Critical Design Review Report Cover Page & Vehicle Description Form

Human Powered Vehicle Challenge

Competition Location: Digital

Competition Date: 27/02/2021

This required document for all teams is to be incorporated into your Critical Design Review Report.
Please Observe Your Due Dates; see the ASME HPVC website and rules for due dates.

Vehicle Description

University name: Kalinga Institute of Industrial Technology
Vehicle name: Vulcan
Vehicle number: 53
Vehicle Configuration:

Upright Semi-recumbent: Fully recumbent
Prone Other(specify)

Frame material: Chromoly AISI 4130
Fairing material(s): Carbon Fiber GSM 200
Number of Wheels: 2
Vehicle Dimensions(m)

Length: 1.950 m
Width: 0.600 m
Height: 1.355 m
Wheelbase: 1.395 m

Weight Distribution (Kg)

Front: 42.7%
Rear: 57.3%

Total Weight (Kg): 20Kg
Wheel Size(inch)

Front: 20 inches
Rear: 26 inches

Frontal area(m^2): 0.21 m^2
Steering (Front or Rear): Front
Braking (Front, Rear, or Both): Both
Estimated Coefficient of Drag: 0.19

Vehicle history (e.g., has it competed before? where? when?):
No, The Vehicle has not competed in any event at the time of submission of the design report.

KIIT UNIVERSITY

(Declared U/S 3 of UGC Act, 1956)

Bhubaneswar, Odisha, India

TEAM

KETAV

PASSION.AMBITION.DETERMINATION

ASME HPVC TEAM

PRESENTS

VULCAN

VEHICLE#53

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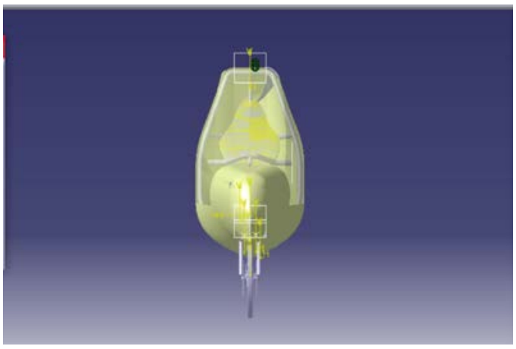
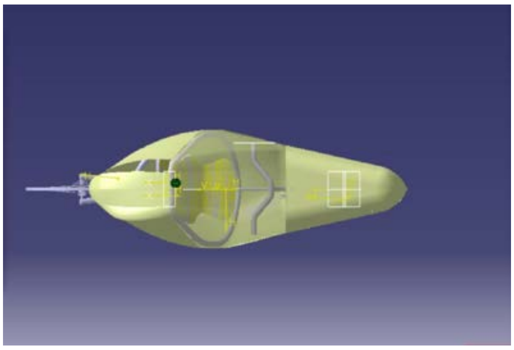
TRINANJANA BAGCHI
ANANT SHRI KAUSHIK
SATTWIK SINGH
VAISHNAVI JAISWAL

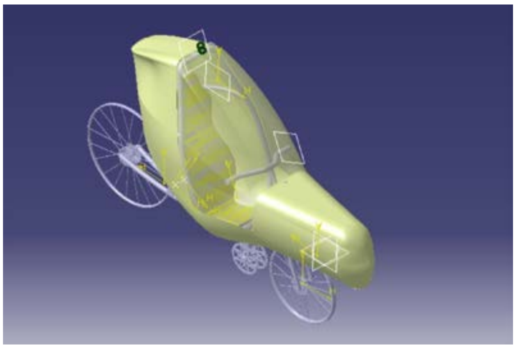
3D VIEW



Front View

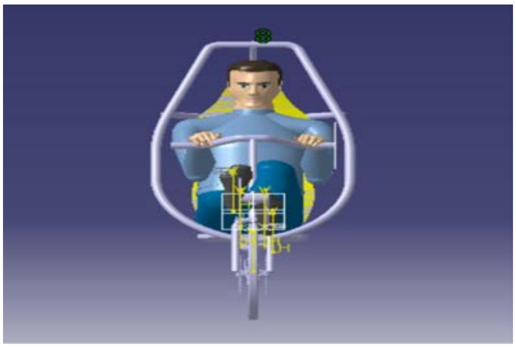


Top View

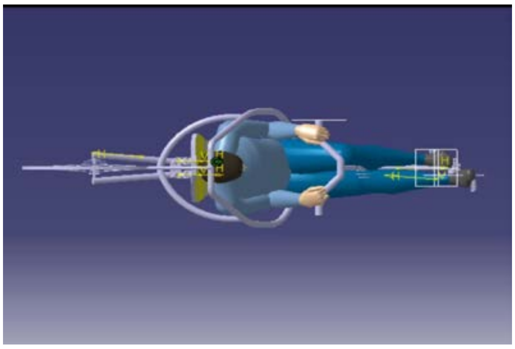


Isometric View

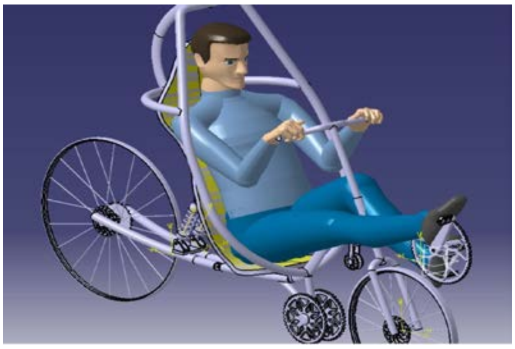
Side View



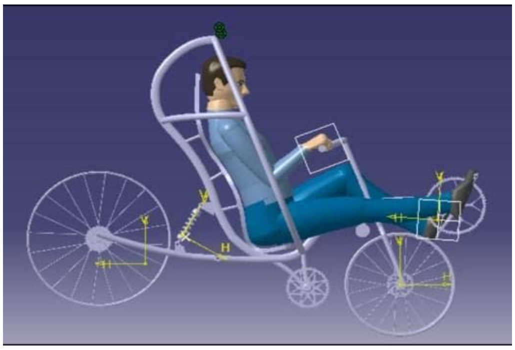
Front View



Top View



Isometric View



Side View

Abstract

From the past few years, the world has started recognizing the value of human-powered vehicles as sustainable transportation, and with increasing demand for fossil fuels, the rise in demand for clean renewable power sources has also been observed. Team Ketav has put its best ideas and innovations into the cycle and tried to demonstrate remarkable engineering design and vehicle performance by making a lightweight, efficient, and agile human-powered vehicle that can safely and effectively be used for everyday transportation. The team has tried its best to practice and implement the principles of competition and rules of HPVC E-Fest Asia Pacific 2021 which require the vehicle to excel in speed, efficiency, handling, practicality, reliability and safety. Pulleys along with standardized equipment are being used for the drive train. Chromoly AISI 4130 has been selected for the chassis due to its high tensile strength and lightweight. Aluminium T6-6061 has been used to create lightweight and durable pulleys. Team Ketav has worked extensively on the various aspects of designing and engineering along with rigorous analysis via CATIA V-5 R-21 and ANSYS 2020 R-2 student version. The vehicle weighs 20kg and is a two wheeled, rear wheel drive fully recumbent bicycle with a field of vision of 240 degrees. We made sure that our product was as minimalistic as possible to maximize efficiency. Aerodynamic considerations also shaped our decisions. As a conclusive note, we believe we have successfully achieved our initial objectives while conceding that several improvements are imminent.

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VI. DESIGN

VI.A. Objective

Team KETAV has designed Vulcan during 2020-2021 season with the end goal of idealizing our statement of purpose: "To manufacture a recumbent bicycle, keeping in mind an appropriate extent of rider's comfort without hampering the safety of riders and also showcasing a good degree of efficiency under vigorous terrain conditions, thereby improving the scope of HPVs in the technological arena". The vehicle, overall is structured and manufactured keeping a reasonable harmony between the safety features and aesthetics.

