Critical Design Review Report Cover Page & Vehicle Description Form

Human Powered Vehicle Challenge

Competition Location: Digital

Competition Date: April 24, 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

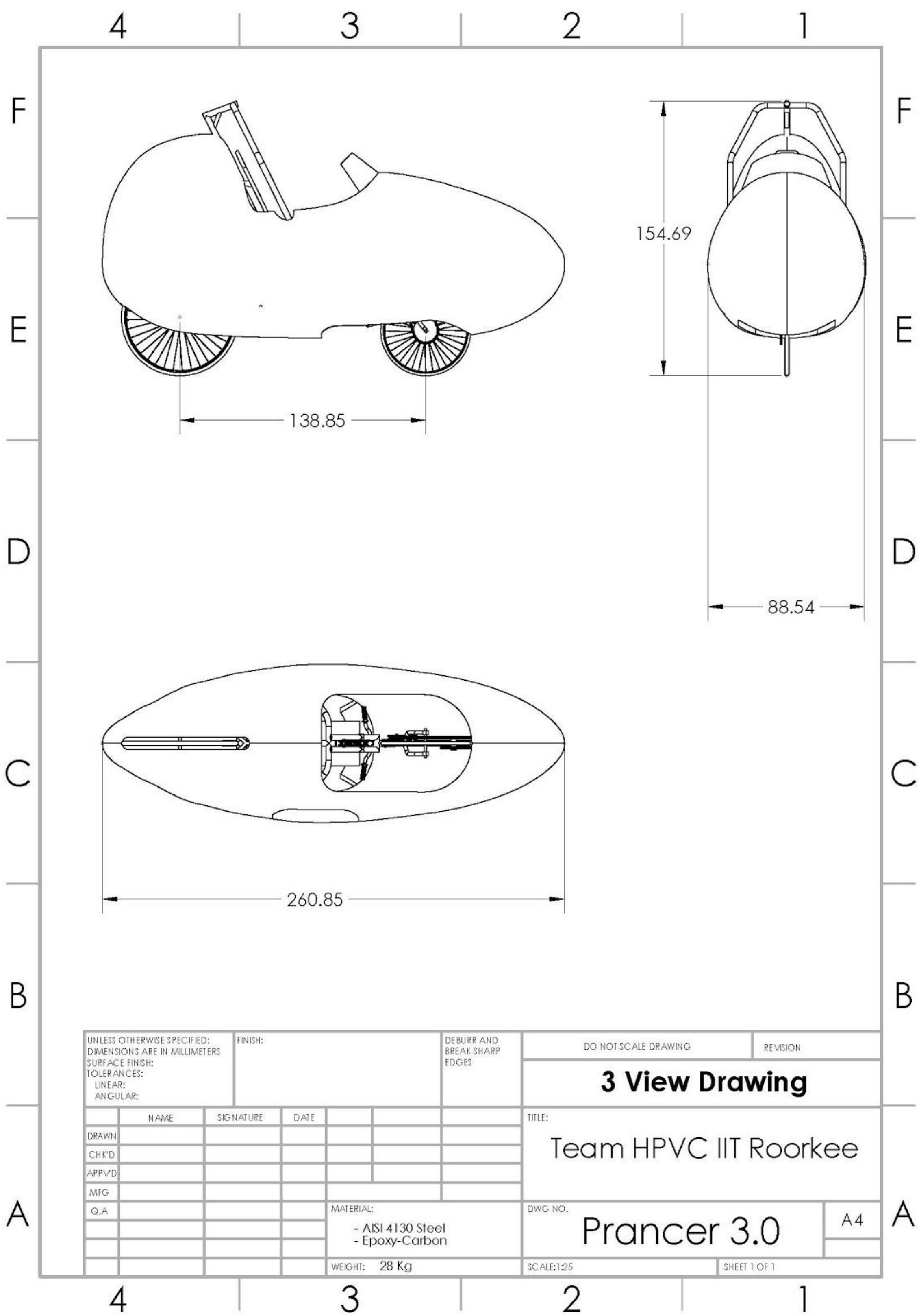
University name: Indian Institute of Technology Roorkee
Vehicle name: Prancer 3.0
Vehicle number: 30
Vehicle configuration:

Upright		Semi-recumbent	✓
Prone		Other (specify)	

Frame material: AISI 4130 Steel
Fairing material(s): Epoxy-Carbon
Number of wheels: 2
Vehicle Dimensions (m)
  Length: 260.85 cm
  Width: 88.54 cm
  Height: 154.69 cm
  Wheelbase: 138.85 cm
Weight Distribution (kg)
  Front: 16.8 kg
  Rear: 11.2 kg
Total Weight (kg): 28 kg
Wheel Size (m)
  Front: 0.504 m
  Rear: 0.604 m
Frontal area (m^2): 0.62 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?):
HPVC 2021 will be the first competition for Prancer 3.0 and before this it has not competed in any other competition.

3 View Drawings of Prancer 3.0



ABSTRACT

Prancer 3.0 has been designed and analysed by Team HPVC IIT Roorkee to participate in the event Human Powered Vehicle Challenge 2021. This report focuses on describing the entire process of creation of Prancer 3.0 in a detailed and elaborate manner. This is Team HPVC IIT Roorkee's fifth attempt in this competition. The team has given utmost care in designing the vehicle for congenial ride and combative performance. Innovative designs and mechanisms have been incorporated to make the vehicle response more predictable at its peak performance.

The team carefully studied the nature and demands of events in previous editions, and did internal surveys to aid Factor ratings and Quality function deployment to finalise various design configurations.

Prancer 3.0 has a semi recumbent design with a front wheel of 20-inch diameter and rear wheel of 24-inch diameter. The vehicle also has an integrated dual suspension system ie. both at front and rear. The team made special efforts to integrate a suspension system at the front to dampen out some vibrations due to external perturbations at the drivetrain.

The frame for Prancer 3.0 has been designed using AISI 4130 steel tubes of varying dimensions as per structural requirements. The frame weighs close to 9 kg in total.The Material considered for fairing is Carbon fibre based epoxy composite with a Fibre volume fraction of 50% alongwith Polycarbonate sheets for windshield. The initial drawing for frame measurements was made using 3D modelling followed by ergonomics analysis to finalize the dimensions.

Prancer 3.0 boasts of an adjustable front steering system, A non movable bottom bracket, front wheel drivetrain equipped with dual stage gear reduction and incorporates twist chain mechanism with 2 guide pulleys. A 48 teeth driving sprocket has been used with a 7- Speed external gear hub, and an additional 48/14 teeth sprocket pair has been used to provide favourable torque and speed combinations. Rear and Front Suspension has been added to this year's vehicle using the single pivot mechanism to increase rideability on rough riding conditions and also reduce the stress induced in the welded joints. Front suspension is designed such that it doesn't change the distance between pulley mounted on the headtube and sprocket attached to the front wheel, thus keeping the chain in tension all the time. Problem of chain disengagement and power loss would have been caused if conventional telescopic suspension were used in front.Considering their performance and driver response,Disc brakes have been used on both wheels to ensure proper braking of the vehicle in the specified limits.

This is a collective effort to optimise the design by analysing and optimising the design as per prevalent failure modes.

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1. Design

1.1. Objective

Team IITR started working on the project to incorporate different design principles and innovative ideas to design “Prancer 3.0”. In the process, the team also focused on developing their own technical knowledge through various design and analysis techniques.Special focus was put on developing divergent thinking methodology to come up with unique solutions. The specific design objectives are:

- Follow the given ASME Guidelines.
- Incorporate different innovations to accommodate riders of different height and shape and also provide the required rider comfort.
- Designing of an efficient and lightweight drivetrain for higher speed and better efficiency and least vibrations in the chain.
- Designing of a full suspension vehicle suitable for the uneven riding conditions and isolating the drivetrain.
- Designing of fairing to reduce the aerodynamic drag without much increase in weight, and any adverse pressure gradients.

1.2. Background Research

The team had several brainstorming sessions and built an idea management system, where members from the team formed smaller groups and analyzed it for its practicality, cost versus benefit in terms of human hours devoted and possible gains.The reasons for selecting or rejecting an idea were properly documented for further references.The project started off by referring to the ASME rulebook and judging what were the requirements for the vehicle and what constraints were imposed.Our previous vehicle designs were studied to acknowledge the pros and cons, and were critically benchmarked with open source data, and images from the event of previous years position holders. Several Academicians and Industry Experts were consulted to find contemporary solutions to our design problems, especially for high speed stability and vehicle modelling, aerodynamics.Different MOOC courses and Websites such as “Advanced Composites by Prof Nachiketa Tiwari (NPTEL)”, “Fundamentals of automotive systems by Prof. C.S.Shankar Ram (NPTEL)”, “http://www.recumbents.com” were referred to get an idea of the working of various subsystems and find right combinations to achieve desired results.

Comprehensive study for the fairing was done to choose from the different configurations that were present. Our team had previously worked semi-fared,and was aware of the pain points like, Adverse weight transfer leading to skidding of rear tyre and improper CFD analysis. Also, a detailed analysis of probable sources of vibrations was done to improve high speed stability and predictability.

This year we focussed on introducing unique solutions and to make the vehicle adjustable for different riders at a faster pace, so multiple mechanisms were studied,and their feasibility was judged. Care was given to not compromise structural integrity and rider comfort in the vehicle.
For the analysis part, our faculty advisor Dr V.H Saran guided us about the different analysis methods, Also we would like to acknowledge Prof D.M.Joglekar and Prof Ankit bansal for their benign guidance in vibration studies and CFD Analysis respectively. Using information from all the above sources, the team was able to design and simulate the performance of “Prancer 3.0”.

1.3. Prior Work

The entire designing and analysis of “Prancer 3.0” was done in the academic year 2020-2021. The previous year’s design and the vehicle were just used for study before the beginning of the designing phase.

1.4. Organizational Timeline

A strict timeline has been created and is being followed by the team to head towards the successful completion of the project. The below table represents the starting and end dates of each activity in the project; there have been breaks in between taking into consideration vacations and exams:

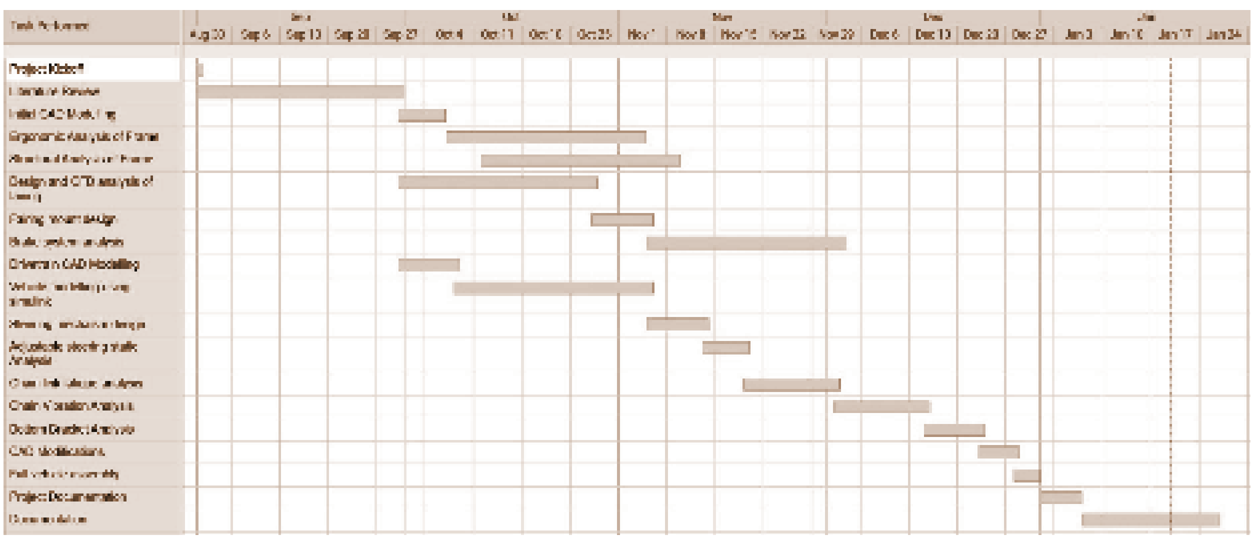

Figure 1.1: Gantt Chart representing entire timeline of project

1.5 Design Specifications

The designing process of “Prancer 3.0” was carried with strict compliance to ASME’s HPVC Rulebook. Below is a table that summarises the design requirements:

Parameters	Requirements
Performance	● Come to a stop from speed of 25 km/hr in a distance of 6.0 m ● Turn within an 8.0 m radius ● Travel for 30 m in a straight line at a speed of 5 to 8 km/hr
RPS	● Under the top load of 2670N, total deformation should be less than 5.1 cm with no sign of permanent deformation or fracture ● Under side load of 1330 N, total deformation should be less than 3.8 cm with no sign of permanent deformation or fracture ● Prevent significant body contact with the ground in the event of a fall and provide adequate abrasion resistance to protect against sliding across the ground
Braking	● Vehicle must at least have front brakes.
Safety	● Safety harnesses with lap and shoulder belts (also known as 4- or 5-point safety harnesses) should be used ● No sharp edges, open ends, pinch points should be present and the field of view of the rider should be at least 180 degrees.

