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Design of suspension system for formula student race car

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Abstract

The design of a suspension system has a significant role in vehicle dynamics. There is a different purpose of a vehicle which has different types of suspension systems. In the case of a race car, there are several factors that have to be considered. In this paper, a suspension system for a student formula race car is designed. The load calculation of a formula student race car is presented. Before load calculation, suspension geometry and upright or knuckle geometry are determined. Then the design of the rocker arm is determined from the value of the motion ratio. The spring is selected from the calculation of spring stiffness. The load during acceleration, cornering and braking are calculated. From that calculation, the maximum loads are accounted for by selecting the wishbone and pushrod pipe.

Keywords: Suspension, suspension design, formula student, race car suspension, suspension load calculation.

1. Introduction

A suspension system is one of the most important systems of an automobile that deals with the dynamics of the vehicle. It is the intermediate flexible system that connects the wheels with the mainframe of the vehicle, the suspension system is a combination of various components like a knuckle or upright, Arms or linkages and shock absorber that comes together and enables the relative motion between the tire and the mainframe. Formula Student (or Formula SAE (F-SAE)) is a worldwide university competition, organized by the Society of Automotive Engineers (SAE), which encourages university teams to design, build, and compete with a Formula-student race car. For a formula student car, the suspension system plays a very important role in case of acceleration, cornering, and braking. In 2004 A. Theander et al. [1] researched vehicle dynamics. The aim of that thesis work was to design the suspension and steering geometry for the race car being built. In 2015 Y.S. Saurabh et al. [2] researched the design procedure of the front double A-arm pushrod suspension system for a formula student race car. This work emphasizes the method for designing and analyzing the suspension system for a race car in various aspects. In this research, we explained the design process and details calculation of suspension designing.
The main objectives of this project are:

i. To design a suspension system for a formula student car.
ii. To keep all four wheels in contact with the ground at the correct angles in order to exploit the maximum tractive force of the tires.
iii. To analysis the load of the components of the suspension system.
iv. To compliance with F-SAE rules.

2. SAE rules for Suspension System

For designing a suspension system firstly the suspension rules of SAE for student formula competition should be checked properly. The rules are the followings:
According to the rules of FSAE rule book 2017-2018, Rule T6.1.1: The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do not represent a serious attempt at an operational suspension system or which demonstrate handling inappropriate for an autocross circuit. Rule T6.1.2: All suspension mounting points must be visible at Technical Inspection, either by direct view or by removing any covers.

3. Methodology

Suspension geometry for a formula student car is designed in such a way that results in a minimum change of toe angle during bump so that bump steer becomes minimum. Also, the roll center is kept such that the camber can compensate with roll angle and the tire can get maximum traction during cornering. There are some parameters that are called packaging parameters and they should be determined for designing the suspension geometry. The packaging parameters are tire size, rim diameter and width, wheel offset, kingpin inclination, mechanical trail, scrub radius, spindle length, the caster, the camber, tie rod position, rack location, track width, the upper and lower ball joint positions and tie rod outer position [Fig. 3.1]. Upright or knuckle geometry can be determined from these packaging parameters. In automotive suspension, a steering knuckle is that part which contains the wheel hub or spindle and attaches to the suspension and steering components. It is variously called a steering knuckle, spindle, upright or hub as well. For designing front view geometry; roll center height, instant center position and fvsa length (Fig. 2) should be calculated. The lower wishbone length is kept larger than the upper wishbone to keep the camber rate low. A low camber rate ensures handling and stability. The roll center of a vehicle is the notional point at which the cornering forces in the suspension are reacted to the vehicle body.

