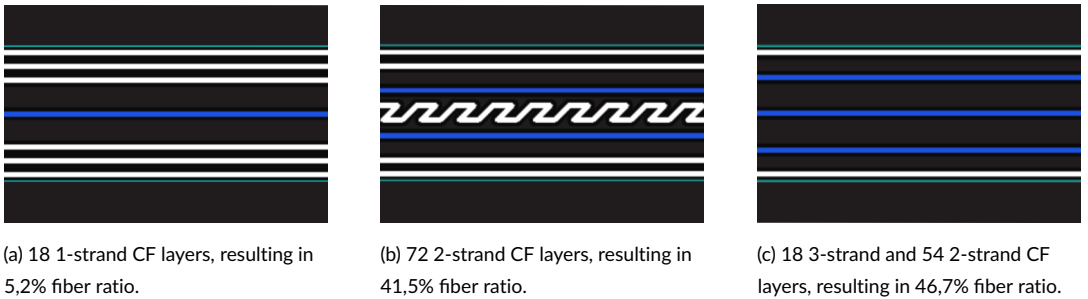


FIGURE 8 Top view of the ISO 527-2 specimen with different fiber volume fractions when printed in 'upright' position. White lines represent Onyx[®] strands, while blue lines show carbon fiber strands.

As shown in Figure 8, specifying a fixed number of wall layers did not necessarily result in a constant number of fiber strands. For specimens with an uneven number of matrix wall layers, the number of fiber strands varied by one in one out of every four layers. This behavior of the slicer software was unexpected and could not be corrected, which partly explains why the fiber volume fractions of the different specimens were not evenly spaced. To obtain a more evenly distributed range of fiber volume fractions across the samples, the print orientation of the part was reconsidered. When reoriented to the 'flat' configuration, the cross section of the part essentially becomes wider, allowing for a greater number of fiber strands to be deposited per layer. The previously observed issue of stress concentration was mitigated by allowing a diagonal infill pattern for the matrix material, thereby eliminating the critical location where layer lines converge (see Figure 9).

This final configuration resulted in five different sample groups, including three with varying levels of carbon fiber content. The remaining two groups consisted of 'matrix only' specimens, with one printed in the original 'upright' orientation and the other in the revised 'flat' orientation to analyze the effect of layer orientation when a diagonal infill pattern is allowed. The results of the tensile test for these samples are presented in Figure 19.

To obtain the flexural modulus of the 3D printed composite, rectangular shaped specimens were created following the ASTM D790 standard [38], which was mentioned as the standard followed by the mechanical characterization as done by Markforged [13]. For this simple geometry, no stress concentrations can occur in the middle of the part when printed in the 'flat' orientation. Therefore, this orientation is preferred, to allow the maximum variation in fiber placement in the component. As a result, no diagonal matrix infill was required and all strands could be oriented in the axial direction of the component. The sharp turns of the matrix and fiber strands at the component's edges do not affect the flexural behavior, as they are situated in the overlap regions outside the lower supports of the three-point bending test. The results of this experiment for these samples are presented in Figure 22.

3.2.2 Structural level: sample design

As mentioned in Section 2.6, honeycomb cores are being manufactured to analyze the influence of the cell angle and the fiber inclusion on the overall bending stiffness of the sandwich structure. In the design phase, it was noticed that fiber could not be included in specific small parts. After doing more research on the design parameters for 3D printed composites, it was found that the minimum width of a part in which an open looped fiber strand could be placed is 3.6 mm. For parts where the endpoint of the fiber strand meets its beginning point (i.e. a closed loop part), the minimum thickness is given to be 2.8 mm [39]. Consequently, given the dimensional limitations of the three-point bending setup, the core structure was limited to a maximum of 2×8 unit cells (see Figure 10).

Due to this limited number of unit cells, the structure cannot be considered representative of an infinite honeycomb. All unit cells are located at the boundaries, meaning that edge effects dominate the mechanical response. As a result, the measured behavior does not fully reflect the intrinsic properties of the honeycomb core. Nevertheless, the results still provide a meaningful indication of the influence of fiber content on the mechanical behavior of the honeycomb structure. Since this limitation is consistent across all specimens, a comparative analysis between the different fiber configurations remains valid. To avoid boundary effects in the study on cell topology, these particular samples will be printed without fiber reinforcement.

The fiber reinforced honeycomb structures were designed with a fixed 55° cell angle to facilitate relatively smooth corners for easy fiber deposition. The two cell unit rows were manually separated by 0.001 mm in the CAD software, in order to force the slicing software to perceive the part as being 'closed loop', and to accept a 2.8 mm wall thickness of the unit cell. Due to the lower resolution of the printer, the two cell rows are still fully adhered. Contrary to the

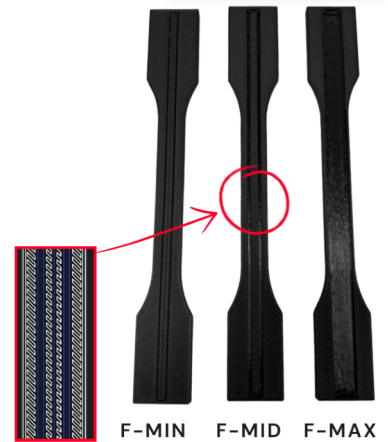
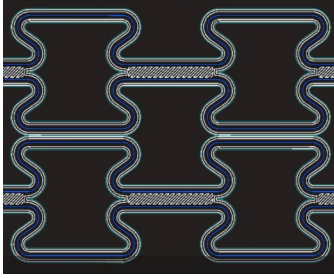
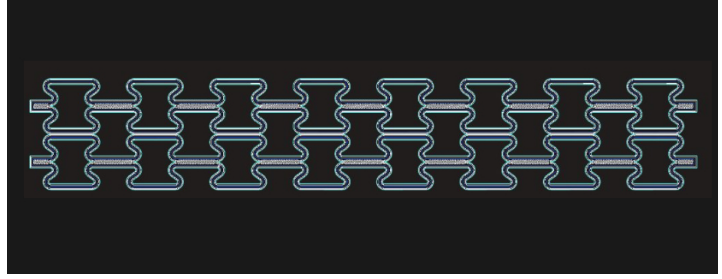


FIGURE 9 The ISO 527-2 samples printed in 'flat' orientation with diagonal matrix infill.

specimens used for the tensile tests, the amount of fiber per layer could not be adjusted, since only one fiber strand could be included in a part of this thickness. Instead, the fiber content of the core was altered by choosing the amount of layers that include fiber reinforcement. This resulted in 4 different samples, each with a different fiber content. The samples were labeled F-0, F-24, F-48 and F-96, indicating the number of layers that contain the fiber reinforcement (with 96 layers being the total number of layers of the part).



(a) Close-up look of the fiber reinforcement.



(b) The core has a limited number of unit cells (2×8), which implies that edge effects play a role.

FIGURE 10 The design of the fiber reinforced honeycomb core.

As mentioned earlier, the second sample group that varies in core topology was not reinforced with fiber, which enabled a smaller cell wall thickness. As a result, less cells are present on the structure's boundary, and edge effects are reduced. The thickness of the cell wall for these samples was reduced to 0.6 mm and the cell angles for the three sample groups were -140° , 55° and 30° , respectively. The relative density for these structures is determined as follows:

$$\rho_{rel, -140^\circ} = \frac{m}{\rho_s \cdot V} = \frac{96}{1.2 \cdot 30 \cdot 5 \cdot 1.2} = 0.44 \quad (28)$$

$$\rho_{rel, 55^\circ} = \frac{m}{\rho_s \cdot V} = \frac{54}{1.2 \cdot 30 \cdot 5 \cdot 1.2} = 0.25 \quad (29)$$

$$\rho_{rel, 30^\circ} = \frac{m}{\rho_s \cdot V} = \frac{37}{1.2 \cdot 30 \cdot 5 \cdot 1.2} = 0.17 \quad (30)$$

Where m is the mass of the part in grams, ρ_s is the density of the printed material in g/cm^3 and V is the volume of the part in cm^3 . Based on the relative density of the different sandwich structures, the 140° panel is expected to exhibit the highest bending stiffness, followed by the 55° and 30° structures. However, this prediction does not account for the influence of layer line orientation on the flexural modulus of the part, which must be demonstrated by the 3 point bending experiment. The different cell geometries were inspired from recent work by Vladimír Hostinský [36], where a set of seven parameters is used to fully describe the geometry of the core. Hostinský's geometry is based on the honeycomb cell model by Gibson and Ashby [35], where cellular materials are described as periodic networks of beams, allowing their mechanical properties to be derived directly from unit cell geometry and base material properties. The effective Young's modulus in the loading direction can be expressed according to Gibson and Ashby as a function of the cell geometry:

$$E_x = E_s \left(\frac{t}{l} \right)^3 \frac{\cos \theta}{(\sin \theta + h/l) \sin^2 \theta} \quad (31)$$

This expression relates the effective Young's modulus in the x-direction to the geometry of the unit cell. It shows that stiffness is governed not only by the relative density $(\frac{t}{l})^3$, but also strongly by the cell angle θ and the ratio $\frac{h}{l}$. This highlights the significant influence of cell geometry on the mechanical response of the structure (see Figure 6).

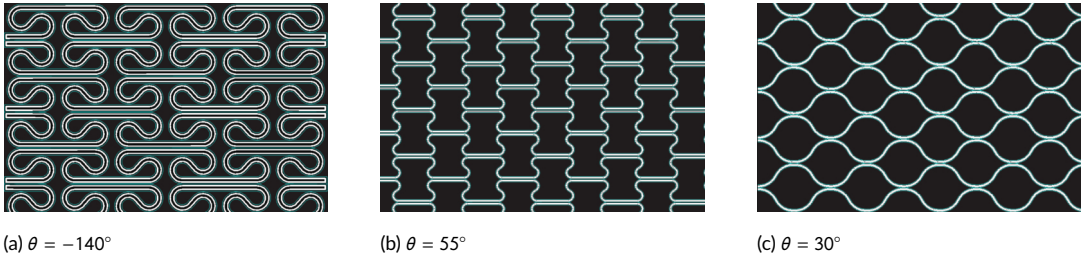


FIGURE 11 Cross sectional top view of the honeycomb core structure with varying cell angle θ .