Table 1.1: Design Constraints by ASME

Apart from the requirements mentioned above, Team HPVC IIT Roorkee did set some additional targets based on competitive benchmarking and prior experiences. Below is a table that summarises the design targets set by the team:

Constraint	Rationale
Weight of the vehicle should be less than 30 kgs	Minimise the weight of the vehicle to decrease the effort required by the rider
Efficient drivetrain with less vibrations	Reduce the total amount of energy lost in power transmission and chances of derailment.
Adjustable handlebar angle with quick and reliable configuration change	Ensure ergonomic position of handlebar, and aid in quick rider change
Addition of additional gear reduction	To remove any overdrive condition to ensure right cadence at high speeds and improve drivetrain response to riders.
Addition of front and rear suspension	To make the vehicle more usable on uneven surfaces, and isolate the drivetrain from any vibrations.
Addition of fully covered fairing	To reduce the coefficient of drag and ensure streamline flow of air around the vehicle.

Table 1.2: Design Constraint by our team

1.6. Quality Function Deployment- House of Quality

A house of Quality was constructed with each customer requirement being weighted and used to judge all the design parameters. The team came to a conclusion that to meet most of the consumer requirements, more focus should be on suspension, drivetrain and the fairing.

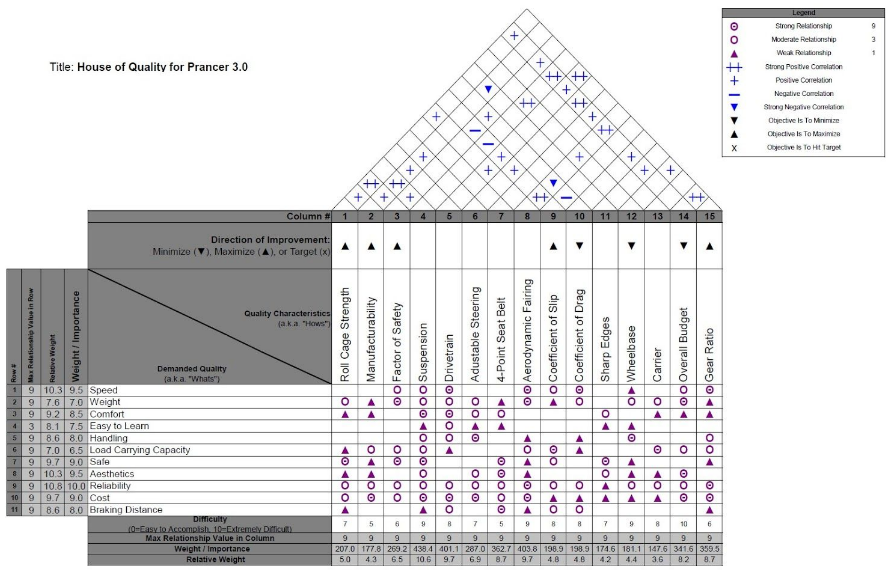

Figure 1.2: House of Quality

1.7. Concept Development and Selection

Each subsystem in the HPV design had several options and combinations. Decision matrix method was used to decide the variants in each subsystem.The weightage was provided from (1-5) and the marking was done on a scale from (1-10) and the option with the highest total was selected and implemented in the design.

1.7.1 Vehicle Configuration

The first step of the designing process was to decide on the vehicle configuration. The vehicle configuration is the single largest factor which decides rider comfort and stability. We gathered data from our past experiences, competitive benchmarking and several research papers which helped us in preparing the decision matrix:

Parameter	**Weightage (1-5)**	**Vehicle Configuration**			
		Two-wheels		**Three-wheels**	
		Long Wheelbase	**Short Wheelbase**	**Delta**	**Tadpole**
Weight	4	8	9	6	6
High Speed Stability	5	8	8	4	5
Low Speed Stability	4	6	7	8	9

Aerodynamics	3	9	9	7	6
Frontal Area	3	7	9	6	8
Rider Comfort	4	8	8	9	9
Maneuverability	3	7	9	6	8
Performance	4	8	9	6	6
Reliability	5	8	9	4	5
	Total	269	298	213	236

Table 1.3: Vehicle configuration Decision matrix

1.7.2 Frame Material

Frame comprises a significant portion of vehicle mass and can largely affect reliability of the design. Apart from conventional parameters we also took into consideration the recyclability and corrosion properties of the material.

Parameter	**Weightage (1-5)**	**Material Used**			
		Carbon Fiber	**AISI 4130**	**Aluminum 6061**	**AISI 4340**
Weight	4	9	5	9	5
Machinability	4	5	9	8	9
Yield Strength	5	9	8	7	9
Fatigue Strength	4	9	8	6	9
Availability	2	5	8	7	6
Cost	3	5	8	6	6
Recyclability	2	3	9	8	9
Corrosion	3	8	9	7	9
	Total	192	213	196	212

Table 1.4: Frame Material Decision Matrix

1.7.3 Drive Train Configuration

The team tried to select the most suitable drivetrain configuration to maximise the performance and sustain the ergonomics and predictability at high speeds.

Parameter	**Weightage (1-5)**	**Drive train Configuration**		
		Rear wheel Drive	**Front wheel Drive with Movable Bottom Bracket**	**Front Wheel Drive with non-Movable Bottom Bracket**
Ease of Rideability	5	9	4	7
Efficiency	4	5	9	8
Predictability at high speed	5	4	8	8
Ease of Turning	3	9	7	7
Weight	2	5	9	8
	Total	113	135	144

Table 1.5: Drive Train Configuration Decision Matrix

1.7.4. Aerodynamics

Aerodynamics can provide significant performance gains, especially in high speed conditions.

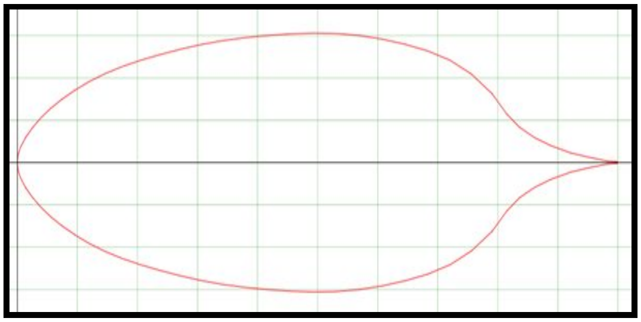

Figure 1.3: Top view Consideration - griffith30symsuction-il

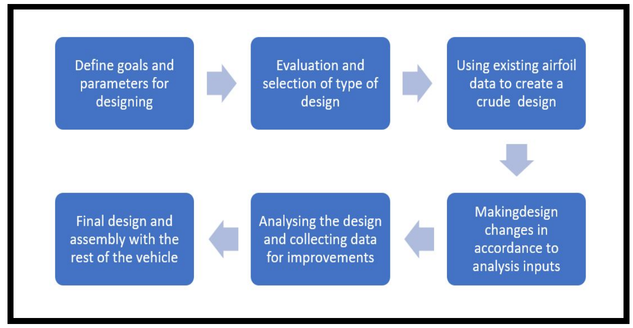

Figure 1.4 : Outline of design process

The criterias, ratings and weightage were modified as per past experience. For the fairing design we opted to start from a pre-existing symmetric airfoil with an aspect ratio close to that of our HPV taken from “airfoiltools.com” for the top view of our fairing. The top cross-section was chosen to be ‘griffith30symsuction-il.’ The rear portion had to be modified to better suit our HPV and accommodate all the parts.

1.8. Vehicle Description

After all the comparisons the team came forward with the design for Prancer 3.0 which aimed to provide the required comfort ,suited competition needs and had competitive performance.

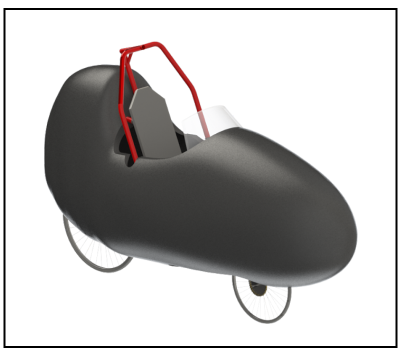

Figure 1.5: CAD model of Prancer 3.0 (with fairing)

Figure 1.6 : CAD model of Prancer 3.0 (without fairing)

1.8.1. Frame

The frame is composed of circular tubes of 1.6 mm thickness made of AISI 4130 Steel.
It has been designed as per ASME HPVC rulebook in order to provide the required strength and safety to the rider. The Frame has been divided into three parts:

- Front portion consisting of head tube and boom
- Back portion consisting of rollcage and main frame
- Rear fork

This design has been implemented to accommodate the front suspension and the rear suspension. The backrest rod is inclined at an angle of 125 degrees. Additional Truss elements were added to meet the design requirements. Furthermore mounts and hinges were added to accommodate the drivetrain and suspension.

1.8.2 Suspension

Front and Rear suspension has been used in our vehicle to increase the rider comfort and reduce the stress induced on the frame due to uneven roads and bumps. Front suspension is placed at the junction of the front and back portion of the frame. To implement front suspension, telescopic suspension was avoided because compression in telescopic suspension would lead to change in distance between sprocket attached to front wheel and the pulley mounted on headtube resulting in the slack of chain which may lead to disengagement of chain, power losses and low efficiency.

The rear suspension is attached to the frame through the rear fork. Single-pivot design was chosen for the rear suspension. It ensures overall stability of the vehicle.

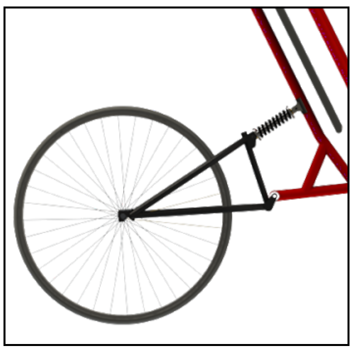

Figure 1.7: Rear suspension Design

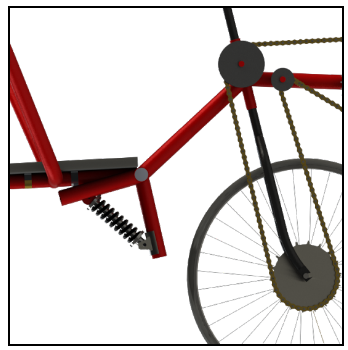

Figure 1.8: Front Suspension Design

1.8.3 Drivetrain

The drivetrain is an integral part of the HPV. Prancer 3.0 has a front wheel drive with a non-movable bottom bracket. It has a 2 stage gear reduction i.e. a fixed gear reduction through a pair of sprockets mounted on boom to provide a gear ratio of 48/14, and additionally through an external gearset at the front wheel hub. Also, a lesser slack in chain elements and a higher pre-tension was thus achieved. Two Pulleys having an outer diameter of 11cm and 7.5 cm have been used in the right part to guide the chain from the driving sprocket having 48 teeth to the external gears.

