ASME Report Cover Page & Vehicle Description Form

Human Powered Vehicle Challenge

Competition Location: Delhi Technological University (DTU), Delhi, India

Competition Date: March 16-18, 2018

http://go.asme.org/HPVC

This required document for all teams is to be incorporated in to your Design & Innovation Reports. Please Observe Your Due Dates; see the ASME HPVC for due dates.

Vehicle Description

School name: VIT University-Vellore

Vehicle name: Colt

Vehicle number 10

Vehicle configuration

Upright ______ Semi-recumbent ✓

Prone ______ Other (specify) ______

Frame material: Aluminum 6061-T6 and Aluminum 6063-T6

Fairing material(s): Fiber Reinforced Plastic

Number of wheels: 2

Vehicle Dimensions *(please use m, m^3, kg)*

Length 2.096 m Width 0.567 m

Height 1.342 m Wheelbase 1.2 m

Weight Distribution Front 7.2 Kg Rear 10.2 Kg

Total Weight 17.4 Kg

Wheel Size Front 0.4064 m Rear 0.6604 m

Frontal area $0.285\ m^2$

Steering Front ✓ Rear ______

Braking Front ______ Rear ______ Both ✓

Estimated Cd 0.1878

Vehicle history (e.g., has it competed before? where? when?)_-N/A-______________________

VELLORE INSTITUTE OF TECHNOLOGY (VIT)

2018 Human Powered Vehicle Challenge Asia Pacific

DESIGN REPORT

TEAM ANANT

Presents

COLT

Vehicle Number 10

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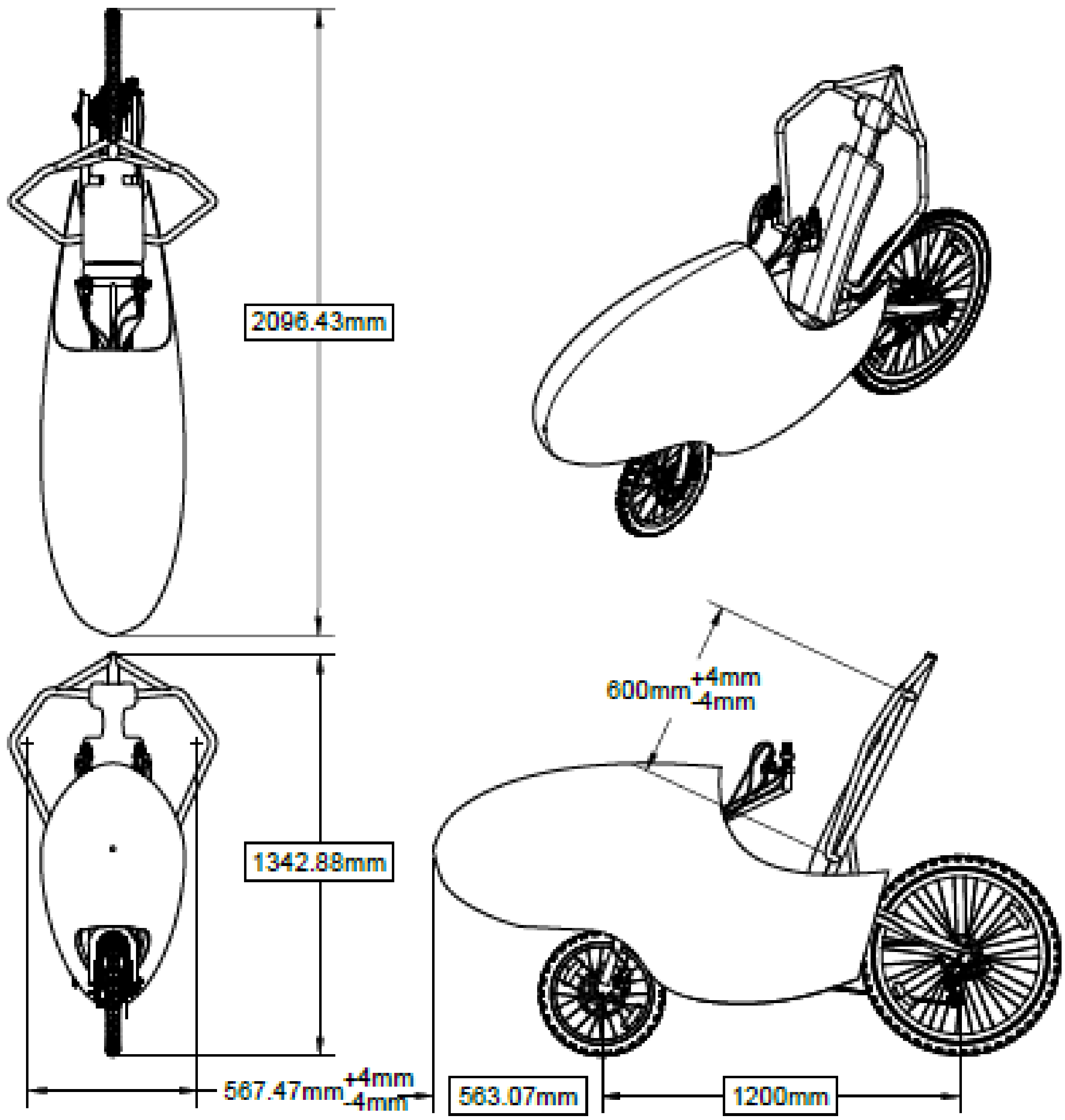
Avik Dutta	Xitij Detroja	Sujay Singh
Devika RB		



TEAM ANANT

PROJECT	Human Powered Vehicle Challenge					
TITLE	HPV COLT TEAM ANANT VIT					
APPROVED	Sabbella	03/02/2018	SIZE A	CODE TANT	DWG NO 1	REV 1
CHECKED	Rohit Malik	03/02/2018				
DRAWN	Sabbella	01/02/2018	SCALE 1:16	WEIGHT 17.4 Kg	SHEET 1/1	

Abstract

This will be the third time that Team Anant will partake in ASME India's HPVC, representing "Vellore Institute of Technology, Vellore". Based on our performance in 2017 and keeping in mind our aim, we required to develop a vehicle that is lighter and gives more acceleration. Our team chose to design, test and fabricate a HPV named "Colt" in the academic year 2017-2018 that excels in speed, aesthetics, ergonomics and safety. It is a two-wheeler semi-recumbent with a rear wheel drive and front wheel steering. The vehicle being a faired semi recumbent, the overall drag area was reduced which further helps in attaining higher speeds.

Aluminum 6063 T6 RHS beam has been used along with hollow pipes to develop the chassis. Roll over protection system was designed for the enhancement of the rider's safety. The two-sided, double chain, double shifting, 7x3 speed transmission system has been decided. An optimum design process has been followed to reduce the overall cost and energy consumption. To validate the design several analyses were performed in ANSYS under various load conditions for different parts of the chassis. Furthermore, developmental, performance and quality testing were done to ensure a safe and sound vehicle. Moreover, we are looking forward towards the development of a single lever braking system that provides an optimum braking system.

Learning from previous experiences has enabled us to build a new safe HPV with which we are confident that it will pave way for the future developments.

Figure 1: Comparison between BOLT and COLT

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1. Design

1.1 Objective

Team Anant of Vellore Institute of Technology designed, built and tested COLT during the 2017-18 academic year, in view of the team motto "Onwards and Upwards". We have the objective of setting up stepping-stones to accomplish better plans for human powered vehicles and manufacturing techniques to guarantee rider safety, speed, effectiveness and affordability. We look forward to get better results from of our HPV COLT to ensure that we always put our best foot forward with a specific end goal to present humanity, a vehicle that is exceptionally reliable for all needs of human and affordable so it can reach people from all backgrounds. The designing and manufacturing of COLT provided positive learning and working environment for students, with the goal that they can utilize the gained knowledge, experience and spirit of teamwork for accomplishing their respective objectives and contribute to the betterment of mankind.

1.2 Background

The environmental concerns are increasing at an alarming rate. It is important to think of an answer of travelling without negatively affecting the state of our already degrading condition. The regular human powered vehicles like bicycles and rickshaws do not have the practical applications for being utilized daily as a personal transport. To tackle this problem, we designed COLT, accounting for the safety and ease of rider. The recumbent frame makes it easy for the rider to ride longer distances without staining his/her back. Its design guarantees minimal energy requirements from the rider, greater maneuverability, more speed, safety and affordable manufacturing cost. These highlights altogether make COLT a more desirable vehicle than traditional bicycles and even automobiles because of the pressing need to prevent nature degradation.

1.3 Prior Work

This is the third time that our team is taking part in HPVC. The team learnt from our first vehicle 'ASHV' and ad-libbed in our second vehicle 'Bolt'. It was useful as we got the opportunity to gain from our mix-ups. That is the reason, this time additionally we scribbled down the issues looked in 'Bolt' and extemporized them in 'Colt'. A portion of the information gathered amid past sessions were utilized this time. Ergonomics information that was gotten last time with the assistance of prototype was utilized again for development of ‘Colt’. Likewise, market reviews, steering and transmission calculations were utilized from past session. However, the designing, assembling and testing was done afresh.

1.4 Organizational Timeline and Planning (As of Feb 05, 2017):

We know that for a group to adequately complete the task, legitimate arranging of the different stages including Design, Analysis, Manufacturing and Testing are basic. For this purpose, our team used a Gantt chart (Appendix-Gantt Chart) which was always refreshed to monitor the postponements in the advancement procedure. Following is the tabular introduction of our course of events (Date format- mm/dd/yy) which was used to build up the graph. Because of dedication and commitment of our team members and proper planning, this time we could adhere to our timeline. This helped a lot to complete the vehicle effectively and efficiently.

Table 1: Organizational Timeline

Task Name		Start	End
1.	Discussion on past experience	07/09/17	07/26/17
2.	Pre-Design phase	07/27/17	09/16/17
3.	Design Phase and Analysis phase	09/17/17	11/18/17
4.	Manufacturing Phase	11/12/17	01/3/18
5.	Testing	01/04/18	01/20/18
6.	Safety Analysis	01/21/18	01/27/18

1.5 Design Criteria

We followed every design constraint for the design and validation of 'COLT' as stated in the ASME HPVC rulebook 2018 along with additional constraints laid out by the team to develop a better HPV. They are plotted as followed:

Criteria	ASME Constraints	Additional Team Constraints
Performance	Stopping Distance: 6 meters at (25 kmph speed)	Light Weight and Safe
	Turning Radius: 8 meters	High speed and acceleration
	Field of View: 180 degrees (min)	Low drag coefficient
	Low speed Stability: Straight line motion for 30 meters at 5-8 kmph	Comfortable ergonomic Riding Position
	Storage Capacity	Drivetrain efficiency and reliability
	New Design Entry	Constrained wheelbase
Safety	Roll Over Protection System: **Top load: 2670 N** Deflection: <**5.1 cm** Direction: Applied at 12 degrees to the vertical **Side Load: 1330 N** Deflection: <**3.8 cm** Direction: At Shoulder Height Horizontally	Good Braking System Helmet and safety equipment requirements Safety analysis of custom designed parts
	RPS Attachment: Structurally Attached	Stable and reliable steering
	No sharp Edges	Wide viewing solid angle of rider
	Harness: Safe and Firm	Chest and lap Seat harness

Table 2: Design Criteria

The above stated constraints along with the experience gained amid previous sessions were used to prepare the House of Quality Chart. The House of Quality Chart was utilized to organize the plan prerequisites in view of the score accumulated amid the investigation. The diagram was plotted by drilling down the opposition imperatives and particulars along the section. The group necessities were plotted out along the line. Definite investigation delineated that the vehicle configuration must be agreeable, exceptionally steady, safe organized, cost proficient and the vitality utilization amid the entire procedure must be least. Additionally, the design was compared with ASHV and BOLT (previous entries to the competition).

