TEST Engineering & Management

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

Design and Simulation of Pushrod Suspension System

[1]Dr.Shriniwas S Metan, [2] Mr. Saleek Bijapure, [3]Mr.Mayur Mukkawar
[1], [2] NK Orchid College of Engineering and Technology, Solapur, [3] Sai Service, Pune
[1]shrinims@gmail.com,[2] saleekbijapure@gmail.com, [3]mayurmukkawar142@gmail.com

Article Info
Volume 83
Page Number: 7621 - 7632
Publication Issue:
March - April 2020

Article History
***Article Received**: 24 July 2019*
***Revised**: 12 September 2019*
***Accepted**: 15 February 2020*
***Publication**: 07 April 2020*

Abstract

The suspension system of modern vehicle still has heavy unsprung components which reduce the performance of vehicle considering comfort, roll stability, bump steer etc. Reducing the unsprung mass of the conventional suspension system will increase the quality and performance of vehicle which includes better kinematic simulation of wishbones for better ride control and use of different alloys for wheel assembly and VMC machining over casting for different components etc. The design starts with vehicle kinematics considering changes in wheel motion while in dynamic condition. The simulation was carried out in Lotus Shark Suspension Analysis, thus the hard points were determined. With these hard points solid modeling of wheel assembly was modeled consisting of Upright, Bearings, Hub and rod ends etc. All the solid modelling was done in Solid works student edition. For the manufacturing of wheel assembly Aluminum alloys (6061/7075-T6) were used. Pushrod suspension system was selected as it gives more flexibility while packaging, easy modifications and less unsprung mass.

Index Terms; *Double wishbone, Upright, Hub, Vehicle kinematics, suspension design, Pushrod suspension, FEA.*

I. INTRODUCTION

An automobile suspension is the system of parts that give a vehicle the ability to maneuver. The function of suspensions in automobile is to handle bump and droop in road surface, which also means improving ride comfort. The objective of suspension system is to provide maximum contact patch between the roads and tires so that the vehicle can achieve better stability in steering and handling. Independent suspension provides these properties. It’s essential to analyses suspension system as of which the behavior of the vehicle can estimated. It is a very important sub system of vehicle as it is responsible for:

1. To transfer the Unsprung and Sprung mass of the vehicle to chassis.
2. To maintain maximum possible contact patch with ground for better grip.
3. To absorb dive (braking) and squat (acceleration) forces of the vehicle.

Suspension system is involved in the dynamics of the vehicle which includes the cornering forces and steering inputs for the maneuverability .The components in suspension system are tire, hubs, uprights, wishbones, rod ends, dampers, springs and shock absorbers. The objective of suspension system is to provide a relative motion between chassis and wheels and also confine shocks and vibrations during motion. Thus, this system serves a dual purpose: optimizing the ride comfort and resisting the loads on all terrain.

II. METHODOLOGY

The design starts with vehicle kinematics considering changes in wheel motion while dynamic condition. This simulation was carried out on Lotus Shark Suspension Analysis software, thus the hard points were determined. With these hard points solid modeling started of wheel assembly consisting of Hub, Bearings, Uprights and Rod ends. The wishbones lengths were determined considering the braking torque and anti-dive/anti-squat scenarios.

All the solid modeling and FEA simulation of subsystems was carried out in SolidWorks. For the manufacturing of wheel assembly Aluminum alloys (6061/7075-T6) were machined using VMC's. Pushrod suspension system was selected for better optimization.

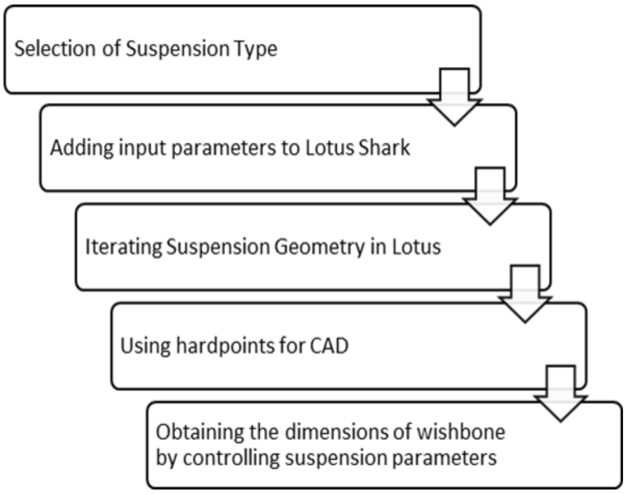


Fig. 1 Methodology Flowchart

III. DESIGN

A. Design consideration

Table 1 Design Considerations

CONSIDERA-TION	*PRIORITY*	*REASON*
Light Weight	High	A light Racecar is fast Racecar
Resilience	High	Must withstand to Endurance limit
Design Parameters	High	Compliance and integration with other subsystem
Structure	High	Easy to manufacture
Attractive Design	Desired	Aesthetics of Vehicle
Cost	Low	Budget allocations
Manufacturability	High	Can be fabricated in our workshop with available facilities

B. Selection of Suitable Suspension System:

The first step in designing of suspension system is to select type of geometry of suspension. For our case as we will be focusing on designing for racing or performance perspectives the suspension should have freedom of adjustment, light weight and low drag. This all can be achieved with the Double wishbone suspension system [1]. The two types of double wishbone suspension system are parallel arms with unequal lengths and unparalleled arms with unequal lengths, the later was selected. By this roll center of the vehicle can be lowered below the center of gravity resulting in lower jacking force. Following parameters were considered while selecting the suspension system:

1. Force analysis.
2. Kinematics of geometry.
3. Compliance.
4. Cost.
5. Design parameters: Camber, Caster, Toe etc.
6. Availability.

