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Miguel Nuño

RWTH Aachen University: Rheinisch-Westfälische Technische Hochschule Aachen

Jannik Bühring (✉ jannik.buehring@sla.rwth-aachen.de)

RWTH Aachen University <https://orcid.org/0000-0002-0468-7853>

Narasimha Rao Mekala

RWTH Aachen University: Rheinisch-Westfälische Technische Hochschule Aachen

Kai-Uwe Schröder

RWTH Aachen University: Rheinisch-Westfälische Technische Hochschule Aachen

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RESEARCH

Delamination Testing of Additive Manufactured Sandwich Structures with Lattice Truss Core

M. Nuño^{*†}, J. Bühring[†], M.N. Rao and K.-U. Schröder

^{*}Correspondence:

miguel.nuno@sla.rwth-aachen.de

Institute of Structural Mechanics
and Lightweight Design, RWTH
Aachen University, Aachen,
Germany

Full list of author information is
available at the end of the article

[†]Equally contributing first authors

Abstract

Sandwich structures possess a high bending stiffness compared to monolithic structures with a similar weight. This makes them very suitable for lightweight applications where high stiffness to weight ratios are needed. Most common manufacturing methods of sandwich structures involve adhesive bonding of the core material with the sheets. However, adhesive bonding is prone to delamination, a failure mode which is often difficult to detect.

In this paper, the results of delamination testing of fully additive manufactured (AM) AlSi10Mg sandwich structures with pyramidal lattice truss core are presented. To characterise the bonding strength, climbing drum peel tests and out-of-plane tensile tests are done. The thickness of the faces and the diameter of the struts is 0.5 mm, while the core is 2 mm thick. The inclination of the struts is 45°. To predict the expected failure loads and modes, analytical formulas are derived. The analytics and tests are supported by finite element (FE) calculations. From the analytic approaches, design guidelines to avoid delamination in AM sandwich structures can be followed. The study shows, that critical ratios for face sheet thickness to strut diameter can be determined, to define if the structure tends to delaminate under certain loads. Those ratios are mainly influenced by the strut inclination. The peel tests resulted in face yielding, which can also be followed from the analytics and numerics. The out-of-plane tensile tests didn't damage the structure.

Keywords: Additive Manufacturing; Sandwich Structures; Pyramidal Lattice Core

1 Introduction

Sandwich structures show outstanding properties, especially regarding bending and buckling behaviour. Usually, two thin face sheets are adhesively connected to a

much thicker core structure. A lot of different core structures were developed in the¹ past. The most commonly used are honeycombs and foams. The adhesive bonding² is often responsible for the failure of sandwich structures and can in many cases be³ traced back to delaminations, which can occur due to local damages of the adhesive⁴ connection or local buckling of the face sheets. Furthermore, especially in the case⁵ of honeycomb core structures, the bonding surface is small and therefore the max-⁶imum transferable load too. To increase the bonding strength, an increase of the⁷ bonding surface would be reasonable. But, whereas the specific weight is propor-⁸tional to the vertical tensile module and the tensile strength, the peeling resistance⁹ of the skin/core bonding is not exceedable.¹⁰

Depending on the type of adhesive, the force that an adhesive fillet can transfer is¹¹ equal to the tensile strength of a 0.08 - 0.1 mm thick foil. Thicker foils are there-¹²fore inappropriate for increasing strength properties. This can only be achieved by¹³ smaller cell sizes [1]. With the establishment of new manufacturing processes, such¹⁴ as additive manufacturing (AM), alternative approaches for sandwich structures are¹⁵ attractive. By using lattice structures for the sandwich core, the sandwich can be¹⁶ printed as a single part and by this a direct connection between the core and the¹⁷ faces is established.¹⁸

