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Research Article

DESIGN REPORT QUAD BIKE DESIGN CHALLENGE – 2016

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ABSTRACT

The Quad Bike Design Challenge initiated by Fraternity of Mechanical and Automotive Engineers, provides a platform for graduate and diploma students to fabricate their own QUAD BIKES and compete against students from various colleges around the country. The report explains objectives, assumptions and calculations made in designing and fabricating a QUAD BIKE for QBDC 2016. Quad Bike is an All Terrain Vehicle (4-wheeler bike), which was initially developed as a farm to town vehicle in Isolated and mountainous areas. The team's primary objective is to design a safe and functional vehicle based on rigid and torsion free frame considering the technical guidelines provided in the rulebook along with that working on each parameter that would enhances the performance and efficiency of our vehicle. Our project vision is to develop technically sound and conceptually engineered quad bike which includes design, analysis and methodology followed and considerations made in the entire design process.

INTRODUCTION

Technical Specifications of the Vehicle

Chassis	Seamless Tube	AISI 4130
Wheelbase	1130.3 mm	
Overall Length Of Vehicle	1676.4 mm	
Track Width	1016 mm	Front
	960.12 mm	Rear
Type Of Engine	23bhp 200cc Engine	
Steering	Mechanical Linkage	
Wheels And Tyres	21*7*10	front
	22*10*10	rear
Brakes	Hydraulic disc brake	Rear& front
Transmission	Manual	
Mass Of The Vehicle	270kg	Approx
Ground Clearance	7 Inch	

Performance Targets

Designing: To ensure good sustainability of the frame designed with even stress distribution.

Suspension System: To keep the wheels firmly pressed to the ground for better traction and to support the weight of the vehicle

Steering: To achieve minimum turning radius with ease of handling to driver.

Engine and Transmission: To achieve maximum acceleration in minimum possible time.

Braking System: To accomplish the task of getting less stopping distance with less effort of the driver and to provide safety to the vehicle in all the prevailing circumstances.

LITERATURE REVIEW

Being a new team required a clear idea of basic requirements, parameters in designing of quad bike. We made a detailed study on quad bike by visiting Polaris-Hyd, BRC- Bhopal. We gained a sound knowledge during our field study and basic doubts on design were cleared. We approached our design by considering all possible alternatives for a system and modeling them in CAD software, Solid works and subjected to analysis using Solid works simulation and ANSYS. Based on analysis result, the model was modified and a final design was fixed. The design process of the vehicle is based on various engineering aspects depending upon

- Safety and Ergonomics
- Market Availability
- Cost of the Components
- Safe Engineering Practices

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About Team: Team Super Ignite has made significant progress towards the completion of the bike. The team consists of 30 dedicated students under the name of team super ignite and have been working since past few months to de3ign and fabricate a quad bike for QBDC. The concept of team work has been developed based on the sub components present in the vehicle fabrication; the team has been branched into 6 Departments. Each branch/department holds the responsibility of applying engineering techniques and performs design calculations, and selects the required materials. With this we had view of our quad bike and set up some parameters of our work and team has been divided into the following core groups.

- Design
- Engine and Transmission
- Steering
- Suspension
- Brakes and wheels
- Business and Management

Project Objective

Participating in championships like QBDC helps a student to understand the variation in theoretical and practical knowledge. For students to enter into the racing field and understand its standards QBDC is a best start. Provides good work experience with national level recognition and a huge platform to display our talent.

Design of Quad

The following design methodology was used during design:

- Requirements
- Design calculations and Analysis
- Considerations
- Testing
- Acceptance

Material Selection The material AISI-4130 (Annealed at 840° C) is used in the frame design because of its good weldability, relatively soft and strength as well as good manufacturability.

AISI 4130 has been chosen for the chassis because it has structural properties that provide a low weight to strength ratio. Presence of chromium makes it a corrosion resistive material. 1 inch diameter tube with a thicker wall is used. It is assured by analysis in ANSYS simulation software. The various Physical properties of the material are as follows

Properties	AISI 1018	AISI 1020	AISI 4130
Yield strength	370 MPa	294.74Mpa	460 MPa
Ultimate strength	440 MPa	394.72 MPa	560 MPa
Carbon %	0.14-0.20%	0.14-0.24%	0.28-0.33 %
Young's Modulus	200 GPa	200 GPa	205 GPa
Density	7.87 g/cc	7.87 g/cc	7.85 g/cc
Mach inability	70 %	65 %	70%

Properties of AISI 4130 seamless tube

Feature	Specification
Name	AISI-4130
Hardness	95 BHN

Outside Diameter	25.4mm
Wall Thickness (Front and Rear Suspension Control Arms and Engine Compartment and different accessories mounting)	2.4 mm
Wall Thickness (Main frame and its structure)	1.65 mm
Tensile Strength (Ultimate)	560MPa
Tensile Strength (Yield)	460 MPa
Shear Modulus	80 GPa
Modulus of Elasticity	190 -210 Gpa
Poisson's Ratio	0.27-0.30
Mass Density	7.85 g/cm3
Dimensional Specifications	
Round tube of dimension	= 25.4mm OD
Thickness	= 1.65 mm, 2.4 mm

Justification

Round hollow tubes are light in weight

Fabrication Processes

Lathe Work, Cutting, Drilling, Milling, Shaping, Grinding, Polishing, Finishing, Welding

Analysis

Front Impact Analysis

Using the gross weight of the vehicle is 270 kg, the impact force is calculated taking the time of impact between 0.1 to 1.2 sec. With different impact times forces are calculated.

Impact time(sec)	Force(N)
0.1	44982
0.3	14994
0.5	8996.4
0.8	5622.75
1	4498.2
1.2	3748.5

Among them maximum, minimum and intermediate forces 44982N, 8996.2N, 3748.5N forces are picked up and simulation is performed.

