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An Overview of the Early Stage of Vehicle Modeling and Design

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ABSTRACT This is a paper intended for initial stage of vehicle modeling and design. The needs to determine a variety of vehicle suspension parameters required for initial design has been difficult and time-consuming task.

In order to facilitate a concise and efficient presentation of initial vehicle design procedure, this paper uses a mathematical model and physical geometry. Vehicle model consists of dimensions, inertias and mechanical constants. These vehicle model parameters divided into several categories : basic parameters, coefficients and constants, design specification, spring and damper, bush stiffness, stabilizer bar, suspension geometry, tire, and vehicle weights of various design condition.

This paper uses a vehicle design fundamental (VDF) program running under Windows 95 graphical interface. The features of VDF will be briefly outlined in this paper.

Keywords Vehicle, Vehicle Dynamics, Vehicle Modelling, Vehicle Design.

1. INTRODUCTION

Engineers can simulate the response of vehicles by computer to study their stability and comfort. The purpose of this paper is to provide details of the initial vehicle design and modeling. In order to facilitate a concise and efficient presentation of initial vehicle design procedure, this paper uses a mathematical model and physical geometry. Vehicle model consists of dimensions, inertias and mechanical constants. To evaluate the initial design performance, the following vehicle parameters were selected ; basic parameters, coefficients and constants, design specification, spring and damper, bush stiffness, stabilizer bar, suspension geometry, tire, and vehicle weights of various design condition. In addition, the present paper contains the light commercial vehicle example model and introduces the review of an aspect of vehicle design and performance briefly.

2. MODELING

2.1 WEIGHT DISTRIBUTION

Weight distribution means the percentage of the total weight distributed to each axle for a commercial vehicle.

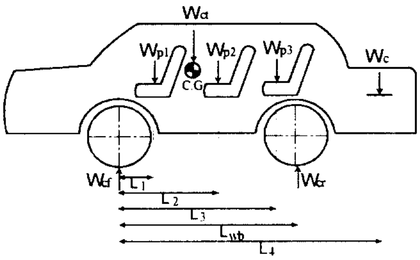

Fig. 1 Configure of Weight Distribution

Front Weight Distribution

$$W_{cf} = W_{ct} - W_{cr} \qquad (1)$$

Rear Weight Distribution

$$W_{cr} = W_{spr} + W_{usr} + \frac{W_{p1}L_{p1} + W_{p2}L_{p2} + W_{p3}L_{p3} + W_c L_{p4}}{L_{wb}} \qquad (2)$$

2.2 SUSPENSION

Suspension means whereby vehicle body is supported on its undercarriage, comprising springs, dampers and locating linkages. The preprocessor supports 2 suspension models, 'MacPerson' type & 'Double Wishbone' type. These models were chosen because they have a wide user community. Major difference between two models is control arm ; MacPerson type has only lower control arm, however Double Wishbone type has lower and upper control arm.

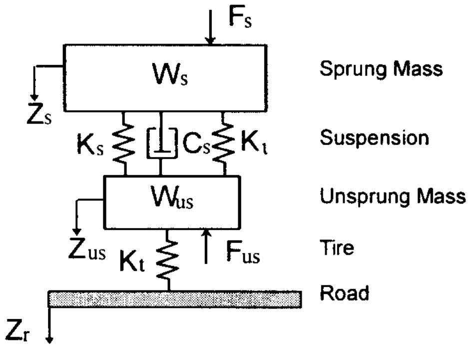

Fig. 2 Quarter-Car Model Representation of Suspension

- **FLAT RIDE** - Important feature which has improved the ride of vehicles is the "Flat Ride Tuning" of the two suspensions. The front spring should be softer than the rear for the "Flat Ride Tuning".