First fundamental investigations about sandwich structures with lattice cores were¹⁹ presented by Wicks & Hutchinson [2, 3]. They show that truss core sandwich struc-²⁰tures have high flexibility in designing the structural behaviour. Compared to hon-²¹eycomb structures, much better structural properties against bending and com-²²pression loads, regarding stiffness or strength, can be achieved. Regarding cellular²³ structures, especially the work of Gibson & Ashby [4, 5] shows basic formulations²⁴ for the mechanical properties of lattice structures as a function of the relative cell²⁵ density. They show that different elastic properties of sandwich structures scale lin-²⁶early with the relative density of the core. For sandwich structures with tetrahedral²⁷ and pyramidal core under 3-point bending, Deshpande & Fleck [6, 7] derived equiv-²⁸alent expressions. The results are extended by the research of Evans et al. [8, 9] and²⁹ Chiras et al. [10] for shear and compression loads. Zok et al. [11] show that main³⁰ failure mechanisms of pyramidal truss core sandwich panels are face sheet yield-³¹ing, face sheet buckling, core member yielding and core member buckling. Basic³² studies about failure modes of sandwich structures have been done especially by³³

Zenkert *et al.* [12, 13, 14]. They show that sandwich beams under quasi-static and¹ fatigue bending loads tend to delaminate for the most tested configurations. Some² research focuses on methods to stop these delaminations instead of avoiding them.³ For example, Grenstedt [15, 16] implemented peel stoppers during manufacturing.⁴ Other approaches focus on the creation of sandwich structures with high transverse⁵ stiffness and strength [17, 18] to avoid delaminations. Furthermore, Jakobsen *et*⁶ *al.* [19] show an alternative approach. They reroute the delamination to confine it⁷ to a predefined zone in the sandwich. By this, they are able to stop the propaga-⁸ tion of the delamination. According to the authors knowledge no studies have been⁹ performed yet on fully AM sandwich structures with lattice core. Since the delam-¹⁰ ination behaviour of such structures has not been investigated, this study focusses¹¹ on a specific geometry.¹²

The main objective of this research is the development of simple criteria for avoid-¹³ ing delaminations in AM sandwich structures with pyramidal cores. For this, the¹⁴ bonding strength and delamination resistance is investigated. Analytical prelimi-¹⁵ nary considerations regarding peeling strength of sandwich structures with lattice¹⁶ core are supported by FE studies and by physical tests. Fig. 1 shows the considered¹⁷ AM sandwich structure.¹⁸

2 Methods and Materials¹⁹

In the following section the used specimens, material and associated test meth-²⁰ ods are presented. Furthermore, analytical formulations are derived to predict the²¹ expected failure loads and modes. This formulations can be used as rough design²² guidelines to avoid delamination.²³

2.1 Materials and Specimens²⁴

All structures are made of AlSi10Mg via Laser Powder Bed Fusion (LPBF) using an²⁵ EOS M290 printer. The nominal geometric parameters of the structures are summed²⁶ up in Table 1. Due to manufacturing restrictions, a face sheet thickness $t_f = 0.5$ mm,²⁷ a core thickness $t_c = 2$ mm, a strut diameter of $d = 0.5$ mm and a relative strut angle²⁸ of $\omega = 45^\circ$ are used. A total sandwich height of $h = 3$ mm results. The radius of the²⁹ struts is defined using r . Fig. 2 shows a schematic sketch of a lattice unit cell with³⁰ its face sheet section. To describe the structural properties of the lattice core, simple³¹ calculation methods can be used for strength and stiffness. The in-plane Young's³² ³³

Table 1: Nominal parameters of the considered AM sandwich structures

t_c	r	ω	t_f
2 mm	0.25 mm	45°	0.5 mm

modulus $E_{11} = E_{22} = E_s$ and shear modulus $G_{12} = G_s$ in 1-direction and 2-direction are dominated by the face sheets. The solid materials Young's modulus E_s and shear modulus G_s can be used. The core influence is neglected since according to classic sandwich theory, the influence on the elastic constants is small. Assumptions for the effective elastic properties of lattice structures are usually connected with the dimensionless relative density $\bar{\rho}$, which defines the ratio between the struts mass and the mass of the enclosing equivalent solid mass using the same material. For pyramidal cores it can be calculated with Eq. (1).

