Design and Optimization of the Steering System of a Formula SAE Car Using Solidworks and Lotus Shark

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Abstract— **The main aim of this paper is to design the steering system for a formula SAE vehicle. The main focus is to design a steering system such as to counter bump and roll steer and ensure proper response to high speed and low speed turns. The design process consists of first determining the steering parameters and geometry and then analyzing it in lotus shark suspension analyzer. After analysis and optimization of the geometry the entire system is designed in Solidworks.**

Index Terms— **Steering, FSAE, Ackermann, LOTUS Shark, SOLIDWORKS**

I. Introduction

THE steering system of a Formula SAE car is of the utmost importance as it has to have a good reaction to all turns and corners at the event. The steering system is also one of the most key designs for overall handling and stability of the car.

The steering system should be such that the driver can actually sense what is happening at the front tires. The entire system must be designed in such a way that the components must be able to take all the load. The steering system should be responsive enough to high speed as well as low speed turns and also possess some self-returning action.

The steering parameters like castor angle, kingpin angle, scrub radius, mechanical trail etc. have to be kept in mind while designing and the best compromise for these values has to be found.

II. DESIGN

While designing, the major factor is the type of geometry to be used for the steering system. The three possible geometries that can be used are Ackermann, anti-Ackermann and parallel steer geometry.

As the Formula SAE event consists of more low speed corners it was decided to use Ackermann steering geometry as in this geometry the inner tire turns more as compared to the outer tire thus giving an added advantage for tracks with low speed turns.

Now since the geometry has been decided the percent Ackermann has to be decided. 100% Ackermann was considered to be the best solution for low speed maneuvers but due to compliance effects an Ackermann percent of around 60 to 80 percent was considered to be the best solution. The exact percent would be later decided on keeping in mind packaging constraints and tie rod length.

III. STEERING ABILITY REQUIRED

To calculate the rack, travel the steer angle required and steering ratio need to be calculated.

A simple model is used to determine approximate steering angle required considering maximum radius of turn in FSAE events. The wheelbase of the car is 1550 mm and tire radius of turn to be used is 4.5m.

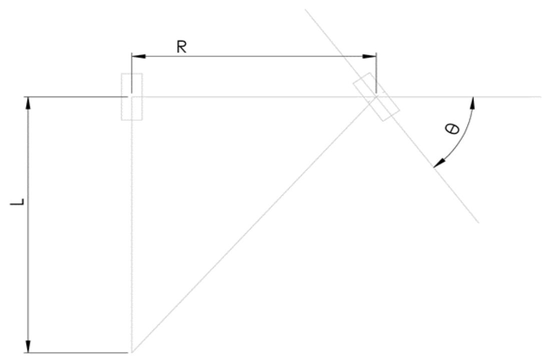

Figure 1. Steer angle for a simple model

A. *Final Stage*

The approximate steer angle is θ=R/l

Where θ= steer angle
R=wheelbase
L=radius of turn

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θ=1.55/4.5
=0.344 rad
=19.71 degrees

Now considering both the tires the steering angle has now to be calculated taking into account that both tires turn by a different amount.

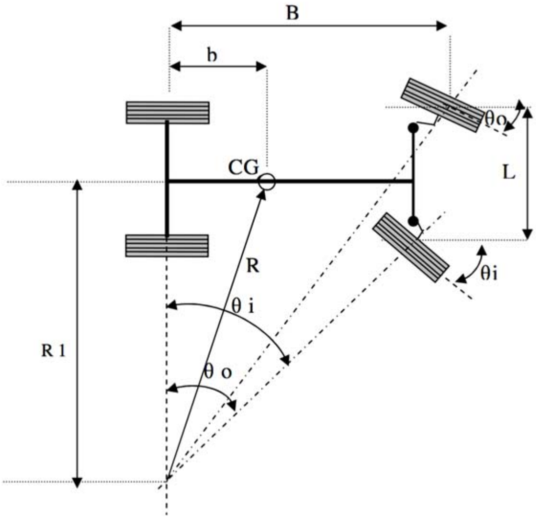

Figure 2. Steer angle for Ackerman principle

Where:
θo = turn angle of the wheel on the outside of the turn
θi = turn angle of the wheel on the inside of the turn
B= track width
L = wheel base
b = distance from rear axle to center of mass

$R=\sqrt{(R_1^2+B^2)}$
$R_1^2=R^2+B^2$
$R_1=\sqrt{(R^2+B^2)}$
R=4.5
B=1.55
R_1=4.43m
$R_1=B/\tan\theta_i+L/2$
$R_1=1.55/\tan\theta_i+1.195/2$
$\Theta_1 = 22.02$

Through the calculations we can find out that for a turn of maximum radius 4.5 m the steer angle for the inner tire is 22.02 degrees and the outer tire is 17.13 degrees.

