



Prifysgol Cymru
Y Drindod Dewi Sant
University of Wales
Trinity Saint David

School of Engineering

*Hydrogen Power: The Use of Filament-Wound
Carbon Composite Pressure Vessels in small scale
Ammonia Storage.*

Aidan Chapman

P149661

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This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.

Signed: A.Chapman

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Abstract

This Thesis Investigates the potential use of Ammonia as a fuel supply for both ammonia fuel cells and hydrogen fuel vehicles. As such, the investigation is focused on the design and integration of a fuel tank capable of storing liquid ammonia in a safe and practical manner.

Testing was conducted through the use of Classical Lamination Theory and simulated FE modelling and found results to be in good agreement between the two mediums.

The cylindrical tank was found to best combine a minimum specified safety factor of 2.5 and limit material usage when an aluminium sleeve of 3mm thickness with a carbon wrap of 3.6mm was applied. When combined, it was found that the materials exhibited increased strength, further increasing the safety of the design.

When combined with necessary external safety features discussed, it was concluded that that the fuel tank, along with ammonia fuel was a future possibility to be considered in the development of alternative fuel vehicles.

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2 Nomenclature

General

R	Radius
P	Pressure
t	thickness
σ	Stress
τ	Shear Stress
$E_{L,T,11,22}$	Long/Lat Moduli
ν	Poisson's Ratio
θ	Laminate Angle

Buckling Analysis

$E_{x,y}$	Translated Moduli
μ	Poisson's Ratio
d	Diameter
P_{CR}	Critical Pressure
l	length
n	
α	

Laminate Theory

$\mu_{12,21}$	Major/Minor Poisson's
$[Q_{ij}]$	Reduced Stiffness Matrix
$[\bar{Q}_{ij}]$	Translated Stiffness Matrix
ϵ_i^0	Mid-plane Strains
σ_i	Total Stresses
ϵ_i	Total Strains
$\sigma_{1,2}$	Principle Laminate Stresses
$\{N_i\}$	In-plane Forces
$\{M_i\}$	Moments
$[A_{ij}]$	Laminate in-plane stiffness
$[B_{ij}]$	bending-stretching coupling
$[D_{ij}]$	Laminate Bending Stiffness
$[C_{ij}], [A_{ij}]^{-1}$	Inverse A Matrix
Z^k	Coordinate to bottom of K^{th} Layer
$\{K\}$	Laminate Curvatures
h_k	thickness at top of K^{th} layer
h_{k-1}	thickness at bottom of K^{th} layer

3 Introduction

Since the beginning of the automobile industry, engineers have sought to push the boundaries of improvement, whether it was through personal intrigue and publicity, or through concurrent legislation at the time, the world of the automobile is an ever changing one.

One current topic is reducing the environmental impacts of personal transport. With the upcoming Euro 7 legislation coming into effect by 2021, stating that the “emissions target for new cars will be 95 g CO₂/km” (EUROPEAN PARLIAMENT, 2009), bringing a decrease in emissions of roughly 26% from the previous standard.

The Basis of this Thesis is to design an alternative fuel tank, that can be implemented into an existing vehicle, or to be used in new production vehicles. The fuel to be used within the tank will be ammonia, which can be broken down to form Hydrogen, that would be capable of running a fuel cell, or used in a hydrogen IC engine.

The reason for this Thesis is to investigate a potential fuel to compliment the current range of alternative fuel automobiles, with the intention of providing a rival fuel to that of Gasoline and Diesel. In the UK, alternative fuel vehicles have been generating a steady rise in total car sales, increasing by 20% in the last year, coming in at 2.2% of total new registrations made (Parry, 2019). Battery power is currently leading the quest for alternative fuels, however making up only 3% of final car sales worldwide, there is clearly still a need for a more diverse portfolio of vehicles to take over from Fossil Fuels. (Irle, 2019) Hydrogen is one such competitor. When used in an IC engine, it produces near to no pollutants, other than small amounts of NO_x at fuel ratios under stoichiometric conditions.

However, Hydrogen does have its drawbacks. For instance, if using fuel cells, the cathodes inside require the use of Platinum, a rare and expensive resource. Alongside this, hydrogen itself is highly volatile, making it difficult and expensive to store and transport.

This is where ammonia comes in. Whilst the chemical is highly acidic, and therefore toxic, ammonia can be stored much easier and can be transported at a much-reduced cost. Alongside this, the chemical already has the third largest transportation network globally due to its uses as fertilizer and in cleaning products.

The issue that will be addressed in this thesis is the possibility of small-scale storage of ammonia, in a container that must be Resistant to corrosion to contain the liquid, as well as being robust enough to remain in-tact in the event of a medium-large scale accident.

3.1 Aims of The Project

- To Investigate the use of Hydrogen Fuel for the applied design of a storage tank, that holds true to a thorough research informed specification.
- To Evaluate the performance of such a design, through Simulated Software and mathematical analysis, to conclude the performance characteristics necessary for the part to be fully functional.

4 Literature review

4.1 Alternative fuels

There are currently a wide range of alternative fuels that are being used within the automobile industry. However, even with this wide range, the alternative fuel market makes up a small amount of total vehicle sales worldwide. The three main types of power generation within the automotive industry can be generalised as Gaseous, Electric, and biofuel; These three terms can be split into the five most commonly used fuels discussed below:

4.1.1 Biodiesel

Biodiesel fuel is an umbrella term used to denote a fuel source made from vegetable oils. These fuels are then used within a standard diesel compression engine, and can be used as either a substitute to, or mixed with, standard diesel fuels.

Conclusions from (KumarAgarwal, 2007) state that the use of biodiesel/diesel blends resulted in lower net emission output of both CO and NO_x, and that B20 (20% biomass, 80% diesel) could be implemented into current diesel engines without the requirement of modified hardware. Alongside this, it was found that the peak thermal efficiency increased by up to 2.5% and general wear of internal parts reduced by 30% due to the increased lubrication of the fuel blend.

Compared to the conclusions drawn by (Shahid & Jamal, 2008), it can be said that biodiesel, like most fuels, has both its advantages and drawbacks. The main drawback being that the use of pure biodiesel is only possible in short-term running engines. The conclusions show that for the most effective long-term use, B20 fuel is the better option. Alongside this, the vehicle will still need to run on pure diesel on 'cold start', which is generally when the engine will produce the most emissions from its journey.

Therefore, it can be said, that whilst biodiesel does reduce overall emissions from the diesel engine, it can only be considered as a short-term reduction in emissions, and not as a long-term solution towards a zero-emission future.

4.1.2 Alcohols

Alcohols such as Methanol and Ethanol are a specific form of biomass fuels, and are most used as additives to gasoline, for use in internal combustion engines. Alcohol fuels are normally created as by products from the fermentation of corn and sugarcane. As with other biomass fuels, the injected fuel into the engine must be blended fuel, and therefore presents similar problems to that of biofuel.

Conclusions drawn from (Yusri, et al., 2017) state that blending alcohols with diesel cause the engine to produce lower brake power, and lower torque ranges, and this trend seemed to intensify with a higher percentage of alcohol used. Contrary to this, their research also found that alcohols with higher cetane numbers improve total combustion, thus increasing torque. Alongside this, the improve combustion efficiency had a knock-on effect by lowering the total output of NO_x, however this seemed to have the reverse effect on a lean-burn engine. The research also gave note that the exhaust gasses of a 'blended' fuel contained less CO however showed an increase in HC from unburnt fuel. However, the blended fuels tended to contain fewer overall emissions than that of a gasoline engine.

This reinforces the idea that diesel and gasoline blend fuels are an achievable short-term goal for reducing overall emissions from automobiles, however in the long-term, more efficient, zero emission outcomes will be needed that stray away from non-renewable fuel sources.

4.1.3 Liquid Petroleum Gas

Liquid Petroleum Gas (LPG) is a gas by-product from petroleum refinement, and generally consists of a mixture of Butane and Propane, although singular mixtures of each are possible. LPG can be easily implemented worldwide, as varying values in the mixture allow for different climatic conditions, and vehicles only require some minor modifications to convert them. However, most vehicles will require the use of both gasoline and LPG, usually within the cold start phase, or when the LPG fuel tank drops below a certain level.

Despite this, research from (Tasic, et al., 2011) and (Mustafa & Gitano-Briggs, 2009) has shown that a vehicle running on LPG can reduce average emissions of CO, CO₂, and HC by 20%, 10%, and 40% respectively. By applying this on a worldwide scale, this reduction in emission would be a massive step towards zero emission vehicles and could provide scientists and engineers the extra time to develop the next step.

However, as with Biofuels, the vehicles still need to use a gasoline fuel tank, and gasoline is a necessity on cold start when the engine has not yet reached lambda. Therefore, it can be said that whilst these alternative fuels will help towards a zero-emission goal, the future of most automobiles will not contain a combustion engine.

4.1.4 Battery Power

Battery Powered vehicles fall into the category of electric vehicles and are considered by many to be the next long-term step in producing a viable zero-emission vehicle, however the current technology does have its disadvantages, that to some, may outweigh the advantages.

Table 4-1: Advantages and Disadvantages of the EV (Argueta, 2010)

Advantages	Disadvantages
Fuel can be harnessed from any source of electricity, which is available in most homes and businesses.	Limited in the distance that can be driven before the complete failure of the battery.
It reduces hydrocarbon and carbon monoxide, responsible for many environmental problems, by 98%.	Accessories, such as air conditioning and radios drain the battery.
Also reduces pollution.	Heavier car due to the electric motors, batteries, chargers, and controllers.
Does not produce emissions. Important in urban cities, where cleaner air is much needed.	More expensive because of cost of the parts.

Alongside this, the life cycle of the batteries should be considered. Almost all automotive batteries require the use of lithium, and as stated by (Tahil, 2007) “A Li-Ion ‘Energy’ battery requires between 1.4 and 1.5kg of Li₂CO₃ per kWh of capacity. Therefore, 60M PHEV20s with a 5kWh battery would require at least 420,000 tonnes of Li₂CO₃ per year – 6 times current production.” These values skyrocket if fleets of heavy goods vehicles are taken into consideration as well. Therefore, it is clear the world production of Lithium cannot sustain a such a model.

Table 4-2: The Environmental impacts of Battery Manufacturing (McManus, 2012)

	Climate change (kg CO ₂ eq)	Metal depletion (kg Fe eq)	Fossil fuel depletion (kg Oil eq)	Cumulative energy demand (MJ/kg)
Lead acid	0.9	0.4	0.3	17
Lithium ion (NMP solvent)	12.5	20	1.6	90
Lithium ion (water solvent)	4.4	20	1.5	88
Nickel cadmium	2.1	1.5	0.7	37
Nickel metal hydride	5.3	3.2	1.6	90
Sodium sulphur	1.2	3.2	0.4	19

From the table above, it is clearly shown that lithium ion batteries have the highest figures for all categories of environmental impacts. The high energy demand and output of CO₂ show that lithium

is not the most sustainable approach to battery production, however they are currently the best option available to the automotive market. A full lifecycle would generate a much lower output overall for battery powered vehicles, however specifics such as production and waste should still be of high concern.

With Further developments and improvements happening within the industry right now, it is unclear how much battery power will play a role in future vehicles. Alongside production problems with current batteries, there is the demand for charging stations, which on a small scale has little effect, however if this is scaled up it becomes a major problem, not only for current station capacity, but for electricity production and distribution.

4.1.5 Hydrogen

Hydrogen power can fall into both categories of vehicles, as hydrogen can be used as an alternative fuel in IC engines, as well as generating electrical power through a PEM fuel cell. PEMFC vehicles work in the same theory as battery powered motors, however they generate electricity 'on-board' using hydrogen. Hydrogen gas is passed through the fuel cell, which allows it to reach with a catalyst (usually platinum), through which a process of stripping electrons from the hydrogen atoms occurs. These electrons are then used as an energy source to power the electric motors of the vehicle (Edmunds, 2015).

To produce fuel grade Hydrogen, the main process used is that of natural gas reformation. This process accounts for the generation of 50% of the global demand of hydrogen. Alongside this, oil reformation (30%), coal gasification (16%) and renewable sources (4%) account for the remaining processes in which hydrogen is generated (Kalamaras & Efstathiou, 2013).

The current advantages and disadvantages of hydrogen power as a fuel source are widely agreed. The following lists have been compiled from (Berman, 2015), (Barden, 2017) and (Rinkesh, n.d.):

Advantages of Hydrogen Power

- **Increased driving range** – Hydrogen powered vehicles can provide up to and over 300 miles range per full tank of fuel. This gives consumers the ability to travel further than most battery powered vehicles on a single charge and can even match the range available with most gasoline vehicles.
- **Quick Refueling** – The refueling of hydrogen vehicles is very similar to gasoline and diesel vehicles, providing a full tank of fuel in under 5 minutes. The clear advantage over battery power is the lengthy wait times involved with battery charging.

- **Zero direct emissions** – Hydrogen vehicles, when used in conjunction with a fuel cell, produce only water and oxygen as a by-product of the power process. However, there is currently still questionability towards the well-to-wheel emissions generated from hydrogen reformation.
- **Energy efficiency** – Hydrogen produces more energy per pound of fuel than gasoline powered vehicles.

Disadvantages of Hydrogen Power

- **Lack of Infrastructure** – Current infrastructure is incapable of storing hydrogen, as the tanks require very high pressures to ensure a high volumetric density of fuel is available, therefore infrastructure will need to be updated to provide this service.
- **Cost of Vehicles** – Current Fuel Cell Vehicles are very expensive when compared to gasoline equivalents, due to the Fuel Cell technology requiring precious metals to function efficiently.
- **Cost of Fuel** – The pressurizing of hydrogen to the levels required for fuel stations and vehicles is an energy intensive and costly process, which in turn, directly effects the average fuel prices for the consumer.
- **Storage and Transportation** – The high pressures involved with hydrogen vehicles makes the transportation and storage of the fuel costly, as well as increased safety concerns over the volatility of the compressed hydrogen.
- **Safety Concerns** – Hydrogen is a very volatile fuel. The high pressures involved increase safety concerns in respect to fuel tank integrity in both the vehicles themselves and the logistics of transport and storage. Alongside this, Hydrogen is odorless and therefore leaks are only detectable using sensors.

