Design and optimization of suspension geometry of a Solar Electric passenger vehicle

Navneet Singh Dhir[1]

[1]Dept. of Mechanical and manufacturing engineering, Manipal Institute of Technology (MIT), Manipal, India

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Abstract - *Suspension geometry provides the various points at which all the suspension parts need to be positioned so that they serve the purpose for which they are designed. This project is in succession to team SolarMobil's first attempt to make a solar electric vehicle, SERVe, so the design was aimed to not only fulfil the basic requirements of a passenger vehicle like comfort and safety but also enhance the performance of the car. Hence design considerations like making the suspension lighter, more compact, and ensuring it always stays in contact with road during various manoeuvres were given prime importance.*

As part of design process, first various suspension types were considered and geometries were made for each using 'Winge0 3' software. Various methods of testing were implemented to make the comparison between alternate designs, and a final decision was made to use a modified double wishbone pushrod actuated suspension. The main reason for this decision was that we could achieve a compact design by shortening the length of wishbones while not compromising on the performance. The conventional push rod design couldn't be implemented in the front (FWD) due to interference with the transmission parts so the pushrod, bell crank and spring were inclined by the same angle towards the front of the car. To fine tune and optimize the design, 70-80 iterations were performed after this to come up with the final suspension geometry considering a balance between Comfort, performance, ease of manufacturing, cost and weight.

Key Words: **bell crank, pushrod suspension, upright, wishbone, spring, suspension geometry**

1. INTRODUCTION

SolarMobil Manipal is the official solar car team of MIT, Manipal. This project is in succession to Solarmobil's first attempt to make a cruiser class solar electric road vehicle SERVe which was a 2 seater passenger car. The current project aims to build a 4 seater passenger car, which is suitable to drive on Indian roads with better performance, so the suspension system needs to be designed accordingly. We need it to not only fulfill the basic requirements of a passenger vehicle like comfort and safety but also to make changes such that the performance of the car is enhanced. So things like making the suspension lighter, more compact, and ensuring it always stays in contact with road during various maneuvers were given prime importance so that the overall efficiency of the vehicle is improved by reducing the losses through tire at various maneuvers

After deciding the purpose of the car, various initial parameters are considered from steering and ride roll rate calculation sheet.

Taking these parameters initial geometry is made and then multiple iterations are done along with testing to come up with the final geometry

For both front and rear various different suspension types were considered, geometries were made for each and then compared to come up with the suspension that fulfills most of the parameters

Once the suspension type is finalized multiple iterations are done along with testing and slight modifications are made to the initial parameters for optimization. It is also ensured that the front and the rear geometries complement each other for maximum stability and comfort.

After following the above process, a decision was made to use ***unequal, non-parallel double wishbone pushrod suspension***, which is modified according to our needs.

Table -1: General Specifications of the car

Length	4500mm
Width	1780mm
Height	1360mm
CG height	475mm
Drive	Front Wheel Drive
Weight distribution (Front; Rear)	55:45
Wheelbase	2950mm
Track width	1530mm
Tire	175/65 R15
Ground Clearance	160mm

2. INITIAL PARAMETERS

2.1 Steering Calculation

First from the steering calculation sheet based on adequate self-aligning torque and driving feedback parameters such as caster angle, steering axis inclination, scrub radius and caster trail are considered for the initial design of the front geometry.

Table -2: Steering parameters for front suspension

Caster angle	4 degrees
Steering axis inclination	3 degrees
Scrub radius	22mm
Caster trail	21mm

2.2 Ride and Roll rate calculations

Ride and roll rate calculations are one of the most important parts of suspension design. The formulae for calculations in the sheet were taken from [1]. Purpose of the car is again the basis for this. As mentioned above comfort and performance both are important so an optimum balance between both is found and the ride frequencies were assumed accordingly. Major aim is to avoid use of Anti roll bar which causes packaging issues and adds weight. Hence the spring should be soft enough to provide comfort and stiff enough to aid in performance. So multiple iterations are done to find the spring stiffness and roll gradient of the car. The roll gradient decides the rolling tendency of the car and based on the maximum lateral acceleration, the maximum roll the car experiences can be calculated and hence the geometry can be tested accordingly. If the geometry is not holding properly for a given roll angle, roll gradient can be decreased by increasing height of roll centers in the front and rear. The negative ARB stiffness in the calculation sheet indicates ARB can be omitted.

