DESIGN OF LIGHT-WEIGHT SANDWICH PANEL FOR TRAILERS

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Abstract
The purpose of this study is to design a lightweight sandwich panel for trailers. At first the sandwich panel is analyzed in extreme wind condition. Tsai-Wu strength index is computed and verified that it remains under unit value. Thus, the sandwich panels do not break under severe wind condition. Then the Pareto optimality concept is employed and optimal solutions are determined by applying multi-criteria analysis techniques and genetic algorithms. Finally, modal analysis is performed. The results of the modal analysis demonstrated that natural frequencies are much lower than working frequencies, which could lead to resonance. In order to avoid resonance frequency, the trailer sandwich panels were made stiffer until their first natural frequency value became close to 40 Hz.

1 Introduction

Low-cost composites are currently in focus of many material researchers. Low-cost means using an inexpensive reinforcement and matrix in order to keep the consolidation costs minimal. The final product is usually a composite material, which has significantly lower stiffness than moderate- or high-cost composites. One approach utilizing such material and mitigating its low stiffness is to use it as facesheets in a sandwich construction. Low-cost core materials can be bonded together at room temperature to form sandwich plates. Thus, low-cost sandwich plates can be produced [1].

Some studies are focused on trailers. Reinforced sandwich panels are frequently used in refrigerated trailer construction, there is a need to reinforce the sandwich panels because they can have extremely large dimensions and consequently lose their ability to support the working loads. It is demonstrated that the Z reinforcement drastically increases the stiffness of sandwich beams in three point bending [2]. Previous research performed in Tallinn University of Technology has concentrated on designing and testing of sandwich structures for trailers. Strength calculations and selection of different materials are carried out in order to develop a new solution for this application. The different types of sandwich composite panels are tested in 4-point bending conditions according to ASTM C393/C393M. Virtual testing is performed to simplify the core material selection process and to design the layers [3].

The main objective of current study is to design sandwich panels for trailers with maximum mechanical properties and lowest cost. To achieve the main target of the current study, the following subtasks are solved. Firstly, sandwich panels are selected in the basis of 4-point bending test results obtained in [3]. Secondly, the 3D FEA, based on wind load, is performed. Thirdly, based on the 3D FEA, the Pareto optimality concept is employed and optimal solutions are determined by applying multi-criteria analysis techniques and genetic algorithms. Finally, based on Pareto frontier, trailer modal analysis is performed, and natural frequencies are calculated and compared with working frequencies.

2 Problem formulation

2.1 Structure of the sandwich panel

Current study analyses trailer's side panel which is depicted in Figure 1. The panel is 4930 mm long and 1590 mm wide. The panel lay-up consists of 5 layers and is described in Table 1. In the FEA, the sandwich panel thickness remains always the same (25,5 mm), but the core and the biaxial $0^o/90^o$ E-Glass fiber thicknesses change. As gelcoat layer does not have significant mechanical properties, it is neglected in FEA. Undergoing research focuses on five different foam cores, which are:

- polyethylene-terephthalate (PET) 80 kg/m^3;
- high-density polyethylene (HDPE) 80 kg/m^3;
- polymethyl-methacrylate (PMI) 52 kg/m^3;
- polyurethane (PUR) 60 kg/m^3;
- extruded polystyrene (XPS) 30 kg/m^3.

Core materials are selected on the basis of variable mechanical properties, weight and price. Described foam cores are competitive in designing sandwich structures for automotive, marine or wind turbine industry.

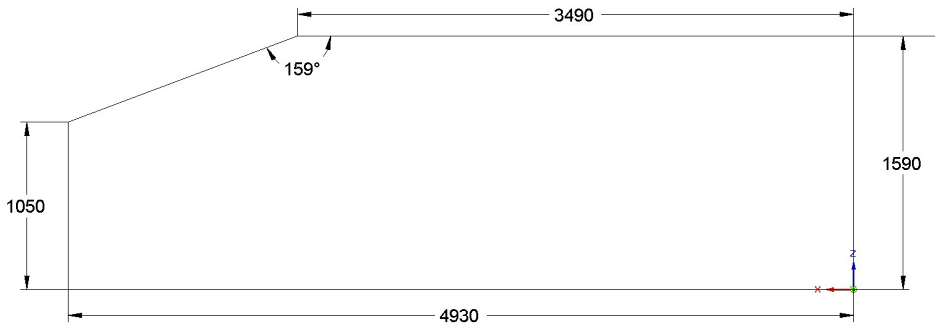

Figure 1. Trailer's side panel.

