VELLORE INSTITUTE OF TECHNOLOGY, VELLORE
FORMULA IMPERIAL HVC 2020
CAR #690 – DESIGN REPORT

POWERTRAIN

System Overview

Parallel hybrid drivetrain with:

- **Saietta 119R** motor with peak rpm **4176rpm** and peak torque **64.69Nm**.
- Modified **KTM Duke 200** engine with peak torque **14.11Nm** and power **15.26hp.**
- Both the power sources are coupled through a **planetary gear system coupler** which provide an efficient way to shift from one source to another and allows to drive both the power sources at the same time
- **Drexler** limited slip differential.

Energy Distribution

Keeping the space constraints, performance characteristics, maximum energy limit and the results from simulations on Optimum Lap the following energy distribution was decided between the fuel and the accumulator.

IC:

Total energy from the fuel = Calorific value of the fuel (Octane 87) * maximum amount of fuel used.
Fuel used = 0.68 litres
Fuel Calorific Value = 32 MJ/litres
Total energy from the fuel = 32.4*0.68MJ = **21.76 MJ**

Battery:

Voltage = 48.1 V
Capacity = 78 Ah
Power = 48.1 * 78 = 3751.8 Wh
Energy = 3751.8 * 3600 * 10^{-6} = 13.506 MJ

Total:

IC + Battery = 21.76 + 13.506 MJ**= 35.26 MJ**
This is total the endurance energy allocation.

Engine Selection:

The engine selected for TU-20 is from KTM Duke 200 that is a standard or "naked" motorcycle made by KTM and sold in most markets worldwide. This is a single cylinder engine having a displacement of 199.5 cc which is well within the engine displacement limit of the competition. This engine has been taken from our previous car with which we had participated in Formula Hybrid where the engine size limit was 250cc. This engine also gives us a better power to weight ratio over more readily available twin cylinder engines within the engine displacement limit of the competition. The technical data available on open sources like the internet of this engine was also higher than other comparable engines. The availability of data makes the design and determination of various components in the powertrain of the vehicle much more accurate and easier.

Motor Selection:

After doing the analysis and theoretical calculation of the resistive forces acting on the car keeping in mind the desired speed, we have found Saietta 119R motor as best suitable for our car. Also, it has high efficiency of 91% & power of 15.9kW, compact size, light weight, high power-to weight ratio and is easily serviceable. The availability of the motor was the biggest deciding factor for the motor.

Motor Controller Selection:

Considering the motor, Team has selected **Kelly KDHE 12601E** controller which is an **opto-isolated** programmable controller through Kelly controller software and falls in the range of our motor specification. The Controller can monitor the over voltage and under voltage of the battery, and also can monitor the current of the battery. This motor controller comes with wide range of operating input voltage up to 124V DC and 600 Amps current.

Tractive Force Analysis:

Formulas used:

- **Velocity of wheel in Meter per second**
  $V_{m/s} = RPS_{GR} * r_{tyre}$ [r_{tyre} = radius of tire (meter), RPS_{GR}=rotations per second of the tire]
- **Coefficient of rolling resistance (f_r)**
  $f_r = 0.015(1+V/160)$
- **Rolling Resistance**
  $RR = M*g*f_r*\cos(\alpha)$ [α =inclination angle (in degrees) w.r.t horizontal=5°]
- **Aerodynamic Drag:** $AR = 0.5 * \rho * A_f * C_d * (V - V_w)^2$ [ρ= Air density; A_f = frontal area=0.0165m^2; V = vehicle speed; C_d = 0.15; V_w = wind speed]
- **Grading Resistance**
  $GR = M*g*\sin(\alpha)$
- **Tractive Effort**

$TE = T_{GR}/\ r_{tyre}$ [TGR=Torque at driven wheels]

- **Net Tractive Force**
  NRF = TE – (RR + AR + GR)
- **Acceleration**
  $a = NRF/\ M_{vehicle}$

<u>Calculation for torque required to attain particular speed:</u>

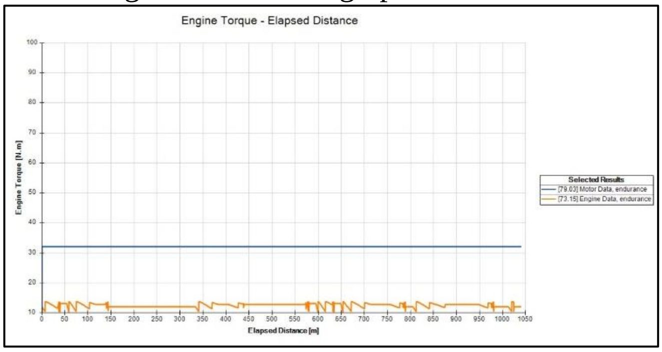
Velocity (Kmph)	Velocity(m/s)	FR	RR	AR	GR	TE	Torque
0	0	0.015	20.743	4.945	256.499	282.187	71.676
5	1.389	0.015	20.923	2.579		280.002	71.120
10	2.778	0.015	21.103	0.977		278.580	70.759
15	4.167	0.015	21.283	0.137		277.920	70.592
20	5.556	0.016	21.463	0.061		278.024	70.618
25	6.944	0.016	21.643	0.748		278.891	70.838
30	8.333	0.016	21.823	2.198		280.521	71.252
35	9.722	0.016	22.003	4.411		282.914	71.860
40	11.111	0.016	22.183	7.387		286.070	72.662
45	12.500	0.016	22.363	11.126		289.989	73.657
50	13.889	0.016	22.543	15.628		294.671	74.846
55	15.278	0.016	22.723	20.893		300.116	76.230
60	16.667	0.017	22.903	26.921		306.325	77.806
65	18.056	0.017	23.083	33.713		313.296	79.577
70	19.444	0.017	23.264	41.267		321.030	81.542
75	20.833	0.017	23.444	49.584		329.528	83.700
80	22.222	0.017	23.624	58.665		338.789	86.052
85	23.611	0.017	23.804	68.509		348.812	88.598
90	25.000	0.017	23.984	79.115		359.599	91.338
95	26.389	0.017	24.164	90.485		371.149	94.272
100	27.778	0.018	24.344	102.618		383.462	97.399