Ride Rate $$K_f = \frac{(2\pi \times f_{ride})^2}{W_{spf}} \qquad (3)$$

Wheel Rate $$K_w = \frac{K_t \times K_f}{K_t - K_f} \qquad (4)$$

Couple Spring Rate $K_c = \dfrac{K_f}{\eta^2(1+K_\eta)}$ (5)

● **BRAKING AND ACCELERATION PERFORMANCE** - Suspension spring in a vehicle can cause excessive pitching of the body during braking and acceleration. Thus, suspension geometries with 'Anti-Dive' and 'Anti-Lift' characteristics have been designed to partially correct these annoying pitching tendencies.

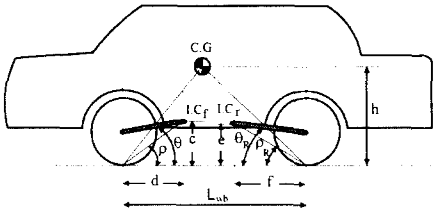

Fig. 3 Geometry of Dive and Lift

Anti-Dive $BAD(\%) = \dfrac{\tan\theta}{\tan\rho}$ (6)

Anti-Lift $RAL(\%) = \dfrac{\tan\theta_R}{\tan\rho_R}$ (7)

Anti-Squat $AAS(\%) = \dfrac{\tan\theta_R}{\tan\rho}$ (8)

where,

$$\tan\theta = \frac{c'}{d'}, \qquad \tan\theta_R = \frac{e'}{f'},$$

$$\tan\rho = \frac{c}{d} = \frac{h}{qL_{wb}}, \quad \tan\rho_R = \frac{e}{f} = \frac{h}{(1-q)L_{wb}}.$$

2.3 ROLL

Roll is an angular displacement of the sprung mass of a vehicle about its longitudinal axis.

● **ROLL GAIN** Roll gain means roll angel per unit lateral acceleration($1G = 9.8\mathrm{m/s^2}$). Roll gain, θ_s is obtained by

$$\theta_s = \frac{2W_{sp}h(a_y + G\theta_t)}{K_{rf}T_f^2 + K_{rr}T_r^2}. \tag{9}$$

where,

$$\theta_t = \theta_{sp} + \varphi_{usf} + \varphi_{spa},$$

$$\theta_{sp} = \frac{2W_{sr}h\{a_y + G(\varphi_{usa} + \varphi_{spa})\}}{K_f T_f^2 + K_r T_r^2},$$

$$\varphi_{spf} = \frac{2W_{spf}a_y G h_f}{K_f T_f^2}, \qquad \varphi_{spr} = \frac{2W_{spr}a_y G h_r}{K_r T_r^2},$$

$$\varphi_{usf} = \frac{2W_{usf}a_y G R_{tf}}{K_f T_f^2}, \qquad \varphi_{usr} = \frac{2W_{usr}a_y G R_{tr}}{K_r T_r^2},$$

$$\varphi_{usa} = \frac{\varphi_{usf} + \varphi_{usr}}{2}, \qquad \varphi_{spa} = \frac{\varphi_{spf} + \varphi_{spr}}{2}.$$

● **ROLL STIFFNESS DISTRIBUTION** A vehicle while cornering, the spring, bush, tire and stabilizer bar has a reaction force by lateral acceleration. The roll stiffness or roll rate of the suspension will influence the weight transfer on the wheels. The roll stiffness distribution as follows

Total Roll Stiffness $K_{rot} = \dfrac{W_{sp}a_y h_c}{\theta_s}$. (12)

Front and Rear Roll Stiffness

$$K_{rof} = \frac{K_{rf}T_f^2}{2\theta_s}, \quad K_{ror} = \frac{K_{rr}T_r^2}{2\theta_s}. \tag{13}$$

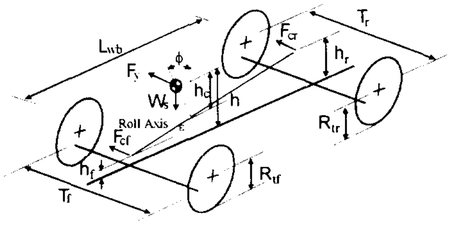

Fig. 4 Force Analysis for Roll of a Vehicle

Stabilizer Bar Roll Stiffness

Stabilizer bar(i.e., anti-roll bar) is a torsion bar coupling nearside and offside wheel suspensions of an independent suspension system, to minimize body roll.

