DESIGN-OPTIMIZATION OF CYLINDRICAL, LAYERED COMPOSITE STRUCTURES USING EFFICIENT LAMINATE PARAMETERIZATION

A. Mönicke[1], H. Katajisto[1], M. Leroy[2], N. Petermann[2], P. Kere[3], and M. Perillo[4]

[1]*Componeering Inc., Itämerenkatu 8, 00180 Helsinki, Finland,*
Email: andre.moenicke@componeering.com
[2]*ASTRIUM Space Transportation, avenue du Général Niox, 33160 Saint-Médard-en-Jalles, FRANCE,*
Email: michel.leroy@astrium.eads.net
[3]*Tampere University of Technology, Department of Mechanics and Design, P.O. Box 589, 33101 Tampere, Finland,*
Email: petri.kere@iki.fi
[4]*EnginSoft S.p.A., Via A. Murri 2, ZI, 72023 Mesagne, Italy, Email: m.perillo@enginsoft.it*

ABSTRACT

For many years, layered composites have proven essential for the successful design of high-performance space structures, such as launchers or satellites. A generic cylindrical composite structure for a launcher application was optimized with respect to objectives and constraints typical for space applications. The studies included the structural stability, laminate load response and failure analyses. Several types of cylinders (with and without stiffeners) were considered and optimized using different lay-up parameterizations. Results for the best designs are presented and discussed.

The simulation tools, ESAComp [1] and modeFRONTIER [2], employed in the optimization loop are elucidated and their value for the optimization process is explained.

1. INTRODUCTION

Layered composites are widely used in space structures and often they are vital for achieving certain structural properties, as for example very low thermal expansion, and meeting mass targets. Cylindrical composite structures are of special interest because they provide the core function within many types of space structures, e.g. load adapters, propellant tanks, fairings in launchers, and satellite central cylinders.

Material selection, layer orientations and thicknesses can be varied for each layer. Hence optimization tasks involving layered composites can quickly become challenging owing to the number of design variables involved. The size of the optimization problems in terms of the number of variables rapidly increase increases when the level of details is expanded. Thus it is desirable to find optimal designs that satisfy global requirements early in the design phases. A possible pre-design optimization process is described in this paper.

2. CASE SPECIFICATION

2.1. Structure

The input data for the cylindrical composite shell to be optimized towards minimum mass was provided by ASTRIUM Space Transportation and included requirements for safety factors, which are further discussed in Section 2.2, main dimensions, stiffeners and lay-ups, as well as loads and boundary conditions.

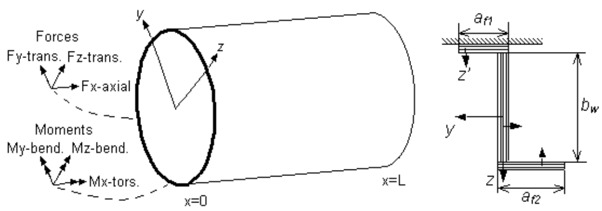

Figure 1. ESAComp cylindrical shell and load coordinate system schematics (left) and the Z-stiffener (right).

The cylinders had a fixed length of $L = 600\,\mathrm{mm}$ and a diameter of $d = 700\,\mathrm{mm}$. One end of the cylindrical shell was clamped and the load, which was a combination of a transverse force Fy=200kN and a counterbalancing bending moment My=-100kNm, was introduced into the other end. Both ends of the cylindrical shell were rigid in the radial direction. The coordinate system used in the model is illustrated in Fig. 1.

To withstand the loads and deliver a required buckling stability Z-stiffeners were allowed to be used in the structure with some limitations. The minimum spacing was 75 mm, their maximum laminate thickness $3\,\mathrm{mm}$, and both maximum developed height b and maximum width were 30 mm (see also Fig. 1), where the height of the web b_w was automatically calculated by subtracting the laminate thicknesses of the flanges from the desired developed height.

