Design and Optimization of steering and Suspension System of All Terrain Vehicle

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Abstract - *One of the most important automotive systems is suspension. The primary purpose is to protect the driver from shocks from the road. In addition to providing proper wheel travel and load transfer, secondary functions also include lateral stability and ergonomics and driver comfort. Vehicles with all-terrain capabilities are built to tackle any surface. Stability, vehicle behavior, and driver comfort are the main issues because the vehicle is designed to operate on various types of terrain. The paper's primary goal was to develop and optimize the front suspension wishbone for a BAJA (ATV) vehicle. The purpose of this study is to analyze the entire suspension system of an ATV vehicle to enhance wheel handling and stability. The entire system was made to be strong enough to absorb shocks from the rough terrain that ATVs are typically used on. Calculations were used to design the springs, and SOLIDWORKS was used to design the components. The front and rear systems were simulated using Lotus software, and the components were examined using commercial FEA software from ANSYS. The steering and suspension system's pivot point, known as the steering knuckle, enables the steering arm to turn the front wheels. The knuckle is put to the test in a variety of loading conditions, including bump, cornering, and braking.*

Key Words: Suspension, load transfer, stability and handling, driver comfort, less weight, less expensive, SOLIDWORKS, ANSYS, LOTUS software

1.INTRODUCTION

1.1 Problem Identification

- Failure of steering knuckle and wishbone due to lack of design validation.
- Bump steer, over steer, under steer, loss of traction and roll of vehicle affects handling and performance of the vehicle.
- Slipping of wheel due to undesirable Ackermann geometry

1.2 Design Consideration

- To have a quick steering response and to attain minimum free play.
- To reduce the bump steer, and steering effort

The rack and pinion mechanism were chosen for its simplicity, higher displacement of the wheels, and ease of packaging in the vehicle's toe box. The Ackerman steering geometry was chosen considering the load transfer characteristics, cornering speed, track characteristics, and required turning angle to achieve minimal turning radius with specific wheelbase and trackwidth and to achieve the perfect rolling of the wheels. The Ackerman angle was found to be 25 degrees which was designed to achieve an oversteer that provides sensitive handling characteristics.

Table 1 : Steering Specifications

Steering mechanism	Rack and Pinion
Steering geometry	Ackerman
Turning radius	2.27m
Ackerman angle	25°
Steering ratio	6:1
Lock to lock	264°
Rack lock to lock travel	72mm
C factor	96mm
Steering Effort	62N
Tie Rod Length	412mm

1.3 Steering Calculations

Ackermann angle
Tan β = king pin – king pin center distance/ 2x Wheelbase
β = 25°
Max inner wheel angle-Si= 44°(considering maximum slip)
By Ackermann condition
Cot So- cot Si = Track width/wheelbase

Outer wheel turning angle So = 25.6°
Turning radius of the wheel
Front inner wheel,
Ri = wheelbase /Sin (Si)- (Track width -K. k)/2
Front outer wheel,
Ro = Wheelbase/ Sin Si -Track width -k. K)/2
Where k. k is the kingpin center to center distance
RI = 1828.9 mm, RO = 2926.7 mm

Steering Effort Calculations
Total weight of the vehicle = 245 kg
On front axle = 40% of W
On Rear axle = 60% of W
Weight on one wheel in front = 40% of W = 49 kg
Torque on pinion = Weight on one-wheel x radius co-efficient of friction
= 49 x9.81x21x0.7 =7066. 143N.mm
Torque on pinion = 7.066 Nm
Steering effort Torque on pinion = torque on steering wheel
Torque on steering wheel= steering effort (force given by driver) x Radius
7066.143= F x (5 x 25.4)
F= 55.63
Steering effort = 55.63 N

1.4 SUSPENSION DESIGN CONSIDERATION

- To maintain maximum contact patch with the ground in all conditions, providing higher traction, better handling, and comfort by isolating road shocks while riding.
- To provide minimum body roll while cornering.
- To attain minimum jacking force by managing the static roll center location. Unequal and Unparallel wishbone geometry is chosen to minimize the camber change during cornering.

The front Trackwidth of the vehicle was chosen to be 52" for better stability and rear trackwidth was reduced to 48" for better traction during acceleration. The vehicle's 18" inch ground clearance provides the wheel vertical room to travel and absorb shocks.

