IARJSET

ISSN (Online) 2393-8021
ISSN (Print) 2394-1588

International Advanced Research Journal in Science, Engineering and Technology

Vol. 6, Issue 1, January 2019

# A Recumbent Trike Design with Maximum Performance and Vehicle Dynamics Analysis

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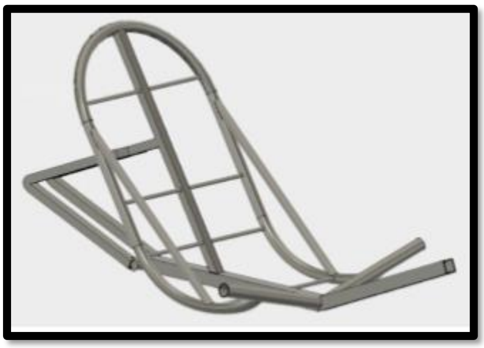
**Abstract:** The vehicle is designed based on an International Trike competition called ASME HPVC. The vehicle was designed to excel in speed, handling, efficiency, practicality and safety as these aspects will be tested in the competition. The team has also put in a lot of effort in the innovative side of the design as we strive to create a unique design that set us apart from other competitors. The design of the vehicle consists of background research, concept generation and analysis and testing to ensure the best possible design. To simplify the design process, the team was divided into five subsections with a single team member in charge of each section. These sections include frame, fairing, steering, braking and drive train. For each subsection, design concepts were generated and evaluated to make the best selection. The team also consists of a subsection dedicated only for innovation which creates inventive technology to improve the design of our vehicle. The vehicle is a front faired tadpole trike with a lightweight aluminum frame constructed from 6061-T6 tubing having a total weight of 25 kg. The front faring creates the perfect aerodynamic structure. Vehicle is designed to be safe and rider is protected by a rollover protection system (RPS) which is designed to meet the ASME HPVC requirements with a minimum safety factor of 1.9.

**Keywords:** Recumbent Trike, Ackermann Angle, Tadpole trike, Vehicle testing, Roll-over protecting system

## I. INTRODUCTION

The objective is to design and fabricate a Human Powered Vehicle which has the attributes are to design a vehicle with optimum performance, to reach a speed of 55+ kmph and to gain 0-40kmph in 14sec. recumbent design with a low centre of gravity with emphasis on safety of the rider and ergonomics. The motive of an HPV is a safe, ergonomic and an efficient replacement for an automobile. An efficient HPV would give maximum output with minimum effort from the rider. There are many different types of HPV's, but the research is leaned toward trikes which have proved to be both ergonomic due to their recumbence and are also performance oriented. We have observed that the uncommonly high speed imparted to recumbent human powered vehicles is due to their low centre of gravity, aerodynamic shape and a fairing which reduces drag. A rollover protection system and the fairing would provide the safety that generally lacks in normal bicycles. The choice of Tadpole trike has been done after research and by examining various reports. From there, it has been involved in the improvement of this design and fabrication process by critical evaluation of aspects like safety, endurance, manoeuvring and comfort of our HPV. Apart from the Tadpole configuration and direct knuckle steering, this HPV is entirely a new design and fabricated in this academic year (2018-19). There has been no reuse of any components from other previous HPVs

## II. PRIOR DESIGN WORK



Trial 1

IARJSET

ISSN (Online) 2393-8021
ISSN (Print) 2394-1588

International Advanced Research Journal in Science, Engineering and Technology

Vol. 6, Issue 1, January 2019

Firstly, considering the design of the frame, referring to the figures we can observe that the current design is lower lined, and the recumbent angle has been increased to 130o. The centre of gravity as noticed is considerably lower which prevents roll over. A bent X-Member has been incorporated to reduce stress concentration and increase load bearing property of the frame. The use of only front fairing as opposed to the full fairing used in last year's vehicle. Front fairing reduces aerodynamic drag and also reduces cost of production. Overall design is improved in design and fabrication aspects to optimize performance of the trike.

Final Design

Conceptual Development and Selection method For the concept development of our HPV, the whole vehicle was divided into subsystems and analysed individually. The selection of each aspect of our HPV was done after thorough research and analysis. Several studies were done to conclude on the final design aspects of our vehicle.

--
++ ++
-- ++ -
++ + + -
-- ++ - --
-- -- + + ++ +
+ + + -

Positive = ++
Negative = --
Blank = no appreciable relation

| Customer | Engineering Requirements | | | | | | | | | Competition | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Needs | Importance | Speed | HPV Geomety | Turning radius | Drag | Frame strength | Braking distance (6m) | High speed stability | Low speed stability | 2017 | Target |
| Performance | 10 | **** | **** | **** | **** | **** | **** | **** | **** | *** | **** |
| Material | 8 | *** | **** | * | ** | **** | - | - | - | ** | *** |
| Weight | 7 | **** | - | - | - | **** | **** | ** | ** | * | **** |
| Shape | 5 | * | **** | - | **** | *** | - | - | - | *** | *** |
| Cost | 10 | *** | - | - | **** | **** | *** | *** | * | **** | *** |
| Safety | 10 | *** | ** | - | - | **** | ** | **** | *** | **** | **** |
| Ergonomics | 3 | - | **** | ** | ** | - | - | *** | * | *** | ** |
| Aerodynamics | 6 | **** | *** | - | **** | ** | - | ** | - | * | *** |
| Maintainence | 4 | - | *** | - | * | **** | * | - | - | ** | ** |

| Competition | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 2017 | | 25km/hr | | 7.2m | | | 5.7m | | | | |
| Target (2018) | | 50km/hr | | 4.8m | | | 5m | | | | |

