Design and Analysis of Semi-Recumbent Bicycle

Dr G Laxmaiah[1], P Anjani Devi[2], Dr Ch Indira Priyadarshini[3], Anirudh Kishan K[4], V Karthikeya Reddy[5], S Vijaya Bhanu Deepak[6]

Professor, Chaitanya Bharathi Institute of Technology, Hyderabad, India[1,]
Assistant Professor, Chaitanya Bharathi Institute of Technology, Hyderabad, India[2,3]
UG Student, Chaitanya Bharathi Institute of Technology, Hyderabad, India[4,5,6]

---***---

Abstract:

The objective is work is to design and fabricate a Human Powered Vehicle (HPV) that is efficient, lightweight and comfortable for long range cycling distances. To achieve the stated objective, a semi-recumbent configuration is used, which ensures simple structure and comfort to the rider. A roll over protection system (RPS) is added to this type of configuration which can assure stability and safety of the rider in case of a fall. Initial design has been made taking into considerations the average anthropometric measurements of riders available. Wide range of analyses has been done on the frame to ensure the design is both safe and light in weight. Physical testing has been carried out to ensure that the desired vehicle performance is met, while simultaneously satisfying all other safety requirements.

Key words: Human Powered Vehicle (HPV), Roll over protection system (RPS), Finite Element Method (FEM)

1. Introduction

Human Powered Vehicle is used to refer all the vehicles that run on muscular power of a human. A real HPV can be powered by an electric engine, but the energy must come from a human powered generator.Electric bicycles with batteries onboard do not include to HPVs. HPVs can be found in rail, water and also road. Bicycles form the largest section of HPVs. As the increasing needs of a man is leading to faster change in climate, HPVs have a greater scope of being employed for commuting for relatively longer distances in cities. The objective of this work is to develop a HPV that can takes less effort to ride with the safety to the rider and provides back support for easier riding of long distances in cities. A recumbent bicycle is a bicycle that places the rider in a laid-back reclining position. Most recumbent riders choose this type of design for ergonomic reasons. The rider's weight is distributed comfortably over a larger area, supported by back and buttocks. On a traditional upright bicycle, the body weight rests entirely on a small portion of the sitting bones, the feet, and the hands.

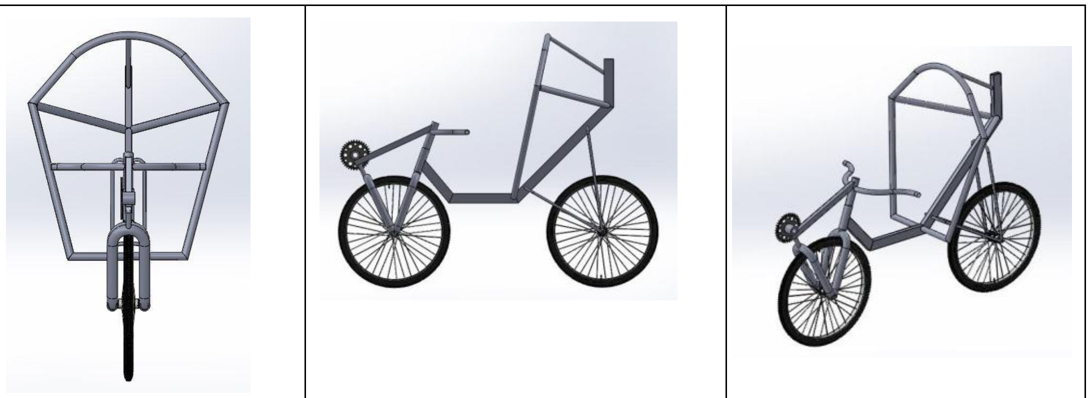
1.1 Rollover Protection System (RPS)

In the event of an accident, all vehicles must include a rollover protection system that protects all drivers in the vehicle. The RPS must Absorb sufficient energy in a severe accident to minimize risk of injury and Prevent significant body contact with the ground in the event of a fall (vehicle resting on its side) or rollover (vehicle inverted). It also Provide adequate abrasion resistance to protect against sliding across the ground. This is particularly important around the rider's arms and legs. Adequate guarding must be included. The RPS must allow for a load path supporting the driver and retaining them frombeing ejected from the HPV in the event of a crash. This load path will be defined from the ground (impact point), to the outside of the vehicle body, through the structural RPS, through the safety harness, to the driver's body (center of gravity). A thorough RPS design includes the structural fortitude of not only the roll bar/frame, but also a rigidly mounted and structurally sound seat and properly affixed safety harness.