Despite the concerns generated through the reformation of hydrogen, a Life Cycle Assessment conducted by (Moghbelli, et al., 2003) found that, of a variety of different vehicles tested, hydrogen-based fuel cell vehicles (DHFCV) outperformed in terms of both efficiency and emissions. It was found that the well-to-wheel emissions of hydrogen reformation from Methane, was 63.6% lower when compared to gasoline. Alongside this, it was found that the hydrogen powered vehicles had a 30% well-to-wheel efficiency, over twice that generated by traditional gasoline vehicles. These results also correlate strongly with another LCA done on well-to-wheel hydrogen, produced by (Contadini, 2002), that found that in 50% of cases, Hydrogen was found to be the most efficient energy cycle over Methanol, and all hydrogen scenarios proved more efficient than traditional gasoline and diesel engines.

4.2 Ammonia as a Fuel Source

4.2.1 Is Ammonia a viable fuel?

Ammonia is chemical that is produced from to combination of hydrogen and nitrogen, usually derived using the Haber Process. The process bonds nitrogen and hydrogen under extremely high pressures and with the help of a catalyst to generate a chemical reaction. Pure hydrogen produced using the methods mentioned above is combined with nitrogen found abundantly in natural air to produce gaseous ammonia. From here the ammonia is then fed through industrial reactors to convert it into a fluid (Briney, 2019). In today's industry, the most common uses for ammonia is within the fertilizers for agriculture and the production of household cleaners, however, there is a growing interest in using ammonia as an indirect pre-fuel storage for hydrogen power.

In the conclusions drawn from (Lan, et al., 2012), it is outlined that the main advantages of using ammonia are storage, transportation and cost. Ammonia is easily stored as a liquid or solid, at much lower pressures than required from pure hydrogen, such that, at an average temperature of 20°C, the required pressure to keep ammonia as a saturated liquid 8.5 bar (Prasad, 1989). Alongside this, Ammonia already has a large developed network for transport and logistics, exceeding \$4.99 Billion in 2018 (Trade Economy, 2018), and is currently the 460th most traded product globally (OEC, 2019). However, the research conducted by (Lan, et al., 2012) concludes that to further reduce carbon emissions, alternative feed stocks for the Hydrogen process must also be considered.

To further this, the research of (Bicer, et al., 2016) into the life cycle assessment of ammonia production, found that hydropower and water electrolysis provide a good alternative to current non-renewably led methods of producing ammonia. It was found that municipal waste-based production generated the lowest global warming and toxicity values, thus being the most environmentally benign practice, whereas hydropower gave the highest sustainability index and highest energy efficiencies. This data allows for the assumption that renewable sources, with their increased efficiencies and reduced footprint can further lower the well-to-wheel impact of ammonia production, making it a viable option for hydrogen-based power generation.

Ammonia has been studied as both a hydrogen-based fuel source, as well as direct ammonia fuel cells. (Cheddie, 2012) provided research into using ammonia as a feed source for hydrogen fuel cells, using a catalyst to break down the ammonia into hydrogen and nitrogen prior to use in PEMFC. It was found that high temperature (400-600°C) were required to gain useful conversion efficiencies. Alongside this, the materials used as a catalyst within the 'cracking' of ammonia are expensive noble metals.

In the research of (Duijm, et al., 2005), it stated that, provided the containers are thoroughly emptied and cleaned, that existing fuel storage located at existing fuel stations would be capable of handling and storing liquid ammonia. However, due to safety, it would be highlighted that extra caution and measures should be taken into such upgrades. For one, the tanks should be located in contained underground units, which is usually the case, and these units must have increased safety features including leak sensors and close temperature regulation. Alongside this, it is also mentioned that it could be necessary to relocate the storage tanks to an area that is 200+ meters away from populated living spaces. This would be a precautionary safety measure in the unlikely event of a catastrophic failure, to limit public exposure to the hazardous gases that would leak into the nearby airspace. Alongside this, fuel pumps would require a smart shut off, to prevent the ability to pump fuel unless securely within the vehicle refuelling unit. Whilst this process is happening, a seal must be formed within the vehicles internal fuel systems and the pump, to prevent leakage of liquid and fumes.

In an article written by (Brown, 2017), he discusses a new technology designed by *RenCat*, a company that designs ammonia cracking devices for use in hydrogen powered generators. Using an in-expensive Fe-Ni Catalyst, atmospheric pressure and running at temperatures of half that of previous technologies, a new device for cracking ammonia in hydrogen has been produced. Alongside the apparent reduction in energy and cost, it can generate hydrogen at a higher efficiency to previous designs. Post cracking, it uses a second mixed metal oxide catalyst to remove trace ammonia impurities down to 0.1ppm, providing fuel grade hydrogen directly out the exit. This device, or even future advancements, in combination with a hydrogen fuel cell and an ammonia fuel tank could provide the next step in hydrogen powered vehicles.

Alongside this, the information provided by (The Royal Society, 2020), shows that the estimated costs of storage and transport of ammonia, when compared to hydrogen, is a much cheaper and more economical process.

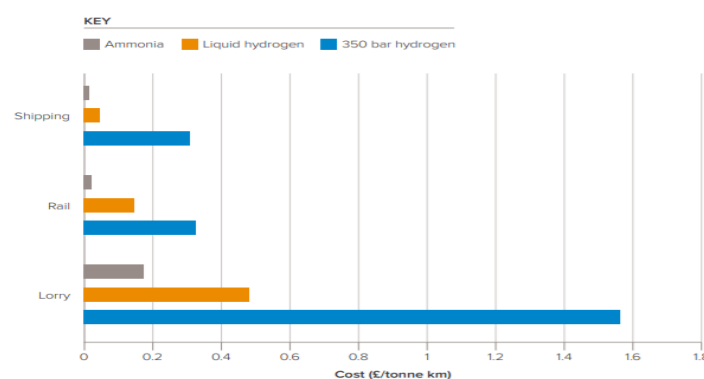


Figure 4-1 : Estimated costs of Ammonia and hydrogen transport (The Royal Society, 2020)

4.2.2 Safety Considerations

Anhydrous Ammonia, ammonia with a water content of under 33ppm, is a colourless alkaline liquid or gas, which is both toxic and corrosive to living organisms. In humans, it effects 3 major areas, the eyes, lungs and skin; And can prove fatal if inhaled. Alongside this, it can cause skin irritation and potential loss of sight if in direct contact with these areas. Ammonia is easily absorbed into water, making it dangerous to skin contact, as it will easily spread through cells (Marrella, et al., 2019). Thus, safety must be of paramount concern when dealing with this chemical. Due to this, the ammonia must be stored in failure proof tanks, with precautionary sensors to carefully monitor for leaks or damage. The storage unit must be self-contained and away from direct airways into the passenger cabin.

The work of (Duijm, et al., 2005) explores the safety of using ammonia as a transport fuel. The first process was determining the potential failures of the design and outlining strategic ways to minimise and counteract any potential outcome. :

Table 4-3 : Results of the HAZOP study of the car system (Duijm, et al., 2005)

System	Event	Safeguard	Conclusion
Fill line to tank	Release due to check valve on tank fails and valve at connection fails as well on disconnection. Release by filling	Disconnect Safety system on car. Line empty Concentration based alarm; emergency shut down of fuelling station	The Safety System prevents a release of ammonia
Tank	Release	Safe design of the tank. The tank is designed to withstand a collision. External frame to protect tank on impact. A level indicator protects overfilling and valves close at max. level. Gas alarm is installed Safety system on compressor and vacuum compressor to prevent damage on tank Max. flow valve in pipe from tank.	The safeguard will prevent damage of the tank and a total release will only happen in very severe accidents. Small punctures with a leak will be detected by the gas alarm. This will also detect gas leakages from other parts of the ammonia system. The max flow valve in pipe from tank and narrow pipes decrease a leak from this system.

Evaporator	Down-flow section on high pressure due to evaporator failure. Down flow system not designed for high pressure. Fuel cells will be destroyed, and leaks can occur	High-pressure alarm on down flow section closes main valve. Gas alarm installed	The safety system will protect the down flow system and the fuel cells.
Absorbers	Poisoned absorbers will allow NH ₃ into the fuel cells and damage these	NH ₃ sensor	
Fuel cells			Leakages from the system are limited and will be detected by the gas alarm

All these failsafe's must be considered in the design of the vehicles systems; however, this project will be focused on the design and integration of the tank.

4.3 Materials and Structure

Due to the nature of anhydrous ammonia, the fuel tank must be made of suitable materials to provide corrosive resistance and strength through the lifetime of the vehicle. This is a necessary feature to safeguard both the passengers of the vehicle and the general environment from the potential of a hazardous chemical spill.

Plastics

A well-known medium used to store hazardous chemicals, is the use of plastics. Plastics are generally used due to the versatility of the various types of plastics available, such as polypropylene, which is used for the storage of corrosive chemicals. (Clifton, 2020)

Polypropylene (PP) has a good temperature resistance and high melting point, along with good corrosion resistance, durability and a high resistance to cracking and stress. The uses of PP are vast, from use in automotive bumpers, to uses in military equipment, and for the storage of strong acids and bases (Professor Plastics, 2018).

Table 4-4 : Pros and Cons of Polypropylene (Rodgers, 2014)

Advantages	Disadvantages
Inexpensive Material	High thermal expansion coefficient
High Flexural Strength	UV degradation
Resistance to Moisture	High flammability
Good fatigue resistance	Susceptible to oxidation
Good impact strength	
Chemical Resistant	

Whilst showing good properties for corrosive resistance and good strengths, the application in this instance may not be beneficial. Under certain extreme circumstances, such as a vehicle fire, only a slow release of ammonia may be permitted to prevent a chemical spill, therefore the use of a thermoplastic, where intense heat will likely cause melting, would be inappropriate from a safety standpoint.

Stainless Steel

Another potential material to consider is Stainless Steel, which also has a popular use for storing chemicals. Stainless Steel can resist corrosion and protect against puncture, whilst having a high resistance to oxidization. Several benefits over the use of plastics include the ability to resist bursting under fluctuating loads and the ability to hold abrasive and toxic chemicals that would likely erode or degrade most plastics (SMF, 2019).

Table 4-5 : Advantages and Disadvantages of Stainless Steel (SGtanks, 2016)

Advantages	Disadvantages
Highly Resistant to oxidization	Heavy
High Durability	Different grades offer varying resistance to corrosion
Inexpensive material	
Corrosion resistant	
Generally longer lifespan than plastics	

Whilst Stainless steel provides several enhanced features over the use of plastic, the concerns of weight bring an issue. For the vehicle to provide relatively comparative distance compared to hydrogen vehicles, the on-board weight is a crucial factor. As the ammonia will be stored as a liquid, the fuel itself will account for an increase in weight, along with the other necessary devices such a cracker, therefore the tank itself must remain as light as possible.

Aluminum Alloys

Aluminum is a lightweight metal alloy and is used in thousands of products worldwide. Varying grades of aluminum are manufactured for their different purposes and properties. Each variation is given a 4-digit number, with the first number being the grade or series.

Table 4-6 : Properties of Aluminium (The Aluminium Association, n.d.)

Series	Properties
1xxx	99% aluminum content Excellent resistance to corrosion Easily workable Highly conductive
2xxx	High strength High toughness Easily corroded
3xxx	Moderate strength Good workability General all-purpose alloy
4xxx	Low melting point Usually used for welding wire
5xxx	Moderately high strength Good weldability Corrosion resistance
6xxx	Versatile Formable Weldable High strength
7xxx	Highly treatable High strength Used in aviation industry

In comparison to stainless steel, aluminum generally has lower strength, however, also contains a third of the weight of steel, therefore providing better strength to weight ratios. Aluminum generally has a higher corrosion resistance than steel and is typically cheaper. Alongside this, aluminum is easier to work and shape, but is harder to weld than steel. (MSM, 2014)

Carbon Fiber

Filament winding is a process generally used for the formation of hollow, cylindrical composite shells. Fiber tows are passed through a resin bath, which are then wrapped around a mandrel, which can then either be dissolved, or integrated as a liner for the design. The fiber feeding mechanism and mandrel spin speed allows for variations in orientation of fibers (Net Composites, 2019).

Table 4-7 : Advantages and Disadvantages of Filament winding (Net Composites, 2019)

Advantages	Disadvantages
Fast and efficient	High Mandrel costs for larger components
Easily controlled resin content	Unmolded external surface
Fiber cost reduced due to no secondary processes (fabric conversion)	Limited application to specific shapes
Orientations can be easily modified to respond to load distributions	Non-recyclable
Superb strength to weight ratio	
Resistant to corrosion	

The main issue with carbon fiber is that it is not a green resource, as the strands must be combined and cured with a polymer resin. Once cured, the newly formed carbon fiber is non-recyclable, and therefore, any inefficiency with the use of the material ends up as landfill, with no ability to melt down or re-mold errors or waste pieces (Harris, 2017).

However, through the research of () and (Shotton-Gale, et al., 2010) , new variations of the wet-filament winding technique have been observed. Dubbed the 'clean filament winding' technique, the use of modified dispensing techniques for the resin found that the volume of waste resin generate through the process was significantly lower than that of conventional methods. Alongside this, the volume of solvents required to clean the resin bath and impregnation unit were also significantly less, showing that less resin was being lost to the process. Furthermore, filament winding is considered more environmentally friendly than woven fabric, as less waste is generated from the secondary weaving and cutting process necessary to create woven carbon fiber.

4.4 Specification

4.4.1 Vehicle Range

- The fuel tank must be able to store enough liquid ammonia to generate the equivalent amount of hydrogen necessary to travel within the same range as hydrogen fuel tanks.
- Tank weight must be considered to prevent a further reduction in range due to increased weight of ammonia-based systems.

