ED5510 - Design of Automotive Systems

Design of Project RP

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Inspiration

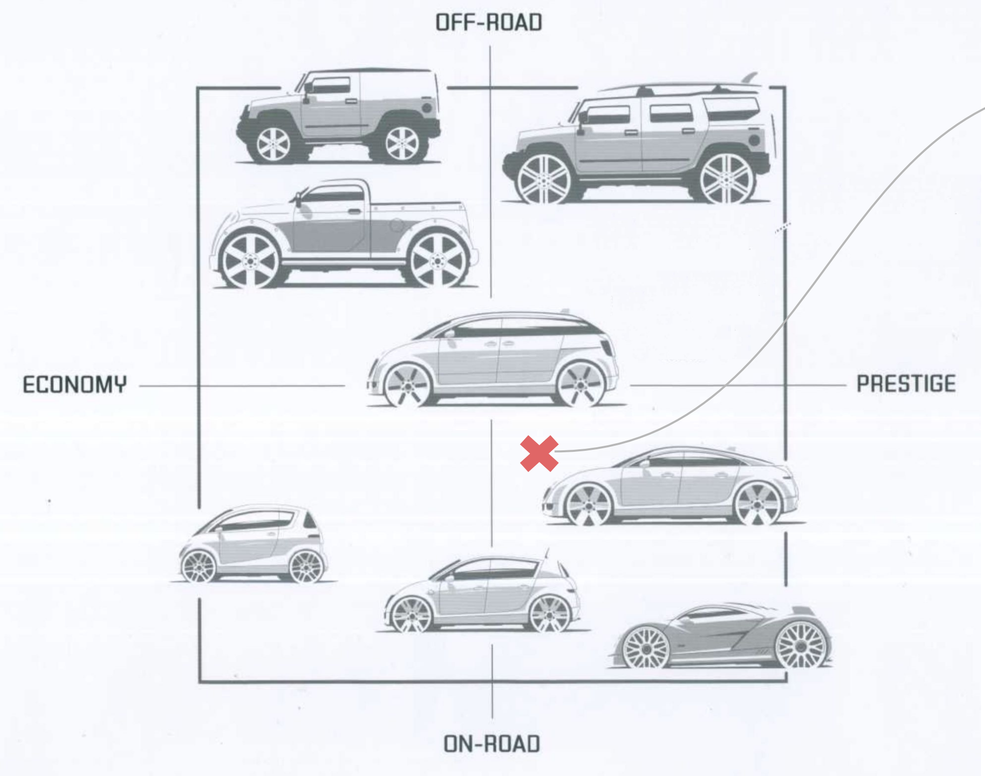
Our target market is the hot hatchback car segment, which is not a prominent segment in India currently. This is a highly popular and competitive segment in other countries, and we believe that the growing youth executive population in India would find this an aspirational goal in the future.

Unique Selling Points

The salient features of our intended design are-

- First indigenous Hot-Hatch
- Electric powertrain
- Multiple driving modes for a range of scenarios
- Dual motor AWD
- Lightweight, Agile Design focused on driving experience
- 3 door (2 side and 1 tail)
- Steer by wire + Brake by wire
- Active suspension control
- Active aerodynamic package
- Advanced safety features (ABS, Collision Avoidance etc.)

Vehicle Positioning



Proposed Design

Market Study

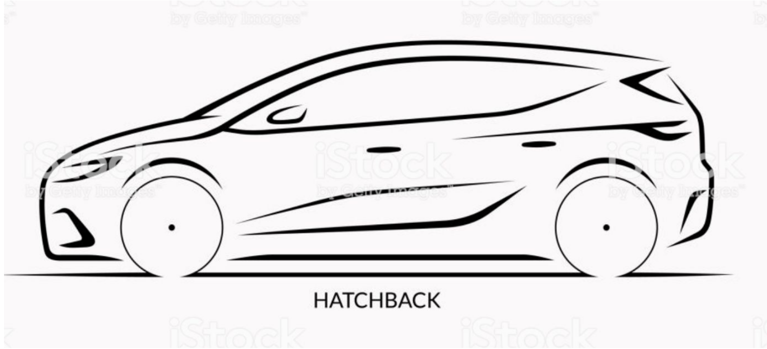
	Civic Type-R	VW e-Golf	Renault Clio Sport	Renault ZOE	Vauxhall GTC VXR	Ford Focus RS	VW Golf GT	Renault ZOE e-sport (concept)
Power	235 kW	100 kW	147kW	65 kW	205 kW	260 kW	86 kW	340 kW
Top speed	270 km/h	150 km/h	230 km/h	135 km/h	155 mph	250 km/h	200 km/h	210 km/h
Price	$ 33,900	$ 33,250	$ 33,000	$ 41,000	$ 35,200	$ 41,000	$ 36,500	N/A
Wheelbase	2,700 mm	2,620 mm	2,589 mm	2,588 mm	2,695 mm	2,650 mm	2,620 mm	N/A
Track	1,599 mm	1,521 mm	1,500 mm	1,510 mm	1,588 mm	1,524 mm	1,588 mm	N/A
Height	1,434 mm	1,492 mm	1,434 mm	1,448 mm	1,482 mm	1,470 mm	1,492 mm	N/A
Weight	1,320 kg	1,610 kg	1,204 kg	1,470 kg	1,475 kg	1,570 kg	1,301 kg	1,400 kg
Tyre Size	245/30R20	P205/55R16	205/45 R17	195/55 R16	245/40 R19	245/40 R19	245/40 R19	N/A

Vehicle Layout

Height
1400 mm

Overall length
4000 mm

Wheelbase
2570 mm



Track width
1500 mm

Unladen weight
1500 kg

Motor Selection - Assumptions

1. Vmax = 200 km/h
2. A = 2.1 m^2
3. Cd = 0.35
4. Mass (laden) = 1700 kg
5. Rolling resistance coefficient = 0.02
6. Transmission Efficiency = 0.8
7. Motor max rpm = 1047.22 rad/sec or 10000 rpm
8. Peak Accln = 5 m/s^2
9. Tyre radius = 350mm
10. Gear ratio = 6.59

Motor Selection

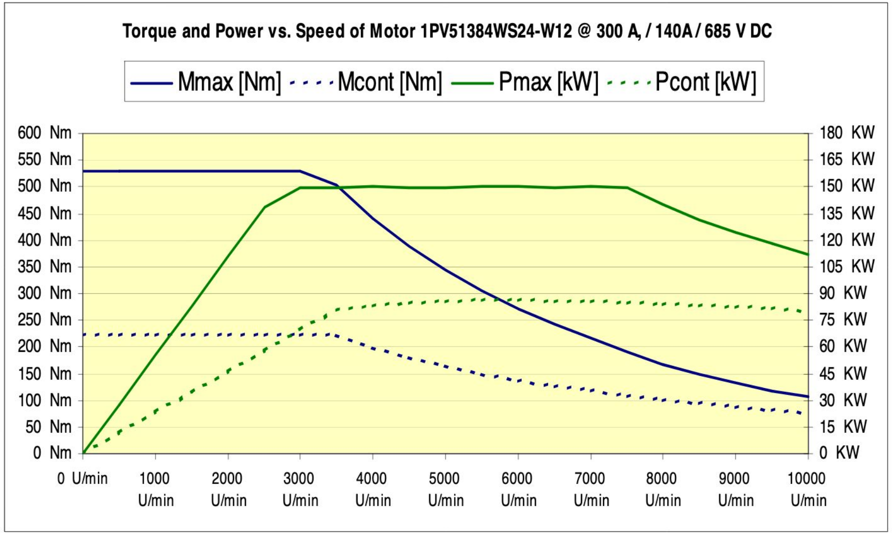
Results:

1. Pmax = 120 kW
2. Total Torque = 450 Nm

Sample Motor from market: [Drive Motor 1PV5138-4WS24]

$$P = 0.5\rho C d A v^3 + \mu M g v$$

$$G = \frac{\omega_{max,motor}}{v_{max}/r_{wheel}}$$



$$T_{wheel} = T_{motor} * G * \eta$$

Powertrain

$$v(i+1) = \Delta t(\frac{T_{wheel}(v(i))}{m * r_{wheel}} - \frac{K * v(i)^2}{m} - f_r * g) + v(i), \quad \text{for } \omega \leq \omega_{rated}$$

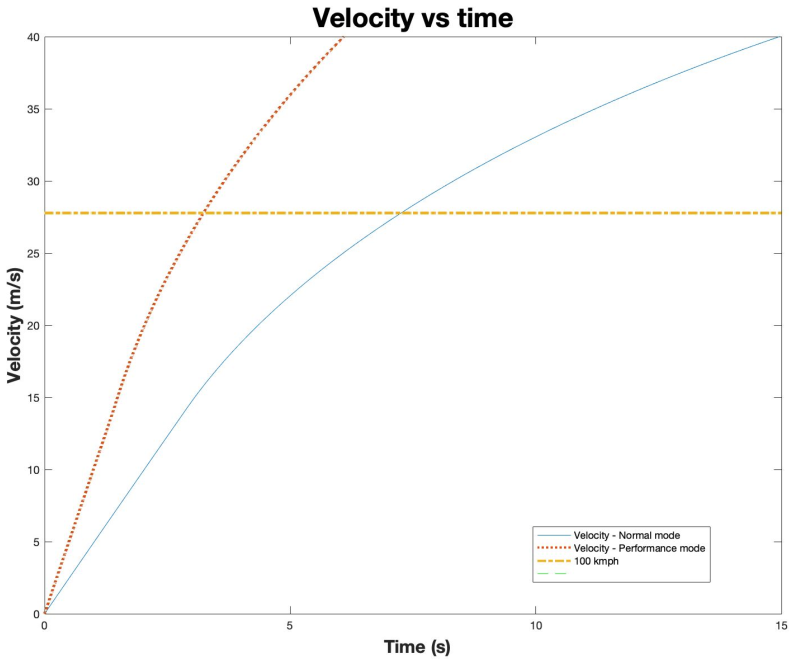
$$= \Delta t(\frac{P_{max} * \eta}{m * v(i)} - \frac{K * v(i)^2}{m} - f_r * g) + v(i), \quad \text{for } \omega > \omega_{rated}$$

Normal Mode:

- Pmax = **120 kW**
- Tmax = 450 Nm
- 0-100 kmph time = **7.32 s**

Performance Mode:

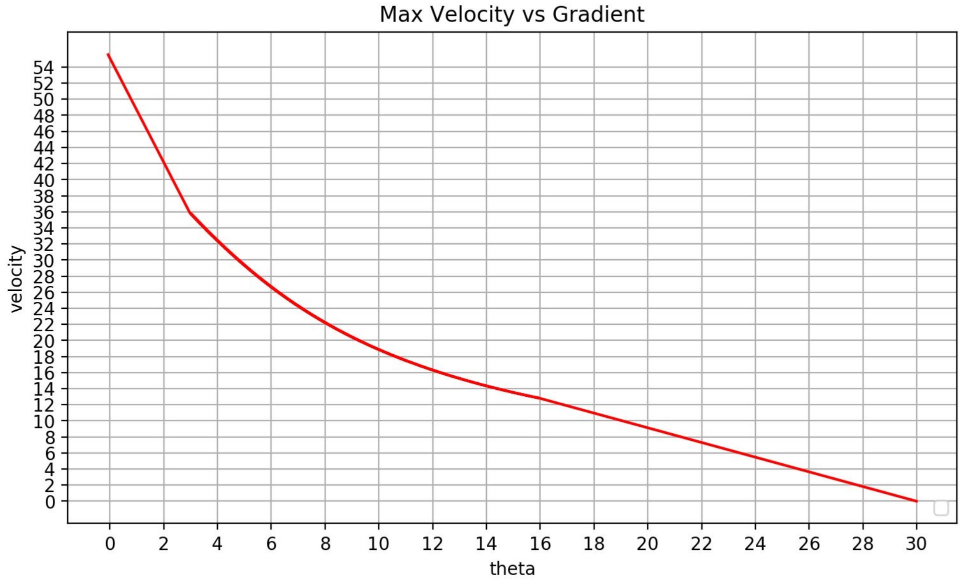
- Pmax = **270 kW**
- Tmax = 900 Nm
- 0-100 kmph time = **3.17 s**



Gradeability

- Law requirement = 7 deg (AIS 003)

$$F_{tot}(v) = mgsin\theta + 0.5C_d\rho Av^2 + \mu mgcos\theta$$



Battery Sizing

Assumptions:

- Min. Required Range = **300 km**
- Mass = **1700 kg** (with passengers)
- Drive Cycle - Artemis Cycle
- Battery Efficiency = **0.8**

Calculated Values:

- Distance per cycle = **28.7 km**
- Energy per cycle = **7 kWh**
- Battery Size = **80 kWh**
- Range for NEDC cycle = **465 km**
- Range for WLTP cycle = **409 km**

$$E_{battery} = \frac{E_{cycle} * d_{desired}}{d_{cycle}}$$

$$ExpectedRange = \frac{d_{cycle} * E_{battery} * \eta_{battery}}{E_{cycle}}$$

Accessories Power

- Avg. actuator motor - 10W
  - https://www.beckhoff.com/english.asp?drive_technology/default.htm
- No. of motors required by:
  - Steering - 2
  - Brakes - 4
  - Throttle control - 3
- LED lighting = 10W*4 = 40 W
- HVAC - 5kW
  - https://www.hindawi.com/journals/mse/2013/935784/
- Speaker system = 500 W

Total = **6kW**

Accessories Energy

- Total time = 11150 s
- Total power = 6000W
- Total Energy = 66.9 MJ = 18.53 kWh
- => Extra energy required = ~20 kWh
- => Extra battery capacity = **20 kWh**
- => Final battery capacity = Drive + Accessories = **100 kWh**

Differential

Our design does not consider off-road driving conditions, thereby eliminating locked differential. We considered mechanical clutch limited slip differential and Torque vectoring differential and have chosen Mechanical clutch LSD based on constraints of cost,handling and driving experience. It will be used for both front and rear wheels.

Mechanical clutch LSD :

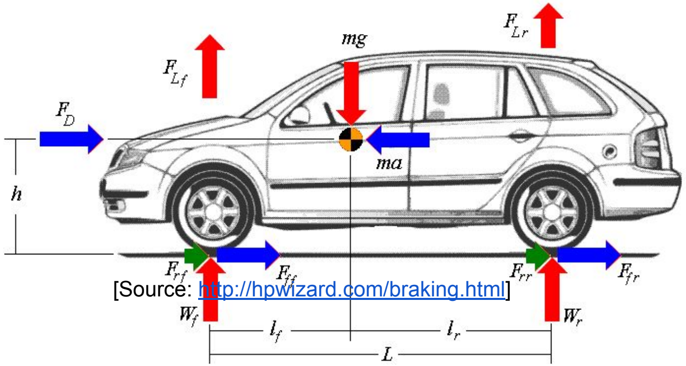
- Although this works well in off-road conditions, it works better in paved road surfaces as traction is near perfect. The traction becomes difficult to manage in off-road conditions thereby making it a bad choice.
- Less tire wear and axle shaft wear
- Not too expensive

Brakes

- Considering the intended use of the car, Disc brakes are chosen over Drum brakes for the following reasons:
  - Disc brake provide consistent braking over a wider temperature range
  - Disc brakes have better heat dissipation as compared to drum brakes
- Vented disc brakes would be preferred to increase the heat dissipation and keep the temperature under the limit, under extreme driving
- Brakes being located at the extremes of the car, they have substantial effect on the moment of inertia of car. They also contribute to the unsprung mass and their mass would be minimized
- Smaller rear brakes can be used as compared to the front as torque requirements are less