RPS Load Cases: The RPS system shall be evaluated based on two specific load cases, a top load representing an accident involving an inverted vehicle and a side load representing a vehicle fallen on its side. In these two cases the applied load shall be reacted by constraints at the seat belt attachment points; simulating the reaction force exerted by the rider in a crash.

1.2 Performance safety requirements

The vehicle must demonstrate that it can come to a stop from a speed of 25 km/h in a distance of 6 m and can turn within an 8.0m radius. It also should demonstrate stability by traveling for 30 m in a straight line at aspeed of 5 to 8 km/h (fast paced walking speed).



Front View of the Vehicle	Side view of the Vehicle	Isometric View

Figure 1: RPS Design to with stand Side and Top Load

1.3 Cadence

It is evident that cycling performance would appear to be dictated largely by the ability of the cyclist to produce high power outputs at minimal metabolic costs. As pedal rate (i.e. cadence) can influence both the ability to produce power, as well as rate of energy consumption, cadence selection could have a significant impact on cycling performance. While information concerning pedal rate selection during cycling exists, a comprehensive review of the present literature is not currently available. As such, the cadence that results in the best possible performance outcome during the vast array of cycling events and conditions remains unclear. An elucidate knowledge about cadence abets the selection of efficient drivetrain configuration [1]. Factors affecting the cadence are Muscular Factors, Non muscular factors and Aerodynamics.

2 Material Selection

After market research and study of different materials, it narrowed down to thethree suitable materials for the purpose, those are Aluminium 6061, AISI 1018 Steel, AISI 4130 Steel. AISI 4130 grade steel is used for the fabrication of the vehicle because it is a versatile alloy with good atmospheric corrosion resistance, strength, toughness, weldability and machinability.

3 Vehicle Description

3.1 Frame

The frame is designed for two wheeled front wheel drive, semi recumbent human powered vehicle which has a short wheelbase. The base frame was manufactured using rectangular tubes of AISI 4130 Material, based on design iterations the frame cross section was decided as shown in the figure 2. The frame geometry has been decided based on anthropometric data and this geometry is designed to accommodate every rider with comfort.

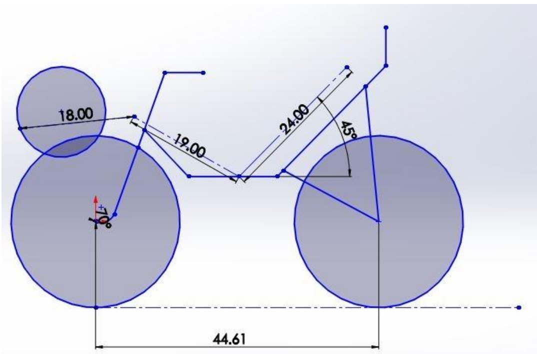


Figure 2: Ergonomically designed model for fabrication

3.2 Rollover Protection System

The RPS plays a very important role in rider's safety. The RPS was designed according to HPVC Rule book. AISI 4130 pipes of circular cross section is used to construct the Roll bar.

3.3 Drivetrain

The drivetrain of the HPV is Front Wheel drive with movable bracket system. FWD has been chosen over RWD as there are several power losses in RWD recumbent as the power is transferred from front to rear wheel through several additional pulleys. The FWD with moving bracket helps the chain not to misalign during propulsion. The powertrain consists of 7 speed cassettes so as to offer flexibility to the driver for different scenarios.

3.4 Wheels

HPV is equipped with 24" tyre on the front and 24" on the rear. Though the vehicle is Front wheel driven, a 24" wheel has been used to avoid interference between sprocket and wheel.

4. Results and Discussion:

The frame is modeled in Solidworks software and this model is imported to ANSYS analysis software

4.1 Top Load Analysis: in this analysis a load [3] of 2670N applied to the top of the roll bar directed downwards towards to rear of the vehicle at an angle of 12 degrees from the vertical, the reactant force is applied to seat belt, seat or roll bar attachment point. The Vonmises stress induced, total deformation and factor of safety were identified

4.2 Side Load Analysis: The RPS is provided to protect the rider from getting in contact with the ground atthe time of collision or imbalance. A side load of 1330N shall be applied horizontally to the side of the roll bar at shoulder height. The reactant force was applied to seat belt, seat or roll bar attachment. The maximum deformation, Vonmises stress and Factor of safety are calculated. Both the top load and side load results are tabulated in the table1