C. Double Wishbone Pushrod Suspension System

In pushrod suspension system the shock absorbers are mounted on the chassis of vehicle and the force is transferred to it by push rods which are mounted on the wishbones. The mounting of absorbers on chassis gives lower center of gravity, lower unsprung mass and lower air drag. Which is better for ride control. This setup gives freedom for controlling tire camber, high rigidity and strength, less vulnerability to damage by debris. Mounting of shock absorbers inboard gives much more flexibility than the normal conventional direct acting suspension [2]. For the front suspension the absorber were mounted up higher in car though it raises center of gravity allowing more space for driver cockpit and legs. The rear suspension system packaging was done considering the powertrain and drivetrain assemblies. The absorber were mounted above drive axle and differential. Both the pushrods were mounted on the upper wishbones linkage of geometry [3].

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

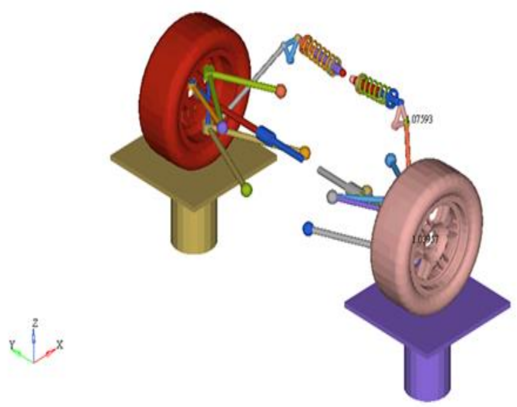


Fig. 2 Double Wishbone Pushrod Suspension System

IV. KINEMATICS OF VEHICLE

Study of motion of linkages involved in machines a very important part of simulation and analysis of the design. There are two types of motion studies kinematics and dynamics. Kinematics is the study of motion without considering the forces acting on it, whereas dynamics study considers the forces that cause the motion [5]. Further improvement in this are multibody dynamics, model based simulations, mechanical simulations and even virtual prototypes.

Kinematic simulation is less complex as compared to dynamic simulation and is applicable for most of the application concerning with motion in parts. Kinematic simulation has a baseline with reference to time. It determines the positions in form of coordinate with respect to time. These simulations are quite beneficial for complex machine assemblies where the operations are divided into small simulation and the study of kinematics is carried out. These studies are further taken into account for dynamic analysis [6].

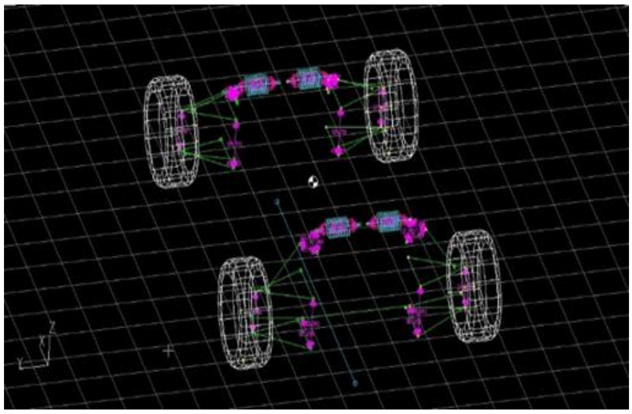


Fig. 3 Lotus Shark Model

To create model in these solvers, the prerequisites are vehicle design parameters which includes wheelbase, track width, inertia of vehicle, center of gravity, tire parameters and hard points or coordinates of suspension and steering systems. While creating the model there are two options available in most of the solvers some pre-defined templates are available one can directly use one of these and modify or build a model from scratch by defining all constraints, joints and motion

[7]. The mathematical modeling used by the solver must be known to user which helps in understanding the conceptsbehind the conclusion and results shown by it. For this case Lotus Shark Suspension Analysis was used. The model was defined in it with the design parameters and properties related with the tires. After completing the model the kinematic simulation for it was carried out. The results include wheel travel against camber, toe and roll. The following images show the front and rear of kinematic model [8].

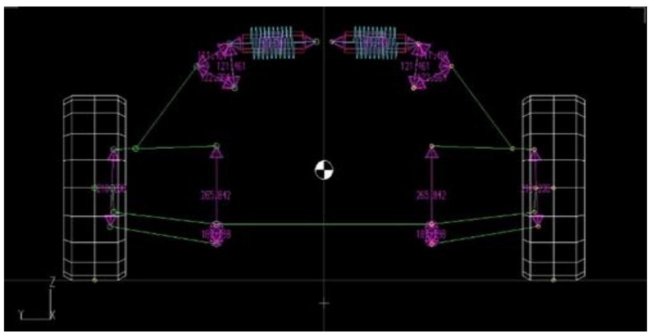


Fig. 4 Front End

TEST Engineering & Management

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

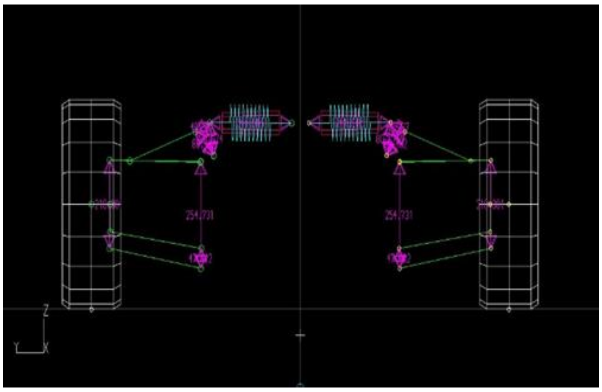


Fig. 5 Rear End

The results are tabulated below and plots of same are also dine with all wheel travel.

