Interchangeable Medium-Duty Electric Vehicle

A Baccalaureate thesis submitted to the
Department of Mechanical and Materials Engineering
College of Engineering and Applied Science
University of Cincinnati

in partial fulfillment of the
requirements for the degree of

Bachelor of Science

in Mechanical Engineering Technology

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April 2020

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ABSTRACT

This senior design focuses on designing a medium-duty electric vehicle that could help integrate cleaner energy solutions into the current market. Smaller, passenger electric vehicles have become more common place, but they are lacking in the transportation industry. A new design will be proposed to overcome some of the challenges in integrating these vehicles. This design will feature a swappable cab and battery to increase vehicle use and productivity as well as include an optimized powertrain and battery sizing for inner city transportation. This design will be validated through adherence to industry standards and use of experimental data provided by the National Renewable Energy Laboratory from the US Department of Energy.

PROBLEM DEFINITION AND RESEARCH

PROBLEM STATEMENT

While significant progress has occurred in electrifying passenger cars, passenger buses and freight trucks predominantly run on fossil fuels. As congestion in cities and freight increases across the world, the need for public and freight transport will increase, thus increasing global emissions and air pollution. Hence, significant economical and efficient progress needs to be made in electrification of medium-duty vehicles to offset these problems.

BACKGROUND

Electric vehicles (EV) are starting to become more prevalent as the technology becomes more robust and there are more companies focused on developing it. This is an important development because of the increasing urgency to move away from the use of fossil fuels. There is a growing concern over the amount of pollutants being emitted to the atmosphere each year, as well as the depleting reserves of oil that are so heavily relied upon all over the world. One of the major consumers of this resource is transportation. Over the past few decades, the focus of many companies and governments has shifted to developing and producing alternatives to internal combustion engines. Advances in energy storage systems, government subsidiary projects, and specialized infrastructures have given electric vehicles an avenue for a greater presence in the global market.

RESEARCH

SCOPE OF THE PROBLEM

Poor air quality caused an estimated 4.2 million premature deaths in 2016 worldwide. In the US alone, nearly 134 million people are at risk of disease and premature deaths because of air pollution (1). Apart from this, an increase in greenhouse gas emissions from industry and transportation has fueled dangerous global issues such as heatwaves, extreme weather, food supply disruption, wildfires, floods and an increased risk of diseases.

According to the latest available statistics, the individual cost of traffic congestion exceeded $900 per driver in 1997, resulting in more than $72 billion dollars in lost wages and wasted fuel. These numbers have only grown in the last 2 decades. Drivers in 1/3rd of the US cities spend almost an entire week per year in traffic. A regular rush-hour driver wastes an average of 99 gallons of gas per year due to traffic. Above all, on-road vehicles are responsible for 44% of all carbon emissions in the US (2).

Considering the current trends of our transportation and freight industry as well as the imminent climate crisis, there is an urgent need to systematically and significantly shift towards renewable and sustainable forms of transportation. Major progress needs to be made in the electrification of passenger buses and freight trucks to prevent further deterioration of our environment and our living conditions.

CURRENT STATE OF THE ART

The electric vehicle industry is expanding as more companies develop larger commercial vehicles. Companies such as BYD, Thor Trucks, Tesla, Nikola Motors, and Workhorse, among other companies, are developing their own EV trucks and medium duty vehicles. The beverage truck and city transportation market is a great target for electric vehicles as the environment is mainly stop and go traffic, with engines requiring a lot of torque at low speeds. Meanwhile, companies are battling issues such as driver shortages and high operating costs (3). While many companies have begun production of EVs, this technology is still in development and full products are not widely available.

One product with this technology is the BYD class 6 step van (4). BYD is a Chinese company that focuses mainly on markets around China and the Pacific rim. This product targets last mile delivery with a maximum range of 125 miles. It is very similar in structure to standard step vans used by Fed-Ex or UPS which would assist in easy integration with those markets. The one issue with this model is the 7-hour AC charge time or 2-hour DC, limiting the vehicle's operating time.

Figure 1 – BYD Step Van

New vehicles are not the only way to get an electric vehicle. Lightning Systems has retro-fitted a Chevrolet 6500XD Low Cab Forward with their electronic system (5). They have created 4 different configurations ranging from 66 miles with a 1.75-hour charge time, to a 130-mile range and a 3.5-hour charge time. They have also replaced several other diesel powertrains with varying electric configurations but all with a similar range to charge time ratio.

Figure 2 – Chevrolet 6500XD

Another vehicle by BYD is the 30-foot-long electric transit bus (6). This model has lithium iron phosphate batteries that have been tested extensively for safety and do not have the usual risks associated with other lithium ion batteries. The range is up to 150 miles and the charge time is between 3 and 4 hours. The range and the electric motor are suited for operation in a metropolitan area, but the charge time still limits the its time of operation.

Figure 3 – BYD Transit Bus

Examining the above three vehicles, one flaw can be observed across the board. While significant progress was made to increase the battery range and motor efficiency, the overall cab and chassis architecture is almost comparable to their ICE counterparts – who don't have to worry about reducing air drag on heavy duty vehicles. Examining how the front areas of all these vehicles have flat faces with sharp corners (BYD 6D) which increase the air drag by a huge margin and thus decrease the already low range on these vehicles. The overall cab architecture of the three vehicles is box shaped and allows very low opportunity for the vehicle to reduce drag. Lightning Systems outright retrofit electric architecture into current ICE vehicles, which can be a highly inefficient and an unsustainable model.

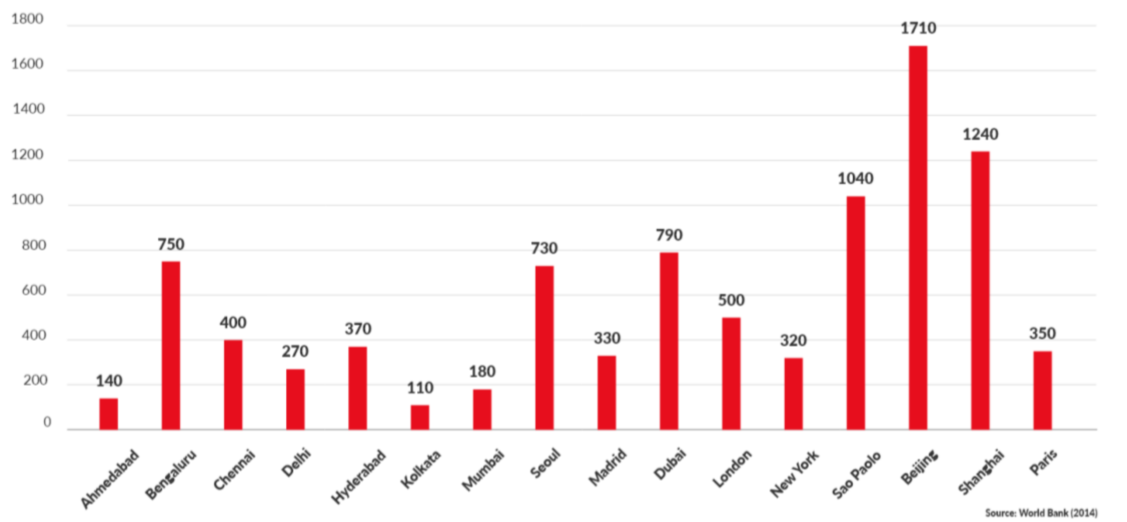
Proterra has a different charging design than BYD. Rather than charging the battery at one time and having a larger battery pack for range, Proterra has in-route, overhead charging. It charges for under 10 minutes and runs for 30 – 40 miles before needing another quick charge. This limits the bus to a very specific route and layover times at certain stopping stations. (7)

Each of these systems offer different configurations for different markets. The main constraint that these vehicles experience are their limited range and the time it takes to recharge the battery pack. Most vehicles spend several hours of downtime to finish charging, and those that offer quick several minute charging, are limited to a small range that keeps the vehicle on an inflexible schedule.

END USER
This vehicle will target both the passenger transportation and the freight industry. In order to reduce the cost of ownership of a medium duty electric vehicle, under the current constraints of battery energy density and cost, it is important to increase the sources of revenue on the vehicle. Developing a medium duty electric vehicle that can be seamlessly converted into a bus or a truck will generate revenues for a company or a city government by reducing idle time. For example, a University that purchases these interchangeable EVs for their shuttle service during daytime can lease them to a local warehouse distribution company during the night for door to door last mile deliveries. This will help the university offset the extra cost of buying an electric vehicle instead of an ICE.

For passenger transportation, the vehicle will be deployed in cities and will primarily target daily commuters and intra-city travelers.
A study conducted by the World Resources Institute indicates that 500 – 700 buses per million people is a reasonable number to reduce congestion in cities. The actual number of buses across most major cities in the world (except in China and Dubai) are drastically lower than that, as shown in Figure 4.

Figure 4 – Number of Buses Per Million People in Select Cities



Apart from increasing the number of buses to cover the necessary population, it is important to promote the kinds of services that would attract personal motor vehicle users to make the switch: increased speed and efficiency, multimodal integration across different bus and metro routes, and provide conveniences like on-demand availability, origin-to-destination travel, comfort, and safety (8). The majority of these services are missing from the current transportation system and most of the cities across the globe have failed to address these problems adequately, mainly because they are not equipped with the appropriate institutional capacity and required financial resources. The main cause of this failure is due to the functional responsibilities for urban transport are fragmented among central, state, and local level governments where no one entity is in charge of overall coordination (9). This has led to the phenomenon called the "Bus Stigma", where buses are seen in a negative way by most people, which is the biggest hurdle in facilitating the transition to shared transportation within cities (10).

The second group of end users are targeted once the vehicles have run their daily routes and their cabs have been replaced for use as **overnight freight shipping**. These vehicles will primarily target last mile delivery of goods from warehouses to customers for online shopping companies. Almost all online retailers are trying to bring down the last mile delivery costs of their products. Just the last mile of the entire supply chain adds up to about 53% of the total cost, making it the most inefficient leg of the chain. These costs can be attributed to a list of challenges including poor transportation infrastructure, costs of fuel and time wastage for each delivery, maintenance costs, different types of goods, proximity and traffic congestion (11).

Effective solutions to these issues include increasing real time visibility, customer proximity, better fleet management and optimizing routes and vehicles to reduce operating costs (12).

CONCLUSIONS AND SUMMARY OF RESEARCH

The current transportation industry is in need of new innovations to create a sustainable, attractive, and effective system while reducing operating costs. Electric vehicles are beginning to fulfill that need, but are not fully capable of meeting the customer requirements at the current stage. They need to have both a long-range, flexible route and maintain a fast charge time, which tends to be contradictory in the current state of the art. Most EVs are being used to replace their diesel counterparts in the same system that is in place throughout cities. EVs offer lower operation costs by requiring less maintenance and by saving on the use of fuel.

In the last-mile freight industry, which operates 24/7, it is essential to have the least amount of changeover times possible to maximize profits. EVs with the required range for freight also require multiple hours of charging: a wait time that companies cannot afford. To be competitive, EVs will need a charge time that is comparable to the time required to refuel a diesel-powered vehicle. Meanwhile, most buses in the passenger transportation industry can spend time overnight charging, but are losing customers to more convenient modes of transportation. Ultimately, the transportation system will need to be redefined to be more attractive and competitive in a market defined by a wide range of passengers that favor personalization. In meeting those needs, smaller, more flexible modes of transportation have an advantage. In summary, for one EV to bridge the gaps in current markets, this vehicle needs to have a relatively long range and short charge-times, low operating costs, and have a smaller, more personal design.

CUSTOMER FEATURES

The customer features in Table 1 have been compiled based on the customer survey responded to by 41 people targeted as potential customers for this product.

Table 1 – List of Customer Features

Customer features	Weighted importance
Fast Charge Time	7.0
Safe	6.0
Low Initial Investment Cost	5.0
Long Vehicle Range	4.0
Low Operating Costs	3.0
Vehicle Effectiveness	2.0
Environmental Impact	1.0

Fast charge time, safety, low initial investment costs and long vehicle range have been given top priority. Long vehicle range, which is generally regarded as hinderance for EV sales, is the 4th highest priority on this feature list. Although the medium duty vehicles currently in the market adhere to all the safety standards, they either have fast charging times and low range or vice versa, while being expensive either way. The current state of the art has not been able to strike a balance between these three features so as to promote mass adoption of EV medium duty vehicles.

EVs by default have low operating costs, greater vehicle effectiveness and negligible environmental impact when compared to their ICE counterparts due to significantly fewer moving parts in their system architecture.

PRODUCT OBJECTIVES

Based on the customer features, the primary focus of this project will be to address fast charging time, low initial investment and long vehicle range – Finding the perfect balance between them.

In order to reduce charging time and increase range, the product will feature a swappable battery box. Since these trucks require huge batteries which take longer to charge, a battery box will be designed to accommodate all the batteries below the chassis which will be replaced with a new battery box once discharged. This will reduce the wait times while charging, increasing efficiency and overall driving distance of the vehicle.

Decreasing initial investment cost of an EV is difficult considering the cost of batteries and electric motors, and the need to use expensive, lighter-weight materials in manufacturing in order to reduce gross weight. Hence, the best way to address this customer feature is to increase the source of revenue from the vehicle, thus helping the customers recuperate the cost over time. The vehicle chassis will be designed to accommodate 2 different types of cab: the first to carry passengers as a bus and the other to move freight as a medium duty truck. These cabs will be interchangeable as needed with an associated downtime. Hence, the vehicles can be used or rented out as needed. For example, it can run a bus route during the day, but by night run last-mile delivery instead of sitting derelict, thus increasing the revenue over time.

Below in Table 2 is the list of all the engineering characteristics in order to address the customer features.

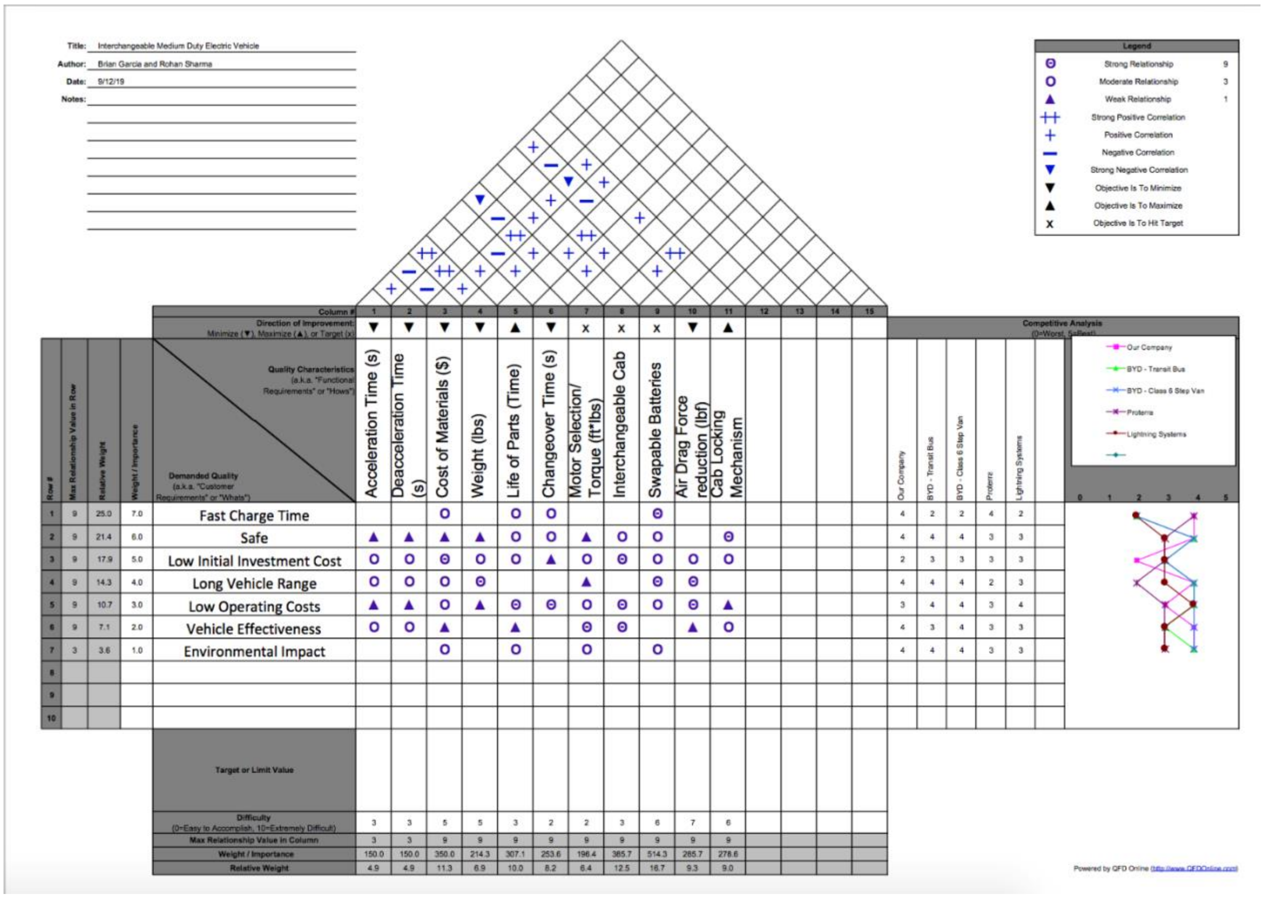
Table 2 – Engineering Characteristics by Weight

Engineering Characteristic	Importance	Relative Weight
Swappable Batteries	514.3	16.8%
Interchangeable Cab	385.7	12.6%
Cost of Materials	350	11.5%
Life of Parts	307.1	10.1%
Air Drag Force Reduction	285.7	9.4%
Cab Locking Mechanism	278.6	9.1%
Changeover Time	253.6	8.3%
Weight	214.3	7.0%
Motor torque	164.3	5.4%
Acceleration	150	4.9%
Deceleration	150	4.9%

QUALITY FUNCTION DEPLOYMENT

The design features with the highest importance are the swappable battery at 16.7% followed by the interchangeable cab at 12.5%. These are the main features that this vehicle will highlight, giving it an advantage in the market and fulfilling the customer requirements. By focusing on the interchangeability of the battery and cab, the vehicle will be able to bridge the gaps seen in the current state of the art, allowing for a more adaptive system that can address the main issue in current electric vehicles: long charge times. With these features, the vehicle will be able to run for a longer period of time per day.