3.3 | Manufacturing process

The sandwich structures were manufactured in the Meta Mechanics group laboratory at the University of Groningen, where both the Markforged Mark Two 3D printer and the Instron 34SC-5 tensile test machine were located. During the manufacturing process, the Markforged continuous fiber printer encountered several issues regarding clogs in both the fiber and plastic extruder and nozzle. To resolve the issue, procedures were executed ranging from simple tasks like reloading material, to a complete removal of the print head and nozzle. During the removal of the nozzle, it was tightened to such extent that its thread broke during removal, and a tap extractor was required to be able to install a new nozzle again. Furthermore, during the second part of the project, prints tended to stop regularly mid-print, without any obvious reason. After several consultations with support from Markforged, it was determined that an oversensitive extrusion sensor measured a slight increase in friction when the nozzle passed over a damaged part of the print bed, causing the print to stop. As an immediate solution, the sensor was disabled and printing was limited to undamaged areas of the print bed. Apart from printing errors, the (surface) quality of the correctly printed parts was perceived to be more than satisfactory, with almost no post-processing required (apart from removing support structures).



FIGURE 12 Both plastic and fiber clogs occurred regularly, resulting in printing delays.

It was decided that the face sheet material should comply with the following requirements:

1. The material must be transparent, such that any defects occurring in the core can be observed closely.
2. Although typically $E_f \gg E_c$, in this sandwich structure the face stiffness may not greatly exceed that of the core, causing the core to play a more significant role in the overall structural response.
3. The material must behave as a linear elastic material. If the face sheets remain linear elastic and do not yield, any

early or progressive failure observed is more likely to originate in the core, rather than being masked by plastic deformation in the faces.

The material selected based on these guidelines is acrylic glass, as it is transparent, brittle, and exhibits little to no plastic deformation. As a result, stress redistribution is limited, and failure occurs suddenly once the tensile strength is exceeded. Furthermore, the material is readily available and affordable. The acrylic panels were cut to size by performing many cutting passes with a utility knife along a thick ruler, to avoid stress cracking or crazing. The material properties were experimentally determined, yielding a Young's modulus of 2117 MPa and a flexural modulus of 3978 MPa.

The adhesion between the face sheets and sandwich panel core is another important design choice, which may strongly influence the results of the 3 point bending test if applied inconsistently [40]. It was found that epoxy adhesives bond well to polyamide plastics, like Onyx®, due to its nylon based matrix, allowing both chemical adhesion and mechanical interlocking with the printed surface [41]. In addition, epoxy can bond to the smooth surface of acrylic through intermolecular (van der Waals and polar) interactions. Therefore, a two-component epoxy adhesive from Bison was selected to bond the sandwich core and face sheets. The adhesive was weighed to ensure an identical quantity for each sandwich panel and was applied using a ruler to achieve an even distribution across the face sheet. Since the face sheets are transparent, the adhesion could be visually inspected directly.

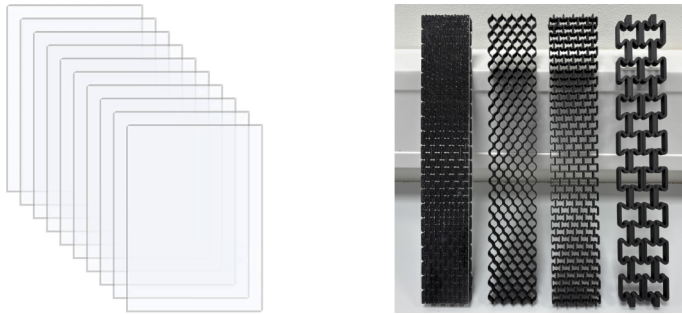


FIGURE 13 1 mm acrylic plates were used as face sheets (left) and bonded with epoxy adhesive to the different core geometries (right).

3.4 | Experimental methods

As shown in section 2.3, many different experimental methods are available for determining the mechanical properties of (continuous fiber reinforced) 3D printed materials. In this section, the methods that were used during this study will be elaborated.

3.4.1 Tensile Test

First of all, a tensile test was performed to determine the mechanical properties, the Young's Modulus in particular, of the 3D printed material Onyx®, with several fiber ratios. The Instron 34SC-5 machine was used for this experiment. The test was executed following the ISO 527 standard, which recommends a 1 mm/min strain rate for the determination of an elastic modulus. 3 specimens were tested per sample group, which consisted of 0%, 5, 2%, 41, 5% and 46, 7% fiber content. After the first test round, it was noted that graphs did not linearly start from the origin, but from around

0.2% strain instead. As this occurred for all samples, it was concluded that this effect was happening due to initial seating in of the specimen in the grips of the machine. Therefore, a second round of tensile tests was executed to verify the results, with a pre-strain applied to decrease clamping effects. This pre-strain was set to have a strain rate of 0.5 mm/min until 10 N, after which the tensile displacement was balanced. In this experiment, the sample groups were further apart in terms of fiber content, namely 0% ('upright'), 0% ('flat'), 16.6%, 31.3% and 66.3% of carbon fiber volume.

3.4.2 Three-point bending test

Since a main research objective in this study is to discuss the flexural behavior of the 3D printed materials, a three-point bending test was performed subsequently, both on material and structural level. First, the rectangular shaped ASTM D790 specimens were placed in the Instron 34SC-5 machine, that had been converted to a different clamping setup to accommodate the experiment. According to the standard, the clamps must be placed apart at a distance equal to 16 times the thickness of the specimen. With a thickness of just over 3 mm, the clamps were placed 48.8 mm apart, which ensured enough overhang for boundary effects to be minimised. Also a 1 mm/min strain rate was used here, and 5 specimens were tested per sample group to improve accuracy.

3.4.3 Dynamic Mechanical Analysis (DMA)

As the results from the initial tensile test did not show satisfactory accurate results for determining the Young's Modulus of the printed material, dynamic testing methods were explored, out of which the Dynamic Mechanical Analysis was attempted first. The Perkin Elmer DMA 8000 was used for this experiment. The clamping setup that was installed in the machine was meant for a single cantilever test, where one end of the specimen is clamped, and the other is free. A small oscillating force is applied, causing the sample to bend back and forth. The specimen's dimensions were designed to be as large as possible according to the machine's manual for this configuration (25 x 12 x 4 mm), to suit the manufacturing method 3D printing the best. In this DMA experiment, a frequency sweep of 0 to 100 Hz was applied to analyze viscoelastic effects. Due to the nature of the single cantilever experiment, the results from the DMA show flexural properties of the material, rather than a tensile elastic modulus. Therefore, other dynamical approaches were explored.

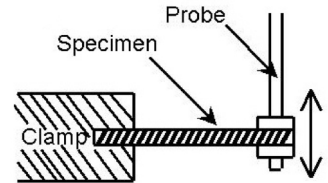


FIGURE 14 Schematic representation of DMA setup with a single cantilever geometry [42].

3.4.4 Time of Flight experiment (ToF)

The time of flight experiment is one such approach that is often used in literature to find the elasticity modulus of a sample. First, a through-transmission experiment was done, which is a method where two transducers are placed on opposite sides of a part. One transducer transmits a signal, while the other receives it. The time it takes for the signal to travel through the material can be related to the Young's modulus of the material, once the Poisson ratio and dimensions of the specimen are known:

$$E = \rho c_L^2 \left(\frac{(1 + \nu)(1 - 2\nu)}{1 - \nu} \right), \quad (32)$$

where the longitudinal wave speed is given by $c_L = h/\Delta t$, h is the thickness of the sample, and Δt is the time of flight of the signal. The Poisson's ratio ν for both Onyx[®] and carbon fiber was found to be 0.35 and 0.33, respectively, from other independent literary sources that used the same material and 3D printer [43][44].

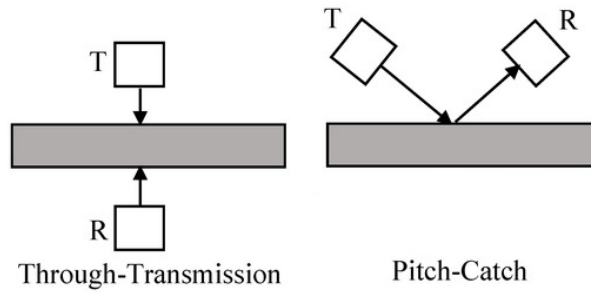


FIGURE 15 A schematic showing the two different ToF methods that were used in the experiment, through-transmission and pitch-catch [45].

The Smart Materials MFC P3 piezoelectric patch was used as the transmitter of the signal. A Laser Doppler Vibrometer (LDV) from the brand Polytec was used to receive the signal. The setup also includes an amplifier for signal excitation and a data acquisition (DAQ) system for recording and processing the measured signals. The sample was placed on a porous foam material to simulate free-free boundary conditions.

In another variant of the ToF experiment, called the pitch-catch method, both transducers are placed on the same side of the part. By adapting the distance between the transmitter and the receiver, the time of flight can be plotted against the distance from the excitation point, which ideally would result in a linear relationship. In this way, ToF results can be verified.

