PODIUM

ADVANCED TECHNOLOGIES

Anti-Features Study

1. FREE BODY DIAGRAM

1. FREE BODY DIAGRAM

1.1 3-D Suspension Movement

The classic roll center height approach to describe how the chassis rolls in cornering have many limitations. The bigger one is that equal Fy is implicitly assumed [Gerrard].

To overcome this limitation different approaches have been proposed, such as [Mitchell]'s Force Based Roll Center.

All those approaches have in common the hypothesis of a fixed ground with a moving sprung mass.

A tentative to easier imagine how the suspension moves in the 3D space has been proposed by [Lamers]. He defines an instant axis of rotation of the un-sprung mass, from which the front and side Instant Centers of rotation are identified in the intersection with front and side planes.

Lamers still approaches the problem with a fixed ground, but from his idea the presented approach for the free diagram was born.

Nevertheless, fixing either the body or the ground only affects the orientation of the forces in free body diagram without effects on the conclusions. Imagining the body as fixed may be easier to mentally visualize the independency of the single suspensions.

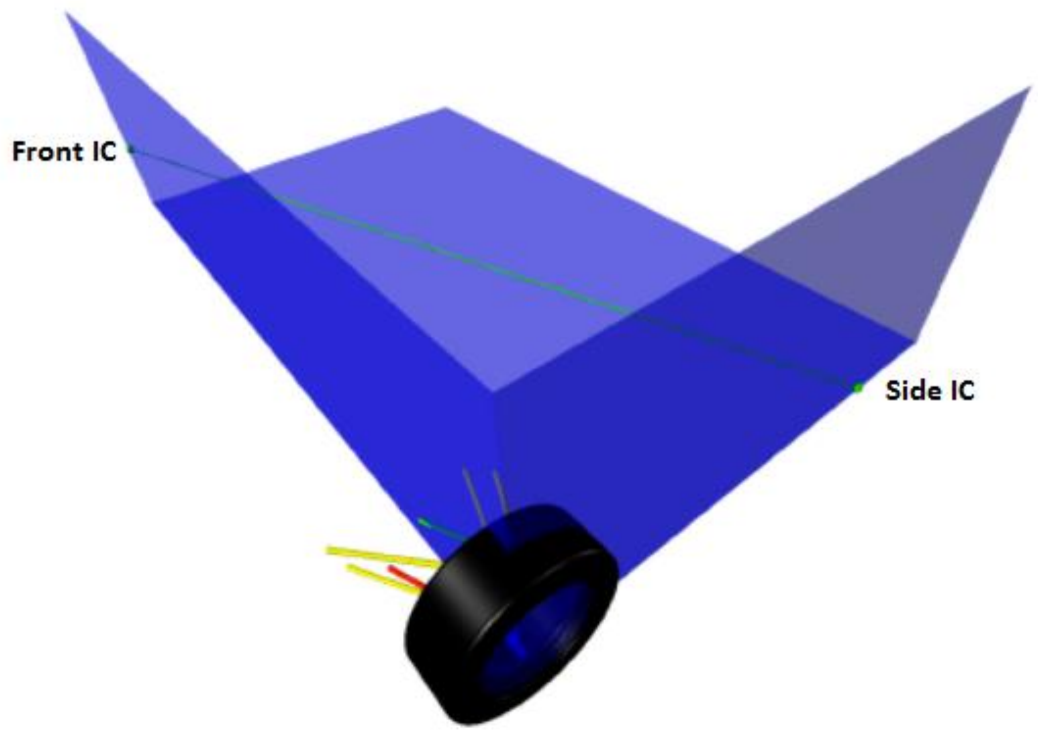


Figure 4.6: Instantaneous screw axis

1. FREE BODY DIAGRAM

1.2 Front View Scheme

In this proposed approach is convenient to imagine the green line representing the chassis fixed in a global reference frame, while the two wheels are free to move independently.

This way the important point is for each of the two suspensions the position of its IC.

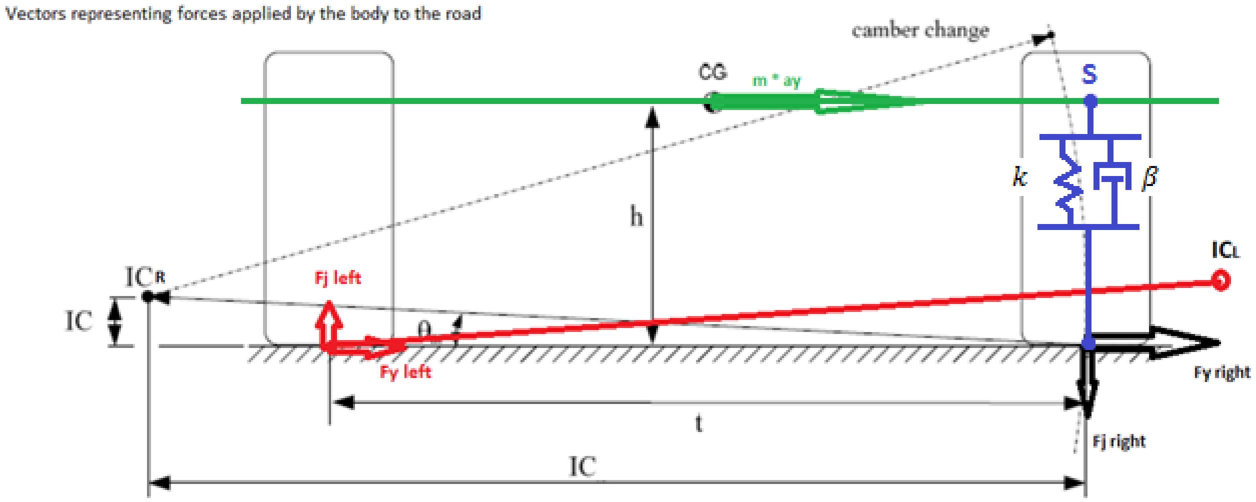
The ideal IC-CP line represent the direction of the resultant of the forces applied on the contact patch so that an Fy always implies a jacking-force Fj if the IC is not at ground.