Forces are applied on chassis taking three considerations,

Case 1) Vehicle hit at the center, Case 2) Vehicle hit at leftwards, Case 3)Vehicle hit at rightwards

For F=44982N

Case 1)

Model of frame designed using Solid Works

Case 1)

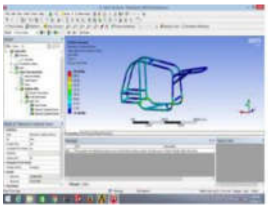


Fig: Stress

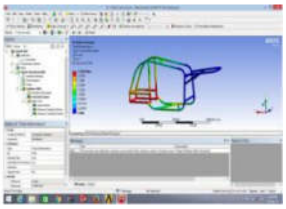


Fig: Deformation

Case 2)

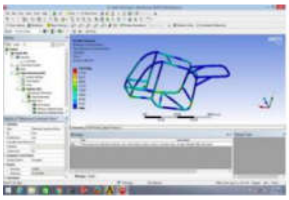


Fig Stress

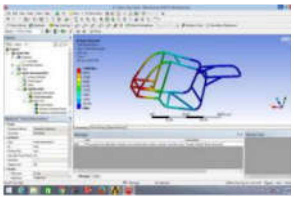


Fig Deformation

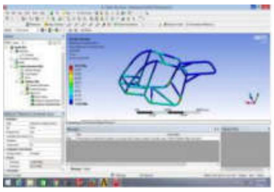


Fig Stress

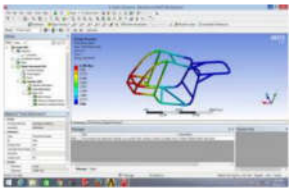


Fig Deformation

	For load applied F=44982N		
	Case 1	Case 2	Case 3
Stress	336.59	312.05	307.91
Deformation	2.309	7.44	6.598
F.O.S	1.366	1.474	1.49

Above table describes about stress and deformation

For F=8996N

Case 1) and case 2) case 3) are as follows.

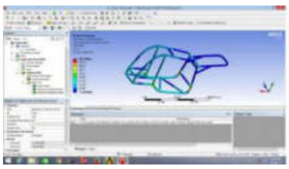
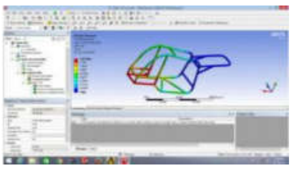


Fig Stress



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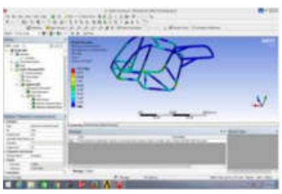


Fig Stress

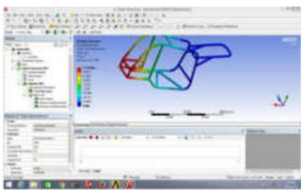


Fig Deformation

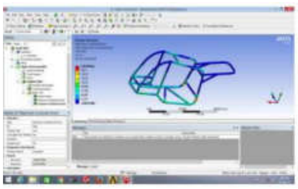


Fig Stress

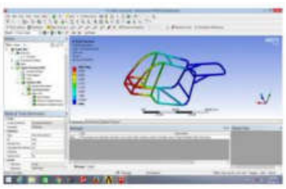


Fig Deformation

	For load applied F=44982N		
	Case 1		
Stress	218.7	115.57	226.08
Deformation	1.50	2.759	4.6875
F.O.S	2.10	3.9	2.03

Above table describes about stress and deformation

For F=3748.5N

Case 1)

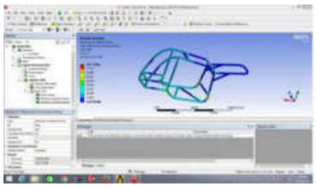


Fig Stress

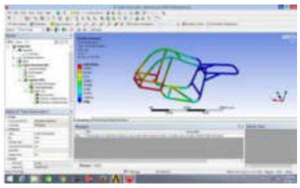


Fig Deformation

Case 2)

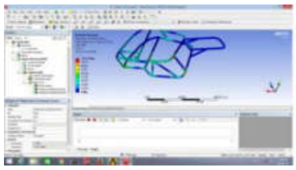


Fig Stress

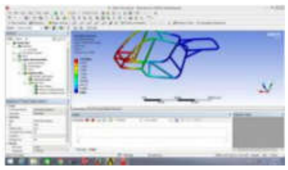


Fig Deformation

Case 3)

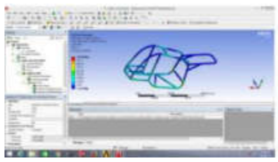


Fig Stress

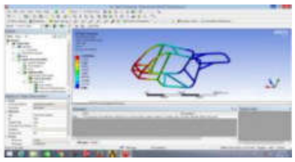


Fig Deformation

	For load applied F=44982N		
	Case 1	Case 2	Case 3
Stress	126.15	115.57	114.04
Deformation	**0.865**	**2.759**	**2.4438**
F.O.S	3.646	3.98	4.033

Side Impact Analysis

For side impact forces are calculated by consideration of relative velocity of the vehicles being impacted. As maximum speeds of vehicles are 60kmph. The relative velocities are 30Kmph, 25Kmph & 20Kmph among them maximum, minimum and intermediate

applied			
Stress	358.87	179.22	29.88
Deformation	4.281	2.138	0.356
F.O.S	1.28	2.566	15.39

Torsional Test

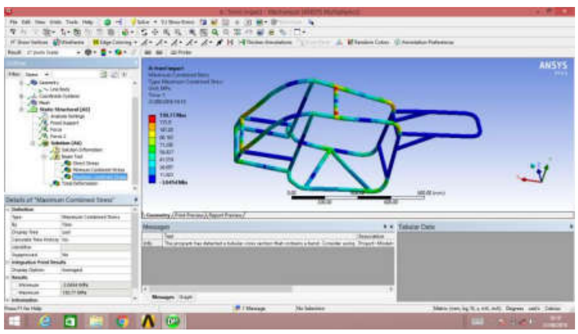


Fig Torsional stress analysis

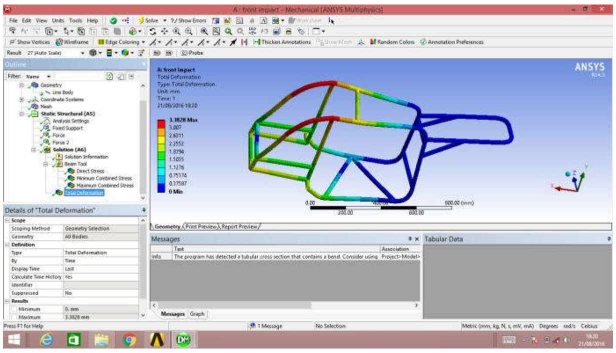
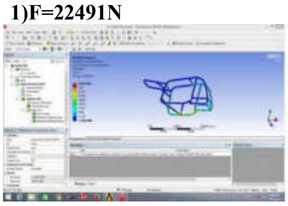


Fig Torsional test Total Displacement

forces 22491N,7491N&1248N forces are picked up and simulation is performed.