Here RR is the rolling resistance, AR is the aerodynamic resistance, GR is the gradient resistance and TE is the required tractive effort to reach the given vehicle speed. The below image provides the tractive effort generated by the engine which is available at the wheels of the vehicle at all 6 gears. A1, A2, A3… Are the accelerations at each gear:

TE 1	TE 2	TE 3	TE 4	TE 5	TE 5	NRF 1	NRF 2	NRF 3	NRF 4	NRF 5	NRF 6	A1	A2	A3	A4	A5	A6
0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
665.87	485.68	365.57	290.95	245.69	215.42	386.50	206.31	85.44	9.16	-38.38	-71.33	1.29	0.69	0.28	0.03	-0.13	-0.24
865.63	631.39	475.24	378.23	319.39	280.04	586.30	351.94	194.79	95.75	34.21	-8.29	1.95	1.17	0.65	0.32	0.11	-0.03
965.51	704.24	530.07	421.88	356.24	312.35	686.21	424.68	249.25	138.62	69.83	22.28	2.29	1.42	0.83	0.46	0.23	0.07
1023.78	746.74	562.06	447.34	377.74	331.20	744.48	467.04	280.80	163.22	89.97	39.24	2.48	1.56	0.94	0.54	0.30	0.13
1065.39	777.09	584.91	465.52	393.10	344.66	786.09	497.22	303.17	180.47	103.85	50.65	2.62	1.66	1.01	0.60	0.35	0.17
1082.04	789.24	594.05	472.79	399.24	350.05	802.71	509.17	311.77	186.72	108.41	53.84	2.68	1.70	1.04	0.62	0.36	0.18
1098.69	801.38	603.19	480.07	405.38	355.44	819.31	521.08	320.32	192.88	112.84	56.87	2.73	1.74	1.07	0.64	0.38	0.19
1111.17	810.49	610.04	485.52	409.99	359.47	831.74	529.92	326.53	197.14	115.62	58.41	2.77	1.77	1.09	0.66	0.39	0.19
1119.50	816.56	614.61	489.16	413.06	362.17	839.99	535.70	330.41	199.50	116.75	58.44	2.80	1.79	1.10	0.67	0.39	0.19
1123.66	819.59	616.89	490.98	414.59	363.51	844.07	538.42	331.94	199.96	116.23	56.98	2.81	1.79	1.11	0.67	0.39	0.19
1131.98	825.66	621.46	494.61	417.67	366.21	852.29	544.14	335.71	202.15	117.12	56.70	2.84	1.81	1.12	0.67	0.39	0.19
1133.64	826.88	622.38	495.34	418.28	366.75	853.83	544.97	335.77	201.35	115.44	54.12	2.85	1.82	1.12	0.67	0.38	0.18
1140.30	831.73	626.03	498.25	420.74	368.90	860.35	549.42	338.52	202.65	115.48	53.01	2.87	1.83	1.13	0.68	0.38	0.18
1140.30	831.73	626.03	498.25	420.74	368.90	860.20	548.97	337.55	200.95	112.95	49.58	2.87	1.83	1.13	0.67	0.38	0.17
1144.47	834.77	628.32	500.07	422.27	370.25	864.19	551.54	338.83	200.99	111.84	47.35	2.88	1.84	1.13	0.67	0.37	0.16
1165.27	849.95	639.74	509.16	429.95	376.98	884.82	566.21	349.18	208.22	116.75	50.35	2.95	1.89	1.16	0.69	0.39	0.17
1169.44	852.98	642.03	510.98	431.49	378.32	888.78	568.72	350.34	208.09	115.40	47.81	2.96	1.90	1.17	0.69	0.38	0.16
1173.60	856.02	644.31	512.80	433.02	379.67	892.72	571.19	351.46	207.87	113.93	45.12	2.98	1.90	1.17	0.69	0.38	0.15
1173.60	856.02	644.31	512.80	433.02	379.67	892.49	570.60	350.23	205.76	110.81	40.93	2.97	1.90	1.17	0.69	0.37	0.14
1165.27	849.95	639.74	509.16	429.95	376.98	883.92	563.90	344.38	199.92	104.50	33.90	2.95	1.88	1.15	0.67	0.35	0.11
1161.11	846.91	637.46	507.34	428.41	375.63	879.49	560.22	340.76	195.82	99.61	28.05	2.93	1.87	1.14	0.65	0.33	0.09
1152.79	840.84	632.89	503.71	425.34	372.94	870.89	553.46	334.80	189.81	93.07	20.71	2.90	1.84	1.12	0.63	0.31	0.07
1140.30	831.73	626.03	498.25	420.74	368.90	858.10	543.64	326.51	181.91	84.87	11.87	2.86	1.81	1.09	0.61	0.28	0.04
1131.98	825.66	621.46	494.61	417.67	366.21	849.47	536.83	320.45	175.73	78.09	4.22	2.83	1.79	1.07	0.59	0.26	0.01
1115.33	813.52	612.33	487.34	411.52	360.82	832.49	523.91	309.76	165.84	68.12	-6.28	2.77	1.75	1.03	0.55	0.23	-0.02
1098.69	801.38	603.19	480.07	405.38	355.44	815.50	510.97	299.02	155.86	58.04	-16.92	2.72	1.70	1.00	0.52	0.19	-0.06
1073.72	783.17	589.48	469.16	396.17	347.36	790.17	491.92	283.66	142.16	44.76	-30.42	2.63	1.64	0.95	0.47	0.15	-0.10
1057.07	771.02	580.34	461.88	390.03	341.97	773.15	478.92	272.81	132.02	34.44	-41.37	2.58	1.60	0.91	0.44	0.11	-0.14
1032.10	752.81	566.63	450.97	380.81	333.89	747.78	459.81	257.35	118.15	20.93	-55.17	2.49	1.53	0.86	0.39	0.07	-0.18
998.81	728.53	548.35	436.42	368.53	323.12	714.08	434.60	237.25	100.56	4.23	-71.82	2.38	1.45	0.79	0.34	0.01	-0.24
973.84	710.31	534.64	425.51	359.32	315.05	688.69	415.44	221.68	86.53	-9.51	-85.92	2.30	1.38	0.74	0.29	-0.03	-0.29
944.70	689.06	518.65	412.78	348.57	305.62	659.11	393.20	203.77	70.59	-24.91	-101.53	2.20	1.31	0.68	0.24	-0.08	-0.34
915.57	667.82	502.66	400.06	337.82	296.20	629.52	370.94	185.80	54.57	-40.43	-117.29	2.10	1.24	0.62	0.18	-0.13	-0.39
882.28	643.53	484.38	385.51	325.53	285.43	595.76	345.61	165.50	36.65	-57.60	-134.55	1.99	1.15	0.55	0.12	-0.19	-0.45
848.99	619.25	466.10	370.96	313.25	274.65	561.97	320.25	145.14	18.64	-74.89	-151.96	1.87	1.07	0.48	0.06	-0.25	-0.51
807.37	588.89	443.25	352.78	297.89	261.19	519.85	288.79	120.16	-3.09	-95.36	-172.21	1.73	0.96	0.40	-0.01	-0.32	-0.57
765.75	558.54	420.40	334.59	282.54	247.73	477.71	257.30	95.13	-24.90	-115.96	-192.63	1.59	0.86	0.32	-0.08	-0.39	-0.64
707.49	516.04	388.42	309.13	261.04	228.88	418.91	213.64	60.91	-54.07	-142.81	-218.57	1.40	0.71	0.20	-0.18	-0.48	-0.73