$$\bar{\rho} = \frac{\rho}{\rho_s} = \frac{4\pi r^2 l}{b^2 t_c} = \frac{2\pi \tan(\omega)}{\cos(\omega)} \left(\frac{r}{t_c} \right)^2 \quad (1)$$

Equations for the out-of-plane Young's modulus and shear modulus can be derived using the stiffness method, assuming pinned struts as shown for example in [6]:

$$E_{33} = E_s \bar{\rho} \sin(\omega)^4 \quad (2)$$

$$G_{13} = G_{23} = \frac{1}{8} E_s \bar{\rho} \sin(2\omega)^2 \quad (3)$$

Due to the overlapping of the struts, the bonding surface between the core and the face sheets cannot assumed to be circular, as shown in Fig. 3. Since no non-destructive measuring method is accessible, the nominal bonding surface can be extracted from the CAD-model, which was used for the manufacturing, or derived analytically using the following assumption:

$$A_c = d^2 \arctan(\csc \omega) \csc \omega \quad (4)$$

Both solutions result in a cross-section of the connection point of $A_c \approx 0.34 \text{ mm}^2$ for the nominal strut diameter of $d = 0.5 \text{ mm}$.

Material properties of AM structures vary depending on used machine parameters while manufacturing. Since the results are very sensible on stiffness and strength

properties, material properties for the calculations are determined doing tensile¹ tests. The specimens are produced from AM sandwich structures by separating the² face sheets from the core structure and afterwards milling them to a bone shape.³ Fig. 4 shows the results of three tested specimens. Regarding the elastic properties,⁴ the results show a reproducible behaviour. Young's modulus of the base material⁵ E_s can be derived to $E_s \approx 60.000$ MPa and yield strength to $\sigma_Y \approx 170$ MPa.⁶ All specimens are tested under the same conditions with displacement controlled⁷ load introduction and $\dot{u} = 0.5$ mm/min. Elongation at the breakpoint and ultimate strength show different values, which is probably attributed to the strong⁹ influence of the manufacturing process. Compared to the datasheet of the manufac-¹⁰ turer, Young's modulus and yield point are in a tolerable range, whereas elongation¹¹ at the breakpoint and ultimate strength are much lower than specified. The more¹² brittle behaviour is mainly attributed to the thickness of the structures. Material¹³ properties in the datasheet is extracted from thick, solid structures, whereas in this¹⁴ study thin structures are investigated. ¹⁵

Using the presented equations and the extracted material data, the following sand-¹⁶ wich properties are derived: ¹⁷

Table 2: Resulting sandwich properties ²⁰

Desription	Symbol	Value	Unit
Base Material			
Young's modulus	E_s	60.000	MPa
Poisson Ratio	ν_s	0.3	–
Yield Strength	σ_Y	270	MPa
Rel. Density	$\bar{\rho}$	0.13884	–
Effective Elastic Properties			
	E_{11}	60.000	MPa
	E_{22}	60.000	MPa
	E_{33}	2082.6	MPa
	G_{12}	23076.9	MPa
	G_{13}	1041.3	MPa
	G_{23}	1041.3	MPa

2.2 Preliminaries and Experimental Set-Up

Throughout the study, two test strategies are used to characterise the delamination behaviour of AM sandwich structures: out-of-plane tensile tests and climbing drum peel tests. Furthermore, the tests are supported by numerical FE calculations to get a better understanding about the failure mechanisms. First, the main failure mechanisms for the core structure are determined. Failure can happen at three different zones: failure at the connection point or a failure of the strut itself. Considering one unit cell with pinned struts, as shown in Fig. 2, a load in 3-direction (out-of-plane) results in an axial strut force of:

$$F_s = \frac{1}{4} \frac{F_3}{\sin(\omega)} \quad (5)$$

The failure zone is only a function of the strut angle ω . This can be derived by comparing the strut stress with the stress in the connection surface. Under a tension load in 3-direction F_3 , the resulting stresses in the struts σ_s and the stress in the connection σ_c can be calculated as followed:

$$\sigma_s = \frac{F_3}{4\pi d^2 \sin(\omega)} \quad (6)$$

$$\sigma_c = \frac{F_3}{\arctan(\csc(\omega)) \csc(\omega) d^2} \quad (7)$$

Under the condition that the struts are slender, and by this, no bending influence has to be considered, simple assumptions can be derived for the predicted failure mechanism. For $\sigma_s = \sigma_c$ a critical value $\omega = 35.26^\circ$ results. For lower angles, the struts will fail before the connection. For larger angles, the connection will fail before the struts. This relation is shown in Fig. 5. In the following sections the experimental set-up and the expected analytical determined values are shown.

Out-of-Plane Tensile Test

As shown in Fig. 5 failure in the struts is expected at strut inclination angles $\omega < 35.26^\circ$. For larger inclination angles the structures are susceptible to delamination. A total number of 3 specimens with 25 unit cells (5×5) is tested. The tests are performed quasi-static and displacement controlled with $\dot{u} = 0.2$ mm/min, according to DIN 53292 on an Instron 5567 electric universal testing machine. To neglect influences resulting from clamping torque, a moment-free fixture, which equals a

Table 3: Nominal parameters of AM sandwich structures

	Specimen 1	Specimen 2	Specimen 3
Dim. [mm ²]	17.6 × 14.6	17.5 × 15.5	17.1 × 18.0
Area [mm ²]	257	271	308
Force [kN]	7.45	7.86	8.93

cardanic bearing, is constructed (Fig. 6).

The face sheets of the specimens are adhesively bonded with the bearings. A 3M Scotch-Weld EC-9323-2 high strength epoxy adhesive is used. The strength of the adhesive is specified with 29 MPa, which results in a higher maximum load compared to the expected strength of the connection between the lattice core and the face sheets. The specimens are cut from large AM plates using a band saw. Due to these circumstances the outer geometry of the specimens varies and can be found in Table 3. Furthermore, the resulting maximum allowable forces, before the strength of the adhesive would be exceeded, are shown. All these force are much higher in comparison to the binding strength between face sheets and core, assuming the lowest ultimate strength of $R_m \approx 250$ MPa from Fig.4, which results in a maximum force of $F_{\max, \text{core}} = 2.125$ kN. Since the exact ultimate strength is unknown, the expected failure loads of the three specimens should be in the range of 2 – 3 kN, assuming that the load is evenly distributed on the connection points and the plastic stress distribution is homogeneous. The final test set-up is shown in Fig. 7. The displacement of the machine, as well as the force in the load cell, are documented. Additionally, the tests are monitored with a DIC technique, using a GOM Aramis 4M system, which allows measuring three-dimensional displacements and strain information of a surface by measuring absolute and relative multiple point movements.

Climbing Drum Peel Test

The climbing drum peel test” is a method to determine the resistance of bonded sandwich cores against peeling forces acting perpendicular to the surface layer. The drum transforms the vertical force of the testing machine into a moment so that the connection between the face sheets and the core is directly loaded. This test is the only way to characterise the direct bonding strength. The tests are done according

to DIN 53295. The most important values are the diameter of the drum itself and¹ the flange, where the specimens are fixed. The difference between both results in² a translation of the tension axis. By this, a moment in the midpoint of the drum³ results, which has to be carried by the specimen. If the stresses get higher than the⁴ critical stress, the face sheets will delaminate. This value is known as specific peel⁵ moment and can be calculated according to DIN 53295 as followed:⁶

