UNIVERSITY OF NAIROBI

DEPARTMENT OF MECHANICAL AND MANUFACTURING ENGINEERING

PROJECT TITLE: HANDMADE ELECTRIC VEHICLE

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PROJECT SUPERVISOR: PROF A. NYANGAYA

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THIS PROJECT REPORT IS A PARTIAL FULFILMENT OF A DEGREE COURSE IN BACHELOR OF SCIENCE (MECHANICAL AND MANUFACTURING ENGINEERING).

UNIVERSITY OF NAIROBI

DECLARATION

We declare to the best of our knowledge that this Final Year Project report to be submitted as a partial fulfilment of the Bachelor of Science (Mechanical and Manufacturing Engineering) degree, to be our own original work and has not been presented in this or any other university for examination, academic or any other purpose.

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SUPERVISOR:

This design project has been submitted for examination with my approval as the university supervisor.

PROF. A. NYANGAYA

SIGN: ………………………………………….

DATE: …………………………………………

DEDICATION

We have to remember the smile on their faces when we first told them about our final year project. We have to remember our parents for the encouragement to work hard to achieve our dreams. Also our lecturers, for their efforts in assisting us academically and our friends for their moral support. We cannot but offer them our hard work, our hopes for success and our determination for excellence.

ACKNOWLEDGEMENT

We would like to thank everyone who assisted us with completing this project. The names are too numerous to list, but they know who they are.

We want to especially thank our project supervisor; Prof A. Nyangaya for his wisdom, friendship, support and assistance in guiding us throughout this project. He not only gave us the opportunity to design the homemade electric vehicle, but also provided the moral and academic support that has benefited us in our project. Besides just putting up with us, he was always able to give us insightful feedback on whatever the problem was.

We also thank Prof Mbuthia from the Electrical engineering department for his guidance on the choice of batteries and motor.

We would also like to thank the Department of Mechanical and Manufacturing Engineering and all our lecturers for their academic support that has made this project possible.

We would also like to appreciate the Go Kart Centre in Langatta for being cooperative in giving us the information we required during our project.

Finally we would like to thank our classmates, friends and our families for their endless love and support throughout the years. We would not have achieved the success that we have without them.

ABSTRACT

The objectives of this project was to design a hand-made electric vehicle using locally available subassemblies and also determine the cost affordability of the designed vehicle.

The frame was designed based on the formula student requirements and the SAE rules. The frame was designed in such a way that air friction was minimised and weight of the vehicle reduced. ISO carbon steel was chosen as the main frame material. Cold drawn square tubing of dimensions 25x25x2.5 was chosen. The square tubes were chosen because of their good weld ability.

The steering was designed to be light weight and with minimum number of mechanical parts in the mechanical steering system. The mechanical parts chosen include, tie rods, steering column and kingpin, all of which were made of aluminium to ensure the weight of the vehicle was kept low. The steering wheel was chosen according to the SAE rules. We chose 185mm diameter steering wheel with a comfortable grip area of 25x25mm.

Chain and sprocket drive was chosen as a means of power transmission from the motor to rear axle because of its compact ability, simplicity and its ability to give less load on the shaft and causes minimum vibrations compared to other trains like gears.

The centre of gravity of the vehicle designed according to Autodesk Inventor simulation gave a perfect roll over angle (64^0) that was well within the Formula Student requirement of an angle of 60^0. It was established that a track inclination of 17^0 was the maximum angle the vehicle could be started and accelerated from 0 kph up the track. The designed electric vehicle can move at a maximum speed of 60 km/hr.

From the cost analysis, the electric vehicle was found to be more affordable and cheaper by almost Ksh.420000 compared any small engine powered vehicle hence easily affordable. This project was successful in providing an innovative and low cost solution to the design problem. This design provides the lightest motor.

NOTATION

AC – Alternating Current

AISI –American Iron and Steel Institute

ANSI –American National Standards Institute

CVT – Continuous Variable Transmission

DC – Direct Current

EV – Electric Vehicle

F – Force

F1 – Formula 1

ISO – International Standard of Organization

K_1 *– Load Factor*

K_2 *– Lubrication Factor*

K_3 *– Rating Factor*

K_a *– Finish factor*

K_{fs} *–Factor for Shear stress*

Kph –Kilo metre per hour

K_s *– Service Factor*

Kshs. – Kenyan Shillings

K_t *– Factor for normal stress*

Kw – Kilo Watts

L – Length

LED – Light emitting diode

M – Bending moment

m – Metre

MPa – Mega Pascal

P- Power

R_r *–Roller Seating Radius*

S.F – Safety factor

SAE – Society of Automotive Engineers

S_{uts} –*Ultimate Tensile Strength*

SUV – Sports Utility Vehicle

S_y –*Yield Strength*

S_{ys} – *Shear strength*

T – Torque

TIG- Tungsten inert gas

USD –US Dollars

τ_{xy} – *Shear stress due to torque*

δ – *Deflection*

μ – *Coefficient of friction*

σ – *Compressive strength*

τ - *Shear stress*

ω – *Angular velocity*

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CHAPTER ONE

1.0. INTRODUCTION

1.1. BACKGROUND

Many vehicles in production today are powered by fossil fuels, however as the prices of this fuels continue to rise, their future availability not certain, and the costs involved in servicing fuel powered engines has also risen. The fluctuating availability of the commodity (petroleum) due to other factors like wars in the middle east has made it hard to predict the reliability of petroleum in the near or long term future.

From these experiences we realized that the world need a revolution or a complete overhaul in the transport system to complete electric systems. In that case therefore, there is need to design vehicles that are powered by electricity. Moreover, electric vehicle are more efficient compared to fuel powered vehicles. It is also cleaner and thus ensures less pollution and hence a clean environment.

As compared to fossil powered engines EV have got simplified parts and connections. A motor in itself is so simple as compared to the engine block of petrol and diesel powered vehicles which is made up of so many parts.

In the world right now there are electric vehicles available in the market like the Tesla. This brand of cars like many other EV from other companies are very expensive with the Tesla retailing at USD 100,000. This is a very large amount of money that can only be raised by a few people. There is a need for a cheaper EV in our markets.

The cheaper EV available in the markets have their shortcomings in that most of them run at low speeds compared to fossil powered and can only operate for a relatively short period of time before the batteries get depleted.

1.2. PROBLEM STATEMENT

Most car owners in Kenya would like to cut on the cost of buying, maintaining and the running of their vehicles. The owners can be owning cars for commercial and personal use and or for fun e.g. racing. It is so hard to have that when the engines are either petrol or diesel engines which are generally expensive to maintain and run.

Other car owners to be will opt for another vehicle other than EV because of the absence of the charging stations in Kenya which is necessary provided the low range of the current EV. Global warming that has seen the world temperatures to rise especially in the poles has been attributed to the fossil fuels being used in the petroleum engines especially vehicles due to their monstrous number in the world.

The goal of this project was to come up with a handmade EV that uses locally available subassemblies. We focused on improving the range by reducing the weight of the vehicle. We also made sure that we met the SAE and the Formula Student rules that governed us through the design. We tried as much as possible to cut the cost of EV by designing it using the locally available materials.

1.3. SIGNIFICANCE OF OUR PROJECT

For a successful project, an engineering solution for the following needs statement must be developed. "There is a need for a vehicle that is cheap, easy to maintain and operate. The vehicle should have less pollutants released into the environment either solid, liquids or gases during its idle and operation time. The vehicle should be safe to the user and to the people around it. It should also be easy to use and finally it should be both appealing to the user and satisfactory to their needs''. We therefore, used our knowledge in engineering to provide some of solutions to the above needs.

In our project, we want to provide a creative solution that assists a person to cut on the cost of buying maintaining and operating a car. With this project we shall benefit all classes of people from the lower to high class.

1.4. OBJECTIVES

The objectives of this study project are as indicated below:

i) To design a handmade EV using locally available subassemblies.

ii) To determine the cost affordability of the designed EV.

1.5. DESIGN REQUIREMENT

The design requirements were;

i) A vehicle capable of carrying one person

ii) Speed up to 60kph

iii) On road Vehicle which would move around the University of Nairobi.

iv) The cost of design and fabrication should be kept at a minimum.

v) The vehicle's operation should be easy and the design should ensure that all parts are easily accessible for repair and servicing.

CHAPTER TWO

2.0. LITERATURE REVIEW

2.1. HISTORY OF ELECTRIC VEHICLE

An electric vehicle uses an electric motor DC or AC powered by batteries for propulsion rather than being powered by a gasoline-powered motor.

Electric vehicle was first produced in the mid-19th century. They were the fastest vehicles around that time as they held the vehicular land speed until around 1900. Their high cost, low top speed and short range compared to later fuel powered vehicles, led to a worldwide decline in their use. At the beginning of the 21st Century, interest in electrical has increased due to growing concern over the problems associated with petroleum products.

In 1828, Ányos Jedlik, a Hungarian invented an early type of electric motor, he then created a small model car powered by his new motor.

Rechargeable batteries were made around 1859, when French physicist Gaston Planté invented one. The capacity of such batteries has since been improved with the improved research and technological approach.

Despite the success at the beginning of the 20th century, the electric car began to lose its position in the automobile market. A number of developments contributed to this situation. By the 1920s an improved road infrastructure required vehicles with a greater range than that offered by electric cars. Worldwide discoveries of large petroleum reserves led to the wide availability of affordable gasoline, making gas-powered cars cheaper to operate over long distances. Electric cars were limited to urban use by their slow speed and low range. They were outpaced.

After years outside the limelight, the energy crises of the 1970s and 1980s brought about renewed interest in the perceived independence electric cars had from the fluctuations of the hydrocarbon energy market.

The global economic recession in the late 2000’s led to increased calls for automakers to abandon fuel-inefficient SUVs, which were seen as a symbol of the excess that caused the recession, in favour of small cars, hybrid cars, and electric cars. California electric car maker Tesla Motors began development in 2004 on the Tesla Roadster, which was first delivered to customers in 2008. The Roadster was the first highway-capable all-electric vehicle in serial production available in the United States.

Our car is a homemade electric vehicle with a frame borrowing a lot of its design from the go-kart and the Formula e and Formula 1 designs.

2.2. CHASSIS MATERIALS

Most frames are made of either aluminium or steel. Titanium and carbon fibres have also been used in the past. However, all of these materials have their strengths and weaknesses.

2.2.1. ALUMINIUM

Aluminium alone can not be used in chassis building because if its low yield strength. Aluminium alloys with additions of materials like zinc are bettter materials for chassis building. They are lighter as compared to the steel frames. Alloying does not improve the stiffness of the frame. Steel is still 3 times more stiff at this stage. Aluminium density is 35% that of steel. Common aluminium chassis are Tungsten Inert Gas (TIG) welded. They are sometimes welded together using lugs. It has most of the desirable qualities of aluminium, that is, good mechanical properties and high corrosion resistance due to the formation of the oxide layer. It is more expensive compared to steel.

2.2.2. STEEL

Steel is easy to get. People who work with steel so well are so easy to find. It is cheap and is the reason as to why most cars chassis are made of steel. Chassis are supposed to be stiff strucures to ensure that other components can work as they are designed to. Steel is stiff and is why it is good for this kind of job. Steel is widely used because of its weldability, ease fabrication, mild hardenability and high fatigueability. However, steel is so heavy because of it high density, it also corrodes when exposed to adverse environmental conditions. Overally the benefits of steel outweighs the disadvantages of using it.

2.2.3. TITANIUM

Regarded as the ultimate material, it is usually used in the space tech. its density is approximately 50% that of steel.More and more of the high-end vehicles are made of titanium,since it is a light weight, strong and non-ferrous metal. Chassis made of titanium are TIG welded. Titanium is the most exotic of the metals used in production of chassis and the most expensive. It requires special tooling and skills to be machined and welded. It has very good mechanical properties and high corrosion resistance. It is resilient to wear and abrasion. Titanium is used because of its availabilty, appearance, very good strength and light weight. A draw back in titanium, besides cost, is that titanium once worn or if flawed may break rapidily. This is because it has a tendency toward brittle fractures.

2.2.4. ADVANCED COMPOSITES (FIBRE GLASS AND CARBON FIBRES)

Fibre glass is created when raw plastics is reinforced using strands of glass. It can be moulded to any shape. A body shell can be created with no seams that can create weak points. The areas which carry load are made thicker while the less stressed areas are made thinner. It is used in sport cars and often in conjunction with a separate subframe. Carbon fibres are made by changing the molecular structure of Rayon fibres byextream stretching and heating. Carbon fibre is very stiff meaning its modulus of elasticity is high and also it is very strong implying that it has a high tensile strength. It has a very low density. Carbon fibres can provide high strength and stiffnes with very minimal weight but with very high cost. This is the reason why they are used in the tub chassis of the F1 cars.

2.3. EV DESIGN

2.3.4. CHASSIS

It is a frame on which the body of a car is mounted. Typical chassis are made of rigid frame. It is usually designed to support the body and all the other components of the car.

The following are the requirements of the chassis;

- Rigid
- Must contain a roll hoop for both the front and rear of vehicle
- Contain a front impact structure

The most important aspect of the chassis is the torsional rigidity of the car for cornering stability. The suspension must attach to portions of the chassis that will allow the transfer of forces to be absorbed by the stiffness in the chassis.
Our frame is made of 25x25x2.5 square carbon steel tubes.

GENERAL CHASSIS RULES

The car must be equipped with a fully operational suspension system with shock absorbers, front and rear.

GROUND CLEARANCE

Ground clearance is defined as the distance at which the vehicles body other than the wheels approaches the ground. The clearance must be sufficient to prevent any portion of the car, other than the tires, from touching the ground. Excessive ground contact of any portion of the car other than the tires wears that car out much and constricts the car to a level road only. We chose a ground clearance of 152.4mm for the chassis although the suspension protrudes more from under the frame.

2.3.4. WHEELS

The wheels of a car convert rotary motion of the axle to linear motion and the tyre (inflated) act as secondary suspension to the car. Wheels come in different sizes and patterns. The chosen wheel should accommodate the tyre. Modern tyres are made of layers; these layers define the tyres strength. Moreover the clearance of the car relies mostly on the diameter of the wheel and thus choosing the right diameter goes back to the desired clearance. Most wheel mounting systems of small cars like the Formula e, Go-kart and small electric vehicles uses a single retaining nut with or without devices to retain the nut and the wheel in the event that the nut loosens.

We have this system in our car with a second nut added to act as a jam nut. The wheel lug bolt is bolted onto the hub which carries the wheel. Carbon steel nuts are used but aluminium wheel nuts may be used

2.3.4. TYRES

Vehicles may have two types of tires as follows.

- Slicks
- Threaded

Slicks or dry Tires are used on racing cars mainly. They are meant to enhance the surface are the vehicle touches the ground hence increasing the frictional surface which is important for propulsion. These tyres are not suitable for a ride in an all-weather road.

Threaded / rain Tires they are used in cars, trucks and other all-weather vehicles. They are rough on the surface and the threads are cut by the manufacturing company according to the safety standards specified by the standardization companies.

In our car we are using the threaded tyres because it is designed to be used in non-even surface and the fact that it cannot be used for racing.

2.3.4. STEERING

Small vehicles that run on smaller motors are usually designed to use mechanical steering system. The system is made so to cut on the undesired weight.
The steering system must have positive steering stops that prevent the steering linkages from locking up. The stops may be placed on the uprights or on the rack and must prevent the tires from contacting suspension, body, or frame members during the track events.
Allowable steering system free play is limited to seven degrees (7°) total measured at the steering wheel.
The steering wheel must be attached to the column with a quick disconnect. The steering wheel must have a continuous perimeter that is near circular or near oval, i.e. the outer perimeter profile can have some straight sections, but no concave sections.
Our vehicle is steered using a circular steering wheel with a continuous perimeter. The wheel is attached to the aluminium steering column through a steering hub. The connection is compact and so no parts of the steering wobbles and or free play in any direction.

2.3.4. ROLLOVER STABILITY

The track and centre of gravity of the car must combine to provide adequate rollover stability. The vehicle must not roll when tilted at an angle of forty degrees (40°) to the horizontal in either direction.
Our rollover stability is evaluated 63^0. This was done by measuring the angle between the perpendicular line to the centre of gravity and the line connecting the bottom of the wheel axis and the centre of gravity.

2.3.4. BRAKE SYSTEM

Brakes are devices whose function it is to slow and stop a car. They are mandatory for the safe operation of vehicles. When a car is in motion, it has kinetic energy or energy derived from this motion. In order for the car to slow down, this energy must be decreased. This is accomplished by transforming it into another form. In the case of brakes, this form is heat. In short, brakes transform the kinetic energy of the car into heat energy, thus slowing its speed and, if enough energy is transferred, bringing it to a stop.

There are different types of brakes which include

- Disc brakes
- Drum brakes
- Rim brakes

Whichever brake is chosen there are rules that governs their performance;

Brake system must act on all four wheels and is operated by a single control.

The brake pedal must be designed to withstand a force of 2000 N without any failure of the brake system or pedal. The brake pedal must be fabricated from steel or aluminium.

Our brake system which is mainly mechanical is made up of modified components from motorbike. The system comprised of a hub, a rotor, a cable, a calliper and a brake pedal.

2.3.4. TRANSMISSION AND DRIVE

There are different types of transmission that can be used. Clutches, gears, belts, chains and sprockets can all be used to transmit the power from the motor to the axle or drive shaft. Any transmission and drive train may be used. We chose the chain and sprocket drive train.

2.3.4. DRIVE TRAIN SHIELDS AND GUARDS

Exposed high-speed final drivetrain equipment such as Continuously Variable Transmissions (CVTs), sprockets, gears, pulleys, torque converters, clutches, belt drives, clutch drives and electric motors, must be fitted with scatter shields in case of failure. The drive train shield must cover the chain or belt from the drive sprocket to the driven sprocket or chain wheel or belt or pulley.

Chain drive Scatter shields must be made of at least 2.66 mm steel (no alternatives are allowed), and have a minimum width equal to three times the width of the chain. The guard must be centred on the centre line of the chain and remain aligned with the chain under all conditions. Our chain drive has a simple guard.

2.3.4. FASTENERS

FASTENER GRADE REQUIREMENTS

All threaded fasteners utilized in the driver's cell structure, and the steering, braking, driver's harness and suspension systems must meet or exceed, SAE Grade 5, Metric Grade 8.8 and or AN/MS specifications.

SECURING FASTENERS

All critical bolt, nuts, and other fasteners on the steering, braking, driver's harness, and suspension must be secured from unintentional loosening by the use of positive locking mechanisms. Positive locking mechanisms include;

- Correctly installed safety wiring
- Cotter pins
- Nylon lock nuts
- Prevailing torque lock nuts

Our fasteners are ISO and the security fasteners got nylon lock nuts as positive locking mechanisms.

2.3.4. EQUIPMENT REQUIRED

DRIVER'S EQUIPMENT

The equipment specified below must be worn by the driver anytime he or she is in the cockpit with the engine running or with the tractive system active for electric vehicles.

HELMET

A well-fitting, closed face

EYE PROTECTION

Impact resistant helmet face shield, made from approved impact resistant materials.

We recommend that the car be driven with protective glasses and a helmet on.

2.3.4. ELECTRICAL SYSTEM

BATTERIES

All batteries on-board that are tasked with power supplies to the vehicle, must be attached securely to the frame. Any wet-cell battery located in the driver compartment must be enclosed in a nonconductive container.

The hot terminal must be insulated.

MOTORS

Any type of electrical motor can be used in a car. The number of motors is not limited. Motors must be contained within a structural casing where the thickness is at least 3.0 mm. The casing must use an Aluminium Alloy of at least 6061-T6 grade or better if a casing thickness of 3.0mm is used. If lower grade alloys are used then the material must be thicker to provide an equivalent strength.

We have only one motor and four batteries.

INSULATION

All live wires and contacts need to be isolated by non-conductive material or covers to be protected from being touched.

CHARGERS

All connections of the charger(s) must be isolated and covered. No open connections are allowed.

All chargers must either be accredited to a recognized standard. The charger must incorporate an interlock such that the connectors only become live if is correctly connected.

When charging it must be possible to turn off the charger in the event that a fault is detected.

The charger must include a push type emergency stop button.

CHAPTER THREE

3.0. DESIGN OF COMPONENTS

3.1. STEERING SYSTEM DESIGHN

Among the considerations made during the design of the steering system were

- Steering must be lightweight and have few mechanical parts
- Driver's hand must not leave steering wheel while accelerating and breaking
- Driver must be able to maintain control while killing and restarting the engine
- Steering must allow wheels to be turned at a 10.5 degree angle.
- In order to properly negotiate a turn, the inside and outside wheels must be able to be turned at different angular degrees simultaneously.
- Principals from the Ackerman style of steering will be applied to allow the vehicle to turn properly.
- The camber must be adjustable.
- The steering system must have minimal wind resistance.