<u>Accumulator capacity and voltage determination</u>

The image below is a graph between distance



travelled by the car and the torque derived from the motor and the engine. This graph was plotted in Optimum Lap. From the image, we can see that the 32.1Nm of torque is continuously drawn from the motor. The motor produces 32.1Nm of torque at a current of 200amps. From the energy distribution between fuel and battery pack, we can use about 680ml of fuel. From tests conducted by us we have found that we can get 12kmpl mileage from the engine which means, using 680ml of fuel we can complete 8 laps on the engine driven mode. This also means that we have to complete the remaining 12 laps using the energy from the battery pack.

The lap time when driven on motor is 79.03 seconds, which implies that to complete 12 laps, the battery pack must discharge continuously at 200amps for 15.806 minutes or 16 minutes. The battery capacity required is as follows:

Capacity in Ah = $\frac{200\times15.806}{60} = 52.68\ Ah$

This is the capacity of battery pack where the cells can discharge at a rate of 4C or higher but

the cells chosen by us can discharge at a maximum rating of 3C with the capacity of one cell being 2.6Ah so to reach the discharge current of 200amps, the capacity of the battery pack needs to be $\frac{200}{2.6} = 76.92Ah$

The minimum number of cells in parallel is $\frac{76.92}{2.6} = 29.58\ cells$

Approximating this value to the nearest higher number, we get 30 cells in parallel.

For the determination of the voltage of the battery pack, the motor rpm data at voltages 12, 24, 36, 48, 60 and 72V were put in as data points into optimum lap. Increasing the voltage, increases the maximum rpm that the motor can achieve, at 48V it was found that the maximum the maximum speed that can reached by the vehicle when driven on motor is nearly 75kmph. From the lap time simulation on optimum lap, the maximum speed of the vehicle during the lap was 68kmph which could be achieved at 48V.

Therefore, the number of cells in series was determined by dividing 48 V with the nominal voltage of the cell that is 3.7V.

Number of cells in series = $\frac{48}{3.7} = 12.97$

Rounding of this value, we get 13 cells in series

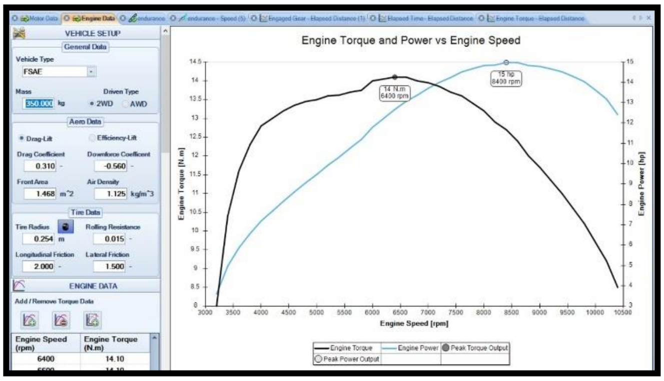
The accumulator configuration thus achieved is 13S30P.

<u>Total Reduction in the Drivetrain:</u>

Our approach to determining the reductions was centered on the average endurance speed rule in the rulebook according to which the average speed of the car needs to be around 45kmph for the endurance event. To find the reduction we used Optimum L software where we created an approximation of the track for the endurance event.
Two vehicles were created on the software, one for the engine drive mode and the other for the motor drive mode. The vehicle mass, driven wheels, longitudinal and lateral friction coefficients, frontal area, aerodynamic drag, etc. were some of the basic data given as input for the vehicle that remained same for both the vehicles. Since, we were taking part in the event after a span of 3 years our aim was to make the car such that we could compete and effectively complete all he dynamic events. Multiple iterations were done by varying the total reduction on each vehicle setup in optimum lap and the best value for total reduction was determined for both the vehicles. After this, a final value of final reduction and coupler reduction was chosen such that minimum compromise was done to the vehicle performance. The total reduction for the engine was **2.856** and **3.17.**

Engine Drive Mode: Vehicle Data and Performance Results:

RPM and torque data points, the gear ratios and the final drive ratio were the parameters that were different in the two vehicles created. Engine data was provided for the first vehicle.



