DESIGN AND ANALYSIS OF THE NATURAL GAS STORAGE TANK FOR AUTOMOBILES

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by
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CHAPTER 1
INTRODUCTION

1-1 Introduction

Environmental problems, such as global warming and acid rain, are rampant on a
global scale, but one of the worst problems is the air pollution in major cities and
industrial towns due to nitrogen oxides (NOx), carbon monoxide (CO), soot, dust and
black smoke, which is causing an increasing demand by society for the reduction of these
pollutants.

On the other hand, our dependence on foreign oil has reached the highest level in
more than 30 years. Even though we have improved the quality and level of automobile
exhaust emissions, the growth in the number of vehicles on the road and the increase in
mileage driven per year pose threats to air quality in most major cities. Therefore, the
widespread introduction of natural gas powered vehicles promises to reduce our oil
dependence and give us cleaner air.

At present, there are over 30,000 compressed natural gas (CNG) vehicles
operating in the United State, but despite low fuel costs, reduced engine maintenance and
significant reduction of exhaust emissions, CNG use has been restricted to fleets of
trucks, delivery vans and buses. This limitation is due largely to the modest driving range
/about 150 miles) and lost truck bed or trunk space. Another problem inhibiting natural
gas vehicle (NGV) production is the high cost of on-board storage cylinders. Because of
these drawbacks, there is a need for further optimization of the developed NGV.
1-2 Objective

NGVs are not suitable for long distance driving compared to gasoline or diesel powered vehicle. The purpose of this thesis, therefore, is to develop a natural gas tank that can endure higher pressure, weigh and cost less, approach an actual shape--irregular and non-symmetrical type--and store more natural gas in order to increase NGV driving distance.

By using Intergraph and Patran finite element packages, Chen [1], a graduate student in Mechanical Engineering at Ohio University, developed his models especially focused on rectangular and cylindrical tanks. Then he looked into the relationship between stress and geometry variations, such as wall thickness, fillets, fins and a fin with a hole. He used three different materials: Aluminum-6061-T6, Steel-AISI-1040 and composite material. He attained his final model--rectangular shape with fins and a hole--by using laminate material (Kevlar/Steel-AISI-1040), which can endure internal pressure at 15000 psi (higher than an ordinary natural gas tank at 3500 psi).

This research paper is a continuation of Chen's thesis [1]. The first part involved, using his latest model and adding more holes on the fins and changing the geometry--particularly the hole arrangement and shape. The second part modified the shape of the tank. Some models were developed in several shapes; for example, the shapes of sphere, ellipse and polygon. Then an effort was made to find the best models from the two parts. Besides the three materials used in Chen's study, two other composite materials--Eglass-Steel1040 and Eglass-Aluminum7050--were added. Therefore, the outline of this thesis follows.
Chapter 1: An introduction to the development of natural gas vehicles and the rationale in on-board storage of natural gas in vehicles.

Chapter 2: Literature review.

Chapter 3: Description of the materials used in the design and analysis process, especially composite material.

Chapter 4: Comparison of the results from several theories, small deflection, large deflection, American Society of Mechanical Engineering (ASME) code and Finite Element Method (FEM) analysis of non-cylindrical tank.

Chapter 5: Modeling and analysis.

Chapter 6: Discussion of the results of FEM modeling.

Chapter 7: Conclusion.

1-3 On-board Storage of Natural Gas in Vehicles

There are two main factors which can affect the amount of fuel stored on-board a vehicle. One is the driving range of the vehicle obtained by the use of the stored fuel size of the tank and the pressure inside the tank. The other is the ease of refueling location of the tank, particularly with respect to the time required.

There are also two factors which determine whether or not fuel can be used as an automotive fuel. One is the high heating value (energy per unit mass), which implies a relatively small fuel mass requirement, and the other is the high energy density (energy per volume), which implies a small storage volume. At present, gasoline and diesel fuel used as automotive fuels have a high heating value as well as high energy density.
However, natural gas is a gaseous fuel whose energy density is about 1/1000 of gasoline. Obviously, natural gas in its natural state can’t be used as automotive fuel.

Taking into account the above reasons, there are three ways to store natural gas on-board a vehicle within the vehicle volume constraints: compressed natural gas, liquefied natural gas and adsorbed natural gas. In compressed natural gas (CNG), the operation pressure is between 2400 psi and 4350 psi. The storage tank is typically cylindrical in shape, which is still not good enough for practical application.

For the liquefied natural gas (LNG), the density is 3.5 lb/gal. Its heating value is 23450 BTU/lb, which is the same as CNG, and its energy density is 89000 BTU/gal. The liquefaction of natural gas requires cryogenic temperature. So LNG fuel must be stored at -256 °F and, consequently, it requires a cryogenic tank. The state of cryogenic liquid tends to evaporate readily, possibly causing some accidents, such as fire, explosions and a loss of a certain percentage of fuel no matter how well the tanks are insulated. Hence, it introduces a significant degree of inefficiency. Moreover, its flammability is somewhat higher than gasoline, and for decentralized usage this means the cost invested in the equipment to distribute the fuel and refuel easily is expensive. Thus, when considering the reasons of safety and economy, it is not suitable for the general compact car.

In adsorbed natural gas (ANG), on-board storage requires the aid of adsorbents. An adsorbent is typically a kind of microporous solid, which effectively condenses the gas molecules within the porous structure by means of surface-gas interactions. Basically, natural gas stored in an adsorbent material will have a density between that of LNG and CNG. Thus, ANG will have a very high energy density without the need of low
temperature or the need of high pressure. However, this sort of fuel requires the finding and development of the right adsorbent material, which is still under research. Hence, it is also unsuitable for practical use.

In conclusion, several technologies are currently competing with the on-board storage of natural gas for a vehicle, such as CNG, LNG and ANG. Because of safety, efficiency and economic reasons, LNG is just more suitable for transportation applications where strict schedules of operation are a major part of the system, such as airlines, railways and long distance trucking companies.

ANG has not yet been proven economical at natural gas densities, which will make it desirable for automotive applications. Thus, a CNG fuel storage system is the only option at this time; however, efforts are still needed to improve the storage of natural gas.
CHAPTER 2

LITERATURE REVIEW

2-1 Natural Gas Vehicle

For long-term considerations of environmental protection, reducing operational costs and decreasing over-dependence of gasoline and diesel, the natural gas vehicle is still a potential alternative compared to the traditional gasoline and diesel engine vehicles. Many groups in various countries are working on developing NGV; industrial officials from seventeen countries [2] have formed an association to promote gas as a world transportation fuel. ENGVA (European Natural Gas Vehicle Association) will develop the global network necessarily to achieve a world-wide market for natural gas as a vehicle fuel.

Meanwhile, developing NGV still needs some cooperation from relevant associations; for instance, auto-manufacturers, governments, institutions and oil and gas industries need to join together. In the field of automaking, when automotive fuel is changed to natural gas, some components should be, consequently, modified.

John Ingersoll [3] suggested several modifications. For example, increasing the engine compression ratio, hardening the cylinder head, employing the new natural gas fuel injection system and installing a low-pressure natural gas regulator are required. Moreover, adding engine coolant lines for the heating pressure regulator, connecting the fuel line to the engine and considering the space of the CNG tank in the trunk or
underbody should be considered, as shown in Figure 2-1. Other modifications, such as placing the fueling point CNG valve and solenoid valve for each tank, applying the solenoid operated high-pressure system CNG fuel valve and high-pressure CNG fuel regulator, setting up the stone and heat underbody shield as well as relocating the mini-space tire in underbody, are also necessary.

![Figure 2-1: Pictorial representation of natural gas fueled vehicle](Source: Ford Motor Company)

The most recent federal regulation enacted is the Energy Policy Act (EPA), which was signed into law in late 1992 [4]. It covers a wide range of energy-related issues, such as an effort to reduce foreign oil dependence and increase the use of domestically produced fuels, and to encourage the use of alternative fuel vehicles. It also classifies alternative fuels, including CNG, LNG, LPG (liquefied petroleum gas), hydrogen, electricity and any other fuels that are made up of at least 85% alcohol, such as ethanol and methanol.
On the academic front, the NGV team of Colorado State University [5] built up a successful NGV vehicle in a 1991 competition. By modifying the combustion chamber, piston, camshaft, valve train, air intake, fuel delivery, ignition and a computer-controlled fuel system, the team produced a natural gas vehicle with a highway fuel economy of 23.6 mpg with excellent emission and power comparable to the stock gasoline vehicle. Although the vehicle can travel up to 356 miles at 55 mph, the cylinder necessitated the displacement of the truck bed, so, it is not suitable for average compact car.

Another NGV team at University of Nebraska-Lincoln provided a useful way to calculate the size of natural gas tank in their paper [6], as follows:

- **Fuel data:**
  - **Gasoline**
    - Lower heating value: 44 MJ/KG
    - Density: 0.73 kg/M$^3$
    - Molecular weight: 
    - Gas constant:
  - **Natural Gas**
    - Lower heating value: 45 MJ/kg
    - Density: 
    - Molecular weight: 18 kg/kg-mol
    - Gas constant: 461.9 J/kg-k

- **Assume:**
  1. Natural gas as an ideal gas which follow PV=NRT.
  2. Maximum travel range is 250 miles.
  3. Fuel consumption rate is 15 Miles/Gallon (mpg).
  4. Initial tank pressure is 3000 psi (206.8e+05 pa).
    - Z (compressibility factor) is 0.84 ($Z = PV/RT$).
  5. Final tank pressure is 300 psi (20.68e+05 pa).
    - Z is 0.96.
  6. Temperature (T) is 300 °k.

- **Procedure:**
  - Volume of natural gas required: 250 miles / 15 mpg = 16.67 gal = 0.0638 M$^3$
  - Mass of gasoline required: 0.06308 M$^3$ * 730 kg / M$^3$ = 46.05 kg
  - Natural gas required in mass: 46.05 kg * 44 MJ/kg (gasoline) / 45 MJ/kg (NG) = 45.05 kg.
  - So that, Mass (initial) – Mass (finial) = 45.05 kg
  - Because, Mass * Z = (P*V) / (R*T), so:
    - [P (initial)*V] / [Z (initial)*R*T] - [P (finial)*V] / [Z (final)*R*T] = 45.05 kg
\[
\Rightarrow \text{Volume (V) } = \frac{(45\times R \times T)}{[\pi \times (Z_i - P_f) / Z_b]} = \frac{(45\times 461.9 \times 300)}{[206.8e+05 / 0.84 - 20.68e+05 / 0.96]} = 0.2776
\]

\[
\Rightarrow V = 0.2776 \text{ m}^3 \text{ or } 16,900 \text{ in}^3.
\]

Therefore, the minimum volume used to store the compressed natural gas in a tank is 16,900 cubic inch.

Although many organizations have conducted research on NGV development, there are still some obstacles in the growth of NGV use on a grand scale [7]. For instance, the problems of space for natural gas storage, lack of compressed natural gas refueling infrastructure and the deficit of actions by the oil and gas industry need to be solved. Hence, my thesis is focused on improving natural gas storage.

2-2 Design of Natural Gas Tank

As David Burnicle [8], a deputy chairman of the Natural Gas Vehicle Association (NGVA) in the United Kingdom, pointed out, in the near future the development of fuel storage tank will continue to use steel cylinders. He believes that advanced gas container technology will include the use of composites and a mix of metal and composites containing a hoop-wrapped aluminum design, although he hasn’t seen the fuel containers being developed in a complex shape.

From the design point of view, a natural gas tank can be viewed as a pressure vessel. So for safety reasons, it should follow ASME code [9]. Section VIII, Division 1 covers general rules in designing pressure vessel, such as minimum thickness of shells and heads, design pressure, loading and maximum allowable stress. It also classifies different types of rectangular vessels, which can be used as a reference for engineers.
For example, a group in the Concordia University presented a paper [10] on rectangular pressure vessels based on three theories: small deflection analysis, large deflection analysis and finite element analysis. They quoted that although the ASME code includes the design rules of unreinforced and reinforced rectangular pressure vessels, these rules are based on infinitely long vessels of non-circular cross section. Moreover, the stress calculations are also based on linear small deflection theory of plate bending. Hence, it is not practical for pressure vessels of finite length.