Figure 5 – House of Quality



CONCEPT DESIGN FOR POWERTRAIN

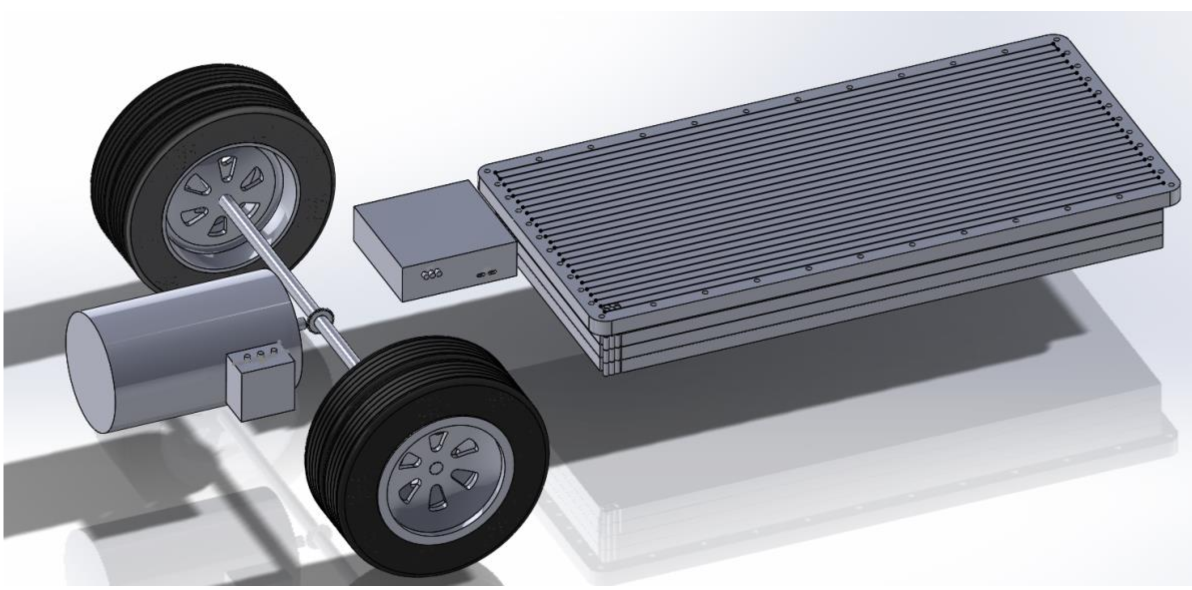
The following concepts are for the EV powertrain which includes the batteries, the battery management system (BMS), the motor, and the inverter/ motor controller. There are not significant differences between BMS or inverters/controllers of different manufacturers. They are selected based on vehicle requirements and are not necessary to include variations of these systems. The auxiliary systems that draw power from the battery are controlled through the motor controller/inverter and are represented by a net power reduction, but are not shown in these concepts. The type of battery has been predetermined to be lithium iron phosphate batteries (LiFePO4), which were chosen for their safety, high energy density, and their optimal operating conditions. The main focus will be on the type of motor, the battery configuration, and the mounting of the battery to the chassis.

CONCEPT 1

The first concept, shown in Figure 6, includes the following:

- Induction motor
- Total of 24000 small battery cells with the following arrangement:
    - 1 module containing 100 batteries in series
    - 240 modules in parallel
- Hydraulic lift and bolt mounted battery from underneath

Figure 6 – Powertrain Concept 1.

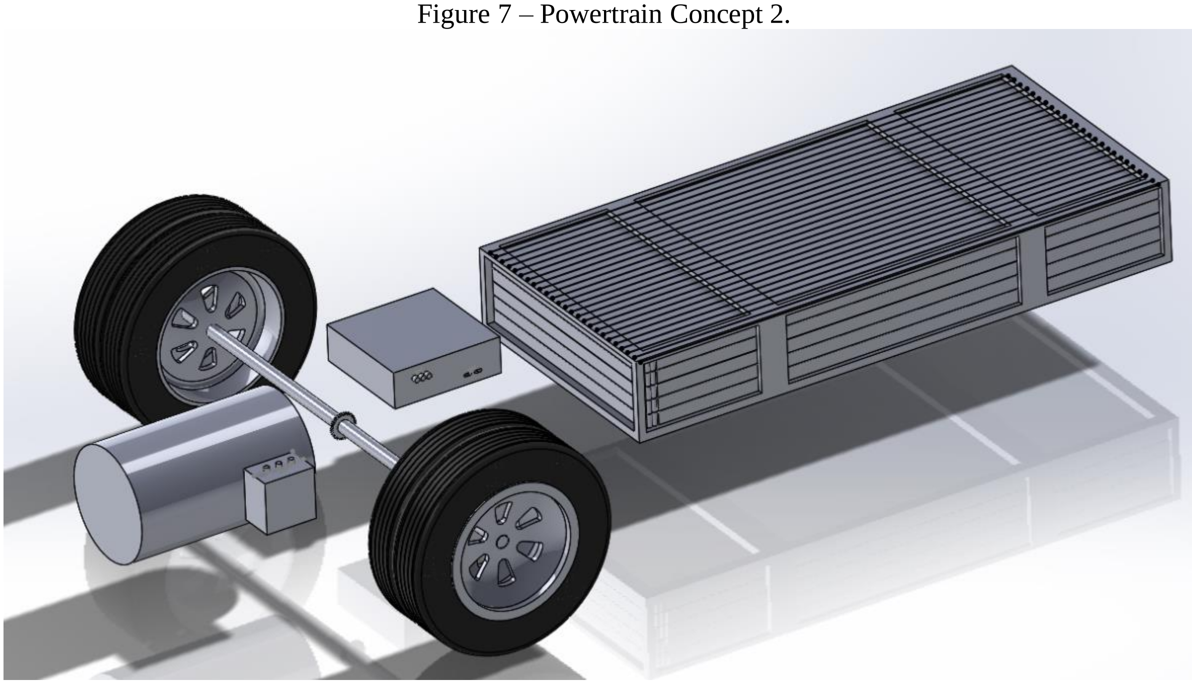


CONCEPT 2

The second concept, shown in Figure 7, includes the following:

- Permanent Magnet Synchronous Motor (PMSM)
- Total of 24000 small battery cells with the following arrangement:
    - 1 module containing 100 batteries in series
    - 240 modules in parallel
- Side-accessible - battery containment box with locking fasteners

Figure 7 – Powertrain Concept 2.

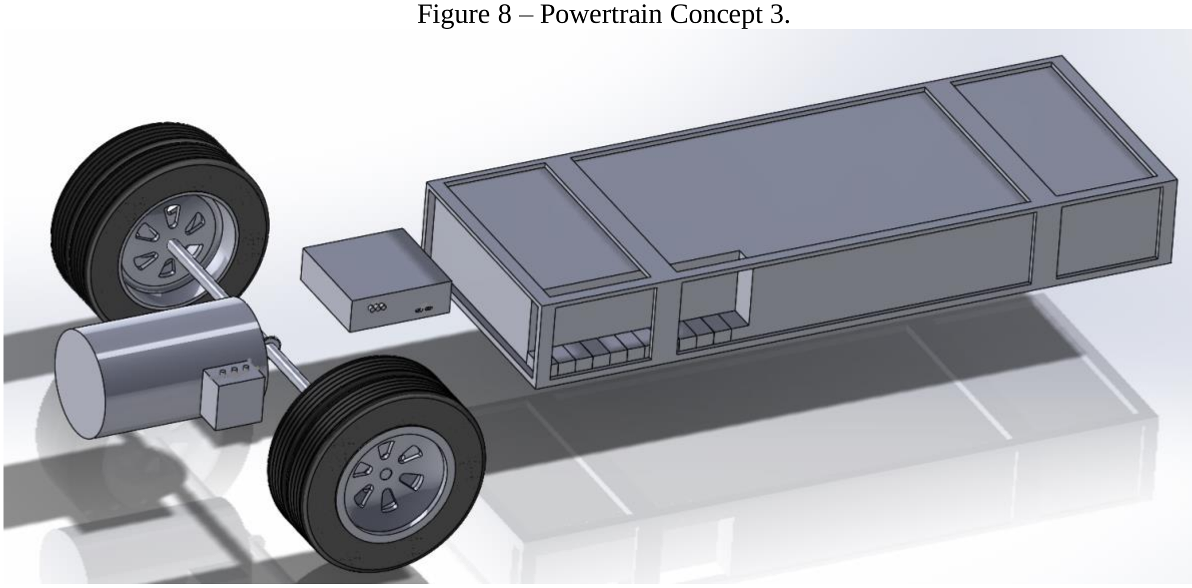


CONCEPT 3

The third concept, shown in Figure 8, includes the following:

- Induction motor
- Total of 1680 high voltage battery cells with the following arrangement:
    - 1 module containing 12 batteries in series
    - 140 modules in parallel
- Side-accessible - battery containment box with locking fasteners

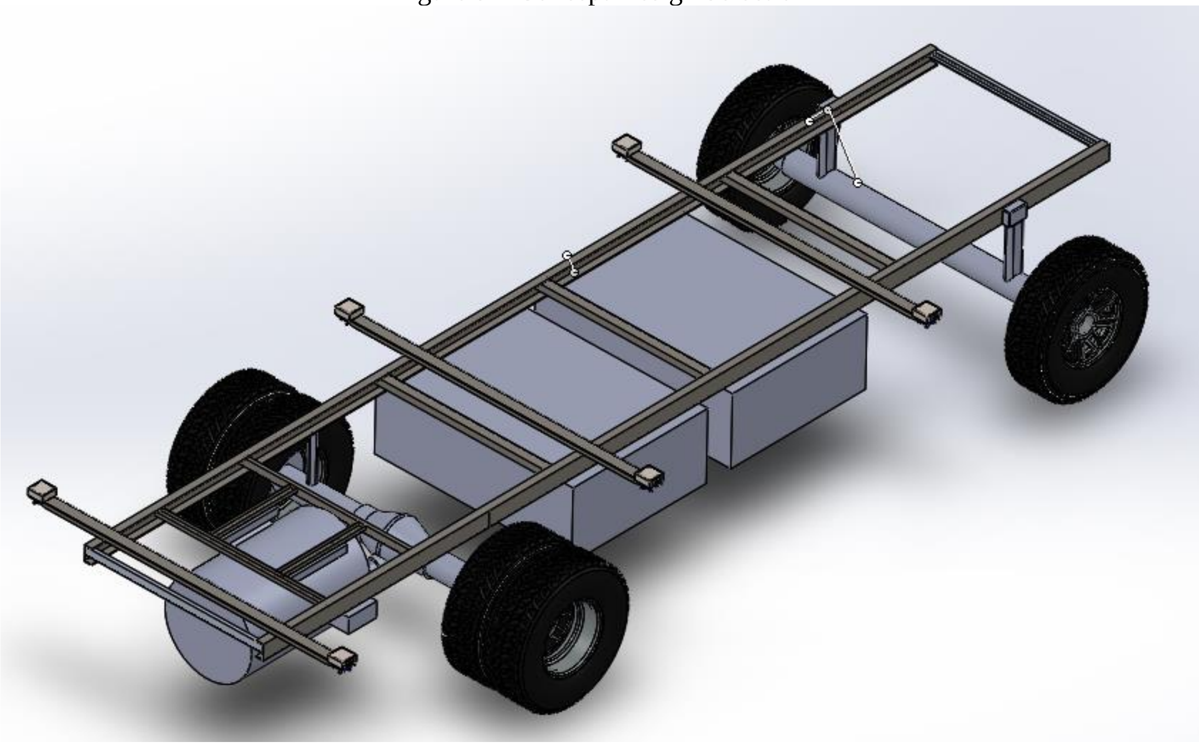
Figure 8 – Powertrain Concept 3.



POWERTRAIN DESIGN

The chosen concept is the first one, which has been modified for the final proposal and shown in Figure 9. This design consists of two battery boxes, totaling 19240 batteries. At the battery specifications, these cells provide the 166 kWh of energy required, using a safety factor of 1.2. The powertrain was designed for this energy requirement.

Figure 9 – Concept Design Selection



ENERGY CALCULATIONS

The energy requirement of the vehicle, which is required to determine all components in the powertrain, was calculated through the following Matlab script:

```
%% Energy Calculations %%

%%%%%%%%%%%% Variables %%%%%%%%%%%%%%%%
g = 9.81;                 % m/s^2, Gravity
rho = 1.204;              % kg/m^3, Air Density
Amax = 1.6677;            % m/s^2, Max Acceleration
Aavg = 0.39;              % m/s^2, Average Acceleration
Davg = 0.42;              % m/s^2, Average Deceleration
Vavg = 11.18055556;       % m/s, Average Velocity
%Theta = 0.0996687;       % rad, Incline
Theta = 0;
%Crr = 0.006735197;       % Average Coef. of Rolling Resistance
R = .38862;               % m, Tire Radius
p = 5.5;                  % bar, Tire Pressure
uk = 0.8;                 % Kinetic Friction on concrete

%%%%%%%%%%%% Inputs %%%%%%%%%%%%%%%%

Weight = 15000;           % Weight of Vehicle in pounds
m = Weight*4.45/9.81;     % kg, mass
```

```
Eff = 0.83862353;        % Efficiency of System
Area = 2.1312;           % m^2, Front Area
Cd = 0.65;               % Coef. of Drag
range = 100;             % km, Total drive cycle range

%%%%%% Effective Mass %%%%%%%%%%%%%%%%%
Iwt = 56.7*(R^2);        % kgm^2, Inertia of Wheels and tires
Idrive = .02;            % kgm^2, Inertia of Drive train
GD2 = 3.621;             % Inertia of Motor (from manufacturer)
Imotor = GD2/4;          % kgm^2, Inertia of Motor
GR = 4.555555556;        % System Gear Ratio

Meff = m + (Iwt/(R^2)) + ((Idrive*(GR^2))/(R^2)) + ((Imotor*(GR^2))/(R^2));
% Meff = kg Effective rotational mass

%%%%%% Drive Cycle Variables %%%%%%%%%%
%%% Based on NREL Class 3 Electric Vehicle Data %%%

a = xlsread('NREL_Class3_Acceleration.xlsx');
v = xlsread('NREL_Class3_Velocity.xlsx');
t = 1:length(a);
sz = size(t);
count = ones(sz);
Crr = 0.005 + (1/p).*(0.01+0.0095.*(v./100).^2);

%%%%%% Total Forces %%%%%%%%%%%

Fi = Meff.*a;                          % (N) Force of mass in motion
%Fs = m * g * sin(Theta) .* count;    % (N) Force due to slope
%%% Fs is Accounted for in Drive cycle
Frr = m * g .* Crr* cos(Theta);       % (N) Force of Rolling Resistance
Fd = .5 * p * Cd * Area .* v.^2;      % (N) Force of Drag

DecelF = zeros(sz); % Create an empty array the size of the other arrays
AccelF = zeros(sz);

%%%% Separate Acceleration from Deceleration %%%%%%%
for k = 1:length(a)
    if a(k) < 0
        DecelF(k) = Fi(k) - (Fd(k)+Frr(k));
        AccelF(k) = 0;
    else
        AccelF(k) = Fi(k) + Fd(k) + Frr(k);
        DecelF(k) = 0;
    end
end

%%%%%%%%% Power for Acceleration %%%%%%%%

A_Pt = AccelF.*v;     % Total Power (W) for Acceleration
subplot(2,1,1)
plot(t,A_Pt)
title('Total Power (W) for Acceleration');
xlabel('Time (s)');
ylabel('Power (W)');

%%%%%%%%% Power for Deceleration %%%%%%%%

D_Pt = DecelF.*v;     % Total Power (W) for Deceleration
figure(1)
subplot(2,1,2)
plot(t,D_Pt)
title('Total Power (W) for Deceleration');
xlabel('Time (s)');
ylabel('Power (W)');

%%%%%%%%%%% Conclusion %%%%%%%%%%%%%

Accel_Energy = cumtrapz(A_Pt); % Joules in the hour of the drive cycle
Decel_Energy = cumtrapz(D_Pt);

Energy = Accel_Energy + Decel_Energy;

figure(2)
hold on
plot(t,Accel_Energy);
```

```
plot(t,Decel_Energy);
plot(t,Energy);
grid on
title('Total Energy of Drive Cycle');
legend('Acceleration Energy (J)','Deceleration Energy (J)', 'Total Energy
(J)','Location','NorthWest');
xlabel('Time (s)');
ylabel('Energy (J)');
hold off

kWh = (Energy*(10^-6))*3.6; % Convert Joules to kilowatt-hours

distance = trapz(v);        % Total distance traveled
km = distance/1000;         % in Kilometers

kWh_km = kWh(3697)/km       % Energy consumption per km

kWh_total = kWh_km * range % Energy consumption over full range
```

