DESIGN AND DEVELOPMENT OF A CHASSIS CONCEPT FOR AN AUTONOMOUS AIRPORT SHUTTLE

ROVAN DIARY ALI
Acknowledgements

I dedicate this work to my beloved mother, Peri Amin, who despite being a single parent with three children, always tried to provide me with everything I needed during my childhood. Her fortitude and benevolence has inspired me throughout my entire life. I would like to acknowledge my gratitude to my older sister, Dalia Diary Ali, who foresaw my potential and encouraged the pursuit of my engineering career.

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Sammanfattning

Abstract

In this project, a chassis concept has been developed for a battery-powered autonomous vehicle. The vehicle is intended to be used at an airport for transporting people between different terminals. The objective is to develop a chassis which is anchored with modern requirements and futuristic research based on conventional chassis design methods in order to find an optimal solution for this specific vehicle. Literature studies have been conducted on future batteries, types of chassis, chassis materials, and optimal cross-sections. The chassis materials have also been analyzed from an environmental perspective and life cycle analysis (LCA). Based on this, it was found that the “skateboard” chassis model was optimal for the intended vehicle while Advanced High Strength Steel (AHSS) proved to be the most suitable material for the load-bearing structure. It is essential to keep in mind that this project has been carried out on a conceptual level within the framework of a degree project. This master thesis project aims to provide a solid benchmark for further development and research within the subject.
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1 Introduction

This master thesis project was conducted at ALTEN Sweden AB in collaboration with KTH Royal Institute of Technology. The project was done in accordance with both the company’s as well as KTH’s rules and guidelines regarding degree projects. ALTEN Sweden AB, headquartered in Gothenburg, is a Swedish consultancy company focused on providing solutions in different fields of technology. It was founded in 1991 as Xdin then changed its name to ALTEN Sweden in November 2013 after being acquired by the French technology consulting group ALTEN. One of the company’s largest areas of work is in the automotive industry and as the technology in society is constantly evolving, it forces the entire transport sector to move towards an innovative future. Components that were previously mechanically controlled tend to be electrified to a larger extent. Electrification is an important part of the development since there is now an international need for reducing the total carbon footprint from the transport sector. Self-driving vehicles, commonly referred to as autonomous vehicles will be eventually more common in the transport sector. Humanity should be able to transport themselves in both autonomous buses and private vehicles. There are huge profit opportunities when it comes to automated transport systems. Smoother driving, increased lane capacity and the ability to drive in tight convoys are some of the advantages that will lead to increased transport efficiency. The most significant profits will probably be in the form of reduction of harmful emissions and traffic deaths. However, with unmanned transport follows new challenges and these smart, autonomous and connected vehicles will give creative engineers and product developers a new freedom of design, allowing them to explore new chassis concepts. This thesis project within the automotive field for ALTEN aims to generate an innovative concept solution for an electric vehicle chassis. The general aim of the study is to meet the future requirements regarding chassis for fully autonomous vehicles. The project will contribute to an increased level of expertise within the area and thus, potentially increase attraction to well-established automotive industries by providing technology that is anchored to modern requirements. Therefore, this master thesis project can be a benchmark for future projects within the field of chassis design.
1.1 Objectives

Based on detailed study analysis, future requirements and regulations, a chassis will be designed for an autonomous road vehicle. The vehicle is intended to be used at an airport with the purpose of transporting people with their luggage between different gates and terminals. The project starts from a clean sheet, indicating no starting positions or guidelines for dimensions, weight, materials, etc. Hence, the idea is based on designing a future chassis concept whose parameters are determined based on the purpose of the vehicle and area of usage. During the course of the project, different parameters of similar modern vehicles will be evaluated, in order to determine plausible parameters for this project. Before starting the research and development, there are certain requirements for the vehicle that have been predetermined together with ALTEN and they will form the basis of the project. The requirements are as follows:

- Adequate flexibility in the chassis
- Electrically powered by batteries
- Capacity for 15 passengers including luggage
- Estimated Gross Vehicle Weight (GVW) of 3000 - 3500 kg
- Maximum speed of 45 km/h
- Cost-effective
- Environmentally friendly
- Comfortable travelling

The chassis will be developed in SolidWorks 3D CAD. The 3D modeling will be in full-scale, however, a scaled-down and simplified model will be built up as a physical prototype. The primary objective of this is to view the design from other perspectives and to have a physical model for presentations.

1.2 Delimitations

The delimitations of this project are as follows:

- This is a thesis project in vehicle engineering, thus the focus will be limited on mechanical design & vehicle dynamics.

- The project’s time-span is limited to 20 weeks, where each week consists of 40 hours of work. A time-plan was made in order to organise the project. 6 weeks of the project’s time-span will be dedicated to the theoretical part, 3 weeks to concept generation, 3 weeks to first draft simulations, 2 weeks for finalizing the simulations, 4 weeks for realizing the prototype and the rest for finishing the work.

- Development of a scaled-down prototype can be complex since some geometric ratios will be difficult to obtain. Thus there is a risk that different parts of the model will be scaled down by different ratios.
2 Research methodology & work approach

When conducting a thesis project, it is essential to support the work and research behind it with methods and methodologies as these will determine the results of the work. A method is the procedure which is used to formulate facts; a methodology is the reasoning behind a method and the justification of using a certain method instead of another. Methodologies can be divided into two categories: quantitative and qualitative. Methods are then chosen based on the methodology which the project adheres to. Quantitative methodologies require significant amounts of data for generating conclusions and demand statistics to verify the hypotheses, whereas qualitative methodologies use modest amounts of data and involve analyzing behaviors to establish theories [1].

The thesis project is initiated with the mnemonic SMART (Specific, measurable, achievable, relevant, time-bound), which works as a guideline for carrying out projects. The project itself must fulfill the meaning of these words before the actual work can be started and the methodology can be evaluated. The methodology for the degree project broadly follows the following template: The work will start of by setting up at schematic project plan where the different phases of the project are roughly allocated into time periods. This defines the last letter in the mnemonic SMART which will make it easier to stay within the time-plan. The next step will be to begin with a brief literature study within the subject of chassis in order to estimate where the research is within the field at present. This will be in the form of analytical research method where pre-planned hypotheses are tested based on existing knowledge and results. This particular method is widely used for product development purposes [1].

The brief study is followed up by conceptual research which is a more advanced literature study where parallels are drawn in order to form new concepts or interpreting existing concepts. In this case, it will consist of first stage concepts of chassis design, however, it also include forming strategies and practices. The next step will be the data analysis method where all the collected data from previous steps will be analyzed in order to start with first draft CAD models and simulations. The first draft iterations must be evaluated with an inductive strategy in the qualitative research methodology. This means that the first draft models must be evaluated in terms of relevance, validity and reliability [1]. In this particular project this will be done by having a half-time presentation of which the first draft model will be presented and the received feedback will be evaluated. The last step will be to finalize the model with belonging simulations and prepare for an oral presentation. Documentation and report writing is an important part that will be done continuously through all phases. It is essential to not present conclusions that lack any foundation or evidence in the research work.
3 Market analysis

3.1 Electric vehicle trends

Research from the automotive industry says that the era of electric and autonomous vehicles is right around the corner [2]. Customers around the world are willing to buy electric vehicles (EVs) more than ever before. Many markets have registered a 50 to 60 percent increase in EV purchases in recent years [3]. With this sort of increase, it is important that research and development within the area follows the emerging era. Performance, reliability and range improvements are major and demanding factors that must constantly be evolved in order to convince the people to go towards the electric market. Getting people to move from vehicles with internal combustion engines (ICEs) to electric vehicles poses major challenges and is very demanding in terms of resources [3]. Most original equipment manufacturers (OEMs) do not make a profit from the sales of EVs [3].

Figure 1: Cost walk of ICE to electric vehicle in 2019 [3].

Figure 1 shows that electric vehicles on average often cost 12000 USD more to produce than comparable ICE vehicles in the small to midsize car segment and the small utility vehicle segment. This has become a problem for various OEMs since they try to recoup the costs through pricing alone. This makes it hard for customers to justify the purchase of a EV. Hence, many OEMs stand to lose money on almost every EV sold [3]. This unsustainable business is mostly due to the battery costs since it is the largest single factor responsible for this price difference [3]. Due to, amongst other things, a major progress in the battery development the prices are dropping and the production of EVs has risen more than ever before. From Figure 2 it can be seen that the price of batteries has decreased quite drastically over the years and is projected to fall further in the coming years.

Figure 2: Lithium-ion battery price outlook [4].

As the battery prices decline, the market price of EVs will reduce as well. According to
some researchers this will stabilize the economic aspect of the industry and hence absorb the losses. With the help of various trends, researchers have begun to estimate how the number of electric vehicles will increase in the future. The International Energy Agency forecasts that EVs will grow from 3 million in 2018 to 125 million by 2030. However, this number deviates from the prediction done by Bloomberg which forecasts that 57% of all global passenger vehicles sales will be electric by 2040 which is approximately 57 million vehicles, which can be seen in Figure 3 [5] [6].