Unfortunately, both experiments led to ambiguous results, with no clear distinction in between peaks in the time domain plot to analyze the time it took for signal to travel through the materials. This can be caused by several factors:

1. 3D-printed polymers (especially fiber-filled like Onyx®) strongly attenuate ultrasonic waves. A sufficiently high frequency is required to obtain a resolution that is high enough to obtain a clear defined arrival time, but in highly attenuating materials such as 3D-printed polymers, higher frequencies are strongly damped, leading to weaker signals.
2. Layered printing, pores, and fiber inclusions cause wave scattering, leading to signal spreading. ToF experiments are very sensitive to irregularities in materials, which is why it is also a common tool for detecting defects in structures.
3. Mechanical properties and wave speed vary with print direction, leading to inconsistent ToF results and making it difficult to define a single representative modulus, especially when continuous fibers are being examined.

Consequently, other mechanical characterization methods were explored to find a suitable fit for this specific 3D printed material.

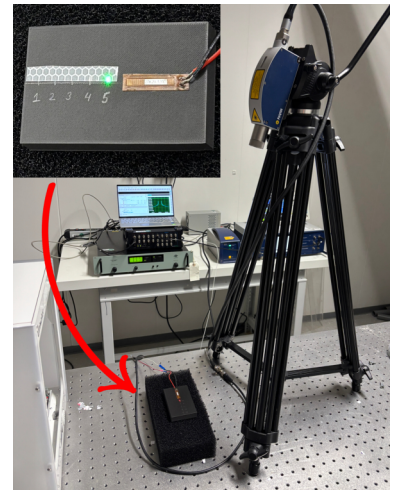


FIGURE 16 The experimental setup for the ToF pitch-catch experiment.

3.4.5 Modal Analysis

Another dynamic mechanical characterization method involving the excitation of wave signals is modal analysis. In this experiment, the different modes of excitation can be correlated to a mechanical property of the tested material, like the Young's Modulus. A mode is a characteristic vibration pattern of a structure associated with a specific natural frequency. These frequencies, also known as eigenfrequencies, can be quantified either by analyzing the frequency response of the sample or by visualizing the vibration patterns of the corresponding mode shapes.

The sample used in this experiment was manufactured as a thin plate. Thin plates are commonly used in modal analysis as they exhibit clear, well defined bending modes that are easy to excite and analyze, allowing accurate characterization of mechanical properties. In engineering, a plate is commonly considered thin when its thickness is no greater than 1/20 of its in-plane dimensions. Therefore, the sample used in this experiment was designed with dimensions of 150 x 150 x 1 mm. For the frequency response analysis of the plate, a piezoelectric patch was used to excite the signal and was positioned off-center to avoid placement at a nodal location for multiple mode shapes. The signal was measured by the LDV at four different locations on the plate's surface. These locations were carefully selected to minimize overlap with nodal lines, such that the maximum amount of natural frequencies could be excited and measured. The excitation signal used in this experiment was a sinusoidal frequency sweep ranging from 1 to 1000 Hz.

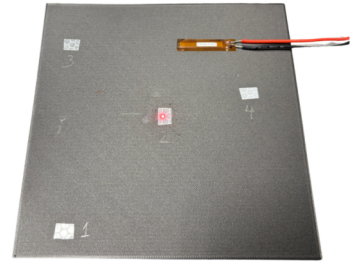


FIGURE 17 The thin square plate used in the modal analysis with its excitation and measurement points carefully positioned, taking into account possible nodal lines.

The eigenfrequencies of a sample can also be identified by exciting it at resonance, where the corresponding mode shapes can be visualized through vibration patterns formed by small particles such as sand or salt. The shape that emerges on the plate is referred to as a Chladni pattern. The particles gather around the nodal lines, which are sections where plate remains stationary. In the experiment, the plate was being excited by using the Philip Harris Vibration Generator, which was attached to a small M4 bolt glued to the bottom of the plate. By matching the experimentally observed mode shapes with those obtained from a finite element analysis (FEA) in COMSOL Multiphysics, the corresponding modes and their associated frequencies can be identified. In this process, an eigenfrequency study is performed in which the sample geometry is defined, material properties such as density and an initial estimate of the elastic modulus are assigned, and appropriate boundary conditions are applied to represent the experimental setup. The model is then discretized using a finite element mesh, and the eigenvalue problem is solved to obtain the natural frequencies and corresponding mode shapes. Subsequently, the elastic modulus in the FEA model is adjusted until the simulated eigenfrequencies align with the experimental results, allowing the modulus of elasticity of the material to be determined. The two above mentioned methods can also be combined to achieve a more accurate result.

3.5 | Data processing and analysis

Raw experimental data that follows from the methods that were elaborated above is converted into into valuable graphs and tables, out of which the requested mechanical properties can be determined. The main tools that are used to process the data are Microsoft Excel and MATLAB. The experimentally gathered data is then checked for validity and compared to simulated values from Comsol Multiphysics® and the theoretically expected results.

4 | RESULTS & DISCUSSION

4.1 | Quasi Static material-level results

This section will present the experimentally obtained mechanical properties of the printed materials. Measured Young’s modulus, strength values, and stiffness parameters will be reported and compared with manufacturer datasheet values and theoretical predictions. Variability between specimens and repeatability of the tests will also be evaluated.

4.1.1 Tensile Test

The first results are obtained from a tensile test performed on pure Onyx samples, and Onyx samples reinforced with carbon fiber at different fiber volume fractions. These results provide an initial comparison of the mechanical response of the materials, illustrating the influence of fiber content on stiffness and strength.

Sample Group	Specimen	Young’s Modulus (MPa)	Average Young’s Modulus (MPa)	Carbon Fiber Content (%)
Pure Onyx	T9	2241.4	2173.366667	0
	T10	1939.5		
	T11	2339.2		
3 Walls	T3	2557.3	2966.166667	5.2
	T7	3181.4		
	T12	3159.8		
2 Walls	T2	4004.9	3741.7	41.5
	T6	3478.5		
	T1	4527.1		
1 Wall	T5	4773.2	4741.85	46.73
	T4	4815.3		
	T8	4668.4		

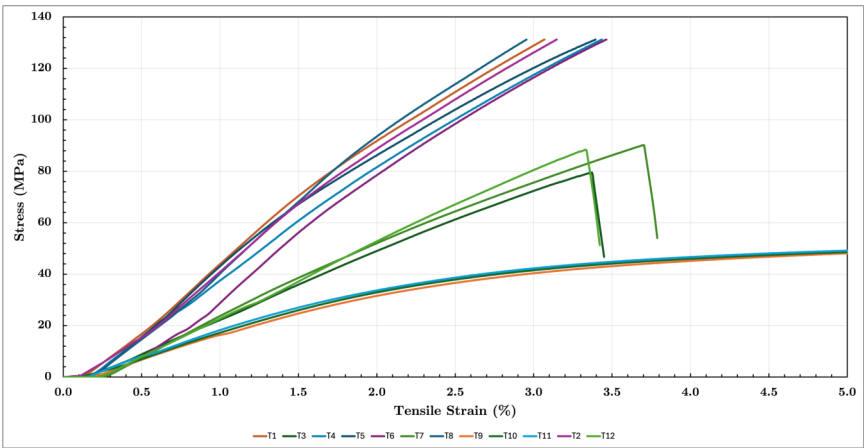


FIGURE 18 The stress–strain curves from the first tensile test results show that the addition of a small amount of carbon fiber has a clearly distinguishable effect on the material response, while samples with higher fiber fractions exhibit more similar behavior and are less easily differentiated.

From the first tensile tests, the following observations were made:

1. Samples printed with Onyx® only show a very consistent stress-strain response across all specimens.
2. The addition of fibers introduces a brittle fracture at much smaller strains. Adding more fiber content postpones this point of fracture, up to the point that the 5 kN load cell of the Instron 34SC-5 was not enough to break two sample groups with the highest fiber content.
3. For all samples, the graph is not originating from the origin, but from approximately 0,2% strain instead. The Young's modulus was determined from the slope of the linear trend line fitted to the initial linear region between 0.35% and 0.50% strain. In the second round of tensile tests, a pre-strain was applied to ensure that the curves started from the origin, as discussed previously in Section 3.4.1.

4.1.2 Tensile Test with pre-strain

In the second round of tensile test, 5 different sample groups were tested with a pre-strain applied. A detailed description of the samples that were manufactured for this experiment can be found in Section 3.2.1.

Sample	Specimen	Young's Modulus (Mpa)	Average Young's Modulus (Mpa)	Carbon Fiber Content (%)
FO	FO-1	1301.9	1252.9	0
	FO-2	1243.5		
	FO-3	1213.3		
FO2	FO2-1	2668.4	2732	0
	FO2-2	2781.8		
	FO2-3	2745.8		
FMIN	FMIN-1	3914.2	3790.3	16.6
	FMIN-2	3666.4		
FMID	FMID-1	4253.6	4242.2	31.3
	FMID-2	4230.8		
FMAX	FMAX-1	5232.4	5067	66.3
	FMAX-2	4901.6		

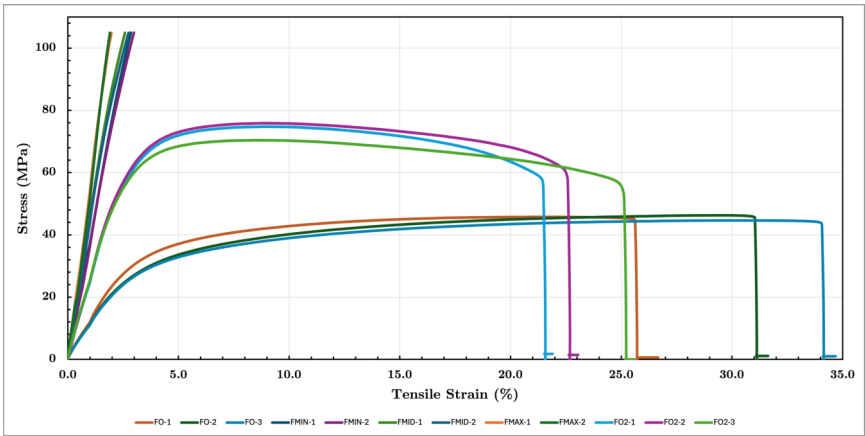


FIGURE 19 The stress-strain curves from the second tensile test results show a more consistent response and indicate that the orientation in which the part is placed has a significant effect on its stress-strain behavior.