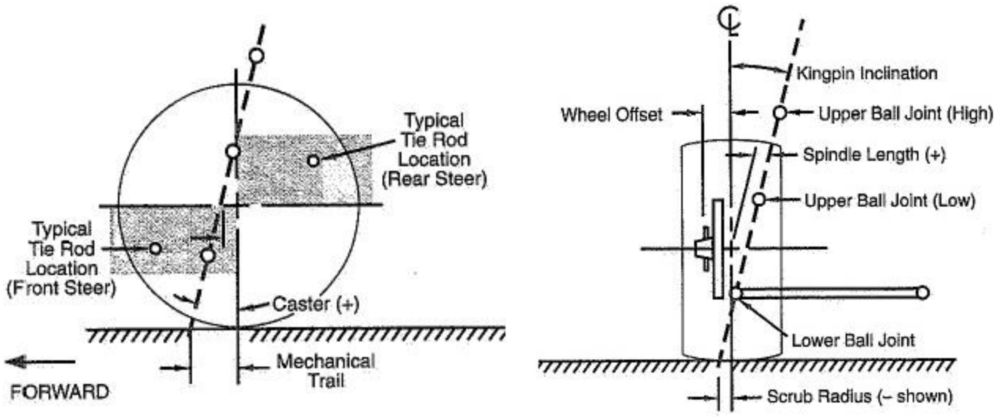

Fig. 3.1: Packaging parameters [3]

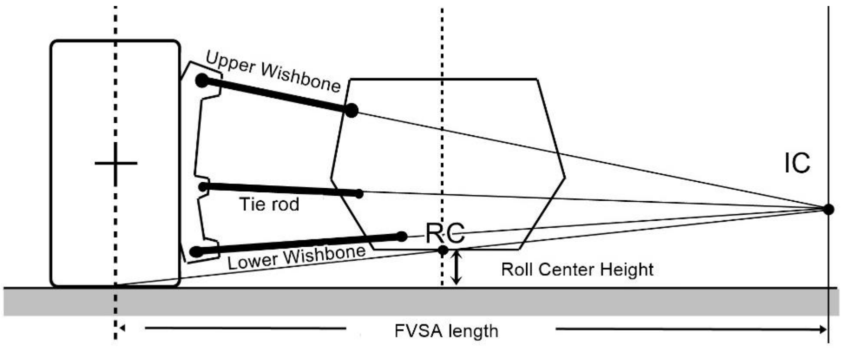

Fig. 3.2: Front view suspension geometry

Based on the literature survey, knowledge about the cars from the Formula Student 2003 is at KTH an overview of static set up is given below.[1]

- Kingpin inclination angle between 0° and 8°
- Scrub radius between 0mm and 10mm
- Caster angle between 3° and 7°
- Static camber is adjustable from 0° to -4°
- Camber gain 0.2-0.3 degrees/roll angle at the front axle
- Camber gain 0.5-0.8 degrees/roll angle at the rear axle
- Roll center height between 0mm and 50mm in front and slightly higher at rear
- Well-controlled and predictable movement of the roll axle
- Minimize bump steer
- 50% - 65% of the roll stiffness on the rear axle.

• Maximum roll angle about 2°

From the above survey, the suspension geometry is designed according to the process described in the book 'Race Car Vehicle Dynamics'[3]. The value of different parameters is given below.

Table 3.1

Parameters	Values	Parameters	Values
Wheelbase	1570 mm	Caster angle	2 degree
Wheel track	1200 mm	Roll center height	25.4 mm
Rim diameter	10 inch	Static Camber	2 degree
Rolling radius	177.8 mm	Roll angle	5 degree
Kingpin inclination	7 degree	Bump	40 mm
Scrab radius	9.5 mm	Rebound/ Jounce	40 mm