![Figure 3: Global long-term vehicle sales by drive train](image)

### 3.1.1 Future batteries

Solid state batteries are a growing alternative for next-generation traction batteries. Currently, Lithium-ion batteries are being used widely because of their effective performance over a wide temperature range (from few tens of degrees below 0°C to about 100°C) and their high energy density. However, there are some disadvantages e.g. high flammability and risk of leakage at the electrodes which leads to capacity loss. Research has shown that Solid electrolyte lithium-ion cells does not show the same disadvantages. In fact, due to higher electro-chemical stability and high potential cathodes, increased performance and safety with the combination of lower cost is expected.

At a summit in Berlin on June 17 2019, the research institute Imec announced a solid-state Li-metal battery cell with an energy density of 400 Wh/liter and a charging speed of two hours. The specific energy for this battery was 480 Wh/kg. This was claimed to be a world record combination for a solid-state battery. The research continues with the goal of finding the batteries of the future in order to increase the reliability of the electric transport sector. The research institute Imec continues to follow its road map and their goal is to reach densities over 1000 Wh/l at a charging speed of less than half an hour by the year of 2024 [7][8].
In Figure 4 we can broadly see the physical difference between the two batteries. Since solid systems are more efficient the demand for spacious cooling system are less, they weigh less and need less space for packaging than lithium-ion batteries for powering electric automobiles. This is something that one strives for when it comes to the electric vehicle industry.

4 Estimation of required battery size

Although this project is within the field of vehicle dynamics and mechanical engineering, the battery dimensions are something that must be taken into consideration when designing a chassis for a Battery Electric Vehicle (BEV). The batteries with the associated packaging system tend to take up a large portion of the chassis space, which by itself is very limited due to the increasing number of components that switches to fully electric mode. To be able to design a realistic chassis, it is essential to estimate a battery size for this particular vehicle. Since the vehicle is not built yet, there are no pre-determined values to consider when it comes to energy requirement, thus the batteries will be selected based on research for similar vehicles, chosen driving cycle and estimated loads that have been done previously.

4.1 Driving cycle

The Worldwide Harmonized Light-Duty Vehicles Test Procedure (WLTP) is going to be used for estimating a driving cycle. For this particular project, the WLTC class 1 driving cycle will be used since the average speed and the overall drive pattern fits best with the one intended for the upcoming vehicle. Figure 5 shows the drive pattern and the characteristics of the driving cycle.
4.2 Motion resistance equation

The energy consumption is calculated based on the road loads. The total road load \( F_{\text{tot}} \) [N] is the sum of the inertial force, road slope force, road load (friction) force and aerodynamic drag force.

The average energy consumption of the vehicle [Wh/km] is calculated based on the road loads. The total load is the sum of the inertial force, rolling resistance, road slope, and the aerodynamic drag force. Equation (1) shows the total longitudinal motion resistance which consists of the loads mentioned above.

\[
F = (m + m_j)\ddot{x} + mg(f_r \cos \alpha + \sin \alpha) + \frac{1}{2} \rho AC_x \dot{x}^2
\]  

where,

\( m \) = vehicle mass

\( m_j \) = equivalent mass of rotating parts

\( g \) = acceleration due to gravity

\( \ddot{x} \) = vehicle acceleration

\( \alpha \) = road inclination

\( f_r \) = Rolling Resistance
4.2 Motion resistance equation

\[ C_x = \text{Weighted aerodynamic drag coefficient} \]
\[ A = \text{frontal area of the vehicle} \]
\[ \rho = \text{Air density (1.226 kg/m}^3\text{ at } 15^\circ\text{C and 1.013 bar)} \]
\[ \dot{x} = \text{Vehicle velocity / oncoming air velocity}. \]

This equation has some parameters which are not defined and has to be estimated and pre-determined in order to do the calculations.

4.2.1 Equivalent mass of rotation

The inertia of rotating parts can be replaced by an equivalent mass \( m_j \). The equivalent mass increases with increase in gear ratio (U). Unlike a conventional ICE vehicle the BEV will not have as many rotating parts such as drive shafts etc. This makes it possible to neglect all the inertial masses except from the tires itself. The tires of the vehicle will have the dimensions of 235/60R18 with a side wall height of 141 cm. The Rotational inertia \( (J_{\text{wheels}}) \) is estimated to be around 1.6 kg-m\(^2\)\[10\]. The equivalent mass of inertia for all four wheels can therefore be calculated as:

\[
m_j = 4 \cdot \frac{J_{\text{wheels}}}{r_{\text{tire}}^2} = 4 \cdot \frac{1.6}{0.3696^2} \approx 47 \text{ kg} \tag{2}
\]

Where \( r_{\text{tire}} \) is the radius of the tire.

4.2.2 Acceleration

The comfort of the passengers plays an important role in limiting the acceleration of the vehicle. Some passengers will be seated and a few of them will be standing up. The vehicle should have a reasonable acceleration from standstill so it does not affect the balance of the standing passengers drastically. According to Madison Area Transportation planning board, typical acceleration values for medium sized buses are between 0.894 - 1.118 m/s\(^2\)\[11\]. The acceleration values used in Figure 5b seems realistic and hence they will be used without any changes.

4.2.3 Weighted aerodynamic drag coefficient & frontal area

The vehicle will be in a rectangular form since it is a medium sized bus. The aerodynamic drag coefficient for a typical bus lies between 0.6 - 0.8\[12\]. Since it will be smaller than a typical commercial bus, the value of 0.6 is chosen. The frontal area of a similar vehicle is chosen for the power requirement calculation. The electric and driverless EZ10 shuttle has a frontal area of 4.7m\(^2\)\[13\], which is appropriate to use.
4.2.4 Rolling resistance

The rolling resistance coefficient ($f_r$) of car tyres on asphalt is 0.02 \[14\]. By applying the motion resistance Equation \[1\] the rolling resistance and aerodynamic drag can be compared. Figure \[6\] shows the aerodynamic drag and the rolling resistance as a function of velocity.

Figure 6: Rolling resistance & aerodynamic drag as function of velocity.

With the estimated parameters from the previous sections inserted in Equation \[1\] it can be seen that the rolling resistance is more dominant for speeds less than 20 m/s.

4.3 Power requirement

The total power $P_{tot} \ [W]$ is calculated as the product between the total road forces and the vehicle speed:

$$ P = F \cdot v, \quad (3) $$

Where $v$ is the longitudinal velocity and $F$ is the longitudinal motion resistance as in Equation \[1\].

By integrating the total power $P_{tot}$ over the duration time for the drive cycle, the total energy consumption can be calculated as:

$$ E_{tot} = \int_{t_{start}}^{t_{end}} P \, dt. \quad (4) $$

The WLTC class 1 driving cycle is a well known cycle which is used in a lot of similar examples \[9\]. The equations mentioned above can be downloaded as Xcos block diagram model (see Figure \[7\]) where the parameters can be changed as required.
4.3 Power requirement

ESTIMATION OF REQUIRED BATTERY SIZE

Figure 7: Xcos block diagram for WLTC energy consumption [15].

The model is run for 1967 seconds, which is the total time for the WLTC class 1 drive cycle. The clock block in the upper left corner in Figure 7 generates a time step of 1s because of the fact that the WLTC speed profile is sampled at 1 second. The necessary input data such as the parameters mentioned above and the speed profile is read from the workspace block. Before running the simulation all the variables must be loaded in. The acceleration will be calculated between two velocity points with the time step of 1 second in between. This indicates that the sign of the power can be either negative or positive depending on if the vehicle is braking or accelerating, thus the sign of the power output is used to distinguish braking and acceleration. All the integration of the power with time step 1 second is added up and finally the last calculated value which is the total energy consumption, is divided by the length of the drive cycle. The last calculated value of the energy (1515.35 Wh) divided by the total length (11.428 km) gives the average energy consumption of 132.6 Wh/km.

One driving cycle (1967 s) is almost 30 minutes long which means that the vehicle should complete 24 cycles in order for it to run half a day without having to be recharged. 24 cycles have the total distance of 274 km. The total energy needed for traction would be the product of the total distance and the average energy consumption which is 36.33 kWh. However, the energy available should not only be enough for traction but also for the auxiliary systems such as heating, ventilation, and air conditioning as well as the overall light systems. These systems should be able to work during standstill. On average the energy required for these auxiliary systems can reach up to 18% of the total energy capacity [16]. With this in mind the total energy needed can be expressed as:

$$P_{used} = 36.33 + (0.18 \cdot 36.33) \text{ [kWh]}$$

(5)

Since it is not preferable to fully discharge a lithium battery and to have safety margin in terms of battery capacity. A state-of-charge (SOC) value of 85% is used which means that the total battery power needed is

$$P_{need} = \frac{P_{used}}{0.85} = 50.43 \text{ [kWh]}.$$  

(6)

If a energy density of 400 Wh/l is chosen according to section 3.1.1, the total battery volume is 126 liters which is the maximum battery size. However, if the energy density of 1000 Wh/l is chosen the battery volume will be only 50.43 liters.
5 Types of chassis

This section will highlight the most widely used chassis types at present. The chassis have different qualifications however it is essential to put these into perspective in order to eventually select one construction type for the project goal.

