

Advancements and Opportunities for On-Board 700 Bar Compressed Hydrogen Tanks in the Progression Towards the Commercialization of Fuel Cell Vehicles

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ABSTRACT

Fuel cell vehicles are entering the automotive market with significant potential benefits to reduce harmful greenhouse emissions, facilitate energy security, and increase vehicle efficiency while providing customer expected driving range and fill times when compared to conventional vehicles. One of the challenges for successful commercialization of fuel cell vehicles is transitioning the on-board fuel system from liquid gasoline to compressed hydrogen gas. Storing high pressurized hydrogen requires a specialized structural pressure vessel, significantly different in function, size, and construction from a gasoline container. In comparison to a gasoline tank at near ambient pressures, OEMs have aligned to a nominal working pressure of 700 bar for hydrogen tanks in order to achieve the customer expected driving range of 300 miles. Beyond the need to contain pressure, the hydrogen tanks also differ from gasoline fuel tanks because of the additional vehicle space needed due to the lower hydrogen energy volumetric density even with the highly efficient fuel cell (four times the external volume of a gasoline tank including the fuel cell efficiency benefit). The main difference and challenge of hydrogen tanks is the construction and design that depends on a high utilization of carbon fiber in order to reduce the weight of the pressure vessel although substantially increasing the cost. In 2012, the U.S. Department of Energy (DOE), Office of Fuel Cell Technologies recognized these challenges and initiated a project to research enhance materials and design parameters to reduce the cost of hydrogen storage tanks. The project was led by Pacific Northwest National Lab (PNNL) and involved several other organizations in the value chain of hydrogen tank development: AOC, Ford Motor Company, Hexagon Lincoln, and Toray CFA. The project took a holistic approach to improving performance by investigating: (1) composite matrix resin alternatives including adding nano-reinforcing particles and fiber-matrix sizing for improved adhesion, (2) carbon fiber alternatives, (3) tank design alternatives using hybrid fiber layups, and (4) opportunities with cold gas operating conditions to maintain the hydrogen density while reducing the tank composite utilization. In each of these areas, the project successfully identified the potential benefits: (1) demonstrated new resin with 50% cost reduction at equivalent or better performance than traditional epoxy, (2) identified 4% to 12% improvement with fiber alternatives, (3) developed validated tank design models with improved failure prediction capability, and (4) confirmed value and system level viability of cold gas storage with a combined 22% cost reduction opportunity. This paper examines these modifications and considers the outlook for on-board 700 bar compressed hydrogen tank systems to achieve the commercialization goals for fuel cell vehicles.

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INTRODUCTION

Fuel cell vehicles have environmental, energy security, and efficiency benefits. Compressed hydrogen tanks are a key enabler for the commercialization of fuel cell vehicles. Hydrogen tanks are certainly unique from gasoline fuel containers in terms of function (pressure vessel), size (lower energy density), and construction (carbon fiber). In the late 1990s, the industry first developed hydrogen tanks based on traditional type 3 and 4 compressed natural gas (CNG) tank constructions at 250 bar. A type 3 tank is defined as a pressure vessel with a metal liner reinforced by a fully wrapped composite while a type 4 tank has a polymer liner reinforced by a fully wrapped composite. These tanks are preferred for hydrogen due to their weight and fatigue resistance over a type 1 (all metal) or type 2 (hoop wrapped only) tank. These type 3 and 4 tanks typically utilize carbon fiber for the composite wrap that is filament wound around the liner. As fuel cell vehicles developed a need for greater driving range, the OEMs increased the hydrogen storage pressure to 350 bar and then to 700 bar, which is currently the industry norm. As the pressure increased, the hydrogen tanks required additional carbon fiber for the thicker tank walls. In general, the hydrogen tanks still continued to be manufactured using the materials and approaches from the CNG tank technologies. The result was hydrogen tanks were substantially higher in cost than conventional tanks and CNG tanks. The high cost of hydrogen tanks was a notable barrier, second only to the fuel cell system as the most costly system on a fuel cell vehicle. In recent years, there have been several advancements with on-board 700 bar compressed hydrogen tanks including efforts funded by the U.S. DOE, Office of Fuel Cell Technologies to develop novel approaches to hydrogen tank designs. This paper summarizes a DOE project led by Pacific Northwest National Lab (PNNL) along with several key stakeholders in the tank development process: AOC, Ford Motor Company, Hexagon Lincoln, and Toray CFA. The project evaluated multiple hydrogen technology opportunities including: (1) composite resin alternatives, (2) carbon fiber alternatives, (3) tank design alternatives, and (4) opportunities with cold gas operation. The ultimate goal was focused on the DOE hydrogen storage system performance targets for light-duty vehicles. Table 1 provides a subset of the complete list of targets related to gravimetric, volumetric, and cost. The targets apply to system-level properties and were developed by the DOE and OEMs within the U.S. DRIVE partnership with a commercialization focus in comparison to conventional vehicles.