Description	total	front	rear	%front
Weight of the car(kg)	1000	550	450	55
Weight of the car(lb)	2204.62	1212.541	992.079	55
sprung weight(kg)	880	490	390	55.68182
sprung weight(lb)	1940.066	1080.264	859.8018	55.68182
CG height(mm) h	475			
CG height(in) h	18.6675			
Wheelbase(mm) l	2950			
Wheelbase(in) l	115.935			
Unsprung weight(kg)	120	60	60	
Unsprung weight(lb)	264.5544	132.2772	132.2772	
RIDE ANALYSIS				
Ride frequency(hz)		1.8	2	
ride rate(lb/in) Krf		178.8071	175.6986	
ride rate(N/mm)		31.28372	30.73986	
tire rate(lb/in)	1143.3			
tire rate(kg/m)	20390.35			
wheel centre rate		211.9562	207.6023	
installation ratio		0.923	0.917	
Spring rate (lb/in)		248.7956	246.8843	
Spring rate (N/mm)		43.52874	43.19435	

Fig -1: Ride Analysis

	Total	Front	Rear
Roll center heights(mm)		47.8	56.2
Roll center heights(in)		1.87854	2.20866
Tire static loaded radius(mm)		304	304
Tire static loaded radius(in)		11.9472	11.9472
Track(mm) t		1530	1530
Track(in)		60.129	60.129
Roll gradient(deg/g)	3.280882		
ROLL ANALYSIS			
Sprung mass CG height(in)	19.5839		
Rolling moment arm(in)	17.55906		
Rolling moment/g (lb-ft/g)	2838.811		
Rolling moment(N-m/g)	391.9671		
Roll rate(lb-ft/deg)	865.2586		
Roll rate(N-m/deg)	119.4701		
Individual spring roll rate(lb-ft/deg)		470.1648	461.9911
Individual spring roll rate(N-m/deg)		64.91773	63.78915
Total roll rates that spring provide(lb-ft/deg) K#s	932.1559		
Total roll rates that spring provide(N-m/deg)	128.7069		
ARB stiffness(lb-ft/deg) K#b	-66.8974		
ARB stiffness(N-m/deg)	-9.23681		

Fig -2: Roll Analysis

3. SELECTION OF TYPE OF FRONT SUSPENSION AND SUSPENSION GEOMETRY DESIGN

As it is A FWD vehicle all the transmission parts like differential, gearbox, half shafts are present in the front which causes the issue of interference with suspension parts. So various types were considered to meet the purpose considering a balance between performance, compactness, ease of manufacturing, cost and weight

3.1 Double wishbone suspension

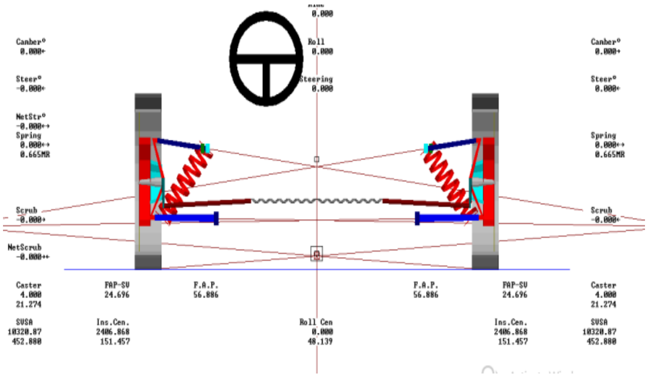

Fig -3: Double wishbone geometry

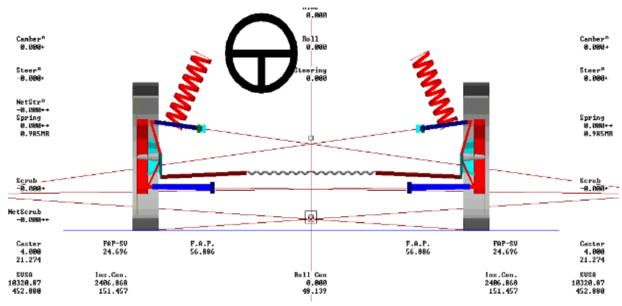

Fig -4: Double wishbone geometry with spring on top

It is a compact system and very customizable in terms of geometry design but due to interference of spring with the half shaft boot cannot be implemented.

