Design and Optimization of Suspension System for an Formula Hybrid Vehicle

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***Abstract*—The suspension system is one of the most important systems to consider when designing a car.The car will not perform up to its full potential, if the suspension does a poor job. So, suspension components are first designed using certain assumptions and other criteria and are then analyzed to obtain final optimized components of suspension system. The objective of this paper is to discuss a design methodology and FEA of suspension components of a Formula one prototype vehicle including upright, hub and wishbones in accordance with the rule book of HVC 2018 provided by ISIE. The paper main aim is to describe an approach for analysis of different components and deal with various methodologies for analysis of components so as to check all possible failure possibilities. In this paper the 2018 Formula Hybrid Car is taken as an example. Suspension is the key to connect car chassis with the wheels and has relative motion between them. Suspension is a device for guiding the wheel kinematically relative to the chassis. To support vehicle weight in all driving possible conditions. To optimize the tire contact with the respective surface. To maintain the wheel plan in a good configuration of steering and camber angle with respect to the ground. To oppose a reaction to longitudinal and lateral forces produced at the tire contact patch. to keep rolling movements when pitch (dive under braking and squat under acceleration) and cornering (lateral acceleration). To isolate the case of road irregularities to ensure passenger comfort to ensure the passengers safety by maintaining the integrity of the vehicle during impact.**

Key Words: Suspension, FEA, Upright, ISIE HVC Rule book.

I. INTRODUCTION

This suspension is designed on the basis of driver's safety and calculations are done comparatively with different journals and textbooks for the basic idea. We know that formula vehicle contains a closed cockpit body and formula vehicle is very compatible and driver safety is major criterion for engineer and calculations are done on those bases and these calculations, design, analysis and manufacturing is done and these components are mentioned in detail in this paper.

II. GENERAL PARAMETERS

An unequal length, double-wishbone suspension system was chosen. Damper is mounted to the lower wishbone at front and upper wishbone at rear. The main reason to choose this suspension is:

a. As the vehicle chassis height is more, double wishbone is more rigid and stable than other suspension systems.
b. As we have lowered the un-sprung weight to make the ride more comfortable we use, double wishbone suspension.
c. One of its primary benefits is the increase of negative chamber as a result of the vertical suspension movement of the upper and lower arms.
d. Tires maintain more contact with the road surface. Handling performance also increases [6].

A slightly longer wheelbase of 1700mm was selected to introduce a margin for later design compatibility [1]. And track width of 1200mm was selected as a compromise between the benefits of reduced weight transfer from a wider track and the tighter travel path about a chicane a narrower track allows. Both front and rear track were made equal. To provide sufficient ground clearance and prevent the bottom of the chassis from hitting the ground under full bump and maximum braking an initial ride height of 127mm (5 inch) was chosen [1]. Estimation of the all the component weights are made and yield a weight of 325kg for the vehicle including a 77kg driver and a weight distribution of 43:57 front to rear. Centre of gravity height 320.97mm is determined using the equations in Milliken & Milliken [2].

A. Expected Performance

Predictions about the vehicles performance are necessary to perform forthcoming calculations. The strongest acceleration usually experienced in a vehicle is the braking as this is primarily limited by the grip of the tyres. Braking is expected to max at 1.4g [3]. Under steady state cornering, 1.2g of lateral acceleration is expected which is smaller than braking partly due to the track being narrower than the wheelbase. Based on the 4:1 step down ratio of the gearbox and maximum power, acceleration expected to max at 0.7g.

III. ROLL DYNAMICS

A. Roll Centers

The height of the roll centre affects the moment arms connecting it to the centre of gravity and the Tyre contact patch. The roll of the vehicle is reduced by a high roll centre reducing the moment arm of the centrifugal force acting on the centre of gravity during cornering [7]. A high roll centre also increases the moment arm to the lateral force acting at the Tyre contact patch creating undesirable jacking of the sprung mass, inducing forces to pass through the wishbones rather than through the spring and lateral Tyre scrub [8]. In consideration of all these points a roll centre of 117.01mm at the front (Z_{RF}) and 109.61mm at the rear (Z_{RR}) was determined above ground. The kinematic design of these roll centers is given in Fig. 1 and Fig. 2.