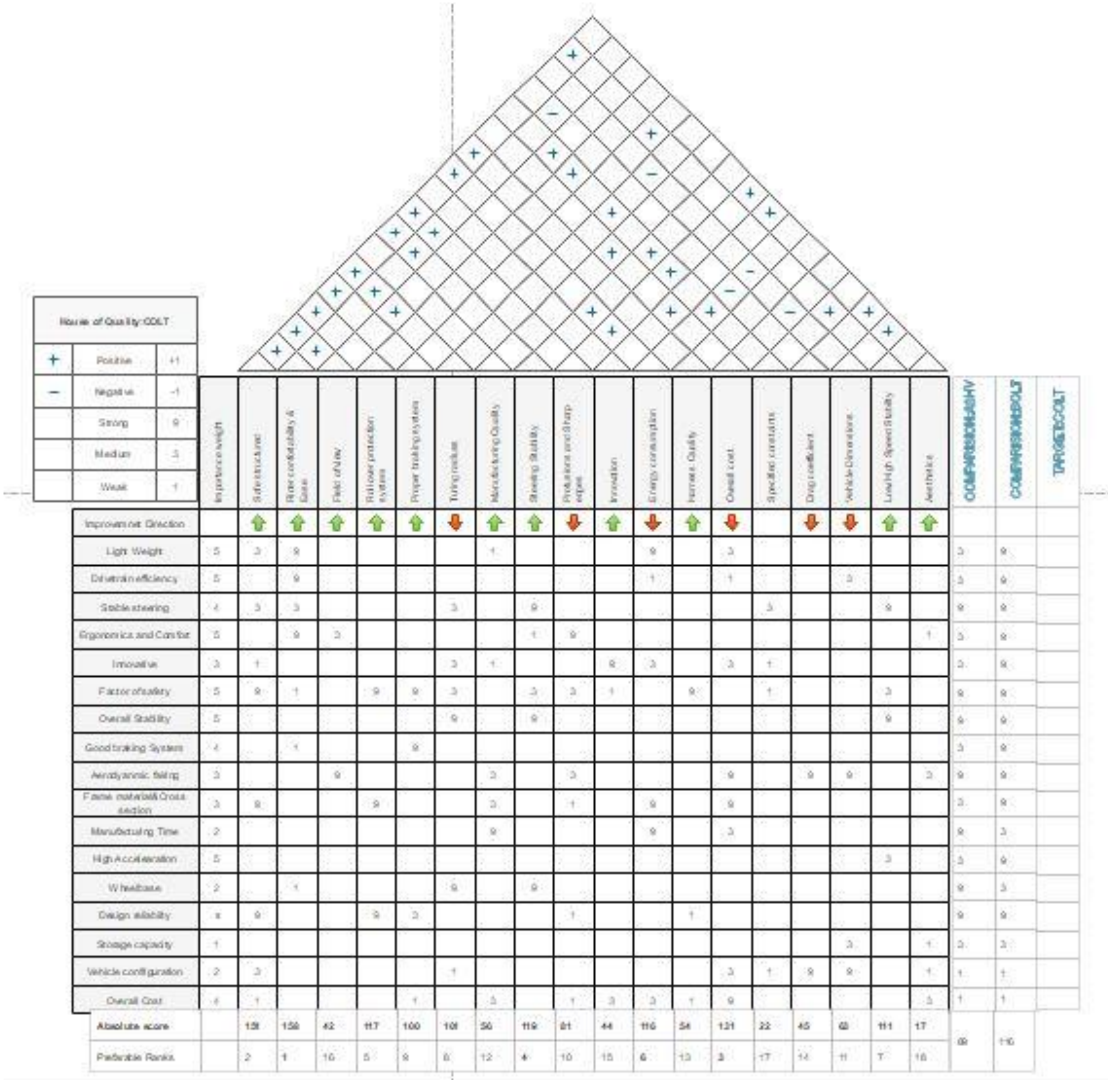


Figure 2: House of Quality Chart

Based on the outcomes the critical outline highlights were arranged as stated in the table. The sketched-out highlights were thought about emphatically to build up a productive plan.

Table 3: Features table

Features	**Requirements**	**Expected Target and Solution**
Comfort	Comfortable	Data collected from previous competitions and prototype were used as ergonomics reference.
Overall Cost	Minimal	Performed Cost analysis to minimize the inflow
Safety	Safe Vehicle Design	Proper selection of frame material and frame cross-section for reducing weight and providing strength.
Stability	Highly Stable	Steering system and rider position supports stability in vehicle
Feasibility	Highly Feasible	To develop a feasible and easy to manufacture vehicle.
Energy Consumption	Analyzed	To minimize the energy consumed throughout the development phase of the vehicle.

1.6 Concept Development and Selection

The design process helped with laying out the fundamental prerequisites for the vehicle development. A few plan, ideas and elective thoughts were produced amid the underlying period of the advancement procedure. The doable thoughts created were then dealt with as per the group's prerequisite. To finish the outline idea and different highlights Pugh's Selection Technique was conveyed for the whole plan. The components thought about comprised of Vehicle design, frame material, cross section, material and Drivetrain arrangement.

1.6.1 Pugh's Selection Technique

a) Vehicle Configuration:

Proper configuration of vehicle was important as ease of riding and comfort mostly depends on it. These data were collected from previous ergonomics studies and experience from last two HPVC. This helped us to shift through other considered options and enabled us to offer them proper weightage and points. Selection technique was conveyed for concluding the setup out of the two best voted two wheeled designs.

Table 4: Decision matrix for vehicle configuration

Features Weightage	**Weightage**	**Compact Long Wheelbase**	**Medium Wheelbase**	**Short Wheelbase**
Rider Safety	5	1	0	-1
Capsizing Stability	4.5	0	1	-1
Turning Ease	4	-1	0	1
Drivetrain Efficiency	4.5	-1	1	0
Comfort	4.5	1	-1	0
Weight	3	-1	0	1
Relative Score		**-2.5**	**9**	**-2**

Going by the results, we selected medium wheelbase that was better than compact long steering according to our requirements and constraints.

b) Frame Material:

It assumes a critical part in weight and factor of safety of the vehicle along with the ease in machinability and joining operations like welding. To tackle the problem of getting optimum strength to density ratio, we referred to Ashby Diagram. While comparing different materials and analyzing previous experiences we found that Aluminum 6063-T6 is best suited for our purpose

c) Steering System:

Steering system ensure safe turning and stable headway. Hence, it is mandatory to plan accordingly to avoid accidents. We compared universal joint steering, bevel steering and bullhorn steering. Universal joints proved a difficult task last year. To facilitate riding in recumbent position the conventional bullhorn steering has been modified to meet our purpose.

d) Drive Train System

After a lot of brainstorming we came to a conclusion with the help of Pugh's chart to use two-sided, double chain with the help of an intermediate jackshaft. The points awarded for each criterion are mentioned in

Table 5: Decision Matrix for drivetrain configuration

Criteria	Weightage	One Sided	Two sided	Single chain
Weight	3	0	-1	1
Rotational Moment of Inertia	4.5	1	-1	0
Integration ease	4.5	0	1	-1
Stability	5	-1	1	0
Relative Score		-0.5	2	1.5

1.7 Vehicle Distribution

The vehicle was designed in view of the choices made in the above-mentioned Pugh's Selection Method. The few basic designed parts were concluded and modelled in Solidworks CAD software and then analyzed using ANSYS. The entire model is shown in the figure underneath this section.

1.7.1 Roll Over Protection System:

The Roll over Protection System was designed keeping in mind the design constraints set by ASME-HPVC rulebook. The Roll over protection system has the same material as of the main frame so that it can be braced to the frame which is must. We have used 22 mm outer diameter Al-6063 T6 pipe for the design. The above selected frame image incorporates the roll over protection design.

1.7.2 Aerodynamic Fairing:

There is a strong need to design a sturdy, ergonomic and light weight fairing for our vehicle. Fairing is needed to cater the aerodynamic and safety needs of the vehicle. For aerodynamic aspects, the shape of a tear drop has very less drag to it, so we decided to use the standard curves for this shape i.e. NACA profiles for the fairing nose. Fairing also caters to the safety needs of the passengers by absorbing crash during vehicle side fall or front collision.

To support the fairing, fairing mounts have been provided in the main frame. Some preliminary NACA curves are chosen, lofted as per the requirement, and modified and extruded cut in places wherever necessary in order to avoid interferences.

1.7.3 Drive Train Configuration:

This time we made drastic changes in our drive train configuration. Our main objective was to get good start-up acceleration along with fast speed and reliable setup, so in order to acquire the goal we came up with the idea of a combination of two-sided, double chain configuration.

Two Gear Shifter:

During HPVC'16, we lacked in speed, so we improved it in HPVC'17 by changing gear ratio range. However, inefficient shifting decreased the possible maximum acceleration. Hence, to provide rider with both speed and acceleration we came with an idea to increase the range of provided gear ratio. As first chain sprocket has 60 teeth while first chain jackshaft has 24 teeth, so including it in calculation. Resulting gear ratios are as presented in the excel table. The transmission ratios are obtained using the following procedure ((34/42) *(24/60)). The optimal gear ratios for best shifting series are obtained by using speed v/s power graph presented in the analysis section [Figure 5: Graph for shifting sequence analysisFigure 5: Graph for shifting sequence analysis].

Gear Ratios at different configurations		Number of teeth in Second chain jackshaft		
		42	34	24
Number of teeth in Second chain cassette	34	0.3238095	0.4	0.5666667
	24	0.2285714	0.2823529	0.4
	22	0.2095238	0.2588235	0.3666667
	20	0.1904762	0.2352941	0.3333333
	18	0.1714286	0.2117647	0.3
	16	0.152381	0.1882353	0.2666667
	14	0.1333333	0.1647059	0.2333333

Table 6: 21 different transmission ratios

1.8 Innovation

a. Steering System:

We have planned to use bullhorn design for the steering. To accommodate varying heights of riders, we have made adjustable steering. It provides comfortable ride to riders. The handlebar has been provided with an adjusting mechanism.

b. Single brake lever mechanism:

For the most optimum braking, the braking force must be distributed in a certain ratio (as calculated in the braking analysis section). To incorporate the idea, force sensitive resistors will be coupled with the lever to measure the applied force and transmit the signals to the processor which further actuates the servo motor according to the brake distribution programmed. This enables to attain the optimum braking using a single lever.