Brakes Sizing

- The aerodynamic forces, rolling resistance and the gradient of the surface is assumed to be zero
- CoG longitudinal position: **1.285 m** (from front axle)
- CoG height : **0.5 m**
- Mass of the car with passenger = **1700 kg**
- Front brake bias for simultaneous locking of wheels:
- Bias increases with mu value, Fixed bias taken for maximum mu of **0.9**
- Brake Bias (Front : Rear) : **70:30** (Ensuring Stability)
- Max Braking Torque (Front): **3.35 kNm**
- Max Braking Torque (Rear): **1.43 kNm**

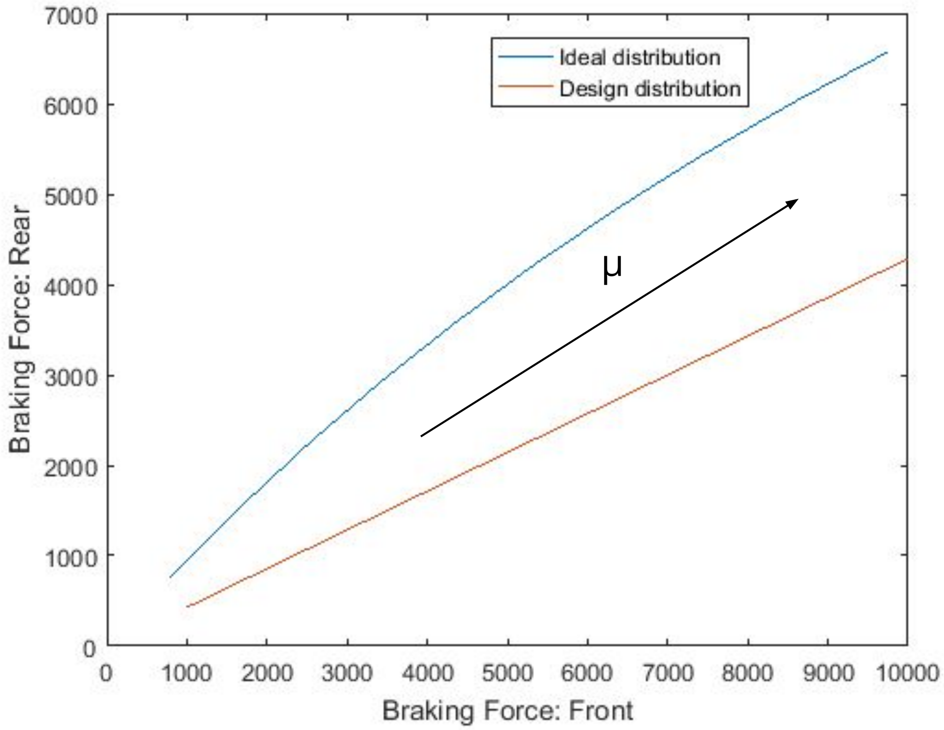


[Source: http://hpwizard.com/braking.html]

$$\lambda^* = (l_r + \mu h)/L$$

$$M_{bf} = \mu l_r M g r/(L - \mu h/\lambda)$$
$$M_{br} = M_{bf}(1 - \lambda)/\lambda$$

Braking Force Distribution



$$F_{max,f} = \mu M g(b + \mu h/L)$$

$$F_{max,r} = \mu M g(a - \mu h/L)$$

Braking standards (IS 11852-9)

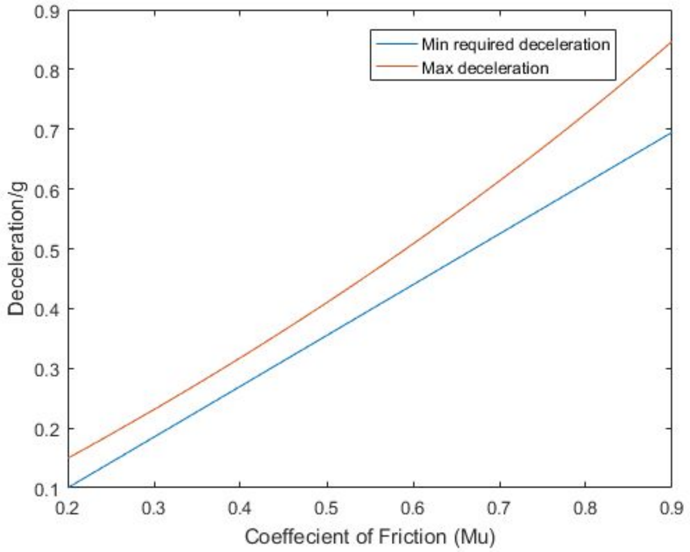
According to IS 11852-9:

- For mu between 0.2 and 0.8

  Deceleration (in g's) $d \geq 0.1 + 0.85(\mu - 0.2)$

- Stopping distance from 100 to 0 kmph required by law : **70 m**
- Response time **< 0.6s** (Assumption)

- Min. Stopping distance obtained : **51.3 m**



Tyres

We are looking at rim size diameter between **17" and 19"**, after considering the tradeoff between tyre grip and acceleration. The following are considered:

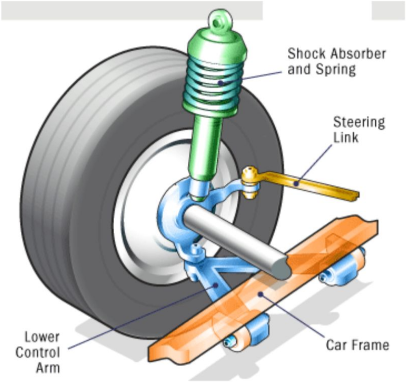
1. **Michelin Pilot Super Sport** : ***Pros*** - High levels of grip, good noise, comfort, durability; ***Cons*** - Pricey
2. Pirelli P Zero : *Pros* - Superior acceleration and braking power, excellent grip Cons - Less durable
3. MRF Perfinza CLX1 : *Pros* - Advanced grip, good in wet weather, very affordable Cons - Available in limited sizes

Suspension

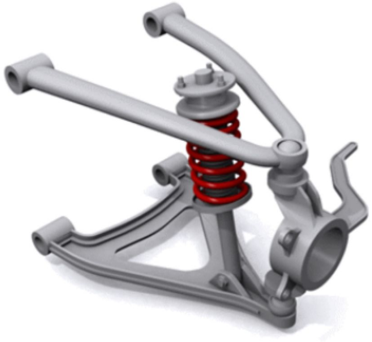
There are two prevalent suspension architectures used in the industry:

- MacPherson Strut System
- Double Wishbone System

The former is simpler in design, cheaper to manufacture and occupies less horizontal space, whereas the latter has better handling characteristics and provides the suspension engineer with greater parameter flexibility.



MacPherson Strut System
Image Source: ED5160 Lecture notes



Double Wishbone System
Image Source: ED5160 Lecture notes

Suspension Choice

The aim here is to design a high-performance hatchback with superior ride characteristics than the average hatch that is currently available. Hence, the double wishbone system is chosen as the preferred architecture over the MacPherson suspension system for both the front and rear wheels.

However, future design constraints may force a change in this choice, especially at the front, as the double wishbone system occupies more space than the alternative.

Suspension Design

Assumptions:

- Tyre chosen - 345/30ZR19 109Y XL BSW Michelin Pilot Sport Tyres
- Sprung mass = 380 kg per wheel
- Unsprung mass = 35 kg per wheel

Suspension - First Order Analysis

The initial values for the spring stiffness and damper coefficient were calculated using the values specified by Gillespie for the natural frequency of a passenger vehicle.