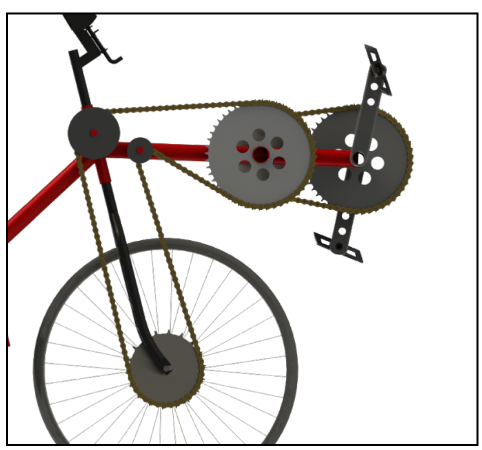

Figure 1.9: Representative Picture of Drivetrain

1.8.4 Steering

Steering system was optimized for better stability, handling comfort and weight distribution. We decided to go with Front-Wheel steer to provide better overall weight distribution. The headtube angle was kept to be 74 degrees with an offset of 4.63 cm which resulted in a trail value of 4.45 cm, which is in ideal range of trail value for stable steering.

1.8.5 Fairing

This year we have decided to go with full fairing design to get further less drag compared to partial fairing which did not perform well in the previous year. The fairing of Prancer 3.0 is designed considering the proper analysis techniques. Factors such as ease of exit, entering and ease of visibility were also considered. Material for the fairing is Epoxy-Carbon composite (V_f=50%) [1]

2. Analysis

2.1 Roll Over Protection System Analysis

The objective of this analysis is to determine the feasibility of the Rollover protection design under the provided load condition. Top and Side load analysis study was performed using Static Structural module of Ansys. Many evaluations and modifications were made to select an optimized design and ensure proper safety and performance of the vehicle.

2.1.1 Top Load Analysis

To ensure the safety of the rider in case of a roll-over accident involving an inverted vehicle.

Boundary conditions : As per ASME rules, the maximum load that has been assumed to act on the vehicle is 2670 N that has been applied at an angle of 12 degrees from vertical on the roll cage having fixed supports at seat belt attachment points on the roll cage.

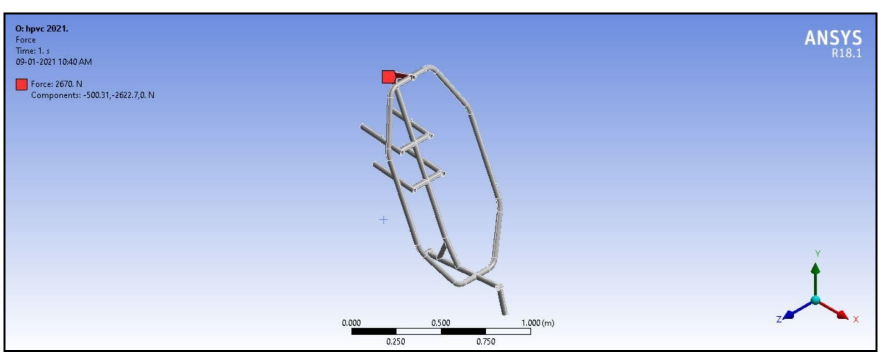

Figure 2.1: Initial Conditions

Results and Modifications:

- The minimum factor of safety was found to be **2.4595**.No permanent deformation was observed on frame and roll bar.
- Maximum elastic deformation was found to be **2.1084 mm** which is less than 5.1 cm.

- For the initial design, the factor of safety was less than 1.5 cm so additional truss elements were added at regions with high stress concentration.

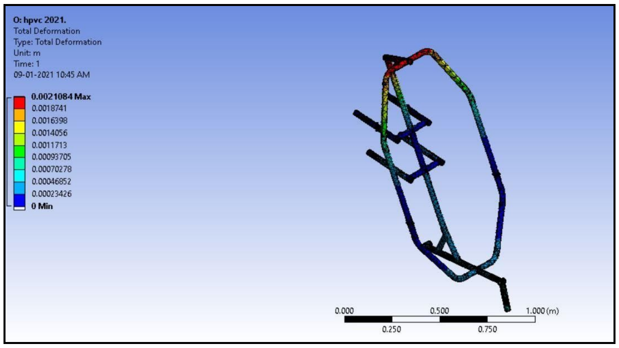

Figure 2.2: Factor of Safety

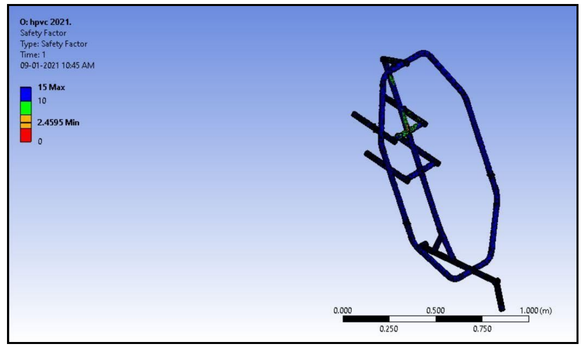

Figure: 2.3: Total Deformation

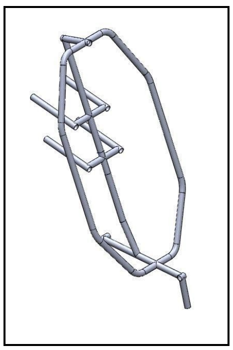

Figure 2.4: Initial design

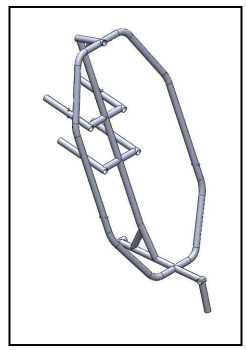

Figure 2.5: Final design with
2 additional truss element

2.1.2 Side Load Analysis

To ensure the safety of the driver in case of roll-over accident involving sideways tilting of HPV.

Boundary conditions: As per ASME rules, the force having a magnitude of 1330N was applied horizontally on the side of the roll cage at shoulder height and keeping the seat belt attachment points as fixed.

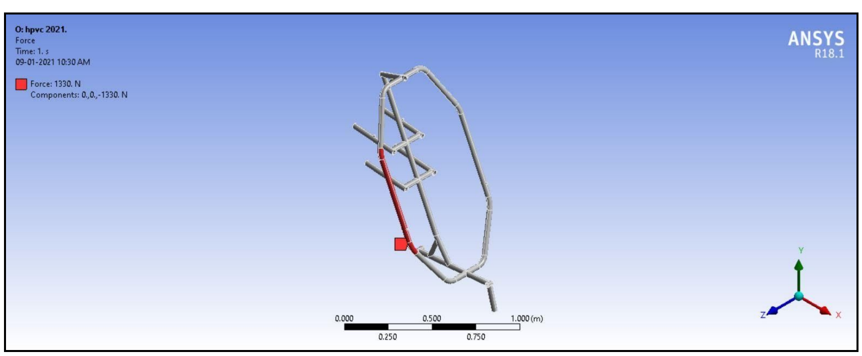

Figure 2.6: Initial Condition

Results:

- The minimum factor of safety was found to be **3.2236**.Nopermanent deformation was observed in the frame and roll cage.

- The Maximum elastic deformation was found to be **0.41174mm** which is less than 3.8 cm.

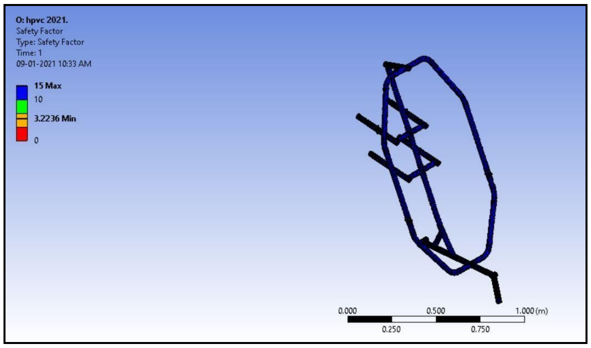

Figure 2.7: Factor of Safety

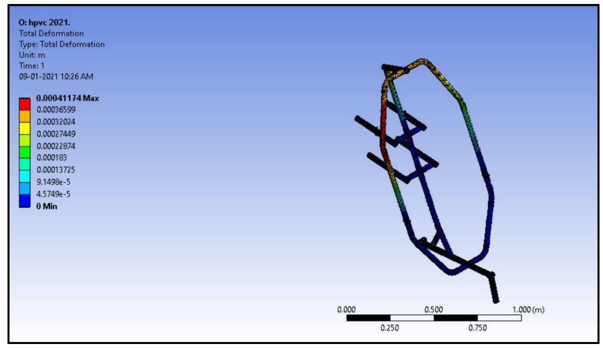

Figure 2.8: Total Deformation

2.2 Structural Analysis

2.2.1 Bottom Bracket Analysis

To determine the structural integrity of the boom under the applied load especially at mounts made for secondary gear reduction. As per the design, we have a boom length of 55 cm, to which the bottom bracket, and additional mounts for a primary gear reduction are attached. Therefore due to pedalling an axial force is exerted on the mount and a bending moment is produced at the end where the bottom bracket is attached to the boom.We decided to analyze if the deformation is of the order of thickness of the chainlink to support derailment of chainlinks form pulley and ensure the structural integrity of the structure

Assumptions:

• Uniform load is applied by the rider's legs on the bottom bracket

• Chain losses and frictional forces have been neglected.

• Initial tension in the chain is taken to be 0, and mass of chain and sprocket is neglected

Boundary Conditions:

A load of 200 N was applied on the bottom bracket in downward direction,Resultant forces on mounts were applied as per the model described further in this document in section 2.5.3

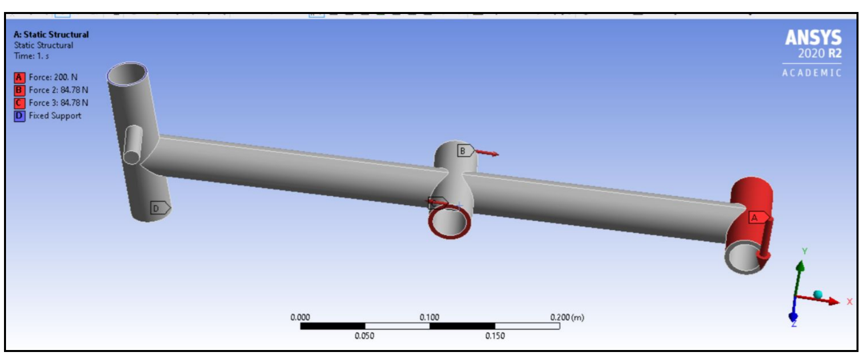

Figure 2.9: Initial Conditions

Result:

• Factor of Safety of 4.6245 was obtained thus no extra support elements were needed.

• The maximum total elastic deformation was 1.8525 mm which is acceptable.

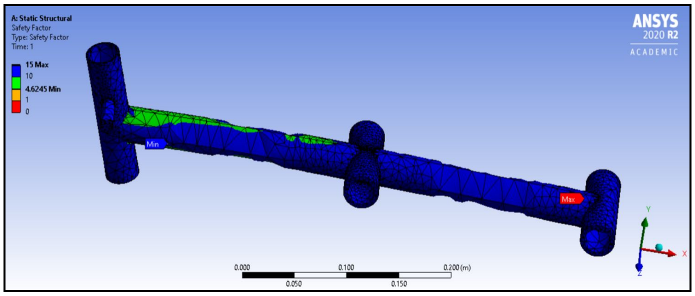

Figure 2.10: Safety Factor

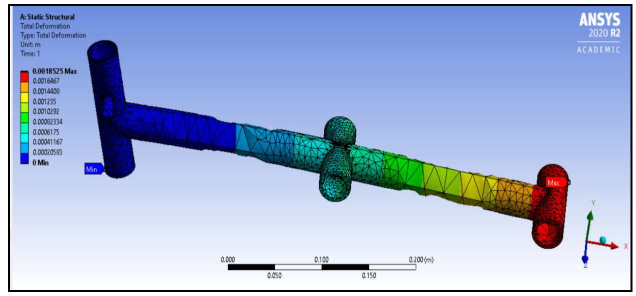

Figure 2.11: Total Deformation

2.2.2 Adjustable steering static Analysis

To perform a finite element analysis of the revolute joint used in the steering mechanism to confirm theoretically, working of the revolute joint based on decided dimensions of the spring bolts.While turning the handlebar the maximum force acts on the knuckle joint which has been used to facilitate change of handlebar angle and so it is important to theoretically analyse the possibility of failure of the mechanism under the applied load.

