Building A Custom Steering System For All-terrain Vehicle (Agv)

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ABSTRACT

In this paper, a steering system is designed for a Baja car, which adopts a rack-and-pinion steering mechanism. The theoretical modeling of the system as well as the derivations of optimal parameter values are presented here. First, the steering angles of the front wheels are derived based on the geometry of the steering system. Second, linear equations representing the axial lines of the front wheels are derived based on the steering angles of the front wheels. Then the Ackermann steering errors are computed on the axial line of the rear wheels using the axial lines of the front wheels. Finally, the optimum values of the parameters of the steering system are obtained via computer programming such that the obtained values of the parameters minimize the Ackermann steering error on the axial line of the rear wheels.

Keywords: Steering, Camber angle, Rack and pinion, Caster angle

1. INTRODUCTION

In the literature on steering systems for vehicles there are three main steering geometries used as previously mentioned these are Ackermann, parallel, and reverse Ackermann steering geometries. These three different steering geometries greatly affect the performance of the vehicle in its ability to turn and will be discussed further with their respective advantages and disadvantages. In 2017, the BYU Baja team decided that Ackermann steering would be the best use for the vehicle. They stated: Ackermann is most useful at very low speeds and tight turns because that is when you have the least wheel slip and load transfer side to side. We maintained no less than 40% Ackermann throughout the wheel stroke, which increases to nearly 100% Ackermann as the wheels reach the end of the steering stroke.

2. MATERIALS AND METHODS

This section explains the materials and methods which are used for designing the components of the steering system for ATVs is composed of rack and pinion and upright

2.1 Design Methodology for Steering System:

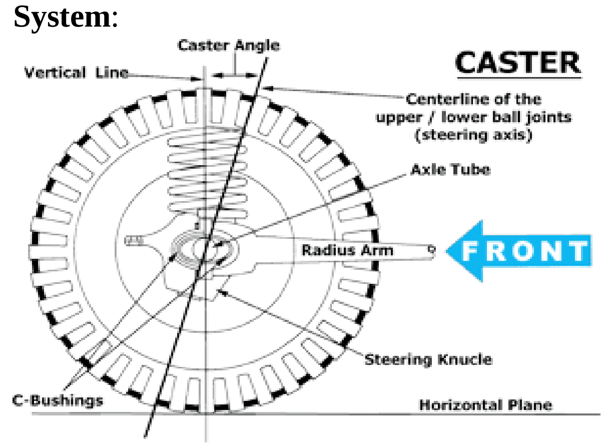


Fig.1 Caster is the angle between the steering axis and the vertical axis of a vehicle

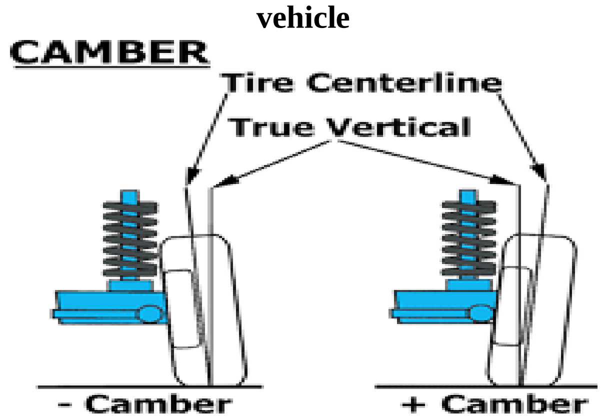
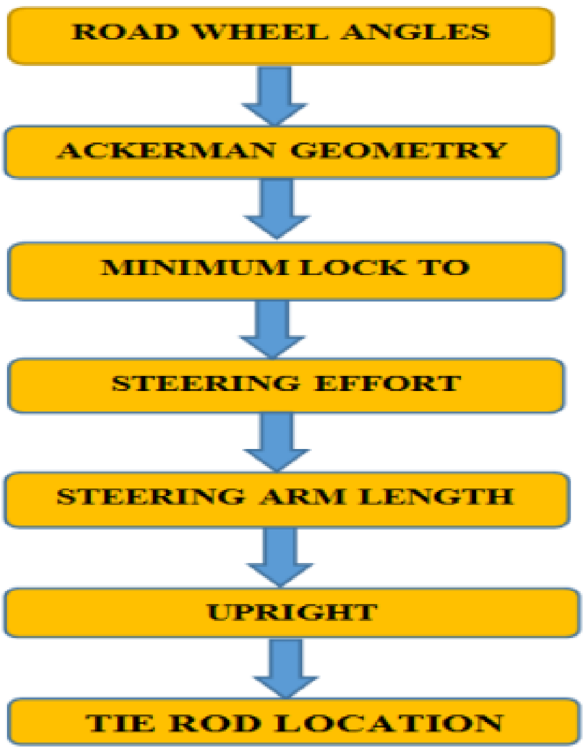


Fig.2 Camber is the angle between the

vertical axis of a wheel and the plane of its rotation

2.2 Materials



The materials used in this setup of the ATV steering system are made up of C45 steel, C45steel is readily available in the market, economical to use, has good mechanical properties, is easily machinable, and has a moderate factor of safety, high tensile strength, and low coefficient of thermal expansion This makes the material able to withstand destructive elements.

2.3 Conceptual Design

The design of this steering system is reduced and optimized to be as simple as possible without compromising its high efficiency and work output. The design makes the working of ATVs easier and the placement of components in a sufficient way to achieve efficient handling

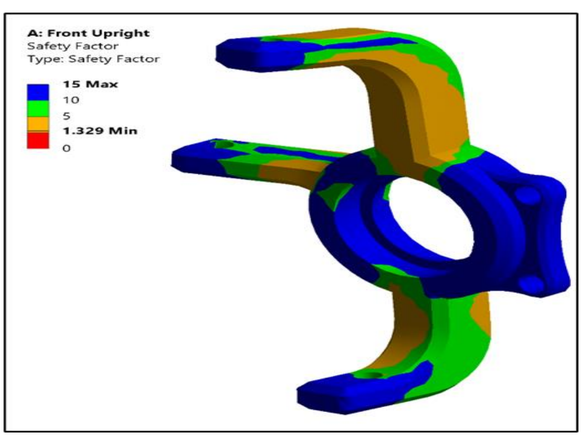


Fig.3 Shows the analysis of the front upright

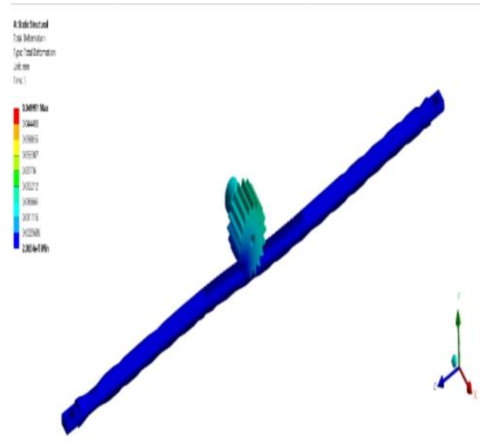


Fig.4 Analysis of rack and pinion

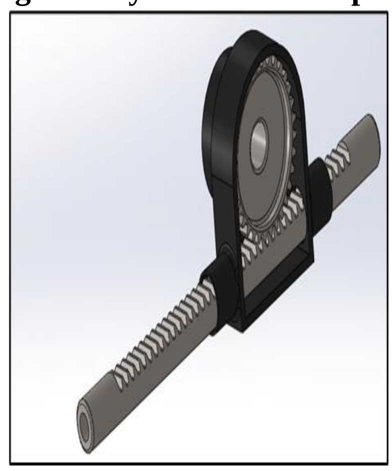


Fig.5 CAD design of rack and pinion

(Rack and pinion is a type of steering gear that is commonly used in modern vehicles)

3. RESULTS AND DISCUSSION

3.1 Completed unit

The completed unit is shown in the below figure.

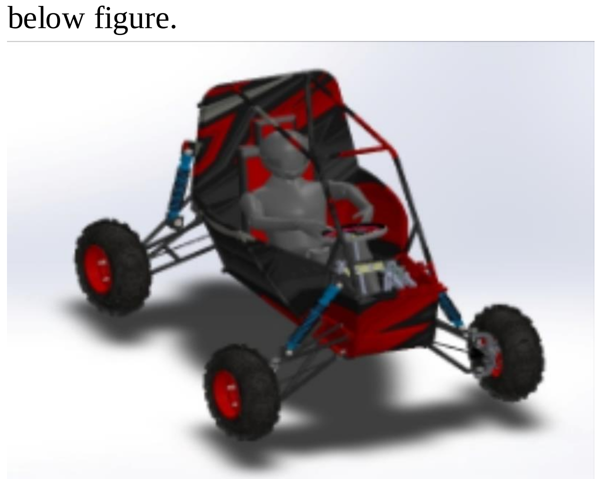


Fig.6 Shows that the 3D view of the vehicle

3.2 DESIGN VALIDATION

The CAD model from the conceptual design of material handling AGV was subjected to stress, strain, and displacement to determine the vehicle's withstanding capability, which was carried out by finite elements using the tool SOLIDWORKS simulator. The load applied is 490 N (50 kg)

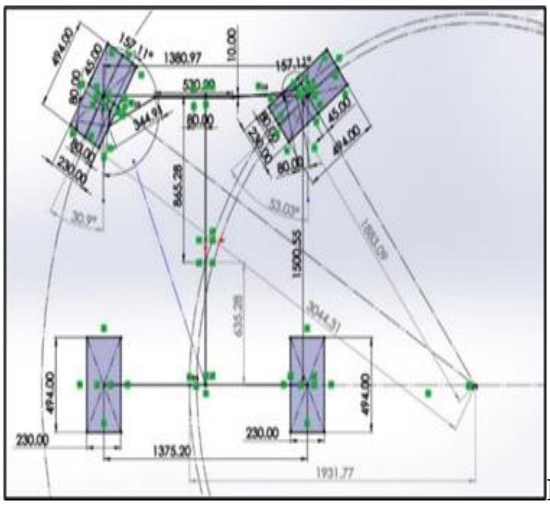


Fig.7 shows the angle calculation diagram of the **ATV (**All-Terrain Vehicle**)**

3.3 Calculations:

3.3.1Known parameters:

Wheel base=50in

Track width=48in

Center of gravity for the vehicle

Wheel's inner and outer angles

Θ_i=48deg

Θ_o=28.2deg

R=1.8m

3.3.2 Lateral force

(F_y) = (mass of vehicle *(velocity)^2) ÷ turning radius **F_y=43,227.16-N**

3.3.3) M-Trial

Wheel radius *sin 90 - head angle

Wheel radius=12in =0.3048m

M-Trial=0.3048

3.3.4) Lateral torque

Lateral force * M-Trial

13,175.63-Nm

Traction Force =coeff of friction *mass of vehicle *acceleration

- Coeff of friction =0.6
- Mass of vehicle =280 kg
- Acceleration=3.48

Traction force=584.64-N

3.3.5) Traction torque

Traction force *scrub radius (by lotus software) =**40,270 Nm**

3.3.6) Force on Tie Rod

Total torque ÷ steering arm length

Steering arm length=100m

Force on tie rod =**534.45 N**

3.3.7) Tie Rod inclination (α) =**28.6 deg**

3.3.8) Force on Rack =Force on Tie rod * cos α =**469.23 N**

3.3.9) Torque on pinion =Force on Rack * Pinion radius

Pinion radius =0.00638 m

Torque on pinion =Torque on steering =**2.993 Nm**

3.3.10) Effort on the steering wheel

Torque on steering wheel ÷ steering wheel radius

Steering wheel radius=**0.15m**

Effort on steering wheel=**19.95 Nm**

TRACK WIDTH (front) = 54 inches = **1371 mm**

WHEEL BASE = 50 inches = **1270 mm**

ACKERMAN ANGLE (inner wheel) θ_i= **40**°

This inner wheel angle is fixed to the desired condition, then the outer wheel angle is calculated.