VI.B. Background Research

Before deciding the design of Vulcan, all the team members researched on the various improvements that can be implemented in our new vehicle. We went through various research papers and videos on HPVs to get a better idea of what new features we can include in our HPV to make it stand out from the rest. The team members gathered information about the effect of various types of stresses on the RPS and frame of the vehicle from the book 'Theory of Machines' by S.S. Rattan and ''Analytical Mechanics of Gears' by Earle Buckingham helped to polish our knowledge on gears. After studying the performance of our previous vehicle, we decided to increase the ground clearance to improve stability and better off road performance. We have also introduced 3 cranks, one bigger crank of 58 teeth and 2 smaller cranks of 34 teeth each which is connected with the jackshaft. After sufficient research we have decided to keep the number and material of the pulleys same as of previous year i.e., two Aluminium T6-6061 pulleys have been used. Clamps have been added on the pulleys to minimize chain derailment. We have kept all the rules and constraints of HPVC in our mind while designing our vehicle.

VI.C Prior Work

This year the team has switched to a rear wheel drive for improved traction and better performance. Other characteristics like the chassis design, use of Chromoly AISI 4130 as the material of the vehicle have remained the same due to its high tensile strength and lightweight. Aluminium T6-6061 pulleys have been used due to their lightweight and durable nature. Three Cranks have been added to the pulleys to avoid chain derailment: one bigger crank of 58 teeth and two smaller cranks of 34 teeth. The size of the front wheel is 20 inches and the rear wheel size has been changed to 26 inches from last year. A Double spring suspension system has been installed this year for better off road performance. The design of the vehicle was also influenced by various Aerodynamic analyses. The seat has been fabricated with FRP, chalk powder and resin just like the previous years.

VI.D. Organizational Timeline

S.no.	Work to be done	Time taken	status
1.	Formation of team and orientation to new recruits	29/08/2020-02/09/2020	completed
2.	Previous report analysis	07/09/2020-13/09/2020	completed
3.	Research on transmission design	15/09/2020-30/09/2020	completed
4.	Scrutiny chassis design and material specification	01/10/2020-7/10/2020	completed
5.	Discussion on fairing design and material specification	9/10/2020-15/10/2020	completed
6.	CATIA modelling of vehicle	17/10/2020-26/10/2020	completed
7.	ANSYS analysis (RPS, Fairing, Structural, others)	27/10/2020-10/11/2020	completed
8.	Presentation	11/11/2020-20/11/2020	completed
9.	Discussion on vehicle's innovation	16/10/2020-6/11/2020	completed

VI.E. Design Specifications

Metric	Importance	Units	Marginal value	Ideal value	Methodology
Braking Acceleration	3	m/s^2	<2	-6.87	Braking calculation
Braking distance	5	m	<15	11.3	Braking calculation
Ground clearance	2	mm	<110	106	Designing
Turning Radius	5	m	5	3.3	Steering calculation
Drag Coefficient	4	Unitless	0.3	.19	Design flow Simulation
Riders view	5	Degree	180	240	Designing
Weight	4	kg	<25	20	Use of lightweight materials
Safety (RPS Load)-Total deformation	5	cm	0.00072(TL) 0.0012(SL)	5.1(TL) 3.8(SL)	Design analysis

VI.F. Concept Development and Selection Methods

Rider Configuration Selection: To arrive with the most efficient vehicle from every possible perspective such as rider's comfort, vehicle performance, cost factors etc. the team indulged themselves into research of different rider configurations, comparing the vehicles on the basis of the below-mentioned factors and after a detailed analysis, recumbent vehicle was found to be the ideal vehicle.

					
Parameters	**Weight Score**	**Upright**	**Recumbent Delta Trike**	**Recumbent Tadpole Trike**	**Recumbent Bike**
Aerodynamics	5	2	4	4	5
Comfort	4	3	5	5	5
Rider's Safety	5	2	5	5	5
Design Simplicity	5	5	2	2	4
Vehicle's Weight	4	5	2	2	5
Availability of Parts	3	5	2	2	4
Innovative	5	1	4	5	5
TOTAL		**97**	**109**	**114**	**147**

Table. Weighted decision matrix for frame design

Drivetrain Selection: The drivetrain is one of the most important factors which determine the balance, weight and the speed of the vehicle. The selection of the drivetrain was based on deep discussions, research and testing of the vehicle over 9 factors including maneuverability, weight, design complexity, pedaling force, traction, availability of parts, potential for failure and steering complexity. Rear Wheel Drive was selected as it performed better than Forward Wheel Drive in most of the factors.

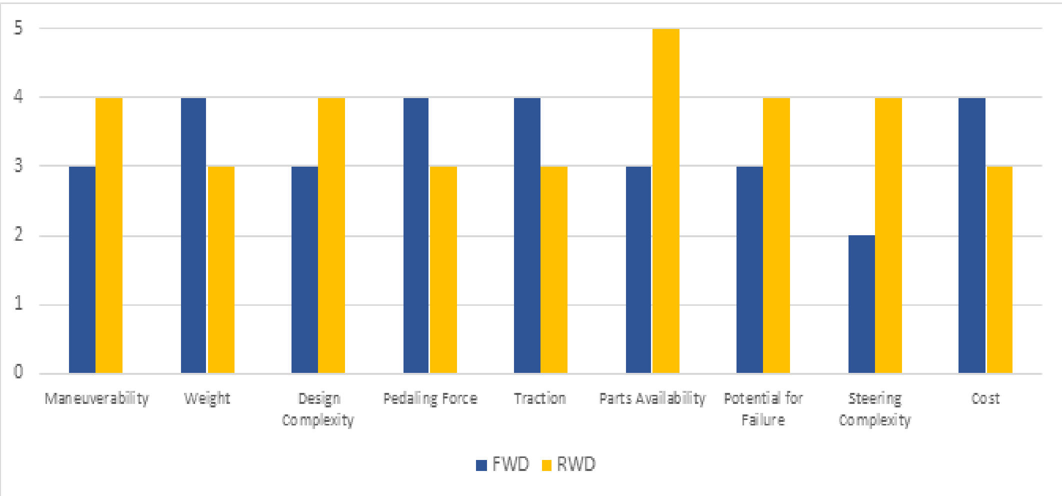

Fig. Forward Wheel Drive Vs. Rear Wheel Drive

Selection of Chassis Material

We have preferred Chromoly AISI 4130 as our frame material because of its better durability, and better tensile strength. Apart from being lightweight, it provides high corrosion resistance and is cost efficient. Chromoly AISI 4130 has good formability, machinability and is quite weldable.

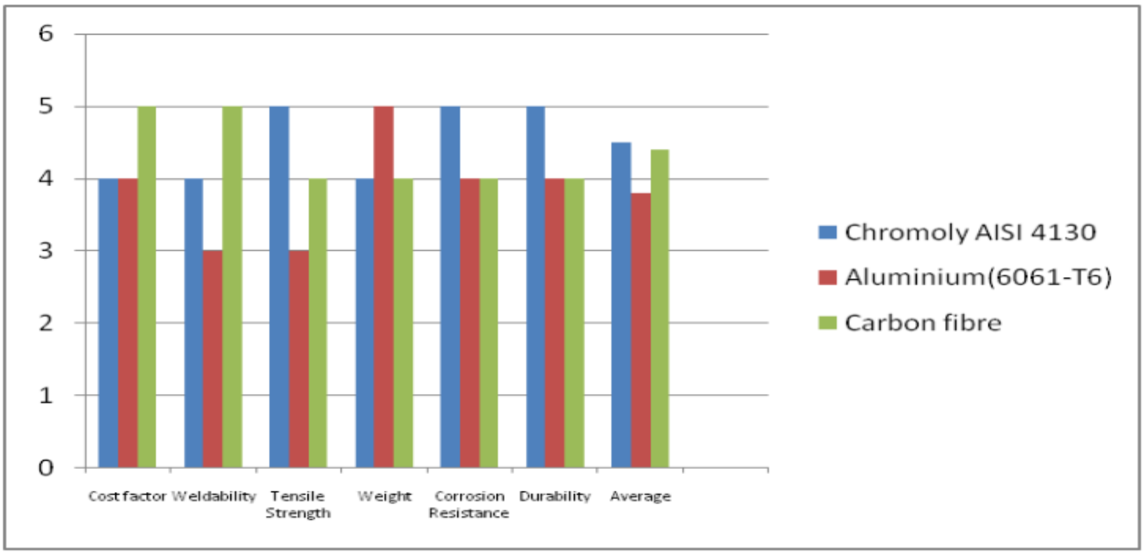

Fig. Chromoly AISI 4130, Aluminium(6061-T6) and Carbon fiber for chassis

STRUCTURED DESIGN METHODS

Team KETAV has used a PUGH chart analysis this year for analyzing the vehicle characteristics with respect to the needs of the consumer.

