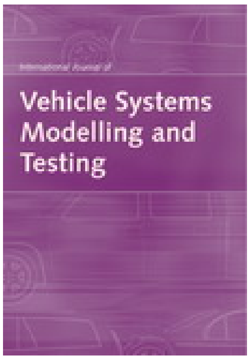

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Design and optimisation of double wishbone suspension for high performance vehicles

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Design and optimisation of double wishbone suspension for high performance vehicles

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Abstract: Technology is an unpredictable friend, nudging us forward with one hand while demanding exorbitant tolls with the other. With the control on the parameters like camber angle, scrub radius, roll centre height etc., the careful design of double wishbone allows the motion control of wheel throughout the suspension process. The paper considers the high-performance vehicles and impinges the issues related to optimisation of double wishbone suspension system. The analysis will help the researchers and practioners to frame the strategies to develop a system with reduced vibrations and noise.

Keywords: camber; scrub radius; caster angle; bellcrank; A-arms; tyre-data; suspension geometry.

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1 Introduction

Double wishbone is a suspension system (independent type), i.e., wheels are not bound by a rigid axle. Every wishbone has two mounting points on chassis and one on knuckle joint. Double wishbone suspension system is characterised by its low installation height and large width between left and right sides which makes particularly advantageous and sporty suspensions. Although it takes up more space, it prevents severe load reversal reactions.

Double wishbone suspension system has various components which include two wishbones or A-arms, dampers, bellcranks, the connecting rods and tyres. Each component has its own importance in proper functioning of the suspension system. When tyre goes in a drop or bump, it transfers the force to the upright, where both the wishbones are attached. These wishbones move downwards or upwards resulting in transfer of force to the connecting rod. The connecting rod rotates the bellcrank, which is pivoted at one end and then bellcrank in turn transfers the force to dampers.

Double wishbone suspension system is the framework that helps to provide more free parameters than other suspension systems. The details of wishbone diagram is shown in Figure 1. In order to cope up with the road conditions, A arms can be fixed to different angles. With this, the parameters (camber angle, roll center height, swing arm length) can be found out. As a result of vertical suspension movement of upper and lower arm, negative camber angle increases. In view of this, the outside tyres has more contact with road surfaces and hence better stability of automobiles. This suspension system is much more rigid and stable, thus steering and wheel alignments are constant even when undergoing high amount of stress and hence, handling performance also improves. This suspension system also has some limitations. It is quite complicated to design and calculate the various suspension geometry parameters which affect the functioning of the suspension. This system includes many parts, and thus every time any of these malfunctions or fails, and the whole system fails.

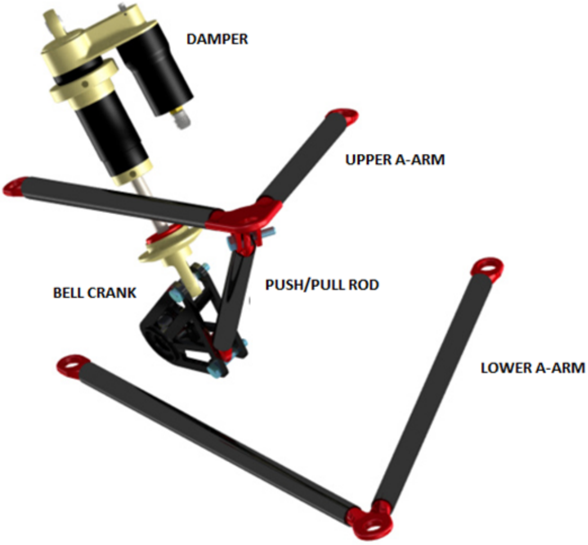
The research conducted in this area is tabulated in Table 1.

Table 1 Research findings from literature

S. No.	*Author*	*Findings*
1	Kavitha et al. (2018)	The authors analysed the distinct attributes of wishbone suspension system
2	Arikere et al. (2010)	They studied the various versions of wishbone suspension system using different geometric angles
3	Saurabh et al. (2016)	The authors considered the racing automobiles and introduced the layouts of suspension systems
4	Blanco and Munoz (2013)	The authors took into consideration various parameters of double wishbone approach and optimised those parameters
5	Kang (2008)	He studied the dynamic behaviour of spring used in a suspension system
6	Mahroogi and Narayanan (2019)	They designed and analysed the double wishbone suspension system considering the layout of shock absorber and spring
7	Nam et al. (2019)	They studied the FEM model with objective to analyse the dynamic behaviour for wishbone suspension system considering link flexibility

Double wishbones are usually considered to have superior dynamic characteristics as well as load-handling capabilities, and are still found on higher performance vehicles and hence this system is commonly used in racing cars and high performance vehicles.

