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Analysis of Steering System in FSAE Vehicle

Darshan Oza[1], Vinayak Hanmantasa Khatawate[1*], Glenn Mark Lewis[1], Ganesh Deepak Pandit[1], Krish Patel[1], Advait Rajendra Phadke[1] and Veeresh Balikai[2]

[1]Department of Mechanical Engineering, Dwarkadas J. Sanghvi College of Engineering, Mumbai, Maharashtra, India
[2]KLE Technological University, Hubballi, Karnataka, India

***Correspondence to:**
Vinayak Hanmantasa Khatawate
Department of Mechanical Engineering, Dwarkadas J. Sanghvi College of Engineering, Mumbai, Maharashtra, India.
E-mail: vinayak.khatawate@djsce.ac.in; vinayakhk@gmail.com

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Abstract

FSAE (Formula Society of Automotive Engineers) is a series of international competitions in which university teams compete to design and manufacture the best performing race cars. FSAE vehicles are designed and fabricated under a certain set of rules mentioned in the FSAE rulebook. Apart from other important components of the vehicle, it also focuses on the parameters required to design the steering system. It also provides information about the types of steering geometries as per the different handling setups of the vehicle. The solid model of the steering system was prepared by using SOLIDWORKS and the Finite Element Analysis (FEA) of the steering components was performed to check the structural integrity of the system. The iterations of the steering geometry were performed in LOTUS with the help of the steering parameters obtained. In this paper, the design and analysis of FSAE steering system is performed.

Keywords

Formula Society of Automotive Engineers, Finite Element Analysis, Steering, Lotus, Ackerman

Introduction

The steering system is the directional control system in automobiles that allows the driver to control the vehicle while driving [1]. Alongside, it is capable to provide feedback regarding the road conditions which helps driver handling the car. The motive behind this paper is to understand the design process of the steering system in the formula student vehicle along with its procedure to finalize it. Its main purpose is to give good steering response to the driver. The sideways motion of the wheels is the result of transmission of the translatory force by the tie-rod whose movement is determined by the angular rotation of the steering wheel and adopted steering mechanism in the vehicle [2]. The steering geometry parameters are responsible for stability and handling characteristics of the vehicle during pitch, roll and yaw. To determine the amount of corner entry understeer and corner exit oversteer in the vehicle, first step is to build an adequate steering geometry of the vehicle [3]. The steering geometry determines the amount of steering input obtained during the testing of geometry in Lotus. The objective of this paper is to show the step-by-step process of designing and FEA of the rack and pinion of steering system in a FSAE vehicle.

Experimentation

Steering system selection

The directional stability of steering system makes the vehicle return to its straight face position even after a slow speed corner. The steering assembly should be light in weight and should have minimum steering effort. Considering these points and following the rules of the FSAE rule book for the adequate steering mechanism, the paper is focused on rack and pinion steering mechanism (Figure 1) and has selected the parameters required for its calculations [4].

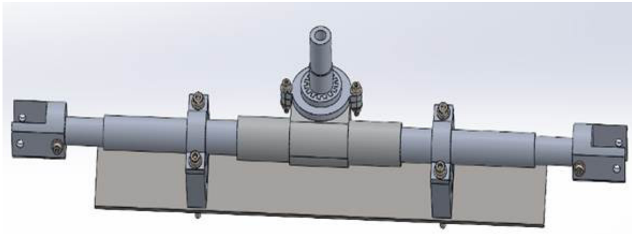

Figure 1: Rack and pinion steering.

Steering geometry selection

Steering geometry mainly depends on the track layout, magnitudes of speed that vehicle generates, rack position, displacement and tire data. According to which, it is systematic arrangements of links in order to provide better steering response and to avoid excessive skidding.

Figure 2 shows the anti-Ackerman geometry is efficient in high speeds. The lateral load transfer occurs from the inner wheel to the outer wheel. The steering angle of inner wheel is less than steering angle of the outer wheel, thus reduces the effect of the slip angle. The outer wheel despite having a high initial slip angle gains more angles during the turn and hence the traction increases significantly.

