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Design and Analysis of Front Suspension System for An All-Terrain Vehicle

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Abstract:
Suspension is one of the important sub-systems of a vehicle. It plays a major role in stability, ride comfort and overall performance of an all-terrain vehicle. The suspension geometry commands the stability of the vehicle and its steering ability. In this study, an attempt has been made to model and analyse the front suspension of an all-terrain vehicle and to study the influence of vehicle parameters on suspension geometry and how they dictate the performance of the suspension system. The double wishbone suspension system is selected for front suspension due to its advantages and is designed and modelled for a vehicle weight of 225 kg , track width of 52 inches, wheel base of 58 inches and a ground clearance of 13 inches. The mounting points of suspension systems are arrived after critical analysis using Lotus Shark software. The components of suspension like A-arms, hub, Knuckle etc., are modelled using SOLIDWORKS 2021 and analysis is carried in the ANSYS 2021.

Key Words: *suspension, double wishbone, A-arms, Knuckle, All terrain vehicle*

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I. Introduction:

A suspension system is the one which is used to maintain the constant contact of vehicle with the ground. Suspension is the system of tires, shock absorbers and linkages that connect a vehicle chassis to its wheels and allows relative motion between them [1]. Suspension plays a vital role in designing an All-Terrain Vehicle (ATV), which provides better handling and performance even in rough terrain. In off-road terrain, the road consists of all kinds of obstacles that could easily bind up the suspension of any road vehicle [2]. The design of front and rear suspension of a vehicle may be different depending upon the requirements.

Among the two types of front suspension system, i.e. dependent suspension and independent suspension, Independent suspension system is a selected here due to advantages like when one wheel goes down, the other wheel does not have any effect. Dependent suspension system has disadvantage like when camber of one-wheel changes, same changes in camber of opposite wheel also changes [3]. As the ATV runs through the rough terrain, which consists of various kinds of obstacles independent suspension is the most suitable suspension for tackling such obstacles on the road surface [4]. Among the independent suspension system double wishbone suspension system is selected for front suspension of an ATV due to advantages like adjustable mounting points, motion ratio etc., [4].

The following design methodology is adopted for the design of front suspension system.

- The type of suspension to be adopted for the vehicle
- Specifying track width, wheel base of the vehicle and ground clearance
- Selection of tyre and rim
- Focussing on the suspension parameters and suspension geometry analysis in lotus suspension software
- Suspension kinetics
- Material selection for the components like A-arms, hub and knuckle
- Design and analysis of A-arms, hub and knuckle

1.1 TYPE OF SUSPENSION

A double wishbone type of independent suspension system is selected with unequal A-arms by mounting the damper on the upper A –arm as the vehicle is four wheel drive. The reasons for selecting this suspension system are

- Ease of manufacturing
- Fine tuning of various parameters
- Adjustability of suspension characteristics
- More accuracy and simple in design
- Unequal A arms provide camber gain during wheel travel

1.2 VEHICLE PARAMETERS

Track width and wheel base of the ATV are finalised for better performance of the vehicle. The track width and wheel base are selected in a way that they improve the cornering performance of the vehicle. Track width of 52 inches and wheel base of 58 inches are selected based on various iterations and also lower track width helps in smaller turning radius and lower wheel base reduces the longitudinal load transfer during braking and accelerating [8].

1.3 TYRE DATA

The tyre and rim are selected so that there is more amount of rubber present between the rim and ground which will provide suspension action and increase grip. Ground clearance and tyre wear are considered in the selection of tyre.

Tyre diameter: 23 inches
Rim diameter: 11 inches
Tyre width: 6 inches

The front rim is selected by considering the compactness of various components that are to be installed. Front rim has 2:4 offset ratio.

1.4 GROUND CLEARENCE

Ground clearance has significant effect on the suspension geometry. So, an optimum value of ground clearance is selected in order for better performance of the vehicle on different terrains. A ground clearance of 13 inches is finalized as the vehicle should climb over the obstacles and move over different kinds of terrain.