IV. STEERING RATIO

The steering ratio is the ratio of how much the steering wheel turns in degrees to how much the wheel turns in degrees.

Approximating maximum turn to be of 25 degrees and steering wheel movement to be 180 degrees the steering ratio can be calculated as

S.R =180/25
=7.2

V. RACK TRAVEL

Once the steering ratio has been calculated the rack travel needs to be decided.

The steering wheel decided is AIM Formula steering wheel 2 which has a radius of 130 mm.

The steering wheel travel for one complete rotation
=2π x r
=0.816m

Considering maximum steer angle and max rack travel is reached at complete rotation of the steering wheel

The steering ratio can be equated to steering wheel travel/rack travel
7.2=0.816/Rack travel
Rack travel=113.33 mm

Therefore, required rack travel is around 114 mm.

VI. RACK POSITION

The rack can have two positions. It can either be in front of the front wheel center line or behind it. If the rack is placed forward of the front axle line it can be mounted easily on the frame giving wide range for choice of heights. However, this arrangement makes it difficult to have the steering rack, track rods and steering arms in a straight line which is required if Ackermann geometry is a goal for steering design. Fixing the rack behind the axle line is better from both a geometrical and packaging viewpoint. Hence it is decided to have the rack positioned behind the front axle line i.e. a rear steer is chosen.

VII. ACKERMAN PERCENT

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The exact Ackermann percent can be calculated according to the position of the steering arm or knuckles.

The percent can be calculated but based on the fact that parallel steer is 0%, and 100% is when the steering arms can be projected back to the rear axle at the vehicle centerline, then the range from 0-100% is between this geometry.

Current distance (where the lines projected meet) = 961.19 mm

Distance for 100% Ackerman= 1496.38mm

Ackermann percent=current distance/distance for 100 percent x 100 percent
=961.19/1496.38 x 100
=64.23 %

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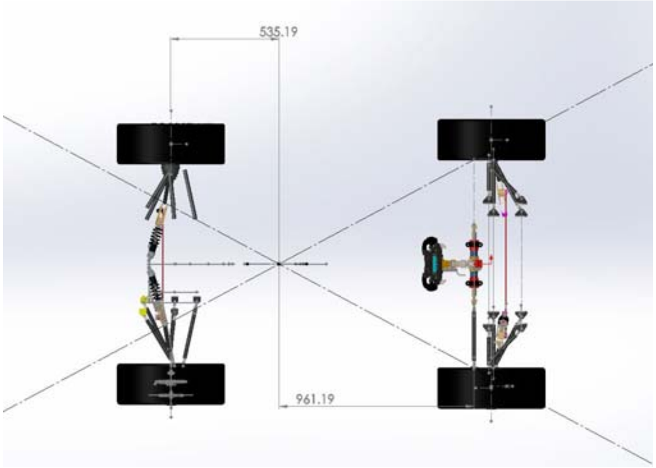

Figure 3. Calculation of Ackerman %

A. *Analysis in Lotus Shark*

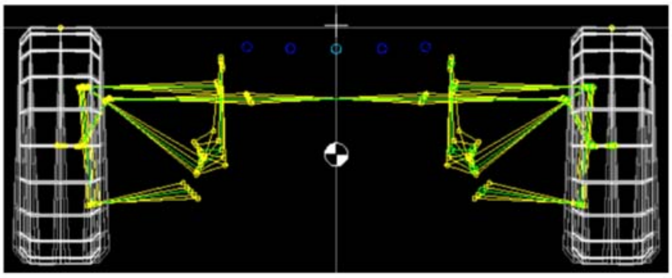

Figure 4. Front view geometry in LOTUS Shark

Sr. no.	Points	X	Y	Z
1	Lower Wishbone Front Pivot	-11.3	260	138.794
2	Lower Wishbone Rear Pivot	149.02	260	138.794
3	Lower Wishbone Outer Ball Joint	-11.33	565.58	137.23
4	Upper Wishbone Front Pivot	11.33	322.85	366.12
5	Upper Wishbone Rear Pivot	149.02	322.85	366.12
6	Upper Wishbone Outer Ball Joint	11.33	538.36	396.17
7	Pushrod Wishbone End	20.8	555	137.5
8	Pushrod Rocker End	33.76	304.13	317.79
9	Outer Track Rod Ball Joint	95	508	165
10	Inner Track Rod Ball Joint	95	193.3575	162
11	Damper to Body Point	40.14	253.78	84.67
12	Damper to Rocker Point	40.14	253.78	275.17
13	Wheel Spindle Point	0	565.58	266.7
14	Wheel Centre Point	0	610	266.7
15	Rocker Axis 1st Point	33.86	303.37	274.29
16	Rocker Axis 2nd Point	40.14	303.37	274.29
17	Centre of Gravity	780	0	280

