DESIGN AND OPTIMIZATION OF STATIC CHARACTERISTICS FOR A STEERING SYSTEM IN AN ATV

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Abstract. The suspension and steering system of an ATV (All-Terrain Vehicle) plays a vital role in vehicle dynamics for off-road conditions by load transfer and providing stability. Incorrect geometry and additional mass on the front axle affect the stability and performance of the vehicle in dynamic condition. This can be optimized by varying the geometric, design and mass characteristics of the wheel assembly and suspension system parameters. CAD model of the wheel assembly and steering gear box had been developed using SOLIDWORKS, with respect to geometric parameters obtained on suspension and steering system analysis using LOTUS software with various structural and geometric constraints. Finite element analysis of the wheel assembly was done using ANSYS WORKBENCH for static and basic dynamic loading conditions. Then topology analysis has been done to reduce the additional mass of the wheel assembly without compromising its geometric and load transfer characteristics.

1. Introduction

Performance analysis of an off-road vehicle with respect to the environment of its operation is termed as terra mechanics. Driving a vehicle in free snow, desert sand and soft mud conditions can be included in off-road driving conditions. Such rugged terrain conditions may cause much threat towards driving the vehicle in those conditions and hence the behaviour of vehicle in such tough conditions depends upon the interaction of the vehicle with such harsh environments and conditions. It is therefore important to be empathetic over the vehicular mechanics in off-road conditions. In order to meet the operating necessities of the vehicle in off-road terrains, choice of design factors and configuration of the vehicle becomes significant [1]. During the past few years focus is being diverted towards proposing and developing a wheeled and tracked vehicle, using theoretical and analytical approaches, for operating in unpredictable and rough driving conditions.

Any off-road vehicle configuration can be specified by their operating power, dimensions and overall kerb weight. Some of the factors involved in selecting the off-road vehicle parameters include operation condition, cost of manufacturing and maintenance of the vehicle, operational environmental impact, reliability, economy and above all the need for its development. Virtual approach serves better for initially confining the operational environment and need of the vehicle. One such design is the design of an all-terrain vehicle (ATV) for the event of SAE BAJA.
Based on the need for the event vehicle configurations are taken as: Track width = 1250 m, Wheelbase = 1450 m, Ground clearance = 400 mm [2, 3]. Weight of the vehicle affect the overall performance of the vehicle. That also increases the mileage of the vehicle. The weight is reduced with the optimization of the components is done by the analysis using the software. The geometry of the systems will help in handling and maneuverability of terrain vehicles. Normally there are few issues in the handling of the vehicle system like direction control of the terrain vehicle and the control of the vehicle by posing resistance to the external instabilities [4]. Some of the parameters for the geometry are camber, caster, toe angle, roll center and roll axis. Caster is geometrical alignment of vertical axis of steering with respect to the steered wheel taken along the longitudinal direction. Caster angle is adjusted to optimize the handling characteristics of a vehicle. There are two types of caster one is positive caster and the other is negative caster. Positive caster provides the rate of inherent ability of steering wheel to center itself with respect to the caster of the vehicle so as to mechanically follow the path of steering axis. Such geometry renders better vehicular control in terms of direction and stability. Negative caster results in lighter and easier steering but reduced stability when driving in a straight line [5-7].

Camber can be defined as the angle between the vehicle vertical axis and the steered wheel vertical axis when viewed in the lateral direction from front or rear of the vehicle. There are two types: one is positive camber and the other is the negative camber [8, 9]. Positive camber stabilizes your car. Negative camber alters the handling qualities particularly in the cornering. By this we can maximize the contact patch area. There are two terms to be known for the cornering one is slip angle and other is force of cornering. Stability and direction control of all-terrain vehicles mostly depend on the aforementioned parameters [10]. If a side force is not induced along the direction perpendicular to the plane of wheel rotation, then it might be subjected to a force at an angle of $\alpha$ will act on the wheel due to the path of contact of the tire with the surface and is presented in Figure 1. The angle $\alpha$ is called slip angle and the force induced along the path of contact of tire at zero camber is called cornering force [11-13].

![Caster and Camber Diagram](image1)

**Figure 1.** Schematic of Caster and Camber [10]

Toe angle denotes the tire or wheel geometry angle and represents how much the wheels are turned around the vertical axis when pointed straight-on in top view and is presented in Figure 2. Toe angle is also defined as the direction of pointing of tyres with respect to the longitudinal center axis of the chassis when the view is from the top of the vehicle [14].