1.5 SELECTION OF SUSPENSION GEOMETRY ANGLES AND KINEMATIC SIMULATION

Successive iterations are carried after the input of basic suspension geometry hard points in Lotus software to achieve optimum results of suspension parameters. Some iterative points which are used for the analysis are:

1. Camber change can be controlled by manipulating upper wishbone inner-pivot points.
2. Toe change can be controlled by manipulating inner and outer tie-rod point.
3. KPI change can be controlled by manipulating outer wishbone points of upper and lower wishbone.
4. Small change in these parameters is also observed when track width is increased or decreased.

Camber

Camber is angling of the wheel from the vertical, in front or rear view. The optimum camber angle obtained for vehicle parameters and mounting points is as shown in Fig.1.

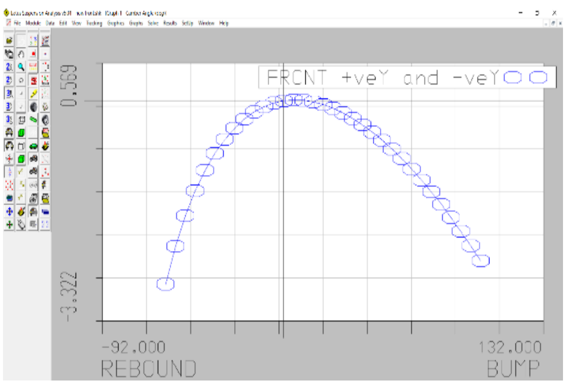

Fig. 1 Optimum Camber angle obtained from Lotus suspension Analyser

Toe

"Toe-in" is when the wheels point towards each other in top view and "toe-out" is when the wheels point away from each other on top view. As the car moves forward, drag from the tires tends to push the alignment toward a toe-out condition. Slight toe-in stabilizes the steering by slightly "preloading" the tires inward. If a toe-out condition exists at speed, the steering becomes noticeably unstable. The optimum Toe angle determined for input vehicle parameters is as depicted in Fig.2.

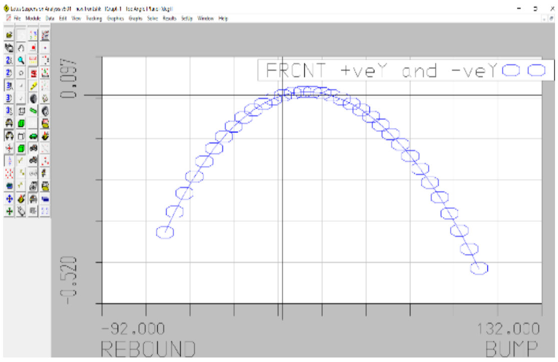

Fig-2: Optimum Toe angle obtained from Lotus suspension Analyser

Caster

Caster angle is the angle to which the steering axis is tilted forward or rearward from vertical, viewed from the side of the vehicle. Steering axis is the line about which the wheel will turn when steered and it connects the upper and lower ball joints of the a-arms. Caster angle has great effect on vehicle handling. Positive caster angle centres the steering wheel after a turn and makes the tires straight again.

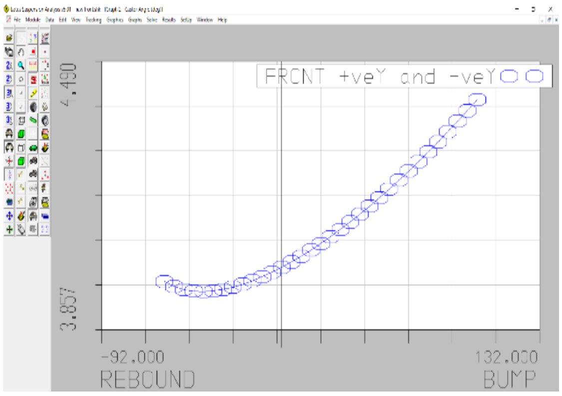

Fig. 3 Optimum Caster angle obtained from Lotus suspension Analyser

Kingpin inclination

Kingpin inclination angle is the angle between the symmetric axis of the wheel and the imaginary line connecting the outer upper and lower ball joints when viewed from the front [2]. The Obtained Kingpin Inclination for the given vehicle parameters is shown in the Table.1.