3.1.4. ACKERMAN DESIGN

The purpose of using an Ackerman Steering system is to reduce the amount of rolling friction on the front tires. This is accomplished by causing the two front tires to turn at different degrees so that they can follow two different circles with two different radii. This is desired because when going around a turn the tire closest to the inside of the turn is going to be travelling a shorter distance than the tire on the outside of the turn so they have to turn at different angles.

3.2.2. THE STEERING COLUMN

We evaluated existing column lengths for small sized vehicles and compared them to existing column lengths in the market.

We choose a column length of 106cm from its fixing position at the bottom to the point where the bottom steering wheel hub is fixed.

We choose a hollow member to reduce weight and improve the strength.

The material we choose was aluminium which makes sure the column is not susceptible to rusting. The steering column is to be held in place by a tube of 20mm inside diameter which is fixed to the frame.

3.2.3. STEERING WHEEL

Choice of a steering wheel depends on the available floor space and the ergonomic standards required. Suitable steering should be of suitable diameter and preferably circular or oval.

The most suitable steering wheel is one that does not take much of the space. We choose a 185mm diameter steering wheel with a grip area of 25mm by 25mm.It is readily available in automobile parts shops.

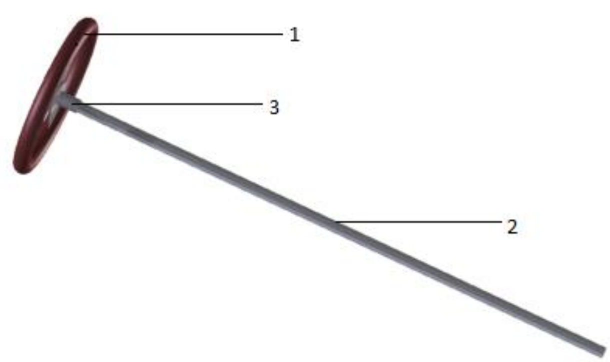


Figure 1. The steering (author)

1→Steering wheel

2→Steering hub

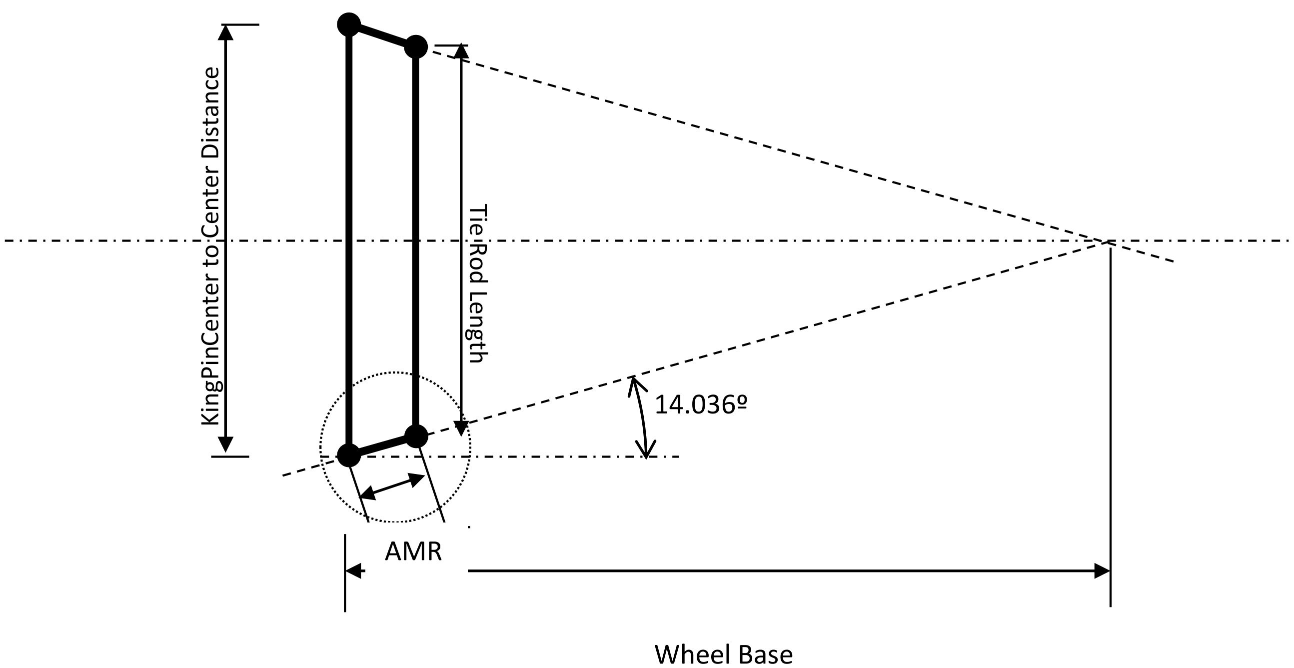
3→Steering column

3.1.4. TIE RODS

They are attached to the Ackerman arm radius and the steering column so that they provide the base for control of the wheels. Its dimension (length) calculation is based on the Ackerman angle.

3.2.5. ACKERMAN ANGLES

The Ackerman angle is as shown below together with its calculation.



$$\tan B = \frac{kingpin\ centre\ to\ centre\ distance}{2 \times Wheel\ base} \ldots\ldots (3.2.5.1)$$

$$B = \tan^{-1} \frac{kingpin\ centre\ to\ centre\ distance}{2 \times Wheel\ base} \ldots\ldots (3.2.5.2)$$

$B = \tan^{-1} \frac{0.9}{2 \times 1.6}$

$B = 15.709^0$

Ackerman angle is 15.709 and hence we can use it to find the tie rod length.

Tie rod length is calculated as shown;

$$\sin 15.709 = \frac{y}{6\ inches} \ldots\ldots (3.2.5.3)$$

Hence y= 4.1262cm

So the tie rod is 4.1262cm shorter on the top and 4.1265cm at the bottom.

Tie rod length = 90cm – (2×4.1262)

= 81.7476cm

=32.18 inches

Hence one tie rod is approximately 16 inches.

Weight 2pounds (1kg)

Length 16 Inches (40.64cm)

Diameter 5/8 inches (15.875cm)

Tubular type tie rods were chosen since they are strongest

This tie rod kit comes with;

- One tubular rod called the track rod
- 2 rod ends
- 1 right hand thread
- 1 left hand thread
- 2 jam nuts (right hand and left hand)

The tubular rod carries rod ends at its ends. The rod ends are connected to the tubular rod using the jam nuts. The jam nut used is an ISO 4032 M8.

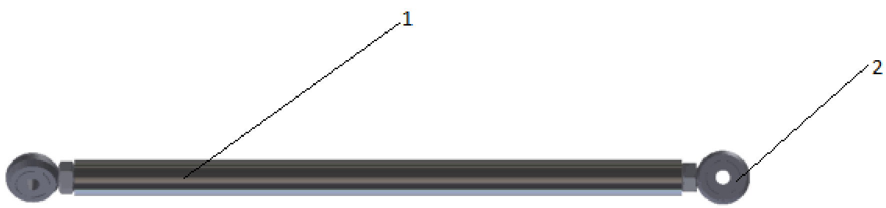


Figure 2. Tack rod (Author)

1→cylindrical tube

2→Rod end

3.2.6. SPINDLE KIT

It has a shaft of diameter 20 cm and length 98mm this shaft carries another shaft on its side which is 13mm diameter and the length is 34.42mm.The smaller shaft of 13mm carries the front wheel hub. The hub is connected to the shaft through a roller bearing (ISO 15 – 838 - 20 X 32) and fixed to the same shaft by a helix nut. The wheel hub carries a wheel (Tuk-tuk wheel) of outside diameter 304. 8mm. The wheel hub and the wheel are assembled using three bolts (ISO 4017 – M8 X 35) and three helix nuts (ISO 4032 – M8)

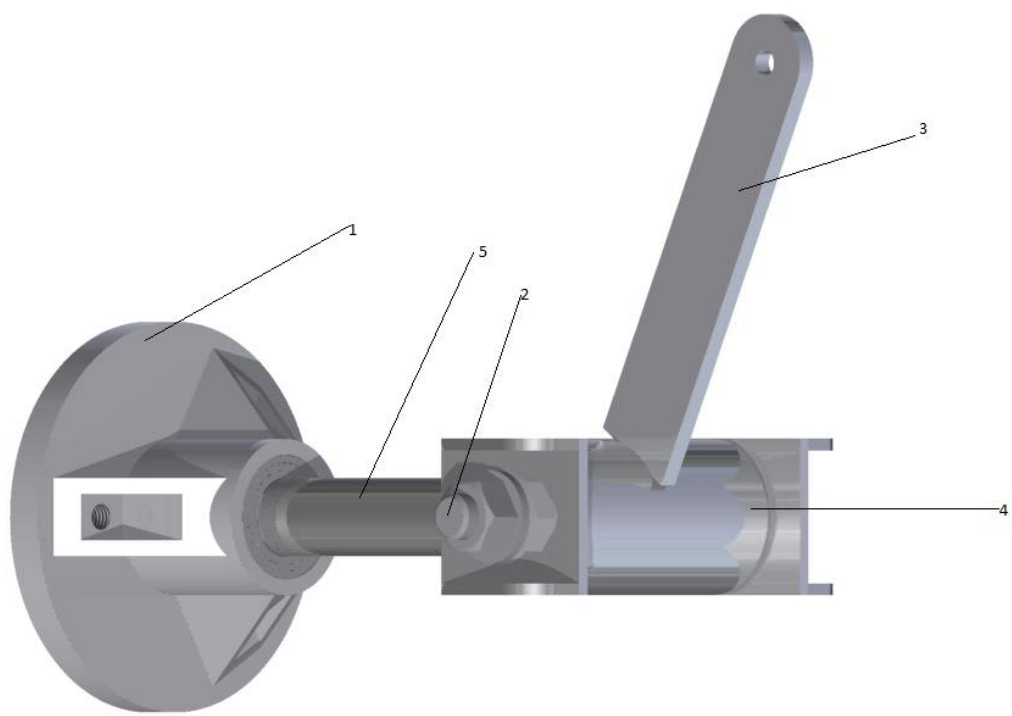


Figure 3 Spindle Kit (Author)

1→Wheel hub

2→Kingpin

3→Steering arm

4→Kingpin carrier

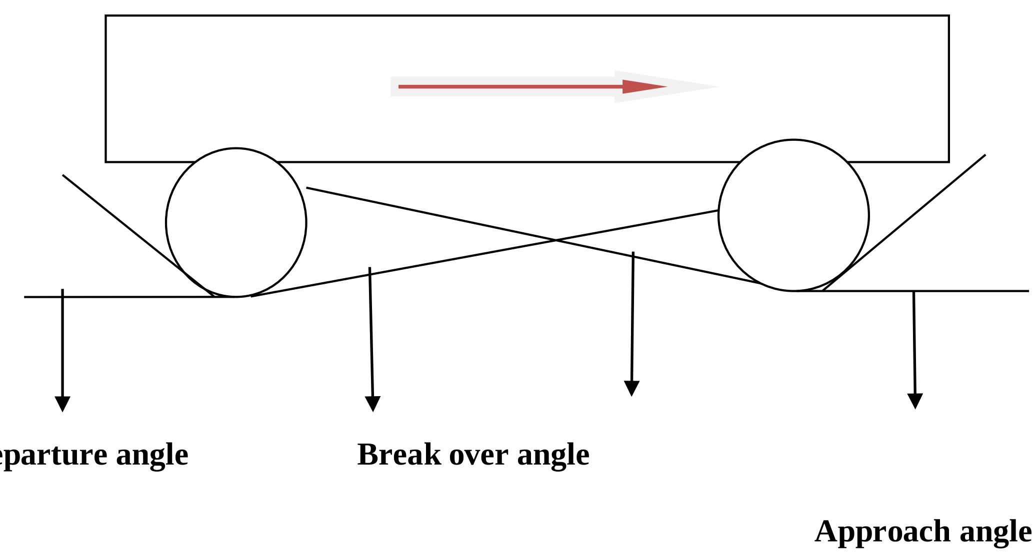
5→Stub Axle

6.2. WHEELS

The design consideration made in the choice of the wheels for the vehicle was the desired clearance of the vehicle from the ground. Too high a clearance causes the vehicle to roll over since it raises the centre of gravity of the vehicle. The maximum allowable clearance for small road vehicles is 165mm.

The factors affecting clearance are;

- Approach angle - Maximum angle a vehicle can approach an obstacle without any part of the wheel approaching the obstacle.
- Departure angle - Maximum angle a vehicle can leave an obstacle.
- Break over angle - maximum angle a vehicle can ride over an obstacle without striking the obstacle between its axles.
- Roll over angle - Angle at which a vehicle will roll when traversing a slope. The allowable angle is 40^0



Based on the requirements, Tuk-tuk wheels were chosen with a diameter of 12 inches and a width of 4.5 inches. It is a wheel of nomenclature (4, 4.5)4PR. The 4PR (Ply Rating) is what is available in the Kenyan market. The wheels can be bought at Paggio outlet along Jogoo Road.

This diameter gave us a clearance of 15.24cm which was within the required clearance.

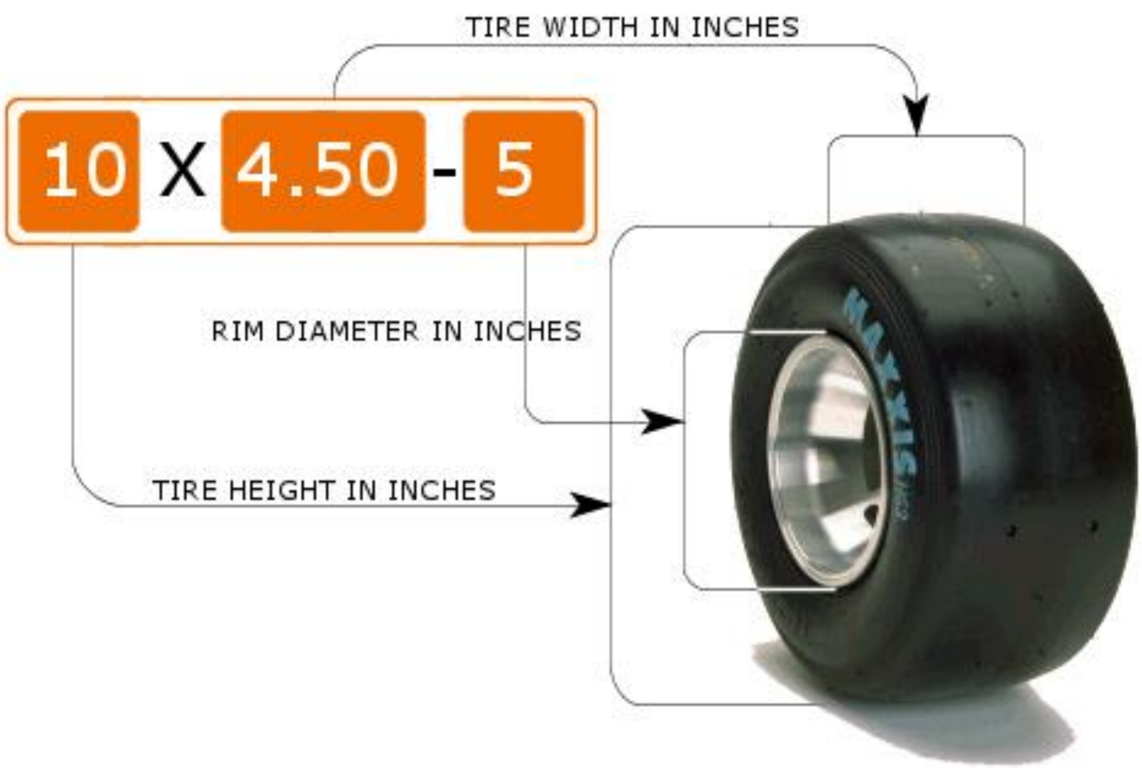


Figure 4 wheel (www.safercar.gov/Vehicle+Shoppers/Tires+Rating/Passenger+Vehicles)

6.2. POWER TRANSMISSON

The common power transmission drives are gears, belts, clutches, cables and chain and sprockets. We chose the chain and sprocket. We choose chain drive as the means of power transmission from the motor due the following reasons

- No slip takes place hence perfect velocity ratio is obtained
- Chains are made of metal hence occupy less space in width than the belt or rope drives
- May be used for both long and short distances
- High transmission efficiency of 98%
- Gives less load on the shaft
- Transmits more power than belts
- High speed ratio of 8 to 10
- Can be operated under adverse temperature conditions
- Does not cause vibrations as compared to gear drive
- Their ease of application in large torque applications.

3.4.2. CHAIN AND SPROCKET CALCULATIONS

Velocity ratio of chain drive

$V.R = N_1/N_2 = T_2/T_1$ ……………………………………………………………………………………….... (3.4.2.1)

N_1 = Speed of rotation of small sprocket

N_2 = Speed of rotation of larger sprocket

T_1 = Teeth on small sprocket

T_2 = Teeth on large sprocket

Speed of motor = 2670 RPM

Speed of axle = 60 kph = speed of the wheel

$V = \omega r$ ……... (3.4.2.2)

$R = 6$ inches

$= 6 \times 2.54cm = 15.24cm$

$= 0.154m$

$V = 16.67m/s$

$16.67 = \omega \times 0.1524$

$\omega = 109.361 rad/s$

1 Revolution = 2πRadians

? = 109 Radians

=17.41Rad/s

=1044.32Rpm

Velocity Ratio = 2670/1044.32 = **2.6**

Minimum number of teeth on small sprocket

The velocity ratio is 2.6~3

From the table 1 from the appendix, number of teeth on small sprocket, $T_1 = 25$

Number of teeth on larger sprocket.

$V.R = T_2/T_1$

$T_2 = V.R \times T_1$ ……….... (3.4.2.3)

$= 2.6 \times 25$

$= 65$ Teeth

Design power

Design power = Rated power × Service factor

$K_S = K_1 \times K_2 \times K_3$ …………………………………………………………………………………………………….... (3.4.2.4)

Where;

K_1 = Load factor

K_1 = 1.25 for variable speed with mild shock

K_2 = Lubrication factor

K_2 = 1 for drop lubrication

K_3 = Rating factor = 1 (8 hours of servicing)

K_S = 1.25

Design power = 1.25 × Rated power

= 1.25 × 2000 watts

= **2**.5 Kw

Type of chain

From table 2 in the appendix, type of chain chosen according to the Design power of 2.5 Kilowatts is *6b double strand chain*

The pitch

From the table 3 in the appendix;

Pitch = 9.525mm

Roller diameter = 6.35mm

Breaking load = 16.9KN

Pitch circle diameter

$D_1 = P \text{ Cosec } (180/T_1)$ ……………………………………………………………………………… (3.4.2.5)

$= 9.525 \text{ Cosec } (180/25)$

$= 75.99\text{mm}$

Pitch circle diameter of larger sprocket

$D_2 = P \text{ Cosec } (180/T_2)$ ……………………………………………………………………………… (3.4.2.6)

$= 9.525 \text{ Cosec } (180/65)$

$= 197\text{mm}$

Pitch line velocity of smaller sprocket

$V_1 = (\pi d_1 N_1)/60 = (\pi \times 0.07599 \times 2670)/60$

$= 10.62\text{m/s}$

Load on the chain

W= Rated power/ pitch line velocity

= 2/10.62

= 188.3 N

Factor of safety

= Breaking load/ Load on the chain

= 16.9/0.188

= 89.894

This is a comfortable factor of safety since the current motor bikes like Focin 150 uses a sprocket with a safety factor of 85.