5.1 Backbone chassis

The backbone chassis has a strong tubular backbone which usually has a rectangular cross-section area. The backbone connects the front and the rear axles with associated suspension systems. This type of automotive chassis is mostly used in passenger cars and sports cars. However, it is also used for heavy duty vehicles with a lot of wheel axles which are used in the mining industry. It has a standard superstructure that can withstand torsional twist and subsequent wear, however, it does not provide safety against side collisions and the manufacturing process is complicated mechanically. The backbone chassis is not cost-effective when it comes to mass production due to its low range of use. The environmental impact due to the production of backbone chassis is relatively low since they are not produced in large scales and does not require complicated technical equipment’s and they are mostly made out of materials such as steel alloys, which in turn can be recycled as well since it does not provide complete safety, it needs additional absorbing structure to be mounted on the chassis which may increase the weight and thus the vehicle will need more power for traction. This may indirectly have environmental impact due to more batteries etc [17].

5.2 Uni-body "monocoque" Chassis

The unibody, also called the monocoque chassis is a one-piece structure which prescribes the overall shape of the vehicle. The design is based on the fact that the frame and the vehicle body are integrated into one single structure. The one-piece structure must be able to withstand all the forces that can arise during driving but even during collisions. Thus there are high demands regarding all the connections between the different components. When designed properly, a monocoque chassis is usually more rigid and offers better protection in a crash. If a light-weight material is chosen, the design is usually lighter than a similar design based on the body-on-frame principal which will be explained later. This type of chassis is not flexible since the design will be customized for a particular type of vehicle. The production of monocoque chassis for conventional vehicles are often made in automated workshops. Since the whole structure is supportive, it can be hard to repair damage from various accidents that have affected the structure and still maintain the same original strength [18].

5.3 Subframe chassis

Subframes are boxed frame sections that are attached to a monocoque car body. It is often used on the front end of the car, but recently it is also used in the rear. Most prominent are axle subframes which are used to attach the wheels and suspension to the vehicle. Subframe chassis are typically added to a monocoque chassis as a way of isolating the noise and vibration of the powertrain and suspension components from the rest of the vehicle body monocoque. This type of chassis can be stronger and lighter than fully monocoque chassis, however it is not suitable for larger vehicles due to the cost of monocoque bodies. Subframes can be found in many passenger sportscars like Lamborghini Aventador [19].
5.4 Frame chassis

The ladder frame with two straight longitudinal beams "long members" connected by several cross members is the oldest and simplest form of all automotive chassis designs but also one of the most used one. The ladder frame resists twisting better than a unibody vehicle which makes it generally preferred for towing or carrying heavy loads especially for off-road driving. The design is also easier to modify and repair if damage occurs. For larger passenger cars, buses and heavy duty vehicles the ladder frame with associated cross and side members forms the base of the chassis which connects the powertrain and suspension system. The vehicle body itself is thereafter put on top of the frame. This classical setup is often named "body on frame" chassis. The body-on-frame chassis is flexible since the OEMs could design different vehicle body styles while keeping the base of the chassis. The setup is often heavier than corresponding unibody setups yet less expensive [18].

5.5 Skateboard chassis

For electrically driven vehicles, a new chassis platform has been dominating the market which is called the skateboard chassis. The design consists of a low flat battery which forms the belly of the vehicle with the ability to be lengthened or shortened depending on the battery size requirement. Since the vehicles does not need a driveshaft and transmission going through the platform, the electric motors can be placed on the front and the rear ends of the platform. Similar to the body on frame concept mentioned in 5.4, the skateboard platform allows for a vehicle body being mounted on top. This gives a OEMs greater efficiency by reducing assembly line complexity while quickly adapting to different customer demands. This platform is used in many vehicle brands such as Tesla, Toyota Avalon and Lexus ES. For heavier vehicles the "belly" may need additional frame design in order to withstand more load [20].
6 Chassis properties

6.1 Analysis of different stresses

When designing an automobile chassis, prior understanding of the various conditions the chassis is most likely to face is of great importance. There are six major conditions which must be taken into account during the design process:

- Short duration load - while crossing a broken road
- Momentary duration load - during cornering
- Impact loads - due to collision
- Inertia loads - due to braking
- Static loads - due to chassis components
- Over loads - due to loads beyond design capacity

For this airport shuttle, there is a high probability that all these conditions will be realized except the impact loads since the vehicle will be driving in highly controlled areas, thus the risk for collision are minimal. During these conditions, the are four types of loading situations that might occur:

- Longitudinal torsion
- Vertical bending
- Lateral bending
- Horizontal lozenging

All these conditions and type of loads will affect the overall shape of the chassis as well as the choice of material. If a chassis is designed to be resistant to bending it may be lacking in resistance for torsion etc. Thus, it is important to compensate in the design process so that the final product will be optimal for its purpose. With this in mind, there will be less environmental impact.

6.1.1 Longitudinal torsion

An arbitrary road is rarely smooth. When a vehicle is driving on a bumpy road, the diagonally opposite front and rear wheels can roll over a bump simultaneously. This phenomenon will cause the chassis to twist which can be seen in Figure 8.

![Figure 8: Demonstration of how torsion load can occur](image)

This type of load on the vehicle is called a longitudinal torsion and it is one of the most important type of stresses that must be taken into account when designing the chassis. Lack of chassis torsional stiffness affects the lateral load transfer distribution, the suspension kinematics and it can trigger unwanted dynamic effects like rollovers, vibrations and resonance phenomena.
6.1 Analysis of different stresses

6.1.2 Vertical bending

A typical vehicle chassis is supported by the wheel axles which usually are places by the end of the vehicle. The vehicle weight, passengers and luggage can thus be concentrated around the middle of its wheelbase. This will cause the chassis to bend vertically which will basically lead to a sag in the central region of the chassis.

6.1.3 Lateral Bending

When the chassis is exposed to a lateral force due to a centrifugal force while cornering, side wind, camber of the road, the tyres will oppose the lateral forces.

![Lateral bending](image)

Figure 9: Lateral bending [21].

The reaction of this will be a bending moment acting on the side members of the chassis (see Figure 9), which can bow the chassis in the same direction of the lateral forces.

6.1.4 Horizontal lozenging

When a vehicle is driving forwards or backwards, it is continuously exposed to wheel impact with road obstacles, road joints, surface humps etc while other wheels is giving thrust. This situation will cause a rectangular frame chassis to distort to a form of parallelogram shape. This phenomenon is known as "lozenging". Figure 10 visualizes the situation.

![Horizontal lozenging](image)

Figure 10: Demonstration of how horizontal lozenging can occur [21].
6.2 Chassis material

In this section, different types of materials that are being used in the automotive industry will be discussed briefly. It is important to put the materials weight in perspective to each other as well as mentioning their yield strength which is the maximum stress that can be applied before the body changes shape permanently.

6.2.1 Advanced High Strength Steel

Automotive steels can be classified in several different ways. Common types include low-strength steels (interstitial-free and mild steels); conventional high speed steels (carbon-manganese, bake hardenable and high-strength, low-alloy steels). The large variety of the different types makes it the most used material in the automotive industry. There are certain types of steel which are more valuable due to its properties than others. Advanced High-Strength Steels (AHSS) are complex, sophisticated type of metal which undergoes several steps in the heating and cooling processes. This in turn makes it able to achieve a high strength, ductility, toughness and fatigue properties. Compared to mild steel, AHSS is lighter and more customized to the vehicle industry. Steels are categorized as AHSS when their yield strength reaches 550 MPa or higher. AHSS are more expensive than alloy steel or mild steel however, it is still more affordable than other types of materials for the same strength properties [22].

6.2.2 Aluminium

Aluminum in automobiles has been used for several years. The material itself comes with a lot of benefits such as light weight, corrosion resistance, high ductility, high mobility and high recyclability. The material is mostly used in passenger vehicles but in different quantities. It can be found in smaller portions in mass production vehicles such as Toyota, Nissan etc. However, in the more expensive category of passenger vehicles the material dominates. Aluminum is a fairly malleable metal, it is not often found in the list of the strongest metals with yield strength reaching 310 MPa, in fact, aluminum’s balance of malleability and strength is part of what makes it such a useful and versatile material. Manufacturers can shape it as needed while still being confident in its strength and durability. The price is what limits the use of aluminum for many OEMS, and for larger vehicles such as buses or heavy duty vehicles, Aluminum is used to a lesser extent [23].

6.2.3 Magnesium

In the 1920s magnesium alloys began to make an appearance in the automotive industry, especially in racing cars. Slowly the material also began to be used in commercial vehicles and the interest for the material has increased over the past ten years. This is mostly due to increasing environmental and legislative influences. The material is about 34% lighter than aluminum which is already relatively light and it is one of the most recyclable materials in the market. Magnesium is very malleable and it offers sufficient strength to produce some components for the automotive industry. It is often used in gearbox, steering columns, coverplates etc. Due to its low yield strength which reaches up to 300 MPa it is not used for load-bearing structures and especially not for heavier vehicles such as buses and trucks. The price is another factor which limits the use of this particular material since it can be twice as expensive as Aluminum [24].