The amount of Jacking Force is then related to Fy thru the tangent of theta:

$$F_j = F_y \cdot \tan(\theta) = F_y \cdot \frac{IC_z}{IC_y}$$

Please note that

$$tan(\theta) = \frac{IC_z}{IC_y} \sim= \frac{RCH}{\frac{T}{2}+RCY}$$



Rear View

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1.3 1-DOF Equilibrium Equation

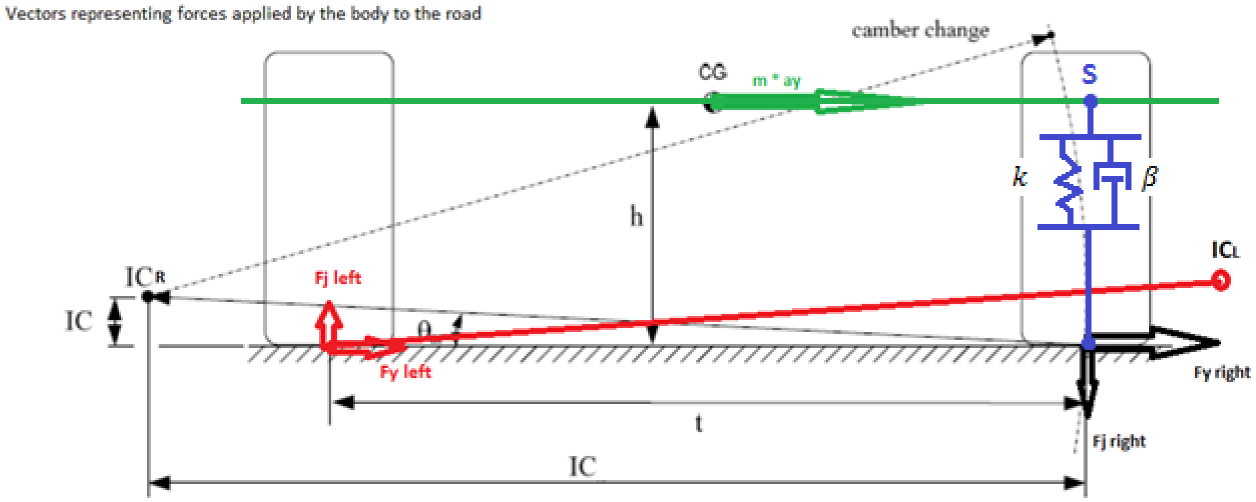
The dynamic equilibrium equation for lateral dynamics, inserting a 1-dof suspension between the contact patch (CP) and the chassis suspension anchoring point (S):

$$m \cdot a_y \cdot \frac{h_{CG}}{T} = k_e \cdot z + \beta \cdot \dot{z} + m \cdot \ddot{z} + F_y \cdot \tan(\theta)$$

This is the forces equation in the point S.

The left term is the load transferred thru the chassis due to lateral acceleration, while the right-term represents all the terms that counteract the suspension jounce.

Where m is m/4 + a fake mass equivalent to the Ixx of the body.



Rear View

Anti-Features Study

2. ANTI-ROLL DEFINITION

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Anti Roll Equation

To solve the system the following assumptions are made:

- Negligible Ixx, infinite torsional and suspension arms stiffness
- Fy = f(Fz) without delay, constant slip angle
- Fy lateral distribution as Fz distribution

Developing the previous equation we can define:

$$F_J = m \cdot a_y \cdot \left(\frac{1}{2} + \frac{h_{CG}}{T} \cdot \frac{a_y}{g}\right) \cdot \tan(\theta)$$

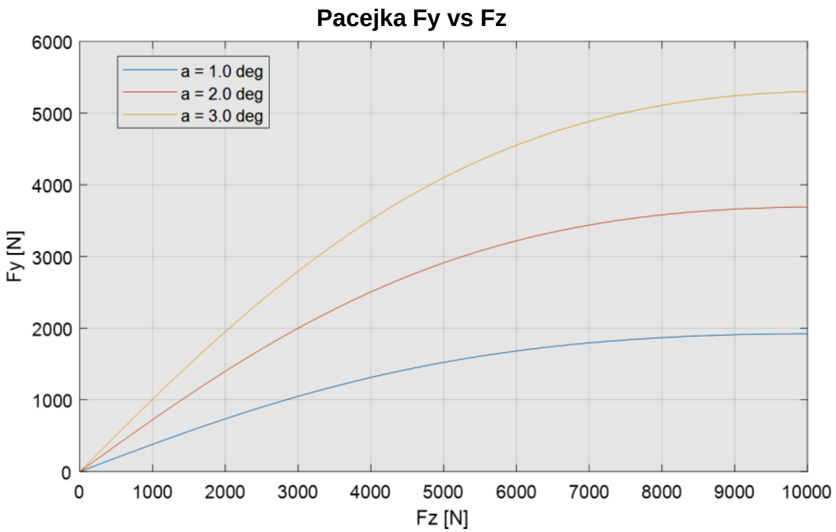
With some algebraic steps we reach the following formulation for the ratio of jacking force over transferred lateral load

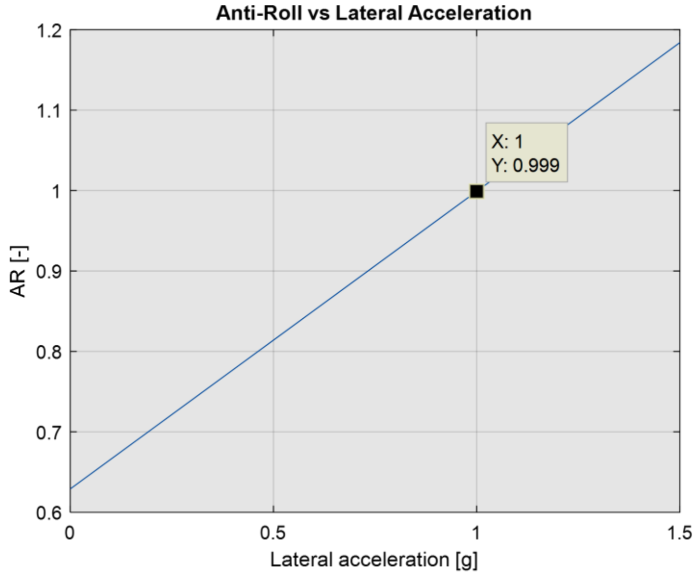
$$Anti - Roll = \frac{F_J}{\Delta F_z} = \left(\frac{T}{2 \cdot h_{CG}} + \frac{a_y}{g}\right) \cdot \frac{IC_z}{IC_y}$$

Where the ratio ay/g is the maximum lateral friction coefficient of the tire (Fy/Fz).