The original requirement that stiffeners were supposed to be shorter than the shell was not used in this study for axial stiffeners, as that feature is not available for axial stiffeners in the utilized analysis tools, therefore axial stiffeners cover the full length of the cylinder. An additional design study involving circumferential stiffeners, here referred to as ring stiffened cylinders, was also performed.

2.2. Preliminary design

In the pre-design phase appropriate factors of safety were defined for the stability studies. General safety factor of 20% was selected. Knock-down factors (KDF) have been experimentally determined for axially compressed cylindrical shells. The widely referred test results have been presented e.g. in [3]. For this specific design study a KDF of 0.75 was selected to consider geometrical imperfections of the structure. For axially loaded structures this factor would represent approximately the mean value of the test results for cylinders with radius-to-thickness ratio of 100. Thin-walled structures in axial compression are vulnerable to imperfections. Other types of loads are not that critical and therefore, this selection can be considered conservative.

ESAComp has been benchmarked against various design studies [3, 4, 5]. The reference cases cover thin-walled composite cylindrical shells loaded with axial force. The benchmarking was made against the simulation results presented in these papers. A general trend was that ESAComp predicted a slightly higher buckling load but not more than 5%. Therefore, an additional safety factor of 1.05 was selected. For the efficient optimization runs it is important to select an appropriate finite element mesh density. The selected mesh should not be too dense due to the penalty in the computation time. On the other hand, too coarse mesh would yield optimistic results. Before launching the optimization run mesh sensitivity studies were performed for the different design studies. The purpose was to determine a mesh density that guarantees solutions that do not deviate more than 5% from the fully converged case. An additional safety factor of 1.05 was selected to present the uncertainty in the accuracy of the results. The resulting safety factor of 1.76 was obtained.

Factor of Safety (FoS) of 3 was required for the first ply failure. Analyses were made with the nominal load and therefore, associated Reserve Factor (RF_{FPF}) of 3 was required, and a Buckling Reserve Factor (RF_{B}) of 1.76 respectively. Max Stress 3D was used as failure criterion throughout the studies.

During the preliminary design also the unidirectional carbon fiber reinforced (CFRP) material HM30S 150 ER432 by SAATI S.p.A. was chosen from the ESAComp material database to be used for the studies. Detailed material data is presented in Tab. 1.

Engineering constants		FF stresses	[MPa]
E_1 [GPa]	169	X_t	2662
$E_2 = E_3$ [GPa]	7.9	X_c	1093
$G_{12} = G_{31}$ [GPa]	4.17	$Y_t = Z_t$	64.6
G_{23} [GPa]	2.93	$Y_c = Z_c$	200
$\nu_{12} = \nu_{13}$	0.38	S	64.6
ν_{23}	0.35	R	64.6
t_{ply} [mm]	0.159	Q	46

Table 1. Mechanical properties of the unidirectional M30S carbon fiber reinforced ER432 epoxy plies used for the cylindrical shells.

2.3. Optimization problem

The main objective was to minimize the mass of the cylinders by optimizing the composite lay-ups and stiffener sizing parameters

A simple single-objective formulation for the optimization problem would thus only include

$$\min_{\vec{x}\in \mathrm{S}} m\left(\vec{x}\,\right) \tag{1}$$

with the vector of design variables $\vec{x}$, the mass of the structure m and the feasible set S defined by constraints as $\mathrm{S} = \{\,\vec{x} \mid \vec{g}(\vec{x}\,) \leq 0,\ \vec{h}(\vec{x}\,) = 0\}$.

However, for part of the study maximizing the stability reserve factor (RF_B) was included and thus a multi-objective optimization of the form

$$\min_{\vec{x}\in \mathrm{S}} \vec{z}\,(\vec{x}\,) = \min_{\vec{x}\in \mathrm{S}} \begin{bmatrix} m(\vec{x}\,) \\ -RF_B(\vec{x}\,) \end{bmatrix} \tag{2}$$

applied. The vector objective function $\vec{z}$ contains the mass $m(\vec{x}\,)$ of the structure and its buckling reserve factor $RF_B(\vec{x}\,)$. Further constraints implied in the set S are explained only verbally in the following.