	1	2	3	4	5	6	7	8	9	10	11	12	13	14		
	Vehicle Characteristics															
Criteria	Roll bar strength	Field of view	Turning radius	Drivetrain Efficiency	Rider's satisfaction	Weight	Vehicle Volume	Number of Parts	Manufacturing Time	Repair Time	Gear Shifting	Cd * A	Cost	Mean time between failure	Totals	Rank
Minimal rider's injury	+	+	+		+	-									3	2
Rider's surrounding awareness		+													1	6
Handling	0		+	+	+			0							3	1
Quick Acceleration				+		-	0				+	+			2	4
Durability	+			0	0					0				+	2	3
Feasibility of construction								+	-	+			0	+	2	4
Easy and quick repairing								0	0	+					1	6
Low cost					0								+	-	0	8

Fig. Pugh Chart

+ signifies a better performance as compared to the baseline performance indicator; **0** signifies that performance observed is approximately at par with the baseline performance indicator; and - indicates a below par performance observed with respect to the baseline performance indicator. Based on the results interpreted by the graph, it can be interpreted that vehicle handling, rider safety and the vehicle's durability have been given maximum priority during the design of Vulcan.

VI.G. Vehicle design Specification:

VI.G.1 Chassis and RPS

The chassis design basically serves the purpose of ensuring the safety and comfort even for longer distances and ensuring durability. This time our vehicle is rear wheel drive as the front drive takes up a lot of space in the front making the fairings blunt which is not desirable as it affects the aerodynamics of the vehicle. We have used Chromoly AISI 4130 owing to its light weight and high tensile strength. Ground clearance has been increased by 10 cm. This year also we have used lower weight Aluminium alloy for wheel sets. The backrest angle was kept at 123° with the horizontal while designing the prototype. On the mainframe, mounts have been welded for installing the seat belts. Keeping the HPVC constraints in mind, a structured roll over protection system has been linked to the chassis to protect the upper part of the rider by absorbing high speed impacts.

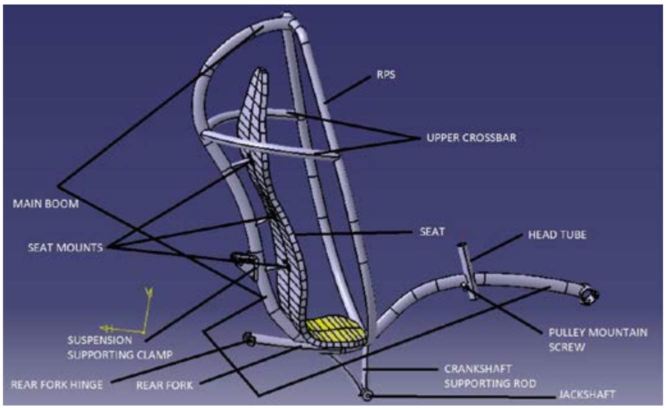

Fig. Chassis

VI.G.2 Suspension

A double spring suspension has been used on the rear wheel of the vehicle, to facilitate a smooth ride for the rider because of its ability to absorb sudden impacts during the vehicle's working. The team has used two spring coils attached to horizontal plates which in turn have been attached to the rear wheel. Hydraulic flow is maintained across the suspension system to dampen the shocks efficiently during travel and also to avoid compression of the suspension system due to the self-weight of the vehicle and rider. Keeping Indian road conditions in mind, the team has decided to use a double spring suspension.

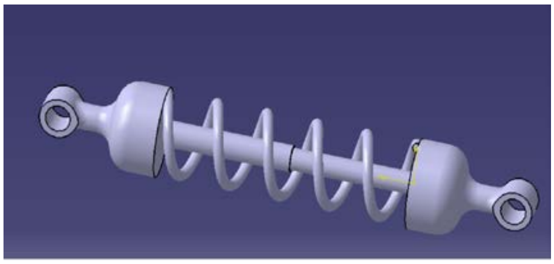

Fig. Suspension

VI.G.3 Fairing

The main purpose of the fairing is to provide an aerodynamic aid to the rider. This year, a modified approach was taken and minimizing fairing weight was considered as the highest priority. Minimizing windshield weight and also surface area required placing the windshield closer to the rider's face thus changing the shape of the fairing from a more gradual frontal slope to one containing a significant slope change

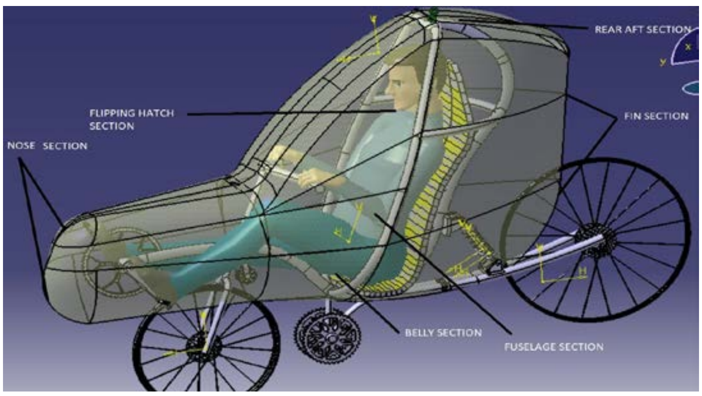

Fig. Fairing

between the rider's face and the front of the fairing. Design material carbon fiber is used again like last year as the main fairing material.

1. Visibility discussion

A primary objective of the windshield is to maximize visibility of vehicle surroundings which include a wide field of view. This year, the size of the windshield of the vehicle was reduced by placing it closer to the rider's head. The field of view for this year's windshield is approximately 180° and through rear mirrors field of view 240°. Additionally, the rider will be able to see the ground at a minimum distance of approximately 18ft in front of the vehicle.

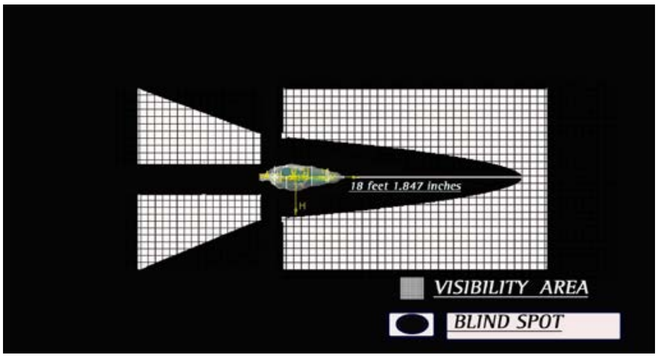

Fig. Visibility area illustration

2. Fairing opening

A front flip opening was designed having a front hinge to allow easy entry and exit from the vehicle. It has a double locking mechanism i.e. can access both from outside and inside. It will be primarily removed by the rider during vehicle use. As the target is to minimize assistance required, the complete flip design was considered. The side slits are provided for ventilation.

3. Communication

In order to maintain safety on the road, the vehicle will contain various features that the rider can utilize to communicate with other riders on the road as well as with his or her team. To communicate with other riders there will be a horn for alerting or passing. Communication with the team while out on the road is carried out using a walkie-talkie system. The system will be located near the rider to minimize the effort to reach it.

VI.G.4 Transmission

Vulcan has a rear wheel drive and the transmission has been described below:

- Multiple cranks of different sizes (58 teeth crank and two 34 teeth cranks) have been used.
- Front wheel has been kept 20 inches and the rear wheel has been changed to 26 inches. Wheelbase has been kept short taking into consideration factors like stability, turning ease and vehicle agility. All the dimensions were chosen keeping in mind the HPVC rules.
- Use of pulleys allowed us to achieve a greater steering angle which is beneficial on track. Clamps are used to cover the pulleys to reduce chain derailment.
- A 7-speed cassette and derailleur have been used.
- This year a double spring suspension system has been implemented in the vehicle for better off-road capabilities.
- Power is transmitted through the pedals via the 58 crank teeth to the small pulleys by chain. Then it is transmitted to the right side of the 34 teeth crank in the intermediate setup. As the intermediate cranks are on the same shaft they rotate uniformly. Then another chain from the 34 teeth crank of the left side goes to the rear derailleur set.