High importance =****

Low importance=*

IARJSET

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The chart below shows the design considerations and the requirements which were mapped out to compare and prioritise aspects to meet design specifications. The factors relevant to the design and construction of the trike were specified and a product design was accomplished accordingly. The design specifications have been mentioned in the following section. The matrices below have been considered to decide the type of frame and the steering configuration of the trike using an efficient marking system and a decision matrix which comprises of the aspects on which both the configurations depend on.

| | Overseat steering | Underseat steering | Direct Knuckle steering |
|---|---|---|---|
| Ease of operation | 4 | 3 | 5 |
| Ease of build | 2 | 2 | 4 |
| Low cost | 4 | 3 | 4 |
| Accomodate with fairing | 3 | 5 | 3 |
| Comfortable | 4 | 3 | 4 |
| Sturdy | 4 | 2 | 5 |
| Safety | 4 | 4 | 4 |
| Total | 25 | 22 | 29 |

Decision matrix for steering

## III. DESIGN DESCRIPTION

1. Frame Design

The frame is of a tadpole tricycle mainframe with the required RPS for safety. The frame must have optimum strength and minimum points of stress concentration hence uniform stress distribution across the frame. The fabrication of the frame involved tube bending, welding etc. The fame material chosen is Aluminium 6061 T6. The frame includes the following component as shown below. With a rectangular or square cross section of the tubes it is possible to obtain a much higher moment of inertia and polar moment of inertia in a specific plane which will result in a greater resistance to both torsion and bending deflections. With the square flat surfaces, this design will allow for simplified seat integration and manufacturing. The angle of recumbence is 130o. The wheel base of the giver design is 1.916m; the model caster is 14o, the wheel track 1.20m, the ground clearance 125mm, the center of gravity without the rider is located 1.06m from the front and 383mm from the ground. The weight distribution is 66% on the front wheels and 33% on the rear wheel. All these features contribute towards a performance oriented ergonomic design with an optimum safety factor.

| | Fame configuration | | |
|---|---|---|---|
| | Tadpole | Delta | 2 wheel |
| Speed | 0 | 1 | 2 |
| Drag | -1 | 0 | 1 |
| Weight | 1 | 0 | 2 |
| Reliability | 2 | 2 | -2 |
| Maneuverability | 0 | 0 | 1 |
| Ease of entry or exit | 1 | 1 | -3 |
| Ergonomics | 1 | 1 | 0 |
| Portability | 0 | 0 | 0 |
| Manufacturability | 1 | 1 | 2 |
| Repairability | 0 | 0 | 1 |
| Price | 1 | 1 | 2 |
| Appearance | 0 | 0 | 0 |
| Safety | 3 | 2 | -2 |
| | 0.85 | 0.825 | 0.4 |

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International Advanced Research Journal in Science, Engineering and Technology

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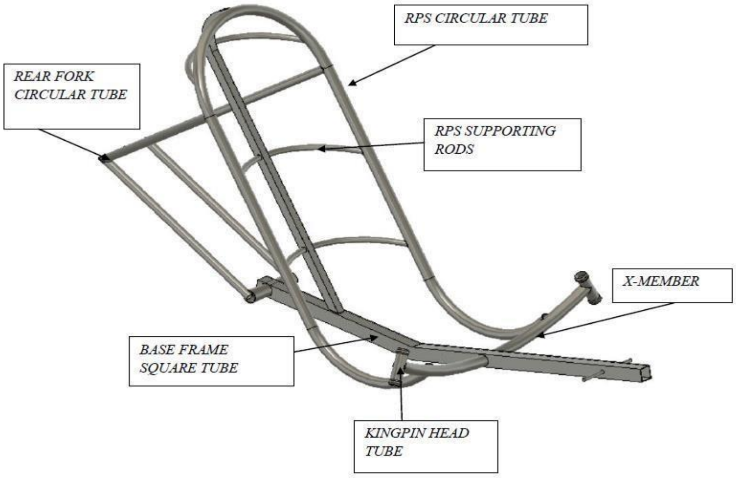
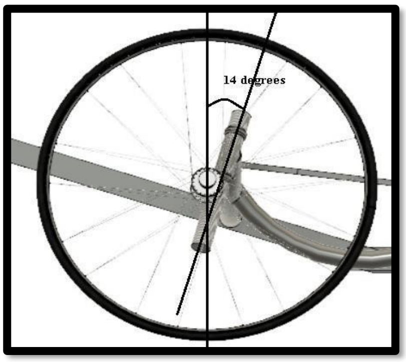