The outputs of the above script are shown in Figures 10 and 11 and as follows:

>> Energy_Calculations

kWh_km =

1.1898

kWh_total =

118.9845

Figure 10 – Total Energy of Drive Cycle

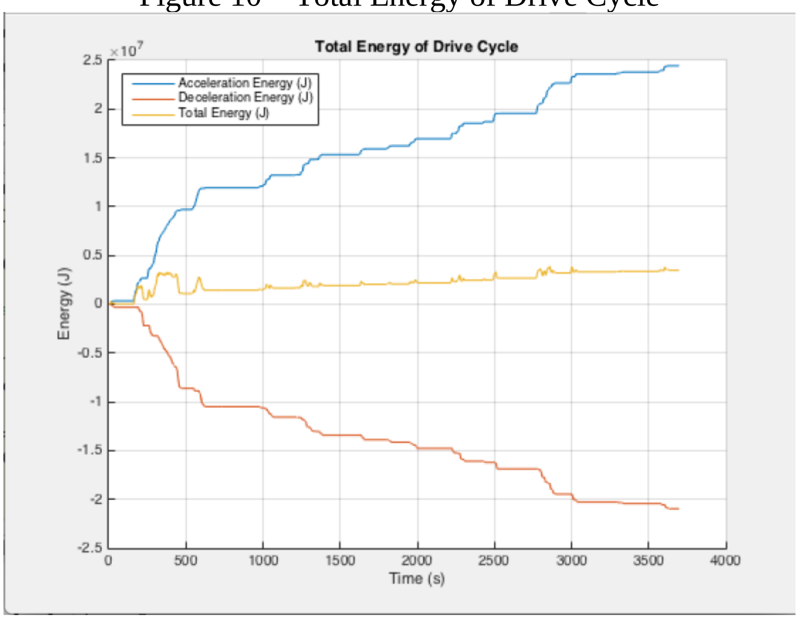
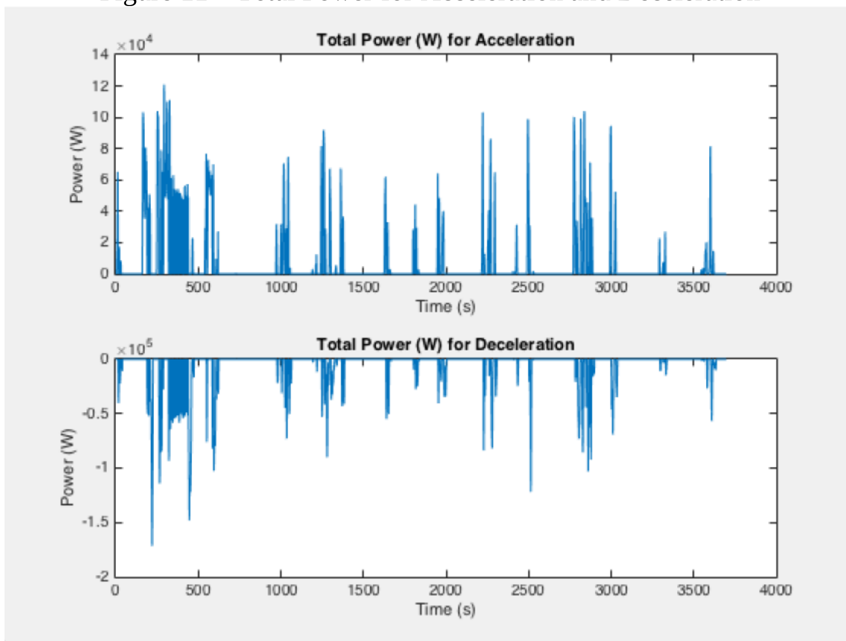


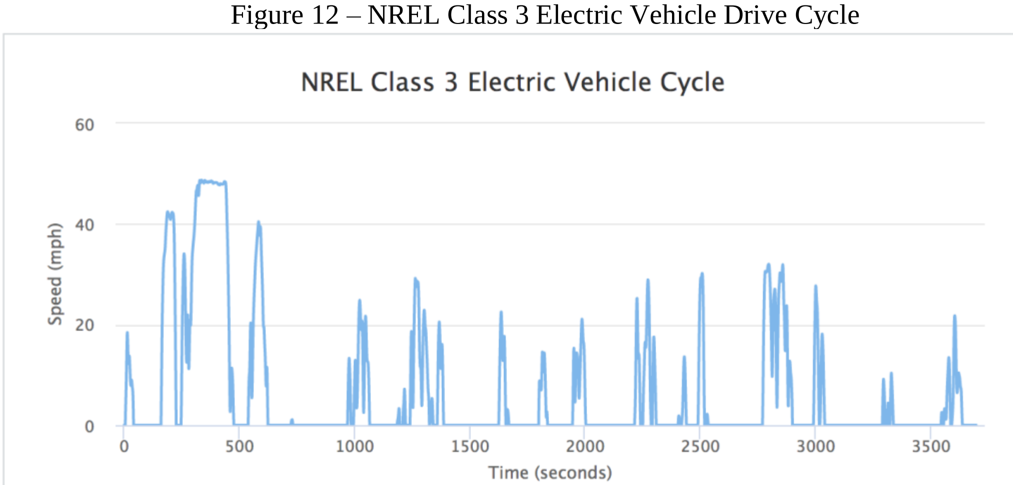
Figure 11 – Total Power for Acceleration and Deceleration



Drive Cycle

The drive cycle used to calculate the energy requirement was based on a report by the U.S. Department of Energy's (DOE's) National Renewable Energy Laboratory (NREL) (13). This study focused on two different vehicles: Navistar eStar (Class 3) and Smith (Class 6). Figure 12 shows the drive cycle of the class 3 vehicle studied over the course of 1 hour. This drive cycle was then used to find the acceleration and velocity over that period and incorporated into the Matlab script.

Figure 12 – NREL Class 3 Electric Vehicle Drive Cycle



(14)

This Senior Design is focusing on a Class 5 vehicle that is designed for inner-city travel with many stops, accelerations and an, on-average, low top speed. Therefore, the drive cycle of the class 3 vehicle from the NREL study was selected because it is a more aggressive drive cycle that represents the focus of this design better than the larger class 6 vehicle. From the NREL report, "The larger Smith Electric vehicles averaged a daily driving energy consumption rate of 1.15 kWh/km while the Navistar eStar vehicles averaged 0.52 kWh/km". The Matlab script showed that the senior design vehicle has a daily consumption rate of 1.19 kWh/km which should make sense. This vehicle is closer in weight and inertia of the larger class 6 vehicle, but also has the more aggressive drive cycle of the smaller vehicle, resulting in a slightly larger consumption rate than the class 6 vehicle in the study. In another study by the California Air Resources Board, Table 3 shows the energy consumption of medium and heavy duty vehicles for this comparison (15). This senior design is for a vehicle GVWR of 15,000 lbs. with a kWh/mile of 1.9.

Table 3 – Estimate of BEV Energy Needs per Mile

Estimate of BEV Energy Needs per Mile	
Weight Class	**kWh/mile**
Medium-Heavy (8,501 to 14,000 lbs. GVWR)	1.8*
Heavy-Duty (>14,000 lbs. GVWR)	2.8*

*ARB internal estimates, which do not include a buffer to prevent over discharge

Effective Mass

In order to move the vehicle, the powertrain must not only overcome the weight of the vehicle, but also the inertia of the rotating components within the powertrain. These rotating components are found in the motor, the gears and axles, and in the wheels and tires. All of these inertias and the weight of the vehicle are summed together to form the Effective mass (Meff, in the matlab script).

$$M_{eff} = M_{vehicle} + \frac{I_{wheel/Tire}}{r^2} + \frac{I_{drive} * GR^2}{r^2} + \frac{I_{Motor} * GR^2}{r^2} \quad (16)$$

Where:

I = rotating Inertia of component
GR = Gear Ratio
r = Tire radius

Individual inertia of component calculations:
Wheels

$$I_{w/t} = \frac{1}{2} m r^2$$
$$\frac{I_{w/t}}{r^2} = \frac{1}{2} m$$

The wheels and tires are considered to be a cylinder.

$$m = 2_{drum\ brakes} + 2_{Wheel\ hubs} + 4_{tires} = 60lbs + 4 * (47)lbs = 113.4\ kg$$

The weight of the drum brakes and wheel hubs are assumed to be a total of 30 lbs per side.

$$\frac{I_{w/t}}{r^2} = \frac{1}{2} m = \frac{1}{2} * 113.4\ kg = 56.7 \frac{kg}{m^2}$$

Drive Train

$$I_{drive} = \frac{1}{2} m * r^2\ (cylinder)$$
$$m = (2^*) * V * \rho = (2^*) * \pi r^2 h * \rho\ =\ 2\pi * \left(\frac{0.060m}{2}\right)^2 * 0.9m * 7850 \frac{kg}{m^3} = 40.265kg$$

* there are 2 axles due to the differential
* the value of density is for AISI 1030 Carbon Steel (17)

$$I_{drive} = \frac{1}{2}(40.265) * \left(\frac{0.060m}{2}\right)^2$$
$= 0.183\ kgm^2$ + inertias from gears
$$= 0.02\ kgm^2$$

Motor

$$I_{motor} = \frac{GD2}{4} = \frac{3.621}{4} = 0.90525\ kgm^2$$

*GD2 is the moment of inertia provided by the manufacturer’s catalogue. (18)

The effective mass is calculated within the Matlab script using the results of the equations above. This allows for the vehicle to more accurately account for the energy requirements during operation.

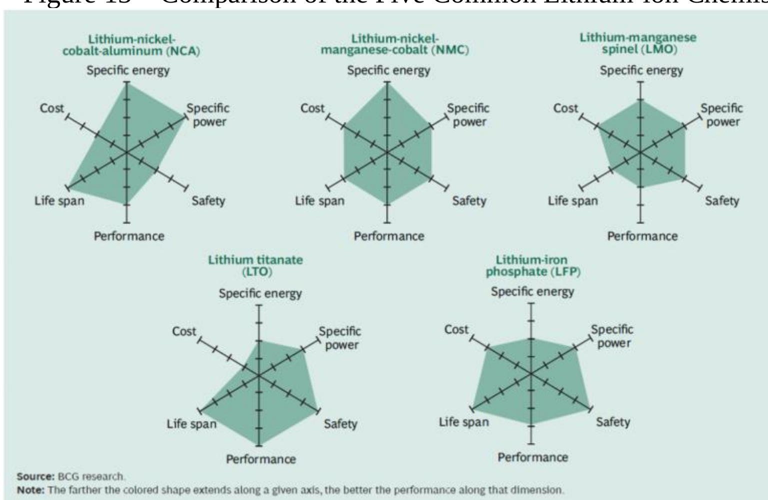
<u>Other calculations</u>

The rest of the Matlab script manipulates the data given in the drive cycle and the results from the force equations. (19) This manipulation includes separating the instances of positive acceleration from the negative, calculating the amount of power during those instances, calculating the net amount of energy over the time of the drive cycle and converting that energy to kWh/km for comparison and for selecting subsequent components of the drivetrain.

BATTERY SPECIFICATIONS

Currently, energy storage is the limiting technology of electric vehicles. It is the heaviest, most expensive, and life-limited part of every electric vehicle. The future of electric vehicles lies with more efficient forms of energy storage. Therefore, the type of battery is very important to the vehicle specifications and requires careful consideration. Of the many battery chemistries available today, lithium ion batteries are the most popular for use in electric vehicles. Lithium ion batteries have a high-energy density and low self-discharge rate as compared to most other chemistries. Figure 13 shows the five main types of Lithium ion battery chemistries and their respective strengths and weaknesses.

Figure 13 – Comparison of the Five Common Lithium-ion Chemistries



(15)

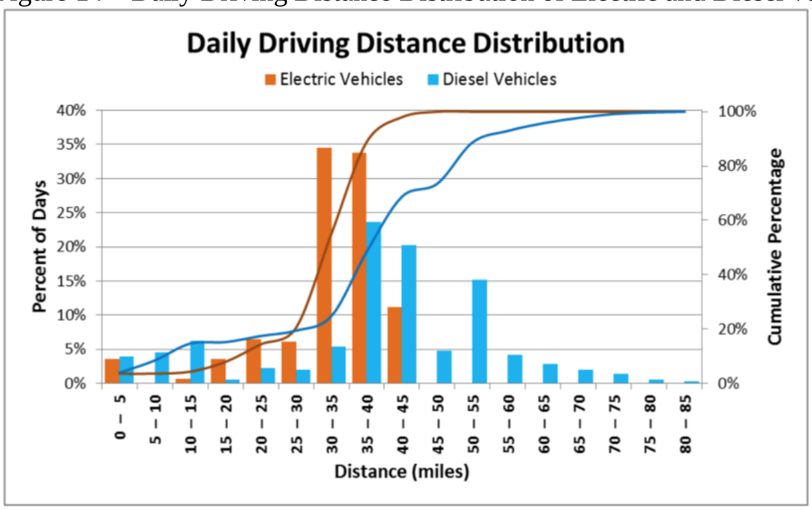
The chemistry that is most suitable for this application is Lithium-iron Phosphate (LFP). Although LFP batteries are not the most energy efficient, the reason for selecting this chemistry is because of its safety factor. Most lithium-ion batteries have a reputation for catching on fire or exploding when the cell malfunctions, is over or under charged, or is damaged. This is considerable as thousands of cells are strung together to make a pack large enough to power a vehicle of this magnitude. There can be natural imbalances in charge across these thousands of cells and the risk of one of them malfunctioning can be likely. However, LFP batteries will not catch on fire or explode. These cells are also less costly than some other chemistries and have a longer life span. The trade-off is that the vehicle will require more of them to reach the required energy level.

BATTERY PACK SIZING

The size of the battery pack is affected by several factors: first, the total range of the vehicle; secondly, the overall efficiency of the powertrain; and third, by the factor of safety. There is also a slight impact of auxiliary systems on the battery pack such as heating and cooling (20).

The battery pack size is directly proportional to the total range of the vehicle, as it is based on the consumption rate of kWh per kilometer. This range has a significant impact on the amount of energy required to store in the batteries and on the weight of the battery cell. Figure 14 from a field evaluation by the NREL shows the ranges experienced by electric vehicles currently in use (21). In this study, the maximum daily driving distance for electric vehicles is 50 miles (80 km). Since the vehicle is designed for city transportation with more focus on number of stops per mile, the vehicle will not need the energy reserves for longer distances. This design considers 100 km (62.14 miles) as an optimal range, which allows for more flexible driving routes without an excess of battery weight and is comparable to current diesel vehicles. For reference, if 10 km are added to the range in the current battery configuration, the vehicle puts on about 375 pounds of battery weight.

Figure 14 – Daily Driving Distance Distribution of Electric and Diesel vehicles



(21)

Therefore, it is necessary to balance between total range and total weight. As the batteries are swappable, one set of batteries can be charged while another set is in use, making the vehicle more efficient and able to have smaller periods of downtime. This is a similar concept to airplanes swapping engines to keep the aircraft running and being productive as one of its engines goes through maintenance.

The efficiency of the system determines how much energy is lost while transferring it from storage in the battery pack to the motion of the tires. The efficiency of energy conversion is much higher in EVs than in conventional ICE vehicles. Table 4 shows the efficiencies that were used in this design (22) (23). Note, the wiring is assumed to have negligible losses in energy. Also, due to frictional losses, gear reductions typically experience efficiencies between 95 and 98% and in this design, they are assumed to be 97% efficient per gear reduction. (24)

Table 4 – Component Efficiencies

Component	Efficiencies
Motor	0.936
Battery	0.9931
Wiring	1
Inverter	0.9791
Gears	0.9409
Total	**0.856**

The total efficiency of this system is 85.6%. The battery pack is increased to take into account these losses.