Assumptions:

• We have considered head tube fixed and force is applied on the handlebar.

• Gravitational torque is assumed much less than the centrifugal torque on the handlebar.

Boundary conditions

T (net torque) = Tg + Tc

• Tc (torque due to centrifugal force) = 3.6367Nm

• Tg (torque due to gravity) = 60.6131 Nm

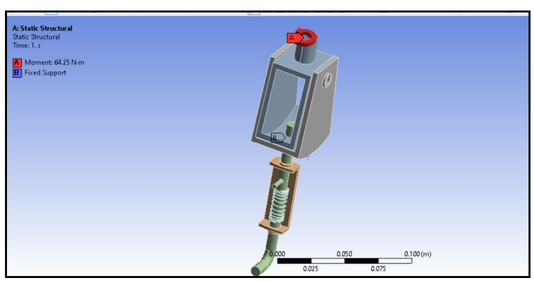

Figure 2.12: Initial Conditions

Result:

• Factor of safety was found to be 1.9777 which was acceptable for the given case.

• The maximum deformation came out to be 0.049013mm which was acceptable.

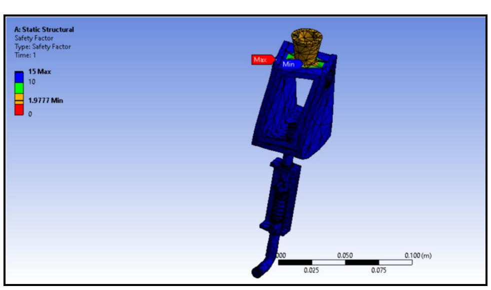

Figure 2.13: Factor of Safety

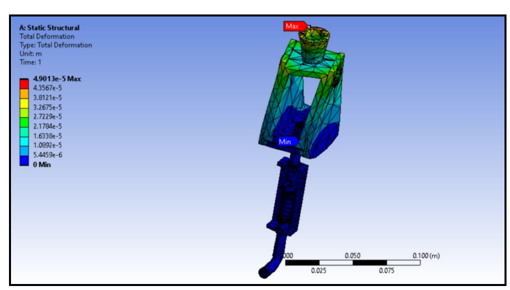

Figure 2.14: Total Deformation

2.3. Aerodynamic Analysis

The primary purpose of aerodynamic analysis is to study the fluid flow around the vehicle and to determine the coefficient of drag of each design iteration. It also helped us as a feedback source of each design iteration.After each design iteration we have done the fluid flow analysis and determined the coefficient of drag using ANSYS Fluent. We have chosen the k-ω SST turbulence model for its advanced and updated equations. We used the first order upwind scheme to solve the equations for initial iterations to get quick convergence which gave us appreciable results. Later for the final designs we have used the second order upwind scheme for the final fairing designs to get accurate results.

Assumptions:

1. Inlet Velocity: (Front Wind) = 7 ms^{-1} and (Side Wind) = 1.38 ms^{-1}
2. Outlet = PressureOutlet Boundary Condition with Gauge Pressure = 100000 Pa
3. Walls = No-Slip Condition and InFlow of Air considered to be uniform at inlet.

The first iteration was a crude structure made to just enclose the components of our vehicle and give us an idea about the structure of the fairing. Further on the rear portion was extended upwards to allow for better aerodynamic flow behind the driver. A windshield was also added for the driver to redirect the strong winds while travelling at higher speeds. Over the several iterations through the process of analysis and redesigning we made improvements in the curvature of the surface and lowered the windshield for optimal flow of air current and reached our final design.

Figure 2.15: Final CAD model of fairing

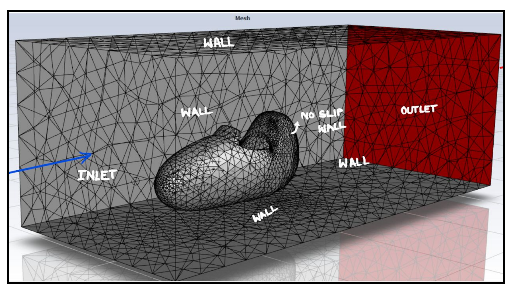

Figure 2.16: Fluid Domain and Setup

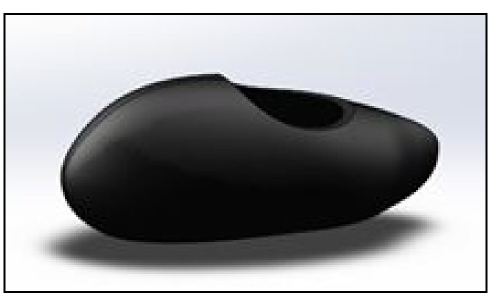

Figure 2.17: Initial Design

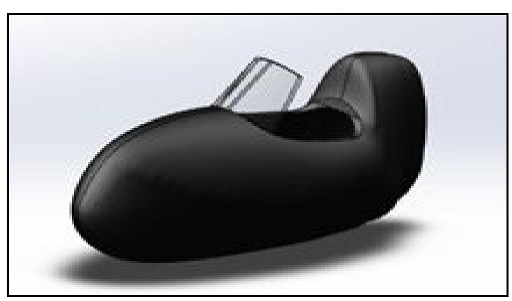

Figure 2.18: Mid-Phase Design

2.3.1. Front Wind Analysis

The front wind analysis aimed at studying the aerodynamic response of the vehicle in the direction of the motion. The fairing CAD model was imported to ANSYS Fluent and we have created a fluid domain using enclosure and boolean features in Design Modeler. The meshing was done in such a way that we have considered the proximity and curvatures of geometry. We also inflated the mesh layers (7) to get accurate details near the surface. Inlet and Outlet were defined in the direction of vehicle motion (along the wind).

- Inlet velocity = 7 ms^{-1}
- Wind flow in the longitudinal direction

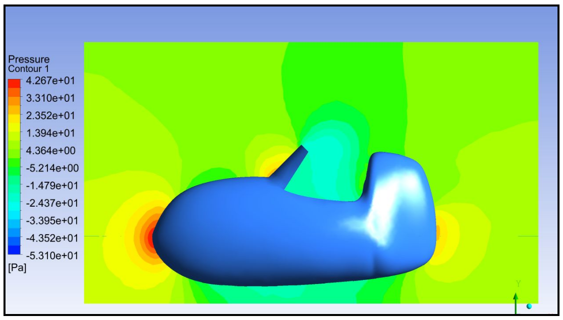

Figure 2.19: Pressure Contour in Side View of one of the design iterations

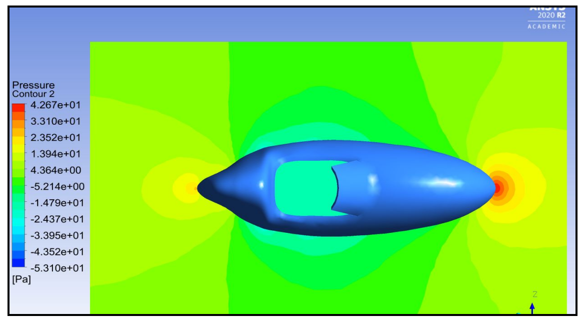

Figure 2.20: Pressure Contour in Top View of one of the design iterations

Observations:

- Vortex formation behind the windshield, which creates more pressure drag.
- Streamlines splitting near the nose of fairing which creates additional drag.

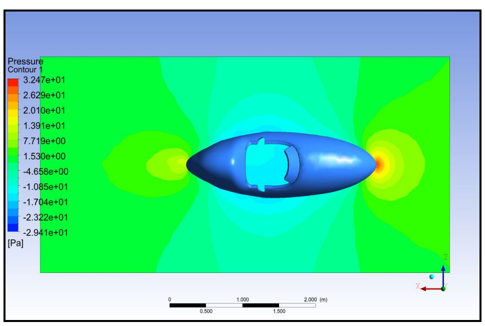

Figure 2.21: Pressure Contour in Side View of Final Design

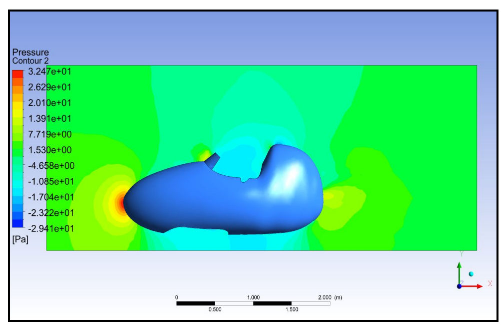

Figure 2.22: Pressure Contour in Top View of Final Design

Observations:

- Maximum gauge pressure of **32.47 Pa** is observed at the nose of the fairing
- Minimal Splitting of streamlines and Vortex formations.

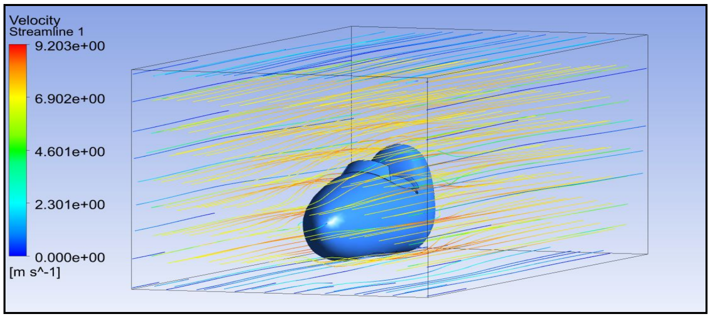

Figure 2.23: Streamlines in Front Wind Analysis

2.3.2. Side Wind Analysis

The Side Wind analysis aimed at determining the force acting on our vehicle due to the side/lateral wind flow after attaching the fairing. The procedure for this analysis is the same as Front Wind Analysis with boundary conditions perpendicular to the front wind.

- Inlet Velocity: **1.38m/s**
- Surrounding Pressure: **1 atm**

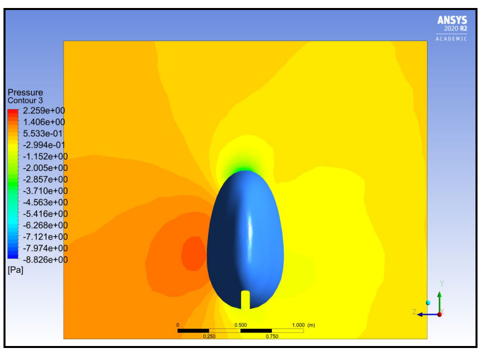

Figure 2.24: Pressure Contour

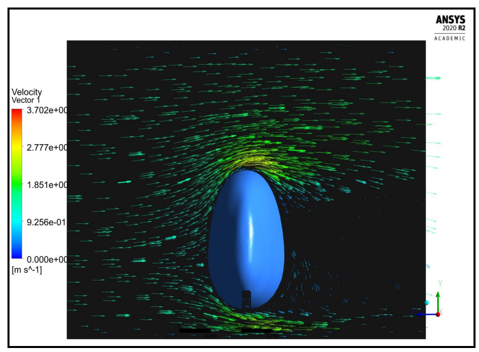

Figure 2.25: Velocity Vectors

Observations:

- Maximum Pressure Difference of **2.07 Pa** was obtained.
- Maximum Velocity of 3.7 m/s is observed.