Maximum centre distance

Maximum centre distance is usually 30- 50 times the pitch

$= 30P$

$= 30\times 9.525$

$= 285.75mm$

To accommodate for initial sagging, centre distance value is reduced by a value 2 to 5 mm

$X = 285.75 – 4 = 281.75mm$

$= 282mm$

Number of chain links

$K = (T_1 + T_2)/2 + 2X/P + [(T_2 - T_1)/2\pi]^{2P}/X$ ………………………………………………………….... (3.4.2.7)

$= 45+59.21+ 1.3698$

$= 105.58$

Length of chain

$L= KP$…….... (3.4.2.8)

$= 105.58 \times 9.525$

$= 1005mm$

Tooth frank radius

$= 0.008d_1(T^2 +180)$ ………………………………………………………………………………….... (3.4.2.9)

Where d_1 = diameter of chain roller

$= 0.008\times 6.35 \times 10^{-3}(625+180)$

$= 6.08\times10^{-4} (805)$

$= 40.89mm$

Minimum tooth frank radius (r_f)

$r_f = 0.12d_1(T+ 2)$ .... (3.4.2.10)

$= 0.12 \times 6.35\times 10^{-3}(27)$

$= 20.5mm$

Roller seating radius (r_r)

Maximum = $0.505d_1+ 0.069\sqrt{d1}$ .... (3.4.2.11)

$= 8.7mm$

Minimum = $0.505d_1$ .... (3.4.2.12)

$=3.2mm$

Roller seating angle

Maximum = $140^0 -90^0/T$ .... (3.4.2.13)

$= 140^0- 90^0/25$

$= 136.4^0$

Minimum = $120^0 – 90^0/25$ .... (3.4.2.14)

$= 116.4^0$

Teeth height above the pitch polygon

Maximum = $0.625P – 0.5D_1 + 0.8(P/ T)$ .... (3.4.2.15)

$= (0.625\times 9.525\times 10^{-3}) – (0.5\times 6.35\times 10^{-3}) + (0.8\times 9.525\times 10^{-3}/25)$

$= 3.1mm$

Minimum = $0.5(P- D_1)$ .... (3.4.2.16)

$= 0.5\ [\ (9.525\times 10^{-3}) - (6.35\times 10^{-3})]$

$= 1.64mm$

Top diameter (d_a)

Maximum $= D + 1.25P - d_1$ …………………………………………………………………………………………...... (3.4.2.17)

$= 75.99 + (1.25\times 9.525) - 6.35$

$= 81.54mm$

Minimum $= D + P\ (1- 1.6/T) - d_1$

$= 78.55mm$

Root diameter (d_r)

$d_r = D- 2r_1$ …………………………………………………………………………………………………..... (3.4.2.18)

$= 75.99- 6.35$

$= 69.64mm$

Tooth width (bf_1)

$=0.93\ b_1$

$= 0.93 \times 5.72$

$= 5.32mm$

Tooth side radius

$R_X = P$

$= 9.525mm$

Tooth side relief (b_a)

Usually a value between 0.1P and 0.15P

$= 0.9525$ and $1.429mm$

Widths over teeth

= ((number of strands) - 1) $P_1 + b_1$ ……………………………………………………………………………..... (3.4.2.20)

= 9.525 + 5.32

= 14.845mm

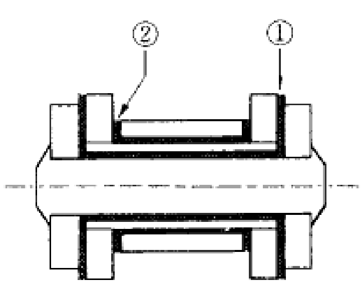
Hence the required sprocket is *3/8 inches pitch sprocket* with *25 teeth* and outside diameter of *0.7599cm* and *double stranded chain.*

From the calculations of the velocity ratio, the minimum number of teeth was obtained as 25 and the number of teeth on the larger sprocket as 65.From the calculation of design power the type of chain was chosen as a 6B double strand chain with a pitch of 9.525mm and roller diameter 6.35mm.Its breaking load is 16.9 KN. The pitch circle diameter of the smaller sprocket 75.99mm and that of the larger sprocket was 197mm. The factor of safety was obtained as 89.894 which is a comfortable factor of safety since the current motorbikes like Focin 150 uses a sprocket with a safety factor of 85.

The centre distance between the axis of rotation of the smaller and the larger sprocket is 282mm. The length of the chain is 1005.6mm and with 106 chain links.

The smaller sprocket is attached to the motor and the larger one to the rear axle through an assembly of hub.

Lubrication is to be done by oiling between the pin and bushing (1).This will prevent wear of pin and bushing, main causes of roller chains wear.



Oiling between the bush and the roller (2) will help prevent wear and reduce noise. The recommended lubrication oil is Tellus oil SAE 20 readily available from Shell Company.

3.4.2. THE ATTACHMENT OF THE LARGER SPROCKET TO THE REAR AXLE

The larger sprocket is bolted to the hub using three bolts (ISO 4017 – M8 X35) and hexagonal nuts (ISO 4032 – M8). The 30mm internal diameter hub is fixed on the rear axle and anchored rigidly through a key and keyway. We used the keyway because it allows for full power transmission as well as provides a rigid frame.

3.5. THE SUSPENSION SYSTEM

3.5.1. REAR SUSPENSION

The suspension system is made of a

- Damper and a spring
- Double wishbone

Damper and a spring

This is a component of Bajaj 125/150 motorbikes. Its length is 304.8mm from the lower axis to the upper axis for anchorage. We chose this model because of its simplicity in design, availability and the mass that we require to support. It ends are made of cylindrical tubes of inside diameter 6.647mm and the outer of 25mm. the length of the cylinder comes in different sizes and the choice of a 25mm was a good decision as it provides a rigid connection to the frame. It is connected to the frame using bolt and nut to provide a semi rigid connection for ease of articulation. From the lower end it is fixed to the upper arm of the double wishbone through bolt and nut connection for ease of articulation. The angle of alignment to the horizontal is 68^0 this angle is chosen to ensure that the forces acting on the damper are mainly axial.

The double wish bone is made of a middle supporting frame and two arms. The lower arm is a component made of a cylindrical tube of outside diameter of 15mm and inside being 10mm. the length from frame attachment to the point at which it is attached to the supporting frame is 267mm and is attached to the lower end of the frame through a cylindrical tube of outside diameter of 15mm and 10mm inside diameter. Its length is 125mm and is fixed to the frame via a bolt of length 150mm on to the carriers. The upper arm is a component with the same specifications as the lower but with shorter length of 129.5mm from the point at which it is attached to the main frame to that point of the supporting frame. The supporting frame is made up of four components;

- Frame
- Bearing
- Bearing stopper
- Two arm carriers

The rear axle passes through the centre of the supporting frame and through a bearing. The loading position of the supporting frame is perpendicular to the axle to avoid the angular shearing effect on the axle.

The wheel itself acts as a damper and shields against shock due to uneven load surfaces.

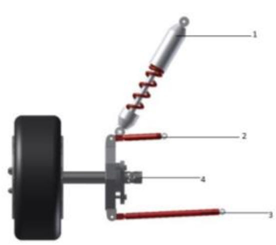


Figure 5 Rear wheel and Suspension (Author)

1→Damper

2→Upper arm

3→Rear shaft

4→Lower arm

3.5.2. FRONT SUSPENSION SYSTEM

It is made up of

- Wheel
- Front wheel hub
- Bearing
- Stub axle
- Double wish bone

The stub axle is connected to the wheel through a roller bearing. The cylindrical hub carries the upper and the lower arm. The upper arm has a length of 129.5mm from the axis of attachment to the cylindrical hub to that of the frame. It carries a carrier for the shock on which the shock is attached on its lower end. The lower arm has a length of 267mm from the axis of attachment to the cylindrical hub to the attachment to the frame.

We choose a cylindrical hub for the front suspension since it is easier to machine. It involves parting two solid cylinders for the left and right suspension of 77mm and drilling through holes using the diameter of the kingpin.

The shock that is attached to the upper arm and to the frame is 304.8mm from its axis of attachment to the upper arm and the frame.

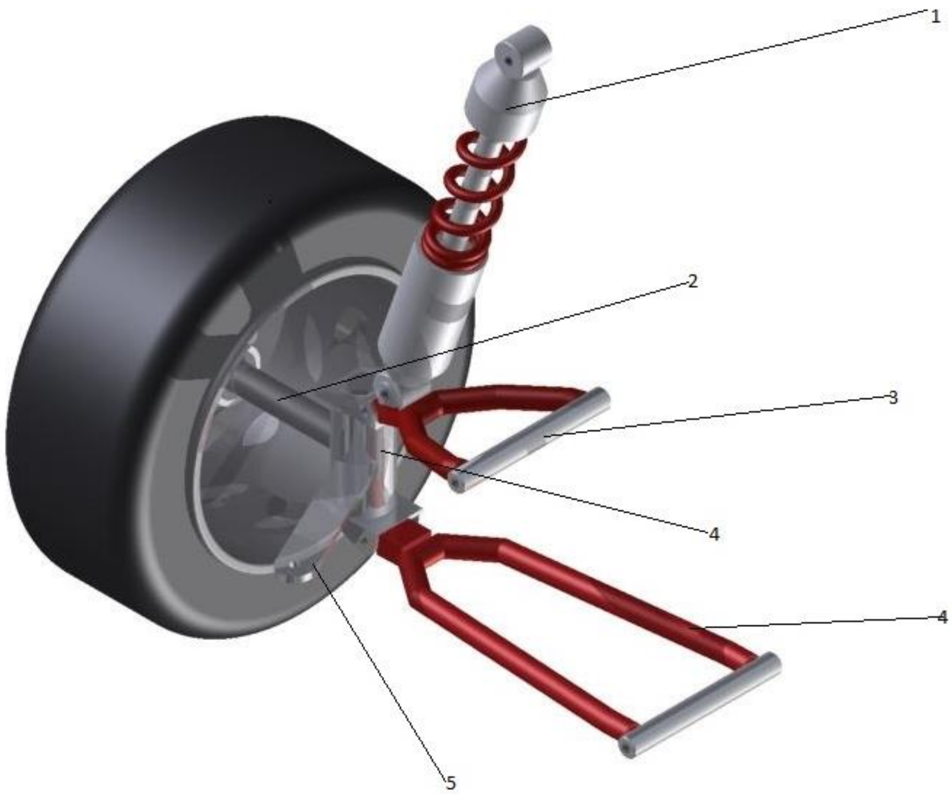


Figure 6 Front Wheel and Suspension (Author)

1→Damper

2→Stub shaft

3→Upper arm

4→Kingpin carrier

5→Lower arm

6→Steering arm

3.6. BRAKES

3.6.1. BRAKES FORCES

The braking requirements were calculated in order to establish that the brakes ordered would stop the car based on the maximum speed and a desired 5 seconds time to stoppage from the time brakes are applied. This is done by first calculating the force required to slow the vehicle down based on a deceleration.

This is the force that has to be applied as a reactionary force to the tire and the driving surface. The required deceleration can be calculated

Using the trajectory equation given below we can calculate the deceleration of the car. We know the distance to stop and we know the speed as well as the standard time for vehicle to stop from the start of depression on the brakes.

$$V^2 = U^2 + 2as \qquad \text{(3.6.1.1)}$$

$$0^2 = 16.67^2 + 2 \times a \times 52.46$$

$$Deceleration, a, = \frac{-(16.67^2)}{2 \times 52.46}$$

$$Deceleration, a, = \frac{-277.89}{104.92}$$

$$Deceleration, a, = -2.65 \ {}^{m}/_{s^2}$$

The calculated value is used to calculate the braking force for our car. The equation is given as

$$Braking\ force = Mass \times gravitational\ acceleration \times deceleration$$

$$Braking\ force = \{250\ \times 9.81 \times (-2.65)\}N$$

$$Braking\ force = -6500N$$

The negative sign shows that the force act in the opposite direction to that of the vehicle.

Brake Torque

Brake Torque (N-M) is calculated by using the braking force, radius of the tire, R, and the ratio of the rotational speed of the wheel, ***R***, to brake.

$$Brake\ Torque = \frac{Braking\ force \times Tire\ Radius}{Tire\ to\ brake\ speed\ ratio(R)} \qquad (3.6.1.2)$$

But,

Tyre to brake speed Ratio is 1

Thus brake torque is given as,

$$Brake\ Torque = \frac{Braking\ force\ \times Tire\ Radius}{1}$$

$$Brake\ Torque = \ Braking\ force\ \times Tire\ Radius \qquad (3.6.1.3)$$

From above calculations and the details on Tuk -tuk wheel,

$Braking\ force = 6500N$

$Tire\ Radius = 0.1524m$

$Brake\ Torque = \ 6500N\ \times 0.1524M$

$Brake\ Torque = \ 6500N\ \times 0.1524M$

$Brake\ Torque = 990.9N.M$

We calculate the disc effective radius which is given by the equation,

$$Disc\ effective\ radius, R_e = \frac{(Disc\ outside\ diameter + Disc\ inside\ Diameter)}{4} \qquad (3.6.1.4)$$

$$Disc\ effective\ radius, R_e = \frac{(153 + 100)}{4}$$

$$Disc\ effective\ radius, R_e = \frac{(153 + 100)}{4}$$

$$Disc\ effective\ radius, R_e = 63.125mm$$

Then using the value attained for brake torque and the disc effective radius together with the frictional coefficient of the disc and its articulation with the brake pad, we calculate theoretical clamp force (N), CFt. It can be calculated by the formula given below

$$Theoretical\ clamp\ force = \frac{brake\ torque}{R_e \times \mu \times n} \quad \text{......} (3.6.1.5)$$

Where,

$\mu =$ Is the coefficient of friction

$n =$ Is the number of frictional surfaces

μ for the disc brake lining according to the company manufacturing Focin motorbikes is in a range of 0.35 to 0.42 and it all depends on the model number. The more demanding models use the higher value. We chose 0.35 because of the non-demanding working environment of our vehicle.

Theoretical clamp force is then given as;

$$Theoretical\ clamp\ force = \frac{990.9Nm}{0.063125m \times 0.35 \times 2} \quad \text{......} (3.6.1.6)$$

$$Theoretical\ clamp\ force = 22425N$$

3.7. AXLE

3.7.1. SHAFT

A shaft is a rotating member, usually of circular cross section, used to transmit power or motion. It provides the axis of rotation, or oscillation, of elements such as gears, pulleys, flywheels, cranks, sprockets, and the like and controls the geometry of their motion.
An axle is a non-rotating member that carries no torque and is used to support rotating wheels, pulleys, and the like (Shigley's Mechanical Engineering Design, Eighth Edition Budynas−Nisbett).
Unlike a rotating axle, a non-rotating is designed and analysed as a static beam. To ensure that the shaft used is of correct strength and not an overdesigned component that is beyond the economic constraints allocated, various theoretical tests are carried to get a picture of what the final design will be like geometrically and the component's mechanical properties. Simulation of this component can also be done using software that are used for these analysis.

SHAFT MATERIALS

Deflection is affected stiffness as represented by the modulus of elasticity, which is essentially constant for all steels. For that reason, rigidity is only controlled by geometric decisions.
Shafts are usually made from low carbon, cold-drawn or hot-rolled steel, such as ANSI 1020-1050 steels. Heat treatment and high alloy content are often not warranted. Fatigue failure is reduced moderately by increase in strength, and then only to a certain level before adverse effects in endurance limit and notch sensitivity begins to counteract the benefits of higher strength. A good practice is to start with inexpensive, low or medium carbon steel for the first time through the design calculations.
Usually, cold drawn steel is used for diameters under about 3 inches. Our shaft is less than 3inches and cold drawn steel became our choice. We chose cold drawn steel ANSI 1020 with properties as shown turned and polished as delivered and only surface hardened the bearing position thus cutting on the production costs.

PROPERTIES OF CARBON STEEL ANSI 1020

ANSI 1020 is a general purpose low tensile low hardenability carbon steel generally supplied in the cold drawn or turned and polished condition. They have tensile strength that range from 410 - 790 Mpa, and Brinell hardness range of 119 - 235.

They have excellent weld-ability, very good machinability with reasonable strength and very good ductility.

ANSI 1020 is used extensively by all industry sectors for applications requiring good machinability or excellent weld-ability. It is also often used un-machined as supplied, its bright cold drawn or turned and polished finish being acceptable for many applications.

They are used in the manufacture of Axles, General Engineering Parts and Components, Machinery Parts, Shafts etc.

ANSI 1020 comes in Round bars of diameter ranging from 6 mm to 200 mm.

CHEMICAL COMPOSITION

Chemical composition	Minimum percentage	Maximum percentage
Carbon	0.15	0.25
Silicon	0.00	0.35
Manganese	0.30	0.90
Phosphorous	0.00	0.05
Sulphur	0.00	0.05

Table 1: Chemical composition of Carbon Steel ANSI 1020

Cold drawn sizes mm		-16mm	17-38mm	39-63mm	Turned and polished (All sizes)
Tensile strength Mpa	Min	480	460	430	410
	Max	790	710	660	560
Yield strength Mpa	Min	380	370	340	230
	Max	610	570	480	330
Elongation in 50mm %	Min	10	12	13	22
Hardness HB	Min	142	135	120	119
	Max	235	210	195	170

Table 2: Typical Mechanical Properties - Cold Drawn and Turned and Polished Condition

HEAT TREATMENT

Annealing

Heat to 870^0 C - 910^0 C hold until temperature is uniform throughout the section, and cool in furnace.

Tempering

After Carburising, Core Refining and Case Hardening, re-heat to 150^0 C - 200^0 C, hold until temperature is uniform throughout the section, soak for 1 hour per 25 mm of section and cool in still air.

Tempering will improve the toughness of the case with only slight reduction in case hardness. It will also reduce the susceptibility to grinding cracks.

Normalizing

Heat to 890^0 C – 940^0 C hold until temperature is uniform throughout the section, soak for 10 - 15 minutes and cool in still air.

Stress Relieving

This is done by heating the bars to 650^0 C – 700^0 C. The bars are then held until temperature is uniform throughout the sections. Finally they are soaked for 1 hour per 25mm of section, and then cooled in still air.

Heating temperatures, rate of heating, cooling and soaking times will vary due to factors such as work piece size/shape, also furnace type employed, quenching medium and work piece transfer facilities etc.

Machining

ANSI 1020 in the bright cold drawn or turned and polished as supplied condition has very good machinability and all operations such as turning, drilling, tapping and milling can be carried out satisfactorily as per machine manufacturers' recommendations for suitable tool type, feeds and speeds.

Welding

ANSI 1020 has excellent weld-ability in the cold drawn or turned and polished as supplied condition, and can be readily welded by any of the standard welding processes.

It should be noted that welding in the carburised or heat treated condition is not recommended.

Welding Procedure

ANSI 1020 is welded using low carbon electrodes. A pre-heat or post-heat is not generally required, however pre-heating larger sections can be beneficial as can a post-weld stress relieve if this is possible.

SHAFT LAYOUT

Our rear shaft is designed to accommodate bearings and the two hubs carrying the sprocket and the disc brake. Shaft shoulders are usually used to carry any thrust loads, in our case the only shoulder is a 40mm diameter tube with an internal diameter of 30mm bolted on to the shaft to stop the bearing from lateral movement. We have not used real shoulders because we do not have feasible axial forces that can warrant for a shoulder or shoulders. We have used a 40mm collar instead. The power transmission from the shaft to the hubs is done by the use of keys and keyways only. The keyways on the shaft are designed correctly such that stress concentration do not result in catastrophic effect on the shaft during its normal operations.

AXIAL LAYOUT OF COMPONENTS

It is best to support load-carrying components between bearings, and not cantilevered outboard of the bearings. In this case our hubs carrying both the sprocket and the disc brake are mounted in between our two main bearings.

As a way to minimise bending moments and deflections our shaft has its bearings kept at a distance of 263mm from each other. As per the shaft design requirement, load bearing components should be placed near the bearings, we have used the same point at which the bearing is located as our point of load application this help to minimize the bending moment at the critical locations that have stress concentrations, and to minimize the deflection at points at which the load-carrying components are located.

We have designed hubs that are bolted on to the shaft so that the anchored components are aligned accurately with other mating components, the bolts are also there to hold the component in place securely. Even though we are not using any shoulder, that complicate the design of the shaft, the use of keys and keyways help to serve as a perfect replacement for the shoulders and provide support to minimise deflection and vibration and wobbling of the component. Because of the minimum axial forces on the shaft we have done away with the shoulders and instead opt for collars with set screws to maintain the axial position of the bearings.

PROVIDING FOR TORQUE TRANSMISSION

Our shafts serve to transmit torque from the motor via a chain and a sprocket located on the shaft to the wheels. Having gathered for the torsional stresses and deflection we chose a means of transmitting torque between the sprocket and the shaft and the wheel. We had a lot of options to choose from, they include:

- Keys
- Splines
- Pins
- Press or shrink fits
- Tapered fits

The above mentioned devices chosen should be able to fail in case a certain torque is exceed so that the more expensive components are protected.

Keys

It is one of the most effective and economical means of transmitting moderate to high levels of torque through a key that fits in a groove in the shaft and hub. Keyed components generally have a slip fit onto the shaft, so assembly and disassembly is easy. The key provides for positive angular orientation of the component, which is useful in cases where phase angle timing is important.