Layer	Material	Test No. 1	2	3	4
		Thickness [mm]			
1	Chopped Strand Mat (CSM) 810 g/m^2	1,5	1,5	1,5	1,5
2	Biaxial $0^o/90^o$ E-Glass fiber woven roving 600 g/m^2	1	3	6	9
3	Chopped Strand Mat 810 g/m^2	1,5	1,5	1,5	1,5
4	Foam core	20	18	15	12
5	Chopped Strand Mat 810 g/m^2	1,5	1,5	1,5	1,5

Table 1. Sandwich panel lay-up.

2.2 Objective of 3D wind analysis

There are currently no standards for mechanical testing of trailer's composite nacelle. Therefore, every trailer company is developing and using their own in-house guide lines. On the basis of the literature review, it was concluded that trailer side panels have not been analyzed in extreme wind conditions. Therefore, the analysis is carried out in the current study. If extreme wind conditions can cause fatal damage to the sandwich panel of the trailer, then wind analysis should be compulsory for the strength analysis of similar products. Otherwise, it can be omitted.

2.3 Objective of trailer's modal analysis

Vibration is related to trailer's stiffness and strength. Similar natural frequencies and working frequencies can cause resonance effects and significantly reduce trailer's operating life. Present research identifies if the trailer's working frequencies are lower than the first natural frequency in order to avoid resonance. Natural frequencies are analyzed by performing modal analysis. Trailer's working frequencies are obtained from literature.

3 Experimental and numerical analysis

3.1 Glass- fiber reinforced polymer (GFRP) tensile test specimens

Test specimens are manufactured according to standard EN ISO 527-4:2000. Three different type of GFRP test specimens are produced:

- E-Glass fiber balanced stitched biaxial roving mat 0°/90° (3 x 600 g/m^2);
- E-Glass fiber balanced stitched biaxial roving mat +45°/-45° (3 x 600 g/m^2);
- Chopped Strand Mat (3 x 810 g/m^2).

Vacuum infusion process (VIP) is used to manufacture test specimens and polyester resin (413-568) is used as matrix material. After post-curing the laminates at the room temperature, the rectangular tensile test specimens (25 mm x 250 mm) are cut with 3D CNC milling machine.

Mechanical testing is performed with Instron 8516 tensile testing machine. Elastic modulus and Poisson's ratio are calculated according to standard EN ISO 527-4:2000. Longitudinal elastic modulus E_x is obtained from tensile tests (Table 2). As E-Glass fiber properties are similar in 0° and 90° directions, then $E_x = E_y$. Determining the E_z value for 0^o/90^o and CSM, it is assumed to be 50 % of the polyester resin elastic modulus [4]. Poisson's ratios υ of facing materials are obtained from tensile tests (Table 2). Shear modulus (G) for E-Glass fiber is calculated according to standard ASTM D3518. E-Glass fiber 0^o/90^o shear modulus is obtained from tensile tests with ±45° test specimens (Table 2). As fiber properties are same in x- and y-direction, then $G_{xz} = G_{yz}$. CSM shear modulus is obtained from literature [5].

E-Glass fiber 0°/90° [GPa]		**CSM [GPa]**	
E_x	19100	E_x	9400
E_y	19100	E_y	9400
E_z	1800	E_z	1800
υ_{xy}	0.11	υ_{xy}	0.26
υ_{yz}	0.30	υ_{yz}	0.33
υ_{xz}	0.30	υ_{xz}	0.33
G_{xy}	2900	G_{xy}	2200

G_{yz}	1600	G_{yz}	800
G_{xz}	1600	G_{xz}	800

Table 2. Mechanical properties of GFRP laminate.

Mechanical properties, presented in the Table 2, are important parameters for performing 3D analysis with ANSYS APDL software.