Conclusions:

- The final drag coefficient of fairing is **0.19** which is significant reduction from 0.32 for the frontal area of **0.62 m^2**

2.4 Cost Analysis

We utilised multiple services provided by the institute, whose market price is mentioned in the table below

Software Package/ Digital Service	Unit Price (Rs)	Quantity	Cost(Rs)
High performance Workstation (registration charge)	2000	2	4000
Solidworks 3D experience	291500	1	0(Reimbursed)
Total expenditure			4000

Table 2.1: Cost Analysis of Prancer 3.0

2.5 Other Analysis

2.5.1 Braking Analysis

As per the ASME standard safety requirement that all the vehicles must be able to stop within 6m while decelerating from 25 Kmph to 0 Kmph. Last year, we faced difficulties to achieve the required stopping distance limit practically, so this year we have done proper analysis on the braking system using hand calculations and also by modelling panic braking in MATLAB using Simulink and Simscape.

Methodology

We have used basic physics concepts to solve for the stopping distance. We also modelled the vehicle in MATLAB to find the additional information about the vehicle during braking. We made use of the inbuilt Driveline and Multibody Simscape blocks in which all the vehicle dynamics equations are precoded.

Stopping Distance: $S = (v_2^2 - v_1^2) \div 2 * a =$ **4.91*m***

Stopping Time: $v_2 = v_1 + at$

Where v_2 is the final velocity of 0 m/s, v_1 is the initial velocity of 25 Kmph or 6.94 m/s, and t is the time.

Solving for t, $t = (v_2 - v_1) \div a = (0-6.94)/(-0.5 * 9.81) =$ **1.415** s

Assumptions and Specifications Data:

1. We have assumed standard brake clutch dimensions with no cable losses/friction
2. Brake Clutch Force applied by the rider is 300N (constant and continuously applied)
3. Mass of the vehicle + rider system = 93 Kg
4. Coefficient of friction between tyre and road = 0.5(wet) - 0.7(dry)
5. Coefficient of friction between brake disc and pad = 0.2(wet) - 0.4(dry)
6. Rolling Coefficient = 0.015
7. Frontal Area and Coefficient of Drag as per the aerodynamics results - 0.62 m^2, 0.19

Results and Conclusion:

**Refer Appendix A for simulink model for braking*

The Stopping Distance is 4.91m as per the hand calculations, which is well behind the safety standards. As our vehicle dynamic weight distribution is highly towards the front we couldn't achieve simultaneous braking, so our rear wheel locks first and then the front wheel. This might generate lateral forces due to which vehicle will be out of yaw control. But as the stopping/ retarding time is very low as 1.5 seconds the rider will not experience any instability or discomfort. The figure below shows the position vs time of the vehicle derived from the Simulink model. It is also evident from figure 2.27 that the vehicle normal force on the rear tyre is never equal to zero which shows that there is no chance of tipping in case of panic braking.

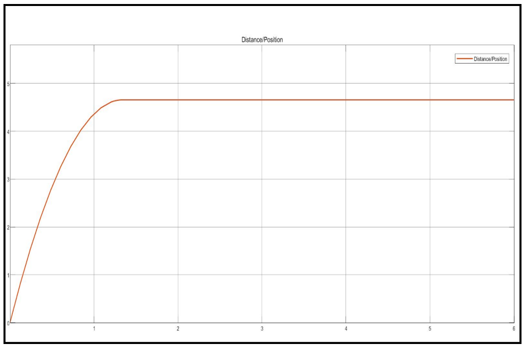

Figure 2.26: Position vs time plot

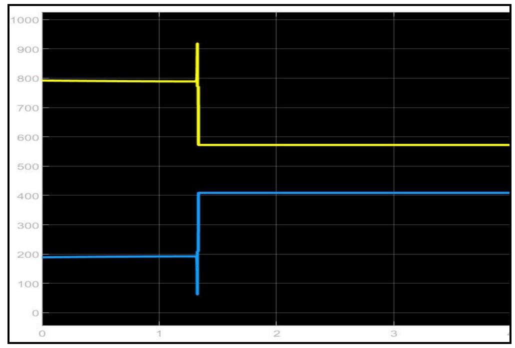

Figure 2.27: Normal Force vs time plot

2.5.2 Ergonomics Analysis

To determine the most ergonomically suited dimensions for the mainframe. Recumbents provide an unconventional posture for riders to maintain during riding. Such postures can be uncomfortable, especially when riding long distances. Therefore an analysis of the stresses occurring on the rider's body is important.

Methodology:

- The assembly of front and back portion of frame was imported to **CATIA** along with a manikin of 10, 50 and 90 percentile.
- Posture Analysis of manikin was done to test the frame ergonomically.

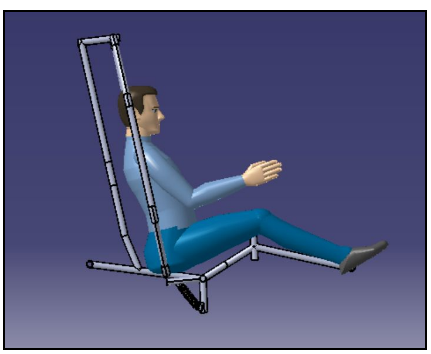

Figure 2.28: Sitting Posture (Side View)

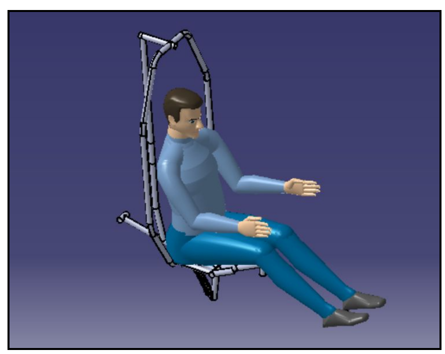

Figure 2.29: Sitting Posture (Isometric View)

Results:

- The Rapid Upper Limb Assessment (RULA) was done to obtain stress level for different body parts affected by sitting posture.

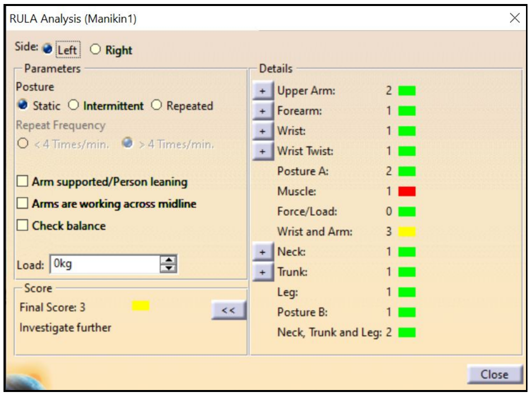

Figure 2.30: RULA Analysis (Left Side)

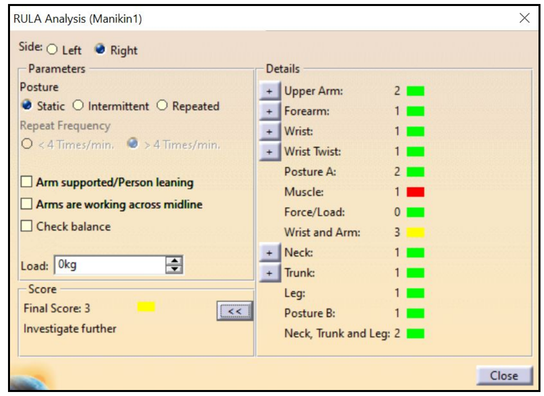

Figure 2.31: RULA Analysis (Right Side)

- Bottom Bracket to Back of Seat Distance was obtained for manikins of different percentiles. Got it as a range of 82 cm to 95 cm and an adjustable seat was designed to achieve this range.
- Optimum range for handlebar position was obtained such that stress level is minimum at forearms and wrists. Handlebar angle was obtained as a range of 15 to 31 degrees from vertical. Adjustable handlebar mechanism was employed to achieve this objective.

2.5.3 Drivetrain Analysis

[2]To find the maximum velocity of the vehicle by changing the gear ratios.Given the challenging circumstances where our designed vehicle can't be manufactured and put to test for physically finding the top speed and other similar parameters, it is important to make a numeric model of it. The model was made using MATLAB Simulink.The gear box "Shimano SG-C6061-8V"was considered.

**Refer Appendix B for Simulink Model for drivetrain*

Forces on vehicle –

1. Drag force (density of air = 1.225 kg/m^3 at NTP)
2. Friction force (rolling resistance=0.03)
3. Driver input force = 23N.
4. Tractive force (transmission efficiency = 0.78 from previous year)

Results-

- Maximum velocity obtained by this analysis is **18.65 m/sec**.
- The following v versus t graph was obtained from the analysis.

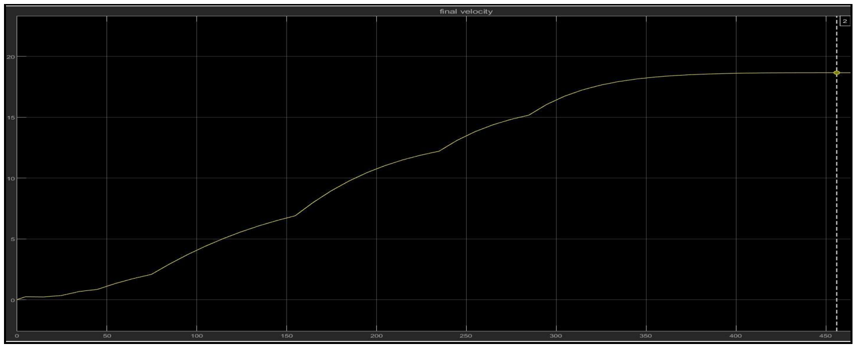

Figure 2.32: Velocity (y axis) vs. Time (x axis) graph

2.5.4 Suspension Analysis

To find the suspension travel at front and rear over a bump at a given speed and thus determine specifications of the spring and damper systems required for critical damping. While the rear suspension plays a pivotal role in enhancing rider comfort, a critically damped unique suspension configuration at the front helps in isolating the vibrations from drivetrain. Thus careful analysis of motion characteristics is necessary.

Methodology-

- A mathematical model of the vehicle was made using MATLAB Simulink (version 2020 a).
- The following governing equation was used i.e. For a critically damped system-
  $x(t)= [x_0 + (v_0 + \omega_n x_0)*t]exp(-\omega_n*t)$
- Dimension of bump is in accordance with NHAI (National Highway Authority of India) norms- ie height = 10 cm, width= 170 cm.
- Resultant stiffness of suspension spring in mount is equal to 48000 N/m

**Refer Appendix C for Simulink model for suspension system*

Assumptions-

- System is critically damped.
- Force transability due to mass of fairing on spring based mount at the boom is taken to be 0.

Result and Conclusion-

- Maximum travel of front suspension is **7cm.**
- Maximum travel of rear suspension is **2.8cm.**
- A lesser suspension travel at rear was finalised in the view of additional dead weights put at the rear during endurance event in HPVC 2020.

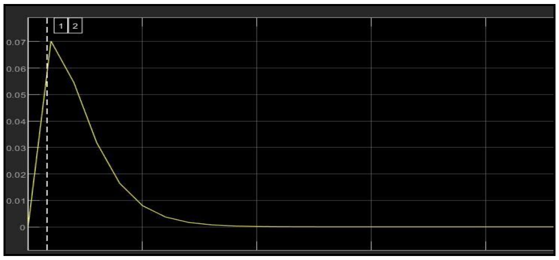

Figure 2.33: Front Suspension Displacement (y axis) vs Time (x axis)

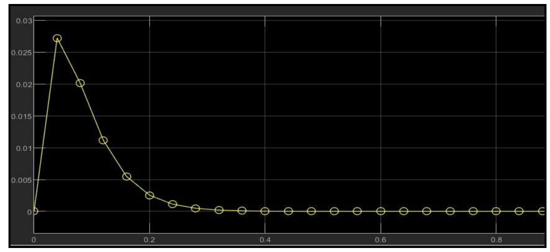

Figure 2.34: Rear Suspension Displacement (y axis) vs Time (x axis)

2.5.6 Chain Vibration Analysis

The peddling force applied by the rider on the vehicle can be approximated as a harmonic.Analysis was done to find the steady state amplitude of vibration of chain element under harmonic excitation input force and variable distance between Front Driving sprocket and Primary gear reduction.