Spring can be mounted on top but due to increase in CG and difficulty of manufacturing mounting member on chassis for spring it is discarded.

3.2 Macpherson strut

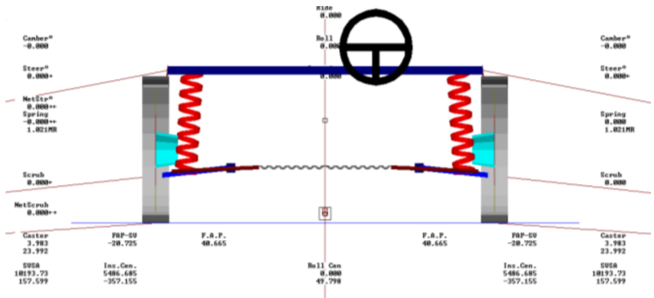

Fig -5: Macpherson strut

Passenger cars generally use this at the front as it is very compact and is relatively cheap and effective. But this also increases the CG and ride height of front substantially. Camber change is also high during ride and roll so it isn't acceptable.

3.3 Trailing and Semi trailing arm

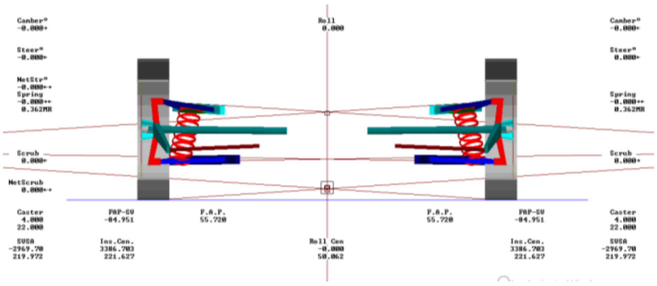

Fig -6: Semi trailing arm

Most modern passenger cars trailing arm due to it occupying very less space and not inducing any camber. But as the A arms are parallel to the tire, the load the arms have to take is very high and hence it should be strengthened which adds weight. The roll center of this arrangement is on the ground which means the front will roll more than the rear and hence more bending of the structural members, and reduces the transient stability of the car.

Semi trailing arm allows customization when compared to the trailing arm as the roll center height can be changed. It resulted in a very compact system with required parameters. But as the A arms are at a very tight angle, design of upright would be very tough and as the spring is inclined with respect to 2 axis the efficiency of the system is less as shown in fig. 6.

3.4 Pull rod suspension

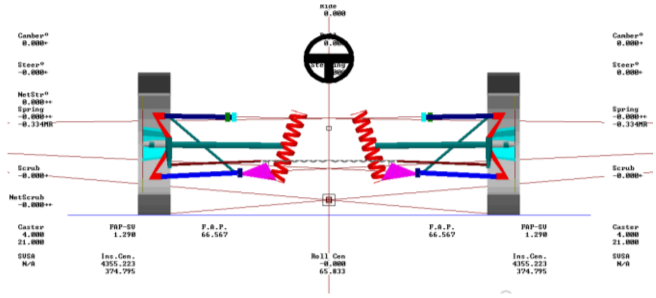

Fig -7: Pull rod suspension

Pull rod suspension allows us to avoid interference as the pull rod comes from the upper wishbone. It also lowers the CG of the car, hence decreasing the rolling moment arm. It provides us with customizability of front and side view geometry to design as per our requirements. But as it can be seen in fig. 7 it takes up a lot of space at the center of the car which is not available and hence is discarded