The factor of safety used in this design is 1.2. This indicates that the battery pack has 20% more energy than what the drive cycle demands. This provides a buffer for the end user who may experience variation in the drive cycle, battery degradation over time, or be approaching the range limit and in need to get the vehicle to its predetermined charging location.

The following equation shows the calculation of the final energy requirement when considering the above factors:

$$E_f = \big((E_i + E_{aux}) * n\big) + \Big(\big((E_i + E_{aux}) * n\big) * (1 - \eta_{total})\Big)$$

Where:

E_f = Final Energy Requirement
E_i = Initial Energy Requirement
E_{aux} = Auxiliary Energy Requirements
n = Factor of Safety,
η_{total} = Total efficiency

$$E_f = \big((118.98\ kWh + 2kWh) * 1.2\big) + \Big(\big((118.98\ kWh + 2kWh) * 1.2\big) * (1 - 0.856)\Big)$$

$$E_f = 166.03\ kWh$$

The battery array is then arranged to meet that Energy requirement as well as the 415V input voltage required for the motor. Table 5 shows the battery cell and array specifications (25) and Table 6 shows the corresponding battery cell module to meet the energy requirements.

Table 5 – Lithium Iron Phosphate Specifications

Battery Specs:	1 Cell	Full Pack
Amp-hours	3.3	488.4
Volts	3.2	416
kWh	0.01056	203.17
lbs	0.19400679	3732.69
ft^3	0.001237563	23.81
$/per unit	$10.18	$124,398.85

* Total cost of pack includes buy-in-bulk deals, but not shipping or assembly costs nor does it include any price reductions associated with being a manufacturer of a product.

Table 6 – Module and Array Configuration

Battery Module	Batteries	Module Specs	
# Parallel	1	3.3	Amp-hours
# Series	130	416	Volts
Total Per Module	130	1372.8	Watt-hours
Total in Array	19240	148	# of Modules

Each Module is comprised of 130 cells in series and 37 of those in parallel. This gives each module 50.79kWh at 416V. In order to meet the total energy requirement of 166.03kWh and due to lithium ion batteries only being able to supply 80% of their energy, the battery pack must increase by 20% to give the required energy without the batteries becoming over-discharged.

$$Battery\ E_f = 166.03kWh * 1.2 = 199.24kWh$$

Therefore, 4.2 modules in parallel are required to meet the energy demand. Table 5 above shows the final size and cost of this battery pack.

MOTOR SELECTION

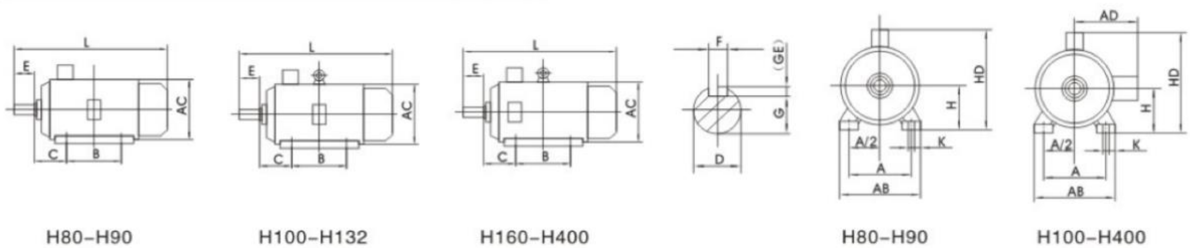
The electric motor is selected based on the energy requirement of $166.03\ kWh$. For this design an induction motor is selected for its efficiency and the ability to perform regenerative braking, which turns the motor into a generator and puts energy back into the battery from the motion of the vehicle. Motor specifications (22) are shown in Figure 15.

Figure 15 – Motor Specifications and Dimensions

Technical Data - Y2 series induction motor - 2 poles - 50Hz

Type	Rated output		Full Load						Rated Torque (Tn)	Locked Rotor Torque / Rated Torque	Max. Torque / Rated Torque	Locked Rotor Current / Rated Current	Noise	Weight
			Speed	Input Current (Amp)			Efficiency	Power factor						
	KW	HP	RPM	380V	400V	415V	η%	Cosφ	N.m	Tst/Tn	Tmax/Tn	Ist/In	dB/(A)	Kg
Y2-280S-4	75	100	1480	139.3	132.9	128.1	93.6	0.87	484.0	2.2	2.3	7.2	86	525

Y2 Series Induction Motor Dimensions



Type	Poles	Mounting Dimensions										Overall Dimensions				
		A	A/2	B	C	D	E	F	G	H	K	AB	AC	AD	HD	L
280S	2	457	228.5	368	190	65	140	18	58	280	24	550	580	410	680	985
	4.6.8	457	228.5	368	190	75	140	20	67.5	280	24	550	580	410	680	985

The peak motor power can be calculated through the following equation:

$$T = 9.55 * \left(\frac{P}{n}\right)$$

$$P_{peak} = ((T_{rated} * T_{max}) * n_{rated}/9.55)/1000$$

Where:

T_{max} = ratio between Maximum torque to rated torque
n_{rated} = rated motor revolutions per minute

$$P_{peak} = ((484.0Nm * 2.3) * 1480rpm/9.55)/1000$$

$$T_{peak} = 172.517\ kW$$

The peak power is more than the required energy. This motor will be sufficient to move the vehicle forward.

INVERTER SELECTION

The inverter manages the energy flow in the vehicle. The inverter takes the DC electricity of the battery and converts it to AC for use in the induction motor, or the opposite during regenerative braking. It can also do a high voltage DC to low voltage DC conversion as well, which is used to power auxiliary systems throughout the vehicle. Table 7 shows the inverter that meets the vehicle requirements.

Table 7 – Inverter Specifications

Product series	GVD550 series 5-in-1 controller
Product model	GVD550-5L50-240
Product picture	
Function integration	Main drive, auxiliary drive (oil pump+air pump), power distribution, DC/DC
Vehicle type	Bus and logistics vehicles
Input voltage	400-750V/12V or 24V
Output current	Drive: 350A (rated), 700A (peak) Auxiliary drive: 13A (rated), 19.5A (peak) DC/DC: 100A (rated), 120A (peak)
Operation temperature	-40-85℃
Cooling mode	Liquid cooling
Dimensions	615W x 460D x 250H(mm)
Weight	≤38Kg
IP protection level	IP67

(26)

Powertrain Requirements

The power train is the mechanical means of transferring the rotational output of the motor to the rotation of the tires. This includes all axles and gears required to give enough power to the vehicle. In EVs, the motor can supply a range of torques and rpms by regulating the amount of power it receives from the inverter. This way, most electric vehicles have no need for a transmission. However, gears are required to output the correct amount of torque and rpm to the tires. In order to know how fast the tires rotate and how much torque is required, the following equations were used:

<u>Speed</u>

$$rpm_{tire} = \left(\frac{V_{avg}}{\pi * d}\right) * 60\,\frac{sec}{min}$$

Where:

V_{avg}= The operational or average speed in m/s

d = diameter of rear tires in meters

$$rpm_{tire} = \left(\frac{11.18\frac{m}{s}}{\pi * 0.77724m}\right) * 60\,\frac{sec}{min}$$

$$rpm_{tire} = 274.73\ rpm$$

<u>Torque</u>

$$T = F * r$$

Where:

$F = N * \mu_k$ = The traction force on the back tires in Newtons

N = Normal force on back tires

μ_k= Coefficient of Kinetic Friction on concrete (27)

r = radius of rear tires in meters

$$T = 4082.569N * 0.8 * 0.38862m$$

$$T = 1269.25\ Nm$$

Table 8 shows the comparison between the output requirements and what the motor provides.

Table 8 – Motor-to-Tire Torque and Speed Comparison

Criteria	Tire Requires:	Motor Provides
rpm	274.73	1480
torque	1269.25	484.2

It has been determined that one gear reduction would be sufficient. This vehicle also requires a differential in order to function properly. A differential allows the two back tires to rotate at different speeds, such as during a turn. In addition, differentials have a gear reduction by nature. For design simplification, the required gear ratio can be fulfilled by the differential. The ford 10.5” differential has been selected to fulfill both of those rolls (28). This differential is for comparatively sized vehicles and also contains a gear reduction of 4.5556:1. By applying this gear ratio, the output rpm of the motor to the tires is 324.878rpm and output torque is 2205.8 Nm. This is higher than the requirements, giving the vehicle an operational average speed of 16.24 m/s or 36mph. This is an acceptable average speed based on data provided by NREL (29) and shown in Figure 16.

Figure 16 – Average Driving Speeds of Delivery Vans and City Transit Buses

(29)

Drive Axle Design

Due to the differential, there are two drive axles. These axles must transmit power from the motor to the tires while typically bearing the load of the vehicle. For larger vehicles, a full-floating axle is required. Full-floating axles are a specific design that direct the weight of the vehicle directly to the wheel hubs rather than the axle. In effect, the vehicle weight is transmitted through the shock absorbers to the axle housing, and the wheel hub rotates about the axle housing as well. The axle, inside the housing, connects via a flange to the wheel hub outside of the bearings and housing. In effect, the bending stress is removed from the axle, which only transmits torque. The down-side to full-floating axles is that they increase the rotational mass of the drive-train, thus increasing inertia. This inertia has been accounted for in the effective mass equations in the Matlab script. To determine the shaft diameter, the modified Goodman Distortion Energy equation (30) was used:

$$\frac{1}{n} = \frac{16}{\pi d^3}\left\{\frac{1}{S_e}\left[4(K_f M_a)^2 + 3(K_{fs} T_a)^2\right]^{1/2} + \frac{1}{S_{ut}}\left[4(K_f M_m)^2 + 3(K_{fs} T_m)^2\right]^{1/2}\right\}$$

This equation has been solved for the diameter and, because the loading conditions dictate that the shaft is fully reversed, Ta=0 and Mm=0.

$$d = \left(\left(\frac{16n}{\pi}\right)\left(\frac{\left(2\left(K_f M_a\right)\right)}{S_e} + \frac{\left[3\left(K_{fs} T_m\right)^2\right]^{\frac{1}{2}}}{S_{ut}}\right)\right)^{1/3}$$

Since the load has been redirected from the axle, the only bending stress is from the weight of the axle itself and becomes negligible when considering the torque on the shaft from the motor. The equation simplifies to:

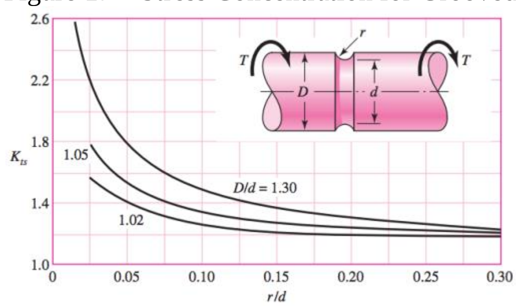
$$d = \left(\left(\frac{16n}{\pi}\right) \left(\frac{\left[3\left(K_{fs}T_m\right)^2\right]^{\frac{1}{2}}}{S_{ut}} \right) \right)^{1/3}$$

Where:

d= shaft diameter in meters
n = factor of Safety
K_{fs} = Stress concentration factor
T_m = Torque on Shaft
S_{ut} = Ultimate Strength of the material

K_{fs} was determined by analyzing the groove for the retaining rings for the gear in the differential using Figure 17 below.

Figure 17 – Stress Concentration for Grooved Shaft



(30)

Where: D/d = 1.2 and r = d/10

$$d = \left(\left(\frac{16 * (4)}{\pi}\right) \left(\frac{[3(1.46 * 2205.8Nm)^2]^{\frac{1}{2}}}{550\ MPa} \right) \right)^{1/3}$$

$$d = 0.059\ m$$

The standard diameter size is 60mm (31) and meets the shaft fatigue factor of safety of 4. To ensure that the shaft is designed against yielding the following equation was used:

$$n_y = \frac{S_y}{\left(\frac{32K_f M_a}{\pi d^3}\right) + \left[3\left(\frac{16K_{fs}T_m}{\pi d^3}\right)^2\right]^{1/2}}$$

Where:

$M_a = 0$
S_y = Yield Strength

$$n_y = \frac{S_y}{\left[3\left(\frac{16K_{fs}T_m}{\pi d^3}\right)^2\right]^{1/2}}$$

$$n_y = \frac{390\ MPa}{\left[3\left(\frac{16 * 1.46 * 2205.8Nm}{\pi * (0.060m)^3}\right)^2\right]^{1/2}}$$

$$n_y = 2.965$$

This factor of safety is less than 4, but it is in acceptable number. The main concern for the axle is failure due to fatigue.

Bearing Selection

Bearings are selected based on the number of cycles until a percentage of its components fail. For this design, the bearings are selected based on the L_{10} Life, which indicates 90% reliability, for a 2-shift continuous operation (32). The following equation determines the load requirement for the bearing needed:

$$C_{10} = F_D\left(\frac{L_{10} * n * 60\frac{sec}{min}}{10^6}\right)^B$$

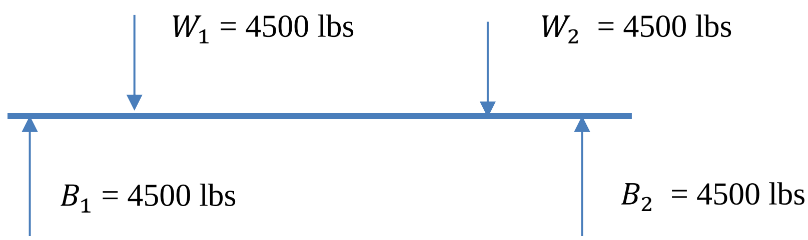
Where:

F_D = Design Force on the bearing
n = rpm of bearing
10^6 = the number of cycles
B = exponent based on type of bearing
= 1/3 for Ball Bearing
= 3/10 for Roller Bearing

Drive Axle

Figure 18 shows the loading on the rear axle hub and the forces on the bearings.

Figure 18 – Free Body Diagram for Rear Axle Hub



4500 lbs = 20.017 kN

$$C_{10} = 20.017kN\left(\frac{40{,}000 * 324.878rpm * 60\frac{sec}{min}}{10^6}\right)^B$$

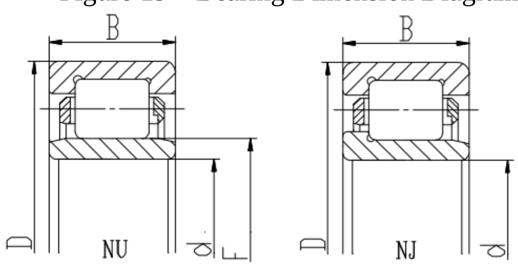
$$C_{10} = 184.237kN\ Ball$$
$$= 147.56kN\ Roller$$

Since these bearings are rolling on the axle hub and not the axle, the bore is 80mm. Table 9 shows suitable roller bearings (33) for this application and Figure 19 shows the basic bearing dimension diagram.

Table 9 – Roller Bearing Specifications and Dimensions for Drive Axle

Basic Dimensions					Basic Load Ratings		Bearing Designations	Weight
d	D	B	F	E	Cr	Cor		
mm					KN			kg
80	140	26	95.3		151	184	NU216E	1.65
	140	26	95.3		151	184	NUP216E	1.72
	140	26		127.3	151	184	N216E	1.67
	140	26	95.3		151	184	NJ216E	1.68

Figure 19 – Bearing Dimension Diagram

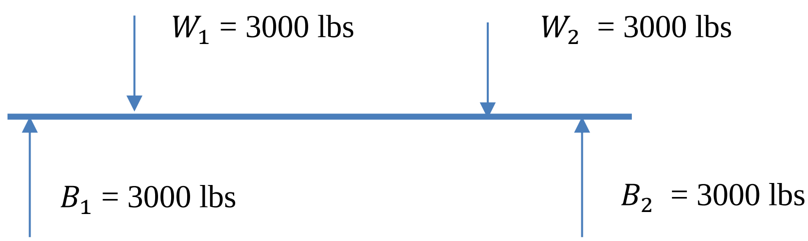


(33)

Front Axle

Figure 20 shows the loading on the front axle and the forces on the bearings.

Figure 20 – Free Body Diagram for Front Axle



3000 lbs = 13.34466 kN

$$C_{10} = 13.34466\left(\frac{40{,}000 * 324.878rpm * 60\frac{sec}{min}}{10^6}\right)^B$$

$$C_{10} = 122.834kN\ Ball$$
$$= 98.3755kN\ Roller$$

Table 10 shows the selected roller bearing and Figure 19 above shows the diagram for its basic dimensions. (33)

Table 10 – Bearing Specifications and Dimensions for Front Axle

Basic Dimensions					Basic Load Ratings		Bearing Designations	Weight
d	D	B	F	E	Cr	Cor		
mm					KN			kg
60	110	60	73.5		106	130	NU2212 WBM/C3	1.57

The diameter for the front axle bearing is 60mm because the front does not require a full floating axle due to less loading than the rear axle.

