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Multi Body Dynamic Analysis on An All-Terrain Vehicle

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Abstract - When the roads vanish, the suspension system comes into picture majorly. To be able to design an ATV (All-Terrain Vehicle) suitable for most off-road conditions is still a target for most existing companies. Most off-road terrains are difficult to maneuver, causing high reaction forces onto the driver. A considerably less off-road terrain causes an ATV to roll as the vehicle is not designed for that purpose. The main objective of this work is to create a customizable ride quality ATV comprising of minimum roll, reduce steering effort, allow end user customizability and thereby provide better stability. Hence the outcome of this work will be an increase roll camber gain, reduction in scrub and weight, minimized toe on the front and dynamic roll center as close as possible to the center of gravity.

***Key Words*:** ATV; Stability; Steering; Suspension System.

1. Introduction

The purpose of this work is to design and analyze the suspension geometry of an ATV, both at the front end and the rear end, using Multi body dynamics method on Lotus Shark Suspension Analysis software. The main outcomes of this work is to cause minimum roll, reduce steering effort and allow end user customizability. This is done to increase the stability of the vehicle while off roading and we can achieve this outcome by increased roll camber gain, reduce scrub and weight, minimize toe on the front and to keep the roll center dynamically as close as possible to the centre of gravity. To be able to design an ATV suitable for most off-road conditions is still a target for most existing companies and this is one of the main crucial things in the automobile industry. This work aims at creating a customizable ride quality ATV. The front suspension geometry is chosen to be independent Double Wishbone type and the Rear wishbone is chosen to be H Arm type double wishbone with no Control link and thereby zero toe. The hard points are thereby chosen to meet these objectives by successive iterations on Lotus Shark Suspension Analysis. The design and analysis of the individual components as well as the complete vehicle will be carried out on Solidworks and Ansys. Each component of the suspension geometry is taken under analysis for fatigue and different load operations. The subsequent materials and their properties are iterated accordingly with respect to the factor of safety thereby obtained and their utility. The damping and the spring calculations are done with respect to the vehicle roll rate, ride rate, natural frequencies, load distributions, wheel and spring travel- motion ratio, springs progression and other characteristics. In the experimental studies by Solomon et al. [1] the hydro-gas suspension model was used for the experimental force–displacement characteristics. S. Sankar et al. [2] helped understand, a computer simulated model of a tracked vehicle which is developed for suspension dynamic analysis and ride quality assessment. The study conducted by Balamurugan et al. [3] focuses on the development of single station representation of tracked vehicles with trailing arm hydro-gas suspension systems by simulating the ride dynamics. Tyan et al. [4] shows the ability to accurately simulate the vibratory motion of transport vehicle and its importance when designing vehicle components. E.J. Haug et al. [5] provided an explanation for the constrained mechanical systems using computers aided analysis of large-scale mechanical system. M. K. McCullough et al. [6] gave a vectoral approach towards the components thereby explaining the dynamics of a multi-body device. M. J. Vanderploeg et al. [7] covered the aspects of modelling a multi-body vehicle and came to the conclusion that usage of multi-body dynamics is a good option to ensure better braking and handling in the vehicles. H. J. Lai et al. [8] elucidates how a double-wishbone suspension allows the independent reaction of each wheel. R.A. Wehaoe et al. [9] put forward certain assumptions to help analyse the dynamics of high mobility tracked vehicles as well as their power and acceleration. M. D. Bennett et al. [10] clarified how it is possible to use the contact points between two parts moving in relation to one another in order to decrease the challenges faced by the linear approach.

2. Methodology

Mentioned below are the software that were used for the execution of the complete analysis :

I. Lotus Shark suspension analysis- Vehicle Suspension design and analysis software.
II. Solidworks- Vehicle and its parts design & analysis software
III. Ansys- Parts Analysis Software

The main targets of this analysis are mentioned below:

1. To design and analyse the suspension geometry of an ATV, using Multi body dynamics method on Lotus Shark Suspension Analysis software.
2. Camber gain in bump

   ront ump to - roop to to.

   ear ump to - roop to to.