Figure 1 Rendered image of double wishbone suspension design (see online version for colours)



Source: CATIA Render

2 Design considerations

Following parameters need to be considered while designing suspension geometry:

1 *Camber*: The angle between the vertical axis of the wheels used for steering and the vertical axis of the vehicle viewed from front or rear.

In our suspension design camber angle varies from 0° to -3°.

2 *Caster*: The angle between the line joining the centre of upper ball joint and centre of lower joint and the vertical line to the ground surface.

3 *Roll centre*: It is the point in a vehicle at which cornering forces in the suspension are reacted to vehicle body.

In our suspension design, front roll centre height is 27 mm and rear roll centre height is 56 mm with minimal lateral migration.

4 *Slip angle*: The angle between a rolling wheel's actual direction of travel and the direction towards which it is pointing. Our car operates in a range of –12° to +12°.

5 *Lateral acceleration*: It can be defined as an acceleration created, when a vehicle corners that tends to push a vehicle sideways. Because of centrifugal force, the vehicle is pushed outward.

$$A_y = V^2/R$$

where

A_y = Lateral acceleration

V = Velocity of the vehicle while cornering

R = Radius of turn/corner.

V exponentially increases with increase in A_y. Thus, we try to maximise the value of lateral acceleration.

6 *Longitudinal acceleration*: This is the acceleration attained in straight line conditions in either braking or acceleration state.

Acceleration is generally limited by the power of the engine but deceleration (braking) is governed by tyres. Hence, if the value of this parameter is more, the car can brake later for the next corner, thereby decreasing the time it takes to cover the straight.

7 *Roll stiffness*: It is the resistance to roll in turns, independent of its spring rate in vertical direction. High roll stiffness provides better stability while cornering. Roll stiffness for our car was kept at 800 Nm/deg.

8 *Spring rate*: It is amount of weight required to deflect spring by 1 inch.

In our car, we used springs with spring rate varying from 125lbs/in – 175lbs/in, depending on various conditions.

9 *Motion ratio*: It is the ratio of the displacement of the spring to the displacement of the wheel.

MR = Spring Travel/Wheel Travel

Motion ratio for our suspension design was kept at 0.758 for the front and 0.744 for the rear suspension.

10 *Wheel rate*: It is same as the spring rate but measured at the wheel instead of where the spring is attached to the linkages.

$\mathrm{WR} = \mathrm{SR}/(\mathrm{MR})^2$

where

WR = Wheel rate

SR = Spring rate

MR = Motion ratio.

Wheel rate for our suspension design was calculated as 12.24 for the front and 24.66 for the rear.

3 Problems faced

1 *Inefficient performance of suspension geometry due to lack of tyre-data*

While working on a project of designing and fabricating a single seater, double wishbone suspension system, formula style car, we have gone through following observations:

- In 2010 and 2011, car's suspension was design without using any parameter of tyre data. We simulated that design on Hockeinheimring using lap-sim software from OptimumG, and got the following result for that suspension system:
  - Time taken to complete Autocross (1.2 km approx) is 130 s.
  - Time taken to complete acceleration is 5.1 s.
- In 2013, we were successful to determine a few of the parameters of tyre data and designed our suspension using those parameters. Following are the results obtained after the simulation of 2013's suspension design using Optimum-G lap-sim software:
  - Time taken to complete Autocross is 109 s.
  - Time taken to complete acceleration is 4.4 s.

2 *Bad suspension design due to inaccurate values of suspension parameters*

As we did not having the tyre data and much knowledge about determining the various suspension parameters such as camber, toe angle, roll centre height, ride height, etc., we took assumptions and designed the suspension geometry on WinGeo software. On WinGeo, we need to define all parameters as per our requirement. But, we did not have accurate values to define our suspension geometry and hence, while testing of the car, various problems were faced with the vehicle behaviour.

The problem includes popping out of the ball of the bearings at the ball joint of A-arms, understeer due to wrong toe rod angle selection, more wearing of outside of the tyre due to excessive camber, less allowable bounce and jounce, difference in actual ride height than the one designed on WinGeo software.