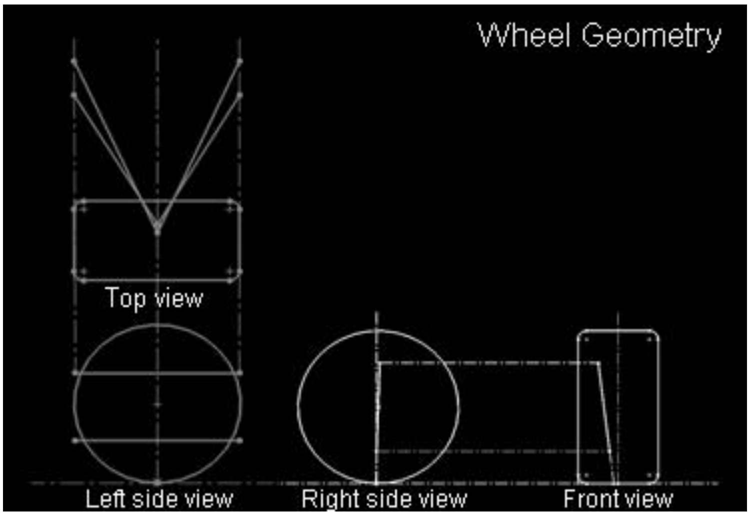

Fig. 3.3: Wheel geometry drawing

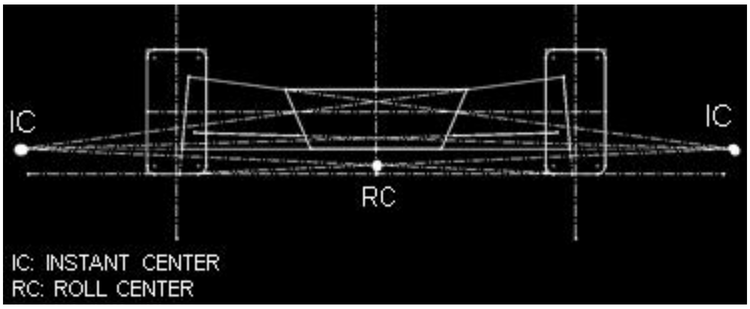

Fig. 3.4: Suspension geometry drawing

In figure 3.3, the drawings of the wheel geometry are shown and figure 3.4 shows the instant centers and roll center of the suspension geometry.

For a racing car, comfort is not important. An average damping ratio of 0.65-0.7 has been found to work well for a racing car. [4] For calculating damping co-efficient the natural frequency should be known. For getting the value of the natural frequency value of the wheel center rate, tire stiffness and ride rate should be known. The ride rate can be calculated from the following equation.

$$\frac{1}{K_R} = \frac{1}{K_W} + \frac{1}{K_T} \tag{1}$$

Here K_R is ride rate, K_w is wheel center rate and K_T is tire stiffness. The sprung natural frequency f_s can be determined from the value of ride rate K_R and sprung mass m_s. [5]

$$f_s = \frac{1}{2\pi}\sqrt{\frac{K_R}{m_s}} \tag{2}$$

The unsprung natural frequency f_u can be calculated from the value of K_W, K_T and the unsprung mass m_u. [5]

$$f_u = \frac{1}{2\pi}\sqrt{\frac{K_W + K_T}{m_u}} \tag{3}$$

The motion ratio is the ratio of wheel movement to the damper movement. The motion ratio depends on the design of the bell crank design. To keep the desired motion ratio, the bell crank should be designed properly. The ratio of L_1 to L_2 should be equal to the motion ratio. The motion ratio should be near 1. Here we kept the motion ratio 1.3 where L_1 is 65 mm and L_2 is 50 mm.

$$R_m = \frac{L_1}{L_2} \tag{4}$$

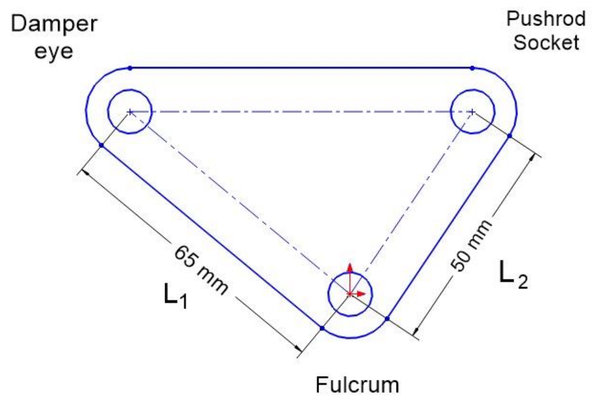

Fig. 3.5: Bellcrank geometry

Roll rate can be determined from the track width T and the ride rate K_R.

$$Roll\ rate = \frac{T^2 K_R}{114.6} \tag{5}$$

Critical damping can be calculated from sprung mass m_s, sprung natural frequency f_s and motion ratio R_m. [4]

$$C_{crit} = 4\pi m_s f_s R_m^2 \tag{6}$$

For ordinary car damping ratio of about 0.25 gives the best compromise between comfort and performance. For racing cars where comfort is not important, an average damping ratio of about 0.65-0.7 has been found to work well.