3. Toe total change: - . to.4.

Many ongoing and previously completed works were reviewed before proceeding with the work mentioned here. Multi body dynamic analysis includes analysis and simulation of more than one body at a single time. Literature survey and review done of the various research papers inclined towards the analysis of individual components and simulation of those components. Flexibility and rigidity of an ATV frame was deeply reviewed to increase the rigidity of the frame (All-Terrain Vehicle Flexible Multibody Dynamic Simulation for Fatigue Prediction by Jia-Shiun Chen and Hsiu-Ying Hwang). Fatigue study was carried on after conveying from the above research paper and was thoroughly kept in mind while carrying out this work. A basic idea and motivation about the design and manufacturing of an All-Terrain vehicle was taken from the literature survey of (Design & Manufacturing of All Terrain Vehicle- Selection, Modification, Static & Dynamic Analysis of ATV Vehicle). Ideations about the design of frame, design aspects which plays important role in dynamics and the parameters which affects the dynamic as well as manufacturing process were taken into considerations through this literature survey. Design Methodology for designing of roll cage was well described in the literature review. Parameters for steering system, suspension system and transmission were well added to give a brief demonstration about the designing and manufacturing of an All-Terrain Vehicle. The Survey addressed the following factors: Analysis of dynamic components, the parameters to be taken into consideration, design aspects, optimization of suspension geometry, determining the factor of safety, simulation.

3. Design parameters to be considered

There are various design aspects to be considered while designing. These factors play a major role in the dynamic behavior of the vehicle. Some of the factors are:

3.1 Material Selection

Material selection is the major key point in every vehicle's design. It can either drastically increase the performance of the vehicle if right material is selected or can decrease the performance if wrong material is selected. Material is selected based on performance requirement and the utility of the vehicle being manufactured. Reliability also plays major role in the material selection. For instance, if reliability is not priority then a light weight material can be selected which ultimately increases the vehicle performance. Size, shape and mass are also the factors which influence the material selection.

3.2 Material Properties

Material properties which include the physical and chemical properties influences the choice of material. The properties which are considered while selecting a material based on material properties are:

Young's Modulus, Strength, Plasticity, Brittleness, Toughness, Stiffness, Elasticity, Ductility, Creep, Fatigue.

3.3 Suspension Geometry

Suspension geometry are of various types starting from the dependent and independent suspension type geometry. Suspension geometry varies with the type of suspension being used. In Dependent suspension when vehicle undergoes any bump or obstacle, the movement of one wheel is influenced by the motion of another wheel i.e. they are dependent to each other and does not have independent movement of their own. In Independent suspension the movement of one wheel is not influenced by the motion of another wheel and thus have independent motion of their own. It is used majorly in all off-road purpose vehicles. Independent Double Wishbone type suspension is used here which gives greater wheel travel and performance advantage over any other suspension type for off- road purpose.

3.4 Determining Factor of Safety (FOS)

For determining factor of safety, FEA operation is performed on the component in FEA software. Ansys is a Finite Element Analysis (FEA) software which is used to analyze and simulate various engineering problems such as structural analysis, thermal analysis, fluid analysis and also air flow for aerodynamics. It acts as a common platform to integrate all kinds of disciplines of physics, mechanics, structures, heat transfer and fluid dynamics. Structural analysis of the structure i.e. the chassis of the vehicle is carried out on Ansys. These experiments are basically carried in three ways to perform the operation.

These are stated as: Front Impact, Rear Impact, Side Impact

4. Experimental setup in SOLIDWORKS

4.1. Vehicle Dimensions

The track width was greater at the front compared to the rear, as it provides more stability during cornering; as well as ensuring less slippage at the rear. The Vehicle parameters used are specified in table 1.

Table 1: Vehicle Parameters

Parameter	Value
Wheel Base	1100mm
Track Width (Rear)	1000mm
Track Width (Front)	1080mm
Width	1283.2mm
Length	1584.2mm
Ground Clearance	330mm
Center of Gravity Height	450mm

4.2.Material Selection

The material comparison was done between AISI 347 steel and AISI 4130 steel using the parameters as availability, cost and weldability. Where AISI 347 Steel had availability of 4, cost of 2 and weldability of 3. And AISI 4130 Steel had an availability of 3, cost of 3 and weldability of 4. The Material Properties used are mentioned in Table 2.

Table 2. Frame Material Properties

Material Properties	AISI 347	Chromoly 4130 Steel
Elastic Modulus	195000 N/mm^2	205000 N/mm^2
Poisson's Ratio	0.270	0.285
Mass Density	8000 Kg/m^3	7850 Kg/m^3
Tensile Strength	515 N/mm^2	560 N/mm^2
Yield Strength	275 N/mm^2	460 N/mm^2
S/W Ratio	82 KN-m/Kg	110 -m/Kg

5. Frame

Weight plays a major design factor in any frame that drastically increases the safety, performance and reliability of any vehicle design. To meet all the requirement and gain an advantage in weight side, extensive research was carried out and comparison of materials with different properties were also carried out. The selected material, Chromoly 4130 steel which shows all of the above features is used to carry out this work. AISI 4130, Chromoly is a low alloy steel, doped with chromium and molybdenum. The low Carbon content (0.3%) increases the weld ability of the material, thus contributing to ease of use. The material finds common applications in aircraft frames and race car frames. Tubes of following dimensions were used:Primary members: OD 25.4mm, Wall thickness 1.65mm, Secondary members: OD 25.4mm, Wall thickness 1.20mm.