Table 1: Result for RPS analysis

Case	Maximum elastic deformation (mm)	Maximum Vonmises stress (MPa)	Factor of Safety
Top load	1.486	117.25 MPa	3.92
Side load	4.959	137.61 MPa	3.34

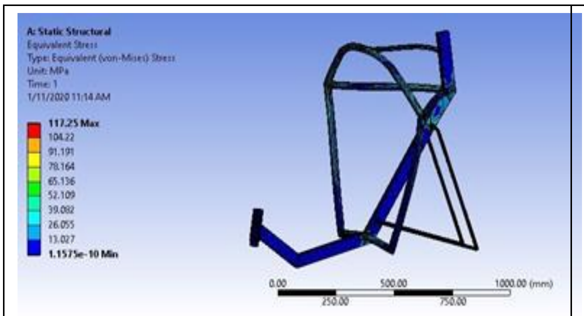


Figure3*: Top load induced Vonmises Stress*

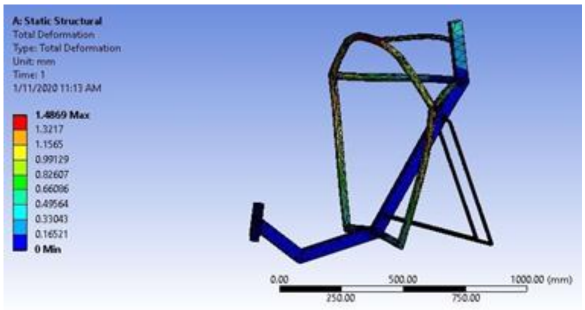


Figure 4*: Top load Total Deformation*

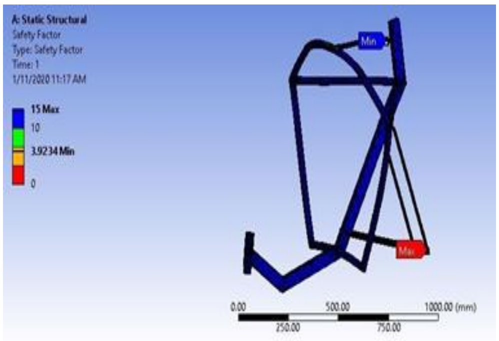


Figure 5*: Top load Factor of Safety*

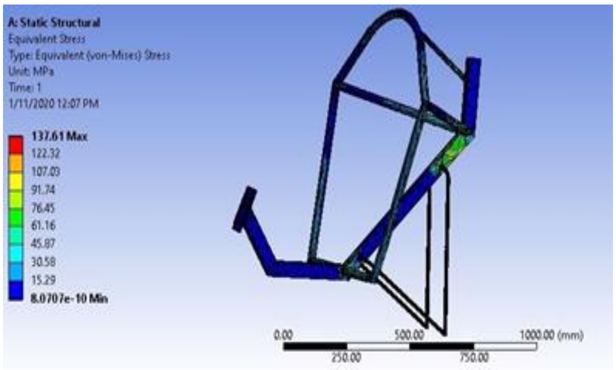


Figure 6*: Side load induced Vonmises Stress*

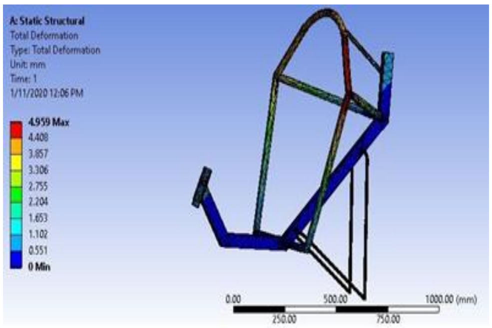


Figure 7*: Side load Total Deformation*

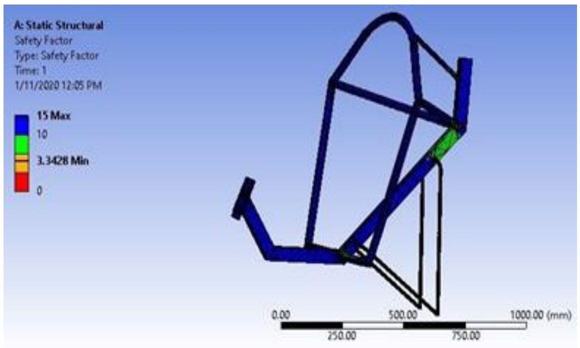


Figure 8*: Side load Factor of Safety*

4.3 Static Analysis: The rider is assumed to be sit on seat member and the behavior of the frame under the rider's weight is analyzed statically. The maximum rider weight is assumed to be 800N.The load is applied on seat member and fixed support constraints were applied at head tube and rear dropout.

