International Journal of Mechanical Engineering and Technology (IJMET)
Volume 7, Issue 2, March-April 2016, pp. 168–176, Article ID: IJMET_07_02_018
Available online at
http://www.iaeme.com/IJMET/issues.asp?JType=IJMET&VType=7&IType=2
Journal Impact Factor (2016): 9.2286 (Calculated by GISI) www.jifactor.com
ISSN Print: 0976-6340 and ISSN Online: 0976-6359

FORCE CALCULATION IN UPRIGHT OF A FSAE RACE CAR

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ABSTRACT

Forces are generated at the tire contact patch during various maneuvers of the car and transferred to the chassis through the suspension links. Calculating the forces on every link is important to design the suspension system as all the forces from wheel to the chassis are transferred by the suspension linkages. These forces have been calculated for all the links of a double wishbone suspension geometry. The load paths and FBD have been drawn and axial stress in the all the linkages.

Key words: Analysis of wish-bones, Upright, FSAE, Tire-data, etc.

Cite this Article Anshul Dhakar and Rishav Ranjan, Force Calculation In Upright of A FSAE Race Car. *International Journal of Mechanical Engineering and Technology*, 7(2), 2016, pp. 168–176.
http://www.iaeme.com/currentissue.asp?JType=IJMET&VType=7&IType=2

1. INTRODUCTION

Upright is a part of the wheel assembly which holds the hub and allows rotation of the wheel. The forces from the tire contact patch are transmitted by the upright to the suspension links.

The suspension geometry of a standard FSAE race car has been used for the calculation of the forces. The target values for concerning and braking have been set according to the tracks present in the FSAE International events. Tire data has been used to find out the friction coefficient at the contact patch which varies to the normal load on it.

The calculation of forces has been done in matrix form using MS Excel. The Free body diagram of the corner assembly and all the linkages are made.

2. FORCE CALCULATION

For the calculation of loads and load paths in wheel assembly, we need to take parameters from the geometry of the car, the track and the procured components such as Tire-data.

The following data have been obtained from standard FSAE race car:

Table1 Data from Suspension Geometry

WheelBase	1600.000	mm	1.600	m
CG height	300.000	mm	0.300	m
Track width (rear)	1200.000	mm	1.200	m
Track width (front)	1250.000	mm	1.250	m
Weight	300.000	Kg		
Front weight distribution	0.450			
Weight distribution in Y	0.500			
Tire dia	510.000	mm	0.510	m
Tire Width	206.000	mm	0.206	m
Rim Dia	13.000	in	0.330	m
Aspect Ratio	43.64%			

The accleration during various actions of the car (i.e. braking, cornering and throttling) are calculated using three equations of motion, considering accleration as constant through out the particular event.

$$v = u + at$$

$$s = ut + \tfrac{1}{2}at^2$$

$$v^2 = u^2 + 2as$$

Table 2 Data from FSAE track

Braking Event					
Braking speed	u	60.000	kmph	16.667	m/s
Braking distance	S	14.334	m		
Braking Time	t	1.720	s		
Braking Accleration	a	9.690	m/s2		
Braking g's	a	0.988	g		
Lateral (Skid-pad Event)					
Skid-Pad Radius	r	9.125	m		
Velocity during cornering	v	40.000	kmph	11.111	mps
Lateral Accleration	a	13.530	m/s2		
Lateral g's	a	1.379	g		
Longitudinal (Accleration Event)					
Maximum velocity	v	60.000	kmph	16.667	mps
Time for maximum veocity	t	3.000			
Longitudinal Accleration	a	5.556			
Longitudinal g's	a	0.566			

The braking acceleration is calculated when the car de-accelerates from a speed of 60kmph to 0kmph in a braking distance of half the skid-pad circumference. Similarly, lateral acceleration is calculated assuming the maximum cornering speed as 40kmph and a lateral acceleration of around 1.4g has been obtained. The various speeds assumed are from standard FSAE race cars participating in the event.

The coefficient of friction has been obtained from tire data which varies according to the normal load on the tire (F_Z).

2.1. Calculation of the forces during various maneuvers

The upright takes longitudinal force during braking and acceleration, and lateral forces during cornering. Hence, extreme forces are considered for a situation, when the car brakes during cornering. The total amount of traction is considered constant. The Circle of Traction shows the total amount of traction distributed between lateral forces and longitudinal forces.