3.5 Push rod suspension with bell crank titled upwards

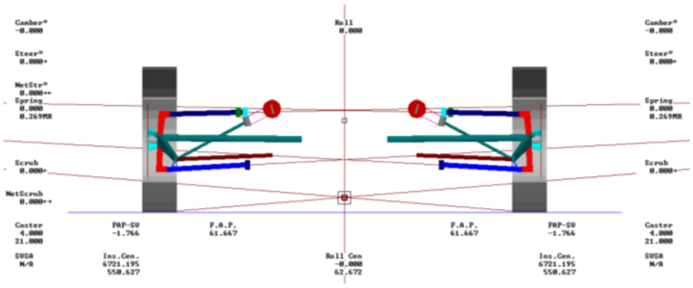

Fig -8: Push rod front view

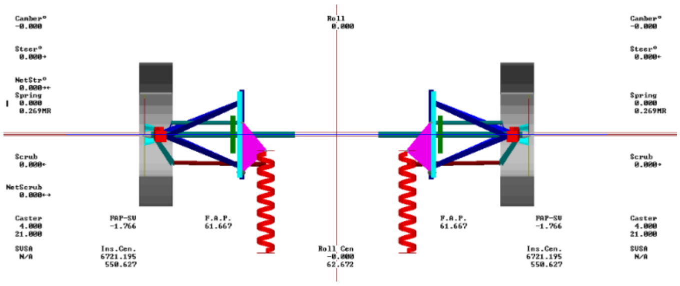

Fig -9: Push rod top view

To avoid interference with half shaft the push rod can be placed at an offset to the half shaft once the exact mounting position is known. Rather than placing the spring at the center, we can incline the bell crank in such a way that the spring is perpendicular to the push rod, hence saving space and achieving required motion ratio.

As the bell crank is placed at an angle to the push rod, it will take forces in 2 components which can lead to the bending of the bell crank. So by mounting the bell crank at the lower part of the chassis member the inclination of

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pushrod is decreased which decreases the bending effect. But due to difficulty in placing the member to hold the spring on the chassis at that specific height and position, and also increase in CG lead to this design being discarded

3.6 Inclined Push rod suspension

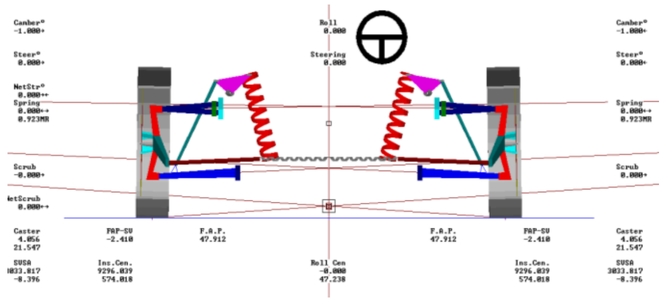

Fig -10: Inclined Push rod rear view

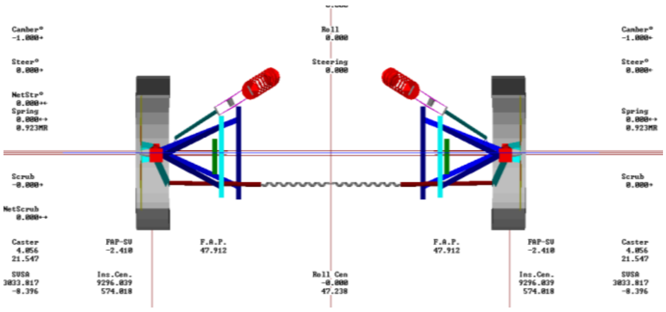

Fig -11: Inclined Push rod top view

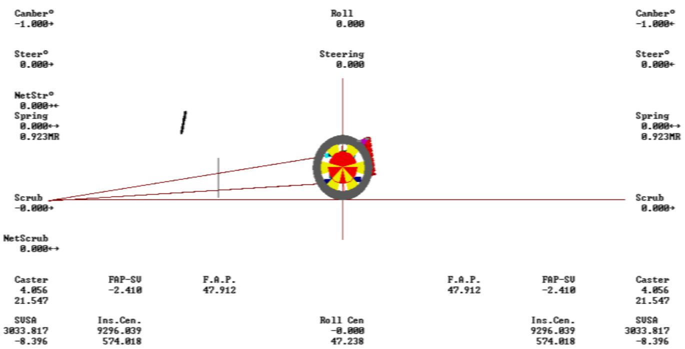

Fig -12: Side view geometry

Kingpin axis	3.023 degrees
Scrub radius	21.102
Caster	4.056 degrees
Caster trail	21.547

Fig -13: Final geometry parameters

Table -3: Specifications under static condition

Static camber	-1
FVSA(Front View Swing Axis)	9298.03mm
Motion ratio	0.923
Roll center height	47.238mm

With this setup there are no problems of interference and mounting of suspension parts. The push rod was placed at an angle of 30 degrees to the line passing through the center of the tire when seen from top view and all parts (push rod, bell crank, spring) were placed on that plane. Spring position couldn't exceed a certain value in the front so the Bell crank was designed in such a way to provide desired motion ratio and also attain required spring length.