Due to using a genetic optimization algorithm, which only supports integer variables, the stiffener dimensions are optimized with a range of 10 mm to 30 mm in steps of 5 mm The allowed quantities of axial stiffeners were 12, 18, 24 or 30. Respectively, from two to seven stiffeners were allowed for ring stiffened cylinders

For layer orientations with variable layer angle allowed range varied from 15° to 75° in steps of 3°. Other requirements for the lay-ups are discussed in the next section.

The results for the different types of cylinders and parameterizations were obtained using a full factorial design for the cases where no more than 512 possible designs existed. For others Latin Hypercube Sampling with 40 to 80 designs, depending on the amount of variables, was used as an initial generation with the multi-objective genetic algorithm MOGA II included in modeFRONTIER.

3. LAY-UP PARAMETERIZATION

General design rules for the lay-up given by ASTRIUM Space Transporation included the following: Reference lay-ups to compare optimization results were $[90/45/0/-45]_{n/\text{SE}}$ and $[90/(45/-45)_n]_{/\text{SE}}$ with the multiplier n to be determined for our chosen material system. Other possible lay-ups were required to have at least one 90° layer on the outside and on the inside. The lay-ups should be symmetric (symmetric even or symmetric odd) and balanced. Also, no consecutive layers are allowed to have the same orientation, except at the symmetry plane.

As the final lay-ups of the optimized cylinders are rather thin (not more than 10 for the symmetric half stack) engineering oriented lay-up parameterization based on elementary laminates and the use of a stacking sequence vector as employed in earlier optimization studies of the authors[6] were not used. However, the same tools were used and with respect to laminate parameterization the same Extensible Markup Language (XML) based scripting together with a parsing interface provided by ESAComp. This dedicated scripting language supports sub-laminate based laminate design, which reflects the manufacturing process of the final composite part better than the traditional zone based design. Using zone based design for composite structures one has to keep track of laminate changes and there effect on the whole structure manually, where as in sub-laminate based design changes automatically are automatically applied. With the scripting language the user can further built groups of allowed material systems from which the materials can be chosen independently for the different layers of the sub-laminate during the optimization. Also different symmetry options as well as the automatic balancing of lay-ups are included.

Based on the results for the optimized quasi-isotropic lay-up and the constraints, several general lay-up formulations are introduced and partly combined. Fig. 2 illustrates one such a formulation, in which the parameterization is made simply in terms of layer orientations. In other formulations layer multipliers were included as design variables. Combining different formulations without any extra measures might not be the most efficient method with respect to the number of designs needed to find near optimal designs[6], but it was found feasible since already the integration of symmetry, balance, and other constraints in the formulation, reduces the designs space to a feasible set with respect to the stacking sequence.

When considering the effectiveness of a parameterization for a composite lay-up optimization and especially in such an early design phase where the calculation time per model is small and the options to be considered are plenty, one should not just take the computation time into account, but also the time to set up the different problems, and how well one can afterwards review what was done. Being able to include certain manufacturability constraints and material choice easily into the optimization set-up with options for extension and modification to dynamically create laminates based on different parameterization approaches can save a lot of time in the design process. The approach how different parameterization options can be implemented is important and should be addressed by optimization environment used.

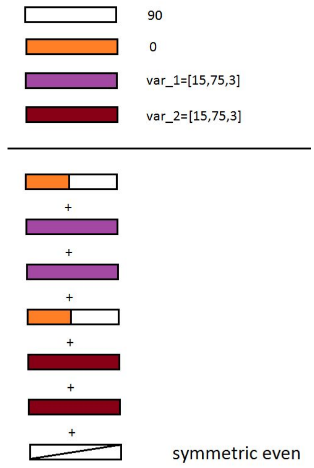

Figure 2. Example for a relatively simple laminate coding with fixed layer mutlipliers and variable orientations only.