6.2.4 Titanium

The automotive applications of titanium and its alloys follow logically from high strength, low density and, low modulus, and they have excellent resistance to corrosion and oxidation. Titanium is mostly used in engine components, such as valves, valve spring, retainers, and connecting rods. It offers great strength with yield strength reaching up to 880 MPa for
some type of alloys (Titanium Ti-6Al-4V). Compared to mild steel in a strength-to-weight ratio, titanium is superior, as it is as strong as steel but up to 45% lighter. Titanium has the highest strength-to-weight ratio of all metals. In general, titanium will usually be more expensive than other metals because it is rarer and because it is typically only found bonded to other elements which can make processing more expensive [25].

6.2.5 Carbon fibre-Reinforced plastics

Carbon fiber-reinforced plastics (CFRPs) have been spreading recently into industries such as aerospace and automobiles. With their excellent specific yield strength reaching up to 1230 MPa on average combined with the modulus, and fatigue strength, the material has been widely used in air frame structural applications, especially for aircraft structures. This is particularly due to requirements for light-weight and high-strength materials to reduce fuel consumption for economic and environmental reasons, while maintaining safety standards and durability. The application in the vehicle mass production industry has not been as noticeable due to the high cost of the material. Carbon-fibre in general is classed as a premium material used for the high end vehicles providing extreme weight reduction. Even though the material is optimal in terms of weight reduction and high strength, it is not as recyclable as the previously mentioned materials [26].

6.2.6 Glass fibre-reinforced plastic

Glass fibre-reinforced Plastic (GFRP), otherwise known as Fibre Reinforced Polymer (FRP), offers many advantages such as corrosion and chemical resistance, non-conductive properties, cost effective, durability and sustainability. The material has a high yield strength reaching up to 1600 MPa while it can be up to 75% lighter than mild steel. It is often used in buildings for exterior cladding panels, load-bearing elements, drainage components and windmill blades. For high volume productions, GFRP is not the first choice for many vehicle OEMs due to the high cost compared to the metals discussed previously, however it is not as expensive as CFRP. Similarly as for CFRP, the material requires a lot of time to be manufactured and hence they are highly labor intensive and time consuming. Nowadays there are methods for recycling GFRP to greater extent but it is still not comparable as for the metals discussed previously [26] [27].

6.2.7 Summarized material data

<table>
<thead>
<tr>
<th>Material</th>
<th>Density [kg/m³]</th>
<th>E [GPa]</th>
<th>σy [MPa]</th>
<th>Recycle fraction [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>AHSS</td>
<td>7100</td>
<td>208</td>
<td>&gt;550</td>
<td>&gt;85</td>
</tr>
<tr>
<td>Aluminum</td>
<td>2700</td>
<td>69</td>
<td>&lt;310</td>
<td>&gt;90</td>
</tr>
<tr>
<td>Magnesium</td>
<td>1800</td>
<td>45</td>
<td>&lt;300</td>
<td>&gt;90</td>
</tr>
<tr>
<td>Titanium</td>
<td>4500</td>
<td>105</td>
<td>&lt;880</td>
<td>&gt;90</td>
</tr>
<tr>
<td>CFRP</td>
<td>1750</td>
<td>150</td>
<td>&lt;1230</td>
<td>&lt;10</td>
</tr>
<tr>
<td>GFRP</td>
<td>2460</td>
<td>17</td>
<td>&lt;1600</td>
<td>&lt;10</td>
</tr>
</tbody>
</table>

Table 1 summarizes material data. E is the Young’s modulus and σy is the yield strength.
Many automakers today work on three major areas to improve the fuel economy of their vehicles. Electrification, advanced powertrain technologies, and lightweight design have been widely analyzed by most of the OEMs in order to have minimal impact on the environment. Since the airport shuttle will be electrified and autonomous, another way of reducing the overall environmental impact is to reduce the weight. In the early stages of vehicle design, the aim was to reduce the weight by replacing some of the heavy cast iron and steel parts with lighter material such as magnesium and aluminum. However, advanced high strength steel (AHSS) emerged as a new lightweight material and it has lately become widely accepted by most of the OEMs due to their strength, production cost and new innovations in the manufacturing process which allows for mass production of the material.

One of the main issues associated with lightweight materials such as magnesium, aluminum, advanced composite materials etc is the high cost compared to conventional steel alloy. The cost difference of these materials could be up to 3-22$ per kilogram of total weight for parts made out of aluminum and magnesium and between 11-33$ for carbon fibre epoxy and advanced composite materials. Despite cost, there are other factors that needs to be analyzed in order to minimize the environmental impact. Materials such as carbon-fibre, magnesium, glass-fibre composites, advanced composite materials and aluminum are energy-intensive, with aluminum being the least energy extensive among these materials. The energy intensive parts are mainly within the extraction and manufacturing phases. Many research institutions focus on the usage phase of a certain vehicle and tends to neglect the energy and GHG emissions during the material extraction and manufacturing process. This situation may lead to a biased view towards choosing some materials over others. Choosing materials for different vehicle components can be difficult, especially when it should be durable and environmentally friendly. Many OEMs often end up facing conflicting objectives, often requiring a lot of compromises. For example, lightweight materials such as magnesium and aluminum tend to reduce the amount of energy required for traction during the use phase, however the materials are expensive and energy-consuming during the mining and refining processes. Carbon and glass fibre reinforced plastics (CFRP & GFRP) are known for their density, strength to weight ratio and low mass but the combination of the complexity in their manufacturing process and their high prices limit their use in the vehicle industry.

Sustainability is a concept which is often used in all types of industries especially within the automotive industry. However, the concept tends to be very broad and it can include a number of different parts to consider. Figure 11 shows one form of interpretation of the concept.

![Figure 11: Sustainability model and its branches](30)
As it can be seen, sustainability can be divided into three major fields which covers economic, environmental and societal factors. These in turn can be divided into several sub-fields. This model is quite comprehensive and some branches goes into each other. For example, design for environment include life cycle assessment which is another research area. The economic factors include the overall cost of a certain product, including everything from material extraction and production, through usage phase and maintenance as well as end-of-life and recycling/reuse phase. The prioritization of these sub-fields varies for different OEMs depending on the products. In this project, the analysis will mostly cover design for environment in the environmental pillar of sustainability. The economic factors will not be as broad as it is in the model due to the fact that the airport shuttle is not designed yet. However, it will cover a brief cost analysis with the term circular economy in mind. The societal factors will not be covered in the analysis of this particular project, but it will be a major field to analyze when the airport shuttle is ready to be manufactured and the OEMs plans the production location, safety conditions and manufacturing process.

The choice of material can drastically affect the life cycle assessment and the greenhouse gas (GHG) emissions throughout the vehicle’s lifetime. A study from the Swedish Energy Agency shows that some lightweight materials such as Aluminum alloys, magnesium alloys and other plastic-composites tend to be more related to high GHG emissions during the material extraction and refining processes. The study also shows that the manufacturing phase stands for about 2-4% of the total emissions released throughout the life cycle. The largest contributions regarding GHG emissions comes from the vehicles operating phase followed up by the end-of-life phase [31]. Fully recyclable materials are highly favored from LCA point of view since energy can be saved by recycling the materials when the vehicle is retired, thus the material can be used to create new vehicles.

8 Chassis shape & requirements

The most important aspects to consider when it comes to the shape of the chassis is the strength, stiffness, ergonomics, space and the weight. Since the vehicle will be electrically driven by batteries, the allocation of space for batteries is of great importance. Since the vehicle will require a high volume of batteries (126 liters) it is essential that the batteries are evenly distributed over a relatively large area. This prevents the vehicle from having possible mass-concentrations which may lead to unwanted load distributions. The batteries should also be placed low to the ground in order to get a low center of gravity (CoG). Even though the vehicle will not drive at high speeds and steep corners, the risk for roll-over should be minimized.

The chassis should be designed in way which makes it easy to get access to the electronics without having to disassemble the whole vehicle. This means that the powertrain, electronics, batteries and other associated equipment should have the possibility to be easily separated from rest of the vehicle body. The torsional stiffness is one of the most important properties of chassis since it significantly affects the dynamic characteristics such as handling and rollover. A high torsional stiffness is desired otherwise it may cause resonance or vibration [32].