1)F=22491N



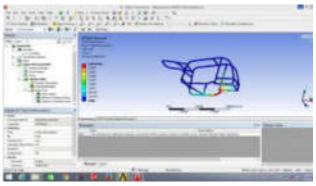
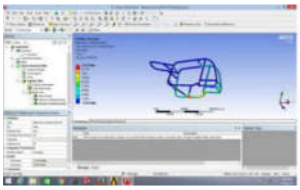


Fig: Stress **Fig: Deformation**

2)F=7491N



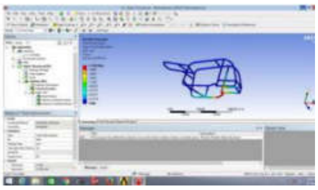
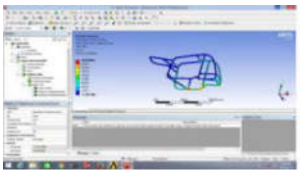


Fig: Stress **Fig: Deformation**

3)F=1248N



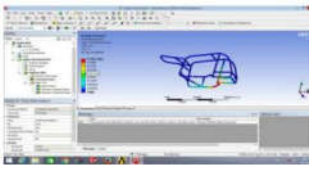


Fig: Stress **Fig: Deformation**

For load applied F=44982N			
	1	2	3
Load	22491N	7491N	1248N

Load applied: 2200
Maximum stress: 130.77 MPa
Displacement: 3.38 mm
Factor of safety: 3.51

Suspension System

Suspension system is referred to the springs, shock absorbers and linkages that connect the vehicle to the wheels and allows relative motion between the wheels and the vehicle body. Also, the most important role played by the suspension system is to keep the wheels in contact with the road all the time. Good suspension system and better handling is the characteristic of a good All-Terrain Vehicle (ATV).

Its main objective is

The overall purpose of a suspension system is to absorb impacts from coarse irregularities such as bumps and distribute that force with least amount of discomfort to the driver.

- It supports the weight of the vehicle.
- Keeps the wheels firmly pressed to the ground for better traction.
- The design of a suspension which will significantly show its effect on comfort, safety and maneuverability.

Of all the independent suspension systems, Double Wishbone Suspension System is the most common type of suspension system used in the passenger cars and most of the All-Terrain Vehicles.

Double Wishbone Suspension System consists of two lateral control arms (upper arm and lower arm) usually of unequal length along with a coil over spring and shock absorber. It is popular as front suspension mostly used in rear wheel drive vehicles. This type of suspension system provides increasing negative camber gain all the way to full jounce travel unlike Macpherson Strut. They also enable easy adjustment of wheel parameter such as camber.

Design of Wishbones

Design of wishbones is the preliminary step to design the suspension system.. The Roll Centre determined in order to find the tie-rod length. The designed wishbones are modeled using software and then analyzed using ANSYS analysis software to find the maximum stress.

Material Selection of Wishbone Material and swing arm

The material selection also depends on number of factors such as carbon content, material properties, availability and the most important parameter is the cost. Initially, three materials are considered based on their availability in the market are AISI 1018, AISI 1040, AISI 4130.

Wishbone Material Selection

Properties	AISI 1040	AISI 1018	AISI 4130
Carbon content (%)	0.40	0.18	0.30
Tensile strength (MPa)	620	440	560
Yield strength(MPa)	415	370	460
Hardness (BHN)	201	126	217
Cost (Rp /Meter)	425	325	570

As for above properties we selected AISI 4130 based on hardness, tensile and yield strength

Determination of Roll Centre

Roll Centre in the vehicle is the point about which the vehicle rolls while cornering. The location of the geometric Roll Centre is solely dictated by the suspension geometry, and can be found using principles of the instant Centre of rotation. Determination of Roll Centre plays a very important role in deciding the wishbone lengths, tie rod length and the geometry of wishbones. Roll Centre and ICR is determined because it is expected that all the three elements- upper wishbone, lower wishbone and tie rod should follow the same arc of rotation during suspension travel. This also means that all the three elements should be displaced about the same Centre point called the ICR.

Analysis of Wishbones

Analysis of wishbone in ANSYS Analysis Software is necessary in order to determine the induced maximum stress and maximum deflection in wishbones. For analysis, wishbones are first needed to be modeled in software. **Solid Works Cad Models are shown below.**

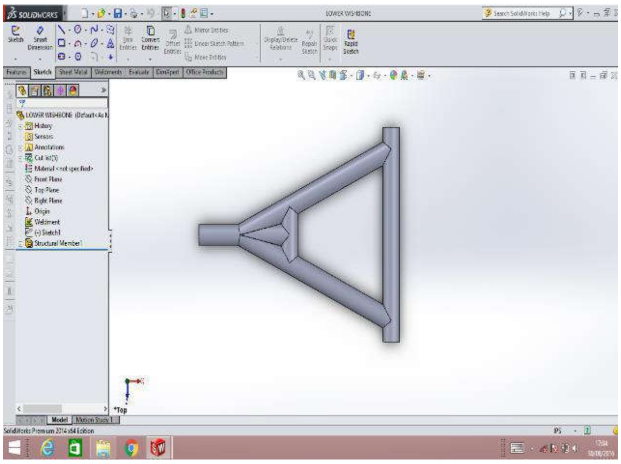


Fig: Lower Wishbone

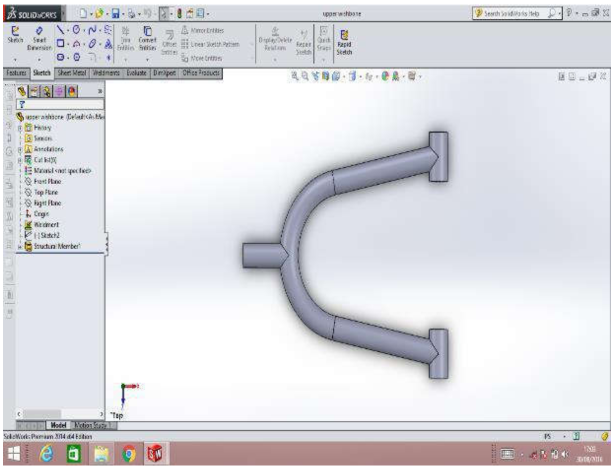


Fig: Upper Wishbone

Analysis in ANSYS is engineering simulation software (computer-aided engineering). Various types of analysis like structural analysis, thermal analysis, etc are possible using ANSYS analysis software. In structural analysis in ANSYSs, boundary conditions are to be defined in order to determine the stress and deflection.