Figure 5. Suspension and steering geometry coordinates in LOTUS Shark

The steering geometry had to be analyzed using a particular software to determine the steering parameters for best values of bump and roll steer. The software chosen was LOTUS Shark suspension analyzer due to its ease of use and accurate results. The process used was to determine the 2D suspension points in Solidworks and then input them into LOTUS Shark analyzer. After the first set of points were entered into the software, a number of iterations were carried out to determine the best possible values for the steering geometry.

Roll Angle	Camber Angle	Toe Angle	Castor Angle	Kingpin Angle
-3	1.8632	0.1129	4.9848	4.1279
-2.5	1.5746	0.077	4.987	4.4196
-2	1.277	0.0481	4.9894	4.7197
-1.5	0.9705	0.0262	4.9921	5.0281
-1	0.6554	0.011	4.9949	5.3446
-0.5	0.3319	0.0023	4.9979	5.6689
0	0	0	5.0012	6.0009
0.5	-0.34	0.0039	5.0048	6.3406
1	-0.6881	0.0138	5.0086	6.6878
1.5	-1.044	0.0296	5.0127	7.0424
2	-1.4078	0.0511	5.0171	7.4043
2.5	-1.7793	0.0783	5.0219	7.7734
3	-2.1585	0.111	5.0269	8.1497

Figure 6. Suspention and steering parameter values in LOTUS Shark during roll

Bump Angle	Camber Angle	Toe Angle	Castor Angle	Kingpin Angle	Damper Ratio
-25	0.6838	0.0535	4.9953	5.3125	1.593
-24	0.6597	0.0489	4.9955	5.337	1.58
-23	0.6354	0.0445	4.9957	5.3617	1.556
-22	0.6107	0.0403	4.9959	5.3867	1.544
-21	0.5858	0.0363	4.9961	5.412	1.532
-20	0.5607	0.0325	4.9963	5.4375	1.52
-19	0.5352	0.0289	4.9966	5.4632	1.509
-18	0.5095	0.0255	4.9968	5.4892	1.497
-17	0.4835	0.0233	4.997	5.5155	1.486
-16	0.4573	0.0193	4.9972	5.542	1.475
-15	0.4308	0.0166	4.9974	5.5687	1.464
-14	0.404	0.014	4.9979	5.623	1.453
-13	0.3769	0.0116	4.9981	5.6505	1.442
-12	0.3496	0.0095	4.9984	5.6783	1.431
-11	0.322	0.00075	4.9946	5.7063	1.421
-10	0.2941	0.0058	4.9984	5.7346	1.41
-9	0.2659	0.0042	4.9986	5.7632	1.4
-8	0.2375	0.0029	4.9989	5.792	1.389
-7	0.2088	0.0018	4.91	5.8211	1.379
-6	0.1798	0.0009	4.94	5.8504	1.369
-5	0.1505	0.0002	4.96	5.88	1.359
-4	0.121	-0.0003	4.99	5.9098	1.349
-3	0.0912	-0.0005	5.002	5.9399	1.339
-2	0.0611	-0.0006	5.004	5.9703	1.33
-1	0.0307	-0.0004	5.007	6.0009	1.32
0	0	0	5.001	6.0319	1.31
1	-0.031	6	5.0012	6.063	1.301
2	-0.0622	14	5.0015	6.0945	1.291
3	-0.0937	25	5.0018	6.1262	1.282
4	-0.1225	38	5.0021	6.1581	1.273
5	-0.1576	52	5.0024	6.1904	1.264
6	-0.19	70	5.0027	6.2229	1.255
7	-0.2227	89	5.003	6.2557	1.246

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8	-0.2557	111	5.0033	6.2887	1.237
9	-0.289	135	5.0036	6.3221	1.228
10	-0.3225	161	5.0039	6.3557	1.219
11	-0.35645	189	5.0043	6.3895	1.21
12	-0.3905	220	5.0046	6.4327	1.201
13	-0.425	253	5.0049	6.48581	1.193
14	-0.4597	289	5.0052	6.4929	1.184
15	-0.4948	327	5.0056	6.5279	1.176
16	-0.5301	367	5.007	6.5632	1.156
17	-0.5658	409	5.0073	6.5987	1.137
18	-0.6017	454	5.0077	6.6246	1.159
19	-0.638	501	5.0081	6.6707	1.15
20	-0.6746	551	5.0085	6.7071	1.142
21	-0.7115	603	5.0088	6.7439	1.133
22	-0.7487	657	5.0092	6.7809	1.125
23	-0.7862	714	5.0096	6.8182	1.116
24	-0.824	774	5.0099	6.8558	1.108
25	-0.8261	836	5.0102	6.8932	1.1

Figure 7. Suspention and steering parameter values in LOTUS Shark during bump

After analysis in LOTUS Shark suspension analyzer the steering parameters were finalized. The values were Kingpin Angle=8 degrees, Caster Angle=1.41 degrees, Mechanical trail=5.25mm.