Figure 3: Vehicle Description

2. Analysis

2.1 Roll Over/Side Protection System

Objective: To determine the feasibility of the Roll over protection design under provided load condition. Top and Side load analysis study was performed using Static Structural module of Ansys. The methodology followed has been presented below.

2.1.1 Methodology

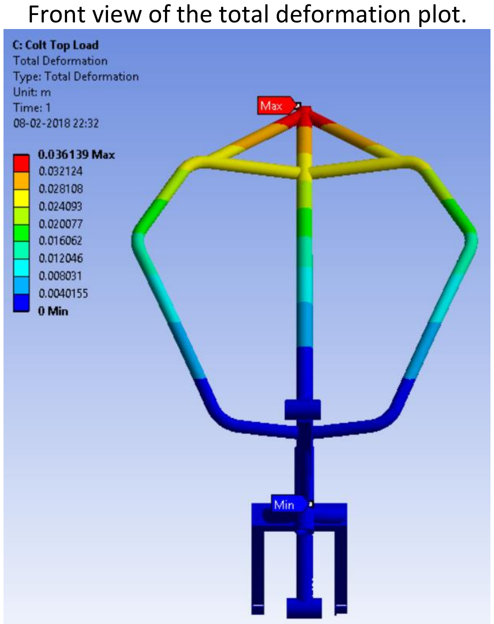
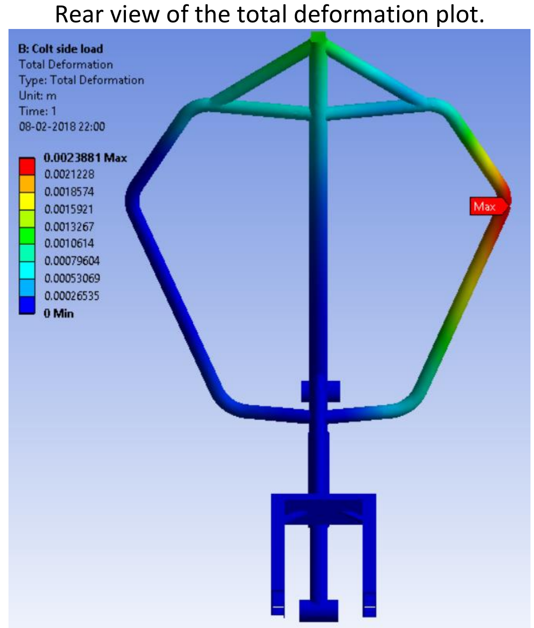
<table>
<tr><th>Top Load Analysis</th><th>Side Load Analysis</th></tr>
<tr><td>The setup was modeled according to the specified load conditions. For the analysis of the RPS under top load conditions, the lower end of the frame was constrained. The load of 2670 N was applied at an angle of 12 degrees at the top most point. The total deformation results were analyzed.</td><td>Similar methodology was followed for the side load setup. The lower frame and the other end of the side of the RPS was constrained imitating the real situations. The load of 1330 N was applied at the side. The analysis was interpreted for maximum total deformation.</td></tr>
<tr><td colspan="2">Total Deformation Plot</td></tr>
<tr><td>Front view of the total deformation plot.

</td><td>Rear view of the total deformation plot.

</td></tr>
<tr><td colspan="2">Results and Discussions</td></tr>
<tr><td>The maximum deformation of 3.6 cm was observed at the top most point of the RPS. The deformation and the factor of safety results were improved by adding two supports at the top points as seen in the plot. This modification upgraded the results by distributing the load and thus reducing the total deformation.</td><td>The maximum deformation observed in this case is 0.24 cm. The deformation being low, no further modifications were required. However, a plate was welded at the bend which even served the purpose of the seat belt mount.</td></tr>
<tr><td colspan="2">Conclusion: The design was considered safe to proceed with.</td></tr>
</table>

Table 7: Analysis setup for Roll/Side protection system

2.2 Structural Analysis

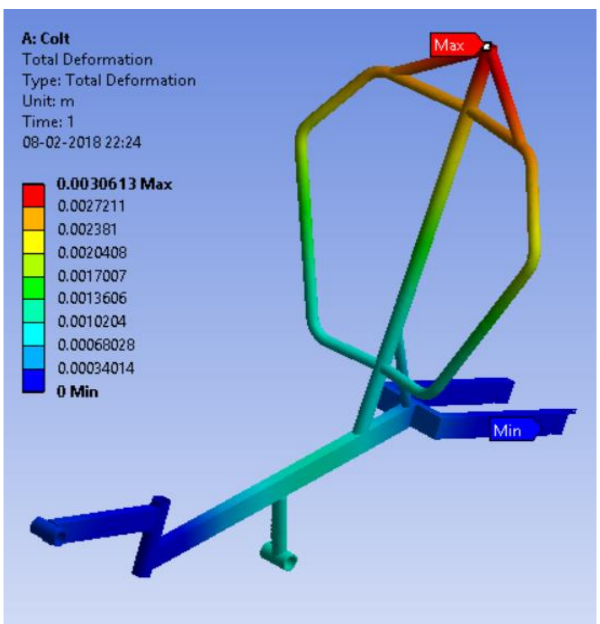
Objective and Analysis	To determine the feasibility of the frame design, reduce weight of the overall chassis and detect critical points. Static structural module of Ansys was utilized to analyze and understand the behavior of the fame under different loading conditions.
Methodology	The complete analysis setup was developed in Ansys. Frame model was imported from Solidworks. Analysis was performed and results were analyzed for modifications.

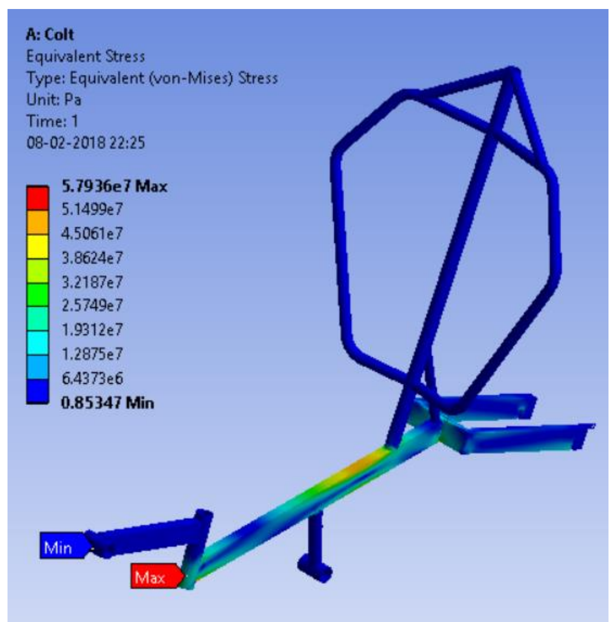
2.2.1 Frame Analysis

a. Static Frame Analysis

The major objective for this case study was to understand the behavior of the chassis under the load condition of the rider. The complete frame was modeled in Solidworks. The area where the rear wheel stud is mounted on the clamp was completely constrained. The lower end of the head tube was fixed in the vertical direction. As the complete weight of the rider is supported by the main beam and the backrest, a load of 1200 N and 400 N was applied downwards and towards the rear respectively. The results obtained are presented below.

Table 8: Analysis Setup for static frame analysis

<table>
<tr><th>Displacement Plot</th><th>Stress Plot</th></tr>
<tr><td>


</td><td>


</td></tr>
<tr><th colspan="2">Results and Discussions</th></tr>
<tr><td>The maximum deformation observed for this case study was 0.306 cm. The frame does not undergo drastic deformation.</td><td>The maximum equivalent stress obtained is at the lower end of the head tube. The factor of safety observed for the frame is 8.7.</td></tr>
<tr><th colspan="2">Modifications</th></tr>
<tr><td colspan="2">As we see, the most stressed region in the frame from the analysis is the lower end of the head tube. Therefore, to distribute the load two supports were added between the frame and the head tube.</td></tr>
</table>

b. Road bump at rear wheel

Similar approach has been followed for the rear bump analysis setup. The lower end of the head tube was constrained in the x, y and z axis. Additionally, the rotation about y axis was restricted. The seat mount was constrained in the y axis. A remote force of same magnitude was provided at the contact point of the rear wheel. The most stressed region highlighted was the lower part of the head tube.

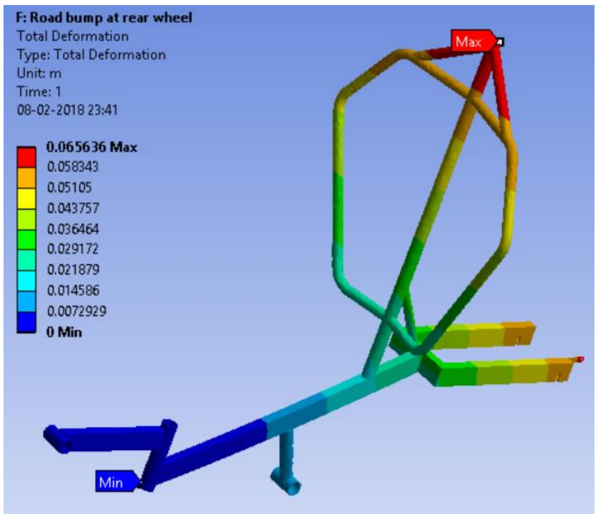
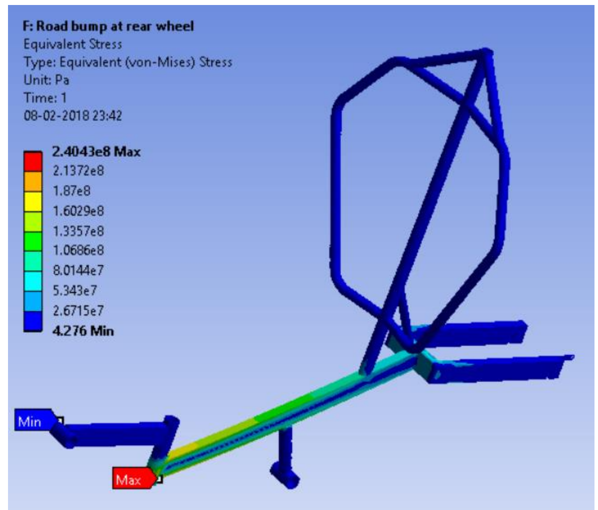
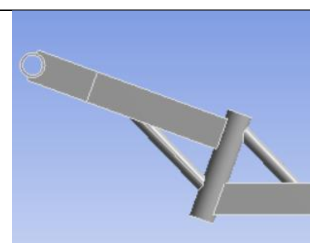
<table>
<tr><th>Displacement Plot</th><th>Stress Plot</th></tr>
<tr><td>
</td><td>
</td></tr>
<tr><th colspan="2">Results and Discussions</th></tr>
<tr><td>The maximum deformation observed for this case study was 6.5 cm.</td><td>The maximum equivalent stress obtained was at the lower end of the head tube.</td></tr>
<tr><td colspan="2"></td></tr>
<tr><th>Modifications</th><td rowspan="2"></td></tr>
<tr><td>As the rear wheel bounces, the head tube compresses. This causes severe stresses in the lower region of the head tube. To further reduce this stress, the frame was modified. Supports were added connecting the main frame and the head tube. This effectively reduced the stress experienced.</td></tr>
</table>

Table 9: Analysis setup for bump at rear wheel load condition

c. Road bump at front wheel

The condition when the vehicle encounters a road bump has been analyzed in this case study. The analysis setup was developed with respect to the bump encounter at front wheel. The setup followed was as presented in this research paper [1]. The rear wheel mount was constrained in longitudinal (x), lateral (y) and vertical (z) axis. Additionally, the axial rotation of the clamp was constrained about the y axis. The seat mount was constrained in the y axis. A remote force of 1000 N towards the rear and 2500 N upwards which presents the force due to impact with the bump was provided at the front wheel impact location. The setup was simulated. The factor of safety of 3.4 was obtained for the analysis. The maximum displacement obtained was 5.6 cm.