FACTORS OF SAFETY OF CONCERN

The only factor of safety that is not ideal is the rear drive axle, n_y = 2.96. This factor of safety, which is close to 3, is less than the ideal of 4. However, this is only a check to make sure the axle won’t yield during normal operating procedures. Failure due to fatigue is more likely and has a safety factor of 4. Although the yield factor is not ideal, it is acceptable for this application.

VEHICLE ARCHITECTURE DESIGN

CHASSIS DESIGN ANALYSIS

The objective of this project was to determine the best material and design for a chassis to support cab and battery swapping with minimum weight. The constraints for this design were maximum shear and bending stress, acceptable deflection, and acceptable standard industry practices.

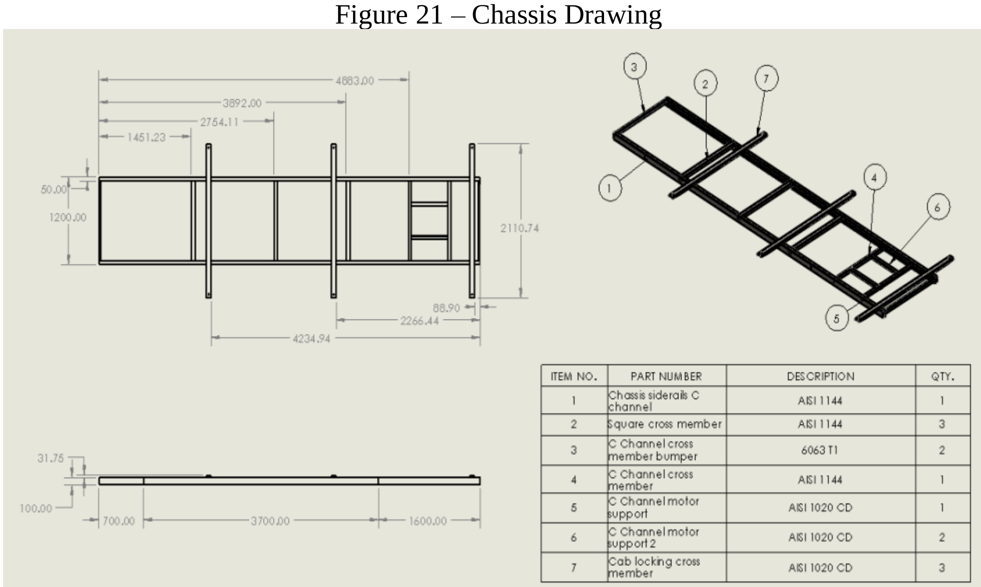
The two most commonly used chassis in the medium – heavy duty industry are:

1. Unibody: This setup integrates the frame into the body construction to act as a single unit. This construction reduces weight of the vehicle and allows for better safety and handling.

2. Body on frame: The vehicle uses 2 different components as the frame – the body sits on top of a ladder frame chassis. The ladder frame chassis is more resistant to bending and twisting under load and allows more freedom to modify the setup (Chassis length, suspension, motor, battery etc.)

For this project application, Body on Frame ladder frame chassis was used since the body needs to be swapped out when needed, hence, they need to be 2 separate components. The frame must be rigid enough to support and carry all the loads and forces that the vehicle is subjected to during operation. The frame must also be flexible enough to handle shock loads and the twists, bends, sway and sag that it encounters under different road or load conditions. The frame should be able to flex under different situations, while being able to return to its original shape when loads or forces are removed.

Figure 21 – Chassis Drawing



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Chassis siderails C channel	AISI 1144	1
2	Square cross member	AISI 1144	3
3	C Channel cross member bumper	6063 T1	2
4	C Channel cross member	AISI 1144	1
5	C Channel motor support	AISI 1020 CD	1
6	C Channel motor support 2	AISI 1020 CD	2
7	Cab locking cross member	AISI 1020 CD	3

Overall chassis dimensions (mm) based on Figure 21 drawing:

Table 11 – Chassis Dimensions

Front overhang (a)	700
Wheelbase (b)	3700
Rear overhang (c)	1600
Overall Width (W)	1200
Overall length (L)	6000
X = $L/2$	3000

The following materials, shown in Table 12, were considered for the chassis based on commonly used chassis materials in the industry.

Table 12 – Material Details

Material	**Modulus of Elasticity (MPa)**	**Poisson's ratio**	**Yield Strength (MPa)**	**Density (kg/m3)**	**Ultimate Tensile Strength (MPa)**	**Modulus of Rigidity G (MPa)**
ASTM A710 (34)	210000	0.3	550	7850	841	80000
Mild Steel (35)	220000	0.275	207	7850	345	80000
AISI 1144 (36)	210000	0.28	620	7850	745	75000

AISI 1144 was the final material selected through multiple trials and design changes to achieve minimum weight and withstand maximum combined stress with acceptable deflection range.

The following loading conditions in Table 13 were estimated on the chassis. (37)

Table 13 – Loading Conditions

Overall vehicle payload Capacity	22241	N	4000	lbs
Battery weight	18264.21	N	4105.96	lbs
Battery cover weight	1334.466	N	300	lbs
Invertor weight	343.35	N	35	kg
Motor	5150.25	N	525	kg
Misc weight (HVAC, harness, hydraulics etc)	4448.22	N	1000	lbs
Weight of the seats 2	667.23	N	150	lbs
Weight of the chassis cab	981	N	100	kg
Weight of the back cab	4448.2	N	1000	lbs
Gross chassis load	57877.93	N	**13012.124**	**lbs**

Vehicle Payload capacity = 4000 lbs. = 17792.8 N

Payload with 1.25 Safety factor = 17792.8 x 1.25 = 22241 N

Gross load on the chassis = 57,877.93 N

Total load on 1 beam = $l_1 = \frac{Gross\ chassis\ load}{2} = \frac{57877.93}{2}$ = 28938.9647 N

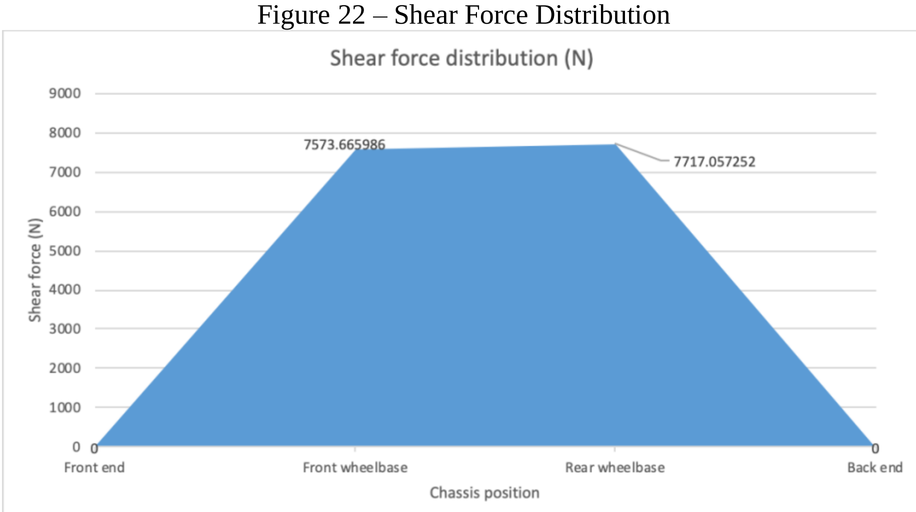
Distributed load per beam = $w = \frac{l_1}{L} = \frac{28938.9647}{6000}$ = 4.823 N/mm

Reaction force on the rear axle = $R_r = \left[\frac{w}{b}\right] x \left[\frac{a^2}{2} - \frac{b^2}{2} - \frac{c^2}{2} - (c\ x\ b)\right]$ = 17,989.086 N

Reaction force on the front axle = $R_f = l_1 - R_r$ = 10,949.878 N

Shear force: Front end = 0 N
Shear force: back end = 0 N
Shear force: front axle = $F_f = R_f - (w * a)$ = 7573.66 N
Shear force: Rear axle = $F_r = R_r + R_f - [w * (a + b)]$ = 7717.057 N

Figure 22 – Shear Force Distribution



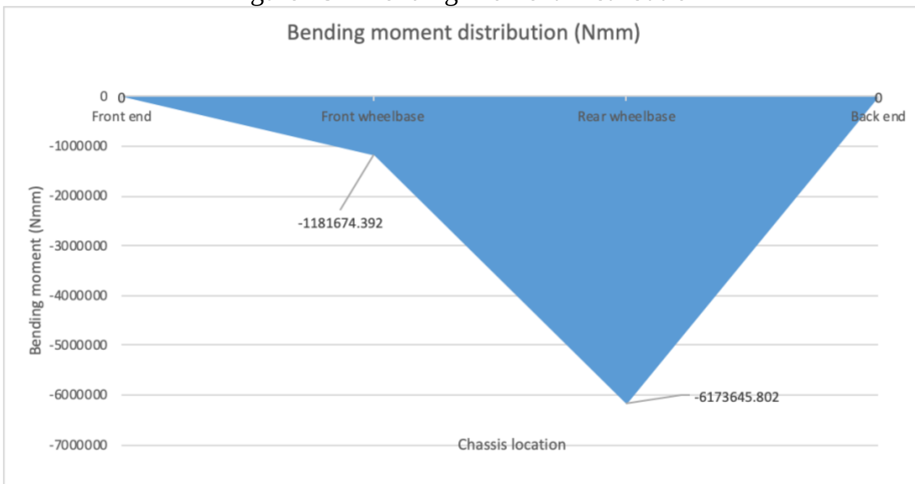
Bending moment: Front end = 0 N
Bending moment: Back end = 0 N

Bending moment: Front axle = $0 - \frac{w.a^2}{2}$ = -1181674.39 N mm

Bending moment: Rear axle = $\left[F_f . b\right] - \frac{[w.(a+b)^2]}{2}$ = -6173645.81 N mm

Maximum bending moment = $M_m = 6173645.81$ N mm

Figure 23 – Bending Moment Distribution

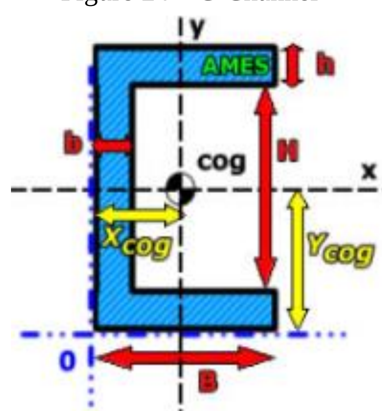


The following C-channel was selected for the side beam of the chassis based on various design changes and trails to minimize size and increase strength (38):

Table 14 – C-Channel Specifications

C 100 x 50 x 5 x 7.5		
H+2h	100	mm
B	50	mm
h	7.5	mm
b	5	mm
Ycog	50	mm

Figure 24 – C-Channel



Moment of inertia:

$$I_{XX} = \frac{H^3 * b}{12} + 2 * \left[\frac{h^3 . B}{12} + \frac{h . b}{4}\ (h + H)^2\right] = 1863697.92 \text{ mm}^4$$

Section modulus:

$$S_{XX} = \frac{I_{XX}}{Y_{cogg}} = 37273.95\ mm^3$$

Angle of twist is assumed to be $\theta = 0.017452\ rad$

Max bending stress applied on the beam

$$\sigma = \frac{M_m}{S_{XX}} = 165.628\ MPa$$

Shear Stress applied on the beam

$$\tau = \frac{G.\theta.W}{L} = 261.78\ Mpa$$

Combined (Von Mises) stress

$$VMS = \sqrt{}(\sigma^2 + 3\tau^2) = 482.7207\ MPa$$

Deflection of the chassis

$$Y = \frac{wX(b - X)}{24EI_{XX}}\left[X(b - X) + b^2 - 2(c^2 + a^2) - \frac{2}{b}\{c^2X + a^2(b - X)\}\right] = 5.7725\ mm$$

FEA analysis was conducted on SolidWorks as per standard procedure by creating the chassis assembly. Square tubing cross section were added to connect the C-channel sidebars to increase strength and reduce twisting. (37) The cross sections were placed strategically in the assembly to increase the safety of the chassis in case of a crash, support battery and motor sub-assemblies (maximum weight on the chassis) while keeping the weight of the chassis as low as possible.

All the parts of the assembly were merged together for the analysis. The chassis model was loaded by static forces uniformly distributed along the length of the side beams. The 4 boundary conditions are applied on the front and back of the chassis where the axles are attached to the chassis.

Figure 25 – Chassis FEA Results

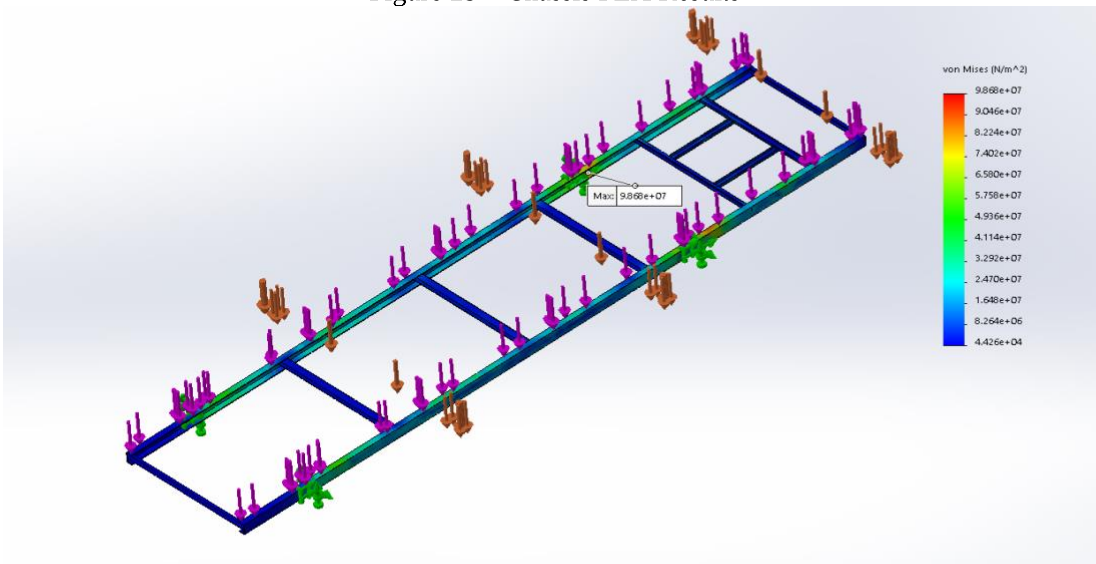


Figure 26 – Chassis FEA Results

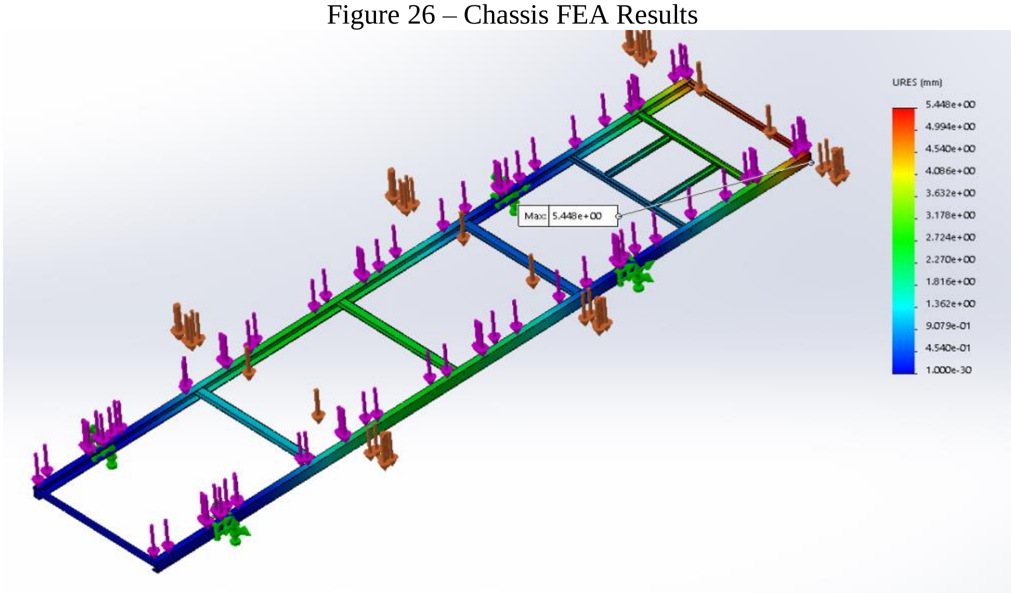


Figure 27 – Chassis FEA Results

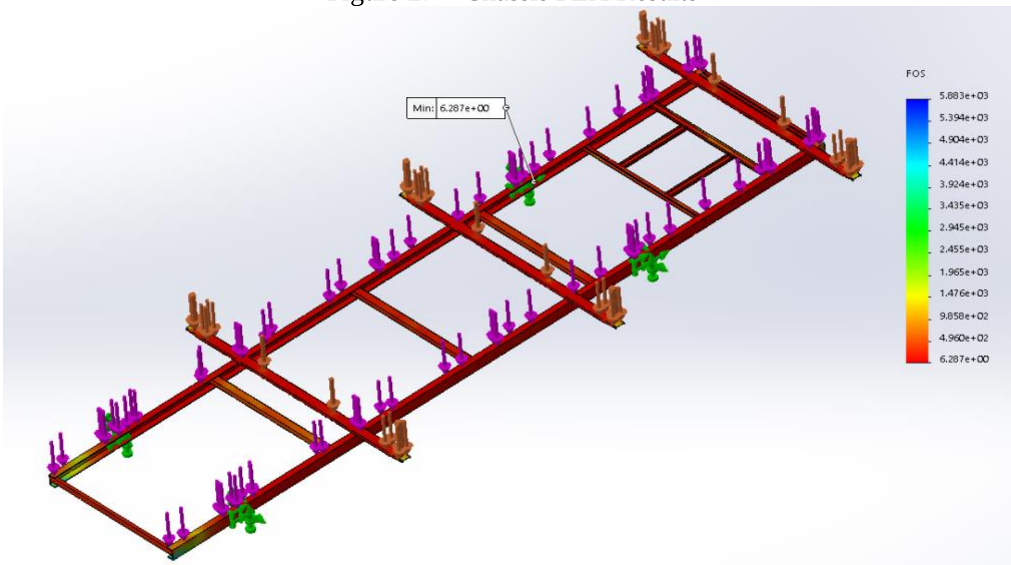


Table 15 – Chassis Stress Analysis

	Calculated results	**FEA results**
Combined (Von Mises) stress	482.72 MPa	98.68 MPa
Deflection	5.772 mm	5.448 mm
Factor of safety	1.25	6.28

The calculated and FEA maximum von mises stress are less than the yield strength of the chassis material.