No aero-forces should be used to choose the maximum ay.

Since AR is function of ay a reference ay must be chosen.





2. ANTI-ROLL DEFINITION

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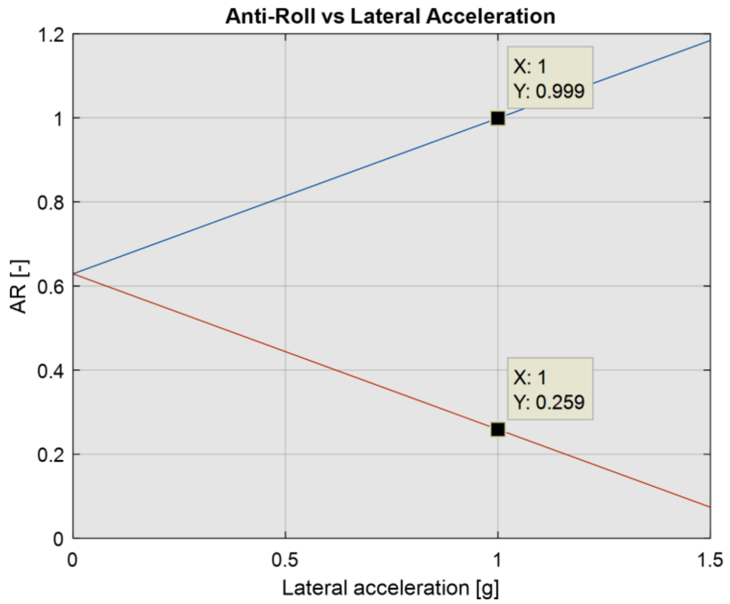
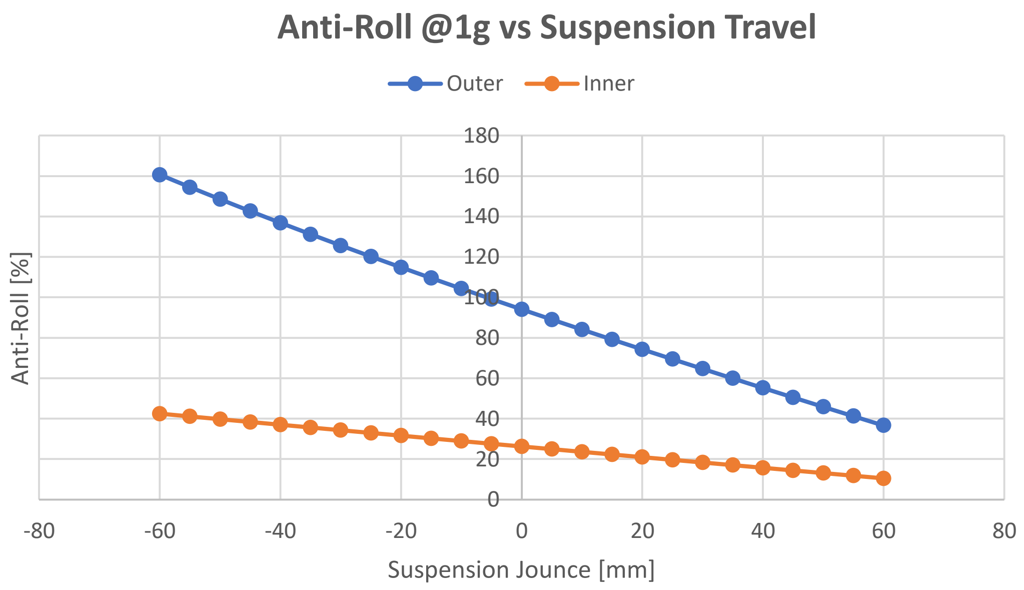
Inner and Outer wheel Anti Roll

Since the sign of Fy is different for the inner and the outer wheel of the turn the slope of AR vs ay is opposite.

This seems obvious thinking that if on the inner wheel the vertical load is zero (extreme cornering or very high CG), Fy on the inner wheel will be zero and no anti-jacking force can be produced by the suspension.

The situation is a bit better than may seem as the jacking force gain (Fj/Fy) can be obtained decreasing in bump and increasing in rebound (Roll Center migrates towards inner side of the turn).





Anti-Features Study

3. 2-DOF AND VI-CRT MODELS

3. 2-DOF AND VI-CRT MODELS

3.1 2-DOF Simulink Model

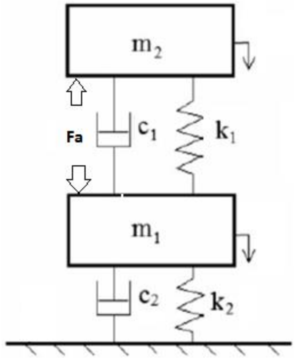
To verify the dynamic response varying the AR%, both Simulink and VI-CarRealTime model have been used.