The most critical point for this analysis was the rear clamp and it successfully handled the provided loading conditions.

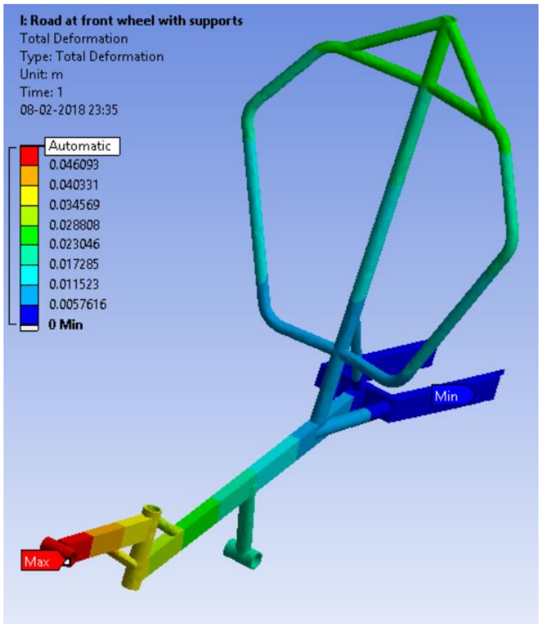
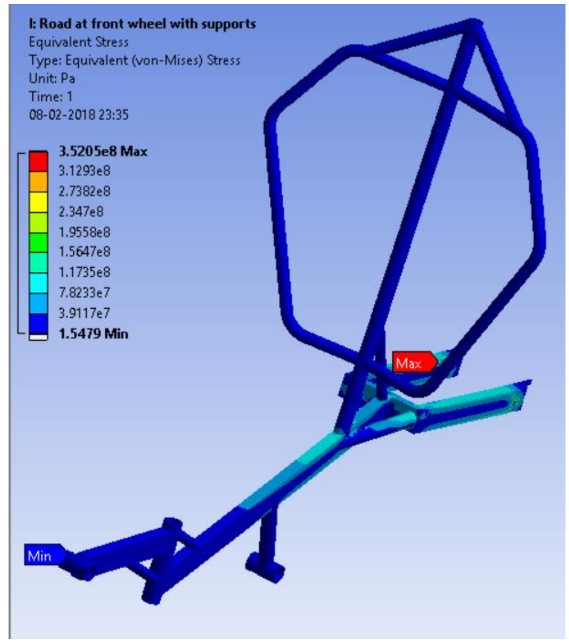
<table>
<tr><th>Displacement Plot</th><th>Stress Plot</th></tr>
<tr><td>
</td><td>
</td></tr>
<tr><th colspan="2">Results and Discussions</th></tr>
<tr><td>The maximum deformation observed for this case study was 5.6 cm.</td><td>The maximum equivalent stress obtained is at the lower end of the rear fork. The factor of safety observed for the frame is 3.4.</td></tr>
</table>

Table 10: Analysis Setup for bump at front wheel load condition.

Modifications: Supports have been provided between the main frame and the rear fork, supporting rectangular plates with optimized design have been added between the rear forks. This further helped in reducing the equivalent stress obtained in the region.

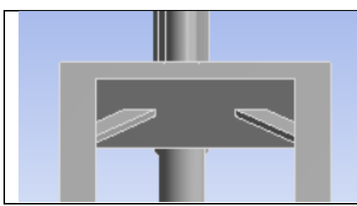
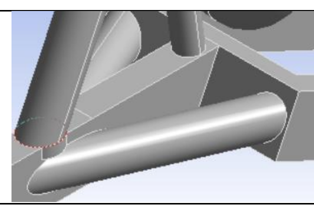
	

Table 11: Modifications provided

2.2.2 Bottom Bracket Analysis

The bottom bracket analysis was very crucial for determining the overall feasibility of the design. The fixtures provided were similar to the ones set for the static frame analysis. However, the load conditions vary. A remote load of 1200 N has been applied in the longitudinal direction at the pedal point to the bottom bracket side face. Additionally, 200 N of force has been applied in the vertical direction. Similarly, on the other side of the pedal, 1/10th the magnitude of this load was applied in the backward direction imitating the returning pedal force. On the secondary bottom bracket because of the transmission ratio, the force acting is 3000 N for this input load condition. The setup is analyzed and the results are presented below.

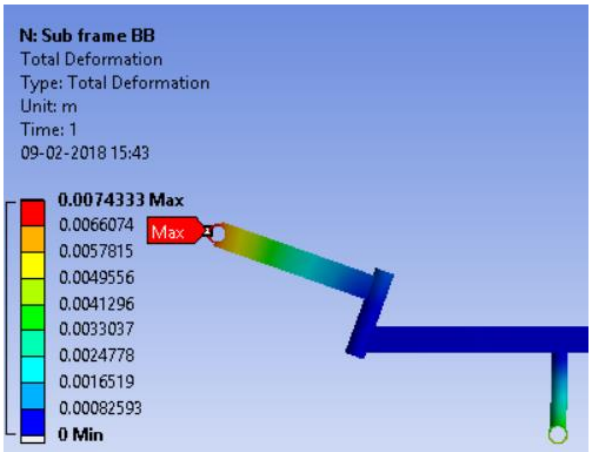
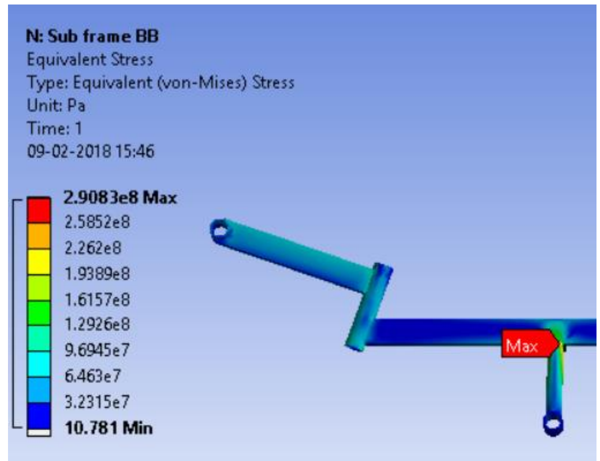
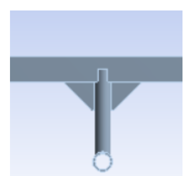
<table>
<tr><th>Displacement Plot</th><th>Stress Plot</th></tr>
<tr><td>
</td><td>
</td></tr>
<tr><th colspan="2">Results and Discussions</th></tr>
<tr><td>The maximum deformation observed for this case study was 0.74 cm.</td><td>The maximum equivalent stress obtained was at the upper end of the secondary bottom bracket.</td></tr>
<tr><td>Modifications added
To reduce the stress experienced by the secondary bottom bracket, two triangular plates were welded. The bottom bracket support is welded to the main beam. Thus, adding these plates further prevents the BB pipe from experiencing any bending due to pedaling force.</td><td></td></tr>
</table>

Table 12: Analysis setup for bottom bracket

2.3 Aerodynamics

The major objective of this analysis is to determine the drag area of the fairing designed for the vehicle. The fairing was modeled using Solidworks. 2424 and 0024 NACA profiles were decided according to the chord length set for the design. ANSYS CFX was utilized to analyze the aerodynamic behavior of the fairing. Designs considered have been presented below.

2.3.1 Designs Considered

The analysis setup was modeled in CFX. The designs considered were analyzed. The second fairing developed covers the lower region of the chassis, this provides an added benefit to the previous design. The pressure counters and drag area was compared for finalizing the design.

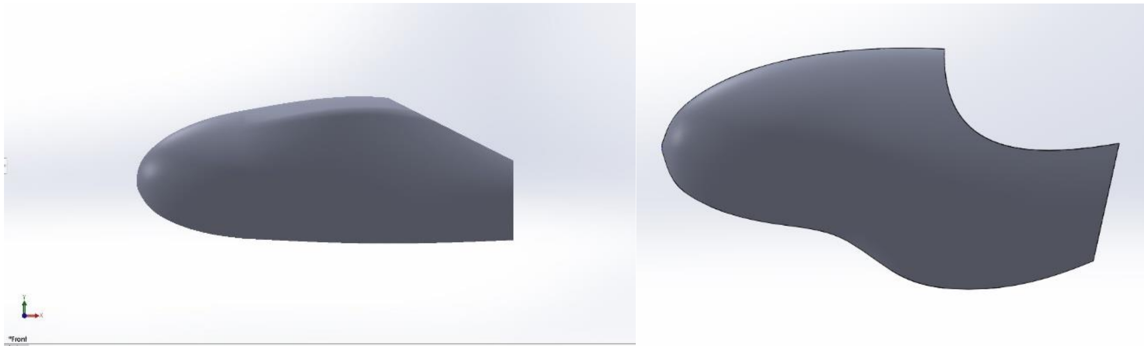


Figure 4: Design considerations for aerodynamic fairing

2.3.2 Front Flow Analysis

The front flow analysis tries to analyze the aerodynamic behavior of the vehicle while in motion. The geometry was imported and enclosed according to suitable dimensions. The setup was appropriately meshed. Further, the inlet was provided to the one side of the enclosure and the opposite side as the output. The other sides were provided with wall boundary conditions. Atmospheric temperature (25 degrees C) and pressure (1 atm) was provided. Air velocity of 12 m/s was considered. No slip conditions were provided for the fairing and the wall. The setup was analyzed for this boundary condition. The counters were plotted for pressure and velocity streamlines. The results are presented below.

