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Design & analysis of solar car suspension system

Rushikesh patil[1], Swapnil admane[2], Yogesh bhosle[3], Ganesh koli[4], Prof.Ashish patil[5]
[1,2,3,4,5] *Padmabhooshan Vasantraodada Patil Institute of Technology, Budhgaon, Sangli*

***Abstract*— In this paper, our main emphasis is on providing various aspects which are required in the design of suspension system. This paper consists of information about the evolution of suspension system. Basic terminologies required such as toe angle, camber angle, caster angle etc. This paper also includes kinematic and dynamic analysis of a suspension system with a detailed design procedure and analysis of the components using CAE software such as ANSYS for further optimization.**

I. INTRODUCTION

Suspension is a necessary system for solar cars because it protects the frame and other on-board components from large jolts encountered along highways. Suspension system absorbs the unnecessary vibration which applied on the upward from road irregularities. If the suspension is too soft, energy is wasted by absorbing the motion of a car as it travels over bumps. For increased efficiency, most solar cars use a suspension that is stiffer than normal.
Suspension is the most vital sub-system in an automobile. Its main functions are load transfer to the wheels and protection of the driver from road shocks. The purpose of this paper is to select suitable suspension system for the front and the rear of an SOLAR CAR with rear electric drive and to thereafter design, analyze, simulate and test the suspension systems for optimum performance of the vehicle, driver safety and maximum driver comfort.
However, recent passenger vehicles prefer independent suspension system for better drive comfort. The suspension systems can be categorized into:
Independent suspension system
a) Macpherson suspension
b) Double wishbone suspension

Dependent suspension system
a) Leaf spring
b) Trailing arm

In future, there are many new suspension technologies which can be actualized such as Dynamic bend tilting, Magnetic damper and Hydraulic move control etc.

II. THEORY

The suspension is certainly one of the most vital systems for the proper car functioning. It is not only responsible for absorbing external vibrations providing comfort to the passengers, but also for shock softening, protecting all the mechanical parts while sustains the entire vehicle weight, maintaining the tires in firm contact with the road enhancing propulsion and safety.
The basic terminologies used in suspension system are-
1. Toe Setting
Toe setting strongly affects the handling characteristics and transitional cornering of the vehicle. It is the difference between the front and rear edges of the wheels. Toe-in means the front edges are closer together than the rear edges and the wheels point inward. Toe-out means the front edges are farther apart than the rear edges and the wheels point outward. Extreme toe-in or toe-out will cause excessive tire wear and steering instability, especially at high speeds.

2. Camber Angle
Camber angle is measured in degrees. It is the angle at which the wheels are angled i.e. the angle between the centerline of the tire and a vertical line. Extreme positive camber causes wear on the outside of the tire.

3. Caster Angle
Caster is the backward or forward slope of the steering axis when viewed from the side. Caster angles are used to alter the directional stability of the

wheels. Proper caster will also help to keep the vehicle in a straight line at high speed.

4. Kingpin Inclination

It is the inclination concerning the swivel pin axis to the vertical is known as kingpin inclination. On current suspension systems, the kingpin is set at an angle to the vertical plane when viewed from the front or rear of the vehicle.

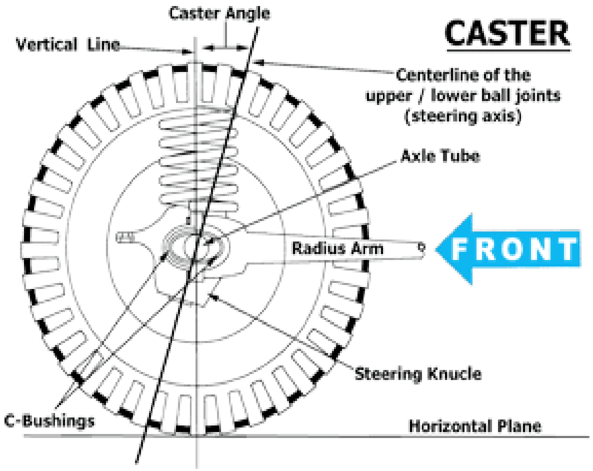

Figure-1

III. DESIGN & CALCULTION

Total Weight of vehicle = 350 kg

Wheel Base (L) =1700 mm(66.92 Inch)