4.4 Bump Analysis

In this case, the behavior of frame is simulated when the vehicle encounters a bump. The front fork and handle assembly takes the shock and it is transferred to the head tube in the frame. A remote force of 1300N representing the impact is applied at the point at which the wheel axle is supposed to be present. The equivalent stress, maximum deformation & factor of safety are calculated.

4.5 Maximum Brake Force Analysis

This case simulates the loading on frame during the braking of vehicle. The average speed of vehicle is considered and target speed for stopping is fixed and the braking force required is calculated and this force is transmitted through the axle. The seat member is fixed and remote force is applied representing the front wheel braking force and rear wheel braking is being transmitted to the frame through fork. It has been observed that stress induced is within the limits

The results obtained from static analysis, bump analysis and brake force analysis are tabulated in the table2

4.6 Braking Analysis

Force applied at the brake F1=65N, Braking factor taken = 5

Force generated at the caliper F2= 65x5 = 325N

Torque produced by the braking system

$$T_b = 2 \times \mu \times F_2 \times r_e \quad (1)$$

= 2 x 0.75 x 325 x 0.25 = 121.88 **N-m**

(Considering the two brake pads in contact)

μ= coefficient of friction between brake pad and rim

re= effective radius of tyre

Torque produced at the wheel Tw= F x R (2)

R= radius of the wheel in m

For successful application of brakes **Tb ≥Tw**

Tb=Tw (3)

For rear tyre **Rr** = 0.304 m (Rear wheel diameter=24 inches)

121.88 = **Fr** x 0.304

Frictional force acting on the rear tyre **Fr**=400.92N

For front tyre **Rf** = 0.304 m (Front wheel diameter=24 inches)

121.88 = **Ff** x 0.304

Frictional force acting on the front tyre **Ff** =400.92NMass **m** = 100kg

acceleration = **a** m/s^2 ∑F = m x a (4)

400.92+400.92 = 100 x **a** a = 8.01 m/s^2

Stopping Distance (**S**): Let **u**= 25km/hr= 6.94 m/s

$v^2 - u^2 = 2 \times a \times S$ (5)

$0-(6.94)^2=2 \times (-8.01) \times$ **S**, S= 3.00 m. The stopping distance is found to be **3**m.