Scrub radius

This is the distance between the centre of the tire tread and the point where the kingpin axis intersects the ground. A large scrub radius gives lighter steering because the tires can roll in an arc as the steering wheel is turned. If the scrub radius is zero, then the tire has to be twisted on the ground, and the steering effort is highest. However, too large of a scrub radius makes the steering very sensitive to road surface irregularities. As the tire hits a bump, it will tend to turn the steering wheel to the side. [2]. The Obtained scrub radius for the given vehicle parameters is shown in the Table.1.

Anti-dive

The longitudinal load transfer incidental to braking acts to pitch the vehicle forward producing "brake dive." Suspension can be designed to resist braking the generation of anti-dive forces during braking. Because virtually all brakes are mounted on the suspended wheel, the brake torque acts on the suspension and proper design can create forces which resist dive [2].

Roll centre

The roll centre is a point at which the suspension links exert a lateral force on the sprung mass. This is represented, in essence, by the roll centre height, which affects the roll angle of the body and the means of lateral load transfer through the suspension links.

Jounce

The term bounce refers to the vertical (up and down) movement of the suspension system. The upward suspension travel that compresses the spring and shock absorber is called the jounce or compression.

Rebound

The term bounce refers to the vertical (up and down) movement of the suspension system. The downward travel of the tire and wheel that extends the spring and shock absorber is called rebound, or extension.

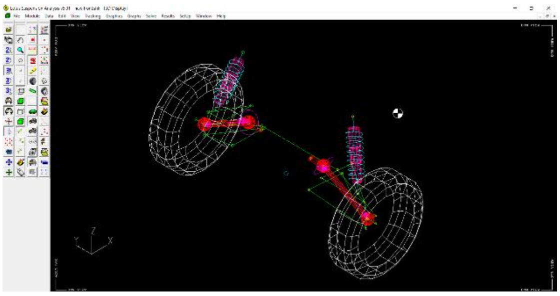

Fig. 4 Quarter car model of suspension in Lotus Shark

Table.1 Optimum Static Suspension Geometry

Parameter	Optimum value chosen
Camber (Degree)	0
Toe (Degree)	0
Caster (Degree)	4.5
Kingpin (Degree)	5
Scrub radius(mm)	53

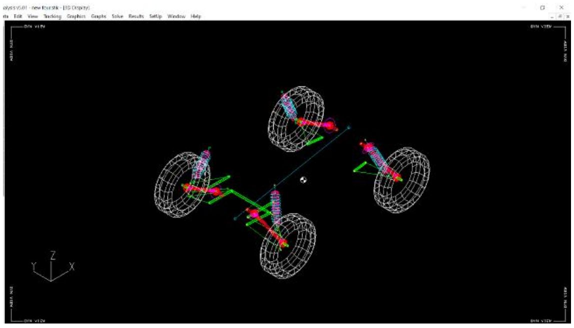

Fig. 5 Complete car model in Lotus Shark

1.6 SUSPENSION KINETICS

Suspension kinetics is concerned about forces acting on the suspension components and the way how those forces travel through them. It mainly focuses on the ride comfort and the factors which are affecting it. For this suspension system Quarter car model is used for the kinetic calculations. There are many such factors which affect the ride comfort few of them on which our focus is placed are listed below.

Ride frequency

A ride frequency is the undamped natural frequency of the body in ride. The higher the frequency, the stiffer the ride. Obtained ride frequency is shown in the Table. 2.

Ride rate

Ride rate refers to the rate of change of tire normal force with vertical body movement. Obtained ride rate is shown in the Table.2.

Motion ratio

Mathematically it is the ratio of shock travel and wheel travel. The amount of force transmitted to the vehicle chassis reduces with increase in motion ratio. A motion ratio close to one is desired in vehicle for better ride and comfort. The ratios are depicted in the Table. 2.

Spring stiffness

The Stiffness of spring formula is defined as the extent to which an object resists deformation in response to an applied force. The stiffnesses are depicted in the Table.2 [3].

Wheel rate

Wheel rate is a measure of the effective stiffness of the suspension system measured at the wheel centre. The wheel rate is the slope of a graph plotting wheel force vs wheel travel. It is expressed in newtons per metre or N/m.