CG Location in longitudinal plane is calculated by =

C=Wf * L/W

Where ,

C = Distance of C.G. from rear wheel Wf = Weight on front Wheel

Wr =Weight on rear wheel

In static condition weight distribution is 45:55 , 45% on front wheel & 55% on rear wheel

Wf = 157.5kg Wr =192.5kg

C=Wf*L/W =157.5*66.92/350

C =30.11 Inch

- Center of Gravity Height:-

It is calculated as follows, H= Δ Wf*L/Wtan θ

Where,

H = Height of C.G. from ground level

Δ Wf =Change if Wf , for θ = 0 for test value

ɵ = Angle to which vehicle is raised from rear

For Better approximation we take θ value from 25 to 30˚& calculate C.G and average them. For θ =30˚

H= Δ Wf*L/Wtan θ

=78.75*66.92/350*tan30

H=26.07inch

- Longitudinal weight transfer

Load transfer during acceleration

WL=W/g*ax*H/L

Where, WL=Longitudinal weight transfer

WL=350/9.80*2.05*26.07/66.92 WL=28.52kg

Load on front wheel = Wfs-WL

=157.5-28.52

=128.98kg

Load on rear wheel =WRS+WL

=192.5+28.52

=221.05kg

Maximum weight distribution during acceleration

% Load on front wheel = (128.98*100)/350

=36.85%

% load on rear wheel = (221.05*100)/350

= 63.15%

Weight Distribution is – front: Rear =36.85: 63.15

- Spring design for front & rear suspension:-

Load on front single wheel = (157.5/2)*9.81=784.8N

Material of spring = Cold drawn steel (Spring steel)

Sut = 1460 Mpa

Modulus of rigidity = G = 81370 Mpa

Permissible shear stress = τ =0.5* Sut

= 0.5*1460 = 730 N/mm2

Wahl’s factor

K=(4C-1)/(4C-4) + 0.615/C ………… assume (C=6)

K= (4*6-1)/(4*6-4) + 0.615/6

K=1.2525

Permissible shar stress

τ = K * (8PC)/□d2

d2 = 1.2525*(8*784.8*6)/(□*730)

d=4.5357mm =5mm

Wire diameter for spring = d = 5mm

Mean coil diameter of spring = D = c*d = 6*5 =30mm

No of active coil assuming deflect of 2 inch

2 inch = 50.8mm

δ = (8*P*D3 *N)/Gd4

N = (δ *G*d4)/(8*P*d3)

N = (50*81370*54)/(8*784.8*303) N = 15 coils

Total number of coils Nt =N+2

= 15+2 = 17

Actual deflection of spring

$\delta = (8*D^3*N*P)/(G*d^4)$

$\delta = (8*30^3*15*784.8)/(81370*5^4)$

$\delta = 49.99$mm

It is assumed when maximum force applied when,
Max gap between two consecutive coils 1 mm.
Solid length of spring = Nt*d
= 17*5 =85mm
Total axial gap between coils will be
= (Nt-1)*1*(17-1)=16mm
Free length of coil = solid length + total axial gap + deflection
= 85+16+50.8 =151.8mm
Pitch of coil = Free length / Nt -1
= 85/16 =5.3mm
Wire diameter = d = 5mm
Mean coil diameter = D = 30mm
Actual deflection = δ = 50.8mm
Solid length = 85mm
Free length = 151.8mm
Pitch of coil = 5.31mm
Number of turns = 17

IV. ANALYSIS RESULTS

1. Analysis of SPRING

The material of spring is spring steel is used. The most popular alloys included high carbon, oil tempered low carbon, chrome silicon and stainless steel.