Splines

They are essentially stubby gear teeth formed on the outside of the shaft and on the inside of the hub of the load-transmitting component. Splines are generally much more expensive to manufacture than keys, and are usually not necessary for simple torque transmission. They are typically used to transfer high torques.

Press and shrink fits

They are used for securing hubs to shafts; moreover they are used both for torque transfer and for preserving axial location.

Tapered fits

They are used between the shaft and the shaft-mounted device, such as a wheel, are often used on the overhanging end of a shaft. Screw threads at the shaft end then permit the use of a nut to lock the wheel tightly to the shaft.

We choose the key and keyway because of the moderate torques that we are transmitting. The keyways are also cheaper compared to splines.

Key dimensions.

Parallel keys are the common keys used. Key and key seat are ISO standardized. The key length should be about less than 1.5 times of the shaft diameter to ensure good load distribution over the entire key length when the shaft becomes twisted when loaded in torsion.

For our 30mm shaft the length of the keyway is 20mm

According to tribology calculator, a shaft with 30mm diameter and carrying a torque of 990N.m will take a key of with the following specifications

Key width b = 8mm

Key height h = 7mm

Keyway depth into the shaft = 4mm

Key depth into the hub = 3.3mm

SHAFT FORCE AND DIAMETER CALCULATION

Our shaft is 1m long from the point in which the wheels are attached. The sprocket and a disc brake are held between two bearings that act as the loading point of the shaft and as the connection between the shaft and the frame. The bearings being the only point having a real contact with the frame, have the following forces B_1 and B_2 acting on the shaft. These bearing points are located 0.351m from each end of the shaft.

Force acting on $B_1$and B_2 are given as

$$B_1 = 125 \times 9.81 = 1226.25N$$

$$B_2 = 125 \times 9.81 = 1226.25N$$

This is because the two points are placed at the same distance from the ends of the shaft. The mass being supported is also assumed to be equally distributed.

The bearing and the reaction forces are not the only forces acting on the shaft, there are other loads on the shaft.

From the sprocket and brake forces calculations, the torque calculated were *654N.m* for sprocket and -*990.1N.m* for disc brake. The negative sign do not mean anything because the shaft is rotational. In our shaft design we choose the torque due to disc brake because it is higher thus help us to design a secure shaft.

The ends where the wheels are bolted act as the reaction zone where the Newton's action and reaction theory is satisfied. Reaction forces $R_1$and R_2 are located at these points. The assumption that the load is equally distributed and that the vehicle is on a relatively level ground gives us the reaction forces $R_1 = R_2 = B_1 = B_2 = 1226.25N$.

We now calculate the bending moment about both B_1 and B_2

$M_{B_1} = \sqrt{(x \times R_1)^2((1-x) \times R_2)^2}$ .... (3.7.1.1)

$$M_{B_1} = \sqrt{(0.351 \times 1226.25)^2((1-0.351) \times 1226.25)^2}$$

$$M_{B_1} = 904.771N.m$$

$M_{B_1} = M_{B_2}$ This is because the points at which the forces act on the axle are equidistant.

Next is to calculate the nominal tensile and shear stress. This is done to enable us to calculate the geometric stress concentration factors even though we do not have the diameter of the shaft.

$\sigma_0 = \frac{32M}{\pi d^3}$ .... (3.7.1.2)

$$\sigma_0 = \frac{32 \times 907.771}{\pi d^3}$$

$$\sigma_0 = \frac{9215.93}{d^3}$$

The maximum shaft torsional shear occurs between the two bearings.

$$\tau_0 = -\frac{16T}{\pi d^3} \quad \text{..... (3.7.1.3)}$$

$$\tau_0 = -\frac{16 \times 990.1}{\pi d^3}$$

$$\tau_0 = -\frac{5042.03}{d^3}$$

We calculate the maximum tensile and torsional stress

$$\sigma_1 = \sigma_{max} = \frac{\sigma_0}{2} + \sqrt{(\frac{\sigma_0}{2})^2 + {\tau_0}^2} \quad \text{..... (3.7.1.4)}$$

$$\sigma_{max} = \frac{1}{d^3}(\frac{9215.93}{2}) + \sqrt{(\frac{9215.93}{2})^2 + 5042.03^2}$$

$$\sigma_{max} = \frac{11438.44}{d^3}$$

$$KO\tau_1 = \tau_{max} = \sqrt{(\frac{\sigma_0}{2})^2 + {\tau_0}^2} \quad \text{..... (3.7.1.5)}$$

$$\tau_1 = \tau_{max} = \sqrt{(\frac{9215.93}{2})^2 + 5042.03^2}$$

$$\tau_{max} = \frac{6830.48}{d^3}$$

Because of the presence of the keyway in the shaft the ultimate tensile strength, s_{ut} of the material together with its yield strength, s_y is reduced to 75% of the material's (AISI 1020) s_{ut} and s_y

Thus the tensile strength becomes

$$0.75 \times 710\ Mpa = 532.5 Mpa$$

Theoretical or geometric stress concentration factors.

K_t This is a factor for the normal stress.

K_{ts} This is a factor for the shear stress.

$$K_t = \frac{\sigma_{max}}{\sigma_0}$$

$$K_t = \frac{11438.44}{9215.93}$$

$$K_t = 1.24$$

$$K_{ts} = \frac{\tau_{max}}{\tau_0}$$

$$K_{ts} = \frac{6830.48}{5042.03}$$

$$K_{ts} = 1.355$$

We now calculate the miscellaneous effect K_f and K_{fs}that account for the endurance reduction due to all other factor effects.

Notch type	$\sqrt{a}$	C_k
Transverse	$174/S_{ut}$	0.10
Shoulder	$139/S_{ut}$	0.11
Groove	$104/S_{ut}$	0.15

Table 3: Heywood parameter for $\sqrt{a}$ and C_k

Thus

$$\sqrt{a} = \frac{104}{532.5} = 0.1953$$

$$K_f = \frac{K_t}{1 + \frac{2(K_t - 1)\sqrt{a}}{K_t\sqrt{r}}}$$

$$K_f = \frac{1.24}{1 + \frac{2(1.24 - 1)1953}{1.24\sqrt{2}}}$$

$$K_f = 1.177$$

$$K_{fs} = \frac{K_{ts}}{1 + \frac{2(K_{ts} - 1)\sqrt{a}}{K_{ts}\sqrt{r}}}$$

$$K_{fs} = \frac{1.355}{1 + \frac{2(1.355 - 1)1953}{1.355\sqrt{2}}}$$

$$K_{fs} = 1.047$$

$$M_a = 904.77N.m$$

$$T_a = 991.3N.m$$

$$M_m = T_a = 0$$

$$\sigma_a = K_f \frac{32M}{\pi d^3} \qquad \text{(3.7.1.6)}$$

$$\sigma_a = 1.177\frac{32X904.77}{\pi d^3} = \frac{10847.13}{d^3}$$

$$\tau_m = K_{fs} \frac{16T_m}{\pi d^3} \qquad \text{(3.7.1.7)}$$

$$\tau_m = 1.047\frac{16X991.3}{\pi d^3} = \frac{5285.94}{d^3}$$

Take factor of safety as 2

Using De Gerber’s formula we calculate the diameter of the shaft required to do the job.

$$d = (\frac{8nA}{\pi S_e}(1 + (1 + (\frac{2BS_e}{AS_{ut}})^2)^{\frac{1}{2}}))^{\frac{1}{3}} \quad \text{...... (3.7.1.8)}$$

But A and B are given as

$$A = \sqrt{(4(K_f Ma)^2 + 3(K_{fs}T_a)^2} \quad \text{...... (3.7.1.9)}$$

$$A = \sqrt{(4(1.177X904.77)^2} = 2129.82858$$

$$B = \sqrt{(4(K_f Mm)^2 + 3(K_{fs}T_m)^2} \quad \text{...... (3.7.1.10)}$$

$$B = \sqrt{(4(K_f Mm)^2 + 3(K_{fs}T_m)^2}$$

$$B = \sqrt{(3(1.047X991.3)^2} = 1797.68$$

$$S_e = K_a K_b K_e S_e' \quad \text{...... (3.7.1.11)}$$

$S_e' = 0.5S_{ut}$ for 1400Mpa

$$S_e' = 0.5X532.5 = 266.25Mpa$$

$$K_a = aS_{ut}{}^b \quad \text{...... (3.7.1.12)}$$

Surface finish	Factor a	Factor b
Ground	1.58	-0.085
Machined / cold drawn	4.51	-0.265
Hot rolled	57.7	-0.718
Forged	272	-0.995

Table 4: Surface finish factorK_a for steel

We used the cold drawn AISI 1020

$K_a = 4.51(532.5)^{-0.265} = 0.8545$

We calculate the size factor for our rotating shaft.

$K_b = (\frac{d}{7.62})^{-0.107}$For diameters of 2.79≤ d ≤ 51

We pick any diameter within the above range for example 30mm and use in our calculation

$$K_b = (\frac{30}{7.62})^{-0.107} = 0.8636$$

Loading factor K_c is calculated.

$K_c = 1$ For bending

$K_c = 0.85$ Axial

$K_c = 0.59$ Torsion

The temperature factor K_d according to Shigley's machine design 8th edition is 1 for temperature of around 20^0C

The reliability factor $K_e$of the shaft with 99% reliability is 0.814 according to the table below (an excerpt from Shigley's Machine design 8th edition)

Reliability %	Reliability factor
50	1.00
90	0.89
95	0.868
99	0.814
99.9	0.756
99.99	0.702

Table 5: Reliability factor.

$S_e = 0.8545X0.8636X0.814X266.25 = 159.93Mpa$

The diameter can then be calculated using the De Gerber's formula as shown below

$$d = (\frac{8 \times 2129.83}{\pi \times 159.93 \times 10^6}(1 + (1 + \left(\frac{2 \times 1797.68 \times 159.93}{2129.83 \times 532.25})^2)^{\frac{1}{2}}\right))^{\frac{1}{3}}$$

$$d = (1.696 \times 10^{-4}(1 + (1 + \left((0.507)^2)^{\frac{1}{2}}\right))^{\frac{1}{3}} = \ 0.028448m$$

Metric shafts of between 20-50mm have an increment of the diameter by 1mm thus choosing 30mm shaft was feasible.

CALCULATION OF FORCES FOR STUB AXLE/FRONT AXLE

Stub axle is subjected to shearing forces and negligible torsional.

The shear stress at the shoulder of the 6.547mm bolt and the 20mm diameter axle is given by

$$\tau = \frac{F}{A} = \frac{1226.5}{\frac{\pi}{4} \times 6.576^2 \times 10^{-6}} = 36.104MPa$$

The shearing force is well less than the shear strength of the material. Shear strength is 60% of tensile strength.

Shear strength = 0.60x532.5 = 319.35MPa

3.8. PEDALS AND LINKAGES

From the calculations of the breaking forces, the breaking force was 990N. The force exerted by the driver is about 4kgs according to highway safety research institute

We choose to use force magnification to achieve breaking force required. Using the maximum force exerted and the breaking force required, the magnification factor was found to be,

= 990/39.24

= 25

Hence the mechanical advantage to be achieved by use of levers should be 25

The maximum possible brake pedal ratio is 6: 1. Hence the force delivered at the cables is

= 39.24x6

= 235.44 Newton's

We require another magnification factor of 5 at the callipers

Brake cables were to be used to transmit the magnified force to the callipers

The distance from the foot pressure application to the pivot is9cm and from the pivot to the connection hook is 1.5cm.Giving a magnification factor of 6.

After the cables have transmitted the force to the disk brake, the sliding callipers move with respect to the disk, a piston on one side of the disc pushes the inner brake pad until it makes contact with the braking surface, then pulls the calliper body with the outer brake so that pressure is applied on both sides of the disc.

The next point of magnification is at the piston which pushes the inside calliper. The distance from the connection hook to the pivot is 8cm and from the pivot to the pushing piston is 1.6cm

The delivered force is

=235.44x 5

= 1177.2 N

3.9. MOTOR DRIVE SYSTEM

To choose the motor for the car, the following torque and power calculations were done

The total tractive force required for the vehicle;

Total tractive force = rolling resistance + force required to climb a grade + force required to accelerate to maximum velocity

Rolling resistance

Rolling resistance = gross vehicle weight * surface friction value

= 300 x 0.22(surface friction value for asphalt)

=66 kilograms

= 66 x9.81 = 647.46N

Grade resistance

Maximum angle the vehicle will be expected to climb in normal operation;

Grade resistance = gross vehicle weight * maximum incline angle (degrees)

= 300 x sin 17

= 87.71KGS

= 860.444N

Acceleration force

Acceleration force is the force necessary to accelerate from a stop to maximum speed

Acceleration force = Vehicle mass* (maximum speed / time required *Gravity)

= 300 x (60/12)

= 1500

Hence total tractive force = 647.46 + 860.44 + 1500

= 3007.9 N

Wheel Motor torque

Wheel torque = total tractive force* radius of the wheel * resistance factor

Resistance factor accounts for the frictional losses between the wheels and the axis. Typical values range from 1.12 to 1.15

Wheel motor torque = (3007.9 * 0.1524 *1.12)/ 9.81

= 51.4 N.M

Hence the power required;

$p = \tau\omega$

= 51.4Nm *109.38rad/sec

= 5602watts

= 7.5Horsepower

To choose the appropriate motor, we multiply by a factor of safety 1.2

= 9.011 horsepower

Hence the required motor is 10 horse powers DC motor.

From evaluation of the local market, dc motors are not available. The most viable option is to order from eBay. We found the motor to suit our torque requirements which was a Manta- Etek permanent magnet DC motor from Briggs and Stratton. The permanent magnet motor is compact and creates its torque by pushing two magnetic fields against each other. One magnetic field is produced by mineral permanent magnets while the other magnetic field is from the batteries. This provides for a very powerful and yet very small motor.

This motor is a 10 horsepower 48 volts DC motor. It is used in electric motor cycles, robots and in small electric cars. It does not require an expensive speed controller and its weight is 10kgs.

It is placed behind the driver’s seat to the right of the vehicle and is held in place by a vertical plate that is bolted onto the front side of the motor. This vertical plate is hollow to allow for the position of the rotating shaft. A horizontal plate is welded onto the frame and the motor is bolted on top of it.

Figure 7 Image of Briggs and Stratton Motor (www.speedace.info/solar_car_motor_and_drivetrain.htm)

To provide 48volts, four 12 volts batteries are required. These batteries are readily available and we chose Chloride Exide lead acid batteries with a weight of 5kgs each. Giving us a total weight of 20kgs.Adding the weight of batteries and motor, the total weight is 30kgs.This takes 16 percent of the total vehicle weight.

A motor controller is placed between the motor and the batteries and is used to control the speed of the motor. We choose a stepper motor controller that increases the voltage from the batteries and runs the motor at higher voltages. The stepper motor controller can also reverse the direction of rotation as when required to. The motor controller produces a signal representing the demanded speed and drives the motor at that speed.

CHAPTER FOUR

4.0. ECONOMIC CONSIDERATIONS

We evaluated the cost of all the parts in the vehicle and they are as follows;

Frame	70000
Steering system	15000
Chain and sprocket	5000
Suspension	13000
Axle	8000
Disc Brake	2800
Motor	56000
batteries	4 x 6000 = 24000
wheels	16000
Total	**Kshs. 209,800**

Table 6: Costing of the EV

The total cost is Kshs. 209,800. Considering that it is the initial design, this is a fair budget. The final electric vehicle’s benefits surpass the cost.

The minimum cost of a car engine for a diesel or petrol powered vehicle is 420000 which twice the cost of our entire vehicle which shows that the electric vehicle saves on money and saves on the environment due to reduced pollution effects. Commercial electric cars which are available for the public in the world markets are more expensive. Tesla roadster is in the markets with a marked price of over Kshs. 8million. This is very high as compared to our design, 40 times higher.

CHAPTER FIVE

5.0. DISCUSSION

The frame members joining at right angles are mitred at an angle of 45^0. The diagonal ones are mitred to the requirement of the joint being made. Welding is done on all the members and every joint should be a complete one that is, no holes should be left unfilled in the frame. This ensures that the strength of the frame is not compromised.

The rear axle is mounted to the frame through bearings on a bearing carrier bolted to a plate. The plate is welded on to the frame this is done to avoid any possible weakening of the frame. Hubs carrying both the sprocket and the rotor are mounted onto it. The hubs are anchored on to the shaft using bolts with the help of a key to enhance the torque transmission.

A hub was mounted on the front axle to carry the wheel. To enhance the safety in case the nut loosens a jam nut was added to make sure the hub was securely in place.

Tuk-tuk wheels were chosen because of their availability and the clearance they could offer thus enhancing the stability of our car. The wheels are threaded this allow the car to be used on an all-weather road.

The motor which is placed strategically behind the driver's seat is guarded from direct contact with a metallic plate insulated from the side facing the motor to eliminate injuries that can happen due to exposure to moving parts and to guard against high temperatures. It is held onto the frame by a casing that is bolted onto a plate. The plate is welded onto the frame.

Wires used in the wiring to connect the motor to the accelerator and the motor controller are insulated. This reduces the chances of the operator coming into contact with live wires.

The four lead acid batteries are kept in a container that is a non-conductor. The container is fixed onto the frame through a simple plate bolted onto the frame. This ensures that the operators or the driver is not exposed to the corrosive substance and prevent a possible exposure to high voltages.

The braking cable is fixed in such a way to minimize the number of corners or the angle of bend. This is important to minimize stress concentrations and wear on the cable and also ensure that the cable transmits the required braking force.

CHAPTER SIX

6.0. CONCLUSION AND RECOMMENDATIONS

6.1. CONCLUSION

The aim of the experiment was to design a handmade EV using locally available subassemblies and to determine the cost affordability of the designed EV.

The EV was designed to the SAE and Formula Students standards to ensure the safety of the vehicle for the immediate user the environment and the public. This was done as part of our own safety.

Most of our subassemblies were made from the locally available material and technology. Also, the parts bought were at fair prices but to the required standards. Therefore, basing on our design, our EV is more economical as compared to other existing designs.

Overall, our design provides an innovative and low cost solution to our design problem. The vehicle weighs 300kgs and is cheap enough to warrant its actual fabrication. The vehicle meets all the design standards required hence safe to drive.

6.2. RECOMMENDATIONS

There are a variety of ways in which this design may be improved that were not implemented in this project. Some of the improvements include the use of a closed body using light materials like carbon fibres or fibre glasses. The chassis or frame could also be made of different materials. The lower part taking on steel then the upper can either be made using the advanced composites or aluminium alloys.

A second improvement could be to use longer lasting alkaline batteries as opposed to the lead acid batteries. The seat can be made adjustable to accommodate both younger and adults comfortably.

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APPENDIX

Go Kart

Go Kart is a 4-wheeled car with small sized engine purposely used in United States in earlier1950s. Kart Ingels was the first one to develop the concept of Karting. He built the first concept car in California in year 1956. From there it became famous all over America and Europe.