The reference values taken are:

- **ω**n = 1.5 Hz
- Damping Ratio = 0.4

Which result in:

- Ks (Spring stiffness) = 36.865 KN/m
- Cs (Damping coefficient) = 2.994 KNs/m

$$\omega_n = \sqrt{\frac{K_s * K_t}{(K_s + K_t) * m_s}}$$

$$\omega_d = \omega_n \sqrt{1 - \zeta_s^2}$$

$$\zeta_s = \text{Damping ratio}$$

$$= C_s / \sqrt{4 K_s M}$$

Vibration Analysis

- Displacement Power spectral density *Gd* (m3)
- Wave number *n* (cycles/m) ϵ (2e-2,1e3)
- *w* = 2, *n0* = 0.1 cycles/m
- Phase of *sine* waves taken as random

$$G_d(n) = G_d(n_0) \cdot \left(\frac{n}{n_0}\right)^{-w}$$

$$z_R(x) = \sum_{i=1}^{N} A_i \sin(\Omega_i x - \varphi_i),$$

$$A_i = \sqrt{\Phi(\Omega_i)\frac{\Delta\Omega}{\pi}},$$

$$\Delta\Omega = \frac{\Omega_N - \Omega_i}{N-1},$$

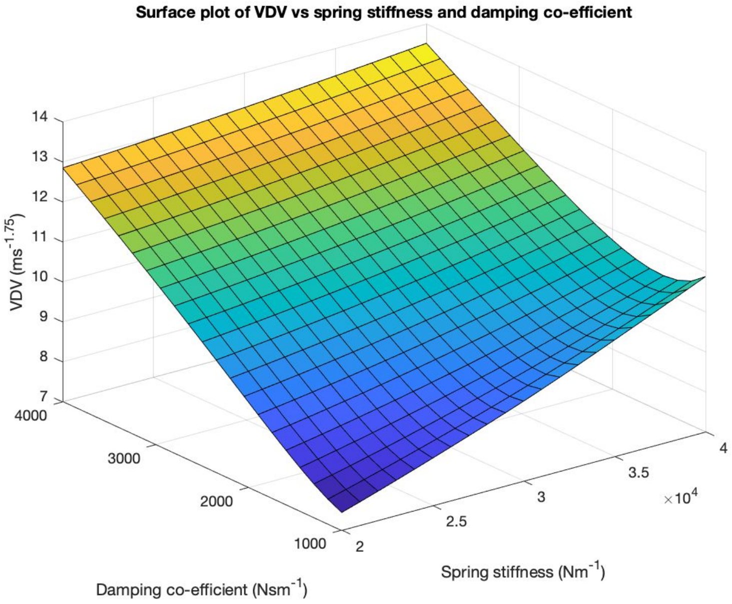
Table 1 ISO 8608 thresholds of the first four classes (A–D)

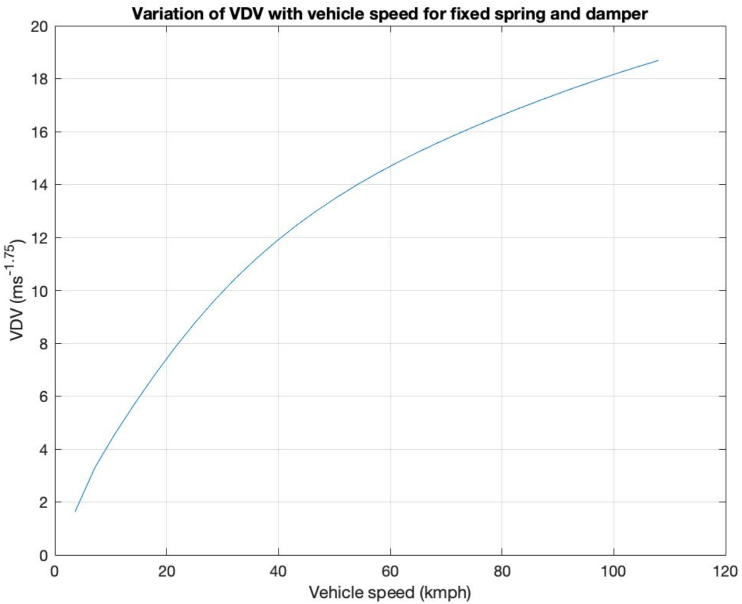
ISO 8608 class	$G_d(n_0)$ (10^{-6} m^3)
A (very good)	<32
B (good)	32–128
C (average)	128–512
D (poor)	512–2,048

VDV Analysis

Road quality - Average
Simulation time - ~10 mins

$$VDV = \left\{ \int_0^T [a_W(t)]^4 dt \right\}^{\frac{1}{4}}$$





Steering

- Our vehicle category: M1 (AB Hatchback)
- The vehicle shall be able to maneuver on either lock inside a circle of 12.5 m radius without any of its outermost points projecting outside the circumference of the circle
- Turning circle diameter (outer) shall not exceed 24 m.

Sl No.	Vehicle Category	Intact			With a Failure		
		Maximum Effort daN	Time s	Turning Radius m	Maximum Effort daN	Time s	Turning Radius m
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
i)	M_1	15	4	12	30	4	20
ii)	M_2	15	4	12	30	4	20
iii)	M_3	20	4	12[1)]	45[2)]	6	20
iv)	N_1	20	4	12	30	4	20
v)	N_2	25	4	12	40	4	20
vi)	N_3	20	4	12[1)]	45[2)]	6	20

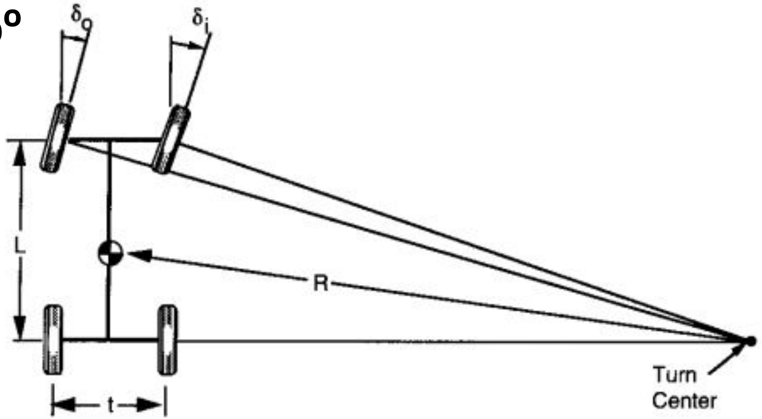
[1)] Or full lock, if 12 m radius is not attainable.
[2)] 50 for rigid vehicles with 2 or more steered axles excluding self-tracking equipment.

Steering

- Turning radius requirement (IS 12222 : 2011) : **12m**
- Based on market study of cars in India, target turning radius (outer): **6.8m**
- For low speeds, no lateral slip assumed
- Ackermann geometry used to find the max steer angle
- Max Steer angle obtained: **28.2^o**
- Steering wheel angle limits set as : **-270^o** to **270^o**
- Steering ratio : **9.6:1**

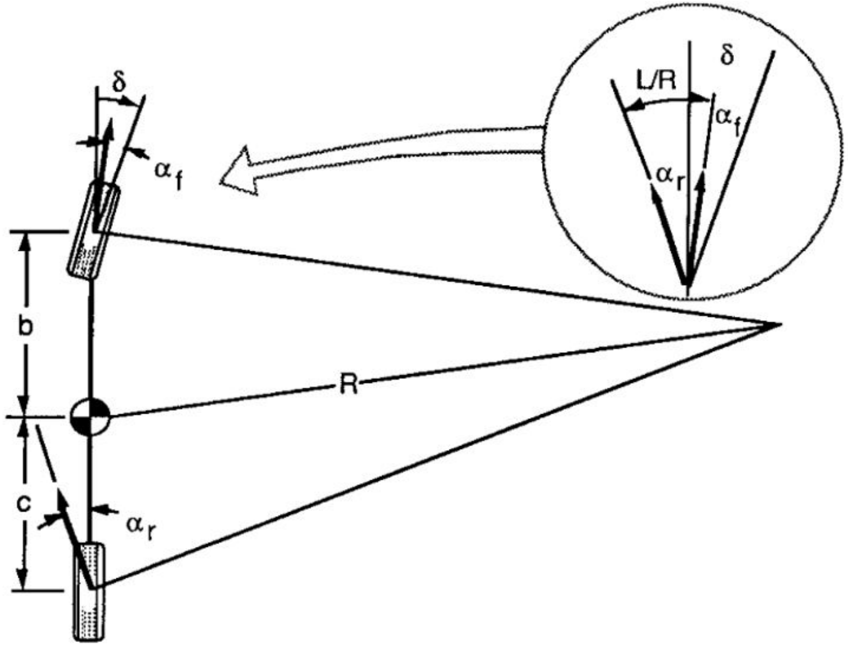
$$\delta_o = tan^{-1}(\frac{L}{R+t/2})$$

$$\delta_i = tan^{-1}(\frac{L}{R-t/2})$$



Steering

- Same tyres on front and rear
- Aiming for 50:50 mass distribution for neutral steer.