4.4.2 Tank weight

- The best combination of material thickness must be used to ensure reliance is not generated on either material; this will allow for reduced material usage across all aspects of the design.
- Minimum thicknesses within the given safety factors will provide the best outcome.

4.4.3 Tank Materials

- The tank will be comprised of both aluminium and carbon fibre.
- Each material should be experimented with individually, generating individual factors of safety to ensure all factors are considered equally.

4.4.4 Tank Design

- The tank will be a cylindrical shape, to provide the most robust design, with spherical end caps.
- FEA model will be verified using calculations provided on the cylindrical portions of the vessel.

4.4.5 Safety Considerations

- The Tank must safely contain the liquid at a MINIMUM of 8.5 bar internal pressure (7.5 bar difference).
- Both inner sleeve and wrap must generate safety factors of 2.5 at the specified pressure, thus allowing for fluctuations in pressure, and to reduce to potential of bursting in the event of a fire.
- The system must contain a pressure release valve in the event of a fire, to prevent explosion and rapid release into the environment.
- The tank will be fitted with a probe to measure capacity, a maximum of 85-90% of the total tank volume should contain liquid ammonia to prevent over-pressurization.

5 Methodologies

5.1 Preliminary design - Laminate

Firstly, it was necessary to gain the dimensions needed for a container to store a suitable amount of liquid ammonia. The percentage weight of Hydrogen in an Ammonia atom is that of 17.65%, therefore, to gain a similar vehicular range comparable to current hydrogen vehicles, the fuel tank would require a similar amount of Hydrogen by weight as such. Using a 125-liter tank, a hydrogen vehicle can store around 5Kg at 700bar of pressure, to travel between 500-600km (Air Liquide, 2020). Using this as a baseline, a tank with the ability to store 28.33Kg of Ammonia would be required.

Using the saturation tables for Liquid ammonia found at (Prasad, 1989), at a temperature of 20 degrees, at 8.5bar of pressure, the density would be 610.2Kg/m³. Using the equation for mass/density, a volume of 0.0464m³ (or 46.4L) was found to be the required size. However, this is under the assumption of a 100% conversion from liquid ammonia.

Therefore, the use of a 60L fuel tank, would allow for 77% efficiency within the required systems, which would be dependent on operating temperature and catalyst used within the cracker. A 60L Ammonia tank would contain 6.46Kg of Hydrogen, with a total wet weight of 36.61Kg.

From here, the length and radius of the tank could be calculated. By using Radial increments of 0.05m from 0.1m, up to 0.25m, a suitable length/width ratio could be found:

$$Length = \frac{Vol}{\pi R^2}$$

(5.1) Cylinder volume eq

Table 5-1 : Radius/Length ratios for the vessel

Volume (m ³)	Radius (m)	Length (m)
0.06	0.1	1.9
	0.15	0.848
	0.2	0.477
	0.25	0.305

It was chosen that the most suitable dimensions would be 0.15m radius and a length of 0.848m, to allow the tank to fit within the space available in the storage trunks of most passenger vehicles. With the dimensions set, an internal pressure of 8.5bar and using the stress principles applied to isotropic materials, a nominal minimum thickness can be calculated using;

$$t = \frac{PR}{\sigma}$$

(5.2) Isotropic Cylinder Stress eq

Using the 4 different Ultimate Stresses associated with Boron Epoxy, the largest minimum thickness for the hoop stress can be found:

Table 5-2 : Minimum thickness required by UTS

Direction	Max Stress	Minimum Thickness (mm)
X ^T	1260x10 ⁶	0.1
X ^C	2500x10 ⁶	0.05
Y ^T	61x10 ⁶	2.1
Y ^C	202x10 ⁶	0.6

From these values, a minimum thickness of 2.4mm was chosen, alongside several more runs of increasing thickness of 3.6, 4.8 and 6mm.

5.1.1 Buckling Analysis

The objective of Buckling Analysis is to predict the pressure at which a fiber wound cylinder will undergo inelastic deformation. Depending on the thickness and material used, the critical pressure can occur after failure of the device, or beforehand. Buckling Analysis can be used to calculate both internal and external pressures, with the latter being its most used application.

A composite cylinder may still hold an effective pressure past its buckling condition, however in this application, where pressures will fluctuate, this would not be a desired outcome.

The use of equations (4.3)-(4.7) were found from (Rao, et al., 2012), (LITTLE, et al., 2008), (Von-Mises, 1931), (Timoshenko & Gere, 1985).

Elastic Modulus

Before running the calculations necessary to determine the buckling loads, the elastic modulus in the x and y plane is to be calculated in reference to the laminate angle used. This is calculated using the equation:

$$E_x = \frac{E_L}{\cos^4 \theta + \frac{E_L}{E_T} \sin^4 \theta + 0.25 \left[\frac{E_L}{E_T} - 2\mu \right] \sin^2(2\theta)}$$

(5.3) Transformed Laminate x-Moduli eq

$$E_y = \frac{E_L}{\sin^4 \theta + \frac{E_L}{E_T} \cos^4 \theta + 0.25 \left[\frac{E_L}{E_T} - 2\mu \right] \sin^2(2\theta)}$$

(5.4) Transformed Laminate y-Moduli eq

Among others, the following three equations can be used to find the critical pressure at which a composite cylinder will enter inelastic deformation.

Windenburg & Trilling

$$P_{cr} = \frac{2.41E \left[\frac{t}{d} \right]^{2.5}}{[1 - \mu^2]^{0.75} \left[\frac{l}{d} - 0.45 \left[\frac{t}{d} \right]^{0.5} \right]}$$

(5.5) Windenburg and Trilling eq

Timoshenko and Von Mises

$$P_{cr} = \frac{Et}{R} \frac{1}{(n^2 - 1) \left[n^2 / \left[\frac{\pi R^2}{l} \right] + 1 \right]^2} + \frac{Et^3}{12(1 - \mu^2)R^3} \left[(n^2 - 1) + \frac{2n^2 - 1 - \mu}{n^2 / \left[\frac{\pi R^2}{l} \right]^2 + 1} \right]$$

(5.6) Timoshenko & Von Mises eq

Where;

$$n = \sqrt[8]{\frac{3 \left[\frac{\pi R^2}{l} \right]^4}{t^2 / (12(1 - \mu^2)R^2)}}$$

Tokugawa equation

$$P_{cr} = \frac{E(t/D)}{n^2 - 1 + \frac{\alpha^2}{2}} \left[\frac{2\alpha^2}{(n^2 + \alpha^2)^2} + \frac{2}{3(1 - \mu^2)} \left[\frac{t}{D} \right]^2 \left\{ (n^2 + \alpha^2)^2 - \frac{n^4(2n^2 - 1)}{(n^2 + \alpha^2)^2} \right\} \right]$$

(5.7) Tokugawa Eq

Where;

$$\alpha = \frac{\pi D}{2S} \quad n = 1.63 \left(\frac{D}{S} \right)^{0.5} \left(\frac{D}{t} \right)^{0.25}$$

5.1.2 Lamina and Failure Theory – first ply failure approach

There are five main failure theories used to quantify the point at which a composite shell will fracture and then fail. Unlike buckling analysis, failure theories predict the exact point at which the shell will be unable to continue holding pressure, and thus, become unable to perform its task.

Lamina Theory

Lamina theory is used to derive the resultant stresses acting through specific coordinates within a laminated shell. The theory implements the use of constitutive equations at specific points along a z-axis, thus allowing for stress to be calculated in individual laminate layers, or across the shell.

Using the chosen materials Moduli, major and minor Poisson's ratios, the reduced stiffness matrix [Q] can be calculated;

$$\begin{aligned} Q_{11} &= \frac{E_{11}}{1 - \mu_{12}\mu_{21}} \\ Q_{12} &= \frac{\mu_{12}E_{22}}{1 - \mu_{12}\mu_{21}} \\ Q_{22} &= \frac{E_{22}}{1 - \mu_{12}\mu_{21}} \\ Q_{66} &= G_{12} \end{aligned}$$

(5.8) Reduced Stiffness Equations

To transpose these values through the varying fibre orientation angles, the Transformed Reduced Stiffness matrix $[\bar{Q}]$ is calculated;

$$[\bar{Q}] = \begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\ \bar{Q}_{21} & \bar{Q}_{22} & \bar{Q}_{26} \\ \bar{Q}_{61} & \bar{Q}_{62} & \bar{Q}_{66} \end{bmatrix}$$

(5.9) Transposed Reduced Stiffness Matrix

Where $[\bar{Q}_{ij}]$ relates to axis 3D constitutive equations;

$$\begin{aligned} \bar{Q}_{11} &= Q_{11} \cos^4 \theta + 2(Q_{12} + 2Q_{66}) \sin^2 \theta \cos^2 \theta + Q_{22} \sin^4 \theta \\ \bar{Q}_{22} &= Q_{11} \sin^4 \theta + 2(Q_{12} + 2Q_{66}) \sin^2 \theta \cos^2 \theta + Q_{22} \cos^4 \theta \\ \bar{Q}_{12} &= (Q_{11} + Q_{22} + 2Q_{12}) \sin^2 \theta \cos^2 \theta + Q_{66} (\sin^4 \theta + \cos^4 \theta) \\ \bar{Q}_{66} &= (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66}) \sin^2 \theta \cos^2 \theta + Q_{66} (\sin^4 \theta + \cos^4 \theta) \\ \bar{Q}_{16} &= (Q_{11} - Q_{22} - 2Q_{66}) \sin \theta \cos^3 \theta + (Q_{12} - Q_{22} + 2Q_{66}) \sin^3 \theta \cos \theta \end{aligned}$$

$$\bar{Q}_{26} = (Q_{11} - Q_{12} - 2Q_{66}) \sin^3 \theta \cos \theta + (Q_{12} - Q_{22} + 2Q_{66}) \sin \theta \cos^3 \theta$$

(5.10) Transposed Stiffness Equations

To calculate the observed stresses, the constitutive relations of each ply are noted as;

$$\sigma_i = [\bar{Q}_{ij}] \{\epsilon_j\}$$

(5.11) Laminate stress-strain relations

Where $\{\epsilon_j\}$ is the total laminate strain found at any z-location, and calculated using the midplane strains $\{\epsilon_j^0\}$, and the curvatures $\{k_j\}$ as below;

$$\{\epsilon_j\} = \{\epsilon_j^0\} + z\{k_j\}$$

(5.12) Laminate total Stain eq

To calculate the midplane strains, the force resultants $\{N\}$ and moment results $\{M\}$ yield to form the constitutive equations;

$$\{N_j\} = [A_{ij}] \{\epsilon_j^0\} + [B_{ij}] \{k_j\}$$

$$\{M_j\} = [B_{ij}] \{\epsilon_j^0\} + [D_{ij}] \{k_j\}$$

(5.13) Laminate Forces and Moments

Forming the resulting matrix;

$$\begin{bmatrix} N_1 \\ N_2 \\ N_6 \\ M_1 \\ M_2 \\ M_6 \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} & A_{16} & B_{11} & B_{12} & B_{16} \\ A_{21} & A_{22} & A_{26} & B_{21} & B_{22} & B_{26} \\ A_{61} & A_{62} & A_{66} & B_{61} & B_{62} & B_{66} \\ B_{11} & B_{12} & B_{16} & D_{11} & D_{12} & D_{16} \\ B_{21} & B_{22} & B_{26} & D_{21} & D_{22} & D_{26} \\ B_{61} & B_{62} & B_{66} & D_{61} & D_{62} & D_{66} \end{bmatrix} \begin{Bmatrix} \epsilon_x^0 \\ \epsilon_y^0 \\ \gamma_{xy}^0 \\ k_x \\ k_y \\ k_{xy} \end{Bmatrix}$$

(5.14) Force and Moment Resultants

Where $[A_{ij}]$ is the laminate in-plane stiffness, $[B_{ij}]$ is the laminate bending-stretching coupling, and $[D_{ij}]$ is laminate bending stiffness, calculated as;

$$A_{ij} = [\bar{Q}_{ij}] z \{h_K - h_{K-1}\}$$

(5.15) Laminate in-plane stiffness

$$B_{ij} = \frac{1}{2} [\bar{Q}_{ij}] z \{h_K^2 - h_{K-1}^2\}$$

(5.16) Laminate Bending-stretching coupling

$$D_{ij} = \frac{1}{3} [\bar{Q}_{ij}]_z \{h_K^3 - h_{K-1}^3\}$$

(5.17) *Laminate Bending Stiffness*

In a symmetric laminate, such as the one considered, the values of the B matrix cancel to become 0 (Herakovich, 1998), and thus reduces the equation down to;

$$\{N_j\} = [A_{ij}] \{\epsilon_j^0\}$$

(5.18) *Reduced Force Resultant eq*

Where the force resultants are;

$$N_{(1)} = \frac{PR}{2}$$

$$N_{(2)} = PR$$

$$N_{(6)} = 0$$

(5.19) *Force Resultants*

By use of rearranging equation (5.18) the resultant midplane strain equation becomes;

$$[C_{ij}] * [N_j] = \{\epsilon_j^0\}$$

(5.20) *Laminate mid-plane strain eq*

Where the Matrix $[C_{ij}]$, refers to the inverse A matrix $[A_{ij}]^{-1}$.