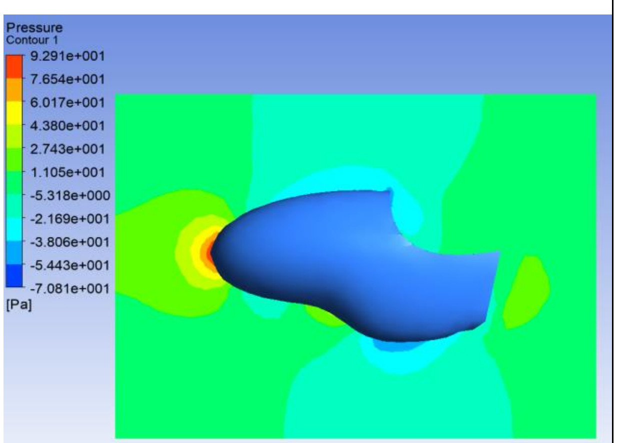
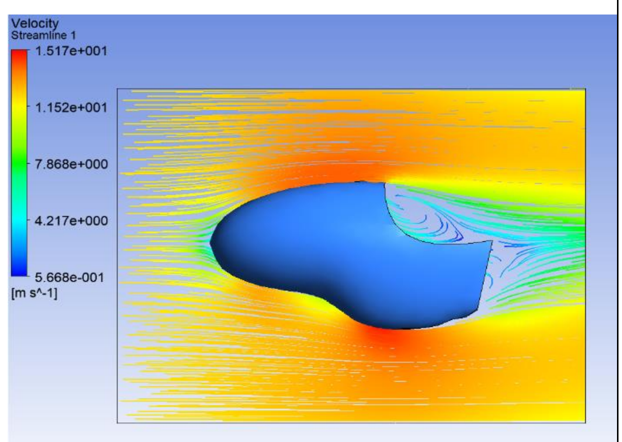
Pressure Counters	**Velocity Streamlines**
	
Results	
The maximum pressure of 92.91 Pa is observed at the front end of the fairing. The pressure difference observed between the front end and cut section is 130.97	The velocity streamline shows the flow of the wind around the fairing. The cut section experiences turbulent flow, which was optimized from the earlier results.

Table 13: Front Flow Analysis

2.3.3 Cross Flow Analysis

Similar procedure was followed for this case study. The cross wind of 4 m/s was provided. The results have been presented below.

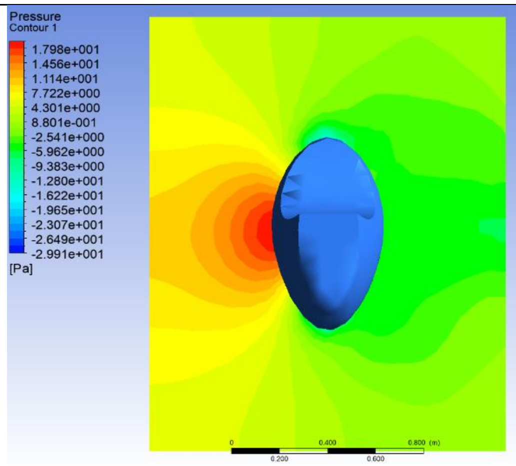
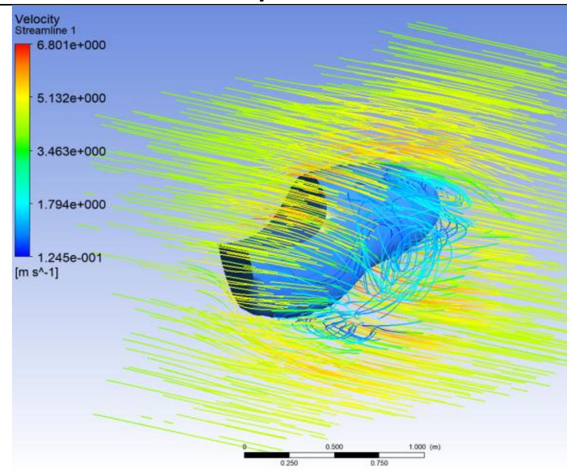
Pressure Counters	Velocity Streamlines
	

Table 14: Cross Flow Analysis

The maximum pressure of 17.98 Pa is observed at the side end of the fairing. The velocity streamline shows the flow of the wind around the fairing.

2.3.4 Design Substantiation

The analyzed design was selected for the fairing. On further analyzing the real-life conditions, considering the motion of chainrings and wheels. The initial design will encounter more drag force. However, this problem is solved in the second design as it covers the most of the transmission components. The drag area calculated from the analysis is 0.0751 m^2. The drag coefficient observed is 0.1878.

2.4 Cost Analysis

Objective	Analysis	Results
To analyze the overall cost of the vehicle and examine the cash flow to minimize the overall expenditure.	Development of a Cost sheet for lying out and adding up of various expenditures. An extensive analysis and market survey assisted in minimizing the overall cost.	The total overall cost of the vehicle amounts to Rs. 23122

The cost investigation was done using a cost sheet, which laid out each part and material bought for the improvement of the vehicle. Cost acquired amid purchase of materials, components, assemblies and operations are as following:

S. No.	Component Description	Rate (Rs.)	Quantity	Cost (Rs.)
A	Materials			
1	Aluminum Pipes (ϕ2.2cm)	340/m	2 m	680
2	Aluminum Pipes (ϕ3.2cm)	360/m	1	360

3	Rectangular Hollow Cross Section	$350/m^3$	3 m	1050
4	Aluminum Sheets (5mm)	$6000/m^2$	$1m^2$	6000
5	Fiber Reinforced Plastic (Seat)	$840/m^2$	$0.8m^2$	672
6	Fiber Reinforced Plastic (Fairing)	$400/m^2$	$2.8m^2$	1120
B	**Components**			
1	Head set Assembly	180	1 set	180
2	Wheel Assembly	250	1 set	250
3	Wheels	2200	1 no(s)	2200
4	Transmission Components	1500	1 set	1500
5	Shifters	650	1 no(s)	650
6	Utilities	600	1 set	600
9	Safety Essentials	860	1 set	860
C	**Operations**			
1	CNC Bending	-	-	3000
2	GAS Welding	-	-	4200
	TOTAL			**23122**

Table 15: Cost Analysis

2.5 Other Analysis

2.5.1 Product Energy/CO2 Lifecycle Analysis

We took values from what we obtained last year, they were verified again and new required values were obtained again by market survey.

S. No.	Component Description	Energy Consumption per unit mass (J/kg)	CO2 Production per unit mass (g/kg)	Mass (kg)	Total Energy Consumption (J)	Total CO2 Production (g)	Reusability	Recyclability	Disposability
1	Aluminum (Recycled)	11860000	2600	6.9	81834000	17940	Easy	Easy	Easy
2	Fiber Reinforced Plastic	17820000	860	0.203	3617460	174.58	Moderate	Hard	Moderate
3	Steel	16000000	4000	1.6	25600000	6400	Moderate	Easy	Easy
4	Rubber	2320000	3600	0.59	1368800	2124	Easy	Hard	Hard
5	Manufacturing operations								
	TOTAL	**48000000**	**11060**	**9.293**	**112420260**	**26638.58**			

Table 16: Product Energy/CO2 Lifecycle analysis

In order to reduce our carbon footprint and energy consumed for production we reused many aluminum members from our previous vehicles.

2.5.2 Front Fork Analysis

Objective and Analysis	The fork was analyzed to determine the safety factor of the design. Static structural module of Ansys was utilized to analyze and understand the behavior of the fork under different loading conditions.
Methodology	The complete analysis setup was developed in Ansys. Fork model was imported from Solidworks. Analysis was performed. Results were analyzed for modifications.

a. Impact Load Conditions:

This case study incorporated the condition of static loading. The complete methodology has been presented as follows. The model was prepared in Solidworks. The model was meshed. Mesh optimization was done at the intersection of the steerer tube and the fork. The top most end and the location where crown is assembled on the steerer tube was constrained. The impact load considered for the front wheel bump impact was used for this analysis. 2600 N of force was applied at the clamps in the vertical direction and 1000 N towards the rear wheel, resulting in an overall load of 2602. 6 N. The clamps were constrained axially. The analysis was performed and the results were analyzed.

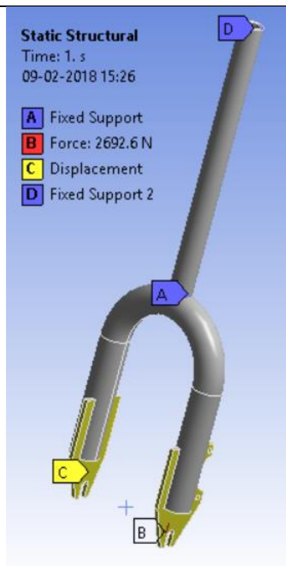
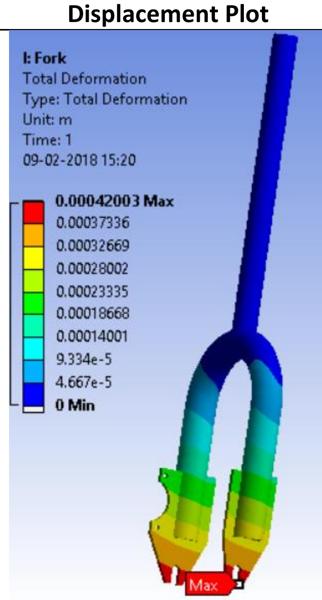
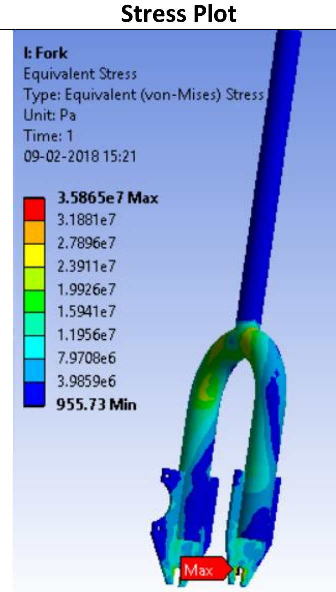
<table>
<tr><th>Fixtures</th><th>Displacement Plot</th><th>Stress Plot</th></tr>
<tr>
<td>
</td>
<td>
</td>
<td>
</td>
</tr>
</table>

Table 17: Analysis setup for front fork

Result	Maximum stress	Maximum Displacement
Values	35.86 MPa	0.42 mm

Table 18: Results for impact load condition

Inference: The most stressed region of concern as depicted from the stress plot was the intersection of the steerer tube and fork. To provide additional support, a sleeve of 32 mm outer diameter has been welded to the steerer tube pipe before welding it with the fork. This adds additional support and reduces the stress experienced by the region.

b. Braking Condition Analysis for clamp:

This analysis setup includes the braking loads acting on the brake caliper mounting points. The clamps are analyzed for determining its behavior. The complete fork has been constrained for the analysis of the clamps. The load condition of 800 N which takes into account the load transfer while braking has been added in the vertical direction. The front braking force of 200 N has been applied at the brake caliper mounts. The clamps were constrained in the axial direction of the wheel. The setup was analyzed and the results are presented below. The design is safe and sound.

