AUTOMATED STRUCTURAL DESIGN OPTIMIZATION OF COMPOSITE HIGH-PRESSURE VESSELS CONSIDERING MANUFACTURING CONSTRAINTS

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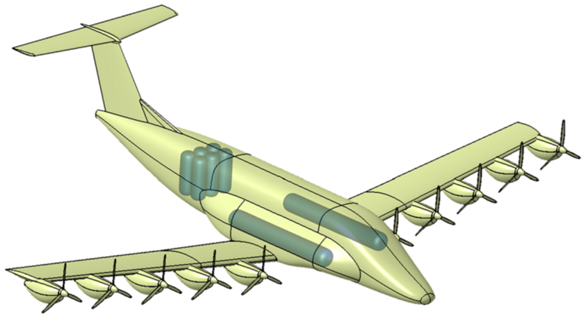

FIG 1. Possible Tank Configuration for D-Light+

1. INTRODUCTION

Lightweight pressurized hydrogen storage is a key technological enabler for carbon-neutral transportation systems. In the design of novel hydrogen aircraft, pressure vessels of different geometries and configurations must be evaluated as part of the overall aircraft layout. Since the requirements are highly specific, standard off-the-shelf solutions are rarely suitable. In the space sector, similar challenges arise, where propellant pressure vessels are needed for the exact dimensions and design parameters specified by the spacecraft. Within the DLR research projects D-Light and D-Light+ [1], a 9-passenger hydrogen-electric aircraft of the CS-23 category is being developed (Figure 1). For the automated multidisciplinary design workflow which has been developed, hydrogen tanks of varying sizes and positions need to be modelled, to find the optimum overall configuration considering weight, center of gravity, safety, complexity.

The current state of the art for hydrogen pressure tanks are Type-IV vessels, which consist of a polymer liner reinforced with circumferential and helical carbon-fiber layers through the filament-winding-process. They are designed for storage pressures up to 700 bar with a burst pressure safety factor of at least 2.0. Filament winding imposes important manufacturing constraints: the polar opening radius is determined by the winding angle, additional thickness accumulates at small radii due to band overlap. Thus, an optimal composite layup must satisfy both mechanical performance and manufacturability. To address these challenges, a novel method was developed that can automatically generate suitable filament-wound layups for high-pressure Type IV vessels.

While previous studies have developed algorithms to improve initial suboptimal pressure vessel designs (Alcantar [2] and Jiang [3] used genetic algorithms and simulated annealing for this purpose) or to optimize a limited number of layers [4], to the authors' knowledge no complete "from scratch" optimization method which takes into account the filament winding boundary conditions has yet been demonstrated.

2. OPTIMIZATION METHOD

The optimization algorithm, tankoh2 is an extension of an open-source framework originally created for thin-walled liquid hydrogen vessels [5]. It is implemented in Python and relies on the commercial program μWind to simulate winding and thickness buildup [6]. The optimization proceeds layer by layer, with each new ply selected to maximize performance while respecting winding constraints.

2.1. FEM Analysis

The structural analysis of the vessel is based on an axisymmetric shell-element FEM mode, which is also evaluated by μWind. In the cylindrical region of the vessel, stresses are dominated by circumferential loading, whereas in the dome and transition regions the critical axial stresses arise due to bending of the dome. Because the failure of composite pressure vessels is generally caused by tensile failure of the fibers, Puck's tensile fiber failure criterion is used within the structural analysis

Because the shell model cannot capture the radial stress distribution that develops in thick-walled cylinders, in which inner circumferential layers experience higher tensile stress than outer layers, an analytical 3D-elasticity tube model [7] is used as a correction:

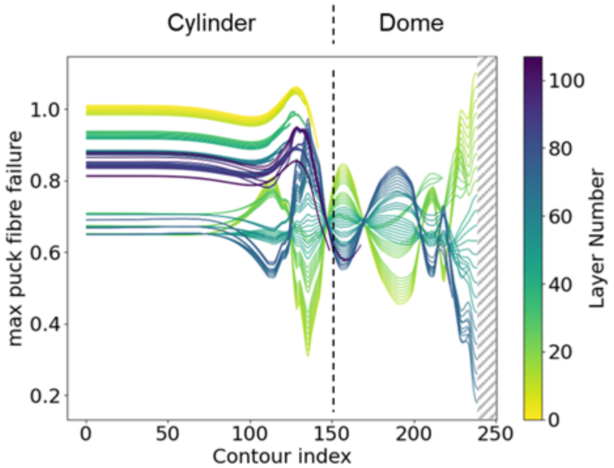

FIG 2. Puck fibre failure indexes along the contour of a reference vessel

the stress of each layer is scaled by the ratio of its circumferential strain to the mean circumferential strain within the 3d-Tube model. For a typical 700-bar pressure vessel, this effect is quite significant. The innermost hoop layers experience up to 15% higher stress than predicted by a shell model. . Figure 2 shows the puck fibre failure criterion, corrected by the thick-shell model, for a reference vessel.

The metallic boss is not modelled. Because the fixed boundary conditions at the polar ends of the domes can lead to unrealistically high stresses, the elements of the final bandwidth are ignored for the optimization. To improve robustness, an additional dome design factor can also be applied. This accounts for higher manufacturing uncertainties during winding of the dome region and shifts the critical failure mode toward the cylinder, where the analysis is more reliable.

2.2. Choosing Optimal Winding Angles

If the highest failure index is found in a hoop layer, another hoop layer is added. Otherwise, a new helical layer is chosen. For each added helical layer, the winding angle is found by minimizing a composite weighted target function, consisting of:

1) the maximum Puck fiber failure index across all layers

$$\mathrm{FF}_{\max} = \frac{\sigma_{1,max}}{R_{1t}}$$

2) the integral of bending-induced strains along the dome contour

$$I_\varepsilon = \int (\varepsilon_{x,\text{top}} - \varepsilon_{x,\text{bottom}})\, ds$$

3) a windability function that penalizes too steep or negative thickness gradients in the dome

$$1 \text{ if}(\ 0\ \ \text{mm/mm}\ < \frac{\mathrm{d}t}{\mathrm{d}s} < 0.5\ \ \text{mm/mm})\,;\ \text{else } 0$$

Only winding angles that yield windable contours are passed to the FEM analysis. Because the target function is non-smooth and may have local minima, a genetic algorithm is employed to find the optimum. Figure 3 shows an example target function during the optimization process.

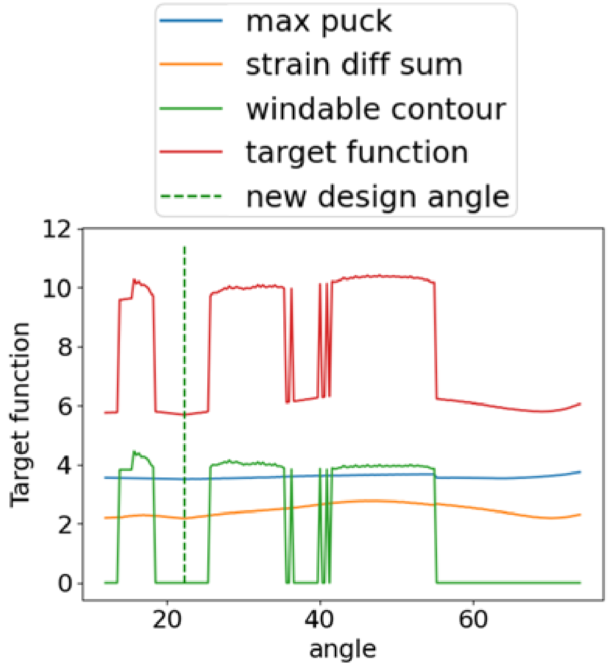

FIG 3. Target function of the optimizer

2.3. Postprocessing Step

After each layer is added to the layup, the stacking sequence can be reordered.

Hoop plies are grouped into more easily manufacturable clusters, and the run-outs of the hoop layers are evenly spread to avoid abrupt stiffness changes between the cylinder and dome. High-angle helical plies, which are often needed to support the dome-cylinder transition region, are sorted to the outside, to keep hoop and low-angle helical layers close together and to maintain a smooth winding surface for other helical layers.