4. SIMULATION AND OPTIMIZATION ENVIRONMENT

The design environment utilizes optimization features of modeFRONTIER - software for process integrated multi-objective design optimization - along with ESAComp - software for design and analysis of composites - applying sub-laminates lay-up parameterization approach.

ModeFRONTIER drives the optimization process. The structure is described by parameterized laminate definitions and geometric parameters, where the input variables concerning the laminate parameterization are imported from the dedicated ESAComp interface. Due to the vast choice of design of experiments methods and optimization algorithms, as well as many integrated post-processing and decision making features modeFRONTIER is well suited for a wide area of problems. Similar optimization tasks can be executed swiftly when making use of the previous projects, for which modification is easy as modeFRONTIER maps the whole optimization problem. [7]

The cylindrical shell analysis of ESAComp provides a powerful pre-design tool to assess global features of cylindrical structures. Depending on the variables, potential laminates, destined for the cylinder structure, are created and evaluated. A finite element (FE) model is solved using the FE solver ELMER[8], which is integrated in ESAComp, and the results of the composite-specific post-processing as well as mass of the structure and buckling stability, are reported to modeFRONTIER to complete the design loop.

Calculations for the cylindrical shells in ELMER are based on a Reissner-Mindlin-von Kármán type shell facet model. The plate bending problem is formulated for a thin or moderately thick laminated composite plate. To obtain the load-displacement curve and to study the stability behavior, the nonlinear equations are solved iteratively by Riks' method with Crisfield's elliptical constraint for arc length [9, 10, 11, 12]. The algorithm as explained in [13, 14] is based on the Newton iteration, which follows the principal equilibrium path. Using the arc-length methods for solving the nonlinear equilibrium equations, a load-displacement constraint is added to the system. ELMER solver provides generality in terms of solving problems with large deformation. It should be noted that for the sake of performance design-optimization was made using linear static analyses.

5. RESULTS

Main results for the optimized monolithic cylinders are presented in Fig. 3. The cylinders with the quasi-isotropic and $[90/(45/-45)_n]_{/\mathrm{SE}}$ lay-up are more than 40% heavier than the designs which allowed for different layer orientations. Though the design with the sub-sequent $\pm 45°$layers leads to sufficient stability already at a much lower weight than the final 7.8 kg, the first ply failure safety cannot easily be reached. On the contrary, the quasi-isotropic was more balanced with respect to the reserve factor.The symmetry condition forces to use building blocks of 8 layers. This constraint prevents finding a low mass design meeting the constraints. The option with the variable orientations is more flexible in the design, thus margins to the constraints can be exploited better.

Generally, more light-weigth designs could be obtained with stiffened than with monolithic cylindrical shells.As shown in Tab. 2 the best monolithic designs is almost 25% heavier than the stiffened designs. Here the designs were chosen first by the lowest mass, followed by the highest stability and at last by the highest reserver factor against first ply failure . It should be noted, that also slightly heavier designs ($\leq 5\%$) with considerably better buckling stability (increased by $\geq 10\%$) were found.

The buckling mode shapes of the monolithic cylinder and the cylinder with axial stiffeners look similar with respect to the wavelength and direction. But for a lighter shell, the stiffeners in the longitudinal direction increase the stiffness of it and thus lead to lighter designs despite the additional weight of the stiffeners. The ring stiffeners, on the other hand, suppress the global buckling with a short wavelength in circumferential direction owing to there orientation and a new, more localized buckling mode occurs.