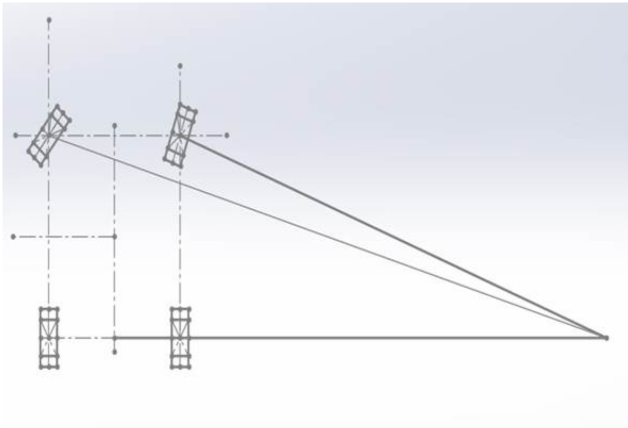

Figure 2: Anti-Ackermann steering.

Figure 3 illustrates parallel steering, where both the front wheels have same steering angle. It helps in taking tight hair pin turns, where the vehicle receives high structural g forces and high transfer of loads. Due to which, the outer wheel's slip angle and traction increases. Despite that, the traction of the inner wheel does not decrease significantly. Thus, it helps in keeping vehicle on the track, reducing understeer.

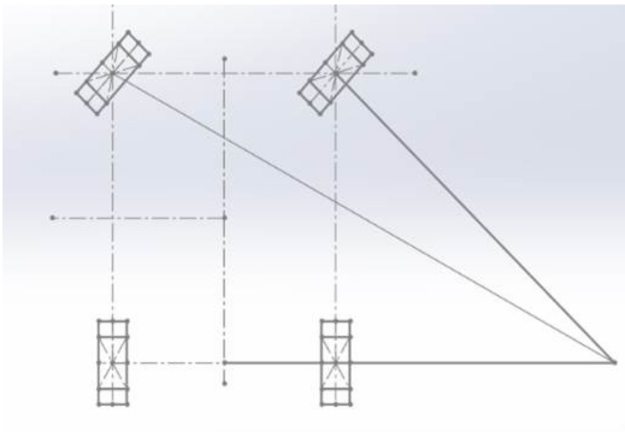

Figure 3: Parallel steering.

Figure 4 shows the Ackerman steering geometry, where the inner wheel angle is greater than the outer wheel angle and is efficient at low speeds. Depending on the tire data of the FSAE vehicle, if the required cornering force is achieved in smaller slip angle than the inner wheel angle, it induces drag [5]. However, the skidding reduces significantly and stability increases. Ackerman steering geometry is selected for this paper. Figure 5 shows the final line geometry of Ackerman steering for the FSAE vehicle.

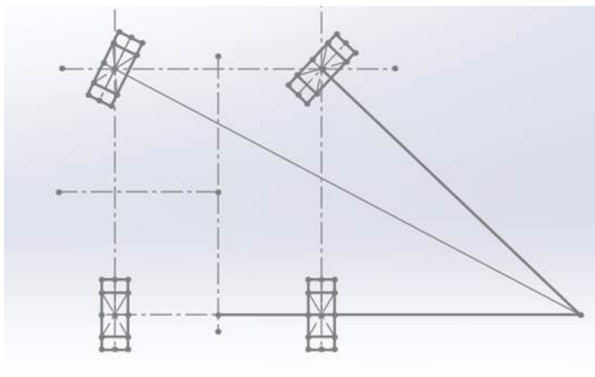

Figure 4: Ackermann steering geometry.

Steering geometry parameters and calculations

The specifications in table 1 were used to find the following parameters. The specifications are obtained from the geometry designed in SOLIDWORKS.

