Presently there are many computational modules available, designing of the suspension system remains very difficult task. When considering a road car suspension system there are many parameters to be discussed. But in case of race car suspension system some parameters can be neglected. The challenging role is to increase the road holding capability. This paper covers the procedure for calculating the suspension parameters and includes a computational example which means spring, damper and antiroll bar specification. Along with the dynamic equations it gives the procedure, how to line up the design evolution. The worked out example specifically deals the numerical design of an open wheel race car suspension system which is being particularly designed and manufactured by the students. These cars may have a weight of 250kgs-350kgs. The covered computations are calculated with respect to the race car vehicle dynamics.

**Key words:** Motion ratio; Ride rate; Roll axis; Sprung mass; Unsprung mass; Vehicle mass; Wheel displacement.

**INTRODUCTION**

This paper is inspired by the Formula student vehicle which participated in the technical event Formula student Spain. The suspension system of the vehicle was designed and fabricated by our team. The system was designed with reference to the Race car vehicle dynamic book. In order to find some key parameters such as ride rate, vehicle weight we have assumed the values. As a result of it the spring rates found were gone wrong. This greatly affected the performance of the vehicle. This paper covers a systematic approach for designing a suspension system for a student level formula car.

**OVERVIEW**

Ride rates and roll rates are the very important vehicle dynamic characteristics which largely influence the suspension behavior. These rates defines the characteristics pitch, roll, heave and warp which needs to be considered when designing a race car. Pitch is the vehicle rotation about the y-axis and Roll is about x-axis. Warp is non-uniform linear motion along the z-axis and heave is the uniform linear motion. Ride motion of the sprung mass relative to the road surface should be zero to achieve the ideal road-holding capability. This can be achieved only when the tire follows the contour of the road exactly. If it cannot be achieved the relative ride motion should be minimized. To attain this springs with lowest possible spring rates can be selected. But to control the relative ride motion of the sprung mass, it needs heavier spring and damper rates. So the challenging task is to find the optimum value. Body bounce frequency consistent for varying masses, plays a vital role in this job.

Family sedans and sports cars

- 0.8 to 1.5 Hz

Non-aero formula cars

- 2 Hz

Aero cars

- 5 to 7 Hz

The pitch tendency of the car can be reduced by keeping the rear axle frequency slightly higher than the front. With these frequencies the ride rates can be easily found when the sprung masses are determined. Ride rate is the vertical force per unit vertical displacement of the tire ground contact reference point relative to the chassis with respect to the race car vehicle dynamics. As said earlier, ride rate affects the pitch, roll, heave and warp. The road holding capacity can be increased by maintaining the wheel loads constant and proportional to the tire’s adhesion capacity. The primary consideration should be made on the vehicle mass, acceleration, C.G height, track width and wheel base since it influences the weight transfer. Roll axis is the line which connects the front and rear roll center heights. If this line passes through the CG, the body roll can be eliminated but it results in instantaneous weight transfer. Comparing the body roll and weight transfer, weight transfer should be minimized since the body roll can be compensated with other parameters such as heavier spring rates or antiroll bars. Springs absorbs the shock and transfer it to the chassis. There is an option for using flexible chassis but it is like an un-damped spring. Along with the spring, dampers are used to absorb the energy released by the springs. They also control bump energy and provide roll damping. If the dampers are not used the energy released by the springs are uncontrolled and the ride motion of the sprung mass and un-sprung mass is majorly affected.
So there comes stability and control issues. The ability of suspension to travel with the road surface irregularities is called warp. If the vehicle is not manufactured with the suspension warp cannot be attained. But normally racers prefer to reduce the warp by stiffening the springs and dampers. This reduces the pitch, roll as well as heave. If adequate amount of warp is not provided, the dynamic load variations will increase and cornering performance is reduced. Mostly double wishbone suspension type is preferred for race cars because they use inboard suspension to reduce the aero-dynamical obstacles. This type also helps in keeping all the members including spring, damper and wheels in perfect position. The kinematic goals of this paper is to reduce the effects due to bump, roll and steering with respect to the ideal wheel position. The installation ratio is a geometric concept that relates the change in length of a force producing device (eg: spring, damper or anti-roll bar) to a change in vertical wheel center movement. For the calculation, motion ratio can be assumed as 1:1 since the spring/damper’s package size and rates are not yet decided.

