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Design and analysis of full composite pressure vessels

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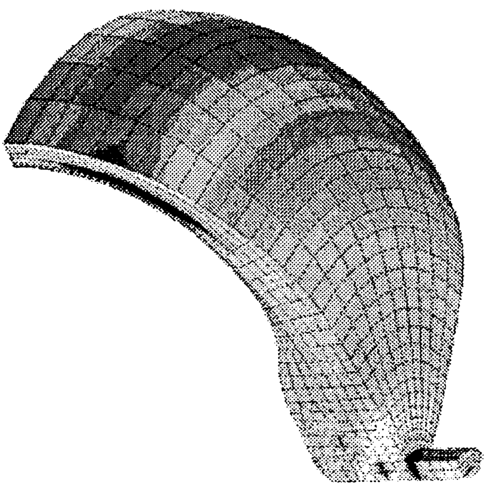
Abstract

The paper presents the design and analysis of full composite pressure vessels with a finite element module. Through a full-parametric, three dimensional modelling and a resolution of the load bearing tank structure in different layers, different dome geometries were optimised with respect of weight and material use by variation of the lay-up. The design method for pressure vessels offers a weight reduction in comparison to standard hoop reinforced aluminium vessels of 50 % and to pure steel vessels of 75 %.



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1 Introduction

As a result of the low pollutant combustion, natural gas or fuel cell driven automobiles are able to assist the relief of the environment in conurbations. A special problem is the comparable costly storage technology in the vehicles. Two methods are in common: "liquefied natural gas" (LNG) in cryogenic tanks with a temperature of approximately – 160 °C or "compressed natural gas" (CNG) in high pressure vessels with a service pressure of at least 200 bar.

CNG-pressure vessels are designed normally in cylindrical form as pure steel tanks or as steel or aluminium liner wrapped over with fibre reinforced plastics (FRP). The relatively heavy weight in comparison to the storage volume decreases the range of such vehicles down to 200 - 250 km. Consequently, only bivalent gas-fuel powered vehicles are actually available on the market. A higher market potential offers the CNG-pressure vessel introduction in fleets of coaches because of the larger spare room available for several vessels as well as larger vessels, e.g. tank mounting on the roof. Objective is the increase of the specific storage capacity due to full composite pressure vessels. Therefore, equal to steel vessels, detailed analysis of the tank design has to be performed considering multi-axial stresses resulting from bending effects in the transition region of cylinder and head. Especially the specific failure modes of wrapped full composite vessels and their difficult evaluation has to be considered in the analysis.

In general, pressure vessels designed in accordance with standards, are designed by rules and do not require a detailed evaluation of all stresses. Up to now, standards with detailed design rules for the lay-up of fibre reinforced plastic pressure vessels exist only for low pressure application, e.g. ASME Code, Section X [1].

It is recognised that high localised and secondary bending stresses may exist but are allowed for by use of a higher safety factor and design rules for details. It is general practice when doing more detailed stress analysis to apply higher allowable stresses. In fact, the detailed evaluation of stresses permits substituting knowledge of localised stresses and the use of higher allowables in place of the larger factor of safety (see table 1). This higher safety factor really reflects lack of knowledge about actual stress, especially in the head regions of a pressure vessel. According to ISO/DIS 11439, the design analysis of a full composite pressure vessels (type CNG-4) is based on a linear elastic material model. The dimensioning against burst failure is sufficient [2]. The designer must familiarise himself with the various types of stress and loadings in order to accurately apply the results of analysis. He must also consider some adequate stress or failure theory in order to combine stresses and set allowable limits. Furthermore, he has to compare and interpret stress values, and to define how the stresses in a component react and contribute to the strength of that part. The interpretation is not trivial for isotropic materials, for orthotropic materials, especially fibre reinforced plastics with a multitude of failure modes, e.g. rupture of fibre and/or matrix, interply and intraply failure, it is even hard to find an appropriate failure

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criteria. A practice oriented detailed analytical multi-axial stress analysis of FRP pressure vessels in the head section is nearly impossible.