2. Steering

Given a tadpole tricycle design, many steering methods have already been implemented. For example under seat steering, over steering, lean steering, knuckle steering and so on. After thorough research direct knuckle steering was chosen as it is simple to incorporate, effective and can be modified easily as per the driver's needs. This steering system consists of a direct steer handle combination, which also holds the brake and shifter levers. During design of the steering, the important factors considered were the kingpin alignment, camber, caster angle and Ackerman compensation. Negative camber has been considered as it improves stability and uniform load distribution on wheels. Having a drastic negative camber helps keep more of the force in the vertical axis of the wheel during turns when there is maximum side loading on the wheels. The negative camber angle considered is 107.5o . To apply proper Ackerman compensation the pivot brackets connecting the rear wheel were aligned to point towards the centre axle of the rear wheel, as shown. This helps reduce the effects of tire rubbing during cornering. Considering all the features, toe in configuration has been achieved as shown in Figure




Caster Angle


Kingpin Representation

IARJSET

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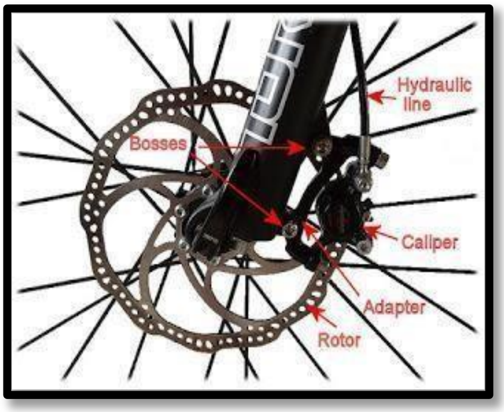
| Kingpin Angle=30° | Caster Angle= 14° |
|---|---|
| This is the angle of the pivot axis from vertical viewing from the front. With the geometry given, the kingpin angle becomes 30° degrees to achieve center point turning. Center point steering is desirable because it allows for more precise and efficient steering | This angle is the Kingpin plane relationship to the wheel contacting the road. A caster of 14° has been used for our trike which also contributed to the toe in configuration of the trike. |

3. Braking

Braking mechanism includes application of friction or resistance to a turning wheel causing it to slow down and eventually stop, creating heat as a by-product. The type of brake used is chosen by comparing the properties of the following 2 types of brakes. According to the comparative study, the disc brakes were chosen for our design. Brakes are applied to all the three wheels of the trike. The brakes of the two front wheels have been combined and connected to a single brake wire to simplify usage of brakes and providing easy access to the rider while braking.

| Disc brakes | Drum brakes |
|---|---|
| Flat friction pad<br>Lighter in weight<br>Rapid heat dissipation<br>More efficient<br>Effective even during repeated application | Semicircular friction pad<br>Heavier in weight<br>Slow heat dissipation<br>Less efficient<br>Lose effectiveness due to continuous application |

Difference between Disc and Drum Brake



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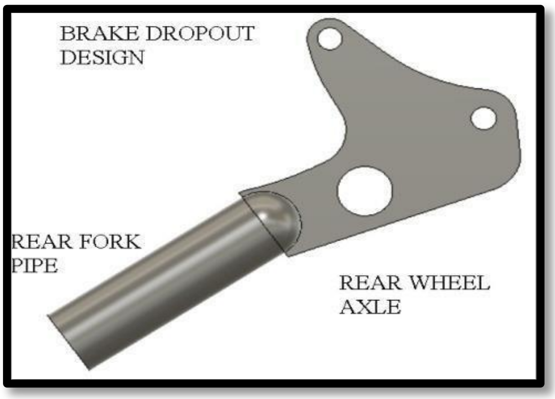
4. Wheels and Drivetrain

Wheel size is an important factor constituting design of the trike. It contributes to the aspects of top load distribution and also the speed of the trike. Large wheels decrease rolling resistances and smaller wheels have less aerodynamic drag at high speeds and will allow for quicker acceleration. Thus, the rear wheel size was decided to be 28 inches and the front wheel size is to be 20 inches. The gear cassette is mounted on the rear wheel.

Cassette: Cassette is a cluster of gears which enable change of speeds manually using a derailleur assembly. The rear cassette used has 7 gear sprockets. A derailleur has been incorporated for efficient change of gears during operation of the trike.

Cadence: It is the number of revolutions of crank per minute. The cadence is directly proportional to the wheel speed and changes with the number of gears used. A high cadence was achieved by a gear combination of 7*1 to reduce effort of the rider and slow twitch muscle recruitment. After taking in to consideration the chain




32 Tooth Chainring 32

Shimano 12-14-16-18-21-24-28

| | |
|---|---|
| Max ratio | 2.74 |
| Min ratio | 1.18 |

| | | 32 |
|---|---|---|
| 12 | | 2.74 |
| | 17% | |
| 14 | | 2.35 |
| | 14% | |
| 16 | | 2.06 |
| | 13% | |
| 18 | | 1.83 |
| | 17% | |
| 21 | | 1.57 |
| | 14% | |
| 24 | | 1.37 |
| | 17% | |
| 28 | | 1.18 |

Speed (km/h)

| Cadence: | 60 | 80 | 100 | 120 |
|---|---|---|---|---|
| Highest gear | 20.6 | 27.5 | 34.4 | 41.3 |
| Lowest gear | 8.9 | 11.8 | 14.8 | 17.8 |

ring and cog specifications the following gear ratio was obtained for the combination. Single chain is used in combination with idler gears to transfer rider power to rear wheel. Idler gears help define the chain path. This is simple to design and manufacture. Chain tubing can be used to prevent the slack of chain and reduce the numberof idler gears. The chain routing shows the path followed by the chain during operation. It is depicted in the above figure.