$$M_{\text{peel}} = \frac{(F_p - F_1)(r_f - r_d)}{b} \quad (8)$$

The force F_1 is the force needed to deform the face sheet itself and has to be sub-¹⁰stracted from the peel-force F_p . The flange radius is r_f , r_i the drum radius and b ¹¹ the width of the specimen. The flange radius can be calculated from the diameter¹² of the drum d_d and the thickness a of the steel strip, which is needed to introduce¹³ the machine load into the peel fixture. It results in $r_f = (d_f + a)/2 = 62.9$ mm,¹⁴ since $d_f = 99$ mm and $a = 0.8$ mm. The drum radius r_d can be calculated from the¹⁵ drum diameter $d_d = 99$ mm and the thickness of the face sheet $t_f = 0.5$ mm and¹⁶ results in $r_i = (d_d + t_f)/2 = 49.8$ mm. The test set-up is slightly changed in com-¹⁷parison to the standard, but the impact on the calculated resulting peel moment is¹⁸ small, since the relative error results in $\epsilon_{\text{rel}} \approx 1.6\%$, which is a tolerable value. The¹⁹ force F_1 is determined by a preliminary test with a face sheet only. F_1 consists of²⁰ a part which is needed to position the peel fixture F_{pos} and a part which is needed²¹ to deform, or rather roll the face sheet F_{roll} . Fig. 8 shows the result of the test to²² extract F_1 . During the first 15 mm of displacement, the drum and steel strips get²³ loaded and aligned. The followed plateau at 15 mm defines the needed force F_{pos} .²⁴ The additionally needed force to reach the second plateau at 25 mm defines F_{roll} ,²⁵ which lead in sum to $F_1 = 69.31$ N.²⁶

Two different failure modes can occur: Failure of the face sheet and failure at the²⁷ connection between face sheets and core. Assuming an ideal elastic - ideal plastic²⁸ material behaviour, the stress distribution in the face sheet, due to a bending mo-²⁹ment, can be assumed to be linear in the case of elastic material behaviour and³⁰ constant (with different signs) in the case of plastic material behaviour. The maxi-³¹mum moment which can be transferred before face sheet failure can be calculated³² with $M_{\text{el}} = R_m t_f b / 6$ in the elastic case and $M_{\text{pl}} = R_m t_f b / 4$ in the plastic case.³³

Table 4: Geometric paramters of specimens for climbing drum peel test

	Specimen 1	Specimen 2	Specimen 3
n [-]	9	5	2
b [mm]	33.4	35.1	30.1
$\tilde{b}$ [mm]	27.9	15.5	6.2
l [mm]	150	150	150

Since for ductile materials failure occurs in the plastic area, the plastic case will be⁸ used. Fig. 9 shows a submodel for two rows of the sandwich structure. The applied⁹ bending moment will result in reaction forces $F_R = M/a$ at the connection points¹⁰ if pinned struts are assumed. The distance between two unit cells is a . By this, a¹¹ general relation to avoid delamination can be established:

$$\frac{t_f}{d} < \begin{cases} 2\sqrt{\pi \sin \omega} & \omega \leq 35.26^\circ \\ 2\sqrt{\arctan(\csc(\omega)) \csc(\omega)} & \omega > 35.26^\circ \end{cases} \quad (9)$$

This relation is shown in Fig. 10. If delaminations should be avoided, the ratio of¹⁶ t_f/d should be lower than the in Fig. 10 shown critical ratio. The critical value for¹⁷ ω is the value where the failure mode changes from strut failure to joint failure (see¹⁸ Fig. 5). The ratio of t_f/d is increasing with increase of ω up to the critical value¹⁹ whereas after this point it is decreasing, as shown in the Fig. 10. Therefore, for²⁰ $\omega = 45^\circ$, $t_f/d < 2.32$ should be complied to avoid delaminations.²¹

A total number of 3 specimens is tested. In Table 4 the specific geometric parameters²² are summed up. Furthermore Fig. 11 shows a sketch of the specimens and Table 5²³ the resulting expected relevant cross sections. $R_m = 350$ MPa is used. The number²⁴ of cells n was successively decreased since no delamination could be incited with²⁵ high cell numbers. Fig. 12 shows the final test set-up. All drum peel tests are done²⁶ displacement controlled using $\dot{u} = 25$ mm/min. The load from the load cell and the²⁷ displacement are documented during the tests.²⁸