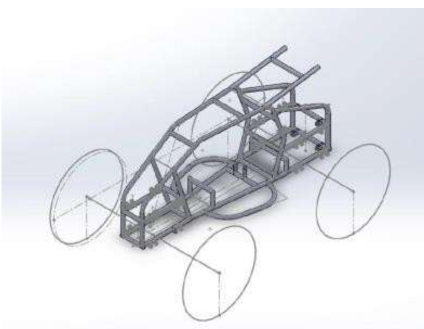

Fig. 1. Isometric View of Frame

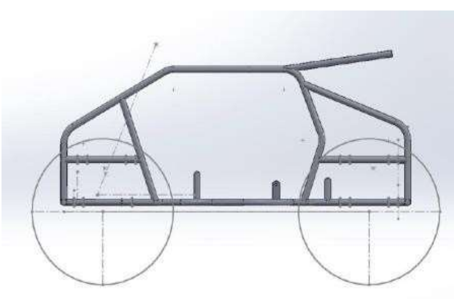

Fig. 2. Side View of Frame

5.1 Upright Assembly

The upright assembly- namely the wheel hubs and upright- fall under unsprung mass and hence, were to be kept to a bare minimum to improve suspension and steering stability. Hence, Aluminum and its alloys were considered. After thorough research on each material for strength, weight and cost, Aluminum 6061 was considered. The material properties are given below in table 3.

Table 3. Upright Material Properties

Material Properties	Aluminium 6061
Elastic Modulus	68.9 GPa
Poisson's Ratio	0.33
Mass Density	2700 Kg/m3
Tensile Strength	310 MPa
Yield Strength	276 MPa
Elongation	12%

5.2 Wheel spindle

Since the wheel spindle experiences high bending forces, Aluminium was not a favoured choice. A much more ductile material was required, and hence, we switched back to the steels. EN24 was chosen, owing to its high yield strength of 680MPa. It was taken in the form of a solid shaft, to enable it to withstand the stresses.