2.1.1 Longitudinal Force Calculation

The longitudinal forces are developed by an upright during acceleration and deceleration. It is evident that a driver experience more g's of force during deceleration (braking). The mass transfer during braking has been calculated using the following equation:

(Mass Transfer) * 'g' * Wheel Base = (Mass of the Car) * Braking g's * 'g' * Centre of Gravity (Z).

The coefficient of friction is a variable obtained from the tire data which depends on the net normal force, F_Z after mass transfer. The system is considered as a beam with Frictional Force and the reaction forces on upright.

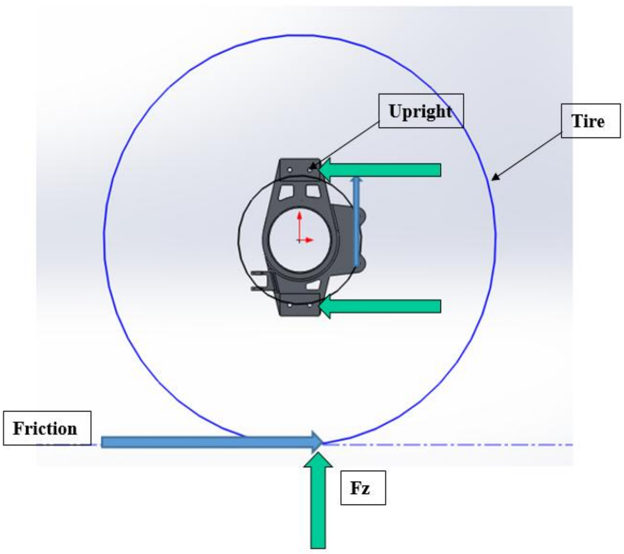

Figure 1 FBD during Braking (Side View)

The distances between the reaction forces and frictional force is obtained from the Suspension Geometry and Tire Specifications (such as Tire Diameter, Aspect Ratio, etc.)

The equations were put in MS Excel and the longitudinal forces during braking were obtained in the following table:

Table 3 Braking Force

Braking Maneuvre	Front Corner		Rear Corner	
Frictional coefficient	2.31		2.59	
longitudinal mass transfer on one tire	27.78	Kg	-27.78	Kg
vertical load on front wheel (on one wheel)	934.70	N	536.80	N
Frictional Forces on one wheel	2157.05	N	1388.87	N
Breaking Force on Upright	6370.06	N	4101.51	N
Dia of the Brake Calliper Bolts	0.17	m	0.17	m
Distance from Hub Centre to Lower A-Arm Point, b	0.087	m	0.115	m
Distance from Hub Centre to Upper A-Arm Point, a	0.094	m	0.095	m
Reaction at Upper Clamp	-2002.13	N	-925.91	N
Reaction at Lower Clamp	4159.18	N	2314.78	N

2.1.2 Lateral Force Calculation

The lateral forces are developed by an upright during cornering or while steering. The mass transfer during cornering has been calculated using the following equation:

(Mass Transfer) * ‘g’ * Track Width = (Mass of the Car) * Cornering g’s * ‘g’ * Centre of Gravity (Z).

Since the track-width of front and rear are different the load transfer is different in both cases. The reaction forces here as well is calculated considering beam structure. And the distances were again taken from the data extracted (Tire Data, Suspension Geometry, etc.)

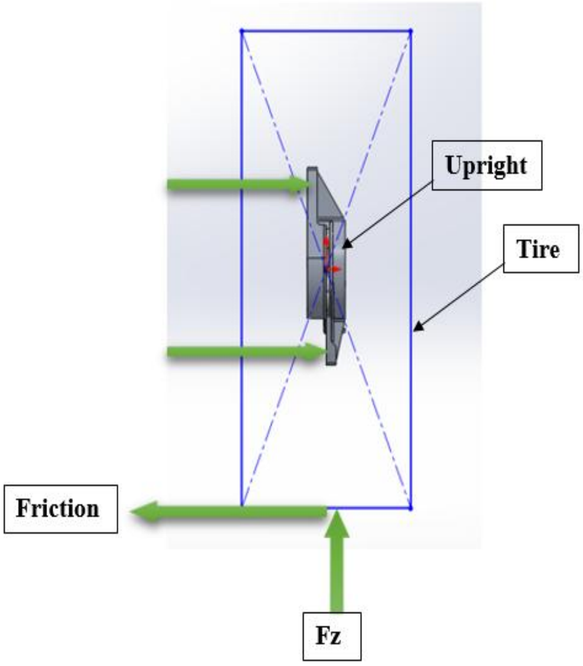

Figure 2 FBD during cornering (Front View)