Finite Element Model

To validate the formulas derived in the previous chapters and compare the results³¹ with the experiments, FE simulations are performed. The preprocessing and calcu-³² lations are performed with the software Abaqus. For each calculation a convergence³³

Table 5: Resulting expected cross sectional areas for drum peel test

	Specimen 1	Specimen 2	Specimen 3
A_b [mm ²]	0.34	0.34	0.34
A_s [mm ²]	0.19	0.19	0.19
A_f [mm ²]	16.7	17.55	15.05

study is done. All models are meshed with 10-node quadratic tetrahedral elements with improved surface stress visualization (Abaqus element C3D10HS). The models used for the peel test simulations are shown in Fig. 13a and 13b. One side of the sandwich (top and bottom face sheet edges) is pinned. On the opposite side a bending moment is applied using a multiple point constraint (MPC) to all nodes on the top sheet. The mesh in the vicinity of the critical connection area is refined using a seed of 0.05 mm. In the areas close to the support, where low stresses are expected, the seed used is 0.5 mm. The FE model simulating the out-of-plane tension is shown in Fig. 13c. The bottom nodes of the bottom sheet are pinned, and a homogeneous pressure is applied on the top face.

3 Results and Discussion

Out-of-Plane Tensile Test

Fig. 14 shows the results of the out-of-plane tensile tests. Additionally, Fig. 15 shows a picture of one of the specimens right before failure and after failure. The results show slight differences regarding the stiffness. This is mainly attributed to tolerances from manufacturing, which are specified with $\pm 100 \mu\text{m}$, which could result in differences regarding the strut diameters and therefore to differences in the stiffness (see Eqs. 1 and 2) since they are square depending on the diameter. The failure load of all three tested specimens lies between 3.8 - 4 kN. No failure in the core structure was detected. For all specimens, the adhesive fails before the core breaks. All failure loads are smaller than expected, which could be attributed to an irregular moistened adhesive surface. Nevertheless, from the measured loads it can be followed that the connection is at least as strong as the materials strength (Fig. 4). Assuming the connection surface of one unit cell to be, as specified in Fig. 3, $A_c \approx 0.34 \text{ mm}^2$ and the resulting total connection surface $A_C = n \times A_c = 8.5 \text{ mm}^2$. This would result in a connection strength, of at least $\sigma_u > 440 \text{ MPa}$, which is not

a reasonable value. Due to these circumstances, the face sheets are removed from¹ the core structure and the bonding surfaces are measured using optical microscopy.² This showed that although the diameter of the struts is between 0.5 and 0.6 mm,³ the connection surfaces are clearly above the theoretically calculated value. The⁴ measured bonding surfaces lie partly at around 2 mm². Assuming such a single cell⁵ bonding surface, a total bonding surface of $A_C = 50 \text{ mm}^2$ would result and by this⁶ a stress in the connection of $\sigma_c \approx 80 \text{ MPa}$. This explains why the adhesive fails and⁷ not the connection between the faces and the core.⁸

Based on the numerical FE simulations, the first zone to undergo full plastic de-⁹ formation is the joint between the face and the struts, as shown in Fig. 16. This is¹⁰ also the expected critical zone according to Eq. (6) and (7).¹¹

*Climbing Drum Peel Test*¹²

Fig. 17 shows the force-displacement curves of the climbing drum peel tests. Fur-¹⁴ thermore, in Fig. 18 the measured force is related to the specific width $\tilde{b}$ (Fig.11),¹⁵ to avoid that results are misinterpreted by different geometries. Additionally, Fig.¹⁶ 19 shows a picture of specimen 1 and Fig. 20 of specimen 3 after testing. Both spec-¹⁷ imens show a failure in the face sheets. Since the face sheets fail, before they have¹⁸ rolled up sufficiently, the evaluation following DIN 53295 cannot be done. Neverthe-¹⁹ less, it can be followed that no delamination will occur with the used geometry. For²⁰ specimen 3, the number of cells in one row (in the rolling axis) was reduced. The²¹ face sheet did not fail immediately and therefore the evaluation for the peel moment²² can be done according to DIN 53295 to $M_{\text{peel}} = 279.62 \text{ Nmm/mm}$. However, the²³ face sheet did not delaminate, but broke parallel to to cell row in rolling direction²⁴ (see Fig. 20) and by this the result is not meaningful.²⁵