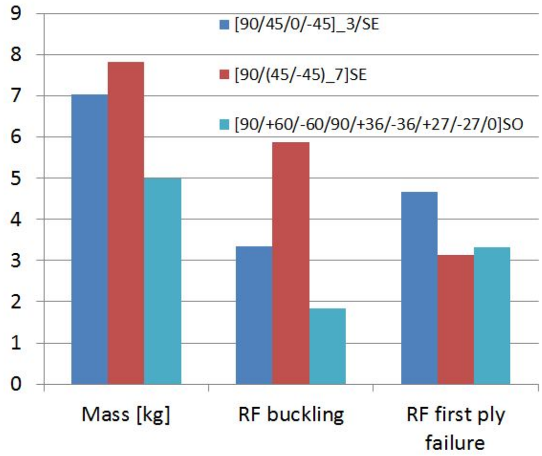

Figure 3. Comparison between different lay-up formulations for solid cylinders.

Particularly for the monolithic option many designs achieved almost identical results for the stability and for the reserve factor against first ply failure with the same mass as the best design. In comparison, the design distributions in the criterion space of the stiffened cylinders studied, for example, with scatter plots of mass over first ply failure and stability(see Fig. 4), indicate that the stiffened cylinder designs are more sensitive to layer orientations. However, the design spaces of the stiffened cylinders are far bigger than the ones of the monolithic cylinders. Therefore, when processed with the same methods, more designs are needed to properly explore the bigger design space. To assess the robustness of the designs additional calculations considering the variations of the parameters of materials and misalignment in the fiber placement could be carried out.

6. POST-PROCESSING

Computational cost is relatively high for nonlinear simulation and therefore, it is not well suited for design-optimization. Instead, it can and should be used as a a post processing step to study how sensitive the optimal designs are for geometrical imperfections. In this study the shape for the geometrical imperfection was obtained using the first mode shape of the linear eigenvalue analysis. Another possibility would be to use an analytical approach for the definition of the imperfection. Such an approach is described in [5], for example, and it produces global-like imperfections. This function is also supported in ESAComp, but the challenge of using such an approach would have been in the determination how many waves are set in the principal directions and if

Data	Monolithic	Ring stiffened	Axial stiffened
Shell lay-up	$[90/\pm60/90/\pm36/\pm27/0]_{\mathrm{SO}}$	$[90/\pm24/0/\pm36]_{\mathrm{SE}}$	$[90/\pm45/\pm30/0]_{\mathrm{SO}}$
Mass [kg]	4.98	4.04	4.07
RF_{B}	1.83	1.80	2.02
RF_{FPF}	3.32	3.04	3.13
Stiffeners	0	6	30
$a_{f1} = a_{f2}$ [mm]	-	10	15
b [mm]	-	25	25
Stiffener lay-up	-	$[\pm15]_{\mathrm{SE}}$	$[\pm18]_{\mathrm{SE}}$

Table 2. Design data and results for the best designs for monolithic cylinder, cylinder with ring stiffeners and cylinder with axial stiffeners.

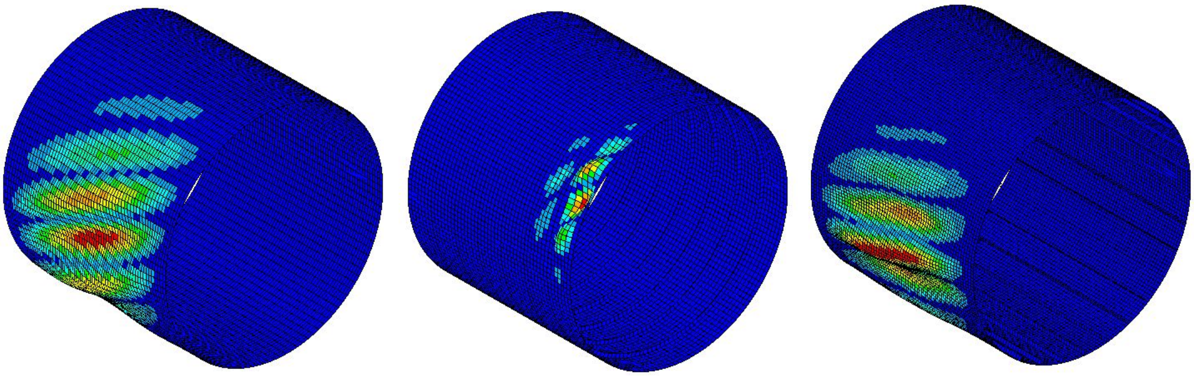

Figure 5. First buckling mode shapes for the best designs for monolithic cylinder (left), cylinder with ring stiffeners (center) and cylinder with axial stiffeners (right).