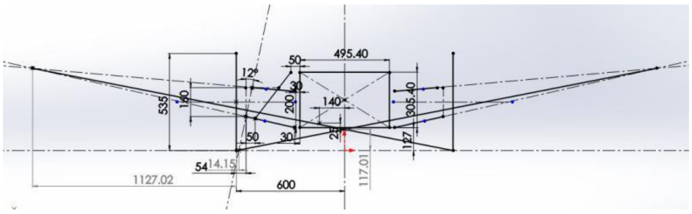

Fig. 1. Kinematic Design of Front Suspension

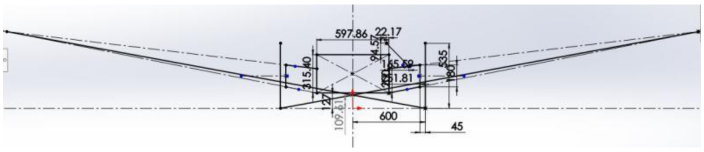

Fig. 2. Kinematic Design of Rear Suspension

B. Roll Gradient

For hard racing vehicle suspensions an roll gradient ($Ø/A_y$) of 1.5 deg/g was selected as recommended in Milliken & Milliken [2].

C. Roll Stiffness

The roll stiffness ($K_Ø$) is described as the roll moment per degree of chassis roll and $K_\phi = 25330.69 Nm/rad$ is calculated using the equations in Milliken & Milliken [2].

D. Roll Stiffness Distribution

The roll stiffness is distributed disproportionately among the front and rear tracks to affect the relative front to rear lateral weight transfer. As the rear of the vehicle is heavier and the roll couple only marginally shorter, the lateral weight transfer at the rear would be larger given even roll distribution [8]. Therefore in the pursuit of neutral handling, the roll distribution was marginally biased towards the front 52:48. This helps reducing the excessive loading on the outside rear tyre that lowers if coefficient of friction. The front and rear roll distributions are calculated as $K_{\phi F} = 13171.96 Nm/rad$ and $K_{\phi R} = 12158.73 Nm/rad$.

E. Weight Transfer

Reactive forces at the wheels attempt to counteract the lateral forces acting at the centre of gravity as described before. As a result load is transferred to the outside wheels from the inside wheels. The simplified lateral weight transfer (ΔW) formula given in Milliken & Milliken. It still takes into account the different roll centers and roll stiffness at the front and rear [2]. The weight transfers are found to be 51.5kg at front and 54.12kg at rear.

F. Ride Rate

The ride rate (K_R) represents the force needed per unit of vertical displacement of the Tyre contact patch [2]. The equation for ride rates are given in Milliken & Milliken (1995). The front and rear ride rates are found to be 18294.39 N/m and 16887.13 N/m respectively.

G. Wheel Rate

The wheel rate (K_W) is the vertical force per unit of displacement of the wheel. For stiffly sprung racing suspension the tyres can provide up to half of the compliance, therefore the compliance of the tyres must be taken out of the ride rate to calculate the necessary spring stiffness [2]. The wheel rates are calculate to be $K_{WF} = 21207.08 N/m$ and $K_{WR} = 19338.92 N/m$.

IV. SPRING DESIGN

To maintain more ride comfort to occupants of the vehicle and to maintain safe handling conditions properly designing the suspension springs are mandatory. If the design of suspension spring fails there are many problems like loosing of CV joints, decreasing ground clearance, transmission of forces from road to vehicle. The interchangeable springs with spring stiffness varying from 29 N/mm to 35 N/mm at rear and from 35 N/mm to 43 N/mm at front were selected by considering the variable loading condition [4] [5].

TABLE I. SPRING DIMENSIONS

Parameters	Front	Rear
Wire Diameter	9mm	8mm
Outer Diameter	64mm	62mm
Inner Diameter	46mm	46mm
Free Length	180mm	155mm
Active Coils	10	8
Total Coils	12	10
Solid Height	108mm	80mm
Spring Index	6.11	6.75
Spring Rate	39.061 N/mm	32.207 N/mm

A. Installation Ratio

The installation ratio (IR) relates the displacement of the spring/damper to the vertical displacement of the wheel [2]. As the installation ratio reduces both the displacement and force at the wheel relative to the spring/damper, the ratio must be squared when relating the wheel and spring rates (Milliken & Milliken 1995). Spring rates of 39061N/m and 32207N/m where used for the front and rear springs and the Installation ratios are found to be $I_{RF} = 0.74$ and $I_{RR} = 0.77$.

V. CAD DESIGN

A. Wishbones

This section began with selection of the material to be used by comparing the advantages and disadvantages of the various materials currently used for suspension components. The material should have high strength to with stand the loads acting on it in the dynamic conditions. The material selection aslo depends on number of factors such as carbon content, mechanical properties, availability and cost. AISI 1018 and AISI 4130 materials were considered based on their availability in the market. AISI 4130 was chosen for wishbones.