Motion ratio is desired to be closer to 1 to use full efficiency of the spring. This reduces the drastic forces on the lower wishbone also. Due to packaging issues a MR of 0.923 was achieved which is acceptable.

Roll center height decides behavior of the system in case of lateral load transfer. The rolling moment arm which is the difference between the CG and RC height defines the roll gradient. Too low RC leads to high roll moments and high RC leads to high jacking forces. So based on desirable roll gradient RC height from ride roll rate calculation sheet is taken. Value of 47.238mm is acceptable.

The wishbone lengths too were kept short in such a way that the roll center movement is not a lot, so difference in lengths of lower and upper wishbone was increased. This helps in improving the transient stability of the car during various maneuvers. Geometry was adequately changed from the side view too, so that the wheels move properly over a bump by tilting the arms such that they coincide at the contact patch of rear wheel. This lets the wishbones react to the bump better and reduces pitching in the car.

This system further has the advantage of the bell crank allowing for various motion ratios without changing the hard points. So based on requirement of tire conservation or maximum performance can be achieved with the same spring

It is easy and economical to manufacture a double wishbone push rod actuated suspension and hence appropriate for our need.

4. TESTING AND STUDY OF FRONT GEOMETRY

Once the geometry is made it needs to be tested for various cases to validate if the design matches our need. The tests are performed at extreme cases of ride and roll because, if it runs well in the worst case it works for every scenario. The geometry is tested for maximum ride of the car (70mm) in both bounce and rebound and maximum roll angle of 3 degrees.

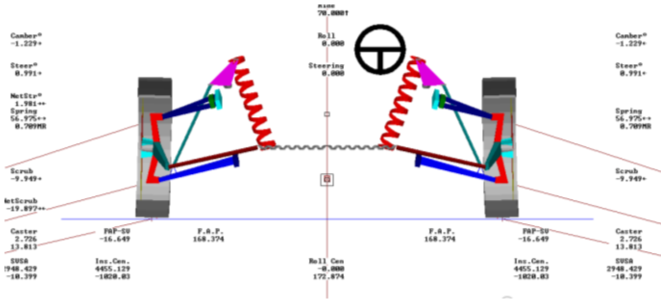

Fig -14: Test for 70m ride

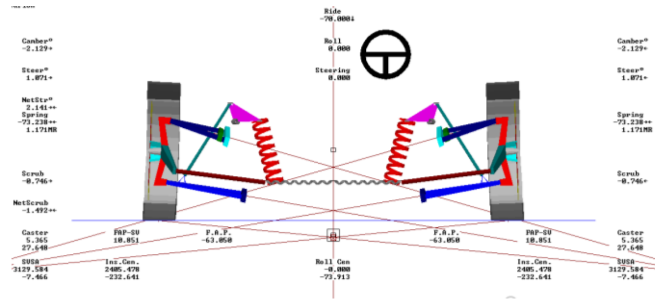

Fig -15: Test for -70m ride

Since the geometry was a long and short arm unequal unparallel geometry, the results are as predicted. When the wheel takes the bump or rebound, the upper and lower arms form different arcs, which is actual cause for the camber. Even though the arms are shortened compared to the previous car, the camber change is not high so the geometry is successful in this case.