These results provide further insight into the mechanical behavior of parts made from pure Onyx and the influence of print orientation. It was observed that samples with layer lines printed in the axial direction (FO) exhibit a significantly higher elastic modulus than those printed diagonally. This is expected, as the filaments are aligned parallel to the loading direction, allowing the load to be carried directly through the continuous polymer strands. As a result, these samples also show more brittle behavior, with fracture occurring once the strands themselves fail. In contrast, the group of samples printed in the 'flat' orientation (FO2) exhibits an average Young's modulus that is less than half of that of the 'upright' group. This is primarily due to the layer lines being oriented at an angle to the loading direction, causing load transfer to occur through shear between rasters and interlayer bonding. This shear-dominated deformation is more gradual, as it allows for redistribution of strain within the structure. Consequently, these samples are able to sustain higher strains before failure compared to the FO group.

For the parts containing carbon fiber, the distinction becomes more pronounced, which is likely due to the larger differences in carbon fiber content between the sample groups. Although a pre-strain was applied, the initial response of some of these specimens still appears somewhat irregular, which could be due to slight misalignment during loading, or local inhomogeneities in the printed structure affecting the initial load transfer. Nevertheless, the Young's modulus values of the different samples appear to be more consistent, with significantly smaller deviations between specimens within the same sample group compared to the first round of tensile tests.

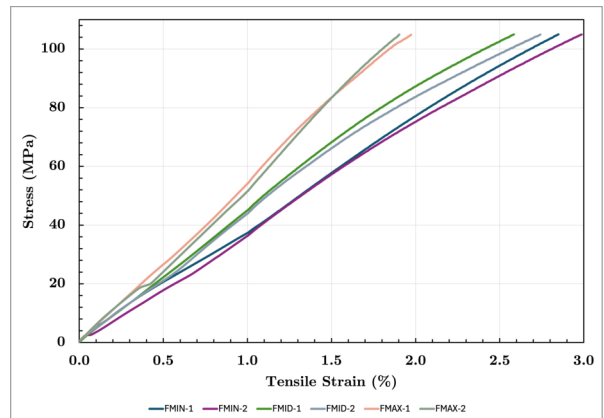
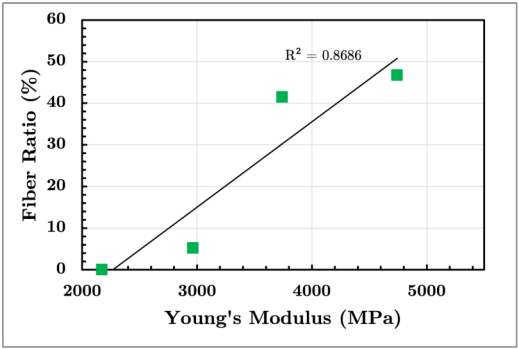


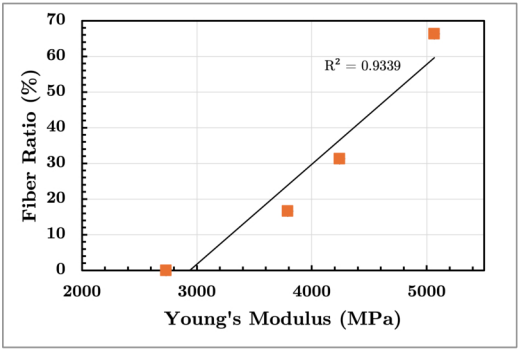
FIGURE 20 The stress-strain curves from samples that contain carbon fiber in the second tensile test.

From theory (e.g. Rule of Mixtures, see Section 2.2), and other numerical literary work, it can be predicted that the relationship between fiber content and Young's Modulus of a 3D printed sample is linear [46]. From the first tensile tests, a linear correlation was difficult to recognize, which could be explained by the inconsistencies in the found Young's Modulus values within the same sample group; individual results occasionally deviate between 10%-15% from the average value of that group (e.g. T1 and T10). This makes the average value less meaningful, and therefore no reliable conclusions can be drawn about the linearity of the relationship between fiber content and Young's modulus based on these figures alone (see Figure 21a).

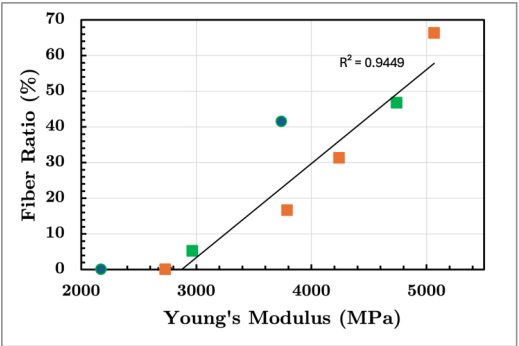
From the second series of tensile test, a similar graph was made. Although data points still do not provide a perfect linear fit, the R-squared value of the best possible trend line increased from 0,86 to 0,93, indicating a more reliable result. The figure becomes more interesting when the results of both tensile test series are being combined. Here, the outliers of the first tensile test series are not taken into account (the sample groups with 0% and 41,5% carbon fiber content), as well as the pure Onyx® sample that was printed in flat orientation. As mentioned before, the diagonal layer lines in that sample do not resemble the true tensile modulus of the material.



(a) First series of tensile tests.



(b) Second series of tensile tests.



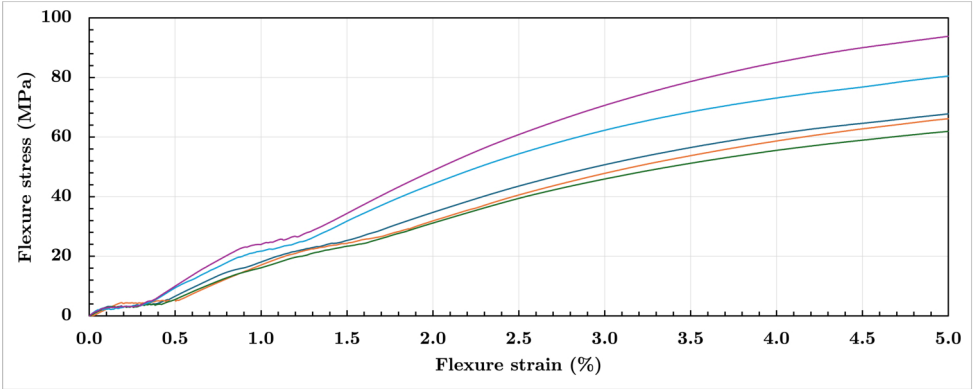
(c) First and second series of tensile tests combined, with outliers not taken into account for determining linearity (blue and circular shaped points).

FIGURE 21 Graphs showing the relationship between Young's Modulus and Fiber Ratio. The R-squared value could be increased by combining the two series of tensile tests and removing unreliable outliers.

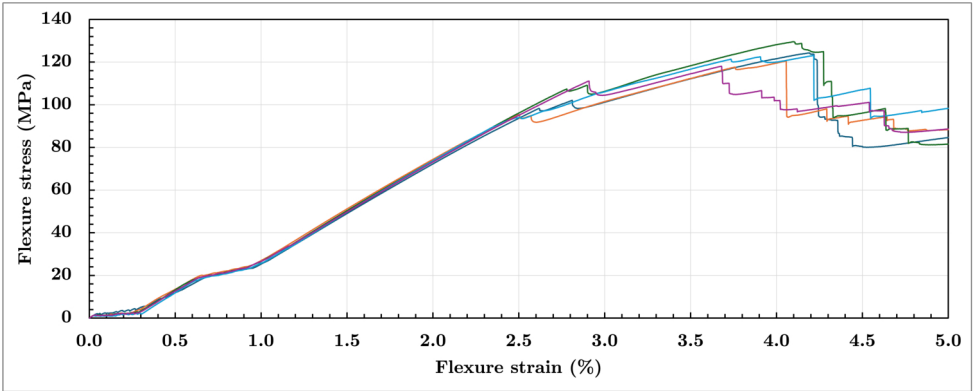
4.1.3 3-point bending test

In order to obtain the flexural modulus of Onyx[®] and analyze the influence of fiber reinforcement on its elasticity, 3-point bending tests were performed. For a more detailed explanation of the testing procedure and parameters, see Section 3.4.2. The results from the experiment are shown in flexural stress-strain graphs in Figure 22. The following three conclusions can be drawn from these outcomes:

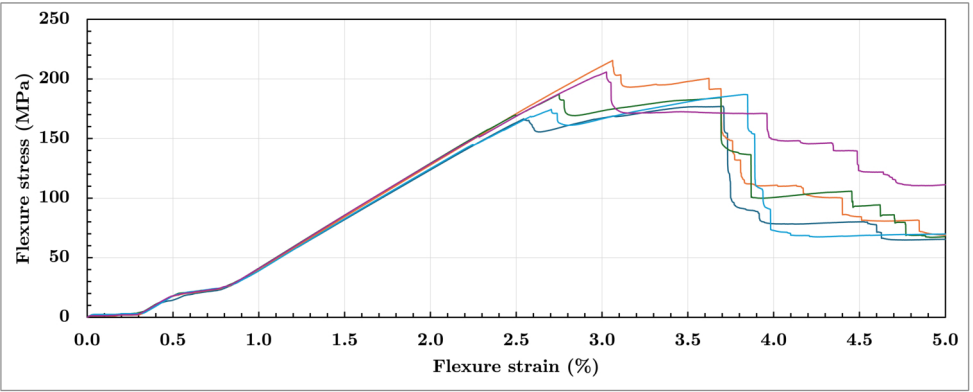
1. The addition of carbon fibers significantly increases stiffness and strength, but changes the failure behavior from ductile to a more brittle, progressive failure mechanism characterized by sudden stress drops due to the breakage of individual strands of fibers in the sample.
2. The Onyx[®] + carbon fiber samples exhibit a more consistent and repeatable linear response, as indicated by the close alignment of the curves in the initial region. In contrast, the pure Onyx samples show greater variation between specimens, with the curves more widely dispersed. This suggests that the addition of carbon fiber reduces variability and leads to a more predictable mechanical response.