The calculated deflection ratio $\left(\frac{5.772}{6000}\right)$ = 0.0019 and the FEA deflection ratio$\left(\frac{5.448}{6000}\right)$ = 0.0009 are within the safe limit according to the deflection span ratio (0.0005 to 0.003)

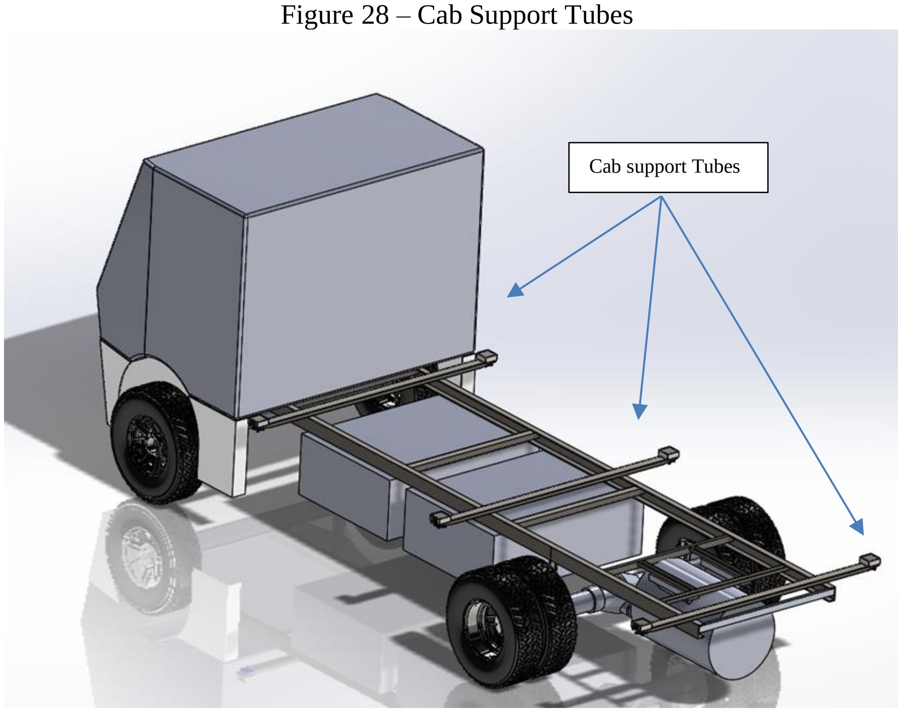
The minimum FEA factor of safety 6.28 is greater than the 1.25 FOS used as standard for ladder frame chassis analysis.

CAB LOCKING MECHANISM ANALYSIS

The objective of this analysis was to design a new locking mechanism to seamlessly attach and detach the bus and truck cab to and from the chassis, enabling swappable cabs. A rigid and reliable locking mechanism was designed considering constraints like machinability, ease of use, factor of safety, adhering to industry bolt torque standards and maintaining the cab at perfect level.

Instead of attaching the cabs directly to the bolt holes on the chassis side beams, three rectangular 1020 CD steel tubing were attached to the top of the chassis to easily attach and detach the cab from the main structure and always maintain cab level – Figure 28.

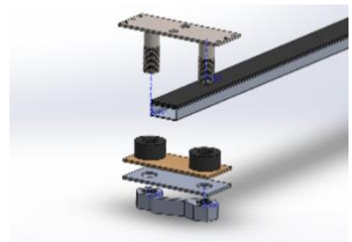
Figure 28 – Cab Support Tubes



The following two design alternatives were considered for the locking mechanism:

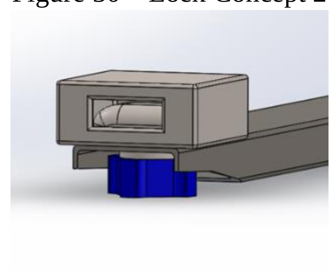
1. This mechanism, Figure 29, sandwiches the Rectangular square tube on the chassis between the custom threaded part (attached to the cab) and plates of plywood and SS that would be bolted down using 1 custom and 1 standard bolt. Rubber shock on either side of the square tube would reduce vibrations and thread damage. Upon assembly, the custom bolt would be tightened first and then the standard bolt. The custom bolt will align in such a way that loosening of the custom bolt would be prevented by the standard bolt, hence setting up a failsafe.

Figure 29 – Lock Concept 1



2. This locking mechanism, Figure 30, consists of a custom-made key which would lock the cab in place. The key would be tightened in place using a standard bolt with a custom-made polymer covering around it.

Figure 30 – Lock Concept 2



Due to the relative ease of use and machinability of the second concept design, it was used for this project. Each cab uses 3 locking assemblies on each side. Figure 31 is the detailed breakdown of the different components used in the mechanism.

Figure 31 – Locking Mechanism

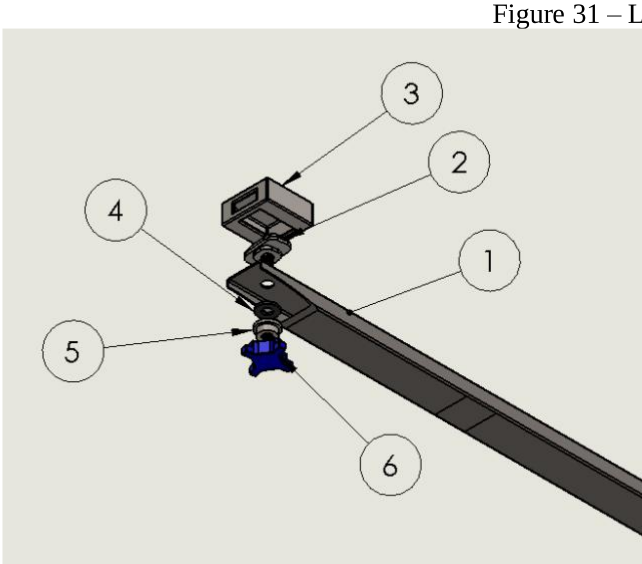


Table 16 – Locking Mechanism BOM

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	cross member locking support Final	AISI 1020 CD	1
2	locking key	201 ANNEALED SS	2
3	Cab attachment with key	201 ANNEALED SS	2
4	Polyutherane bushing	PU 11671	2
5	bushing support steel	AISI 1020 CD	2
6	plastic attachment to the nut	PET	2

Part number 3 is attached to the cab being placed on part number 1 which is attached to the chassis.

The following 2 parts were to be custom made, all the other are industry standard parts (39).

Figure 32 – Locking Key

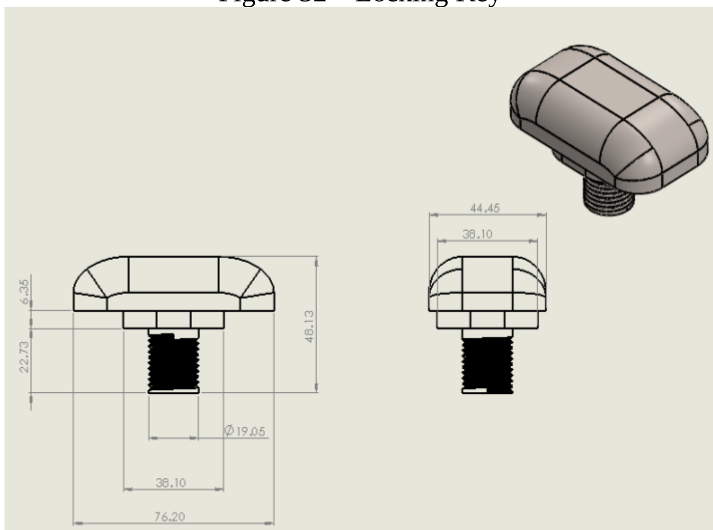
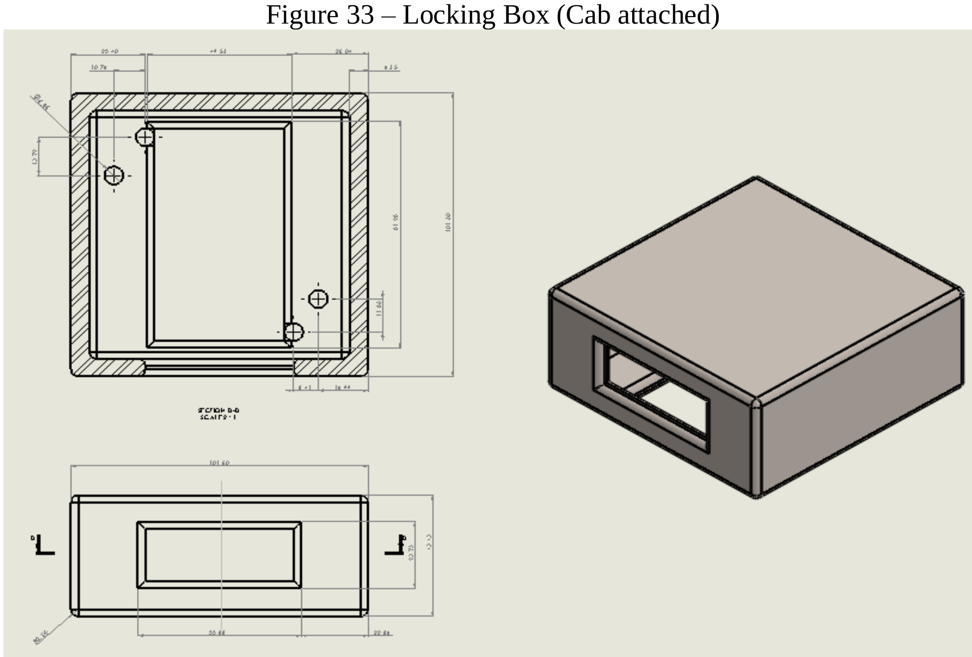
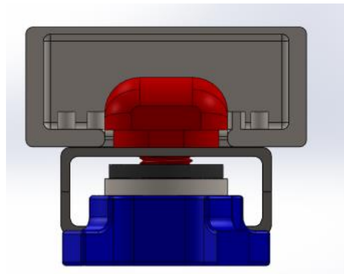
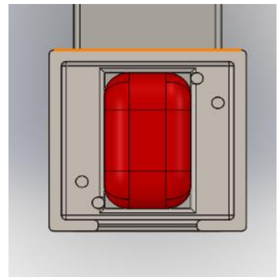
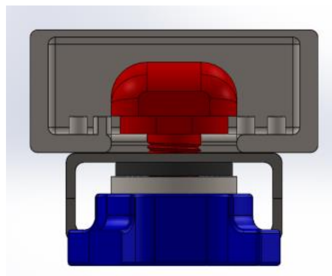
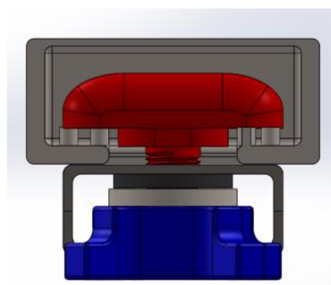
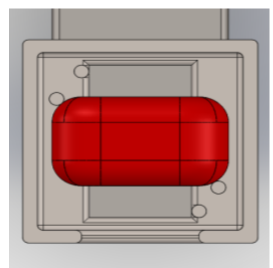
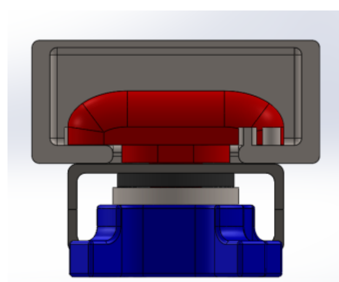


Figure 33 – Locking Box (Cab attached)



Below is a description of how this locking mechanism would function:

Table 17 – Locking Mechanism Function

Step	Description	Position (front view)	Position (top view)
1	When the cab is lowered onto the chassis, the locking key is parallel to the cavity in the box attached to the cab.		
2	The operator pushes the key up from the blue grip to clear the cavity on the box.		
3	The key is rotated clockwise 90 degrees. The extrude bars on the box help align the key to be perfectly perpendicular to the cavity.		
4	The key is tightened to hold the box, hence the chassis in place.		

The **FEA analysis** on this assembly was done considering a bolt torque of 80 Nm, which is the standard cab attachment bolt torque used by Ford and Chevy (40). Figure 34 and 35.

Figure 34 – Locking Mechanism FEA

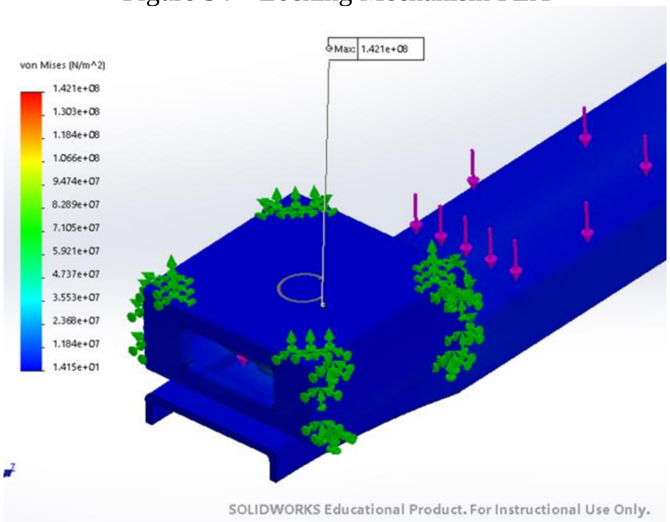


Figure 35 – Locking Mechanism FEA

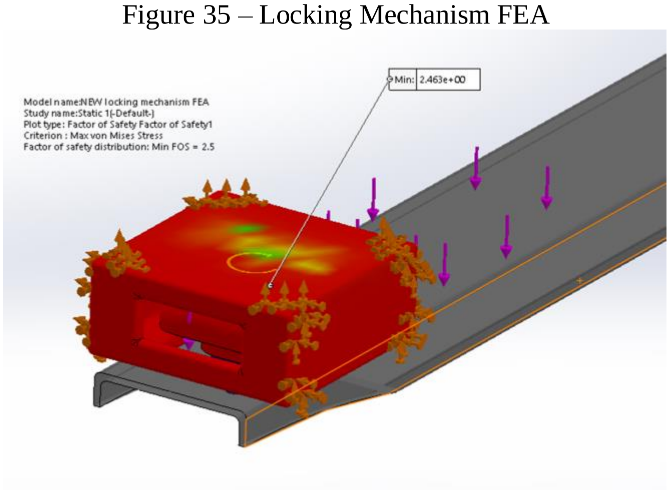


Table 18 – Locking Mechanism FEA Results

	Combined Stress (Max)	**Factor of Safety (min)**
FEA	142 MPa	2.46

Since the maximum FEA von mises stress is significantly less than the yield strength of the material used (350 MPa: AISI 1020) – The assembly is safe to use.

The general recommendation of FOS for use with ordinary materials where static loading and environmental conditions are not severe is 2 – 2.5 (41). Since the minimum FOS based on solid works FEA is 2.46, the assembly is safe.