Table 2 Static Values

CAMBER ANGLE (deg):
0.00
TOE ANGLE (SAE) (+ve TOEIN) (deg):
0.00
CASTOR ANGLE (deg):
3.00
CASTOR TRAIL (HUB TRAIL) (mm):
0.26
CASTOR OFFSET (mm):
12.31
KINGPIN ANGLE (deg):
3.00
KINGPINOFFSET (AT WHEEL) (mm):
48.91
KINGPIN OFFSET (AT GROUND) (mm):
36.22
MECHANICAL TRAIL (mm):
12.30
ROLL CENTRE HEIGHT (mm):
-198.58

Table 3 General Data Values

TYRE ROLLING RADIUS (mm):
251.00
WHEELBASE (mm):
600.00
C OF G HEIGHT (mm):
300.00
BREAKING ON FRONT AXLE (%):
50.00
DRIVE ON FRONT AXLE (%):
0.00
WEIGHT ON FRONT AXLE (%):
50.00

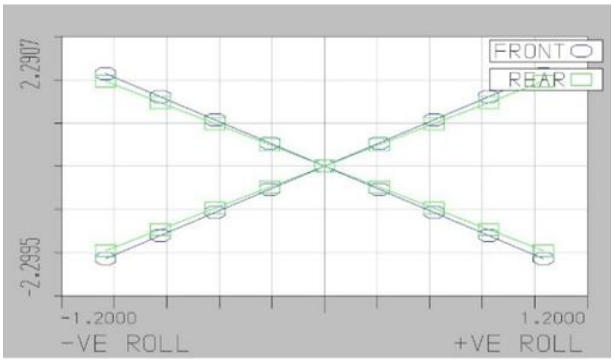


Fig.. 6 Wheel travel VS. camber

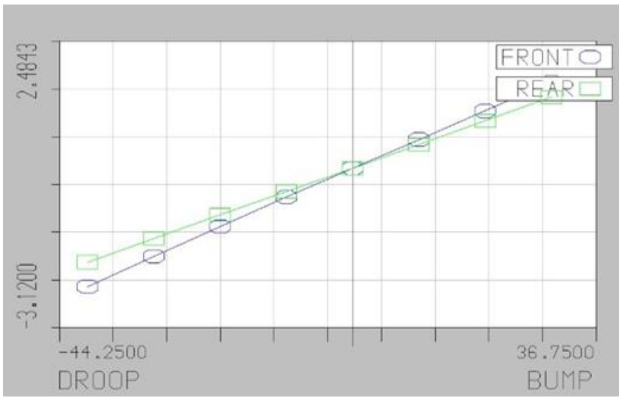


Fig.. 7 Wheel travel VS. Toe

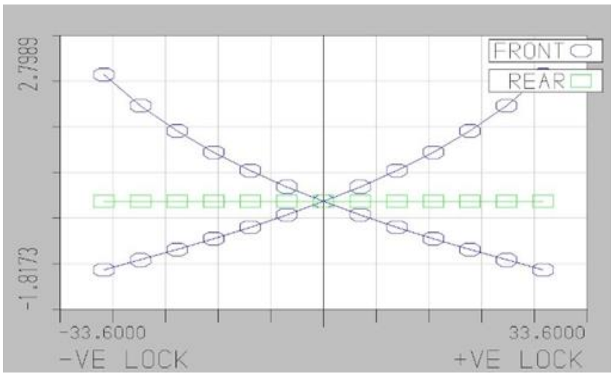


Fig.. 8 Roll VS. Camber

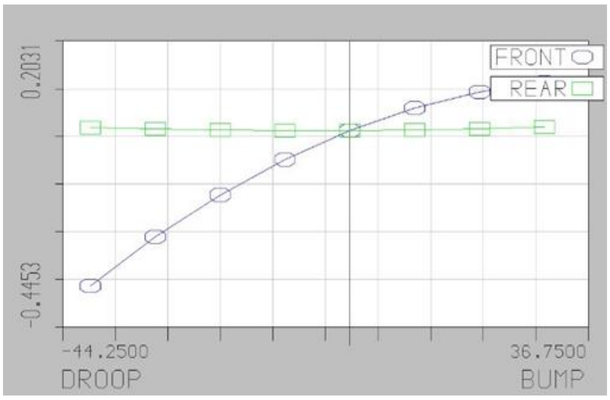


Fig.. 9 Steer travel VS, camber

Published by: The Mattingley Publishing Co., Inc.

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

After simulations are done the data can be stored in forms of tables. From this the nature of vehicle can be studied while in motion. The setup may take certain iteration to replicate terrain scenarios and design requirements. The critical factors affecting motion of car like roll center and center of gravity can be analyzed for better ride control and stability. While setting different strategies for the track the vehicle needs different suspension setups for the likes of calibrating the anti-dive and camber controls etc. For these different sets of hardpoints are needed to be simulated which can be later used for changing the car behavior during testing. To do so the positions of mountings of wishbones on chassis are modified and simulated. These reduces time while testing different types of geometry and comparing them with each other [3].

It might seem Kinematic analysis is the solution to all but when it comes to changing setup of suspension for different strategies it really is the first and arguably most important step which helps in solving the rest of equations and simulations.