3 *ARB was not used, leading to poor roll characteristics and difficulty in tuning*

Due to lack of proper ride and roll rates calculation, the anti-roll bar was not used in previous design which led to poor roll characteristics and caused difficulty in tuning.

4 Possible solutions

Tyres are one of the most critical factors affecting the suspension. Reading upon tyre data, various parameters of the suspension system can be defined. By analysing the various graphs from the tyre data shown below (refer Figure 2–9), the parameter values were determined.

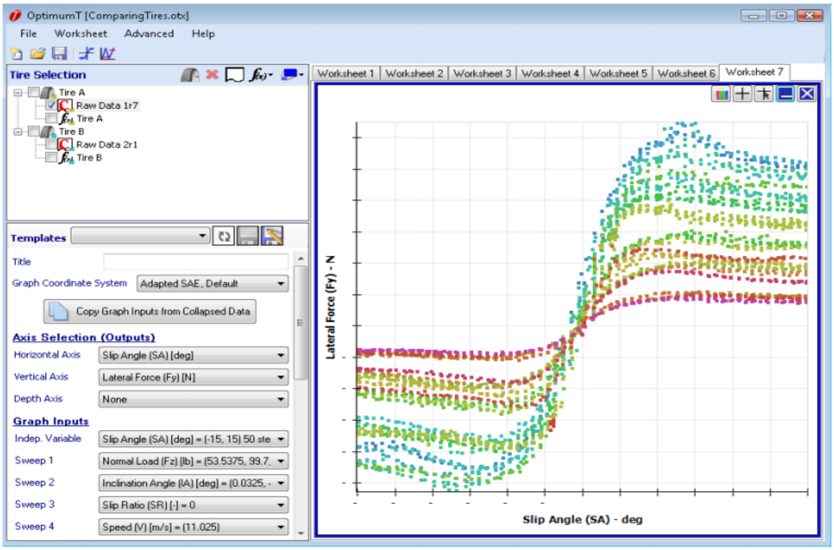
- Taking the above points into consideration the designs were modified and simulated again for the 2014 car, now with maximum known parameters to get the following results:
  - Time taken to complete autocross will be 98 s.

- Time taken to complete acceleration will be 3.96 s.

These timings make car competitive against other teams.

- Use of Anti-Roll Bar increases the roll stiffness of the car and gives it more stability. Hence, it is being incorporated in our suspension geometry to optimise it further.
- Analysis and simulation of suspension geometry on CAD and analysis software ensured the proper functioning of suspension and better performance of the car.

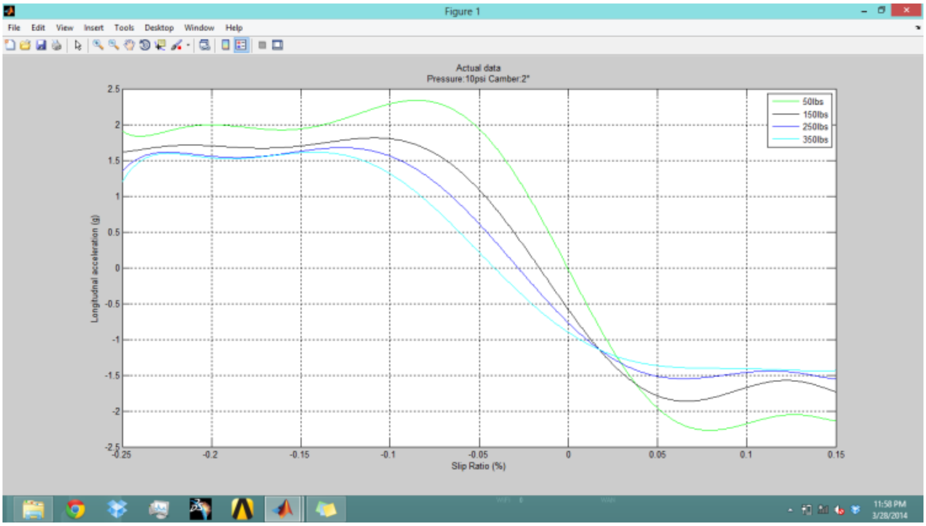
Figure 2 Tyre-comparison data and graph (see online version for colours)