$$Damping\ ratio\ \zeta = \frac{C}{C_{crit}} \tag{7}$$

From equation 28 we can find the value of damping co-efficient C. The value of spring rate or spring stiffness is determined from the following equation. [4]

$$K_S = R_m^2 K_W \tag{8}$$

The minimum length of spring can be calculated by the following process. [4]

$$Initial\ compression = \frac{Sprung\ corner\ weight}{K_W} \tag{9}$$

$$Total\ wheel\ movement\ =\ Bump\ +\ Initial\ Compression \tag{10}$$

$$Total\ spring\ movement\ =\ \frac{Wheel\ movement}{R_m} \tag{11}$$

Table 3.2: Calculation of spring and damper

Variable	Value	Variable	Value
Wheel center rate K_w	22 N/mm [4]	Wheel track T	1200 mm
Tire stiffness K_T	139.5 N/mm [6]	Roll rate	238.74 Nm/deg
Ride rate K_R	19 N/mm	Critical damping C_{crit}	4024.4 N/m/s
Sprung mass per wheel m_s	74.9 kg	Damping co-efficient C	0.7
Unsprung mass m	4 kg	$Damping\ ratio\ \zeta$	2817 N/m/s
Sprung natural frequency f_s	2.53 Hz	Spring rate K_W	37.18 N/m
Unsprung natural frequency f_u	312.68 Hz	Sprung corner weight	78.9 kg
L_1	65	Initial compression	35.2 mm
L_2	50	Bump	40 mm
Motion ratio R_m	1.3	Total wheel movement	75.2 mm
		Total spring movement	57.84 mm

4. Load calculation

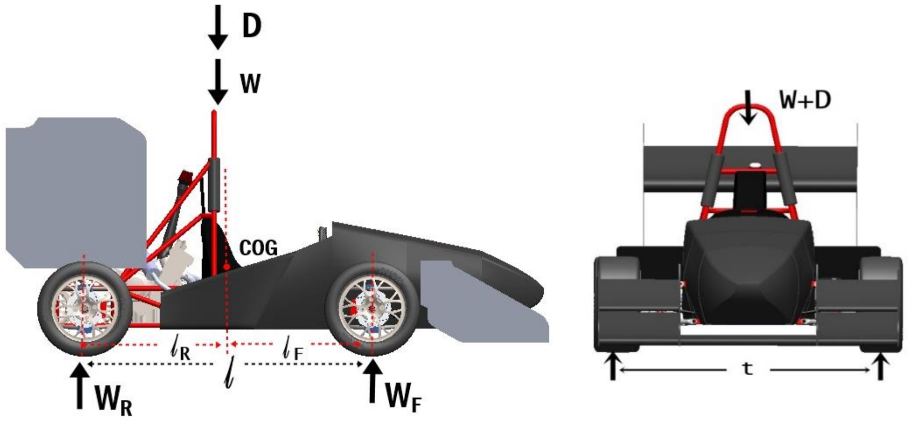

Fig. 4.1: Loads on the car

Car total design effective vertical load

$$W_{vert} = W \times d_f + \mathrm{D} \times a_f \tag{12}$$

$$\frac{Design\ vertical\ load}{side} = \frac{\mathrm{W} \times d_f + \mathrm{D} \times a_f}{2} \tag{13}$$

Where W_{vert} is car effective weight, W is car weight D is downforce, d_f is the dynamic multiplication factor for the portion of vertical load derived from mass and the factor for that derived from aerodynamic downforce.
Front and rear wheel design vertical load per side,

$$W_F = \frac{Design\ vertical\ load/side \times (l - l_F)}{l} \tag{14}$$

$$W_R = \frac{Design\ vertical\ load/side \times (l - l_R)}{l} \tag{15}$$

Where, l is wheelbase and l_F is the distance from the front wheel center to the center of gravity.[4]

Design load on front and rear axels,

$$W_{Faxel} = \frac{W_{vert} \times (l - l_F)}{l} \tag{16}$$

$$W_{Raxel} = \frac{W_{vert} \times (l - l_R)}{l} \tag{17}$$

Design braking force for each wheel on front axel (longitudinal)

$$F_B = W_{Faxel} \times \mu \times d_f \times \frac{1}{2} \tag{18}$$

Where W is car weight and μ is co-efficient of friction and d_f is dynamic multiplication factor. [4]

Maximum cornering force

$$F_C = W_{vert} \times \mu \tag{19}$$

Total lateral weight transfer

$$\Delta W_y = \pm \frac{F_C h_m}{t} \tag{20}$$

Where, h_m is the height of the center of gravity and t is wheel track. ΔW_y is the total lateral load transfer of the car. [5]

Now, design lateral load on the front outer wheel is

$$W_{FO} = \frac{W_{Faxel} + \Delta W_y \frac{W_{Faxel}}{W_{vert}}}{2} \tag{21}$$

Design lateral load on the front inner wheel is

$$W_{FI} = \frac{W_{Raxel} - \Delta W_y \frac{W_{Raxel}}{W_{vert}}}{2} \tag{22}$$

Similarly, we can calculate the load for rear wheels.