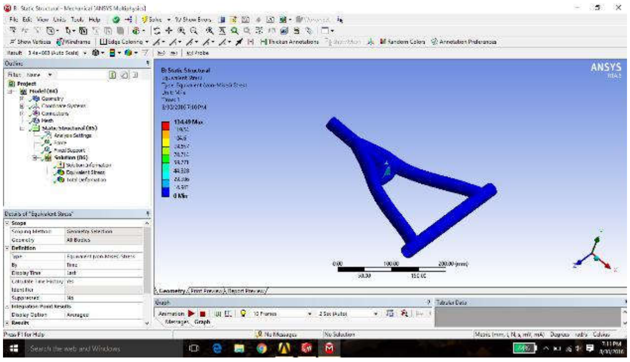
Input parameters are as follows,

Material	AISI 4130
Vertical load	7100N
Spring force	1500 N

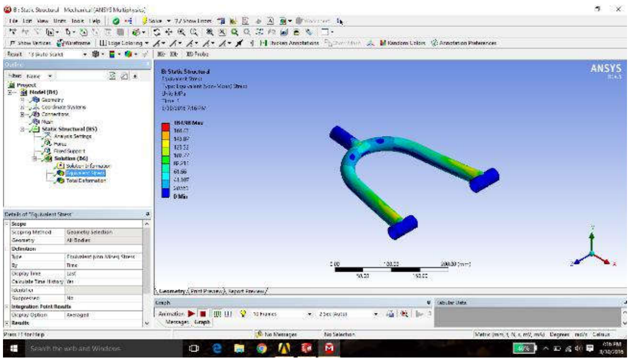
Analysis of wishbones in ANSYS:

Swing Arm

It is a control arm which is used at rear suspension system; it is an arm that there is mono shock is mounted on it and to the frame.



Analysis of Lower Arm in ANSYS



Analysis of Upper Arm in ANSYS

Design of Spring

A spring is an elastic object used to store mechanical energy. Springs are usually made out of spring steel. When a spring is compressed or stretched, the force it exerts is proportional to its change in length. The rate or spring constant of a spring is the change in the force it exerts, divided by the change in deflection of the spring.

Design Considerations in spring: Design Sprung mass = 140kg

Un sprung mass = 60 kg

Wheel displacement= 228mm (9")

Specifications of spring

Specifications	Front shock absorber	Rear shock absorber
Motion ratio	0.65	0.7
Coil diameter	12.7	15.78
Mean diameter	88.9	71
Solid length	190.5	157.8
Free length	330.2	279.4

Simulation of Suspension System

Lotus Engineering Software has been developed by automotive engineers, using them on many power train and vehicle projects at Lotus over the past 15 years. It offers simulation tools which enable the user to generate models very quickly, using a mixture of embedded design criteria and well-structured interface functionality. 8.1 Suspension Geometry in Lotus simulation software has been used to simulate the suspension geometry of double wishbone suspension system. Various co-ordinates of the entire system are given as input and the virtual model is built.

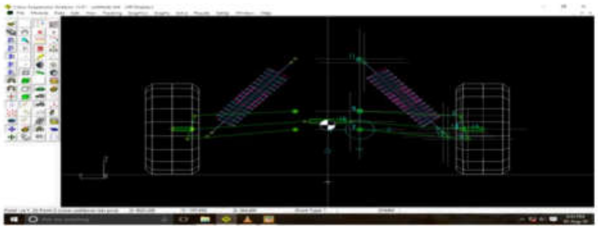


Fig: Simulation of Suspension using Lotus Shark Software

Fig: Camber Change in Bump

Fig: Toe Angle change in bump

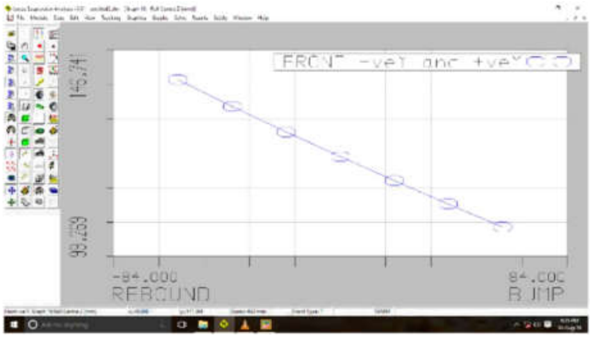


Fig: Roll center change in bump

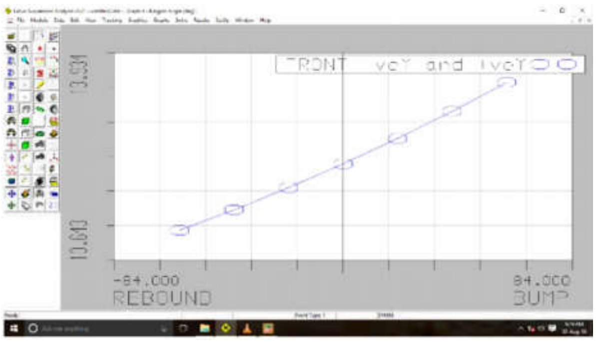


Fig: King pin Angle change in bump

Thus, we have designed the double wishbone suspension system and simulated it in the LOTUS software. This was followed by analysis of the system in ANSYS.

Steering System

The steering system is designed to withstand the stress of safely manoeuvring the vehicle in all type of terrains with appreciable safety and minimum effort. The purpose of the steering system is to provide directional control of the vehicle with minimum steering effort.