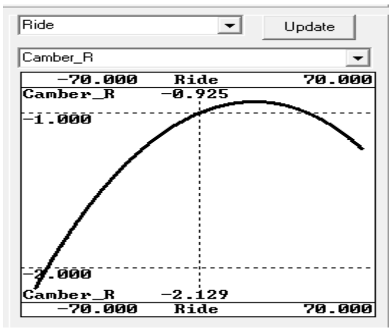

Fig -16: Camber R vs Ride graph

A graph between the ride and camber are generated in both bounce and rebound are studied. Camber at the extremes are noted and analyzed and the results were found acceptable. Due to the introduction of steering axis inclination positive camber is induced because of presence of bump steer. Positive caster angle induces negative camber, these two factors almost balance each other and hence the camber is only slightly changed. The negative camber is due to static -1 degree camber which is acceptable as it provides stability

Table -4: Camber change for ride

Ride(mm)	Camber(deg.)
Bounce(70)	-0.925
Rebound(-70)	-2.129

4.1 Test for 2 degree roll and 60mm movement of rack (without considering banking)

The chassis undergoes rolling when the car is steered in the corner. So in this test both steer and roll of the chassis is taken into account for analyzing the geometry. From the roll gradient we find the maximum roll of the car based on the maximum lateral acceleration achieved while cornering. This was calculated to be 2 degrees in case of maximum steer.

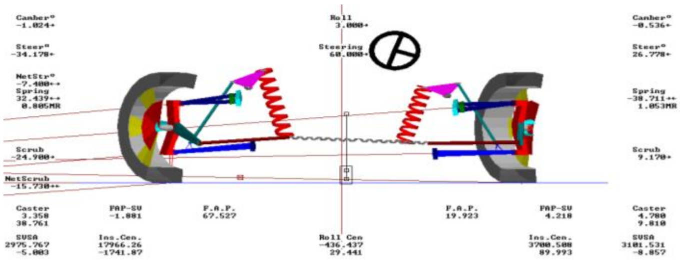

Fig -17: Test for roll

The geometry behaved as expected, since it was SLA type, the outside wheel is expected to go positive camber and the inner wheel to go negative camber, Due to the SAI and caster angle balancing each other's camber effects and the initial -1 degree camber both tires have negative camber, so overall desirable camber was achieved when the car takes a corner.

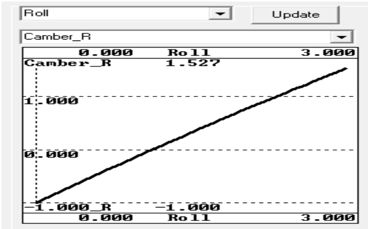

Fig -18: Camber R vs Roll graph

The curve has a positive slope indicating the gain of positive camber on the right wheel (outer wheel).

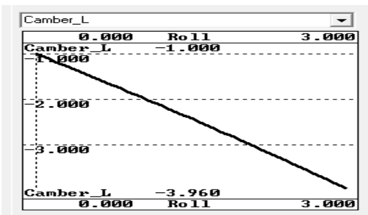

Fig -19: Camber L vs Roll graph

The curve has a negative slope indicating the gain of negative camber on the left wheel (inner wheel).

Note: the above graphs only show change in camber due to roll and doesn't consider changes due to steering.

Table -5: Camber change for roll

Roll angle(deg)	Rack movement(mm)	Camber on right wheel(deg) due to roll and steer
0	0	-1
2	40	-0.579
3	60	-0.536

5. SELECTION OF TYPE OF REAR SUSPENSION AND SUSPENSION GEOMETRY DESIGN

Unlike the front, rear has no interference issues with transmission parts but there needs to be space provided at the center of the car for the battery to be removed from the rear of the car. Considering the selection procedure for the front, few of the suspension types were discarded for their drawbacks that would affect the rear too. Comparison is made between Double Wishbone and Push rod suspension setup.

5.1 Double wishbone suspension

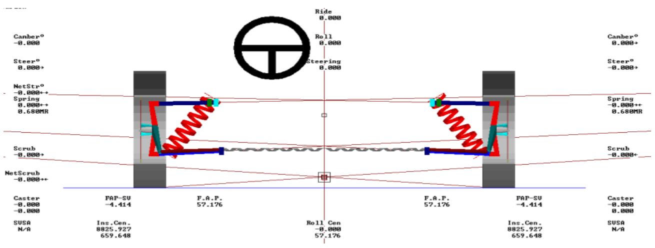

Fig -20: Double Wishbone rear geometry

This setup allows us to have a very compact suspension system. The front and side view geometries can be attained and is easy to manufacture. But the major seen was that due to the large spring inclination neccesary to fit the spring in the motion ratio decreased to 0.6 which reduces the sprin efficiency massively. Hence is discarded.