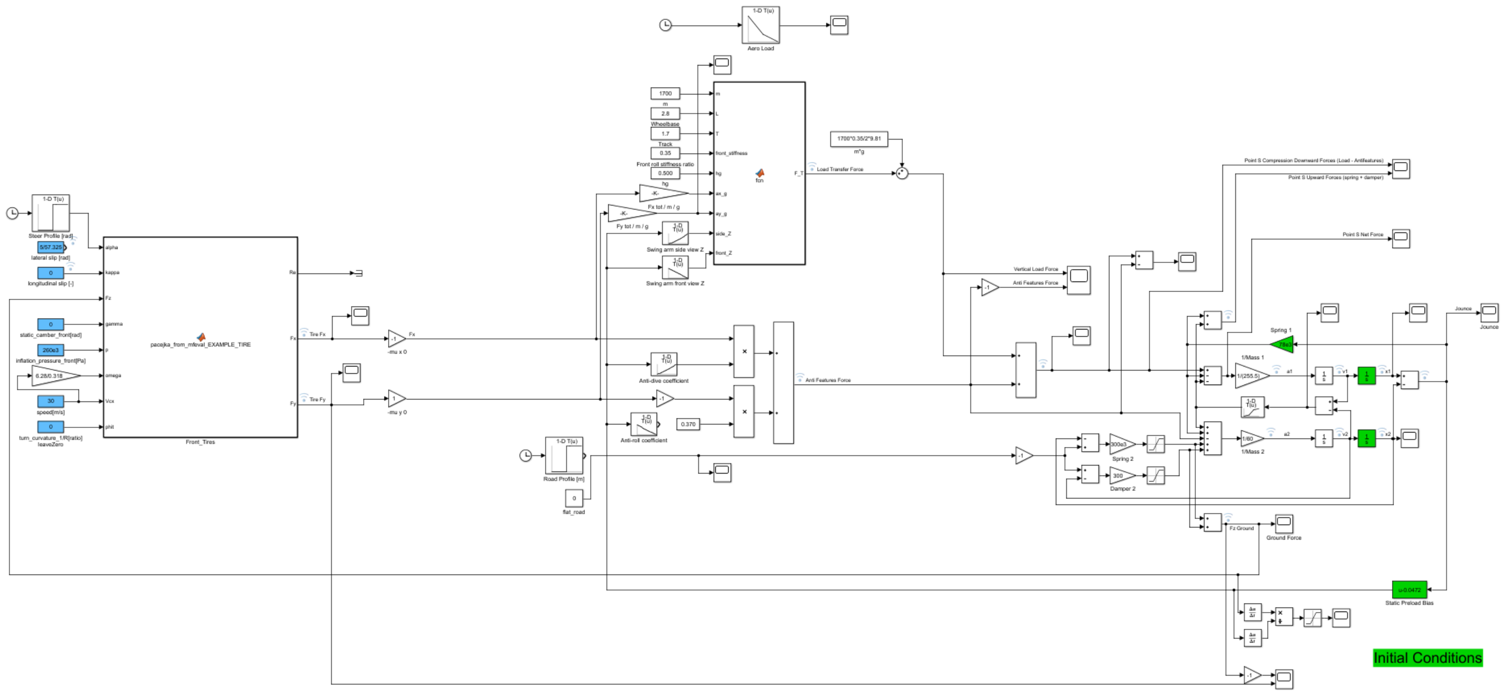
The Simulink model is 2-dof quarter car, with the addition of:

- Pacejka model for tire forces calculation
- Rigid chassis for lateral/longitudinal load transfers

The anti-features force have been applied rigidly between the sprung mass and the wheel.

More DOF may be added but the model has been proven able to qualitatively match CarRealTime results and so no more development efforts have been dedicated to it.





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3.2 VI-CarRealTime: WP1 VS 100% AR

Two CarRealTime vehicle models have been used for comparison. The first one is fitted with the delivered WP1 suspension, the second one is fitted with modified WP1 suspensions as to achieve ~100% AR both on the front and rear axles. This has been achieved rising inner hardpoints by 50mm at front and 40mm at rear. Stiffer springs and dampers have been used on the AR100 configuration to compensate for different motion-ratio.

WP1 (Lotus):

- RCH: 128mm front | 177mm rear
- Camber variation:
  0.450deg/10mm front
  0.239deg/10mm rear
- Minimal bump-steer
- Anti-dive 46%
- Anti-squat 49%
- Anti-roll outer @1g: **40% front | 56% rear**

AR100 (Lotus):

- RCH: 301mm front | 303mm rear
- Camber variation:
  0.526deg/10mm front
  0.243deg/10mm rear
- Minimal bump-steer
- Anti-dive 43%
- Anti-squat 50%
- Anti-roll outer @1g: **94% front | 96% rear**