Standard	GFRP	AFRP	CFRP
Aerospace (USA)	3.0 – 4.0	1.5 – 3.0	1.5 – 2.0
ANSI/AGA NGV2	3.5	3.0	2.25
ASME, Section X	5.0	5.0	5.0
prEN 12245 (for CNG-3, CNG-4)	2.0 x p_h	2.0 x p_h	2.0 x p_h
prEN 12257 (for CNG-2)	1.67 x p_h	1.67 x p_h	1.67 x p_h
ISO/DIS 11439			
CNG-2	2.75	2.35	2.35
CNG-3	3.65	3.10	2.35
CNG-4	3.65	3.10	2.35

CNG-1 full metal pressure vessel
CNG-2 latitudinal FRP reinforced pressure vessel
CNG-3 fully overwrapped metal liner
CNG-4 full plastic pressure vessel
p_h proof pressure

Table 1: Burst safety factors according to different standards [2, 3, 4, 5]

2 Pre-dimensioning with analytical models

A relatively easy to handle method for the analytical dimensioning of bi-directional loaded fibre reinforced structures represents the netting theory. It is based on the assumption that the single load bearing part is represented by the fibres and no load transfer exists between fibres and matrix (resin system). I.e., no shear and no normal stress components perpendicular to the fibres are taken into account. Furthermore, all fibres are assumed to have equal strains.

A detailed explanation to the netting theory can be found in the paper of Zimmermann [6]. A lot of analytical models base on the netting theory, mostly modified by empirical terms. In consequence of the assumptions, the theory is suitable only in the cylindrical vessel part. A predictive design analysis has to take into account the heads including bending effects in the transition region and the resulting critical loads perpendicular to the fibres. Thus, the analytical dimensioning can only be a pre-dimensioning start value for the finite element analysis.

Based on the equilibrium condition at a shell of revolution in a bi-directional $^{(2)}$ ply and the superposition of the unidirectional layers $+\omega$ and $-\omega$ of equal thickness and hence the cessation of the shear part in addition to the known internal pressure quota in meridional m and latitudinal direction θ, the netting theory formulas in the cylinder for geodesic fibre trajectory are

$$N_m^{(2)} = \sum_{j=1}^{n} t_{f,j}^{(2)}\, X_j^t \cos^2 \omega_j = \frac{p r_E}{2} \tag{1}$$

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$$N_\theta^{(2)} = \sum_{j=1}^{n} t_{f,j}^{(2)} X'_j \sin^2 \omega_j = p r_E \quad (2)$$

Cuntze introduced modifications for the equator radius r_E' in order to use a mid plane in the load bearing ply, the pole radius r_P' for the correction of the pole circle by the half of the fibre bundle width and added a safety factor S. This factor takes into account the discontinuity effects in the transition region and boss effects. Derived from experimental experience, he suggested a safety factor of 1.5 [7]. Other pre-dimensioning analyses use calculated layer strengths X_t, e.g. according to Tsai-Hill [9]. Different lay-ups and effects of the equator-pole-ratios or head geometries (e.g. isotensoidal or spherical contour) cannot be taken into account for a detailed vessel dimensioning.

	NT (general)	NT (Modification of Cuntze)	NT (X_t according to Tsai-Hill)	FE
Substantial formulas	$t_i = \frac{n_{f,i}}{X_{t,i}\,\varphi_i}$	$t_\omega = p_b\,S * \frac{0{,}5\, r_E}{\varphi\, X_{t,f,\omega} \cos^2\omega}$ $t_{90°} = p_b\,S * \frac{r_E(1-0{,}5\tan^2\omega)}{\varphi\, X_{t,f,\omega}}$ $r_E' = r_E + \frac{1}{2}t$, $r_P' = r_P + \frac{1}{2}b$	$X_t = \left[\frac{\cos^4\omega}{X_t^2} + \left(S_{12}^{-2} - X_t^{-2}\right) * \sin^2\omega \cos^2\omega + \frac{\sin^4\omega}{Y_t^2}\right]^{-\frac{1}{2}}$	
t_ω **[mm]**	**2x 1.2**	**4.3**	**11.8**	**6.5**
t_θ **[mm]**	**3.8**	**6.8**	**4.2**	**9.5**
t_{total} **[mm]**	**6.2**	**11.1**	**16.0**	**16.0**

Table 2: Results of the pre-dimensioning analysis with netting theory (NT) in comparison to Finite-Element-dimensioning for the cylindrical part of a full composite pressure vessel (dimensions see table 3)

3 Parametric FE-model

A full parametric model generation module for the dimensioning of pressure vessels especially of full composite vessels made by filament winding has been developed and is actually in the validation phase supported by burst pressure tests. A quarter segment of a vessel is modelled defined by the following parameters: equator and pole diameter, choice between no fitting or two default fittings in the head (the linking in of additional fittings is made easy by prepared interfacial cut outs in the model head), a possible predefined lay-up (e.g. the desired wrapping angles based on process experience or at least a starting lay-up for the optimisation based on the analytical pre-dimensioning) and the definition of the head contour. Furthermore, the model gives the possibility to wrap over the transition region of head and cylinder by hoop layers defined by a hoop layer

