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Finite element analysis of composite high-pressure hydrogen storage vessels

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Abstract
Composite pressure vessels (CPV) are used for large commercial and industrial applications such as softening, filtration and storage. It is expected that composite high-pressure hydrogen storage vessels will be widely used in hydrogen fueled vehicles. Progressive failure properties, the burst pressure and fatigue lifetime should be taken into account in design of composite pressure vessels. In this study, the design of composite hydrogen storage vessels based on "unit load method" along with complete structural analysis and evaluation of fatigue lifetime are performed using finite element commercial code ABAQUS. "Unit load method" covers many of the weaknesses of traditional methods and carries out more detailed design using many factors that composite materials provide for designer. The obtained FEM results are compared with experimental ones and a good agreement between them is noticed. By increasing the vessel pressure acceptable and appropriate behavior is observed in strain-pressure curves. As a result, the proposed method can be implemented with acceptable confidence and less cost for comprehensive design of composite high-pressure hydrogen storage vessels.

Keywords: composite high-pressure hydrogen storage vessels; unit load method; finite element analysis; fatigue lifetime

1. Introduction
The intemperate use of fossil fuels has led to gradually increasing drastic environmental pollution and energy crisis. Numerous research works have recently been carried out on looking for renewable resources as replacement for conventional fossil fuels. Hydrogen has been recognized as the superior option for the future energy industry because of the characteristics of unlimited supply, zero-emission of green house gases, and high energy efficiency. Hydrogen storage has become one of the predominant technical barriers limiting the widespread use of hydrogen energy. Safe, high-efficiency and economical hydrogen storage technique is a key to ensure favorable run of hydrogen fuel cell vehicles. Among many hydrogen storage patterns including high-pressure gaseous storage, cryogenic liquid storage and chemical hydrogen storage, high-pressure gaseous storage has become the most popular technique [1–4]. The basic requirements for design of storage vessels are safety, reliability and economy. However, the composite pressure vessels may work under the high-pressure and high-temperature environment. Conventional metallic pressure vessels cannot longer be competent for the rigorous need of high strength and stiffness weight ratios. Therefore, the composite filament wound technology was introduced to improve performance of the storage vessels [5]. Generally, the composite materials are used for fabrication of pressure vessels by placing them in different orientations for different layers and in a common orientation within a layer. These layers are stacked in such a way to achieve high stiffness and strength [6]. The design of the composite vessel as a fundamental research work relates the physical and mechanical properties of materials to the geometric specifications [7].

In the literature, numerous research works have been carried out on design and analysis of composite pressure vessels. Nunes et al. [8] investigated the production of large composite pressure vessels. They were made of thermoplastic liner, glass fiber and polymer resin. They showed that there is a good agreement between experimental results and elasto-plastic modeling for mechanical behavior of high density polyethylene liner under internal pressure. Koppert et al. [9] conducted experimental investigation along with finite element modeling on the composite pressure vessels made by dry filament winding method. They represented that the results of finite element model for vessels with one or two layers is consistent with experimental results but there are high errors for vessels with three or four layers. Chang [10] theoretically and experimentally analyzed failure of the first laminate of composite pressure vessels. He studied proximity of the theoretical and experimental analysis through fracture strength of the first laminate on the symmetrical laminate of composite pressure vessels using different materials with different number of layers under uniform internal pressure. The experimental results were compared with theoretical results based on the Hoffman, Hill, and Tsai-Wu maximum stress criterions which accurately predicted the pressure in which the failure of the first layer occurs. Vasiliev et al. [11] studied the filament wound composite pressure vessels that have commercial applications. Su et al. [12] used the nonlinear finite element method to calculate the stresses and the bursting pressure of filament wound solid-rocket motor cases which are a kind of composite pressure vessel. The effects of material performance and geometrical nonlinearity on the relative loading capacity of the dome were studied. Kim et al. [13] presented an optimal design method of filament wound structures under internal pressure. They used the semi-geodesic path algorithm to calculate possible winding patterns taking into account the windability and slippage between the fiber and the mandrel surface. In addition, they performed a finite element analyses using commercial code, ABAQUS, to predict the behavior of filament wound structures. The optimal dome contour was studied in ANSYS with a trial design. Onder et al. [14] studied burst pressure of filament wound composite pressure vessels under alternating pure internal pressure. The study dealt with the influences of temperature and winding angle on filament wound composite pressure vessels. Finite element method and experimental approaches were employed to verify the optimum winding angles. The hygrothermal and other mechanical properties were measured on E-glass–epoxy composite flat layers. Some analytical and experimental solutions were compared with the finite element solutions, in which commercial software ANSYS 10.0 was utilized; close results were obtained between analytical and experimental solutions for some orientations. Liu et al. [7] presented a comprehensive review on recent development of numerical simulation and optimization for designing of composite pressure vessels. The methods on damage modeling for predicting the failure properties and degradation mechanisms of the composite vessel along with research on predicting the burst pressure and lifetime of the composite vessel were reviewed. Zheng et al. [4] firstly reviewed recent progress toward low-cost, large capacity and light-weight on high pressure gaseous hydrogen storage vessels. Then, three important aspects of high pressure gaseous hydrogen safety, i.e., hydrogen embrittlement of metals at room temperature, temperature rise in hydrogen fast filling, and potential risks such as diffusion, deflagration, and detonation after hydrogen leakage were introduced. Liu and Zheng [15-16] established parametric finite element model for the cylinder part using ANSYS finite element code. However, the design platform has not been completely established since ANSYS software does not provide an efficient modeling module for composites until now.