Ride height

Ride height is the position of the body (sprung mass) above the basic ground level datum plane. The dynamic ride height is the ride height relative to the static position. In general, during acceleration, braking or cornering and on rough roads, the ride heights vary continuously and have different values at each wheel.

Table. 2 Parameters of Suspension System

Parameter	Front suspension
Roll centre height(inch)	11
Motion ratio	0.7
Natural frequency (Hz)	1.22
Ride rate(N/mm)	3.356
Ride frequency (Hz)	2.19
Spring stiffness(N/mm)	14.05
Unsprung mass (Kg)	38

Spring Calculations

Motion Ratio (M.R) = (Spring travel/wheel travel) = d1/d2 [2]

d1 = distance between lower wishbone rear/front pivot point & damper end point = 90 d2 = distance between lower wishbone rear/front pivot point & lower wishbone outer ball joint = 200 mm

M.R(d1/d2) = 0.85

Spring travel = 0.85*Wheel Travel

Wheel Travel = (Jounce + Rebound)

Assuming

Jounce = 100 mm

Rebound = 100 mm

Travel =0.7*100 =85 mm

We know that Spring Travel is 0.3 times of spring free length

L_f = 470mm

(l) (l_f) =470mm From Standard Wire Gauge numbers we choose SWG = 0 it represents wire dia of spring (d) is 12 mm

We know that spring free length

l_f=N*d+δ+0.15* δ

Where N is number of turns

Number of active turns (N)=20 turns

Number of active turns (n) =16 turns

We know that = $(8*W*D^3*n)/(G*d^4)$

Where W=Axial load on spring

D= Mean diameter of spring

n =Number of active turns G=Modulus of rigidity

While applying brakes maximum load transfer to front is 0.66 times of total sprung mass of vehicle W = 0.66*225 = 148.5 Kg

Considering 2g condition in bump then; W =4950 N

Load shared by each wheel = 2475 N

For spring steel G = 80000 N/mm2

From D = 60.2 mm

Spring stiffness or Spring rate (KS) = (G*d4)/(8*D3*n)

Where G = modulus of rigidity

d = wire diameter of spring

D = mean diameter of spring

n = number of active turns

(KS) =14.08 N/mm

Wheel Rate (Kw) =Spring rate*(M.R)2

Wheel Rate (Kw) =6.7 N/mm

Table .3 Comparison of AISI 4130 and AISI 1018 Steels

	AISI 4130	AISI 1018
DENSITY (kg/cm^3)	7.85	7.87
YOUNGS MODULUS(GPa)	205	186
POISSIONS RATIO	0.33	0.29
YIELD STRENGTH(MPa)	435	413
ULTIMATE TENSILE STRENGTH(MPa)	670	483

1.7 DESIGN AND ANALYSIS
Calculations
Momentum analysis [2]
F=(m*v)/T
F= (225*11)/0.5
F=4950N
The load on the front wheels is 4950N
The load on single wheel is 2475N
Load transfer
The max longitudinal acceleration Ax = 1.2g The max lateral acceleration Ay = 0.6g
The total longitudinal load transfer can be calculated as
ΔWlongitudinal = (W∗HCG∗Ax)/wheelbase ΔWlongitudinal=(225*290.2*1.2)/1473.2
ΔWlongitudinal = 50.5 **kg** = 493.06 **N**
The total lateral load transfer can be calculated as
ΔWlateral = (W*HCG*Ay)/trackwidth ΔWlateral = (225*290.2*0.6)/1320.8
Δ**W**lateral = 29.64**kg** = 290.48 **N**

Upper A-arm:
Material selected for upper A Arm is chromoly 4130 and its properties are shown in the below Table. 4.