![Toe Angle Diagram](image2)
2. Geometry of the Model

2.1. Ackermann Steering geometry

Ackermann steering geometry is defined as the arrangement of wheels according to the geometry in such a way that it caters the needs of the inner or outer wheels to rotate at different speeds when the vehicle is turning in a curve at various radii and speed. During design of the steering mechanism scrub in tires induced by cornering should be as minimum as possible. These factors should ensure that tires of the vehicle, while turning, should be devoid of sliding but undergo only pure rolling. Hence the wheels should be placed in such a way that they follow a curved path at different curve radius with a common radius of curvature [15]. In order to satisfy this condition a simple relationship between steering angle and vehicle motion direction has to be adopted as shown in equation 1.

\[ \text{Cot } \phi - \text{Cot } \theta = \frac{w}{b} \]  

Where, \( \theta \) and \( \phi \) is the angle between inside and outside front wheel with the common center point, \( w \) is track width and \( b \) is wheel base. Substituting the values of \( \phi \) as 33.84º and \( \theta \) as 57.81º, we get

\[ \text{Cot } 33.84 - \text{Cot } 57.81 = \frac{1250}{1450} \]

\[ 0.8620 = 0.8620 \]

Thus, Ackermann condition is satisfied by the current design of steering assembly and is presented in Figure 3. The geometry is achieved with the help of lotus software as shown in Figure 4. To match the instantaneous center for our chassis design, number of iterations are made.
2.2. Instantaneous Center

The instantaneous center is called roll center. The roll center of vehicle denotes the point of corner forces acting in the suspension of the vehicle towards the body of the vehicle [16-18]. Suspension geometry is the major deciding factor of the geometric center placement in a vehicle and it is usually calculated by the concept of instantaneous center method. The SAE’s definition of the force-based roll center is, "The point in the transverse vertical plane through any pair of wheel centers at which lateral forces may be applied to the sprung mass without producing suspension roll" [19, 20]. The roll center for the geometry shown in Figure 5.

Roll axis could be defined as the line connecting the roll center at the front and rear and it is an imaginary line in side view. If the roll center height in front is lower than the rear then it is called anti dive, it will happen during the breaking. If front is higher than the lower then it is called anti squad, it will happen during the acceleration. If both the roll center height is same then it is neutral roll axis [21 -23]. Roll axis shows the mass shift from vehicle CG based on the inclination as shown in figure 5 b. It will be helpful during the turning of the vehicle to have the more grip in the front to turn in the curved path and to avoid the slip of the tires since it is an off-road vehicle which have the loose sand.

3. Design and Optimization of the component

The wheel assembly components were designed based on obtained geometry parameters obtained from lotus analysis. Then the deformation and stress analysis were done for the components using ANSYS, and optimization of the components carried out to reduce un-sprung mass without affecting load transfer characteristics.

3.1. Hub Design

The basic hub design was done based on considering parameters like PCD of rim, PCD of brake disk, Distance of wheel pivot point from wheel center (obtained from LOTUS), Bearing and stub axle size, Bolt size and strength. The analysis was done for various load acting on the wheel assembly for static condition of the vehicle.

3.1.1. Load calculations

Considering,
Radius of wheel, R = 292.1 mm
Acceleration due to gravity, \( g = 10 \text{ m/s}^2 \)

Acceleration of vehicle, \( a = 2 \text{ m/s}^2 \)

Mass of vehicle, \( M_1 = 160 \text{ kg} \)

Mass of driver, \( M_2 = 80 \text{ kg} \)

PCD of wheel, \( D = 110 \text{ mm} \)

PCD of brake disk, \( d_1 = 65 \text{ mm} \)

Brake disc diameter, \( d_2 = 150 \text{ mm} \)

Assuming, the height of bump, \( h = 100 \text{ mm} \)

Bump force [17],

\[
F_B = (M \ast a \ast \sqrt{(2Rh^2)}) / (R-h) \\
= ((160+80) \ast 2 \ast \sqrt{(2\ast292.1 \ast 100 - 100^2)}) / (292.1 - 100) \\
= 550 \text{ N}
\]

Thrust Force [17, 18],

\[
F_T = (M_1 + M_2) \ast a \\
= (160+80) \ast 2 \\
= 480 \text{ N}
\]

Torque in Rim PCD [19], \( T_R = F_T \ast (D / 2) \\
= 480 \ast (110 / 2) \\
= 26400 \text{ N-mm}
\]

Braking force at caliper, \( F_\mu = 3000 \text{ N} \)

Torque on brake disc PCD, \( T_B = F_\mu \ast ((d_2 - d_1) / 2) \\
= 3000 \ast ((150 - 65) / 2) \\
= 127500 \text{ N-mm}
\]

Load acting on one side of wheel due to mass of vehicle,

\[
F = (M \ast g) / 4 \\
= ((160+80) \ast 10) / 4 \\
= 600 \text{ N}
\]

Force acting on steering arm of knuckle = 150 N

Force distribution of load acting on wheel due to mass of vehicle, upper mount = 96 N

Lower mount = 384 N

The material for all the wheel assembly components were chosen as Al6061-T6, because it has less mass and high ultimate tensile strength considerably. The properties of the selected materials is shown in Table 1.

