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A NUMERICAL APPROACH TO SUSPENSION KINETICS ANALYSIS OF FSAE CAR

V.Sakthi Murugan, Rathan Kumar, Y.Suresh, M.Kumaragnanamithra, Raveen rohith and M.Viswanathan, Kaleem Sherif

Department of Automobile, Saveetha School of Engineering, Chennai, TamilNadu, India

ABSTRACT

This paper discusses the effect of suspension kinetics on handling and steering response of a FSAE(Formula Society of Automotive Engineers) car. Parameters like suspension frequency, roll stiffness, roll sensitivity, lateral load transfer distribution will be discussed in detail. The paper aims at finding an acceptable range of these values which leads to good vehicle handling and steering response. Also discuss the effect and finding the suitable LLTD(Lateral Load Transfer Distribution) and making the vehicle dynamics better for FSAE car

Keywords: Suspension kinetics, spring rate, suspension frequency, roll stiffness, roll sensitivity, lateral load transfer distribution, FSAE

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1. INTRODUCTION

The lateral accelerations at the rollover and static stability factor, but different suspension characteristics and loads are exposed to roll during handling manoeuvres. The results of simulation performed correlate well with the predictions based on the proposed model.[1].A time-domain analysis considering the vehicle behaviour in terms of its motion as time passes. In time stepping, the velocity of each component follows from position or displacement changes divided by the time step or from acceleration multiplied by the step time. In this time-stepping simulation, nonlinear springs and nonlinear damping were used to predict accurately the vehicle behavior [2].The mode superposition method is improved on the description of large horizontal displacements (in longitudinal and lateral directions), while the effects of dampers and other structural elements with non-linear characteristics are considered by the pseudo force method. Examples, relating to cornering and braking, are presented by the use of a bus finite element model containing 2294 degrees of freedom[3]. Vehicle dynamic analysis of 4wheel steering system is addressed to improvement of vehicle maneuverability and stability. All research here is based on production vehicle from manufactures. To study actual system response, a three dimensional, full vehicle model was related. In past research of this type, simple dimensional, bi-cycle vehicle models were used. The modeling and analysis for this model and subsequent 4WS vehicles were performed using ADAMS(Automated Dynamic Analysis of Mechanical System) software[4].A procedure for establishing suspension parameters, which includes computation and example that has damper, spring, and anti-roll bar[5].By creating a simulated model of an older car and correlating it to collected test data, as well as creating Track Master templates (software used to view DAQ data) for viewing the logical data manner[6]. Additional features provided by this project include a more organized method of testing, a system of data analysis, a test stand for organization of test equipment, and a database for vehicle dynamics to organize data of vehicle and it has analysis tools [6].

The simulations and to study the effects that uneven roads have on the systems. Furthermore, a 3D environment based on real life data is also useful in driving simulators, when for example, analysing driver behaviour, testing driver response, and training for various driving conditions[7].To measure and collect data, a car equipped with sensors and a computer. On top of the car, a SICK LMS200 2D lidar a ranging sensor mounted tilted updside down, facing the road in the car's front[8].To create the 3D environment, all the individual measurements were transformed to a global coordinate system using the pose (position and orientation) information from a high-class navigation system. The pose information made it possible to compensate for the vehicle motion during data collection.[9].The fundamental problem around chassis stiffness was studied and discussed why a chassis should be stiff, which increasing the stiffness of chassis makes the race engineer's ability to change handling balance of the car and how far chassis stiffness is vital. A dynamic behavior analysis of vehicle handling was performed in ADAMS Car and ADAMS Flex to authenticate the influence of chassis stiffness in race car's handling balance by simulation of steady state handling [10].

In this work suspension kinetics analysis predicts the effect of various factors like spring stiffness, motion ratio, suspension roll stiffness, vehicle roll rate, anti-roll bar stiffness etc on the handling and steering response of the vehicle. Calculations will be made to predict favourable kinetics parameters for the present vehicle.

2. THEORETICAL APPROACH

2.1. Suspension Frequency

A suspension or ride frequency is the undamped natural frequency of the body in ride. The higher the frequency, the stiffer the ride. So, this parameter can be viewed as normalized ride stiffness. Based on the application, there are ballpark numbers to consider.

2.2. Spring Rates

As the name suggests, spring rates provide a measure a measure of the softness or stiffness of the spring being used in a suspension system. As we will see later that for a given mass of a vehicle, suspension frequency depends on spring rates only.

2.3. Roll stiffness, roll sensitivity and lateral load transfer distribution

Similar to choosing ride frequencies for bump travel, a roll stiffness must be chosen next. The normalized roll stiffness number is the roll gradient, expressed in degrees of body roll per "g" of lateral acceleration. A lower roll gradient produces less body roll per degree of body roll, resulting in a stiffer vehicle in roll.