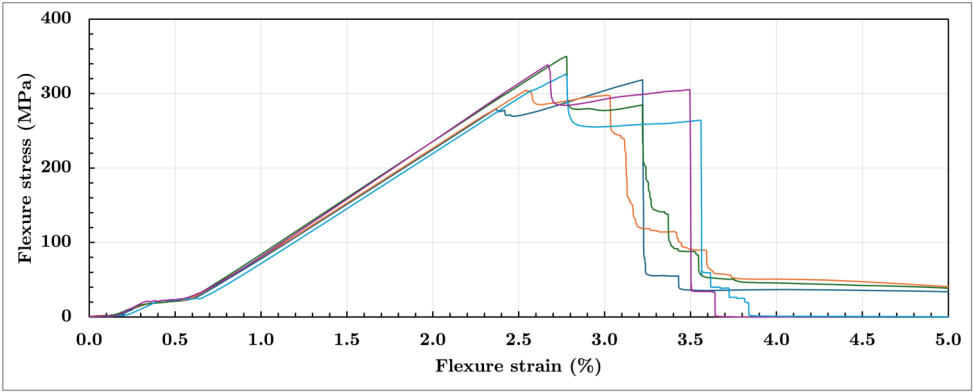
(a) Fiber ratio = 0%, average flexural modulus = 2,735 GPa.



(b) Fiber ratio = 12,2%, average flexural modulus = 5,482 GPa.



(c) Fiber ratio = 32,3%, average flexural modulus = 8,652 GPa.



(d) Fiber ratio = 60,8%, average flexural modulus = 14,189 GPa.

FIGURE 22 Flexural stress–strain curves of Onyx[®] + carbon fiber samples, obtained from three-point bending tests in accordance with ASTM D790.

3. All samples display a step-like behavior in the initial phase of the curves, likely caused by initial settling in the test setup, minor slipping at the supports, or local rearrangement of the printed microstructure before stable loading is achieved. The test was initiated by bringing the upper clamp into minimal contact with the sample, after which the displacement was set to zero. This procedure can be considered a form of pre-strain. It was found during later bending tests with sandwich panels that the step-like effect could be minimized by further tightening the clamps.

The relationship between flexural modulus and fiber content is analyzed by taking the average flexural modulus for each sample group and plotting these values against the corresponding fiber ratio. An almost perfect linear correlation is observed, indicating good agreement with theoretical expectations and confirming the validity of the experimental results. For this reason, the three-point bending experiment was not repeated, as it was shown that the initial step-like artifact in the graph had a negligible effect on the results.

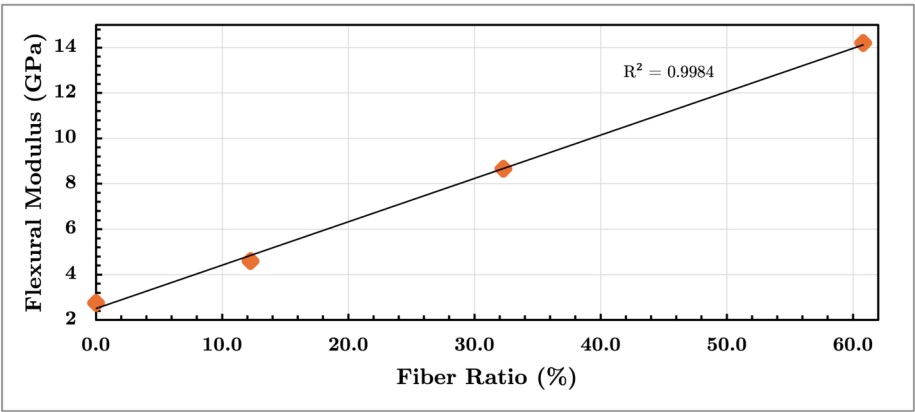


FIGURE 23 A graphical representation of the relationship between flexural modulus and fiber ratio. As expected, these variables exhibit a near-linear correlation, with an R^2 value of 0.99.

4.2 | Dynamic material-level results

4.2.1 Dynamic Mechanical Analysis (DMA)

The first dynamical mechanical characterization method that was used was DMA. For more information on DMA, the test parameters and sample dimensions, see 3.4.3. The results of the DMA on a pure Onyx[®] sample are shown in Figure 24. The storage modulus is analyzed here, as it directly represents the material's elastic stiffness, which is the property most comparable to conventional moduli such as Young's or flexural modulus. The graph shows that the storage modulus increases with frequency, reflecting the viscoelastic behavior of Onyx[®]. At low frequencies the polymer matrix relaxes more, resulting in lower stiffness, while at higher frequencies reduced molecular mobility leads to a higher storage modulus.

DMA with single cantilever geometry measures flexural stiffness rather than pure axial stiffness. In bending, shear deformation and interlayer effects are more significant, while tensile testing loads the filaments directly along their stiffest direction. Therefore, the storage modulus represents the dynamic equivalent of the flexural modulus, rather than the Young's modulus. Although both the storage modulus from DMA and the flexural modulus from the 3-point bending test both describe bending behavior, they should not be expected to coincide directly, as DMA provides a dynamic viscoelastic modulus while three-point bending yields a quasi-static flexural modulus.

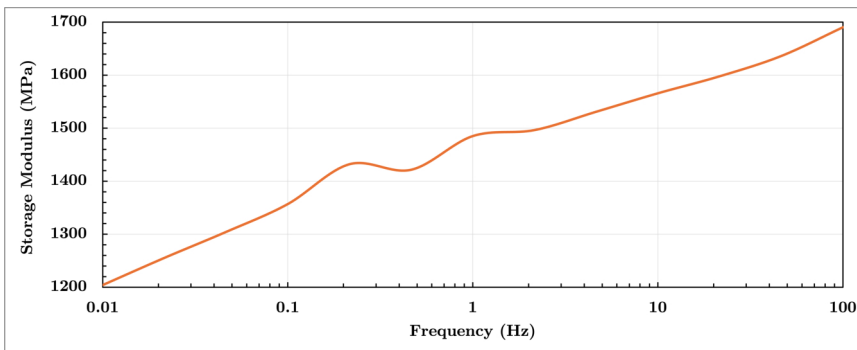


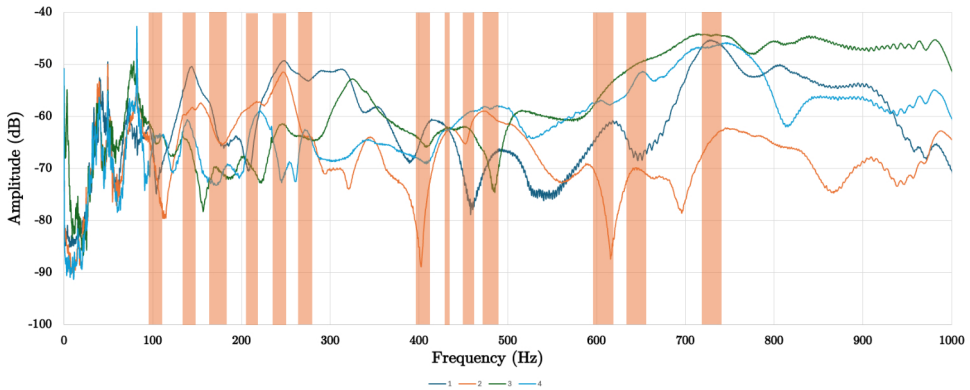
FIGURE 24 Storage modulus as a function of frequency obtained from Dynamic Mechanical Analysis (DMA) in single cantilever mode, showing the frequency-dependent stiffness of Onyx[®].

Nevertheless, the storage modulus values that were found in the DMA, ranging from approximately 1200 - 1700 MPa for 0.01 - 100 Hz, are significantly lower than the average that was measured in the 3-point bending experiment (2,735 GPa). This result can be explained by the following factors:

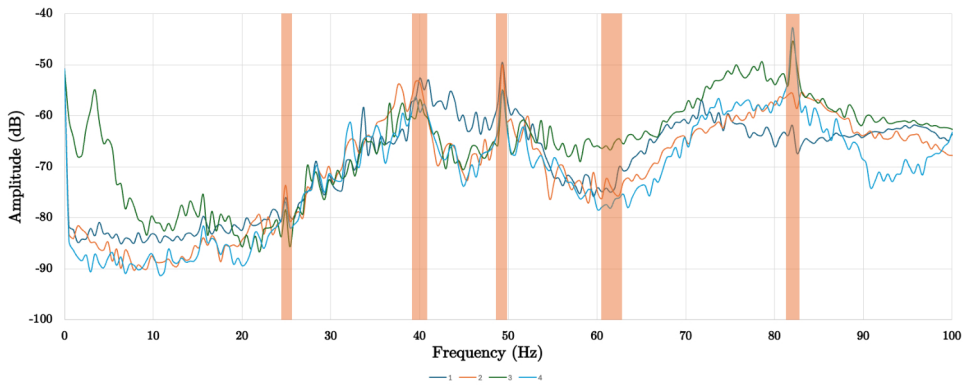
1. In DMA single cantilever mode, the measured stiffness is quite sensitive to clamp compliance and machine calibration. Any compliance in the fixture can reduce the measured storage modulus.
2. The sample geometry may play an important role. The DMA specimen is significantly smaller than the 3-point bending sample, which means that small geometric inaccuracies can have a strong influence on the calculated modulus. Furthermore, although the filament strands in both samples are oriented axially, the minimum number of wall layers is the same, which may cause edge effects to be relatively more pronounced in smaller samples.
3. And lastly, the modulus definition is different. The value from three-point bending is a flexural modulus derived from the initial slope of a static stress-strain response, whereas DMA reports storage modulus, which reflects the elastic part of the viscoelastic response under cyclic loading.