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ending parameter. A spherical dome is directly implemented whereas other contours like an isotensoidal shape has to be filled in pointwise, as a consequence of the not closed soluble differential equation that has to be solved separately, e.g. with the Runge-Kutta-Nyström procedure (see eq. 3). Based on the Clairaut condition for geodesic fibre trajectory, each single wrapped layer with its continuous changing thickness and winding angle in the head is calculated and generated (eq. 4 - 5). The orthotropic material modelling is based on laminate properties and has to be defined in all three space directions. A microscopic resolution in fibre and matrix modelling was not distinguished. Experimental data for CFRP (EP/UTS-CF, 60 % p. Vol.) is implemented in the module. The helical properties with a minimum fibre volume content of 50 % were scaled. Special care requires the correct element orientation of the layered volume elements for the CFRP laminate. Corresponding control and orientating algorithms were implemented. The liner (in this case made of HDPE) and all metal parts were modelled with isotropic volume elements except of the screwed connections that were represented by beam elements.

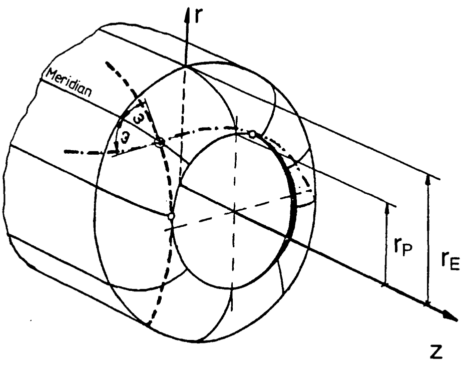

Figure 1: Geodesic fibre trajectory

Differential equation for the meridian of an isotensoidal head with geodesic fibre trajectory:

$$r'' = \frac{1+r'^2}{r}\left(\frac{1}{\left(\frac{r}{r_P}\right)^2} - 2\right) \quad (3)$$

Clairaut condition:

$$r \sin\omega = const. \quad (4)$$

$$\omega_{\min} = \arcsin\frac{r_P}{r_E} \quad (4a)$$

Ply thickness approach [8]:

$$t_f = t_{f,E}\frac{\sqrt{\left(\frac{r_E}{r_P}\right)^2 - 1}}{\sqrt{\left(\frac{r}{r_P}\right)^2 - 1}} \quad (5)$$

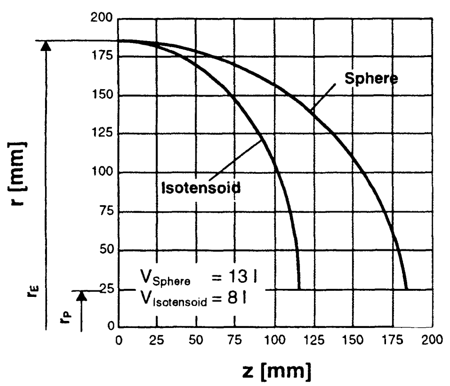

Figure 2: Comparison between spherical and isotensoidal head contour

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4 Influence of the dome geometry

Spherical heads are optimal with respect to strength in full metal pressure vessels. For fully overwrapped metal liners or full plastic pressure vessels isotensoidal heads should be preferred (isotensoid = equal stress in both in-plane main directions). The differential equation (3) includes the Clairaut condition (4) for geodesic fibre trajectory. If the winding pattern fulfils this condition, a non-fibre slippage is guaranteed. The resultant minimum wrapping angle at the vessel equator has an amount of 7.6° in the presented case (see figure 2 for pole and equator radii, eq. (4a)). A minimum wrapping angle of 10° has been fixed taking into account the manufacturing tolerance. If a geodesic fibre trajectory is not possible, e.g. in consequence of not suitable winding machines or different pole diameters, a planar wrapping path with its constant wrapping angle at any point of the vessel could be an alternative. But the theoretical advantage of a constant loaded wrapped body will be lost, wherefore the geodesic fibre trajectory has been favourited.