Using the above equations, the curvatures can then be found from the midplane strains;

$$K_x = 0$$

$$K_y = -\frac{\epsilon_y^0}{R}$$

$$K_{xy} = \frac{\gamma_{xy}^0}{R}$$

(5.21) *Laminate Curvatures*

By combining equations (5.20) and (5.21) through the z-locations, the laminate total strains can be identified, along with the stresses present;

$$\epsilon_x = \epsilon_x^0 + zk_x$$

$$\epsilon_y = \epsilon_y^0 + zk_y$$

$$\gamma_{xy} = \gamma_{xy}^0 + z k_{xy}$$

(5.22) *Laminate Total Strains*

$$\sigma_i = [\bar{Q}_{ij}]\{\epsilon_i\}$$

(5.23) *Laminate Total Stresses*

From here, the stresses are then calculated through the carbon fibres principle coordinates, so they can be used in the Tsai Wu failure criterion;

$$\sigma_1 = \sigma_x \cos^2 \theta + \sigma_y \sin^2 \theta - 2\tau_{xy} \sin \theta \cos \theta$$

$$\sigma_2 = \sigma_y \cos^2 \theta + \sigma_x \sin^2 \theta + 2\tau_{xy} \sin \theta \cos \theta$$

$$\tau_{12} = \sigma_x \cos \theta \sin \theta - \sigma_y \cos \theta \sin \theta + \tau_{xy} \cos^2 \theta \sin \theta^2$$

(5.24) *Laminate Principle Stresses*

These equations were all coded to run through MATLAB coding software. This allowed the ability to quickly compute the values necessary and allowed for a prediction of failure for all angles from 0 through to 90 degrees. A copy of the code can be found in appendix 1.

(Herakovich, 1998), (Gheshlaghi, et al., 2006), (Rao, et al., 2012).

Tsai Wu Failure Criterion

The use of Tsai Wu failure criterion, requires the calculation of several components, based on the Ultimate Stresses of the chosen Materials;

$$\begin{aligned} H_1 &= \frac{1}{\sigma_{C1 \text{ ult}}} - \frac{1}{\sigma_{T1 \text{ ult}}} & H_{11} &= \frac{1}{\sigma_{C1 \text{ ult}} \sigma_{T1 \text{ ult}}} & H_2 &= \frac{1}{\sigma_{C2 \text{ ult}}} - \frac{1}{\sigma_{T2 \text{ ult}}} \\ H_{22} &= \frac{1}{\sigma_{C2 \text{ ult}} \sigma_{T2 \text{ ult}}} & H_6 &= 0 & H_{66} &= \frac{1}{\tau_{12 \text{ ult}}^2} & H_{12} &= \frac{1}{2\sigma_{T1 \text{ ult}}^2} \end{aligned}$$

(5.25) *Tsai Wu Failure components*

From these calculations, the values can be input into the failure equations, where each component of UTS will be checked against the relevant stresses acting through that plane. A resultant value of >1 will constitute that the vessel will fail, based on a first ply failure method, and anything under will give a percentage towards failure.

$$H_1 \sigma_1 + H_2 \sigma_2 + H_6 \tau_{12} + H_{11} \sigma_1^2 + H_{22} \sigma_2^2 + H_{66} \tau_{12}^2 + 2H_{12} \sigma_1 \sigma_2 < 1$$

(5.26) *Tsai Wu Failure Criterion*

5.1.3 Computational Modelling

The Preliminary testing consisted of a test, wherein the lay-up of the fibres was changed to test how the laminate responded under load. The test consisted of 12 different cases of alternating fibre angle and thickness, through the addition of further layers. The test used layers of 0.2mm thickness, in sums of 12 (2.4mm), 16 (3.2mm), 19 (3.8mm), and 25 (5mm). The varying angles chosen were 0°, 30°, 45°, 70° and 90°.

Firstly, the vessel was designed through Solidworks. After designing the vessel in 3D, a surface offset was generated, at a 0mm distance from the surface of the vessel. This 2D surface is necessary for allowing the part thickness to be altered within Design Studio quickly. Creating a new static analysis, whilst excluding the solid shell, allowed a 5mm mesh to be assigned to the 2D surface, and a base material to be added. The material added was Steel, however this would be changed within the design studio software. For Design Studio to read the file, it was exported from Solidworks as a .DAT file.

The .DAT file was then imported into Genesis Design Studio. Firstly, a new material was created using data found from (Kaw, 2006), with Boron Epoxy chosen for its better overall strengths:

Table 5-3 : Table of Material Properties from (Kaw, 2006)

Property	Symbol	Boron Epoxy	Units
Longitudinal modulus	E_1	204	GPa
Transverse Modulus	E_2	18.5	GPa
Major Poisson's Ratio	ν_{12}	0.23	
Shear Modulus	G_{12}	5.59	GPa
Density	ρ	1600	Kg.m ⁻³
Longitudinal Ultimate Tensile Stress	$(\sigma_1^T)_{ult}$	1260	MPa
Longitudinal Ultimate Compressive Stress	$(\sigma_1^C)_{ult}$	2500	MPa
Transverse Ultimate Tensile Stress	$(\sigma_2^T)_{ult}$	61	MPa
Transverse Ultimate Compressive Stress	$(\sigma_2^C)_{ult}$	202	MPa
In-plane Ultimate Shear Stress	$(\tau_{12})_{ult}$	67	MPa

As Design studio automatically assigns all elements of the mesh to a P-shell element group; A new element group, P-comp, was formed to allow the assignment of layers, and all elements from the old group were transferred into the new group, along with the material properties above. Once assigned, the theta vectors of the mesh were loaded to check the direction of what would be considered a 0° lay-up of fibres. It was found that the original set up was across the y-axis, therefore these were altered using the global coordinates to conform across the x-axis instead.

Alongside this, the displaying of the theta vectors showed any discrepancies and errors within the mesh itself, in which it was found there were several. Firstly, on the curved ends, it was found that one half of the generated mesh conformed well, whereas the other half showed major errors that would present later errors within testing. One Half of the vessel's elements were deleted, and the correct, properly aligned, elements were copied and translated around the x-axis by 180 degrees. From here, the grids along each section were merged to once again create a sealed unit.

Alongside this, there were some minor errors within the cylindrical section of the mesh. The whole section was removed from the study; And, using the edges of the spherical ends, new 5mm elements were hand drawn around the circumference. From here, the new elements were then copied across the entire length of the cylinder, until they matched the spherical section on the opposing side.

At a singular point on each end, SPC's (single point constraints), were set to hold the vessel during the analysis. One side was set with all 6 major Degrees of Freedom held (1,2,3,4,5,6), and the other was set with 4 (1,2,3,6) , to prevent the vessel from 'moving' outside of the remit of the test and retrieving erroneous results. Once complete, a pressure of 750,000 Pa was assigned to the inside of the vessel, to simulate the difference between the 8.5 bar interior and ~1bar atmospheric exterior.

A Load case was then created, this compiled all the settings described above into a singular analysis, including the SPC's and Pressure inputs, and then the vessel was run using the single analysis feature, and the results imported through the 'Post' feature, where the maximum Tsai Wu failure criterion was found.

Testing Parameters

The tests consisted of a symmetric laminate, in which each layer ran $[+\theta]$ throughout the whole thickness . The vessel was easily altered within the 'group properties' tab, which allowed the P-Comp to be edited after each run. The layer thickness was set to 0.2mm, and 12 layers were initially added at 30°. After each run, both the angle was changed, and every three runs more layers were added until all 12 tests had been completed.

5.2 Preliminary Design – Aluminum Inner Sleeve

Following on from the Composite design, an aluminum inner sleeve for the vessel was designed. The aluminum inner sleeve was necessary as it would better allow for future fixings to be added for input and output of the fuel as required.

5.2.1 Mathematical Analysis

Using the rearranged hoop stress equation in an isotropic material, along with the UTS of aluminium-3003, the minimum wall thickness necessary to hold 8.5bar internal pressure was found to be 0.865mm.

$$t = \frac{PR}{\sigma_H} \quad t = \frac{PR}{2.\sigma_L}$$

(5.27) isotropic thin cylinder equations

From here, the stress equations were used to find the hoop and longitudinal stresses acting across the cylinder at a varied thickness of 1mm from 1 up to 5mm. Then, using the Von-Mises yield Criterion, the approximated failure index could be found;

$$UTS \geq \sqrt{(\sigma_H^2 - (\sigma_H * \sigma_L) + \sigma_L^2)}$$

(5.28) Von Mises Yield Criterion (Meyers & Chawla, 2009)

5.2.2 Computational Modelling

The design was once again modelled through Solidworks, using similar techniques to the laminate cylinder. The cylinder was created, and a 2-D shell was taken from the inside surface. This shell was then meshed down to 5mm elements and exported as a NASTRAN file and imported into Design studio.

Table 5-4 : Table of Material Properties (AZOM, 2012)

Property	Symbol	Aluminium 3003	Units
Elastic Modulus	E_1	204	GPa
Shear Modulus	G_{12}	29	GPa
Poisson's Ratio	ν_{12}	0.33	
Density	ρ	7600	Kg.m ⁻³
Ultimate Tensile Stress	$(\sigma^T_1)_{ult}$	130	MPa
Ultimate Shear Stress	$(\tau_{12})_{ult}$	85	MPa

The material properties for Aluminium 3003 were input into the materials tab, and the elements were assigned to a P-shell group and the thickness of the shell was set to the required test thicknesses. The Model was once again held with single point constraints at either end. The required internal pressure was assigned, and all the components were input into a single load-case.

After each run, the material thickness was altered using the properties tab and the results processed through excel and compared to the mathematical results. From here the Von-Mises stress was divided by the UTS of the material to acquire the failure index of the vessel, with 1 being the failure point.

5.3 Preliminary Design - Combined Tests

The final Preliminary test combined the use of the aluminium sleeve with the carbon fibre outer wrap, the models were adjusted and computed, and the mathematical analysis was altered to verify the new model. From the results of the previous sections, it was found that a 3mm aluminium sleeve with a 3.6mm outer carbon wrap, aligned to 30° would be sufficient enough to provide respectable individual safety factors, whilst also reducing on material usage, this is discussed further in chapter [6].

5.3.1 Mathematical Analysis

The original code generated for test 1 was altered slightly to include the material properties of aluminium inner liner. From there, the code was shortened to only include a single angle input, as only one test was too be conducted within this test.

Firstly 2 new equations were needed to determine the stiffness matrix required for the aluminium inner liner :

$$P_{11} = \frac{E}{(1 + \nu)(1 - 2\nu)} * (1 - \nu)$$

$$P_{12} = \frac{E}{(1 + \nu)(1 - 2\nu)} * (\nu)$$

(5.29) isotropic stiffness matrix equations

As aluminium is an isotropic material, Q11 = Q22. The values within this matrix were then multiplied by the wall thickness of the liner. The new 'P' matrix values were then added into the corresponding [A] matrix (5.15) of the carbon fibre, before being inversed and the subsequent calculations performed as before.

$$[A] = \begin{bmatrix} A_{11} & A_{12} & A_{16} \\ A_{21} & A_{22} & A_{26} \\ A_{61} & A_{62} & A_{66} \end{bmatrix} + t \begin{bmatrix} P_{11} & P_{12} & 0 \\ P_{12} & P_{22} & 0 \\ 0 & 0 & 0 \end{bmatrix}$$

(5.30) combined stiffness matrix

Where t is the thickness of the aluminium layer.

By using this value to verify the outputs of the FE model, confidence with the accuracy and reliability of such results allow the project to progress into the final design stage. Once again, the MATLAB script generated for this purpose is available in appendix 2.

5.3.2 Computational Modelling

The aluminium Cylinder was designed in Solidworks, this time including a singular shaft on one end that would later become the output shaft of the tank. The cylinder was designed with 3m thick walls and the same dimensions used in previous tests. A base material was then added to the design, and a 5mm mesh generated throughout the cylinder. Once again, this was exported and imported into Genesis Design Studio.

In Genesis, the previously mentioned materials were created, and the elements of the aluminium tube were assigned to the respective material. From here, new elements were generated from the outer skin surface of the tube and assigned to a shell group labelled 'null'. The null is a shell group that is necessary for the interaction between the solid and the P-comp shell group, as 2-D Comp element groups will only react with shell elements. Although technically in their own group, the null elements are attached to the surface of the solid group, and therefore undergo the same forces and stresses experienced by the solid.

The shell elements were then assigned as the same material, and given thickness of 0.1mm, as this would be unlikely to impact the results due to providing nearly no addition stiffness. The elements were refined down to only include the elements on the outer most surface of the solid (where the carbon wrap would be in contact).

To generate the elements needed for the composite wrap, the null elements were then copied into another separate group, already set up with the Boron Epoxy material, set up to 3.6mm with a 30° fibre orientation.

The reason for taking this approach, instead of designing the carbon wrap in Solidworks is due to the mesh generation. It was found that, by making a surface offset in Solidworks and using that as a base for the carbon wrap, that the mesh didn't fully line up with that of the solid aluminium sleeve, along with having a small gap between the solid and shell elements. This caused problems in computing the design, as the software had to continuously move the elements around until it had 100% cohesion before it could compute the results. By using this method above, there were no gaps, and the meshes aligned perfectly, meaning that the data could be computed with 0 alignment iterations, lowering the final run time to under 3 minutes, in comparison to 45+ minutes and close to 100 iterations using the previous method.

Once again, constraints were added to either end of the cylinder, and the pressure was added to the element faces on the inside of the solid aluminium structure. This time contact data was necessary to compute all the data required. Therefore, the faces of the carbon wrap and the null elements

were each assigned to a contact group, these were then added into a contact pair, using a CF-aluminium friction coefficient of 0.23 (Schön, 2004). The data was then added to the load-case and the analysis was run.

5.4 Final Design Proposal

The final design was created using Solidworks, this time incorporating a fuel input shaft, along with an output shaft and aluminium holders. Due to the amount of additional influences on the design, and the main body of the tank remaining the same as the final preliminary test, Mathematical analysis of the design was considered both unachievable and an unnecessary use of limited time as previous testing had verified the FEA outputs for such a design. It is also important to note that at a stage such as this, it would be likely that real-world physical testing would commence to validate the FEA model.