Gear Ratio

The transmission ratios are obtained by following procedures:

(Crank teeth/1st Intermediate Crank) x (2nd Intermediate Crank/Rear sprockets)

No. of teeth in rear sprocket	*11*	*13*	*15*	*18*	*21*	*24*	*28*
Gear ratio 58t chainring	*5.27*	*4.46*	*3.86*	*3.22*	*2.76*	*2.41*	*2.07*

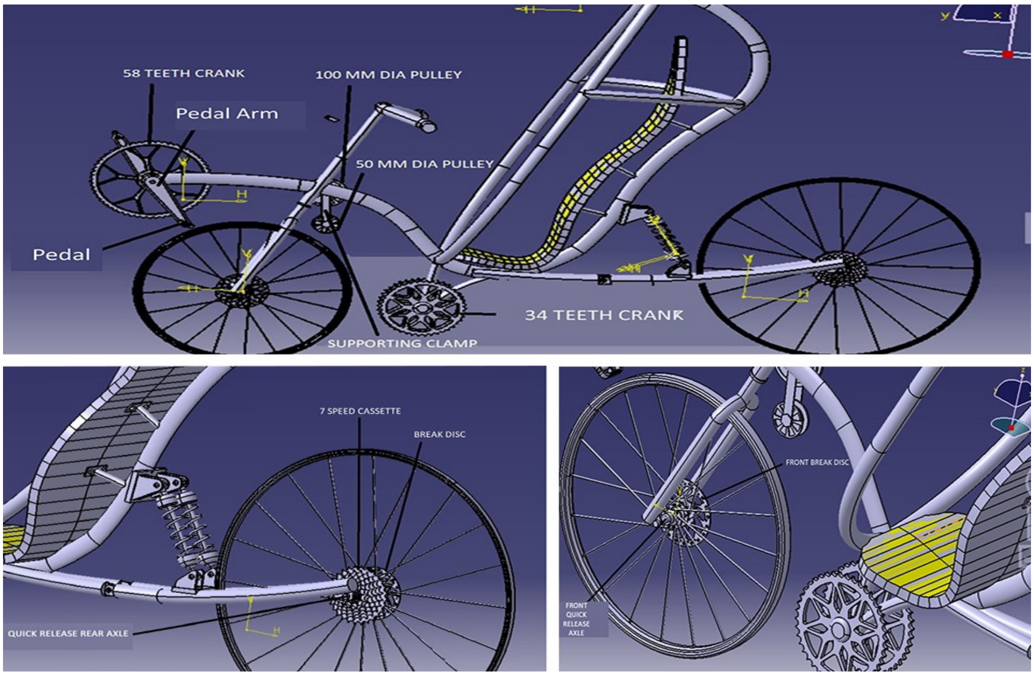

Fig. Transmission System

VI.G.5 Steering

Steering system is one of the most important components of our vehicle. A direct steering system has been chosen that is fastened to the top of the headset. The steering has been designed for optimal maneuverability and ease of handling. The steering handle has been kept straight to fit with the sleek RPS and fairing. The caster angle of the steering is 29° and the length of the steering rod has been increased by 55 cm to accommodate riders of all heights. The material of the steering is Chromoly AISI-4130.

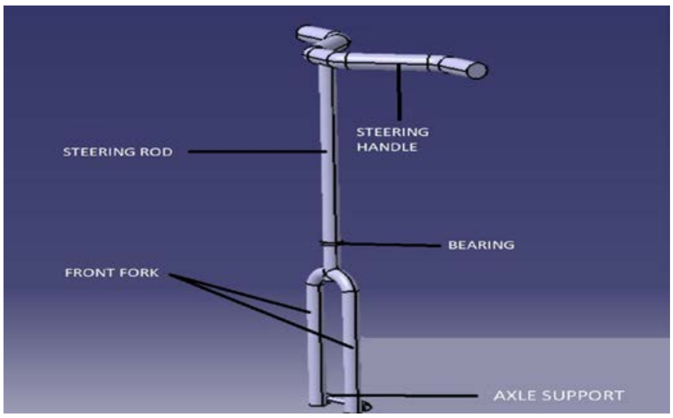

Fig. Steering

VII. ANALYSIS

VII.A. Rollover protection system analysis:

Top load analysis

OBJECTIVE:

RPS top load is utilized to statically stimulate the responses of the framework in a rollover situation with a rider inside. For rider's safety, the RPS must absorb efficient energy and prevent body contact with the ground.

METHOD: As mentioned in the rulebook, a load of 2670 N per driver/stroker was applied at the top of the roll bar at 12 degrees angle from the vertical to the rear of the vehicle. The reactant force was applied downward along the Z direction. Modeling was completed by constraining the seat belt as mentioned in the rules.

CONCLUSION From the above results we observed that the displacement for the given load under the maximum allowable deformation was 0.007 mm whereas equivalent stress was found to be 2.7128 Mpa max.

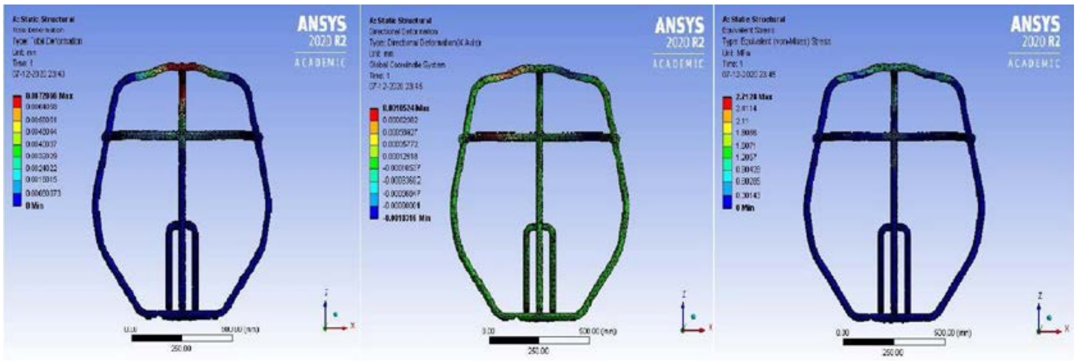

Fig. Top load analysis

Side load Analysis

OBJECTIVE: Side loading of the RPS is representative of a vehicle that has rolled onto its side with a rider strapped into the harness. Analyzing the rollover protection system is of great importance to ensure that it is sufficient to protect riders from serious physical harm in the event of the vehicle tipping.

METHOD: A load of 1330 N was applied at one side at shoulder height in the direction of X-axis and the entire other side was constrained. The analysis was done to obtain the maximum deformation.

CONCLUSION: The Roll bar that can support 1330 N per driver/stroker side load will have maximum elastic deformation of 0.124 mm.

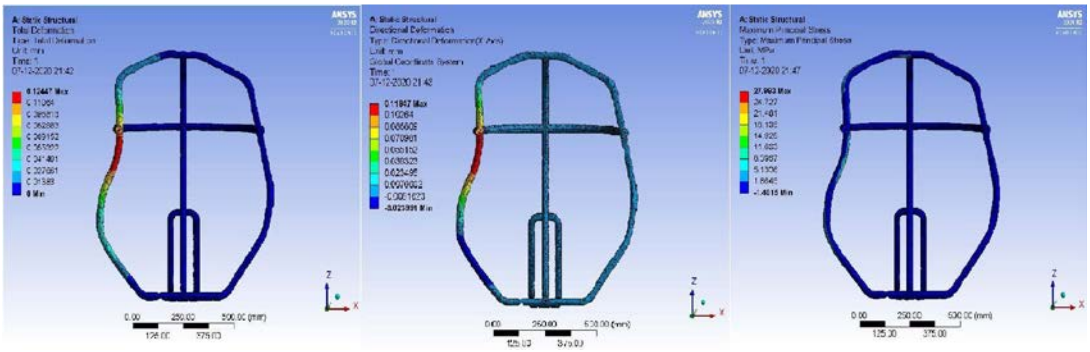

Fig. Side load analysis

Rollover Protection System (RPS) Calculation:

Outer Diameter(d_o)= 25mm   Inner Diameter(d_i) = 18mm   Wall Thickness = 7mm

Moment of Inertia (I) = π ($d_o^4 - d_i^4$)/64

= 14,014.654 mm^4

Modulus of Elasticity (E) = 205 GPa = 205 x 10^3 N/mm^2

Distance from Neutral axis (C) = 12.5 mm

Bending Stiffness = E x I = 205 x 10^3 x 14,014.654

= 0.2873004070 x 10^{10} N-mm^2

Bending Strength (M) = (S_Y x I)/C = (533 x 14,014.654)/12.5

= 5,97,584.847 N-mm

VII.B. Structural Analysis

VII.B.1. Frame Analysis

<u>OBJECTIVE:</u> The objective of this test was to determine the stability of the chassis when tested under the load of the rider. This test was also conducted to find the optimum recumbent angle to maximize the rider's comfort.