Since the vehicle will be powered by batteries, the overall mass of the vehicle will play an important roll in terms of the range capability with a certain battery setup. The chassis should therefore be as lightweight as possible, however, it should be justifiable in terms of cost, durability, climate impact etc. The vehicle must be designed with regard to saving weight without substantially affecting the strength of the structure itself. Additionally, space for batteries and electric motors must be taken into consideration.
Material selection

9.1 Bending resistance (ASHBY method)

In order to select a material for maximum bending resistance, the ASHBY method can be used. Figure 12 shows a beam with length $L$, width $w$ and wall thickness $t$. The beam is exposed to a force $F$.

\[
\sigma_y > \frac{m \cdot \frac{t}{2}}{I} = \frac{3F \cdot L}{w \cdot t^2}
\]

\[
I = \frac{w \cdot t^3}{12}
\]

where $m = \text{mass}$, $w = \text{width}$, $L = \text{length}$, $\rho = \text{density}$, $t = \text{thickness}$ and $I = \text{second moment of inertia}$. When designing a vehicle, some dimensions must be kept constant to fulfill the requirements. In this example, the length and the width is kept constant while the thickness $t$ is a variable which can change to fulfill the requirements above. Equation 1 can be rewritten into:

\[
t = (\frac{3F \cdot L}{w \cdot \sigma_y})^{1/2}
\]

To include mass in the equation, the following equation can be written:

\[
m = V \cdot \rho = (w \cdot t \cdot L) \cdot \rho
\]

Now Equation 9 can be written as:

\[
m = (3F \cdot w)^{1/2} L^{3/2} (\frac{\rho}{\sigma_y^{1/2}}) = K \cdot (\frac{\rho}{\sigma_y^{1/2}})
\]

As it can be seen the first few parameters except the thickness ($t$) is kept as constant $K$. The ratio between the material density and the yield strength can therefore be as shown in Equation 11. In order to compare the materials, dual phase 280/600 steel has been selected as a reference material since it is one of the most used materials. The other materials will be compared with according to the following equation:

\[
\frac{m_i}{m_{ref}} = (\frac{\rho_i}{\sigma_y^{1/2}}) \cdot (\frac{\sigma_y^{1/2}_{ref}}{\rho_{ref}})
\]

Aluminum, AHSS, Magnesium, Titanium, GFRP and CFRP has been compared with dual phase 280/600 steel as Figure 13 shows.
The figure shows the average maximum and minimum ratios within each material class. The straight lines "error bars" represent the maximum and minimum ratio of each class.

9.2 Eco-material selection

As it was mentioned previously in section 7, the end-of-life phase of a particular vehicle is important in terms of GHG emissions. If the material can be recycled and used for other purposes, energy can be saved and reduce the amount of GHG emissions that would have been released if new materials were to be extracted and refined. If a material is selected also based on recyclability, the same problem formulation can be used as in Section 9.1. If the same beam: width w, length L and thickness t is subjected to a force F of which the objectives is to minimize mass and maximize recycle fraction ($\psi$) which is a value between (0 - 100%), the following expression can be made:

$$m \cdot \frac{1}{\psi} = V \cdot \rho \cdot \frac{1}{\psi} = w \cdot t \cdot L \cdot \rho \cdot \frac{1}{\psi}$$ \hspace{1cm} (13)

The stiffness (S) of the beam can be expressed as:

$$S = \frac{F}{\delta} = \frac{KEI}{L^3}$$ \hspace{1cm} (14)

$$I = \frac{w \cdot t^3}{12}$$ \hspace{1cm} (15)

where $m =$ mass, $w =$ width, $L =$ length, $\rho =$ density, $t =$ thickness, $K =$ constant, $I =$ second moment of inertia, $E =$ Young’s modulus. The variables in the equations above are specific to the material and the thickness (t) of the beam. When the thickness (t) is eliminated, Equations 13, 14, 15 can be re-arranged and written as:

$$\frac{m}{\psi} = \left(\frac{12 \cdot S \cdot w^2}{C}\right)^{1/3} \cdot L^2 \cdot \frac{\rho}{\psi \cdot E^{1/3}}$$ \hspace{1cm} (16)

Equation 16 is optimised and the optimal material is the one with the highest ratio.

$$Ratio = \frac{\psi \cdot E^{1/3}}{\rho}$$ \hspace{1cm} (17)
Figure 14: Recyclability plot for given materials [30].

The dual phase 280/600 steel is used as a reference in the Figure 14 where the dashed line represents the value for the reference material. The materials in the right side can replace dual phase steel in terms of lower weight and higher recyclability. GFRP and CFRP are good candidates for mass saving, however their recyclability can be a limit of their use.
9.3 Material selection matrix

Based on what has been discussed in Section 6.2, 9.1 and 9.2, a material selection matrix was developed and is shown in Figure 15 which includes all the materials discussed previously. The materials are evaluated on nine different selection criteria which are placed in three different priority ranks. The materials are given a score from 1-5 for each criteria. The net results within each priority rank is then multiplied by a weight factor which is 1.0 for high priority, 0.5 for medium priority and 0.25 for low priority. The score values for the criteria are defined as:

- Reliability: 1 (very low) :::::: 5 (very high)
- Recyclability: 1 (very low) :::::: 5 (very high)
- Cost: 1 (relatively expensive) :::::: 5 (relatively cheap)
- Weight: 1 (high weight) :::::: 5 (low weight)
- Durability: 1 (very low) :::::: 5 (very high)
- Maintenance: 1 (hard/cost ineffective) :::::: 5 (easy/cost effective)
- User-friendly: 1 (easy to handle) :::::: 5 (very hard to handle)
- Yield strength: 1 (very low) :::::: 5 (very high)
- Corrosion: 1 (high risk) :::::: 5 (no risk).

<table>
<thead>
<tr>
<th>Priority rank</th>
<th>Selection criteria</th>
<th>Aluminum</th>
<th>Magnesium</th>
<th>Titanium</th>
<th>CFRP</th>
<th>GFRP</th>
<th>Steel (AHSS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>High</td>
<td>Reliability</td>
<td>3</td>
<td>3</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>High</td>
<td>Recyclability</td>
<td>4</td>
<td>5</td>
<td>4</td>
<td>1</td>
<td>1</td>
<td>3</td>
</tr>
<tr>
<td>High</td>
<td>Cost</td>
<td>3</td>
<td>2</td>
<td>4</td>
<td>1</td>
<td>1</td>
<td>5</td>
</tr>
<tr>
<td>High</td>
<td>Weight</td>
<td>3</td>
<td>4</td>
<td>2</td>
<td>5</td>
<td>5</td>
<td>1</td>
</tr>
<tr>
<td>Medium</td>
<td>Durability</td>
<td>4</td>
<td>3</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Medium</td>
<td>Maintenance</td>
<td>3</td>
<td>2</td>
<td>3</td>
<td>1</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>Medium</td>
<td>User-friendly</td>
<td>5</td>
<td>5</td>
<td>3</td>
<td>1</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Low</td>
<td>Yield strength</td>
<td>2</td>
<td>1</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Low</td>
<td>Corrosion</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Net-result</td>
<td>12</td>
<td>30</td>
<td>36</td>
<td>29</td>
<td>34</td>
<td>39</td>
</tr>
<tr>
<td></td>
<td>Weighted result</td>
<td>21.5</td>
<td>21.5</td>
<td>23</td>
<td>18</td>
<td>20.5</td>
<td>24</td>
</tr>
<tr>
<td></td>
<td>Continue</td>
<td>no</td>
<td>no</td>
<td>no</td>
<td>no</td>
<td>yes</td>
<td></td>
</tr>
</tbody>
</table>

Figure 15: Material selection matrix.

As it can be seen, AHSS has the highest score in terms of both net result and weighted results, hence the material will be selected for the fundamental base of the chassis. This does not mean that every component of the chassis will be produced in AHSS, this strictly concerns the high load carrying frame of the chassis.
10 Chassis-type selection

The chassis types discussed in section 5 have similarly been evaluated in a selection matrix. In this case, the different chassis will be compared using ten selection criteria which are also placed within a priority rank as mentioned in section 9.3. The net results within each priority rank is multiplied by the same weight factor as mentioned before which is 1.0 for high priority, 0.5 for medium priority and 0.25 for low priority. The score values for the criteria are defined as:

- Safety: 1 (not safe) ····· 5 (very safe)
- Cost: 1 (relatively expensive) ····· 5 (relatively cheap)
- Environmental impact: 1 (low impact) ····· 5 (high impact)
- Suitable for BEV: 1 (not suitable) ····· 5 (suitable)
- Suitable for bus?: 1 (not suitable) ····· 5 (suitable)
- Weight: 1 (low weight) ····· 5 (high weight)
- Durability: 1 (not durable) ····· 5 (very Durable)
- Flexibility: 1 (not flexible) ····· 5 (flexible)
- Easy to manufacture: 1 (hard) ····· 5 (easy)
- maintenance: 1 (hard/cost ineffective) ····· 5 (easy/cost effective).

Figure 16: Chassis type selection matrix.

Figure 16 shows the selection matrix for the different type of chassis. As it can be seen the skateboard and frame chassis performs best in terms weighted result as well as the total net result. The chassis will be a mixture of a classical ladder frame chassis and a modern skateboard chassis.
11 Chassis design

The design of the chassis has been influenced by Equipmake’s new electric low floor bus combined with Tesla model S chassis.

Figure 17: Equipmake’s new low floor electric bus chassis [34].

Figure 17 shows Equipmake’s new low floor chassis for single deckers. This ladder frame chassis is optimized for its use and lighter in terms of weight than the previous models but also compared to its competitors [34]. Tesla model S chassis is a high quality chassis for electric powertrain. Tesla was one of the first to introduce the term “skateboard” chassis which is optimal for BEV [35]. Figure 18 shows the Tesla Model S chassis. As it can be seen the battery packaging is integrated inside the chassis and thus it becomes part of the supporting structure.