3.2 4-point bending tests of sandwich panels

In our previous study [3] different type of sandwich composite panels were tested in 4-point bending conditions according to ASTM C393/C393M standard. Experimental and virtual tests are performed to simplify the core material selection process, and to design lay-up. Virtual 2D results are compared with real 4-point bending tests, to validate the finite element model (FEM).
Results of the 4-point bending tests showed that the sandwich panel which had PMI foam core achieved best results as regards to stiffness. Although, the cost of PMI foam exceeds 5-times the cost of HDPE and PET foams. Thus, PET foam is economically more reasonable having similar modulus of elasticity as PMI and better flexural strength than HDPE [3].
Based on paper [3], the lay-up and the best 3 foam cores (PET, HDPE, PMI) are selected and used in current study. In addition, low-cost foams like PUR and XPS are added to 3D wind analysis.

3.3 Finite element model of sandwich panel

FEM subjected to 3D wind analysis of sandwich panel (Figure 1) is conducted with ANSYS APDL v14. Shell 181 3D elements, with four nodes and six degrees of freedom (DOF), are used to model the sandwich panel. The panel is meshed with 10 mm element side length. Linear orthotropic material model is used to define laminate properties (defined by elastic modulus E, Poisson´s ratio υ and shear modulus G). Linear isotropic material model is used to define core properties (defined by elastic modulus E and Poisson's ratio υ). Mechanical properties of used GFRP laminates and core materials are presented in Table 2 and Table 3 respectively. The sandwich panel lay-up used in FEM is described in Table 1. The fiber longitudinal and transverse directions are same with the panel's length and width respectively. All edges of the panel are constrained so that all DOF are removed and surface pressure of 1482 Pa is applied evenly to the entire panel (Chapter 3.4). Finally, the panel is analyzed with large displacement static option, which is non-linear static analysis where large deformation effects are included.

Core material	Density [kg/m³]	Shear strength [MPa]	Elastic modulus [MPa]
PET	80	0,60	60
HDPE	80	0,60	27
PMI	52	0,80	70
PUR	60	0,55	20
XPS	30	0,24	13

Table 3. Mechanical properties of GFRP laminate.

3.4 3D wind analysis of sandwich panel

The sandwich panel, depicted in Figure 1, is analyzed in extreme wind condition that has probability to occur once during 50 year. Based on Estonian wind atlas [6], the maximum wind speed does not exceed 6 m/s in dry land. Thus, the sandwich panel can be analyzed according to wind class IV (standard EVS-EN 61400-2:2006). The wind class IV gives v_{ref}=30 m/s if the v_{ave}=6 m/s. The 50 year extreme wind speed v_{e50} can be calculated using the following equation:

$$v_{e50} = 1{,}4 * v_{ref} \tag{1}$$

where v_{e50} is the 50 year extreme wind speed and v_{ref} is the reference wind speed. Based on equation (1) v_{e50}=42 m/s. Subsequently the whole panel surface pressure can be calculated using the following equation [4]:

$$p = C_d * \rho_{air} * \frac{v_{e50}^2}{2} \tag{2}$$

where p – air resistance that creates surface pressure to the panel [Pa]; C_d – air resistance factor (1,4 for rectangular shapes); ρ_{air} – air density (1,2 kg*m^{-3}); and v_{e50} – 50 year extreme wind speed.
Based on equation (2), p≈1482 Pa.

3.5 Optimal design of sandwich panel

General aim is to design sandwich panel with maximum stiffness. Several optimality criteria are analyzed. The maximum deflections of the sandwich structure and the cost of the materials are subjected to minimization (other considered criteria are the strain energy density and weight of the structure).

$$\bar{F} = (F_1(\bar{x}), F_2(\bar{x})) \to min \tag{3}$$

where $F_1(\bar{x})$ and $F_2(\bar{x})$ are maximum deflections of the sandwich structure and the cost of the materials, respectively, $\bar{x}$ is the vector of design variables describing configuration of the sandwich structure (thickness, material). The maximum stresses of the each layer σ_k and the weight of the structure W are subjected to constraints.

$$\sigma_k(\bar{x}) \le \sigma_k^*, \quad W \le W^* \tag{4}$$

In (4), σ_k^* stands for the upper limit strain for layer k, and W^* stands for the upper value of the weight of the structure. The posed problem can be considered as a mixed integer optimization problem [7].