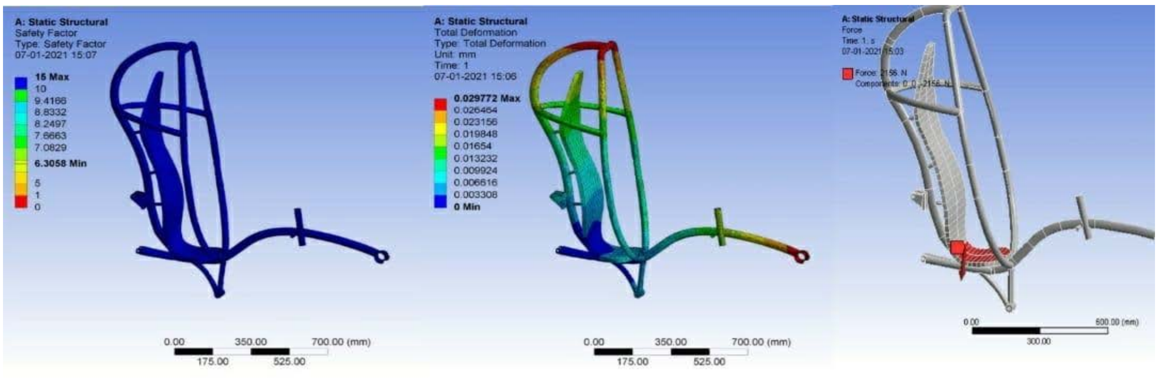

Fig. Frame analysis

<u>METHOD:</u> The test had a simple yet an innovative approach. A driver was assumed to be sitting at an angle of 123 ° from the horizontal i.e.33 ° from the vertical on the seat. Brackets under the seat were kept fixed and a load of 2156N was applied. A load value of 2156N was

applied due to the fact that the maximum weight that was considered was 220kg in the negative Z direction, according to the given configuration as shown in the pictures. The deformation as well as the safety factor of the frame were calculated. Our ultimate aim was to obtain a lower deformation and a higher factor of safety.

CONCLUSION: After thorough analysis the factor of safety and total deformation were found to be 6.30 and 0.016mm respectively. The Ansys results also confirmed that 123 ° was the optimal recumbent angle considered. The structure was proclaimed safe for a rider up to the weight of 220 kg.

Material selection analysis:

Material used: Chromoly AISI 4130

d_o = 35 mm; d_i = 25mm; Wall Thickness = 10mm; Yield Strength (S_y) = 533 MPa

Moment of Inertia (I) = $\pi (d_o^4 - d_i^4)/64 = 3.14 (35^4 - 25^4)/64$

$= 54{,}459.375 \text{ mm}^4$

Modulus of Elasticity (E) = 205 GPa = 205×10^3 N/mm^2

Distance from the Neutral axis (C) = 17.5mm

Bending stiffness = E x I = $205 \times 10^3 \times 54{,}459.375$

$= 1.116417188 \times 10^{10}$ N mm^2

Bending Strength (M) = (S_y x I)/C

= (533 x 54,459.375)/17.5 = 16,58,676.964 N-mm

VII.B.2 Pedal Moment Analysis

OBJECTIVE: In an attempt to comprehend how much force the pedal crank would withstand without failure, pedal crank analysis was conducted.

MODELLING METHOD: One side of the pedal was fixed and a force of 1330N was applied on the Y-axis. After evaluation maximum deformation was found to be 0.19759 mm whereas equivalent stress on the pedal was found to be 29.188 MPa. This helps the rider to perform better when they know the active efficiency of the pedal crank.

Pedaling power Calculation

$P = (x+y).V/n_m$ where, P = power exerted by rider in walls

$x = (C_d A\rho V^2)/2$ x = aerodynamics drag force in vehicle

x = 14.06N n_m = drivetrain mechanical efficiency (0.95)

V = Velocity of vehicle(25km/hr)

$y = C_{rr} * g * M_t$ where, y = +ve rolling resistance force in Newton

y = 0.005 * 10 * 80 = 4N C_{rr} = Coefficient of rolling resistance of vehicle = (0.005)

P = (14.06+4)25/0.95 M_t = total mass of vehicle and rider (80kg)

P = **475.26 W**

Average force on the pedal = 500N

Length of pedal arm = 207.57 mm

Torque at pedal = 500 * 0.207 = 103.3 N-m

Radius of rear wheels = 32.76 cm = 0.32 m

Circumference of rear wheel = $2\pi r$= 2π* 0.32 = 2.01m

i. For maximum speed

Gear Ratio = 1:5.27

Torque at wheel = Pedaling torque * Gear ratio

= 103.3 * 0.18 = 18.594 N-m

Force by wheel on ground = Torque at wheel / Radius of wheel

= 18.594/0.32 = 58.10 N

Distance travelled for 1 cadence = 5.27 * 2πr = 10.59 m

For maximum speed cadence > 100 rpm

Therefore, distance travelled by vehicle for 100 rpm = 1059.59m/min

Maximum speed = 63.57 km/hr

ii. For maximum torque

Gear ratio = 1:2.07

Torque at wheel = 103.3*0.48 = 49.90 N-m

Force by wheel on ground = 49.90/0.32 = 155.94 N

Distance travelled by 1 cadence = 2.07 *2πr = 4.16 m

Therefore, distance travelled by vehicle for 100 rpm = 416.198m/min

Speed at maximum torque = 24.97 kmph

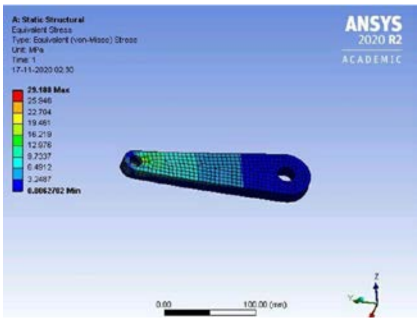

Fig. Equivalent stress

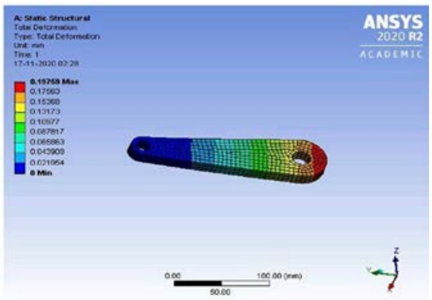

Fig. Total deformation

VII.B.3 Suspension analysis

OBJECTIVE: To calibrate the properties of the suspension system, and to ensure the safety and the stability of the vehicle, especially while travelling on substantially irregular surfaces, the team conducted a suspension analysis on Ansys 2020 R2 academic.

METHOD: The lower part of the suspension was constrained completely, and the force was applied in a positive Z direction from the given configuration. The given configuration is based on the fact that the suspension has been fixed at an angle of 28 degrees from the vertical axis. The force on the suspension was assumed to be approximately 588 N (assuming an average driver weight of 60 kg). The component of the force was considered as per the placement of the suspension. And the total force on one system was divided by 2, because the vehicle has used a double suspension system.

<u>CONCLUSION:</u> Upon analysis, the safety factor was found to be 2.69, hence the design of the suspension is safe. The total deformation of the system upon the application of the load was found to be 0.061mm.