Title
Graph Coordinate System: Adapted SAE, Default
Horizontal Axis: Slip Angle (SA) [deg]
Vertical Axis: Lateral Force (Fy) [N]
Depth Axis: None
Indep. Variable: Slip Angle (SA) [deg] = (-15, 15) 50 ste
Sweep 1: Normal Load (Fz) [lb] = (53.5375, 99.7,
Sweep 2: Inclination Angle (IA) [deg] = (0.0325, -
Sweep 3: Slip Ratio (SR) [-] = 0
Sweep 4: Speed (V) [m/s] = (11.025)

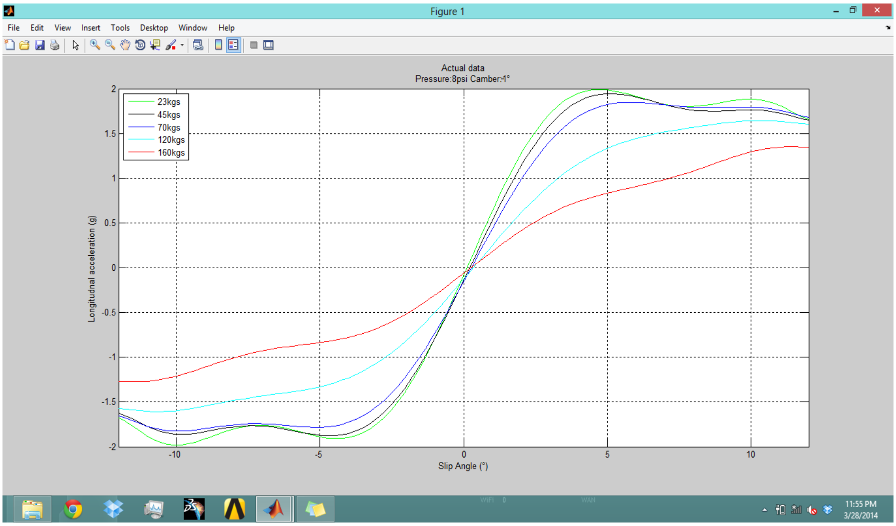
Source: Optimum-T

Figure 3 Tyre-data camber 2° and pressure 10 psi (see online version for colours)



Source: Hoosier tires

Figure 4 Tyre-data camber –1° and pressure 10psi (see online version for colours)



Source: Hoosier tires

5 Optimisation of suspension geometry using tyre data

The only interaction the car has with the road is through the tyres. All the forces a car experiences from the road act at the tyres and then propagate to the different parts of the car. Hence, understanding the tyres is vital while designing a car and especially while working on its suspension geometry.

Every tyre is different from the others due to differences in its composition, tread pattern etc. A tyre will behave differently under different conditions and it is the suspension designers' job to make full use of the tyre's performance. The performance of the tyre with respect to different parameters is given by the tyre data. This data when plotted and properly interpreted can give us an insight into when the tyre will perform at its peak.

Here is an example of how tyre data was used to decide three key vehicle parameters, namely tyre vertical load, camber angle and tyre pressure. The tyre used was a Hoosier 20.5×6 R-13 R25B Road Racing tyre.

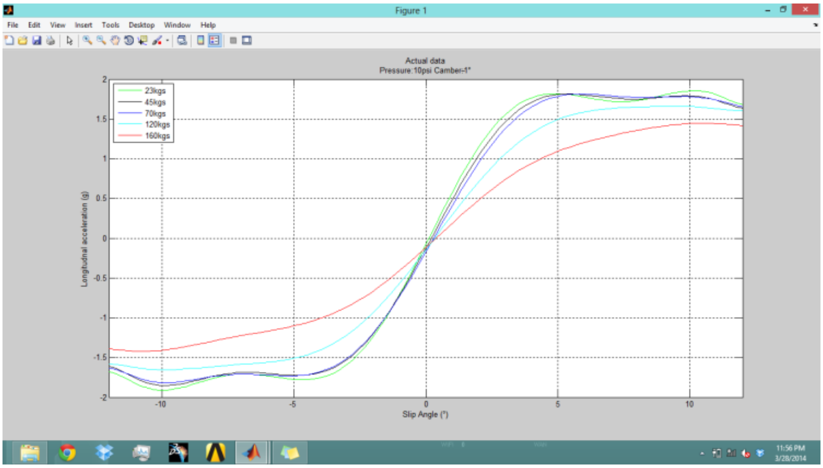
The following graphs are between normalised lateral force F_y/F_z and slip angle (SA).