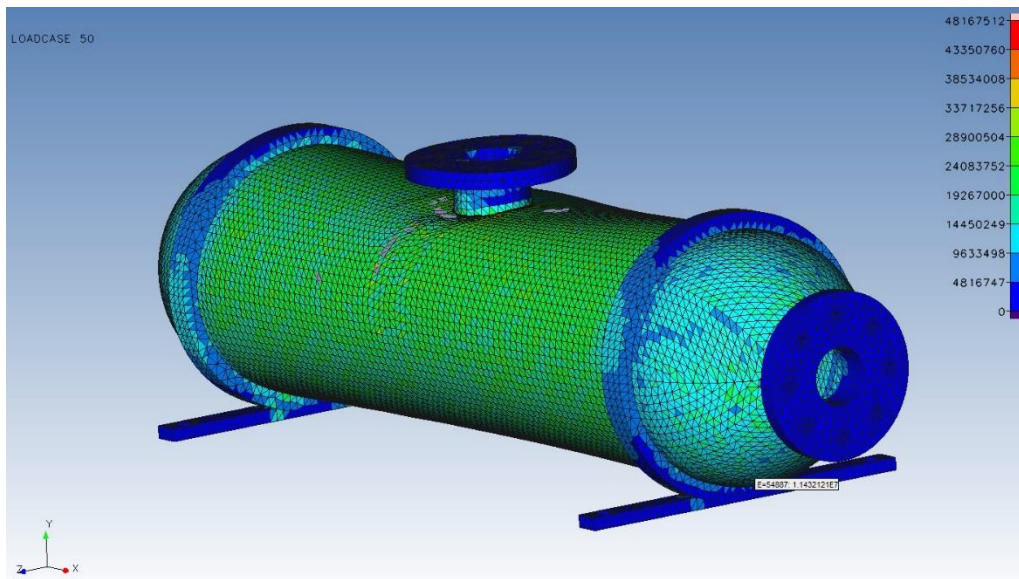
Firstly, the main body of the fuel tank was redesigned, using the same measurements as before, this time with a 70mm input and output shaft, with a 10mm thick wall, and end plates designed for a bolt and gasket attachment. A separate part was created for the exterior holders, designed as a 20mm thick x 30mm wide hoop that encompasses the entire tank to hold it in place within the vehicle storage space. The design would incorporate 2 of these at either ends of the cylindrical section of the tank using the Solidworks assembly feature. All 3 parts were then meshed as a single design and had a base material applied to them. The mesh was then exported to be used in Genesis.

As previously described, the 2-D surface elements were generated on the outer surface of the cylindrical tank, to be used as a null for interaction with the carbon fibre layer. These elements were then copied into a separate group and classified as Boron Epoxy with a 30°, 3.6mm layup as before. This time, a second null was generated on the inside surface elements of the aluminium holders, using aluminium and a 0.1mm thickness, as such with the tanks null elements.

The required internal pressure was applied, and constraints were added to the bottom of the holders and the output pipe exit. Contact elements were classified as the wrap, tank null, holder null, and the wrap elements in contact with the holder. These were then assigned into the 2 relevant contact pairings, using the friction coefficient of 0.23 as before. These pairings were then grouped into a single 'union of contact pair sets.' These were once again all grouped into a single load-case.

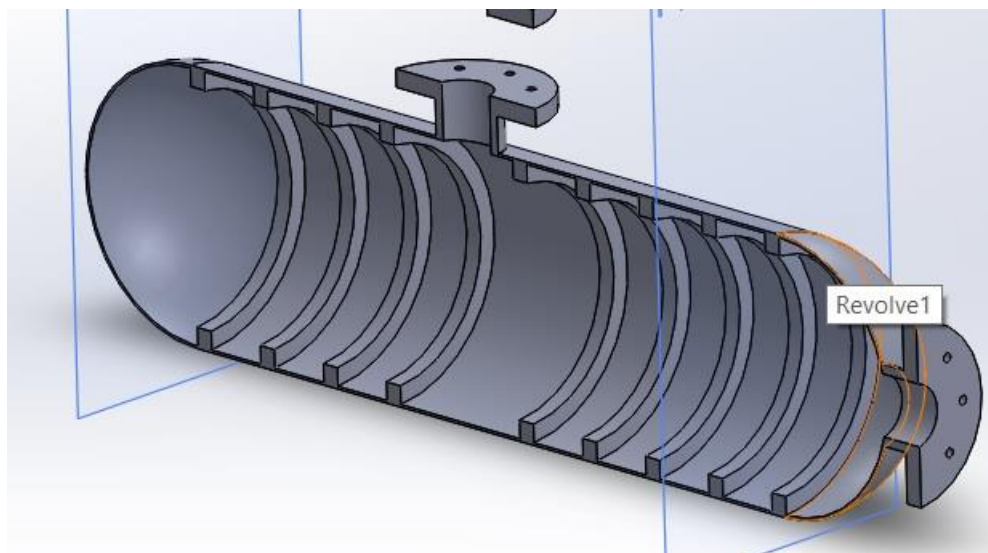
The model was then run, and it was found that the new input shaft, located at the top of the tank, provided too much excess weight, causing the design to deform and fail.

Figure 5-1 : Collapse of the vessel under excess weight



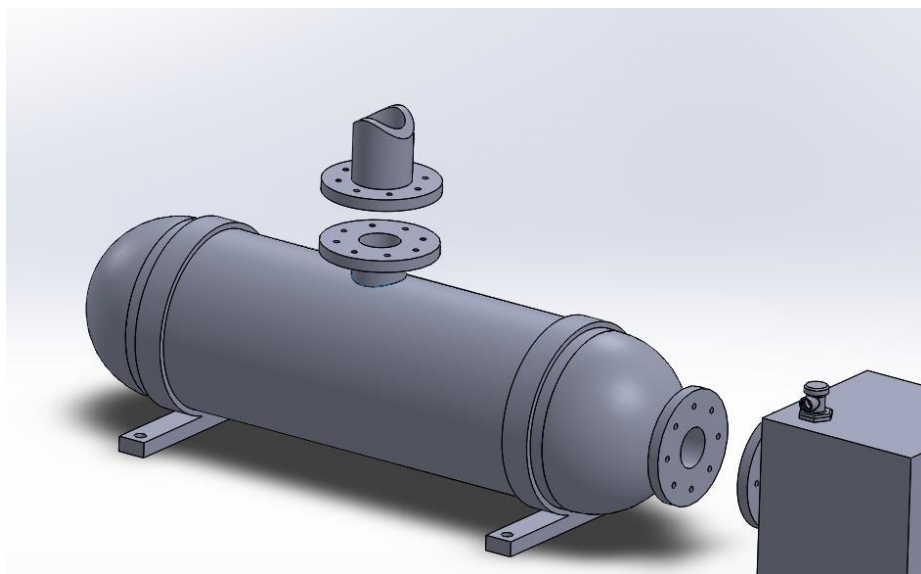
Therefore, the tank was redesigned, with the addition of equally spaced reinforced hoops within the fuel tank, either side of the input shaft to provide further structural integrity within the tank. The hoops were designed as a 20x20mm rings along the length of the cylindrical section to help provide extra strength to counteract the weight of the input shaft. The model was then re-meshed, and the steps above followed.

Figure 5-2 : Interior Reinforcement of the pressure vessel



A mock-up of the full design includes a rudimentary evaporation chamber to help highlight a significant safety feature required of the design. On the evaporation chamber, there is a one-way pressure release valve. This safety feature is there, for if in the event of a catastrophic accident, the vehicle housing the tank catches fire within the storage compartment. If such an event occurred, the rising temperature, and thus pressure within the vessel could potentially cause the vessel to explode. The release valve would trigger via the use of a temperature sensor, and slowly release the ammonia, which would now be in a gaseous state, into the storage compartment, where the fire would burn the majority of the harmful gas before it is released to the environment, and also prevent the vessel from eruption.

Figure 5-3 : Full design mock-up



6 Results and Analysis

It was decided through discussion, that in-house testing of composite pressure vessels would not be a viable option. Therefore, using the methods above, results were compared where possible between mathematical analysis and the FE model. This analysis will therefore examine the Verification process of the Verification & Validation technique when resolving FEA simulation.

6.1 Preliminary Results – Lamina

6.1.1 Buckling Results

The 3 types of buckling theory used displayed similar results and trends. In all 3 cases, the cylinder was likely to buckle during most runs, with the exception of the 6mm laminate. The data also showed that the Tokugawa equation tended to provided values between the other two.

Table 6-1 : Buckling Theory Results

n	n	Thickness	EL	ET	μ	Laminate Angle	E(x)	E(y)	Tokugawa eq		Timoshenko and Von Mises		Windenburg and Trilling	
									Pcr (x)	Pcr (y)	Pcr (y)	Pcr (y)	Pcr(x)	Pcr(y)
3.25714	3.2737	0.0024	204000	18500	0.23	30	57635	23851.19	2.6521	1.09752	2.78693913	1.15332448	3.01342007	1.24704953
3.25714	3.2737	0.0024	204000	18500	0.23	45	36115.7	36115.70	1.66188	1.66188	1.74637455	1.74637455	1.88829389	1.88829389
3.25714	3.2737	0.0024	204000	18500	0.23	70	19786.9	77604.65	0.9105	3.57101	0.95679508	3.75257303	1.03454915	4.05752633
2.94316	2.95812	0.0036	204000	18500	0.23	30	57635	23851.19	7.14202	2.9556	6.41911188	2.65643365	8.33131733	3.44776538
2.94316	2.95812	0.0036	204000	18500	0.23	45	36115.7	36115.70	4.47539	4.47539	4.02239629	4.02239629	5.22063808	5.22063808
2.94316	2.95812	0.0036	204000	18500	0.23	70	19786.9	77604.65	2.45195	9.61662	2.20377066	8.64324086	2.86025747	11.2179977
2.73892	2.75284	0.0048	204000	18500	0.23	30	57635	23851.19	14.3908	5.95538	11.6442239	4.81875202	17.1500825	7.09725227
2.73892	2.75284	0.0048	204000	18500	0.23	45	36115.7	36115.70	9.01769	9.01769	7.29659867	7.29659867	10.7467247	10.7467247
2.73892	2.75284	0.0048	204000	18500	0.23	70	19786.9	77604.65	4.94057	19.377	3.99762453	15.6787783	5.8878626	23.0923369
2.59031	2.60347	0.006	204000	18500	0.23	30	57635	23851.19	24.7469	10.2411	18.5265979	7.66689831	30.0335222	12.4288314
2.59031	2.60347	0.006	204000	18500	0.23	45	36115.7	36115.70	15.5071	15.5071	11.609288	11.609288	18.8198509	18.8198509
2.59031	2.60347	0.006	204000	18500	0.23	70	19786.9	77604.65	8.49596	33.3214	6.36043952	24.9457948	10.3109272	40.4397012

In all cases except one, the 6mm laminate provided critical buckling pressures above the required values 7.5bar, whereas at 4.8mm, the Tokugawa and Windenburg equations showed the lamina prevented buckling at 45°. The Von Mises equation seemed to underpredict the failure at any given thickness when compared the values generated from the others.

Unfortunately, due to the Genesis program being a linear solver, the ability to predict inelastic buckling was not possible, and therefore, the results above are present for reference, but are unable to be verified during this experiment.

6.1.2 Failure Results

The first test consisted of testing the lamina failure, where results were compared between the simulation runs, and the calculated values using classical lamination theory. A table of results can be found in appendix 3.

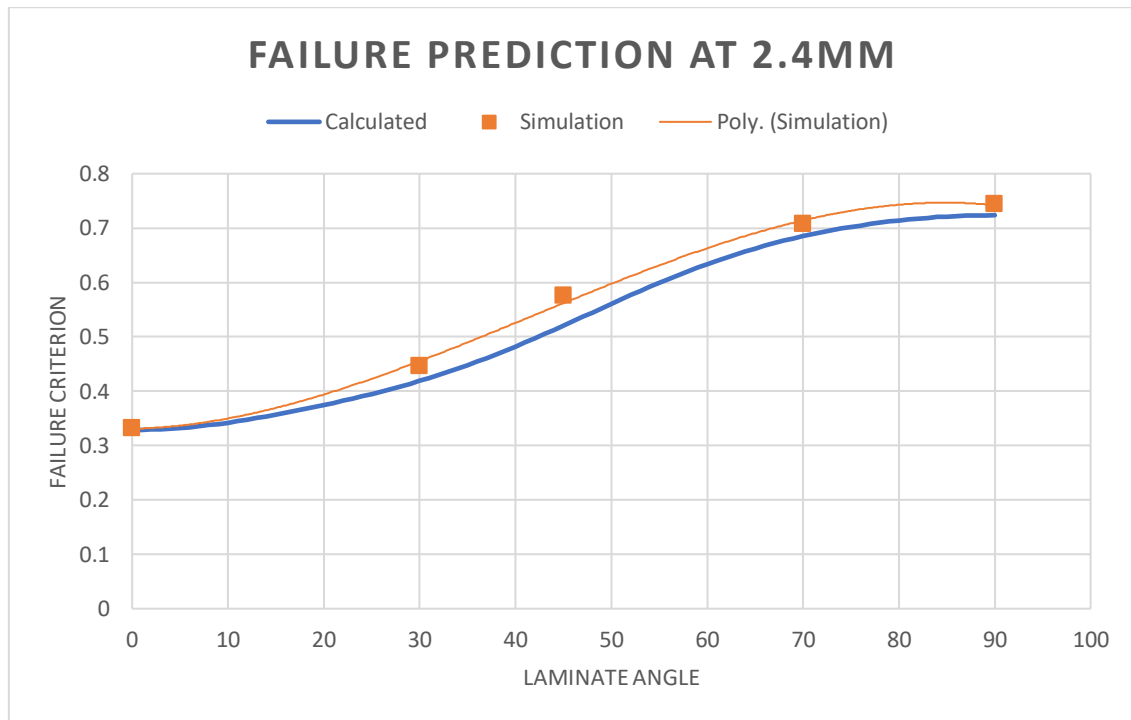


Figure 6-1: 2.4mm Failure Predictions

The data shows a good agreement between results at 2.4mm, with clear trends and low errors across all runs. As the data was calculated for every angle between 0° and 90°, a polynomial curve fit was used to join the simulated data. The strongest correlations show at either end of the graph, where there is little to no influence from the reduced stiffness that occurs through rotation of the lamina.