Suspension Calculations:

G = 78600N/m^2; Total coil = 12; Active coil = 10; Total weight = 80kg; Suspension% = 65,

d_{wire}= 5mm; d_{coil}= 37.53mm

Single shock absorber force = W/2 = 80 * 9.8/2 = 254.8N

Spring Compression(S) = $W*d_{coil}^3*n/G*d_{wire}^4$; C = spring index = d_{coil}/d_{wire} = 37.53/5 = 7.53

S = 784.8 * $(37.53)^3$ * 10/78600 * 5^4 = 8.44mm

Solid length = $n*d_{wire}$ = 12 * 5 = 60mm

Free length = solid length + clearance b/w adjustable coils = 60+8.44+(8.44*0.01) = 68.52mm

Spring rate(K) = W/8 = 254.8/8.44 = **30.18 N/mm**; Pitch = 24.24mm

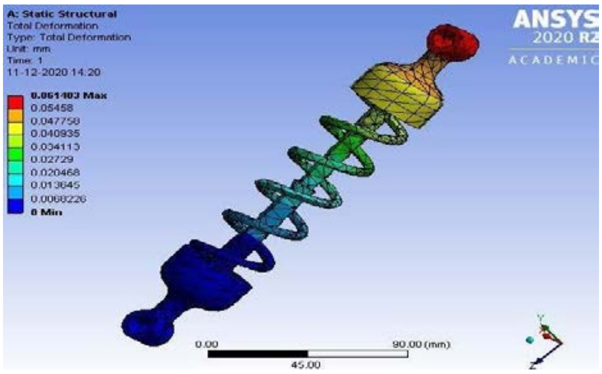

Fig. Total Deformation

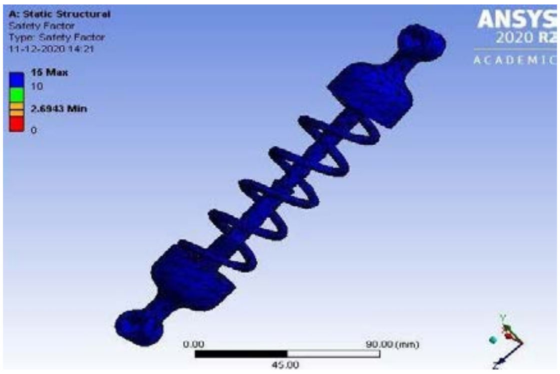

Fig. Safety factor

VII.B.4 Bottom Bracket Analysis

<u>OBJECTIVE:</u> The bottom bracket analysis was crucial for determining the overall feasibility of the design as it is a critical component in a drivetrain.

<u>METHOD</u>: The head tube was fixed while performing the analysis via ANSYS. On applying a load of 1300N in the positive Y-axis and a load of 300N in the negative Y-axis was applied.

<u>OBSERVATION</u>: A maximum total deformation of 0.145 mm was observed. Observed factor of safety was 3.70.

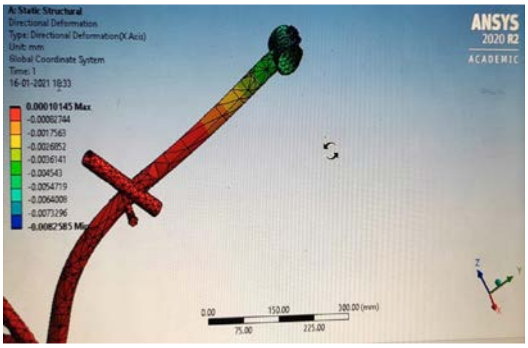

Fig. Directional Deformation

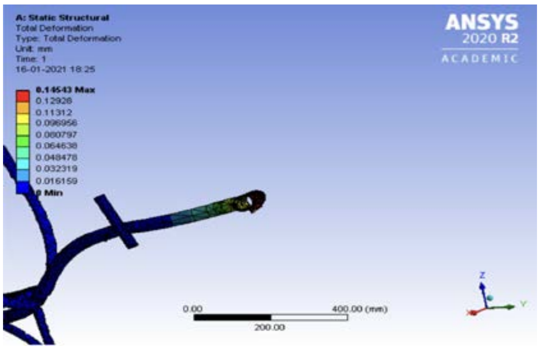

Fig. Total Deformation

VII.C Aerodynamic Analysis:

Aerodynamic analysis was conducted for the vehicle to derive the optimum design and material selection for the vehicle, for best aerodynamic properties. The team compared 2 fairing designs, and made their choice based on the results obtained by conducting the front wind and cross wind analysis for the designs. The software used for analysis was ANSYS 2020R2. Number of iterations for each analysis was set at 500.

Front wind analysis: the front wind was carried out on the vehicle's fairing to test its performance under front wind impact. The wind velocity was kept at 7.0m/s.

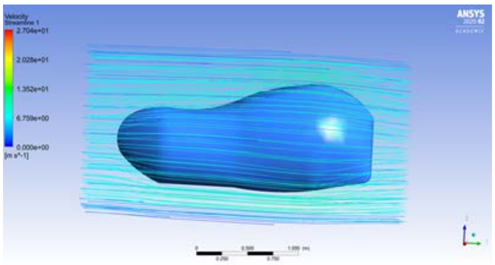

Fig. A

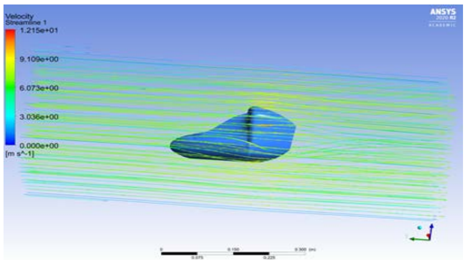

Fig. B

Cross wind analysis: Additionally, analysis was done for the performance of vehicle in crosswinds. The wind velocity was taken to be 5.0 m/s. The value was seen to be C_d falls in magnitude which was due to the increase in area affected by drag in cross wind analysis.

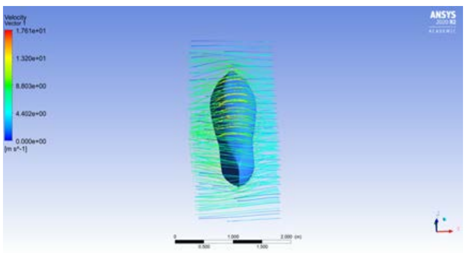

Fig. A

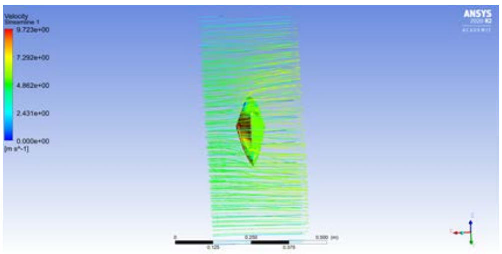

Fig. B

Fairing design	Frontal area	Drag coefficient (ff)	Drag coefficient (cf)
A	0.21	0.19	0.186
B	0.25	0.29	0.265

Based on the results obtained after analysis, fairing design A was observed to have a better drag coefficient for both front wind analysis and cross wind analysis. hence, fairing design A was finalized by the team.

VII.E. Other analysis

Centre of Gravity Analysis

X= Horizontal position of CG from front axle; Y = Height of CG;

W_f= Front Weight; W_r = Rear Weight ; Wb= Wheelbase;

Total Weight of vehicle = 20 Kg; Frontal Weight (W_f)= 8.54 Kg; Rear Weight(W_r)=11.46Kg;

Actual Wheelbase= 139.5 cm; H = Height by which frontal wheel is raised above rear = 38 cm;

R_f = Radius of front tire= 24.65 cm; R_r= Radius of rear tire = 32.76 cm

i. Horizontal Centre of Gravity position:

$X = (W_r * Wb)/(W_f+W_r)$

$X = (11.46*139.5)/(8.54+11.46)$

$= \mathbf{79.93\ cm}$ (from rear wheel centre)

ii. Vertical Centre of Gravity Position:

$Y = Cot[Sin^{-1}(h/Wb)] * [(W_r*Wb)/W_r+W_f) - X)] + (R_f+R_r)$

$= Cot[Sin^{-1}(38/139.5)] * [(11.46*139.5)/(11.46+8.54) - 79.93)] + [(24.65+32.76)/2]$

$= \mathbf{28.705\ cm}$ (from ground)

CG of vehicle (X, Y) = **(79.93, 28.706)** in cm

Turning Radius Calculation

<u>OBJECTIVE</u>

To determine the minimum turning radius of a vehicle.

<u>METHODOLOGY</u>

Formulating trigonometric equations with vehicle design parameters by simulating the vehicle position along a curved path and hence, determining the turning radius. To ensure the turning radius of the vehicle was compliant with the pre-planned design standards and ASME HPVC's rules, the following mathematical analysis was carried out of the design, wherein the instantaneous position of the vehicle along a curved path was used to formulate trigonometric relations in between design specifications and turning radius which are elaborated below.