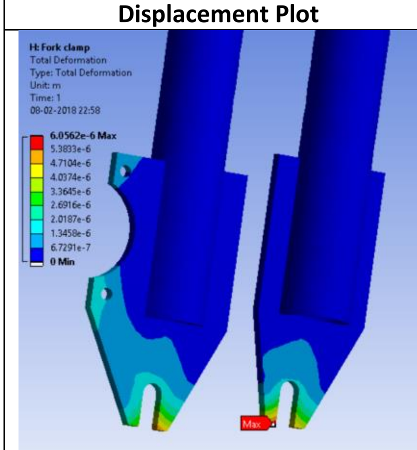
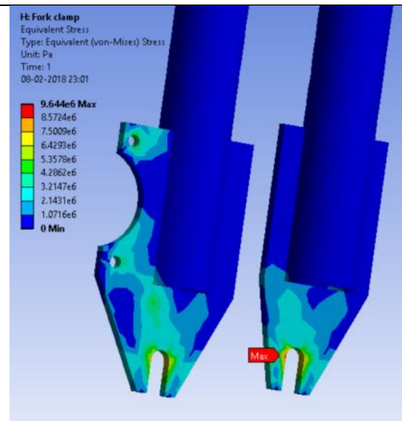
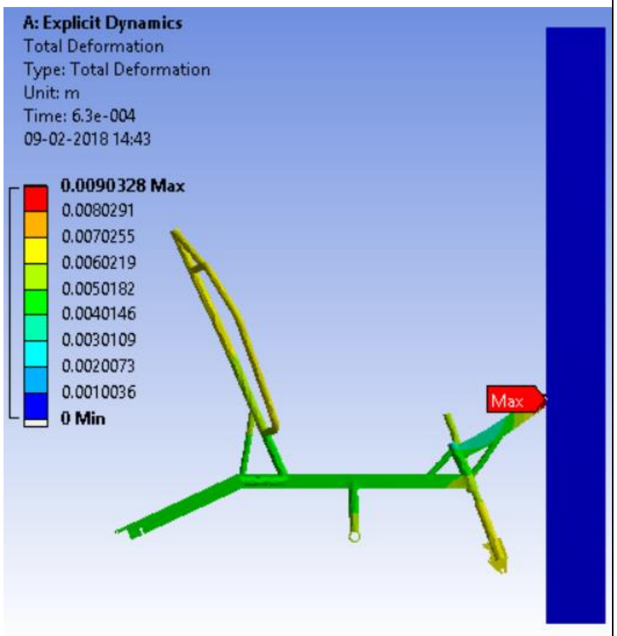
Displacement Plot	Stress Plot	Manufactured
		
The maximum deformation observed is negligible.	The most stressed region is the wheel mounting area.	The modifications provided between the fork and steerer.

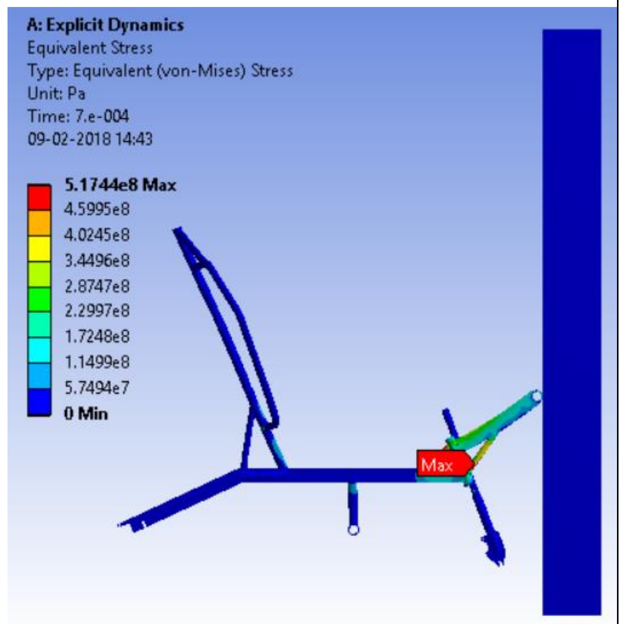
Table 19: Front clamp analysis under braking load

2.5.3 Crash Analysis

Objective and Analysis	The complete chassis was analyzed to determine its behavior under the condition of high speed fatal crash. Explicit Dynamics module of Ansys was utilized for the analysis of this case study.
Methodology	The complete analysis setup was developed in Ansys. Model without the fairing was imported from Solidworks. Analysis was performed. Results were analyzed.

The velocity of 10 m/s was provided for this crash test. A Structural Steel wall was placed at the front of the vehicle for performing crash analysis. The total deformation and equivalent stress plot has been presented below. The modification provided during the structural analysis proved beneficial. The front end was prevented from undergoing drastic deformations. The graph between the energy and the velocity is presented below. Due to the crash, the kinetic energy decreases and the internal energy increases. The result converges, and further during bounce back, the internal energy reduces and the kinetic energy increases.

<table>
<tr><th>Deformation Plot</th><th>Stress Plot</th></tr>
<tr><td>


</td><td>


</td></tr>
<tr><th colspan="2">Results and Discussions</th></tr>
<tr><td>The maximum deformation observed for this case study was 0.90 cm. Due to the supports provided the large deformations were prevented.</td><td>The maximum equivalent stress obtained was at the lower end of the supports provided. The front end experienced a severe stress.</td></tr>
<tr><th colspan="2">Modifications</th></tr>
<tr><td colspan="2">The factor of safety of the front end was 0.8 and it fails under crash test. It is the lowest at the lower part of the support. Therefore, to ensure a safe design the outer diameter of the supporting pipe is changed to 32 mm from 22 mm.</td></tr>
</table>

Table 20: Crash Analysis

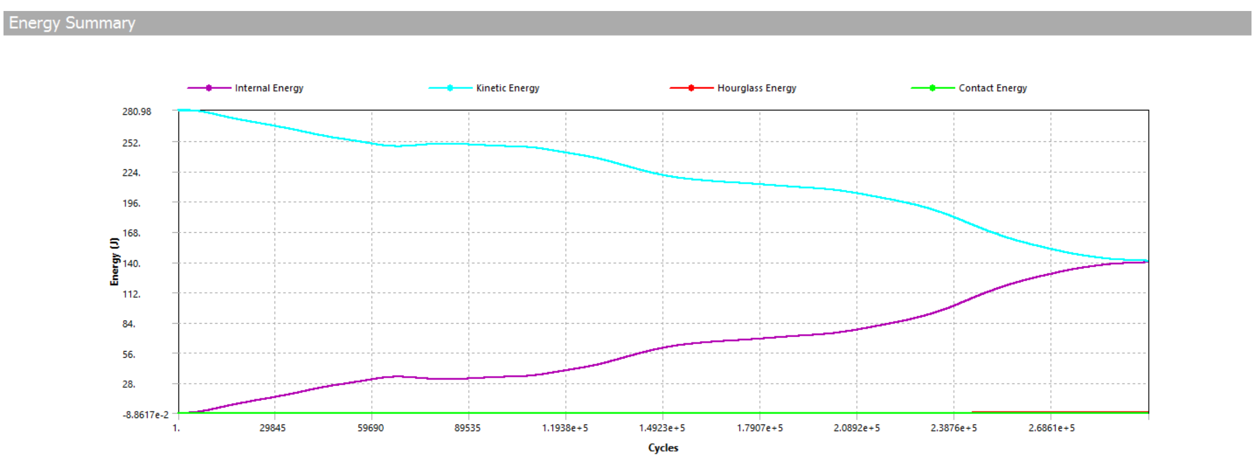


Table 21: Energy vs time plot for the crash.

2.5.4 Rectilinear Motion Analysis

a. Acceleration Analysis:

The acceleration analysis was essential for optimum performance of the vehicle. The major objective of the analysis performed was to understand the characteristics of power and speed for the recumbent bicycle and accordingly assist in deciding the shifting pattern.

Methodology: The driving force available at the wheel was calculated using the equation.

$$S = \frac{T}{R_r} t_{sr} t_{ps},$$

Where, T is the input torque, R_r is the radius of the rear wheel, $t_{sr} t_{ps}$ the transmission ratio for rear transmission and front transmission respectively. The front transmission ratio is fixed. However, transmission ratio for the rear transmission varies depending on the gears selection. The power available at the wheel is calculated as S*V, where V is the velocity of the vehicle. Having calculated all the readings for 21 different gear ratios, the graph between the power and velocity was plotted. The plot also carries the graph of power absorbed due to drag force. The drag force was calculated using the equation,

$$F_D = 0.5(\rho)(C_d A)v^2$$

Where, ρ is the density of the air medium, $C_d A$ drag area calculated in the *performance testing* for vehicle without fairing and v is the velocity. The power is calculated as F_DV. Both the driving power and drag power are represented in the plot below.

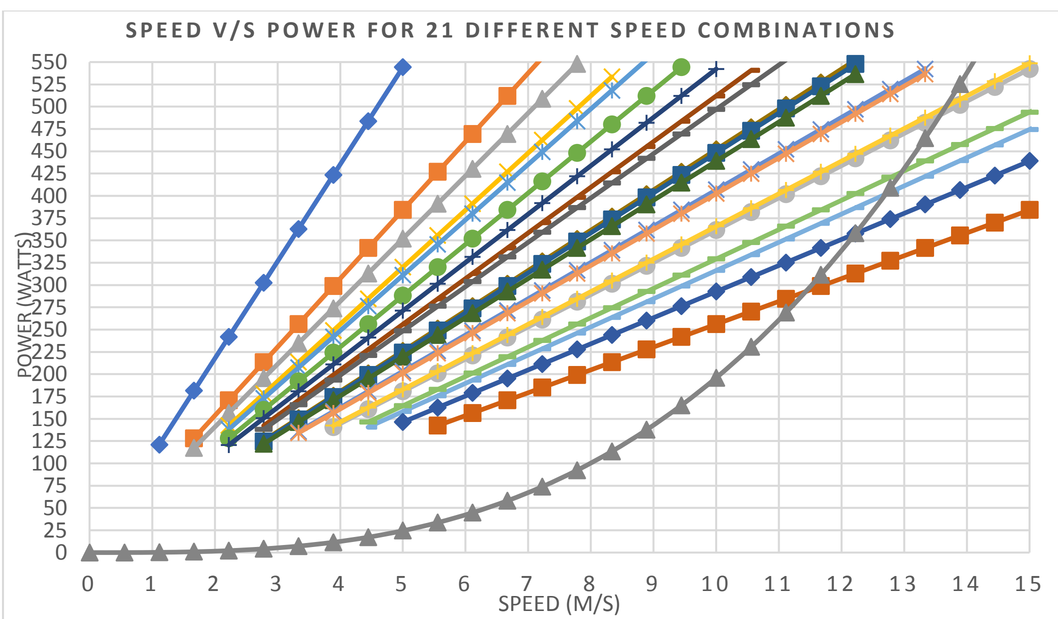


Figure 5: Graph for shifting sequence analysis

Results and Discussions: The above plot carries the data for all the 21 speed ratios. The left most blue color series represents the first gear and successively other gears are presented. The graph was plotted for the input power of 332 watts (Input Force: 373 N, Input rpm: 50). The power range has been fixed from 125 watts to 550 watts. The speed axis ranges from 1 to 15 m/s. For this analysis case, the input power is considered constant throughout the range and further to achieve this constant power, the shift pattern should be followed for the line of 312 watts (332*transmission efficiency calculated in the testing section (0.94)) in the graph. The plot assists in mapping which gear to shift to for a particular speed. As the speed increases the shift should be done in the manner observed in this graph. The inference laid out from the graph is, the maximum speed that can be reached by following this sequence was 11.66 m/s (41.976 kmph) which varies for different power line. If the power is 377 watts, the maximum speed attained was 12.22 m/s (43.992 kmph). To further attain higher speeds aerodynamic fairing is utmost necessary. The lower the drag area the more speed can be attained. The shifting pattern decided was based on the above graph plots with respect to speed. Following this sequence also helps in achieving best acceleration.

b. Braking Analysis

The major objective of this analysis is to calculate the braking force distribution in the front and the rear wheel for the optimum braking.