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Material Properties:

Properties	AISI 1018 Mild Carbon Steel	AISI 4130 Chromoly
Outer Diameter (mm)	25.4	25.4
Thickness (mm)	3	2.3
Density (kg/m³)	7870	7870
Ultimate tensile strength (MPa)	440	670
Yield tensile strength (MPa)	370	435
Poisson's Ratio	0.29	0.29

Comparison:

	AISI 1018 Mild Carbon Steel	AISI 4130 Chromoly
Equivalent Stress (MPa)	144.58	215.79
Total Deformation (mm)	1.0381	1.918
FOS	1.6599	2.0158
Wall thickness (mm)	3	2.3

Fig. 3. Comparison between AISI 4130 and AISI 1018

The 3D CAD modelling began by inputting the suspension pivot locations determined during the Kinematic Design into SolidWorks using points on a 3D sketch. Using the points, lines representing the wishbones are drawn. A weldment member feature generated the shape of the wishbones a-arms. The trim feature and fillet feature were used to correct the model.

Both on the front and rear the inner wishbone pivots were spaced 300mm apart. The forces acting in the wishbones are reduced by increasing the distance between inner wishbone pivots under any longitudinal loads.

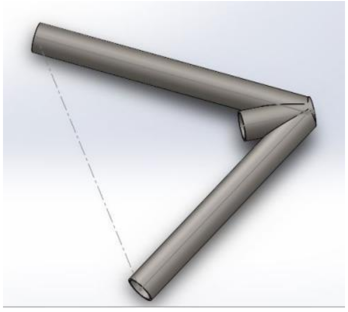

Fig. 4. 3D CAD model of Wishbone

On the rear suspension the toe control arms are located parallel to the upper wishbone to provide easier access to adjustment. The lower outer pivots of front wishbones are located at 120mm from the wheel centre and the upper outer pivot at 90mm. And The lower outer pivots of rear wishbones are located at 80mm from the wheel centre and the upper outer pivot at 85mm.

B. Upright

The front / steering upright is the connection between the tie rod, stub axle and axle housing. Upright is connected to the axle housing by using king pin. Another end is connected to the tie rod. Then the wheel hub is fixed over the upright using a bearing and stub axle.The uprights connect the upper and lower ball joints of the control arm, also provides a mounting point for the brake calipers. Front upright also contain mounting point of steering arm. Aluminium 1060 material is used for uprights.

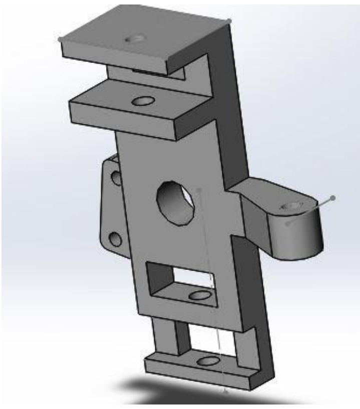

Fig. 5. 3D Model of Front Upright

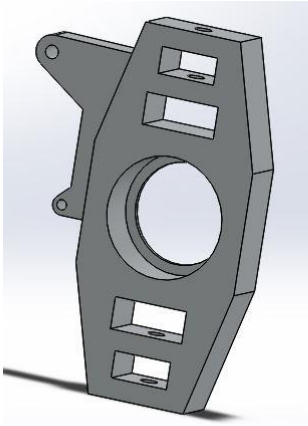

Fig. 6. 3D Model of Rear Upright

VI. SIMULATION

LOTUS Simulation software is used to simulate the suspension geometry of double wishbone suspension system. The graphs were obtained using the Motion Analysis feature in LOTUS Simulation.

The rolling circumference, rebound and bump travel, wheelbase, track width, COG height, braking on front, weight on front, total sprung weight and Various co-ordinates of the system are given as input and an virtual model is built.