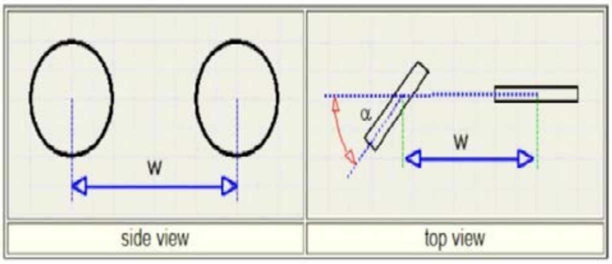

Fig. Vehicle turning geometry.

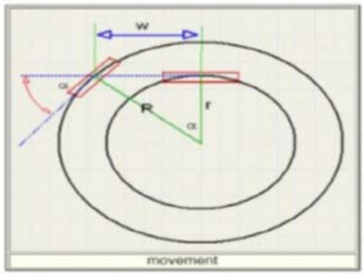

Fig. Vehicle Turning Radius Geometry

Wheel Base, Wb = 1.395m; Maximum Possible Steering Angle, α = 25°
Sin(α) = Wb/R,
i.e., R = Wb/Sin(α)
R = 3.30085 m.

<u>RESULT</u>

Minimum turning radius is found to be 3.30085m which satisfies the team's pre-planned standard as well as the ASME HPVC's standard.

Impact Analysis

Mass of Vehicle: 20Kg; Total mass with the Rider: 80Kg

Frontal Analysis

For Frontal Impact analysis, we consider the vehicle to strike the wall at a velocity of 12.5m/s(45Km/h) and the mass of the vehicle is 80 kg with the passenger.

Calculation of Impact Forces
Momentum(P)=m*V = 80*12.5 = 1000Kg m/s.
Impact Force (IF)= I*P = 1*1000
=1000N.
Let us consider Factor of Safety (FOS)= 1.2
Net force = FOS*IF = 1.2*100 = **1200 N**

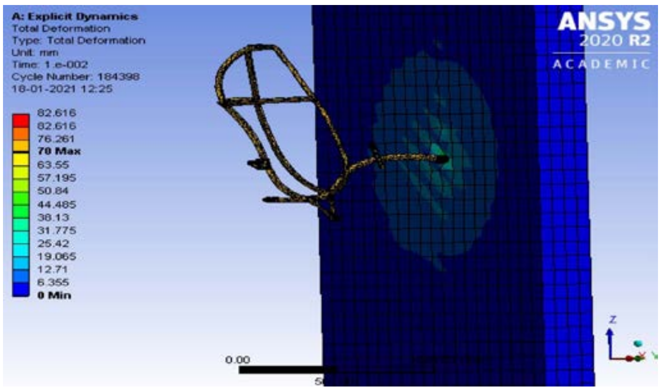

Fig. Front Impact Analysis

<u>Analysis result</u>

Maximum Deformation = 70mm

Side Impact Analysis

For Side Impact analysis, we considered our vehicle to be stationary and hit by an object from the side (Normal to the longitudinal axis). The mass of the object is considered to be 80Kg and the velocity of the object is considered to be 12.5 m/s.

Momentum (P) = m*v = 80*12.5

=1000 Kg m/s

Impact Force (IF)=I*P = 1 *1000

= 1000N

Let us consider Factor of safety (FOS) = 1.2

Net Force = FOS*I = 1.2*1000

= 1200N

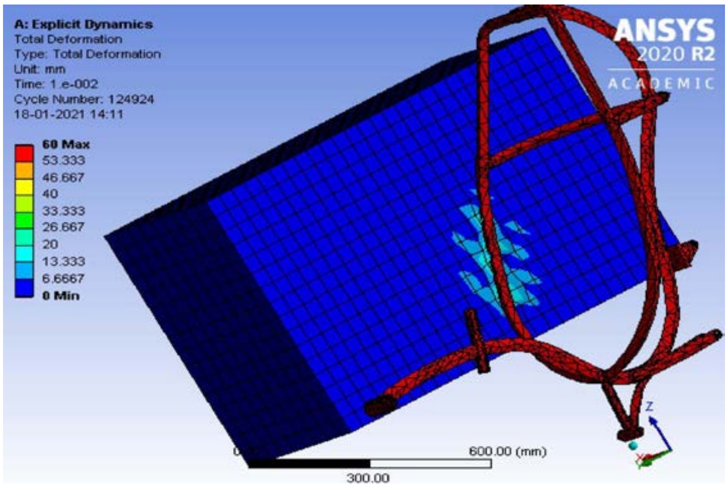

Fig. Side Impact Analysis

<u>Analysis Result</u>

Maximum Deformation = 60 mm

Braking Analysis:

Velocity of vehicle(V) = 45 km/h =12.5m/s; Total mass of vehicle + Rider(m) = 80kg

Kinetic Energy = $1/2mV^2$ where, m = Total mass of the vehicle

$0.5mV^2 = \mu mgx$; V = Velocity of the vehicle

$x = 0.5V^2/\mu g$ x = Stopping distance of the vehicle

$= (0.5 * (12.5)^2)/(0.7*9.81)$ μ= Coefficient of friction between road and tire (0.7)

x = 11.37 m

1.) <u>Deceleration of vehicle</u>

$v^2 = u^2 + 2ax$; where, v = final velocity of the vehicle = 0

$v^2 = u^2 + 2ax$ u = initial velocity of the vehicle = 12.5 m/sec

$0 = (12.5)^2 + 2*a*11.37$ a = acceleration of the vehicle

22.74*a = -156.25

a = -6.87m/s²

2.) Time taken for deceleration

$v = u + at$
$0 = 12.5 - 6.87 * t$
$6.87 * t = 12.5$
t = 1.81 sec
Total time = time taken for deceleration + reaction time of the rider
= 1.81 + 0.3 = 2.11 sec

3.) Brake Efficiency

Brake efficiency (η) = $v^2/3x$ where, v = velocity of the vehicle
= (45 * 45)/ 3* 11.37 x = stopping distance
= **59.36 %**

Stopping distance when vehicle is at 25 kmph

$\eta = v^2/3x$
$x = v^2/3\eta = 25^2/(3*59.36)$
x = 3.50 m

4.) Required brake force

Kinetic Energy = Braking work = Braking force * Stopping distance
$0.5mv^2 = F * x$
$F = 0.5mv^2/x = (0.5 * 80 * (12.5)^2)/3.50$
Braking Force (F) = **1785.71 N**

Maximum retarding force applied by brakes

Retarding force = μW
W = weight of vehicle
Retarding force = 0.7*80*9.81 = **549.36 N**

Drag Force

$F_d = (0.5)\ C_d\ V^2 A\rho$; where, F_d = drag force coefficient; C_d = coefficient of drag;
$= 0.5*0.19*25^2 * 0.21*1.128$ Air density at 25℃ = 1.128kgm^3; Velocity = 25km/h
= **14.06N**

Crank Analysis:

OBJECTIVE

To determine maximum deformation, maximum stress and safety factor of the crank under normal riding conditions.

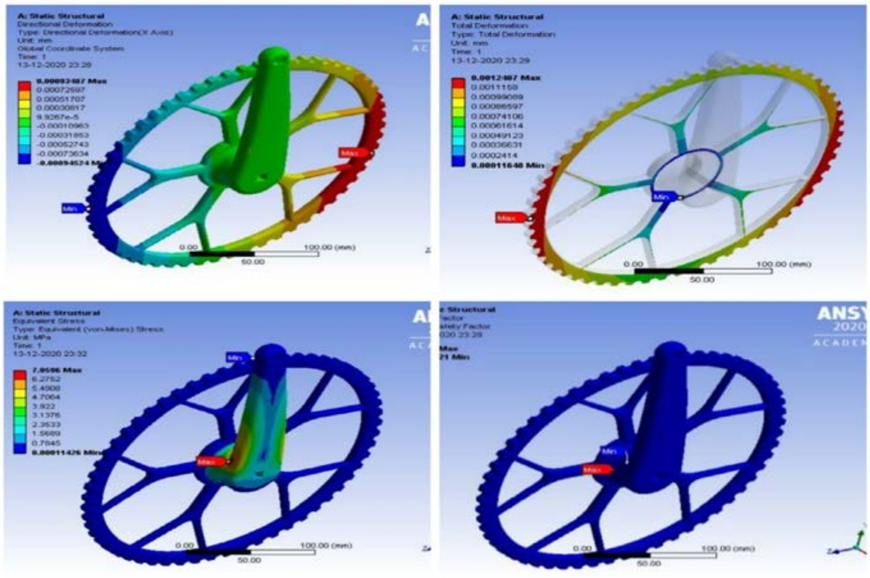

Fig. Crank Analysis

The directional deformation is along the x axis. From the above analysis we can see that the overall deformation was found to be 0.0012 mm and equivalent stress was found to be 7.05 Mpa max.