Vehicle 1: engine driven mode

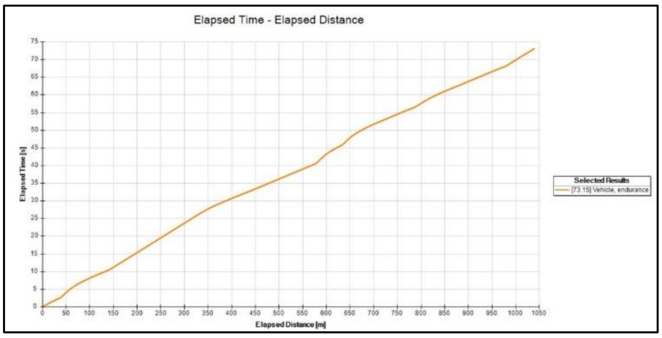
Engine Speed (rpm)	Engine Torque (N.m)
3200	8.00
3400	10.40
3600	11.60
3800	12.30
4000	12.80
4200	13.00
4400	13.20
4600	13.35
4800	13.45
5000	13.50
5200	13.60
5400	13.62
5600	13.70
5800	13.75
6000	14.00
6200	14.05
6400	14.10
6600	14.10

Engine Speed (rpm)	Engine Torque (N.m)
6800	14.00
7000	13.95
7200	13.85
7400	13.70
7600	13.60
7800	13.40
8000	13.20
8200	12.90
8400	12.70
8600	12.40
8800	12.00
9000	11.70
9200	11.35
9400	11.00
9600	10.60
9800	10.20
10000	9.70
10200	9.20

	Gear Ratios
Gear 1	13.9000
Gear 2	10.2430
Gear 3	7.7143
Gear 4	6.5141
Gear 5	5.7115
Gear 6	4.4990

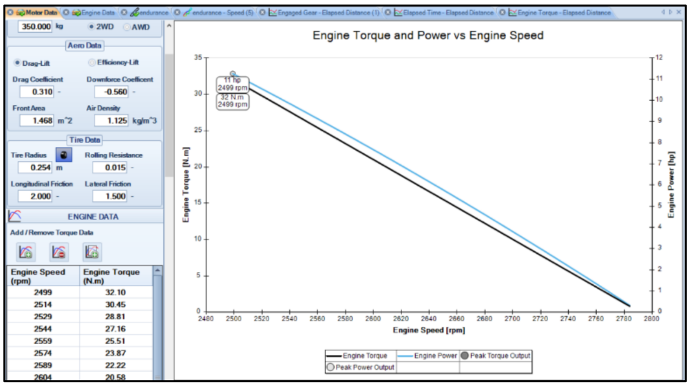
Final Drive Ratio	Drive Efficiency
1.904 -	90.000 %

Vehicle 1: Engine data points



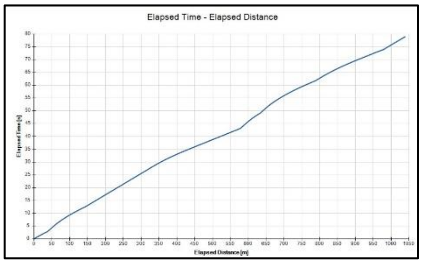
Vehicle 1: Elapsed time vs Elapsed Distance

Motor Drive Mode: Vehicle Data and Performance Results



Engine Speed (rpm)	Engine Torque (N.m)
2499	32.10
2514	30.45
2529	28.81
2544	27.16
2559	25.51
2574	23.87
2589	22.22
2604	20.58
2619	18.93
2634	17.28
2649	15.64
2664	13.99
2679	12.35
2694	10.70
2709	9.05
2724	7.41
2739	5.76
2754	4.12

Vehicle 2: motor driven mode



	Gear Ratios
Gear 1	2.6667

Final Drive Ratio	Drive Efficiency
1.190 -	90.000 %

Vehicle 2: Elapsed time vs Elapsed Distance

GLV System (Shutdown Circuit)

Following the rulebook, our car contains following components to satisfy the shutdown circuit table:

- Grounded Low Voltage Master Switch
- Tractive System Master Switch
- Two side mounted Kill Switches
- Cockpit-mounted Kill Switch.
- Brake over-travel switch.
- A normally open relay controlled by the insulation monitoring device.

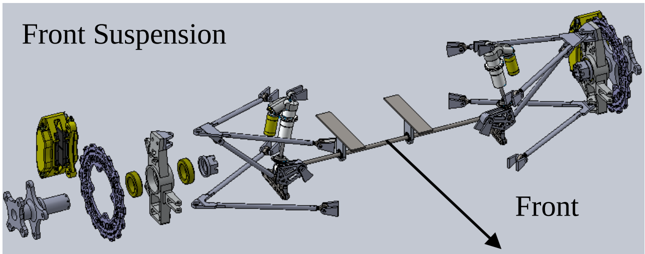
Telemetry and Data Acquisition System:

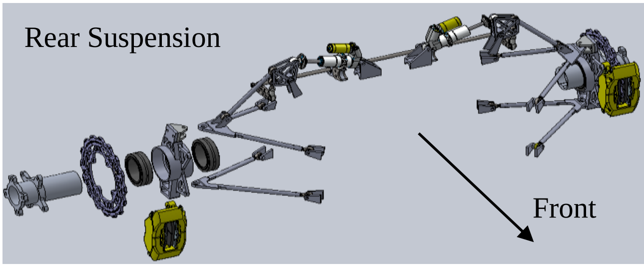
A telemetry and data acquisition system allows for the collection and interpretation of data from sensors on the car, which enables the team to not only diagnose and solve issues with the other systems of the car, but to fine-tune and optimize the geometry of the mechanical systems as well as

making suggestions to the driver based on data.
Our team wished to measure various parameters including: suspension travel of all 4 wheels, throttle pedal position, wheel speed of all 4 wheels, brake disc temperature for all 4 wheels and engine temperature. To accomplish this task, various devices and computer programming tools and software were used. The telemetry system in the car is based largely on an **Arduino Mega and Arduino UNO** microcontroller and **RF modules (Xbee Pro S3B)**. The data is transmitted via wireless communications using RF modules and displayed graphically using Telemetry Viewer.
The data from the various sensors is also displayed on a 2.4’’ TFT screen that is fitted the **3D printed steering wheel,** it helps driver to observe various important parameters like Ground Low Voltage, throttle position, engine temperature and many more.