BATTERY SWAPPING MECHANISM ANALYSIS

One of the main hurdles in widespread adoption of electric vehicles is the long charging downtime. One of the objectives of this project was to figure out ways to reduce charging downtime to 10 minutes or less. The following design alternatives were considered:

1. **Flash Charging:** With addition of supercapacitors along with the batteries, the bus battery can be charged up to 80% within 10 mins. This technology is still upcoming since smaller batteries are used to be able to fast charge, thus reducing the range significantly. But it is expected to become mainstream in the next 5 - 10 years. Figure 36. (42)

Figure 36 – Flash Charging

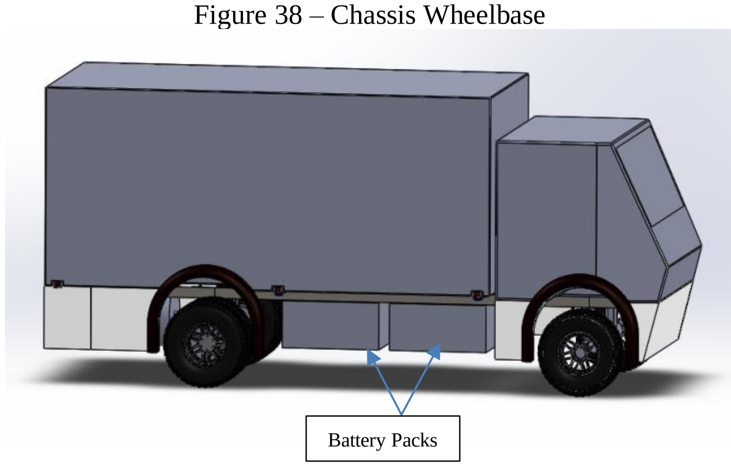
2. **Battery Swapping**: When needed, the battery box will be removed from the chassis and replaced with a charged box using automated robots. This has a high initial investment and big space requirement to set up a swapping station. Figure 37. (43)

Figure 37 – Battery Swapping

Designing a robotic arm and locking mechanism for battery swapping was beyond the scope of this project.

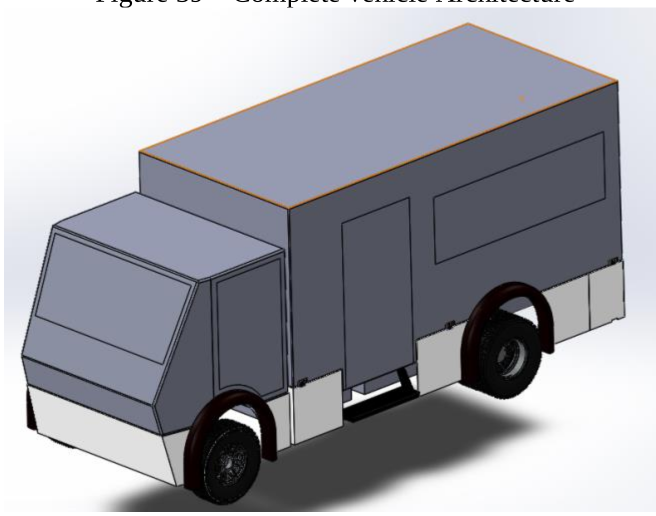
However, the chassis, axle and motor have been designed and assembled in such a way as to accommodate for battery swapping equipment into the vehicle in the future. The wheelbase is ideal for the swapping mechanism to move freely – Figure 38. The potential weight of the battery support rigs has also been accommodated in the chassis stress analysis, along with the concentration of battery weights within the wheelbase instead of distributing smaller packs throughout the chassis.

Figure 38 – Chassis Wheelbase



COMPLETE VEHICLE ARCHITECTURE ANALYSIS

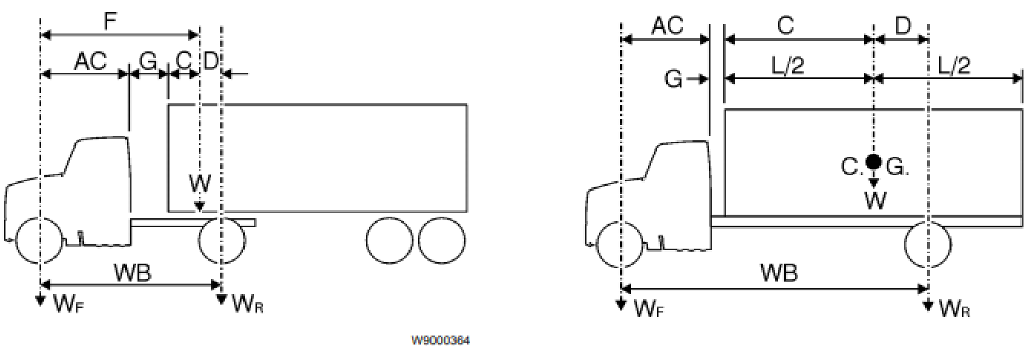
Figure 39 – Complete vehicle Architecture



Weight distribution and Tire selection

The center of gravity of the final vehicle assembly was determined using SolidWorks. The following formulas were used to determine the weight distribution between front and back axles (44):

Figure 40 – Weight distribution



AC Front axle to back-of-cab
G Gap between cab and body or trailer
CG Center of gravity of body and payload
C Front of body to CG, or front of trailer to kingpin
D Distance CG or fifth wheel is ahead of rear axle
F Distance CG or fifth wheel is behind front axle

L Body length
WB Wheelbase
W Weight of body plus payload, or kingpin load
W_F Portion of W transferred to front axle
W_R Portion of W transferred to rear axle

$$W \; x \; D = W_f \; x \; WB$$

$$WB = F + D = AC + G + C + D$$

$$W = W_f + W_R$$

$$\% \; weight \; on \; front \; axle = \frac{D}{WB} \; x \; 100$$

$$\% \; weight \; on \; rear \; axle = \frac{F}{WB} x \; 100$$

Using the above formulas, the following results were calculated:

Table 19 – Vehicle Specs

L	6200	mm
WB	3700	mm
W	6803.88	Kg
AC	850	mm
G	55	mm
CG	From SolidWorks	
D	1442.88	mm
C	1351.78	mm
F	2256.78	mm

Figure 41 – Center of Gravity

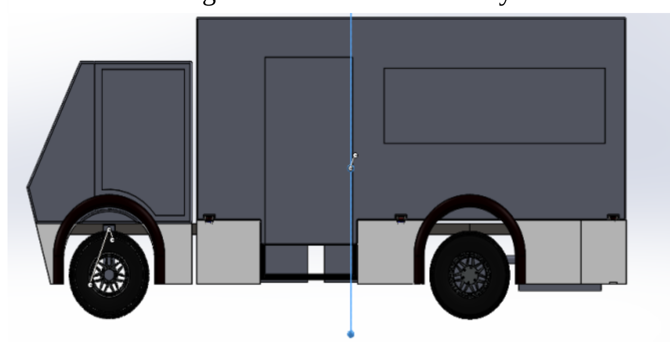


Table 20 – Weight Distribution

	Ratio	Kg	%
Weight on front axle	0.389968	2653.293	38.99676
Weight on rear axle	0.609941	4149.962	60.99405

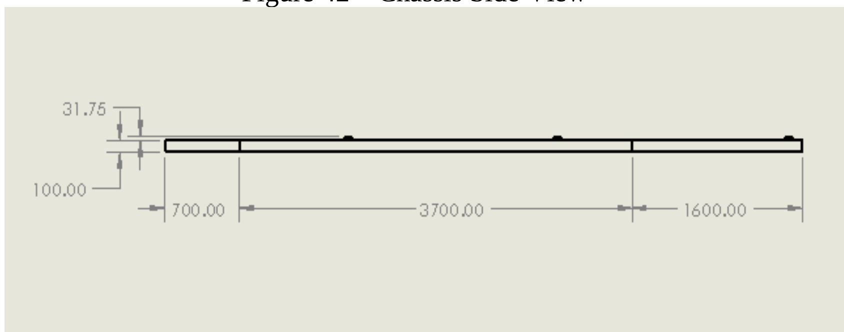
Based on the loads on front and rear axles, Duravis R500 HD (45) tires were selected, with a tire size of LT225/75R16 (Appendix 1). Each tire has a load capacity of 1215 kg @ 550 kPa. Hence 2 tires on the front axle (total load capacity 2430 kg) and 4 tires on the back axle (total load capacity 4680 kg) can safely withstand the maximum weight on the vehicle.

Industry Standards

In order to ensure the safety of the designs, the following industry standards were used as design constraints for the vehicle subassemblies architecture:

1. **Rear overhang:** For general access rigid vehicles, the rear overhang should be less than 60% of the wheelbase (46).

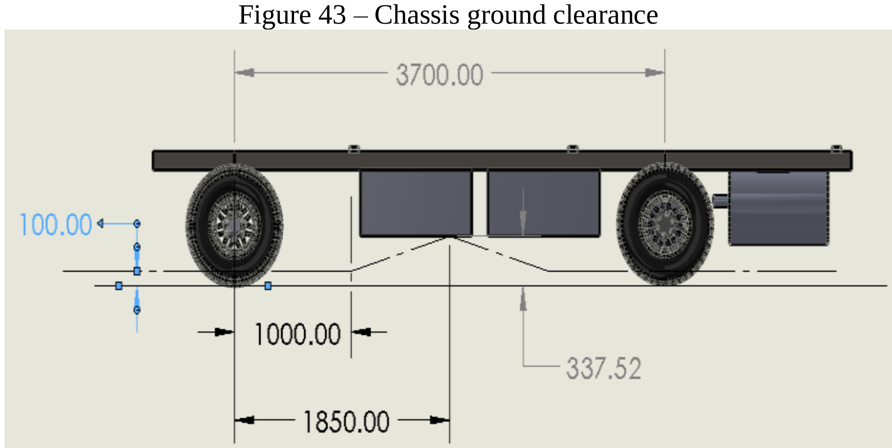
Figure 42 – Chassis Side View



Based on Figure 43, for this vehicle, the rear overhang is $\frac{1600}{3700}$ = 43.24 % of the wheelbase, hence adhering to the standard.

2. **Ground Clearance:** All vehicles must conform to the following clearance requirements (46). The ground clearance must:
   - **a.** be a height of at least 100 mm within 1 meter of an axle, and
   - **b.** be a height of at least 1/13th of the distance between centers of adjacent axles, measured at the midpoint between them (DA).

Figure 43 – Chassis ground clearance



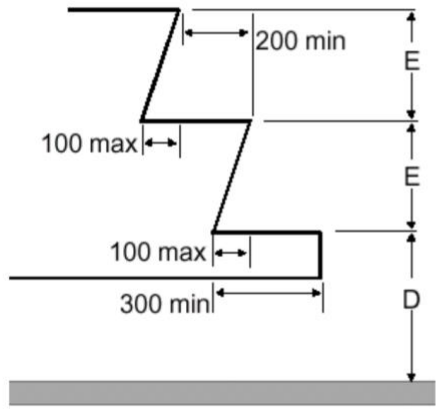
Based on Figure 43, the ground clearance is at least 100m within 1000m of each axle. The height DA of the vehicle is also greater than the required minimum height.

Table 21 – Ground Clearance

Min. required DA (m)	Actual Vehicle DA (m)
= 3700/13 =**284.61 m**	From the above drawing =**337.52 m**

3. **Step height:** The following Figure 44 standards need to be followed for the passenger cab steps (47)(63):

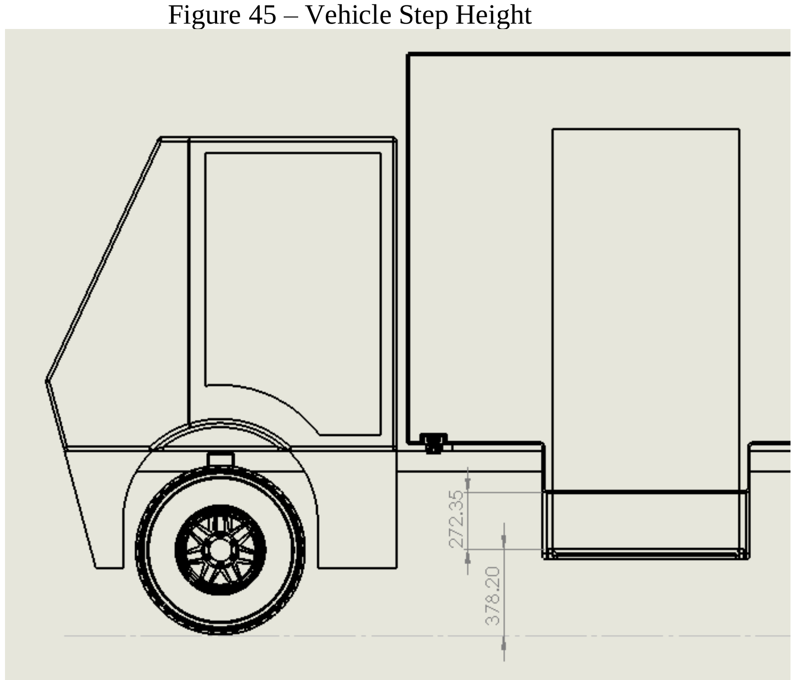
Figure 44 – Step Height



Steps

Classes		I	II, III
First step from ground 'D'	Max. height (mm)	340 (1)	380 (1)(2)(5)
	Min. depth (mm)	300 */	
Other steps 'E'	Max. height (mm)	250 (3)	350 (4)
	Min. height (mm)	120	
	Min. depth (mm)	200	

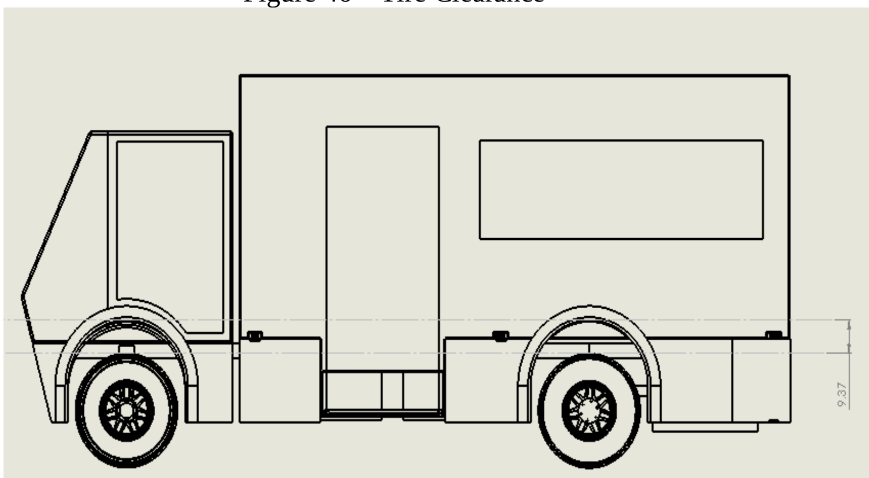
Figure 45 – Vehicle Step Height



For the vehicle, the length D is 378.2 mm and length E is 272.35 mm, both of which adhere to the industry standards

4. **Tire clearance:** Normal suspension movement could cause contact between the tires and the body to prevent this, the minimum clearance between the top of the tire and the bottom of the body should be 203mm (8in). (44):

Figure 46 – Tire Clearance



As per the above drawing, the wheels and the base of the cabs (loaded) have a clearance of 237.98mm (9.37in), thus exceeding the minimum standard requirement.

Bill of Materials

Table 22 – Vehicle Bill of Materials

	Item	Source (Manufacturer)	Additional information	Quantity	Cost	Manufacturing cost
Designed	Batteries	(25) Zeus Battery Products	Must be assembled into modules and Array	19240	$124,398.85	$6,219.94
	BMS	(48) Lithium Balance	Attached to each battery module	148	$8,880.00	$444.00
	Motor	(22) ATO	Must be mounted to the car chassis	1	$1,120.00	$56.00
	Inverter	(26) invt-vehicle.com	Must be mounted to the car chassis	1	$500.00	$25.00
	Differential	(49) Bronco Graveyard.com	Must be assembled	1	$485.00	$24.25
	Drive Shaft	(50) Metal Miner	Must be Manufactured and assembled	2	$77.00	$35.00
	Strip Chassis	Ford	Includes Chassis, Front Cab, suspension, front axle, brakes and hubs, tires, heat pump for cabin, harnessing, steering and other miscellaneous components	1	$60,000.00	$3,000.00
Purchased	Rear Suspension	(51) SD Truck Springs.com		2	$434.02	$21.70
	Brakes and hubs	(52) Auto Anything.com		1	$288.86	$14.44
	Heat Pump (Passengers)	(53) Thermo King	HVAC for bus cab	1	$3,000.00	$150.00
	Notes:					
	Miscellaneous	Miscellaneous components include: gages, oil pumps, electrical harnesses, air ducts, splash guards, mirrors, lights, dashboard, and interior				
	Manufacturing costs	Note: Most manufacturing costs are estimated to be 5% of the cost of the component. Otherwise, the cost is $35.00/hr of estimated machinist's work				
					Total	$209,174.06

It is important to note that the batteries alone account for 62.45% of the total cost of the vehicle. This is the limiting technology at this moment in time. If there was a lighter, more efficient way of storing energy, then electric vehicles would be much easier to integrate into current industries.