Methodology- Analysis has been done using MATLAB Live Scripts on top speed achieved through drivetrain analysis i.e.18.65 m/s. Mass density of chain = 0.2273 kg/m (Generic Shimano IG51 Steel chain)

Assumptions-

- Input force is uniform and periodic and Mode of vibration is 1st overtone
- Chain is homogeneous and system is undamped

Results-

From the above analysis we found a satisfactory distance between front driving sprocket and primary gear reduction which is equal to **15.0 cm** from driving sprocket. At this distance amplitude ratio is also minimum.

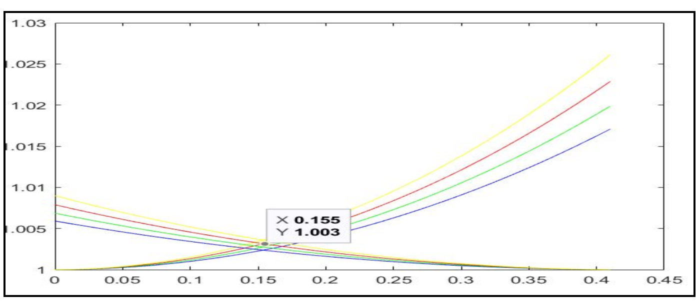

Figure 2.35: Amplitude Ratio (y axis) vs Position of Mount for secondary sprocket pair (x axis)

Note-

1. Same analysis was also done for the 2nd and 3^{rd} overtone and we found similar results.
2. Physical assembly of the vehicle at this configuration is not possible. So the position at **18.7 cm** was chosen as the next best alternative.

3. Conclusion

3.1 Comparison

The below table summarizes the comparison of the final vehicle with the initially set design constraints and targets along with justification for the same

Parameter	Target values	Obtained values	Justification
RPS structural strength	Top load: deformation< 5.1 cm Side load: deformation< 3.8 cm	Top load: deformation=2.108 mm Side load: deformation= 0.417 mm	RPS Analysis
Top Speed	>55 km/hr	67.14 km/hr	Drivetrain Analysis
Stopping Distance	<6 metres	4.91 metres	Braking Analysis
Aerodynamics Drag Coefficient	<0.4	0.19	Aerodynamic Analysis
High speed stability ie.Drivetrain resonance	>1	1.4106	Drivetrain Analysis, Chain Vibration Analysis
Rider Comfort	Front suspension, Handlebar Adjustable according to rider	Achieved	Adjustable Handlebar mechanism, Ergonomic Analysis,Bucket Seats
Drivetrain Reliability	Infinite life (>10^6 cycles of chain link)	Achieved	Chain link fatigue analysis
Weight	< 30 kg	28 kg	Weight Calculation

Table 3.1: Comparison of current vehicle with design specification and constraints

3.2 Evaluation

Prancer 3.0 was evaluated with respect to the ASME guidelines using all the above analysis and testing methods and it passed all the design criterions.Prancer 3.0 currently is capable of sustained performance over higher speeds and prolonged usage by virtue of carefully designed subsystems. Given the detailed ergonomics and adjustable parts, our design can accommodate riders of varied physiques.Also,Comparisons with Prancer 2.0 (our previous entry in ASME HPVC 2020) were also done to make sure that the upcoming design was better than the previous one in all aspects and feedbacks from the jury members were taken into account with utmost careful consideration.

3.3 Recommendations

The team greatly benefited due to easened design constraints by keeping non conventional manufacturing techniques namely Laser Cutting, Laser beam welding and CNC Milling in mind. Efforts are being put to make a proof of concept of this design. In upcoming days we would like to do a more comprehensive study of drivetrain, suspension and fairing to incorporate more inherent non-linearities and reduce the number of assumptions therefore made.

The team has also for the first time worked with carbon fibre composite. We foresee multiple avenues to incorporate them in other subsystems of the vehicle.

4 References

- [1]J.C. Halpin and J. L. Kardos Halpin-Tsai equations:A review, Polymer Engineering and Science, 1976, v16, N5, pp 344-352
- [2]H. Pacejka (2005) Tire and vehicle dynamics. Elsevier
- ASME HPVC 2020 Report: “Prancer 2.0”by team HPVC IIT ROORKEE
- http://jetrike.com/ergonomics.html
- https://sites.google.com/site/bikephysics/english-version/5-torques-acting-on-the-steering-axis
- https://www.obvibase.com/p/MiwdF2D3fkHh#table/MiwdF2D3fkHh/*
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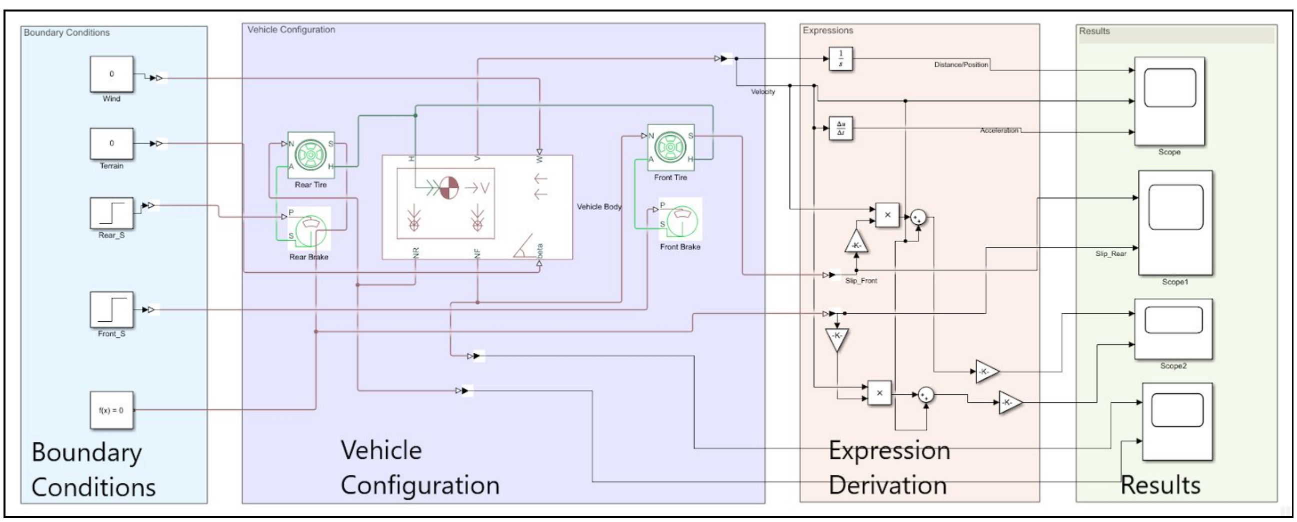
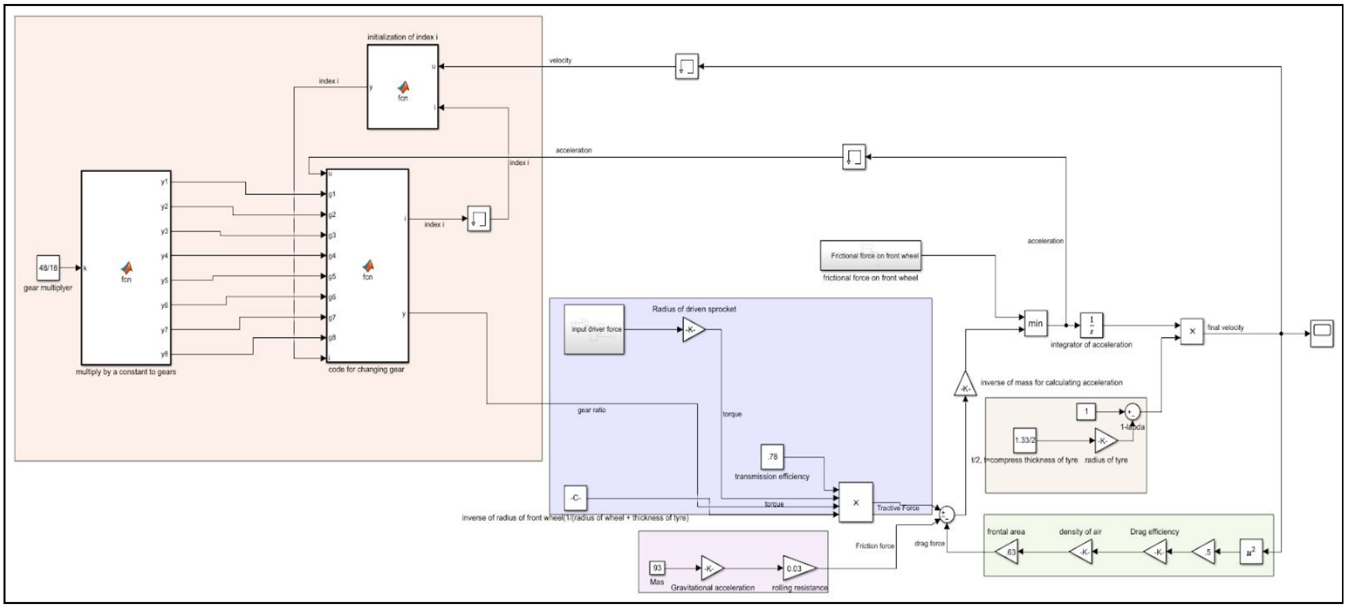
5 Appendices

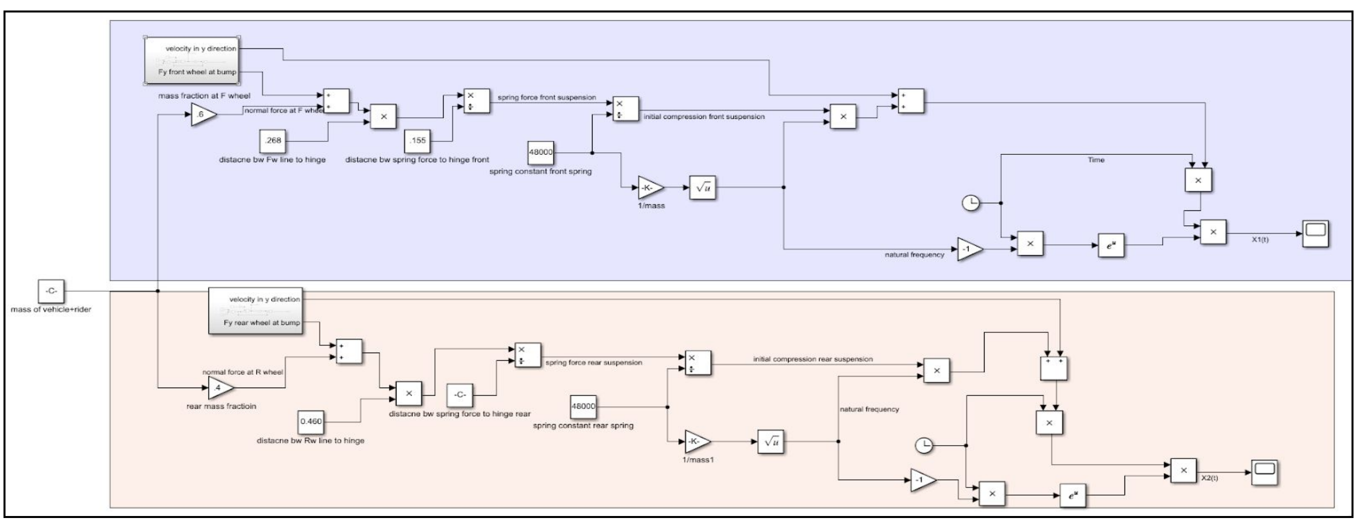
Appendix A (Simulink Model for Braking)



Appendix B (Simulink Model for drivetrain)



Appendix C (Simulink model for suspension system



Innovation Report Cover Page & Innovation Description Form

Human Powered Vehicle Challenge

Competition Location: Digital

Competition Date: 24th April, 2021

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

University name: **Indian Institute of Technology, Roorkee**

Vehicle number: **35**

Innovation Title: **Design and Analysis of Novel Front Suspension Design**

Innovation summary (Up to 150 words):
The proposed front suspension design will help our entry 'Prancer 3.0' achieve stability at higher speeds, Reduce vibrations at Chainlinks due to bumps and obstacles, and enhance rider's comfort by providing an additional rotational degree of freedom. The design is inspired by Single pivot suspension, to isolate front wheel assembly, drivetrain and steering components from saddle and rear wheel assemblies. The design is backed by intensive mathematical modelling and simulations using Matlab and Simulink.Some modifications were also made for better manufacturability.The end result is the outcome of multiple incremental innovations involving suspension design, redesigning of mounts for fairing, addition of more idlers which were further replaced with a pair of sprockets to increase net gear ratio.
This innovation will add to the vehicle's performance with immaterial addition to complexity.