TABLE 1: NUMBER OF TEETH ON SMALL SPROCKET IS STANDARDS

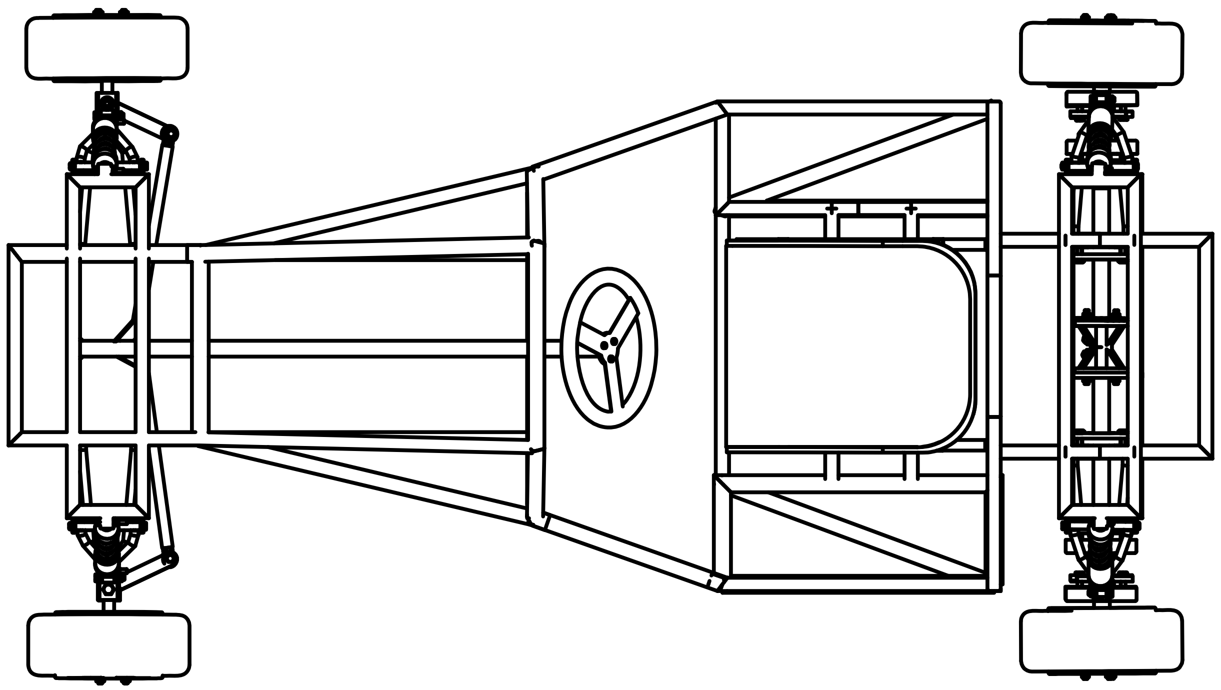
TYPE OF CHAIN	CHAIN	NUMBER OF TEETH AT VELOCITY RATIO					
		1	2	3	4	5	6
ROLLER CHAIN		31	27	25	23	21	17
SILENT CHAIN		40	35	31	27	23	19

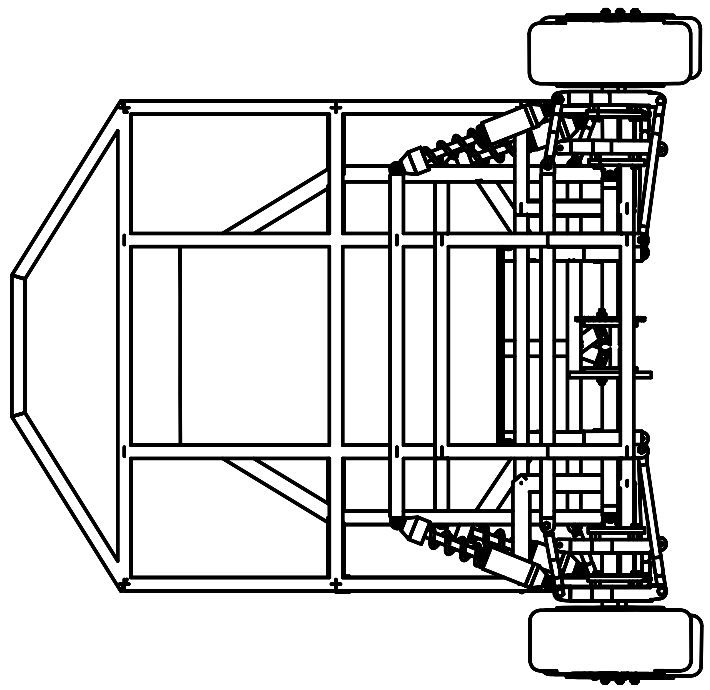
TABLE 2: POWER RATING OF SIMPLE ROLLER CHAINS

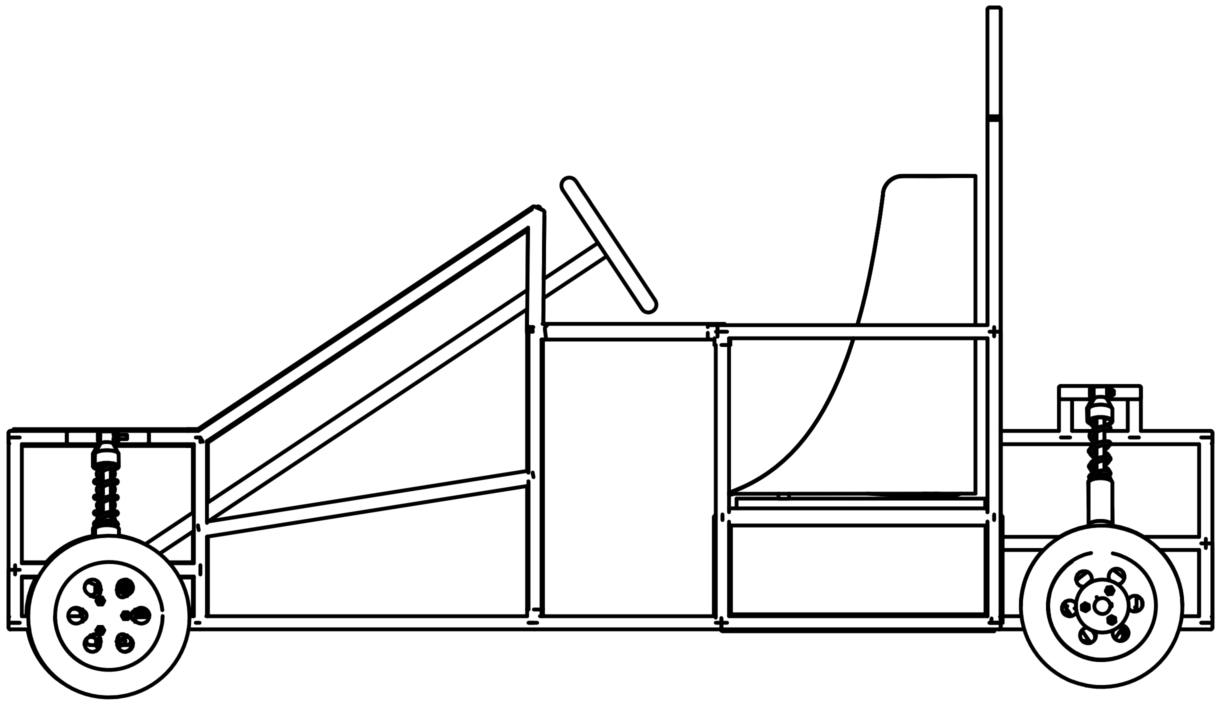
				POWER			
SPEED OF SMALL SPROCKET			06B	08B	10B	12B	16B
	100		0.25	0.64	1.18	2.01	4.83
	200		0.47	1.18	2.19	3.75	8.93
	300		0.61	1.7	3.15	5.43	13.06
	500		1.09	2.72	5.01	8.53	20.57
	700		1.48	3.66	6.71	11.63	27.73
	1000		2.03	5.09	8.97	15.63	34.89
	1400		2.73	6.81	11.67	18.15	38.47
	1800		3.44	8.1	13.03	19.85	
	2000		3.8	8.6	13.49	20.57	

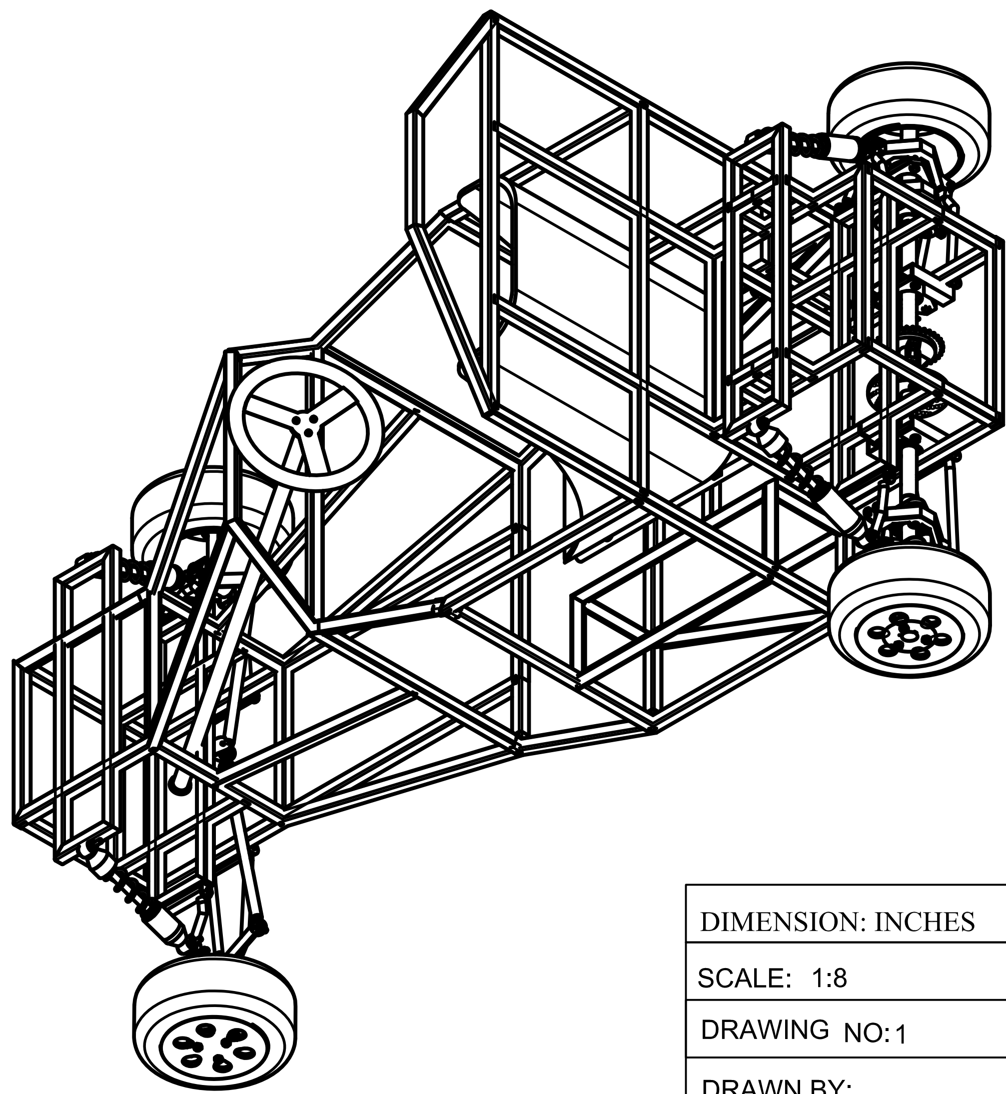
TABLE 3: CHARACTRISTICS OF ROLLER CHAINS IS 2403- 1991

ISO CHAIN NUMBER	PITCH	ROLLER DIAMETER	WIDTH BETWEEN	TRANSVERSE		MAXIMUM BREAKING LOAD		
		(mm)	(mm)	PLATES B1 (mm)	PITCH (MM)	Simple KW	Duplex (KW)	Triplex (KW)
	05B	8	5	3	5.64	4.4	7.8	11
	06B	9.525	6.35	5.72	10.24	8.9	16.9	24
	08B	12.7	8.51	7.75	13.02	17.8	31.1	44
	10B	115.87	10.16	9.65	16.59	22.2	44.5	86
	12B	19.05	12.07	11.68	19.46	28.9	57.8	126
	16B	25.4	15.88	17.02	31.88	42.3	129	193
	20B	31.75	19.05	19.56	36.45	64.5	195	293
	24B	38.1	25.40	25.46	48.36	97.9	258	501
	28B	44.45	27.90	30.99	59.56	129	328	787
	32B	50.8	29.20	30.99	68.55	169	524	796
	40B	63.5	39.37	38.1	72.29	1262	580	805
	48B	76.2	48.26	45.71	91.21	4003	800	1200









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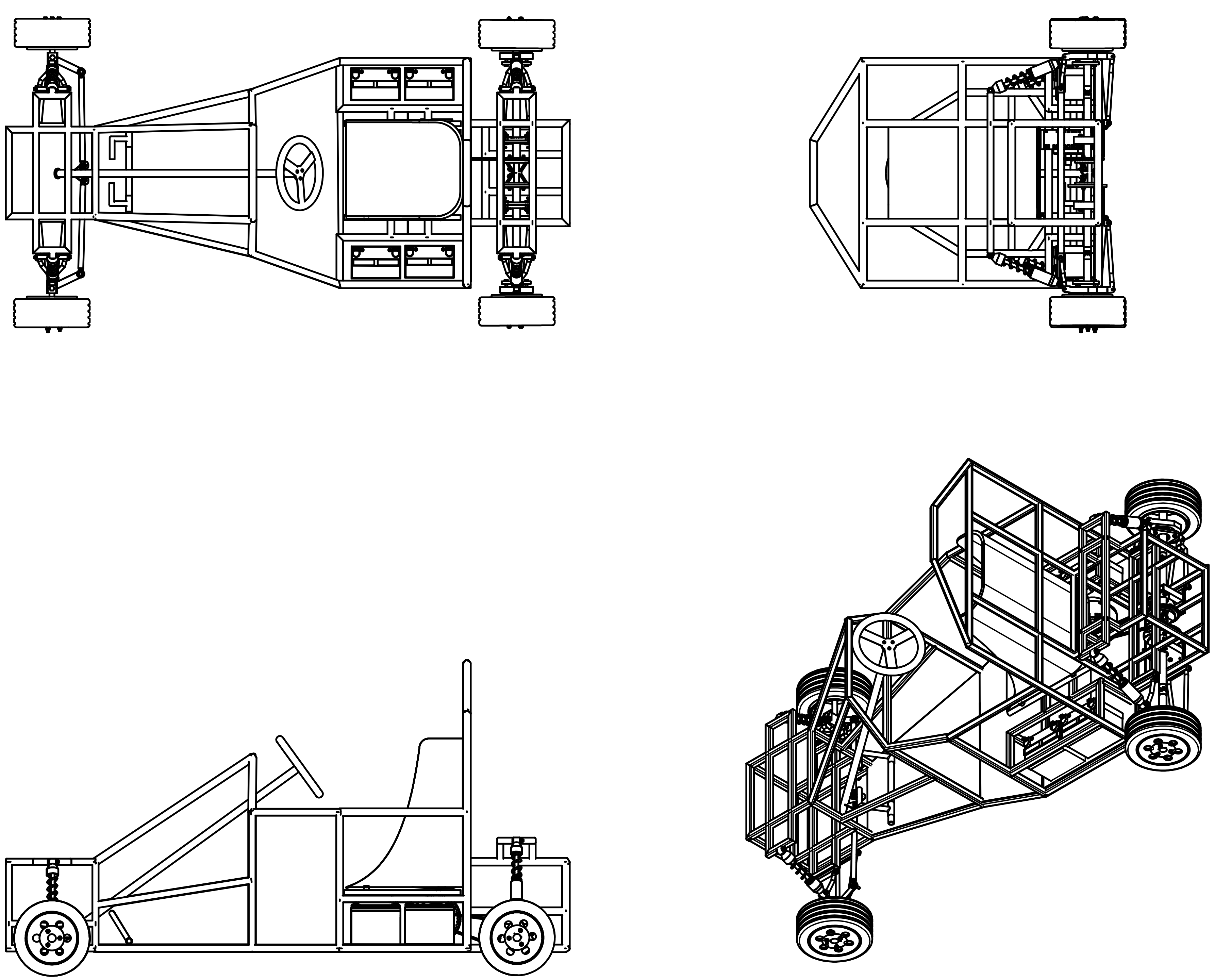

PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	Frame	
2	1	Base + plate + Shaft Carrier	
3	1	Rear Axle	
4	2	Suspension carrier	
5	8	Attachee	
6	2	Upper semi	
7	2	Top Advanced	
8	4	Holder	
9	2	Shock 2	
10	1	COMPLETE LEFT	
11	1	COMPLETE RIGHT	
12	2	Upper Front	
13	1	Lower Right	
14	1	Lower left	
15	2	Shock 2	
16	1	Trac rod 3	
17	1	Track rod left	
18	1	Steering semi	
19	1	Accelerator pedalPedal	
20	1	Brake Pedal	
21	1	Holder of the pedals	
22	1	Floor rear 2	
23	1	Floor front	
24	4	Battery2	
25	1	MantaMotor	
26	1	Support for batteries	
27	1	Spur Gear1	
28	1	Spur Gear2	
29	1	Roller Chain	
30	1	Roller Chain Sprocket1	
31	1	Roller Chain Sprocket2	
32	2	ISO 15 - 8 38 - 20 x 32 x 10	Rolling Bearing
33	2	Front_Hub	
34	2	Wheel2	
35	1	Frame Assembly	
36	1	Seat Assembly3	
37	13	ISO 4014 - M8 x 40	Hex-Head Bolt
38	17	ISO 4032 - M8	Hex Nut
39	8	14 CM BOLT	
40	3	ISO 4017 - M8x35	Hex-Head Bolt

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MFG				
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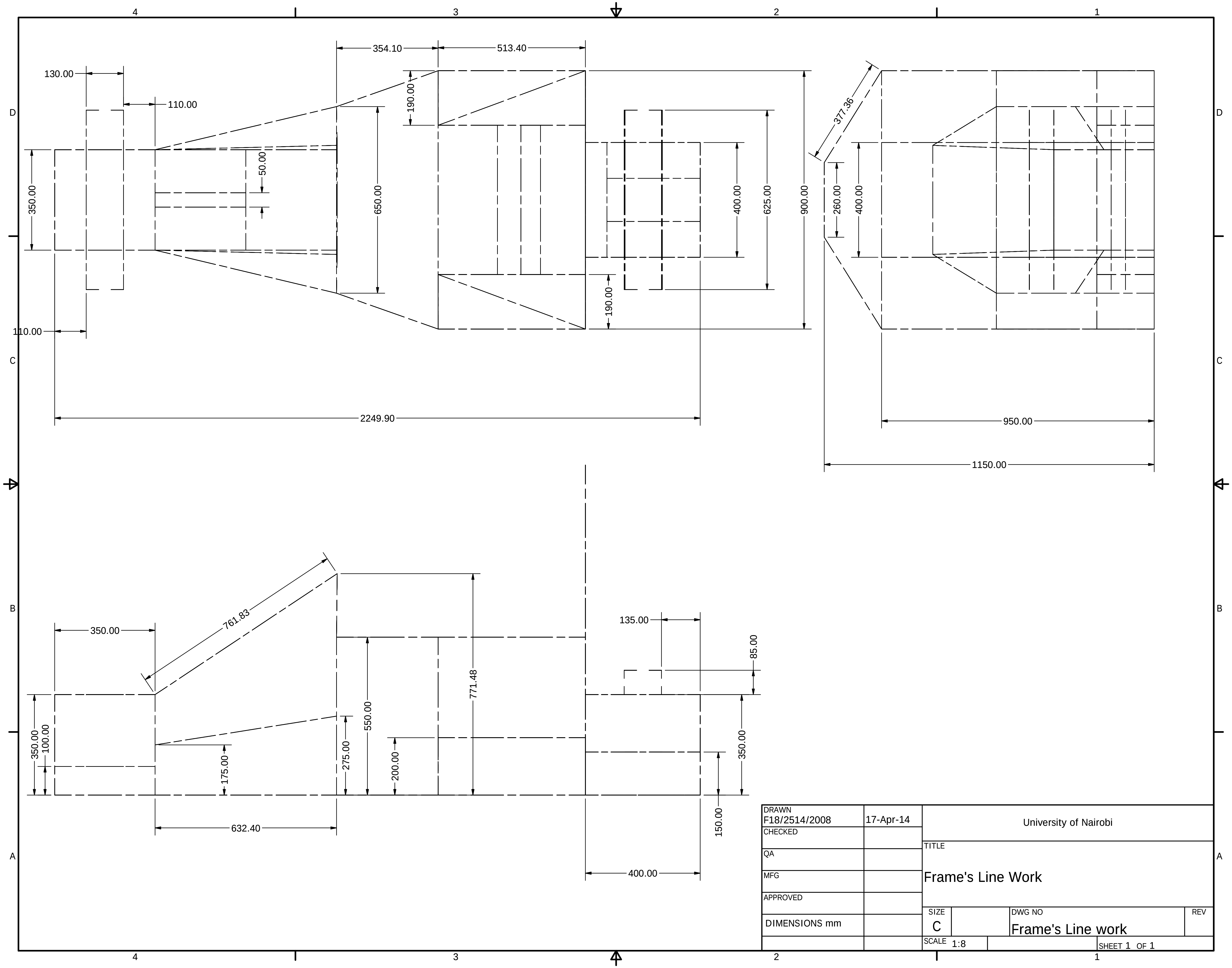