Table-4: Properties of AISI-4130 Steel

PROPERTIES	AISI 4130
DENSITY (kg/cm^3)	7.85
YOUNGS MODULUS(GPa)	205
POISSIONS RATIO	0.33
YIELD STRENGTH(MPa)	435
ULTIMATE TENSILE STRENGTH(MPa)	670

A Arm is designed by taking the design condition as the vehicle falling from a height of 2m when moving with a speed of 40 kmph. For which the loads acting on the a arm are obtained as

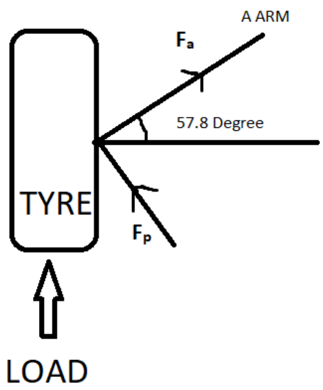

Fig. 6 Load acting on wheel.

F_a=force acting along the A Arm
F_p =force acting perpendicular to A Arm
Fa=2094N
F_p=1318.8N
The load acting on the bracket that is holding the shock absorber is the difference of total load needed to compress the spring and the component of F_p along the shock absorber.

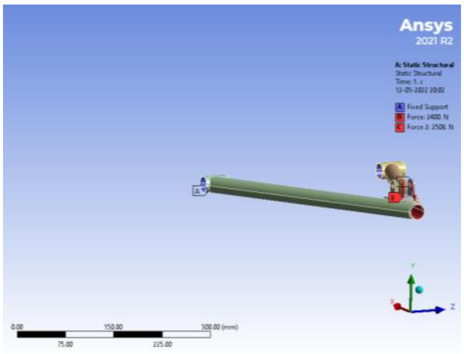

Fig.7 Boundary Condition of A - Arm

By applying above loads on the modelled A- Arm the following stresses are induced in the A Arm. The maximum stress induced in the component is observed to be 288.17 MPa.

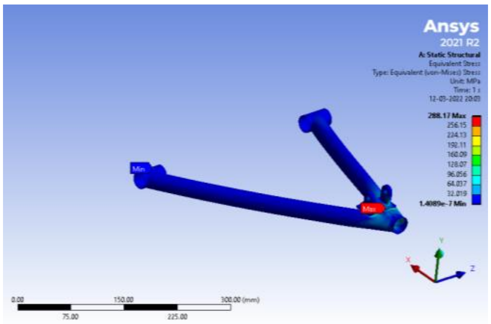

Fig. 8 Stresses on A Arm.

Hub

Material selected for hub is Al-6061 T6 and its properties are shown in the below Table. 5.

Table. 5 Properties of AI-6061 T6

PROPERTIES	Al-6061 T6
DENSITY (kg/m^3)	2700
YOUNGS MODULUS (GPa)	68.9
POISSIONS RATIO	0.33
YIELD STRENGTH(MPa)	276
ULTIMATE TENSILE STRENGTH(MPa)	310

During panic braking condition due to longitudinal load transfer a vertical load component act which compresses the hub. The maximum torque generated by brake calliper during braking will induce twisting moment in between the inner and outer race of hub.

Longitudinal load transfer= 493.06N

Braking torque=$1.9*10^5$Nmm

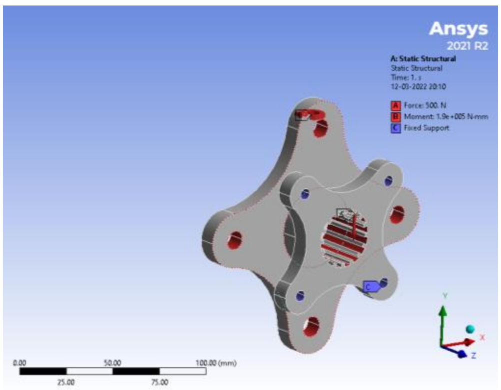

Fig. 9 Boundary Condition of hub

In this case the maximum stress obtained from ANSYS simulation is 163. 95MPa.The failure of hub takes place due to torsion.

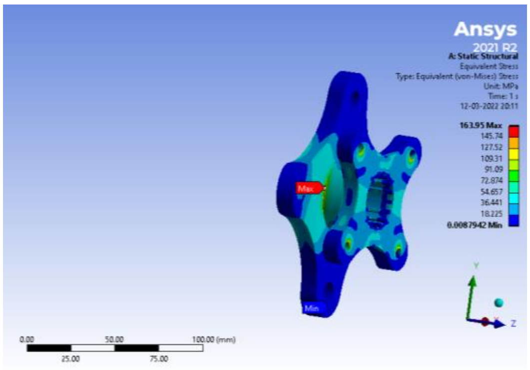

Fig.10 Stresses on hub

Knuckle

Material selected for knuckle is Al-6061 T6 and its properties are shown in the below Table.6.