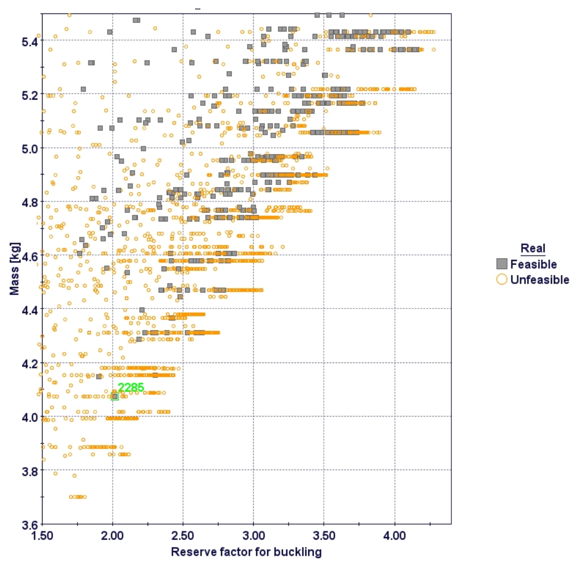

Figure 4. Buckling reserve factor - mass plot of a multi-objective optimization run for cylinders with axial stiffeners.

skewedness is included. For this type of structural applications the amplitude of the imperfection can be expected to be in the scale of 1mm. For example, in [4] amplitude less than 1 mm was experimentally verified.

ESAComp implementation that uses Elmer FE solver[8] is based on the Riks' method. For this study the parameters of the Crisfields's arc-length constraint were set so that the problem was run based on the load-controlled method. The nonlinear simulation was performed for a set of optimum designs and results for the monolithic cylinders are presented as normalized load-displacement curves in Fig. 6. The respective laminate lay-ups are presented in Tab. 3. It should be noted that not all of these laminate designs meet the design requirements in terms of the stability constraint. The purpose of the post processing was to study how sensitive designs created according to different laminate parameterization are to geometrical imperfections.

In Fig. 6 the maximum load level (1 in y-axis) corresponds to the buckling load of the specific laminate design based on the linear eigenvalue analysis. Respectively, the displacements were normalized against the maximum displacement obtain using the linear elastic model. For all studies the amplitude of the imperfection was 0.25 x laminate thickness. Results indicate that for some laminate designs the load-displacement curve was linear up to the maximum load level. For these designs it is obvious that the load bearing capability would be higher than the one predicted with the linear eigenvalue analysis. Some of the designs indicated considerable nonlinear pre-buckling behavior when approaching the limit load predicted with the linear eigenvalue analysis. However, for all designs the load bearing capability determined by the nonlinear analysis was at least 95% of the limit load. Therefore, optimized structures can be considered feasible.

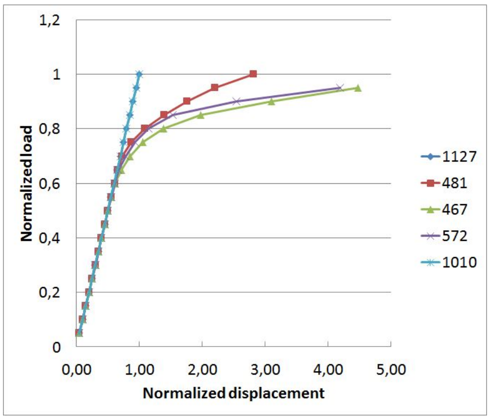

Figure 6. Non-linear load-displacement curves for monolithic cylinders with geometrical imperfections.