Typical target for a quad vehicle designer is to try and achieve the least turning radius so that the given feature aids while maneuverings in narrow tracks, also important for such a vehicle for driver's effort is minimum. The next factor to take into consideration deals with the response from the road. The response from the road must be optimum such that the river gets a suitable feel of the road but at the same time the handling is parameters on other system like the suspension system should not be adverse.

Design considerations

1. We need a steering system that would be easy to maintain, provide easy operation, excellent feedback, cost efficient and compatible to driver's ergonomics.

Thus we have selected trapezoidal linkage as steering mechanism for our Quad bike.

2. We have increased our front and rear track width to improve the lateral stability according to off-road conditions.
3. Rear track width is kept slightly less than front track width to create a slight over steer in tight cornering situation which allows easier

Initial Data

Wheelbase(L)	44.5" = 1130.3mm
Front Wheel Track	40" = 1016mm
Rear Track Width	37.8" = 960.12mm
Total Weight	270 kg
Turning Radius	2.7m
Steering mechanism	Trapezoidal Linkage

Components and dimensions: Steering Column

The steering column is a device intended primarily for connecting the steering Handle to the steering mechanism or transferring the driver's input torque from the steering Handle. The 3-D model of Steering Column is assigned using AISI 4130 material.

The design of steering column is given in the following Figure.

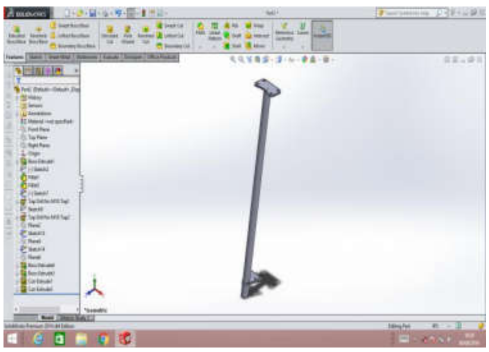


Fig cad model of steering column.

Knuckle

The knuckle was evaluated using the ANSYS 14.0 finite element program. The total weight of the system was estimated to be around 270 kg. In a previous study, it was determined that the maximum acceleration an off-road vehicle is likely to endure during competition is 3g's. With this knowledge, 8240.4 N forces were distributed on each component in various directions in order to simulate the stresses encountered during landing from a jump, lateral acceleration due to turning, and frontal impact.

The resulting Von-misses stress plot is shown in following figure.

And the total deformation for the applied load of 8240.4 N for the knuckle is given below

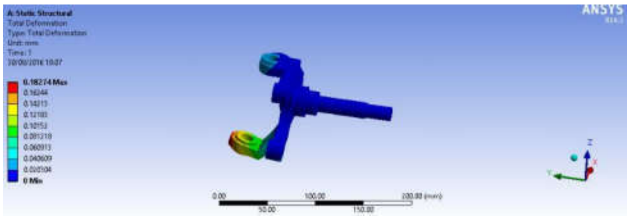


Fig Knuckle Analysis showing total deformation

Hence the design is safe according to the applied loads.

Tie Rods

As that of theoretical study of tie rod is done. The overall purpose if tie – rod is to transmit the motion from steering arm to steering knuckle and sustain the forced vibrations caused by bumps from tires due to uneven road surfaces the main task is to find the deformation and stresses induced in the tie rod and optimizing it for various material combinations. The 3-D model is prepared for Tie-Rod of AISI 1020 material is assigned and analysis is carried out using ANSYS 14.5 and Solid works by applying about 2000 N as load,

Fig: Force applied on Tie rod

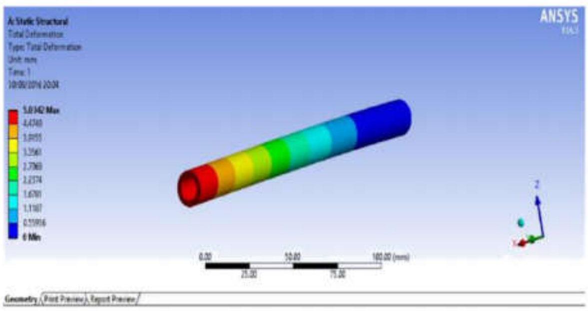


Fig: Stress Analysis on Tie Rod

Handle Bar

Handle Bar is used to turn the vehicle in any direction. The Circular motion of handle bar is converted to angular motion of wheels using tie rods. The design of Handle bar is given below,

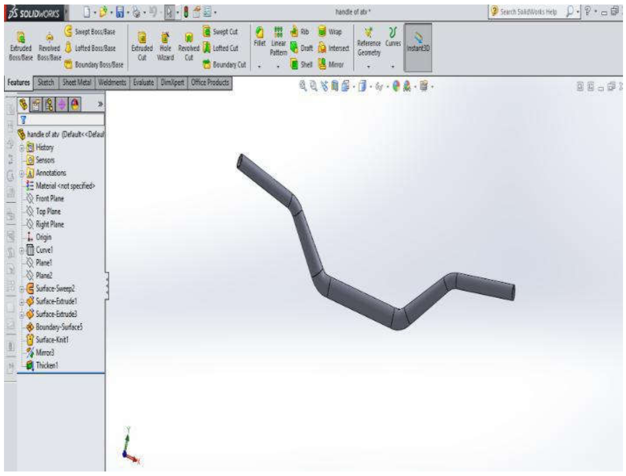


Fig: Handle Bar CAD model in Solid Works

Wheels and Selection

We have chosen 4 ply tires because they have more durability. Deep, thick lugs provide traction in the harshest mud conditions. Off-road trail riding varies from tight single track trails in the mountains, down to high speed wide open desert riding and everything in between Tires.