1. Design

1.1 Literature:

- Commercial shock absorbers are characterized by integrated dampers in the fork. Such systems cannot be used as any suspension action will lead to a change in the chain length. Such changes can lead to excessive vibrations and eventually derailment of the chain over bumps.
- Gowri Shankar, Ajay, Prasanna, and Rajagopal in ‘Design and Fabrication of in-Wheel Suspension in Bicycle’ [1] have thoroughly analyzed the possibility of an in-wheel suspension system, although for the case on an upright bicycle. This design doesn’t fit our case owing to higher cost and thrice the amount of time and effort spent in calibration to suit every rider, which may be undesirable in a racing/team sport scenario. Suspension systems in front-wheel driven and Front wheel steered recumbent vehicles are fairly uncommon. Double Wishbone suspension systems are common in front-wheel driven delta configuration vehicles, which in some cases are torque steered.

1.2 Benefits:

Team HPVC IIT Roorkee designed a front suspension system for HPV which eliminates the problems occurring in telescopic suspension. Use of telescopic suspension in front would have caused a change in distance between the idler mounted on the headtube and the flywheel mounted on the front wheel which would have caused slack in the chain ultimately leading to chain derailment, power losses and low efficiency.

Design is such that the entire drivetrain is intact as one single unit which is connected to the rest of the frame using a hinge joint and front suspension is placed at that junction.

1.3 Manufacturability:

The design involves use of weldments wherever possible, and requires generic manufacturing processes like turning, grinding etc. This suspension system is specifically designed to be compatible with commercially available shock absorbers.

1.4 Iterations:

1.4.1 Initial Design- Our initial design for the front suspension system was such that if the front wheel encounters a bump on the street then the helical spring of the front shock absorber would undergo extension. As most of the commercial shock absorbers are designed to undergo compression ,so the loading conditions in our case would lead to a shorter product life span of the shock absorber used in the front suspension system resulting in increase of maintenance cost of vehicle.

1.4.2 Final Design- We modified the initial design for front suspension such that helical spring of front shock absorber undergoes elongation for the same scenario mentioned above. As the loading conditions are the same as for commercially designed shock absorbers, so the product would work fine till expected life span.

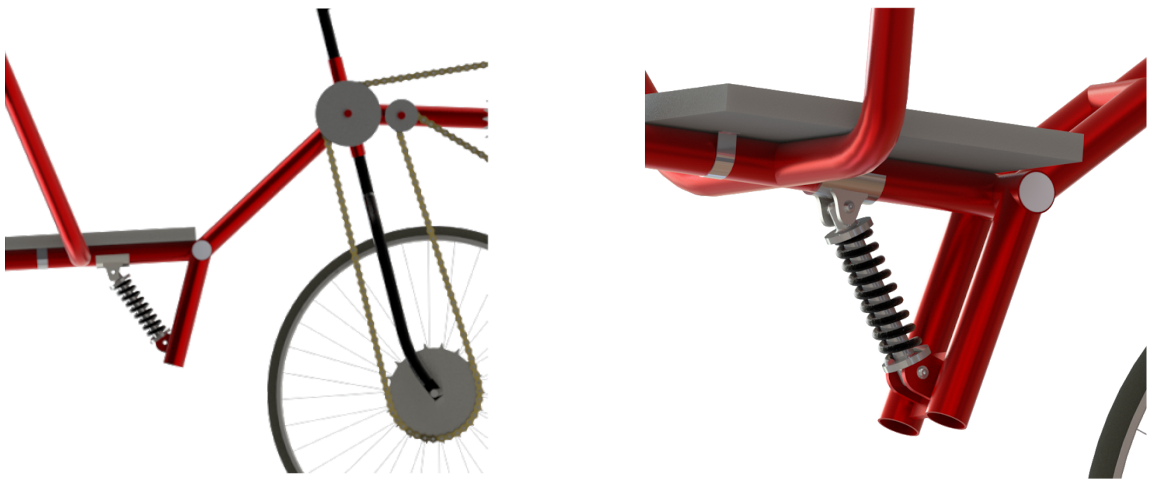


Figure 1: Initial Front Suspension Design

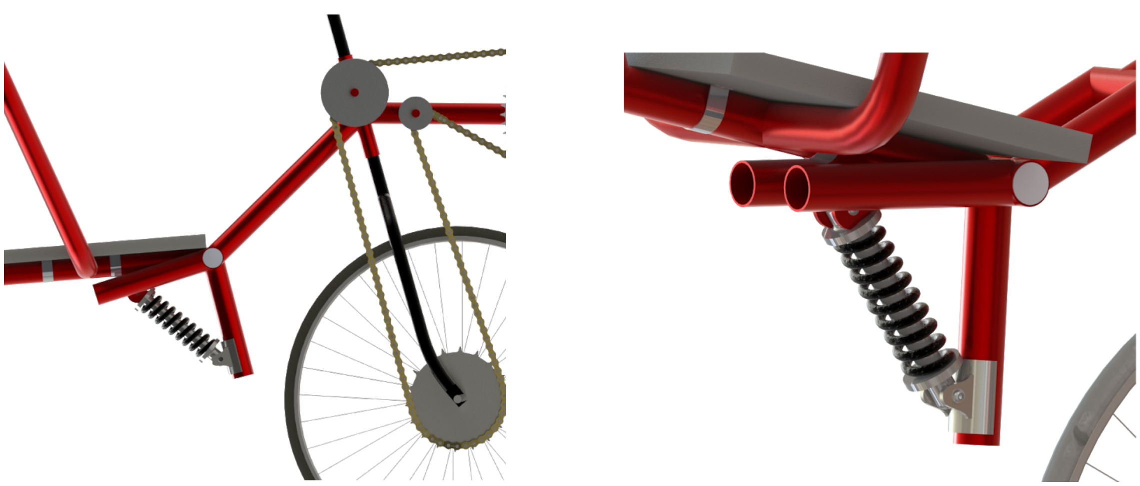


Figure 2: Final Front Suspension Design

1.5 Drawbacks:

A few drawbacks of this design are that it momentary changes the trail value, steering column inclination in steering due to bumps.So, Special efforts have been made to minimize the effects of this drawback by considering a critically damped suspension design.Also, it is not a fail Safe Design and failure of hinge pins can cause catastrophic failures.And,The hinge pin is under bearing loads and any deformation could hinder the suspension action, Thus greater manufacturing tolerances are required.

1.6 Secondary Problems Encountered and their Solution:

After completing the design of the front suspension system we found that when our front wheel encounters the bump the distance between the idler mounted on the headtube and the flywheel mounted on the front wheel is changing. While our fairing is rigidly mounted to the boom and the rear part of the frame thus making the whole suspension system stiff which is not desirable for proper functioning of suspension system.
So, we were in a need of some assembly that can provide relative motion between the boom and fairing when our suspension is in work .

We designed a dynamic fairing mount to solve the problem of relative motion between the boom and fairing also keeping our fairing stable.

1.6.1 Fairing Mount Design:

Need:
Presence of Rotational DOF due to Integrated Front suspension.

Features:
- Provides Support to fairing at 2 points in the front.
- Reduces jerk transmissibility due inertial force on fairing.

Components:
- 25cm long T shaped cylindrical extension from bottom bracket.
- Cylindrical rod symmetrically fixed to fairing, with a curved guideway(Radius of curvature = 44cm) welded.
- The contact between fairing and boom is further enhanced via soft springs, thus its effect on drivetrain analysis is negligible.

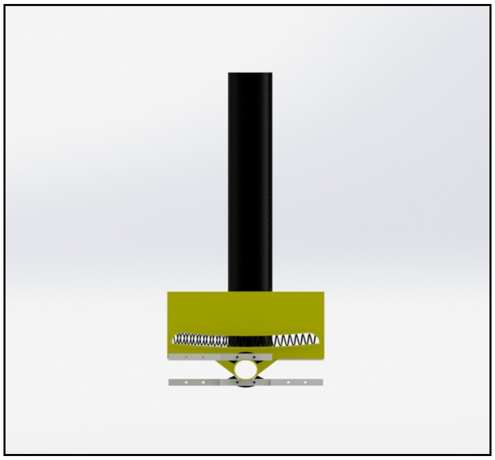


Figure 3: Fairing Mount (Side View)

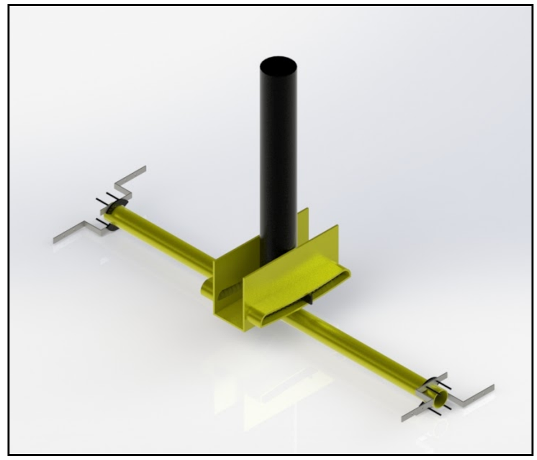


Figure 4: Fairing Mount (Isometric View)

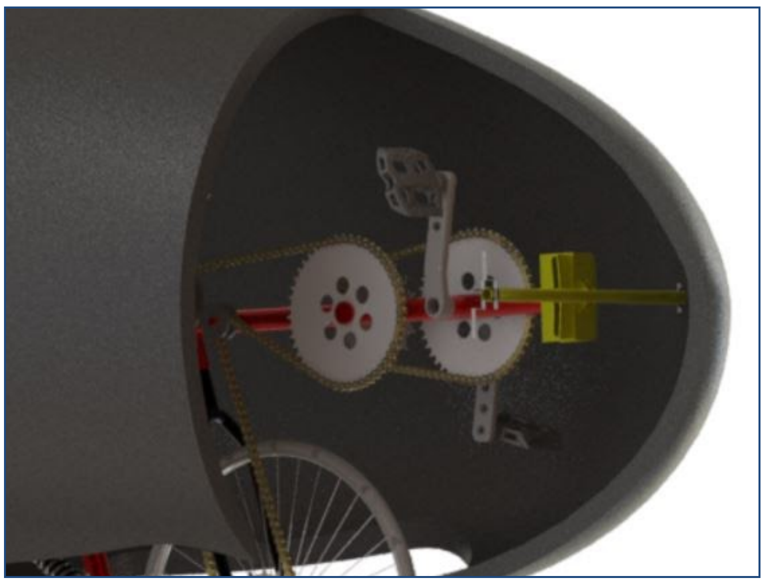


Figure 5: Fairing Mount Assembled

1.6.2 Addition of Secondary gear:

Implementing the front suspension reduces chain vibrations in drivetrain significantly and to reduce it further, free length of the chain is to be reduced. This could be achieved by either adding idles or adding a separate sprocket pair. Through drivetrain analysis we found that a certain amount of gear reduction could be increased due to which the top speed will be more for the same driver input force.

Additional gear ratio is possible due to two different chain orientations. Maximum angular speed could be obtained only upto a certain limit, so considering constant angular speed, radius and P. We could obtain higher velocity by reducing acceleration.

1.6.2.1 Chain Vibration Analysis

The peddling force applied by the rider on the vehicle can be approximated as a harmonic. Analysis was done to find the steady state amplitude of vibration of chain element under harmonic excitation input force and variable distance between Front Driving sprocket and Primary gear reduction.