Design ID	Laminate lay-up
1127	$[90/\pm 55/\pm 50/15/0/-15/0]_{SO}$
481	$[0/90/0/90/0/45/0/-45/0]_{SO}$
467	$[90/42/90/-42/90/27/90/-27/90]_{SO}$
572	$[90/51/90/-51/90/36/0/-36/0]_{SO}$
1010	$[90/\pm 66/\pm 48/30/0/-30/0]_{SO}$

Table 3. Mechanical properties of the unidirectional AS4 CFRP plies used for the cylndrical shells.

7. CONCLUSIONS

In the early design phase of a structure usually various types of design options have to be considered. Thus tools which support a quick setup of separate optimization runs for those design options by providing re-usability of workflows as in modeFRONTIER are of great benefit for the designer. ESAComp offers means to dynamically create various types of laminates taking manufacturability into account with different composite specific parameterization types. Concerning the involvement of non-linear analysis in the optimization loop the analysis feature in ESAComp needs further development to increase its performance. On the other hand, it is not clear how beneficial for the whole design process the use of non-linear analysis in an early design phase is.

REFERENCES

[1] *ESAComp - Software for Analysis and Design of Composites*, Release 4.3, http://www.esacomp.com ESAComp web site.

[2] *modeFRONTIER*, modeFRONTIER web site at http://www.esteco.it.

[3] Hühne C. et al., A new approach for robust design of composite cylindrical shells under axial compression, *Proceedings of the European Conference on Spacecraft Structures, Materials & Mechanical Testing 2005*, SP-581 - September 2005.

[4] Meyer-Piening H.-R., Farshad M., Geier B. and Zimmerman R. Buckling loads of CFRP composite cylinders under combined axial and torsion loading - experiments and computations, *Composite Structures 53 (2001)*, pp. 427–435

[5] ECSS-E-HB-32-24A, 24 March 2010

[6] Mönicke, A., Katajisto, H., Kere, P., Palanterä, M., Perillo, M., "Engineering Oriented Formulation for Laminate Lay-up Optimization", *Journal of Structural Mechanics*, Vol. 41, 2008, pp. 137–151.

[7] Bassanese, A., Ozen, M., Mönicke, A., Katajisto, H., Palanterä, M., Spagnolo, M., Micchetti, F., Perillo, M., "Process Integration and Multi-Objective Design Optimization as New Design Methodologies for Composite Structures", *Proc. SAMPE '08*, Long Beach, CA, USA, 18–22 May, 2008.

[8] *Elmer Open Source Finite Element Software for Multiphysical Problems*, Release 5.4, http://www.csc.fi/elmer Elmer web site.

[9] Kouhia, R., On the Solution of Non-linear Finite Element Equations, Computers and Structures, vol. 44, pp. 243–254, 1992.

[10] Ramm, E., Strategies for Tracing the Nonlinear Response Near Limit Points, Nonlinear Finite Element Analysis in structural Mechanics, Springer Verlag, Bochum, 1981.

[11] Rheinboldt, W. C., Numerical Analysis of Parametrized Nonlinear Equations, Wiley, New York, 1986.

[12] Riks, E., The Application of Newton's Method to The Problem of Elastic Stability, *Journal of Applied Mechanics*, vol. 39, pp. 1060–1065, 1972.

[13] Kere, P. and Lyly, M., On Post-Buckling Analysis and Experimental Correlation of Cylindrical Composite Shells with Reissner-Mindlin-Von Kármán Type Facet Model, *Computers & Structures*, Vol. 86, 2008, pp. 1006–1013.

[14] Kere, P. and Lyly, M., Reissner-Mindlin-Von Kármán Type Shell Facet Model for Buckling Simulation of Imperfect Cylindrical Composite Shells, *Mechanics of Advanced Materials and Structures*, Vol. 18, 2011, pp. 115–124.