4.2.2 Modal Analysis

In the first modal analysis experiment, an sinusoidal frequency sweep was excited on a thin Onyx[®] plate, that was measured on 4 different places on the same surface of that plate. For a more detailed description of the experiment and sample design, see Section 3.4.5. The signal that was received on the different locations on the plate was recorded and plotted in Figure 25 as a Frequency Response Function (FRF). At one of the natural frequencies of the material, the sample is in resonance, and sharp peaks in amplitude may occur in the FRF. Especially when this peak is shown as a local maximum for several points on the plate, this strongly indicates a resonant frequency of the plate. In the FRF that is presented below, peaks that are clearly distinguishable from noise, such as at 49 Hz, 82 Hz and 402 Hz.



(a) FRF: 1-1000 Hz



(b) FRF: 1-100 Hz

FIGURE 25 Frequency response functions showing the amplitude of the received signal over two ranges of frequencies. The local maxima are highlighted with orange bands, indicating potential natural frequencies of the plate.

In addition to experiment, also a eigenfrequency analysis was done in Comsol Multiphysics[®] to simulate the resonant behavior of the material. The material properties that were used given as an input to this simulation were: $\nu = 0,35$, $\rho = 1200 \text{ kg/m}^3$, and $E = 2,732 \text{ GPa}$. Here, the assigned value for Young's modulus is entered as an initial estimate

based on the results of the second series of tensile tests (see Section 4.1.2). The results of the simulation show the first 30 eigenfrequencies of the plate, with its corresponding mode shapes. In Table 5, four mode shapes are selected, and the associated natural frequencies are noted. Since several different values for the Young's modulus have been found for Onyx[®] up to now, this simulation has been run for all those values; from the tensile test experiment in 'flat' orientation ($E_{exp,1} = 1252$), the value from Markforged's datasheet ($E_{dat} = 2400$), 90% of this value ($E_{0.9-dat} = 1920$), and the tensile test experiment in 'upright' orientation ($E_{exp,2} = 2732$). The value found in the modal analysis are compared to the simulated values below, to find the true Young's modulus of the material.

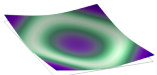
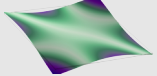
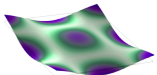
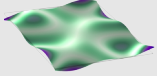
		Young's Modulus				Modal Analysis
		$E_{exp,1}$	$E_{0.9-dat}$	E_{dat}	$E_{exp,2}$	
Mode 1		54.7	67.5	75.5	80.6	82
Mode 2		75.6	93.3	104.37	111.4	107
Mode 3		138.6	171.1	191.4	204.2	212
Mode 4		290.46	358.7	401	427.88	402

TABLE 5 The eigenfrequencies and corresponding mode shapes obtained from the COMSOL simulation, compared with the values identified from the modal analysis.

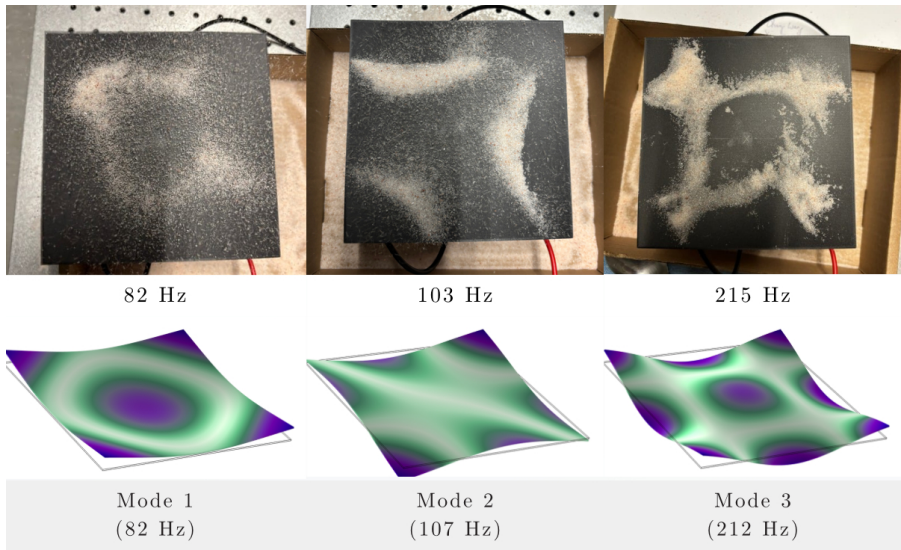


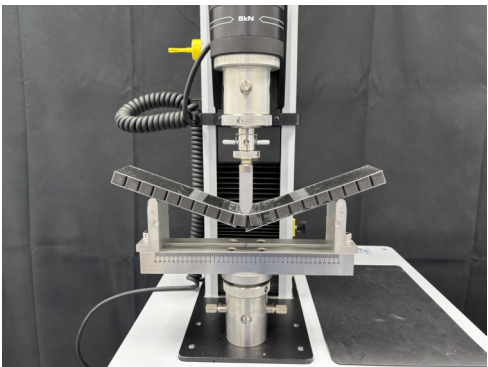
FIGURE 26 The mode shapes visualized during the Chladni experiment show similar patterns to those obtained from the COMSOL simulation, at frequencies close to those identified in the transmission experiment.

In order to verify that the eigenfrequencies that were found in the transmission experiment actually correspond to the mode that was simulated in FEA, its mode shape was visualized by performing a Chladni experiment. For modes 1, 2 and 3, a Chladni pattern was visualized that approached the mode shape that was found in the COMSOL simulation. For these modes, there is strong confidence that the identified eigenfrequencies correspond to the modes listed in the table for the different Young's modulus values. For the fourth mode, the Chladni pattern could likely not be excited due to the presence of a large nodal region at the center of the plate, which coincides with the excitation point (the M4 bolt connected to the shaker). A similar effect is observed for mode 2, although a slight visualization of the mode shape is still present. This is likely due to small asymmetries and imperfections in the setup, which allow the mode to be excited to a limited extent. As a result, the response at this frequency is present, although with a lower amplitude.

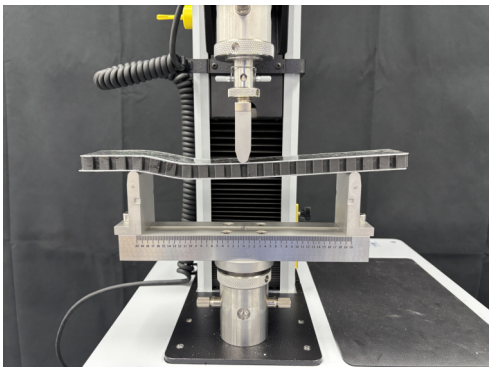
From the combination of the two modal analysis methods, it can be concluded that the Young's modulus of 3D-printed Onyx[®] lies between the datasheet value and the experimentally determined value from the second tensile test series, or possibly slightly above the latter. Quantitatively, this suggests that the actual Young's modulus of 3D-printed Onyx[®] is approximately in the range of 2500 to 2800 MPa.

4.3 | Sandwich panel results

On a structural level, the manufactured sandwich panels with (fiber reinforced) Onyx[®] cores and acrylic face sheets were evaluated on their flexural performance. A more detailed look on the manufacturing process of the sandwich structures can be found in Section 3.3. During the 3-point bending test that was performed on the sandwich structures, two main failure modes were observed. The first is a global bending mode (Figure 27a), characterized by a smooth and distributed deformation over the entire span of the specimen, up until the point where the lower face sheet on directly underneath the loading clamp fractures due to excessive tensile stresses [47]. The second is a localized failure mode (Figure 27b), where deformation is concentrated in a small region, and is not directly located underneath the loading clamp. This indicates a local instability rather than overall bending of the structure. In literature, this is referred to as the transverse shear failure of a sandwich structure, which generally occurs when shear stiffness of the core or face sheet is insufficient, but may also be caused by imperfect bonding between the core and the face sheets [48].

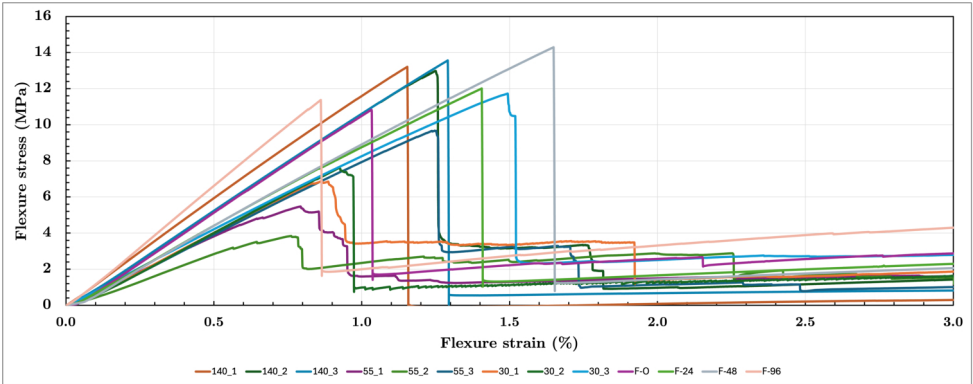


(a) Global failure mode: face sheet yielding fracture.