Table. 6 Properties of AI-6061 T6

PROPERTIES	Al-6061 T6
DENSITY (kg/m^3)	2700
YOUNGS MODULUS (GPa)	68.9
POISSIONS RATIO	0.33
YIELD STRENGTH (MPa)	276
ULTIMATE TENSILE STRENGTH (MPa)	310

Aluminium is selected as the material for knuckle replacing steel in order to reduce weight. The steel knuckle weighs around 2.6Kg whereas aluminium knuckle of same dimensions weighs 0.6Kg but has same strength. The component is designed by keeping in view of yield of the production method.

There are many conditions in which a knuckle fails but a combined conditional case is taken by considering various forces which increases the failure rate of the knuckle. The below loads considered are bump load on the knuckle flanges and steering load on the steering arm.

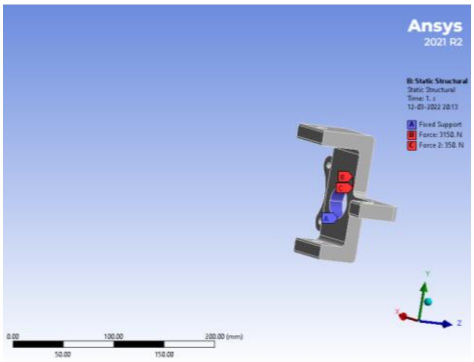

Fig.11 Boundary Condition of Knuckle

Maximum stress is induced in the knuckle is 124. 38MPa.The failure of knuckle is expected to take place at the top flange and the shearing of steering arm from the component. Generally, knuckle is more vulnerable as it has to transmit most of the impact loads to the shock absorber and it is also under the action of forces that are originating from different orientations.

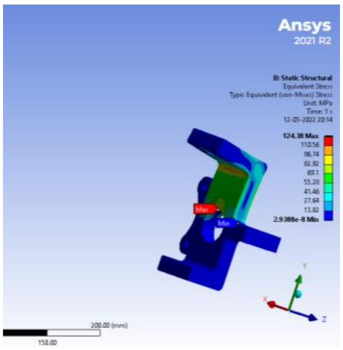

Fig. 12 Stresses on Knuckle

Front suspension integration

The integration of front wheel suspension started with bolting the hinge joints of the A arms to the roll cage then after upper and lower are connected to the knuckle with the help of ball joints. After which the calliper is attached to the calliper mounting points which are provided on the knuckle. The disc is bolted to the hub in such a way that it located between the brake pads. This hub then is bolted to the rim. A bearing id provided on the knuckle to support the drive shaft. The drive shaft then is connected to the hub via splines.

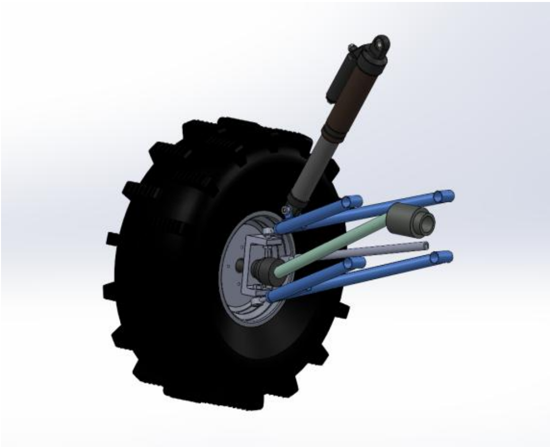
Fig.13 Complete wheel assembly

II. Conclusion

Suspension components of a double-wishbone front suspension system of an all-terrain vehicle are analysed. The forces are calculated using basic concepts like momentum exchange and varignon's theorem. Double-wishbone is designed using suspension hard points obtained from LSA and dynamic force applied considering the factor of safety. Design is validated by using Ansys 2021 R2 software. From the results it is observed that the performance of wishbone system would be satisfactory.

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