Table 1. DOE U.S. DRIVE Hydrogen Storage Targets

STORAGE PARAMETER	UNITS	2020	ULTIMATE
System Gravimetric Capacity Usable specific-energy from H ₂ (useable H ₂ / max system mass)	kWh/kg (kg H ₂ /kg system)	1.8 (0.055)	2.5 (0.075)
System Volumetric Capacity Usable energy density from H ₂ (useable H ₂ / max system volume)	kWh/L (kg H ₂ /L system)	1.3 (0.040)	2.3 (0.070)
Storage System cost Total system cost per H ₂ stored	\$/kWh net (\$/kg H ₂)	10 (333)	8 (266)

COMPRESSED HYDROGEN TANK BASELINE ANALYSIS

In order to pursue advancements for any technology, it is important to first establish the baseline. At the start of the PNNL project in 2012, a baseline was not established by the DOE for the basic attributes of a 700 bar compressed hydrogen tank such as the material utilization, structure, weight, and cost. The tank chosen as the basis for cost comparisons in this study is a Type 4, 700 bar tank with 5.6 kg usable hydrogen at 15° C, which is estimated to provide over 300 mile driving range for a fuel cell vehicle (location illustrated in Figure 1).



Figure 1. The baseline tank chosen for cost comparisons shown installed in a fuel cell vehicle platform (courtesy of Ford Motor Company).

Several approaches were utilized to evaluate and determine the compressed hydrogen tank baseline. The following sections describe each approach and then compare their outcomes to the DOE baseline reference information.

Hydrogen Tank Regression Model

The first approach developed a model based on the physical attributes of a population of as-manufactured Type 4 hydrogen storage tanks. Properties such as the known composite strength and density plus the measured wall thicknesses in the cylinder and dome sections were scaled based on the rated pressure to other operating pressures and tank sizes. Regression analysis was used to develop equations that relate the actual thicknesses of the certified tanks to the thin wall pressure vessel equations for hoop and axial stress. “Thickness factors” were defined for the composite cylinder and dome sections and also for the HDPE liner thickness in the dome. A dome “geometry factor” was also fit to accurately calculate the composite mass in the tank domes as well as the inner tank volume. These parameters were fit for a population of tanks with diameters ranging from about 235 mm to 520 mm and lengths ranging from 650 mm to 3200 mm.

Checking the model by back-calculating the composite mass of other existing tanks shows that the calculated and actual tank attributes differ within +/-5% as shown in Figure 2. Therefore, this method provides a highly accurate approximation, even better than classical laminate theory, because the thick-wall and modulus ratio effects that are known to increase the stresses on the inside surface of high pressure composite tanks are accounted for in the empirical thickness and geometry factors. In addition, the added composite layers required to satisfy fatigue, durability, and ballistic impact requirements are also encompassed in the empirical thickness factors.

carbon fiber is the most significant contributor to tank cost, accurately predicting the composite mass is the most important element to accurately estimating the tank cost.

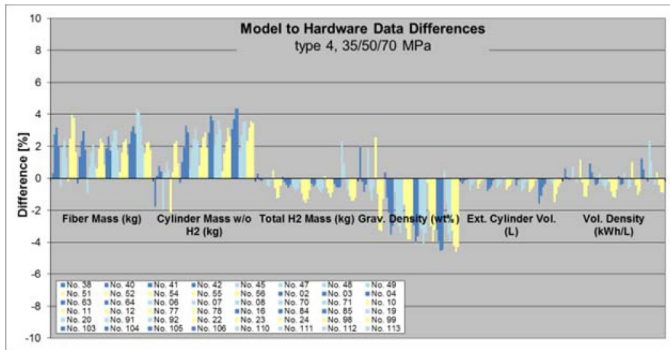


Figure 2. Estimator model to industry performance tank data differences.