Maximum design acceleration load

$$F_a = W_R \times \mu \tag{23}$$

Where W_R is rear-wheel design load and μ is co-efficient of friction. [4]

From the above equations, the calculation values are given below:

Table 4.1

Variables	**Values**	**Variables**	**Values**
Car weight W (with driver)	2940 N	Fr_{axel}	10238.5 N
Downforce D	W× 3.3 = 9702N	$_{axel}$	11194.1 N
d_f	3	μ	1.2
a_f	1.3	F_B	7986.03 N
Wheelbase l	1570 mm	F_C	25719.12 N
l_F	820mm	h_m	400 mm
l_R	750mm	Wheel track t	1200 mm
W_{vert}	21432.6 N	ΔW_y	±8573.04 N
Design vertical load per side	10716.3 N	W_{FO}	7166.95 N
W_F	5119.25	W_{FI}	3071.55 N
W_R	5597.05 N	F_a	6716.46 N

Table 4.2 Maximum loads

Maximum vertical load (F_{vert})	5597.05 N (W_R)
Maximum longitudinal load (F_{long})	7986.03 N (F_B)
Maximum lateral load (F_{lat})	7166.95 N (W_{FO})

All the vertical load is carried by the pushrod and the bottom wishbone. If the pushrod remain inclined at angle θ to the ground plane, the force on pushrod will be,

$$F_{pushrod} = \frac{F_{vert}}{sin\theta} \tag{24}$$

The horizontal component of the pushrod is,

$$H_{pushrod} = F_{pushrod}\, cos\theta \tag{25}$$

During cornering, the load on the upper and lower wishbone, (fig 6.4)

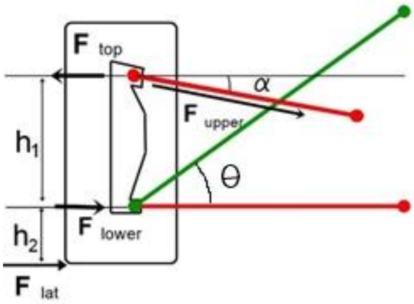

Fig. 4.2

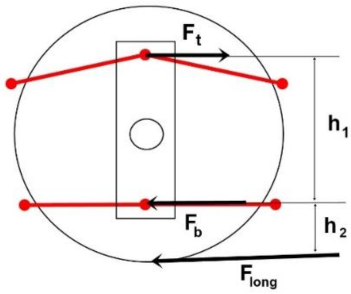

Fig. 4.3

$$F_{top} = \frac{h_1}{h_2} F_{lat} \tag{26}$$

$$F_{upper} = \frac{F_{top}}{cos\alpha} \tag{27}$$

$$F_{lower} = \frac{h_1 + h_2}{h_2} F_{lat} \tag{28}$$

Similarly, we can calculate the force created by longitudinal forces.

$$F_t = \frac{h_1}{h_2} F_{long} \tag{29}$$

$$F_b = \frac{h_1 + h_2}{h_2} F_{long} \tag{30}$$

By applying the triangular method, the load on each member of the wishbone can be calculated.