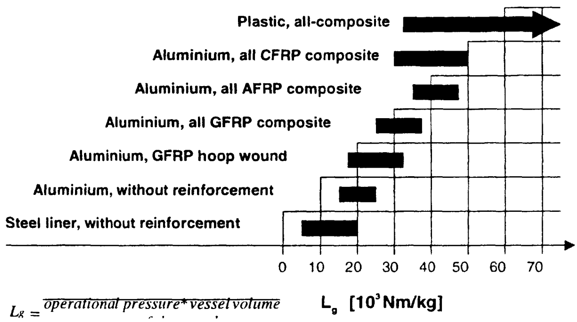

Figure 3: Comparison of different pressure vessel types [9]

Head/vessel type	Sphere-2	Isotensoid-2	Isotensoid-3
Volume V [l]	67	66	65
Vessel length [mm]	880	815	782
Vessel diameter [mm]	452	420	420
Radius equator, radius pole r_E, r_P [mm]	185, 25	185, 25	185, 25
FRP ply thickness t_θ, t_ω [mm]	15, 23	10, 12	10, 12
Mass m [kg]	58 *)	32 *)	26 *)
Material costs (approx.) [DM]	2430	1010	990
Leight weight index $L_g = p_{Service} V/m$ [10^3 Nm/kg]	23,1	41,3	50,8

*) Lay-up after FE-optimisation, liner thickness 4 mm

Table 3: Comparison of cylindrical vessels with spherical and isotensoidal heads after lay-up optimisation

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After the lay-up optimisation of vessels with isotensoidal and spherical heads, described in chapter 5, the cylindrical vessel with isotensoidal head showed significant weight advantages as a result of the distinct thinner load bearing plies (see table 3). A vessel benchmarking based on the product of specific vessel volume and service pressure leads to an additional increase of 120 % for the isotensoidal type after head/fitting design optimisation. A pleasant secondary advantage of the isotensoidal head shows figure 2. It's flat shape leads to a lower head volume and allows an increase of the cylinder length equal to an increase of total vessel volume in space limited mounting, e.g. in a NGV vehicle. Using the presented design module, weight savings in comparison to conventional GFRP hoop reinforced aluminium vessels are expected to be 50 % and to full steel vessels 75 % (see figure 3).

5 Lay-up optimisation

The intended service pressure for the vessel was 20 MPa. Based on the technical requirements the vessel has to withstand a proof pressure p_h of 30 MPa (1,5 x $p_{Service}$) and should not break down below 60 MPa (2 x p_h). The vessel should burst in three parts at maximum with an fracture initiation in the cylindrical part [10, 11]. The presented design studies were dimensioned in respect of burst pressure with a conservative point of view for the compensation of manufacturing inaccuracies. The applied failure criteria was the three dimensional Tsai-Wu criteria that supports this conservative perspective as a consequence of its first-ply failure approach. The tensor polynom of the criteria, consisting of the occurring tension components and strength coefficients, estimates the failure state in the layered composite elements of the FE model. Failure is assumed if the polynom value ζ is greater equal one. Different studies for the last-ply failure dimensioning based on different degradation models and their predicting quality for the UD-FRP vessel dimensioning have been performed and are actually in the validation phase supported by burst pressure tests.

Accordingly to table 2, the vessel dimensioning with netting theory without correction factors was proved as unusable. The FE analysis showed that, according to the assumptions, the fibres were stressed equally and below the strength values. However, the laminate fails perpendicular to the fibre direction. The FE-module-dimensioning requires a ply thickness of 9.5 mm in hoop and 6.5 mm in helical direction which exceeds Cuntze's model by 35 %. A sphere head integration in the analysis leads to a ply thickness increase up to 15 mm (hoop) and 45 mm (helical) in consequence of the additional bending moments in the transition region. The helical ply amount can be decreased up to 23 mm if the lay-up is optimised to an hoop ply ending in the head region wrapping the hoop majority directly on the liner surface (inner hoop ply) and only a small hoop quota at the outside compacting the total laminate.

Different lay-ups concerning stacking, thickness, wrapping angle and ending of hoop plies were analysed with the optimisation module [12, 13]. It was shown that a bone-shaped liner with isotensoidal heads, filled up with the majority of

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the hoop layers to a normal cylinder vessel geometry and the rest of the hoop layers outside compacting the helical laminate to inhibit voids, is the theoretically best design for a full composite cylindrical pressure vessel. This stacking inhibited additional bending moments in the helical layers as they are found in alternating hoop-helical stacking. The bone-shaped liner geometry leads to high tooling costs and complexity in the rotational sintering process or to a welding process of injection moulded heads with a pulltruted cylinder. The influence of the hoop layers beyond the transition region of the isotensoid are of low importance for the vessel burst performance in comparison to vessels with spherical heads. The influence of the geometry and structural stiffness of the fitting in a isotensoid is much more significant. Supposing favourable fitting design, similar helical layer thickness as in the undisturbed cylindrical part is feasible (see figure 4). This point is pleasant in consideration of the lack of practicable analytical models for the head analysis. Owing the more intensive radius change r(z) in the transition region of the isotensoid in comparison to the sphere and the resulting transverse stresses in the hoop layers, the hoop layer ending in the cylinder has to be preferred.