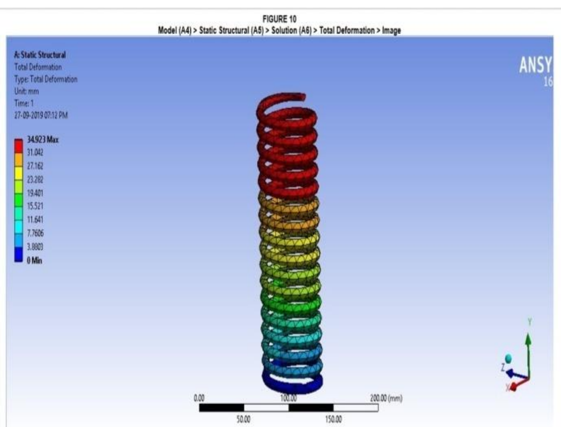

Figure:-2 Analysis of Spring

Result

Maximum Stress	555.27 mpa
Total Deformation	34.92 mm
Factor of Safety	3

2.Analysis of LOWER ARM WISHBONE SYSTEM

The designed lower arm of wishbone system was analysed with static structural by using ANSYS. The input geometry draw in solid-works with result parameters.

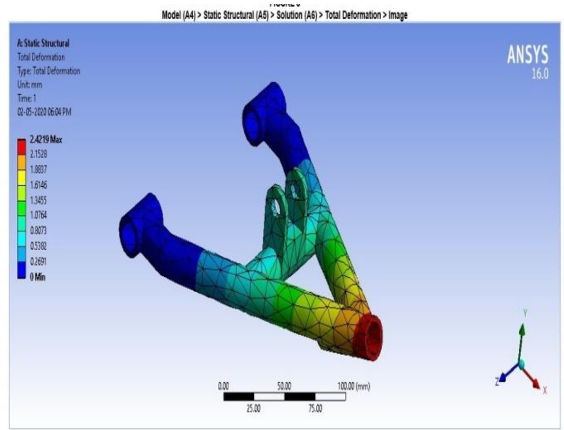

Figure:-3 Analysis of Lower Arm Wishbone System

Result

Maximum Stress	456.37 mpa
Total Deformation	2.34 mm

3.Analysis of SWING ARM

The designed swing arm of wishbone system was analysed with static structural by using ANSYS. The input geometry draw in solid-works with result parameters.

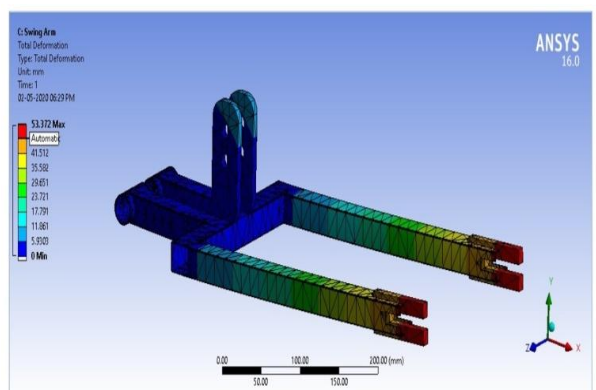

Figure:-4 Analysis of Swing Arm

Result

Maximum Stress	2198.54 mpa
Total Deformation	53.72 mm

V. CONCLUSION

1. We calculated the values of spring such as mean diameter of spring, coil diameter, deflection of spring, number of turns, etc. and load acting on each wheel.
2. We calculated all loads and dimensions of spring theoretically and it is analysed by the using ANSYS software from that we conclude that deformation produced in the spring will minimum. So the spring will be safe.
3. We conclude that the wishbone system gives good handling performance and it is easy to tune with camber variations.
4. We design a Cad modelling in Solid Works Software & we analysed this model by using ANSYS Software.

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