[Source: Fundamentals of Vehicle Dynamics, Thomas D. Gillespie]

Understeer Gradient

- Understeer gradient depends on:
  - **Tire cornering stiffness**
  - **Lateral load transfer**
  - Camber stiffness
  - Roll steer
  - Lateral force compliance steer
  - Aligning torque
  - Steer system

$\Longrightarrow$ Not considered

$$K_{tires} = \frac{W_f}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}}$$

$$K_{llt} = \frac{W_f}{C_{\alpha f}} \frac{2b\Delta F_{zf}^2}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}} \frac{2b\Delta F_{zr}^2}{C_{\alpha r}}$$

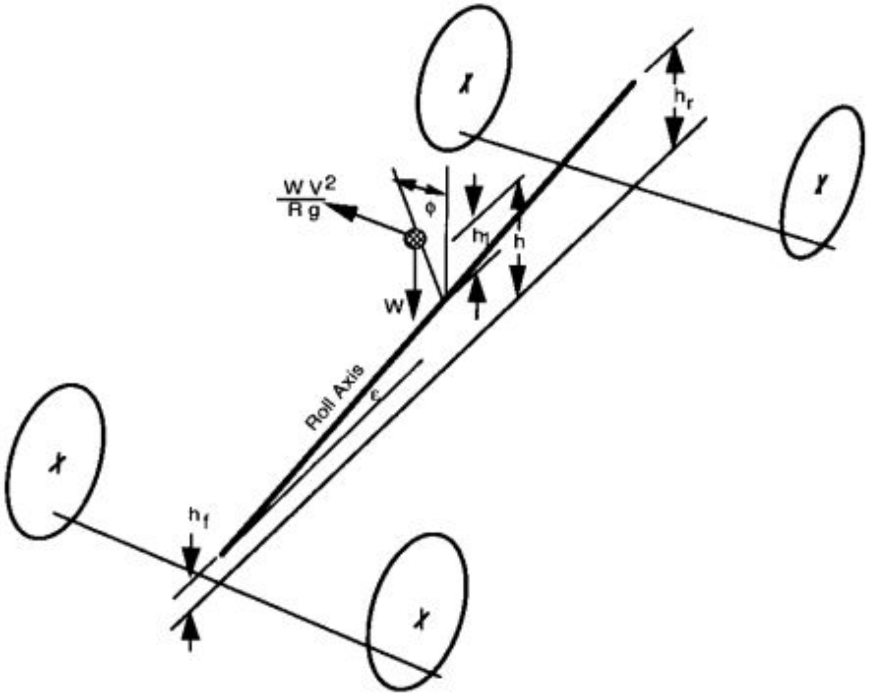
- 50:50 weight distribution assumed between front and rear axle
- Same tyres used on each corner

Understeer Gradient

- Roll rate of **4 deg/g** targeted
- Ks (Spring stiffness) = 36.865 KN/m
- S = 1.2 m
- h1 = 0.225m obtained

$$K_\phi = 0.5 K_s s^2$$

$$R_\phi = d\phi/da_y = Wh_1/(K_{\phi r} + K_{\phi f} - Wh_1)$$



Understeer Gradient

- To determine front and rear roll centre height, understeer gradient is determined for variation in weight on front axle.
- Critical speed in case of oversteer limited to 60 m/s
- hf = 0.375 m, hr = 0.175m obtained
- Cornering stiffness assumed as $C_\alpha = 2aF_Z - 2bF_Z^2$
- 'a' and 'b' found using:
  - Max cornering stiffness = 90,000 N/rad
  - Fz for max cornering stiffness (on the axle) = 8000N

$$V_{critical} = \sqrt{-Lg/K}$$

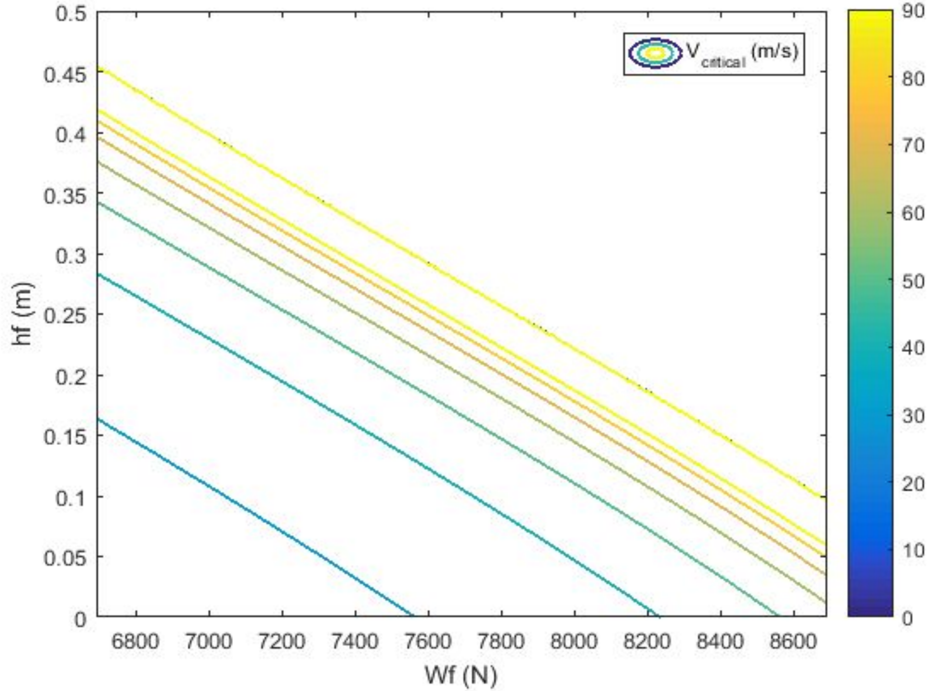
$$K_{llt} = \frac{W_f}{C_{\alpha f}} \frac{2b\Delta F_{zf}^2}{C_{\alpha f}} - \frac{W_r}{C_{\alpha r}} \frac{2b\Delta F_{zr}^2}{C_{\alpha r}}$$

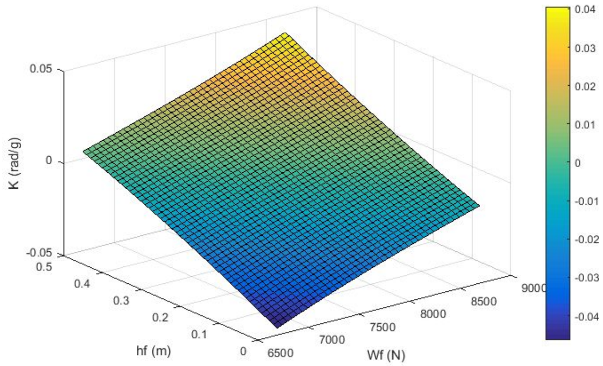
$$\Delta F_{zf} = \frac{W_f h_f (V^2/Rg)}{t_f} + K_{\phi f} \frac{W h_1 (V^2/Rg)}{t_f (K_{\phi f} + K_{\phi r} - W h_1)}$$