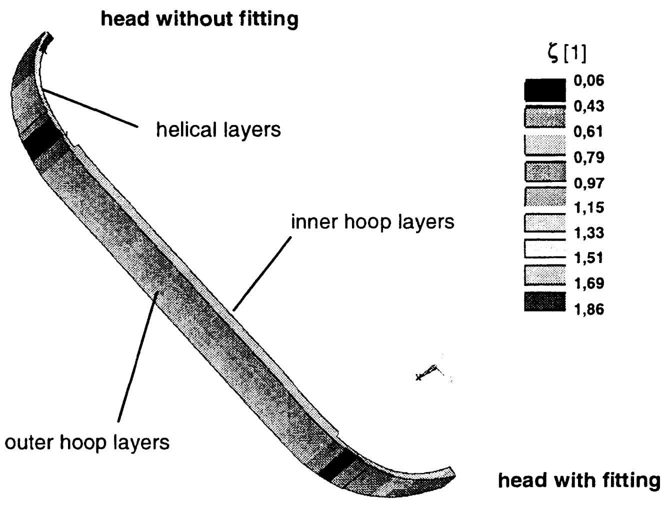

Figure 4: Burst pressure stress conditions in a cylindrical CFRP all-composite pressure vessel with isotensoidal dome

Based on practical experience in filament-winding of CNG-4 pressure vessels the final lay-up stacking consisted of 17°, 30°, 54°, 70° and 88° angle plies. A lay-up with only two bi-directional plies is not feasible in the polar region as a result of the increasing fibre coverage resulting in an dramatically increased ply thickness with inter-ply hollows in the pole region. These harmful hollows must be avoided by subdividing the helical ply. Figure 5 compares different hoop ply

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endings (ω = 88°) in a cylindrical vessel with isotensoidal heads. Beside the increasing fibre slippage tendency of hoop layers in the head, their transverse stress component increases dramatically and leads to failure.

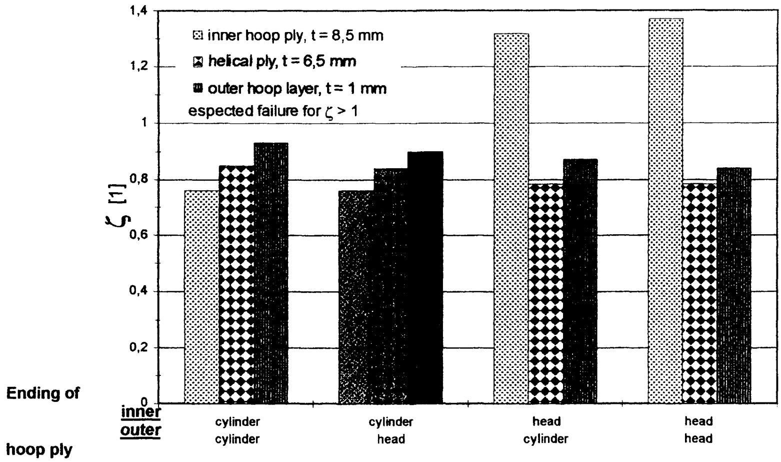

Figure 5: FE optimisation of lay-up, comparison of different hoop ply endings in a cylindrical vessel with isotensoidal heads

6 Conclusions

A full parametric analysis module has been developed based on the Finite Element Method. Pressure vessels, especially full fibre reinforced plastic vessels with plastic liner can be dimensioned with respect of the three dimensional stress state. Different head geometries were optimised with regard to weight and material saturation trough lay-up variation.

Using all-composite pressure vessels with isotensoidal heads especially with propitious fittings in both vessel heads ply thickness reduction of 42 % in comparison to spherical headed vessels can be achieved. Different impact on the burst performance has the hoop layer ending in the two head types. With spherical heads an ending in the heads has to be arranged decreasing dramatically the bending loads in the helical plies (i.e. possible thickness reduction of 49 % in comparison to lay-ups with hoop layer ending in the cylinder). Lay-ups for isotensoidal vessel types should prefer an hoop layer ending in the cylinder. Beside the generally slip-tendency, the transverse stresses in the hoop layers increase with increasing end shifting into the head because of the rapidly decreasing isotensoidal radius. The theoretically best lay-up represents the concentration of the hoop layer majority en block as a first ply on the liner. The surface of the hoop ply embedded in a necking of the plastic liner (bone-shaped liner) should describe an isotensoidal contour, therewith the helical plies above don't meet additional bending loads. Because of this reason an alternating lay-up of helical and hoop layers should be abandoned most possibly or at least only slight inter hoop plies should be provided.

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