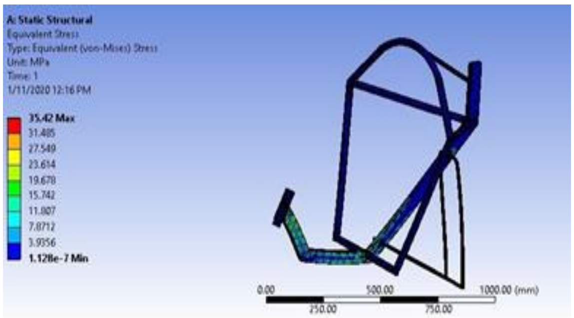


Figure 9*: Static Induced Vonmises Stress*

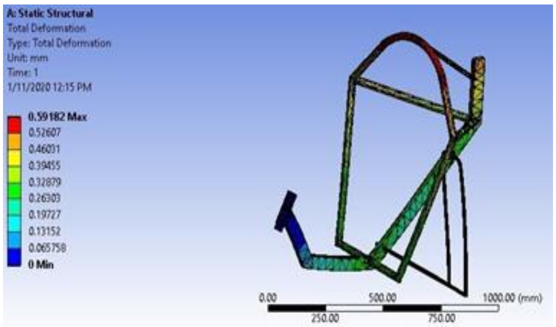


Figure 10*: static Total Deformation*

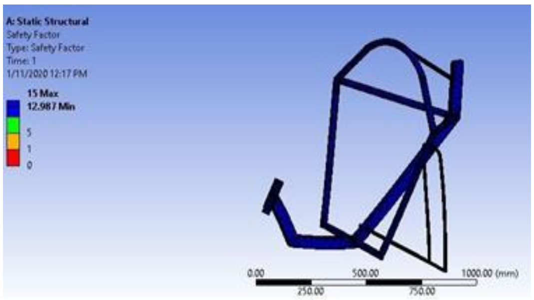


Figure 11*: static Factor of Safety*

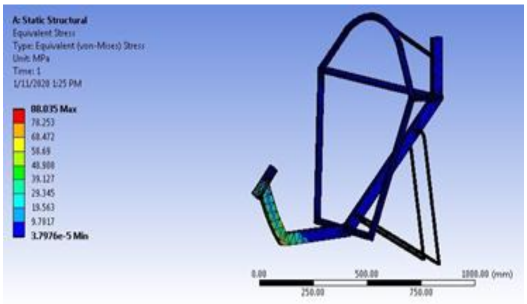


Figure 12*:bump Induced Vonmises Stress*

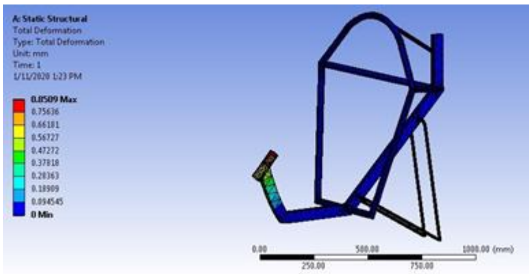


Figure 13*:bump Total Deformation*

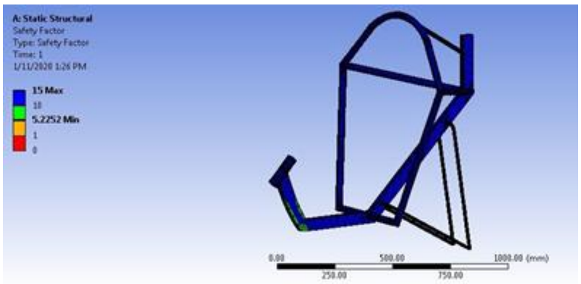


Figure 14*: bump Factor of Safety*

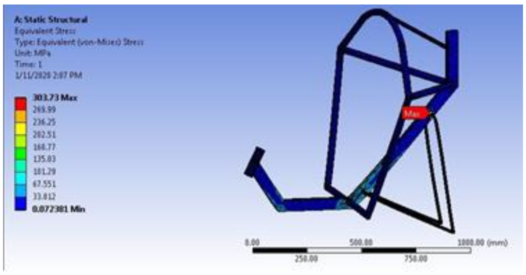


Figure 15*: break force Induced Vonmises Stress*

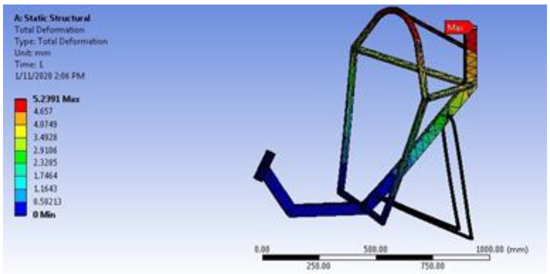


Figure 16*: break force Total Deformation*

Table 2: Result for structural analysis

Case	Maximum elastic deformation (mm)	Maximum Vonmises stress (MPa)	Factor of Safety
Static Analysis	0.591	35.42	12.9
Bump Analysis	0.85	88.03	5.22
Braking Analysis	5.24	303.73	1.51

4.7 Steering Analysis

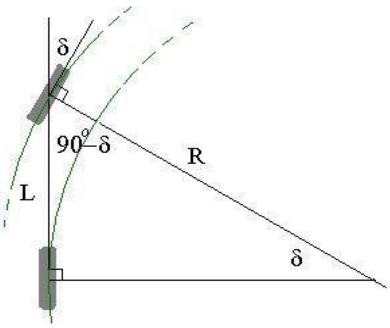


Figure 17 Steering Geometry

Calculation of steer angle:

The turning radius can be determined by the relation

$$\sin(\boldsymbol{\delta}) = L/R \qquad (6)$$

Where **L**= Wheel base of the vehicle (m)

R= Radius of turning = 8m

δ = angle through which wheel turnsThe vehicle has a wheelbase of **1.13**m

Minimum radius of turning according HPVC rulebook 2020 is **8**mThe maximum value of sin(**δ**) = **L/R** = 1.13/ 8 = 0.1412

δ = $\sin^{-1}(0.1412) = 8.12^{o}$

Therefore minimum steer angle is $\mathbf{8.12^{o}}$

4.8 Drivetrain Analysis

The chain ring and cog specifications are chosen based on requirement for both drag and endurance events

Table 3: Cadences for different events

Event	Optimum Cadence(rpm)
Drag Race	120
Endurance	90
Ultra-Endurance	70

The speeds for the chosen cadence and gear ratios are tabulated in the table4.