Steering angle

$$\text{Inner steering angle} = \frac{\text{Wheelbase}}{\text{Turning radius} - \left(\frac{\text{Track width}}{2}\right)} = \frac{1416}{2101.17 - \left(\frac{1200}{2}\right)} = 43.36°$$

$$\text{Outer steering angle} = \frac{\text{Wheelbase}}{\text{Turning radius} - \left(\frac{\text{Track width}}{2}\right)} = \frac{1416}{2101.17 - \left(\frac{1200}{2}\right)} = 24.42°$$

Tie rod forces

The tie rod is designed and analyzed with the help of SOLIDWORKS and ANSYS. After number of iterations, the safety factor of 2 is selected. The values given in table 2 were taken from final line geometry and forces are calculated from the vehicle parameter. The calculations for the total torque are as below:

Torque by mechanical trail: Lateral force x Caster trail; 233.05 x 17.63 = 4108.67 N-mm.

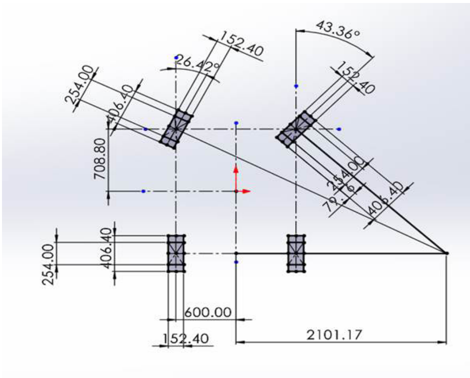

Figure 5: Ackerman steering parameters.

Table 1: Vehicle parameters.

Values	Parameters
Front track width	1200 mm
Rear track width	1200 mm
Wheel base	1416 mm
Tire radius	200 mm
Steering wheel radius	203.2 mm
Turning radius	2101.17 mm

Table 2: Torque calculation parameters.

Parameters	Values
Caster angle	3°
King pin inclination	10°
Caster trail	17.63 mm
Scrub radius	10.51 mm
Lateral force	233.05 N
Longitudinal force	3451.26 N

Torque by scrub radius: Longitudinal force x Scrub radius; 3451.05 x 10.51 = 36273.05 N-mm.

Total torque: Torque due to mechanical trail + Torque due to scrub radius; 4108.37 + 36273.05 = 40381.7279 N-mm.

Total tie rod force = 40381.7279 N-mm.

Steering effort calculations

The steering effort is the force required by the driver on the steering wheel in order to rotate the wheel in a desired angle [6]. The torque required to calculate the steering effort is: Torque = Force of tie rod from rack x Pinion radius. Considering the line geometry, since tie rod is 12° inclined, Torque = 568.75 x cos(12°) x 17.25 = 9.874 N-mm. Steering effort is given by [7].

$$\text{Steering effort} = \frac{\textit{Steering wheel torque}}{\textit{Steering wheel radius}} = \frac{9.87}{0.2} = 82.28\ \text{N}$$

Steering ratio: It is a ratio of the angle made by steering wheel to the angle turned by the wheels during the cornering.

$$\text{Steering ratio} = \frac{\textit{Degrees of steering wheel}}{\frac{\textit{Outer steering angle} + \textit{Inner steering angle}}{2}} = 10.62:1$$

C factor: Linear distance travelled by rack in single pinion rotation = π x Diameter of pinion = π x 35.5 = 111.47 mm.

Working principle

To avoid the self-steering of the wheels due to forces from the roads, the driver rotates the steering wheel in the direction in which he desires to go. Torque transmitted through the steering links converts into the translatory motion of the rack in the direction of the turn [8]. This force through the rack is then transmitted to tie rods attached at one end of the upright makes the wheels turn in the desired direction.

Design of rack and pinion

The material selected for the steering system is given in the tabulated form below (Table 3). Aluminum based composite was used for rack and pinion as it possess good strength-to-weight ratio which is the requisite for transportation industry [9, 10]. The material used for extrusion rod was nylon fiber as it has good mechanical properties for the intended automobile components [11]. Maximum rack travel is assumed at 180° rotation of the steering wheel.

$$\text{The total rack travel} = \frac{\pi * \textit{Pinion diameter}}{4} = \frac{\pi * 35.5}{4} = 27.89\ \text{mm}$$

Pitch diameter of pinion: S = r x θ; 27.86 = r x 180 x (π /180) = 8.87 ≈ 9 mm.