The range at which angle influences failure was found to be greatest at this thickness, with a failure index of 0.744 found at 90°, and 0.332 at 0°. These equate to safety factors between 1.34 and 3.01 respectively and a failure index range of 0.415. Such ranges could prove to be impractical, as slight errors in manufacturing could generate unintended consequences.

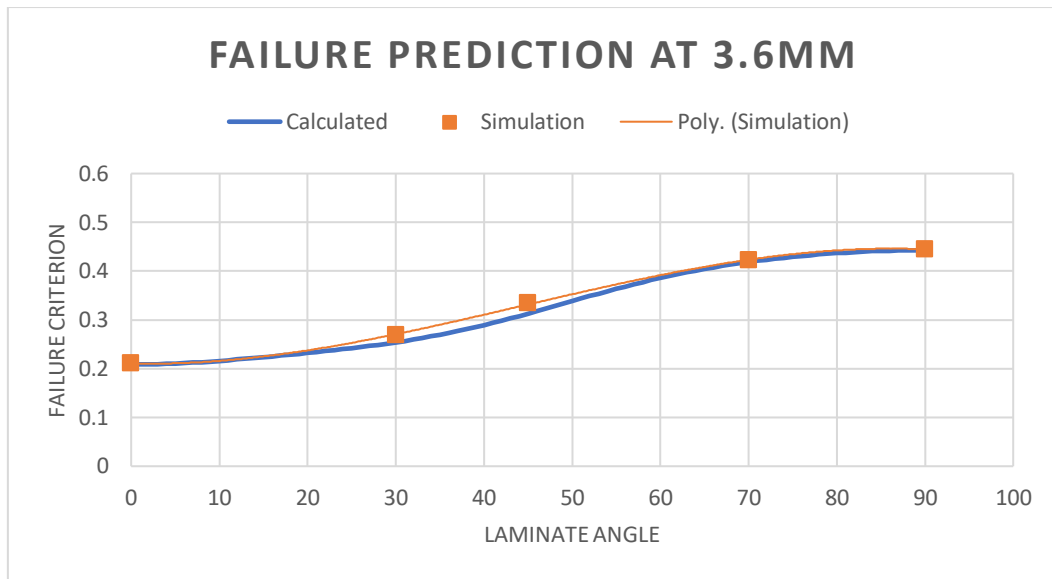


Figure 6-2: 3.6mm Failure Predictions

At 3.6mm, the data shows the strongest agreement between calculated and simulated testing. The largest error, found at 45°, is still considered to be in good agreement, with less than 10% error between readings, which accounts for a difference in predicted failure of 0.03. Failure indexes start at 0.445 at 90° and reduce to 0.211 at 0°, generating safety factors of 2.25 and 4.74; and a failure index range between 0 and 90 of 0.234.

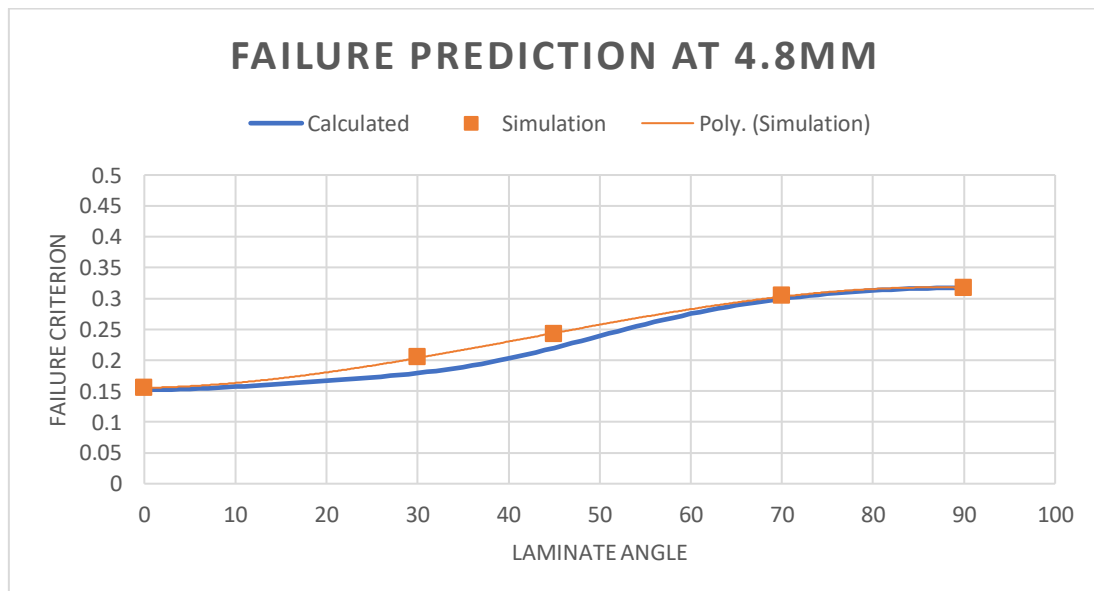


Figure 6-3: 4.8mm Failure Predictions

Once again, the data for the 4.8mm test show good agreement between simulated and calculated results. Clearly there is strong agreement at either end of the scale with errors accounting for less than 2% at 0°, 70° and 90°. However, that data does show some discrepancy closer to the middle, with errors at 30° reaching up to 12.6%.

It is also clear that, as thickness is increased, there is less dependency on fiber orientation to provide increased strength. The flatter curve lends to safety factors of 3.15 at 0° and 6.45 at 90°, and a failure index range of 0.162.

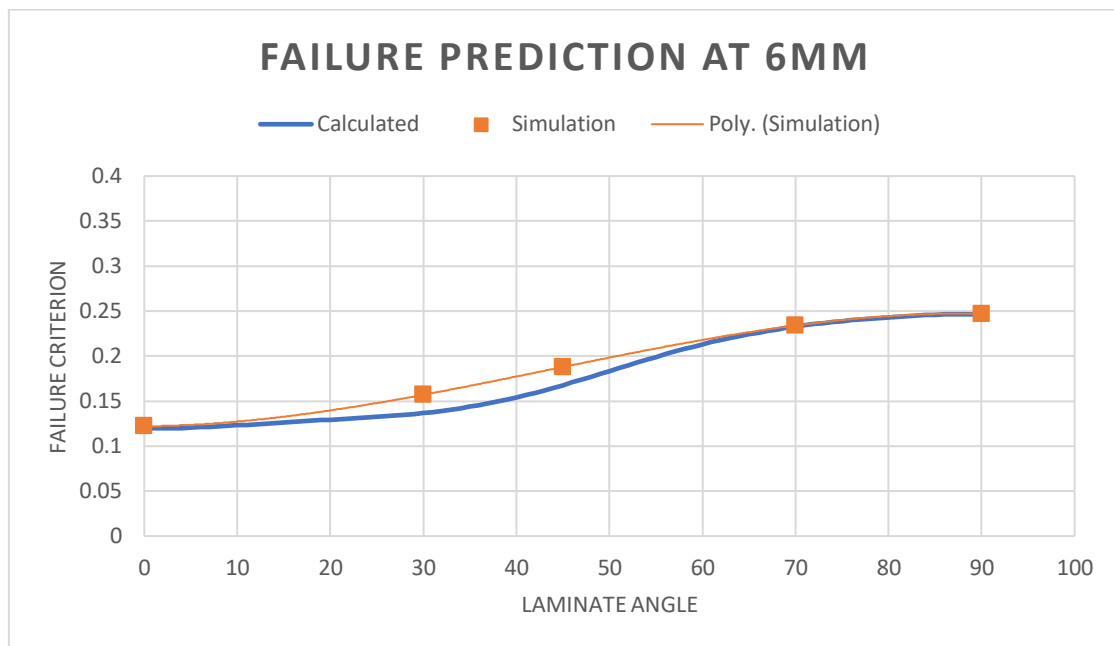


Figure 6-4: 6mm Failure Prediction

As with the previous tests, the same trends are visible, with highest errors being found in the 30°-45° range. However, at 6mm, the cylinder is nearly at the maximum thickness to classify as 'thin'. Therefore, it would be expected, that as the vessel neared such value, that the results would likely start to generate increased errors, as seen with the 13% errors found at 30°.

The greatest failure index of 0.247 was again found at 90°, with the lowest being 0.122 at 0°, creating a range of 0.125, and generating high safety factors of 4.05 and 8.20 respectively.

6.1.3 Discussion

Equations for these tests were generated from (Herakovich, 1998), using the classical theory of Laminates. Alongside this, further reading was found at (Gheshlaghi, et al., 2006), (Kaw, 2006) and (Rao, et al., 2012). It was found that there were several discrepancies in calculations between the 4 sources. Firstly, the Q stiffness matrix calculations showed minor differences between Herakovich and Gheshlaghi. Through testing, it was found that Herakovich's method provided the closest match to the simulation in terms of final failure index. To further this, another discrepancy found was in the calculations of the forces vector $\{N\}$, with Rao and Gheshlaghi using thickness as a component within the calculations, whereas Herakovich's and Kaw's methods both omit this step. Again, it was found that omission of the thickness at this stage of calculations provided a much more agreeable outcome with the simulation.

Originally first calculations towards this step were completed using Excel, however upon further review, it was found that several systematic errors were apparent. Using the transfer to MATLAB to calculate the results provided easier inspection of ongoing calculations, whilst also allowing values to be calculated across the full 90-degree range. The choice to proceed with this route allowed for greater accuracy and increased agreement in the results, however delayed further testing by a considerable amount. Whilst the delay was of major concern, the inspection of the calculations was necessary to ensure poor calculations would not impact further testing and causes excessive delays later.

6.2 Preliminary Results – Inner Sleeve

6.2.1 Results

The second test investigated the change in stress in an aluminum inner tube, and again, results were correlated between the FEA model and Mathematical analysis.

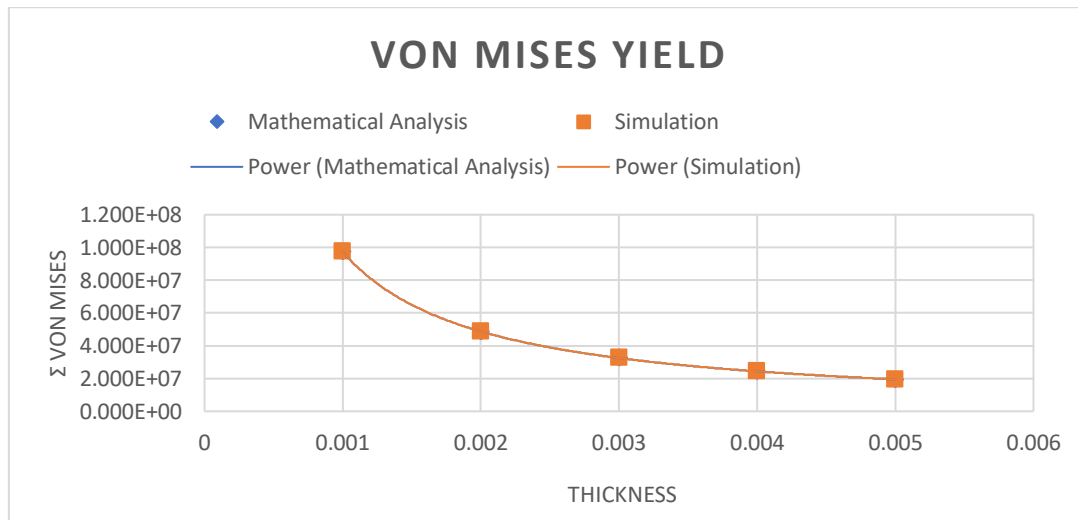


Figure 6-5: Von Mises Yield Criterion

The Von Mises Yield stress follows an expected trend, with stress following decaying at an exponential rate, as thickness increases. With twice thickness resulting in a reduction of stress by half. The results between the mathematical analysis and FE model showed perfect agreement, with the largest error resulting in 0.0002%. Therefore, there is full confidence in the FE model to perform as expected in this instance.

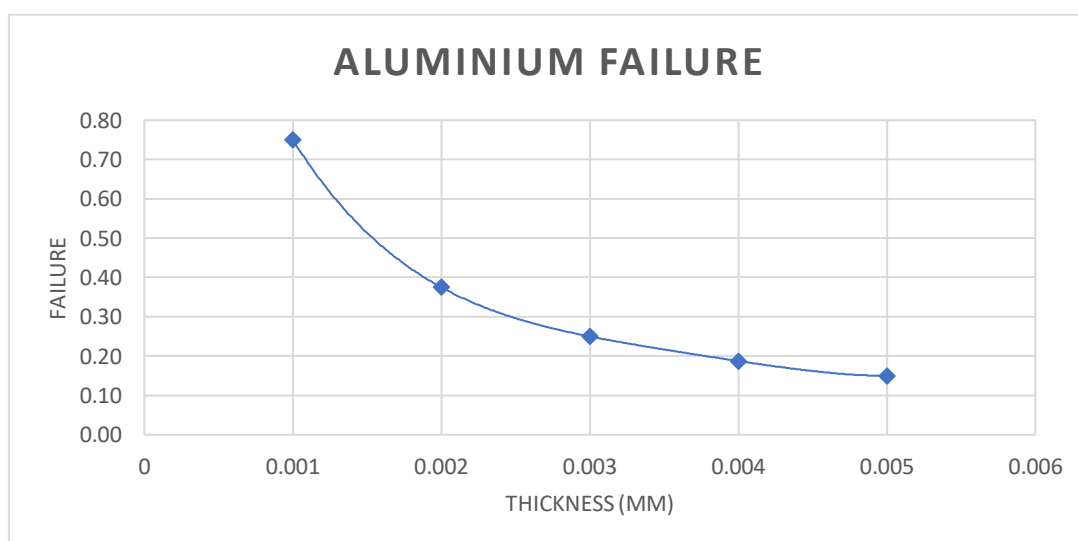


Figure 6-6 Aluminium Failure Predictions

The failure within the aluminum was found to follow the same trend as the above Criterion, with expected results. As the minimum thickness was found to be 0.8mm, it was expected to find that the 1mm thickness would be close to failure. With a thickness of 3mm providing an index of 0.25, and therefore a safety factor of 4, it was decided that this would be the ideal thickness to use to both reduce material usage, as well as keeping the tank safe.

