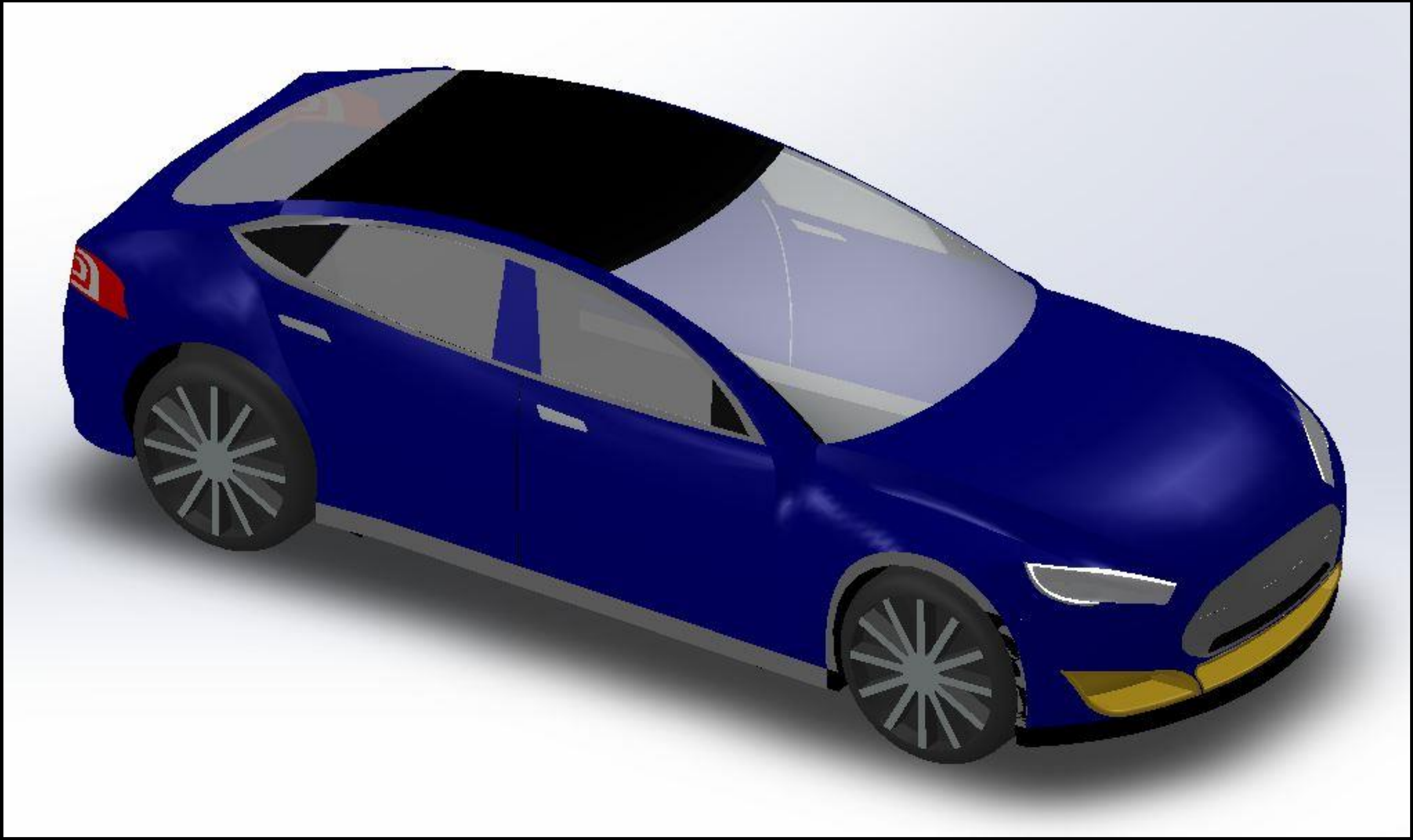
Combined bounce, pitch, and roll dynamics of vehicles negotiating **single** speed bump events

VEHICLE DYNAMICS PROJECT

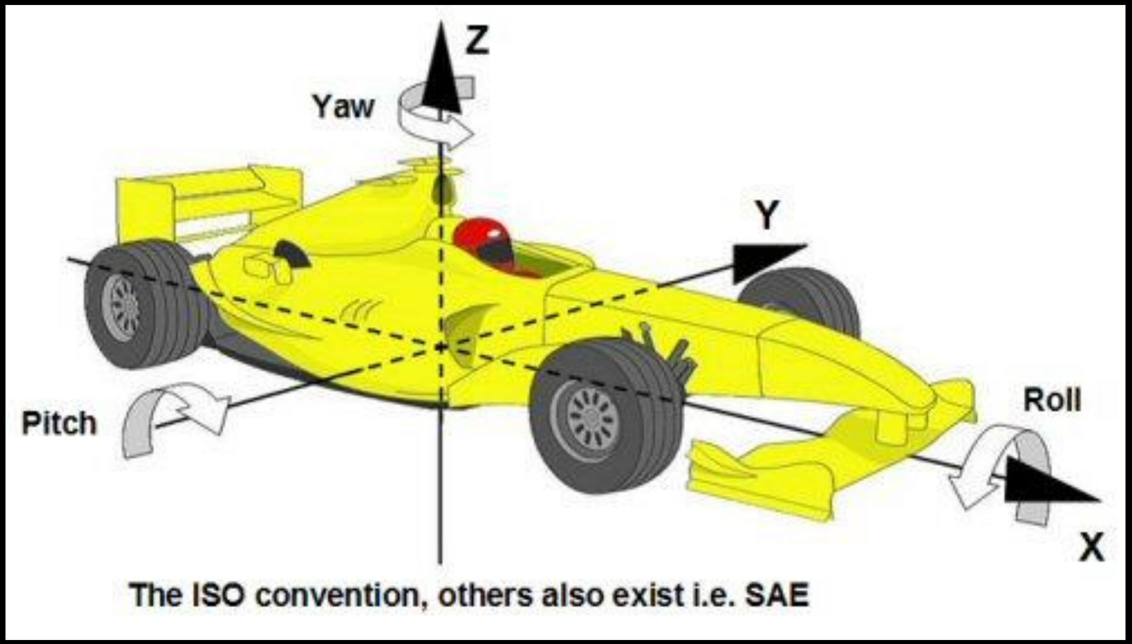
ME17MTECH11001
ME17MTECH11006
ME17MTECH11008
ME17MTECH11010