In the small deflection theory, they modified the model of finite length in a short rectangular pressure vessel with appropriate edge conditions in three dimensions in order to obtain the deflections and stresses. However, this method is tedious and complex and is not suitable for the designer. Therefore, they combined it with the design formulas given in the ASME code based on maximum stresses obtained from frame analysis of prismatic pressure vessels and small deflection plate analysis of large side panels.

In the large deflection theory, when deflection exceeds one-half thickness of the plate, the linear small deflection is no longer applicable because part of the loading is carried by membrane tension. Hence, stresses for a given load are generally smaller and deflections are generally greater than the ones from small deflection analysis.

In the finite element analysis, by applying mesh, material property and appropriate boundary conditions, the values of deflections and stresses are slightly different from those obtained from the experimental method. So, the finite element method is still of help in designing a pressure vessel.
Another theory regards failure. As Dennis Moss [11] stated previously, stresses are meaningless until compared to some stress/failure theories. There are two theories always used in designing pressure vessel: maximum stress theory and maximum shear stress theory.

Maximum stress theory asserts that failure occurs whenever the greatest tensile stress trends to exceed the uniaxial tensile strength or whenever the largest compressive stress trends to exceed the uniaxial compressive strength.

Maximum shear stress theory declares that yielding starts at a point when maximum shear stress at that point reaches one-half of the uniaxial yield, $F_y$. If $\sigma_1 > \sigma_2$ is biaxial state of stress, or $\sigma_1 > \sigma_2 > \sigma_3$ is triaxial state of stress, the yielding will begin when $(\sigma_1 - \sigma_2) / 2 = F_y / 2$ or $(\sigma_1 - \sigma_3) / 2 = F_y / 2$. This theory is closer to the experimental results and is easy to use; however, it is suitable only for isotropic materials.

For anisotropic materials, such as orthotropic lamina, another maximum stress theory [12] can be applied in the failure criterion. It says failure will not occur when any one of the stress components is lower than its relevant ultimate (or yield) strength, so

- $\sigma_L$ (longitudinal allowable tensile stress) < $\sigma_{LU}$ (longitudinal yield tensile strength)
- $\sigma_T$ (transverse allowable tensile stress) < $\sigma_{TU}$ (transverse yield tensile strength)
- $\tau_{LT}$ (allowable shear strength) < $\tau_{LTU}$ (shear yield strength)

If $\theta$ is in the direction of fiber, $\sigma_x$ is the uniaxial tensile stress (or major principal stress) in x-direction, then

- $\sigma_L = \sigma_x \ast (\cos \theta)^2$
Another idea for a natural gas tank is the concept of fins with some hole arrangements. Since the fins connect to the inside walls of the tank, stress variations, such as stress concentration, will appear. Hence, it can also affect the quality of design in a pressure vessel. There are three types of hole-arrangements—square, rectangular and diamond patterns—which were developed by two German researchers in 1956 [13], as shown in Figure 2-2. Basically, fins with hole arrangements applied in a natural gas tank are to enforce the structure in order to prevent explosion and improve the circulation of natural gas. This concept will be applied in the following study.

\[
\begin{align*}
\sigma T &= \sigma x \cdot (\sin \theta)^2 \\
\tau LT &= \sigma x \cdot (\sin \theta \cdot \cos \theta)
\end{align*}
\]

**Figure 2-2:** Hole arrangements on the fins [13]
2-3 Materials Used in Designing Fuel Storage of NGV

An important part of any natural gas engine system is on-board fuel storage, especially as it affects the weight, driving range, safety and cost of a vehicle. Material is one of the key factors in determining whether the fuel storage is lighter, of lower cost or can endure compressed natural gas. There are three materials always used in on-board tanks: steel, aluminum and composites. Composite material, in particular, is the recent trend.

Cymdyne I, Inc. has developed a vehicle gas storage cylinder having a cold-drawn aluminum liner fully over-wrapped with a low cost fiberglass, E-glass [14]. The cost of this model is less than that of a hoop-wrapped aluminum cylinder and is much lighter than a hoop-wrapped steel cylinder. Moreover, it provides better characteristics of material, such as thermal insulation, corrosion resistance and impact resistance.

Aluminum Company of America (ALCOA) has produced a CNG cylinder by using composite-reinforced aluminum in a variety of sizes to suit most vehicular applications [15]. They cut the weight of a conventional vehicular type of steel tank by 50% and increased the maximum endurance pressure from 2,400 psi to 3,000 psi, thus causing the vehicle to store about 25% more natural gas and increasing the range of driving from 150-200 miles to 200-250 miles.

In the competition of NGV in 1991, a team from the University of Virginia used the composite material, Aluminum/Kevlar in designing the fuel storage [16]. This tank has the advantage of being lighter in weight than steel and has a rated pressure of 3,000 psi.
Another team from West Virginia University chose the tank based on the considerations of cost, weight and the size per internal storage volume [17]. The aluminum composite tank and steel composite tank were compared. Finally, steel was selected because it costs less and has a greater yielding point. Although an aluminum tank is lighter than steel, the greater thickness of the aluminum tank counterbalanced the advantage of weight. So, from the previous discussion, composite materials are becoming more popular than ever before.
CHAPTER 3

MATERIAL CONSIDERATIONS IN NATURAL GAS TANK

3-1 Introduction

Different materials have different properties, which consequently affect the results of any analysis. In a natural gas tank, steel and aluminum are traditionally the major options. However, composite material is getting popular recently because of its better material properties. Therefore, in this chapter besides the consideration of conventional materials, some of the properties of composite material are also included, such as physical, mechanical and chemical properties.

In composites, fiber and laminate are the major study. For example, laminate, Eglass/Steel-AISI-1040 and Eglass/Aluminum7050 are the combination of glass fiber and alloy.

3-2 Considerations of Different Materials

In mechanical design, usually strength and rigidity are the main factors in selecting materials. Whenever a machine part is made from alternative material, the equally important factors are its relative reliability and durability. Other factors are its cost and availability. In recent years owing to the growing emphases on environmental protection and ecological preservation, the choice of material has been increasingly influenced by its recycleability, energy requirement and environmental pollution. Therefore, this thesis also deals with these factors.
Among the materials used for a natural gas tank, Steel-AISI-1040 and Aluminum-6061 are the most popular. Steel-AISI-1040 is stronger than Aluminum-6061, but it is heavier than aluminum. Though aluminum is lighter than steel, it can't resist higher internal pressure. A composite material--Kevlar/Steel-AISI-1040--can resist higher pressure than steel does and its weight is between aluminum and steel; however it is more expensive either steel or aluminum. The detailed mechanical and chemical properties of the previous three materials have been discussed by Chen [1] and will not be repeated here. However, there are several significant material properties which were applied in the following FEM analysis and have been listed in Tables 3-1 and 3-2.

Because of their acceptable cost, intermediate weight and stronger strength compared to the previous three materials, composites, such as Eglass/Steel-AISI-1040 and Eglass/Aluminum-6061, have been considered as advanced materials in the further analysis of this study.

3-3 Composite Material

Composites are always made up of one or more discontinuous phases embedded into a sort of continuous phase. Usually, the discontinuous phase called "reinforcement" is harder and stronger than the continuous phase, which is termed matrix. So, the properties of composites are strongly affected by the properties of each constituent material, their distribution and the interaction among them. That means composite properties depend on the properties of constituents, the form of fibrous reinforcement used, such as chopped or woven, fiber volume fraction, length, distribution and
### Table 3-1: Material Properties of Alloy[1]

<table>
<thead>
<tr>
<th>Material</th>
<th>Elastic Modulus (psi)</th>
<th>Poisson</th>
<th>Shear Modulus (psi)</th>
<th>Density (pci)</th>
<th>Yield strength (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel-AISI-1040</td>
<td>2.93e+07</td>
<td>0.32</td>
<td>1.10985e+07</td>
<td>0.284</td>
<td>64000</td>
</tr>
<tr>
<td>Aluminum - 6061-T6</td>
<td>1.0e+07</td>
<td>0.33</td>
<td>3.7594e+06</td>
<td>0.098</td>
<td>36000</td>
</tr>
<tr>
<td>Aluminum - 7050</td>
<td>1.0e+07</td>
<td>0.33</td>
<td>3.7594e+06</td>
<td>0.102</td>
<td>65000</td>
</tr>
</tbody>
</table>

### Table 3-2: Material Properties of Fiber [1] & [18]

<table>
<thead>
<tr>
<th>Material</th>
<th>E-glass</th>
<th>Kevlar</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (lb./ in²)</td>
<td>0.092</td>
<td>0.0362</td>
</tr>
<tr>
<td>Tensile Strength (psi)</td>
<td>5.0e+05</td>
<td>6.50e+05</td>
</tr>
<tr>
<td>Modulus of Elasticity (psi)</td>
<td>1.05e+7</td>
<td>1.8e+07</td>
</tr>
<tr>
<td>Melting Temperature (°C)</td>
<td>1260</td>
<td>500</td>
</tr>
<tr>
<td>Specific Strength (in)</td>
<td>5.24e+06</td>
<td>1.01e+07</td>
</tr>
<tr>
<td>Poisson</td>
<td>0.2</td>
<td>0.32</td>
</tr>
<tr>
<td>Yield Strength (psi)</td>
<td>4.821e+05</td>
<td>3.6562e+05</td>
</tr>
<tr>
<td>Longitudinal Tensile Strength, σLU (psi)</td>
<td>150e+03</td>
<td></td>
</tr>
<tr>
<td>Transverse Tensile Strength, σTU (psi)</td>
<td>7.0e+03</td>
<td></td>
</tr>
<tr>
<td>Shear Strength, τLTU (psi)</td>
<td>10e+03</td>
<td></td>
</tr>
</tbody>
</table>
orientation, interfacial bond strength and void content.

Composites are classified as particle-reinforced composites or fiber-reinforced composites (More detailed items have been classified in Figure 3-1 [18]).

In physical properties, the particle may be described as spherical, cubic, tetragonal, a platelet or another regular, irregular shape. Fiber can be described by its length, which is much greater than its cross-sectional dimensions. When the properties vary from point-to-point through the material, it is called heterogeneity, whereas, a material whose properties are different with direction is called anisotropy.

In the area of elastic properties, although composites are heterogeneous in the microview, for design purposes they are usually considered to be homogeneous, anisotropic materials. Isotropic materials, such as most metals, can be characterized by two independent elastic constants. They can be realized from Equation [19] as

\[ G = \frac{E}{2(1+\nu)} \] ; where
- \( E \) is Young's modulus,
- \( \nu \) is Poisson's ratio,
- \( G \) is Shear modulus.

Therefore, if two of the constants are known, the third can be determined from the relationship. However, for anisotropic material, more constants are required.

Fiber is a kind of composite. There are several major categories of man-made reinforcing fibers. For instance, glass, carbon, organic, boron and ceramic are classified in the field. Glass fibers are the cheapest and most widely used man-made composite reinforcements. The principal advantages of glass fibers are the low cost and high strength. The disadvantages are the low modulus and poor abrasion resistance, which
Figure 3-1: Classification of composite materials [18]
decrease their potential strength and cause poor adhesion to polymer matrix resins, particularly in the presence of moisture. Therefore, it requires the use of chemical coupling agents on the surface of the fibers to improve the adhesion.

Due to bad adherence in glass fiber, it requires chemical treatment to be applied during the forming process, called “sizes”. There are two types: temporary sizes and compatible sizes. The temporary size is applied to minimize the degradation of strength resulting from abrasion of fibers to one another and to bind the fibers together for easy handling in forming glass-fiber products. Then it is removed by heating and replaced by coupling agents before the fibers can be imbued with resin.