5.3 Front Wheel Assembly

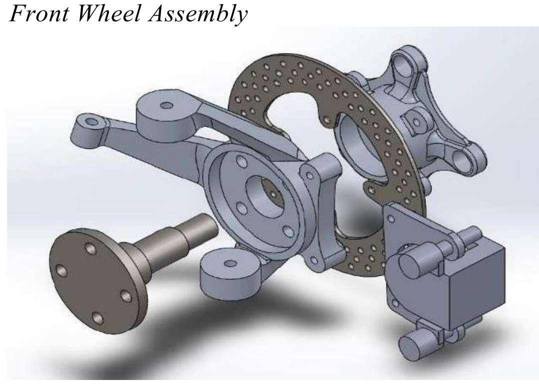

Fig. 3. Top View of exploded Front Wheel Assembly

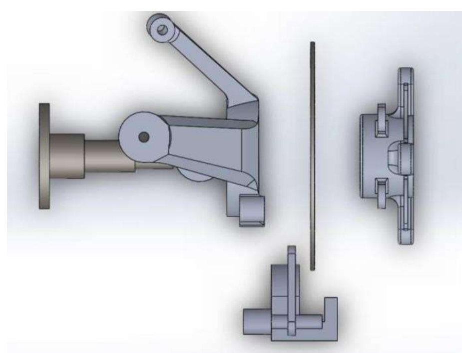

Fig. 4. Isometric View of exploded Front Wheel Assembly

5.4. Rear Wheel Assembly

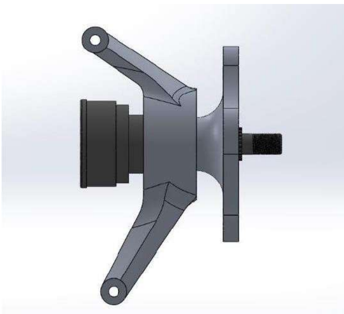

Fig. 5. Side view of Assembled Rear Wheel Assembly

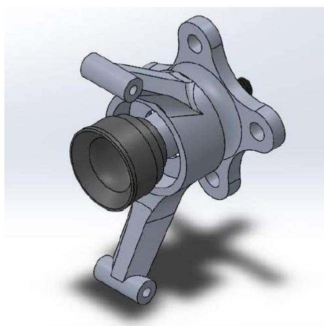

Fig. 6. Isometric view of assembled Rear Wheel Assembly

6. Lotus Shark

The front suspension geometry is chosen to be independent Double Wishbone type and the Rear wishbone is chosen to be H Arm type double wishbone with no Control link and thereby zero toe. The main objective of this work is to cause minimum roll, reduce steering effort and thereby provide better stability. Hence the objectives of this work remain to increase roll camber gain, reduce scrub and weight, minimize toe on the front and to keep the roll centre dynamically as close as possible to the centre of gravity. The hard points are thereby chosen to meet these objectives by successive iterations on Lotus Shark Suspension Analysis. Each component of the suspension geometry is taken under analysis for fatigue and different load operations. The subsequent materials and their properties are iterated accordingly with respect to the factor of safety thereby obtained and their utility. The damping and the spring calculations are done with respect to the vehicle roll rate, ride rate, natural frequencies, load distributions, wheel and spring travel- motion ratio, springs progression and other characteristics.

Targets for Suspension Geometry

- Camber gain: ront ump to - roop to, ear to - . roop to .
- Toe total change - . to . 4
- Reduce Scrub(<45mm) to ensure minimal steering effort, maximize Roll camber gain.

The Static Values used for the targeted suspension geometry is shown in table 4.

Table 4. Static Values of various parameters

Static Values	
Camber Angle (deg)	0.00
Toe Angle {Plane} (deg)	0.00
Toe Angle {SAE} (deg)	0.00
Castor Angle (deg)	3.94
Castor Trail (hub) (mm)	5.85
Castor Offset (grnd) (mm)	14.24
Kingpin Angle (deg)	11.41
Kingpin Offset (w/c) (mm)	106.16
Kingpin Offset (grnd) (mm)	47.22
Mechanical Trail (grnd) (mm)	14.21
Roll Centre Height (mm)	238.50

7. Camber Gain with Bump and Droop, Castor Angles and Toe changes over Bump and Droop

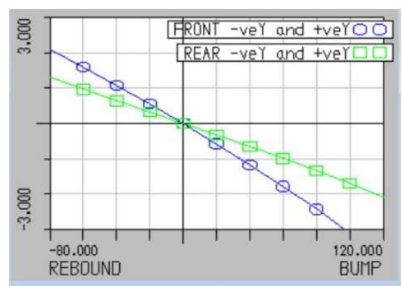

Fig.7.1. Camber Gain with Bump and Droop

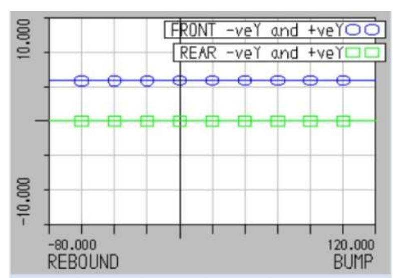

Fig.7.2. Castor Change

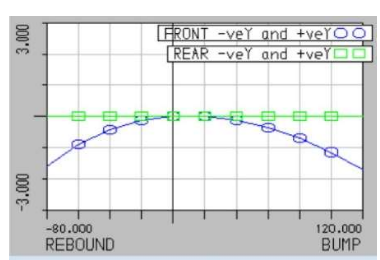

Fig.7.3.Toe change

8. Calculation

8.1. Suspension

The total load of the vehicle (which includes frame, engine, driver and other weights) was estimated to be 230kg (approx.)

Weight Distribution= 40:60 (front: rear). Therefore, The total load acting on the front suspensions,

=Total weight * Weight distribution for Front = 230*0.4 = 92kg (902.52N)

The total load acting on the rear suspensions,

=Total weight * Weight distribution for Rear = 230*0.6 = 138kg (1353.78N)

8.2. FRONT SUSPENSION:

8.2.1. Motion Ratio

The term 'Motion Ratio' in suspension system determines the deflection in vertical direction of wheel whenever the vehicle hits a bump or any obstacle. Basically, it is the ratio of distance between lower shock mount and lower wishbone inner mount' and 'Length of lower wishbone member'. Motion Ratio plays an important role while designing the suspension geometry of any vehicle. Motion Ratio is decided by keeping the following requirements and parameters in mind: Wheel travel for the suspension, Total deflection, that is the shock compression, Angle of shock mounting.

Therefore, Length of lower wishbone member = 367.21mm

Distance between lower shock mount and lower wishbone inner mount = 250mm

Motion Ratio, = 250/367.21 = 0.68

8.2.2. Wheel Rate

The term 'Wheel Rate' determines the spring rate which is measured at wheel instead of the point where spring attaches to the linkage which is pick-up point. In other words, wheel rate is determined by the ratio of Load on Wheel to Spring travel of that wheel.