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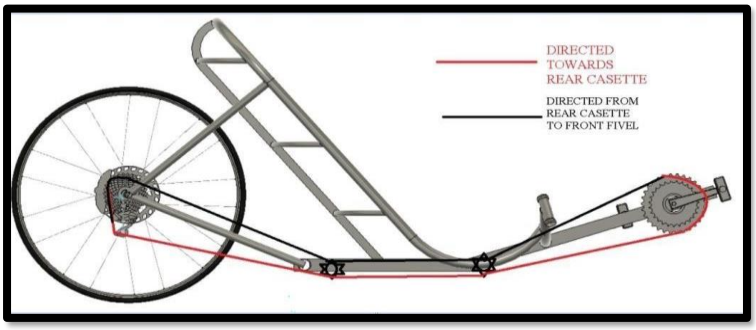
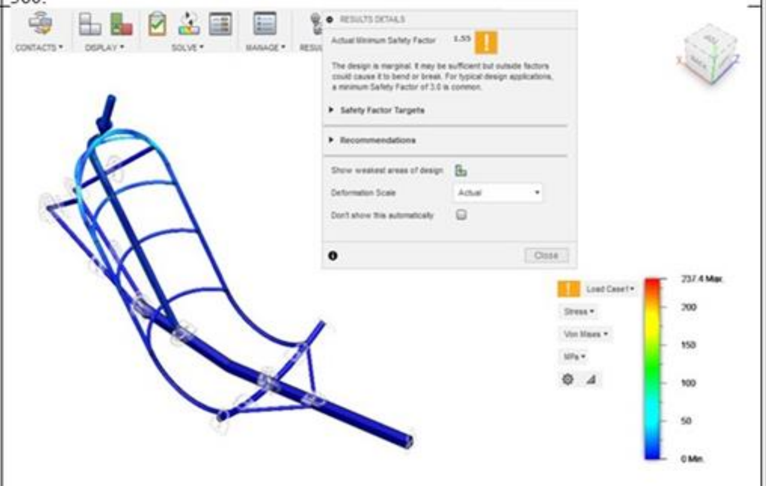
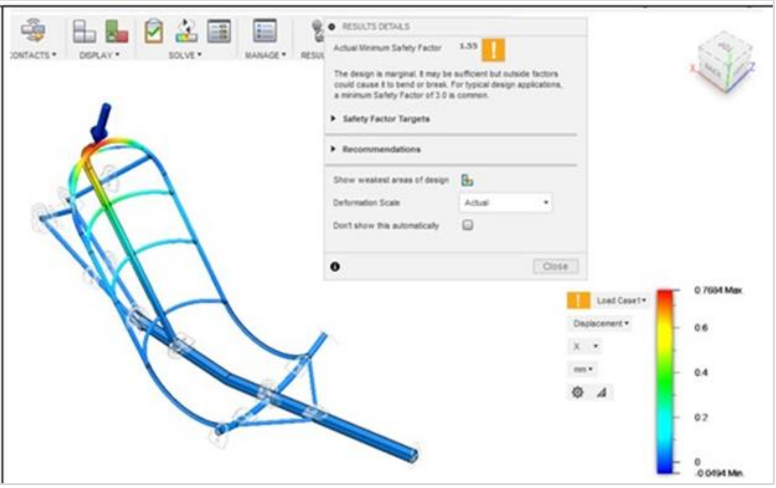



Analysis

1. RPS-

| | |
|---|---|
| **I. Objective:** | The objective of the analysis is to make sure of the safety of the rider during a roll over situation of the trike keeping in mind of the HPVC ASME requirements and Condition. |
| **II. Modelling Method:** | The whole design of the tadpole trike has been conceptualized and designed using theoretical engineering concepts from subjects such as Design of Machine Elements, Mechanics of Materials, Dynamics of Machine, etc. Autodesk Fusion 360 is the software which has been used for carrying out the simulations on the RPS. |
| **III. Case 1: Top Load Analysis**<br>**a. Objective:** | A load of **2670 N** per driver/stoker shall be applied to the top of the roll bar(s), directed downward and aft (towards the rear of the vehicle) at an angle of 12° from the vertical. There should not be any occurrence of plastic deformation and the maximum elastic deformation should be less than 5.1cm and should not deform such that contact with the driver's helmet, head or body will occur. |
| **b. Methods and Assumption:** | The force was applied on the top of the RPS and constraints were applied on the required mainframe parts along with the Rear fork. This force is assumed to have some effect on rear wheel but its enough to bear it. The analysis has been carried out in Fusion 360. |
| **c. Result:**<br>i) Stress: |  |
| i) Deflection and Factor of Safety: |  |