Firstly we restrict the slip angle from –12° to + 12° as a Formula Student car won't go beyond these values. So defines the usable range of slip angle.

The above graph was plotted for 5 load cases which were 23 kg, 45 kg, 70 kg, 120 kg, 160 kg denoted by the curves of different colours.

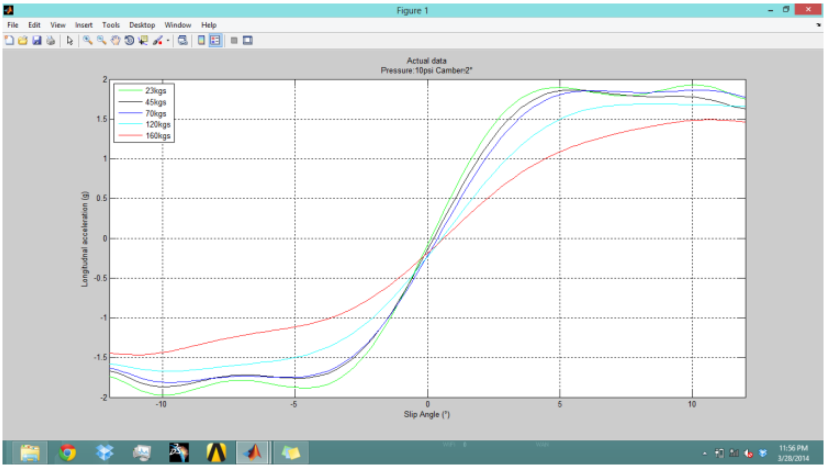
First we kept the pressure constant at 10 psi and analysed the graphs for –1° and –2° camber. It was clear that highest normalised lateral force (NF_y) was occurring at 23 kg of load. But the weight of the car being 240 kg and taking into consideration load transfer, it was calculated that the load transfer at best could be kept to 120 kg. Thus analysing the two graphs at 120 kg load it was found that the tyre gives more normalised lateral force at –2° camber at 10 psi.

Figure 5 Tyre-data camber –1° and pressure 8 psi (see online version for colours)



Source: Hoosier tires

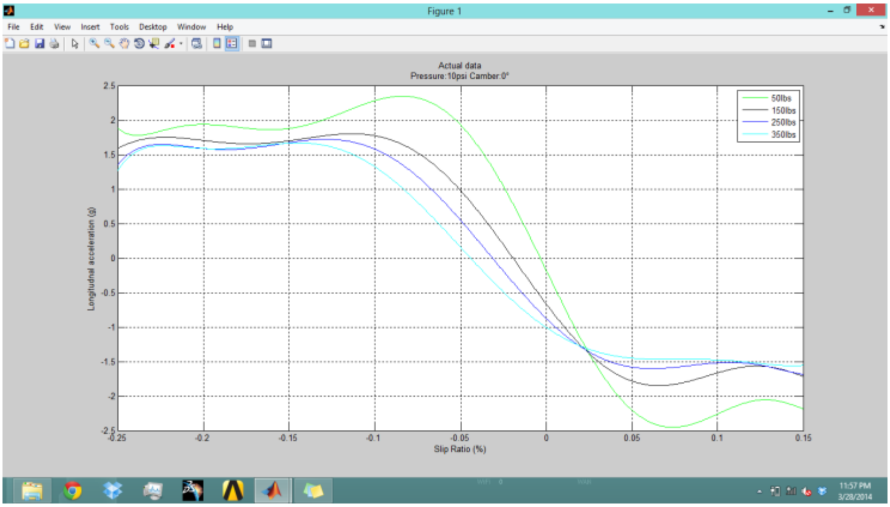
Figure 6 Normalised lateral force vs. slip angle (–1° camber@ 10 PSI) (see online version for colours)



Other such graphs were analysed for example keeping the camber constant and varying pressure from 10 psi to 8 psi. After analysis it was clear that during cornering the required values of the parameters were 120 kg vertical load on the tyre, –2° camber angle and 8 psi tyre pressure on the loaded tyre.

Tyre pressure can be easily set at the start of a run and will stay constant throughout the run. But, camber angle and vertical load on tyre are dynamically changing quantities. Hence the suspension links were designed such that the camber control during heavy cornering ensured that the tyre had a –2° camber on the outside or laden tyre. Similarly various load transfer calculations were done including ride and roll characteristic calculation to get the appropriate amount of load i.e., 120 kg on the outside tyre.