**Table 1.** Material properties of Al6061-T6 [20]

| PROPERTY               | VALUES          |
|------------------------|-----------------|
| Density                | 2703 kg/m³      |
| Modulus of Elasticity  | 70 GPa          |
| Poisson’s ratio        | 0.33            |
| Bulk modulus           | 68.627 GPa      |
| Modulus of rigidity    | 26.316 GPa      |
| Ultimate Tensile Strength | 310 MPa      |
The basic hub design has a mass of 0.50425 kg, for the material of Al 6061-T6 as shown in Figure 6(a). The hub was analyzed in FEA method by meshing the components into smaller parts (as shown in Figure 6(b) and then subjected to various load and torque acting on wheel assembly using ANSYS.

![Figure 6. Modelling of hub a) Mass property, b) Meshing](image)

Then total deformation and equivalent stress acting on the hub have been solved and verified for optimum characteristics. Figures 7(a) and 7(b) show the various loads acting on the hub and its deformation for the applied loads. Figure 8 shows the stresses acting on the hub.

![Figure 7. (a) Various load acting on hub and (b) Deformation of hub](image)

![Figure 8. Stress acting on hub](image)

![Figure 9. Topology optimization](image)
After analyzing the hub design, the components were optimized using topology optimization tool on ANSYS as shown in Figure 9. The optimization was done to reduce mass of the component without affecting the load transfer and geometric characteristics of hub. Then the optimized hub has been given boundary structure by modifying the hub model using SOLIDWORKS. Then the hub has been again analyzed using ANSYS for verifying load and stress characteristics. Figures 10(a) and 10(b) depict the mass property and meshing of optimized hub.

![Figure 10](image1.png) Optimized hub (a) Mass property and (b) Meshing

The mass of the optimized hub has a mass of 0.22177 kg which is about 44% of the basic model hub. Based on the above design factors and considerations and load acting, corresponding deformation of the modified hub is shown in Figures 11(a) and 11(b) respectively.

![Figure 11](image2.png) (a) Load acting on optimized hub and (b) Deformation of the optimized hub

The value of stresses acting on the modified hub remains almost the same as that of the currently used hub. This shows that the design of hub has been suitably modified without compromising the load acting on the hub. Figure 12 shows the stresses acting on the optimized hub.
3.2. Knuckle Design

The design and optimization of the knuckle is similar to that of the hub. The designing of the knuckle is based on the parameters like Upper and lower arm mounting points obtained from LOTUS, Stub axle diameter, Steering tie rod mounting point obtained from LOTUS, Brake caliper mounting point, Clearance between brake disc. CAD model of knuckle is shown in Figure 13.

The knuckle was analyzed using ANSYS by meshing the components into smaller components and applying loads on the mounting and required points of the knuckle. The mass of the non-optimized knuckle is about 0.31549 kg. The deformation and stress characteristics of the knuckle for the loading condition was found and were optimized using topology optimization tool in ANSYS. Figure 14 shows the ANSYS model of steering knuckle.

![Figure 13. Knuckle CAD model](image1)

![Figure 14. ANSYS model of knuckle](image2)

Figures 15(a) and 15(b) show the meshing of modelled knuckle and the loads acting on it. Corresponding deformation of the knuckle and stresses acting on it is illustrated in Figures 16(a) and 16(b) respectively.
Knuckle is then subjected to topology optimization which is shown in Figure 17 for reducing its mass without compromising the stresses acting on it. The optimization of the knuckle was done and suitable boundary model have been developed by SOLIDWORKS as shown in Figure 18 and analysis have been done for the same loading conditions to verify the load transfer and stress characteristics.
Figures 19(a) and 19(b) show the ANSYS model of optimized knuckle and its meshing. Load acting, deformation and stresses on the optimized knuckle is shown in Figures 20(a) and 20(b) and Figure 21 respectively.

**Figure 19.** Optimized knuckle (a) Mass property and (b) Meshing

**Figure 20.** (a) Loads acting on the knuckle and (b) Deformation of optimized knuckle

**Figure 21.** Stress acting on the knuckle
4. Conclusion

The optimization of both the geometric and physical characteristics of the suspension and the steering system has been achieved in static condition. The performance characteristics of the vehicle have been recorded for the static condition and the un-sprung mass has been reduced considerably to reduce the steering effort and improve load transfer using various software.

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