DOI 10.17148/IARJSET.2019.6112

ISSN (Online) 2393-8021
ISSN (Print) 2394-1588

International Advanced Research Journal in Science, Engineering and Technology

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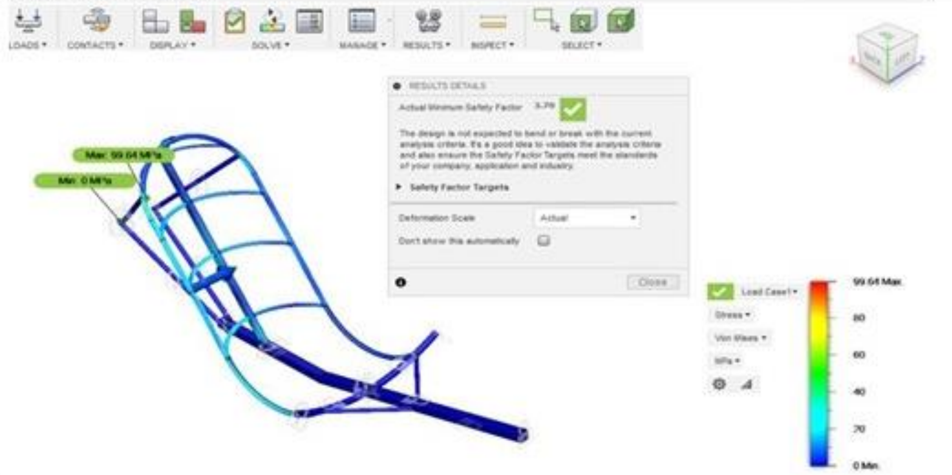
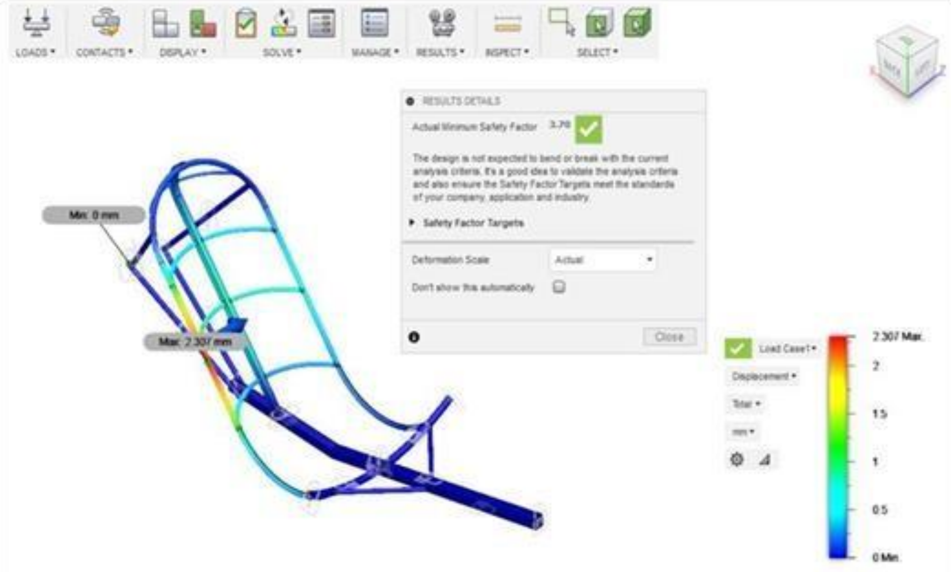
| | |
|---|---|
| **d. Conclusion:** | According to the pipe dimensions of RPS and according to the yield strength of aluminum 6061 T6 i.e. **237.6 MPa** under the top loading condition the RPS does not undergoes permanent or plastic deformation. FOS of **1.55** and maximum deflection of **0.7684 mm** and minimum deflection of **0 mm** Maximum Stress of **164.7MPa** and minimum stress of **0 MPa**. |
| **IV. Case 2: Side Load Analysis** | A load of **1330 N** per driver/stoker shall be applied horizontally to the side of the roll bar at shoulder height, and the reactant force must be applied to the seat belt, seat, or roll bar attachment point and not the other side of the roll bar. There is no indication of permanent deformation, fracture or delamination on either the roll bar or the vehicle frame, and the maximum elastic deformation is less than 3.8 cm and shall not deform such that contact with driver's helmet, head occurs. |
| **a. Objective:** | |
| **b. Methods and assumptions:** | The load was applied on the side of the RPS as shown in the figure and the constraints where applied on the required mainframe parts along with the Rear fork. The main frame was assumed to undergo certain deformation. However, the frame was strong enough to withstand the force. |
| **c. Result:**<br>i) Stress: |  |
| ii) Deflection and Factor of Safety: |  |
| **d. Conclusions:** | The FOS of the design for side load is **3.7**. The maximum deflection experienced by the design is **2.307mm** and Minimum deflection of **0 mm**. The maximum stress experienced by the design is **99.64 MPa** and Minimum Stress of **0Mpa.** |