The front and rear suspension is an unequal and non-parallel length wishbone geometry. To reduce the roll of the vehicle and optimize jacking forces the roll center is kept at an adequate height. The variations of camber, caster and roll have been observed with respect to the wheel travel to ensure better stability. The material used for manufacturing the wishbones is AISI 4130 with tube diameter of 18mm.

A. Front Suspension

Table 5 Bump Travel

BUMPTRAVEL (mm): 30.00 INCREMENT (mm): 10.00
REBOUND TRAVEL (mm):37.50 INCREMENT (mm): 10.00
ROLL ANGLE (deg): 1.00 ROLL INCREMENT (deg):00.25

ROLL ANGLE (deg)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-40	-2.3194	-0.3529	2.8748	5.2309
-30	-1.722	-0.2414	2.8722	4.6279
-20	-1.1386	-0.1459	2.87	4.0397
-10	-0.5657	-0.0656	2.8681	3.4627
0	0	0	2.8665	2.8938
10	0.5615	0.0513	2.8652	2.3297
20	1.1218	0.0882	2.8642	1.7676
30	1.6837	0.1106	2.8635	1.2045

Table 6 Roll Variations

ROLL ANGLE (deg)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-1	1.635	0.0499	2.8633	1.2563
-0.75	1.2269	0.0392	2.8639	1.6649
-0.5	0.8184	0.0273	2.8646	2.074
-0.25	0.4094	0.0142	2.8655	2.4836
0	0	0	2.8665	2.8938
0.25	-0.41	-0.0154	2.8677	3.3045
0.5	-0.8206	-0.032	2.8691	3.7159
0.75	-1.2318	-0.0498	2.8706	4.1281
1	-1.6438	-0.0688	2.8723	4.51

B. Rear Suspension

Table 7 Bump Travel

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-40	-1.8361	0.0076	0.0708	1.9179
-30	-1.3724	0.0038	0.0872	1.4542
-20	-0.9135	0.0012	0.1037	0.9954
-10	-0.4569	0	0.1201	0.5388
0	0	0	0.1364	0.0818
10	0.4599	0.0013	0.1528	-0.378
20	0.9253	0.004	0.1691	-0.8435
30	1.3992	0.008	0.1854	-1.3174

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

Table 8 Roll Variations

ROLL ANGLE (deg)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)
-1	1.5004	-0.0055	0.1543	-1.4185
-0.75	1.1251	-0.0043	0.1498	-1.0432
-0.5	0.7499	-0.0029	0.1454	-0.6681
-0.25	0.3749	-0.0015	0.1409	-0.2931
0	0	0	0.1364	0.0818
0.25	-0.3749	0.0016	0.132	0.4567
0.5	-0.7497	0.0033	0.1275	0.8315
0.75	-1.1246	0.0051	0.1231	1.2064
1	-1.4995	0.007	0.1187	1.5813

V. UPRIGHT DESIGN

The following parameters where considered while design the upright:

1. Camber
2. Caster
3. Toe

These adjustments will be required at the time of testing. As to provide more flexibility the ball joints mounting points were designed closer to the wishbone assembly to make it rigid and avoid failures in bending.

The forces acting on upright are:

1. Braking Torque at Caliper mounting points
2. Tie rod force on steering arm

The following figures shows the 3D CAD models of front and rear uprights

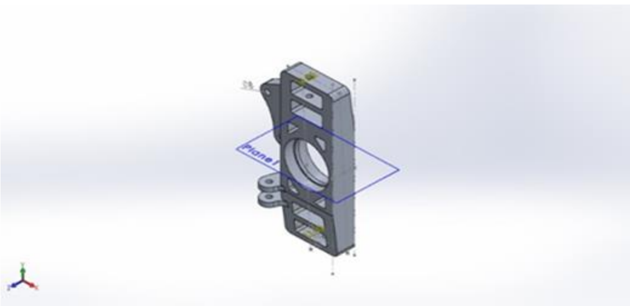


Fig. 10 3D CAD Model of Front Upright

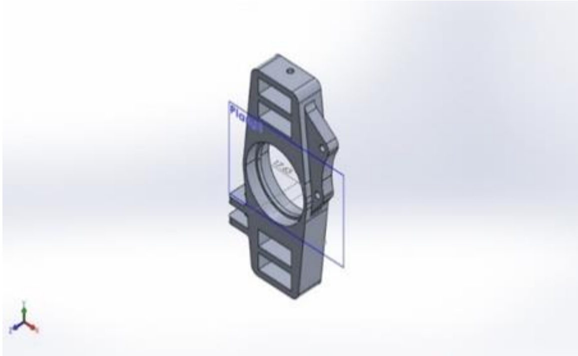


Fig. 10 3D CAD Model of Rear Upright

A. FEM of Uprights

The FEM of upright is one of the most important analysis, because upright is the component on which brake caliper, tie rod, and wishbone rod ends are connected. The 3D CAD model is modelled in CAD software SolidWorks. In present the simulation was performed in SolidWorks simulation. The simulation results were then cross checked using ANSYS software.

i. Boundary Condition

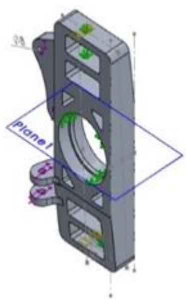


Fig. 11 Front Upright

Fig shows the boundary conditions during the upright FEM analysis the boundary condition are upper and lower wishbone mounting point and the bearing housing were fixed as shown by the green arrows the tie rod end mounting and brake mounting are point were forces were applied as shown by pink arrows. Simulation of upright with respect to wishbone mounting points is done.

Published by: The Mattingley Publishing Co., Inc.