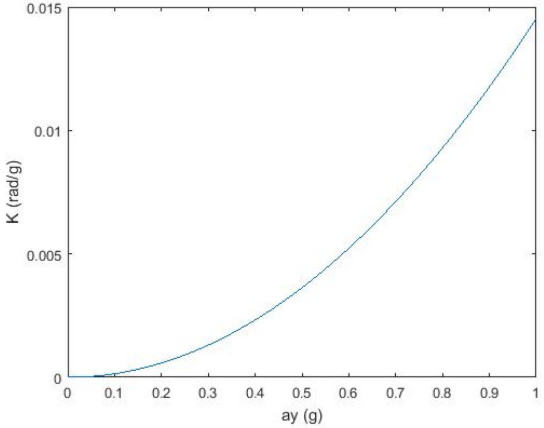
$$\Delta F_{zr} = \frac{W_r h_r (V^2/Rg)}{t_r} + K_{\phi r} \frac{W h_1 (V^2/Rg)}{t_r (K_{\phi f} + K_{\phi r} - W h_1)}$$

Understeer Gradient

Critical velocity for different roll centre heights and front axle loads



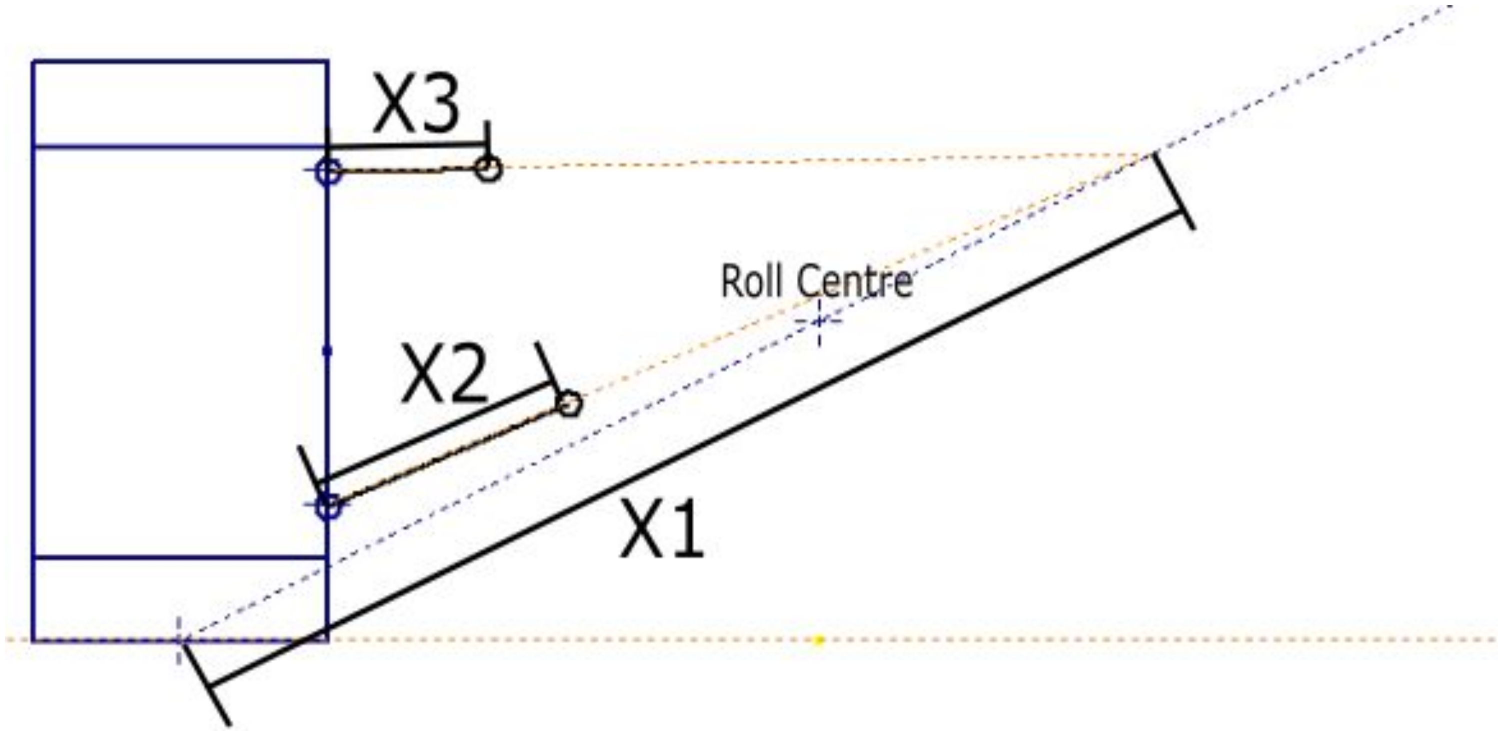




Suspension geometry

- Double wishbone suspension assumed on all four wheels
- Outboard points:
  - Vertical position: Kept as far apart as possible, to reduce the forces in the wishbones
  - Horizontal position: Assumed at the inside edge of the wheels
- Inboard points:
  - Constraint imposed by roll centre position, space requirements
  - Objective to minimize camber variation with roll, minimize lateral movement of contact patch with vertical movement of the wheel

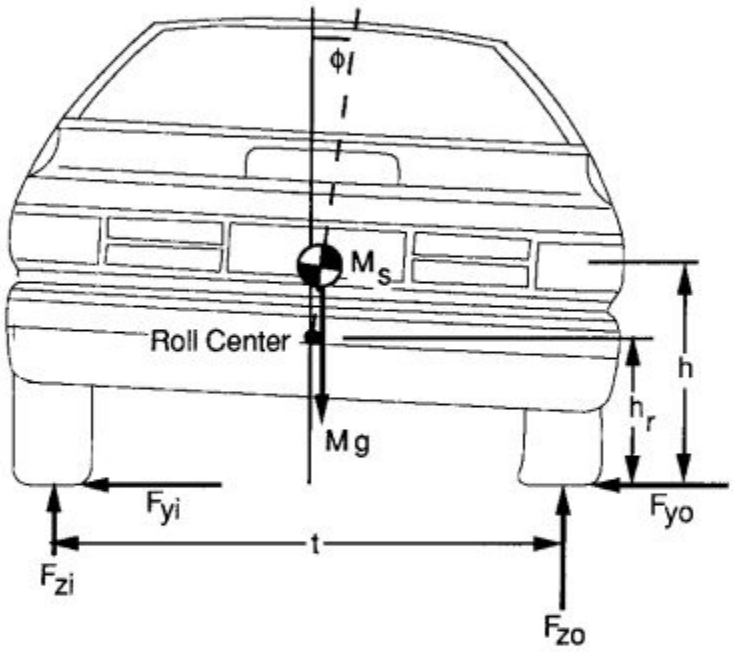
Suspension geometry



Rollover

- Rigid wheels considered
- Roll rate = 0.07 rad/g (4 deg/g)
- h_r = 0.275 m
- Lateral acceleration for rollover, a_y=1.45*g

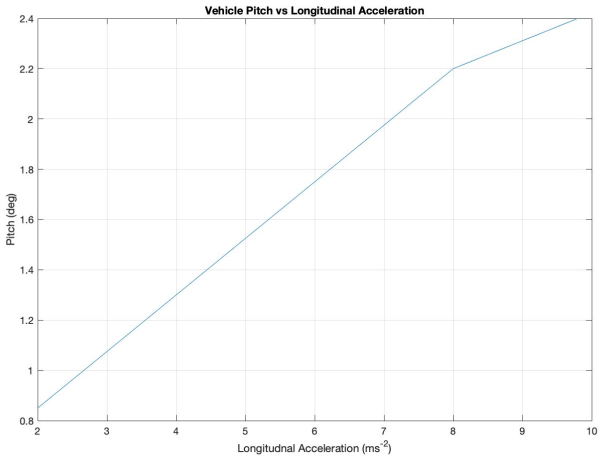
$$\frac{a_y}{g} = \frac{t}{2h}\frac{1}{[1 + R_\phi(1 - h_r/h)]}$$