SUSPENSION AND STEERING

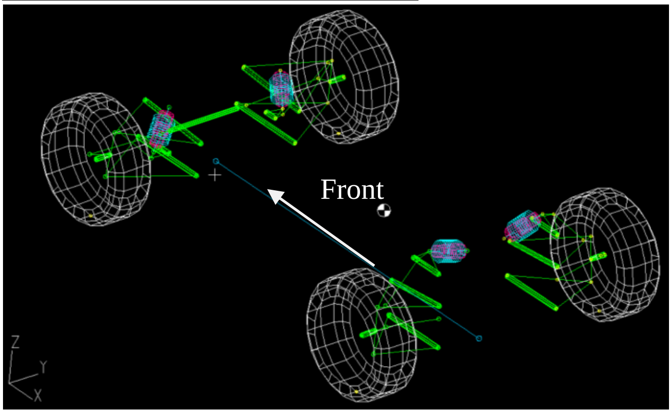
Suspension Design Consideration:





Pushrod suspension in the rear and **pull rod suspension at the front** is being used which can help the proper transmission of force to the damper. Angles have been maintained in such a way that the force transmission from the tire is as low as possible. In the front, pull rod is mounted on the upper control arms at an angle of 45.5° from the horizontal and in rear, pushrod has been mounted at an angle of 45° with the horizontal. Pushrod geometry at the rear yielded optimum results, which seemed satisfactory and was hence implemented. Asymmetric control arms at the rear and symmetrical in front are being used. Rear control arms were designed to give more usable space to the powertrain department ahead of the half shafts without increasing the car’s total length. Mounts for the pull rods were mounted to the upper control arm A plate at the front and at the upper control arm’s A plate at the rear. The force analysis was done considering normal reaction force from the ground and cornering force. **Four-point rocker** (bell crank) made out of **aluminum** has been used among which two points are used for push/pull rods and damper respectively having **1:1 ratio at the rear** i.e. the magnitude of force that is in coming is equal to magnitude of force outgoing, **0.885:1 at the front** and third point is for Anti roll bar. The wheelbase to track width ratio of our car is **1.29** which suggested the use of Anti Roll Bar to **minimize the roll rate** of the vehicle. Adding ARB at any one axle improves traction of the wheels at that axle considerably. Hence ARB has been implemented on both axles, front and rear.

Suspension Setup Parameters:



Dynamic analysis on Lotus Shark.
For the front suspension:

STATIC VALUES		
CAMBER ANGLE	(deg):	0.00
TOE ANGLE (SAE) (+ve TOE IN)	(deg):	0.00
TOE ANGLE (PLANE OF WHEEL)	(deg):	0.00
CASTOR ANGLE	(deg):	4.51
CASTOR TRAIL (HUB TRAIL)	(mm):	0.00
CASTOR OFFSET	(mm):	20.04
KINGPIN ANGLE	(deg):	4.01
KINGPIN OFFSET (AT WHEEL)	(mm):	71.69
KINGPIN OFFSET (AT GROUND)	(mm):	53.88
MECHANICAL TRAIL	(mm):	19.97
ROLL CENTRE HEIGHT	(mm):	60.40

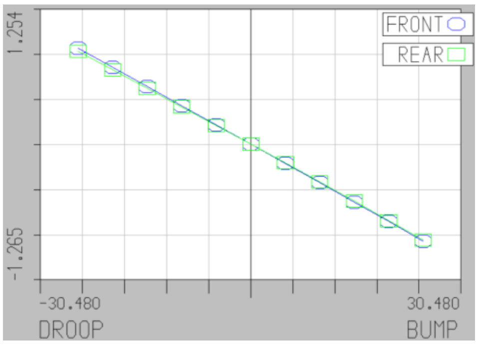
GENERAL DATA VALUES		
TYRE ROLLING RADIUS	(mm):	254.00
WHEELBASE	(mm):	1625.60
C OF G HEIGHT	(mm):	330.00
BREAKING ON FRONT AXLE	(%):	66.00
DRIVE ON FRONT AXLE	(%):	0.00
WEIGHT ON FRONT AXLE	(%):	38.30

For the rear suspension:

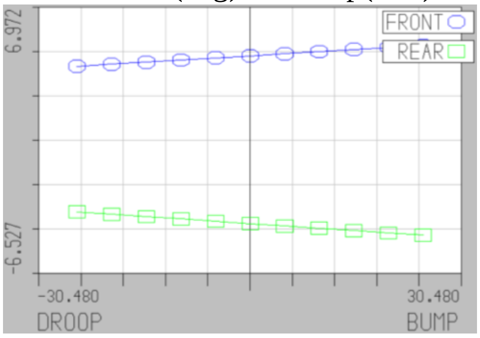
STATIC VALUES		
CAMBER ANGLE	(deg):	0.00
TOE ANGLE (SAE) (+ve TOE IN)	(deg):	0.00
TOE ANGLE (PLANE OF WHEEL)	(deg):	0.00
CASTOR ANGLE	(deg):	-4.01
CASTOR TRAIL (HUB TRAIL)	(mm):	0.00
CASTOR OFFSET	(mm):	-17.79
KINGPIN ANGLE	(deg):	3.01
KINGPIN OFFSET (AT WHEEL)	(mm):	69.92
KINGPIN OFFSET (AT GROUND)	(mm):	56.57
MECHANICAL TRAIL	(mm):	-17.75
ROLL CENTRE HEIGHT	(mm):	57.91

GENERAL DATA VALUES		
TYRE ROLLING RADIUS	(mm):	254.00
WHEELBASE	(mm):	1625.60
C OF G HEIGHT	(mm):	330.00
BREAKING ON FRONT AXLE	(%):	66.00
DRIVE ON FRONT AXLE	(%):	0.00
WEIGHT ON FRONT AXLE	(%):	38.30