5.2 Push rod suspension

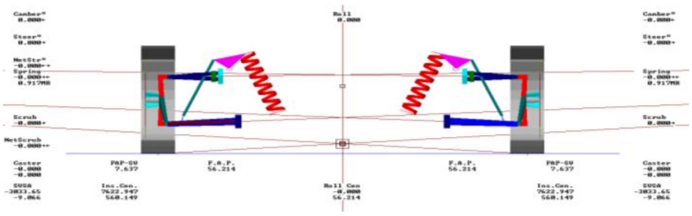

Fig -21: Push rod rear geometry

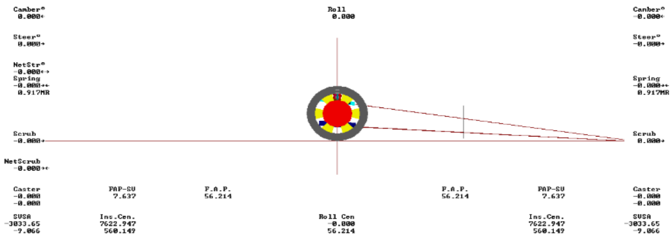

Fig -22: Side view geometry

Table -6: Specifications under static condition

Static Camber	0deg
FVSA(Front View Swing Axis)	7622.947mm
Motion ratio	0.917
Roll center height	56.214mm

This setup takes more space than the double wishbone setup but by making the wishbones short enough the space requirement was met. The upper arm was made considerably shorter than the lower to decrease lateral roll center movement during rolling of the car which improves the transient stabiltiy of the car.

The required roll center to maintain the roll gradient was achieved by inclining the arms from the front view suitably. The side view geometry was also done such that the arms coincide at the contact patch of front tires to decrease pitching.