Due to various constraints that exist in testing of composite pressure vessels, finite element analysis (FEA) can be considered as a suitable method for analysis of composite high-pressure hydrogen storage vessels. This paper studies finite element analysis of a fuel-cell vehicle's composite high-pressure hydrogen storage vessel using commercial code ABAQUS. Complete structural analysis is performed and effect of some parameters such as fiber angle is investigated. Moreover, fatigue evaluation of composite high pressure hydrogen storage vessel is performed which concentrates on the fatigue properties of the aluminum liner. The proposed method can be implemented with acceptable safety and low cost for comprehensive design of CHSVs.

2. Unit load method

A new method called unit load method is used in design of composite pressure vessels [17]. The old design method was based on allowable stress which by having ultimate tensile strength and dividing it to safety factor, the design allowable stress was obtained. This method was successfully applied for metallic vessels and it was also attempted to use it for glass fiber composite vessels, but, desired results did not acquired. In addition, all codes based on the design allowable stress consider two basic assumptions: a) The material is isotropic; b) properties are uniform along the material thickness. None of these two conditions are completely true for composite materials with glass fibers. Therefore, the unit load method is applied for designing of composite pressure vessels.

2.1 Safety factor

The design factor *k* is obtained through Eq. (1) as,

$$K = 3K_1K_2K_3K_4K_5 \tag{1}$$

The coefficient "3" is constant and is chosen as initiation factor for designing in order to confidence of reduction of materials strength due to long time loads. K_1 to K_5 factors are chosen based on the manufacturing methods and performance conditions. The design factor K varies between 6 and 40 and its common range is 8 to 15. The limited allowed unit load is given in Eq. (2) as,

$$u_L = \frac{u}{K} \tag{2}$$

where, u denotes the ultimate tensile strength and is obtained form Table 1.

The value of maximum allowable strain is 10% of the strain of pure resin fracture ($0.1\,\varepsilon_{\mathbf{R}}$) or 0.2% of it and usually the lower one is selected. The reason for studying the allowed unit load in terms of strain is the significant differences in stress-strain curves of metals and composites with glass fibers. The allowed unit load is calculated based on strain limit and is obtained through product of unit modulus in allowed strain (Eq. (3)):

$$u_s = X_z e \tag{3}$$

Table 1: Properties of reinforced laminate layers

Type of fiber	Unit ultimate tensile strength	Unit modulus
	N/mm (Per Kg/m^2 glass)	N/mm (Per Kg/m^2 glass)
Chopped srand mat	200	12700
Woven roving	300	16200
Unidirectional filament (fiber direction)	500	28000

The design unit load $\mathbf{u_z}$ is determined for the all types of layers as:

a) If u_s for the all layers is lower than u_L, the value of u_s will be the desired design unit load for all layers. It is evident that the design of layers when u_s is lower than u_L for all layers is limited to the strain.

b) If u_s for the all layers or some of them is higher than u_L, the strain for all layers will be obtained through Eq. (4),

$$e_L = \frac{u_L}{X_z} \tag{4}$$

Considering that the composition of the layers forms the structure, the allowable strain e_d is determined as the lowest value of e_l. The design unit load for the all layers is obtained from Eq. (5):

$$u_z = X_z e_d \tag{5}$$

The design unit loads for all types of layers are determined using the above two conditions. The design is performed based on obtained design unit load for each layer (Eq. (6)) :

$$u_1 m_1 n_1 + u_2 m_2 n_2 + \dots + u_z m_z n_z \geq Q \tag{6}$$

In which, u_z is the design unit load for each layer of type *z*, m_z is fiber weight per unit area in a layer of type *z*, n_z is umber of layers of type *z* and *Q* is the exerted maximum unit load. When the thickness is required, it is obtained from summation of thickness of the layers. The thickness of each layer is obtained through the amount of used glass in it as depicted in Figure 1.

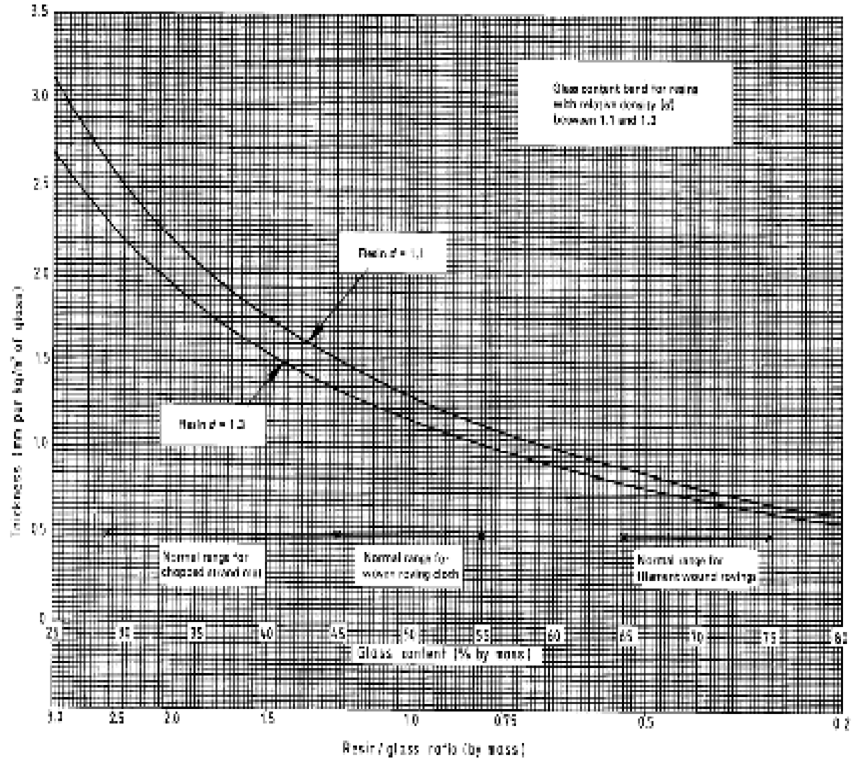


Figure 1: Thickness versus fiber amount

3. Fatigue life evaluation

The fatigue lifetime of high pressure hydrogen storage vessel is an important design parameter in order that they are persistently used in the fuel cell vehicles. In this paper, the composite hydrogen storage vessel is assumed to be composed of an innermost aluminum liner and outer winding E-glass/epoxy composite layers. However, the fatigue lifetime of vessels is mainly controlled by the aluminum liner due to lower fatigue strength of aluminum than that of E-glass/epoxy composites [7–16]. The metal materials and their damage are assumed to be isotropic. Consider a quadrate finite element with three sizes l_1, l_2 and l_3, which includes *n* cracked cells. Each crack's length is *2a* and the maximum crack opening displacement is *2e*. The crack surface is assumed to be elliptic. According to the continuum damage mechanics (CDM) theory, the fatigue lifetime has been obtained as shown below [18],

$$N_f = \frac{k l_1 l_2 n^{l_2/2-1}(s_{eq\max}^{l_2} - s_{eq\min}^{l_2})^{-1}}{2^{1-l_2/2} p D_c l_1(l_2+1) l_3^{l_2/2} e[\frac{2}{3}(1+u)+3(1-2u)(\frac{s_m}{s_{eq}})^2]^{l_2/2}} \tag{7}$$

where, u is the Poisson's ratio, s_{eq} is equivalent stress, s_m is hydrostatic stress, l_1, l_2 are material constants, *k* is crack ratio in an element, *n* is number of cracked cells and *D* is the damage variable.

4. Geometry structure of composite hydrogen storage vessel

Figure 2 shows a composite vessel used in the hydrogen fuel vehicles. Figure 3 demonstrates the basic structure and wound pattern of composite vessel. The vessel includes an inner aluminum layer and several outer composite layers. The aluminum layer prevents the gas leakage and provides a mould for filament wound, and the composite layers are responsible for resisting the internal pressure. Filament winding vessels are designed stiffness enhancement by combining helical winding and hoop winding.

Figure 2: Lightweight composite high-pressure hydrogen storage vessel used in the hydrogen fuel-cell vehicle [7].

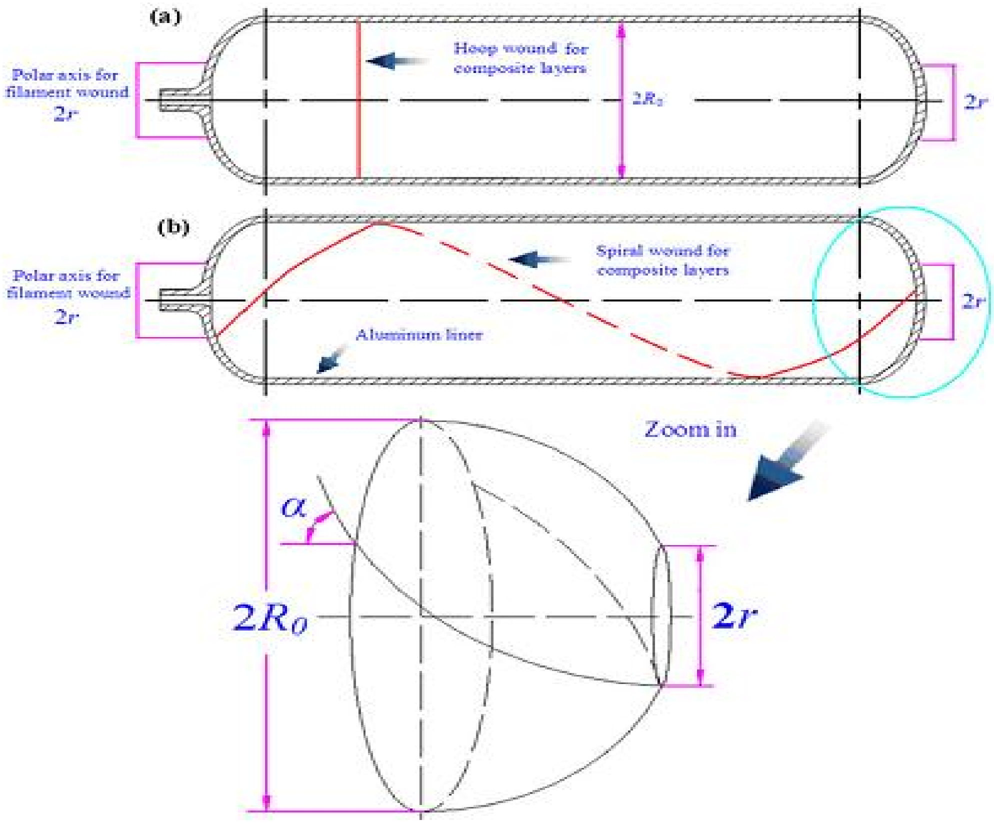


Figure 3: Basic structure and wound pattern of composite vessel: (a) hoop wound; (b) spiral wound [7]

The wound trace for composite layers is a combination of the rotation of aluminum liner and the axial movement of wound machine. The spiral wound angle a_0 at the cylinder is determined by:

$$a_0 = \arcsin(\frac{r}{R_0}) \tag{8}$$

where r is the radius of the polar axis and R_0 denotes the inner radius of the cylinder. The spiral wound at the head follows the geodesic path algorithm, marking the shortest distance between any two points on the head. The spiral wound angle at the head changes from 90° at the polar axis to a_0 at the cylinder :

$$a = \arcsin(\frac{r}{R}) \tag{9}$$

The thickness H of the composite layer at any radius R of the head is calculated by :

$$H = h\sqrt{\frac{(R_0^2 - r^2)}{(R^2 - r^2)}} \tag{10}$$

where h is the wound thickness at the cylinder. From Eqs. (8) – (10), the polar radius r affects the wound angle a and the thickness H at the head [19].

5. Finite element modeling of composite hydrogen storage vessel

Accurate and fast modeling of the composite vessel is an important work in the design, which involves many parameters such as the wound angle and thickness. Parametric modeling by exacting these parameters as the design variables facilitates the strength prediction and optimization for the composite vessels with different sizes. The vessel is considered to be horizontal and the weight of the vessel and its contents are ignored. The mechanical properties of the vessel are given in Table 2. The properties of composite materials are a function of the used fibers and matrix strength. Materials which have high strengths have less toughness (such as glass or ceramics). So, they are not used in industry. The properties of E-glass/Epoxy composite are given in Table 3.

Table 2: Specifications of the composite hydrogen storage vessel

Diameter	380 mm
length	880 mm
thickness	42.25 mm
Layers orientation	$\pm(54.7°)_{40}$
Fiber volume percent	75%
Glass in the thickness	0.65 mm
Glass mass	65 $\frac{Kg}{m^2}$
Performance pressure	2.5 Mpa
Internal volume	80 Lit

Table 3: Mechanical properties of materials

	$E_l(GPa)$	$E_T(GPa)$	G_1(GPa)	$G_T(GPa)$	u_L	u_T
E-glass/Epoxy	56.63	15.22	5.5	5.5	0.29	0.38
Al	70	70	26	26	0.3	0.3

In the model analyzed in ABAQUS, the composite vessel is under uniform pressure and is designed using unit load method. The vessel is fully modeled using the symmetry and to reduce the volume of computations, octant of the vessel is modeled. The vessel is constrained in longitudinal and peripheral directions at transverse and longitudinal cuttings, respectively and the internal pressure is applied according to the perpendicular vector to the internal area of the vessel. The mesh model of the composite vessel is illustrated in Figure 4.

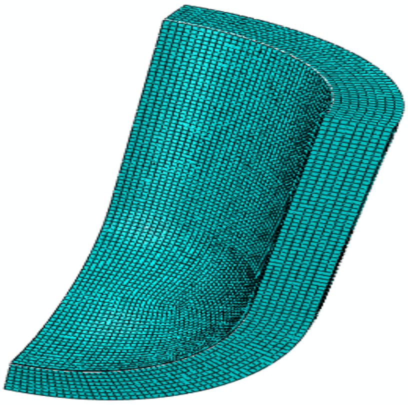
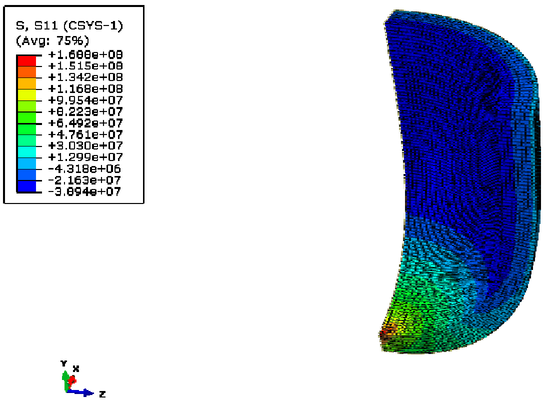


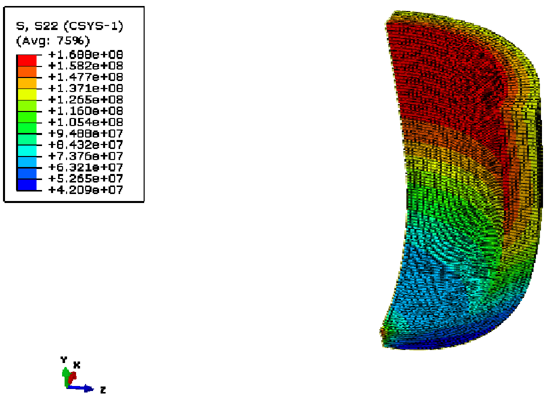
Figure 4: Finite element model of composite vessel

6. Results and discussion

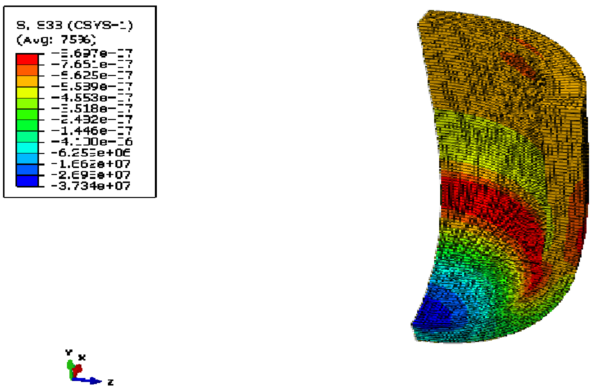
The FEA was conducted for two cases including composite vessel without and with aluminum liner. Figures 5 and 6 show the contours of radial, tangential stress and stress along the main axis of element for composite vessel without and with aluminum liner, respectively. It is evident that the stress magnitude is lower in the case of composite vessel with aluminum liner. Figures 7 and 8 depict the contours of radial, tangential strain and strain along the main axis of element for composite vessel without and with aluminum liner, respectively.



(a)



(b)



(c)

Figure 5: Stress contours without aluminum liner: a) radial direction; b) tangential direction; c) along the main axis element

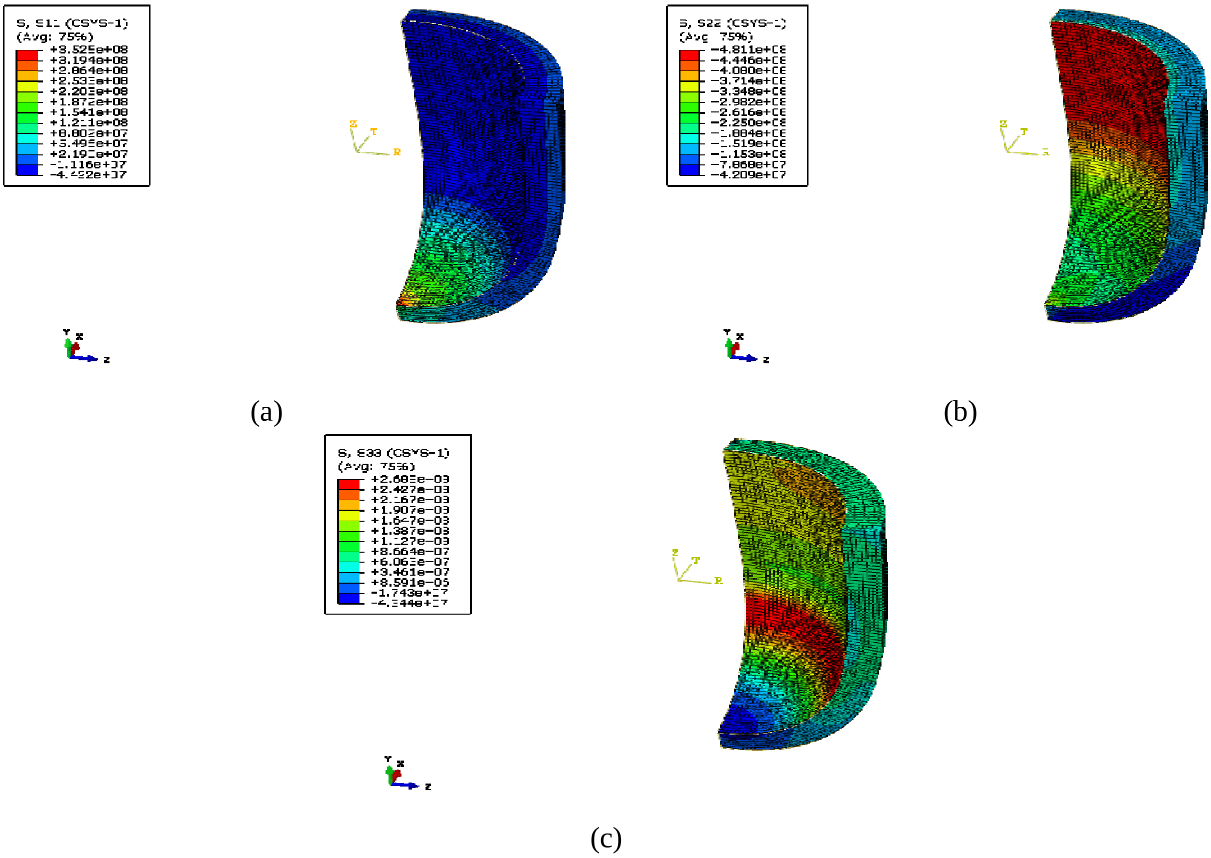


Figure 6: Stress contours with aluminum liner: a) radial direction; b) tangential direction; c) along the main axis element

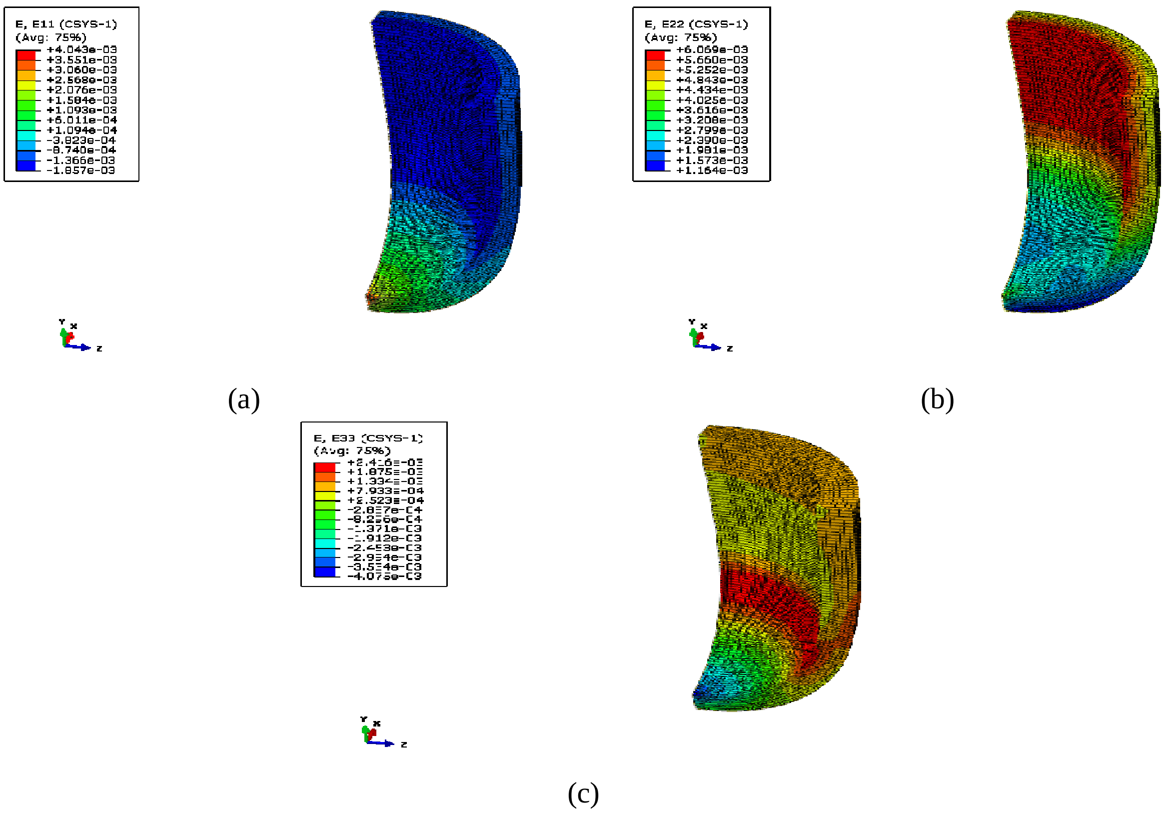
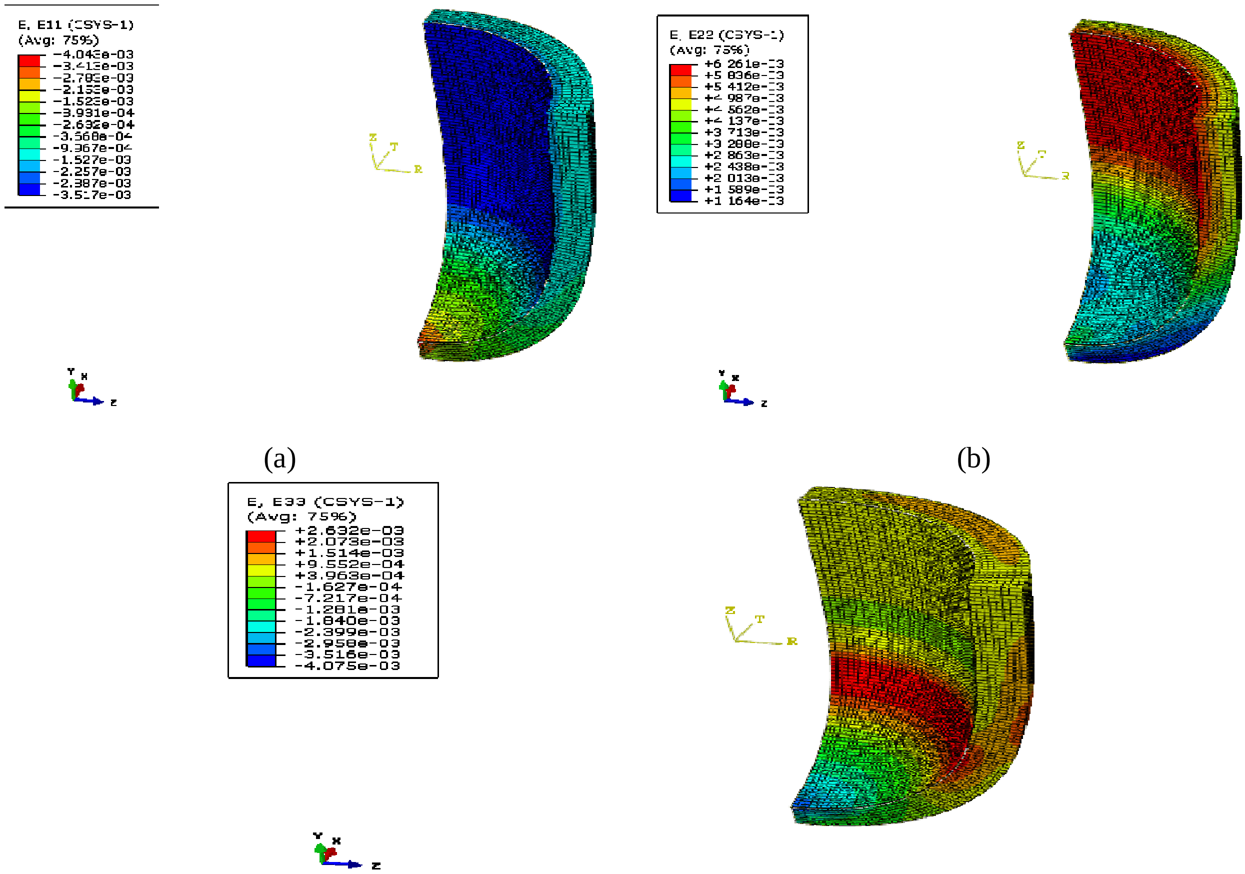


Figure 7: Strain contours without aluminum liner: a) radial direction; b) tangential direction; c) along the main axis element

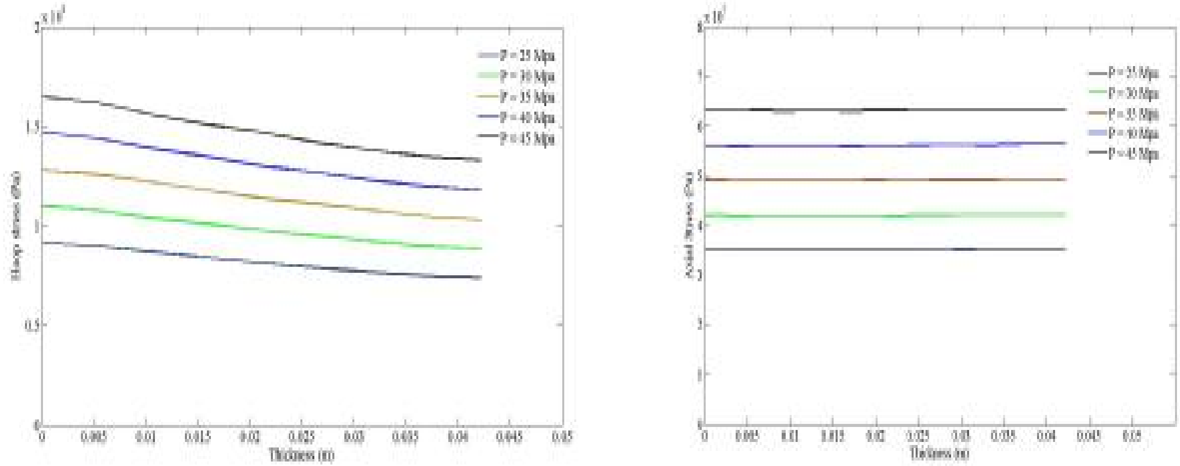


(a) (b)

(c)

Figure 8: Strain contours with aluminum liner: a) radial direction; b) tangential direction; c) along the main axis element

Layers of the vessel are under a suitable portion of the load regarding their location due to the relatively complete theoretical design and selection of optimum fiber angle and as moving forward toward the outer layers, the layers will be under lower uniform stress amount in a way that realize the stressful condition in a normal vessel. This point can be well understood through Figure 9 which depicts the reduction of stress level along the vessel thickness. The same trend can be observed for strain variation along the thickness of the vessel in Figure 10.



(a) (b)

Figure 9: Effect of vessel thickness and pressure on: a) hoop stress; b) axial stress

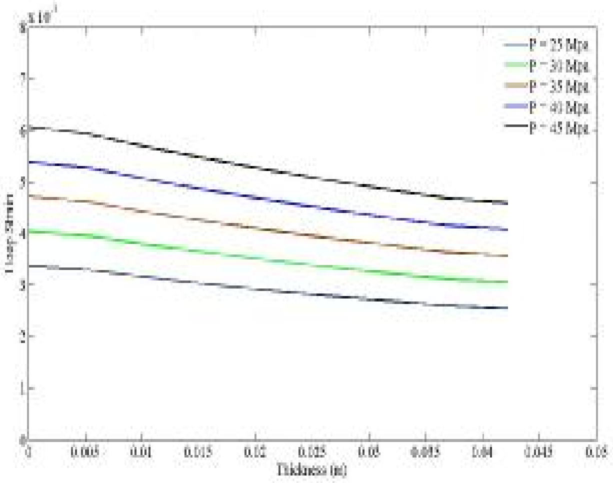
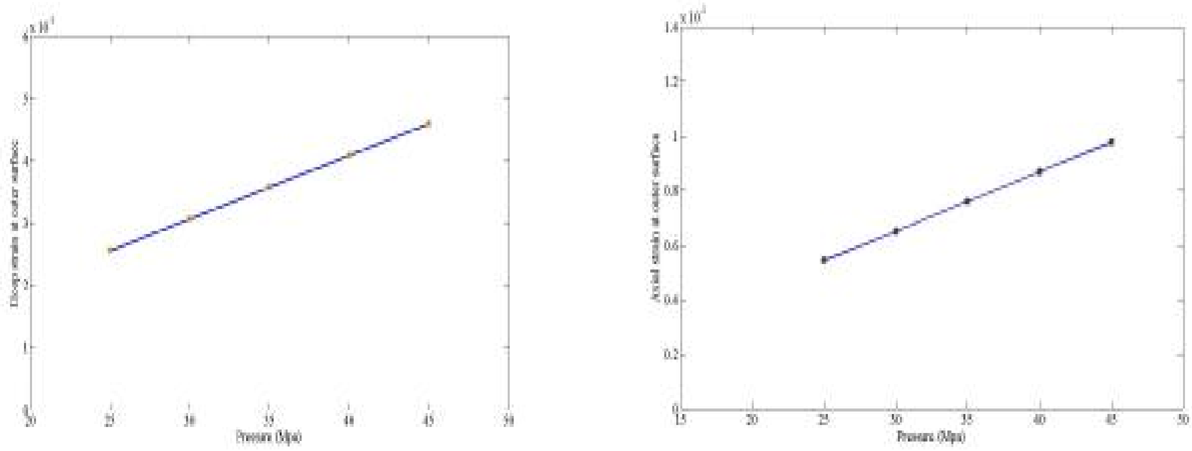


Figure 10: Effect of vessel thickness and pressure on hoop strain

It is obvious that as moving forward from the middle of vessel toward the head, the value of stress is reduced and in the intersection of the head to the cylinder stress is increased due to the geometric deformation of vessel. The strain-pressure curves are plotted in Figure 11. They can be utilized for prediction of vessel behavior at different pressures with acceptable accuracy.



(a) (b)

Figure 11: Strain-pressure curve: a) Hoop strain; b) axial strain

Since, unit load method have been used for wide range of fiber angle, introducing another method is needed to optimize this angle. In previous model, the fiber angle of $\pm 57.4°$ was chosen, now other models with same geometric specifications and layers number but different fiber angle are analyzed and the obtained results are compared with each other. It is clear that as the fiber angle is lower, the vessel is less flexible against the longitudinal loads, because the fibers resists with more intensity in the head against the axial displacements. However, they show less resistance in the radial direction. The fibers are not placed in the lateral loads direction due to their specific orientation (Figure 12).

The fatigue lifetime of vessel is evaluated using the finite element analysis. The constants in Eq. (7) are given in Table 4. Table 5 lists five groups of finite element sizes in order to explore the mesh sensitivity and predict fatigue lifetimes of composite storage vessel. Results indicate that an appropriate selection of mesh size configuration is crucial for accurate prediction of the fatigue lifetime. Figure 13 depicts the calculated S–N curves different mesh sizes.

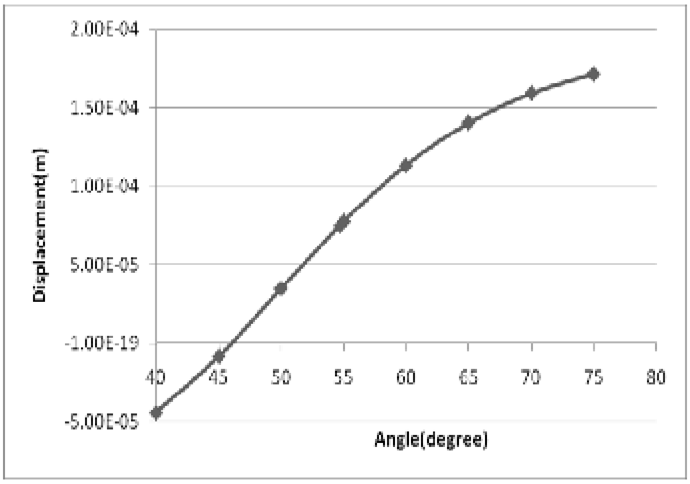


Figure 12: Displacement versus angle change (in longitudinal direction)

Table 4: Constants used in Eq. (7) for fatigue lifetime calculations [20]

material constants	$l_1 = 1.22E-9$ $l_1 = 4.5$
maximum crack opening displacement (2e)	$3mm$
crack ratio in an element (k)	0.8
number of cracked cells	1000

Table 5: Fatigue lifetime for different mesh sizes

Number	l_1 (mm)	l_2 (mm)	l_3 (mm)	fatigue lifetime
1	0.5	0.5	0.5	17801
2	1	0.5	0.5	4994
3	0.8	1	0.8	15774
4	1	1	1	22480
5	3	4	5	3245

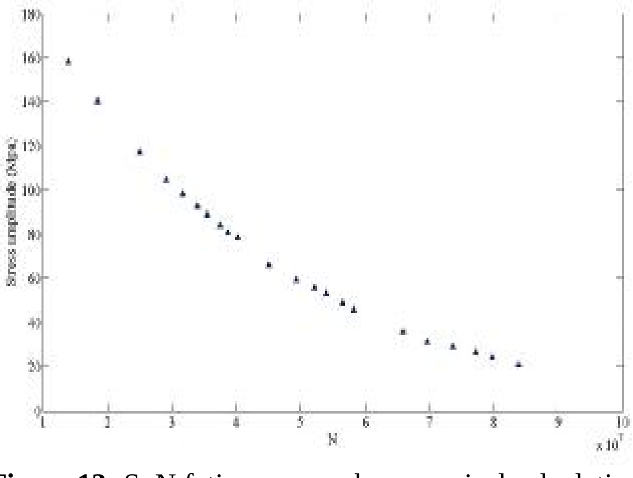


Figure 13: S–N fatigue curves by numerical calculations

Conclusions

In this paper, the Finite element analysis of a composite hydrogen storage vessel based on "unit load method" along with complete structural analysis and evaluation of fatigue lifetime were conducted using ABAQUS package. The numerical simulations were performed on a fuel cell vehicle's composite high-pressure hydrogen storage vessel. The following results were obtained:

- The vessel is modeled using unit load method under various internal pressures. By increasing the vessel pressure acceptable and appropriate behavior was observed in strain-pressure curves. This indicates high safety of this method in the design of composite storage vessels.
- The aluminum liner plays a fundamental role in design and performance of composite high-pressure hydrogen storage vessels.
- The effects of fiber angle in the vessel was undeniable and particularly if it is performed based on the theoretical model (unit load method), accurate and appropriate response was not received. Therefore, another method is needed for selection of the appropriate fiber angle regarding the geometry and loading of vessel.
- The results showed that the fatigue lifetime of vessel depends on the finite element mesh size, crack density and ratio in an element, cyclic loading amplitude and stress status at the liner.

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