Anti-Features Study

4. CORNERING GROUND STEP RESPONSE

4. CORNERING GROUND STEP RESPONSE

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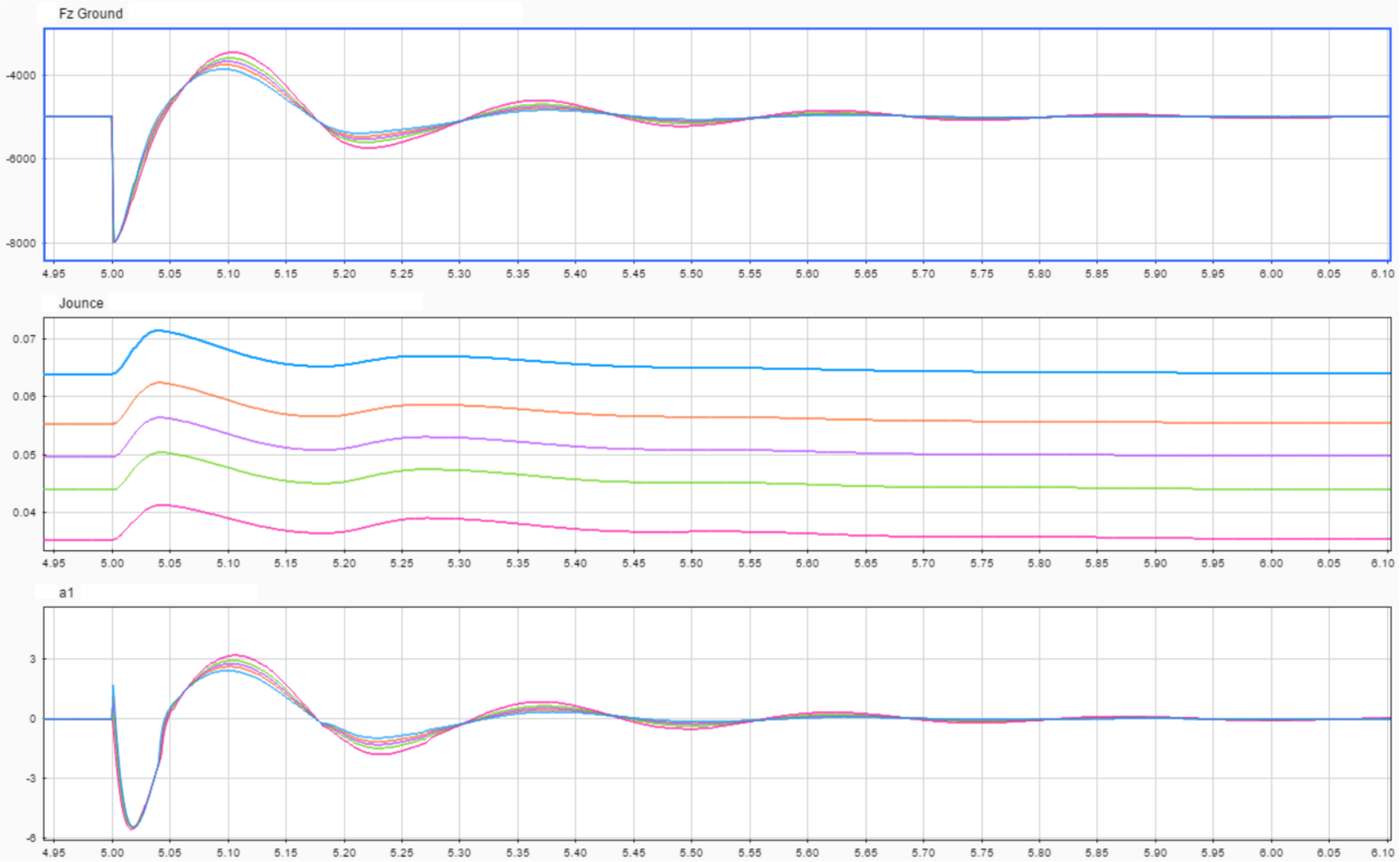
4.1 2-DOF (1/2)

Here a step of 10mm in ground height is applied. Lateral slip angle is kept constant at 5 deg, to achieve a steady state a_y=1g.

We notice that as expected the steady-state suspension jounce (graph 2 before and after step) is influenced very much by the amount of AR. On the contrary AR doesn't affect too much the response to a road step (jounce oscillation amplitude).

We notice that the ground Fz (graph 1) has slightly higher amplitude oscillations.

The big conclusion that seems to be presented is that even reaching 100% anti-roll doesn't impede to the suspension to absorb road asperities.



4. CORNERING GROUND STEP RESPONSE

4.1 2-DOF (2/2)

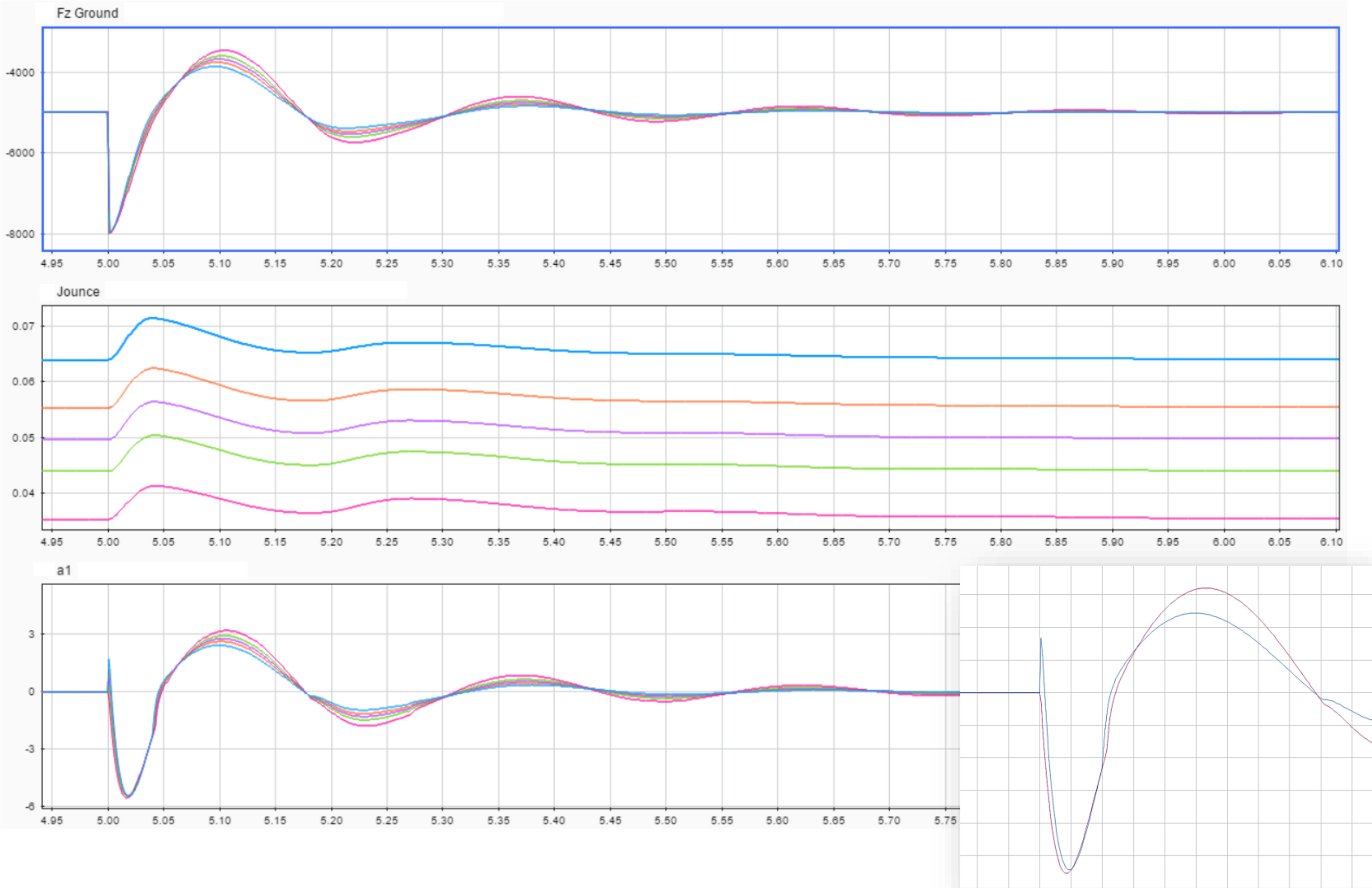
Here a step of 10mm in ground height is applied. Lateral slip angle is kept constant at 5 deg, to achieve a steady state a_y=1g.