A compatible size is applied to improve initial adhesion of resin to glass and to reduce the destructive effects of water and other environmental forces on this bond. The compatible sizes are often called “coupling agents”. The general chemical formula is [18]

\[ X_3 \text{Si} (\text{CH}_2)_n \text{Y} \]

- where: \( n = 0 \) - 3
- \( Y \): organofunctional group that is compatible with polymer matrix.
- \( X \): hydrolyzable group on silicon.

So, when the glass fiber absorbs water molecules to form hydroxyl group, the glass surfaces will be formed immediately. It is thought that coupling agents allow better retention of interfacial strength when composites are subjected to moisture. The subsequent interaction is shown in Figure 3-2.

Moreover, glass fiber consists primarily of a silica(SiO2) backbone in the form of \((-\text{SiO}_4-)_n\) tetrahedra. There are three types of glass fiber: E, S and C. C type is used where corrosion resistance is particularly important. S type is a high-strength fiber initially developed in military application. E type is designated electrical grade used widely in the
Figure 3-2: Sequence of interaction with coupling agents [18]
field of agriculture, aerospace, electronics, transportation, etc. Some material properties of E-type are shown in Table 3-2.

3-4 Laminate Material (Eglass/Steel-1040 & Eglass/Aluminum-7050)

Laminate is formed from two or more laminas bounded together to act as an integral structural element. The principal material directions of laminas are oriented to produce a structural element with the desired properties in all directions. Laminates are fabricated so that they can act as single-layer materials. The bond between two laminas in a laminate is assumed to be perfect, which means infinitesimally thin and with no shear deformation. Thus, the laminas can’t slip over each other and the displacement remains continuous across the bond.

In my study, two kinds of laminate material were applied: Eglass/Aluminum7050 and Eglass/Steel-1040. They are viewed as a symmetrical laminate, which is constructed by placing the laminas symmetrically with respect to mid-plane. The top and the bottom layers are Eglass and the middle layer is aluminum or steel. Each layer has a unique, constant thickness, and the orientation is defined by a single constant angle. Therefore, their material properties are different, accordingly.

In the two kinds of models, two types of thickness sequence were used: 0.2/0.6/0.2 in and 0.2/0.5/0.2 in. The orientation sequence is 45/0/45 degree. Applying the methods mentioned by Chen [1], the material properties can be calculated as follows:
Conditions:

1. Specific strength of E-glass: 5.24e+06 in
2. Thickness of each Eglass layer: 0.2 in (2 layers)
3. Yield stress of aluminum7050: 65000 psi
4. Density of Eglass: 0.092 lb/in³
5. Density of steel-1040: 0.284 lb/in³
6. Density of aluminum7050: 0.102 lb/in³

Properties of Eglass/aluminum7505:

A. Combined density:
For (0.2 / 0.6 / 0.2): 0.102 pci * 0.6 in/1.0 in + 0.092 pci * 0.4 in/1.0 in = 0.098 pci.

B. Combined yield stress:
Specific strength = Yield strength / Density [1]
Yield stress of Eglass = 0.092 * 5.24e+06 = 482080 psi
Combined yield stress of laminate = 4.8208e+05 * 0.4 + 0.65e+05 * 0.6 = 231832 psi

Therefore, by means of the same method, the properties of laminate, Eglass/steel-1040 are

- If (0.2 / 0.6 / 0.2):
  - Density = 0.4 * 0.092 pci + 0.6 * 0.284 pci = 0.2072 pci
  - Combined yield stress = 0.4 * 4.8208e+05 psi + 0.6 * 0.65e+05 psi = 231232 psi.

Table 3-3 includes several properties of laminates, especially on Kevlar/Steel-1040, Eglass/Steel-1040 and Eglass/Aluminum7050. Several material costs per pound are also listed in Table 3-4.
Table 3-3: Material Properties of Laminate [1]

<table>
<thead>
<tr>
<th>Material</th>
<th>Kevlar/Steel-AISI-1040</th>
<th>Eglass/Steel-AISI-1040</th>
<th>Eglass/Aluminum-7050</th>
</tr>
</thead>
<tbody>
<tr>
<td>Combined Density (pci)</td>
<td>0.209</td>
<td>0.2072</td>
<td>0.098</td>
</tr>
<tr>
<td>Combined Yield Strength (psi)</td>
<td>154600</td>
<td>231232</td>
<td>231832</td>
</tr>
</tbody>
</table>

Table 3-4: Price List of Several Materials [20]

<table>
<thead>
<tr>
<th>Material</th>
<th>Steel</th>
<th>Aluminum</th>
<th>E-glass</th>
<th>S-glass</th>
<th>Kevlar</th>
<th>Hi-strength carbon fiber</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost: ($/lb)</td>
<td>0.3</td>
<td>1.10</td>
<td>0.5</td>
<td>1.0</td>
<td>18</td>
<td>19</td>
</tr>
</tbody>
</table>
CHAPTER 4
THEORY AND FEM ANALYSIS OF RECTANGULAR TANK

4-1 Introduction

This chapter mainly deals with the applications of theories as previewed in Chapter 2 [10] on a simple rectangular model. These theories include the ASME code, the small deflection and the large deflection. Some formulas were derived from the concept of material strength [21] in beam and frame structure. Then maximum bending moments corresponding to the principal stresses were calculated by the relevant theories. The stresses with respect to the bending moments are the principal stresses combined with membrane stresses as the sum of maximum stresses.

On the FEM model, a rectangular frame was built up by Patran package. After maximum stresses were produced from analysis, the stresses of the FEM model were compared with those of mathematical models in order to find how close these results are.

4-2 Small Deflection Analysis

This method is assumed the length of rectangular frame is infinite. So we can consider the cross section of the frame in 2-D, and each side of the rectangular frame can be looked as a beam actuated by a uniform distribution load with built-in ends. By applying the equation (1): \( M = q \cdot \ell^2 / 12 \) where \( M \) is the moment per unit length, \( q \) is the load per unit length, \( \ell \) is the length of the beam, the moment and the stress of the rectangular tank can be analyzed as follows:
As shown in Figure-f1, by applying Equation (1), the fixed end bending moment of member AH, and AG:

\[ M_{AH} = P \cdot \ell_2^2 / 12, \quad M_{AG} = P \cdot \ell_1^2 / 12 \quad \ldots(2) \]

By using moment distribution, corner moment:

\[ M_A = k_1 \cdot M_{AH} + (1-k_1) \cdot M_{AG}, \quad \text{where} \]
\[ k_1 = \frac{\ell_1}{\ell_1 + \ell_2} \quad \ldots\ldots(3) \]

Another method is using graphical form, which was created by Blach, Hoa, Kwok and Ahmed [10], to meet both cases--infinite and finite length--as shown in Figure-f2.
\[ M = \beta \cdot P \cdot b^2, \text{ where } M: \text{moment}, \ b \ & a: \text{dimensions of plate} \]

\[ \beta : \text{plate coefficient, stress at center or edge} \]

From Figure-f2, moment respect to several positions on the frame can be found (Figure-f1):

- M11 and M21: fixed end bending moment at edge
- Md and M14: bending moment at center

From curve, M11, M21, and M14 can be found. Then, applying previous method,

- Corner moment: \( M_A = k_1 \cdot M_{11} + (1-k_1) \cdot M_{21} \)
- Center moment of member AH: \( M_d + M_A = M_{11} + M_{14} \implies M_d = M_{11} + M_{14} - M_A \)

Finally, another corner and central moments of member AG can also be found. Then the largest bending moment of the frame is chosen. The stress corresponding to the bending moment is the principal stress, which is combined with membrane stress as the maximum stress of the frame. The plate coefficient \( \beta \) is approaching a constant value while the ratio of \( a/b \) is over 2 if \( a > b \). Therefore, the corner moment, \( M_A \) can be calculated by using Equations (2) and (3).

4-3 Analysis of ASME Code

In this section, several formulas given in Appendix 13 of the code are used to calculate the bending stresses and membrane stresses as discussed:

In Figure-f3: (a) \( t=t_1=t_2 \), for uniform thickness
(b) \( P \) is the internal pressure
(c) Membrane stress of long-side plate:
Sm = P * H / 2*t2 = P * H / 2*t ......(4)

(d) k: vessel parameter = ( I1 / I2 ) * α

I1 & I2 : moment of inertia, α: rectangular vessel parameter = H/h ⇒

k = α* ( t3 / 12 ) / ( t3 / 12 ) = α, only at t = constant

I2 / c = ( t3 / 12 ) / ( t/2 ) = t^2 / 6

(e) Bending stress for long-side plate:

\[ S_{t,m} \text{ (at the middle point)} = \frac{P*c*h^2}{(12*I2)*[-1.5 + (1+k*α^2)/(1+k)]} \] ...

\[ S_{t,c} \text{ (at corner)} = \frac{P*c*h^2}{(12*I2)*[(1 + k*α^2) / (1+k)]} \] ......(6)

So, plug the factors as discussed in (d) into Equations (5), (6):

\[ S_{t,m} = \frac{P * h^2}{(2 * t^2) * [1.5 - (1 + \alpha^3) / (1 +\alpha)]} \] ......(7)

\[ S_{t,c} = \frac{P * h^2}{(2 * t^2) * [(1 + \alpha^3) / (1+\alpha)]} \] ................(8)

Figure-f3: Rectangular frame (H>h)
(f) Total stress:

\[(\text{St})n = (25) + (28) = \frac{P^* H}{2t} + \frac{P^* h^2}{(2^* t^2)} * [1.5 - (1 + \alpha^3) / (1 + \alpha)] \ldots (9)\]

\[(\text{St})c = (25) + (29) = \frac{P^* H}{2t} + \frac{P^* h^2}{(2^* t^2)} * [1 + \alpha^3] / (1 + \alpha) \ldots (10)\]

By using Equations (9) and (10), critical stresses on the middle and edge point of the plate can be calculated.

4-4 Large Deflection Analysis

When the deflection becomes larger than about one-half the thickness, the strain and stress at the middle surface can’t be ignored. The stress is called “diaphragm stress” or “membrane stress”. Hence, it is assumed that the stress at center point of the largest panel governs the design [22].

In this case, the rectangular frame can be looked at as the combination of several rectangular plates with fixed and simple support. Hence, several numerical values of coefficient, such as, \( \frac{y}{t} \), \( \frac{(q^*_b a^4)}{E_t^4} \), and \( \frac{(\alpha * b^2)}{(E_t^2)} \), can be obtained from table form or graph. In this study, several graphs quoted by Blach, Hoa, Kwok and Ahmed [10] were used to calculate deflection, membrane stress and combined total stress, as shown in Figure-f4, f5, f6 [10].

In these diagrams, several descriptions were given:

- \( a & b \): Length of long side and short side of rectangular plate
- \( \sigma m \): Membrane stress
- \( \sigma \): Combined stress = membrane stress + bending stress
- \( y \): Deflection in \( y \) coordinate
- t: Thickness of vessel plate
- q: Uniform distribution load over entire surface, pressure

Figure-f4: Curve of deflection (y)

Figure-f5: Curve membrane stress (σm)

Figure-f6: Curve of total stress (σ)
4-5 Analysis of FEM Method

In this section, a rectangular vessel was built using Patran package in 3-D solid. The dimensions of this model are 12 in (length), 12 in (height), 10 in (depth) and the thickness is 1 inch.

Element is Hex. 4 isoparametric type. In boundary conditions, the internal pressure is 1,500 psi, and the constraints were applied at four edges as fixed and simple support conditions. The material used is Aluminum-6061-t6.

After the process of going through the stage of analysis and post-process, the Von-Mises stress distributions were created.

4-6 Comparison of the Four Methods

In this part, an example of a model with open ends was given in order to calculate the maximum stress using the previous four methods and compare the results of maximum stress.

(A) Given:

(a) \( \ell=12 \text{ in}, \ m=12 \text{ in}, \ n=10 \text{ in}, \ t=1 \text{ in.} \)

(b) Aluminum-6061-t6, \( E=1.0 \times 10^2 \) psi, \( \nu = 0.3 \)

(c) \( P \) (internal pressure) = 1500 psi

(d) Constraints: fixed and simple support at four edges of plates.