This helps reduce scraping against the obstacles and reduces damage underbody. The negative camber aids in better handling at corners and this setup is expected to oversteer in the track because of the camber angles pre-set at the front.

1.5. Spring Rate Calculations

Sprung mass (front) =71.11kg/2 =35.55kg (one wheel)

Sprung mass(rear) =106.67kg/2 =53.3kg (one wheel)

For front, ride frequency (R_f) =1.6Hz

For rear, ride frequency (R_r) = 1.8Hz

Spring stiffness (K) = 4×π^2×(Rf)^2×(M_f)×(M_r)^2
= 4x(3.14)^2×(1.6)^2×(35.55)×(1.57)^2
= 8847.04N/m = 8.29 N/mm
Motion ratio (MR_r) = D1/D2 = 431.8/253.05= 1.70
Spring stiffness (K) = 4×π^2×(R_r)^2×(M_r)×(MR_r)^2
= 4×(3.14)^2×(1.8)×(53.3)×(1.7)^2
= 19682.91N/m
Kr = 19.165 N/mm

Table 2 : Technical Specifications

Parameters	Front	Rear
Spring stiffness	8.29 N/mm	16.15 N/mm
Motion ratio	0.65	0.70
Wheel rate	3.287 N/mm	6.72 N/mm
Roll rate	50.01 Nm/deg	87.13 Nm/deg
Ride frequency	1.6	1.8

1.6 Front Wheel Geometry

Table 1 : Front wheel specifications

Trackwidth(mm)	1320.8
Kingpin Inclination	+8
Scrub radius(mm)	+25.70
Camber angle	-1
Toe angle	0
Ground clearance(mm)	457.2
Roll center height(mm)	341.60
CG Height(mm)	450
Damper length(mm)	482.60
Damper travel(mm)	162.5
Damper angle	21
Motion ratio	0.65

Table 4 : Rear wheel specifications

Trackwidth(mm)	1219.2
Ground clearance(mm)	457.2
Roll center height(mm)	298.91
Damper length(mm)	457.20
Damper travel(mm)	134.62
Damper angle	22
Motion ratio	0.70

Table 5 : Anti-dive geometry specifications

Wheelbase(mm)	1270
Caster angle	+5
Weight distribution	40:60
Ground clearance(mm)	451.8
Anti-dive percentage	40
Anti-squat percentage	0

2. DESIGN of Knuckle

- Front Upright weight: 965 g
- Overall height of Upright: 140mm
- Overall height and length of upper ball joint mounting point is reduced.

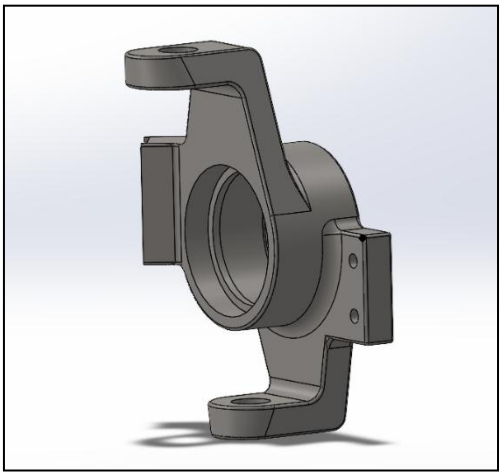

Figure 1 : Front Knuckle

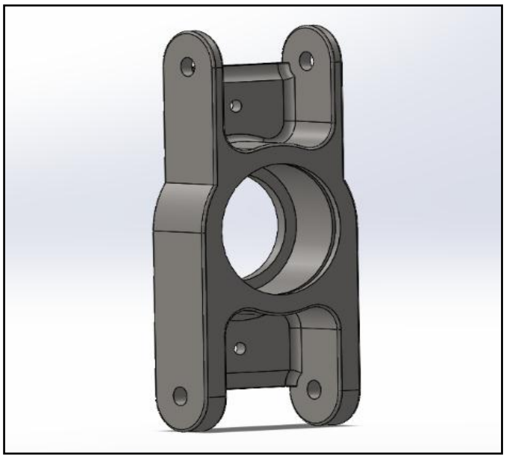

Figure 2 : Front Knuckle

2. ANALYSIS

2.1 Multi Body Dynamics Analysis

Multi-Body dynamics is done so that we can analyze how suspension and steering behaves during bump, and cornering. We have performed suspension analysis using LOTUS SHARK software where we have given parameters of bump travel to be 4 inches analyzed the response of the vehicle.