Pitch diameter = 2 x r = 2 x 8.87 = 17.74 mm ≈ 18 mm.

The module is given by, Pt = σb x D.P x b x y [12].

D.P = π x m; b = 8 m; Pt = 2000 N; y = 0.484 from Design Data Book.

Where, Pt is Transverse load; σb = Permissible stress; D.P = Diametral pitch; b = Face width; y = Lewis form factor; m = Gear module; 2000 = 150 x π x m x 8 x m x 0.484. Thus, m = 1.04 ≈ 1.25 mm.

$$\text{Number of teeth onpinion} = \frac{\textit{Pitch Diameter}}{m}\frac{18}{1.25} = 14.4 \approx 15$$

Table 3: Material selection and manufacturing.

Component	Material	Manufacturing process
Tie rod	AISI 1018	Turning, Tapping
Steering column	AISI 1018	Turning, Drilling
Rack	Aluminum 7075-T6	Milling, Gear hobbling, Drilling
Rack base	Aluminum 7075-T6	Milling, Drilling
Pinion	Aluminum 7075-T6	Facing, Turning, Gear facing, Turning, Gear hobbing, taped hole
Pinion insert	Aluminum 7075-T6	Lathe, Gear hobbing, Grub key hole
Extrusion rod	Nylon fiber	Turning, Drilling

Results and Discussion

FEA of rack and pinion

FEA of rack

When the design of the rack was completed (Figure 6), it was tested under various loads in ANSYS static structural. The material as mentioned in table 3 was aluminum 7075-T6. After meshing in coarse sizing and setting the resolution to 5, more refined mesh is obtained. The first iteration of the rack that was more in height withheld all the forces very easily, thus, it was decided to reduce the rack height in order to reduce the weight and try again to get an appropriate factor of safety (FOS). After the first iteration of the rack, it is subjected to force of 800 N in order to check its sustainability. For the analysis, ANSYS 18.2 was used and various forces on the rack were optimized. On the basis of total deformation, maximum principal stress and fatigue, the FOS was obtained. The figure 7 and figure 8 illustrates the equivalent elastic stress and FOS of rack teeth respectively. The minimum and maximum FOS obtained was 1.94 and 15, respectively. On the basis of force analysis, it was decided to keep minimum FOS between 1.5 and 2. The deformation of the rack was found to be 0.14mm. As rack is only subjected to fixed support at both ends and pinion force acting on the teeth of the rack. Hence, it was concluded that the design is safe.

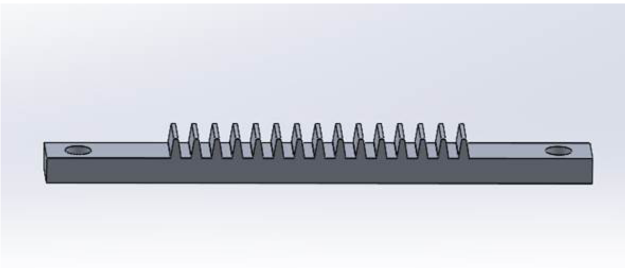

Figure 6: Design of rack.

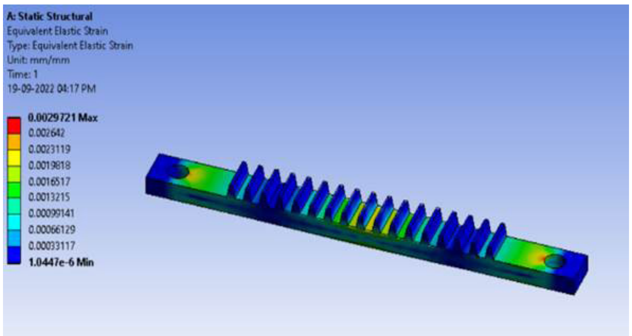

Figure 7: Equivalent elastic strain of rack.