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

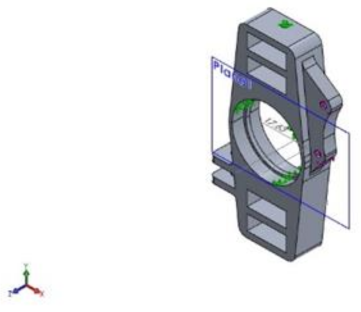


Fig. 12 Rear Upright

VI. HUBS

A. Introduction to Wheel Hub:

Wheel hubs are one of the crucial components of suspension. Wheel hub basically connects the wheels to the vehicle. It is fixed on bearings to rotate freely. Often it house the brake rotors or brake drums for the braking system. And for drivetrain it may hold the drive axle to propel the vehicle. As being one of the components which comes first while experiencing the forces it is very important to design a safe and durable wheel hub.

While designing a wheel hub the dynamic forces must be considered which may occur during the operation of car and those are as follows:

1. Acceleration/Deceleration.
2. Cornering force.
3. Wheel reaction(Bump/Droop)
4. Brake and Axle torques

13" wheels were selected to provide room for the upright and A-arm configuration.

The CAD model assembly of the wheel is shown below

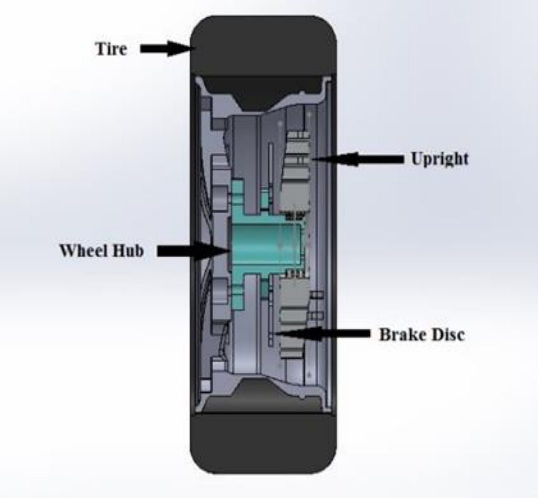


Fig. 13 Cut Section of Front Wheel Assembly

B. Hub Design

The wheel is fixed on hub with the four equally spaced Wheel studs.

Hence, the loads on the wheel are transferred to hub from the wheel studs.

The forces acting on hub are:

1. ± 3G vertical.
2. ± 1.5G lateral.
3. ± 1.5G longitudinal.
4. Braking torque.

The following figures show 3D CAD models of front and rear hubs.

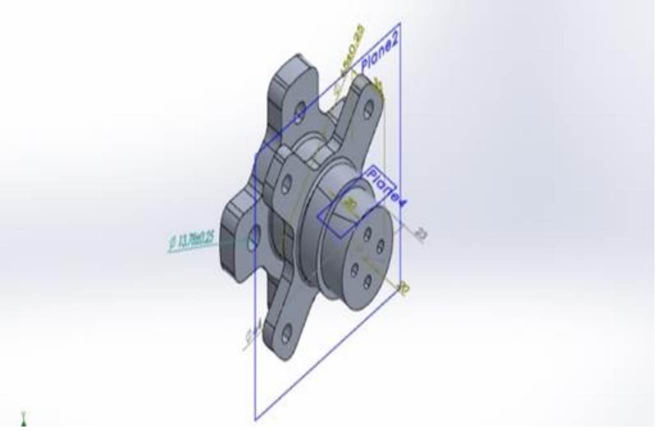


Fig.14 3D CAD model of Front Hub

Published by: The Mattingley Publishing Co., Inc.

TEST Engineering & Management
March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

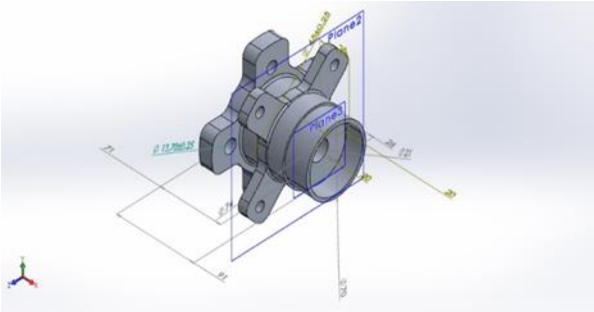


Fig. 15 3D CAD model of Rear Hub

C. Boundary Condition

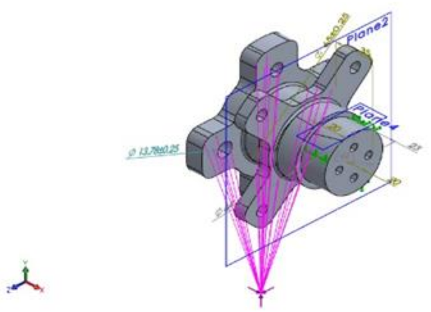


Fig. 16 Front Hub

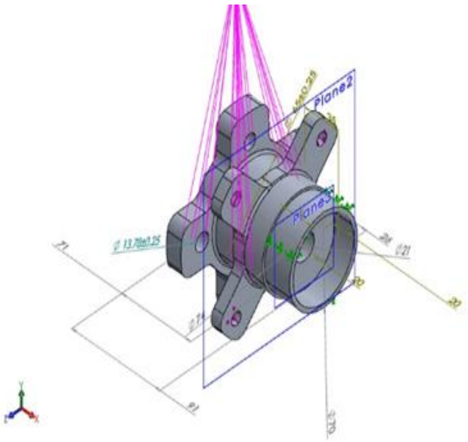


Fig. 17 Rear Hub

The above figures show the boundary conditions for front and rear hubs. The bearing housing area of the front hub was fixed, and remote load is applied at the tire contact patch and braking torque was applied on the brake disc mounting holes.