Methodology- Analysis has been done using MATLAB Live Scripts on top speed achieved through drivetrain analysis i.e.18.65 m/s. Mass density of chain = 0.2273 kg/m (Generic Shimano IG51 Steel chain)

Assumptions-

- Input force is uniform and periodic and Mode of vibration is 1st overtone
- Chain is homogeneous and system is undamped

Results-

From the above analysis we found a satisfactory distance between front driving sprocket and secondary gear reduction which is equal to **15.0 cm** from driving sprocket. At this distance amplitude ratio is also minimum.

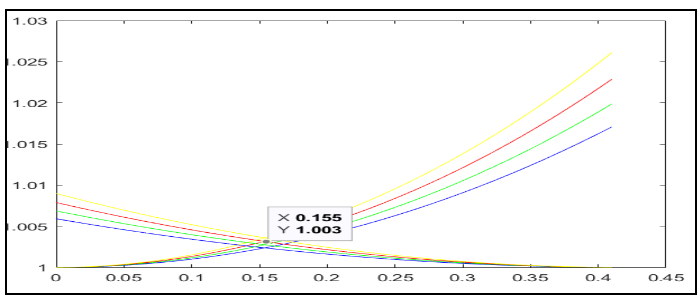


Figure 6: Amplitude Ratio (y axis) vs Position of Mount for secondary sprocket pair (x axis)

Note-

1. Same analysis was also done for the 2nd and 3rd overtone and we found similar results.
2. Physical assembly of the vehicle at this configuration is not possible. So the position at **18.7 cm** was chosen as the next best alternative.

2 Concept Evaluation

2.1 Front Suspension analysis:

Objective: To find travel of front and rear suspension.

Methodology:

- Made a Free Body Diagram and found forces on all suspension links and calculated sprung and unsprung mass for both suspensions.

- We assumed that suspension will be critically damped because we want that our vehicle not to vibrate for long, come to in equilibrium soon, and it is possible in only a critically damped system. (Damping ratio = 1)
- We defined width and height of bump as per National Highway Authority of India Guidelines for City Traffics and calculated the approximate velocity in Y direction by this given method –

  Vh = velocity of vehicle in x direction.
  Vy = velocity of vehicle in y direction.
  H = height of bump= 12cm.
  W = width of bump = 45cm.
  Vy = H / ((W/2)/Vh) (velocity = distance/time)

- From steps, velocity we found force due to change in momentum in y direction.
- As mentioned above, the motion of suspension is critically damped so we can write displacement of suspension as a function of time.

$$x(t) = (c_1 + c_2 t)e^{-\omega_n t}$$
$$\Rightarrow x(t) = e^{-\omega_n t}\left[x_0 + (v_0 + \omega_n x_0)t\right]$$

Assumptions-

- Rider's weight acts as a point load at saddle point.
- Force transability due to mass of fairing on spring based mount at the boom is taken to be 0 (As very soft springs have been used).

Result and Conclusion-

- Maximum travel of front suspension is **7cm.**
- Maximum travel of rear suspension is **2.8cm.**
- A lesser suspension travel at rear was finalised in the view of additional dead weights put at the rear during endurance event in HPVC 2020.

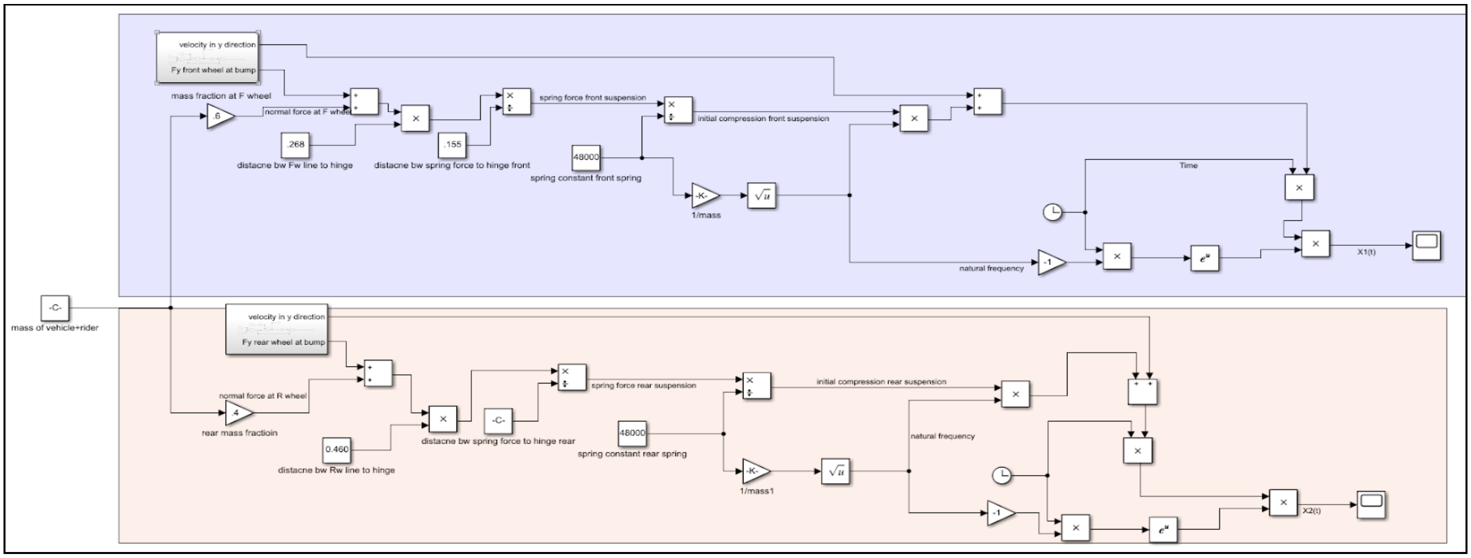


Figure 7: Simulink model for suspension system

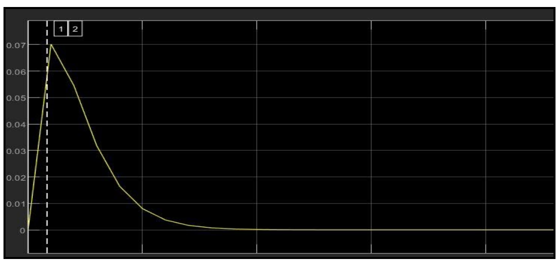


Figure 8 : Front Suspension Displacement (y axis) vs Time (x axis)

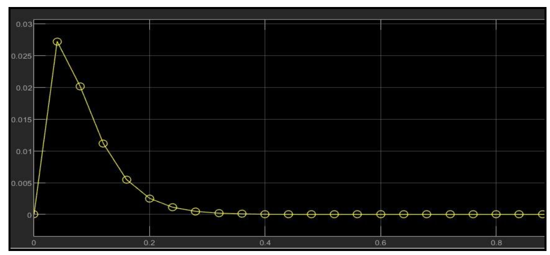


Figure 8 : Rear Suspension Displacement (y axis) vs Time (x axis)

2.2 Unanticipated Benefits:

This innovation unexpectedly improved the torsional stiffness of the frame. This will enhance structural stability and integrity if acted upon by any unforeseen Torsional load , say during an accident. The directional deformation decreased by 43.9 %.

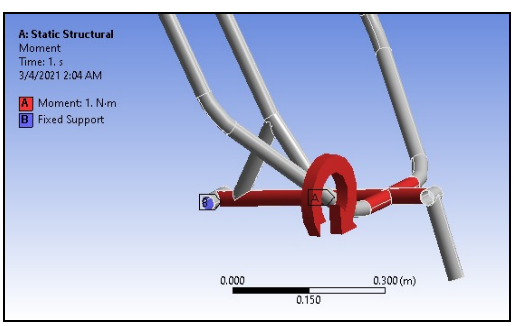


Figure 9 : Boundary Conditions for Prancer 3.0

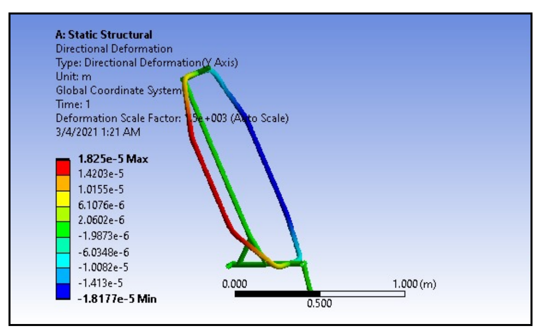
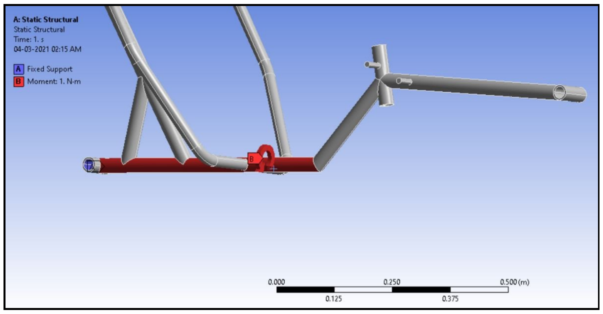


Figure 10: Directional Deformation under given loads for Prancer 3.0



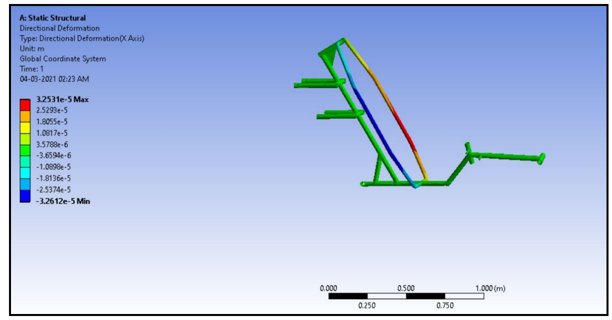


Figure 11:Boundary Conditions for Prancer 2.0

Figure 12: Directional Deformation under given load for Prancer 2.0

During the design phase, to further reduce the vibrations in the chain , we added additional idlers on the boom, we saw this as an opportunity to add a sprocket pair to further enhance the net gear ratio.

3 Learnings

3.1 Failures experienced:

- Faced difficulties in fixing the faring to the frame of HPV. Rigid fixing of fairing would have made the whole suspension suspension system pointless, so our team decided to fix the fairing using dynamic mounts thus allowing relative motion between the boom and fairing when our suspension is in work.
- Front suspension may cause the vehicle to slow down during uphill ride as some of the power transmitted through pedals will be absorbed by it. But it generally happens at high intensity.

3.2 Learning from failures

Implementation of the front suspension system comes with its own challenges. We had to trade off between it’s pros and cons. In our case, the proposed design had some unanticipated advantages and disadvantages which we became aware of during the analysis and evaluation phase. So, our team worked upon the secondary problems and failures to minimize them which have been mentioned in the design section of this report.

3.3 Unanticipated negative aspects of design:

- **Increased Weight:** Addition of front suspension has increased the the net weight of vehicle hence requiring more power to keep it moving at same speed in comparison to our last year’s HPV i.e. Prancer 2.0
- **Increase in Overall Cost:** After implementation of the proposed design for the front suspension system we will be requiring more structural material to manufacture the vehicle and some other additional parts also need to be purchased such as shock

absorbers, connecting pins, etc, thus increasing the overall cost.
- **Requires additional maintenance:** It includes regular fluid change of shock absorber and lubrication of hinge joints. Thus extra time and money needs to be spent.

4 Conclusion

This is a collective effort to redesign drivetrain and suspension for a front wheel drive and front wheel steered fixed bottom bracket type semi recumbent vehicle. The system under simulations performs satisfactorily by reattaining its steady state values and also represents a stable physical system. Thus we can conclude that the suspension system will perform as per the requirements posed by track conditions.

5 References

[1] Gowri Shankar M , Ajay S , Prasanna N and Rajagopal T "Design and Fabrication of in-Wheel Suspension in Bicycle", published May 2020 by Researchgate.

[2] Team HPVC IIT Roorkee Design report ASME E-FEST 2020