(B) Aim: Find the maximum stress by using the four methods
(I) Small deflection method:

The maximum stress happens at the plate whose size is 10in * 12in. So, from Figure-f6 and by applying the theory mentioned in 4-2, some values are calculated:

(*) n/ ℓ = a / b = 10 / 12 = 0.833

by using graphical form

⇒ β1 (edge) = 0.043, β2 (middle) = 0.018

(ref: Figure-f2)

(*) M11 = M21 = β1 * P * a^2 = 0.043*1500*12^2 = 9288 (lb*in/in), (Because pates of the frame are same size.)

(*) M14 = β2 * P * b^2 = 0.018*1500*10^2 = 3888 (lb*in/in)

\[ MA = k_1 * M11 + (1-k_1) * M21 = M11 = 9288 \text{ (lb*in/in)} \]

\[ MA + Md = M11 + M14 \Rightarrow Md = M11 + M14 - MA = 3888 \text{ (lb*in/in)} \]

The maximum stress (σa) with respect to the maximum is MA. ⇒ σa = MA / Z = 6*MA / t^2 = 55728 (psi)

The membrane stress, Sm = P * H / 2*t = 1500 * 12 / 2 = 9000 (psi)

The total stress, σt = σa + Sm = 64728 (psi)

(II) ASME code method:

By using Equations (4), (7) and (8), some results can be found:
Sm = 1500*12/2 = 9000 psi, \( \alpha = H/h = 1 \)

\[ \text{St},m = 1500 \times 12^2 / 2 \times (1.5 - 2/2) = 54000 \text{ (psi)} \]

\[ \text{Sc},m = 1500 \times 12^2 / 2 \times (2/2) = 108000 \text{ (psi)} \]

By applying Equations (30) and (31) for total stress:

\[ (\text{St})_m = \text{Sm} + \text{St},m = 63000 \text{ (psi)} \]

\[ (\text{St})_c = \text{Sm} + \text{Sc},m = 117000 \text{ (psi)} \]

The maximum stress happens at corner \((\text{St})_c\).

(III) Large deflection method:

It is assumed the stress at the center of plate has the maximum value, then several values can calculated from Figure-f5, f6:

\[ a = 12 \text{ in}, \ c = 10 \text{ in}, \Rightarrow \ a/c = 1.2 \Rightarrow P* c^4 / (E* t^4) = 1500*10^4/10^7 = 1.5 \]

From Figure-f6, the coefficient of total stress, \( \sigma * c^2 / (E* t^2) = 0.95 \)

\( \Rightarrow \sigma \) (total stress) = 0.95 * 10^7 / 10^2 = 95000 (psi)

From Figure-f5, the coefficient of membrane stress, \( \sigma m * c^2 / (E* t^2) = 0.113 \)

\( \Rightarrow \sigma m \) (membrane stress) = 0.113 * 10^7 / 100 = 11300 (psi)

(IV) FEM method:

Based on the same model, load and constraints, Patran brought out the result of stress distribution, as shown in Figures-f8. The maximum stress is 87180 psi.

(V) Conclusion:

Several maximum stresses with corresponding to each method were listed in Table4-1. From the table, the average value is 91 kpsi, which is as close as that derived by the FEM method. The value of small deflection is less than the average value by 0.71 times. The value of the code method is more than the average value by 1.29 times.
value of the large deflection method is more than the average value by 1.043 times, and
the one of FEM method is less than average by 0.95 times. Therefore, according to the
differences, the results are reasonable close to each other compared to the study of Blach,
Hoa, Kowk and Ahmed [10].

![Von-Mises stress](image)

**Figure-f8:** Von-Mises stress

<table>
<thead>
<tr>
<th>(unit: kpsi)</th>
<th>Small deflection</th>
<th>ASME code</th>
<th>Large deflection</th>
<th>FEM</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. stress</td>
<td>65</td>
<td>117</td>
<td>95</td>
<td>87</td>
<td>91</td>
</tr>
</tbody>
</table>
5-1 Introduction

Different software has different merits and shortcomings. Therefore, it would increase the efficiency of FEM modeling and analysis if we chose appropriate packages. Keeping this in mind, the FEM packages used in this research on designing compressed natural gas tank are Patran 5.0 and Ansys 3.0.

In Ansys, the supplier support limits the total node number, and it needs a large working memory to create FEM models in more complicated geometry. So this software was applied here mainly on simple geometry, such as spheres, ellipses and polygons. However, it has a powerful feature called Smart Mesh, which can produce a most accurate mesh, especially in complex geometry.

Patran was used to develop the 3-D models from rectangular shapes with different hole arrangements to irregular shapes with the final hole arrangement. Then their FEM models were built and the results of analysis were obtained.

In the application of materials, Steel-1040 and Aluminum-7050 were used as both homogeneous and isotropic material. Composite materials--Kevlar/steel-1040, Eglass/Steel-1040 and Eglass/Aluminum-7050--were considered as an anisotropic
material with unidirectional properties in each layer. This chapter describes how to create FEM models by using Patran and Ansys packages.

5-2 Natural Gas Tank Created by Patran Package in 3-D Model

Patran 5.0 provides new modular architecture to support computing and exchange standard. Basically, it consists of several components, such as pre- and post-processing, analysis modular for executing structure, thermal, fatigue, dynamical and other types of mechanical analysis, as well as linkage to other leading CAD system or FEM programs.

This section was divided into two parts in developing 3-D models. One was to develop the rectangular shape with varied hole-arrangements. Based on the Chen's final model, several hole arrangements were added on the fins mentioned in Chapter 3 [13] to build up the sequential models. Therefore, four types of multi-hole patterns were applied: 5-holes pattern, square pattern, rectangular pattern and diamond pattern. The hole diameter was used in 1, 0.9, 0.8 and 0.35in. Wall thickness was used in 1, 0.9 and 0.8in. Aluminum-6061 was applied in the first development of the models. After the best model was obtained according to the lowest stress from the comparison of the results, other materials were used in that model, such as Steel-1040, Kevlar/Steel-1040, Eglass/Steel-1040 and Eglass/Aluminum-7050. The detailed procedure for creating a 3-D FEM model is shown below:
A. Create Geometry

1. Enter key points, then draw the relevant lines, curves and circles on the Patran drawing environment.

2. Form several surfaces by using the function of extrude, glide, trim and transform to shape an enclosure surface.

3. Produce a quarter part of solid by using the function of B-spline solid according to the enclosure surface.

4. Use the function of mirror to get the whole body according to the related symmetrical surfaces.

Up to this stage, we have a rectangular tank with a hole arrangement that can consist of four B-spline solids, which are different from Chen's model of 57 solids.

B. Create FEM Model

1. Apply solid elements, nodes on the B-spline solids by using element type Tet4 and element size 0.5.

2. Set up loads and boundary conditions on the elements and nodes.

3. Key in material data in the form and apply it to the elements.

4. Create load case from the options of loads and boundary conditions.

5. Put the FEM model into analysis processor to obtain the results.

6. Review the results from the post-processing.

The relevant models and dimensions were shown in Figures 5-1 through 4 and Tables 5-1a and 1b.
The second step was to develop different shapes of tank with the final hole arrangement. By means of the previous procedure, different internal pressures and boundary conditions are applied. Hence, several models were created, as shown in Figures 5-5 through 5-9. Their detailed dimensions were listed in Table 5-1b. Some other relevant FEM models were also shown in Figures 5-10 through 19.

5-3 Creating 3-D Simple Geometry Tank Using Ansys Package

From the cylindrical tank studied in Chen’s paper, the models were developed by drawing the shapes of sphere, ellipse and polygon and applied aluminum-6061, internal pressure (3000 & 3500 psi) and displacements to look into the relationship between stress and shape.

Basically, Ansys package provides several capabilities--performs linear and non-linear static, dynamic and thermal analysis in 2-D and 3-D, allows us rapidly to create, verify and modify the models as well as to optimize the designs, includes the excellent interactive pre- and post-processors and data transfer links to the major CAD systems, and executes high accuracy FEM mesh. The function Smart Mesh can be looked upon as a kind of probe. When approaching complicated geometry, such as hole, fillet or b-spline curve, it can automatically decrease the element size in order to match these shapes accurately. Therefore, the total number of elements and nodes become much higher than the supplier can support. Normally, if a process runs over 6000 in total node number, it
Figure 5-1: Model 1

Figure 5-2: Models 2-4

Figure 5-3: Models 5-7, 11-18

5-hole pattern

Square pattern

Rectangular pattern
Figure 5-4: Models 8-10

Diamond pattern

Figure 5-5: Models 19-27

*Hole arrangement:
Rectangular pattern
*Hole diameter: D1
*High: H
*Depth: D
*Width: W
*Fin thickness: T2

Figure 5-6: Models 28-32

*Hole arrangement:
Rectangular pattern
*Hole diameter: D1
*Radius of fillets:
Fr1 & Fr2
*Fin & Wall: T2 & T1
Figure 5-7: Models 33-38

Figure 5-8: Models 39-41

Figure 5-9: Models 42-44, (half)

- Wave surface: R2 & R3
- Rectangular pattern (A & B value)
  - Hole: D1
  - Width: W/2
  - Depth: D
  - High: H
- Fin and wall: T2 & T1
Table 5-1a: Basic Dimensions Used by Patran in 3-D Modeling

<table>
<thead>
<tr>
<th>Mod No. (unit: in)</th>
<th>Mod 1</th>
<th>Mod 2</th>
<th>Mod 3</th>
<th>Mod 4</th>
<th>Mod: 5, 13-18</th>
<th>Mod 6</th>
<th>Mod 7</th>
<th>Mod 8</th>
<th>Mod 9</th>
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<td>12</td>
<td>12</td>
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<tr>
<td>Depth (D)</td>
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<td>10</td>
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<td>10</td>
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<tr>
<td>Height (H)</td>
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<td>1</td>
<td>1</td>
<td>1</td>
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<td>(T1)</td>
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<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
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<td>0.2</td>
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<tr>
<td>(Fr1)</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td>1.25</td>
<td>1.25</td>
<td>1.4</td>
<td>1.4</td>
<td>1.4</td>
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<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
</tr>
<tr>
<td>of holes (A)</td>
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<td></td>
<td></td>
<td></td>
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Table 5-1b: Basic Dimensions Used by Patran in 3-D Modeling

<table>
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<tr>
<td>High (H)</td>
<td>12</td>
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<td>9.7</td>
<td>9.7</td>
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<td>Hole1 diameter (D1)</td>
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<td>0.9</td>
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<tr>
<td>Fillet1 radius (Fr1)</td>
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<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
<td>0.2</td>
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<td>Fillet2 radius (Fr2)</td>
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<td>0.4</td>
<td>0.5</td>
<td>0.5</td>
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<td></td>
</tr>
<tr>
<td>Radius of outside curve (R1)</td>
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<td>2.9</td>
<td>2.9</td>
<td>2.9</td>
<td>1.9</td>
<td>1.9</td>
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<td></td>
</tr>
<tr>
<td>Radius 2 of wave surface (R2)</td>
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<td></td>
<td>0.5</td>
<td>0.5</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Radius 3 of wave surface (R3)</td>
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<td>0.5</td>
<td>0.5</td>
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<td>Row distance of holes (B)</td>
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<td>1.4</td>
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<td>1.4</td>
<td>1.4</td>
<td>1.4</td>
</tr>
<tr>
<td>Column distance of holes (A)</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
<td>1.25</td>
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</tr>
</tbody>
</table>
Figure 5-10: Model 1 in FEM wire frame

Figure 5-11: Models 5, 13-18 in FEM wire frame
Figure 5-12: Models 19-27, FEM model

Figure 5-13: Models 33-38, shaded model

Figure 5-14: Models 33-38 in FEM wire frame
**Figure 5-15:** Models 39-41, half model

**Figure 5-16:** Models 39-41, full model in FEM wire frame

**Figure 5-17:** Models 39-41, full model
Figure 5-18: Models 42-44, half FEM model

Figure 5-19: Models 42-44, full model
will shut down automatically. Consequently, this default always resulted in some obstacles in FEM modeling. However, the detailed process is shown as follows:

1. Create geometry by using the functions of points, lines, circles, surfaces and solids, as well as Boolean operations.

2. Choose the element type, which is very important for latter mesh. If the wrong type is chosen, the mesher won’t work. (In this case, I selected SOLID72 element type to mesh the solids created by Boolean operation.)