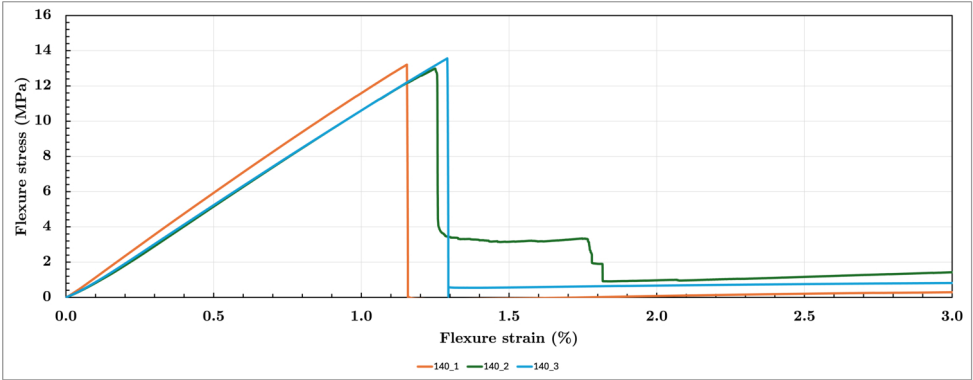


(b) Localized failure mode: transverse shear failure.

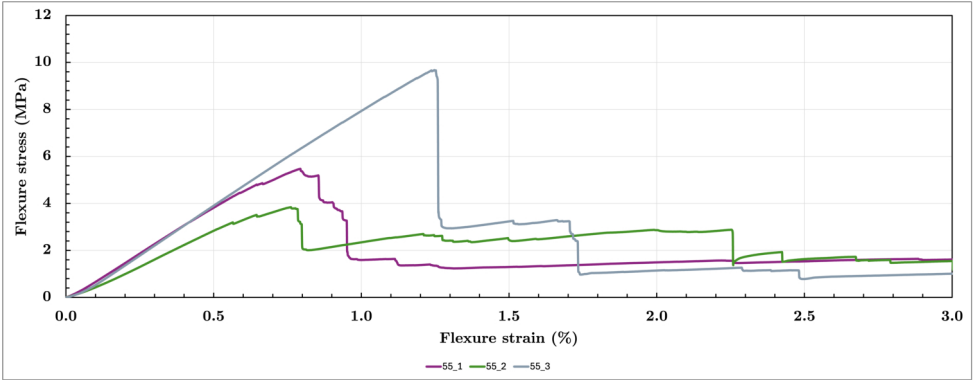
FIGURE 27 The two main failure modes that were observed during the 3-point bending tests of the manufactured sandwich panels.



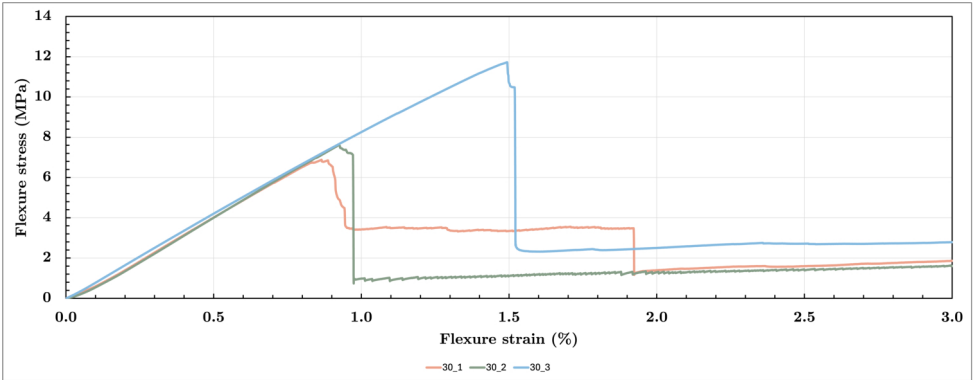
(a) Overview of the flexural behavior of all tested sandwich panels.



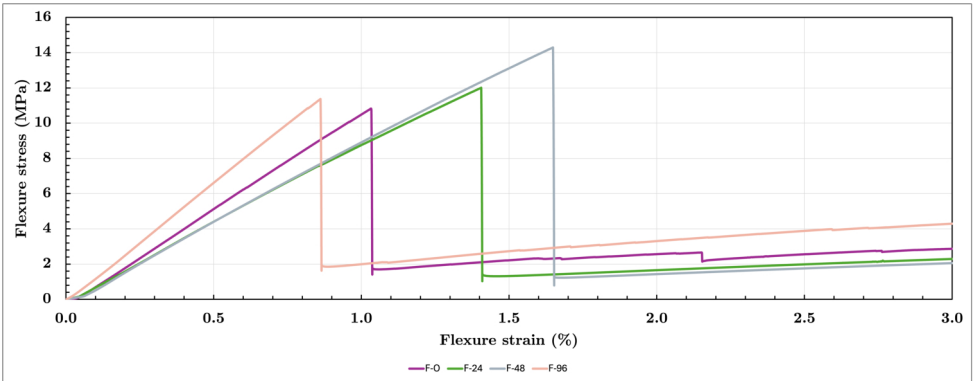
(b) Flexural behavior of samples with $\theta = -140^\circ$, manufactured from pure Onyx[®]



(c) Flexural behavior of samples with $\theta = 55^\circ$, manufactured from pure Onyx[®]



(d) Flexural behavior of samples with $\theta = 30^\circ$, manufactured from pure Onyx®.



(e) Flexural behavior of samples with $\theta = 55^\circ$, manufactured from Onyx® and varying carbon fiber content.

FIGURE 28 Flexural stress–strain curves for all tested sandwich panels, as well as for the individual sample groups.

Sample	Flexural Modulus (MPa)	Average (MPa)
140_1	1216.80	1085.98
140_2	993.45	
140_3	1047.70	
55_1	796.50	722.54
55_2	559.85	
55_3	811.27	
30_1	788.85	816.23
30_2	794.48	
30_3	865.36	
F-0	1084.20	
F-24	914.63	
F-48	976.71	
F-96	1331.30	

TABLE 6 Overview of flexural modulus values for all tested sandwich panels.

4.4 | Structural-level discussion

4.4.1 Effect of cell-angle on flexural response of sandwich panel

From the results of the 3-point bending experiment (Figure (28) and Table 6), the following observations were made:

1. Initial linear portions of the flexure stress-strain curves seem to be consistent over different specimens of the same sample group, especially of $\theta = -140^\circ$ and $\theta = 30^\circ$. For the graph of the $\theta = 55^\circ$ sample group, there is one major outlier when it comes to flexural modulus (specimen 55_2). This specimen was inspected during the test and it was concluded that the adhesion between the core and the upper face sheet was insufficient, reducing the effective load transfer between the two components, and thus the overall stiffness of the panel. Following the expectations from Section 3.3, the adhesion between the core and face sheets plays an important role in the performance of the panel; however, when applied correctly, the results for specimens with the same cell angle show good consistency. As the face sheets are identical for all panels, a valid comparison between the core configurations can be made.

2. Although linear regions of the flexural stress-strain curves correspond closely for the different specimens, there is a larger variation in the point of failure of the sandwich structures. It was observed that the specimens that fracture according to the first failure mode, as shown in Figure 27a, it can sustain a higher strain before failure (i.e. for specimens 140_3 and 30_3). Other specimens showed a more step-like failure (i.e. specimens 30_1 and 55_1), corresponding to the second failure mode shown in Figure 27b, in which small sections of the face sheet often detached from the core before complete failure occurred. This led to failure initiating at lower strains, although complete failure often occurred later than in the panels exhibiting the first failure mode. In this case, the panel may have the same initial stiffness, but a much lower failure strain.

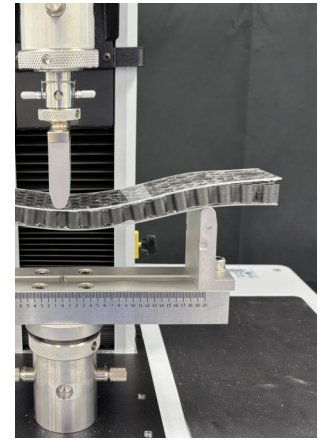


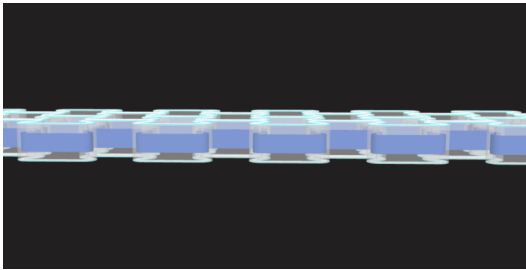
FIGURE 29 In the second failure mode, small sections of the face sheet often detached from the core, resulting in a step-like failure response.

3. Taking a look at Table 6, it can be noted that the sample with the cell angle of $\theta = -140^\circ$ clearly outperforms the other two geometries in terms of flexural modulus. This was expected, as the relative density of this core geometry was significantly higher than the other two core types (see Section 3.2.2). For the geometries with cell angle $\theta = 55^\circ$ and $\theta = 30^\circ$, the flexural stiffness is found to be much closer together, which can partly be explained by the fact that the relative densities of these geometries are also more similar. It should be noted that the average flexural modulus of the $\theta = 55^\circ$ group was negatively influenced by the outlier 55_2, which showed poor adhesion between its core and face sheets, which resulted in lower performance. However, even when this outlier is removed from the calculation, the average for the $\theta = 30^\circ$ sample group is still around 12,5 MPa higher than that of the $\theta = 55^\circ$ group. This can be explained by the fact that the $\theta = 30^\circ$ structure transfers load more efficiently, while the $\theta = 55^\circ$ geometry deforms more through bending of the cell walls. In addition, a larger portion of the layer lines in the $\theta = 30^\circ$ geometry are aligned with the bending direction, allowing the load to be carried more effectively along the filaments (see Figure 30). The zigzag pattern of the $\theta = 55^\circ$ geometry causes many filament strands to be oriented at a larger angle to the load, meaning a greater portion of the load is transferred through interlayer shear and bending, which reduces the overall stiffness.