For the rear part the forces applied were the same as that of the front but the area which was supporting the axle was fixed. 3

VII. MATERIAL SELECTION

The deciding factors used in the material selection for manufacture of the uprights were:

1. Machinalbity.
2. Light weight.
3. Must be relatively inexpensive and available.

Another factor was considered while selecting the material which is its endurance limit as it is a component subjected to fluctuating loads.

1. The front hub material was selected as Al 6061-T6
2. The rear hub material was selected as Al 7075-T6

The reason behind going for stronger material at rear was as the rear hub had splines to fit the transmission axle. So, we are providing extra strength to hub for more safe design.

VIII. RESULTS AND DISCUSSIONS

Study Properties

Table9 Study Properties

Analysis type	Static
Mesh type	Solid Mesh
Mesher Used:	Standard mesh
Mesh Quality	High
Unit system	SI (MKS)
Length/Displacement	mm
Pressure/Stress	N/m^2

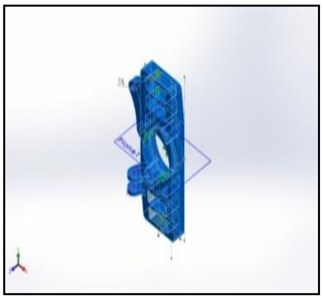
A. Analysis of Upright

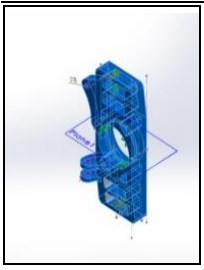
The analysis of front upright was carried out in SolidWorks Simulation.

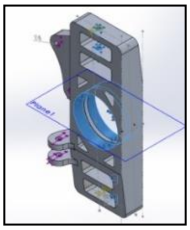
Published by: The Mattingley Publishing Co., Inc.

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

1) Model Information

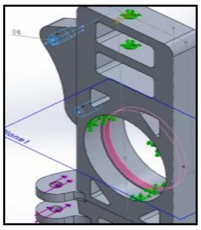
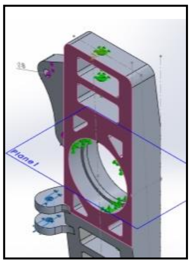
Model	Treated As	Volumetric Properties
	Solid Body	Mass:1.17565 kg Volume:0.000435424 m^3 Density:2700 kg/m^3 Weight:11.521N

Model Reference	Properties		Components
	Name:	6061-T6 (SS)	Solid Body
	Model type:	Linear Elastic Isotropic	
	Yield strength:	2.75e+008 N/m^2	
	Tensile strength:	3.1e+008 N/m^2	
	Elastic modulus:	6.9e+010 N/m^2	
	Poisson's ratio:	0.33	
	Mass density:	2700 kg/m^3	
	Shear modulus:	2.6e+010 N/m^2	
	Thermal expansion coefficient:	2.4e-005 /Kelvin	

Fixture name	Fixture Image	Fixture Details	
Fixed		Entities:	6 face(s)
		Type:	Fixed Geometry

Resultant Forces:

Components	X	Y	Z	Resultant
Reaction force(N)	-9675.56	-23572.6	4200.14	25824.9

Load name	Load Image	Load Details	
Torque		Entities:	2 face(s)
		Reference:	Face< 1 >
		Type:	Apply torque
		Value:	1800 N.m
Force		Entities:	2 face(s)
		Reference:	Face< 1 >
		Type:	Apply force
		Values:	---, ---, -2100 N

Total Nodes	89108
Total Elements	57248

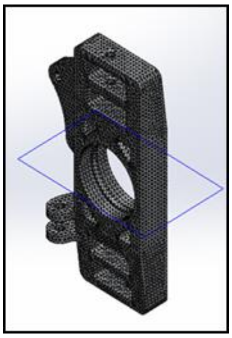


Fig. 16 Meshed model of Front Upright

Name	Type	Min	Max
Stress	VON: von Mises Stress	15006.7 N/m^2 Node: 3160	1.01547e+008 N/m^2 Node: 86165

Published by: The Mattingley Publishing Co., Inc.