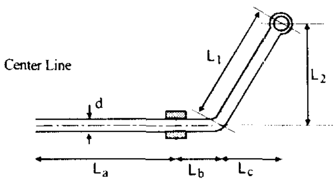

Fig. 5 Configuration of Stabilizer Bar

The Roll stiffness of the stabilizer bar can be represented by

$$K_{sb} = K_{rot} - (K_{rof} + K_{ror}), \tag{14}$$

$$K_{sb} = \frac{3EIL^2}{2[L_1^3 - L_a^3 + \frac{L}{2}(L_a + L_b)^2 + 4L_2^2(L_b + L_c)]}. \tag{15}$$

From above equation (15), we determined the diameter of stabilizer bar d_s obtained by

$$d_s = \sqrt[4]{\frac{8\left[L_1^3 - L_a^3 + \frac{L}{2}(L_a + L_b)^2 + 4L_2^2(L_b + L_c)\right]K_{sb}}{3\pi EL^2}}. \tag{16}$$

2.4 LATERAL DYNAMICS

Road wheel that responds to applied steering input and therefore influences the path taken by the vehicle. The angle between the projection of the longitudinal axis of a vehicle and plane of steered wheel is called steering angle. Lateral acceleration gives rise to the sideways force on a vehicle when cornering.

● **THEORETICAL STEER**

System of double-pivot steering in which two steered wheels pivot about a vertical axis and are steered by linked steering arms. The system was devised by Lankernsperger but takes its name from the patent agent Ackermann. It was originally introduced to prevent capsizing of horse drawn vehicles when turning sharply.

The steady-state cornering equations are derived from the application of Newton's Second Law along with the equation describing the geometry in turns.

For a vehicle traveling forward with a speed of V, the sum of the forces in the lateral direction from the tires must equal the mass times the centripetal acceleration.

$$\sum F_y = F_{yf} + F_{yr} = MV^2 / R \tag{16}$$

For small steer angles, the cornering forces acting at the front and rear tires are approximately given by

$$F_{yf} = \frac{W_f V^2}{GR}, \quad F_{yr} = \frac{W_r V^2}{GR}. \tag{17}$$

where

$$W_f = \frac{Wa}{2(a+b)}, \quad W_r = \frac{Wb}{2(a+b)}$$

With the required lateral forces known, the slip angles at the front and rear wheels are also established. That is

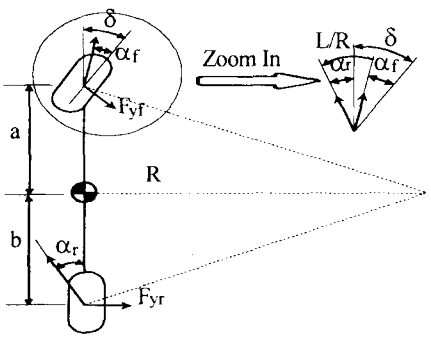

Fig. 6 Cornering of a Bicycle Model

$$\alpha_f = \frac{F_{yf}}{2C_{\alpha f}}, \quad \alpha_r = \frac{F_{yr}}{2C_{\alpha r}}, \tag{18}$$

Thus, the steer angle

$$\delta_{ak} = 57.3\frac{L_{wh}}{R_s} + \alpha_f - \alpha_r, \tag{19}$$

For a understeer vehicle, a characteristic speed V_{char} may be identified. It is the speed at which the steer angle required to negotiate a turn in equal to $2(a+b)/R$.

$$V_{char} = \sqrt{57.3 L_{wb} \frac{G}{K_\phi}} \tag{20}$$

For an oversteer vehicle, a critical speed V_{crit} can be identified. It is the speed at which the steer angle required to negotiate any turn is zero.