3. Enter material properties in the form.

4. Select type of analysis and method, such as structural analysis and h- or p-method. (In this case, I chose h-method.)

5. Activate the Smart Mesher to create FEM model.

6. Apply relevant loads and boundary conditions on the model or finite elements. (Up to this step, we have completed the so-called pre-processing.)

7. Run the solution engine to obtain the results.

8. Review the results from post-processing about stress and deformation.

The related dimensions and shapes are shown in Table 5-2 and Figures 5-21 through 26.
Figure 5-21: Model 45
* Outer radius: Ra * Spherical thickness: T3

Figure 5-22: Models 46-47
Figure 5-23: Model 48 (Pentagon)

Figure 5-24: Model 49 (Hexagon)

Figure 5-25: Model 50 (Heptagon)

Figure 5-26: Model 51 (Octagon)
Table 5-2: Major Dimensions in 3-D Model Created by Ansys

<table>
<thead>
<tr>
<th>Mod No.</th>
<th>Wall thickness (T3)</th>
<th>Tank high (h)</th>
<th>Length of long axis of ellipse: (a)</th>
<th>Length of short axis of ellipse: (b)</th>
<th>Outer radius of sphere or Radius of polygon inscribed in the circle: (Ra)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mod 45</td>
<td>0.9 in</td>
<td>10 in</td>
<td>9.35 in</td>
<td>5.8 in</td>
<td>7 in</td>
</tr>
<tr>
<td>Mod 46</td>
<td>0.9 in</td>
<td>10 in</td>
<td>18.9 in</td>
<td>3.8 in</td>
<td>7.22 in</td>
</tr>
<tr>
<td>Mod 47</td>
<td>0.9 in</td>
<td>10 in</td>
<td></td>
<td></td>
<td>6.9 in</td>
</tr>
<tr>
<td>Mod 48</td>
<td>0.9 in</td>
<td>10 in</td>
<td></td>
<td></td>
<td>6.79 in</td>
</tr>
<tr>
<td>Mod 49</td>
<td>0.9 in</td>
<td>10 in</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mod 50</td>
<td>0.9 in</td>
<td>10 in</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mod 51</td>
<td>0.9 in</td>
<td>10 in</td>
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</table>
Chapter 6

RESULTS OF FEM ANALYSIS IN RECTANGULAR AND IRREGULAR
SYMMETRICAL TANK

6-1 Introduction

This chapter deals with the results of analysis that were developed using Patran and Ansys packages. Fifty-one models were created using different shapes, such as rectangular, irregular, spherical, elliptical and polygonal type.

Besides Aluminum-6061, Steel-1040 and composite material Kevlar/Steel-1040, two additional composite materials were used: Eglass/Steel-1040 and Eglass/Aluminum-7050. In boundary conditions (BC1 to BC7), several different types were applied, as shown in Tables 6-1 to 6-4, in order to match different types of geometry and to find the lowest stress in the distribution.

Finally, by changing geometry, materials and boundary conditions, effort was made to find the most appropriate type of tank which can meet the requirements to endure higher gas pressure, store more natural gas, possibly reduce the weight and deduct the material cost. Thus, NGV can be driven for a longer distance at an acceptable cost.

6-2 FEM Analysis of Non-cylindrical Model by Using Patran

Several factors can affect stress distribution, such as material, boundary condition and geometry including shape, fillet, hole and wall thickness. Based on changing these factors, this section was divided into two parts to develop the pressure vessels: rectangular tank and irregular tank.
Table 6-1: Positions of Constraints

<table>
<thead>
<tr>
<th>BC1: Model 1</th>
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</thead>
<tbody>
<tr>
<td><img src="image1" alt="Diagram of BC1: Model 1" /></td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>BC1: Model 2 – Model 18</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image2" alt="Diagram of BC1: Model 2 – Model 18" /></td>
</tr>
</tbody>
</table>
Table 6-2: Positions of Constraints

BC2: Model 19 – Model 32

XX: area of constraints at top and bottom surfaces.

(front view)

( top area)

front view

( bottom area )

BC2: Model 33 – Model 38

XX: area of constraints at top and bottom surfaces as same as previous one.
Table 6-3: Positions of Constraints

**BC3: Model 39 – Model 41 (half model)**

- **X**: area of constraints at top and bottom surfaces.
  - (top)
  - front view
  - (bottom)
  - (symmetrical line)
  - //: symmetrical area

**BC4: Model 42 - Model 44 (half model)**

- **X**: area of constraints at top and bottom surfaces.
  - (top)
  - front view
  - (bottom)
  - (symmetrical line)
  - //: symmetrical area
Table 6-4: Positions of Constraints

BC5: Model 45
Top view of spherical tank
(symmetrical line)

BC6: Model 46 - Model 47
Constraint edges

BC7: Model 48
Constraints edges

BC7: Model 49
Constraint edges

BC7: Model 50
Constraint edges

BC7: Model 51
Constraint edges
In a rectangular tank, twelve different tanks (Models 1-12) were created by Patran to find the relationship between maximum stress and hole arrangement. Some models in stress distribution were shown in Figures 6-1 to 6-3. These models were built up in 3-D and B-spline solid by varying hole arrangement, hole diameter and wall thickness. Tet element type was used to mesh these models. For element size, 1.0, 0.7, 0.6, 0.55, 0.5 and 0.45 were used in the first model, and the maximum stress converged when element size was at 0.5 and 0.45. Hence, element size 0.5 was chosen for further development.

The internal pressure and boundary conditions were fixed at 1000 psi and BC1 (Table 6-1). Aluminum-6061 was applied in these twelve models. After creating geometry, meshing, entering material data, applying property and forming the load case were finished, the FEM models were analyzed and the results of stress variation were checked--in particular, focused on the maximum and minimum stress, as shown in Table 6-5.

Finally, the better model (Figure 6-2) was selected according to the lowest stress at internal pressure 1000psi. Based on this model, the wall thickness was changed using several values--1.0, 0.9 and 0.8 inches--to find the effect of wall thickness on stress distribution. By comparing the results of stress distribution from the three models, Model 5 was chosen as its maximum stress is the lowest.

In Model 5, several different materials were reapplied, such as Steel-1040, Kevlar/Steel-1040, Eglass/Steel-1040 and Eglass/Aluminum-7050, to find the stress variation according to the different strength of materials. Moreover, the internal pressures (Tables 6-6) were increased until the maximum stress approaching their yielding points.
Table 6-5: Maximum and Minimum Stresses Developed Using Patran

*Package of Finite Element Analysis: Patran 5.0, *Boundary Condition: BC #

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<th>Material:</th>
<th>Aluminum 6061-t6</th>
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<tr>
<td>Model Choice:</td>
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<td>Yield Strength:</td>
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<table>
<thead>
<tr>
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<th>Internal Pressure (psi)</th>
<th>Max. Stress (psi)</th>
<th>Min. Stress (psi)</th>
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<td>Model 4 (BC1)</td>
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<td>21806</td>
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<td>Model 5 (BC1)</td>
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(Table 6-5 Cont.)

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<tr>
<td>Model 29 (BC2)</td>
<td>5500</td>
<td>49491</td>
<td>85</td>
</tr>
<tr>
<td>Model 35 (BC2)</td>
<td>5500</td>
<td>44181</td>
<td>143</td>
</tr>
<tr>
<td>Model 40 (BC3)</td>
<td>5500</td>
<td>23464</td>
<td>89</td>
</tr>
<tr>
<td>Model 42 (BC4)</td>
<td>5500</td>
<td>21727</td>
<td>341</td>
</tr>
</tbody>
</table>

Material: Steel-AISI-1040

Model Choice: Model 15, 23, 29, 35, 40, 42

Yield Strength: 64000 psi
(Table 6-5 Cont.)

<table>
<thead>
<tr>
<th>Material:</th>
<th>Kevlar / Steel-AISI-1040 (laminate)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model Choice:</td>
<td>Model 16, 17, 24, 25, 36</td>
</tr>
<tr>
<td>Combined Yield Strength:</td>
<td>154600 psi</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Model # and BC #</th>
<th>Ip (psi)</th>
<th>Max. (psi)</th>
<th>Min. (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 16 (BC1)</td>
<td>8000</td>
<td>183056</td>
<td>9699</td>
</tr>
<tr>
<td>Model 17 (BC1)</td>
<td>15000</td>
<td>343230</td>
<td>18185</td>
</tr>
<tr>
<td>Model 24 (BC2)</td>
<td>15000</td>
<td>149783</td>
<td>182</td>
</tr>
<tr>
<td>Model 36 (BC2)</td>
<td>15000</td>
<td>134409</td>
<td>664</td>
</tr>
<tr>
<td>Model 25 (BC1)</td>
<td>15600</td>
<td>155775</td>
<td>189</td>
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</table>

<table>
<thead>
<tr>
<th>Material:</th>
<th>Eglass / Steel-1040 (laminate)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model Choice:</td>
<td>Model 18, 26, 37, 41, 43</td>
</tr>
<tr>
<td>Combined Yield Strength:</td>
<td>214440 psi</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Model # and BC #</th>
<th>Ip (psi)</th>
<th>Max. (psi)</th>
<th>Min. (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 18 (BC1)</td>
<td>8000</td>
<td>182888</td>
<td>9698</td>
</tr>
<tr>
<td>Model 26 (BC2)</td>
<td>20000</td>
<td>201100</td>
<td>245</td>
</tr>
<tr>
<td>Model 31 (BC2)</td>
<td>20000</td>
<td>202521</td>
<td>434</td>
</tr>
<tr>
<td>Model 37 (BC2)</td>
<td>20000</td>
<td>180525</td>
<td>881</td>
</tr>
<tr>
<td>Model 41 (BC3)</td>
<td>20000</td>
<td>96134</td>
<td>118</td>
</tr>
<tr>
<td>Model 43 (BC4)</td>
<td>20000</td>
<td>77076</td>
<td>1339</td>
</tr>
</tbody>
</table>
**Table 6-5 Cont.**

<table>
<thead>
<tr>
<th>Material:</th>
<th>Eglass / Aluminum-7050 (laminate)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model choice:</td>
<td>Model 27, 32, 38, 44</td>
</tr>
<tr>
<td>Combined yield strength:</td>
<td>561100 psi</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Model # and BC #</th>
<th>Ip (psi)</th>
<th>Max. stress (psi)</th>
<th>Min. stress (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 27 (BC1)</td>
<td>21900</td>
<td>221396</td>
<td>265</td>
</tr>
<tr>
<td>Model 32 (BC2)</td>
<td>21900</td>
<td>223090</td>
<td>437</td>
</tr>
<tr>
<td>Model 38 (BC2)</td>
<td>21900</td>
<td>198817</td>
<td>962</td>
</tr>
<tr>
<td>Model 44 (BC4)</td>
<td>21900</td>
<td>84879</td>
<td>1430</td>
</tr>
</tbody>
</table>

**Figure 6-1:** Stress distribution of Model 4 (Aluminum 6061-t6)
Square pattern: D1 = 0.8 in
Figure 6-2: Stress distribution of Model 5 (Aluminum 6061-t6)
Rectangular pattern: D1 = 0.8 in

Figure 6-3: Stress distribution of Model 8 (Aluminum 6061-t6)
Diamond pattern: D1 = 0.8 in
Table 6-6: The Load as Internal Pressure

<table>
<thead>
<tr>
<th>Material: Aluminum-6061-t6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model No.</td>
</tr>
<tr>
<td>Model 1 – 12</td>
</tr>
<tr>
<td>Model 14, 19, 33</td>
</tr>
<tr>
<td>Model 13</td>
</tr>
<tr>
<td>Model 45 – 47</td>
</tr>
<tr>
<td>Model 20, 28, 34, 39, 48 - 51</td>
</tr>
<tr>
<td>Model 21</td>
</tr>
<tr>
<td>Model 22</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Material: Steel-1040</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 15</td>
</tr>
<tr>
<td>Model 23, 29, 35, 40, 42</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Material: Kevlar / Steel-1040</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 16</td>
</tr>
<tr>
<td>Model 17, 24, 30, 36</td>
</tr>
<tr>
<td>Model 25</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Material: Eglass / Steel-1040</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model No.</td>
</tr>
<tr>
<td>Model 18</td>
</tr>
<tr>
<td>Model 26, 31, 37, 41, 43</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Material: Eglass / Aluminum-7050</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 27, 32, 38, 44</td>
</tr>
</tbody>
</table>
The results of stress are shown in the Table 6-5. The total number of nodes and elements with respect to each model were listed in Tables 6-7a.