Table 2. Key common assumptions for the Hydrogen Tank Baseline Estimate

Tank Parameters	Units	Value	Description
Usable H ₂ Mass	kg	5.6	baseline assumption
Total Stored H ₂ Mass	kg	5.9	baseline assumption*
Nominal Pressure	bar	700	baseline assumption
Nominal Temperature	K	288	baseline assumption
Hydrogen Density	g/L	40	baseline assumption
Internal Diameter	mm	391	baseline assumption
Inside L/D ratio	#	3.3	baseline assumption
Gas volume	L	147.3	baseline assumption*
Design Factors			
Safety Factor	x NWP	2.25	Carbon fiber standard
Manufacturing Variability	%	14	baseline assumption
Tank Materials			
Fiber		T700S	baseline assumption
Fiber Density	g/cc	1.8	baseline assumption
Fiber Material Cost	\$ /kg	33	up to 30,000 tanks/yr
		30.8	up to 80,000 tanks/yr
		29.48	up to 100,000 tanks/yr
		28.60	up to 500,000 tanks/yr
Resin		Epoxy	baseline assumption
Resin Density	g/cc	1.25	baseline assumption
Resin Material Cost	\$ /kg	8.25**	baseline assumption
Liner		HDPE	baseline assumption
Liner Density	g/cc	0.95	baseline assumption
Liner Material Cost	\$ /kg	2.09	baseline assumption
Boss		Al	baseline assumption
Boss Density	g/cc	2.7	baseline assumption
Boss Material Cost	\$ /kg	4.75	baseline assumption

*unusable storage definition may vary with system performance (worst case assumed)

** includes 1.25x of raw material cost for curing agent and scrap run-off

Table 2 lists the important common parameter assumptions that are used for the baseline tank calculations in all of the models including the DOE reference case. Each of these can be changed as improved information is available. The material costs for the composite fiber, resin, liner, and boss materials were established based on industry data at the time of the initial model formulation. Because the tank cost estimates are sensitive to the material prices, especially the carbon fiber, these assumptions were held constant and aligned with the DOE to ensure the differences in tank estimates were due to the modeling rather than the assumptions. Table 3 provides the results of

the estimated weight of the baseline tank using the regression model (Table 3 is shown later in the report as a summary of the various baseline models in comparison to the DOE reference).

Manufacturing process costs are estimated in a separate estimator tool based on an activity based cost assessment of each process step shown in Figure 3. In terms of cost contributors, the most significant step is the filament winding of the carbon fiber composite on the liner. This step involves high cost for capital and limited cycle time due to restrictions on the rotational speed of the liner while the fiber pattern is applied with wet resin. It is important that the fiber is applied in a manner that resin maintains appropriate adhesion with the fiber plies and is not centrifugally depleted during the winding rotation. The manufacturing cost analysis assumed a 4 spinal winding machine which is probably the limit for this step to maximize the capital utilization and reduce the cycle time per tank. The process step with the longest cycle time (~6 to 7 hours) is high temperature curing of the composite matrix although cost per tank is reduced since a large batch of tanks can be processed simultaneously.

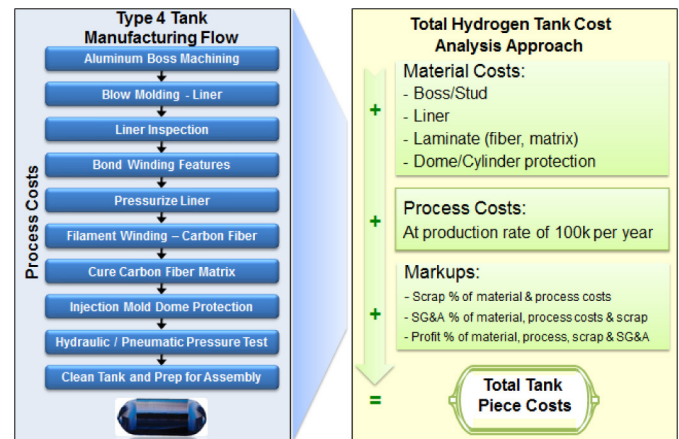


Figure 3. Tank cost assessment approach based on material, process and typical markup costs.