Figure 18: Tesla model S chassis [35].

11.1 Long members

The long members will be placed in the middle of the chassis and must therefore be strong enough to resist vertical bending but also to have the ability to hold lots of other bearing components. An I shaped beam is optimal for this use since it has a high moment of inertia which results in stiff section for vertical loads. The strength to weight ratio is also higher for I beams than it is for C-shaped or BOX-shaped beams. When an arbitrary beam is subjected to a vertical load it rotates around its z-axis like in Figure 19a. The outer layer of the beam is compressed while the inner layer undergo tension, see Figure 19b.

30
11.1 Long members

The middle layer in Figure 19\textsuperscript{[36]} is neutral, which means it does not undergo neither compression or tension. According to beam theory, an I shaped beam is not only effective against bending but also shear stresses. However, the torsional stiffness of the beam is not as effective. For torsional stiffness, hollow crosssections are more preferred which will be discussed later \textsuperscript{[37]}. The beam combined with other bearing components can be designed in order to be very effective against torsional stiffness as well. The long I-beam members can be seen in Figure 20\textsuperscript{[37]}

During bending load, the highest stresses occurs along the axial fibers farthest from the neutral axis as it can be seen in Figure 19\textsuperscript{[36]}. In order to prevent the beam from plastic deformation, most of the material shall be placed far away from the neutral axis. The ideal beam is the one with the smallest cross-sectional area for a given section modulus which depends on the moment of inertia. If most of the material is placed far away from the neutral axis, the section modulus becomes high which in turn means higher bending moment can be resisted \textsuperscript{[38]}. The design of the long members which can be seen in in Figure 20\textsuperscript{[38]} is based on this theory.

\begin{table}[h]
\centering
\caption{Parameter values according to Figure 20\textsuperscript{[37]}}
\begin{tabular}{|c|c|}
\hline
$Tw$ & $15 \text{ mm}$ \\
$Tf$ & $15 \text{ mm}$ \\
$H$ & $123 \text{ mm}$ \\
$W$ & $80 \text{ mm}$ \\
\hline
\end{tabular}
\end{table}

Figure 19: Beam properties

Figure 20: Long members
11.2 Cross members

As previously mentioned, the chassis frame during movement is subjected to lozenging, bending and torsional distortion. In order to resist these situations, there are various design proposals for cross-sectional shapes. These shapes come with different pros and cons regarding stiffness. The cross-members of the chassis play an important role in increasing the torsional stiffness. Research done in the department of mechanical engineering in Pune university, India, analysed cross-member design for improved torsional stiffness. In this research, a square cross-member with 4mm thickness and 80x80 mm dimensions was compared with a tubular cross-member with 4mm thickness and 80mm outer diameter. The analysis was aimed for heavy commercial vehicle chassis and was performed using the same load and boundary conditions for each type [39].

![3rd cross member in front torsion load case](image)

![7th cross member in rear torsion load case](image)

Figure 21: Torsion load [39].

It was clear from the analysis that the tubular cross-member performed better since the deformation in front and rear torsion decreased significantly.

![Comparison of torsional stiffness](image)

![Comparison of lateral stiffness](image)

(a) Comparison of torsional stiffness.  (b) Comparison of lateral stiffness.

Figure 22: FEA analysis results from the research at Pune university [39].

Figure 22 shows the increased torsional stiffness and lateral stiffness for tubular cross-
members as compared to the square and the existing C-shaped cross-members. Even though the vehicle on which the analysis was made for has a weight of 25 tons, the weight increase by using tubular cross-members was only 0.7 kg. The conclusion can be drawn that the hollow tubular cross-members is the best option for improved torsional stiffness for this particular purpose [39]. In order to increase the lateral stiffness even further, four additional c-shaped cross members are added. The price of these members are lower compared to hollow tubes for similar dimensions, and yet they are advantageous to use since they do not differ so much in terms of lateral stiffness compared to the hollow tubular ones (see Figure 36b).

![Figure 23: Cross member.](image)

(a) Hollow tube dimensions.  
(b) C-shape dimensions.

**Table 3: Parameters for the Cross-member.**

<table>
<thead>
<tr>
<th>Physical Properties (Hollow tube)</th>
<th></th>
<th>Physical Properties (C-shaped)</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Parameter</strong></td>
<td><strong>Values</strong></td>
<td><strong>Parameter</strong></td>
<td><strong>Values</strong></td>
</tr>
<tr>
<td>$b$</td>
<td>115 mm</td>
<td>$L$</td>
<td>500 mm</td>
</tr>
<tr>
<td>$L$</td>
<td>500 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$I$</td>
<td>1.42</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$W$</td>
<td>40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$S$</td>
<td>2.56</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$X$</td>
<td>7.44</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Thickness</td>
<td>2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$L$</td>
<td>500 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$P$</td>
<td>115 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$I$</td>
<td>40 mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$t$</td>
<td>3 mm</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 23 shows the designed cross members. The upper section in Figure 23a shows the side view of the tube and the bottom shows the cross-sectional view. A simulation is made in order to show the maximum shear stress which occurs on the chassis during torsional distortion as seen in Figure 24. Like the illustration in Figure 8, the chassis is fixed at the diagonal ends, whereby a force of 10000N is applied to the other diagonal ends. The momentary load is excessive and representing 5-6 mid-size passengers including luggage and it is concentrated at the very ends of the long members.
11.2 Cross members

(a) Simulation with only tubular cross members

(b) Simulation with C-shaped & Tubular cross members

Figure 24: FEA analysis.

Table 4: FEA results.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum shear stress</td>
<td>120 MPa</td>
</tr>
<tr>
<td>Minimum shear stress</td>
<td>0.2 MPa</td>
</tr>
<tr>
<td>Maximum deflection</td>
<td>6.9 mm</td>
</tr>
<tr>
<td>Material Yield strength</td>
<td>620 MPa</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum shear stress</td>
<td>72 MPa</td>
</tr>
<tr>
<td>Minimum shear stress</td>
<td>0.12 MPa</td>
</tr>
<tr>
<td>Maximum deflection</td>
<td>5.0 mm</td>
</tr>
<tr>
<td>Material Yield strength</td>
<td>620 MPa</td>
</tr>
</tbody>
</table>

The left image in Figure 24a and 24b shows the deflection results and the right side shows the stress results. By adding the C-shaped cross member, the maximum shear stress reduced by 60%. The safety margin to the yield-strength is 8.6 which can be considered safe. The torsional stiffness [Nm/deg] can be calculated since the deflection, length and the forces applied are known. Figure 25 is a simplified model of the deflected structure. The value dy in this case represents the maximum deflection in Figure 24a and 24b. The force F is known as 5000N and the length L is 500mm.
The torsional stiffness for the structure with only tubular cross members \( T_1 \) and for the structure with added C-shaped cross members \( T_2 \) can be calculated according to the following equation:

\[
T_i = \frac{M}{\tan^{-1}\left(\frac{dy}{L}\right)} = \frac{F \cdot L}{\tan^{-1}\left(\frac{dy}{L}\right)} \quad i = 1, 2 \quad [Nm/\text{deg}] \tag{18}
\]

With inserted values the torsional stiffness \( T_1 \) is 6324 Nm/deg and \( T_2 \) is 8706 Nm/deg.

### 11.3 Side members

As it can be seen in Figure 17, the new electric chassis from Equipmake have saved a lot of weight by having side members that are distanced a bit from each other. The side members will be covered with a steel plate, followed up by the battery-packaging equipment. The vertical forces exposed to the chassis will be evenly distributed over the total area covered by the side members.

Instead of using the C-shaped side members like the ones in Equipmake’s chassis, the theory of using hollow tubular trusses is being applied for optimizing the strength of the side members. Tubular trusses in general are being widely studied for structural optimization purposes, if designed correctly, they tend to be much more efficient in terms of weight and more supportive than other types of structures.

For maximum support, the angle between the tubes are 45°, hence they form "triangle" shapes which are known to be the most optimal in truss design [40]. Figure 27a shows the aesthetic difference of the side members. The long members have 14 side members evenly
distanced from each other which will take up the load in the center of the vehicle. An assumption is made that the load will be uniformly distributed over these 14 side members. The total load is assumed to represent 10 standing passengers and the belonging chassis components which makes 1000N seem relevant for the simulation of one side member.

\[
(a) \text{ Hollow tubular FEA results. Red area } = 1.57 \times 10^7 \text{N/m}^2 \text{ & } 6.8 \times 10^{-2} \text{mm}.
\]

\[
(b) \text{ C-shaped FEA results Red area } = 2.4 \times 10^7 \text{N/m}^2 \text{ & } 4.6 \times 10^{-1} \text{mm}.
\]

Figure 27: Stress & displacement.

The material yield strength is 620 MPa and from the FEA analysis, it is clear that the tubular truss side members performs best in terms of stiffness as well as deflection compared to the C-shaped side members. The safety margin until the stress reaches the yield strength for the tubular side member is close to a factor of 26.