6.3 Preliminary Results – Combined

6.3.1 Results

The final preliminary test, consisting of an aluminum inner tube of thickness 3mm, and a boron epoxy wrap of 3.6mm at an angle incline of 30° was conducted. To verify the model, the failure index in the outer layer of the carbon composite wrap was calculated.

Table 6-2 : Combined test results

property	value
Aluminium Thickness (mm)	3
Wrap Thickness (mm)	3.6
Angle (deg)	30
Failure (CF) [sim]	0.0536
Failure (CF) [calc]	0.0525
error %	2.05%
von mises [sim]	2.01E+07
Aluminium failure	0.155

The test shows that the combination of both materials drastically reduces the stresses observed in both materials. The aluminum layer, as expected, experiences less of a reduction due to it being the main load bearing layer, however the failure index is still reduced by 0.1, increasing its safety factor to 6.45. The carbon fiber outer wrap experiences a much larger reduction in forces, with the failure index reducing by 0.215, raising its safety factor to 18.66.

The results are in very good agreement, with an error of only 2.05%, therefore the results can be considered accurate and reliable, and the FE model can be considered verified by testing procedures.

Discussion

Testing at this stage ran smoothly, with a minor problem with the calculations not computing properly within MATLAB. However, this problem was rectified, and a good agreement was found between results.

Whilst the result found was accurate, the reliability of the test is somewhat questionable due to the absence of further runs. Due to limited access of the software, further runs at different angles to test the reliability of the equations was not possible. However, due to past testing of the individual sections with little-to-no alterations, it can be said that the obtained from the FE model is likely highly reliable, although further tests would be beneficial to prove this theory.

6.4 Final Design Proposal

6.4.1 Results

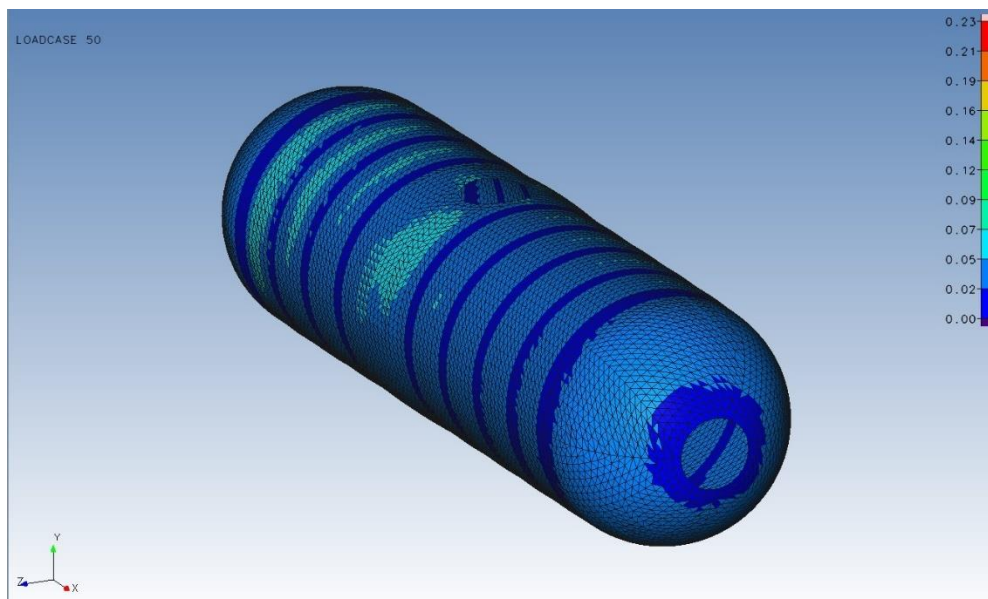


Figure 6-7 : Composite layer failure distribution

Once the model had been adjusted due to overloading from the input shaft, the new model was run again, and results were obtained from 3 separate sections of the model. The first section observed, were the stresses acting around the input shaft area as this was clearly going to be causing a stress concentration. The second area would be the unsupported sections, defined as the spaces between the inner aluminium rings. The final section observed would be the area where the new aluminium inner rings would provide increased strength to the model.

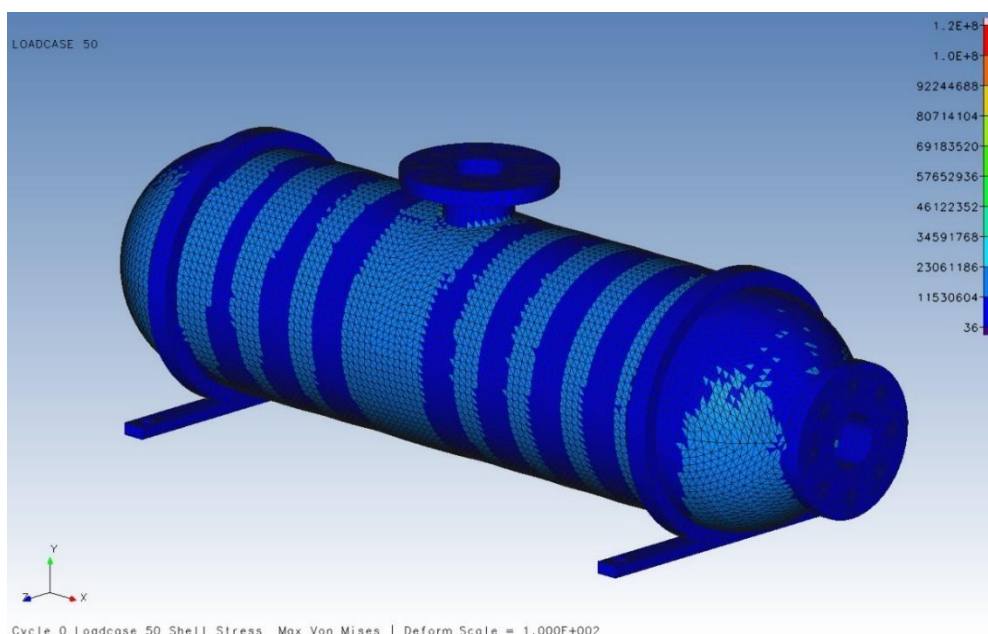


Figure 6-8 : Stress distributions in the aluminium structures

Table 6-3 : Final Design results

Area	Angle	Aluminium	Support	Wrap	max von	Failure index		safety factors	
Observed		Thickness	thickness	Thickness	mises	Carbon	Aluminium	Carbon	Aluminium
		(mm)	mm	(mm)					
near entrance					2.03E+07	0.059	0.156	16.95	6.41
unsupported sections	60/60	3	20x20	3.6	1.63E+07	0.051	0.125	19.61	7.98
Supported Sections					1.48E+07	0.021	0.114	47.62	8.77

The final design provided 3 sets of results at critical sections of the vessel. As expected, the weakest section of the design is near to the opening, however, providing safety factors of 6.41 in the aluminium layer, and 16.95 in the carbon layer, it can be determined that the vessel is more than satisfactory in regards to being able to safely store the liquid ammonia at the required pressure.

With the unsupported sections providing results similar in magnitude, albeit slightly increased, to the previous test, and the supported sections being the 'safest' sections, it can be concluded that the design is more than capable of delivering the required performance expected from it.

6.4.2 Discussion

The Final tests proved to be a success. The original tank was found to fail, not from the interior pressure, but under the weight of the input shaft. This was quickly counteracted using interior stiffeners made from aluminium to provide extra strength.

From this point, the vessel reacted as expected to the loading, with increased failure around the input shaft, with the lowest failure found around the stiffened sections, providing an overall performance similar to that of the combined testing. The tank held the pressure whilst keeping very good factors of safety.

Further design and testing of the tank could include a protective outer barrier for tank to provide a safety net in the event of an impact, as well as impact tests being conducted on the vessel itself to see its reaction in the event of an accident. Alongside this, tests with impurities and 'manufacturing defects' could be included, as lamina theory assumes the use of perfectly bonded layers, to fully understand all potential real-world outcomes. Included in these tests would be physical tests performed to weigh up the validity of the FE model results in relation to real life.

7 Pandemic Impact

The major impact of the recent pandemic was the access to the applications and software necessary for the completion of this thesis. Whilst trial periods for the software were obtainable through industry contact, a lead time of several weeks disrupted the potential for earlier testing. Following this, the limited access fell during term time, when potential help with designs and simulation would have been helpful in speeding up the processes. Despite this, testing was still obtainable, albeit at a reduced pace than would have been preferred.

Another impact would be in the form of reduced access to learning spaces. With the removal of access to the library, not only were physical sources now unobtainable, but the learning space itself. The space was usually used to complete work in a zone with little to no distractions, and allowed ideas, concepts and written words to flow much more readily than elsewhere.

The final noticeable impact would be in the form of mental health. Packing up and moving back home (even if only a 5-hour drive) in a moment's notice provided extra stress, along with the reduced ability of outside activities, lead to a noticeable reduction in mental capacity. It was found that during these times, the ability to focus solely on necessary work was reduced, which further ties into the above point of the loss of neutral spaces.

In the Final model, it would have been beneficial to include inherent weaknesses generated through the processing and manufacturing and the materials, such as welded aluminum joins and bolted sections for the holders. However, with limited online resources for GENESIS, and a lack of in-person 1-to-1 guided design, these features were unable to be included in the design.

8 Conclusions

The project has highlighted the advantages and problems of using hydrogen as an alternative fuel. With the costs and safety considerations of storing hydrogen at high pressure, the use of an ammonia-to-hydrogen based fuel delivery system has been investigated and has found that:

- The reformation of ammonia currently has similar well-to-wheel emissions and efficiencies when compared to hydrogen, however green alternatives are becoming increasingly available.
- With all safety factors considered, ammonia has the potential to deliver equivalent range and economy when compared to hydrogen-based vehicles.
- The storage of ammonia is a much easier process to achieve, and in turn, has lower costs associated to distribution and storage, including the usability of existing infrastructure available.
- The Costs associated with ammonia could theoretically produce lower refuelling costs for the consumer, thus generating more consumer interest in alternative fuel vehicles, however it is unknown how such costs would scale with increased demand and usage.
- Ammonia is viable option for hydrogen-based fuelling, provided strict safety measures and controls are imposed.

The Fuel Tank was designed with a combination of material usage and safety as the major considerations, and therefore, a minimum safety factor of 2.5 was required from individual sections to ensure optimum performance, and allowance for both design flaws and pressure fluctuations. Through testing it was found:

- Separate analysis of the carbon fibre provided good agreement between calculated results and the FE model, with the majority of results displaying errors less than 10%.
- Testing of the composite shell found that a thickness of 2.4mm would not be sufficient, as a large portion of angle range did not conform to the 2.5 minimum safety factor. Whereas at 3.6mm, the minimum safety factor was found to 2.25, however with the correct angle lay, would provide a more than satisfactory SF.
- It was Chosen that, to reduce material usage, a Carbon Fibre layer of 3.6mm, with a layup of 30 degrees would provide a more than acceptable individual SF of 3.72.
- The optimal Aluminium thickness was found to be a 3mm inner liner, yielding a SF of 4.
- The combined test provided a strong agreement between the FE model and the mathematical analysis, with a small error of 2.05%, it can be said at this point that the FE model had been verified.

- Combined tests provided good analysis of interaction between the carbon fibre and aluminium layers, the combination of which drastically reduced failure within the carbon layers, whilst also reducing the failure in the aluminium load-bearing layer.
- It was found that the carbon composite layer showed a SF of 18.66, with the aluminium displaying a new SF of 6.45.
- The Final Design initially incurred failure, however this was found to be induced by weight of exterior fittings, therefore the introduction of interior stiffeners found to both resolve the issue and provide slightly enhanced performance.
- The final design was found to have safety factors in the composite layer ranging between 16.95 and 47.62, with the aluminium exhibiting SF's in the range of 6.41 to 8.77.

Following this, the inclusion of safety features such as the limited pressure release valve that would be located on the evaporation chamber, a maximum fill sensor/switch to prevent overfilling and the necessary temperature and leak sensors located within the storage compartment of the vehicle would allow such a tank to perform to the optimum standard; And opens the potential for Ammonia based fuel delivery systems to become a possibility in the future conversion to alternative green fuel vehicles.