Spring travel (droop to static) = 108mm

Wheel Rate, = 270.75/108 = 2.507 N/mm

8.2.3. Spring Rate

The term 'Spring ate' determines the amount of weight or load needed to compress a spring by one inch, that is the weight need to compress it to one inch. For example, if the rate of spring is 80N/inch means that 80N is required to compress that spring to one inch. Spring rate is determined by the ratio of Wheel travel to the square of motion ratio multiplied by angle of correction.

Therefore, Spring Rate, = $WR/(MR2 * cos\alpha)$.

Where, cos α = Angle Correction Factor=33 degree. Total mass on front suspension = 92kg.

Sprung mass at each wheel in front, = 92 × (60/100) × (1/2) = 27.6kg (270.75 N)

Therefore, Spring Rate, = 2.507(0.68*0.68*cos33) = 6.465 N/mm

8.2.4. Force Analysis

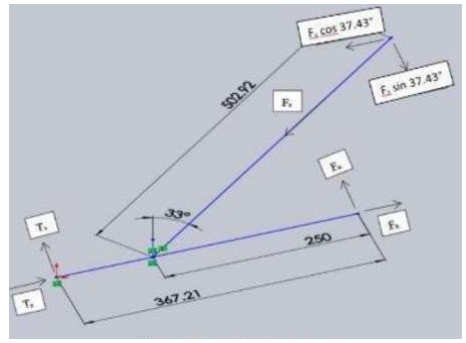

Fig.8. Front Force Analysis

TX: Force exerting on wheel in x-axis, FX: Force exerting on Frame in x-axis, TY: Force exerting on wheel in y-axis, FY: Force exerting on frame in y-axis. FS: Force exerting by shock absorber.

Angle of Correction (angle of shock absorber w.r.t to vertical axis) = 33degrees. Angle between shock and wishbone, α = 33

Sprung mass at the front, = 55.2Kg (541.51N)

Spring Force, Fs= 270.75N

Horizontal forces, Fx+ Tx = Fs cos (37.43) Fx+ Tx = 215N ①

Vertical forces, Fy + Ty = Fs sin (37.43) Fy + Ty = 164.55N ②

Moment about frame end, Spring force and frame forces acting at distances 117.21mm and 367.21mm respectively.

Fs sin (37.43) * (117.21) = Fy (367.21), Fy = [270.75×sin (37.43) × (117.21)]/367.21 Fy= 52.525N

Substituting Fy in ②we get Ty = 164.55 – 52.525 Ty = 112.03N

As Fx= Tx, we get 2 ×Fx= 215, Fx= Tx= 107.50N

Resultant Forces,

Force (Upper wishbone Outer mount) = 155.26N Force (Upper wishbone Inner mount) =119.64N

8.3 Rear Suspension

8.3.1 Motion Ratio

Wheel travel for the suspension, Total deflection, that is the shock compression, Angle of shock mounting.

Therefore, Length of lower wishbone member = 273.35mm

Distance between lower shock mount and lower wishbone inner mount = 210mm

Motion Ratio, = 210/273.35 = 0.76

8.3.2 Wheel Rate

Wheel rate is determined by the ratio of Load on Wheel to Spring travel of that wheel.

Spring travel (droop to static) = 39mm, Wheel Rate = 406.134/39 = 10.38N/mm

8.3.3 Spring Rate

Spring rate is determined by the ratio of Wheel travel to the square of motion ratio multiplied by angle of correction.

Spring Rate = $WR/\ (MR2 * cos\alpha)$, Where, cos α = Angle Correction Factor =25.75-degree, Total mass on rear suspension = 138kg, Sprung mass at each wheel in rear, = 138 × (60/100) × (1/2) = 41.4kg (406.134 N)

Therefore, Spring Rate, = 10.38(0.76*0.68*cos25.73) = 19.54N/mm

8.3.4 Force Analysis

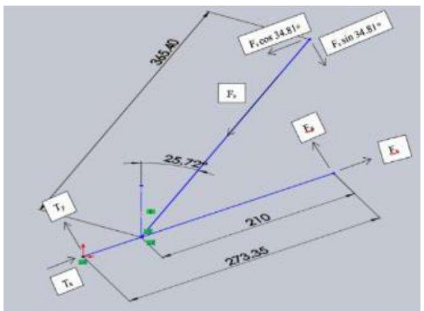

Fig.9. Rear Force Analysis

TX: Force exerting on wheel in x-axis, FX: Force exerting on Frame in x-axis, TY: Force exerting on wheel in y-axis, FY: Force exerting on frame in y-axis, FS: Force exerting by shock absorber.

Angle of Correction (angle of shock absorber w.r.t to vertical axis) = 25.75degrees. Angle between shock and wishbone, α = .7

Sprung mass at the rear, = 82.8KG (812.268N), Spring Force, Fs= 406.13N

Horizontal forces, Fx+ Tx = Fs cos (34.81) Fx+ Tx = 333.45N ①

Vertical forces, Fy + Ty = Fs sin (34.81) Fy + Ty = 231.84N ②

Moment about frame end,

Spring force and frame forces acting at distances 63.35mm and 273.35mm respectively.