Scaled Model

For proof of concept and testing, a 1/8th dimensional scale model will be manufactured. The model will simulate how the full-size vehicle will react to certain conditions. For the battery and powertrain, electric skateboard components will be used as they best fit the size and load requirement of the 1/8th scale model.

Table 23 – Scaled Model Bill of Materials

Item	Source (Manufacturer)		Additional information	Quantity	Cost	Manufacturing cost
Battery and charger	(54)	Mboards		1	$124.99	$-
ESC and Controller	(55)	Mboards		1	$74.99	$-
Motor	(56)	Mboards		1	$84.99	$-
Wheels	(57)	Mboards		1	$99.99	$-
Pully and gears	(58)	Mboards	Assembly Required	1	$39.99	$-
Axles	(59)	Mboards	Assembly Required	1	$34.99	$-
Cab	(60)	Metals Depot	Fabricated sheet metal	4x8 sheet	$96.00	$-
Chassis	(61)	Metals Depot	Fabricated and Welded Steel Channel	10ft	$44.10	$-
Testing Equipment		Home Depot	Miscellaneous material used for testing		$100.00	$-
Miscellaneous Hardware		Home Depot	Nuts, Bolts, Paint, etc.	N/A	$50.00	$-
					Total	**$750.04**

PLANNED FABRICATION AND ASSEMBLY

Specialized Processes

The following are the specialized fabrication and assembly processes that were to be completed. The drawings and scaling reference are shown in Figure 48.

- Chassis: Steel channels will be used as side rails and square tubing for the cross section. The individual components will be cut and welded together as shown in Figure 47.

Figure 47 – Welded Chassis

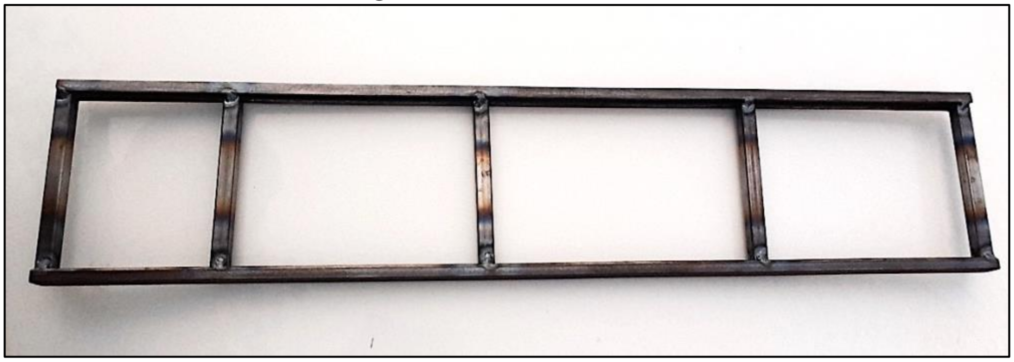
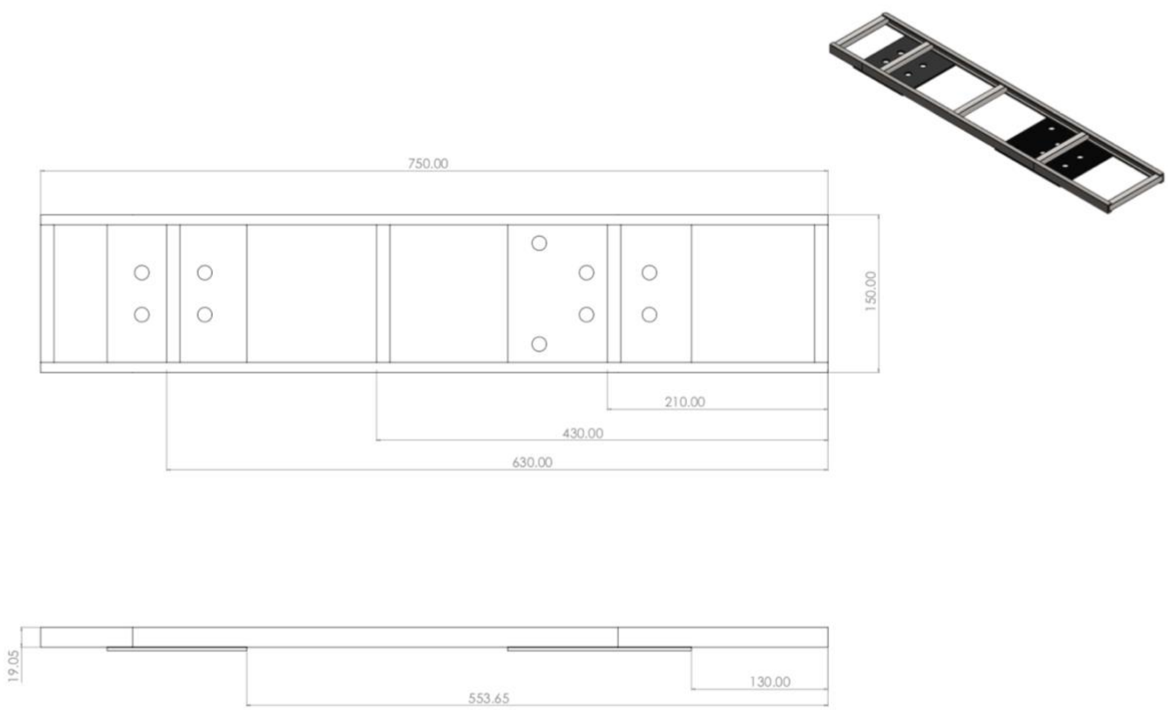


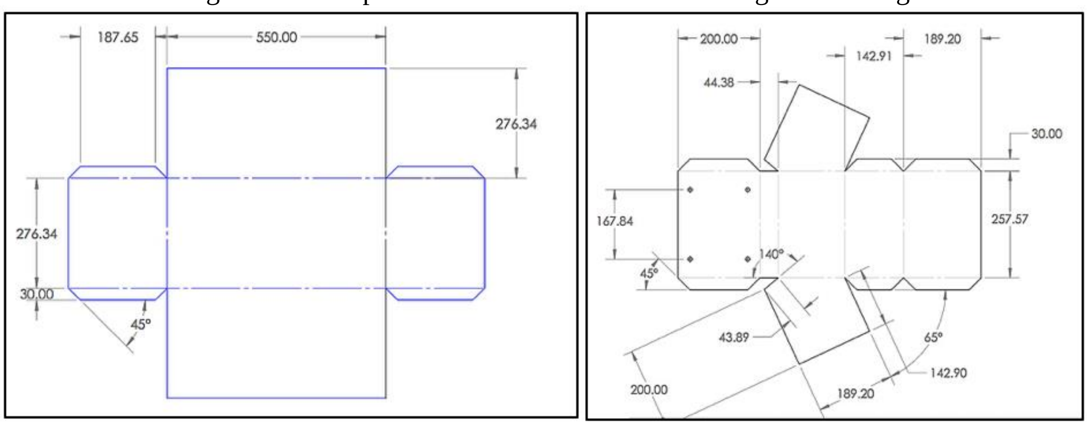
Figure 48 – Scale Model Chassis



- Cab: Steel sheets will be bent, spot welded and bolted onto the chassis. The cut sheets are shown in Figure 49. Figure 50 shows the shop references used to fold the metal for the final assembly.

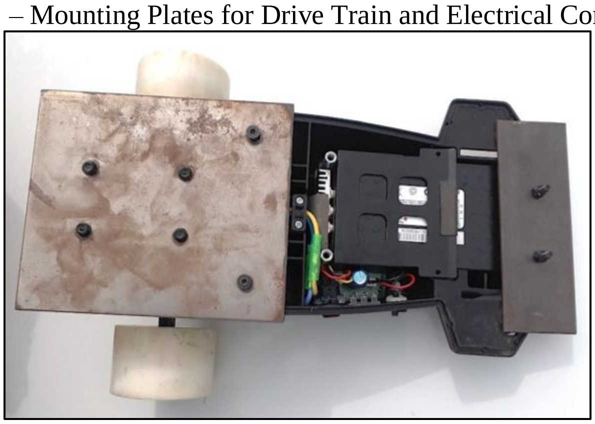
Figure 49 – Cut Sheet Metal for Cabs

Figure 50 – Shop Reference for Sheet Metal Cutting and Folding



- Battery and powertrain: Components purchased would be bolted onto the chassis. Figure 51 shows the powertrain and electrical components bolted to the mounting plates that are to be welded onto the chassis.

Figure 51 – Mounting Plates for Drive Train and Electrical Components

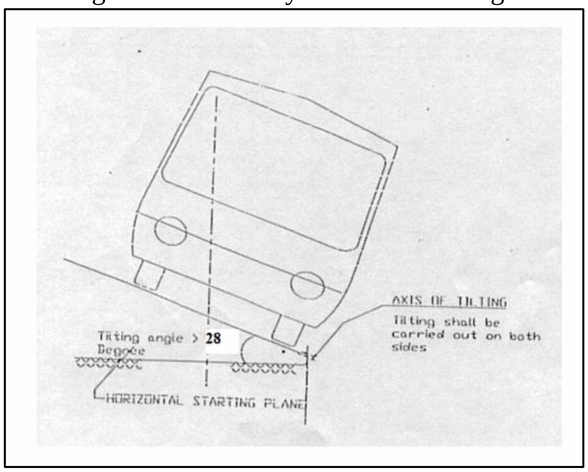


PLANNED TESTING AND PROOF OF DESIGN

The planned testing procedure will consist of the following:

1. Industry Standard (47)(63) Tip-over Test: 28 degrees

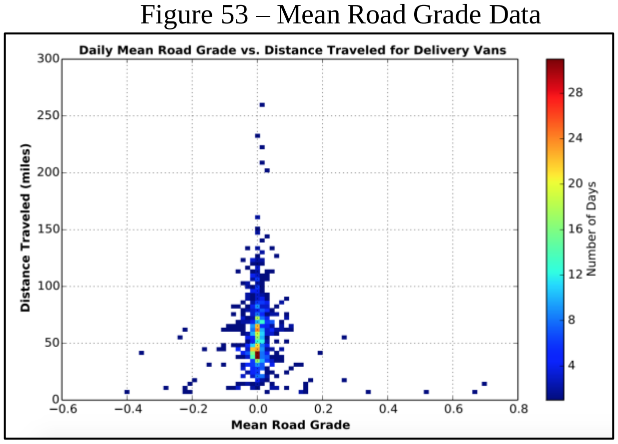
Figure 52 – Stability Test and Tilt Angle



(47)(63)

2. Gradient Test: 11-degree Maximum.

Figure 53 – Mean Road Grade Data



(29)

3. Loading/stress Test

These tests will be used to simulate what the full model will experience and must pass. A preliminary, in-process test shown in Figure 54 shows that the chassis was able to withstand its prescribed weight requirement by holding around 150 lbs.

Figure 54 – Preliminary Stress Test

This test was performed by approximating the locations of the axle mounting plates and the main concentration of the weight of the vehicle. A more precise test was to be performed once assembly was completed.

PROJECT MANAGEMENT

BUDGET

Proposed

This project will not exceed $750 for all expenses. It is personally funded by each member, being responsible for up to $375 each. This budget is based on the Bill of Materials for the scaled model.

Actual

The components purchased and used in the assembly are as follows:

Skateboard parts:	$114.15
Sheet Metal:	$26.75
Chassis C channel:	$20.07
Chassis square tube:	$8.83
Total:	**$169.8**

The skateboard parts include the cost of the wheels, axles, motor, drivetrain, motor speed controller, battery, electronics casing, wires, hand-held controller, and all mounting hardware. The sheet metal, used to create the cab and box section cost the amount shown above. All labor for assembly and fabrication was performed by the members and did not factor into this total cost.

The design for this section was significantly under budget, as only 22.64% was used for all the necessary components.

Due to the coronavirus pandemic and the early ending of the manufacturing section of this senior design, few components were not purchased. This consisted of hardware to assemble the sheet metal as well as paint to finalize the model. These components would not have cost more than $20 altogether ($10 for the hardware and $10 for paint) and would not have had a significant impact on the budget.

SCHEDULE

Proposed

Oct. 17 – Complete the calculations for the chosen concept
Oct. 29 – Complete full Solidworks Model
Nov. 26 – Complete full analysis of system
Dec. 13 – Complete requirements for scale model
Dec – Jan – Begin ordering parts, manufacturing, and assembly
Jan 31. – Design Presentation
Feb. 03 – Begin assembly
Feb. 20 – Complete assembly
Feb. 28 – Finalize Test 1
Mar. 20 – Complete final Test
Mar. 31 – Finish Presentation for Tech Expo
Apr. 9 – Tech Expo
Apr. 20-24 – Final Presentation

Actual
Oct. 17 – Completed the calculations for the chosen concept
Oct. 29 – Completed full Solidworks Model
Nov. 26 – Completed full analysis of system
Jan. 13 – Completed requirements for scale model
Jan- Feb – Began ordering parts for assembly
Jan 31. – Design Presentation
Feb. 03 – Began assembly
Mar. 13 – Finalize Load Testing
** Labs closed due to coronavirus pandemic and manufacturing requirements terminated
Mar. 20 – Nearly Completed assembly
Apr. 22 – Final Presentation

The full design and manufacturing schedule was close to the original intent. The design of the scale model was longer than proposed, however, this did not affect the scheduled completion date. The project was on schedule to have been completed and tested one week prior to tech expo as proposed. However, the assembly could not be completed due to the labs closing from the coronavirus pandemic.

PLAN TO FINISH

As stated before, there are several things that were uncompleted due to the closing of UC from the COVID-19 pandemic. The last major specialized process and operation, welding the mounting plates onto the chassis, could not be completed. There were two mounting plates for the axles to attach to the frame, one support plate for the battery, and then there were 3 smaller mounting plates that were to be welded above the frame for the cabs to bolt into the chassis. Once this operation was completed, the final assembly could be performed and the components could have been painted. After this, the final tests could be performed, which included a more precise stress test, the industry tip-over test, and the gradient test. The results from the test would be paralleled to the full-scale vehicle.

SUSTAINABILITY AND MATERIAL USAGE

The sustainability and material usage for this project could be improved slightly. When purchasing materials, there was always extra material in case a process or component failed. After the assembly and welding processes went well, the excess material was wasted. In future assemblies, that waste could be eliminated. Also, several materials were not ideal due to in-availability of supply. It was discovered that the manufacturer sent hot-rolled steel channel and cold-drawn square tubing. Finding small cold-drawn steel channel was a challenge and the hot-rolled was used, however, it introduced extra processing to the fabrication. The steel had a mill scale that had to be removed prior to the TIG welding process, as it would cause impurities and porosity in the welds. If all the steel was cold-drawn, that would not have been an issue and the fabrication would have been improved. 1/16th inch thick sheet metal was also a challenge to find. Aluminum sheet was used in this assembly, though it was not ideal. The spot welders at UC's campus did not provide enough power to weld the aluminum sheet, therefore, fasteners were used to assemble the cabs. For future assemblies, more of the budget would be used to purchase steel sheet to allow for spot welding and an improved assembly. The electrical and powertrain components worked very well and should not be changed significantly in the future. Instead of purchasing individual components and configuring them to work in the model, a complete electric skateboard was purchased. This ensured that all components were already well suited for each other prior to assembly. The only improvement would be to purchase a skateboard whose motor did not require a kick-start. This would allow for smoother testing as the motor could move the model on demand, without human interaction, though this improvement is not necessary.

CONCLUSION

This vehicle design would help integrate electric vehicles into current industries. It offers multi-functionality that would help offset its initial cost with the swappable cab and batteries. The vehicle is optimized for inner-city transportation, with a range of 62.14 miles (100km) and offers a payload capacity of 4000 pounds. By replacing conventional diesel vehicles running transportation services in most major cities throughout the world, there is the potential for reducing or eliminating the 46.5% of global CO_2 emissions caused by trucks and busses (62). This design has been proven by adherence to many industry standards. Unfortunately, due to UC closing from the COVID-19 pandemic, formal prototype testing was not able to be completed.

APPENDIX 1

Tire Specifications (45)

TECHNICAL DATA

SW Style	Tire Size	Load Range	Service Description	Material Number	Wt. (lbs.)	Measuring Rim	Overall Diam.	Overall Width	Static Loaded Radius	Min. Dual Spac.	Revs Per Mile	Tread Depth (32")	Max. Tire Load (Single)		Max. Tire Load (Dual)	
													Kg/kPa	Lbs/PSI	Kg/kPa	Lbs/PSI
Duravis® R500 HD																
BL	LT225/75R16	E	115/112R	192-659	41	(6.0) 6.0-7.0	29.2	9.0	14.0	10.2	709	14	1215@550	2680@80	1120@550	2470@80
BL	LT245/75R16	E	120/116Q	191-860	47	(7.0) 6.0-7.0	30.6	9.8	14.2	11.3	671	17	1380@550	3042@80	1260@550	2778@80

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