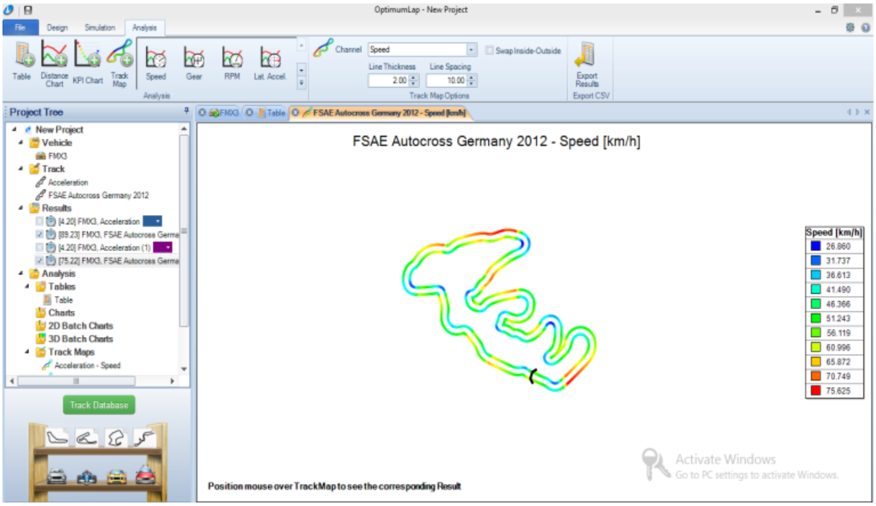
Figure 7 Longitudinal force vs. slip ratio (0° camber@ 10 PSI) (see online version for colours)



Source: Hoosier tyres

Similarly various other graphs were looked into for examining the tyre performance like the graph between normalised longitudinal force (NF_x) and slip ratio (SR) at various camber angles, loads and tyre pressures to determine parameters for optimum straight line performance i.e., acceleration and braking. Combined lateral and longitudinal characteristics graph of tyre is checked as car usually possesses combination of both rather than purely any one.

Figure 8 Lap-simulation comparing the laps of both the vehicles (Outer-with tyre data, inner- without tyre data) (see online version for colours)



Channel: Speed

Line Thickness: 2.00

Line Spacing: 10.00

Position mouse over TrackMap to see the corresponding Result

Source: Optimum-G lap-sim software

Table 2 Comparison of with tyre and without tyre data

Parameter	*Without tyre data*	*With tyre data*
Lateral acceleration	1.1 g	1.7 g
Longitudinal acceleration	1.0 g	1.5 g
Lap time	89.23 s	75.22 s

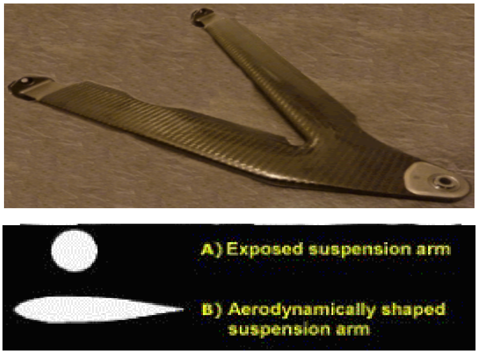
Taking into consideration the on track accelerometer data found on previous year's car and expected longitudinal and lateral acceleration data from the analysis of tyre data, a lap simulation was run on Optimum-Lap software. This shows (refer Table 2) a huge performance increase which justifies the time and energy spent on designing the suspension using tyre data.

6 Future scope

Continued development and optimisation requirements have now forced engineers to consider the smallest areas possible which may offer even slightest ways of better performance.

- We can further optimise our designs by incorporating A-arms with due consideration to minimise already low aerodynamic drag they face.
- The components are made using materials with better strength to weight ratio like carbon fibre and titanium.
- Reduction in tyre wear and tear using more optimised suspension geometry.
- Better understanding of vehicle dynamics under wet weather conditions.

Figure 9 Aerodynamically shaped suspension arm (see online version for colours)



7 Conclusion

This paper tells about the basic parameters and how these parameters affect the optimisation of suspension geometry. Tyre data, material used to manufacture the suspension components and determination of the geometry parameter values play an important role in the suspension geometry optimisation. As shown above, we can see that the benefits of working for the performance improvement of a double wishbone suspension far outweigh the time and energy requirements and are thus very beneficial.

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