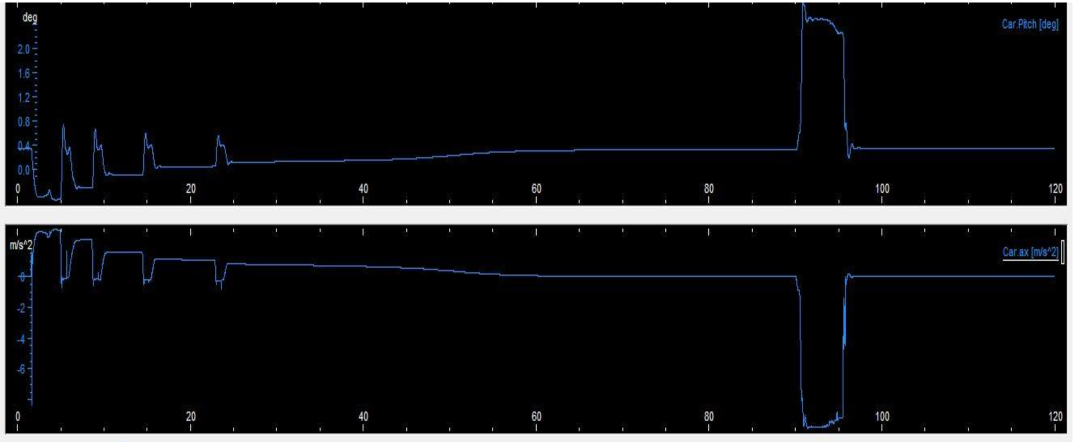


Carmaker simulations

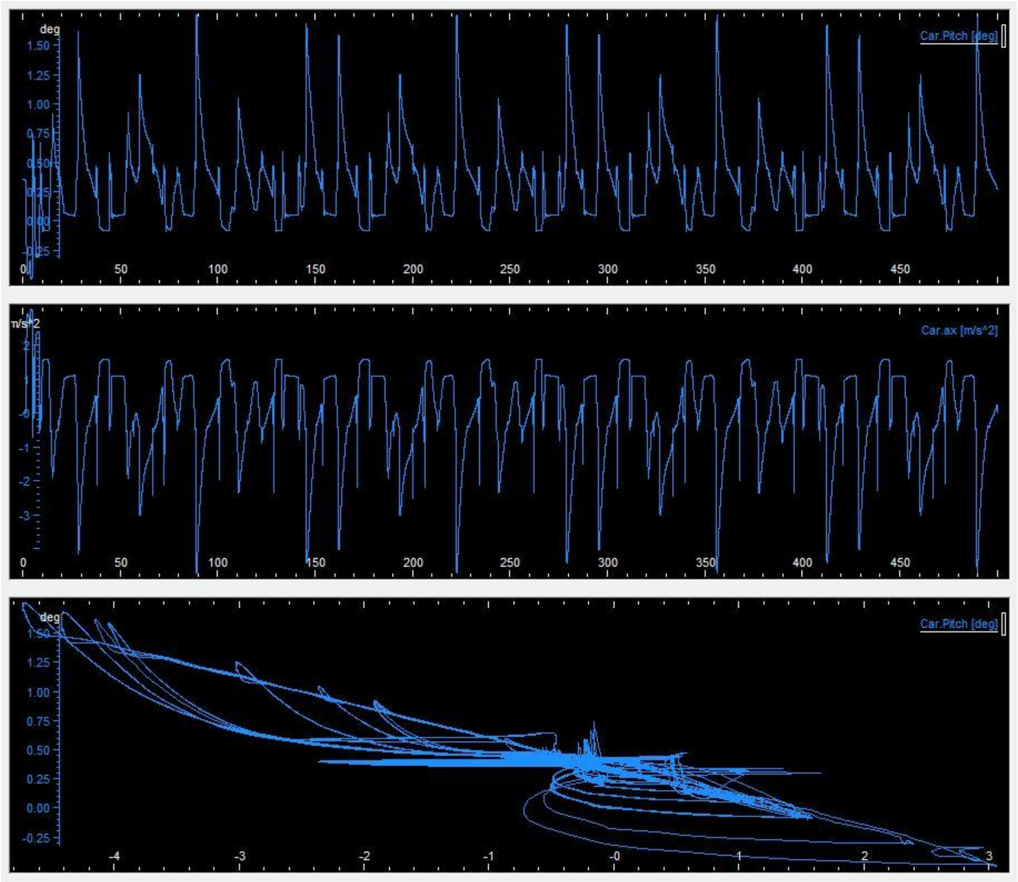
- Pitch vs Longitudinal Acceleration – under steady deceleration
- Pitch vs Longitudinal Acceleration – under transient deceleration
- Roll vs Lateral Acceleration – under steady cornering
- Roll vs Lateral Acceleration – under transient cornering
- Lane change maneuver