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

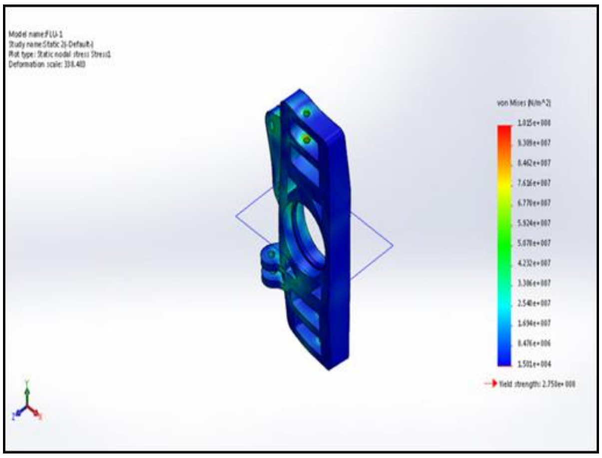


Fig. 17 Von Mises Stress

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 1	0.0811933 mm Node: 88295

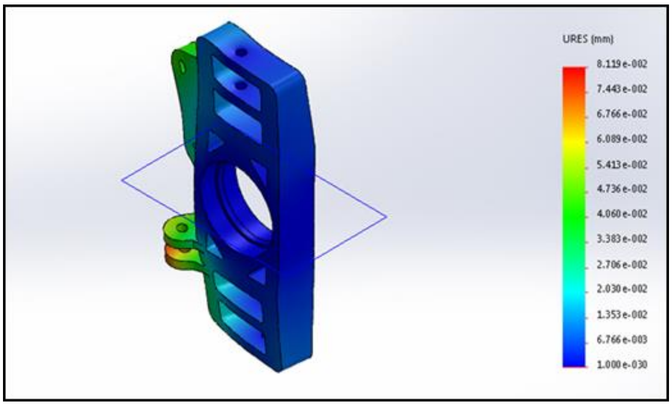


Fig. 18. Static Displacement

Name	Type	Min	Max
Factor of Safety	Automatic	2.70811 Node: 86165	18325.2 Node: 3160

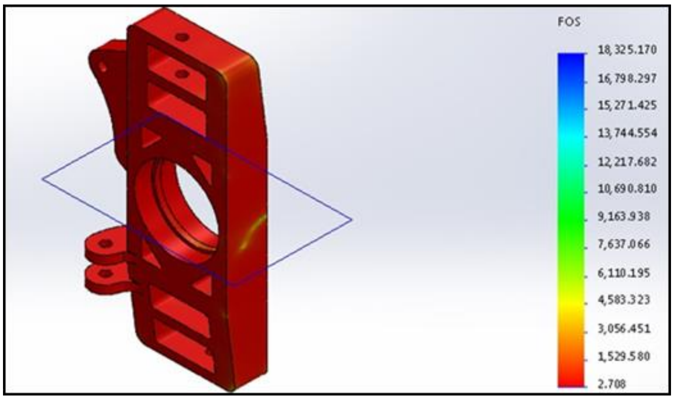
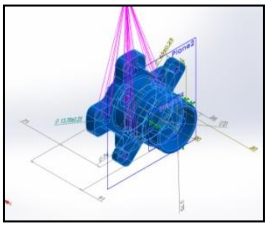
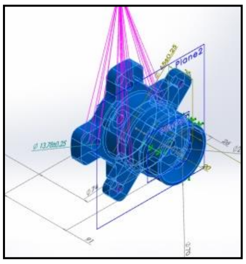


Fig. 19. Factor of Safety

B. Analysis of Hub

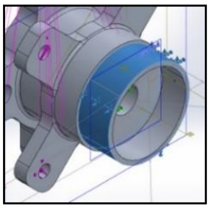
1)Model Information

Model	Treated As	Volumetric Properties
	Solid Body	Mass:1.19002 kg Volume:0.00042349 5 m^3 Density:2810 kg/m^3 Weight:11.6622 N

Model Reference	Properties		Components
	Name:	7075-T6 (SN)	Solid Body 1(Rear Hub)
	Model type:	Linear Elastic Isotropic	
	Default failure criterion:	Max von Mises Stress	
	Yield strength:	5.05e+008 N/m^2	
	Tensile strength:	5.7e+008 N/m^2	
	Elastic modulus:	7.2e+010 N/m^2	
	Poisson's ratio:	0.33	
	Mass density:	2810 kg/m^2	
	Shear modulus:	2.69e+010 N/m^2	
	Thermal expansion coefficient:	2.36e-005 /Kelvin	

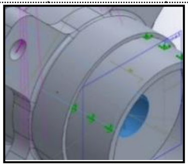
Published by: The Mattingley Publishing Co., Inc.

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

Fixture name	Fixture Image	Fixture Details
Fixed-1		Entities: 1 face(s) Type: Fixed Geometry

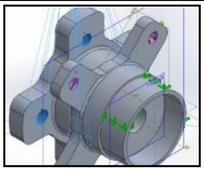
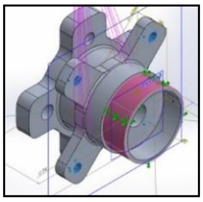
Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	-1463.27	2102.96	-3.50955	2561.96

Fixture name	Fixture Image	Fixture Details
Fixed-2		Entities: 1 face(s) Type: Fixed Geometry

Resultant Forces

Components	X	Y	Z	Resultant
Reaction force(N)	-2954.71	-10946	-4428.52	12171.9

Load	Load Image	Load Details	
Remote Load (Direct transfer)-1		Entities:	4 face(s)
		Type:	Load (Direct transfer)
		Coordinate System:	Global Cartesian coordinates
		Force Values:	4418, 8829, 4418 N
		Moment Values:	---, ---, --- N.m
		Reference coordinates:	0 165 0 mm
		Components transferred:	Force
Torque-1		Entities:	4 face(s)
		Reference:	Face< 1 >
		Type:	Apply torque
		Value:	253 N.m

Total Nodes	110780
Total Elements	73861

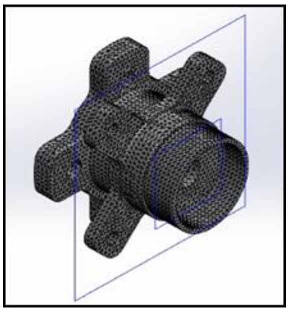


Fig. 20 Meshed Model of Rear Hub

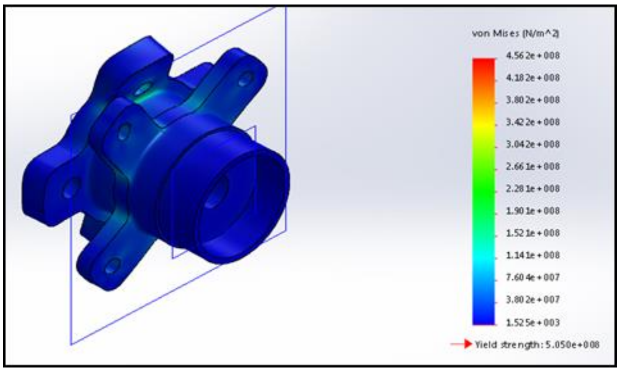


Fig. 21 Von Mises Stress

Name	Type	Min	Max
Displacement	URES: Resultant Displacement	0 mm Node: 288	0.10957 mm Node: 93280