Optionally, the winding angles of helical layers can be adjusted to guarantee valid coverage patterns. Not all winding angles naturally lead to full surface coverage without leading to significant overlap. An angle for a valid winding pattern, with a defined overlap, must satisfy

$$\text{progress}(\alpha) = k \cdot (\frac{\text{bandwidth} - \text{overlap}}{\cos\alpha}) \text{ for } k \in \mathbb{Z}^+$$

$$\gcd(k, \text{cycles}) = 1.$$

The optimization routine can find the closest winding angle that fulfill these conditions.

2.4. Deleting Obsolete Layers

After enough layers have been added and a layup is reached that does not fail the Puck criterion, sometimes obsolete layers can be deleted without reducing the strength of the vessel. In an optional step, the

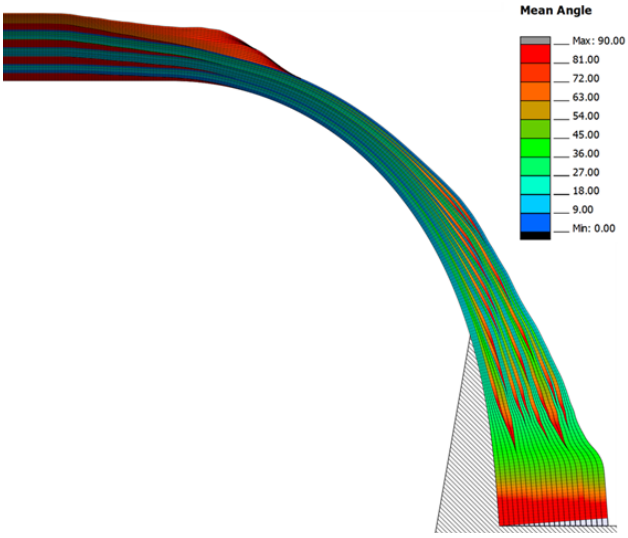

FIG 4. Optimized Laminate Contour

hoop shifts are optimized to minimize the Puck criterion further and allow more layers to be removed.

3. RESULTS

4. VALIDATION

The procedure was validated against a reference design. Table 1 summarizes the test case parameters. Because inter-fiber failure generally occurs everywhere in the vessel during pressurization, but is not critical to the structural integrity of the tank, matrix-dominated values ($E2$, $G12$, $\nu12$, $\nu23$) are reduced to 10% of their undamaged values.

TAB 1. Design parameters for the optimization study

Diameter	415 mm
Length	4400 mm
Pressure	700 bar
Hydrogen capacity	22 kg
Burst Safety factors	2.25
Design Margin	1.15
Dome Margin	1.13
Material	IM7/YD128, $R_{1t} = 2600$ MPa

Figure 4 shows the optimized laminate contour, with layers colored by the local band angle, which was generated in about 30 minutes. For the same conditions and safety factors, the optimizer achieved a slightly lighter design (see Table 2).
Stress distributions were also compared with higher-fidelity axisymmetric solid models including metallic fittings and contact conditions, exported and calculated in Abaqus (see Figure 5. In the cylindrical region, the results agreed closely. In the dome, where no analytical scaling was applied, the shell model underestimated stresses in the inner plies and overestimated them in the outer plies. This effect is seen for multiple layups, and may be related to the thick-walledness of the dome, just as in the cylinder. A local stress peak was also observed near the metallic boss transition, which could not be accounted for in the optimizer model.

TAB 2. Comparison with reference design

Reference Design	Optimized Design
108 Layers	104 Layers
(48 Hoop	46 Hoop)
(45 Low-Angle	43 Low-Angle)
(15 High-Angle	15 High-Angle)
290.12 kg	279.74 kg
$\mu = 6.93\%$	$\mu = 7.17\%$