3.6 Trailer modal analysis

The global vibrational characteristics of a vehicle are related to both parameters - stiffness and mass distribution. The frequencies of the global bending and torsional vibration modes are commonly used as benchmarks for vehicle structural performance. Bending and torsion stiffness K_B and K_T influence the vibrational behavior of the structure, particularly its first

natural frequency [8]. The prediction of the dynamic properties of the trailer is essential to determine, it must be assured that working frequencies are lower than the first natural frequency of the trailer to avoid resonance effects.
FEM subjected to 3D modal analysis is performed with ANSYS APDL v14. Shell 181 3D elements, with 20 mm side length, are used. The trailer's floor is constrained so that all DOF are removed; it is used to interpret rigid steel chassis underneath the composite nacelle. The Block Lanczos mode extraction method is used and 4 modes are calculated.

4 Results and discussion

4.1 Results of 3D wind analysis of sandwich panel

The all edges of the panel are constrained (all DOF removed) and evenly loaded with surface pressure 1482 Pa (Chapter 3.4). The whole panel thickness remains the same but lay-ups and foam cores are varied (Chapter 2.1). Table 4 describes results from test number 1 where layer 1, 2, 3, 4, 5 thicknesses are: 1,5; 1; 1,5; 20; 1,5 mm respectively (Table 1).

Foam core material	Foam elastic modulus [MPa]	Def-lection [mm]	Core maximum stress [MPa]	Core shear strength [MPa]	Laminate maximum stress [MPa]	Strain energy [mJ]
PET	60	4,4	0,05	0,6	7,5	0,77
HDPE	27	5,3	0,05	0,6	7,6	0,90
PMI	70	4,3	0,05	0,8	7,5	0,75
PUR	20	5,9	0,05	0,55	7,6	0,98
XPS	13	7,0	0,05	0,24	7,6	1,15

Table 4. Results of 3D wind analysis.

Table 4 shows that although foams have quite different elastic modulus, the difference in deflection is only 2,6 mm. This allows to use cheaper foam cores with lower mechanical properties. Foam core maximum stress is 0,05 MPa in all foam cores, which is much lower than the weakest foam (XPS) shear strength (0,24 MPa). It means that foam core remains undamaged during surface pressure. Also, laminates stress is quite low (therefore x-, y- and z-directional stresses are not showed) and remains under E-Glass fiber $0^o/90^o$ allowed stress (250 MPa) and under CSM allowed stress (110 MPa).
Tsai-Wu strength index is depicted in Figure 2. The most critical layers in sandwich panel are plotted. Layer 5, which is CSM (Table 1), has Tsai-Wu strength index 0,04 (on the left in Figure 2). Layer 4, which is XPS core, has Tsai-Wu strength index 0,08 (on the right in Figure 2). As all strength indexes are under 1, failure does not occur in the sandwich panel. The data (stress in tension, compression and shear), for the Tsai-Wu failure criteria, are obtained from in house testing and from [5] [9].

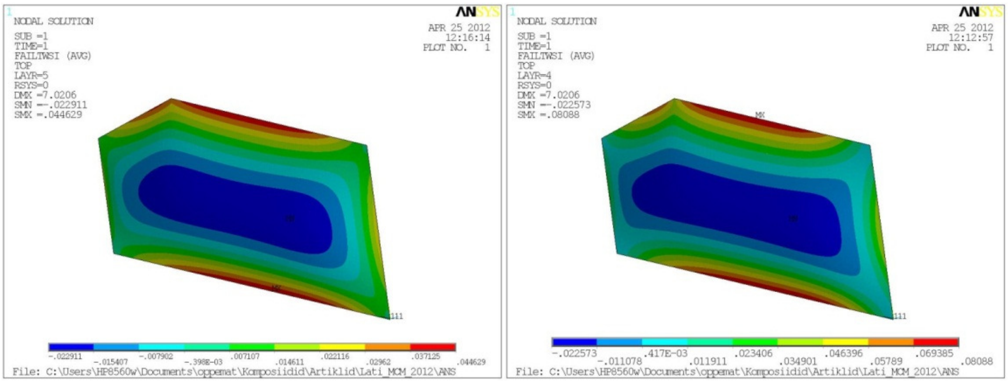

Figure 2. Tsai-Wu strength index. On the left is 5th layer and on the right is 4th layer.