We notice that as expected the steady state suspension jounce is influenced very much by the amount of AR. On the contrary AR doesn't affect too much the response to a road step (e.g. kerb).

We notice that the ground Fz has higher amplitude oscillations that may affect in some way the achievable grip.

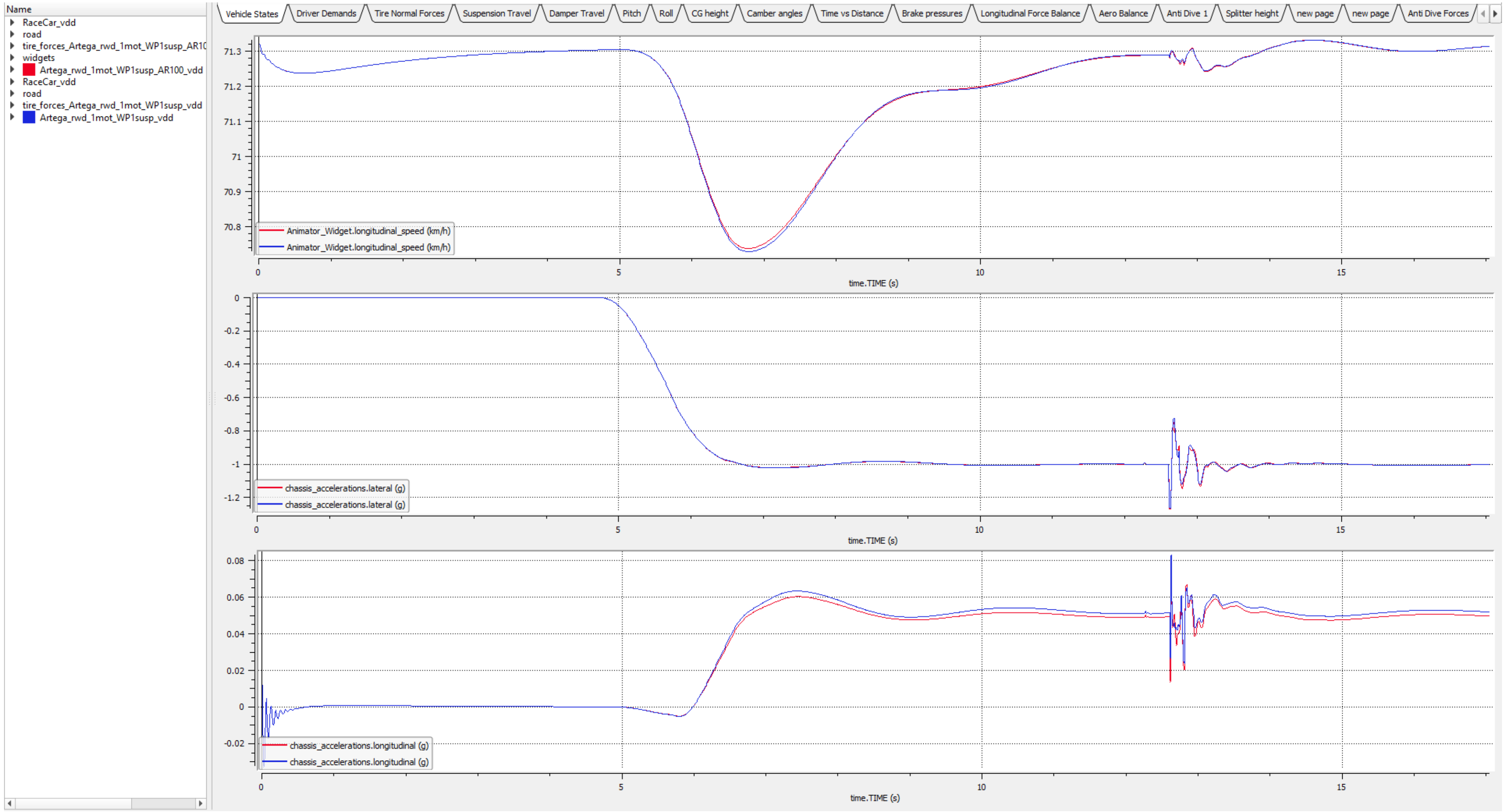
Even reaching 100% anti-roll (no steady-state jounce) doesn't impede to the suspension to absorb road asperities.

Sprung mass acceleration (a1) is comparable and moreover doesn't show the initial rising spike (bottom right image, AR100 in fuchsia).