Fs sin (34.81) * (63.35) = Fy (273.35), Fy = [406.13×sin (34.81) × (63.35)]/273.35, Fy= 53.73N

Substituting Fy in ②we get Ty = 231.84 – 53.73 Ty = 178.11N

As Fx= Tx, we get 2 ×Fx= 333.45, Fx= Tx= 166.72N

Resultant Forces, Force (Upper wishbone Outer mount) = 243.96N, Force (Upper wishbone Inner mount) =175.16N

8.4. Lateral Weight Transfer

The term 'Lateral Weight Transfer' defines the amount of change in the vertical loads of the tires because of the lateral acceleration caused by centre of gravity (COG) of the vehicle. In other words, it is the change in loads on each side of the car, that is the increased load on the outside side of the car while cornering and decreased vertical loads on the inner side of the corner.

$$\text{Lateral Weight Transfer} = \frac{(\text{Lateral Acceleration} * \text{Weight} * \text{C.G hei}}{Track\ Width}$$

$$Lateral\ Acceleration = \frac{(Linear\ Velocity)\text{^}2}{Turning\ Radius}$$

Or a linear velocity 'v' = kmph and a Turning radius of m Lateral Acceleration = 2.772/5*9.81 = 0.157g

Lateral Wt. Transfer (Front), = 0.157*46*0.45/1.08 = 3.0091kg

Lateral Wt. Transfer (Rear), = 0.157*69*0.45/1 = 4.874kg

Observation Tables for Lateral Weight Transfer for varying velocities at a constant Turning Radius for Front and Rear Wheels are given as Table 5 and Table 6 respectively.

Table.5 Lateral Weight Transfer for Front Wheels

Velocity (Kmph)	Lateral Acceleration (g)	Turning Radius (m)	Lateral Weight Transfer (Kg)
5	0.039	5	0.7475
10	0.157	5	3.2499
15	0.350	5	6.7083
20	0.629	5	12.0558
25	0.983	5	18.8408
30	1.415	5	27.1208
35	1.927	5	36.9341
40	2.510	5	48.1083

Table.6 Lateral Weight Transfer for Rear Wheels

Velocity (Kmph)	Lateral Acceleration (g)	Turning Radius (m)	Lateral Weight Transfer (Kg)
5	0.039	5	1.2109
10	0.157	5	4.8748
15	0.350	5	10.8675
20	0.629	5	19.5304
25	0.983	5	30.5221
30	1.415	5	43.9357
35	1.927	5	59.8333
40	2.510	5	77.9355

8.5. Structural Analysis

In this structural analysis the reliability of the structure i.e. the chassis of the vehicle is carried out on Ansys. These experiments are basically carried by three ways to check the factor of safety. These are stated as: Front Impact, Rear Impact, Side Impact.

For a high-speed impact, it is assumed that the vehicle is stopping in 0.5 seconds after being hit. A vehicle can be hit in 3 possible ways, that are from front, from rear or maybe from side. The factor of safety highly reflects how good the vehicle is when it comes on crumpling the whole structure and absorbing the impact forces. The more flexible frame is likely to be safe more the passengers as it will absorb the forces acting on it and preventing it from going to the passengers' cabin. Therefore, to check this crumbliness or the flexibility of the chassis, these tests are caried. These impacts give the factor of safety of vehicle.

8.5.1. Front and Rear Impact

It is assumed that the vehicle will stop in 0.5 seconds and therefore will have stopping time the same. The vehicle is carrying the speed of 100kmph, i.e. 27.72 MPs. we know, 100kmph = 27.7778 m/s

According to the equations of linear motion, $v^2 = u^2+2*a*s$

where, v = final velocity = 0m/sec, u = initial velocity = 27.7778m/sec, a = deceleration, s = stopping distance = 10m

Therefore, 0 = (27.7778)2 + 2*10*a =7771.6061/20, a=38.5 m/s^2, F = ma = 150*38.5 = 5785N.

Therefore, a force of 5785N is experienced by the vehicle in such incidents of both front and rear impact.

G = Force / Weight*g = 5784/150*9.81 = 5784/1471.5 = 3.93G.

Therefore, an approx. of 4G is applied on the chassis nodes under both front and rear impacts.

8.5.2. Side Impact

During the side impact, it is assumed that the impact angle will be 45degrees deflected from the front or rear impact. The force to be applied, G=Roll Force/Weight*g, Roll Force=Roll rate(Nm/deg)*Roll Angle(deg.)