4.2 Results of Pareto frontier

Pareto frontier, of the above posed problem, is not continuous line and consists of four parts (Figure 3). Each part of the Pareto frontier corresponds to different core material. There are five different core materials considered (Table 4), but PMI core material does not generate any point in Pareto frontier (it is more expensive, but the deflections remain in the same range).

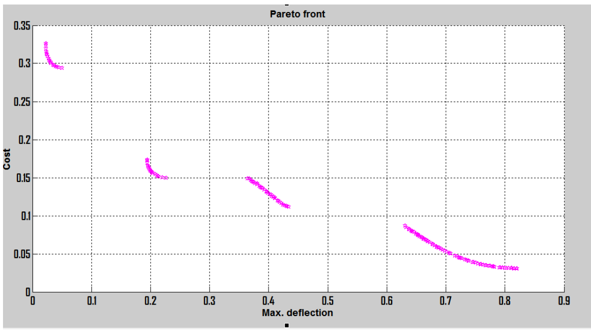

Figure 3. Pareto frontier: maximal deflection vs. cost of the material.

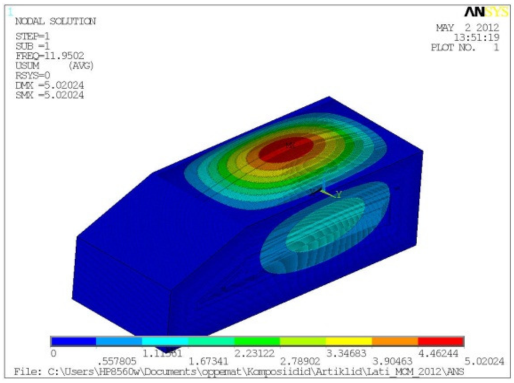

Figure 4. Trailer's first natural frequency.

4.3 Results of trailer modal analysis

Based on optimal design and Pareto frontier, test No. 4 lay-up (Table 1), with PET foam, has the best properties for lightweight and cost effective sandwich panel. Such lay-up is used in current modal analysis. According to study [10], 2,5 ton truck working frequencies can be between 15 and 40 Hz. Those frequencies are used as reference values for trailer development.

Table 5 describes trailer's natural frequencies. It can be seen that the first natural frequency is lower than working frequencies, which could lead resonance. In order to avoid this situation, the trailer is made stiffer.

Condition	Mode 1 [Hz]	Mode 2 [Hz]	Mode 3 [Hz]	Mode 4 [Hz]
Conventional trailer	12,0	17,5	19,7	21,7

Table 5. Trailer's natural frequencies.

Figure 4 depicts trailer's first natural frequency. It can be seen that the biggest deformation area is in the roof panel. Subsequently the biggest deformation areas are made stiffer until the first natural frequency is over 40 Hz. As 40 Hz needs quite thick panels (≈ 60 mm), it is considered that 40 Hz can be too high frequency for trailers and proper road testing should be made.

5 Conclusion

Optimal design of sandwich panel has been performed. The sandwich structure with minimum deflection was determined keeping the cost of the material minimal. Non-linear static FEA has been made to analyze trailer's sandwich panels in extreme wind condition. The failure in the panels has been predicted using Tsai-Wu failure criteria. Based on FEA results, the mathematical model has been composed by applying artificial neural networks. The mixed integer optimization problem with two objectives has been formulated and solved by applying multi-criteria optimization strategies and Pareto optimality concept.

The dynamic properties have been studied by performing modal analysis. It can be concluded that: extreme wind condition do not damage sandwich panel; Pareto frontier can be used to select the laminate with best cost/deflection or cost/weight ratio; and modal analysis is very important in making trailer stiff enough to avoid resonance.

In future studies is planned to make proper road testing and analyze what happens with sandwich panel when the trailer is working in the resonance mode.

6 References

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