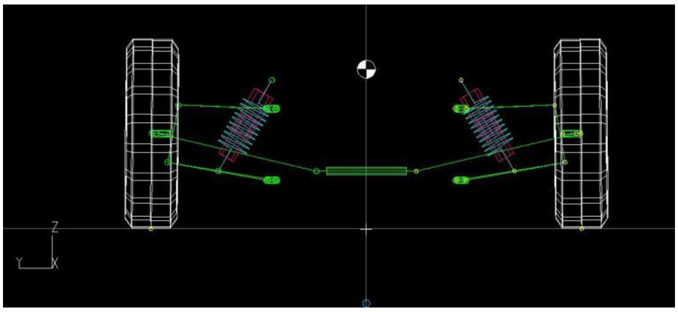

Fig. 7. Front Suspension

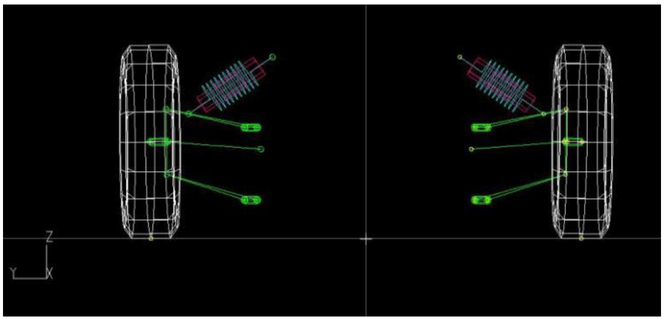

Fig. 8. Rear Suspension

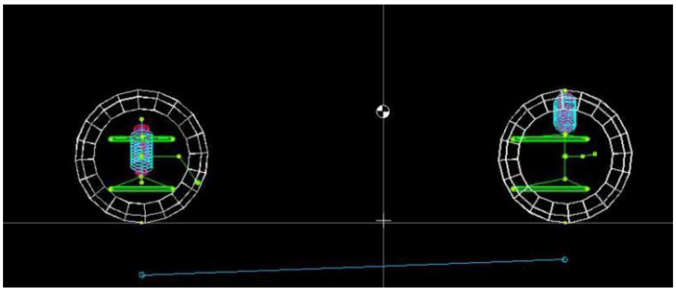

Fig. 9. Side view of front and rear suspensions

A. *Camber change in bump*

Within the maximum limit of travel the Camber curves are studied. Fig. 10 shows the change in camber angle for a rebound and bump for the particular design of the suspension model where the camber is between -2.5 to 2.5.

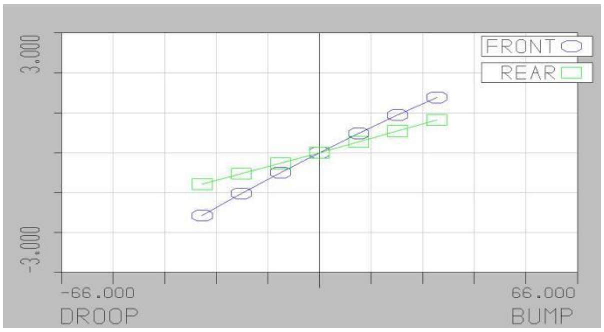

Fig. 10. Camber change in bump

B. *Castor*

Positive caster generates negative camber on the outside tire when the wheel is steered, and positive camber on the inside tire, both of which offset the camber loss due to body roll. Caster change for maximum wheel travel and body roll is shown in Fig. 11.

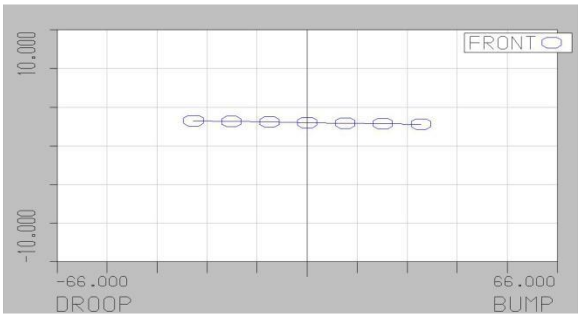

Fig. 11. Castor change in bump

C. *Toe Change*

Toe-in increases straight line stability while toe-out quickens transition behavior and both increases tire wear. Toe change for maximum wheel travel and body roll is shown in Fig. 12.

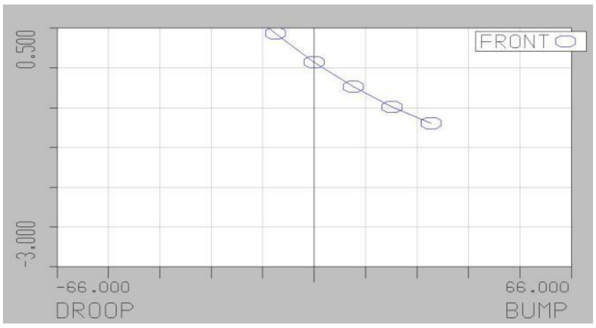

Fig. 12. Toe change in bump

VII. ANALYSIS OF UPRIGHT

A. *Front Upright*

Upright acts as the link between the static and dynamic assemblies of the vehicle. It is subjected to many forces. Analysis of the front upright is done by considering the bearing portion to be fixed. And the forces are applied on the wishbone mounting points. Besides this force the calculated steering torque of 50 N-m is also applied on the upright [9].