2.4 Analysis of ride frequencies in FSAE

Lower frequencies create a gentler suspension with more mechanical grasp, however the reaction will be slower in transient (what drivers report as "absence of help"). Higher frequencies make less suspension go for a given track, permitting lower ride statures, and thusly, bringing down the focal point of gravity.

Ride frequencies front are rear are generally not the same, there are several theories to provide a baseline. Two cases underneath demonstrate overstated plots of what occurs with unequal ride frequencies front and back as the car hits a bump. In Figure 1, we can see the undamped vertical movement of the skeleton with the front ride recurrence higher than the back. The first period is the most dominant on the car when looking at frequency phase, due to effects of damping to be explained later.

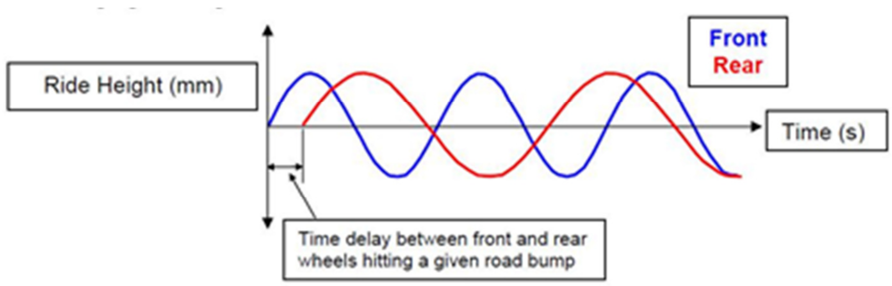

Figure 1 Front ride frequency higher than rear

The out of phase motion between front and rear vertical motion, caused by the time delay between when the front and rear wheel experiences the bump, is highlighted by the frequency difference. A consequence of the stage contrast is pitching of the body. To diminish the pitch originated by hitting a bump, the back needs a higher characteristic frequency to catch up with the front, as appeared in Figure 2.This notion is called producing a flat ride, meaning that the induced body pitch from road bumps is minimized.

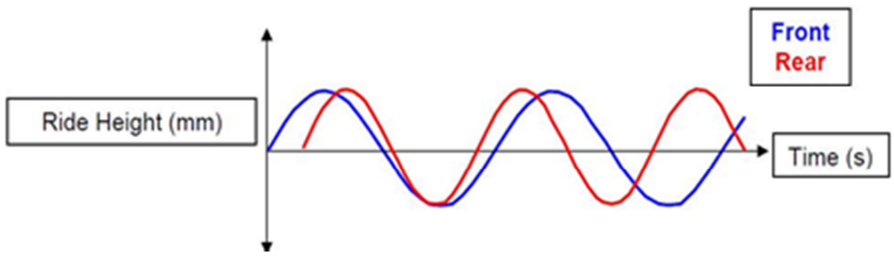

Figure 2 Rear ride frequency higher than front

For a given wheelbase and speed, a frequency split front to rear can be calculated to minimize pitching of the body due to road bumps. A common split is 10 – 20% front to rear. The above theory was originally developed for passenger cars, where comfort takes priority over performance, which leads to low damping ratios, and minimum pitching over bumps. Racecars as a rule run higher damping ratios, and have a substantially littler worry for comfort, prompting some racecars utilizing higher front ride frequencies. The higher damping ratios will decrease the measure of wavering resultant from street bumps, consequently diminishing the requirement for a level ride. Damping ratios will be clarified in the following tech tip in detail. A higher front ride frequency in a racecar permits speedier transient reaction at corner section, less ride stature minor departure from the front (the optimal design are typically more pitch delicate on the front of the car) and takes into consideration better back wheel footing (for raise wheel drive cars) on corner exit. The ride frequency split should be chosen based on which is more important on the car you are racing, the track surface, the speed, pitch sensitivity, etc.

As an example of ride frequency split front to rear, Figures 3 and 4 shows a simple example of a single degree of freedom vehicle model over an impulse disturbance. The ride frequency difference is 10 percent, 70% critical damping, 100 km/h speed, and 1.75 m wheelbase.