Dimensions

Front: 21*7*10 inches
Rear: 22*10*10 inches

Fig: Tires

Formulae and Calculations

Turning Radius Formula:
$R = T/2 + W/\sin(\theta_i + \theta_o/2)$

$\Theta_i + \theta_o = 62.08$ Correct steering angle:

$(\cot\theta_0 - \cot\theta_i) = C/W$

$\Theta_i = 36.53^o$ $\theta_o = 25.55^o$
Ackerman 100% inside angle:
$\tan^{-1}(WB/(WB/\tan\theta_o - TW)) - \theta_o \quad = 14.4212^o$

Ackerman percentage: %Ackerman = ((inside angle - outside angle) / (Inside 100% Ackerman)) * 100%

= 76.13% Ackerman angle calculation:
$\tan\alpha = (\sin\phi - \sin\theta) / (\cos\phi + \cos\theta - 2)$
=0.5571
$\alpha = \tan^{-1}(0.5571) = 29.125^o$
Turning radius(Rmax) calculation

R min = length of heel base / tan θi = 1.5258 m
R max 2 = [R min + Wheel track width] 2 +[Length of wheel base 2]
= 2.7 m

Wheel Geometry:

Description	Values
Outside Wheel Turning Angle	25.55o
Inside Wheel Turning Angle	36.53o
Steering Ratio	1.1
% Ackermann Geometry	76.13
Ackermann Angle ˚	29.125
Turning radius	2.7 m

Steering System Components and their Geometry

Components	Dimensions
Tie Rod	10 inch x 0.78 inch ϕ
King-Pin	6 inch x 4 inch x 0.43inch ϕ
Bracket	4 inch x 4 inch x 3inch x 0.23inch
Bolt	10 mm ϕ
Steering column	29 inch x 1 inch ϕ
Steering handle	29.3 inch x0.8 inch ϕ
Wheel Characteristics:	
GEOMETRY	VALUES
Caster Angle	0 degrees
Camber Angle	+2 degrees
King pin Inclination	10 degrees
Combined Angle	12 degrees
Toe-in	3 mm
Scrub Radius	20 mm
Minimum Turning Radius	1.5258 m
Maximum Turning Radius	2.7 m

Engine	Capacity	performance rating	budget rating	Overall
CBR 250 R	26 BHP @8500 RPM 22.92N-m@ 7000 RPM	5	3	8
KTM 200	24.6 BHP @ 10000 RPM 19.2 N-m @ 8000 RPM	5	3	8
PULSAR200 NS	25 BHP @ 9500 RPM 18.1 N-m @ 8000 RPM	4.5	5	9.5
PULSAR 220	20.8 BHP @8500 RPM 19.1 N-m @ 7000 RPM	3.5	5	8.5

Engine and Transmission

Engine is the power generating unit and the power from the engine is to transmitted to the wheels .So, there is a need of transmission system.

Max engine capacity =250 cc with maximum of 30 BHP.

The reasons for selecting the engine is tabulated accordingly in the below table.

Considering above factors Pulsar 200NS is chosen.

Transmission

Manual transmission system has been used with 6 gears that comes pre-assembled with pulsar 200NS.

Gear ratios for chosen transmission system are:

Gear	Gear Ratio
1	10
2	7.15
3	5.5
4	4.6
5	3.88
6	3.33

Teeth on front sprocket (t1) i.e driver sprocket = 14 Max. Mass (m) = 270Kg

Driven sprocket determination

Min. Force

Starting Force FX = m * a

Considering acceleration

Acceleration(a) in m/s2	Starting Force(Fx) in N
3	810
4.2	1130

Resisting Forces

Drag force

$F_D = [\rho_{air}*u^2*CA]/2$ $P_{air} = 1.1$ kg/m^3
u = 60kmph = 16.6m/s $C_o = 0.3$
A = 1m^3
$F_D = 45.8 = 50$N

Rolling Friction

$F_R = C_R N$ $C_R = 0.03$
Weight distribution – front – 45%
Rear – 55%
N= m*g = 270*10 = 2700N
$N_{rear} = 1485$N

Starting Torque (T)	No. of teeth on driven sprocket (t2)
260	21
350	30

$N_{front} = 1215$N

Rolling friction = 0.03*2700 = 81N ≈ 90N
Total resistance (R) = Drag force + Force due to friction = $F_D + F_R = 50 + 90 = 140$N

Wheel Radius = 11″ = 11*25.4 = 279.4 = 280mm

Rolling Radius (r) = 0.96*280 = 269 = 270mm = 0.27 m Total force F = Starting force (F_X)+ Total resistance (R)

Starting Force (Fx) in N	Total Force (F) in N	Starting Torque (T) in N-m
810	950	258.5 ≈ 260
1130	1280	346 ≈ 350

Engine Torque = 18.1N-m Teeth on front sprocket (t_1) = 14
Torque on front sprocket (T_1) = 18.1*10 = 181N-m

Iterations

Considering starting torque to get number of driven sprocket teeth
$t_2 = (T_2/ T_1) * t_1$

But to reduce our manufacturing cost, we are employing with driven sprocket of 39 teeth which we already have and compromising on speed of vehicle for torque.
So t_2= 39

Fig Driven Sprocket

Then torque on driven sprocket
T_2 = (39*181)/14 = 504.2 N-m

Speed

Let consider vehicle at 6^{th} gear then N_1 = (max rpm)/gear ratio = 10000/3.3 = 3030 RPM
RPM of rear shaft $N_2 = (N_1 * t_1) / t_2$ = 3030*14/39 = 1087.6 RPM
Speed = (2∏*N_2*r)/60 = (2*∏*1087.6*0.27)/60 = 110 Km/hr

Acceleration of vehicle

F = [T_{engine} * gear ratio * Transmission efficiency]/wheel radius = 18.1*10*2.8*0.9/0.27 = 1689.3N ≈ 1690N
Actual force = F– resisting force = 1690 – 150 = 1540N a = F/m = 1540/270 = 5.7 m/s^2
v=u+at, here u=0 then t= 16.6/5.7 = 3 sec

Rear Axle Shaft

Rear axle shaft is designed on Bending moment basics. Diameter of rear axle shaft = 40mm.