This means the damper stroke will be equal to the vertical wheel displacement. Damper is the main suspension component. As a part of the calculations, damper rates are to be specified. But there is no text book formulas to find the damper specifications. Choosing of dampers are really a hypothetical one but with the desired ride rates and roll rates, the damping characteristics can be determined easily. The critical damping of body shows the suspended spring will return backs to original position without any other disturbances (i.e.) the displacement of wheel will be equal to displacement of spring. In this paper the wheel travel is controlled by compression damping and body travel is controlled by rebound damping. Rebound/compression ratios for street cars and race cars vary from 1.5:1 to 4:1. To calculate the Lateral Load Transfer Distribution (LLTD), roll stiffness, roll flexibility, front and rear roll center heights, weight distribution, an estimation of total vehicle weight with driver. Suspension geometry predicts the roll center heights. The sprung mass rotation is about the roll axis. Roll axis is the line joining front and rear roll center heights. For the formula car the front roll center height is very near to the ground and the rear is slightly higher than it. The front roll center is merely zero, it may be positive or negative, and the rear roll center height will be slightly more positive.

**DESIGN PROCEDURE**

**Specification of tires, wheels and Vehicle parameters:** In order to design the open wheeled racing car suspension system, the first and foremost thing is to know the tire data according to manufacturer’s specification along with the vehicle parameters respectively.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Front</th>
<th>Rear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tire Size</td>
<td>20*7.0-13.0in</td>
<td>20*8.0-13in</td>
</tr>
<tr>
<td>Tire Mass</td>
<td>3.99 kg</td>
<td>4.08 kg</td>
</tr>
<tr>
<td>Tire Camber</td>
<td>-3.0 deg</td>
<td>-2.0 deg</td>
</tr>
<tr>
<td>Tire Pressure</td>
<td>90 kpa</td>
<td>90 kpa</td>
</tr>
<tr>
<td>Tire Stiffness</td>
<td>100 N/mm</td>
<td>125 N/mm</td>
</tr>
<tr>
<td>Wheel Size</td>
<td>13*7.0in</td>
<td>13*8.0in</td>
</tr>
<tr>
<td>Wheel Mass</td>
<td>3.35 kg</td>
<td>3.60 kg</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wheel Base</td>
<td>1590 mm</td>
</tr>
<tr>
<td>Track Width Front</td>
<td>1272 mm</td>
</tr>
<tr>
<td>Track Width Rear</td>
<td>1192 mm</td>
</tr>
<tr>
<td>Vehicle Mass With Driver</td>
<td>350 kg</td>
</tr>
<tr>
<td>Suspension Travel Front</td>
<td>31.75 mm / 31.75 mm</td>
</tr>
<tr>
<td>Suspension Travel Rear</td>
<td>31.75 mm / 31.75 mm</td>
</tr>
</tbody>
</table>

**Suspension type:**
- **FRONT:** Double Wishbone, Pushrod Inboard Suspension, Outboard Disc Brakes.
- **REAR:** Double Wishbone, Pushrod Inboard Suspension, Outboard Disc Brakes.

**Ride Frequency and Ride Frequency Ratio:** The ride frequency is a major parameter for determining the ride quality of the vehicle. The optimum range for non-aero formula cars is 2 Hz. From this, the ride frequency ratio can be fixed.
Sprung and Unsprung Corner Weights: The load that acts above the spring is said to be sprung mass and the load that acts below the spring is said to be unsprung mass. The unsprung mass can be determined by below vehicle parameters that are shown. The sprung mass are determined initially for the calculation and then iterated for the best optimum values. The addition of both sprung and unsprung mass will give the corner mass (i.e.) the mass suspended per corner.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Front</th>
<th>Rear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tire</td>
<td>4.0 kg</td>
<td>4.1 kg</td>
</tr>
<tr>
<td>Wheel</td>
<td>3.4 kg</td>
<td>3.6 kg</td>
</tr>
<tr>
<td>Uprights &amp; Hubs</td>
<td>10 kg</td>
<td>12 kg</td>
</tr>
<tr>
<td>Rotors</td>
<td>1 kg</td>
<td>1 kg</td>
</tr>
<tr>
<td>Callipers</td>
<td>1 kg</td>
<td>1 kg</td>
</tr>
<tr>
<td>Suspension Links</td>
<td>2 kg</td>
<td>2 kg</td>
</tr>
</tbody>
</table>