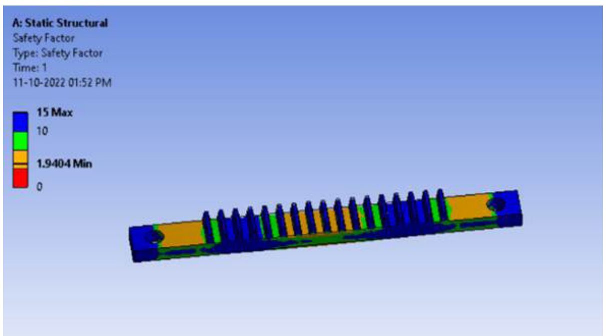

Figure 8: FOS of rack.

FEA of pinion

After finalizing the rack, the solid model of pinion was prepared in SOLIDWORKS and it was optimized with various forces acting on it with the help of ANSYS 18.2, in order to check the maximum deformation as shown in figure 9. Thus, the maximum deformation of the pinion obtained as per figure 9 is 0.076 mm. The FOS of the pinion is 1.47 as shown in figure 10.

Lotus analysis

Lotus Software was used to verify: Camber angle in bump; Camber angle in roll; Toe angel; Ackermann percentage; Rack travel at 2101.17 mm turning radius; and Roll center height at 20 mm bump.

Lotus analysis was performed on the line geometry considering all the coordinates obtained from SOLIDWORKS. Figure 11 shows the line geometry of the vehicle. The double wishbone suspension system was used here and it had a clear

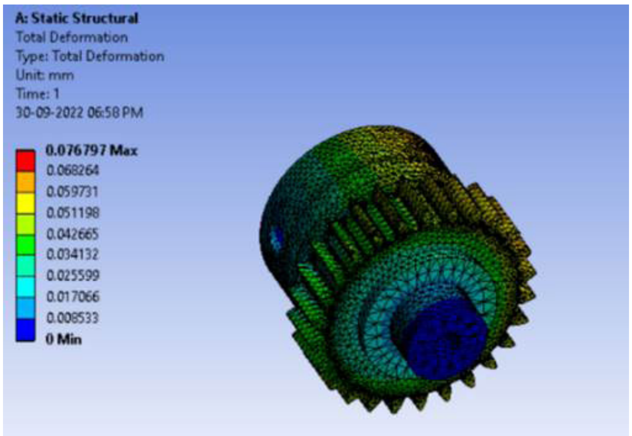

Figure 9: Solid model of pinion.

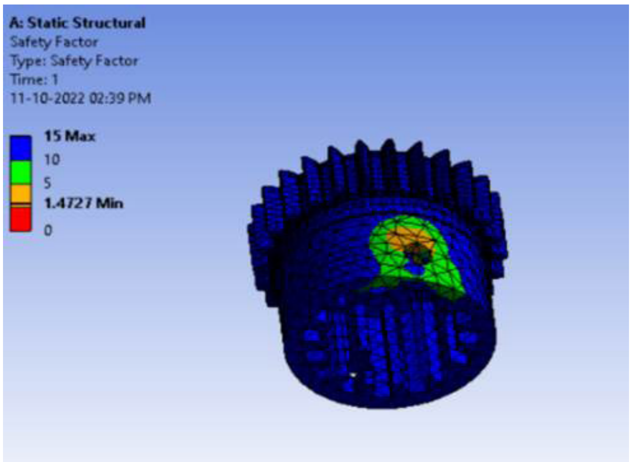

Figure 10: FOS of pinion.

intention to keep the roll center height above and close to the ground. The static camber and toe were set to zero. The wheel base, track width and wheel travel were set. As shown in figure 12, the structural data file gives the data of the wheel travel on the basis of camber change. It also shows the changes in the roll center height with change in king pin inclination and spring travel. The simulations on LOTUS helped to figure out the Ackermann percentage of the geometry after deciding the wheel travel and camber range [13].