In dealing with irregular shapes (Models 19-44), the focus was on the shape variation and composite materials. The internal pressures and boundary conditions were also varied accordingly, as listed in Tables 6-2, 3, 6. Because of symmetrical geometry and insufficient working memory on the workstation for analysis and post-processing, some of the models (Models 39-44) were used in half-model.

After 3-D B-spline models were created, Tet element type was applied. Hence, nearly the total number of nodes and elements for each model are listed in Table 6-7b and 7c. By following the same procedure as mentioned above, the results of stress distribution were produced. The maximum and minimum stress with different materials and internal pressures are listed in Table 6-5. The colorful stress distributions are shown in Figures 6-4 to 9.

6-3 FEM Analysis of Other Simple Geometry Models by Using Ansys

The reasons behind the development of this section were ease in creating the geometry of sphere, ellipse, and polygon by using Ansys drawing tools, and to present ideas on developing spherical and elliptical tanks. Moreover, the development of Chen's rectangular tank inspired the development of other types of polygonal pressure vessels.

Based on the above reasons, several models were created, as shown in Models 45 to 51. After an appropriate element type--SOLID 72--was selected, the function of
Table 6-7a: Total Numbers of Node and Tet Element

<table>
<thead>
<tr>
<th>Model No.</th>
<th>Total Numbers of Node</th>
<th>Total Numbers of Element</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 1</td>
<td>9015</td>
<td>37252</td>
</tr>
<tr>
<td>Model 2</td>
<td>9169</td>
<td>35996</td>
</tr>
<tr>
<td>Model 3</td>
<td>10763</td>
<td>42660</td>
</tr>
<tr>
<td>Model 4</td>
<td>9121</td>
<td>36400</td>
</tr>
<tr>
<td>Model 5, 13, 14</td>
<td>9145</td>
<td>36472</td>
</tr>
<tr>
<td>Model 6</td>
<td>10549</td>
<td>41748</td>
</tr>
<tr>
<td>Model 7</td>
<td>9187</td>
<td>36156</td>
</tr>
<tr>
<td>Model 8</td>
<td>9267</td>
<td>36924</td>
</tr>
<tr>
<td>Model 9</td>
<td>10753</td>
<td>42776</td>
</tr>
<tr>
<td>Model 10</td>
<td>9127</td>
<td>36016</td>
</tr>
<tr>
<td>Model 11</td>
<td>8643</td>
<td>33128</td>
</tr>
<tr>
<td>Model 12</td>
<td>8489</td>
<td>32180</td>
</tr>
<tr>
<td>Model No. and Material</td>
<td>Node numbers</td>
<td>Element numbers</td>
</tr>
<tr>
<td>Model 15 (Steel-AISI-1040)</td>
<td>9145</td>
<td>36472</td>
</tr>
<tr>
<td>Model 16,17 (Kevlar / Steel-1040)</td>
<td>9145</td>
<td>36472</td>
</tr>
<tr>
<td>Model 18 (Eglass / Steel-1040)</td>
<td>9145</td>
<td>36472</td>
</tr>
</tbody>
</table>

Material: Aluminum-6061-t6
<table>
<thead>
<tr>
<th>Model No. and Material</th>
<th>Node Numbers</th>
<th>Element Numbers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 19-22 (Aluminum 6061-t6)</td>
<td>8411</td>
<td>30556</td>
</tr>
<tr>
<td>Model 23-25 (Steel-1040)</td>
<td>8411</td>
<td>30556</td>
</tr>
<tr>
<td>Model 26 (Eglass / Steel-1040)</td>
<td>8411</td>
<td>30556</td>
</tr>
<tr>
<td>Model 27 (Eglass / Aluminum 7050)</td>
<td>8411</td>
<td>30556</td>
</tr>
<tr>
<td>Model 28 (Aluminum 6061-t6)</td>
<td>9321</td>
<td>34604</td>
</tr>
<tr>
<td>Model 29 (Steel-1040)</td>
<td>9321</td>
<td>34604</td>
</tr>
<tr>
<td>Model 30 (Kevlar / Steel-1040)</td>
<td>9321</td>
<td>34604</td>
</tr>
<tr>
<td>Model 31 (Eglass / Steel-1040)</td>
<td>9321</td>
<td>34604</td>
</tr>
<tr>
<td>Model 32 (Eglass / Aluminum 7050)</td>
<td>9321</td>
<td>34604</td>
</tr>
<tr>
<td>Model 33,34 (Aluminum 6061-t6)</td>
<td>9707</td>
<td>36408</td>
</tr>
<tr>
<td>Model 35 (Steel-1040)</td>
<td>9707</td>
<td>36408</td>
</tr>
<tr>
<td>Model 36 (Kevlar / Steel-1040)</td>
<td>9707</td>
<td>36408</td>
</tr>
<tr>
<td>Model 37 (Eglass / Steel-1040)</td>
<td>9707</td>
<td>36408</td>
</tr>
<tr>
<td>Model 38 (Eglass / Aluminum 7050)</td>
<td>9707</td>
<td>36408</td>
</tr>
<tr>
<td>Model 39 – half (Aluminum 6061-t6)</td>
<td>6329</td>
<td>23374</td>
</tr>
<tr>
<td>Model 40 – half (Steel-1040)</td>
<td>6329</td>
<td>23374</td>
</tr>
<tr>
<td>Model 41 – half (Eglass/Steel-1040)</td>
<td>6329</td>
<td>23374</td>
</tr>
</tbody>
</table>
### Table 6-7c: Total Numbers of Node, Tet and SOLID72 Element

<table>
<thead>
<tr>
<th>Model No. and Material</th>
<th>Total Numbers of Node</th>
<th>Total Numbers of Element (Tet)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 42, half (Steel-1040)</td>
<td>6656</td>
<td>24920</td>
</tr>
<tr>
<td>Model 43, half (Eglass / Steel-1040)</td>
<td>6656</td>
<td>24920</td>
</tr>
<tr>
<td>Model 44, half (Eglass / Aluminum 7050)</td>
<td>6656</td>
<td>24920</td>
</tr>
<tr>
<td>(From Ansys Package)</td>
<td></td>
<td>(Solid72)</td>
</tr>
<tr>
<td>Model 45, ¾th part (Aluminum 6061-t6)</td>
<td>643</td>
<td>1837</td>
</tr>
<tr>
<td>Model 46 (Aluminum 6061-t6)</td>
<td>1622</td>
<td>4815</td>
</tr>
<tr>
<td>Model 47 (Aluminum 6061-t6)</td>
<td>2087</td>
<td>6504</td>
</tr>
<tr>
<td>Model 48 (Aluminum 6061-t6)</td>
<td>160</td>
<td>380</td>
</tr>
<tr>
<td>Model 49 (Aluminum 6061-t6)</td>
<td>186</td>
<td>444</td>
</tr>
<tr>
<td>Model 50 (Aluminum 6061-t6)</td>
<td>259</td>
<td>645</td>
</tr>
<tr>
<td>Model 51 (Aluminum 6061-t6)</td>
<td>344</td>
<td>880</td>
</tr>
</tbody>
</table>
Figure 6-4: Stress distribution of Model 25 (Kevlar/Steel-1040)

Figure 6-5: Stress distribution of Model 37 (Eglass/Steel-AISI-1040)
Figure 6-6: Stress distribution of Model 38 (Eglass/Aluminum-7050)

Figure 6-7: Stress distribution of Model 41 (Eglass/Steeel-AISI-1040)
Half model
**Figure 6-8:** Stress distribution of Model 42 (Steel-AISI-1040)
Half model

**Figure 6-9:** Stress distribution of Model 44 (Eglass/Aluminum-7050)
Half model
smart mesh was activated to form the FEM model, and then internal loads, constraints and material properties were applied. When the process was analyzed, the function of the solution was started to get the final results. Finally, the function of the general post-processor helped to review the animation, the deformation and the stress distribution. The colorful stress diagrams are shown in Figures 6-10 to 16. The relevant boundary conditions, internal pressures, maximum and minimum stresses are listed in Tables 6-4, 6 and 8. The total numbers of node and element are shown in Tables 6-7c.

Since the right to use Ansys package had passed its expiration date and the Russ College of Engineering and Technology at Ohio University where the research was conducted has no intention of further supporting this software, the results from Ansys are limited. For example, the internal loads were just stopped on 3000 and 3500 psi without an attempt to find higher pressures, and for material, only aluminum-6061 was tried.

6-4 Final Design

The main emphasis in this research is to improve the NGV's fuel tank, which can resolve NGV's problems as mentioned before in three ways: to store more natural gas, to endure higher gas pressure and to keep the fuel tank as light and inexpensive as possible. Based on Chen's previous models--cylindrical tank and rectangular tank--some models were developed in several directions.

In geometry, the fillet has the effect of reducing the maximum stress. A fin can enforce the structure of the tank and it can also reduce the maximum stress. A larger hole
diameter can cause more serious stress concentration; however, the more holes, the more gas can be stored. A thicker wall can strengthen the tank structure but would increase the

Table 6-8: Maximum and Minimum Stresses Developed Using Ansys

<table>
<thead>
<tr>
<th>Material:</th>
<th>Aluminum-6061-t6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model choice:</td>
<td>Model 45, 46, 47, 48, 49, 50, 51</td>
</tr>
<tr>
<td>Yield strength:</td>
<td>36000 psi</td>
</tr>
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<table>
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</tr>
</thead>
<tbody>
<tr>
<td>45</td>
<td>46</td>
<td>47</td>
<td>48</td>
<td>49</td>
<td>50</td>
<td>51</td>
<td></td>
</tr>
<tr>
<td>Ip (psi)</td>
<td>3000</td>
<td>3000</td>
<td>3000</td>
<td>3500</td>
<td>3500</td>
<td>3500</td>
<td></td>
</tr>
<tr>
<td>Max. Stress (psi)</td>
<td>25645</td>
<td>36445</td>
<td>273848</td>
<td>50194</td>
<td>40299</td>
<td>35333</td>
<td></td>
</tr>
<tr>
<td>Min. Stress (psi)</td>
<td>4782</td>
<td>1572</td>
<td>12595</td>
<td>3519</td>
<td>3377</td>
<td>4022</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3000</td>
<td>3000</td>
<td>3500</td>
<td>3500</td>
<td>3500</td>
<td>3500</td>
<td></td>
</tr>
</tbody>
</table>
Figure 6-10: Result of Model 45 by Ansys

Figure 6-11: Result of Model 46 by Ansys
Figure 6-12: Result of Model 47 by Ansys

Figure 6-13: Result of Model 48 by Ansys
Figure 6-14: Result of Model 49 by Ansys

Figure 6-15: Result of Model 50 by Ansys

Figure 6-16: Result of Model 51 by Ansys
weight of the tank. Therefore, a kind of smooth wave surface with a middle wall thickness (0.9 in) was applied. By regulating the positive and negative effect on stress distribution, the optimal models were created. These models included the most fillets and holes, the smaller hole-diameter (0.8 in) and the moderate wall thickness (0.9 in), as well as the smooth wave surface (see Models 42 to 44).