Table 6 : Boundary Conditions

PARAMETERS	VALUES
Static Camber angle(deg)	-2
Static Castor angle (deg)	5
KPI	8

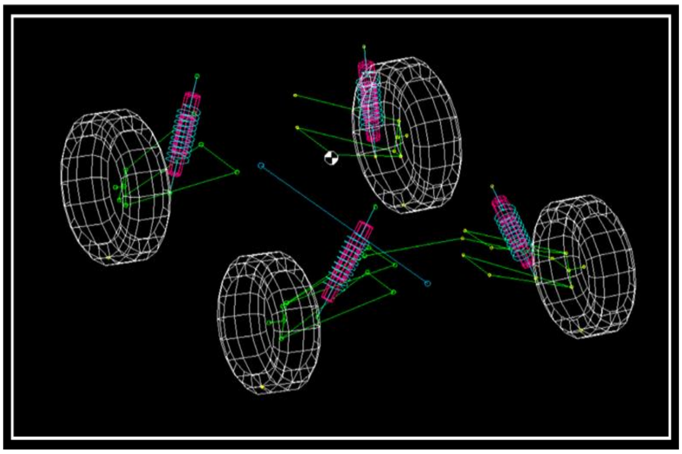

Figure 3 : Dynamic Analysis using LOTUS SHARK

Table 7 : Results of MBD

Parameter	Front	Rear
Camber Change (deg)	2.8	2.6
Toe Change (deg)	0.9	0
Roll Camber (deg/deg)	0.71	0.73

2.2 Finite Element Analysis

ANSYS provides a cost-effective way to explore the performance of products or processes in a virtual environment. This type of product development is termed virtual prototyping. With virtual prototyping techniques, users can iterate various scenarios to optimize the product life before the manufacturing is started. This enables a reduction in the level of risk, and in the cost of ineffective designs.

2.2.1 Maximum Braking Conditions on Front Knuckle

The center part of upright is held fixed, Force of 2200N is applied at Upper mounting point, force of 4400 N is applied at lower mounting point, braking force of 2600N applied at brake mount, braking torque of 230Nm is applied brake mount and steering force of 700N is applied at steering arm.

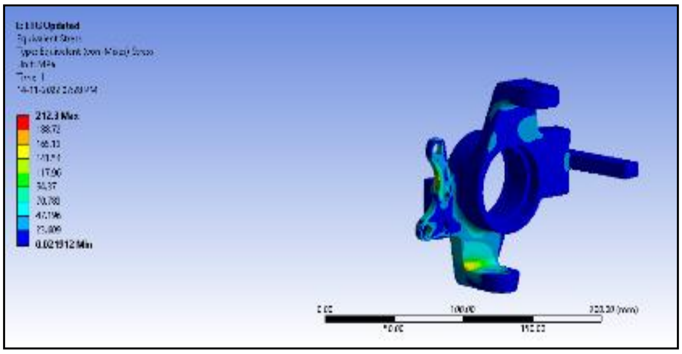

Figure 4 : Total Deformation

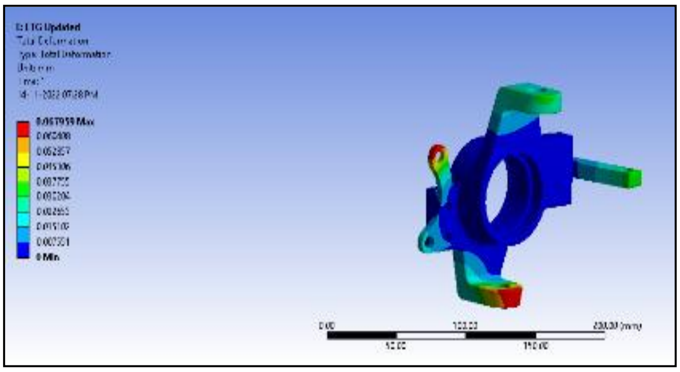

Figure 5 : Equivalent Stress

Table 8 : Results of Front Knuckle at Braking

S.No	FEA Parameters	Results
1.	Equivalent Stress	220MPa
2.	Deformation	0.07mm
3.	FOS	2.18
4.	Life	20592 cycles

2.2.2 Maximum Braking Conditions on Rear Knuckle

Upright connects the wheel hub and the suspension control arm. The Upper and lower control arm mounting points are held fixed and vertical and cornering forces are applied at the center. The maximum equivalent von mises stress obtained in the analysis is 292 Mpa.