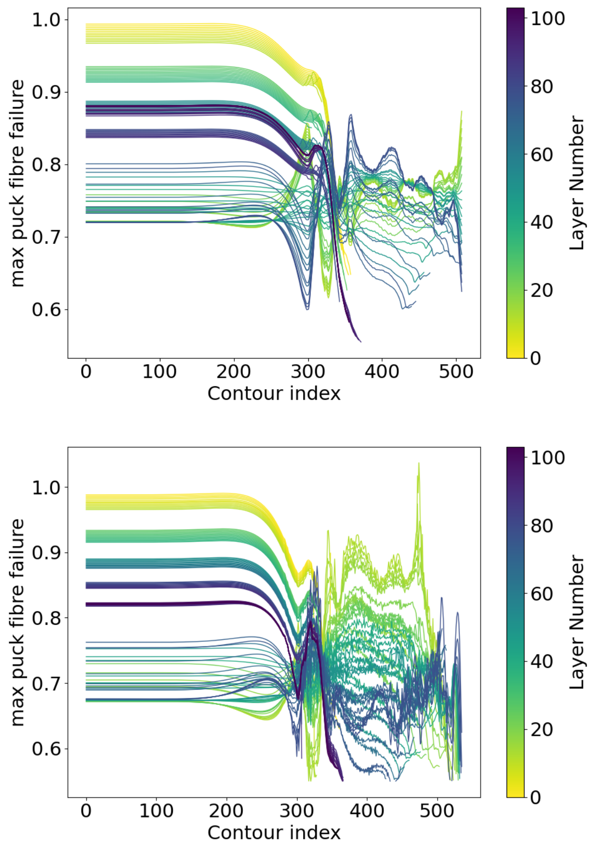

FIG 5. Puck Fiber Failure Indexes for Corrected Shell Model (Top) and Solid Model with Boss (Bottom)

4.1. Application to Trade Studies

To enable fast design exploration within the D-Light design workflow, the optimizer was used to generate Kriging metamodels, using the routines developed by Freund [5]. A design of experiments with 301 optimization runs per material was performed, varying diameter, length, burst pressure, material, and dome design factor and with vessel mass, volume, and wall thickness as outputs.
With the optimizer, it also possible to run designs of experiments to investigate the influence of different ge-

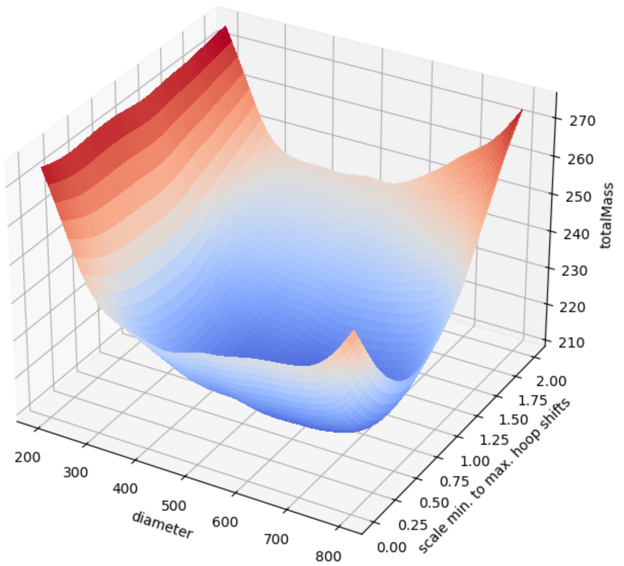

FIG 6. Vessel mass by diameter and hoop shift balance for constant volume

ometric and winding parameters. Figure 6 shows a study in which the diameter was varied between 200 and 700 mm for a constant volume (for 22kg of h2), and the winding parameter "hoop shift balance" was varied from 0 to 2. This parameter describes the ratio of the distance between the end of the furthest-extending hoop layer in the dome, to the distance to the earliest-ending layer in the cylinder, measured from the border between the cylinder and dome on the liner contour. Effectively, increasing this parameter spreads the stiffness drop-off from cylinder to dome over a larger length. The study shows that for all diameters, the optimum lies at around 0.7. With this setting, diameters from 400 mm to 700 mm achieve very similar mass efficiency. For lower and higher diameters, the efficiency starts to reduce. It still needs to be further investigated if this has a physical background or is related to the ability of the optimizer.
The presented optimization routine can be used for parameter studies such as these, or to quickly create reasonable layups for detailed numerical or experimental investigations into structural mechanics of pressure vessels.

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