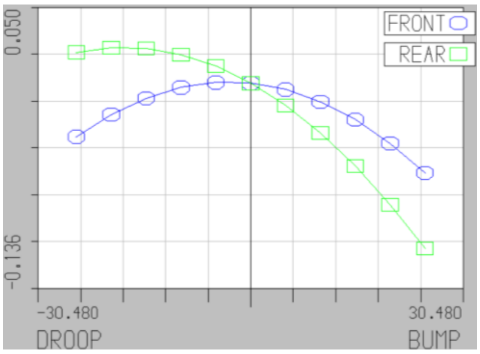
Major Lotus Simulation Results:



Camber(deg) vs Bump(mm)



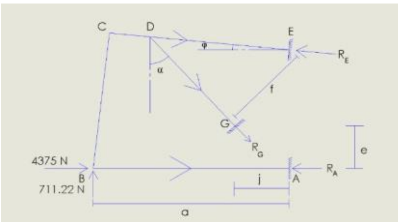
Toe(deg) vs Bump(mm)



Caster(deg) vs Bump(mm)

Calculations for Suspension Geometry:

For front:



a-j=232.93 mm; e=15.75 mm; f= 156.88 mm; φ=7.6° α=44.51°

<u>For $\sum F_X$=0-</u>

F_X=R_A + R_E cosφ – R_G sinα

<u>For $\sum F_Y$= 0-</u>

F_Y= R_G cosα - R_E sinφ

<u>For $\sum M_G$=0-</u>

F_X (e) – F_Y (a-j) -R_A(e) + R_E(f) = 0

<u>On solving these equations-</u>

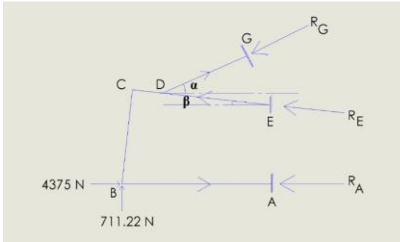
R_A= 4181.5 N

R_G= 1189.6 N

R_E= 1036.6 N

For Rear:

AB= a = 252.4 mm; EA= d = 171.74 mm; EG (perpendicular from E to DG) = f = 161.64 mm;



α= 45°;β= 7.42°

<u>For $\sum F_X$=0 –</u>

F_X=R_A + R_E cosβ + R_G cosα

<u>For $\sum F_Y$=0 –</u>

F_Y= R_G sinα – R_E sinβ

<u>For $\sum M_E$=0 –</u>

R_A(d) + F_Y(a) – R_G(f) – F_X(d) = 0

<u>On solving these equations-</u>

R_A= 4194.11 N

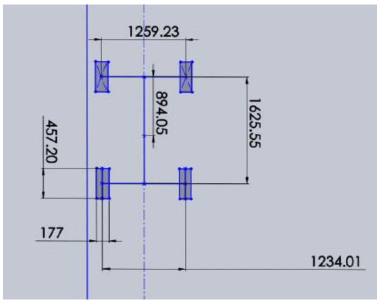
R_E= 473.18 N

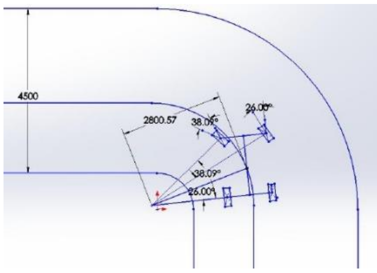
R_G= 919.39 N

Steering General Considerations:

A steering system must be able to accommodate for driver error and deviations and in order to design to this purpose, worst case cornering situations for TU20 were simulated. This model assumes multiple racing lines through the turns with minimum cornering radius in order to arrive at conservative values.

Racing Line Iterations:





Results:

Maximum steer angle required at wheel: **45 degrees**

Steering Wheel Angular Limits: **0 – 180 degrees**

Corresponding Rack Ratio: **4:1**

Total length of the rack: **16.9inches (431.9 mm)**

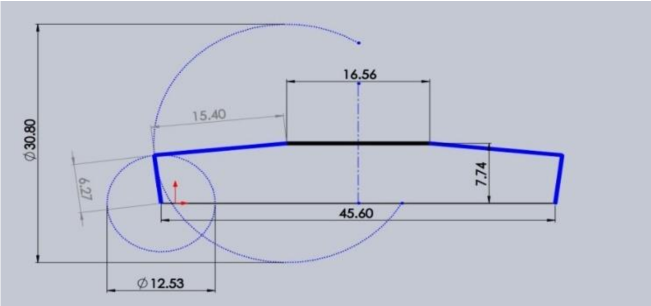
C-factor (rack travel per 360 degrees rotation of pinion gear): **3.94 inches (100mm)**

Steering System Geometry:

In order to optimize the geometry of steering linkages iteratively, a mechanism (slider-crank) has been synthesized via the use of Chebyshev's Spacing and Fruedenstein's Equation. Although this mechanism does not consider other factors related to vehicle dynamics, it provides a solution for a baseline linkage that can then be further modified.

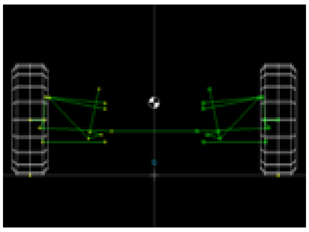
The synthesized steering linkage and the results from optimization as shown below:

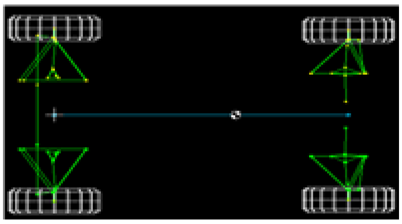
Parameter	Base Values
$Ø_o$ (Steering Arm Angle)	**7.37º**
Rack Position (Fore)	**7.74 in**
Steering Arm Length	**6.265 in**
Tie Rod Length	**15.4 in**



Optimization of Steering System:

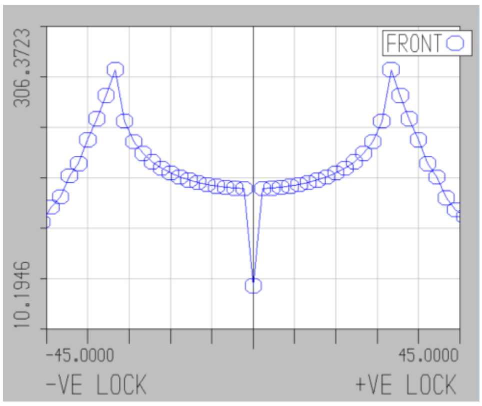
Lotus SHARK and MSC Adams were used for kinematic and dynamic analysis in order to further optimize the conditions obtained via the synthesized mechanism. Priority was given to parameters such as Ackermann (static and dynamic %), dynamic toe change, bump steer minimization, steering effort reduction via maximized steering arm length (56.78 mm). Ackermann conditions were set so that dynamic Ackermann change rate and hence toe change is high. Ackermann percentage was optimized to change from 52.5 % at static condition till 115% at steering lock with a peak of 264%. This was achieved by angularity of steering arm and hence produces a steering system that is more responsive at low speeds. Bump steer was reduced to a maximum of 0.3 degrees of steer at 25.4 mm of bump/rebound by modifying the position of the rack.



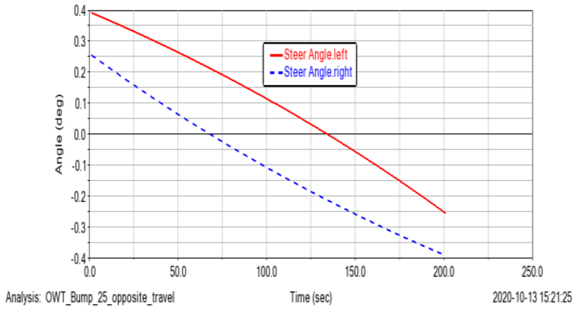


Front View Top View (Front: Rear)

Optimized Graphs:

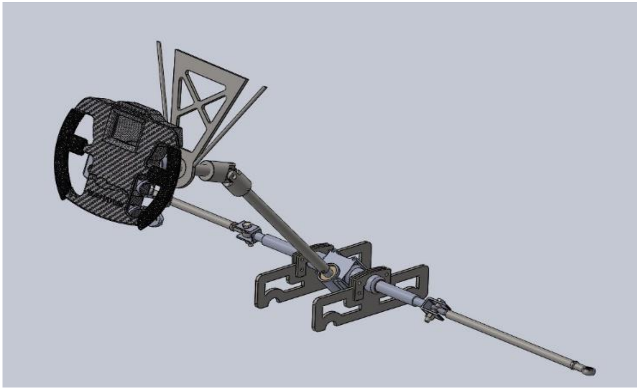


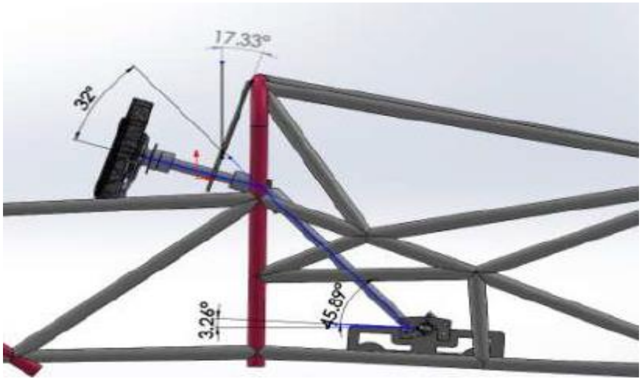
Ackerman (%)vsSteer(deg)



Toe(deg) vs Bump(mm)

Final Steering System Design:

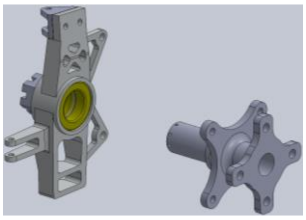


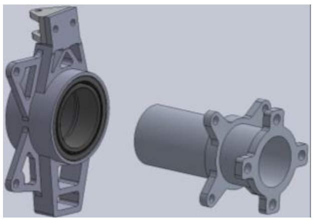


Hub and Upright Design:

The **front upright** is fabricated out of **Aluminum 7075 T6** owing to its high tensile strength and good fatigue strength. But since the fatigue life was not sufficient enough for the forces at the **rear uprights**, **AISI 1020 mild steel** was used for the fabrication process instead.

Front Upright and Hub: Rear Upright and Hub:





Aluminum 7075 is chosen for the hubs and fabrication was done using CNC. The tripod hubs have been successfully **integrated with the wheel hub** which has **better mechanical efficiency** with very less chances of failure.

BRAKES AND TIRES:

Brake Calculation:

For design, various inputs are required from other vehicle dynamics departments. All the essential inputs that are used in the calculation of the braking system are stated below:

Rolling radius R=10in; Wheelbase=64in; CG height=12in; μ_{TR}=1.5; Pedal Force=450N;

Weight distribution:

Static Front=115kgf; Static Rear=185kgf;

Load transfer=(weight * μ_{TR} * H_{CG}) / L=827.72N;

Hence, Dynamic front=1955.87N;

Dynamic Rear=987.13N;

Here, 'μ_{TR}' is Coefficient of friction between tire and road; 'g' is gravity; 'M' is mass of the car including the driver; 'H_{CG}' is the height of the center of gravity from the ground; 'L' is the wheelbase.

Force required to lock the wheel (Newton):

Front, GF=μTR * NFT=1466.90;

Rear, GR μTR * NRT 740.35;

Torque on the wheel, (N inch):

Front, TF=GF*R=14669.02;

Rear, TR =GR*R=7403.48;

Frictional force on the disc rotor, (Newton):

Front, FF=TF/ RD=2933.80;

Rear, FR =TR/ RD=1480.70;

Normal force on disc rotor, (Newton):

Front=FF/μD=7334.51;

Rear=FR/μD=3701.74;

Force on Caliper piston, (Newton):

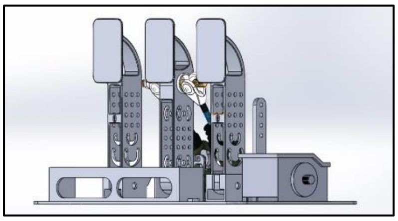
Front=Normal Force/no. of pistons(4)=1833.63;

Rear= Normal Force/no. of pistons(4)= 925.44;

Pressure in Brake line, (Newton/sq. in):

Front=Fp /AP=762.72;

Rear=Fp/AP=384;



Force required on Master Cylinder, (Newton):

Front,WF=Pressure in brake line*AMC=336.79;

Rear,WR =Pressure in brake line*AMC=231.36;

Total force,WT=WF +WR=568.15;

Pedal Ratio:

Pedal Ratio=Total force/driver effort=1.26;

Brake Biasing:

Front=WF /WT=59 %

Rear=WR /WT=41 %

Tires:

After comparison on the basis of lateral force vs slip angle and cornering stiffness vs section width, **new Hoosier 20.5/7.0-13 tires** were chosen over the old Hoosier 18.0/6.0-10 tires.