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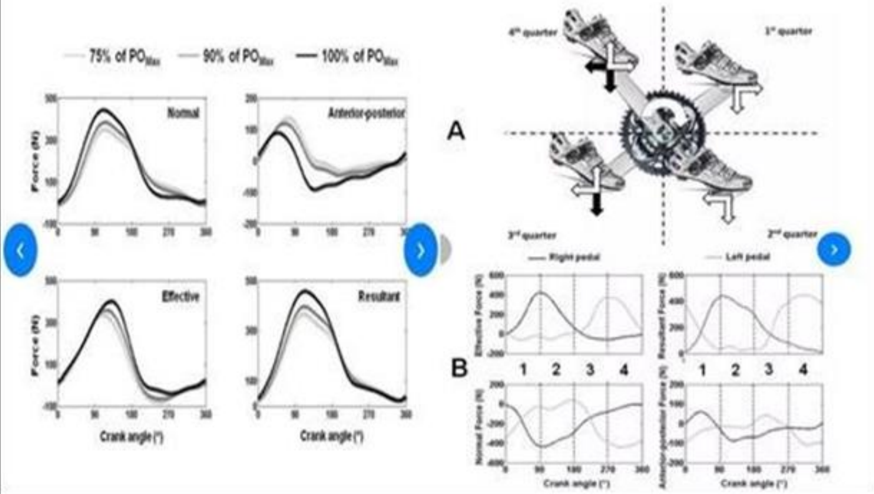
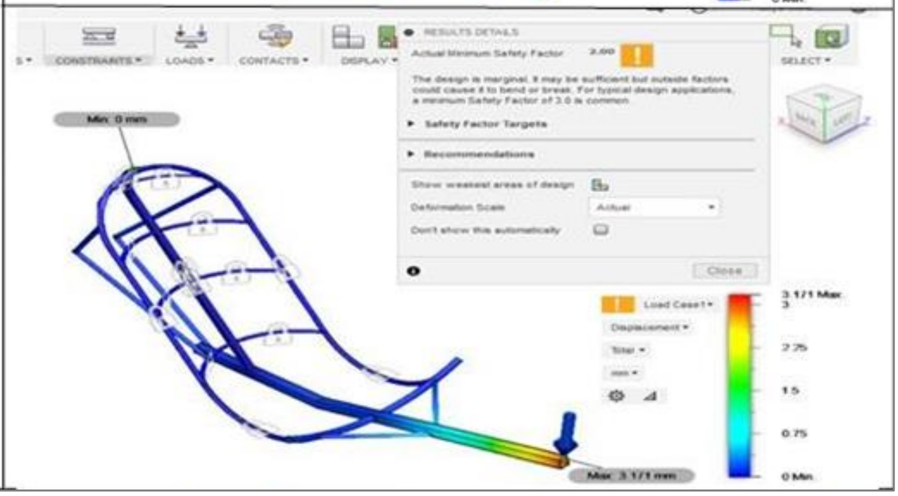
Structural Analysis

1. Load on the Pedal-

| | |
|---|---|
| **I. Objectives:** | The objective of the analysis is to assess the effect of load on the pedal region of the trike and to understand the factor of safety, deformation, and maximum stress experienced by the part for a particular amount of load. |
| **II. Method and Assumptions:** | Again, Autodesk Fusion 360 is the simulation software which is used for the load analysis of the pedal region and the RPS is constrained throughout. A load of 400N is applied on the pedal region both horizontally and vertically.  |

| | |
|---|---|
| **III. Results:** <br> i) Vertical Force: <br> Stress: |  |
| Deformation and Factor of Safety: <br> ii) Horizontal Force: |  |

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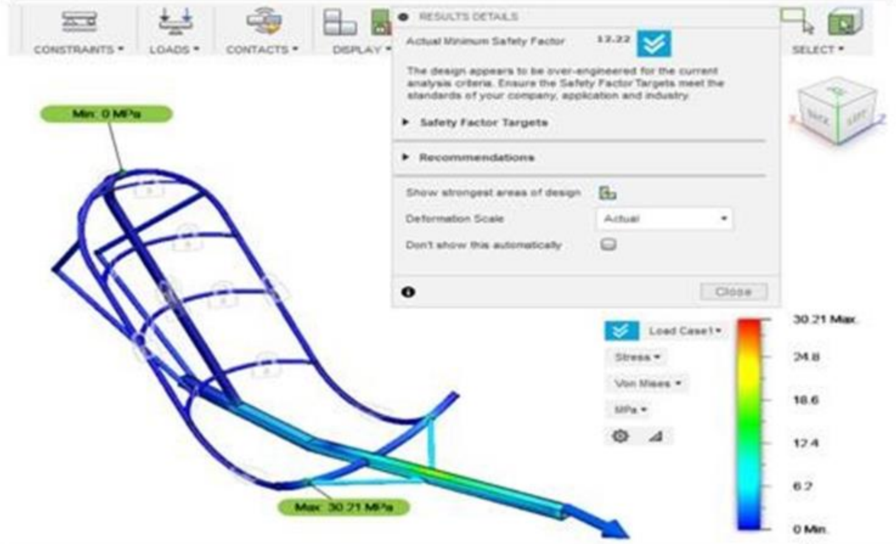
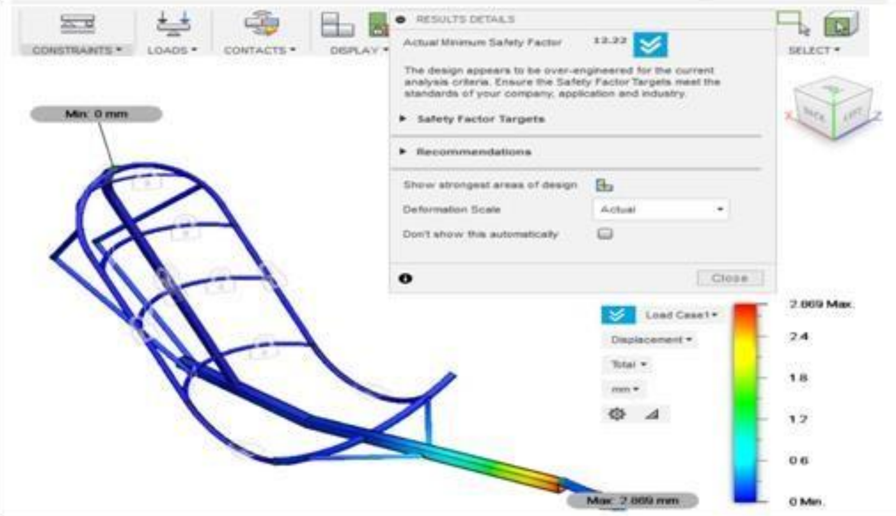
| | |
|---|---|
| Stress: |  |
| Deformation and Factor of Safety: |  |
| **IV. Conclusion:** | The FOS of the design for Vertical load is **2**. The maximum deflection experienced by the design is **3.171 mm** and Minimum deflection of **0 mm**. The maximum stress experienced by the design is **103.5 MPa** and Minimum Stress of **0 MPa.** The FOS of the design for Horizontal load is **12.22**. The maximum deflection experienced by the design is **2.869 mm** and Minimum deflection of **0 mm**. The maximum stress experienced by the design is **30.21 MPa** and Minimum Stress of **0 MPa.** |

2. Load on the Kingpin

| | |
|---|---|
| **I. Objective:** | The objective of the analysis is to assess the effect of load on the kingpin and the rear wheel axle region of the trike and to understand the factor of safety, deformation, and maximum stress experienced by the part for a particular amount of load. |
| **II. Methods and Assumptions:** | Autodesk Fusion 360 is the simulation software which is used for the load analysis of the kingpin and the rear wheel axle region having **66%** of the weight on the front axle and **33%** on the rear and the central pipes of the frame is constrained throughout. A load of **900N** is distributed and applied on the region. |

IARJSET

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| | |
|---|---|
| **III. Results:**<br>1. Stress: |  |
| ii) Deformation and Factor of safety: |  |
| **IV. Conclusion:** | The FOS of the design for load is **4.66**. The maximum deflection experienced by the design is **2.556 mm** and Minimum deflection of **0 mm**. The maximum stress experienced by the design is **79.18 MPa** and Minimum Stress of **0 MPa**. |

Aerodynamic Analysis

| | |
|---|---|
| **I. Objective:** | The objective of the analysis is to ensure that the fairing can accommodate the tallest rider of the team and to also make sure that there is decrease in the drag force by minimizing differences in the pressure. |
| **II. Methods and Assumptions:** | ANSYS fluent CFD software was used for the analysis and the analysis was performed iteratively multiple fairing. |

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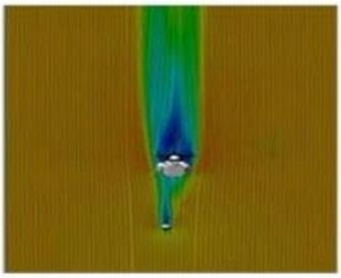
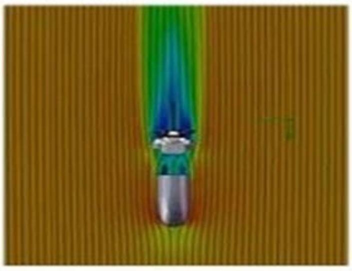
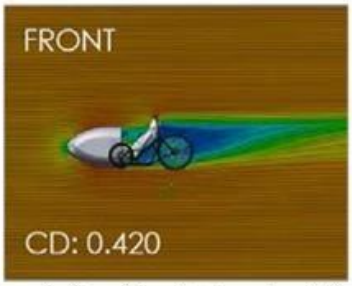
| | |
|---|---|
| **III. Result:** | |
| i) Top View: | Without Fairing:  With Fairing:  |
| ii) Side View: | Without Fairing:   With Fairing:   |
| **IV. Conclusion:** | Comparing various designs of faring and conducting cfd analysis we found the best design having lesser drag and co-efficient of drag. Also, we decided the fairing material based on fabrication processes and difficulties available. |

Steering Analysis

| | |
|---|---|
| **I. Objective:** | To reduce the turning radius. Hence, we shifted the CG in such a way so that it lies more from front. |
| **II. Methods and Assumptions:** | We used the Ackerman steering system for our steering. **Ackerman steering:** The final steering design would include Direct knuckle steering. This steering mechanism which we have used in this vehicle is one of the most economical and reliable that can be turned even within smaller radius of curvature of the path. |

$$R = \sqrt{R_l^2 + a_2^2}$$

$$\delta_i = \tan^{-1}\left(\frac{l}{R - \frac{w}{2}}\right)$$

$$\delta_o = \tan^{-1}\left(\frac{l}{R + \frac{w}{2}}\right)$$

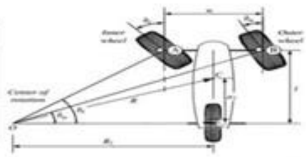


*Figure 6: Turning radius equations and corresponding Ackerman steering diagram*

**III. Result:**

INPUT VALUES

| | |
|---|---|
| L (WHEEL BASE) | 1107.2mm |
| $R_l$ | 1997.3mm |
| W | 850mm |
| $A_2$ | 688.7mm |
| $\partial_i$ | 33.26 degree |
| $\partial_o$ | 23.57 degree |
| R (TURNING RADIUS) | 2112.76mm |

Calculation:

| | Radius =2m | Radius = 4 | Radius =6 | Radius =8 |
|---|---|---|---|---|
| $\theta_{outside}$ | 24 | 14 | 9.73 | 7.54 |
| $\theta_{inside}$ | 34 | 17.3 | 11.13 | 8.54 |

**IV. Conclusion:**

Minimum turning radius=2112.703mm.

Inner wheel angle=33.26degree

Outer wheel angle=23.57degree

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3D Drawing

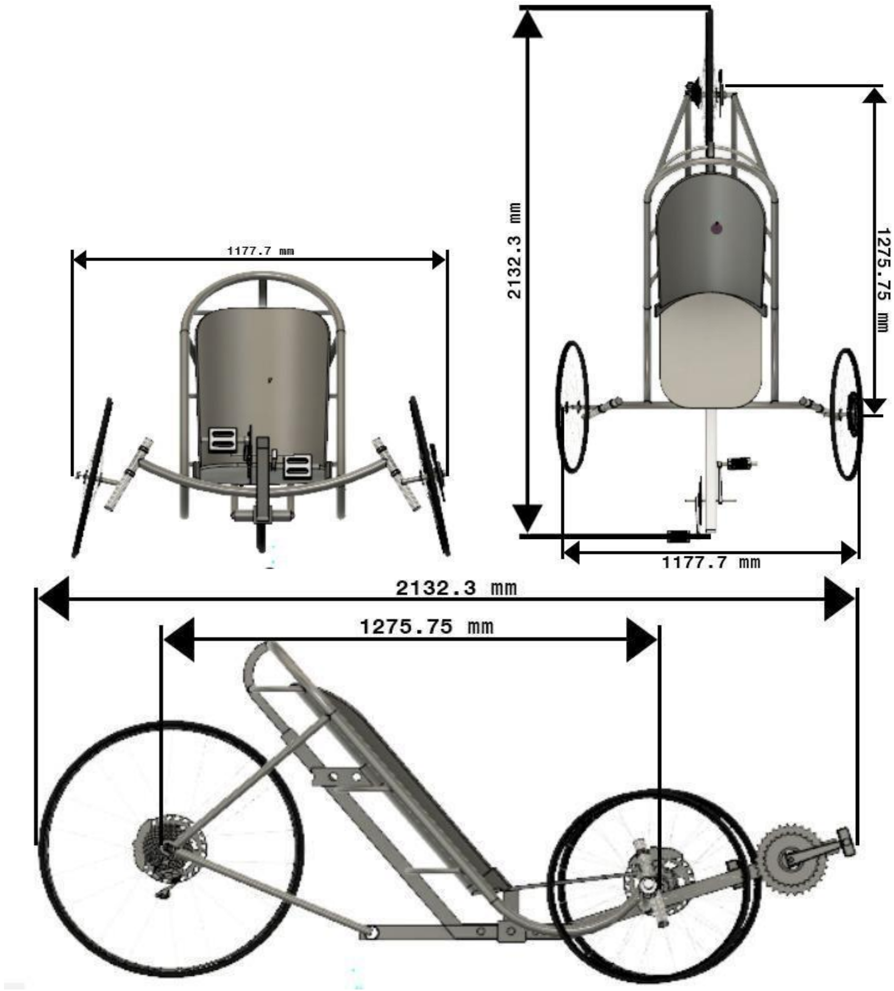


Figure 1- Front View Figure 2- Top View Figure 3- Side View

## CONCLUSION

A. Comparison

| ANALYTICAL PERFORMANCE | EXPERIMENTAL RESULTS |
|---|---|
| RPS is expected to hold a top load of 2760 N and a side load of 1330 N without undergoing any permanent deformation. | RPS successfully holds a top load of 2760 N and a side load of 1330 N without forming any permanent deformation. |
| The load on the pedal is expected to hold a load of 400N and the kingpin is expected to handle a force of 900N without forming any deformation. | The pedal and the kingpin successfully handle the load without any deformation or breakage. |
| Center of gravity is expected to be in the abdomen region of the rider. | The center of gravity is found to be exactly at the desired location. |
| Turning radius is expected to be 2.11 m which is below the 8m limit set by ASME. | Turning radius is found to be around 2 m which is close to the analytical radius. |
| The vehicle is expected to stop within 6 m at 25kph when a braking force of 528N is applied. | The vehicle stops within 6m at 25kmp , which is the desired distance. |

B. Evaluation: The results of the final testing of the vehicle were satisfactory as all of them were exact or very close to our theoretical analysis. The vehicle was put in a Universal Testing Machine (UTM) for physical testing and we

IARJSET

ISSN (Online) 2393-8021
ISSN (Print) 2394-1588

International Advanced Research Journal in Science, Engineering and Technology

Vol. 6, Issue 1, January 2019

got the desired results proving that the vehicle is strong. The RPS was tested from all required directions which did not get deformed at any point proving the vehicle to be safe for the rider. The safety requirements such as safety harness, field of vision and absence of sharp edges and protrusions is met.

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Appendix