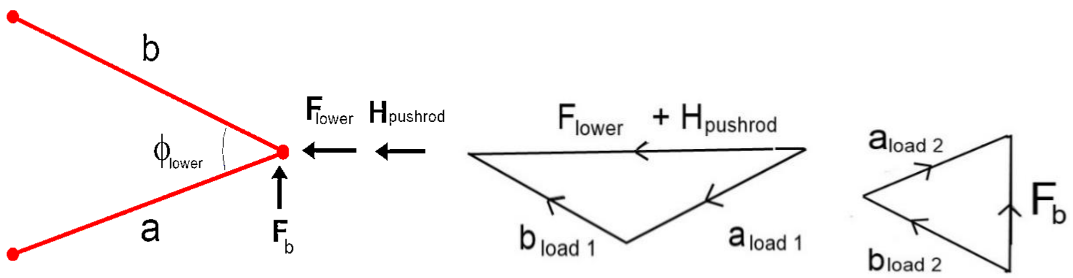

Fig. 4.5. Lower wishbone

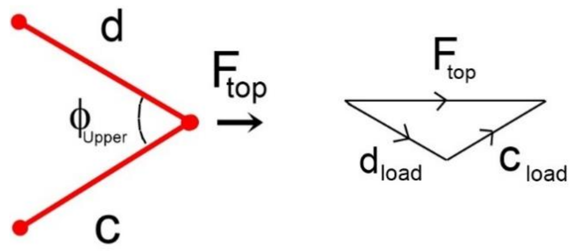

Fig. 4.6. Upper wishbone

So, the loads on wishbone pushrod members will be,

$$Load\ on\ member\ a = |a_{load\ 1} - a_{load\ 2}|$$
$$Load\ on\ member\ b = |b_{load\ 1} + b_{load\ 2}|$$

$$Load\ on\ member\ c = c_{load}$$
$$Load\ on\ member\ d = d_{load}$$
$$Load\ on\ pushrod = F_{pushrod}$$

Table 4.3

Variables	Values	Variables	Values
θ	60 degree	φ_{lower}	51.27 degree
α	7.6 degree	φ_{upper}	65.54 degree
h_1	203 mm	$a_{load\ 1}$	5764.61 N
h_2	68 mm	$b_{load\ 1}$	5764.61 N
$F_{Pushrod}$	6462.91 N (compression)	$a_{load\ 2}$	9229.49 N
$H_{pushrod}$	3231.45 N	$b_{load\ 2}$	9229.49 N
F_{top}	21395.45 N	c_{load}	11865.68 N
F_{upper}	21585.07 N	d_{load}	11865.68 N
F_{lower}	28562.4 N	a	3464.88N (compression)
F_t	23840.64 N	b	14994.1 N (tension)
F_b	31826.67 N		

5. Pipe selection for wishbone

The maximum load on the wishbone rod is 14994.1 N for tension and 3464.21 N for compression which are applied due to braking.
For tension force the cross-sectional area of a round tube for wishbone can be determined by the following equation:

$$A = \frac{1.5\ \times F_t}{\sigma_y} \tag{31}$$

Here, A is the minimum cross-section area of round tube, F_t is the tension force and σ_y is the yield stress of the tube material. For steel alloy 1020 cold drawn tube σ_y is 350 MPa and for 14994.1 N tensile loads, the minimum tube area is 64.26 mm^2.

From appendix of the book 'Race Car Design' [4],
The selected round pipe is 12.7×10.67×2.03 (Outer dia × Inner dia × thickness)
For 3464.21 N compression force, we can select the round tube by the following equation,

$$P_E = \frac{\pi^2 EI}{1.5\ L^2} \tag{32}$$

Here, P_E is buckling load, E is the modulus of elasticity, I is 2nd-moment area and L is the length of that member.
For P_E of 3464.21, E for cold drawn steel is 200000 N/mm^2 and L is 404.5 mm.
So, calculated I is 430.7 mm^4.
The selected round pipe is 12.7×11.48×1.22
So we can see that the tube for tension force is much stronger. So the ultimate wishbone pipe will be 12.7×10.67×2.03 cold drawn round steel tube.

6. Result and Discussion

Satisfying all FSAE rules the suspension geometry is designed for a formula student race car. The suspension geometry is designed keeping the ability to give maximum tractive force on the tire during cornering. From the calculation, the wishbone and pushrod are selected and that is 12.7×11.48×1.22 round pipe. The bell crank is designed keeping motion ratio 1.3 where L_1 is 65 mm and L_2 is 50 mm. The damper is selected with damping co-efficient 0.7 with a damping ratio of 2817 N/m/s. The spring is selected which has 37.18 N/m of spring rate and 57.84 mm of spring movement.

Fig. 8.1: The suspension system according to the design

7. Conclusion

The target of this research is successfully achieved. Maintaining the FSAE rules, the suspension system is designed with proper load analysis. The suspension system is built in real and it performs well. From this research, we are able to build a better suspension system for our car, which participated in Formula Student Japan 2019. Also, this research will help future team members to improve the car for the following projects. For limitations of facilities, we couldn't able to analyze the dynamic behavior of the suspension system. In the future, we will try to design the suspension system with an antiroll system using simulation software and compare the dynamic behavior with the simulation result.

8. References

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