FRAME:

Design Consideration:

The frame for TU20 has been designed **keeping in mind the safety** and **ergonomics of the driver** while minimizing **CG height**. The **cockpit length** has been kept as **745mm** which provides ample space for the driver and some space is used by the battery pack. Calculation:

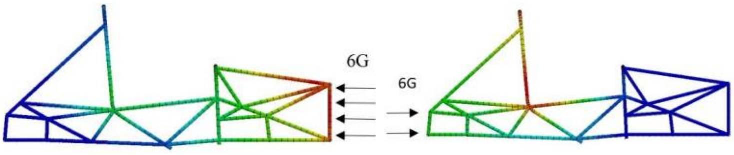
Formula: v=u+a*t; v=0;u=22.22m/s;t=0.4s;

⇨ a=55.55m/s;

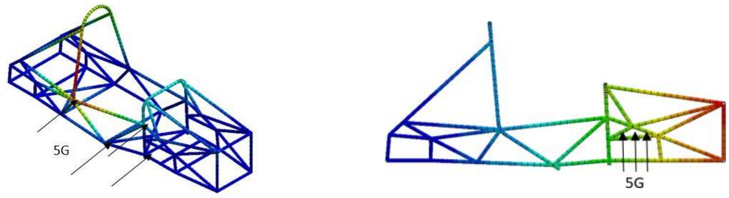
⇨ G Force=a/g=55.55/9.8=6 G Force;

Analysis:

Front Impact(FOS=2.3) Rear Impact(FOS=4.4)



Side Impact(FOS=1.2) Torsional Stiffness(FOS=3)



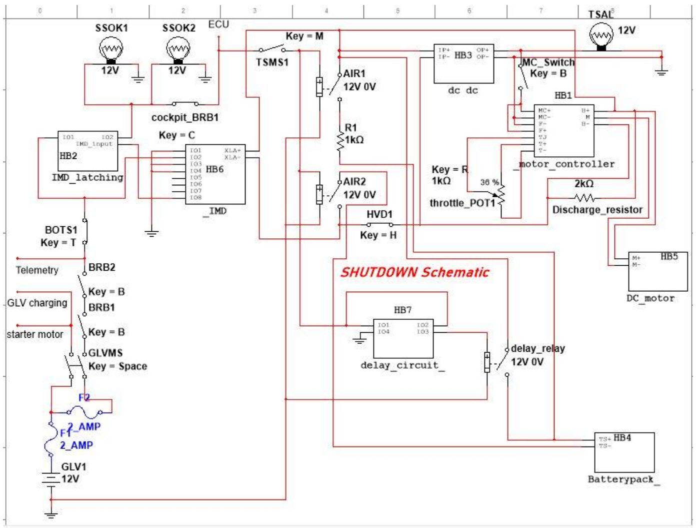
Manufacturing:

We have used **AISI 4130 chromoly steel tubing**. Tubes are cut by using **laser cutting** at both ends and are welded together using **TIG welding**. First front bulkhead is welded. Further to construct the chassis the design team took a **bottom up approach**. The chassis design starts with the bottom layer tubes and simultaneously welding at the location of the suspension node points to define the front and rear suspension boxes. Further components are welded up to front hoop and following next to main hoop by measuring angle and dimension using instruments by checking appropriate design from SOLIDWORKS design file. To make sure all the pipes were at the desired angle, a **fixture table** was designed with a **base plate of thickness 10mm**, which helped maintaining it at **0 degree**.

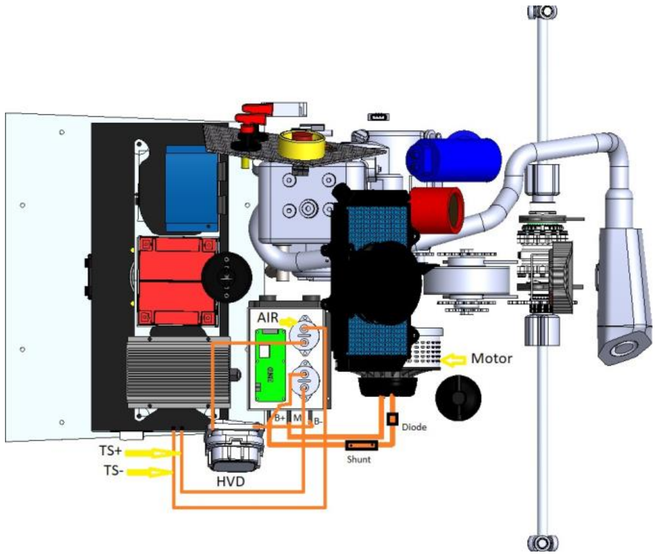
Properties of 4130:

Hardness, Brinell	197
Hardness, Knoop	219
Hardness, Rockwell B	92
Hardness, Rockwell C	13
Hardness, Vickers	207
Tensile Strength, Ultimate	670 MPa
Tensile Strength, Yield	435 MPa
Elongation at Break	25.5 %
Reduction of Area	60 %
Modulus of Elasticity	205 GPa
Bulk Modulus	140 GPa
Poisson's Ratio	0.29
Izod Impact	87 J
Machinability	70 %
Shear Modulus	80 GPa

SHUTDOWN CIRCUIT:



TRACTIVE SYSTEM WIRING:



FULL ASSEMBLY CAD: