Design and Analysis of Independent Suspension System for an All-Terrain Vehicle

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Abstract

An All Terrain Vehicle (ATV), as the name defines is a vehicle designed to handle a more variety of terrain than most other vehicles. Over the period ATVs have been used for performing number of applications ranging from military to desert, jungle safaris, hilly areas. The efficacy of an ATV is determined on the basis of its ability to sustain the irregularities of the terrain with ease. The system that implements the vital role of damping such undesired commotion and vibrations is the suspension system. The suspension systems are mainly classified as dependent suspensions and independent suspensions. The study in this paper is carried out on the topic design and analysis of independent suspension system for an all-terrain vehicle(Double wishbone A-Arms and H-Arms). The purpose of this study is to design an independent type suspension system which can rely on any type of off-roading conditions. The focus of the study is to provide comfort to driver by reducing the effects of bumps and also improving the dynamic parameters of the vehicle. Suspension system is an important part of any vehicle, as it helps in resisting the unwanted forces so that driver feel comfortable while driving.

An independent suspension allows wheels to rise and fall on their own without affecting the opposite assembly wheel. The front suspension and rear suspension of a vehicle can be different as per the application requirements. The paper focuses on the designing, analysis and simulation of an ATV suspension system which was designed for a national level event namely BAJA SAEINDIA.

Keywords: *simulation, ansys, solidworks, lotus suspension, off-roading, bumps, all-terrain.*

Introduction

1 Automobiles were initially developed as self-propelled versions of animal drawn vehicles. However animal drawn vehicles had been designed for relatively slow speeds and lacked a suspension system that could withstand the higher speeds permitted by internal combustion engines. The suspension of modern vehicles need to satisfy a number of requirements whose aims partly conflict because of different operating conditions such as loaded/ unloaded, acceleration/ braking, constant/ variable terrain road, straight running/ cornering. For the purpose of ensuring the optimum handling characteristics of the vehicle in a steady state as well as in a transient state, the wheels must be in a defined position with respect to the road surface for the purpose of generating the necessary lateral forces. The build-up and size of the lateral wheel forces are determined by specific toe and camber changes of the wheels depending on the jounce and movement of the body as a result of the axle kinematics(roll steer) and operative forces(compliance steer). This makes it possible for specific operating conditions such as load and traction to be taken into consideration. By establishing the relevant geometry and kinematics of the axle, it is also possible to prevent the undesirable diving or lifting of the body during braking or accelerating and to ensure that the vehicle does not exhibit any tendency to over-steer and displays predictable transition behaviour for the driver. Suspension System is generally defined as the set of various mechanical linkages arranged in a manner to obtain maximum contact patch between road surface and tires. When we talk about the suspension system it means the broad subject of how the unsprung mass of a vehicle is connected to the sprung mass. These connections not only dictate the path of relative motion, they also control the forces that are transmitted between them. Suspension System helps in controlling the kinematic parameters which are responsible for dynamic behaviour of the vehicle such as Camber,

Caster and Toe angles. By giving positive Caster angle, the self-returning action of Steering can be achieved and controlling other parameter leads to better dynamic behaviour of vehicle.[1]

Negative camber angle produce higher lateral forces to improve the cornering ability of the vehicle. Moreover, Toe in angle of the wheels help in improving the straight-line stability whereas the toe-out angle help in improving the cornering stability. Hence for proper suspension travel, first the alignment and handling characteristics should be right then only suspension will work properly.

1. Suspension Parameters

1.1. **Undamped System:** Undamped systems are those in which there are no forces opposing the vibratory motion to dissipate energy.

1.2. **Damped System:** Damped systems are those in which energy is dissipated by forces opposing the vibratory motion.

1.3. **Damping Ratio:** Damping ratio is the ratio of the amount of viscous damping present in a system to that required for critical damping. Where comfort takes priority over performance, leading to low damping ratios, and minimum pitching over bumps. Racecars in general run higher damping ratios, and have a much smaller concern for comfort, leading to some racecars using higher front ride frequencies. The higher damping ratios will reduce the amount of oscillation resultant from road bumps, in return reducing the need for a flat ride.

1.4. ***Sprung weight and un-sprung weight*:** All weight which is supported by the suspension, including portions of the weight of the suspension members are regarded as sprung weight. Un-sprung weight includes the suspension upright and all components attached to it; the brake caliper, brake disc, wheel, tire and a portion of suspension arms. Sprung weight is protected

from the shocks and vibrations that the wheels experience as they travel over every bump and pothole. This makes for a more comfortable ride and protects the sprung components from destructive and life-shortening shocks and vibrations. Conversely, un-sprung weight must be designed to be tough enough to survive the constant shocks and vibrations, which can be difficult for complex parts such as wireless pressure sensors. In general, it's best to have a high ratio of sprung to un-sprung weight. A higher proportion of sprung weight can then push down on the wheels and tires with more force, keeping them in contact with the pavement or whatever surface they are traveling across. Maintaining contact with the roadway improves handling and traction, and this becomes more of an issue for off-roading and traveling over rough roads. So as a rule, designers try to minimize un-sprung weight to improve handling and steering.

1.5. **Spring Rate***:* The change of load on a spring per unit deflection is spring rate. To minimize the pitching motion of vehicle, the equivalent spring rate and the natural frequency of the front end should be slightly less than those of the rear end. This ensures that both ends of the vehicle will move in phase within a short time after the front end is excited.

1.6. **Camber Angle***:* Camber is the angle of the wheel relative to vertical line to the road, as viewed from the front or the rear of the car. If the top of the wheel is farther out than the bottom (that is, away from the axle), it is called positive camber; if the bottom of the wheel is farther out than the top, it is called negative camber. The cornering force that a tire can develop is highly dependent on its angle relative to the road surface, and so wheel camber has a major effect on the road holding of a car. A tire develops its maximum lateral force at a small camber angle. This fact is due to the contribution of camber thrust, which is an additional lateral force generated by elastic deformation as the tread rubber pulls through the tire/road interface.

1.7. **Caster Angle:** It is the Forward or rearward inclination of the steering axis.
1.7.1. **Positive Caster***:* Positive caster is when the steering axis is in front of the vertical. The purpose of this is to provide a degree of self-centering for steering the wheel casters around, so as to trail behind the axis of steering. This makes a car easier to drive and improves its directional stability (reducing its tendency to wander). Excessive caster angle will make the steering heavier and less responsive, although, in racing, large caster angles are used to

improve camber gain in cornering. Power steering is usually necessary to overcome the jacking effect from the high caster angle.

1.7.2. **Negative Caster***:* Negative caster is when the steering axis is behind the vertical. This is generally only found on older vehicles due to tire technology, chassis dynamics, and other reasons. Modern vehicles do not use negative caster. It will lighten the steering effort but also increases the tendency for the car to wander down the road.

1.8. **Toe Angle:** When a pair of wheels is set so that their leading edges are pointed slightly towards each other, the wheel pair is said to have toe-in. If the leading edges point away from each other, the pair is said to have toe-out. The amount of toe can be expressed in degrees as the angle to which the wheels are out of parallel, or more commonly, as the difference between the track widths as measured at the leading and trailing edges of the tires or wheels. Toe settings affect three major areas of performance: tire wear, straight-line stability and corner entry handling characteristics.

1.9. **Jounce(Bump):** Jounce is the upward movement or compression of suspension components. During bump, the dampers and springs absorb the upward movement from cornering or road irregularities (the springs store some of the energy), the dampers then goes into rebound. If there isn't enough damping then the cycle begins again until the car returns to the original ride height, with a bouncing motion to the car. Another trait of under damping is that loads go into tire and suspension relatively slowly, this combined with the bouncing effect means a constant varying downward force on tires. It is important to have enough bump stiffness to be able to deal with uneven surfaces. If there is too much damping, then it is effectively like running no suspension and any upward motion will be transmitted directly to the chassis. Over damping will result in an increase in the loads acting on the suspension and the tires. The handling will feel very harsh and hard, this will affect street driving in terms of comfort levels, this might not be desired for a daily driver. It is undesirable in both under and over damping settings, as it will reduce the handling of the car and will affect acceleration, braking and cornering loads.

1.10. **Rebound (Droop):** Rebound is the downward movement or extension of suspension components. During rebound (following the bump compression phase) the dampers extend back to their original positions, using up the stored energy from the springs. The rebound stiffness needs to be set at a higher value than the bump setting as the stored energy is being released. If there is no effect of damping on the rebound, the wheel will quickly return through the static level and start to bump again, with the bouncing effect unsettling the suspension with little control. If there is too much rebound stiffness, then the wheel could hold longer in the wheel arch than needed, effectively losing contact with the road as the force to push the wheel back down is slower to respond to the changing surface levels. This state is again far from ideal and it is best to make sure a good level is set for optimal tire contact with road.

1.11. **Suspension Roll***:* The rotation of the vehicle sprung mass about the x-axis with respect to a transverse axis joining a pair of wheel centers.

1.12. **Suspension Roll Gradient:** The rate of change in the suspension roll angle with respect to change in steady-state lateral acceleration on a level road at a given time and test conditions. The main factors affecting suspension roll gradient are instantaneous center and tire data for inclination angles.

1.13. **Roll Camber:** The change in camber of the wheels due to relative motion of sprung mass with respect to un-sprung mass is called as roll camber. It is the result of suspension roll. The basic wishbone is such as to give about 50% compensation of roll camber by the basic geometry and 50% by the action of the extra links, so that camber change is optimum.

1.14. **Roll Center and Roll axis***:* The roll axis is the instantaneous line about which the body of a vehicle rolls. Roll axis is found by connecting the roll center of the front and rear suspensions of the vehicle. Assume we cut a vehicle laterally to disconnect the front and rear half of the vehicle. Then, the roll center of the front or rear suspension is the instantaneous center of rotation of the body with respect to the ground.

1.15. **Motion Ratio***:* Motion ratio in suspension of a vehicle describes the amount of shock travel for a given amount of wheel travel. 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. One should know the desired wheel travel of the vehicle before calculating motion ratio which depends much on the type of track the vehicle will run upon.

1.16. **Wheel travel***:* Wheel travel is the distance that is designed for the wheel/tire assembly to move vertically without bottoming out either at the top or the bottom of the motion. The suspension's job is to ensure that despite any bumps or droops, the vehicle stays as leveled and smooth as possible. While wheel travel is measured on both independent and solid axle suspensions, it is mostly referred to with independent suspension where the vertical travel of each wheel/tire assembly is separate from any of the others, and therefore best able to provide that smooth ride and safe driving experience.

*1.17***.** **Anti-dive***:* Anti-dive is a suspension parameter that affects the amount of suspension deflection when the brakes are applied. When a car is decelerating due to braking there is a load transfer off the rear wheels and onto the front wheels proportional to the center of gravity height, the deceleration rate and inversely proportional to the wheelbase. If there is no anti-dive present, the vehicle suspension will deflect purely as a function of the wheel rate. This means only the spring rate is controlling this motion. As anti-dive is added, a portion of the load transfer is resisted by the suspension arms. The spring and the suspension arms are sharing the load in some proportion. If a point is reached called “100-percent anti-dive”, all of the load transfer is resisted by the suspension arms and none is carried through the springs. When this happens there is no suspension deflection due to braking and no visible brake dive. There is still load transfer onto the wheels, but the chassis does not pitch nose down. The method to achieve anti-dive is controlled by upper and lower control arm pivot points on chassis.[3]

2. Design

The design phase involves variety of fronts leading to one absolute design. Firstly, It involves determination of desired system characteristics. Lotus Engineering Suspension Analysis software was used to design and analyse the vehicle suspension hard points to achieve the required suspension characteristics. The Computer Aided Designing (CAD) was done using SOLIDWORKS V16 with design for manufacturing and assembly considerations. Then using ANSYS 15.0 we performed structural analysis of suspension systems in order to assure the flawless performance of our design.

2.1. **Design Targets**

2.1.1. To isolate high amplitude obstacles by increasing travel.

2.1.2. To maintain the undamped natural frequency from 1.2Hz- 1.5Hz.

2.1.3. To implement anti-dive geometry.

2.1.4. To minimize the chassis roll by maintaining the roll gradient in the range of 1.50-20/g.

2.2. **Lotus Suspension Analysis(LSA***):* LSA is a design and analysis tool that can be used for both the initial layout of a vehicle and suspension hard points. Models are created and modified through a 3d-viewing environment. This allows hard points to be ‘dragged’ on screen and graphical/numerical results updated in ‘real time’. A template-based approach to the modelling allows users to create their own suspension models, supplementing the ‘standard’ suspension templates provided. Any number of results can be displayed graphically, (e.g. Camber angle, Toe angle), against bump motion, roll motion or steering motion. These results are updated in ‘real time’ as the suspension hard points are moved.[2]

2.3. **Front Suspension:** We selected Short Long Arm (SLA) type wishbone front suspension. This suspension system consists of upper and lower wishbones. Both the wishbones were designed in A-arm shape with its narrow end joining the knuckle with ball joints and the wishbone was connected to the chassis at two pivots with heim joints. The suspension system was designed in such a way that the tires remain oriented properly in all modes of motion. This design provided an optimized wheel control and maximum camber gain during cornering. The system is equipped with FOX progressive air shocks which gives us a better performance due to its variable stiffness. The system allows adjusting the geometry of the arms and their mounting locations for fine tuning of performance characteristics.

2.4. **Rear Suspension***:* The H-Arms (TA) were opted for rear suspension system. The system consisted of H- arm along with camber link. The H-arm had one end attached to the chassis with heim joint and other end attached at knuckle with pivot system using nuts and bolts. The camber link provide the ability to control the undesired camber change by restricting the lateral motion of the upper and ends of the wheels. The two camber links are connected one at the lower end of the TA and other at the upper end of the TA, also the respective opposite ends are joined to the chassis. Camber link is joined at each end with heim joints. The trailing arms design provides better wheel travel and has the advantages of being durable, strong and easy to design for a desired amount of travel and static camber.

I. DESIGN PARAMETERS

Before designing the suspension system, some parameters are needed to be known. Hence type of independent suspension system which are going to be used in front and rear side of vehicle are –

Front - Equal & Parallel Double Wishbone suspension system is used as it offers easy controls on the various kinematic parameters.

Rear - Trailing Arm is used as it produces steering effect as it passes through bump or jounce and hence generate roll understeer on rear axle.[4]

TABLE 1. SUSPENSION PARAMETERS

S.No.	**Design Parameter**	**Value**
1.	Wheel Base	1295.4 mm
2.	Track Width	1346.2 mm
3.	Total Weight	240 Kg
4.	Tire Size	22*7 inch
5.	CG Height	635 mm
6.	Static Camber	0 deg.
7.	Caster Angle	0 deg.
8.	Ground Clearance	13 inch
9.	Kingpin Inclination	4 deg.
10.	Kerb Weight	179.8 kg
11.	Kingpin Length	5.5 inch
12.	Motion Ratio	0.75

These are some of the following design parameters which are required for designing the independent suspension system.

A. Front Suspension System

Double Wishbone suspension system was preferred as it consist of two control arms, one upper and other lower on which suspension is mounted and whole system gets attached to the chassis.

This type of suspension system is easy to manufacture as well as possess easy controlling over kinematic parameters and helps in improving the ride quality. And Equal Parallel combination helps in minimizing the camber gain in the front tires whenever vehicle goes in bump or droop.

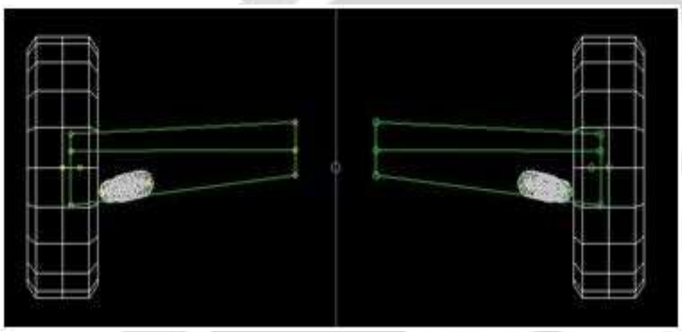
This type of system is connected to tires through wheel assembly which comprise of Hub, spindle and knuckle along with the Ball joints which connects control arms to the wheel assembly. To chassis side, lower control arm connected through bushings whereas upper arm is through ball joints to adjust the kinematic property.[5]

B. Rear Suspension System

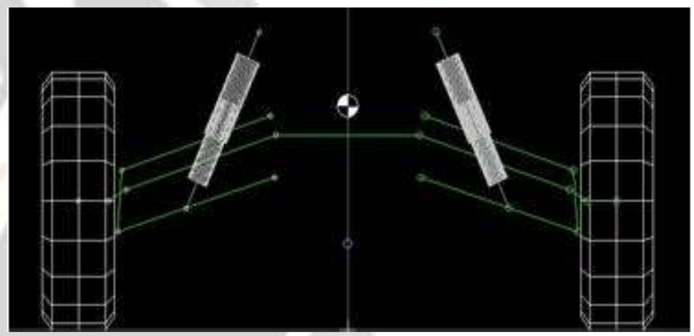
In the H-Arm system , one end of the arm is connecting to the locations of the car body; another end connects two locations of the wheel hub . The H-arm alone can limit 4 degrees of freedom, so it need another horizontal control arm to limit one more degree to make the whole system with 1 degree of the freedom.

One the other hand, with proper design enhancement which creates variation of the H-arm system, superior handling performance can be achieved. In fact some luxury cars and high –end exotic sports cars use such design.

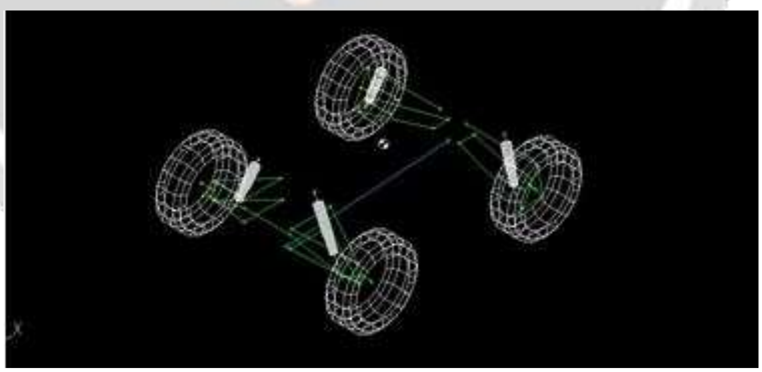
For example Audi changes one of the wheel hub H-arm connections with flexible joint, which release one degree of freedom (makes the H-arm limits 3 degrees of freedom),and compensate it by one extra control arm. Seethe below image for reference



Rear suspension geometry on lotus



Front suspension geometry on lotus



Isometric view of suspension linkage on lotus

METHODOLOGY

Methodology in this study consist of various sets of calculation which are required during analysis of suspension system.

A. Front Axle Weight to Rear Axle Weight Ratio

Front axle weight (FAW) and Rear Axle Weight (RAW) is needed to be calculated for obtaining the required spring rate of suspension. The total weight acting at the CG of vehicle with 70kg driver seated is 279 kg.

And the total sprung mass of the vehicle is 210.8 Kg

Taking moment about rear axle-

W- Total sprung mass at CG

W_{FR}- Weight on front axle

W_{RR}- Weight on rear axle

$$W*20 = W_{FR}*51$$

$$W_{FR}= (150*20)/ 51$$

$$W_{FR}= 58.82 \text{ Kg}$$

Hence Weight at rear axle is

$$W_{RR}= W-W_{FR} = 150-58.82$$

$$W_{RR}= 91.18 \text{ kg}$$

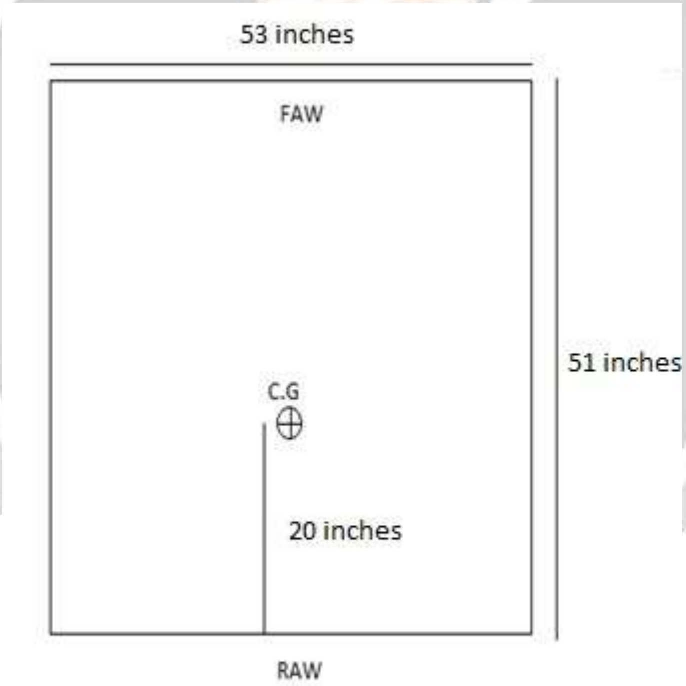


Fig. 6 Weight Distribution

It is taken that under full bump condition the suspension are providing maximum 5” travel.

B. Force Analysis

To calculate the amount of force acting on the suspension components, two methods are used i.e.

1) Drop test is performed under which Energy Conversion took place, kinetic energy of vehicle changes into potential energy.

2) **Impulse Momentum Method:** Drop test is performed from a height of 6 feet, and made to fall under gravity hence energy conversion takes place.

$$\frac{1}{2}mv^2 = mgh$$

$$v = \sqrt{2gh}$$

$$v = 6 \text{ m/s}$$

Now, calculating the time taken in free falling,

$$s = ut + \frac{1}{2}gt^2 \qquad \text{(u=0 m/s)}$$

$$t = 0.61 \text{ sec.}$$

Applying Impulse Momentum Equation,

$$mv = F * t$$

$$F = 2360.5 \text{ N}$$

This much amount of force will act on the vehicle when it made to free fall under gravity.

C. Spring Rate

Spring Rate is defined as amount of force required to produce a unit mm deflection in spring.

As the suspension supports the sprung mass of the vehicle, hence ratio of sprung mass on front and rear axle defines the spring rate for front and rear suspension.

K_f = Front Spring Rate

K_R = Rear Spring Rate

F_f = Bump Force on front axle (2.5G force)

F_R= Bump Force *on Rear axle (2.5G force)*

x = Maximum suspension travel

$$K_f = \frac{F_f}{x}$$

$$K_f = \frac{2.5 * 9.81 * 58.82}{4.5 * 25.4}$$

$$\boldsymbol{K_f = 12.62 \text{ N/mm}}$$

$$K_R = \frac{F_R}{x}$$

$$K_R = \frac{2.5*9.81*91.18}{5*25.4}$$

$$\boldsymbol{K_R = 17.60 \text{ N/mm}}$$

Rear Spring Rate is greater than front because at rear axle percentage of sprung mass is greater as compared to front hence higher stiffness is required to support the rear axle weight.

