Design and Manufacturing of an 'All Terrain Vehicle'

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Abstract – *GHRCE Motorsports team aims to build an ATV with maximum torque at wheels, maximum attainable velocity and gradeability but within the specified limitation stated by the SAE INDIA. The design part not only includes the modelling but the calculation at first hand which itself includes design of mechanical drives and elements. Ultimately achieving the practical parameters in synchronization with attained theoretical values is most important. The factor of safety should always be achieved within certain range. The vehicle handling, driver compatibility and maneuverability are major factors to be considered for a successful championship.*

Key Words: **Design, Analysis, Calculation, Material, Manufacturing**

1. INTRODUCTION

The development of ATV involves design process with simultaneous rectifications and analysis. It is after the validation of each and every component design and feasibility check that it is allowed for manufacturing. The vehicle consists of various dependent and independent parts which need to be separately designed, analyzed and simulated along with each having its own dynamics and static calculations. Conceptualization and visualization are major contributor to initiation of development of a vehicle. Brief about this will be discussed in the paper with detailed descriptions and prototype presentation.

2. FRAME

Roll cage is basically a chassis wiz. structural foundation of an ATV which houses all the components of the vehicle. It is also responsible to keep the driver safe inside from any outer forces. Hence, we have to carefully develop the roll cage according to the proper procedure. Firstly, selection of material, then errorless designing in cad software and linear analysis and determining the force tolerance capability of roll cage. At the end check the ergonomics of the roll cage according to completion guidelines then, manufacture it. Manufacturing is also a crucial process, lots of failures can occur due to improper manufacturing techniques and negligence.

2.1 Selection of Material

While designing a vehicle material selection is the key process in terms of safety, reliability, performance, strength, costing, and availability. On further research on several tube materials and compared them in multiple categories as follows.

Table -1: Material Properties and specifications

Material	AISI 1018	AISI 4130	Duplex 2205 steel	Duplex 2205 steel
Outside diameter	2.540 cm	2.540 cm	2.540 cm	2.540 cm
Wall thickness	0.2 cm	0.2 cm	0.2 cm	0.1 cm
Bending stiffness	2791 Nm^2	2791 Nm^2	2171 Nm^2	
Bending strength	390 Nm	382 Nm	454 Nm	260.4 Nm
Weight/meter	1.6615 kg/m	1.2444 kg/m	1.1475 kg/m	0.8790 kg/m

We have selected 25.mm x 1mm dimension AISI 4130 pipe for our ATV, as per the requirements of our vehicle design, AISI 4130 is the most suitable material because of its lightweight nature, bending stiffness, and strength. We opted for steel which ensured better weight saving. Also, the availability of AISI 4130 is good with efficient costing.

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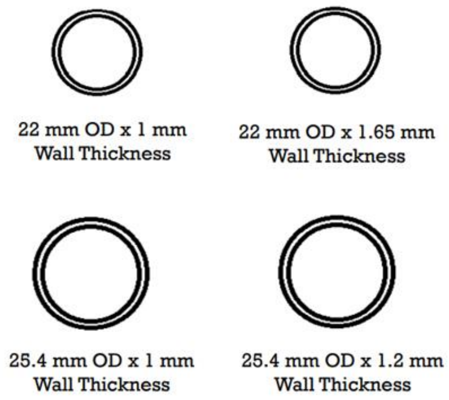

*All the selections have been done according to design parameters and calculations

SPECIFICATIONS OF ROLL CAGE –

Table 2- Roll cage material specifications

Material:	AISI 4130
The thickness of the tube:	1.65 mm
Diameter of tube:	25.4 mm
Weight (approximate for roll cage):	30 kg
Total dimensions of roll cage:	1885 mm ×809 mm × 1135 mm
Weld joints:	51 joints
Weld length:	8000 mm
FOS:	2.07 (average)

Welding: For the fabrication of the final roll cage, we are going to use MIG welding and TIG welding because of effective and efficient welding. The reason behind the usage of MIG and TIG welding techniques are good weld quality, works with many metals and alloys, ease of use, long pass welding, etc. Also, the weldability of AISI 4130 is good enough.

2.2 FINITE ELEMENT ANALYSIS

The following analysis were performed to check the feasibility of design by Front impact test, rear impact test, side-impact test, rear-wheel bump, heave, and twisting. The results are shown below in the table and figures paragraph.

Frontal impact analysis:

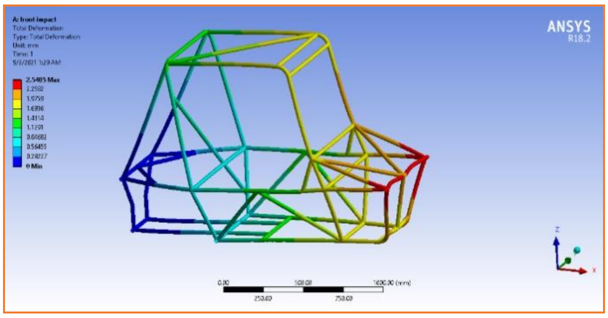

Fig.2.1: Deform

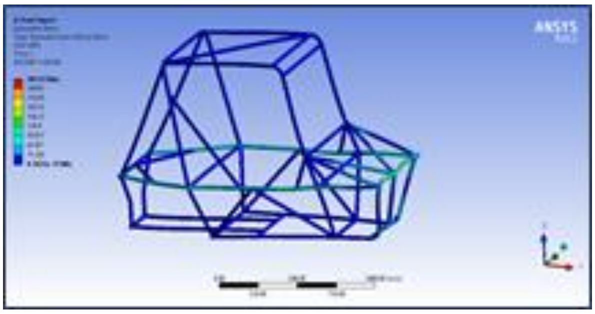

Fig.2.2: Stress

TEST:	FRONT-IMPACT TEST
Force applied:	20,000 N
Stress:	281 MPa
Deformation:	2.5 mm
FOS:	1.63

Table 2.1: Frontal impact analysis

Side impact analysis:

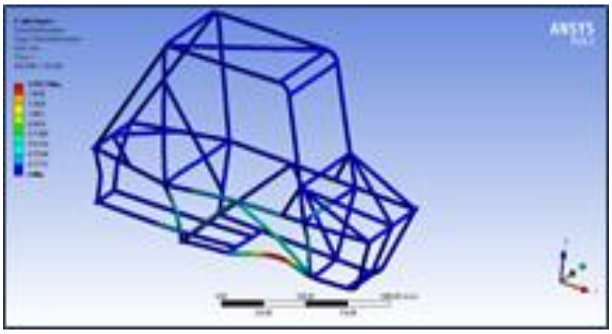

Fig.2.3: Deform

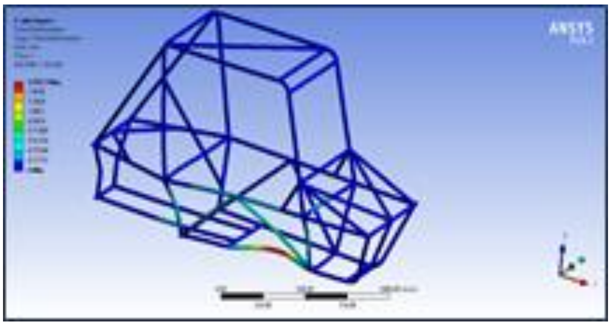

Fig.2.4: Stress

TEST:	REAR IMPACT TEST
Force:	20,000 N
Stress:	282 MPa
Deformation:	2.3 mm
FOS:	1.63

Table 4.2: Side impact analysis

Rear impact analysis:

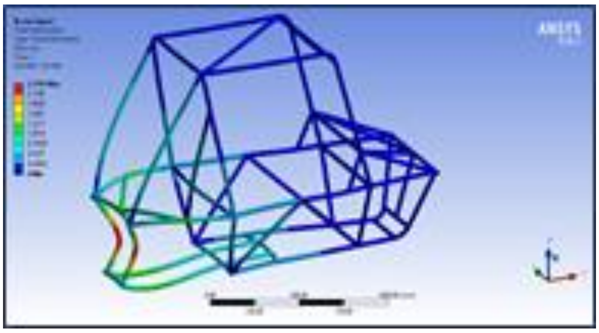

Fig.2.5: Deform

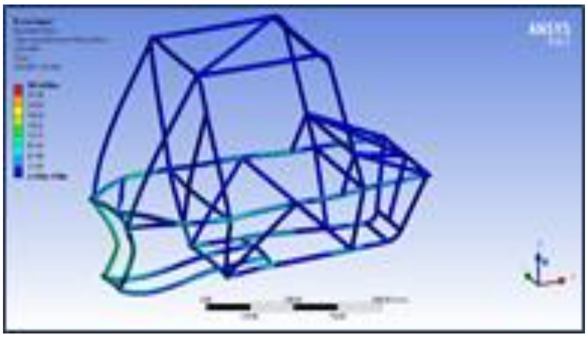

Fig.2.6: Stress

TEST:	SIDE IMPACT TEST
Force:	15,000 N
Stress:	243 MPa
Deformation:	1.59 mm
FOS:	1.89

Table 2.3: Rear impact analysis

3. STEERING

3.1 Objective:

The objective of steering system is to provide directional control of the vehicle, to withstand high stress in off terrain conditions, to reduce steering effort and to provide good response from road to driver.

3.2 Design:

Ackermann steering mechanism:

We are using Ackermann steering mechanism to increase the safety of vehicle while turning. The Ackermann steering mechanism is a geometrical arrangement of linkages in the steering of a vehicle designed to turn the inner and outer wheels at the appropriate angles. The tighter the desired vehicle turn radius, the larger the difference in steer angles required. Steering ratio achieved is 9:1 means for every 9-degree rotation of steering wheel rotates by 1-degree with rack and pinion mechanism. Ackermann percent value is more than 100 means the condition of over steer is achieved. The caster and camber angles in vehicle are 2 degrees and 5 degrees respectively. All the setup helps vehicle to maintain proper grip with road while turning and while climbing which increase the safety of vehicle. Maximum values for outer and inner angle for Ackermann steering is 45 degrees and 28.18 degrees respectively. Turning radius for inner wheel is 1422.4 mm and for outer wheel it is 2743.3 mm. below figures indicates steering wheel assembly and Ackermann geometry.

Specification:

Table 3- Steering system Specifications

PARAMETER	VALUES
Steering system:	Ackermann
Steering mechanism:	Rack and pinion
Ackermann percentage %	110%
Steering ratio	9:1
Caster	2 deg.
Camber	5 deg.
Kingpin inclination	8 deg.
Steering angle	36.18 deg.
Ackermann angle (deg.)	45°(inner), 28.18°(outer)
Turning radius	1422.4 mm (inner) 2743.3 mm (outer)

Fig.3.1 Steering wheel assembly

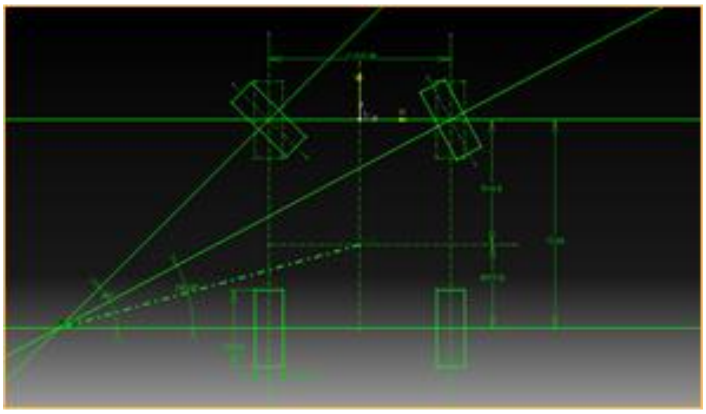

Fig.3.2 Ackerman geometry

4. SUSPENSION

4.1 Objective:

A BAJA suspension must be seamlessly engineered which will provide the ability to compete in every event with practical features like ground clearance and suspension travel which results in good comfort and control to the drive allowing proper navigation in a rough terrain.

4.2 Design:

The designing process is done where the parameters like camber gain, motion ratio were analysed which are required for designing ATV suspension. The mounting points of the front and rear suspension were designed in SolidWorks. Then using these mounting points, the analysis was done in Lotus to verify the assumed parameters. Analysis of output is done in terms of graph between the parameters like wheel travel Vs. camber change etc.

4.2.1. Front Suspension:

The front suspension is a short & long A-arm wishbone arrangement. The roll centre is kept at the optimised height to reduce the body roll. The upright is manufactured by CNC and is symmetric, has good strength to absorb loads. The upright also provides a location to mount the brake calliper. In order to compensate for dive-effects during aggressive cornering, the camber angle for the front suspension has been set at 0° at ride height. In addition to that, the camber angle has been set to decrease when the shock absorber compresses during turns.

4.2.2. Rear Suspension:

4 link H-arm suspension was chosen instead of 5 in order to replace the toe link and better capability to adjust to various parameters. Also, the loads are shared on the 4 mountings which will reduce the stress concentration. As like the front suspension, the rear upright is also a single manufactured piece which provides for the connection of the A-arms and callipers.

4.2.3. Shock Absorbers:

The rear shock absorbers were mounted on upper arm. The rear shock absorbers are stiffer than the front absorbers. Because of uneven distribution of weights, the stiffness of the rear absorbers is kept high. The rear shock absorbers were mounted on upper arm. The rear shock absorbers are stiffer than the front absorbers.

Double wishbone:

In automobiles, a double wishbone suspension is an independent suspension design using two wishbone-shaped arms to locate the wheel. Each wishbone or arm has two mounting points to the chassis and one joint at the knuckle. The shock absorber and coil spring mount to the wishbones to control vertical movement.

Shock absorber:

A shock absorber or damper is a mechanical or hydraulic device designed to absorb and damp shock impulses. It does this by converting the kinetic energy of the shock into another form of energy which is then dissipated. Most shock absorbers are a form of dashpot.

Table 4.1- Specifications of suspension system

SPECIFICATIONS	FRONT	REAR
Type	Double wish-bone damper to lower	Double wish-bone damper to upper
spring length	200 mm	378.2 mm
spring + damper length	393.7 mm	546.248 mm
wire diameter	10 mm	15 mm
mean coil dia	80 mm	120 mm
travel of spring	4 inch	8 inch
spring stiffness	105.65 n/mm	220.212 n/mm

pitch	25 mm	37.8 mm
total no. of turns	9 mm	11 mm

SPECIFICATIONS	FRONT	REAR
Static scrub radius	50mm	0 mm
Percent anti dive/ anti squat	29% (antidive)	57% (antisquat)
Ride rate	312.9 N/m	638.9N/m
Roll center height from ground	305 mm	303 mm

4.2.4. Spring:

Suspension springs are the link between wheels and car body. Their primary task is to compensate uneven road surfaces and thus provide an assurance of high levels of ride comfort. Secondly, they must ensure that the wheels always have safe contact with the road regardless of its condition.

Suspension Parameters	Front	Rear
Wheel Rate	321.65 N/m	676.48 N/m
Spring Rate	105.65 N/mm	105 N/mm
Motion Ratio	0.41	0.41
Sprung Mass Natural frequency	1.2 Hz	1.2 Hz
Ride Rate	312.9 N/m	638.9 N/m
Spring Stiffness	105.65 N/mm	105.65 N/mm
Mean Coil Diameter	80 mm	80 mm
Outer Diameter	90 mm	90 mm
Inner Diameter	70 mm	70mm

Centre of gravity:

Wheel Diameter	22 inches
Weight of Front Wheel with Rear elevated	130 Kg
RLF=Axle height above ground for Front	11 inches=279.4 mm
RLR=Axle height above ground for Rear	11 inches=279.4 mm
H1=height of CG above	215.08 mm

wheel centre	
CG height above ground	19.5 inch

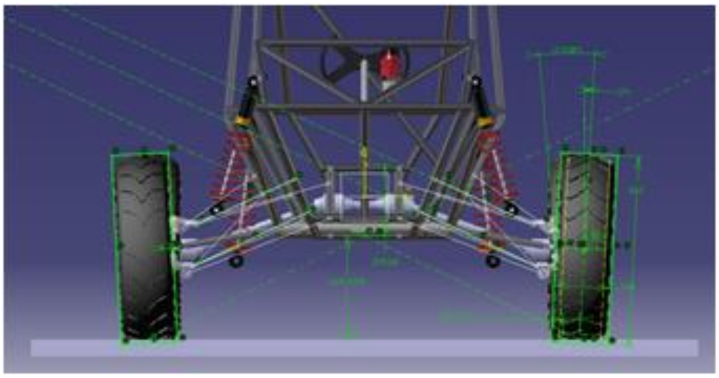

Fig.3.1. Front suspension geometry

TYRES:

Tyres are designed to support the weight of the vehicle, absorb road shocks, transmit traction, torque and braking forces to the road surface and maintain and change the direction of travel. To fulfil these four basic functions tires are made of resilient rubber and filled with compressed air.

Tyres specification:

Suspension Parameter	Front	Rear
Suspension Type	Double Wishbone	Double Wishbone
Tyre Size	22 inches	22 inches

5.0 BRAKING

5.1 Objective:

The purpose of the breaking system is to increase the safety and mobility of the vehicle by statically and dynamically all four tires on both paved and unpaved surface.

Specification:

Table 5- Braking system specifications

Specification	Front & Rear	
Type	Disc & hydraulic	
Brake of torque	359.93 n-m	463.78 n-m
Rotor size	220 mm	
M. C diameter	19.05 mm	

Calliper pad diameter	36mm
Disc thickness	4 mm
Pedal ratio	6:1
Brake fluid	DOT-4

Table 5.1- Thermal properties for brake rotor analysis.

Heat flux	Magnitude	23000w/m²
Convection	Film coefficient	230 w/m²
	Ambient temperature	22°c
Radiation	Emissivity	1
	Ambient temperature	22°c

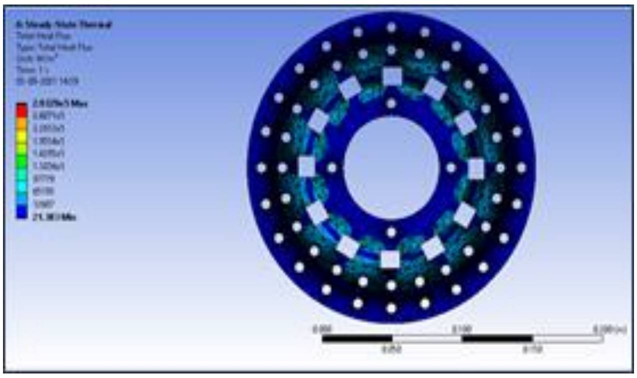

Fig.4.1. Disc analysis

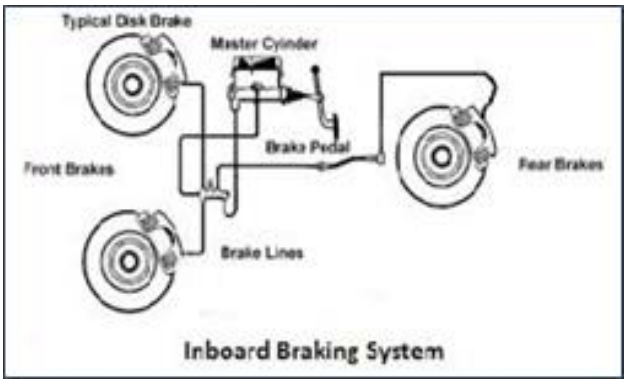

Fig.4.2. Inboard Braking system

6.0 TRANSMISSION

6.1. Objective:

The main objective of the drive train is to vary the torque in the most efficient way possible. This is being done through proper gear reduction for the needs of the vehicle in the competition. We picked a manual transaxle over CVT because of the following reasons:

• Wider gear ratios range

• Gives better acceleration

• Lighter and economical

• Slippage losses are less in manual gear box

• Heat generated in manual transmission is less due to the time gap between the shifts.

6.2. Design:

The gearbox has been designed considering all the events of the Baja competition, and so as to reduce the weight of the gearbox, ensure durability and outputs. And so to obtain the accurate gear ratio, and to meet all the requirements, reverse engineering method is adopted to design the 2-stage gearbox.