SWOT analysis

STRENGTHS (+)	WEAKNESSES (–)	OPPORTUNITIES (+)	THREATS (–)
• Rear Wheel Drive • Smooth transmission system due to the use of Aluminium T6 pulleys • High speed • High stability	• Complex design • Proper training and caution required for the perfect riding experience • Low market demand since its an upcoming product • Peripheral view is limited	• Alternative to conventional fuel vehicles • Keeps the riders physically fit and hence spreads health awareness • Environment friendly	• Requires considerable leg strength • Not suitable for physically challenged people • Might be unsafe on heavy duty roads

VII.D. COST ANALYSIS

Components/Parts	Quantity	Unit price(INR)	Total(INR)
Brake			
Clutch Lever	2 pcs	50	100
Brake Wire Cable	2 pcs	35	70
Disc	2 pcs	400	779
Calliper(with calliper mounts)	2 pcs	150+50	400
Frame			
Chromoly	_	6000	6000
Fabrication			
Carbon Fibre	_	_	32000
Aluminium Pulleys	2	1200	2400
4-point seat belt			12,000
Crank-set(58 teeth) Crank-set(34 teeth)	1 2	6000 3000	15,000
Transmission(chain, clamps, jack shaft, quick release axle)	1	6000	6000
Wheel(20 inch) Wheel(26 inch)	2	5000 7000	12000
Safety			
Seat (FRP,Resin)	1 pc	500	500
Bell	1 pc	30	30
Rollover Protection	1 pc	300	300
Other			
Nut/ bolt/ Flat Plate Locknut/ Washer Drill -bit/			600
Total			**88,179**

SAFETY

1. A windshield has been installed in the vehicle whose job, secondary to providing the rider with a sufficient view of their surroundings and shielding the rider from the environment, is to help minimize rider's injury in the case of a crash.

2. When considering a seat belt for Vulcan, four-point seat belts are being used while riding. Important considerations were reliability, ease of use, and convenience in attachment to the frame. Reliability was ensured by choosing a harness with a certified safety rating that satisfies all specifications of ASME.

3. A rollover protection system has been used in the vehicle to protect the riders from injuries if the vehicle overturns. Keeping the rider's safety in mind, all the sharp edges were blunted.

4. The vehicle provides a visibility range of 180 degrees to the rider, which allows the riders to be more aware of their surroundings.

5. Safety accessories: Rear view mirrors and reflectors on both sides have been installed. Helmets, headlights, tail lights and horns are being used while riding.

VIII. Comparison – Design goals and analysis

This year, the team's focus was on designing a better performing HPV. As unfinished at the moment the vehicle is anticipated to be a robust vehicle with good overall performance. All initial design goals were met other than wheelbase which became slightly longer than last year's vehicle due to preference of rear wheel transmission over front wheel and introduction of a trailing arm suspension. Many critically important subsystems were thoroughly analyzed to verify they successfully met the design specifications and competition guidelines.

VIII.A. Evaluation

The vehicle met all goals set forth by the team. To ensure functionality, the vehicle is designed to accommodate riders of height 5'2'' to 6'1''. Adjustability for various riders was achieved by utilizing an adjustable seat. The vehicle weight is approximately 20 kg. The structural design and rollover protection design was evaluated using calculations and finite element analysis (FEA) to ensure the required loading constraints were met.

VIII.B. Scope of Improvement

The vehicle designed has met all specifications set forth by the team. Improvements can be made. Particular improvements could be achieved regarding the drivetrain and steering components of a vehicle. To improve the drivetrain better chain route planning can be done earlier to ensure maximum efficiency from the riders.

VIII.C. Conclusion

Vulcan set out with the target of designing a reliable, functional and safe human powered vehicle with unique features that enables the cyclist to travel conveniently and securely to places. In this era where the environment seeks urgent attention, recumbent vehicles are supposedly to set a good example towards an eco-friendly environment and our team has left no stone unturned to take this step towards the agenda. This year's design was more complex than the previous one but was simple to ensure all components would work reliably and continuously. With vigorous designing and analysis being done, the vehicle is hence concluded to be safe and comfortable from the riders' perspective.

REFERENCES

1. https://pdfs.semanticscholar.org/5ee5/84368629fdc7ad69a3adf63da2c8e90de9f4.pdf
2. “Engineer to Win” (Caroll Smith, 2010). Racing car design, paperback. https://www.amazon.com/Engineer-Win-Carroll-Smith/dp/B011MBDQOM
3. “Race Car Vehicle Dynamics” (William Milliken, 1994) https://www.amazon.com/Race-Car-Vehicle-Dynamics-Premiere/dp/1560915269
4. “Racer’s Encyclopedia of Metals, Fibers & Materials” (Forbes Aird, 1994). Paperback. https://www.amazon.com/Encyclopedia-Materials-Motorbooks-International-Powerpro/dp/0879389168
5. “Competition Car Composites” (Simon McBeath, 2016). Composites design & fabrication handbook, hardcover. https://www.amazon.com/Competition-Car-Composites-Practical-Handbook/dp/1845849051
6. “Sheldon Brown’s Bicycle Technical Info” (Sheldon Brown). Bicycle design and reference guides https://www.sheldonbrown.com/
7. “Design of Human Powered Vehicles” (Mark Archibald, 2016). Extensive HPV discussion, hardcover. https://www.amazon.com/Design-Human-Powered-Vehicles-Mark-Archibald/dp/0791861104
8. “Bicycling Science” (David Gordon Wilson, 2004). Broad introductory cycling resource, paperback. https://www.amazon.com/Bicycling-Science-Press-Gordon-Wilson/dp/0262731541
9. “The Biomechanics of Force and Power Production in Human Powered Vehicles” (Danny Too, Gerald Landwer). Factors affecting power production via recumbent pedaling, online. https://digitalcommons.brockport.edu/cgi/viewcontent.cgi?article=1100&context=pes_facpub
10. “Biomechanics of Cycling” (Borut Fonda & Nejc Sarabon). Literature review of cycling
11. “HPVC Safety Dos and Don'ts” (Mark Archibald, 2016). HPV safety best practices. https://community.asme.org/hpvc/w/wiki/13014.educational-resources.aspx#Safety-Dos-Donts
12. “Lords of the Chainring” (William Patterson, 2012). Technical theory of the handling qualities of Design, Haughton.
13. “How Do You Carry out an Effective Critical Design Review (CDR)?” *Haughton Design*, 5 Apr 2019, haughtondesign.co.uk/how-do-you-carry-out-an-effective-critical-design-review-cdr.

APPENDICES

1. AISI : American Iron and Steel Institute
2. Ft : Feet
3. FF : front flow
4. CF : cross flow
5. 58t : 58 teeth
6. RPS : Rollover Protection System
7. di : inner diameter
8. do : outer diameter
9. E : modulus of elasticity
10. S_y : Yield strength
11. I : moment of inertia
12. C : distance of body from neutral axis/spring index (suspension analysis)
13. P : power
14. C_d : coefficient of drag
15. X : drag force/horizontal position of center of gravity
16. n_m : mechanical efficiency
17. Y : positive rolling resistance/height of center of gravity
18. C_{rr} : coefficient of rolling resistance
19. G : shear modulus
20. N : number of active coils
21. d_{wire} : wire diameter
22. d_{coil} : coil diameter
23. S : spring compression
24. K : spring rate
25. H : height above which frontal wheel is raised above rear
26. W_f : Front Weight
27. W_r : Rear Weight
28. Wb : Wheelbase
29. R_f : Radius of front tire
30. R_r : Radius of rear tire
31. α : maximum steering angle
32. P : momentum
33. IF : Impact force
34. FOS : Factor of Safety
35. μ : coefficient of friction between tire and road
36. W : weight of vehicle
37. ρ : density at room temperature