$$V_{crit} = \sqrt{-57.3 L_{wh} \frac{G}{K_\phi}} \tag{21}$$

- **PRACTICAL STEER**

In a practical steering system, the understeer coefficient, K, for a vehicle is the result of tire, vehicle and steering system parameters. Cole[3] and Gillespie[4] represents the expression for understeer coefficient as

$$\delta_t = 57.3\frac{L_{wh}}{R} + Ka_y \tag{22}$$

$$K = K_{\text{weight distribution}} + K_{\text{rigid body}} + K_{\text{roll steer}} + K_{\text{roll camber}} + K_{\text{lateral force}} + K_{\text{lateral force camber}} + K_{\text{tire aligning torque}} + K_{\text{tire aligning torque camber}} \tag{23}$$

where,

$$K_{\text{weight distribution}} = \left(\frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}}\right)G$$

$$K_{\text{rigid body}} = \frac{1}{L_{wb}} \frac{(C_{\alpha f} + C_{\alpha r})(N_{\alpha f} + N_{\alpha r})}{C_{\alpha f} C_{\alpha r}}$$

$$K_{\text{roll steer}} = (E_{\phi f} - E_{\phi r})\theta_t$$

$$K_{\text{roll camber}} = \left(\frac{C_{\gamma f}}{C_{\alpha f}}\Gamma_{\phi f} - \frac{C_{\gamma r}}{C_{\alpha r}}\Gamma_{\phi r}\right)\theta_t$$

$$K_{\text{lateral force}} = \left(\frac{E_{Yf} W_{spf} - E_{Yr} W_{spr}}{2}\right)G$$

$$K_{\text{lateral force camber}} = \left(\frac{\Gamma_{yf} W_{spf} - \Gamma_{yr} W_{spr}}{2}\right)G$$

$$K_{\text{tire aligning torque}} = \left(E_{nf} N_f - E_{nr} N_r\right)$$

$$K_{\text{tire aligning torque camber}} = \left(\Gamma_{nf} N_f \frac{C_{\gamma f}}{C_{\alpha f}} - \Gamma_{nr} N_r \frac{C_{\gamma r}}{C_{\alpha r}}\right)$$

In the above equations, $C_{\alpha f}, C_{\alpha r}$ represents the cornering stiffness and $C_{\gamma f}, C_{\gamma r}$ camber stiffness of a single tire. $N_{\alpha f}, N_{\alpha r}$ is the tire align torque steer per slip angle, $N_{\gamma f}, N_{\gamma r}$ tire align torque steer per camber angle and N_f, N_r tire align torque. The values of $\Gamma_{\phi f}, \Gamma_{\phi r}, \Gamma_{nf}, \Gamma_{nr}$ and Γ_{yf}, Γ_{yr} represent the roll camber coefficient, align torque deflect camber coefficient and lateral force deflect camber coefficient of a understeer coefficient K, respectively. And, $E_{\phi f}, E_{\phi r}, E_{nf}, E_{nr}$ and E_{yf}, E_{yr} represent the roll steer coefficient, align torque deflect steer coefficient, lateral force deflect steer coefficient, respectively.

3. SIMULATION

3.1 SIMULATION MODEL AND PROCEDURE

This paper uses a vehicle design fundamental (VDF) program running under Windows 95 graphical interface. Visual C++ 4.0 toolkits was used to build an Windows 95 interface for the program. The code was created in the C++ programming language.

The VDF code is a formulation with very extensive input parameter requirements. Table presents a sample list of theses parameters. The data in this simulation derives from a light commercial vehicle. The vehicle model includes the chassis, four wheels, and the steering system. Coefficients parameters and design properties used in the VDF are listed in Table 1.