6.3 Transmission calculation:

Engine (as per the rule book of BAJA 2021):

- Engine = 19L232-0054-G1 (Briggs and Stratton)
- Power = 10 HP = 7.46 kW
- Engine Torque (Te) = 19.67 Nm
- Speed in rpm (N) = 3800 rpm
- CVT = CVT Shifter

Ratio = 0.9 to 3.9 [150 turning possibilities]

- Wheel Diameter: 0.584 m

Radius of wheel = 0.292 m

- Weight of Vehicle:270
- Rolling Resistance Coefficient (μ): 0.08

[It is maximum condition of all roads]

- Efficiency of Transmission: 75%

[Assumed by considering all losses]

Calculations of gear ratio:

- Maximum Speed of vehicle = $\frac{N*C*E*3.6}{C.R*G.R*60}$

Where,

N: Speed of Engine (in rpm)

C: Circumference of Vehicle (m)

E: Efficiency

C.R.: CVT Ratio

G.R.: Gear Ratio

$57 = \frac{3800* 3.14*0.584*0.75*3.6}{0.9*G.R.*60}$

=> G.R. = 9.168(high)

Torque on Wheel = Te*C.R.*G.R.*E

(Considered) 500 = 19.67*3.89*G.R.*0.75

⇨ G.R. = 8.69(low)

So, we take 9.168 ratio for high torque.

	STAGE 1		STAGE 2	
No. of teeth	PINION	GEAR1	PINION	GEAR2
	16	48	16	54

Stage1 G.R. => 48/16 = 3

Stage2 G.R. => 54/16 = 3.375

Stage 1 G.R	3
Stage 2 G.R	3.375
Gear Ratio	9.168

By taking lower ratio of CVT 0.9 and Gear ratio of 9.168

We get => 0.9*9.168

=8.25

N= 3800/8.25

= 460.6

ώ= 2ΠN/60

= 52.083

V = r*ω

= 0.292*52.083

= 15.266 m/s

V = 54.75

- Maximum speed of Vehicle = 54.75
- Tractive Effort (T.E.) = $\frac{T_e*G.R*C.R*\eta}{0.292}$

$=\frac{19.67*3.9*9.168*0.75}{0.292}$

T.E. = 1806.43 N

R.R. = w*μ

= 270*9.81*0.08

R.R = 211.896 N

Acceleration = F_{net} /m= T.E- R.R/m

= 1594.54/270

= 5.9 m/s²

Gradeability = 100*(T.E.-R.R. Coefficient)/W

= 100*(1806.43-0.08)/270*9.81

= 60.20%

Torque on wheel = 19.67*3.9*9.168*0.75

= 527.47 Nm

Maximum Speed of Vehicle = 54.75 km/hr

Tractive Effort [T.E.] = 1809.43 N

Rolling Resistance = 211.896 N

Acceleration = 5.9 m/s²

Gradeability = 60.20%

Torque on wheel = 527.47 Nm

Gear Ratio = 9.168 [this is suitable for high torque]

Table 6.1 – Specification of Transmission system

Specifications	Front & rear
Maximum speed of vehicles	54.75 km/hr
Tractive effort	1806.43 N

Rolling resistance	211.896 N
Acceleration	5.9 m/s²
Gradeability (in %)	60.20%
Grade angle	31.047º
Torque on wheel	527.47 Nm.

Fig.5.1. Baja ATV engine

7.0 ELECTRIC SYSTEM

The electrical components in our ATV are installed for our safety majors. We are installing two kill switches with easy accessibility, to isolate the current from engine easily. By using SAEBAJA standards we are installing brake lights, reverse light and reverse alarm which will activate in emergency situation. To co-ordinates the drive we are mounting GPS system in cockpit area displays the directions and speed of vehicles.

We are using X2Mx type transponder in STV to for laps completing b to a timing device on the track. All the components we are mounted are safely securely by 9v batteries. Wiring is done by abiding rulebook of SAE BAJA.

CONCLUSIONS

The design and analysis of every component is very crucial for development of a safe and reliable vehicle. All the important aspects contribute to the overall performance and build quality of the vehicle.

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BIOGRAPHIES

"Rohan Gajbhiye is a highly knowledgeable and skilled student who has decent experience in motorsports. He is immensely passionate about automobiles. "

"Taru Sekhar Das is an Alumni of the respective university. Lead the design team in all motorsport's events. Highly skilled in mechanical and automobile design, currently working as a CAD engineer in a reputed MNC. "