Shaft simulation results

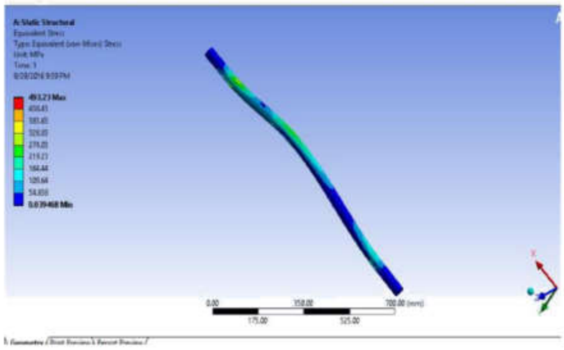


Fig: Stress induced

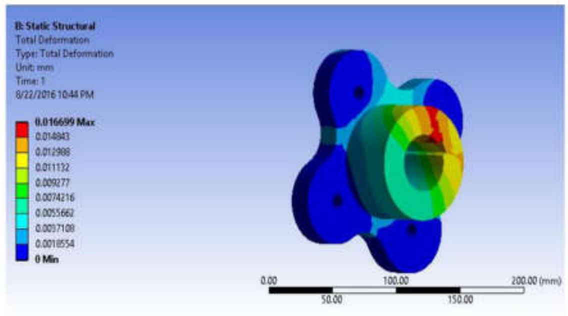


Fig Deformation

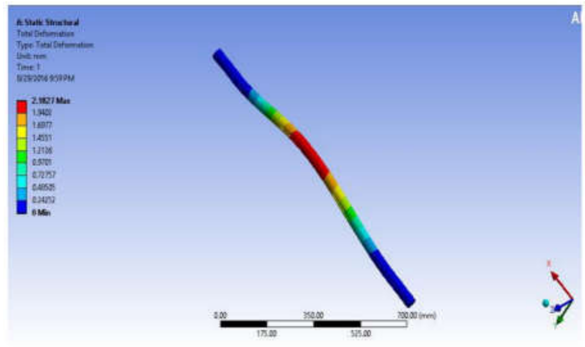


Fig Shaft deformation

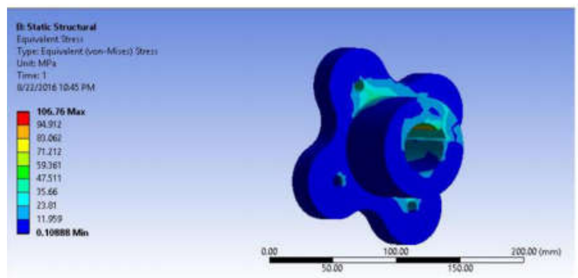


Fig Stress Induced

The shaft has keyways machined exactly opposite to each other with dimensions of 12 x 8 mm in breadth and height.

Hubs

The rear wheel hubs are designed keeping in mind simplicity, machinability. The hub is designed with key way which will be safely housed on the keyway on the shaft ends.

Wheel Hub

Fig CAD model of hub

Braking System

The objective of the braking system is to increase the safety and maneuverability of the vehicle. It converts kinetic motion of vehicle in to heat energy by using frictional force offered by the brake pads. Thus, the vehicle can be slowed down or stopped at any desired distance.

Design Methodology

The braking system design includes the single disc at the rear axle to lock/stop the vehicle. Master cylinder is used at the front near the brake pedal providing the occupant to easily accessible space.

List of design criteria or requirements

1. Assign maximum deceleration of the Vehicle
2. Calculate target stopping distance and target braking force
3. Select optimum brake parts which in combination help achieve target
4. Optimize brake design

Design of Brake System Elements

Brake pedal

Passenger cars generally use a pedal ratio of 3 to 6. We select a pedal ratio of 6 at both rear and front.

Master cylinder: The master cylinder with bore Diameter 10mm proved to be best suited for the design. The single piston master cylinders were used.

Calliper and Disc Selection

The dual piston calipers were selected due to their light weight easy availability and reliability.

The disc of suitable to the calliper was used at the rear and front.

Dimensions of Disc and calliper

Front

1. Diameter of the Disc: 130mm
2. Thickness of the Disc: 4 mm
3. Effective disc radius: 60mm
4. Coff. Of friction of brake pads: 0.4 Rear:

1. Diameter of the disc: 200mm
2. Thickness of the Disc: 4 mm
3. Effective disc radius: 95 mm
4. Coff. Of friction of brake pads: 0.4

Design

The braking system composed of disc brakes. The disc brakes are arranged for two wheels on the Front and a single disc at the rear in order to satisfy the braking requirements. The front discs are mounted on each of the wheel and the rear disc to the axle.

Testing

Analysis of the Hub

The hub for the brake was tested using 3D stress analysis in the ANSYS. The hub was analyzed for impact forces (KN) and the brake torque. The results of analysis were within permissible range.

Brake Fluids

We have decided to use DOT 3 Brake fluid

- Inexpensive
- Easily available
- Compatible

Weight Distribution		
Gross Mass of the Bike =	270Kg	
Front: Rear =	2:3	
All data has been tabulated below:		
Type	**REAR**	**FRONT**
Disc	OD-200 mm	OD – 130 mm
Master cylinder	10 mm	10 mm
Dia.		
Caliper piston diameter	35 mm	35 mm
Brake Pedal Lever ratio	6:1	6:1
Stopping distance	2.4m	2.86 m

Formulae

Brake line pressure = pedal ratio *force on the pedal / area of master cylinder
Rear: = 26.73 MPa ; Front: = 22.91MPa
Clamping force = brake line pressure *(area of caliper piston * 2)
Rear: = 51449.99 N ; Front = 44100 N
Rotating force:
RF = CF* coefficient of friction of brake pads Rear: = 20579.996 N ; Front: 17640N

Braking torque (tn) = Rotating force* effective disc radius
Rear: = 2057.99 Nm; Front: 1146.6 Nm
Braking force = Braking Torque /tire radius Rear: = 7365.556N; Front: = 2149.60N
Deceleration: F=-ma (-ve sign indicates force in opposite direction)
Rear: = -76.22 m/s^2 ; Front: - 67.20
Stopping distance V^2-U^2=2*a*D_s (V=0,U= 19.44m/s) Rear: Ds = 2.4 m ; Front: Ds = 2.86 m

Analysis

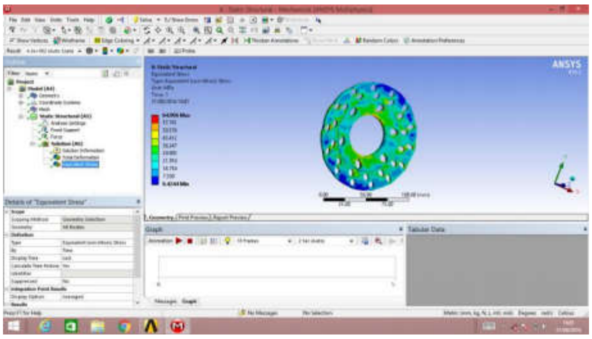


Fig: Disc Analysis

Various parameters of braking system

Parameter	Front	Rear
Vehicle Velocity	19.4 m/s	19.4 m/s
Stopping Distance	2.86 m	2.4 m
Stopping Time	0.3	0.9
Maximum Deceleration	67.20	76.22
Wheel Radius	10"	11"
Rotor Radius(Effective)	60 mm	95mm
Rotor radius (Actual)	65	100
Frictional Coefficient (Tires)	0.3	0.3

Body Works

External appearance of the vehicle depends upon bodyworks. It is an important part of the vehicle design. It also dominates sale and marketing of the vehicle.