The following table lists the net lateral forces on upright outboard points:

Table 4 Lateral Force

Cornering Maneuvre				
Lateral load transfer (front)	44.68	Kg		
Lateral load transfer (rear)	56.89	Kg		
	Front Corner		Rear Corner	
Vertical Load	1100.53	N	1367.42	N
Frictional Coeff, µ	2.19		2.00	
Frictional Force	2411.38	N	2739.65	N
Distance from Hub Centre to Lower A-Arm Point	0.087	m	0.115	m
Distance from Hub Centre to Upper A-Arm Point	0.094	m	0.095	m
Reaction at Upper A-Arm Point	-2238.18	N	-1826.44	N
Reaction at Lower A-Arm Point	4649.56	N	4566.09	N

2.1.3 Forces during extreme maneuvers

During braking, throttling and steering actions of the car, the maximum forces developed on an upright is when the car applies brake while steering. So the upright is designed for the above mentioned condition in which net transfer of mass is considered to calculate the force vector at the tire contact patch.

Table 5 Braking and Cornering Mass Transfer

total weight transfer on the Front corner	72.47	kg
total weight transfer on the Rear corner	29.11	kg
Total weight of front corner	139.97	kg
Total weight of rear corner	111.61	kg
µ	Front	2.00
	Rear	2.20
Fx, Front & Fy, Front (Magnitude)	2745.51	N
Fx, Rear & Fy, Rear (Magnitude)	2403.36	N

3. LINKAGES FORCE

The linkages in a corner of a FSAE race car consists of two wish-bones having four links, a tie/toe rod and a push/pull rod. The tire forces are transferred to the links through upright. Since there are no bending forces, axial forces are developed in all linkages.

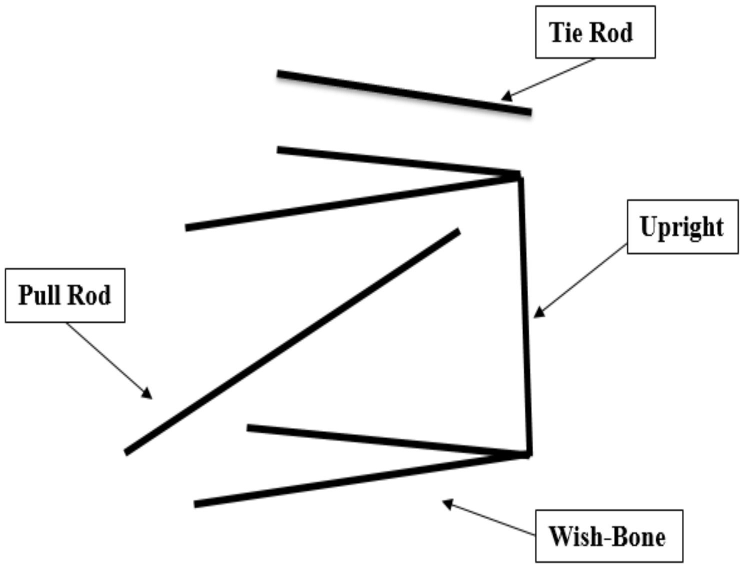

Figure 2 Linkages in a corner of car

The unit vectors are calculated for every link and the tire forces are distributed in the Cartesian plane and directional vector ratios. For calculating the unit vectors of the link, the data have been taken from suspension geometry of a FSAE race car.

Table 6 Data from suspension geometry

x	y	z			Length
-103.80	220.00	123.00	1.00	lower a arm	358.33
0.00	560.00	168.00	2.00		
122.58	220.00	123.00	3.00		364.21
-100.00	282.00	301.00	4.00	upper arm	269.52
8.50	524.00	349.00	5.00		
120.00	282.00	301.00	6.00		270.74
-58.00	597.00	200.00	7.00	tie rod	389.62
-68.00	210.00	156.00	8.00		
19.00	494.00	319.00	9.00	pull rod	286.29
19.00	311.30	98.59	10.00		
0.00	625.00	0.00		Tire contact Patch	

Table 7 Unit vectors

Force			
Unit vectors	X	Y	Z
2 to 1	0.29	0.95	0.13
2 to 3	-0.34	0.93	0.12
5 to 4	0.40	0.90	0.18
5 to 6	-0.41	0.89	0.18
7 to 8	0.03	0.99	0.11
10 to 9	0.00	0.64	0.77