Applying all these conditions, the knuckle is meshed properly and the loading condition is then solved using Solidworks Simulation. The maximum stress induced is $8.532*10^7$ N/m^2 and the factor of safety is calculated as 1.23. The maximum deformation is 0.25mm.

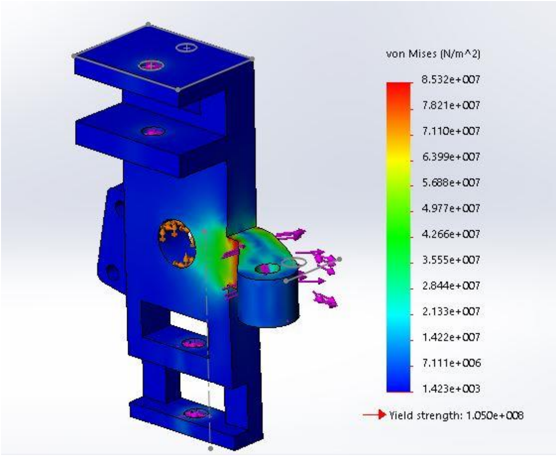

Fig. 13. Stress in Front Upright

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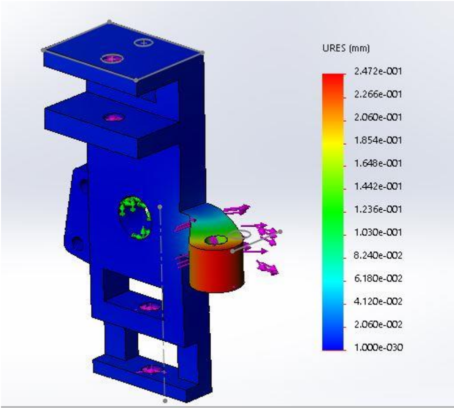

Fig. 14. Displacement in Front Upright

B. *Rear Upright*

Analysis of the rear upright is done by considering the bearing portion to be fixed. And the forces are applied on the wishbone mounting points. Besides this force the calculated braking torque is also applied on the upright [9].

Applying all these conditions, the knuckle is meshed properly and the loading condition is then solved using Solidworks Simulation. The maximum stress induced is $6.592*10^7$ N/m^2 and the factor of safety is calculated as 1.6. The maximum deformation is 0.05mm.

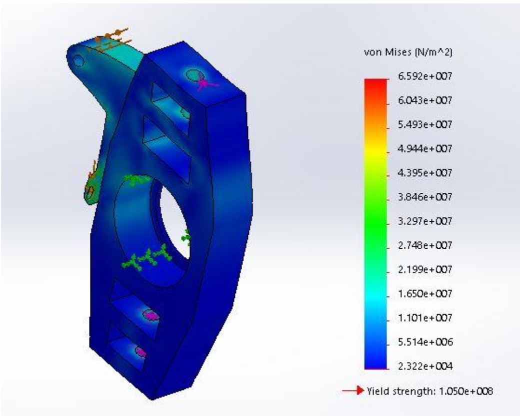

Fig. 15. Stress in Rear Upright

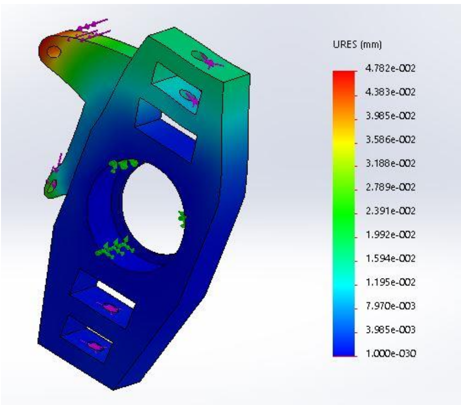

Fig. 16. Displacement in Rear Upright caliper

VIII. CONCLUSIONS

In the present work optimum values for the suspension stiffness and damping are obtained for good comfort and handling conditions. From results it is observed that soft spring gives good comfort i.e. less sprung mass displacement and acceleration. Very less and very high damping values gives more acceleration. So optimum value is selected which gives minimum sprung mass acceleration.

All the obtained suspension parameters are checked by the LOTUS suspension analyzer for kinematic analysis. For the calculated values and it is observed that there is no change in the camber and toe angle values, and even change in wheel base is also within the desired limit. Many of the solutions decided upon are not ideal and therefore there are many opportunities for improvement to future vehicles.

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