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PARTS LIST			
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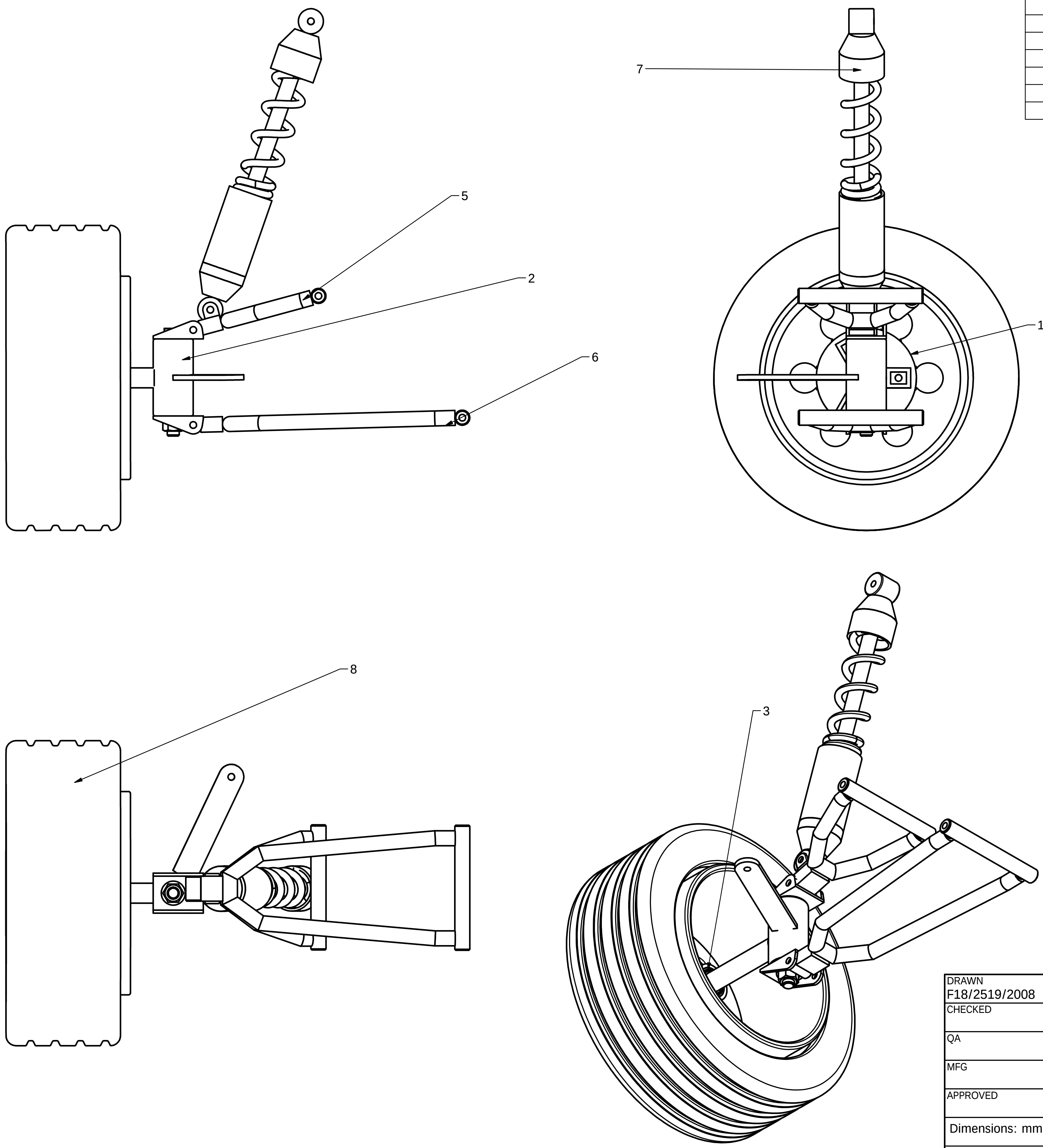
1	1	Front_Hub	
2	1	COMPLETE LEFT	
3	1	ISO 15 - 8 38 - 20 x 32 x 10	Rolling Bearing
4	2	ISO 7414 - M12	Hex Nut
5	1	Upper semi	
6	1	Top Advanced	
7	1	Shock 2	
8	1	wheel	



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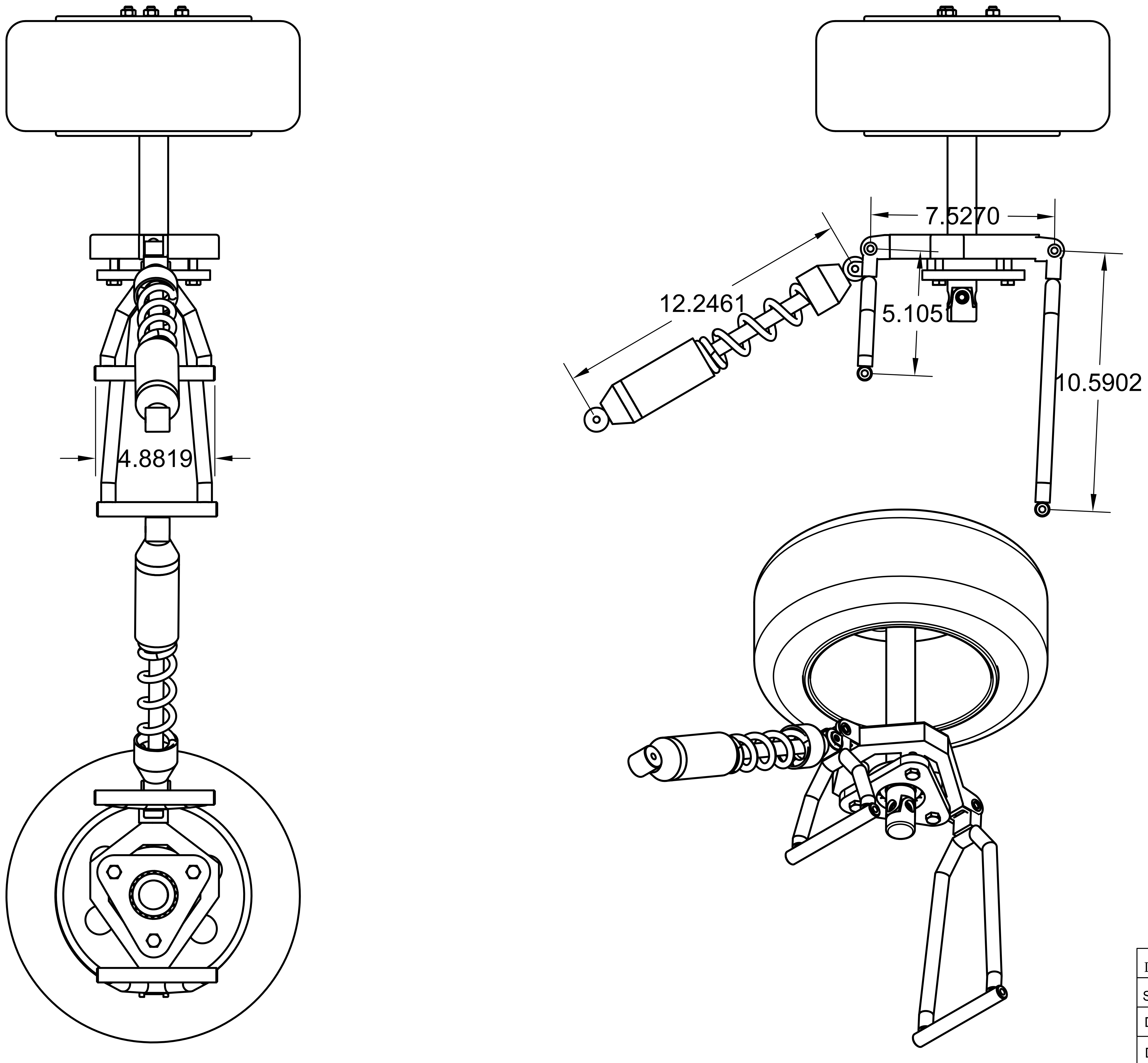

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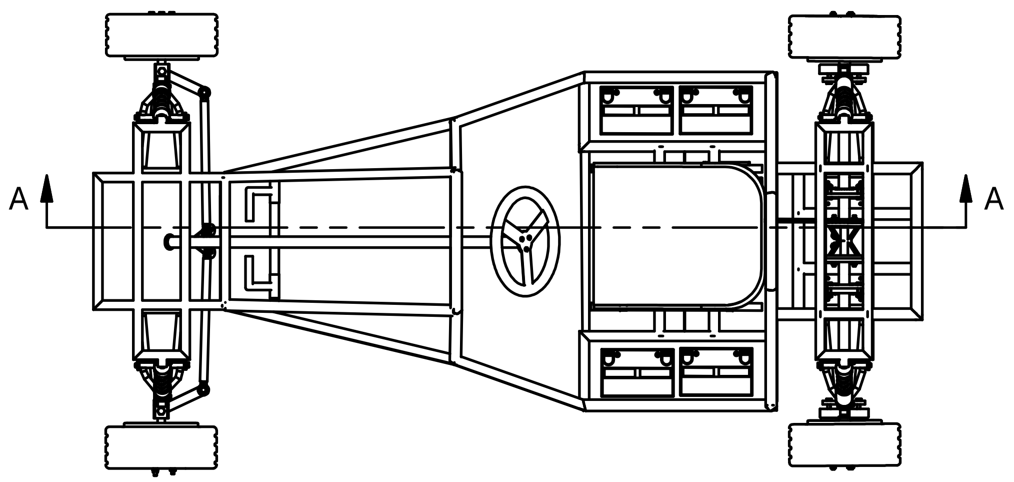

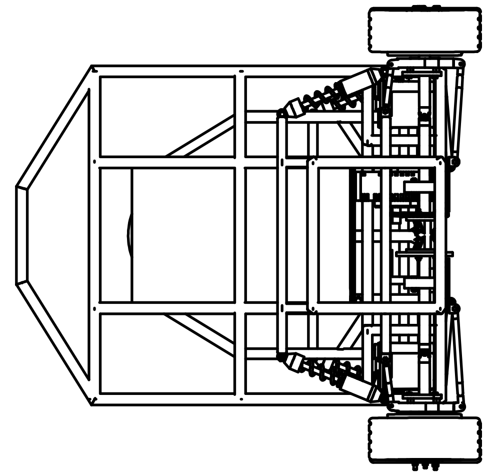

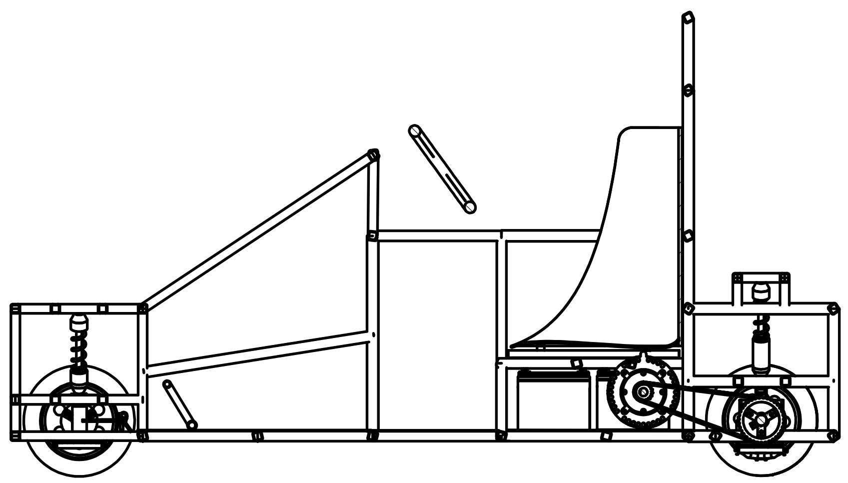

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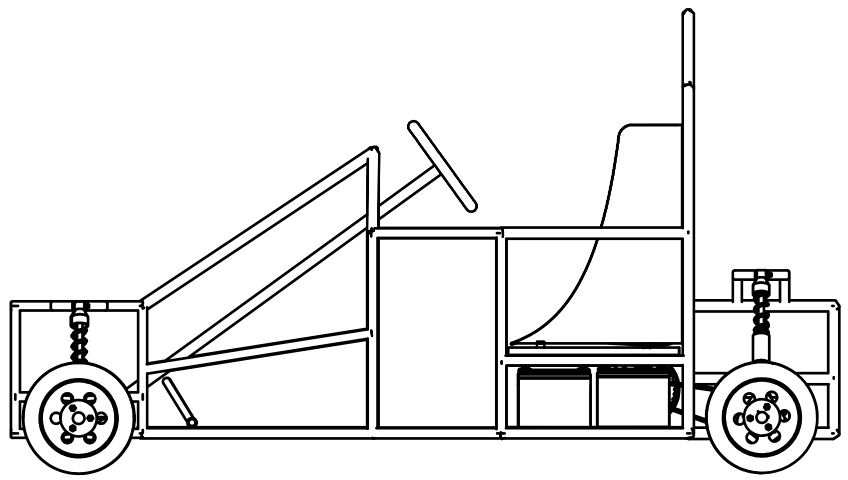

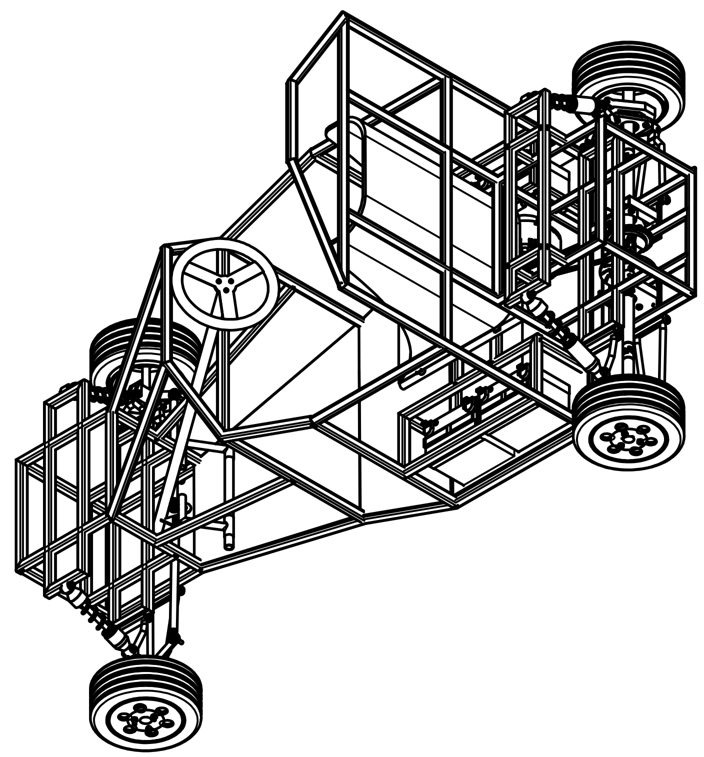

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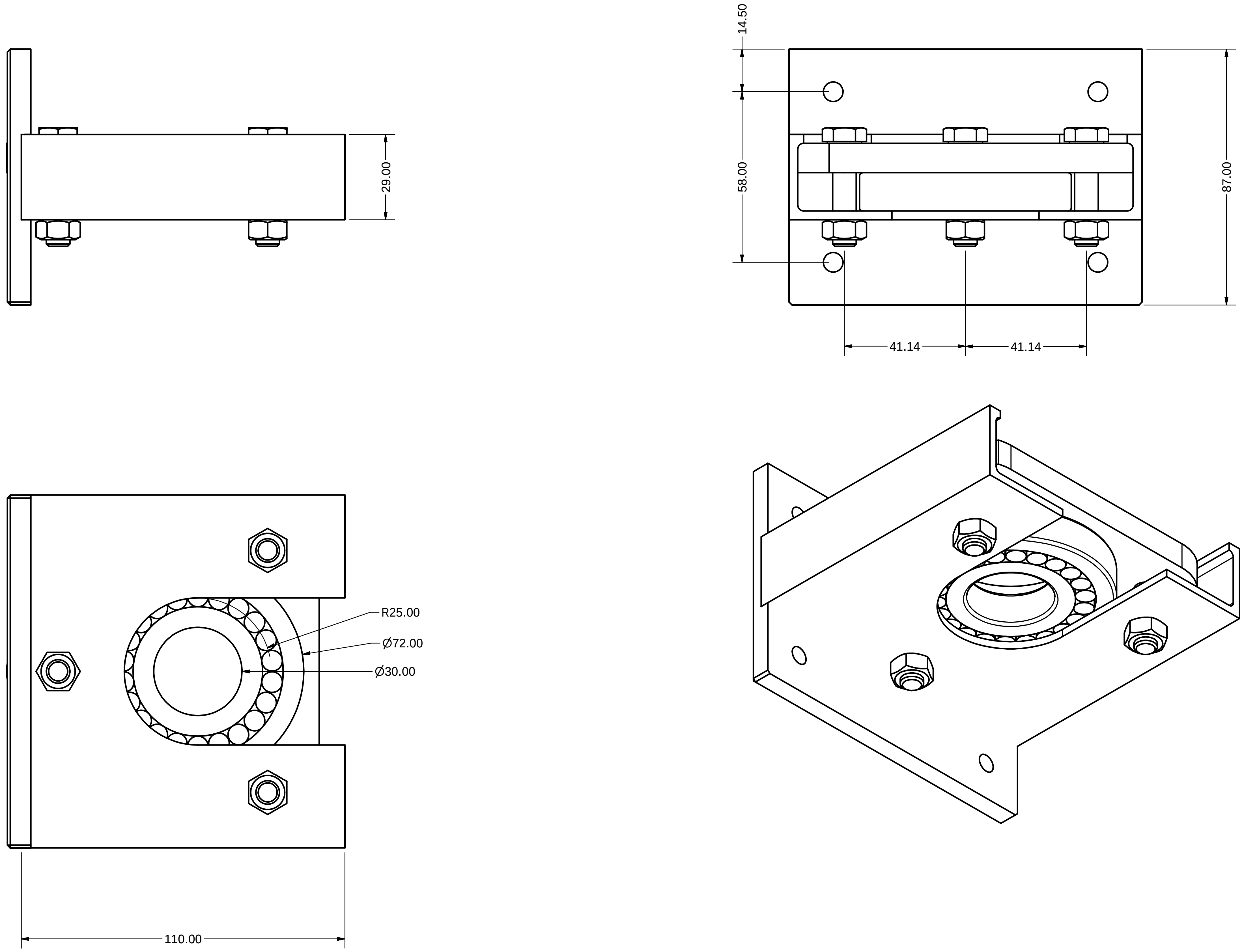

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MFG		
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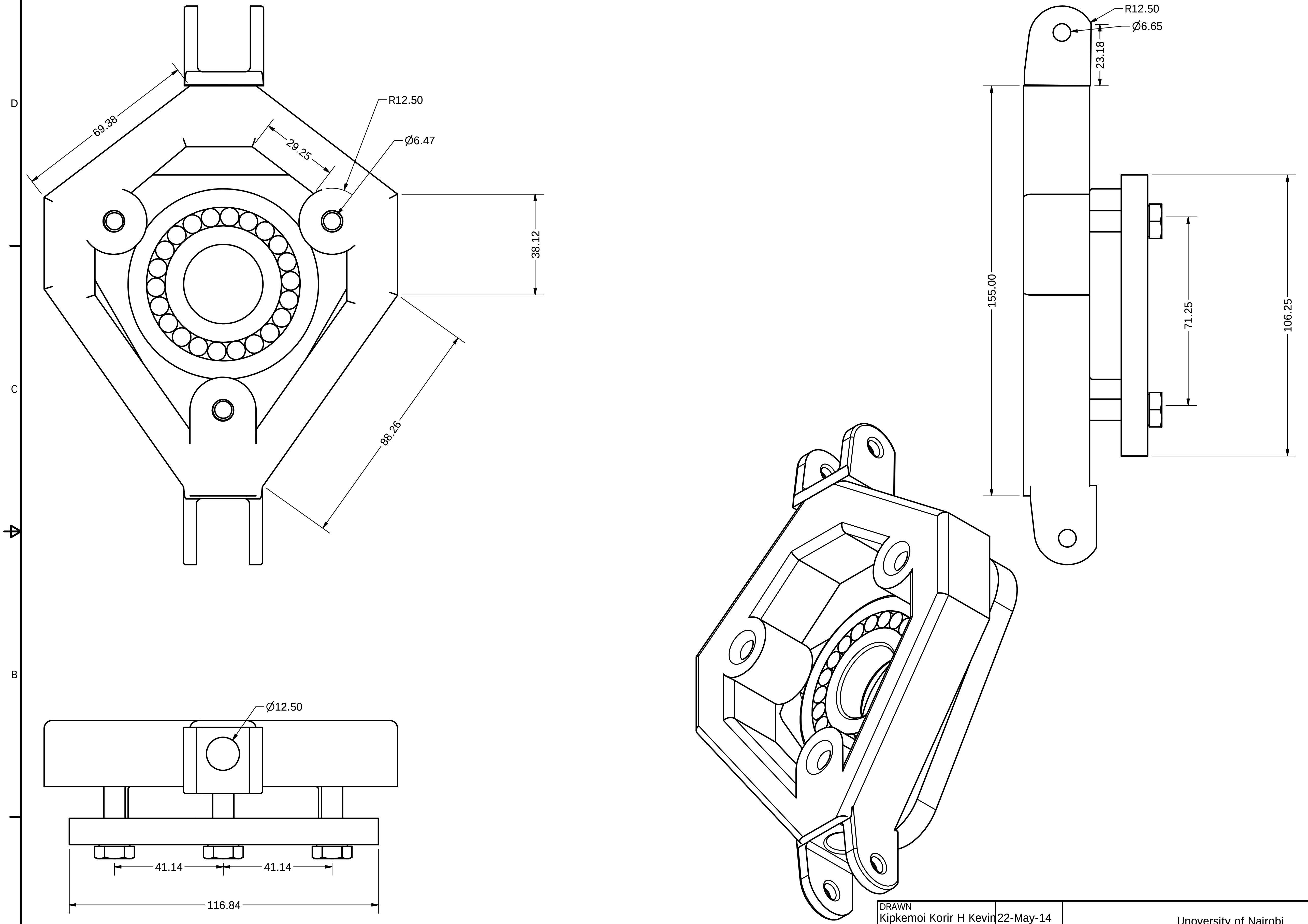