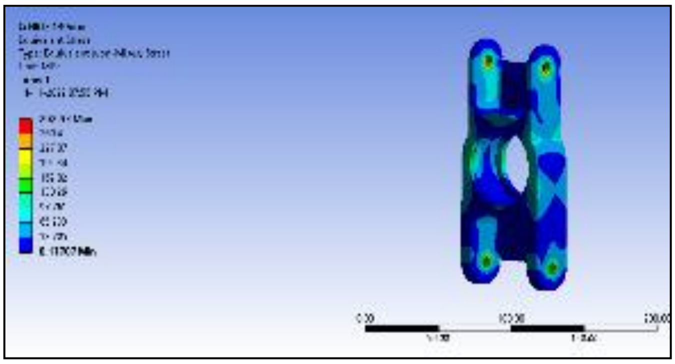

Figure 6 : Total Deformation

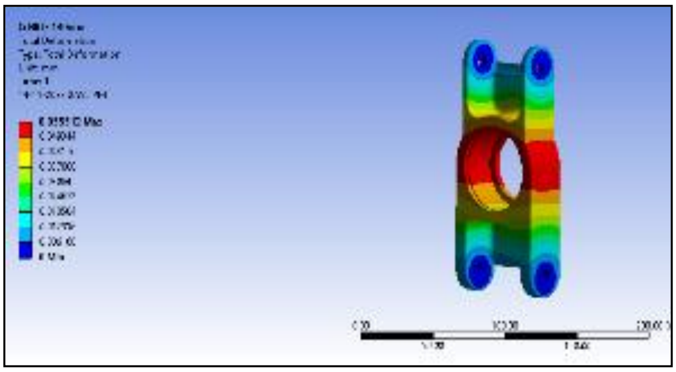

Figure 7 : Equivalent Stress

Table 9 : Results of Rear Knuckle Analysis

S. No	FEA Parameters	Results
1	Equivalent Stress	292MPa
2	Deformation	0.05mm
3	FOS	2.3
4	Life	7082 cycles

3. MANUFACTURING OF KNUCKLE

CNC (Computer Numerical Control) is a manufacturing process that has revolutionized the way products are made. It involves the use of computerized machines that can precisely cut and shape a wide range of materials, including metals, plastics, and wood. The use of CNC technology has many benefits, including increased accuracy, consistency, and speed of production.

Our product is manufactured using CNC, it means that the entire process is automated and controlled by computer software. The machine is programmed to follow a specific set of instructions, which tell it exactly what to do and how to do it. This ensures that each product is identical and meets the required specifications.

Figure 9 : Finished Knuckle

4. RESULTS

From the overall design and the analysis of the front and rear knuckle we have optimized the suspension system for better off-roading and speed cornering. And the designed system can take high impact forces which is very important for an all-terrain vehicle.

The analysis of the control arm helps to understand the behavior of the suspension system in heavy loading, braking, and cornering conditions which helps to study the system in a better way.

5. CONCLUSION

The design of a knuckle and control arm is critical to the performance, stability, and safety of a vehicle's suspension system. The knuckle and control arm work together to support the weight of the vehicle, absorb shocks and vibrations, and provide steering control and stability. The knuckle is the component that connects the wheel hub to the suspension system. It typically has a flange that mounts to the hub and a spindle that connects to the control arm. The knuckle must be designed to withstand the lateral and vertical loads that are generated during cornering, braking, and acceleration.

When designing a knuckle and control arm, several factors must be considered. These include the material selection, manufacturing processes, and the specific loads and forces that the components will encounter during operation. Finite element analysis and other simulation tools can be used to test and optimize the design, ensuring that it meets the necessary strength and durability requirements.

Additionally, the design of a knuckle and control arm should also consider factors such as ease of installation, maintenance, and repair. Access to critical components and ease of disassembly can significantly impact the cost and time required for maintenance or repairs.

In conclusion, a successful design of a knuckle and control arm requires careful consideration of many factors, including strength, durability, manufacturing processes, and maintenance requirements. A well-designed suspension system can improve the ride quality, handling, and safety of a vehicle, reducing wear and tear on tires and other components, and enhancing the overall driving experience.

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