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10 Appendix

Appendix 1 – Lamina Theory Test

```
clc
clear
%-----inputs-----
Variables = inputdlg({'Radius (m):','Pressure (Mpa):'}, 'input variables',[1 42]);
R = str2double(Variables{1});
P = str2double(Variables{2});
L(1) = R*(1/20)*1000;
L(2) = 0;
Variables2 = inputdlg(['input thickness(mm) min:', num2str(L(2)) 'mm', ' and max: ', num2str(L(1)) 'mm']);
t = str2double(Variables2);
t=t/1000;
n = t*5000;
h = t/((t*10000)/2);
l = t/2;

ES = 75000
vS = 0.33
ppp = 3/1000
P11 = ((ES/((1+vS)*(1-(2*vS))))*(1-vS))*ppp
P12 = ((ES/((1+vS)*(1-(2*vS))))*(vS))*ppp
P16 = ((ES/((1+vS)*(1-(2*vS))))*(vS))*ppp
P22 = ((ES/((1+vS)*(1-(2*vS))))*(1-vS))*ppp
P26 = ((ES/((1+vS)*(1-(2*vS))))*(vS))*ppp
P66 = ((ES/((1+vS)*(1-(2*vS))))*(1-vS))*ppp

PS = [P11 P12 P16; P12 P22 P26; P16 P26 P66]

%-----inputs-----
%-----Material-----
E1 = 204000;
E2 = 18500;
G12 = 5590;

T1 = 1260;
T2 = 61;
C1 = 2500;
C2 = 202;
T12 = 67;
v12 = 0.23;
v21 = (E2/E1)*v12;
%-----Material-----
```

```

%-----Constitutive Eqs-----
for j = 1:91
    for x = 1:91
        for i = 1:91
            b = [0:90];
            b = b';
            Bb = deg2rad(b(i));
            Gxy(i) =
E1/(((4*(sin(Bb).^2)*(cos(Bb).^2))*(1+(2*v12)+(E1/E2)))+(
((sin(Bb).^2)-(cos(Bb).^2))^2*(E1/G12)));
            vxy(i) = (-
((sin(Bb).^2)*(cos(Bb).^2)*(1+(E1/E2)-(E1/G12))-
((cos(Bb).^4)+(cos(Bb).^4)*v12)/(cos(Bb).^4+((cos(Bb).^
2)*(sin(Bb).^2))*(-
2*v12+(E1/G12))+(sin(Bb).^4*(E1/E2))));
            Ey(i) =
E1/(sin(Bb).^4+(((cos(Bb).^2)*(sin(Bb).^2))*(-
2*v12+(E1/G12))+(cos(Bb).^4*(E1/E2))));
            Ex(i) =
E1/(cos(Bb).^4+(((cos(Bb).^2)*(sin(Bb).^2))*(-
2*v12+(E1/G12))+(sin(Bb).^4*(E1/E2))));
            end
            q11(x) = E1/(1-(v12*v21));
            q12(x) = (E2*v12)/(1-(v12*v21));
            q22(x) = E2/(1-(v12*v21));
            q66(x) = G12;
        end
        q11 = q11';
        q12 = q12';
        q22 = q22';
        q66 = q66';

        A= deg2rad(b(j));
        Q11(j) =
(q11(j)*cos(A)^4)+(2*(q12(j)+2*q66(j))*(cos(A)^2)*(sin(A)
^2))+(q22(j)*sin(A)^4);
        Q22(j) =
(q11(j)*sin(A)^4)+(2*(q12(j)+2*q66(j))*(cos(A)^2)*(sin(A)
^2))+(q22(j)*cos(A)^4);
        Q12(j) = ((q11(j)+q22(j)-
(4*q66(j)))*(sin(A)^2)*(cos(A)^2))+(q12(j)*((sin(A)^4)+(c
os(A)^4)));
        Q66(j) = ((q11(j)+q22(j)-(2*q12(j))-
(2*q66(j)))*(sin(A)^2)*(cos(A)^2))+(q66(j)*(sin(A)^4+cos(
A)^4));
    end
end

```

```

        Q16(j) = ((q11(j)-q12(j)-
(2*q66(j)))*(sin(A))*cos(A)^3))+((q12(j)-
q22(j)+(2*q66(j)))*(sin(A)^3)*cos(A));
        Q26(j) = ((q11(j)-q12(j)-
(2*q66(j)))*(sin(A)^3)*cos(A))+((q12(j)-
q22(j)+(2*q66(j)))*(sin(A)*cos(A)^3)));
    end
    for X = 1:91
        Nx(X) = (P*R);
        Ny(X) = ((P*R)/2);
        Nxy(X) = 0;
        No{X} = [Nx(X);Ny(X);Nxy(X)];
    end
    for q = 1:91
        Q{q} = [Q11(q) Q12(q) Q16(q); Q12(q) Q22(q) Q26(q);
Q16(q) Q26(q) Q66(q)];
        A1{q} = Q{1,q}*(n*h)+PS;
        C{q} = A1{1,q};
        C3{q} = inv(C{1,q});
        e{q} = C3{1,q}*No{1,q};
        %-----K-----
        kx{q} = 0;
        ky{q} = -(e{1,q}(2,1)/R);
        kxy{q} = (e{1,q}(3,1)/R);
        K{q} = [kx{1,q};ky{1,q};kxy{1,q}];
        %-----K-----
        ex{q} = e{1,q}(1,1);
        ey{q} = e{1,q}(2,1)+(1*(K{1,q}(2,1)));
        exy{q} = e{1,q}(3,1)+(1*(K{1,q}(3,1)));
        E{q} = [ex{1,q};ey{1,q};exy{1,q}];
    end
end
%-----Constitutive Eqs-----
for y = 1:91
    Sij{y} = Q{1,y}*E{1,y};
    S1{y} = Sij{1,y}(1,1);
    S2{y} = Sij{1,y}(2,1);
    S3{y} = Sij{1,y}(3,1);
    Aa= deg2rad(b(y));
    Sx{y} = (S1{1,y}*cos(Aa)^2)+(S2{1,y}*sin(Aa)^2)-
(2*sin(Aa)*cos(Aa)*S3{1,y});
    Sy{y} =
(S2{1,y}*cos(Aa)^2)+(S1{1,y}*sin(Aa)^2)+(2*sin(Aa)*cos(Aa)
)*S3{1,y});
    Sxy{y} = (S1{1,y}*cos(Aa)*sin(Aa))-
(S2{1,y}*cos(Aa)*sin(Aa))+((cos(Aa)^2+sin(Aa)^2)*S3{1,y})
;

```

```

        S{y} = [Sx{1,y};Sy{1,y};Sxy{1,y}]
    end
    %-----Tsai-Wu-----
    H1 = (1/T1)-(1/C1);
    H11 = (1/(T1*C1));
    H2 = (1/(T2))-(1/(C2));
    H22 = (1/(T2*C2));
    H6 = 0;
    H66 = (1/(T12^2));
    H12 = -(1/(2*2500*1260));
    for q = 1:91
        q
        F{q} =
        ((S{1,q}(1,1))*H1)+(H2*(S{1,q}(2,1)))+(S{1,q}(3,1))*H6)+
        (((S{1,q}(1,1))^2)*H11)+(((S{1,q}(2,1))^2)*H22)+((((S{1,q}
        }(3,1))^2)*H66))+(2*H12*(S{1,q}(1,1)*(S{1,q}(2,1))));
    end
    F = cell2mat(F);

    figure(1)
    plot(b,Sx)
    hold on
    plot(b,Sy)
    hold off
    legend('stress-x','stress-y')

    figure(2)
    plot(b,F)
    legend('Failure')

    xlswrite('failure_data.xlsx',F)

    figure(3)
    plot(b,Ex)
    hold on
    plot(b,Ey)
    plot(b,Gxy)
    hold off
    legend('Ex','Ey','G12')

    figure(4)
    plot(b,vxy)
    legend('vxy')

    figure(5)
    plot(b,Q11)
    hold on

```

```
plot(b,Q22)
plot(b,Q66)
hold off
legend('Q11','Q22','Q66')

figure(6)
plot(b,Q12)
hold on
plot(b,Q16)
plot(b,Q26)
hold off
legend('Q12','Q16','Q26')
```

Appendix 2 – Combined Stress Test

```
clc
clear
%-----inputs-----
Variables = inputdlg({'Radius (m):','Pressure (Mpa):','Angle:'}, 'input variables',[1 42]);
R = str2double(Variables{1});
P = str2double(Variables{2});
A = str2double(Variables{3});
L(1) = R*(1/20)*1000;
L(2) = 0;
Variables2 = inputdlg(['input CF thickness(mm) min:', num2str(L(2)) 'mm', ' and max: ', num2str(L(1)) 'mm']);
Variables3 = inputdlg(['input Aluminium thickness(mm) min:', num2str(L(2)) 'mm', ' and max: ', num2str(L(1)) 'mm']);
t = str2double(Variables2);
ppp = str2double(Variables3);
t=t/1000;
n = t*5000;
h = t/((t*10000)/2);
l = t/2;
ppp = ppp/1000;
Bb = deg2rad(A);

ES = 75000;
vS = 0.33;
UTS = 130;
USS = 83;

P11 = ((ES/((1+vS)*(1-(2*vS))))*(1-vS))
P12 = ((ES/((1+vS)*(1-(2*vS))))*(vS))
P22 = ((ES/((1+vS)*(1-(2*vS))))*(1-vS))

%-----inputs-----
%-----Material-----
E1 = 204000;
E2 = 18500;
G12 = 5590;

T1 = 1260;
T2 = 61;
C1 = 2500;
C2 = 202;
T12 = 67;
v12 = 0.23;
```

```

v21 = (E2/E1)*v12;
%-----Material-----
%-----Constitutive Eqs-----
Gxy =
E1/(((4*(sin(Bb).^2)*(cos(Bb).^2))*(1+(2*v12)+(E1/E2)))+(
(((sin(Bb).^2)-(cos(Bb).^2))^2)*(E1/G12)));
vxy = (-((sin(Bb).^2)*(cos(Bb).^2)*(1+(E1/E2)-(E1/G12))-
((cos(Bb).^4)+(cos(Bb).^4))*v12)/(cos(Bb).^4+(((cos(Bb).^
2)*(sin(Bb).^2))*(-
2*v12+(E1/G12))+(sin(Bb).^4)*(E1/E2))));
Ey = E1/(sin(Bb).^4+(((cos(Bb).^2)*(sin(Bb).^2))*(-
2*v12+(E1/G12))+(cos(Bb).^4)*(E1/E2)));
Ex = E1/(cos(Bb).^4+(((cos(Bb).^2)*(sin(Bb).^2))*(-
2*v12+(E1/G12))+(sin(Bb).^4)*(E1/E2)));

q11 = E1/(1-(v12*v21));
q12 = (E2*v12)/(1-(v12*v21));
q22 = E2/(1-(v12*v21));
q66 = G12;

q11 = q11';
q12 = q12';
q22 = q22';
q66 = q66';

Q11 =
((q11*cos(Bb)^4)+(2*(q12+2*q66)*(cos(Bb)^2)*(sin(Bb)^2))+
(q22*sin(Bb)^4));
Q22 =
((q11*sin(Bb)^4)+(2*(q12+2*q66)*(cos(Bb)^2)*(sin(Bb)^2))+
(q22*cos(Bb)^4));
Q12 = (((q11+q22-
(4*q66))*(sin(Bb)^2)*(cos(Bb)^2))+(q12*((sin(Bb)^4)+(cos(
Bb)^4))));
Q66 = (((q11+q22-(2*q12)-
(2*q66))*(sin(Bb)^2)*(cos(Bb)^2))+(q66*(sin(Bb)^4+cos(Bb)
^4)));
Q16 = (((q11-q12-(2*q66))*(sin(Bb))*(cos(Bb)^3))+(q12-
q22+(2*q66))*(sin(Bb)^3)*(cos(Bb)));
Q26 = (((q11-q12-(2*q66))*(sin(Bb)^3)*(cos(Bb))+(q12-
q22+(2*q66))*(sin(Bb)*(cos(Bb)^3))));

Q11 = Q11';
Q12 = Q12';
Q16 = Q16';
Q22 = Q22';
Q26 = Q26';

```

```

Q66 = Q66';

Nx = (P*R);
Ny = ((P*R)/2);
Nxy = 0;
No = [Nx;Ny;Nxy];

Q = [Q11 Q12 Q16; Q12 Q22 Q26; Q16 Q26 Q66];
QP = [Q11*(n*h)+P11*ppp Q12*(n*h)+P12*ppp Q16*(n*h);
Q12*(n*h)+P12*ppp Q22*(n*h)+P22*ppp Q26*(n*h); Q16*(n*h)
Q26*(n*h) Q66*(n*h)];
C = QP;
C3 = inv(C);
e = C3*No;
%-----K-----
kx = 0;
ky = -(e(2,1)/R);
kxy = (e(3,1)/R);
K = [kx;ky;kxy];
%-----K-----
ex = e(1,1);
ey = e(2,1)+(1*(K(2,1)));
exy = e(3,1)+(1*(K(3,1)));
E = [ex;ey;exy];

%-----Constitutive Eqs-----

Sij = ((Q))*E;
S1 = Sij(1,1);
S2 = Sij(2,1);
S3 = Sij(3,1);

Sx = (S1*cos(Bb)^2)+(S2*sin(Bb)^2)-
(2*sin(Bb)*cos(Bb)*S3);
Sy =
(S2*cos(Bb)^2)+(S1*sin(Bb)^2)+(2*sin(Bb)*cos(Bb)*S3);
Sxy = (S1*cos(Bb)*sin(Bb))-
(S2*cos(Bb)*sin(Bb))+((cos(Bb)^2+sin(Bb)^2)*S3);
S = [Sx;Sy;Sxy];

%-----Tsai-Wu-----
H1 = (1/T1)-(1/C1);
H11 = (1/(T1*C1));
H2 = (1/(T2))-(1/(C2));
H22 = (1/(T2*C2));
H6 = 0;

```

```

H66 = (1/(T12^2));
H12 = -(1/(2*2500*1260));

F =
((S(1,1))*H1)+(H2*(S(2,1)))+(S(3,1))*H6+(((S(1,1))^2)*H
11)+(((S(2,1))^2)*H22)+((((S(3,1))^2)*H66))+(2*H12*(S(1,1)
)*(S(2,1))));

W = msgbox(['failure index:', num2str(F)]);
W

```

Appendix 3 – Lamina Theory test results

mm	deg	Calculated Tsai Wu	Simulation Tsai Wu	% error	safety
2.4	0	0.329	0.332	1.0%	3.01205
	30	0.419	0.446	6.0%	2.24215
	45	0.521	0.575	9.5%	1.73913
	70	0.685	0.707	3.1%	1.41443
	90	0.724	0.744	2.7%	1.34409
3.6	0	0.209	0.211	1.2%	4.73934
	30	0.254	0.269	5.6%	3.71747
	45	0.313	0.334	6.3%	2.99401
	70	0.419	0.422	0.8%	2.36967
	90	0.443	0.445	0.5%	2.24719
4.8	0	0.152	0.155	1.8%	6.45161
	30	0.179	0.205	12.6%	4.87805
	45	0.220	0.242	9.1%	4.13223
	70	0.300	0.304	1.5%	3.28947
	90	0.317	0.317	0.1%	3.15457
6	0	0.119	0.122	2.1%	8.19672
	30	0.137	0.157	12.9%	6.36943
	45	0.168	0.188	10.8%	5.31915
	70	0.233	0.234	0.5%	4.2735
	90	0.247	0.247	0.1%	4.04858