Table 1. Coefficients and Design Parameters

Variable Name	Symbol	Units
Bush Effective Ratio	η_f, η_r.	dimensionless
Linkage Ratio	R_f, R_r.	dimensionless
Brake Torque Ratio	q.	dimensionless
Road Friction Coefficient	μ.	dimensionless
Roll Center Height	h_f, h_r.	mm
Passenger Hip Position	L_{p1}, L_{p2}, L_{p3}	mm
Wheel Base	L_{wb}.	mm
Wheel Tread	T_f, T_r	mm

Table 1. (continued)

Variable Name	Symbol	Units
Total C.G Height	h_{CG} ·	mm
Cargo C.G Height	h_{cargo} ·	mm
Imaginary Point	IC_f , IC_r	mm
Rear Spring Span	L_s ·	mm

3.2 SIMULATION RESULTS & DISCUSSION

Comparisons between simulation data and generally accepted data showed reasonable agreement for weight distribution, ride mode, roll mode and steering. A characteristic speed V_{char} and yaw velocity was about 0.7165 m/s and 0.000943(deg/s)/deg, respectively. Stiffness rate of stabilizer bar were 68714608 N·mm/rad and the diameter of stabilizer bar were 25.68 mm.

- **WEIGHT DISTRIBUTION** - Two vehicle parameters, front wheel weight distribution(W_{cf}) and rear wheel weight distribution(W_{cr}) are varied individually from 46% to 55% of their value. Simulation results indicated that larger value of rear sprung 2nd rate led to improvements in vehicle stability.
- **RIDE MODE** - The ride mode has the values of front ride frequency and rear ride frequency, 1.44 Hz and 1.53Hz, respectively.
- **ROLL MODE** - For the study of roll mode response, roll gain calculations were completed. The roll gain was about 4.4 degree(usually between about 5 to 7 degree). Most cases of responses shows good results.
- **STEERING** - To observe steering characteristics, a simulation study was conducted while keeping a lateral acceleration of 0.3 G . Consequently, the vehicle exhibits strong understeer, 7.5 deg/0.3G.
- **PITCHING MOTION** - The pitching motion study reveals that the vehicle model shows a good pitching motion. The relatively poor results for the anti-squat. To get valid results, changing imaginary link of rear suspension has effects on the pitching motion.

4. CLOSURE

Unreasonable parameters can arise from various sources. Thus, input parameter verification is the most challenging aspect of the validation process of a computer simulation for a particular vehicle.

This paper has presented a detailed overview of the initial stage of a vehicle design. Individual input and output parameters were presented and discussed. The main purpose of this paper is to provide a standard model available for use by the vehicle simulation community.

For the future work, parameter identification will be performed using the empirical data obtained from a road test. The parameter identification process will include both pure and comprehensive slip tests with various types of tire and suspension models.

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NOMENCLATURE

a_y : centripetal acceleration or lateral acceleration
c : height of front instant center from the ground
d : distance from front wheel center to front instant center
e : height of rear instant center from the ground
f : distance from rear wheel center to rear instant center
f_{ride} : ride frequency
K_f : ride rate or front suspension stiffness
K_r : rear suspension stiffness
K_η : coupling stiffness
K_ϕ : understeer coefficients
R : radius of turns
T : wheel track or tread
T_s : spring track for an rear axle suspension
V_{crit} : critical speed of a vehicle
V_{char} : characteristic speed of a vehicle
η : coupling coefficient
ρ : angle between instant center and tire-ground contact point
θ_R : angle made by line SI with the ground
θ_{sp} : roll angle of the sprung weight
θ_t : total roll angle
θ_s : roll gain ; roll angle per unit gravitation(1G)
θ_{sp} : roll angle of the sprung weight
φ_{spf} : roll angle for front sprung mass due to the weight transference of the sprung weight
φ_{spr} : roll angle for rear sprung mass due to the weight transference of the sprung weight
φ_{usf} : roll angle on the front tires due to the weight transference of the unsprung weight
φ_{usr} : roll angle on the rear tires due to the weight transference of the unsprung weight
φ_{usa} : average roll angle on the tires
φ_{spa} : average roll angle for sprung mass

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