For determining roll force, we need roll rate,

Therefore, Roll Rate=0.5*(T/2)2*WR, T = Track Width = 1080mm = 1.8m

WR = Wheel Rate = 2.507*10^3, Roll Rate = 0.5*(1.08/2)2*2.507*10^3 = 365.521 Nm/rad

Roll Angle = Rolling Torque*Roll Stiffness

Rolling Torque=Lateral Weight Transfer*Distance between Roll Centre and C.O.G.

Lateral Weight Transfer is given by = Lateral Acceleration*C.G. Height/Track Width

Linear velocity = 40kmph, kmph to MPs = 40*(5/18) = 11.11m/s

Lateral Acceleration = (Linear Velocity)2/Turning Radius = (11.11)2/10*g = 123.431/10*9.81= 1.28g

Therefore, Lateral Weight Transfer = 1.28*46*0.45/1.08 = 24.11*9.81 =236.57N

Therefore, Rolling Torque = 236.57*(450-238) = 236.57*0.212 = 50.16Nm

And also, Roll angle = Rolling torque*rolling stiffness = 5016*365.521 = 7.862degrees/m

Therefore, Rolling Force = Roll Rate*Roll angle = 365.521*7.862 = 2873.72N

As we know, G = Roll Force/Weight*g. Therefore, G = 2873.72/150*9.81 = 1.9529

For side impact approx. of 2G is applied on the nodes of the chassis to carry out the structural analysis.

8.6. Wheel Assemblies

For the dynamic components such as wheel hubs, wheel spindle and wheel uprights, the remote force is applied on from the point of contact of wheel to the ground. A force of 1G in each axis of rol, pitch and yaw, that are x,y and z axis, is applied. Therefore, 1G=150*9.81 =1471.5N, 1471.5N of force is applied in all three directions.

9. Result

9.1. LOTUS SHARK

Lotus Shark is the software used for determining the suspension geometry of various types of vehicles and to optimise the current suspension setup by iterations, which are basically hit and trial method of displacing the hard points manually in the space. It is widely trusted and used by well-established vehicle manufacturers and also by almost all the student teams of universities and colleges.

9.1.1. Iterations:

The main targets of suspension geometry were to obtain camber gain and toe within the stipulated range of ideal setup. Lotus Shark was used to iterate the suspension geometry points to get the good stabilised setup.

The stable iteration range of camber and toe values were set as:

The camber gain of: Front Bump: 0 to -4, Droop: 0 to +2.5 , Rear Bump: 0 to -2.5, Droop: 0 to +1.5, Reduce scrub.

There is no change in castor value as the steering axis in ATV is always fixed and thus, do not change on dynamic conditions. The front suspension geometry is optimized for Independent Double Wishbone type suspension and the Rear wishbone is chosen to be H Arm type double wishbone with no Control link and thereby zero toe. The main objective of this work is to cause minimum roll, reduce steering effort and thereby provide better stability.

9.1.2. Camber Gain

The Camber Gain with Bump and Droop in the above iterated setup for both front and rear suspension was successfully achieved within the target range. The camber values for both front and rear lies within the desirable range of camber change.

9.1.3. Castor

Castor Angles is the angle of steering axis when seen from the side ways of the vehicle. For most of the ATV's steering axis is kept fixed and so the camber does not change in dynamic conditions. As far as this work is concerned, the steering axis is kept fixed and therefore there is no castor change dynamically.

9.1.4. Toe Gain

Whenever the vehicle encounters a bump, there is certain change is observed in toe angle due to the bump. This is change in toe is known as 'Bump Steer'. There should be minimum change in toe, as it increases the driver effort to steer and handle the vehicle over bump and droop. There is no change in toe for rear as the rear suspension. system is equipped with Independent H-Arm Wishbone type geometry which restricts the toe change in dynamic conditions due to the restricted degree of freedom.

9.2. ANSYS

Ansys is a Finite Element Analysis (FEA) software which is used to analyze and simulate various engineering problems such as structural analysis, thermal analysis, fluid analysis and also air flow for aerodynamics. It acts as a common platform to integrate all kinds of disciplines of physics, mechanics, structures, heat transfer and fluid dynamics. Ansys is widely trusted and used software used by many established organizations and manufacturers. Due to its ease of use and user-friendly interface, it is also popular as a learning platform for university level students to hone their skills in analysis. Considering the above factors, Ansys is used for structural analysis of this work.

9.2.1. Frame

Structural analysis of the structure i.e. the chassis of the vehicle is carried out on Ansys. These experiments are basically carried in three ways to perform the operation. These are stated as:Front Impact, ear Impact, Side Impact.