Sprung Total Weight (m) = 21.4 kg
Unsprung Total Weight (m) = 28.7 kg
Sprung Weight (M) = 100 kg
Corner Mass = 121.4 kg
Total Weight of Vehicle = 164.5 kg

Derived Ride, Suspension and Spring Rates: From the above given information, the specified values of ride rates are known which helps us to know the suspension stiffness potentially.

\[ f_{nf} = \frac{1}{2\pi} \sqrt{\frac{K_{RF}}{M}} = 2.0 \text{ Hz} \]  
\[ f_{nr} = \frac{1}{2\pi} \sqrt{\frac{K_{RR}}{M}} = 2.0 \text{ Hz} \]  

Where:
\( f_n = \) Body bounce frequency ; \( K_R = \) Ride rate (N/mm) ; \( M = \) Sprung corner mass (Kg)

\[ K_{RF} = (2*2.0)^2*100 = 15791.36 \text{ N/m} \]  
\[ K_{RR} = (2*2.2)^2*120 = 22929.06 \text{ N/m} \]  

Suspension Stiffness:
\[ K_S = \frac{K_{SR}}{1 - K_{IR}^2} \]  
\[ K_{SR} = 15791.36 \text{ N/m} \]  
\[ K_{IR} = 0.05 \]  
Spring rates = \( K_{SPR} = \frac{1}{1 - K_{SR}^2} \)  
\[ K_{SPR} = 18761.72 \text{ N/m} \]  

Derive Roll Rates: In this roll rates are determined with the help of antiroll bar or without antiroll bar. The roll rates are said to be the body roll to wheel displacement and this wheel displacement is compared with spring or damper travel (jounce/rebound).

\[ R_f = \frac{W/g}{h_i} \left( \frac{K_{SR}}{K_{RF} + K_{IR} - Wh_i} \right) \]  

Where:
\( W = \) Total vehicle weight with driver; \( h_i = h - h_a = \) distance of roll axis from CG

Given:
\( H = \) CG height = 300 mm; \( h_f = \) front roll center height = 10 mm; \( h_r = \) rear roll center height = 20 mm; \( h_a = \) roll axis height at CG = 16 mm
\( h_i = h - h_a = 300-16 = 284 \text{ mm} \)
Lateral load transfer without anti-roll bars (LLTD): The LLTD is determined only by roll rates produced. The LLTD calculated must be less than one for neutral handling of the vehicle (i.e.) understeer and oversteer are not taken in account.

$$\text{LLTD} = \frac{DFzf}{DFzr} = \frac{[K\Phi_f^*R_\Phi + h_f^* (W_f/g)]}{[K\Phi_r^*R_\Phi + h_r^* (W_r/g)]}$$  \hspace{1cm} (10)

LLTD without anti-roll bars, LLTD = 15178.08 * 0.028 + (0.01 * 1613.79) / 19972.69 * 0.028 + (0.02 * 1819.75) = 0.74 (unit less)

Include Anti-roll bars to produce desired Roll Rates:

$$\text{LLTD (desired)} = 0.85$$

$$K\Phi_f^w/\text{ARB} = 18081.32$$ Nm/rad (11)

Recalculate Roll rates with Anti-roll bars:

$$R_f = (W/g)*h_i / (K\Phi_f + K\Phi_r - Whi)$$  \hspace{1cm} (13)

Where:

$$K\Phi_f^w/\text{ARB} = 18081.32$$ Nm/rad
$$K\Phi_r = 19972.69$$ Nm/rad

$$\Phi = \frac{3433.5*284}{(18081.32 + 11515.3433.5*0.284)} = 0.026 \text{ rad/g} = 1.50 \text{ deg/g}$$

Wheel displacement (Dzw) due to roll (f) is:

$$f = 1.4g * \Phi / \text{Whi} = 2.10 \text{ degrees}$$
$$\text{Dzw} = 0.5t* \Phi = 0.5 * 1272 * \sin 2.10 = 23.30 \text{ mm}$$

**CONCLUSION**

Numerical calculations were made on the suspension parameters, with respect to the race car vehicle dynamics. This provides the optimum values of the parameters, required for the suspension system of the formula cars. But testing of the package will always produce the best results. So it recommended to make the suspension geometries as adjustable links. The adjustable links will let us to perform the vehicle in different conditions.

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