6. TESTING AND STUDY OF REAR GEOMETRY

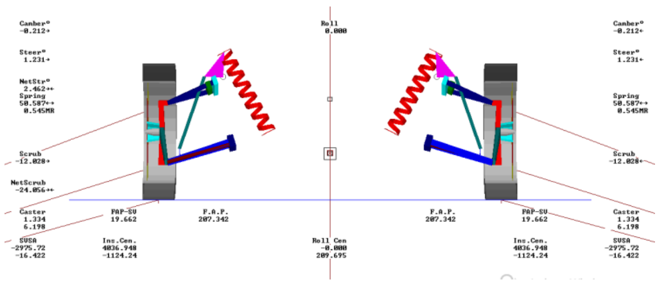

Fig -23: Test for 70mm ride

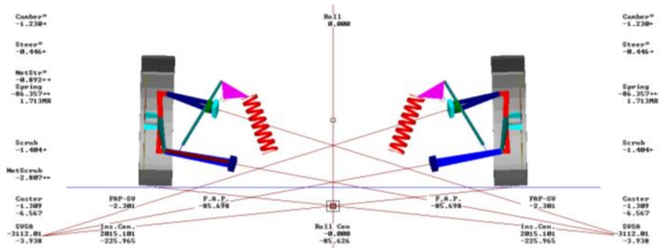

Fig -24: Test for -70mm ride

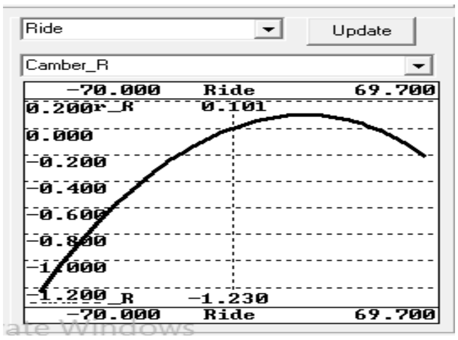

Fig -25: Camber R vs Ride graph

Table -7: Camber change for ride

Ride(mm)	Camber(deg)
Bounce(70)	0.101
Rebound(-70)	-1.23

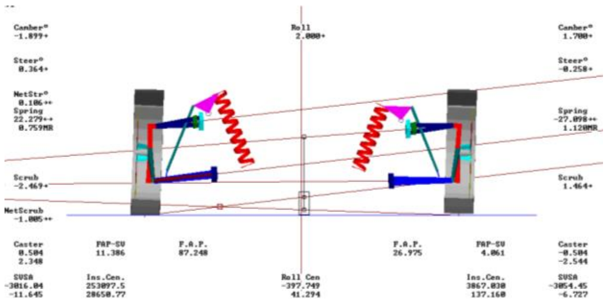

Fig -26: Test for roll

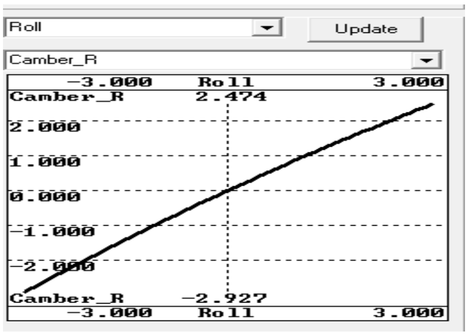

Fig -27: Camber R vs Roll graph

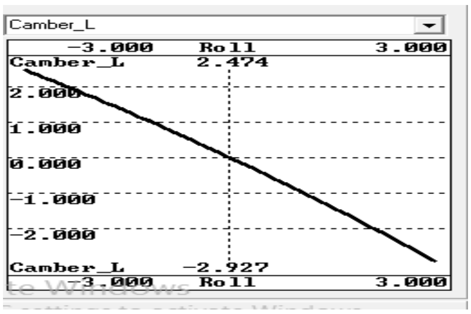

Fig -28: Camber L vs roll graph

Table -8: Camber change for roll

Roll angle (deg)	Camber on right wheel (deg)
0	0(static)
2	1.7
4	3.194

7. CAD MODELS

Fig -29: Front suspension geometry CAD model

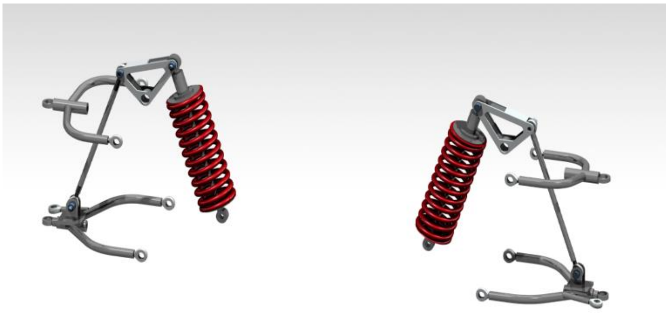

Fig -30: Rear suspension geometry CAD model

8. ACTUAL ASSEMBLY

Fig -31: Front suspension geometry actual assembly

Fig -32: Rear suspension geometry actual assembly

8. CONCLUSION

Table -9: Results from ride and roll calculations

Description	Total	Front	Rear
Ride frequency(hertz)		1.8	2
Wheel center rate (N/mm)		37.08	36.32
Installation ratio		0.923	0.917
Spring rate (N/mm)		43.53	43.19
Roll gradient (deg/g)	3.3		
Roll rate (N-m/deg)	119.47		

For the purpose of a solar electric passenger vehicle aiming for the best of comfort and performance, an ***unequal, non-parallel double wishbone pushrod suspension*** with modifications according to specific requirements, is determined to be the best in terms of providing a balance between comfort, control, performance, compactness, ease of manufacturing, cost and weight.

After multiple iterations the optimum geometry was obtained with very less camber change for maximum ride and roll scenarios, pitching has been reduced, giving a soft and comfortable ride. All this helps in decreasing the losses in power that is being generated by the solar panels and improving the overall efficiency of the car.

Table -10: Final comparison

Suspension system	Compact	Interference	Design customizability	Performance	Motion ratio	Mfg. ease
Macpherson	High	No	Med	Med	Med	Med
DWB	Med	Yes	High	Low	Low	High
Trailing arm	High	No	Low	Low	Med	Med
Semi- trailing	Med	No	Med	Med	Low	Low
Pull rod	Very low	No	Very high	High	Variable	High
Push rod	Low	Yes	Very high	High	Variable	High
Modified pushrod	Med	No	Very high	High	Variable	High

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BIOGRAPHY

Navneet Singh Dhir
Mechanical Graduate from MIT Manipal, who aims to build a career developing technologies that help achieve widely used sustainable mode of transportation