Table 4: Drivetrain Analysis

Chainring	Cog	Distance travelled in 1rotation of the crank (inches)	Speed at70rpm Cadence(Kmph)	Speed at90rpm Cadence(Kmph)	Speed at 120rpm Cadence(Kmph)	Gearratio
50	12	314	33.50	43.06	57.42	4.166
50	14	270	28.80	37.00	50.00	3.571
50	16	236	25.17	32.36	43.16	3.125
50	18	209	22.30	28.65	38.19	2.777
50	21	180	19.20	24.70	33.00	2.381
50	24	157	16.75	21.55	28.72	2.083
50	28	135	14.40	18.50	24.70	1.785
50	32	118	12.58	16.18	21.58	1.562

4.9 Vehicle Testing

This section contains various static and dynamic tests conducted on thevehicle to check its performance, reliability and functioning.

Top Load Testing: In the gymnasium, weight lifting rod was initially placed on the top member of the RPS and weights of magnitude 27kg were added on the either side of the rod. Deflections were noted with weights and after removing the weights. The net deflection obtained as 7.62mm

Side Load Testing: In the gymnasium the deadweights are to apply a load on side member of RPS. Weightlifting rod was initially placedon the side memberof the RPS and weights of magnitude 27kg were added and deflections were noted with weights and after removing the weights. The net deflection obtained as 5.08 mm

Figure 18: Top load and Side Load Testing

5.Conclusions:

The following conclusions can be drawn from the interpretation of the results of the HPV obtained from the analysis, testing.

- It has been observed that there isn't a considerable deviation in the results obtained from analysis and testing. The total weight of the vehicle is 24kg.
- The roll bar could withstand the weights with minimum deformation
- Stopping distance of the vehicle was found to be 4.8m and the top speed achieved by the vehicle was 60kmph
- The ergonomics, efficient powertrain, steering, and braking design ensured that vehicle could be driven long distances without facing any discomforts

6.References:

1) Dr Chris R Abbiss, Dr Jeremiah J Peiffer, Prof Paul B Laursen. (2009). Optimal cadence selection during cycling. International Sport MedJournal, Vol. 10 No.1, 2009.

2) Alexander S. Whitman. (2016). A Systematic Approach to Human Powered Vehicle Design with an Emphasis on Providing Guidelines for Mentoring Students. Literature review of Ergonomics in HPV Design (pp.157-158)

3) American Society of Mechanical Engineers, "Rules for the 2020Human Powered Vehicle Challenge," 2020.

4) Ch. Indira Priyadarsini, B. Srujeeth Khanna.C Raviteja (2019). Analysis of a Human powered vehicle. International Journal of Management, Technology And Engineering, Volume IX, Issue I, January/2019,ISSN NO : 2249-7455. *pp3104-3113*

5) The stability of the bicycle - David E. H. Jones.

6) Chavarren, J., and Calbet, J. a L., 1999, "Cycling efficiency and pedalling frequency in road cyclists," Eur. J. Appl. Physiol. Occup. Physiol., **80**(6), pp. 555–563

7) Abbiss, C. R., and Laursen, P. B., 2005, Models to Explain Fatigue during Prolonged Endurance Cycling, *Sports Medicine* volume 35, pages 865–898 (2005)

8) Danny Too 1988, "The Effect of Body Configuration on Cycling Performance," 6 International Symposium on Biomechanics in Sports (1988), pp. 51–64.

9) Indiana State Legislature, "Chapter 11: Bicycles and Motorized Bicycles," 2013. [Online].

10) Porter, J. M., Case, K., Freer, M. T., and Bonney, M. C., 1993, "Computer aided ergonomics design of automobiles," Automotive Engineering, pp.47–77.

11) Morton, R. H. H., and Billat, L. . V., 2004, "The critical power model for intermittent exercise.,"Eur. J. Appl. Physiol., 91(2-3), pp. 303–307.

12) McCartney, N., Heigenhauser, G. J., and Jones, N. L., 1983, "Power output and fatigue of human muscle in maximal cycling exercise.," J. Appl.Physiol., 55(1 Pt 1), pp. 218–224.