Analytical Analysis: The equilibrium equation of the horizontal forces and vertical forces acting on the vehicle allows us to present the braking deceleration as a percentage of front and rear braking force. The equations employed are:

$Equilibrium\ of\ the\ horizonatl\ forces\text{:}\ ma = -F_f - F_r$
$Equilibrium\ of\ the\ vertical\ forces\text{:}\, mg -\ N_r - N_f = 0$
$Equilibrium\ of\ the\ moments\ around\ the\ COG\text{:}\ -Fh - N_r b + N_f(p-b) = 0\backslash$

Where, F indicates the sum of rear and front braking force, other parameters were taken from the results of the COG test presented in the testing section. $F_f = \mu_f N_f$, Similarly, $F_r = \mu_r N_r$, where mu is the braking coefficient. On solving the equations, we get:

$$Front\ braking\ \%;\ \frac{F_f}{F} = \frac{\mu_f(b + h\mu_r)}{p\mu_r + b(\mu_f - \mu_r)}$$

$$Rear\ braking\ \%;\ \frac{F_r}{F} = \frac{\mu_r((p-b) - h\mu_f)}{p\mu_r + b(\mu_f - \mu_r)}$$

As we see the braking distribution does not depend on mass but on the geometry. Here p is the wheelbase, h is the height of COG from the base and b is the distance of COG from the rear wheel. The braking coefficient of 0.5 is assumed to be same for the front and rear wheel.

Results and Discussion: The inference laid out from this analysis is that to attain optimum braking distance and force distribution, the braking system must be developed accordingly.

Braking Force Distribution	Tuning

<table>
<tr><td>Front Braking force %</td><td>Rear Braking Force %</td><td rowspan="2">The brake levers are tuned to attain this braking force distribution.</td></tr>
<tr><td>65</td><td>35</td></tr>
</table>

Table 22: Braking force distribution

3. Testing

3.1 Roll Over/Side Protection Testing

Objective: Perform the top load and side load testing on RPS to determine its deflection under stated loading conditions. **Equipment Utilized:** Universal Testing Machine. The tests were performed in the strength of materials laboratory. The results and discussions have been presented in the table below.

<table>
<tr><td colspan="2">3.1.1 Top Load Setup</td><td colspan="2">3.1.2 Side Load Setup</td></tr>
<tr><td colspan="2"></td><td colspan="2"></td></tr>
<tr><td colspan="4">Methodology</td></tr>
<tr><td colspan="2">The lower part of the chassis was fixed rigidly. An initial load of 1600 N was applied at the top point of the RPS and was gradually increased to 2700 N. The total deformation was observed noted. RPS was checked for any severe deformations.</td><td colspan="2">The other side of the RPS was fixed rigidly at the base of the machine. An initial load of 400 N was applied at the top point of the side point and was gradually increased to 1400 N. The total deformation observed was noted.</td></tr>
<tr><td colspan="4">Results</td></tr>
<tr><td colspan="2">Maximum Deflection</td><td colspan="2">Maximum Deflection</td></tr>
<tr><td>Test Setup</td><td>3.95 cm</td><td>Test Setup</td><td>0.30 cm</td></tr>
<tr><td>FEA (without supports)</td><td>4.9 cm</td><td rowspan="2">FEA</td><td rowspan="2">0.24 cm</td></tr>
<tr><td>FEA (with supports)</td><td>3.4 cm</td></tr>
</table>

Table 23: Roll over/side protection test

The inference laid out is that the design is appropriately safe and no further modifications are necessary. The FEA results of the modified design are clearly in close proximity to the observed results presented in this test. The load conditions applied in the test were slightly higher than that specified in the rule book to accommodate the difference in the least count of the machine.

3.2 Developmental Testing

3.2.1 Prototype Development

Objective	To determine the feasibility of the initial design
Methodology	Utilized the data collected from previous years geometry data generation prototype to select the best riding position for this year's requirement and compared it with the rider's experience to develop a prototype.

Discussion: The riding position considered best last year, created a comfort difference for riders with different heights and that limited them to provide optimum input power. This was experienced largely in the drag race and the race outcome clearly highlighted the issue. To overcome this limitation, the data collected last year was analyzed for other good ergonomic positions and iterated to decide the best out of them. The best recumbent angle chosen for further analysis were 60,65 and 70 degrees. The seat position from the primary BB was also considered. This testing assisted in developing an efficient design. The parameters selected for the development of initial prototype of the vehicle are presented in this table.

Specifications decided	**Value**
Recumbent angle, α	**70 degrees**
Angle b/w backrest and line joining hip point to BB, β	**99.606 degrees**
Bottom Bracket (BB) height from seat base	**18.36 cm**
Bottom bracket to hip point distance	**100 cm**

Table 24: Parameters

Conclusion: The decided geometry produced the most comfortable riding position.

3.2.2 Liquid Penetrant Inspection

Objective	Methodology	Results and Discussion
To check the quality of the welds after welding of the frame.	LPI as per ASME standards-ASME Boiler and Pressure Vessel Code Section V, Art 6, Liquid Penetrant Examination.	Welds were analyzed and were found to be sturdy.

In LPI the penetrant utilizes the natural accumulation of the fluid around a discontinuity to create a recognizable indication of crack or another surface opening defect. Capillary action attracts the fluid to its discontinuity as compared to its surrounding.
The area to be inspected is thoroughly cleaned with the help of a cleaner solution. The penetrant liquid is spread on the surface and it is left for 10 minutes to let the dye(penetrant) to swipe into the cracks. Excess penetrant is removed after this, a developer is spread and left for 10 minutes to permit the extraction of the tracked penetrant out of any surface flaws. Surface was inspected under suitable lighting.

3.2.3 Determination of Center of Gravity (COG)

The position of the vehicle's center of gravity plays a significant influence on vehicle's dynamics including braking, acceleration and mass transfer. Therefore, to determine the COG of COLT, this developmental testing was performed. The determined result is used to calculate load distribution of the vehicle.

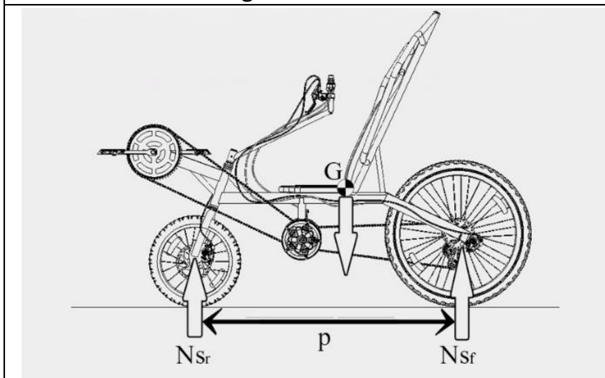
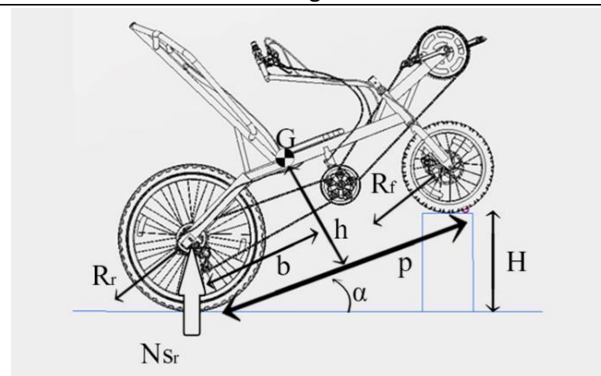
Position of Center of Gravity	
Longitudinal Position	Height
	
Methodology: The longitudinal position of the vehicle without the fairing was measured using two weighing balances, one to measure the reaction of each wheel. The normal reactions were noted down. The position was determined using the noted formula.	Methodology: The height was determined by raising the front wheel of the vehicle to a specified height. The normal reaction at the rear wheel was measured using the weighing balance and the height was estimated using the formula presented below.
$b = \frac{N_{sf}p}{mg} = p - \frac{N_{sr}p}{mg}$	$h = \left(\frac{N_{sr}p}{mg} - (p-b)\right)\cot\left[\sin^{-1}\left(\frac{H}{p}\right)\right] + \frac{R_r + R_f}{2}$
b, is the distance of the COG from the rear wheel h, Height of the COG from the ground p, is the wheelbase of the vehicle. m, is the mass of the vehicle H, is the height of the raised front wheel	N_{sf} , is the reaction at front wheel N_{sr} , is the reaction at rear wheel R_r , is the radius of the rear wheel R_f , is the radius of the front wheel
Results	
***b*, the longitudinal position is: 51 cm**	***h*, the height of the COG is: 62 cm**

Table 25: COG determination test

3.2.4 Frame Deflection Test

The challenges faced during the endurance race were critical while developing the frame. One of such challenges is the rumble strip challenge. No damping or spring systems is present in the vehicle, therefore the forces experienced will be transmitted directly to the chassis. To analyze the stiffness of the final frame this developmental testing has been performed. The Rinard frame deflection test procedure was followed. The test setup was developed and weights (8kg x 2, 5kg x 1) were utilized to apply load.

Tested Vehicle Frame	**Methodology and Description**

COLT	The secondary bottom bracket was rigidly fixed in the horizontal plane. For the front end a load of 21 kilogram of weight was suspended on the front end of the frame. Similar load of 21 kilogram was applied at the rear fork. Anglo meter was utilized to note down the angles and further determine the deflection of the frame in the vertical plane. The difference between the deflection observed before was highlighted.

<table>
<tr><td>Front End Test Setup</td><td>Rear End Test Setup</td></tr>
<tr><td></td><td></td></tr>
<tr><td colspan="2">Results and Discussions</td></tr>
<tr><td colspan="2">Front end deflection: 1.7 cm
Rear end Deflection: 0.8 cm
There were no permanent deformations developed while performing this test.</td></tr>
</table>

Table 26: Frame Deflection Test

3.2.5 Rockwell Hardness Test

The major objective of the test was to determine the hardness of the aluminum plate utilized for the manufacturing of the design. The sample was constrained in the anvil. The indentation was provided with the tool. A major load of 10 kgf was applied on the specimen for 10 seconds. The measured hardness value was displayed on the screen. The experiment was repeated 10 times at different locations and the mean of those readings calculated was 26.7 HRB.