4. CORNERING GROUND STEP RESPONSE

4.2 CRT (0/4): Vehicle States



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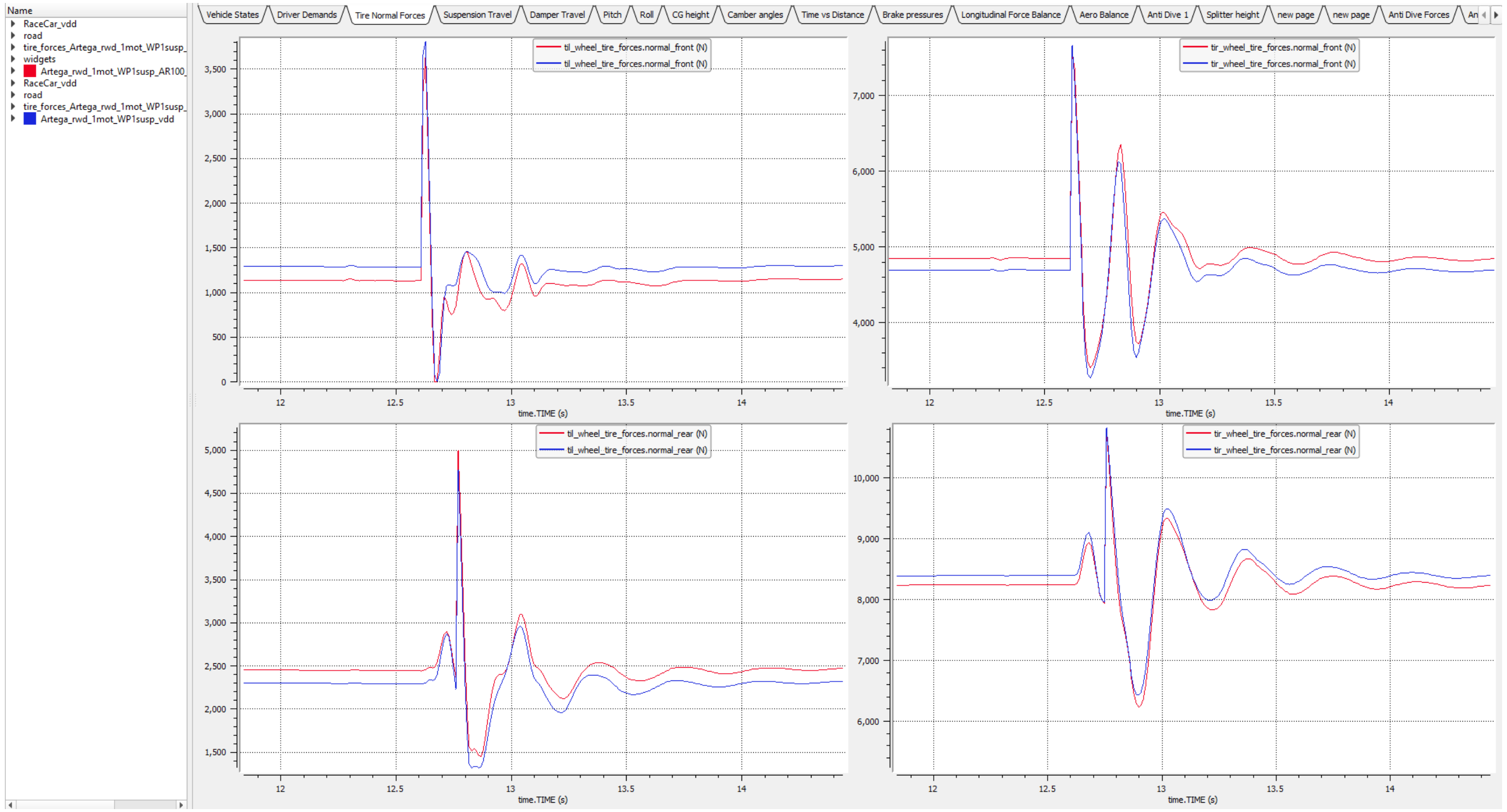
4.2 CRT (1/4): Tires Normal Forces

The same situation has been modeled in CRT building a circular track with a step up of 10mm (for the full track width).

Driver model has the objective of maintaining the correct speed (19m/s) to maintain constant ay=1g for a turning radius of 40m (outer side is right wheels).

Due to different RCH between F/R on the WP1 suspension the load transfer is distributed slightly different between front and rear axles (different roll gradient of the two axes).

Nevertheless is possible to notice that the amplitude of ground force oscillations is comparable.