The trail gets the wheel to follow the steering axis and gives it the self-straightening properties. However too much can make the steering heavy so a trail of 5.25 mm was finalized.

The kingpin inclination should be as close as possible to vertical to avoid unfavorable wheel camber changes when wheel is being steered. The ideal situation is not possible due to attain due to packaging requirements so a kingpin angle of 8 degrees has been chosen.

A positive caster angle of 1.41 degrees was finalized for good steer camber characteristics.

B. *Final Design*

Once the analysis in LOTUS Shark analyzer is completed and the geometry points have been finalized, the final 3-D design of the entire steering system is completed in Solidworks. The final design consists of the Steering wheel, steering column, universal joints, rack and pinion, track rod and steering arm.

Figure 8. Final Sterring system design in solidworks

VIII. STEERING PARAMETERS

Some important parameters have to be considered in designing the steering geometry. The steering geometry should be responsive enough both in bumps and roll and should also possess some self-returning capability. The steering force required should also be appropriate.

The option of having a kingpin angle of 0 degrees was not possible as the resultant scrub radius was too high. To keep the scrub radius to a minimum some amount of kingpin has to be added.

Positive Castor angle was added into the system because it has a good impact on steer camber characteristics. Some mechanical trail should be there to help steer return characteristics but too much mechanical trail can wipe out the effects of pneumatic trail. Pneumatic trail is important for the driver to sense tire wear characteristics. Therefore, a proper value of mechanical trail must be chosen.

To avoid unfavorable bump steer characteristics, the tie rod should point at the front view instantaneous center.

IX. BUMP STEER

Bump steer is an undesirable characteristic resulting from the radial paths described by the upper and lower steering axis bearings based upon a different center than that of the outer end of the track rod during suspension movements. This effect can be reduced by arranging the upper and lower steering axis bearings and keeping the inner track rod bearing to lie on the same line when viewed from the front of the car.

X. CONCLUSION

After all the calculations were completed and analysis in LOTUS Shark suspension analyzer was conducted the final steering assembly was designed in Solidworks. The above picture shows the final design incorporated into the chassis of the FSAE car.

This steering system designed for the turns generally encountered in the FSAE events was optimal to counter negative impacts of bump and roll steer and also possessed self-returning capability. Universal joints have been added in the steering column to line it up nicely with the pinion shaft. It also provides for the columns to fold up in the event of a hard frontal collision preventing it from being forced into the cockpit and injuring the driver.

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REFERENCES

[1] William F. Milliken and Douglas L. Milliken, 1995, “Race Car Vehicle Dynamics”, Society of Automotive Engineers, Inc.

[2] Gaffney, Edmund F., and Anthony R. Salinas. Introduction to Formula SAE® Suspension and Frame Design. No. 971584. SAE Technical Paper, 1997.

[3] Robertson, Dennis, and George J. Delagrammatikas. The suspension system of the 2009 cooper union FSAE vehicle: A comprehensive design review. No. 2010-01-0311. SAE Technical Paper, 2010.
[4] Gillespie, Thomas D., Fundamentals of Vehicle Dynamics. Society of Automotive Engineers, Inc., Pennsylvania, 1992.
[5] Smith, Carroll, Engineer to Win. MBI Publishing Company, Minnesota, 1984.
[6] Norton, R. Machine design. Pearson Prentice Hall, Upper Saddle River, NJ, 2006.
[7] Heisler, H. (1989), Advanced Vehicle Technology, Arnold, 338 Euston Road, London NW1 3BH
[8] Fenton, J. (1980), Vehicle Body Layout and Analysis, Mechanical Engineering Publications Ltd. London
[9] Smith, C. (1978), Tune to Win: The Art and Science of Race Car Development and Tuning, Aero Publishers, Inc. 329 West Aviation Road, Fallbrook, CA 29028
[10] Vehicle Dynamics Terminology- SAEJ670, Revise 07-1976.
[11] Pashley, T. (2008). Hoe to Build Motorcycle-engined racing cars, UK, Veloce Publishing

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