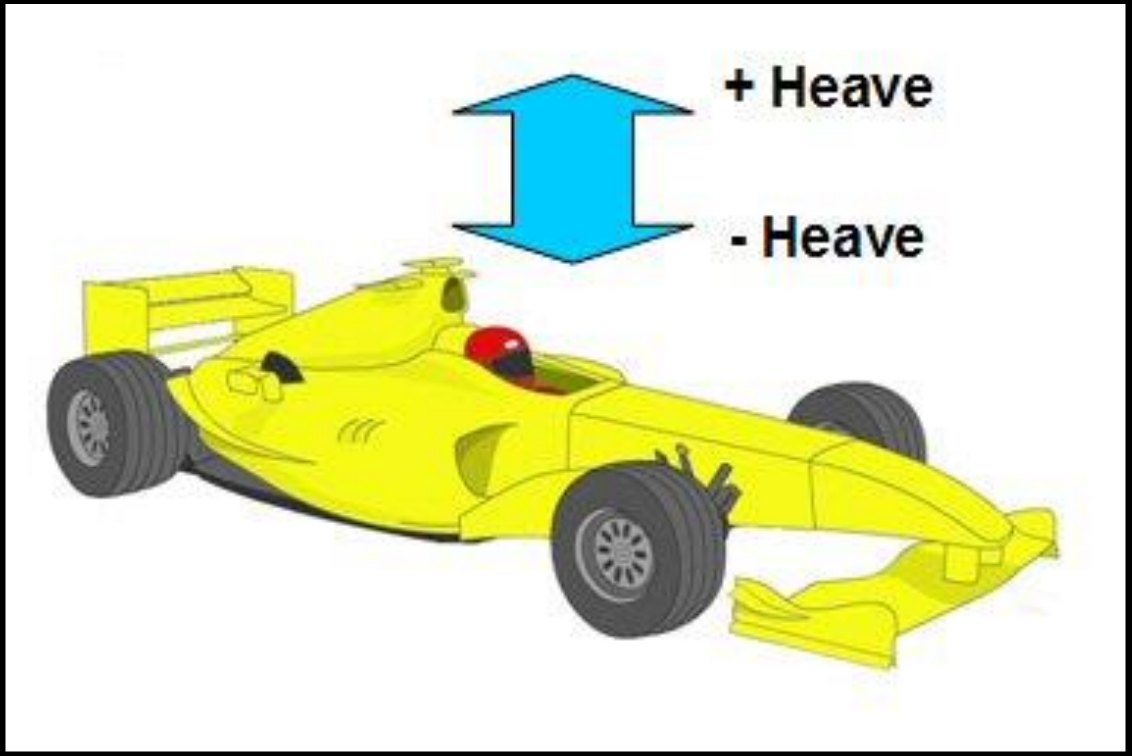
CAR TESLA MODEL S



Bounce, pitch, and roll

- Bounce or Jounce or Heave is equal vertical motion of the front and rear of the car together.
- Pitch (a rotation about the Y axis)
- Roll (a rotation about the X axis)





Pic Credits: http://white-smoke.wikifoundry.com/page/Heave%2C+Pitch%2C+Roll%2C+Warp+and+Yaw

Dynamics of vehicles negotiating single speed bump events

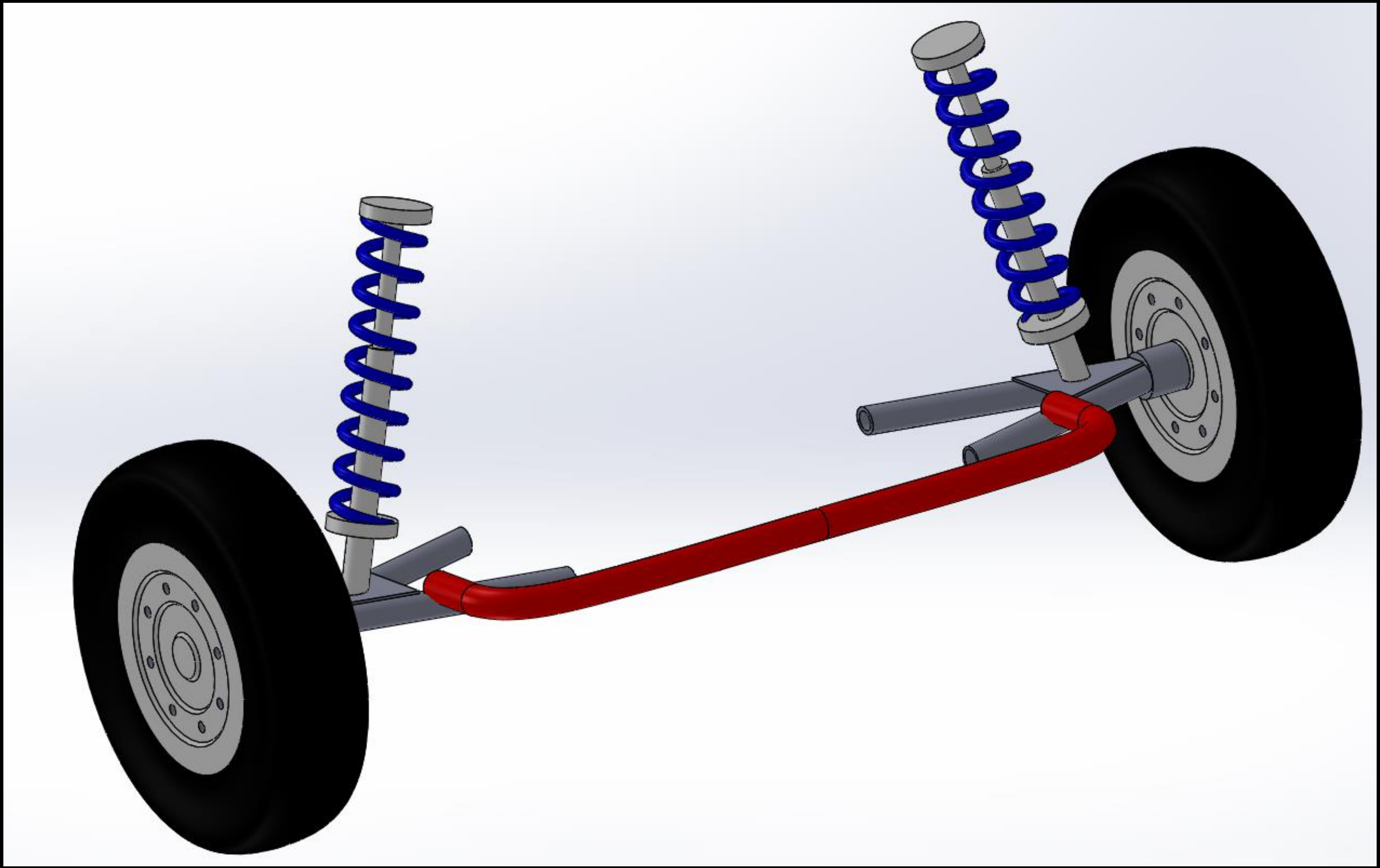
One of the common traffic calming feature

Pic Credits: https://www.theworkplacedepot.co.uk/speed-bump-kit

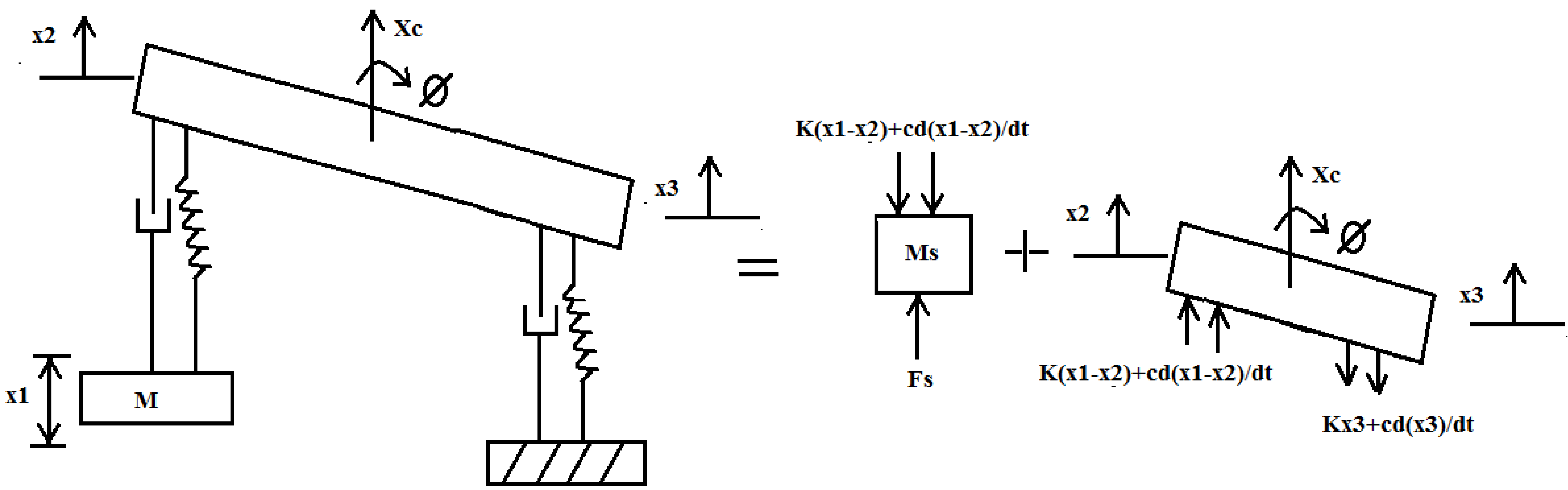
ARB(Anti-Roll Bar)

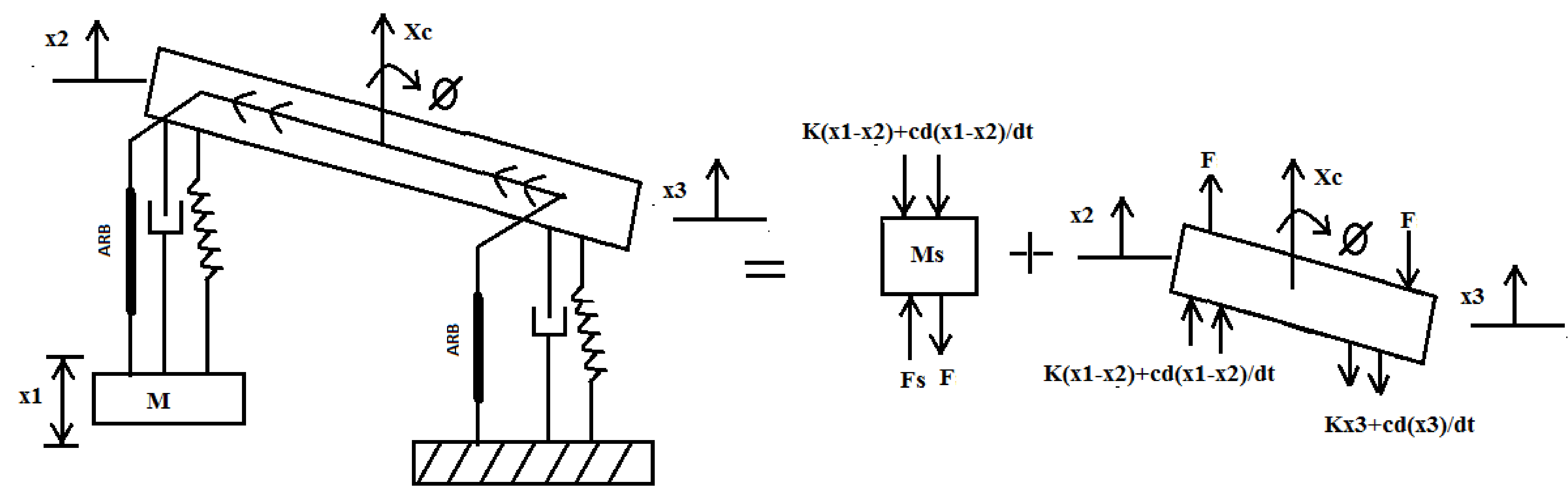
- Stabiliser bar
- Helps reduce the body roll of a vehicle during fast cornering or over road irregularities.
- It connects opposite (left/right) wheels together through short lever arms linked by a torsion spring.
- Increases the suspension's roll stiffness—its resistance to roll in turns, independent of its spring rate in the vertical direction.

SOLID WORKS MODEL



Mathematical model


x2
Xc
Ø
x3
x1
M
=
K(x1-x2)+cd(x1-x2)/dt
Ms
Fs
+
x2
Xc
Ø
x3
K(x1-x2)+cd(x1-x2)/dt
Kx3+cd(x3)/dt


x2
Xc
Ø
x3
ARB
ARB
x1
M
=
K(x1-x2)+cd(x1-x2)/dt
Ms
Fs
F
+
F
F
x2
Xc
Ø
x3
K(x1-x2)+cd(x1-x2)/dt
Kx3+cd(x3)/dt

Virtual Work

To determine the suspension force Virtual work technique is used.

$$F_x\delta x + F_y\delta y + F_z\delta z + F_s(-\delta z) + T_d\,.\,\delta\nu = 0$$

1. F_s is the net suspension force based on the vertical wheel travel.
2. F_x, F_y, F_z are longitudinal lateral and vertical force on the tyre contact patch.
3. T_d is the drive torque (assumed to be generated from an inboard differential)
4. $\delta\gamma$ is the change in the caster angle.
5. c_l is caster length.

Using variational mechanics

$$\delta x = \frac{dx}{dz}\delta z \quad \text{and} \quad \delta y = \frac{dy}{dz}\delta z$$

From above virtual equation and with fraction of traction $'f'$ one can easily arrive at following equation.

$$F_s = f \times \frac{T_d}{R_l} \times \frac{L}{\pi H} \times sec(\pi \times \frac{X}{L}) + m_{cf} \times g - T_d \times (sin(\gamma))^2/cl;$$

Equation of motions without Roll Bar

$$M_s\ddot{x}_1 = F_s - K(x_1 - x_2) - C(\dot{x}_1 - \dot{x}_2)$$

$$M_{us}\ddot{x}_c = K(x_1 - x_2) + C(\dot{x}_1 - \dot{x}_2) - Kx_3 - C\dot{x}_3$$

$$I_{us}\ddot{\Phi} = (K(x_1 - x_2) + C(\dot{x}_1 - \dot{x}_2) + Kx_3 + C\dot{x}_3)\frac{B}{2}$$

$$M_{us}\ddot{x}_c = K\left(x_1 - x_c - \Phi\frac{B}{2}\right) + C\left(\dot{x}_1 - \dot{x}_c - \dot{\Phi}\frac{B}{2}\right) - K(x_c - \Phi\frac{B}{2}) - C(\dot{x}_c - \dot{\Phi}\frac{B}{2})$$

$$x_2 = x_c + \Phi\frac{B}{2}$$

$$x_3 = x_c - \Phi\frac{B}{2}$$

$$\dot{x}_2 = \dot{x}_c + \dot{\Phi}\frac{B}{2}$$

$$\dot{x}_3 = \dot{x}_c - \dot{\Phi}\frac{B}{2}$$

Equation of motions with Roll Bar

$$M_s\ddot{x}_1 = F_s - K(x_1 - x_2) - C(\dot{x}_1 - \dot{x}_2)$$

$$M_{us}\ddot{x}_c = K(x_1 - x_2) + C(\dot{x}_1 - \dot{x}_2) - Kx_3 - C\dot{x}_3$$

$$I_{us}\ddot{\Phi} = (K(x_1 - x_2) + C(\dot{x}_1 - \dot{x}_2) + Kx_3 + C\dot{x}_3)\frac{B}{2}$$

$$M_{us}\ddot{x}_c = K\left(x_1 - x_c - \Phi\frac{B}{2}\right) + C\left(\dot{x}_1 - \dot{x}_c - \dot{\Phi}\frac{B}{2}\right) - K(x_c - \Phi\frac{B}{2}) - C(\dot{x}_c - \dot{\Phi}\frac{B}{2})$$

$$x_2 = x_c + \Phi\frac{B}{2}$$

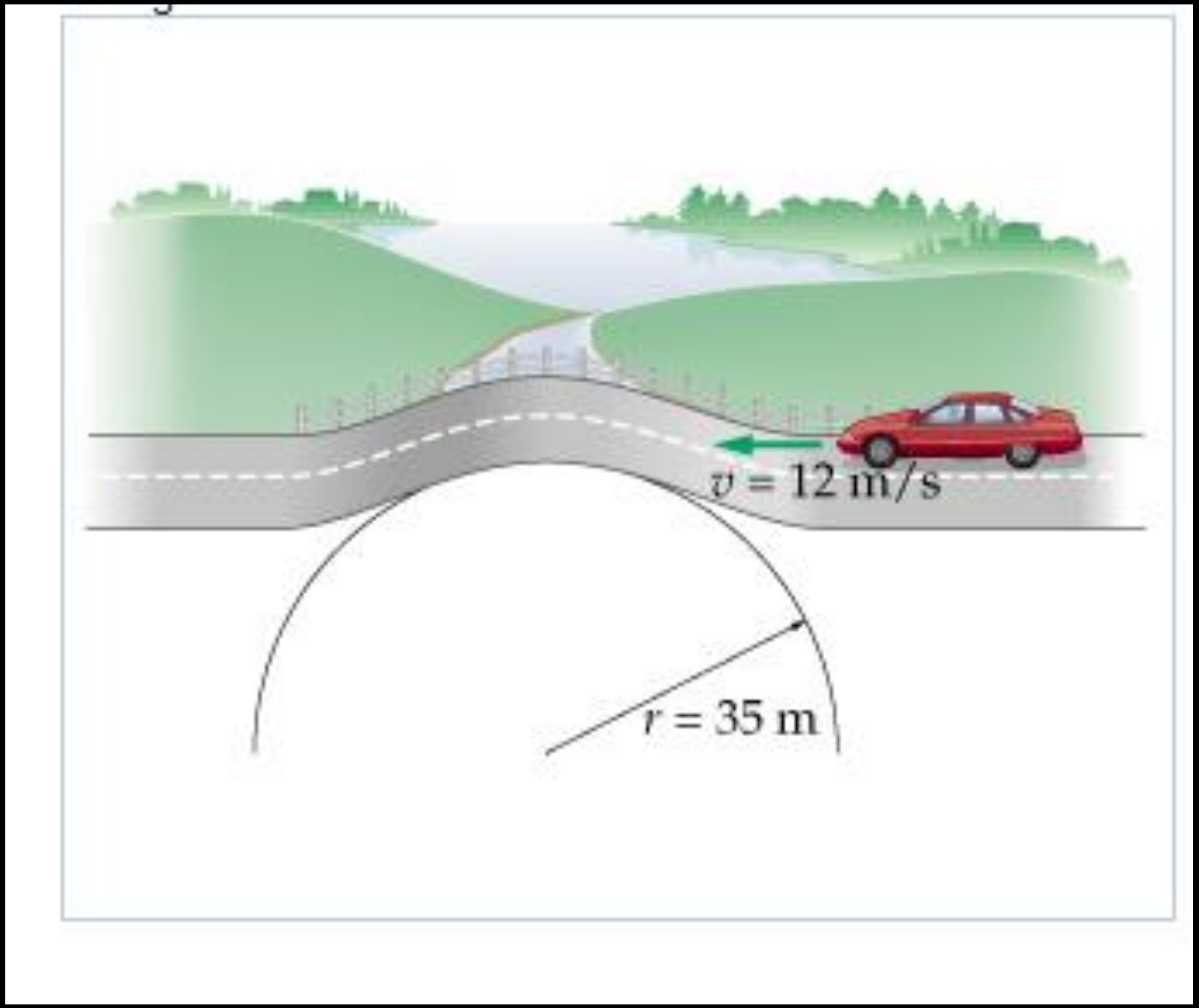
$$x_3 = x_c - \Phi\frac{B}{2}$$

$$\dot{x}_2 = \dot{x}_c + \dot{\Phi}\frac{B}{2}$$

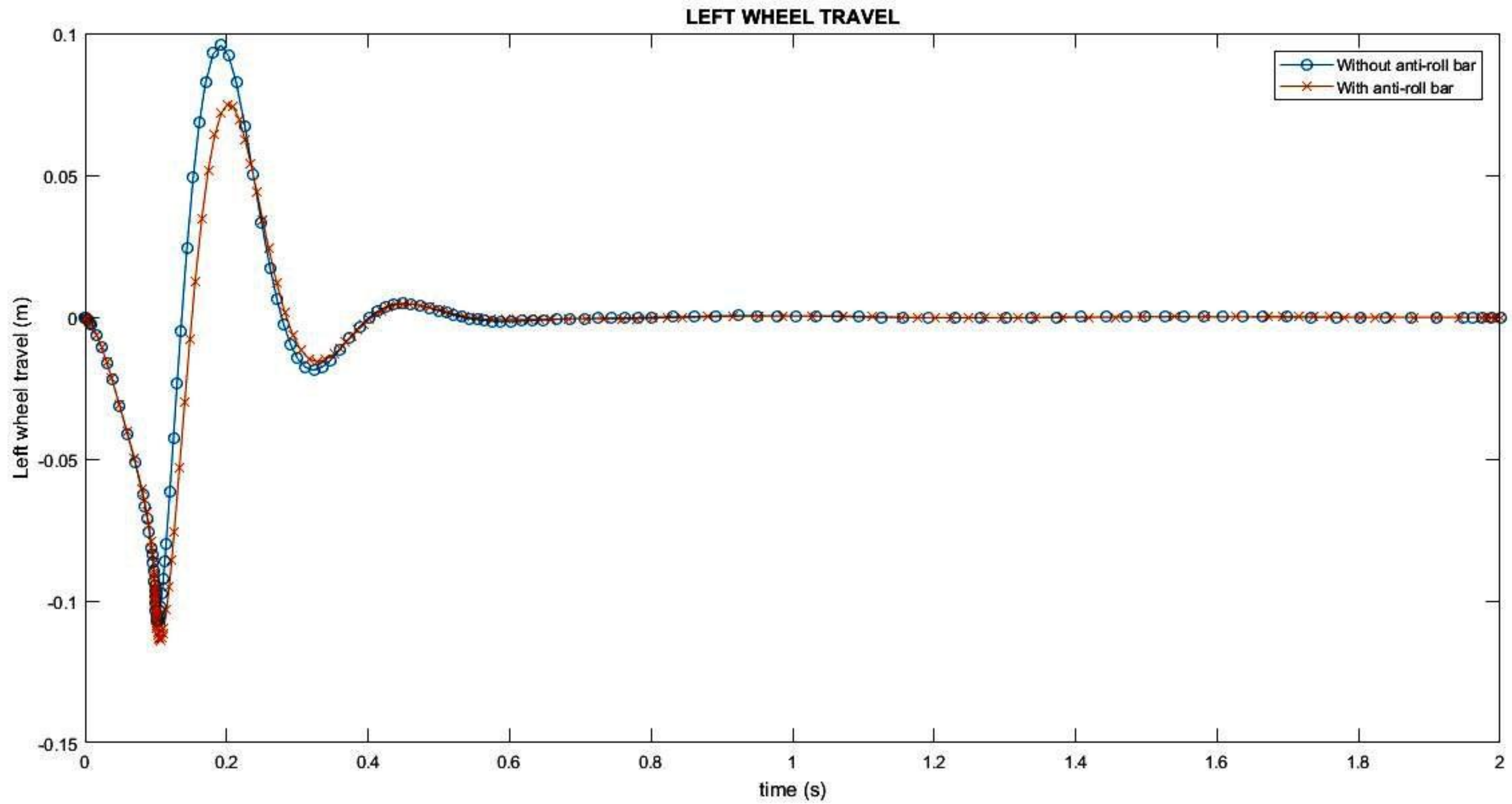
$$\dot{x}_3 = \dot{x}_c - \dot{\Phi}\frac{B}{2}$$

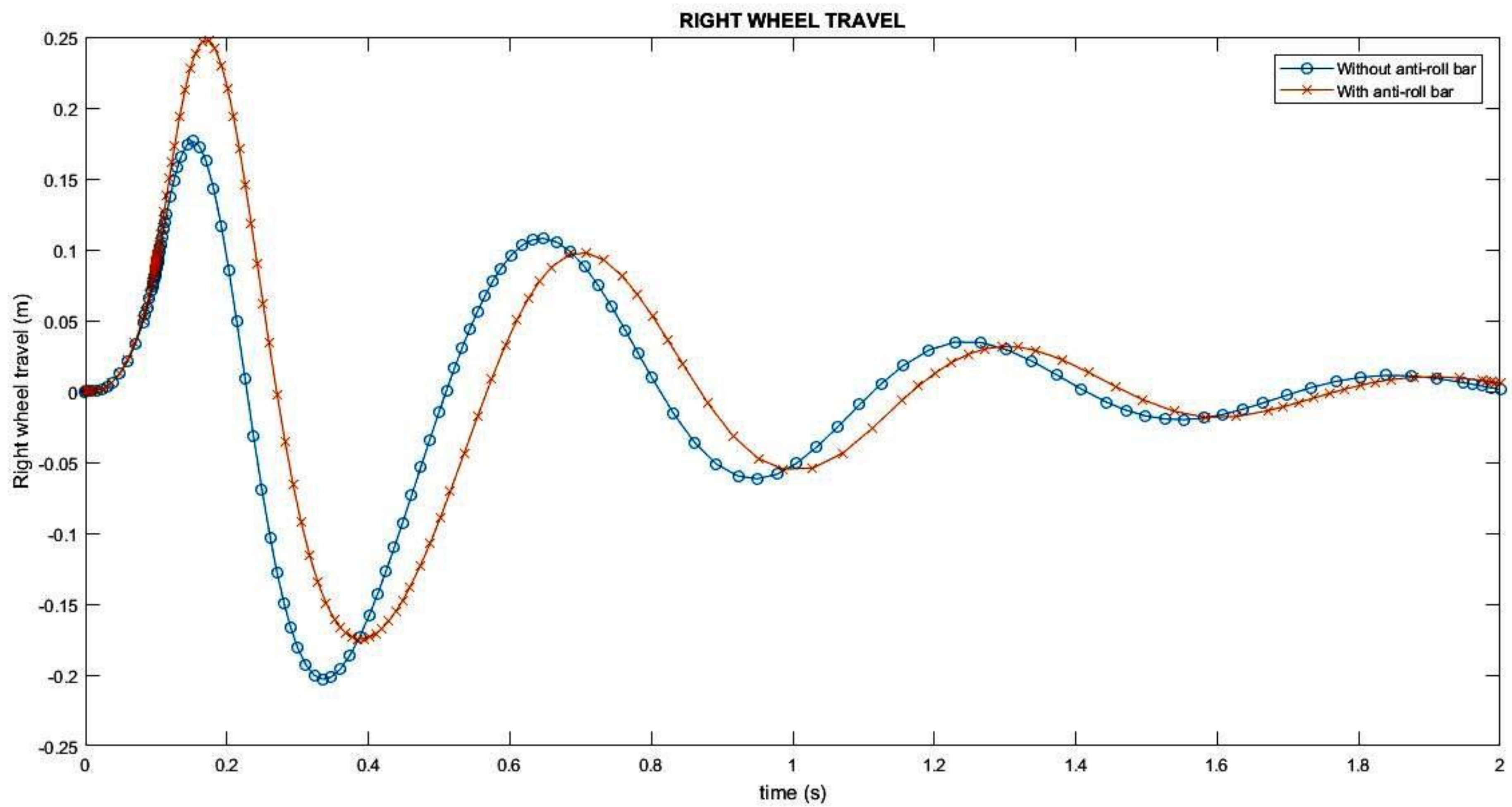
Bump

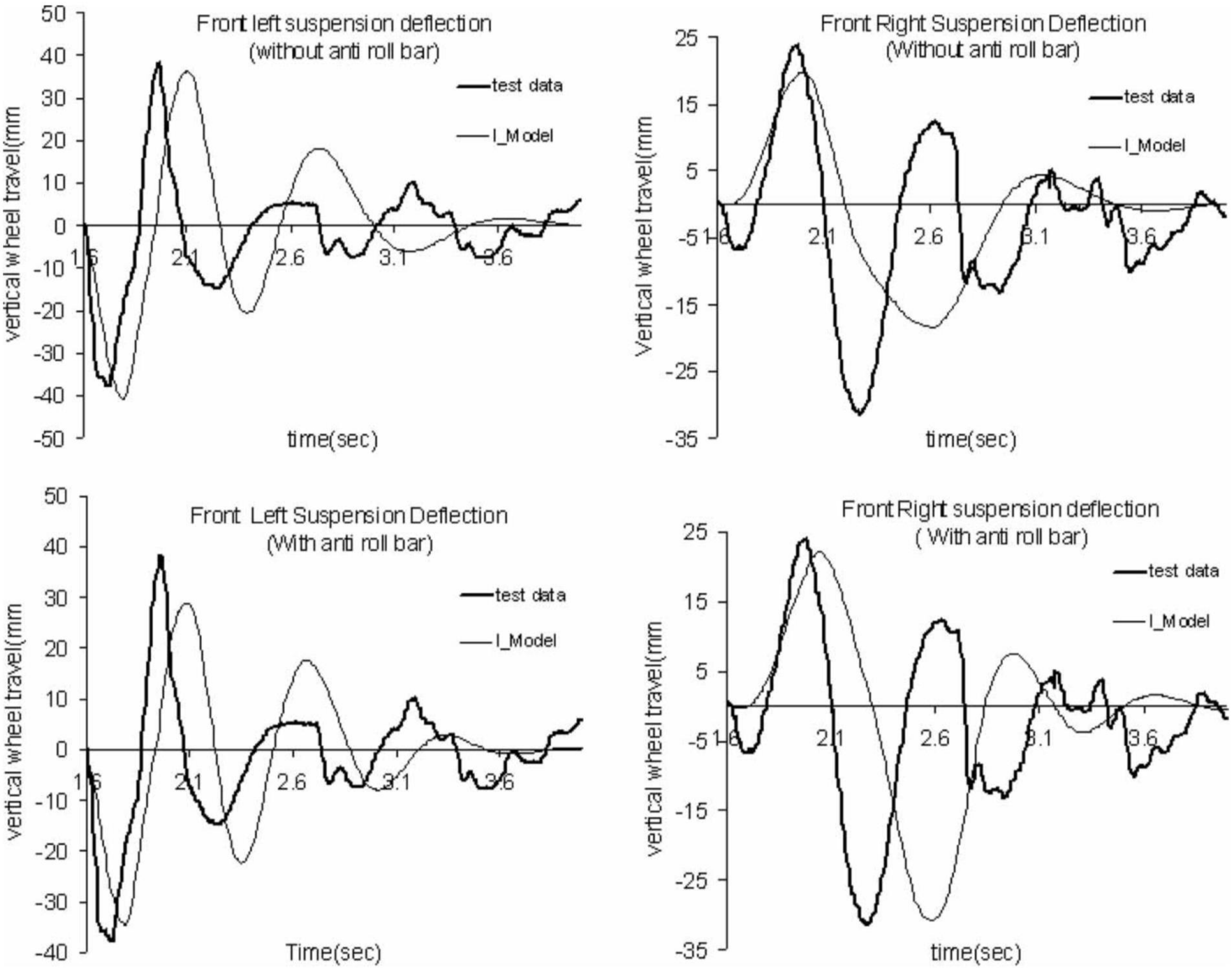
- We took sinusoidal profile split speed bump
- Z = H*sin(pi*x/L)
- L = 1 m
- H = 10 cm



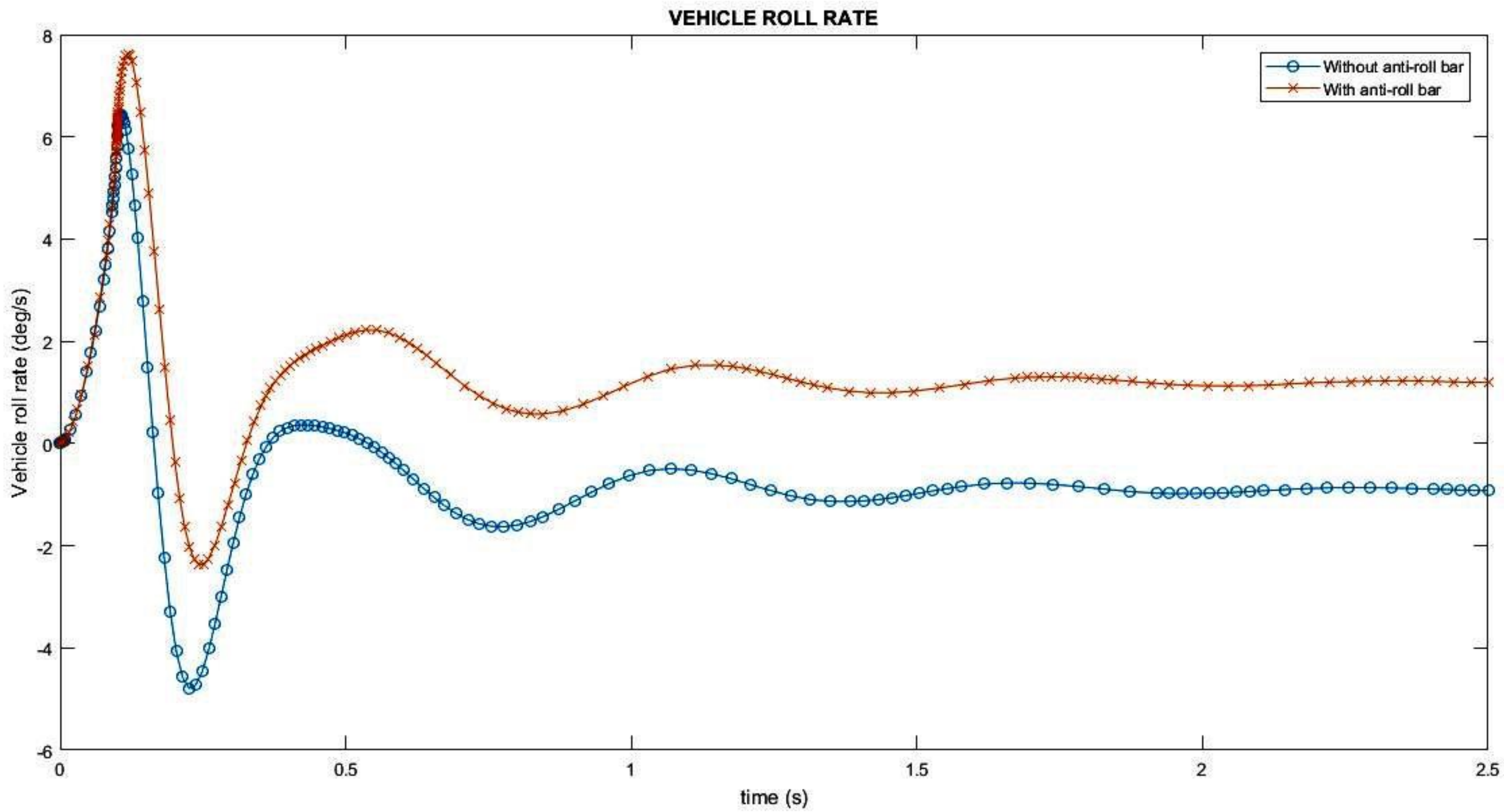
Matlab Results

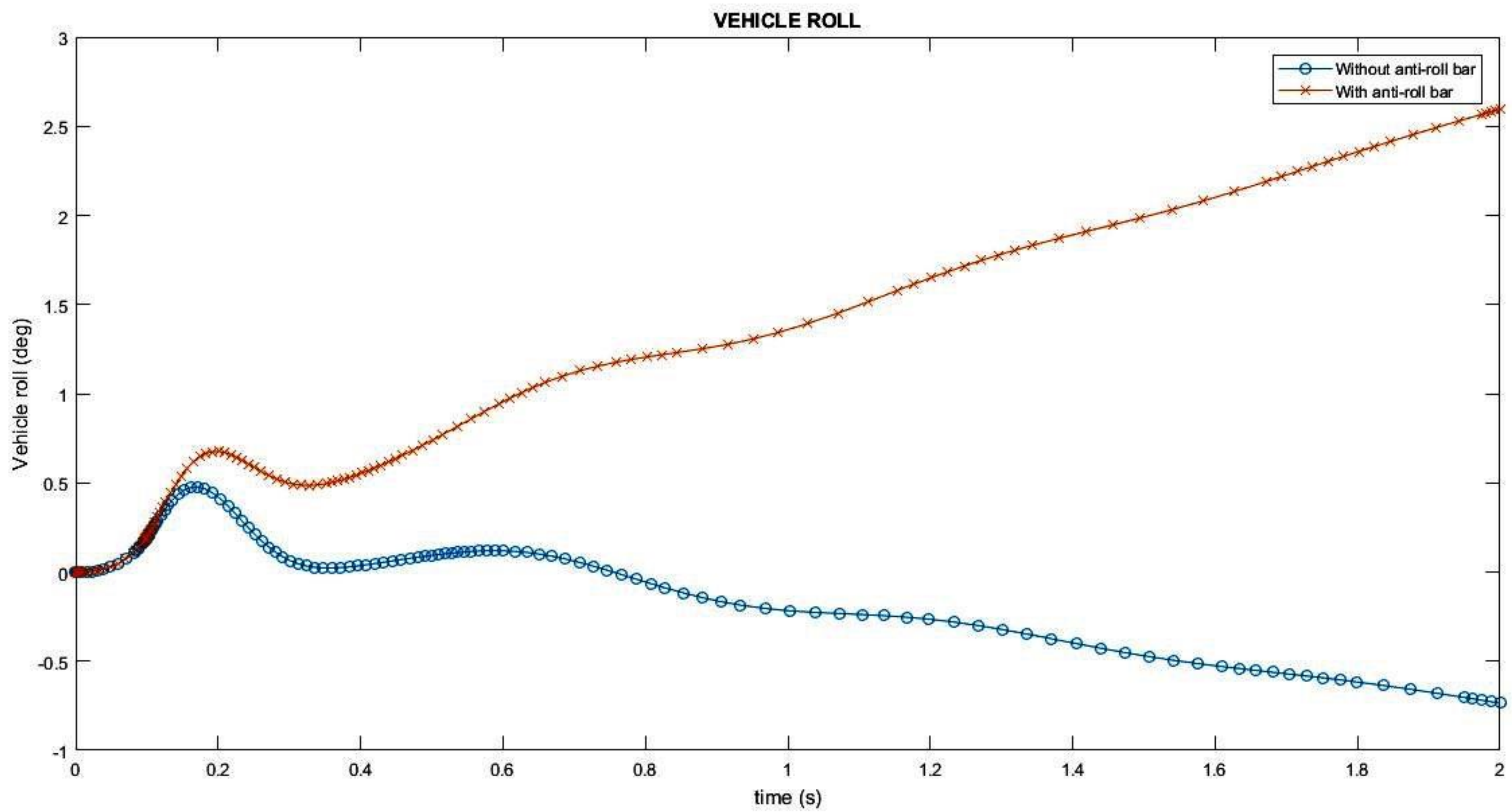

LEFT WHEEL TRAVEL
Without anti-roll bar
With anti-roll bar
Left wheel travel (m)
time (s)
0.1
0.05
0
-0.05
-0.1
-0.15
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2

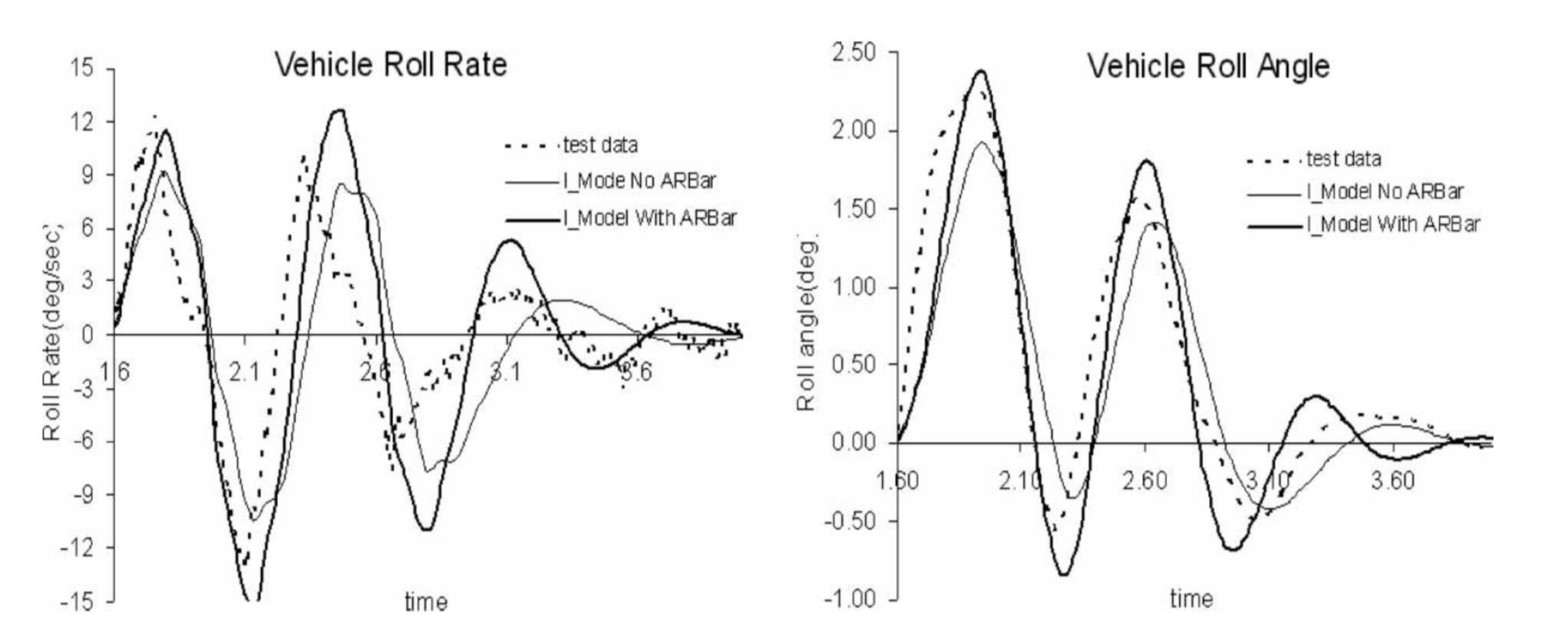

RIGHT WHEEL TRAVEL
Without anti-roll bar
With anti-roll bar
Right wheel travel (m)
time (s)
0.25
0.2
0.15
0.1
0.05
0
-0.05
-0.1
-0.15
-0.2
-0.25
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2



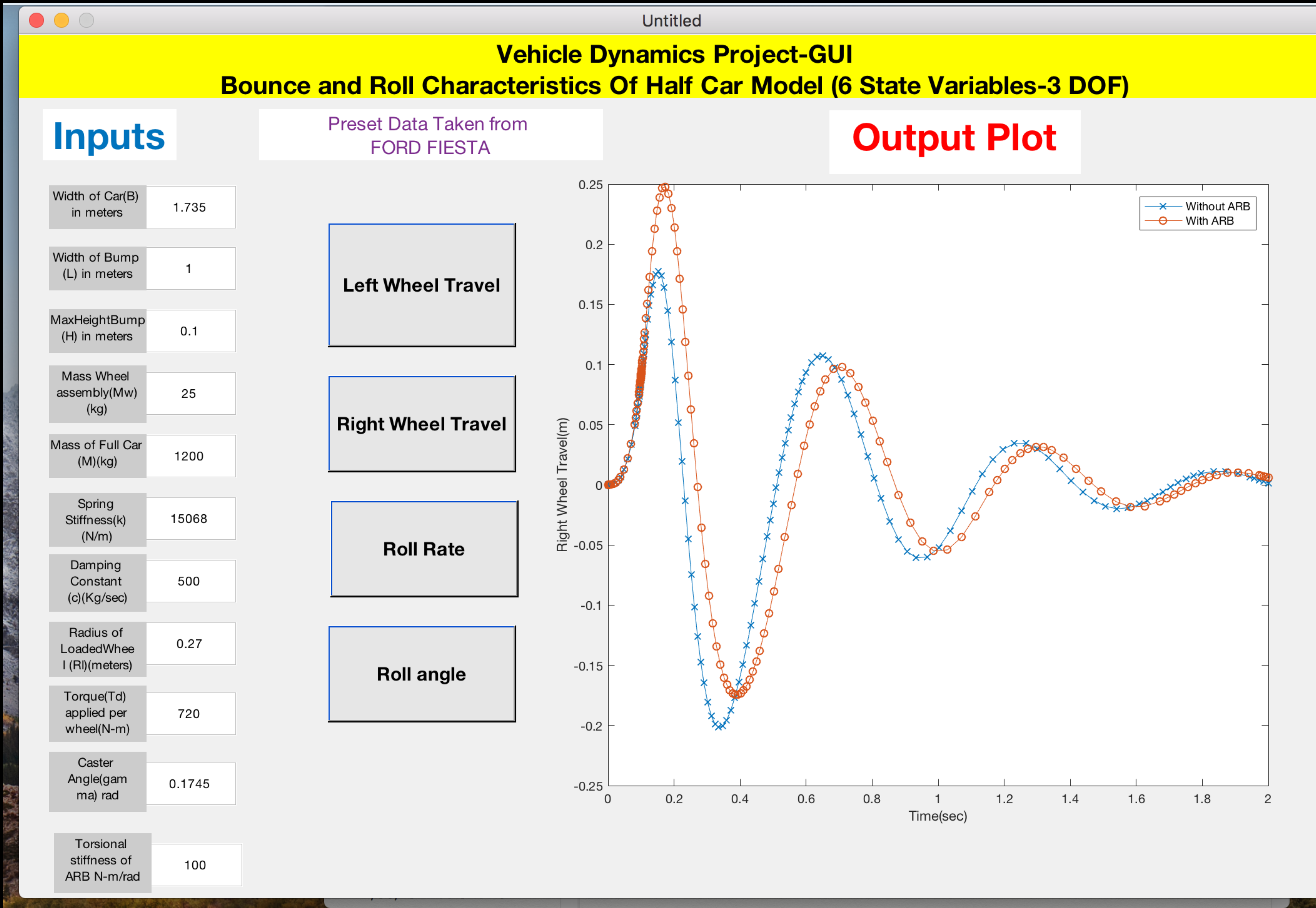
Front suspension deflection from research paper


VEHICLE ROLL RATE
Without anti-roll bar
With anti-roll bar
Vehicle roll rate (deg/s)
time (s)
8
6
4
2
0
-2
-4
-6
0
0.5
1
1.5
2
2.5


VEHICLE ROLL
Without anti-roll bar
With anti-roll bar
Vehicle roll (deg)
time (s)
3
2.5
2
1.5
1
0.5
0
-0.5
-1
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2



Roll rate and Roll angle from research paper


Untitled
Vehicle Dynamics Project-GUI
Bounce and Roll Characteristics Of Half Car Model (6 State Variables-3 DOF)
Inputs
Preset Data Taken from FORD FIESTA
Output Plot
Width of Car(B) in meters
1.735
Width of Bump (L) in meters
1
MaxHeightBump (H) in meters
0.1
Mass Wheel assembly(Mw) (kg)
25
Mass of Full Car (M)(kg)
1200
Spring Stiffness(k) (N/m)
15068
Damping Constant (c)(Kg/sec)
500
Radius of LoadedWheel (Rl)(meters)
0.27
Torque(Td) applied per wheel(N-m)
720
Caster Angle(gamma) rad
0.1745
Torsional stiffness of ARB N-m/rad
100
Left Wheel Travel
Right Wheel Travel
Roll Rate
Roll angle
Without ARB
With ARB
Right Wheel Travel(m)
Time(sec)

Conclusion

- An anti-roll bar plays major role in increasing overall vehicle roll stiffness while negotiating high speed cornering and off-terrain events.
- It is observed here that for a single speed bump analysis, when the vehicle is fitted with an anti-roll bar, it demonstrates a larger body roll rate during transient conditions, compared to the vehicle model without an anti-roll bar negotiating the same speed bump which is undesirable from rider comfort point of view.
- Normally manoeuvres can strictly be considered as combined ride and handling event, so a compromise may need to be found for development of anti-roll bar systems, which improve roll stability, while maintaining a good level of ride comfort.

Conclusion cont

- The second conclusion that results from this study is the affirmation of the use of simple, but sufficiently detailed, intermediate models for the study of seemingly complex ride and handling manoeuvres. Traditionally, such studies have required much more complex multi-body models, but the concordance of analytical predictions with experimental findings here point to a much less arduous approach, at least in the first instance.
- From this study we can point out that simple, but sufficiently detailed models are in good agreement with seemingly complex ride and handling manoeuvres. Here the use of 3 DOF vehicle model is giving very good approximation towards the difficult 6 DOF paper used in the paper.

Thank you