The shapes of tank and boundary conditions are another key factor which can affect the result of analysis. For example, from the results of Ansys it can be seen that the spherical shape is stronger than the elliptical shape and the elliptical shape with the ratio of $b/a < 1$ is stronger than $b/a \ll 1$, if $a$ is the length of long axis and $b$ is the length of short axis. Moreover, in the polygons, the pentagon is weaker than the hexagon and the hexagon is weaker than the heptagon while the heptagon is weaker than the octagon. Hence, according to this tendency, a conclusion about the strength of structure can be induced as rectangle < pentagon < hexagon < heptagon < octagon < cylinder.

Actually, the rectangular tank is not only the weakest structure but also is unsuitable for proper boundary conditions, such as constraints, to be applied. So, based on these considerations, two types of irregular shape were developed in order to find a stronger structure, as well as to apply appropriate boundary conditions easily. Then, the previous factors of geometry were combined with the factors of irregular shapes and boundary conditions to find the optimal models in this study (Models 42-43), as shown in Figures 6-8 and 5-26.

Up to this point, the only factor not considered is the material property. Aluminum-6061 (density=0.098 pci) is lighter than Steel-1040 (density=0.284 pci), but
its yield strength is only 36000 psi, which is not suitable for higher gas pressure. Steel-1040, whose yield strength is 64000 psi, is always regarded as suitable material for a pressure vessel because it is cheaper and stronger than aluminum. However, it is too heavy and will cause vehicle to waste much fuel. Therefore, a composite material becomes the most attractive option.

Laminate, Kevlar/Steel-1040 discussed in Chen’s paper consists of two sheets of Kevlar and one sheet of Steel-1040. Its combined yield strength is 154600 psi and density is 0.209 lb/in³. This material is lighter and stronger than steel, whereas its material cost is more expensive than the other two materials (see: Table 3-4). So, two kinds of laminates were developed in this process: Eglass/Steel-1040 and Eglass/Aluminum-7050.

Fundamentally, the structures of these two laminates are the same as Kevlar/Steel-1040--two sheets (0.5 inches each) of glass fiber and one sheet (0.6 inches) of steel or aluminum. From the method mentioned in Chapter 3, the density and combined yield strength of the two laminates can be found:

- Eglass/Steel-1040: density = 0.2072 pci, yield strength = 231232 psi
- Eglass/Aluminum-7050: density = 0.098 pci, yield strength = 231832 psi

Comparing these two laminates with Kevlar/Steel-1040, it can be concluded that in weight, Kevlar/Steel-1040 is heavier than Eglass/Steel-1040 and Eglass/Steel-1040 is heavier than Eglass/Aluminum-7050. In strength, Eglass/Aluminum-7050 is stronger than Kevlar/Steel-1040 by 1.5 times and Eglass/Steel-1040 is stronger than Kevlar/Steel-1040 by 1.4 times. From the price list of each material shown in Table 3-4 and using simple calculation, we can find the material cost of these three laminates, as follows:
1. Eglass/Steel-1040

- Layer sequence: Eglass / Steel / Eglass
- Thickness: 0.15 / 0.6 / 0.15 in
- Assumption: The surface area of rectangular tank = 960 in²
- Steel: price = 0.3 $/lb, density = 0.284 lb/in³
- Glass fiber: price = 1.0 $/lb, density = 0.092 lb/in³
- Calculation: combined cost = 0.3 $/lb. * 0.284 lb/in³ * 0.6 in * 960 in² + 1.0 $/lb * 0.092 lb/in³ * 0.3 in * 960 in² = $ 75.84

2. Eglass/Aluminum-7050

- Layer sequence: Eglass / Aluminum / Eglass
- Thickness: 0.15 / 0.6 / 0.15 in
- Assumption: The surface area of rectangular tank = 960 in²
- Aluminum: price = 1.1 $/lb, density = 0.102 lb/in³
- Calculation: combined cost = 1.0 $/lb * 0.092 lb/in³ * 0.3 in * 960 in² + 1.1 $/lb * 0.102 lb/in³ * 0.6 in * 960 in² = $ 91.2

3. Kevlar/Steel-1040

- Layer sequence: Kevlar / Steel / Kevlar
- Thickness: 0.2 / 0.6 / 0.2 in
- Assumption: The surface area of rectangular tank = 960 in²
- Kevlar: price = 18 $/lb, density = 0.0362 lb/in³
- Calculation: combined cost = 18 $/lb * 0.0362 lb/in³ * 0.4 in * 960 in² + 0.3 $/lb * 0.284 lb/in³ * 0.6 in * 960 in² = $ 299.52

From the above results, we can see Kevlar/Steel is more expensive than Eglass/Aluminum by 3.3 times, and Eglass/Aluminum is more expensive than
Eglass/Steel by 1.2 times. Consequently, Eglass/Aluminum was chosen as the material in this procedure because of its high yield strength, acceptable cost and light weight.

To sum up, after consideration was made on the fuel capacity, strength, cost and weight of tank, Model 44 (Figure 6-9) was selected as the best model. We can see from the stress distribution that if internal pressure 21900 psi is applied, the maximum stress is only 84879 psi, which is much lower than Chen's last model (ip=15000 psi, max. stress = 130854 psi). A flow chart concludes this thesis to illustrate how to develop the design and analysis, as shown in Figure 6-17.
Purpose: Looking for a natural gas tank which can meet several requirements:
   a. Endure higher pressure.
   b. Be stored more natural gas.
   c. Be weight and cost less.
   d. Approach to the irregular and non-symmetrical shape.
   e. Increase NGV's driving distance.

Parameter: 1- Geometry :
   a. Shape
   b. Fillet
   c. Fin
   d. Hole Diameter
   e. Hole Arrangement
   f. Shell Thickness

Parameter: 2- Material (MP):
   a. Al6061-t6
   b. Steel 1040
   c. Kevlar-Steel1040
   d. Eglass-Steel1040
   e. Egalss-Al17050

Parameter: 3- Boundary Conditions :
   a. Constraints (BC #)
   b. Internal Pressure
   c. External Pressure

Volume = constant

MP: Al6061-t6
Software: Patran

Model 1: Parameter: 1-a: rectangular shape
   1-b: 0.2 in
   1-c: 0.25 in
   1-d: 1.0 in
   1-e: 5-hole pattern
   1-f: 1 in

Result:

Model 2:
   Change
   1-a: diam.
   1-e: square pattern
Result:

Model 3:
   Change
   1-a: diam.
   1-e: square pattern
Result:

Model 4:
   Change
   1-a: diam.
   1-e: square pattern
Result:

Model 5:
   Change
   1-a: diam.
   1-e: square pattern
Result:

Model 6:
   Change
   1-a: diam.
   1-e: square pattern
Result:

Model 7:
   Change
   1-a: diam.
   1-e: square pattern
Result:

Model 8:
   Change
   1-a: diam.
   1-d: d1=.8
   1-e: diam.
   1-d: d2=.35
   Result:

Model 9:
   Change
   1-a: diam.
   1-d: d1=.9
   1-e: diam.
   1-d: d2=.35
   Result:

Model 10:
   Change
   1-a: diam.
   1-d: d1=1
   1-e: diam.
   1-d: d2=.35
   Result:

(Sm1: maximum stress, Sm2: minimum stress)

Figure 6-17: Flow chart
Compare the results and chose the best model:
Model 5

Model 11:
Change:
1-f: t=0.9 in
Result:
Sm1=24155 psi
Sm2=1214 psi

Model 12:
Change:
1-f: t=0.8 in
Result:
Sm1=25394 psi
Sm2=1555 psi

Compare the results in Model 5, Model 11 and Model 12:
Model 5 is the choice

(Figure 6-17 Cont.)
Model 19:
Change:
1-a: irregular 1
1-f: 0.9 in
3-a: BC2
3-b: 1500 psi
Result:
Sm1=14065
Sm2=26.88

Model 20:
Change:
3-b: 3500 psi
Result:
Sm1=32817
Sm2=63

Model 21:
Change:
3-b: 4500 psi
Sm1=42194
Sm2=81

Model 22:
Change:
3-b: 5000 psi
Sm1=46882
Sm2=89.58

Model 23:
Change:
1-b: add fillets
Result:
Sm1=49491
Sm2=85

Model 24:
Change:
1-a: irregular 1
1-f: 0.9 in
3-a: BC2
3-b: 15000 psi
Result:
Sm1=149783
Sm2=182

Model 25:
Change:
3-b: 15600 psi
Result:
Sm1=27865
Sm2=94.54

Model 26:
Change:
3-b: 20000 psi
Result:
Sm1=37945
Sm2=137.6

Model 27:
Change:
3-b: 25000 psi
Result:
Sm1=48340
Sm2=181.7

Model 28:
Change:
3-b: 30000 psi
Result:
Sm1=59198
Sm2=225.9

Model 29:
Change:
1-b: add fillets
Result:
Sm1=14065
Sm2=26.88

Model 30:
Change:
1-b: Add fillets
Result:
Sm1=202521
Sm2=440

Model 31:
Change:
1-a: irregular 1
1-f: 0.9 in
3-a: BC2
3-b: 15000 psi
Result:
Sm1=149783
Sm2=182

Model 32:
Change:
3-a: BC2
3-b: 15000 psi
Result:
Sm1=27865
Sm2=94.54

Model 33:
Change:
3-b: 20000 psi
Result:
Sm1=37945
Sm2=137.6

Model 34:
Change:
1-a: irregular 1
1-w: wave
3-b: 3500 psi
Result:
Sm1=27865
Sm2=94.54

Model 35:
Change:
3-b: 5500 psi
Result:
Sm1=44181
Sm2=143.2 psi

Model 36:
Change:
3-b: 15000 psi
Result:
Sm1=150739
Sm2=366

Model 37:
Change:
1-a: irregular 1
1-w: wave
3-b: 20000 psi
Result:
Sm1=180525
Sm2=880.8

Model 38:
Change:
1-a: irregular 1
1-w: wave
3-b: 21900 psi
Result:
Sm1=198817
Sm2=962.2

Model 41:
Change:
1-a: irregular 1
1-c: Add fins
3-a: BC3
3-b: 20000 psi
Result:
Sm1=96134 psi
Sm2=117.6 psi

Model 44 (half):
Change:
1-a: irregular 2
1-c: Add fins
3-a: BC4
3-b: 21900 psi
Result:
Sm1=84879 psi
Sm2=1430 psi

(Figure 6-17 Cont.)
After comparing the upper results, Model 44 was chosen as the final model which can meet the requirements.

(Figure 6-17 Cont.)
Additional development using
Ansys package

Model 45:
Change:
1-a: sphere
ir=6.1, or=7
1-f: t=0.9 in
3-a: BC5
3-b: 3000 psi

Result:
Sm1=25645 psi
Sm2=4782 psi

Model 46:
Change:
1-a: ellipse
long axis: 7.55 in
short axis: 4 in
1-f: t=0.9 in
3-a: BC6
3-b: 3000 psi

Result:
Sm1=36445 psi
Sm2=1572 psi

Model 47:
Change:
1-a: ellipse
long axis: 15.1 in
short axis: 2 in

Result:
Sm1=273848 psi
Sm2=12595 psi

Model 48:
Change:
1-a: pentagon
1-f: 0.9 in
3-a: BC7
3-b: 3500 psi

Result:
Sm1=50194 psi
Sm2=3519 psi

Model 49:
Change:
1-a: hexagon

Result:
Sm1=40299 psi
Sm2=3377 psi

Model 50:
Change:
1-a: heptagon

Result:
Sm1=35333 psi
Sm2=4022 psi

Model 51:
Change:
1-a: octagon

Result:
Sm1=32966 psi
Sm2=5849 psi

(Figure 6-17 Cont.)
7-1 Discussion 1 — How far can it drive?

By applying the method quoted in Chapter 2 [6], a simple calculation is offered here in order to find the new travel range of a vehicle using the final model (Model 44) for a natural gas tank.