D. Sprung Mass Natural Frequency

w = sprung mass natural frequency

K_f = Front Spring Rate

Kr = Rear Spring Rate

K = Total Spring Rate
M = Total sprung mass

It can be assumed that front and rear suspension are attached in parallel to each other. Hence the net spring rate will be $K = K_f + K_r$

$$K = 12.62 + 17.60$$

$$K = 30.22\ N/mm$$

$$w = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

$$w = \frac{1}{2\pi}\sqrt{\frac{30.22 * 1000}{240}}$$

$$\boldsymbol{w = 1.78\ Hz}$$

3.1.5 WHEEL RATE:

$$(W.R.)_f = (M.R.)^2 * K_f$$

$$(W.R.)_f = (0.75)^2 * 12.62$$

$$\boldsymbol{(W.R.)_f = 7.09\ N/mm}$$

$$(W.R.)_r = (M.R.)^2 * K_r$$

$$(W.R.)_r = (0.75)^2 * 17.60$$

$$\boldsymbol{(W.R.)_r = 9.9\ N/mm}$$

3.1.6 ROLL RATE:

$$K_{\emptyset f} = Front\ Roll\ Rate$$

$$K_{\Phi r} = Rear\ Roll\ Rate$$

$$t = track\ width$$

$$K_{\emptyset f} = \frac{12 * K_f * t^2}{2}$$

$$K_{\emptyset f} = \frac{12 * 12.62 * 5.33^2}{2}$$

$$\boldsymbol{K_{\emptyset f} = 2151.12\ lb\text{-}ft/rad}$$

$$K_{\emptyset r} = \frac{12 * K_r * t^2}{2}$$

$$K_{\emptyset r} = \frac{12 * 17.60 * 5.33^2}{2}$$

$$\boldsymbol{K_{\emptyset r} = 2999.97} \textbf{\textit{ lb-ft/rad}}$$

3.1.7. ROLL GRADIENT

W = Total weight in lbs.

H = Distance between roll axis and CG

$K_{\Phi f}$ *= Front Roll Rate*

$K_{\Phi r}$ *= Rear Roll Rate*

$$\frac{\emptyset}{A_y} = \frac{-W * H}{K_{\emptyset f} + K_{\emptyset r}}$$

$$\frac{\emptyset}{A_y} = \frac{-529.109 * 0.89}{2151.2 + 2999.97}$$

$$\frac{\emptyset}{A_y} = -0.091 \text{ rad./g}$$

$$\boldsymbol{\frac{\emptyset}{A_y} = -5.23 deg./g}$$

3.1.8. DAMPER STROKE AT 0.3m/s

Vs=bump velocity

F_n =natural frequency

Z_h= displacement amplitude

Damper stroke is defined as the distance between the extreme positions of damper and it is twice of displacement amplitude

$$V_s = 2\pi fn \times Z_h$$

$$Z_h = 0.3/(2\pi \times 1.78)$$

$$Z_h = 26.82 \text{ mm}$$

$$\text{Stroke} = 2 \times Z_h$$

$$\text{Stroke} = 53.64 \text{ mm}$$

3.1.9. DAMPING COEFFICIENT

Damping coefficient at 0.3 m/s

$$F_D = C_D \times V_Z$$

Where, V_Z = damper velocity

F_D = damping force

$$V_z = (M.R.) \times V_S$$

$$V_z = 0.75 \times 0.3$$

$$V_z = 0.225 \text{ m/s}$$

3.1.9.1 Front Damping Coefficient

$$C_D = \frac{F_D}{V_z}$$

$$C_D = \frac{2.5 * 9.81 * 58.82}{0.225}$$

$$C_D = 6.411 \text{ N.s/mm}$$

3.1.9.2 Rear Damping Coefficient

$$C_D = \frac{F_D}{V_z}$$

$$C_D = \frac{2.5 * 9.81 * 91.18}{0.225}$$

$$C_D = 9.938 \text{ N.s/mm}$$

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