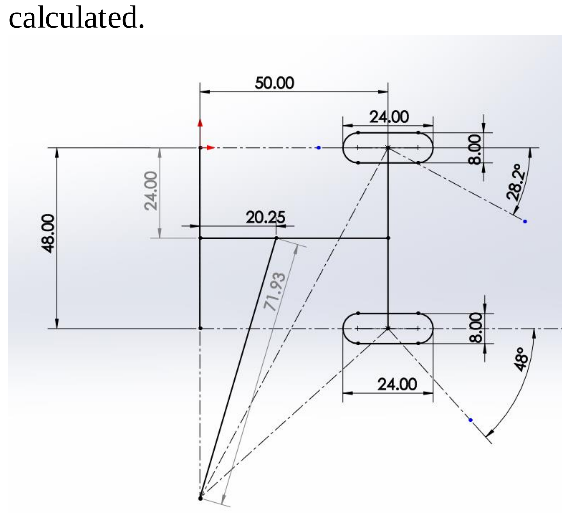


Fig.8 shows the angle calculation diagram of the **ATV (**All-Terrain Vehicle**)**

3.3.11) To calculate the outer wheel angle

We have

$1 / \tan \theta_o = 1 / \tan \theta_I + L / B = 1 / \tan (40) + 1371 / 1270= 1.19 + 1.08 =$ **2.27**

$\tan \theta =$ **0.44**

$\theta = 23.7°$

TURNING RADIUS CALCULATION:

$R_1 = B / \tan \theta_I + L / 2$

$= 1270 / \tan 40 + 1371 / 2$

$= 1513.53 + 685.5$

$= 2099.03$ mm

R_1 = 2.1 m

$R = [(R_1)^2 + (B)^2]$ ^1/2

$R = [(2099.03)^2 + (1270)^2]$

R = 2453.33 mm

R = 2.4 m

R_o (Distance from center of rotation and front outer wheel)

By Pythagoras' theorem,

$R_o = [(2784.53)^2 + (1270)^2]$

R_o = 3060.48 mm

$R_o = 3.1$ m

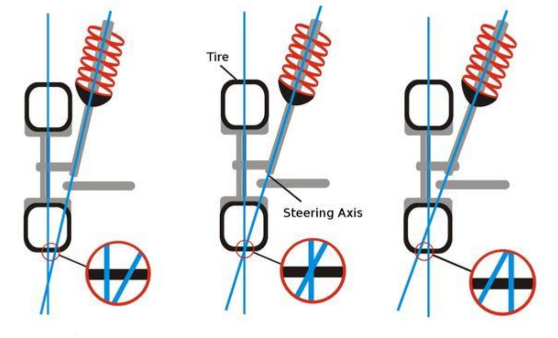


Fig.9 Shows that the scrub radius of the steering system

4.0 CONCLUSIONS

This thesis has explored the two common geometries of steering which are Ackermann and parallel steering geometries. It was found that Ackermann steering has been proven to be the best option for static loads for a smaller turn radius. For parallel steering, the literature recommends that parallel steering geometries are better for dynamic turning because it causes less of a slip angle. However, this was proven to not apply to the BYU Baja vehicle, because of the low coefficient of friction over dirt that causes a higher slip angle. During high-speed turning, the Ackermann steering was found to perform better. To maximize the performance of the BYU Baja vehicle Ackermann steering has been determined to be the best option to use. More testing can be done to prove that parallel steering can have a smaller turning radius during high-speed turning on asphalt to validate more research into other opportunities for adjustable steering geometries like a hybrid option. The next section includes some of the preliminary ideas of how a hybrid option could work for the BYU Baja car, and solve the problem of deciding if the car should be designed for high or low-speed turning. 5.1 Future work the proposed hybrid option is to allow parallel steering at

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