9.2.2. Front Impact

For front impact the force of 4G is applied on the nodes of the chassis, keeping some of its members to be fixed such as the members carrying suspension pickup points. These members are assumed to fixed and does not go any deformation under the impact to transfer all the effective load to the other members on impact. Therefore, on applying the 4G load on the chassis for Front Impact, the results for deformation are:

For frontal impact, we applied a force of 4G, F = 4×G = 4 × 9.81 ×150 = 5886N

where, kerb weight of vehicle = 150Kg, FOS =yield strength / maximum strength = 460/195 (MPa) = 2.36.

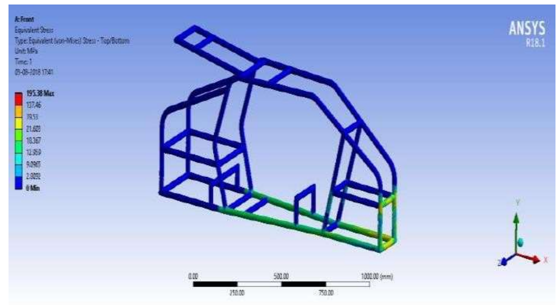

Fig.10. Front Impact

9.2.3. Rear Impact

For rear impact the force of 4G is applied on the nodes of the chassis, keeping some of its members to be fixed such as the members carrying suspension pickup points. These members are assumed to fixed and does not go any deformation under the impact to transfer all the effective load to the other members on impact. Therefore, on applying the 4G load on the chassis for rear Impact, the results for deformation are:

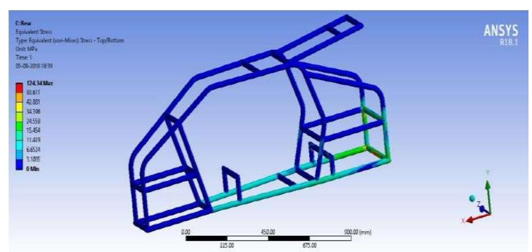

Fig.11. Rear Impact

For rear impact, we applied a force of 4G, F = 4×G = 4 × 9.81 ×150 = 5886N.

where, kerb weight of vehicle = 150Kg FOS = yield strength / maximum strength = 460/124.32 (MPa) = 3.70.

9.2.4. Side Impact

For side impact the force of 2G is applied at 45degrees from the front or rear impact axis, on the nodes of the chassis, keeping some of its members to be fixed such as the members carrying suspension pickup points. These members are assumed to fixed and does not go any deformation under the impact to transfer all the effective load to the other members on impact. Therefore, on applying the 2G load on the chassis for rear impact, the results of deformation are - For side impact, we applied a force of 2G, F = 2×G = 2 × 9.81 ×150 = 2943N, where, kerb weight of vehicle = 150Kg, FOS = yield strength / maximum strength = 460/191.36 (MPa) = 3.02.

9.2.5. Wheel Assembly

Wheel assembly consists of wheel hub, wheel spindle and wheel upright holding them together as a one single assembly. Wheel assembly takes huge number of loads under the dynamic situations. Even apart from dynamic loads, it carries the entire the sprung mass of the vehicle. The weight and reliability factor are basically directly proportional to each other. With more weight, comes the more factor of safety of that component if designed good. Response time of the suspension is directly influenced by the unsprung mass of the vehicle. The time taken by the suspension to provide rebound to a bump faced by the vehicle is known as response time. Less the unsprung mass of the vehicle, less will be the response time for the suspension. The response time should always be as low as possible to obtain a good stability and ride control of the vehicle. Hence, the main target behind designing complete wheel assembly was to attain good factor of safety with less weight to meet all the dynamic performance parameters.

10. CONCLUSIONS

The ATV has been designed and analyzed successfully using Multi Body Dynamics method, using Lotus Shark, Solidworks and Ansys for analyzing suspension geometry, designing CAD and structural analysis respectively. As set in objectives, the following set parameters for suspension geometry were successfully achieved - The camber gain of Bump: 0 to -4, Droop: 0 to +2.5 for front and Bump 0 to -2.5,Droop: 0 to +1.5 for rear is achieved. Overall, toe change of -2.0 to +2.0 is achieved. The scrub has been reduced and the static ride height was increased to 330mm. The toe change was also minimized, and camber gain was induced to reduce the roll camber gain difference. The trade-off between steering effort and bump steer was also controlled by reducing scrub. The complete dynamic parameters have been dully calculated and reported. Different materials were analyzed for the frame and wheel assembly and, the materials were chosen according to the need, availability and application. The light weight front and rear wheel assembly were designed in order to reduce the response time of the suspension. The total weight of the vehicle came around 160kgs theoretically, which is 15% less than a commercially available ATV.

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