Figure 6: Rockwell Hardness Testing HMI

3.3 Performance Testing

3.3.1 Determination of C_dA for COLT without fairing

The main objective of this performance testing was to analyze the aerodynamic properties of the vehicle without the fairing. The C_dA value gives us a great input for rectilinear motion analysis.

Methodology: The procedure followed for the testing is as follows; COLT was ridden on a flat road at a sustained speed, after reaching a pre-determined value of the speed, pedaling is stopped. The time $\Delta \boldsymbol{t}$ that the vehicle takes to slow down from an initial velocity $(V_{initial})$ to a lower velocity (V_{final}) was noted down. The formula to calculate the value of C_dA was utilized. The experiment was performed for several groups (specifically, two different speed ratios) without making gear shifts during the run and an excel sheet was prepared for the collected data. The independent two sample t-test was performed to analyze these groups of collected data and the value of C_dA was determined.

Theoretical Formulation:

a. Calculation of the C_dA:

$$C_dA = 2\frac{1}{\rho.\Delta t}m^*\left(\frac{1}{V_{final}} - \frac{1}{V_{initial}}\right)$$

where,
ρ, is the density of the operating medium (air, 1.225 kg/m^3)
m^*, is the mass of the vehicle including the rider (87 kg, considering the rotational inertia)

b. The t test statistic value

$$t = \frac{m_A - m_B}{\sqrt{\frac{s^2}{n_A} + \frac{s^2}{n_B}}}$$

c. Common Variance

$$s^2 = \frac{\sum(x - m_A)^2 + \sum(x - m_B)^2}{n_A + n_B - 2}$$

The t test value determined was compared with the critical value in t-test table for alpha level of 5%. The degrees of freedom (df) used is: $n_A + n_B - 2$. The two groups were named A and B, n (10) is the size of the groups and m is the mean of the groups.

Results and Discussion: The null hypothesis $m_A = m_B$ was accepted and the C_dA determined from the test was 0.32 with a confidence interval of 95%.

3.3.2 Transmission Efficiency Test

The complete transmission was assembled and tuned by the team and therefore it was highly necessary to determine the transmission efficiency of the system. From last years' experience, we decided to incorporate multiple gear ratios (21). As the chain orientation changes for different gear ratios, the efficiency varies. Determining the efficiency was the major objective of the test. This test also helped us to find out which of the intermediate chain rings matched with rear cogs so that wrong shifts can be avoided to obtain the most optimum ratios for higher efficiency.

Methodology: The wheels were lifted above from the ground. The input speed (RPM) was provided and measured at the pedals. The output speed of the wheel was noted. The speed was measured using Contactless Tachometer. The data was entered in an excel sheet for further analysis. The experiment was repeated 5 times for each gear ratio. Therefore, the total of 105 readings were noted.

Statistics Employed: To compare all the data for ensuring correct results, one-way analysis of variance (ANOVA) was used to determine whether there are any statistically significant differences between the means of 21 groups of readings noted earlier. The null hypothesis considered was that the population means are all equal. A significance level of 0.05 was used to state the results.

Results and Discussions: The transmission efficiency of 94 % was obtained from the result. The best result came for the gear combination of 34 teeth in the front chainring and 20 teeth chainring in the rear cassette.

3.3.3 Field of View

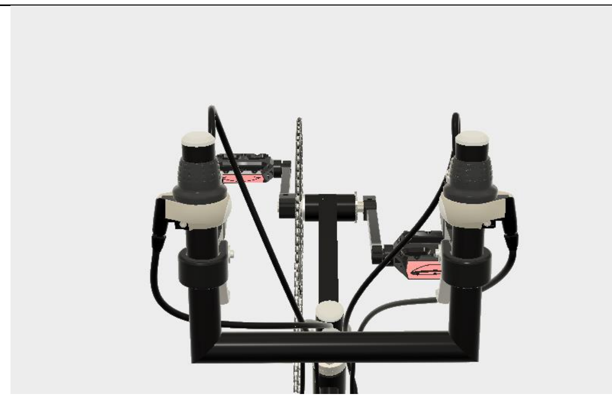
The rider's execution and security depend generally on the field of view amid riding. Typically, the field of perspective of a human eye is 200 degrees in the horizontal field and 100 degrees in the vertical field. The vehicle design was composed keeping in mind this fact. In this manner 200 degrees of level field of view can be effortlessly accomplished in the plan, yet it even relies upon the individual rider. However, because of the fairing in the front, the field of view is blocked. It gets lessened by some specific degrees. The fairing is lifted by 50 cm from the seat base.	

Table 27: Field of View

4. Safety

4.1 Hazard Analysis

Hazards	Possibility	Risk level	Description	Control Measures
Components Entanglement	Moderate	Medium	Body parts, long hair, free dress and gems may end up noticeably entrapped in moving parts.	Every single moving part in the vehicle are fittingly monitored.
Material Reliability	Minor	Low	<u>Risk of injury due to:</u> Mobility of the HPV Crash or collision (e.g. at high speed) Brake or structural failure.	Low Centre of gravity to ensure stability of HPV. Incorporate appropriate safe design features (e.g. RPS, seat belt etc.). Ensure helmet, knee/elbow pads and appropriate footwear is worn.
Instability of vehicle	Minor	Low		
Braking Force	Minor	Medium		
Roll protection system	Minor	Medium		

Table 28: Hazard Analysis

4.2 Rollover/Side Protection System

It has been designed & tested to provide the maximum stability of the vehicle and ensures high class safety for the rider.

4.3 Sharp Edge, Protrusion or Pinch Point

Any sharp edges while manufacturing and assembling were either chamfered, granulated or secured with adjusted edging.

5. Conclusion

5.1 Comparisons

The comparisons presented is between the aim and the testing, experimental results. The required goals were met.

Table 29: Comparisons

Metric	Marginal/Target Values	Actual Values	Justification
Factor of Safety	4/10	8.7	Structural Analysis
Weight (Kgs.)	20/16	17.4	
$C_dA(m^2)$	0.08/0.06	0.0751	Aerodynamic analysis
Speed (kmph)	35/45	43	Gearing Analysis
Cost (rupees)	20000/25000	23122	Cost Analysis
Salom Test	Pass	TBD	Endurance Tests
Safety	Pass	TBD	Safe design
Shifting Sequence	Optimum	Selected	Acceleration Test
COG	Low	Obtained	COG Test

5.2 Evaluation

During the design of Colt all the Product Design Specification(PDS) criteria were taken care of and settled on indispensable planned choices and to guarantee every significant value were met. After making the vital amendments to the design, Colt effectively surpassed for all measurements and inspections that appeared during its testing phase for example Strength, environmental impacts, C_d*A (m^2), cost, etc. The outline objectives for Colt were to make a lightweight, creative vehicle that amplifies speed, rider security and comfort, low a cost vehicle. These were altogether constrained under the ASME HPVC Asia Pacific rules. As a team we believe that we have incorporated the best aspects of previous designs along with new ideas and concepts in a human-powered vehicle that we feel will be the most reliable and refined design to meet the goals.

5.3 Recommendations

Team Anant Recommends doing proper material and aerodynamic analysis and testing during and before designing the vehicle. While the material analysis done, concentrates on minimizing material failure, the team has little data as far as satisfactory measures of distortion is concerned. These perils and misshapenness will influence and attributes to the overall manufacturing of the vehicle and have a tendency to contrarily influence rider's comfort and safety. Hence, Team Anant ought to deal with building a perpetual learning base of exact stacking and loading situations for testing and analysis purpose.

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Appendix: Gantt chart

	Task	Assigned To	Start	End	Dur	%
	COLT		7/9/17	1/27/18	139	
1	Discussion on past experience		7/9/17	7/26/17	13	
1.1	Positive take away		7/9/17	7/14/17	5	
1.2	Rectification of failures		7/16/17	7/20/17	4	
1.3	Future steps discussion		7/21/17	7/26/17	4	
2	Pre-Design phase		7/27/17	9/16/17	36	
2.1	Ergonomics data generation		7/27/17	8/3/17	6	
2.2	Drive train Calculations		8/3/17	8/13/17	7	
2.3	Brake Calculations		8/14/17	8/21/17	6	
2.4	Vehicle Dynamics calculations		8/22/17	8/24/17	3	
2.5	Destructive Testing of Welding Process		8/24/17	8/26/17	2	
2.6	Ordering primary tools and components		8/27/17	9/3/17	5	
2.7	Pre-Design Study		9/4/17	9/12/17	6	
2.8	Selection of material and cross section		9/13/17	9/16/17	3	
3	Design Phase and Analysis phase		9/17/17	11/18/17	44	
3.1	Design of Frame Geometry and other components		9/17/17	10/22/17	24	
3.2	Simulation and verification of Frame, RPS and various components.		10/3/17	11/9/17	27	
3.3	Innovation Decision and design		9/21/17	10/13/17	16	
3.4	Innovation Analysis		9/27/17	10/30/17	23	
3.5	Decision on assemblies and their compatibility		10/31/17	11/9/17	8	
3.6	Set Purpose		9/17/17	9/18/17	1	
3.7	Justification of choices		9/19/17	9/22/17	4	
3.8	Modeling of Aerodynamic Fairing Design		9/30/17	10/21/17	14	
3.9	Fairing Design Analysis and Iteration		10/7/17	10/27/17	14	
3.10	Implementing required Improvements and finalization.		10/28/17	11/18/17	15	
4	Manufacturing Phase		11/12/17	1/3/18	35	
4.1	Task		11/12/17	11/13/17	1	
4.2	Preparation		11/14/17	11/16/17	3	
4.3	Profiling and Cutting		11/17/17	11/19/17	1	
4.4	Bending Operation		11/17/17	11/20/17	2	
4.5	Sheet Preparation		11/21/17	11/23/17	2	
4.6	Bending operation (Chennai)		11/23/17	11/24/17	1	
4.7	Procuring Billet material		11/26/17	11/27/17	1	
4.8	Component(s) manufacture ordering		11/26/17	11/28/17	2	
4.9	Preparation of Fairing Skeleton		11/29/17	12/1/17	3	
4.10	Analyzing the Timeline for Welding Date		12/2/17	12/4/17		
4.11	Integration of Skeleton to Frame		12/4/17	12/10/17	5	
4.12	Assembly of parts		12/20/17	1/3/18	9	
5	Testing		1/4/18	1/20/18	11	
5.1	Analyzing		1/4/18	1/5/18	2	
5.2	Trial Run(s)		1/6/18	1/9/18	2	
5.3	RPS Testing		1/10/18	1/12/18	3	
5.4	Transmission Efficiency		1/13/18	1/15/18