11.4 Base of the chassis

The components discussed in Section 11.1, 11.2 and 11.3 forms the base of the chassis. Their task is to carry the largest and most important loads to which the vehicle will be exposed. Figure 28 shows the components assembled together.

![Figure 28: Long-, cross- & side members assembled.](image)

Figure 28: Long-, cross- & side members assembled.
To get a flat and solid surface on top of the chassis, the components will be enclosed by a 2mm thick cover plate which goes around the whole chassis. Due to a flat surface, the load on top will be uniformly distributed over the whole base rather than having stress concentrations on a few components. The cover plate will also protect the base chassis from dirt and rocks coming from the ground. Figure 29 shows the chassis with the cover plate wrapped around it.

Figure 29: Covered chassis.

11.5 Tires and electric motor area

The airport shuttle will have two electric motors for each pair of tires. The electric motors of EV cars are usually located along the front and rear axles of the vehicle (center mounted motors). In order to protect the electric motor from the upcoming vehicle body, two L-shaped beams are designed and attached on both sides on the upper surface of the long members. Two crash beams made out of aluminum is also mounted on each end of the chassis. The aluminum crash beam will absorb energy during frontal collisions. The added parts can be seen in Figure 30.

Figure 30: L-formed beams and crash beams attached to chassis.

The L-beams will be welded on the surface of the long members as well as mounted with a L shaped steel plate in the rear and on the sides in order for it to be fixed. To further increase the torsional stiffness, 2mm thick cross tubes has been attached between the L-formed beams and the long members, which can be seen in Figure 30.
11.6 Battery placement

The surface of the chassis is spacious enough for the largest battery volume discussed in section 4.3 to be placed without having the edges sticking out on the sides. Figure 31 shows the battery pack of the largest battery volume mounted on top of the surface. The Tesla model S chassis in Figure 18 has similar layout regarding the battery packaging.

![Figure 31: Largest battery volume mounted on the chassis surface.](image)

The rectangular box-shaped battery pack has the dimensions 1760x1193x60 mm which corresponds 126 liters of battery with the energy density of 400Wh/liters (see section 4.3). When the vehicle body is attached to the chassis, it will cover the battery and further protect it from stress concentrations etc.
12 Suspension system

The suspension system which will be mounted on the chassis will be a form of a double wishbone configuration, also called the A arm suspension system due to shape of the control arms linked to the knuckles. Figure 32 shows a double wishbone suspension system with a coil spring in the middle.

Figure 32: Double wishbone suspension system [41].

Since one of the requirements in Section 1.2 is comfortable traveling, the coil spring is replaced by an air compressed system. This system has a lot of advantages such as: adjustable height of the vehicle which makes it easier to get in and out of the vehicle and also the ride quality is increased due to reduced amount of vibrations during driving [42]. The selection of the type of suspension system configuration is already determined within the company, however, in this section the analytical study which will form the base for the suspension system design will be evaluated. The suspension system with the associated components will thereafter be designed by another subgroup at ALTEN.

12.1 Analytical study

The suspension arms will be exposed to different kind of loads throughout its operation period. The static load consisting of the total vehicle weight including passengers will always be divided by the four wheels which in turn will affect the arms with equal amount of force. Two major occasions of which the loads on the tires will increase drastically is through longitudinal and lateral load transfer. When the vehicle is moving in a longitudinal direction and must slow down due to an obstacle at the road or a traffic light, a reaction force to the vehicle motion will increase the load on the front wheels while decreasing the weight at the rear. This phenomenon is reversed when the vehicle is accelerating. The lateral load transfer occurs during cornering. E.g. if the vehicle is going through a left corner, the centrifugal force caused by the lateral acceleration will increase the load on the outer wheels while decreasing the load on the inner wheels relative to the curve center. These load transfers need to be taking into consideration while designing the upper and lower arms. This section will provide analytical methods of calculating the longitudinal and lateral load transfer. The parameter values itself will be provided from the other subgroup at ALTEN who will design the actual suspension system, hence it is not part of this project to estimate the parameters for the equations.
12.1 Analytical study

12.1.1 Longitudinal load transfer

Figure 33 shows a vehicle braking on a straight road with no lateral dynamics involved.

![Free body diagram](image)

Figure 33: Free body diagram.

The resulting force in longitudinal direction according to Figure 33 can be expressed according to the equation below

\[ F_b = F_{z1} + F_{z2} \]  \hspace{1cm}(19)

where \( F_b \) is the total braking force coming from the front and rear wheels. The vertical forces on the front and rear axle during braking is given by:

\[ F_{z1} = \frac{m \cdot g}{L} \cdot (b + h_g \cdot \frac{a_x}{g}) \]  \hspace{1cm}(20)

\[ F_{z2} = \frac{m \cdot g}{L} \cdot (a - h_g \cdot \frac{a_x}{g}) \]  \hspace{1cm}(21)

where \( a_x \) is the longitudinal acceleration value, which in this case corresponds to the maximum acceleration value in Figure 5b (0.76 m/s\(^2\)).

The values for the parameters used in Equations 20 and 21 can be seen in Table 5.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>( m )</td>
<td>3500 Kg</td>
</tr>
<tr>
<td>( h_g )</td>
<td>0.912 m</td>
</tr>
<tr>
<td>( a )</td>
<td>1.45 m</td>
</tr>
<tr>
<td>( b )</td>
<td>1.45 m</td>
</tr>
<tr>
<td>( L )</td>
<td>2.9 m</td>
</tr>
</tbody>
</table>

Table 5: Numerical values for longitudinal load transfer.

With the inserted parameters the load on the front axle \( f_{z1} \) is 18269N and the load on the rear axle \( f_{z2} \) is 16067N.
12.1 Analytical study

12.1.2 Lateral load transfer

During cornering the centrifugal force can be expressed as:

\[ F_c = m \cdot a_y = m \cdot \frac{v^2}{R} \]  \hspace{1cm} (22)

Where \( m \) is the total vehicle weight, \( a_y \) is the lateral acceleration, \( v \) is the velocity and \( R \) is the curve radius. Figure 34 visualizes a vehicle going through a left corner.

Figure 34: Back view of a vehicle going through a left corner [43].

One method of defining the difference in vertical loads between the left (\( f_{zl} \)) and the right (\( f_{zr} \)) side of the vehicle during cornering is with the help of the rollover index (\( R \)), which is defined as [43]:

\[ R = \frac{f_{zl} - f_{zr}}{f_{zl} + f_{zr}} \]  \hspace{1cm} (23)

For simplicity, the unsprung mass of the vehicle is neglected and if the roll motion of the sprung mass is entirely caused by the lateral acceleration, the rollover index can be determined as [43]:

\[ R = \frac{2h_g a_y \cos \phi + 2h_g g \sin \phi}{w \cdot g} \]  \hspace{1cm} (24)

where \( \phi \) is the roll angle of the vehicle, \( h_g \) is the CoG height, \( w \) is the distance between the tires. Rearranging equation (23) and (24), the ratio between the vertical loads between the left and right side of the vehicle can be expressed as [43]:

\[ \frac{f_{zl}}{f_{zr}} = \frac{1 + R}{1 - R} \]  \hspace{1cm} (25)

With this equation it is possible to determine the lateral load transfer during cornering. The values for the parameters shown in Equation (24) is listed in Table 6.

Table 6: Numerical values for lateral load transfer.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>( m )</td>
<td>3500 Kg</td>
</tr>
<tr>
<td>( h_g )</td>
<td>0.912 m</td>
</tr>
<tr>
<td>( w )</td>
<td>1.988 m</td>
</tr>
<tr>
<td>( \phi )</td>
<td>2.95 degree</td>
</tr>
<tr>
<td>( R )</td>
<td>200 m</td>
</tr>
</tbody>
</table>
With the inserted values the lateral load transfer for the right side of the vehicle \( f_{zz} \) is 14906N while for the left side \( f_{zl} \) it is 19464N. Since the force \( f_{zl} \) is higher than the upper limit force in the previous section, the design should be based on this load as a maximum load.

### 12.2 Air compressed suspension system

The complete air compressed suspension system can be seen in Figure 35.

![Suspension System](image)

(a) Side view  
(b) Top view

Figure 35: Suspension mount to chassis.

As it was mentioned previously, this model has been provided from another subgroup at ALTEN. The upcoming design of the suspension mounts will be based on the dimensions of this provided suspension system.

### 12.3 Suspension mount to chassis

Based on the provided CAD model of the suspension system, the suspension mount to the chassis were designed. The mount consists of two parts as it can be seen in figure 36.

![Suspension Mount](image)

(a) Front view  
(b) Side view

Figure 36: Suspension mount to chassis.

The upper part will connect the upper arm of the suspension while the lower part connects the lower arm. The design has been based on saving weight, therefore the structure is hollow and contains several holes in order to save weight. The material used for this structure is AHSS and the total weight of the structure is 37.5 kg.
13 Final design

With the provided suspension system, the complete chassis could be assembled together. Figure 37 shows the full vehicle assembly.