We have selected fiber on the basis of market survey because of its light weight and good electrical insulation

Innovation

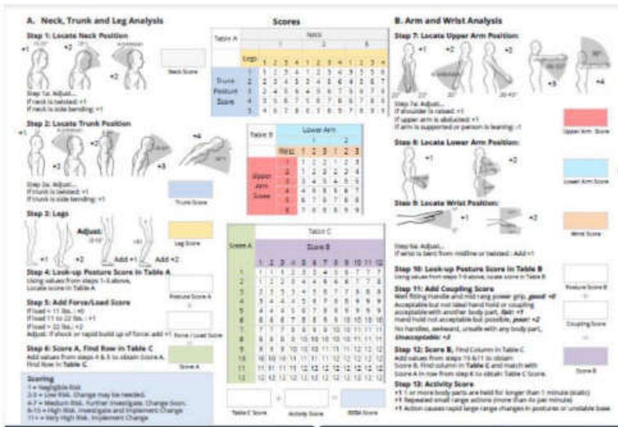
Use of plastic teflon pipes below the engine to reduce vibrations.

Safety and Ergonomics

Safety is the most important concern for our Quad bike. Bumper is provided for safety. In addition fire extinguishers and kill switches will also be used in case of emergency. Ergonomics are designed perfectly for the comfort of the driver using RAPID ENTIRE BODY ASSESSMENT METHOD (REBA) and the value obtained intimates the non necessity of the change in driver's posture.

Driver's
Pictu

Ergonomic Measurements Obtained For Two Drivers (REBA Method)



Steps 1-3: Neck, Trunk And Leg Analysis *STEP 1: Neck* ***position***

Locate neck position: Neck is bent not more than 20° Neck score for Driver 1=.+1
Trunk inclination is 0°
Trunk score for Driver 1=.+1 Trunk score for Driver 2=+1 STEP 3:**Leg position**
Knee Angle > 60° and two legs are supported Leg score for Driver 1 =.+1

Leg score for Driver 2=+1 STEP4

Table A		Neck											
		1				2				3			
	Legs	1	2	3	4	1	2	3	4	1	2	3	4
Trunk Posture Score	1	1	2	3	4	1	2	3	4	3	3	5	6
	2	2	3	4	5	3	4	5	6	4	5	6	7
	3	2	4	5	6	4	5	6	7	5	6	7	8
	4	3	5	6	7	5	6	7	8	6	7	8	9
	5	4	6	7	8	6	7	8	9	7	8	9	9

Values obtained from step 1-3 are checked against the table; Score obtained in table A = 2

STEP5: **Adding forces/ Load score**

Shock or rapid build up of force is observed while riding the bike. therefore score added=+1

STEP 6**: score A** =table A +force score =2+1-=3 Score A=3

STEP 7: **Upper arm position**

Upper arm position is 20 to 45° For driver 1+2

STEP 8: **Lower arm position**

The angle is not more than 90°

Lower arm position score for driver 1 = +2 Lower arm position score for driver 2 = +2 STEP 9: **Position of wrist**

Wrist angle is below 15° Score for driver 1 = +1 Score for driver 1= +1 STEP 10: **Table B**

Table B		Lower Arm					
		1			2		
	Wrist	1	2	3	1	2	3
Upper Arm Score	1	1	2	2	1	2	3
	2	1	2	3	2	3	4
	3	3	4	5	4	5	5
	4	4	5	5	5	6	7
	5	6	7	8	7	8	8
	6	7	8	8	8	9	9

Values obtained from steps 7-9 are checked against Table B, obtaining a Score = 2
STEP 11: Well fitted handle is used hence the coupling score is 0

Coupling Score = 0

Score B=Table B + coupling score = 2+0 =2

STEP 12: **Add the values in step 10 and 11 to obtain score b.** Next, find column in table c and match with score a in row from step 6 to obtain table c score. STEP 13: **The activity score is +1 due to job requiring small range actions** (more than 4x per minute).

The final REBA score = table c score + activity score

Score A	Table C											
	Score B											
	1	2	3	4	5	6	7	8	9	10	11	12
1	1	1	1	2	3	3	4	5	6	7	7	7
2	1	2	2	3	4	4	5	6	6	7	7	8
3	2	3	3	3	4	5	6	7	7	8	8	8
4	3	4	4	4	5	6	7	8	8	9	9	9
5	4	4	4	5	6	7	8	8	9	9	9	9
6	6	6	6	7	8	8	9	9	10	10	10	10
7	7	7	7	8	9	9	9	10	10	11	11	11
8	8	8	8	9	10	10	10	10	10	11	11	11
9	9	9	9	10	10	10	11	11	11	12	12	12
10	10	10	10	11	11	11	11	12	12	12	12	12
11	11	11	11	11	12	12	12	12	12	12	12	12
12	12	12	12	12	12	12	12	12	12	12	12	12

FINAL TABLE C**:** The final REBA score =**2 + 1 = 3**

Therefore, the score obtained is acceptable and changes of posture may not be necessary.

Considering Ergonomics, driver was made to sit in the actual model. Remained in the position for 20 minutes, Thereby, simulating driving conditions.

Electricals

12V DC Battery will be used to power all the electrical components.

CONCLUSION

The design and construction for QBDC has been a challenging task. A detailed study of various automotive systems is taken as our approach. Thus this report provides a clear insight in design and analysis of our vehicle. This project has helped us students in extensive learning and gaining practical knowledge on the theoretical concepts imparted in classrooms.

References

Fundamentals of Vehicle Dynamics-Thomas A Gillespie

Dr.Sreenatha Reddy principal for his continued support and Mr.Vijaya Kumar, head of department of GNIT and Mr.Shaik Himam Saheb for their encouragement and for their valuable support during the project.

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