Hydraulic and pneumatic pressure test is another key cost factor in the tank manufacturing cost. This step is needed for quality control and leakage testing of the tanks. Similar to the winding step, the pressure test also includes high capital and is limited in the cycle time and number of tanks that can be processed. The manufacturing process cost for hydrogen tanks at high volumes (100k to 500k annual units) were projected in the range of \$120 to \$220 depending on the range of the assumptions. Figure 8 provides the results of the estimated cost of the baseline tank using the regression model.

Hydrogen Tank Composite Netting Model

An alternative tank estimating model was developed to compare with the regression model. The model uses composite netting analysis to estimate the required wall thickness and mass of the filament wound carbon fiber composite tank. Netting analysis assumes all fibers are loaded in tension, the resin does not contribute to strength, shear stresses are small, and all stresses are in the plane of the composite.

Enhancements were added to the standard netting analysis method to account for the stress concentrating effects of the thick-walled geometry plus through-thickness compression of the composite. The

combination of high operating pressure (up to 700 bar) and low through-thickness modulus compared to in-plane composite modulus increase the hoop stress at the inside of the tank wall. Based on a literature review, a helical-to-hoop stress ratio of 0.5 is typically applied when using netting analysis to design manufactured cylinders to prevent failure of the domes by boss blowout. It should be noted that this is a conservative value, and accounts for some additional weight from transition circuits. This conservative value also gives some consideration for impact resistance, e.g. drop test and gunfire test. Applying the 0.5 stress ratio in conjunction with the above stress concentrations gave calculated tank masses that compare within 5% of actual manufactured tanks.

The netting analysis by Roylance [1] for hoop plus two helical ply angles was implemented to estimate the required wall thickness of the composite pressure vessel (Figure 4). The solution treats a closed ended vessel with internal pressure, P , and radius, R , with axial direction, ϕ , and hoop direction, θ . Strains in the fiber direction, α , are related to the axial and hoop strains through the elastic transformation equation:

$$\varepsilon_{\alpha} = \varepsilon_{\phi} \cos^2 \alpha + \varepsilon_{\theta} \sin^2 \alpha + \gamma_{\phi\theta} \sin \alpha \cos \alpha \quad (1)$$

The third term, which accounts for shear strains, $\gamma_{\phi\theta}$, is neglected in netting analysis. From eq (1), the fiber-direction stresses in the hoop and helical plies at angles α_1 and α_2 can be written as:

$$\sigma_h = E \varepsilon_{\theta} \quad (2)$$

$$\sigma_{\alpha_1} = E(\varepsilon_{\theta} \sin^2 \alpha_1 + \varepsilon_{\phi} \cos^2 \alpha_1) \quad (3)$$

$$\sigma_{\alpha_2} = E(\varepsilon_{\theta} \sin^2 \alpha_2 + \varepsilon_{\phi} \cos^2 \alpha_2) \quad (4)$$

where E is the fiber uniaxial elastic modulus.

Figure 5 shows that for helical plies with fiber area, A_f , the normal forces in a unit width of the axial, ϕ , and hoop, θ , directions are given by

$$N_{\phi} = \sigma_f A_f \cos^2 \alpha \quad \text{and} \quad N_{\theta} = \sigma_f A_f \sin^2 \alpha \quad (5)$$

The fiber area for a unit width is the lamina thickness times the fiber-volume ratio. Summing the hoop forces in a unit length of the pressure vessel with radius, R , and pressure, P , gives

$$PR = \sigma_h A_h + \sigma_{\alpha_1} A_{\alpha_1} \sin^2 \alpha_1 + \sigma_{\alpha_2} A_{\alpha_2} \sin^2 \alpha_2 \quad (6)$$

Similarly, the axial forces in the circumference of a closed-ended cylinder give

$$PR/2 = \sigma_{\alpha_1} A_{\alpha_1} \cos^2 \alpha_1 + \sigma_{\alpha_2} A_{\alpha_2} \cos^2 \alpha_2 \quad (7)$$

Substituting equations (2) through (4) to express the stresses as a function of the strains, equations (6) and (7) become

$$PR = E \left[\varepsilon_{\theta} A_h + (\varepsilon_{\theta} \sin^2 \alpha_1 + \varepsilon_{\phi} \cos^2 \alpha_1) A_{\alpha_1} \sin^2 \alpha_1 + (\varepsilon_{\theta} \sin^2 \alpha_2 + \varepsilon_{\phi} \cos^2 \alpha_2) A_{\alpha_2} \sin^2 \alpha_2 \right] \quad (8)$$

$$PR/2 = E \left[(\varepsilon_{\theta} \sin^2 \alpha_1 + \varepsilon_{\phi} \cos^2 \alpha_1) A_{\alpha_1} \cos^2 \alpha_1 + (\varepsilon_{\theta} \sin^2 \alpha_2 + \varepsilon_{\phi} \cos^2 \alpha_2) A_{\alpha_2} \cos^2 \alpha_2 \right] \quad (9)$$

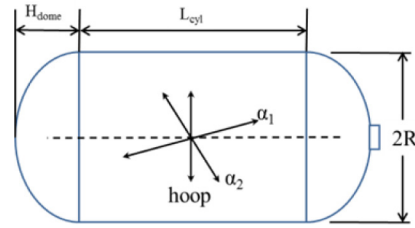


Figure 4. The composite pressure vessel with hoop and helical plies at angles α_1 and α_2 .

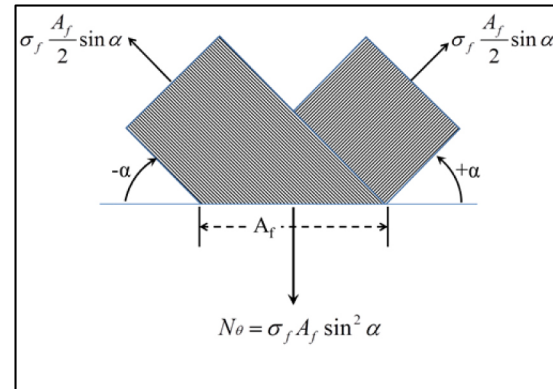
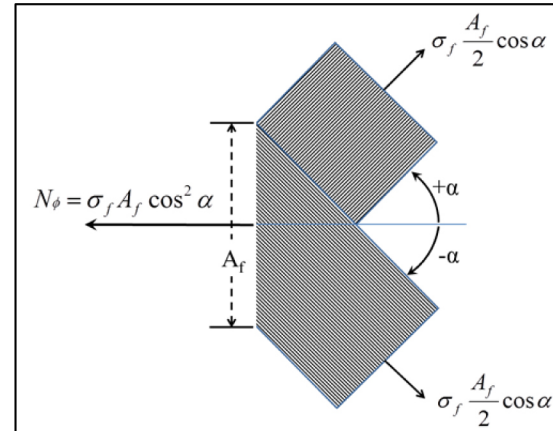


Figure 5. The resolved axial (top) and hoop (bottom) forces in netting analysis of helical plies.

Collecting the strain terms gives:

$$RP/E = K_3 \varepsilon_{\theta} + K_1 \varepsilon_{\phi}$$

$$RP/2E = K_1 \varepsilon_\theta + K_2 \varepsilon_\phi \quad (11)$$

where:

$$K_1 = A_{\alpha 1} \sin^2 \alpha_1 \cos^2 \alpha_1 + A_{\alpha 2} \sin^2 \alpha_2 \cos^2 \alpha_2 \quad (12)$$

$$K_2 = A_{\alpha 1} \cos^4 \alpha_1 + A_{\alpha 2} \cos^4 \alpha_2 \quad (13)$$

$$K_3 = A_h + A_{\alpha 1} \sin^4 \alpha_1 + A_{\alpha 2} \sin^4 \alpha_2 \quad (14)$$

Equations (10) and (11) can be solved simultaneously to give the strains as:

$$\varepsilon_\theta = \frac{PR(K_1 - 2K_2)}{2E(K_1^2 - K_2K_3)} \quad (15)$$

$$\varepsilon_\phi = \frac{PR(2K_1 - K_3)}{2E(K_1^2 - K_2K_3)} \quad (16)$$

The lamina stresses given by equations (2) through (4) can be calculated by back substituting the hoop and axial strains from equations (15) and (16).

Although standard netting analysis theory effectively approximates the average in-plane stresses and strains of a composite laminate, it neglects the stress concentration due to the thick-walled pressure vessel geometry plus the added hoop stress at the inner wall due to through-thickness compression of the orthotropic laminate. Both of these factors combine to increase the stresses and strains at the inside surface of the composite tank wall. Therefore, strain concentration factors were calculated and added to the netting analysis strains for calculating the stresses.

For thick-walled cylinders under internal pressure, the hoop stresses are highest at the inside surface due to the plane strain axial constraint. Timoshenko and Goodier [2] provide the solution for the hoop stress at the inside surface of the thick-walled cylinder in Figure 6.

$$(\sigma_\theta)_{\max} = \frac{p_i(a^2 + b^2)}{(b^2 - a^2)} \quad (17)$$

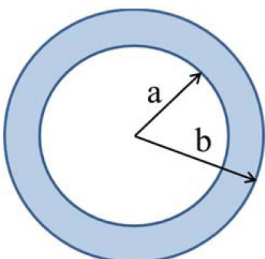


Figure 6. The thick-walled cylinder geometry.

The average hoop stress from netting analysis is

$$\sigma_\theta^{ave} = p_i \frac{a}{(b-a)} \quad (18)$$

Expressing the maximum hoop stress in terms of the average stress gives

$$(\sigma_\theta)_{\max} = \sigma_\theta^{ave} \frac{(a^2 + b^2)}{a(b+a)} = \sigma_\theta^{ave} S_\theta \text{ where } S_\theta = \frac{(a^2 + b^2)}{a(b+a)} \quad (19)$$

As a first approximation, the hoop fiber strains were increased by the thick-wall concentration factor, S_θ .

There is a similar but less pronounced concentration in the tangential stress in the dome which can be estimated from the equation for surface stress in a thick spherical cavity [2].

$$(\sigma_t)_{\max} = \frac{p_i}{2} \frac{(2a^3 + b^3)}{(b^3 - a^3)} \quad (20)$$

The average axial stress from netting analysis is

$$\sigma_t^{ave} = p_i \frac{a}{2(b-a)} \quad (21)$$

$$(\sigma_t)_{\max} = \sigma_t^{ave} \frac{(2a^3 + b^3)(b-a)}{(b^3 - a^3)a} = \sigma_t^{ave} S_\phi \text{ where } S_\phi = \frac{(2a^3 + b^3)(b-a)}{(b^3 - a^3)a} \quad (22)$$

Note that the dome thickness includes only the two helical layers. The axial stress and strain from netting analysis at the transition to the dome was multiplied by the concentration factor, S_ϕ .

The uniaxial stiffness, E_1 , of a 60% carbon fiber epoxy lamina is typically 14 to 17 times the transverse lamina stiffness, E_2 . Figure 7 shows that the pressure at the inside surface compresses the inside layers as load is transferred through the thickness into the fibers in the hoop and helical directions.

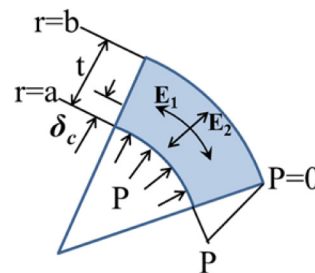


Figure 7. Through-thickness compression of the orthotropic composite cylinder.

Assuming the radial stress decreases linearly through the thickness, which is valid within thicknesses for these tank geometries, the inside layer is compressed an added amount, δ_c , which can be approximated as

$$\delta_c = \frac{P(b-a)}{2E_2} \left(\frac{E_1 - E_2}{E_1} \right) \quad (23)$$

The added hoop strain is greatest at the inside surface and it is equal to the radial compression divided by the inside radius

$$(\varepsilon_h)_{compression} = \frac{\delta_c}{a} = \frac{P(b-a)}{2E_2 a} \left(\frac{E_1 - E_2}{E_1} \right) \quad (24)$$

The average strains from the netting analysis were increased by this amount. The added hoop strain in the tank cylinder was calculated based on the full composite thickness, but the axial strain increment at the transition from the cylinder to the dome was calculated based on the thickness of the two helical lamina.