[A] Given:

1. Lower heating value: 45 MJ/kg for natural gas, 44 MJ/kg for gasoline
2. Molecular weight of natural gas: 18 kg/kg-mol, gas constant \( R \): 461.9 J/kg-k
3. Fuel consumption rate: 15 mile/gallon, (MPG), gasoline density: 730 kg/m\(^3\)
4. Capacity (V) of Model 44: 949.2 in\(^3\) = 0.1544 m\(^3\)
5. Initial pressure of tank (Pin): 11000 psi = 758.42 \( \times 10^5 \) pa
6. Final pressure of tank (Pfn): 300 psi = 20.68 \( \times 10^5 \) pa
7. Compression factor: \( Z_{in} = 1.5 \) for initial state, \( Z_{fn} = 0.96 \) for final state
8. Natural gas as an ideal gas at temperature (T), 300 °k

[B] Aim: find travel range (Trav) in miles.

[C] Calculation:

- Volume of gasoline required: Trav (miles) / 15 (PMG) \( \times 3.785412 \times 10^3 \) (m\(^3\))

- Mass of gasoline required: Trav (miles) / 15 (PMG) \( \times 3.785412 \times 10^3 \) (m\(^3\)) \( \times 730 \) (kg/m\(^3\)) = 0.1842 * Trav (kg)
- Natural gas required in mass: \(0.1842 \times \text{Trav (kg)} \times 44 \text{ (MJ/kg)} / 45 \text{ (MJ/kg)} = 0.1801 \times \text{Trav (kg)}\)
- Mass (initial) – Mass (final) = \((\text{Pin} \times V) / (\text{Zin} \times R \times T) – (\text{Pfn} \times V) / (\text{Zfn} \times R \times T) = 0.1801 \times \text{Trav}\)

\[\Rightarrow \text{Trav} = 0.154 \times 10^5 \times (758.42/1.5 - 20.68/0.96) / (461.9 \times 300 \times 0.1801) = 299.44 \text{ miles}\]

So, the new travel range is 299.4 miles, which is higher than the maximum travel range of 250 miles as determined in the competition of NGV in 1991 by 1.2 times. Moreover, in this case, the initial working pressure is just assumed at 11,000 psi. If the safety factor is 6, then the maximum allowable working pressure is yield strength (231,832 psi for Eglass/Aluminum-7050) divided by 6, which is equal to 38,638 psi. Therefore, the maximum travel range must be much greater than 299.4 miles, accordingly.

In considering the installation space of the vehicle, an example of a car with critical dimensions is given as follows:

- LEXUS - LS400: five passenger sedan
- Over length: 196.7 in
- Width: 72 in
- Height: 55.9 in

The dimensions of last model (Model 44):
- Length: 24in
- Height: 8in
- Width: 20in
Therefore, by comparing the upper dimensions, Model 44 tank can be installed on the underbody without losing the trunk space.

**7-2 Discussion 2 – Is there a possibility of failure in the last several models?**

In considering the homogeneous and isotropic material, such as alloy, Steel-AISI-1040, and the failure theories, maximum stress and maximum shear stress as discussed in Chapter 2 [11] can be applied in an example of Model 42. The model can be checked whether it will fail or not at the initial pressure, 5500 psi. Furthermore, safety factor and allowable stress can be defined according to the ASME code [9].

From the stress results of Model 42 (Figures 7-1, 2, 3), the maximum three principal stresses are: \( \sigma_1 = 18,651 \) psi, \( \sigma_2 = 8,701 \) psi, \( \sigma_3 = 2,521 \) psi. The yield strength (Fy) of steel-1040 is 64,000 psi. So,

- Maximum stress theory: \( \sigma_1 > \sigma_2 > \sigma_3 \Rightarrow \sigma_1 = 18651 \text{ psi} < \text{Fy} = 64000 \text{ psi} \).
- Maximum shear stress theory: \( \sigma_1 - \sigma_3 = 16130 \text{ psi} < \text{Fy} = 64000 \text{ psi} \).
- Maximum combined stress: \( \sigma = 21727 \text{ psi} < \text{Fy} = 64000 \text{ psi} \).

Hence, Model 42 will not break under the initial pressure, \( \text{ip} = 5500 \) psi. According to the ASME code, any combination of membrane stress and bending stress should not exceed 1.5 times the table value. If the table value is allowable stress \( \sigma_a \), and the maximum stress \( \sigma \) of Model 42 is 21,727 psi, then

- \( \sigma = 21727 \leq 1.5 \times \sigma_a \Rightarrow \sigma_a \geq 14484.7 \text{ psi} \)
- Safety factor \( n \) = \( \frac{\text{Fy}}{\sigma_a} \Rightarrow n = \frac{64000}{14484.7} = 4.4 \)
If considering the stability of the material for practical applications, the weight of the tank and the amount of material used in manufacturing, the safety factor (n) can be decreased to two. So, the allowable stress (σa) is equal to 32,000 psi (yield strength divided by two). Then, the maximum stress (σ) is equal to 48,000 psi (allowable stress multiply by 1.5). Therefore, the initial pressure can be increased higher than 5,500 psi.

In considering a composite material, such as laminate E-glass / Aluminum-7050, an example of Model 44 was assumed as a kind of homogeneous and orthotropic material. So, by applying the failure theory--maximum stress theory as stated in Chapter 2--some descriptions were made as follows:

- **Condition:** θ (orientation) = 45°
  
  σx (max. major principal stress) = 66,835 psi (see: Figure 7-4)

- **Maximum stress theory:**

  \[ σ_L \text{ (longitudinal allowable tensile strength)} ≤ σ_{LU} \text{ (longitudinal tensile strength)} \]
  
  \[ σ_T \text{ (transverse allowable tensile strength)} ≤ σ_{TU} \text{ (transverse tensile strength)} \]
  
  \[ τ_{LT} \text{ (allowable shear strength)} ≤ τ_{LTU} \text{ (shear strength)} \]

- **Calculation:**

  \[ σ_{LU} = 150 \times 10^3 \text{ psi, } σ_{TU} = 7 \times 10^3 \text{ psi, } τ_{LTU} = 10*10^3 \text{ psi (ref: Table 3-2)} \]
  
  \[ ⇒ σ_L = σx \times (\cosθ)^2 = 66835 \times (\cos45°)^2 = 33.4 \times 10^3 \text{ psi} < σ_{LU} \]
  
  \[ σ_T = σx \times (\sinθ)^2 = 33.4 \times 10^3 > σ_{TU} \]
  
  \[ τ_{LT} = σx \times (\sinθ \times \cosθ) = 33.4 \times 10^3 > τ_{LTU} \]
**Figure 7-1**: Model 42 for major principal stress

**Figure 7-2**: Model 42 for intermediate principal stress
Figure 7-3: Model 42 for minor principal stress

Figure 7-4: Model 44 for major principal stress
Therefore, when considering first ply failure (FPT), the maximum principal stress, $\sigma_x$, has caused part of plies to fail. However, the combined yield strength of laminate in Model 44 is 231,832 psi, so it can still resist internal pressure up to 21,900 psi without a potential failure.

7-3 Conclusion

A natural gas tank is one of the important elements of a natural gas vehicle. It controls whether a NGV can drive as long a distance as a gasoline or diesel vehicle. Hence, the design and analysis of the fuel vessel becomes significant.

Basically, the natural gas tank of a NGV can be viewed as a pressure vessel, so some of the theories about pressure vessel can be applied to this study. Two FEM computer packages—Patran 5.0 and Ansys 3.0—were applied in this study to find a better model of natural gas tank. The present research developed from Chen's study in several ways:

- Consideration of theory: Four methods—small deflection, large deflection, ASME code, and FEM theory—were applied on a simple rectangular frame with same boundary conditions and loads. The results of the maximum stress from each method were found to be reasonably close. The average value is 91 kpsi.

- Consideration of material: Steel-1040 is a conventional material used in the pressure vessel. It has stable material property and a reliable safety factor. However, its heavy weight causes much fuel consumption. So, Aluminum-6061 was developed in order to modify the shortcomings of steel, but its yield strength was not as high as that of
steel. Therefore, it can't resist high internal pressure as steel can. A stronger composite material--Kevlar/Steel-1040--was derived to overcome the defects of both previous materials but it is expensive. Therefore, two other laminates were involved in this study to enforce the material strength, decrease the weight, and material cost--Eglass/Steel-1040 and Eglass/Aluminum-7050.

Eglass/Steel-1040 (yield: 231,232 psi) and Eglass/Aluminum-7050 (yield: 231,832 psi) are much stronger than previous materials. Eglass/Aluminum-7050 (density: 0.098 pci) is lighter than Eglass/Steel-1040 (density: 0.2072 pci) and Kevlar/Steel-1040 (density: 0.209 pci). In material cost, Eglass/Aluminum-7050 is cheaper than Kevlar/Steel-1040, but is a little more expensive than Eglass/Steel-1040. Hence, in consideration of strength, weight, and cost, Eglass/Aluminum-7050 was found to be the best material in this study.

- Consideration of geometry and FEM models: Fundamentally, in Chen's models, he focused on cylindrical, rectangular and rectangular-with-fins tank. In the present research, 51 models were created in 3-D geometry by means of Patran, and Ansys packages in several directions.

First, based on Chen's last model--rectangular-with-fins tank--several models were developed on the change of hole arrangement and hole diameter on the fins (Model 1-10) using Patran. Four types of hole arrangements were used--5-holes pattern, square pattern, rectangular pattern and diamond pattern. Then the change of wall thickness was considered (Models 11-12). Second, two types of irregular but symmetrical shape--irregular 1 (Model 19 to 38) and irregular 2 (Model 39 to 44)--were built up based on the
most proper hole arrangement, hole diameter and fillets developed previously (Models 19-44). Finally, the development was switched using Ansys. From Chen's cylindrical tank, a series of polygonal tanks (Model 48-51) were developed. Based on spherical type of tank, an elliptical type of tank was developed by regarding the ratio of long-axis distance and short-axis distance (Model 45-47). After geometry were created, finite element type, size, constraint, load, material data, property of each element and load case were applied to the geometry to form a FEM model.

In finite element type, Tet mesh with 0.5 size was used in the analysis of Patran package. SOLID72 element type was applied while activating the function of Smart Mesher in the analysis of Ansys package.

- Consideration of final results: The development in this study is based on changing geometry, materials and boundary conditions. In geometry--such as shape, fillet, fin, hole diameter, hole arrangement and shell thickness--have positive and negative effects on the stress distribution. The optimal material discussed above was used in Eglass/Aluminum-7050. In boundary conditions, constraints (BC1 to BC7) and internal pressures were used to match different shapes and yield strengths. Then, the analysis engine was activated to analyze the FEM models and the results of stress distribution were reviewed from the post-processing.

Finally, stress and deformation distributions were obtained from the post-processing. Maximum stresses were compared with relevant yield strengths. Then, Model 44 was chosen as the best model because it resists the highest initial pressure (21,900 psi) with lower stress (84,879 psi), stores more natural gas, is lighter (density: 0.098 pci),
costs less and approaches the actual shape—irregular and non-symmetrical. Moreover, it increases NGV's driving distance capability (299.44 miles).

In the results of Ansys analysis (Figures 6-10 to 16), some conclusions were made on the strength of frame structure:

- Cylindrical tank > Octagonal tank > Heptagonal tank > Hexagonal tank > Pentagonal tank > Rectangular tank
- Spherical tank > Elliptical tank with lower ratio of \( a/b \) value > Elliptical tank with higher ratio of \( a/b \) value, where \( a \): distance of long axis, \( b \): distance of short axis.

7-4 Future Work

In this study, the main effort was focused on structural analysis. However, in an actual situation, other analyses should be included, such as fluid and thermo, due to the gaseous fluid and flammable nature of natural gas.

Moreover, the shape of a real gas tank is irregular, not symmetrical, and the models developed were limited to an irregular shape with symmetrical geometry. Hence, future work should be included on irregular, non-symmetrical geometry.

Besides laminate, there are a great many other composite materials that can be studied. Therefore, investigating another composite material can also be included in future research efforts.
BIBLIOGRAPHY