Pitch vs Longitudinal acceleration

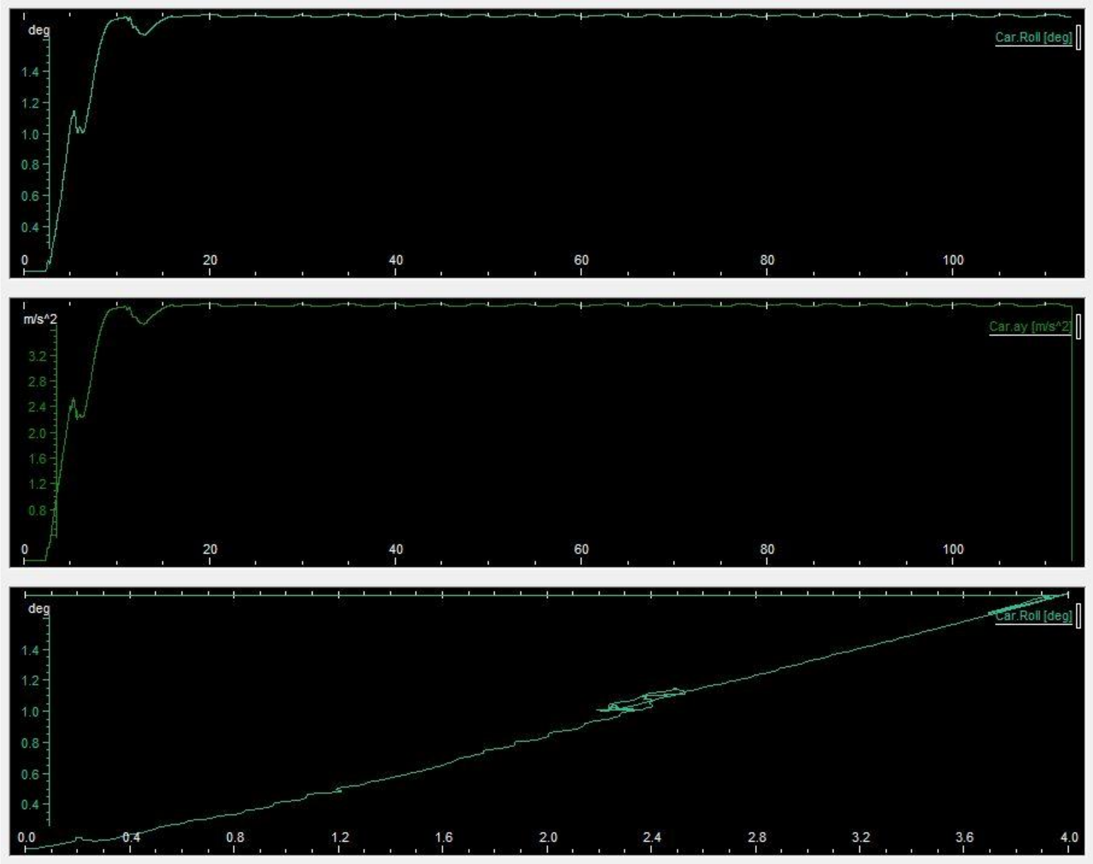




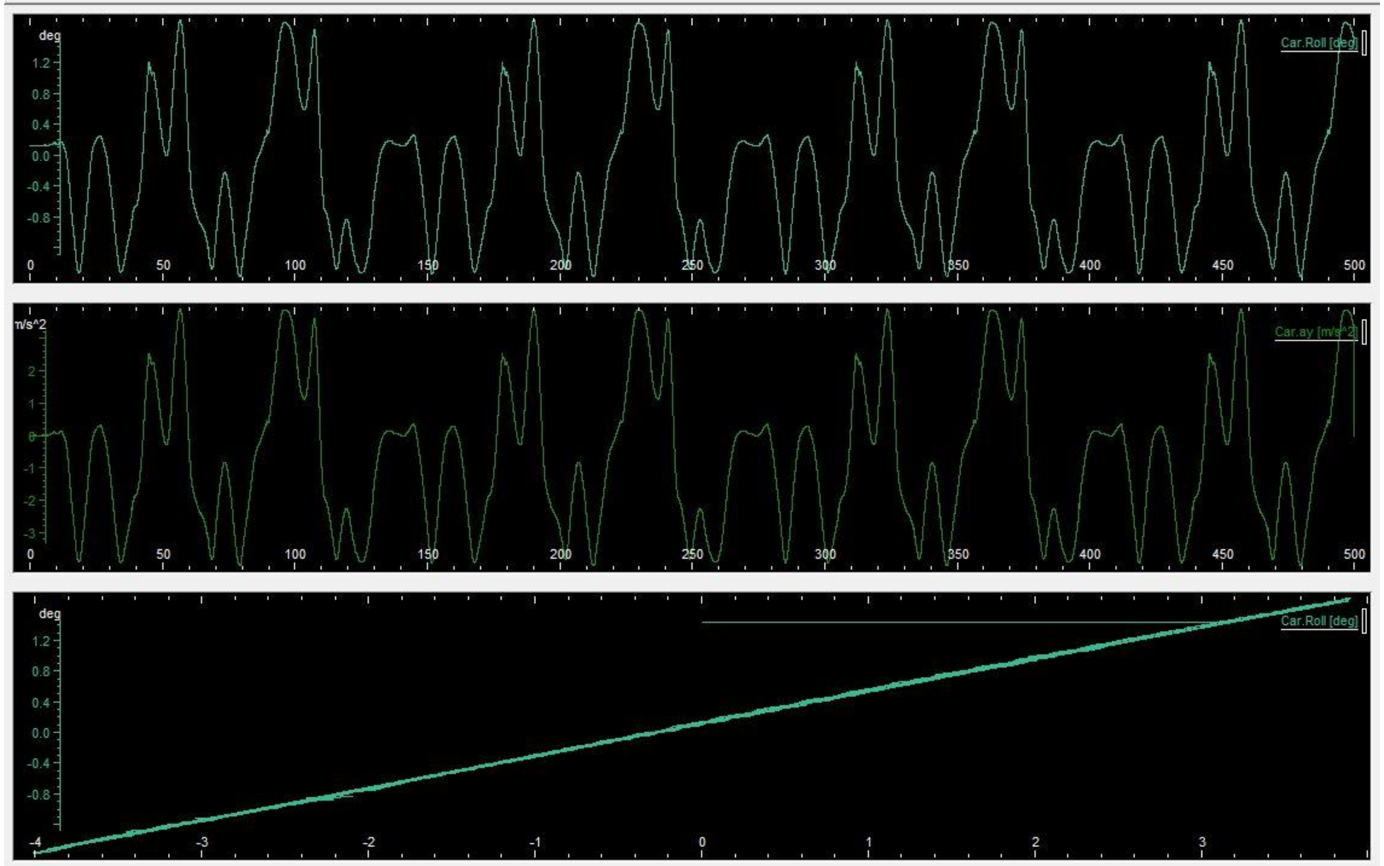
Transient analysis: Pitch vs Longitudinal acceleration



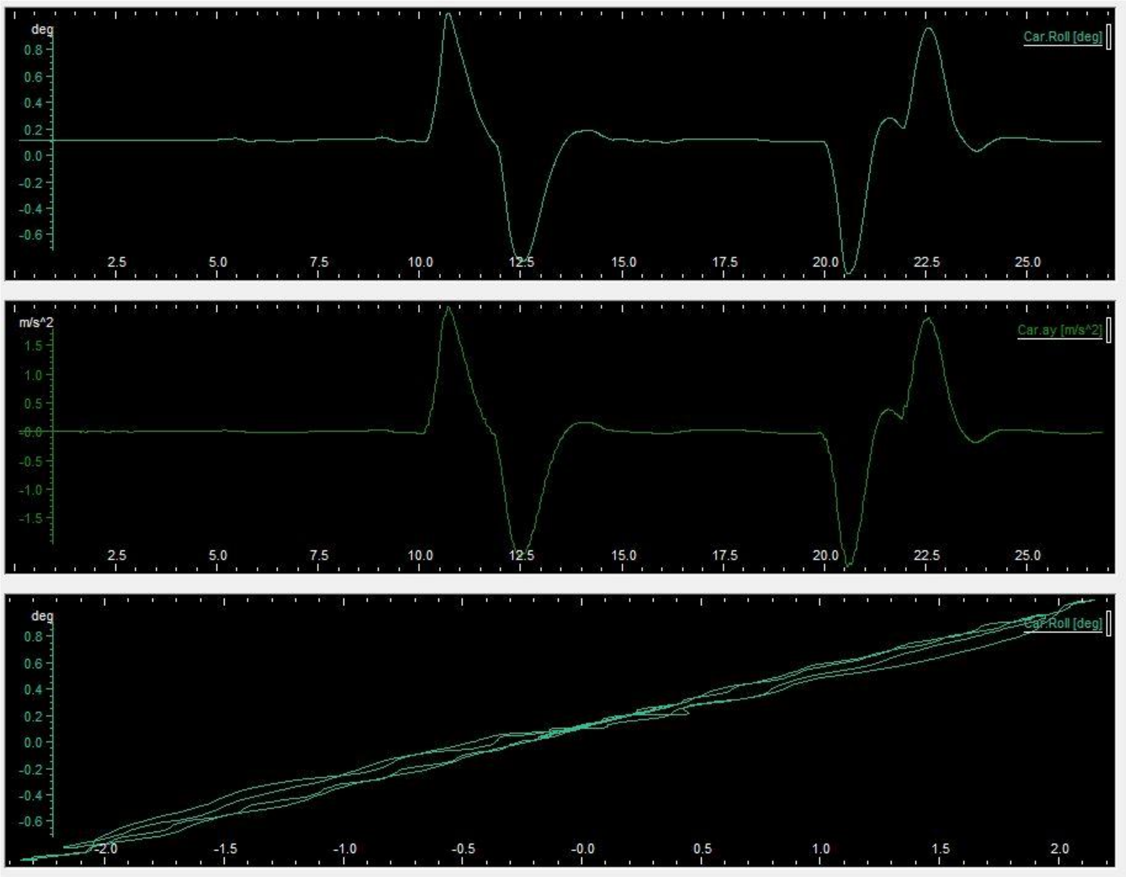
Roll vs Lateral acceleration in a circular track



Transient analysis: Roll vs Lateral acceln.



Lane change maneuver



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