The number of links is equal to the number of column in the matrix. The solution of the matrix results in the axial force in the linkages.

$$
\begin{array}{c}
\begin{array}{cccccc} \mathbf{1} & \mathbf{2} & \mathbf{3} & \mathbf{4} & \mathbf{5} & \mathbf{6} \end{array} \\
\begin{bmatrix} \mathbf{Fx\ 1} & & & & & \\ \mathbf{Fy\ 1} & & & & & \\ \mathbf{Fz\ 1} & & & & & \\ \mathbf{Mx1} & & & & & \\ \mathbf{My1} & & & & & \\ \mathbf{Mz1} & & & & & \end{bmatrix}
\end{array}
\begin{bmatrix} \mathbf{T1} \\ \mathbf{T2} \\ \mathbf{T3} \\ \mathbf{T4} \\ \mathbf{T5} \\ \mathbf{T6} \end{bmatrix}
=
\begin{bmatrix} \mathbf{Fx} \\ \mathbf{Fy} \\ \mathbf{Fz} \\ \mathbf{Mx} \\ \mathbf{My} \\ \mathbf{Mz} \end{bmatrix}
$$

Figure 3 Matrix array

Putting the value of the tire forces obtained from the force calculations in this matrix on RHS, we will get the axial forces in the linkages. The calculation were done in MS Excel taking the data from suspension geometry for both front and rear linkages. The corresponding Force-Moment Matrix is formulated as follows in all the six links:

Table 8 Force-Moment Matrix

0.29	-0.34	0.40	-0.41	0.03	0.00	Fx
0.95	0.93	0.90	0.89	0.99	0.64	Fy
0.13	0.12	0.18	0.18	0.11	0.77	Fz
-0.09	-0.09	-0.22	-0.22	-0.13	0.18	Mx
0.05	-0.06	0.14	-0.14	0.00	0.01	My
-0.16	0.19	-0.22	0.21	0.04	-0.01	Mz

The material used for the link have the following dimensions and material properties:

Table 9 Data of tube used

Outer Diameter	15.88	mm
Inner Diameter	13.80	mm
Young's Modulus	200.00	Gpa
Yield Strength 1018	375.00	MPa
C/S area of tube	48.51	mm^2
Moment of intertia	1341.83	mm^4

Axial stress in all the linkages were calculated and then factor of safety considering the buckling load.

Table 10 Linkages Force (Front)

Force on each arm		Axial Stress	Change in length	Buckling Load	Permissible Stress in axial	FOS
5088.52	Lower front arm	104.91	0.19	1849405.72	224.75	2.14
-8662.35	Lower rear arm	-178.58	-0.33	2001480.75	224.78	1.26
-2813.65	Upper front arm	-58.01	-0.08	2704691.79	224.85	3.88
2635.22	Upper rear arm	54.33	0.07	2692487.65	224.85	4.14
-562.37	Tie rod	-11.59	-0.02	1870955.29	224.76	19.39
2470.08	Pull Rod	50.92	0.07	2546278.18	224.84	4.42

Table 11 Linkages Force (Rear)

			(mm)		Permissible	
Force on each arm		Axial Stress	Change in length	Buckling Load	Stress in axial	FOS
3003.63	Lower rear arm	61.92	0.16	1298526.29	224.59	3.63
-5391.23	Lower front arm	-111.15	-0.20	2076038.85	224.79	2.02
-978.41	Upper rear arm	-20.17	-0.04	1699010.52	224.72	11.14
1663.90	Upper front arm	34.30	0.04	3396231.94	224.89	6.56
255.17	Toe rod	5.26	0.01	3443995.64	224.90	42.75
1441.52	Pull Rod	29.72	0.04	2914910.86	224.87	7.57

4. CONCLUSIONS

A mathematical model has been generated in MS Excel which can be used for any suspension geometry just by changing the coordinates of the suspension point and other vehicle parameters such as mass of the car, center of gravity distance, weight bias etc. Matrix method has been developed to calculate the forces on all the links. Factor of safety for each link has been calculated by using the forces obtained by unit matrix method. These forces are essential to design the upright and other suspension components such as upright clamps, bell cranks, hubs for any car using double wish-bone suspension.

ACKNOWLEDGMENTS

Acknowledge with thanks to Ashwa Racing, a project of Department of Mechanical Engineering, RV College of Engineering, Bengaluru for guiding in the progress of our project.

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