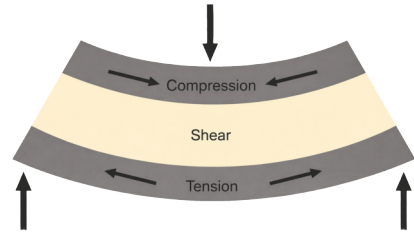
4.4.2 Effect of fiber reinforcement on flexural response of sandwich panel

In the second phase of the experiment, four different sandwich panels were tested, each with a different carbon fiber content. As mentioned in more detail in Section 3.2.2, only one sample was created for each fiber content group. Due to that, and the large chance of edge effects playing a role, the results should be interpreted carefully. The results for the flexural behavior of fiber reinforced sandwich panel cores are given in Figure 28e.

The 3-point bending experiment on the carbon fiber reinforced samples showed interesting results, as the sample consisting of pure Onyx[®] outperformed the panels with 24 and 48 CF layers in terms of flexural modulus. This is likely caused by the location of the carbon fiber layers within the thickness of the core. Since these layers were positioned in the center of the core, closer to the neutral axis, the contribution of these fibers to the overall bending stiffness of the panel is only small. In this case, the added fibers do not reinforce the most highly loaded regions of the panel, which are located near the outer surfaces of the structure. In addition, the partial fiber reinforcement may have introduced extra interfaces and local stiffness differences inside the core, which could reduce the efficiency of load transfer through the printed structure. However, since only one specimen was tested for each group, this cannot be concluded with certainty, and other factors, such as printing defects or poor adhesion between the core and face sheets, may also have contributed to this effect.



(a) The fibers (in blue) are located in around the center of the core.



(b) The largest tensile and compression stresses are found near the face sheets of the sandwich panel.

FIGURE 30 For the F-24 and F-48 panels, the core is mainly reinforced in the central region, while the highest stresses occur near the outer surfaces of the panel. This may explain the lower flexural performance than expected.

Another notable observation is the difference in flexural strain at which the panels failed. In contrast to the first phase of the experiment, where only pure Onyx[®] panels were tested, the second failure mode shown in Figure 27b was not observed in any of the fiber reinforced panels. Instead, failure consistently occurred according to the first failure mode, characterized by global bending and fracture of the lower face sheet beneath the loading point. However, slight variations in this failure mode resulted in differences in the fracture strain between specimens. For example, the F-96 panel exhibited the characteristics of the first failure mode up to failure, but the upper face sheet fractured locally just outside the center, which caused the early sudden drop in Figure 28e. From these sandwich panel bending experiments, it can be concluded that the flexural modulus can be determined quite reliably, as it mainly depends on the global structural response and is less sensitive to local effects such as adhesion or printing defects. In contrast, the exact point of failure is strongly influenced by these local factors, and therefore no firm conclusions can be drawn from the failure strain, other than identifying the different failure modes.

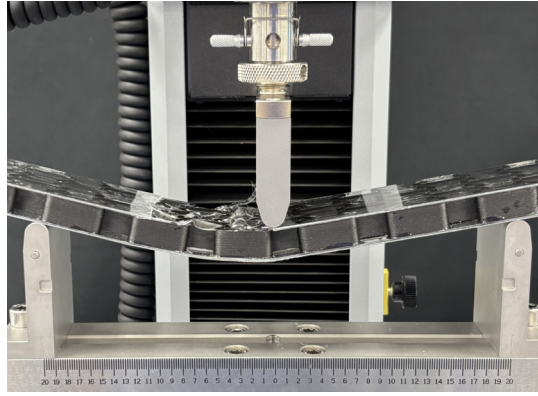


FIGURE 31 Although the sandwich panel F-96 showed a strong initial linear flexural response, a small section of the upper face sheet fractured suddenly at a lower strain than expected.

To relate the experimental results to analytical sandwich beam theory, the bending stiffness of the F-96 panel was estimated using Equation 33.

$$E_f I = E_f \cdot \left(\frac{b \cdot t_c^3}{12} + 2 \cdot \left[\frac{b \cdot t_f^3}{12} + (b \cdot t_f) \cdot \left(\frac{t_c + t_f}{2} \right)^2 \right] \right) \quad (33)$$

$$\begin{aligned} E_f I &= 1,33130 \cdot 10^9 \left(\frac{0,05 \cdot 0,012^3}{12} + 2 \cdot \left[\frac{0,05 \cdot 0,001^3}{12} + (0,05 \cdot 0,001) \cdot \left(\frac{0,012 + 0,001}{2} \right)^2 \right] \right) \\ &= 15,221 \text{ N m}^2 \end{aligned} \quad (34)$$

Or, the bending stiffness per unit width:

$$E_f I / b = \frac{15,221}{0,05} = 304,42 \text{ N m} \quad (35)$$

An attempt was made to determine the bending stiffness of the sandwich panels analytically by using Equations 10–14 to estimate the mechanical properties of the anisotropic composite, in combination with the sandwich beam theory described in Section 2.4. However, this approach could not be completed, as it required all five engineering constants of both phase materials, which have not yet been determined experimentally. As discussed, the other methods for determining the properties of the overall composite do not fully into account the anisotropic phases of continuous fiber 3D printed materials. Therefore, determining the five engineering constants and working out the analytical approach could be an interesting direction for future work.

5 | CONCLUSION

In this study, the influence of continuous carbon fiber reinforcement and honeycomb core geometry on the flexural behavior of additively manufactured sandwich panel cores was investigated at both material and structural level. At the material level, it was shown that the mechanical properties of 3D-printed Onyx® can be determined using a combination of experimental methods. The different experimental methods used in this study each provide valuable insight into the mechanical behavior of 3D-printed Onyx® and carbon fiber reinforced composites. However, clear differences in reliability and applicability to the anisotropic nature of the material were observed.

The tensile tests form the most direct method for determining the Young's modulus, as they measure the uniaxial response of the material. However, the initial test series showed inconsistencies due to clamping effects and seating of the specimen, resulting in a non-zero starting strain. Although this was partially resolved by introducing a pre-strain in the second test series, the results remained sensitive to experimental setup and specimen alignment. This indicates that tensile testing, while fundamentally suitable, requires careful execution to obtain reliable results for 3D-printed materials.

The three-point bending test proved to be more robust and repeatable. The obtained flexural modulus values showed a clear and consistent trend with fiber content, and the results aligned well with theoretical expectations. Since bending tests are less sensitive to local defects and gripping conditions, they appear to be a reliable method for finding flexural stiffness, also for anisotropic materials such as additively manufactured composites.

Dynamic Mechanical Analysis (DMA) provided insight into the viscoelastic behavior of the material. The storage modulus showed a clear dependency on frequency, as expected from a polymer matrix. However, due to the single cantilever configuration, the measured modulus represents a dynamic flexural stiffness rather than a direct Young's modulus. In addition, the results are sensitive to fixture compliance and specimen geometry, which complicates direct comparison with quasi-static test results.

The Time of Flight (ToF) method was found to be unsuitable for this type of material. As discussed in Section 3.4.4, strong wave attenuation, scattering due to the layered microstructure, and anisotropy in the printed material resulted in ambiguous signals and unreliable measurements. Consequently, no consistent elastic modulus could be derived using this approach.

Modal analysis proved to be a valuable complementary method. By identifying eigenfrequencies and corresponding mode shapes, it was possible to estimate the Young's modulus through comparison with finite element simulations. This method required careful interpretation, as excitation and measurement locations suppressed certain modes, particularly when located near nodal regions. Nevertheless, this method provided a valuable global measure of stiffness.

Overall, the combination of methods provides a more complete understanding of the material behavior. Quasi-static methods, particularly three-point bending, offer reliable and reproducible flexural stiffness values, while dynamic methods such as modal analysis provide additional validation for quasi-static test such as the tensile test. In contrast, the ToF method was not suitable for this type of material. The results highlight that no single method is sufficient on its own, and that a combined experimental approach is necessary to accurately characterize 3D-printed composite materials.

At the structural level, the results demonstrate that both core geometry and fiber reinforcement significantly influence the flexural response of sandwich panels. The cell angle was found to have a strong effect on stiffness, with geometries that have a more efficient load transfer and favorable filament orientation showing higher flexural perfor-

mance, even at lower relative densities. Furthermore, a separate series of fiber reinforced sandwich panels was tested. Although the number of samples was limited, it was found the location of the reinforcement within the core plays an important role. Fiber layers positioned closer to the neutral axis contribute less to bending stiffness than material near the outer regions of the structure.

The flexural modulus of the sandwich panels could be determined consistently, as it is primarily governed by the global structural response. In contrast, the failure behavior showed larger variability between specimens and was strongly influenced by local effects such as adhesion quality, printing defects, and geometric imperfections. As a result, while different failure modes could be clearly identified, no definitive conclusions can be drawn regarding the exact failure strain of the panels.

It should be noted that the results are subject to certain limitations, such as a limited number of specimens, the presence of edge effects due to the small number of unit cells for the fiber reinforced panels, and variability introduced by the manufacturing process as a whole. For future work, further research could focus on determining the full set of anisotropic material properties, enabling more accurate analytical predictions of sandwich panel behavior. Also the exploration of optimized fiber placement strategies could lead to improved structural performance of these additive manufactured materials. At the time of this study, the Markforged Mark Two was not yet capable of supporting this approach, and a suitable workaround would need to be developed. Overall, this research aimed to provide a clear link between material composition, core geometry, and structural performance for continuous fiber reinforced 3D printed sandwich panel cores, and demonstrated the importance of both geometric design and fiber reinforcement in optimizing flexural stiffness.

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