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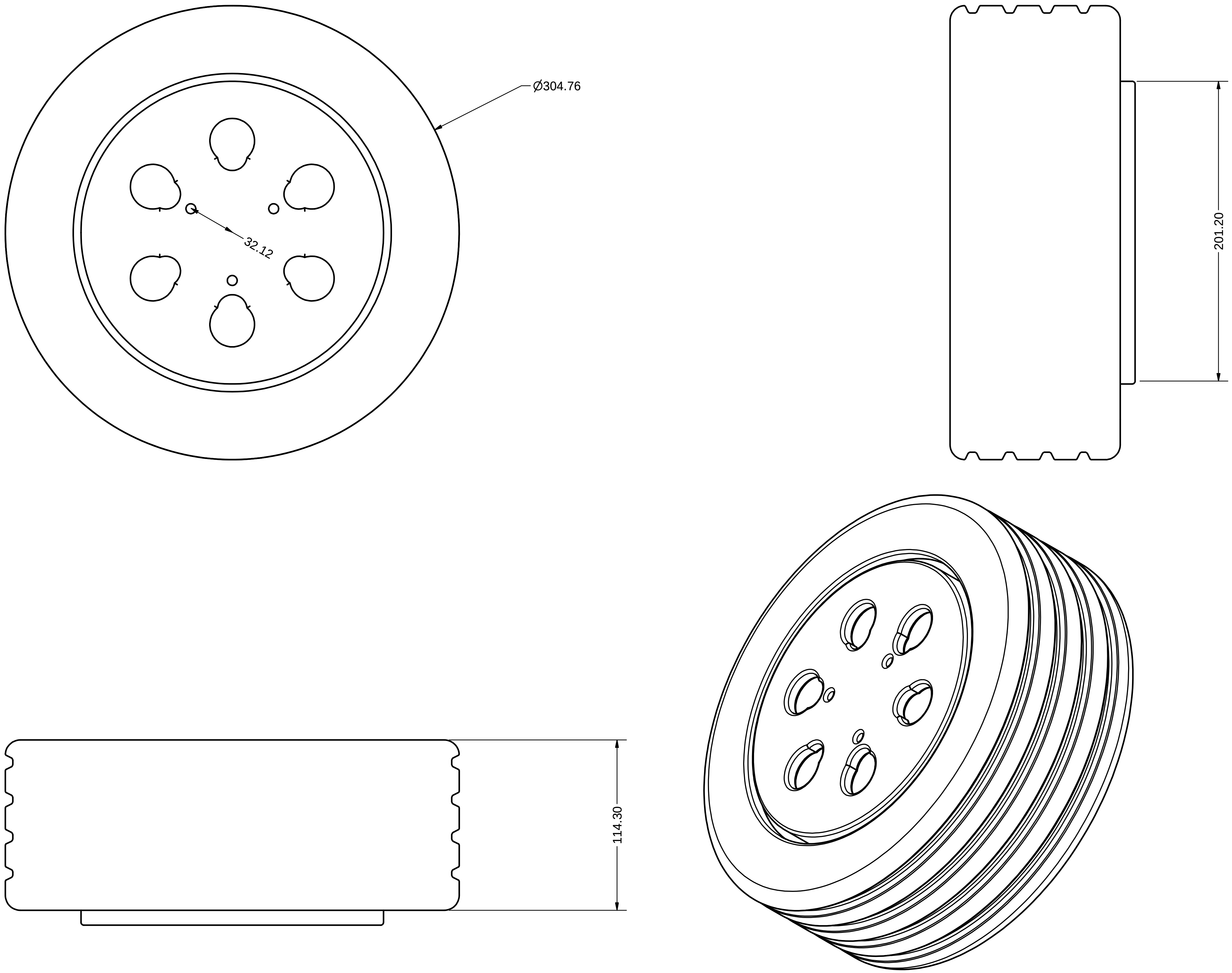

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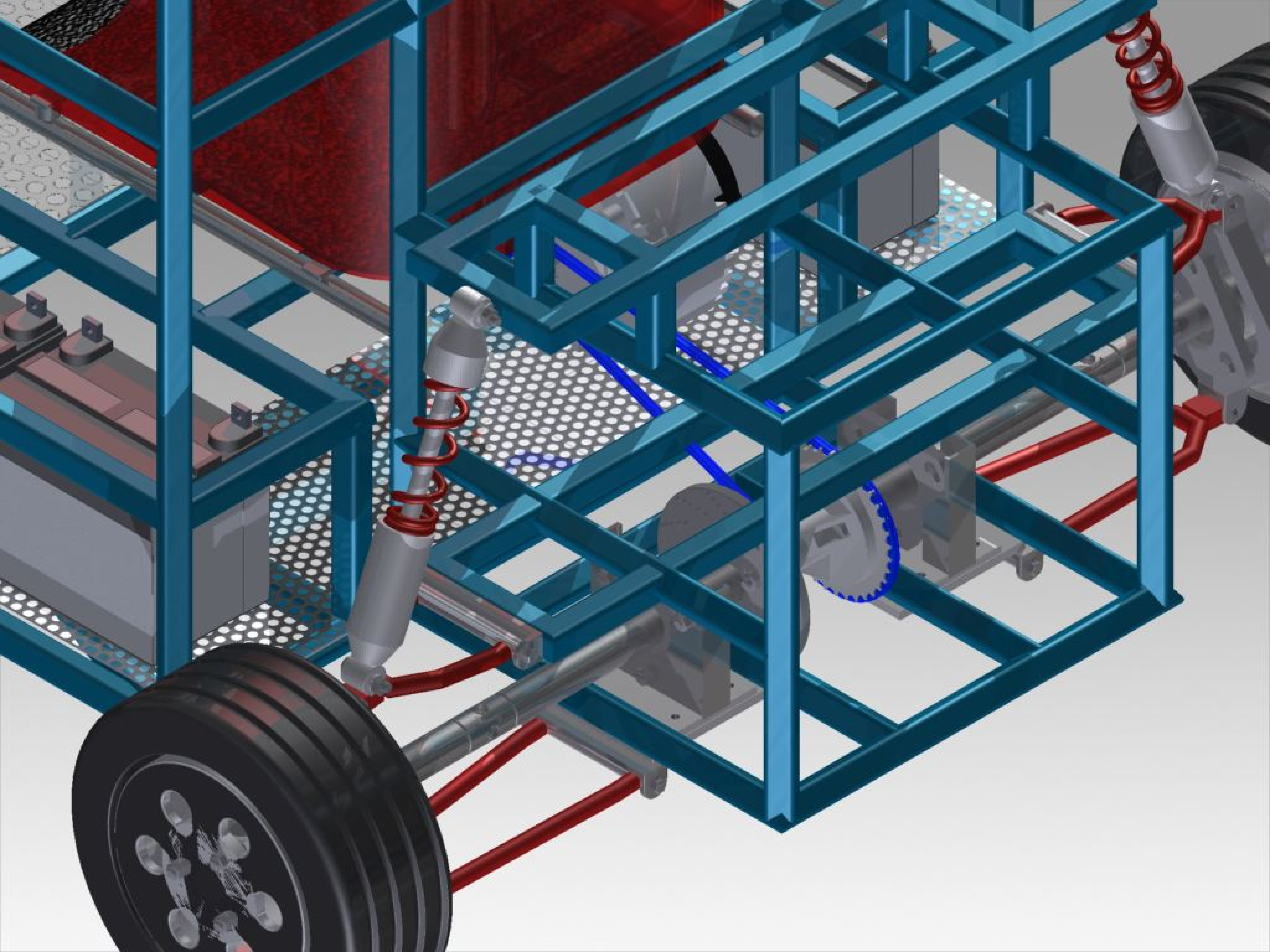
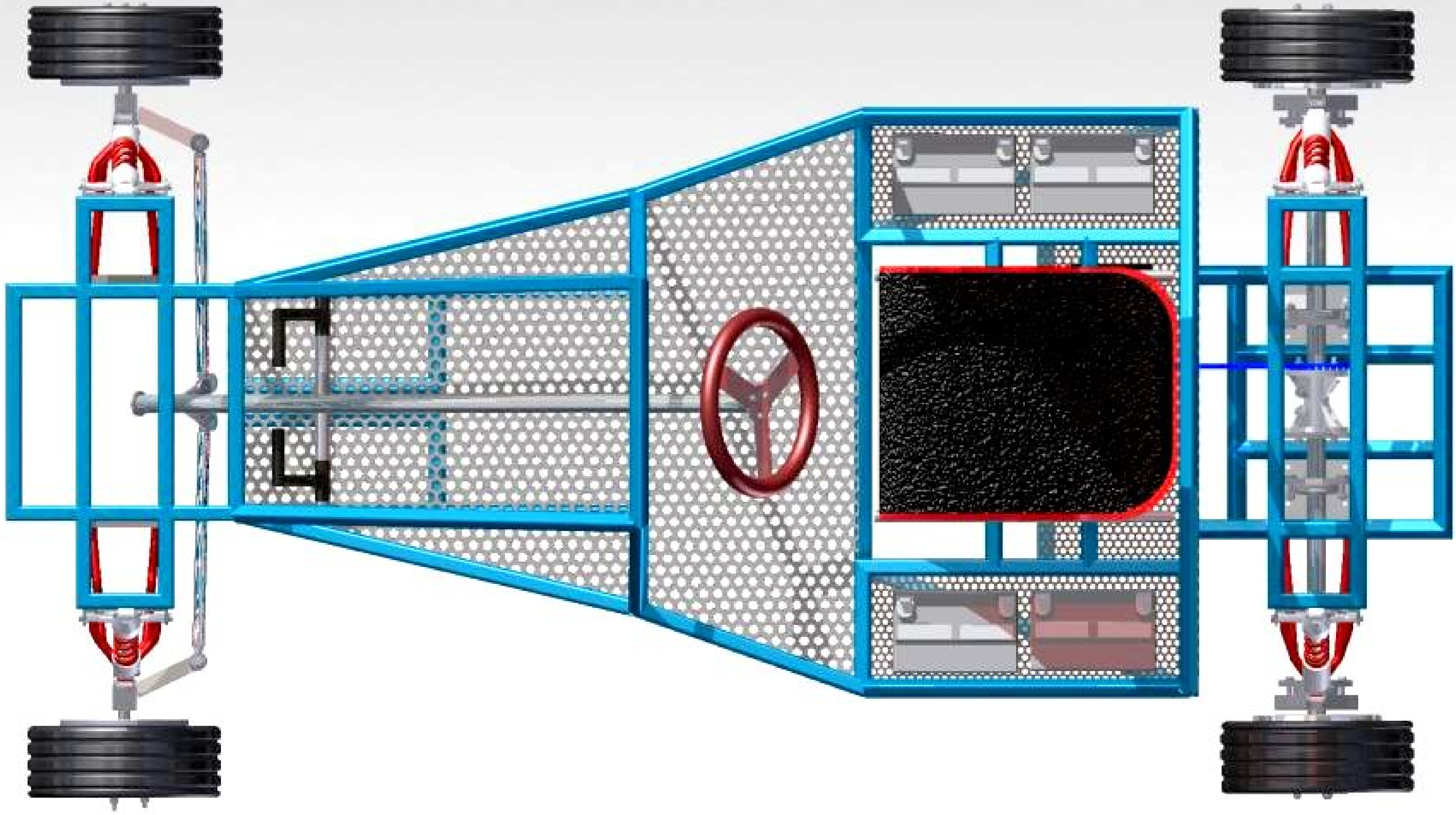
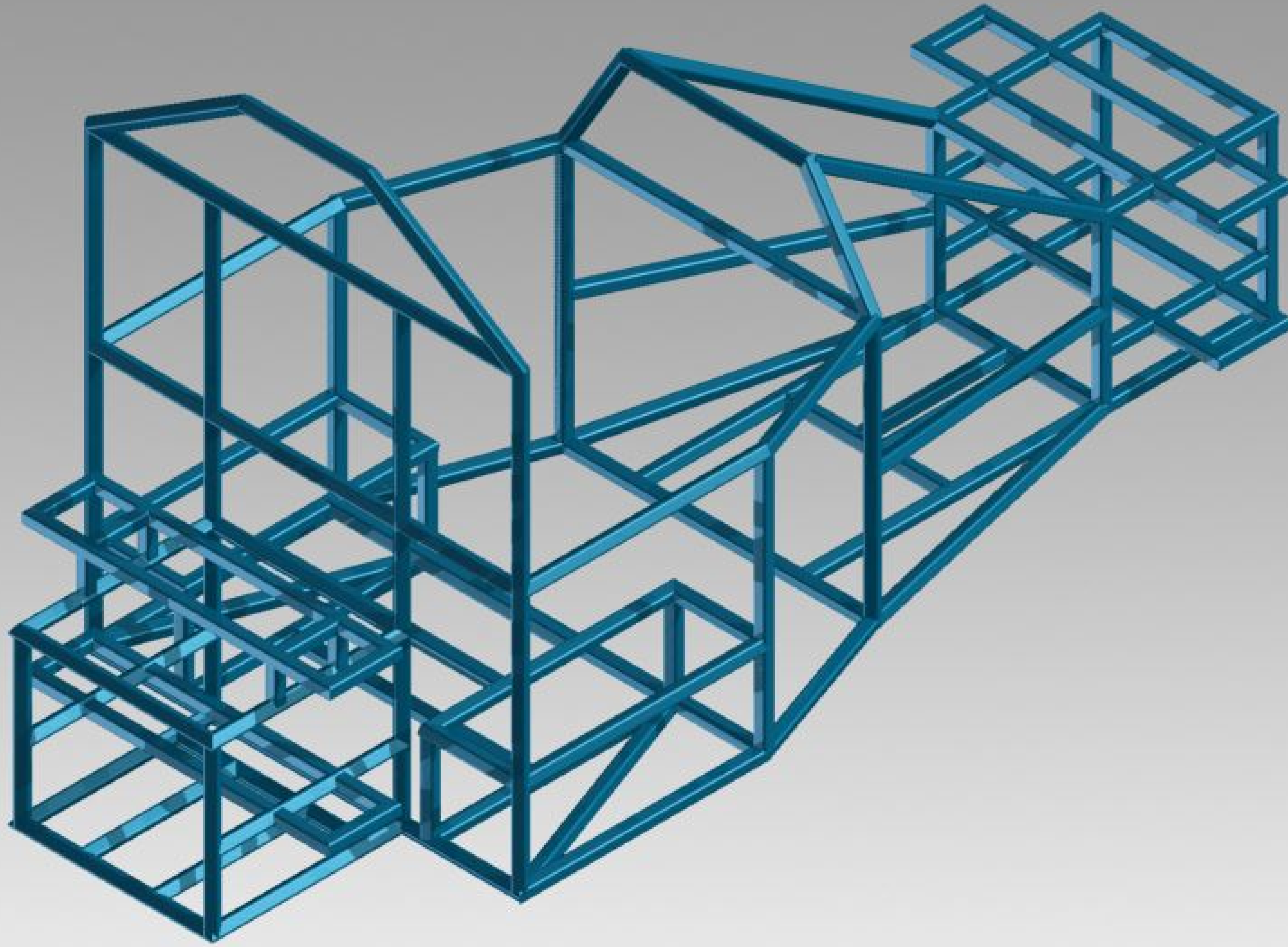
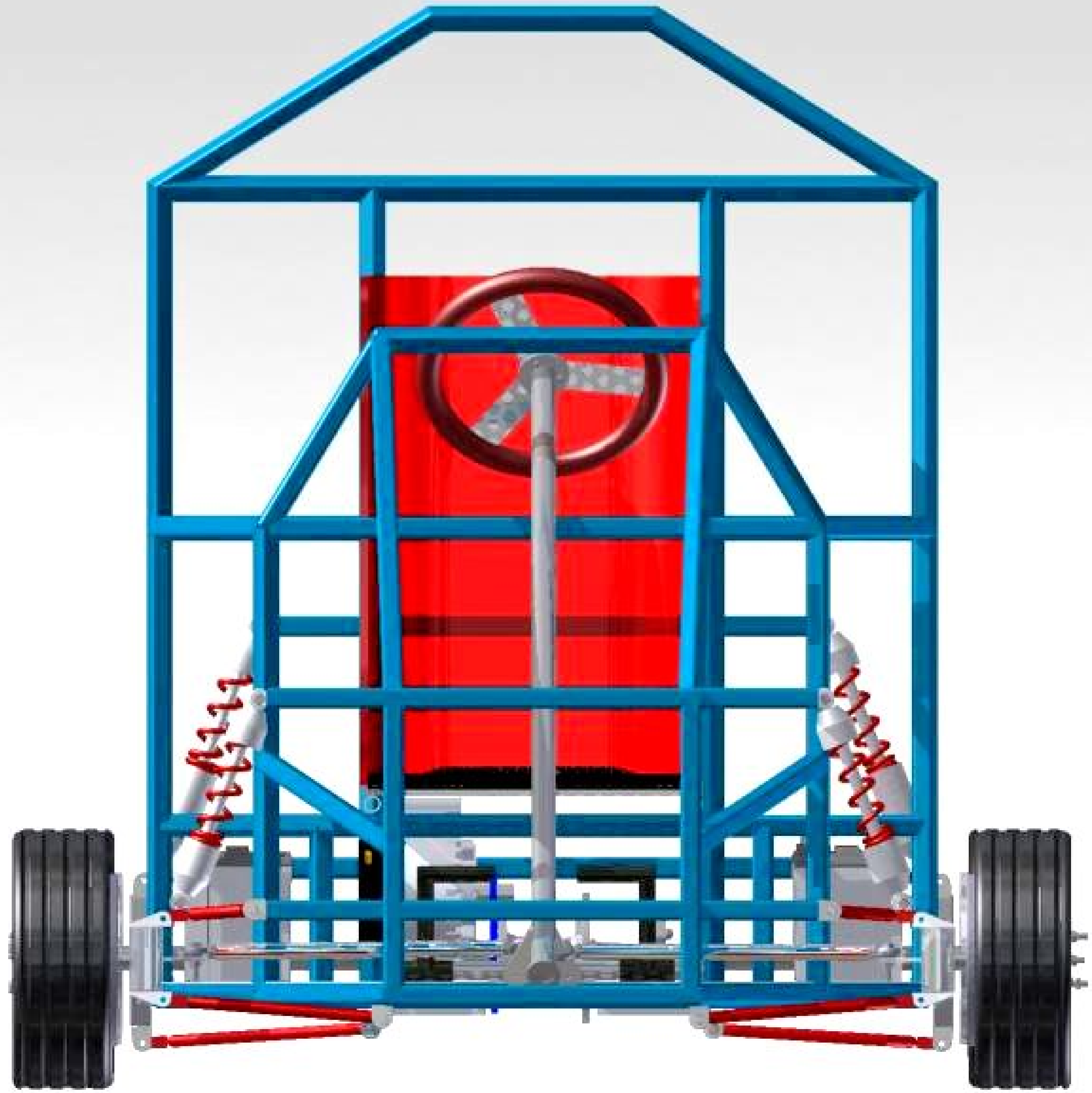
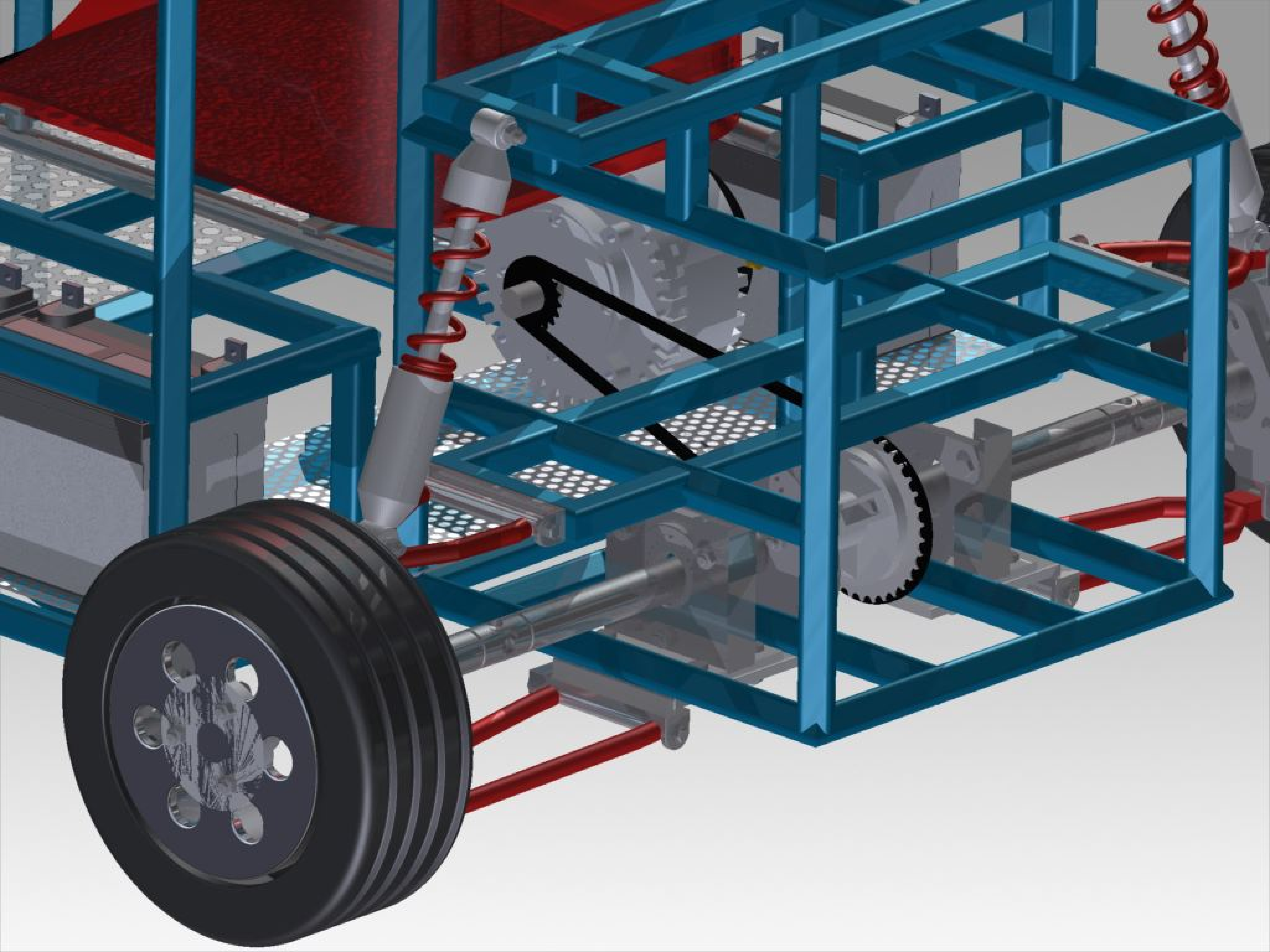
QA			
MFG			
APPROVED		SIZE C	DWG NO Tuk-Tuk Wheel	REV
DIMENSION mm		SCALE 1:5	SHEET 1 OF 1

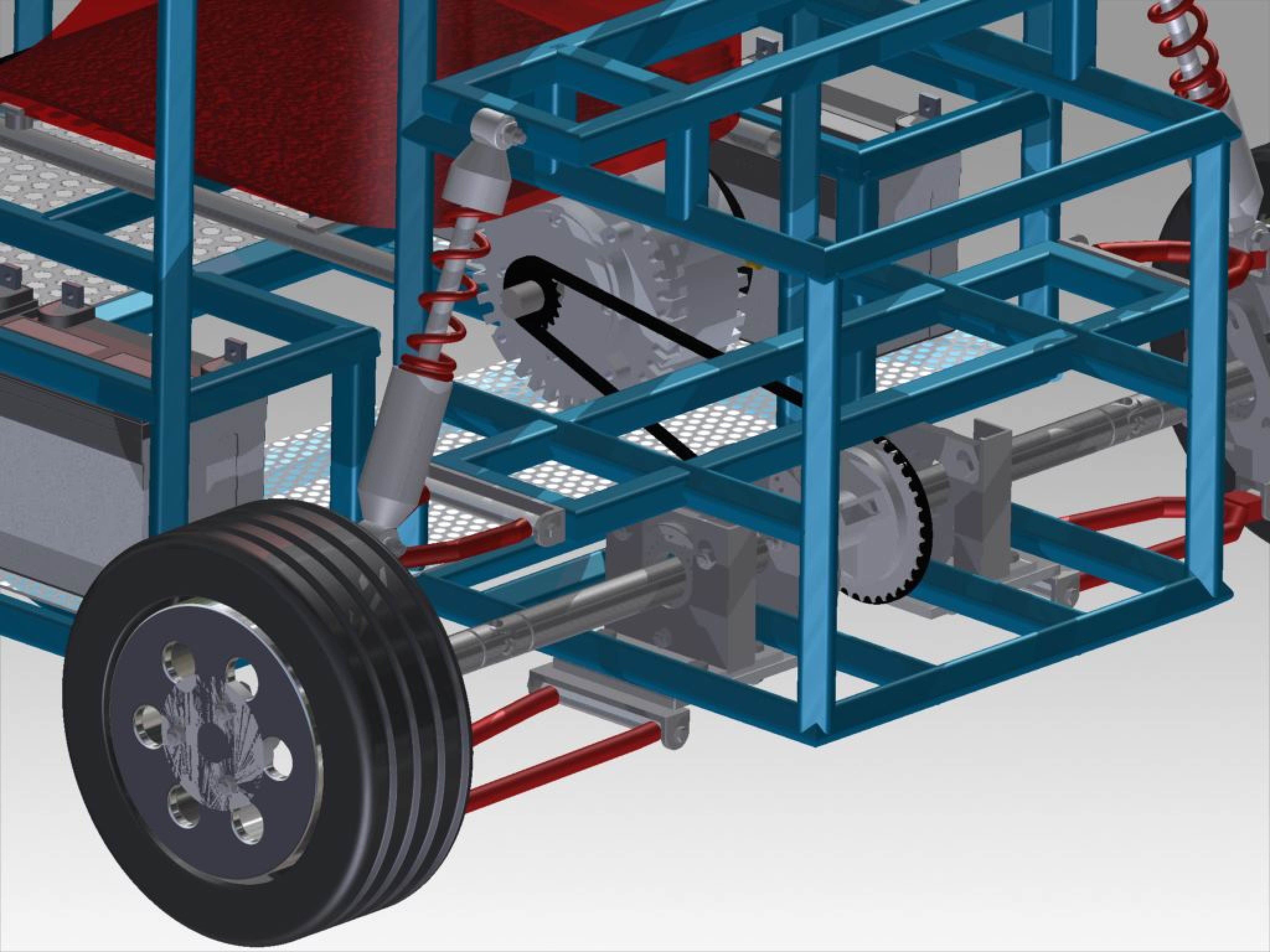


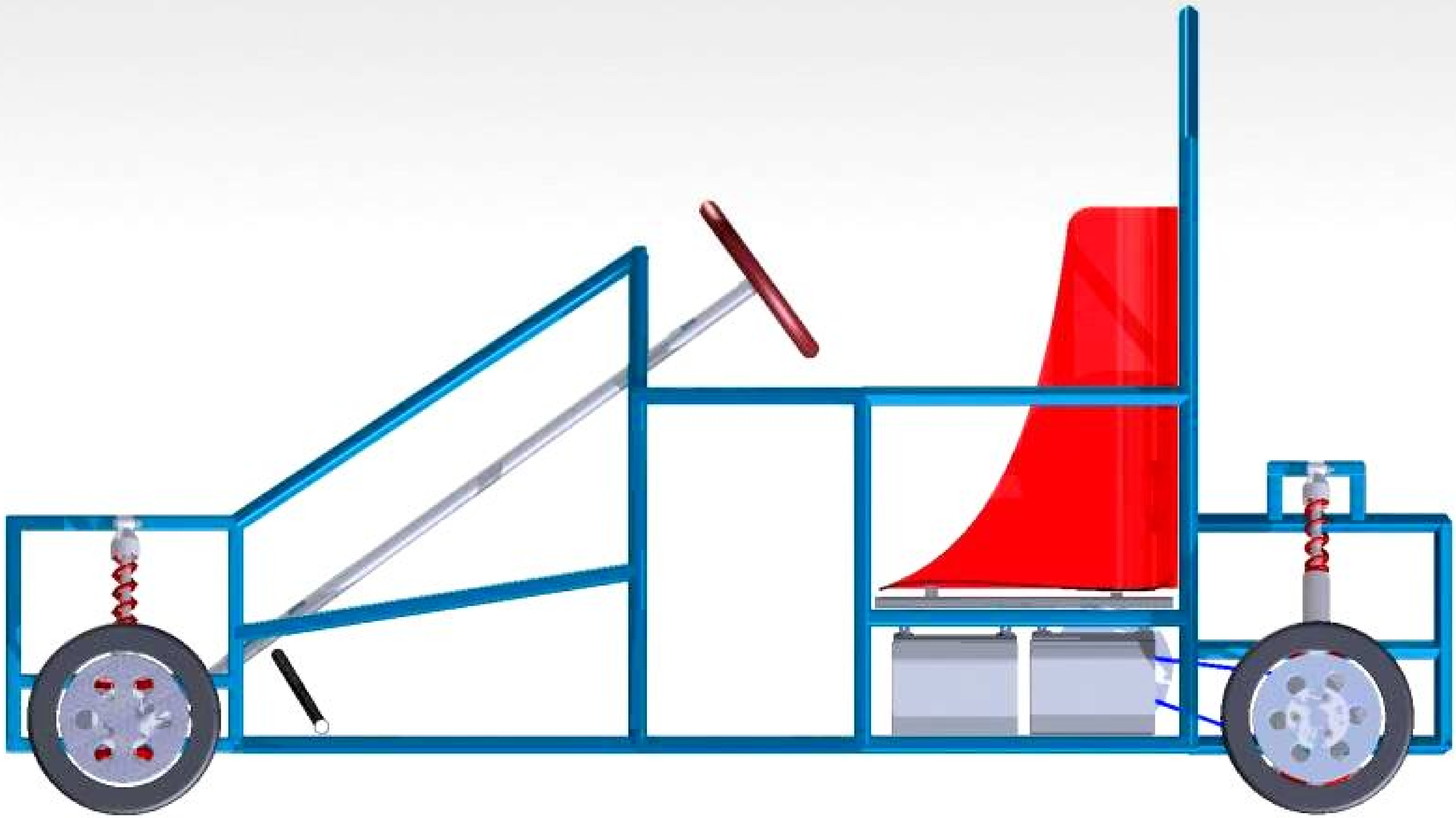


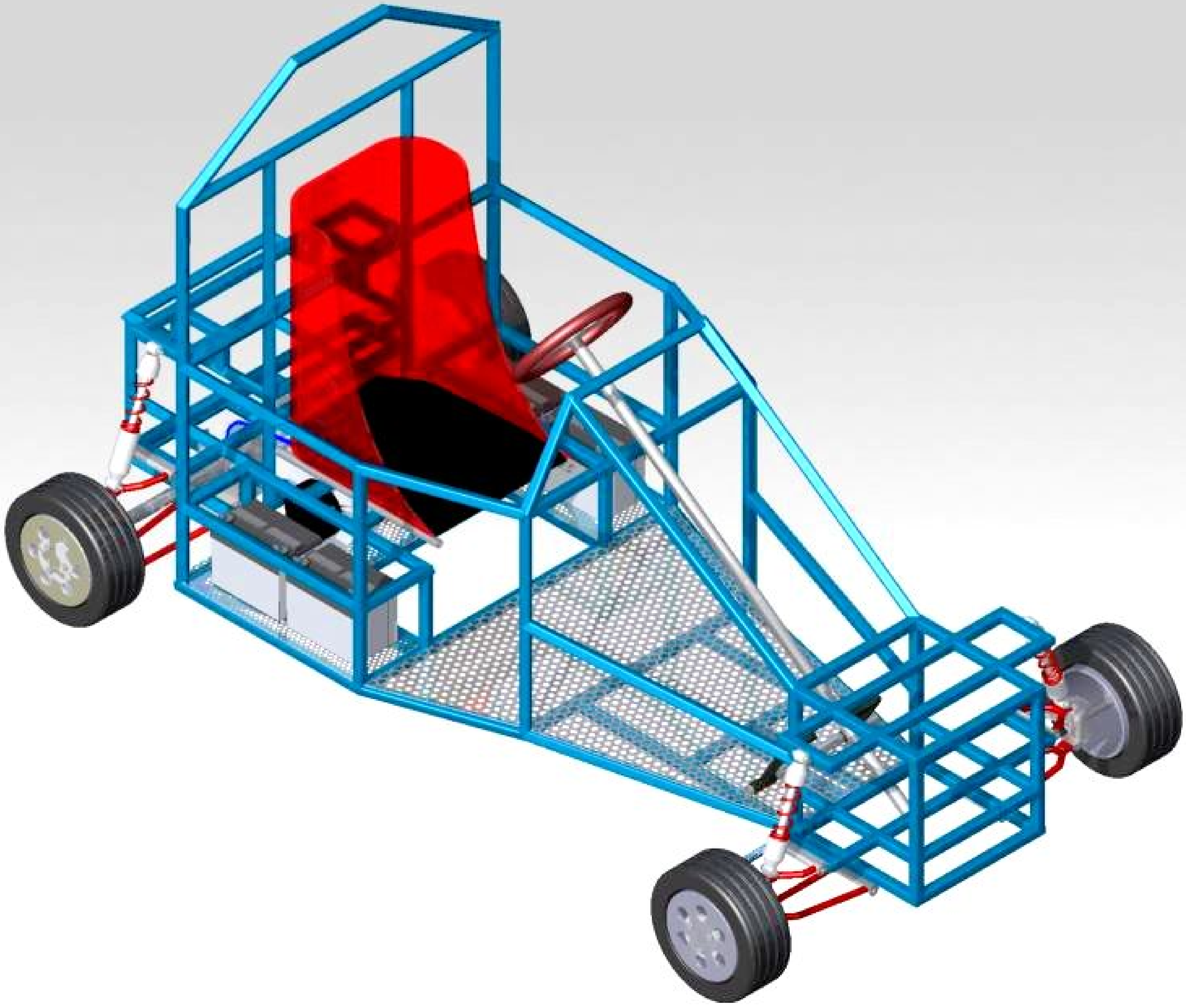


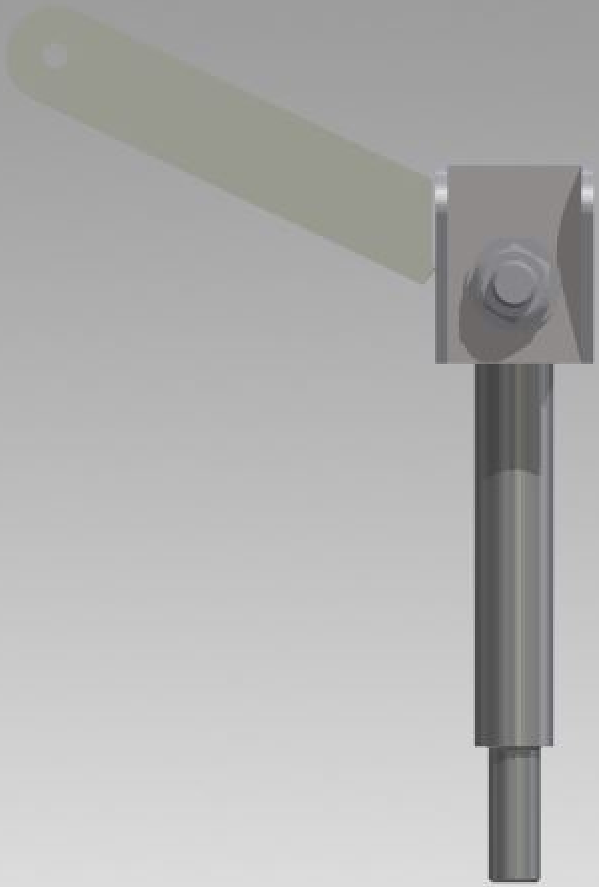


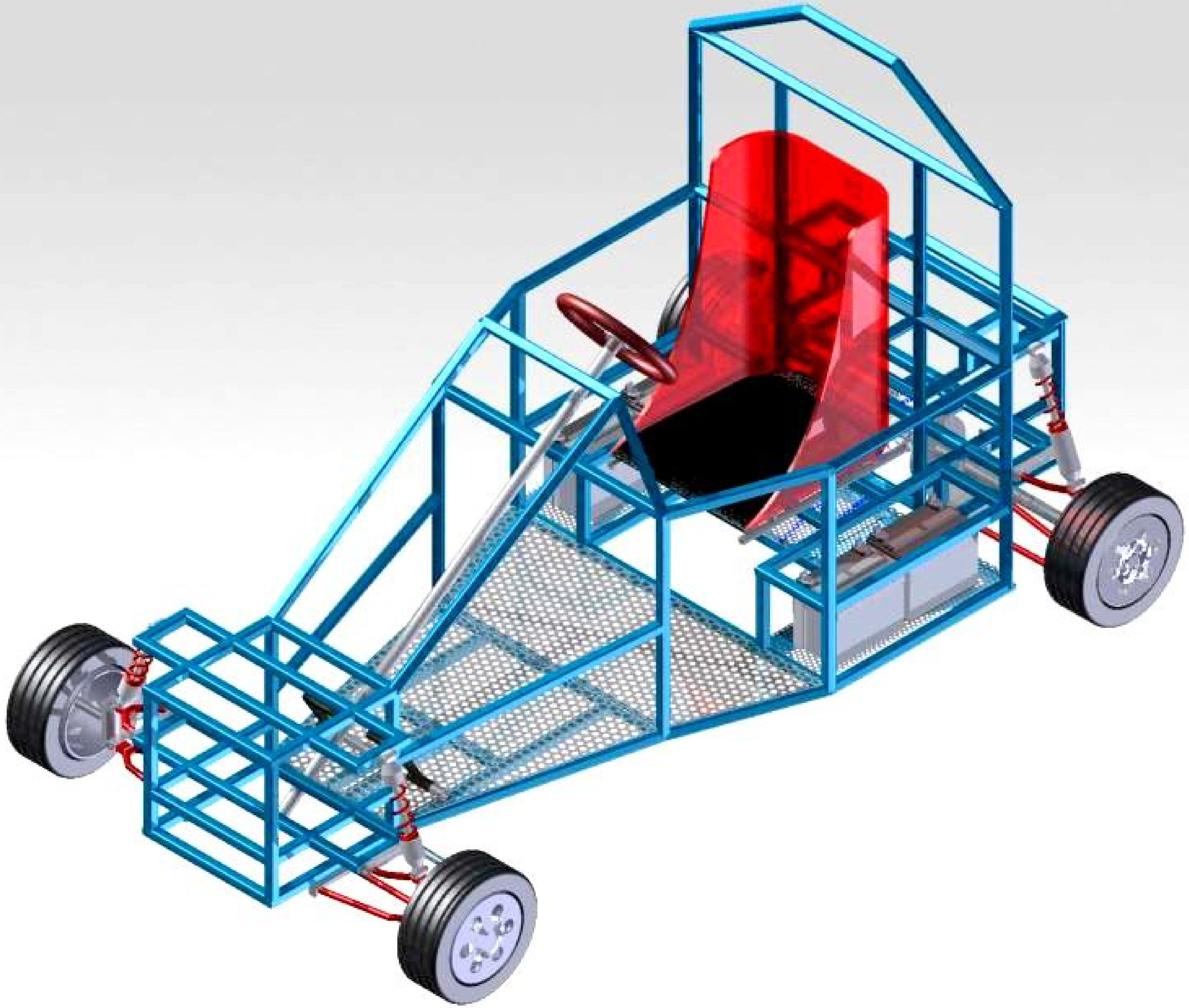










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