The enhanced netting analysis equations were implemented in a spreadsheet to estimate the required lamina thicknesses. The composite input data includes the axial and transverse fiber moduli (E_{1f} , E_{2f}), the matrix modulus (E_m), and the fiber volume fraction (V_f). The lamina stiffnesses, E_1 and E_2 , are calculated using the formulas from Whitney et al [3]:

$$E_1 = E_{1f}V_f + E_m(1 - V_f) \quad (25)$$

$$E_2 = E_m \frac{(1 + \xi \eta_2 V_f)}{(1 - \eta_2 V_f)} \quad \eta_2 = E_m \frac{(E_{2f}/E_m - 1)}{(E_{2f}/E_m + \xi)} \quad (26)$$

where $\xi = 2$ for square fiber packing and $\xi = 1$ for hexagonal packing. Hexagonal packing was assumed due to the random packing of the fiber strands in the filament winding process.

The lamina design strength is determined from the average fiber tensile strength (σ_f), the fiber-to-lamina translation efficiency (T_{eff}), the manufacturing coefficient of variation (V_m), and the code-based safety factor (SF). The fiber-to-lamina translation efficiency is the ratio of the tested lamina strength divided by the theoretical strength. The tested strength is lower than the theoretical strength due to fiber breakage during handling plus stress concentrations that occur at fiber-matrix interfaces in the composite as matrix cracking and fiber-matrix debonding occurs near the failure load. A fiber-to-lamina translation efficiency of about 87% is reported by one manufacturer of fibers used in composite pressure vessels. The manufacturing variability factor accounts for the statistical range in fiber strength and layup variability in the manufacturing process. The code safety factor provides the required margin between the code-rated pressure and the actual burst pressures of the vessel. The safety factor accounts for the statistical, time-dependent, stress rupture behavior of the type of reinforcing fiber used.

Under current standards, carbon fiber requires a minimum safety factor of 2.25 on stress whereas glass fiber requires a safety factor of 3.5. The design fiber strength is calculated as:

$$(\sigma_f)_{design} = \sigma_f T_{eff} (1 - V_m) / SF \quad (27)$$

As an example, a carbon fiber composite with 87% translation efficiency, 10% manufacturing variability, and 2.25 safety factor would have a fiber design strength that is 35% of the average tensile strength.

The ratio of the helical to hoop layer stresses (SR) is another design factor included in the model. It has been found through tank manufacturing and testing that helical layers tend to exhibit lower translation efficiency than hoop layers. Peters [4] attributed this to the helical layers having more overlaps and cross-overs, plus other discontinuities that exist in the dome-to-cylinder transition and near the polar bosses. Peters (1990) reports helical-to-hoop stress ratios of 0.75 to 1.0, but for high pressure hydrogen storage vessels, US Department of Energy Record #13010 [5] recommends stress ratios of 0.5 for 700 bar design pressure and 0.55 for 350 bar pressure. The stress ratio is included in the spreadsheet calculation by increasing the helical layer thicknesses by the factor of 1/SR. The hoop layer thickness is not increased.

The layer thicknesses in the spreadsheet model are adjusted until the calculated hoop and helical fiber stresses are less than the fiber design strength from equation (27). The two helical angles and the three layer thicknesses are defined independently, so some engineering judgment is required to define the helical angles. Trial and error iteration of the layer thicknesses is also required to minimize composite thickness within the fiber design strength. The combination of low angle (10° near-axial reinforcement at the polar boss), mid angle (50° dome shoulder reinforcement) and hoop lamina has been found to give tank wall thicknesses and masses that are similar to manufactured tanks. Table 3 provides the results of the estimated weight of the baseline tank using the netting model.

Hydrogen Tank Finite Element Model

The regression and netting models were developed to estimate the overall tank performance and cost attributes of the tanks. A finite element (FE) model was also developed to predict the detailed stress and strain distribution within the different plies building up the tank's complete laminate as well as the boss. The FE model helps to analyze different design concepts to optimize the tank's performance attributes and costs. The tank simulation is setup through an automated model chain that first calculates all necessary FE initialization parameters for specific tank liner and boss geometries, plus the fiber layers with respect to ply layup structure and material properties. The FE model was constructed in 3D to simulate the lamina fiber directions through the modeling plane. A 2D axisymmetric model would increase calculation speed, however, it would not allow to precisely model the lamina fiber directions nor create tank models to be used in later crash simulations. Due to the axial symmetry of the tank, a wedge section model instead of a full 360° model is used to decrease modeling time.