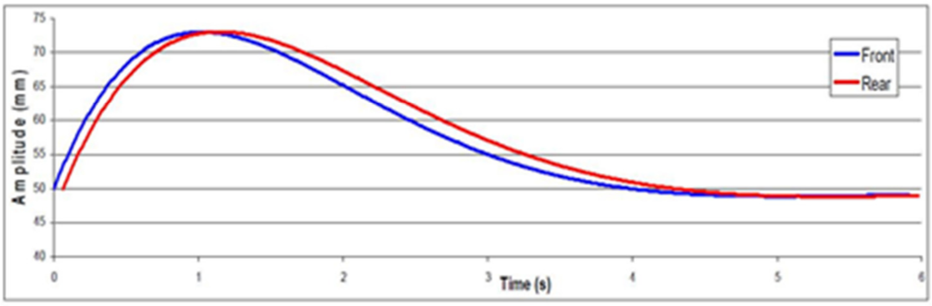

Figure 3 Front ride frequency 10% higher than rear

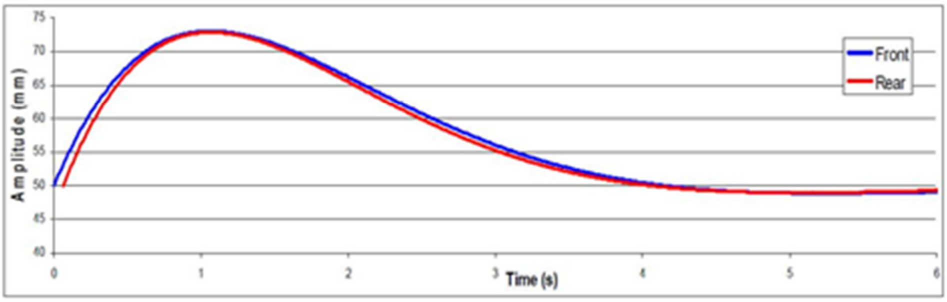

Figure 4 Rear ride frequency 10% higher than the front

Once the ride frequencies are chosen, the spring rate wanted can be resolute from motion ratio of the suspension, sprung mass sustained by each wheel, and the desired ride frequency.

2.5Analysis of spring rates in ride frequencies in FSAE

Starting with the basic equation from physics, relating natural frequency, spring rate, and mass (according to Fundamentals of Vehicle Dynamics by Thomas Gillespie):

$$f = 1/(2\pi)\sqrt{\frac{k}{m}}$$

f = Natural frequency (Hz)
K = Spring rate (N/m)
M = Mass (kg)

Solving for spring rate, and applying to a suspension to calculate spring rate from a chosen ride frequency, measured motion ratio, and mass:

$$K_s = 4\pi^2 f_r^2 m_{sm} MR^2$$

K_s = Spring rate (N/m)
m_{sm} = Sprung mass (kg)
f_r = Ride frequency (Hz)
MR = Motion ratio (Wheel/Spring travel)

3. ROLL STIFFNESS, ROLL SENSITIVITY AND LATERAL LOAD TRANSFER DISTRIBUTION

Like picking ride frequencies for knock travel, a roll solidness must be picked next. The standardized roll firmness number is the roll inclination, communicated in degrees of body roll per "g" of sidelong acceleration. A lower roll inclination delivers less body roll per degree of body roll, bringing about a stiffer vehicle in roll. Typical values are given in Race Car Vehicle Dynamics by Milliken & Milliken:

- 0.2 – 0.7 deg/g for stiff higher downforce cars
- 0.7 – 1.8 deg/g for low downforce sedans

A stiffer roll gradient will produce a car that is faster responding in transient conditions, but at the expense of mechanical grip over bumps in a corner. Once a roll gradient has been picked, the roll gradient of the springs ought to be figured, the anti-roll bar stiffness is utilized to expand the roll gradient to the picked esteem. The roll gradient is normally not shared similarly by the front and back. we can call the roll gradient appropriation the Magic Number (Milliken calls it Total Lateral Load Transfer Distribution). The Magic Number is communicated as the level of the roll gradient taken by the front suspension of the car.

As a baseline, utilize 5% higher Magic Number than the static front weight dispersion. Roll gradients are degrees of body roll per g of lateral acceleration. Roll rates are Newton-meters of torque per degree of body roll or ARB bend. The accompanying conditions do no consider roll because of the tires.

Roll gradient of ride springs according to Milliken & Milliken (Race Car Vehicle Dynamics):

$$\frac{\phi_r}{A_y} = \frac{-W \times H}{K_{\phi F} + K_{\phi R}}$$

H = Cg to Roll axis dist (m)
W = Vehicle weight (N)
ϕ_r/A_y = Roll gradient from ride springs (deg/g)

$$K_{\phi F} = \frac{\pi (t_f^2) K_{LF} K_{RF}}{180(K_{LF}+K_{RF})}$$

$K_{\phi F}$ = Front roll rate (Nm/deg roll)
t_f = Front track width (m)
K_{LF} = LF Wheel rate (N/m)
K_{RF} = RF Wheel rate (N/m)

Remember that wheel rate is spring rate/ MR2 ; the effect of the spring at the wheel

$$K_{\phi R} = \frac{\pi (t_r^2) K_{LR} K_{RR}}{180(K_{LR}+K_{RR})}$$

$K_{\phi R}$ = Rear roll rate (Nm/deg roll)
t_r = Rear track width (m)
K_{LR} = LR Wheel rate (N/m)
K_{RR} = RR Wheel rate (N/m)