Conclusion

In the present study the Ackerman steering geometry helped the suspension system to get its desired outputs in the effective speeds of the vehicle. Moreover, the reduced steering effort and high steering angle provided directional stability and improved response. As shown in the figure 11 and figure 12, the suspension geometry went through the simulations in the LOTUS engineering software. In the end, the part model was finalized which weigh 20.83 g. The required FOS was obtained in ANSYS after few iterations. LOTUS software helped in achieving the nominal Ackermann percentage of 55% under the wheel travel range from -20 mm to +20 mm. Thus, the steering system and line geometry were finalized. The components of the steering assembly were designed in SOLIDWORKS and FEA was used for obtaining the required FOS of each component. The steering geometry of an FSAE vehicle should satisfy the conditions of Ackermann percentage in the fixed camber range and mechanical trail. Moreover, more the Ackermann percentage, more the stability in the steering. The driver must have better handling experience especially during corners. Thus, this paper was an attempt to satisfy all the parameters to make an adequate steering system for an FSAE vehicle.

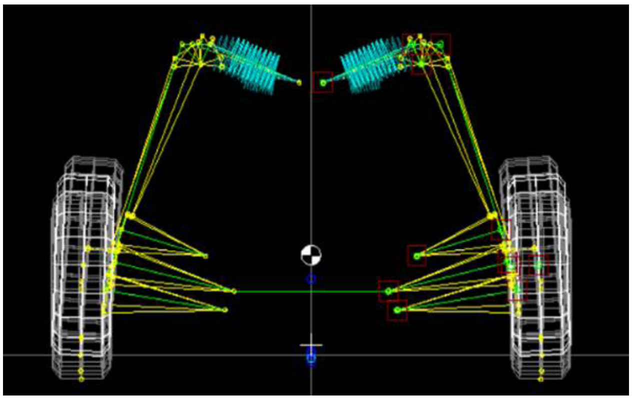

Figure 11: LOTUS simulations.

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)	DAMPER RATIO [-]	SPRING RATIO [-]
-60.00	0.2315	-1.5548	10.3499	13.8818	1.553	1.553
-40.00	-0.4338	-0.9943	10.3749	14.4416	1.432	1.432
-20.00	-1.2108	-0.6067	10.4080	15.1463	1.313	1.313
0.00	-2.1042	-0.3884	10.4503	15.9993	1.189	1.189
20.00	-3.1216	-0.3412	10.5037	17.0081	1.046	1.046
40.00	-4.2735	-0.4715	10.5708	18.1838	0.822	0.822
60.00	-6.9143	-0.2062	10.5953	19.1727	0.000	0.000

INCREMENTAL SUSPENSION PARAMETER VALUES

BUMP TRAVEL (mm)	ANTI DIVE (%)	ANTI SQUAT (%)	ROLL CENTRE HEIGHT TO BODY (mm)	ROLL CENTRE HEIGHT TO GRND (mm)	HALF TRACK CHANGE (mm)	WHEELBASE CHANGE (mm)	DAMPER TRAVEL (mm)	SPRING TRAVEL (mm)
-60.00	4.23	0.00	10.55	70.55	-3.06	-1.42	43.99	43.99
-40.00	3.38	0.00	3.73	43.73	-1.16	-0.75	30.57	30.57
-20.00	2.37	0.00	-2.51	17.49	-0.15	-0.28	15.99	15.99
0.00	1.14	0.00	-8.11	-8.11	0.00	0.00	0.00	0.00
20.00	-0.37	0.00	-13.03	-33.03	-0.68	0.04	-17.89	-17.89
40.00	-2.23	0.00	-17.19	-57.19	-2.17	-0.19	-39.03	-39.03
60.00	-733.48	0.00	796.55	736.55	2.76	-0.33	-47.81	-47.81

Figure 12: Structural data file.

Acknowledgements

None.

Conflict of Interest

None.

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