Based on the numerical FE simulations, the first zone to undergo full plastic de-²⁶ formation is the face sheet, as shown in Fig. 21. This is also the expected critical²⁷ zone according to Eq. (9) since $t_f/d = 1 < 3.2$. In the second simulation, with²⁸ $t_f = 2 \Rightarrow t_f/d = 4 > 3.2$, the connection is expected to fail, as seen in the stresses²⁹ of Fig. 13b. This corresponds too to the simplified model presented in Eq. (9).³⁰

4 Conclusion³¹

In this study investigations of the delamination behaviour of fully additive man-³² ufactured sandwich structures with pyramidal lattice grid core were performed.³³

Simplified formulas were derived to estimate the geometry needed to avoid delamination failure. These formulas were validated with experimental tests and numerical simulations. The analytical approaches show that delaminations can be avoided by considering critical values for the ratio between face sheet thickness and strut diameter. These ratios are mainly dependent on the strut angle. By choosing an adequate set of geometric parameters, delaminations can be avoided. Since there were no studies related to the delamination behaviour of those structures, the results presented in this paper can be used as guidelines to design delamination free structures. In future studies, the models will be expanded to assume a inhomogeneous stress distribution on the joints and struts.

5 Declarations

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Availability of data and materials

The datasets used and/or analysed during the current study are available from the corresponding author on reasonable request.

Competing interests

The authors declare that they have no competing interests.

Authors' contributions

The authors MN and JB have contributed equally to this article and where in charge for the whole trial . The Co-authors NMR and KUS supported with their extensive experience and gave advices on the manuscript.

Author details

Institute of Structural Mechanics and Lightweight Design, RWTH Aachen University, Aachen, Germany.

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Figure 1: Additive manufactured sandwich panel with pyramidal core

Figure 2: Unit cell of sandwich structures with geometric parameters

Figure 3: Non-circular connection between face-sheets and lattice core structure

$$A_c \approx 0.34 \text{ mm}^2$$

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Figures

Figure 1

Additive manufactured sandwich panel with pyramidal core

Figure 2

Unit cell of sandwich structures with geometric parameters

Figure 3

Non-circular connection between face-sheets and lattice core structure: $A_c \approx 0.34 \text{ mm}^2$

Figure 4

Stress-strain curves of AlSi10Mg tensile tests on AM-sandwich structures according to E-DIN-EN-ISO 6892-1:2019-10

Figure 5

Failure in struts and connection as a function of relative angle

Figure 6

Cardanic bearing for out-of-plane tensile tests

Figure 7

Final test set-up for out-of-plane tensile tests

Figure 8

Preliminary peel test on a face sheet to extract F_1

Figure 9

Submodel

Figure 10

Critical ratio for t_f/d to avoid delaminations

Figure 11

Schematic sketch of the peel test specimens

Figure 12

Final test set-up for peel tests

Figure 13

Finite element models for the peel and out-of-plane tensile test

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Load-displacement curve of out-of-plane tensile tests

Figure 15

Pictures of one specimen during testing. Top: before load drop, Bottom: after load drop

Figure 16

Section view at the middle joint

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Force-displacement curve for the climbing drum peel test

Figure 18

Relative force-displacement for the climbing drum peel test

Figure 19

Pictures of specimen 1 after test

(a)

(b)

Figure 20

Pictures of specimen 3 after test

Figure 21

Section view of the specimen model at the middle joint of the outer row of cells. Zones with plastic deformation are marked grey

Figure 22

Section view of the model with thick faces at the middle joint of the outer row of cells. Zones with plastic deformation are marked grey