Total ARB roll rate required to increase the roll stiffness of the vehicle to the anticipated roll gradient:

$$K_{\phi A} = \frac{\pi}{180}\left[\frac{K_{\phi DES}K_T(t^2/2)}{[K_T(t^2/2)\pi/180 - K_{\phi DES}]}\right] - \frac{\pi K_W(t^2/2)}{180}$$

$K_{\phi A}$ = Total ARB roll rate needed (Nm/deg roll)
$K_{\phi DES}$ = Desired total roll rate (Nm/deg roll)
K_W = Wheel rate (N/m)
K_T = Tire rate (N/m)
t = Average track width between front and rear (m)

$$K_{\phi DES} = WH/(\phi/A_y)$$

W = Weight of vehicle (N)
H = Vertical distance from roll center axis to Cg (m)
ϕ/A_y = Desired total roll gradient, chosen earlier (deg/g)

Front and Rear Anti-Roll Bar stiffness:

$$K_{\phi FA} = K_{\phi A}N_{mag}MR_{FA}^2/100$$

$K_{\phi FA}$ = FARB roll rate (Nm/deg twist)
$K_{\phi A}$ = Total roll rate (Nm/deg roll)
N_{mag} = Magic Number (%)
MR_{FA} = FARB Motion ratio

$$K_{\phi RA} = K_{\phi A}(100-N_{mag})MR_{RA}^2/100$$

$K_{\phi RA}$ = RARB roll rate (Nm/deg twist)
$K_{\phi A}$ = Total roll rate (Nm/deg roll)
N_{mag} = Magic Number (%)
MR_{RA} = RARB Motion ratio

Keep in mind, the body goes about as a torsional spring in roll. It merits looking at the roll rate of your suspension to the roll rate of your frame if the body turns as much as the suspension, it could be a bigger territory of worry than the suspension. With steady state roll angles different front to rear, or different roll frequencies front to rear, chassis torsion will be induced- this should be kept in mind.

4. FACTORS AFFECTING VEHICLE DYNAMICS DUE TO SPRING AND RIDE RATE

There a number of other factors other than the various subsystems of a vehicle which affect its performance. These include:

1) Compliance in suspension, steering and other components.
2) Stiction or friction in spherical bearings being used for suspension and steering.
3) Incorrectly inflated or worn out tires.
4) Free-play in components.

Other similar factors can seriously affect the performance of a vehicle. If attention is not paid to these factors, the hard work that went into the accurate design of the subsystems would be a waste.

5. NUMERICAL APPROACH

This section will show a calculation of suspension kinetics for a car weighing 300kg with a front to rear weight distribution of 45:55.

Desired suspension frequencies: Front: 2.2Hz

Rear: 2.0Hz (for better turn-in and driver feedback during corner entry) Ride rates based on these frequencies:

Front: 17.196 N/mm

Rear: 14.212 N/mm

Suspension stiffness after accounting for tire spring rate (~230 N/mm): Front: 18.5856 N/mm

Rear: 15.1480 N/mm

Assuming a good motion ratio of 0.9 the spring rates would be Front: 18.7013 N/mm

Rear: 22.9452 N/mm Suspension roll stiffness

Front: 13899.51 Nm/radian

Rear: 11328.65 Nm/radian Vehicle roll rate or roll flexibility: 0.8614 deg/g

6. DISCUSSION

A LLTD value of 0.86 means that the rear transfers more load to the outer wheel in a corner than the front. This leads to a greater grip loss in the rear than the front leading to a slight over steer. This characteristic can be changed by employing anti-roll bars to change the lateral dynamics of the vehicle without having to disturb the vertical performance of the vehicle. As an example, say the LLTD value is required to be 0.9 instead of 0.86. We can use anti-roll bar up front to increase the outside the load transfer. The stiffness of the required roll bar can be easily calculated based on the change in weight transfer required.

7. CONCLUSION

The conventional concepts used for calculating suspension kinetics parameters are valid in FSAE application too but some changes have to be made to the ball park values that are assumed as per requirements in FSAE. The FSAE poses different requirements from its suspension system than a passenger car or a racing car with aerodynamic devices. This has to be kept in mind and while assuming suspension frequencies, lateral load transfer distribution etc for FSAE.

Based on the theoretical and numerical approach the following conclusions were made on

1. Maximum anticipated lateral acceleration: 1.75g if using racing slicks
2. Body roll due to maximum anticipated lateral acceleration: 1.5075 deg
3. Maximum wheel displacement due to 1.5075 deg of body roll: 1.6087 cm
4. Lateral load transfer distribution (LLTD) between the front and the rear suspension due to maximum lateral acceleration: 0.8642 (Front: Rear)

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