![Full Vehicle assembly - view from above.](image)

The total chassis assembly consists of 58 parts excluding the suspension system and the tires. The complete data for each component can be seen in Figure 38.

<table>
<thead>
<tr>
<th>Part</th>
<th>Amount (#)</th>
<th>Material</th>
<th>Density kg/m³</th>
<th>Weight [kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Long member</td>
<td>2</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>141.58</td>
</tr>
<tr>
<td>Side member</td>
<td>14</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>10.48</td>
</tr>
<tr>
<td>Cross member tubes</td>
<td>3</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>4.35</td>
</tr>
<tr>
<td>Cross member C-shape</td>
<td>4</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>8</td>
</tr>
<tr>
<td>Coverplate (Upper)</td>
<td>1</td>
<td>Aluminum 6061-T6</td>
<td>2700</td>
<td>20.12</td>
</tr>
<tr>
<td>Coverplate (under)</td>
<td>1</td>
<td>Aluminum 6061-T6</td>
<td>2700</td>
<td>16.76</td>
</tr>
<tr>
<td>L beams</td>
<td>4</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>59.76</td>
</tr>
<tr>
<td>Cross beams</td>
<td>4</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>9.6</td>
</tr>
<tr>
<td>Cross tubes</td>
<td>2</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>4.3</td>
</tr>
<tr>
<td>Suspension mount</td>
<td>2</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>75</td>
</tr>
<tr>
<td>Crash beam</td>
<td>2</td>
<td>Aluminum 6061-T6</td>
<td>2700</td>
<td>8.5</td>
</tr>
<tr>
<td>I-mounts</td>
<td>6</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>3.54</td>
</tr>
<tr>
<td>Square-mounts</td>
<td>8</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>3.84</td>
</tr>
<tr>
<td>Side mounts</td>
<td>4</td>
<td>AHSS (SS)</td>
<td>7700</td>
<td>3.08</td>
</tr>
<tr>
<td>Battery</td>
<td>1</td>
<td>Solid State battery</td>
<td>480 Wh/kg</td>
<td>10.5</td>
</tr>
<tr>
<td><strong>Total:</strong></td>
<td></td>
<td></td>
<td></td>
<td><strong>473.87</strong></td>
</tr>
</tbody>
</table>

![Component data for total vehicle assembly.](image)

As it can be seen in Figure 38, the total weight of the chassis is about 474 kg. The tires seen in the figure has the size (235/60R18) with a total diameter of 739mm. The property data for the suspension system and the tires have not been taken into consideration since the design is not part of this particular project. The total weight can be compared to the Tesla model 3 battery pack which weighs 480 kg with a battery density of 168 Wh/kg [44]. The complete chassis including the largest battery volume weighs 6kg less than just the
battery pack alone of the Tesla model 3. However, the battery pack of the Tesla model 3 also includes additional cooling systems, connections and safety shielding which are not taken into consideration for the battery pack in the airport shuttle.

The final dimensions of the vehicle can be seen in Table 7:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheelbase</td>
<td>2909 mm</td>
</tr>
<tr>
<td>Track width</td>
<td>2298 mm</td>
</tr>
<tr>
<td>Total length</td>
<td>4110 mm</td>
</tr>
<tr>
<td>Ground clearance</td>
<td>194 mm</td>
</tr>
</tbody>
</table>

Table 7: Final chassis dimensions.
14 Discussions

The purpose of this project was to design a full scale chassis concept for an electric autonomous airport shuttle based on modern and future requirements and regulations. The chassis should be spacious enough for 15 passengers including luggage. It should also be lightweight while still being able to withstand all the loading conditions that will occur during the vehicles operating period, keeping in mind a factor of safety. Flexibility of the chassis has been one of the main requirements from ALTEN, which means that the vehicle body and the battery should have the potential to be easily separated from the chassis. Since the restrictions to the transport sector regarding hazardous emissions during the vehicles total life cycle is becoming more severe, the chassis should be justifiable in terms of environmental impact. The chassis should also be cost-effective without compromising the quality of the design. There are a few autonomous shuttles in the market at present such as EZ10 and Continental [13] which are similar to the intended airport shuttle in terms of size and capacity. However, there is no specific data regarding the chassis construction or other forms of details available.

The power requirement estimated in Section 4.3, was calculated to 50.43 kWh. This value is 64% higher than the battery capacity used in the EZ10 air shuttle which is 30.72 kWh [13]. It is essential to know that the estimation in Section 4.3 was based on estimated parameters and the WLTC driving cycle. The EZ10 might also have different loading strategy where the strategy for the airport shuttle is to be loaded twice a day. With this in mind and the fact that the EZ10 has a GVW which is 370kg less than the mass used for the calculations, the power requirement for the airport shuttle seems appropriate.

The torsional stiffness for the base of the chassis does not reflect the torsional stiffness for the complete vehicle. However, it provides information on the behaviour of a central part of the chassis subjected to torsion. The FEA showed that the torsional stiffness of the cross members together with the long members were 8.7 kNm/deg. This could be compared to the one for a similar sized van which has a torsion magnitude of about 8500 Nm/deg [15]. These values are slightly similar only considering the base chassis for the airport shuttle. In order to get a reasonable torsional stiffness value for the total vehicle, the vehicle body must be mounted on top on the chassis since it will affect the overall twisting resistance.

The materials has been discussed and compared through environmental impact and LCA perspective (see Section 7). From Section 9.2 all the materials were compared in terms of their recyclability ratio (R). Even though AHSS is not the material with the highest recyclability according to Figure 14, it has the highest ratio (R) which makes it optimal for its use. Despite the fact that GFRP and CFRP has low recyclability, they also require advanced technical equipment to be processed and shaped. This factor will indirectly affect the cost of the materials. For this reason, AHSS is the material which provides the best results in terms of property, cost and recyclability.

The final design which can be seen in Figure 37 have slightly similar appearance as the Tesla model S chassis seen in Figure 18. The major difference comes within the battery packaging strategy. In the Tesla model S chassis, the battery packs has been designed to be part of the supporting structure. Since one of the main requirements from ALTEN was flexibility, the battery pack for the airport shuttle has been centrally mounted on the surface of the chassis which can be seen in Figure 37. Despite its own protective cover made out of aluminum laminated film, the vehicle body should be extra strengthened in the center preventing the chassis from bending vertically due to passenger load. The surfaces on the vehicle body that will cover the battery should also be coated with energy absorbing and fireproof material. With this setup, the risk for potential stress concentrations will be minimized. Another reason is also the fact that the airport shuttle will be designed for 15 passengers which is 3 times the capacity of the Tesla. It would be challenging to design the battery packs which could potentially replace the side member and/or the cross members strength properties.
The base of the chassis seen in Figure 28 is rigid enough for carrying the load of necessary equipment and the passengers (with luggage). The weak links can potentially be the tire and electric motor area seen in Figure 11.5. Even though the L beams are both welded and mounted on the long members, the cross tubes attached on the structure may not be enough when being exposed for lateral forces during cornering and torsional twist. Additional to the theoretical simulations, physical testing may be needed in order to ensure its strength.

The project in general is complex and encompasses many different engineering fields such as vehicle dynamics, mechanical engineering, solid mechanics and also electronics. Despite the wideness of the project, the work has been carried out on a higher conceptual level within the framework of a degree project. The biggest challenge during the course of the project has been its freedom which means that there has not been any existing physical model or data sheet of which the work has been based on.

15 Conclusions

From the literature study and influences from Equipmake’s new electric bus chassis and the Tesla model S platform, a skateboard chassis has been developed for an airport shuttle. The base of the chassis is made out of frames and it was concluded that tubular cross-members were optimal in terms of increased torsional stiffness. For the vertical loads, tubular truss side members proved to be more resistant than the C-shaped side members on the Equipmake chassis. AHSS was the most suitable material for the load bearing structure on the chassis considering recyclability, strength properties and cost-effectiveness, however the cover plates and the crash beams were made out of Aluminum alloy due to its lightweight and shock absorbing properties.

16 Future work

Since the project started from a clean sheet with no starting positions or guidelines for dimensions, weight, materials etc. the work has been carried out on a higher conceptual level. This indicates that the aforementioned parameters have been estimated based on similar sized electric shuttles within comparable areas of usage. When the vehicle body is designed and more detailed information is available regarding driving distances and geometric restrictions etc. specific driving cycles can be developed. Furthermore, the parameters in Equation 1 such as mass, frontal area, aerodynamic drag coefficient etc. can be updated. The total load consisting of the travelers and vehicle body forms the basis for the design of the supporting structure. Hence, the components in Section 11 can be further tuned and optimized with updated mass, in order to save materials and be more cost-effective. To obtain an optimized chassis, detailed pre-information is crucial. This can prevent the various components from being over-dimensional which in turn makes it more environmentally friendly and sustainable.

This thesis project provides a sustainable design model for the chassis of an electric airport shuttle with space for further optimization based on precise vehicle values. Since the thesis project is time-limited, the development of a full scale prototype would be needed in order to secure the strength and structure dynamics of the chassis.
17 References

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