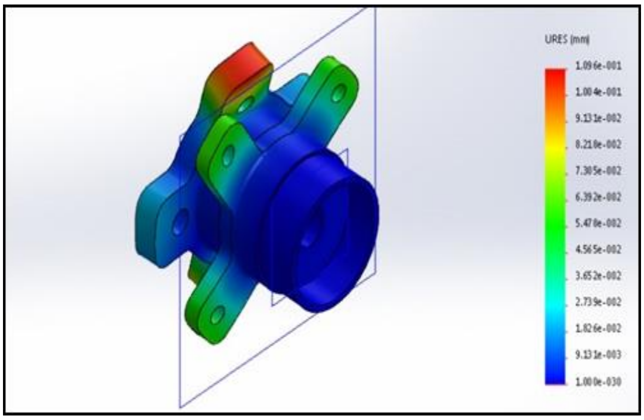


Fig. 22 Resultant Displacement

Published by: The Mattingley Publishing Co., Inc.

TEST Engineering & Management

March - April 2020
ISSN: 0193-4120 Page No. 7621 - 7632

Name	Type	Min	Max
Factor of Safety	Automatic	1.1069 Node: 103616	331250 Node: 110334

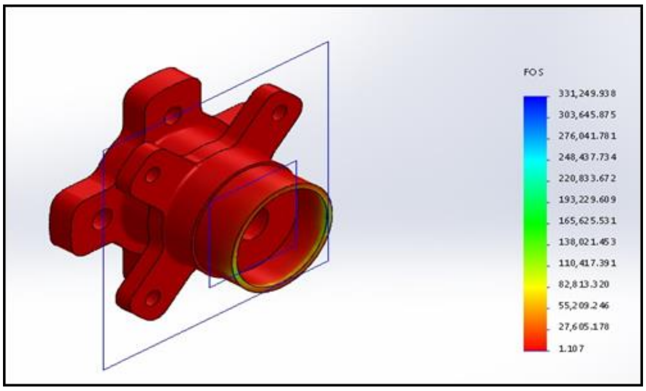


Fig. 23. Factor of Safety

IX. CONCLUSION

After applying same methodology, calculation and simulation to Rear Upright and Front Hub we got these results as follows:

Table 10 Conclusions

Sr. No.	Component	Mass (kg)	Stress (N/m^2)	FOS	Results
1.	Front Upright	1.17	1.01547e+008	2.7	Safe
2.	Rear Upright	1.11	9.03107e+007	3	Safe
3.	Front Hub	0.79	2.05752e+008	1.3	Safe
4.	Rear Hub	1.19	4.56229e+008	1.2	Safe

REFERENCES

[1]. Arindam, Pal, "Optimized Suspension Design of an off-Road Vehicle." International Journal of Engineering and Science, vol 2(6), pp 57-62, 2013.

[2]. Samant Saurabh Y., Santosh Kumar, Kaushal Kamal Jain, Sudhanshu Kumar Behera, Dhiraj Gandhi, Sivapuram Raghavendra, Karuna Kalita , "Design Of Suspension System for Formula Student Race Car", Procedia Engineering, Elsevier, vol 144, pp 1138-1149, 2016.

[3]. K. J. Prashanth, Ashish Baidyanathan and Dr. T. Jagdish, "Role of Kinematic Analysis in tuning the Dynamic Behavior of a Formula Car" International Journal of Technology And Engineering System, Vol 2(3), pp 323-327, 2011.

[4]. Shaopu Yang, Yongjie Lu, Shaohua Li, "An overview on vehicle dynamics", International Journal of Dynamics and Control, Vol 1 (4), pp 385-395, December 2013.

[5]. Mohit, Tomar, Rajkumar, Kewet, Anil, Kumar, Kundu Kuldeep, Kumar., Rohit, Lather., "Dynamic analysis of Double Wishbone with S Link and toe link", International Journal of Engineering Research and Application , Vol 1, pp 247-251, March 2104.

[6]. Udhav U. Gawadalkar., Ranjith M, Anshej Habin, "Design & Modification of All terrain Vehicle". International Journal of Research in Engineering and Technology, Vol 3(9), pp-313-318, March 2014.

[7]. Lalitkumar Maikulal Jugulkar, Shankar Singh, Suresh Maruti Sawant, "Analysis of suspension with variable stiffness and variable damping force for automotive applications", SAGE Journal, Vol 8 (5), pp 1-19, March 2016.

[8]. Lotus Cars Ltd. "Getting Started with Lotus Suspension Analysis," Version 4.02, 2016.

[9]. Gillespie, T.D., "Suspension", in Fundamentals of Vehicle Dynamics", Society of Automotive Engineers, USA, pp- 97-117 and pp- 237-247, 1992.

[10]. Kirpal singh, , "Automobile Engineering", Standard Publishers Distributors, Vol 1 (13), pp 15- 105, 1992.