4. CORNERING GROUND STEP RESPONSE

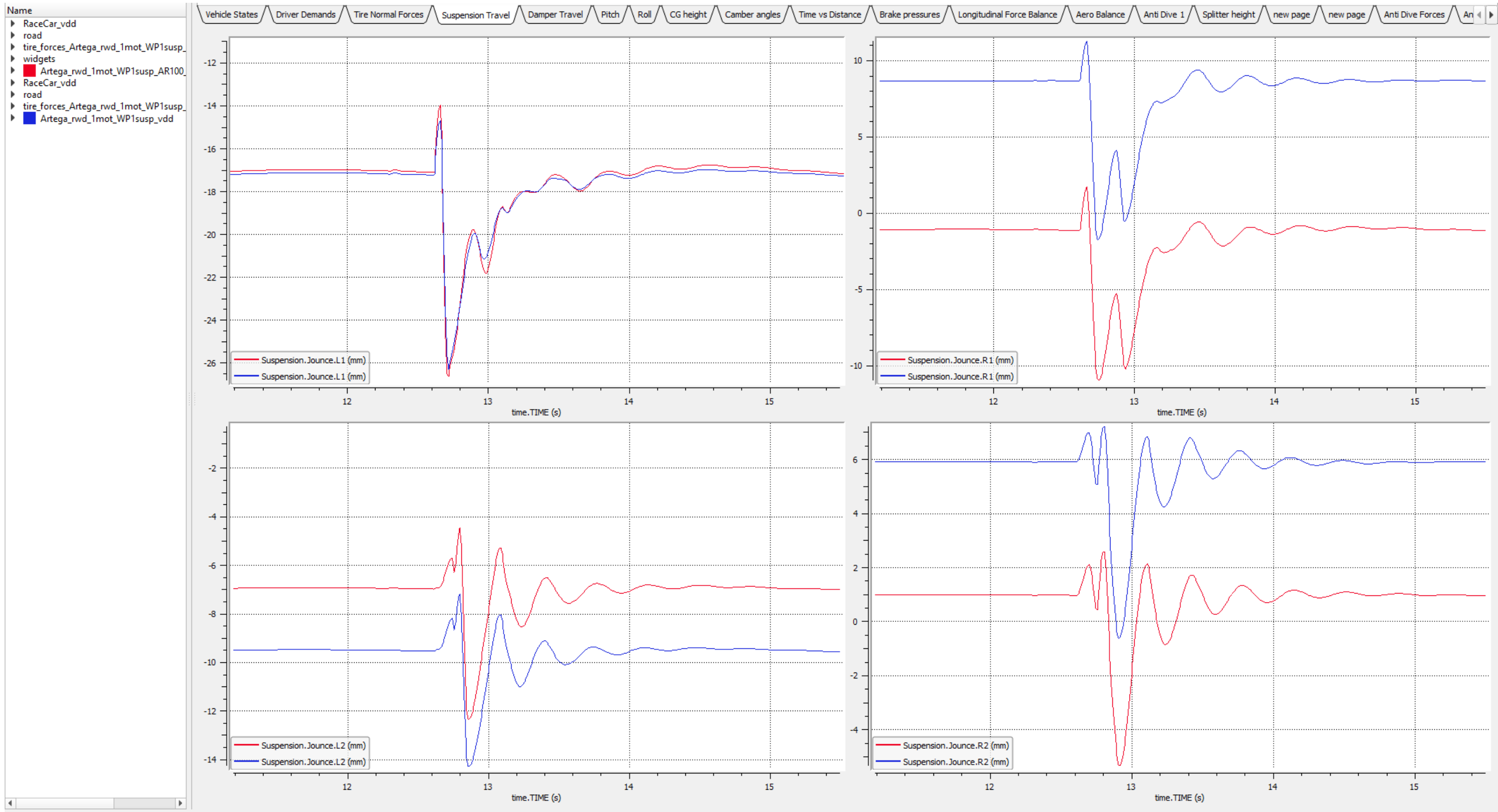
4.2 CRT (2/4): Suspension Jounces

As expected the suspension jounce of the 100% AR configuration is about 0 (<1mm).

On the inner side, the AR percentage definition is different (reversed ay sign) so that is almost impossible to achieve 100% AR (lower vertical load reduces Fy and so anti-jacking force).

This is evident on the left side suspensions travel.

At the same time suspension jounce is of about the same amplitude when hitting the step for both the configurations.



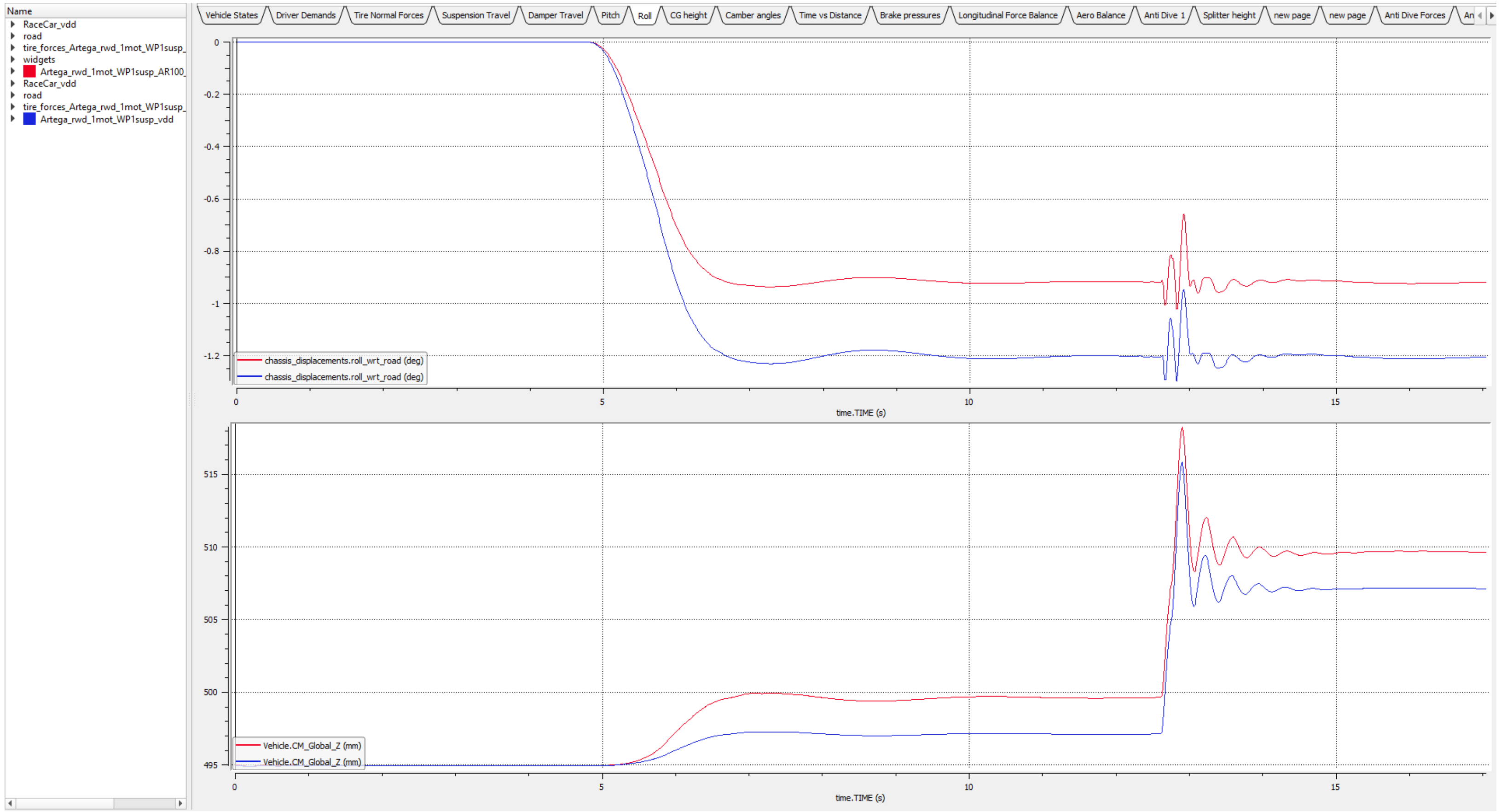
4. IN-TURN GROUND STEP RESPONSE

4.2 CRT (3/4): Roll Angle and CG Rising

As expected the configuration with 100% AR on the outer side has a lower total roll angle. This is achieved as presented in the previous slide with about the same suspension extension on the inner side and with no compression on the outer.

This behavior has the drawback of rising the CG height and thus increasing the total load transfer.

In this case the difference is still about negligible, with 3mm of rising over 500mm of static CG height (<1%).



4. IN-TURN GROUND STEP RESPONSE

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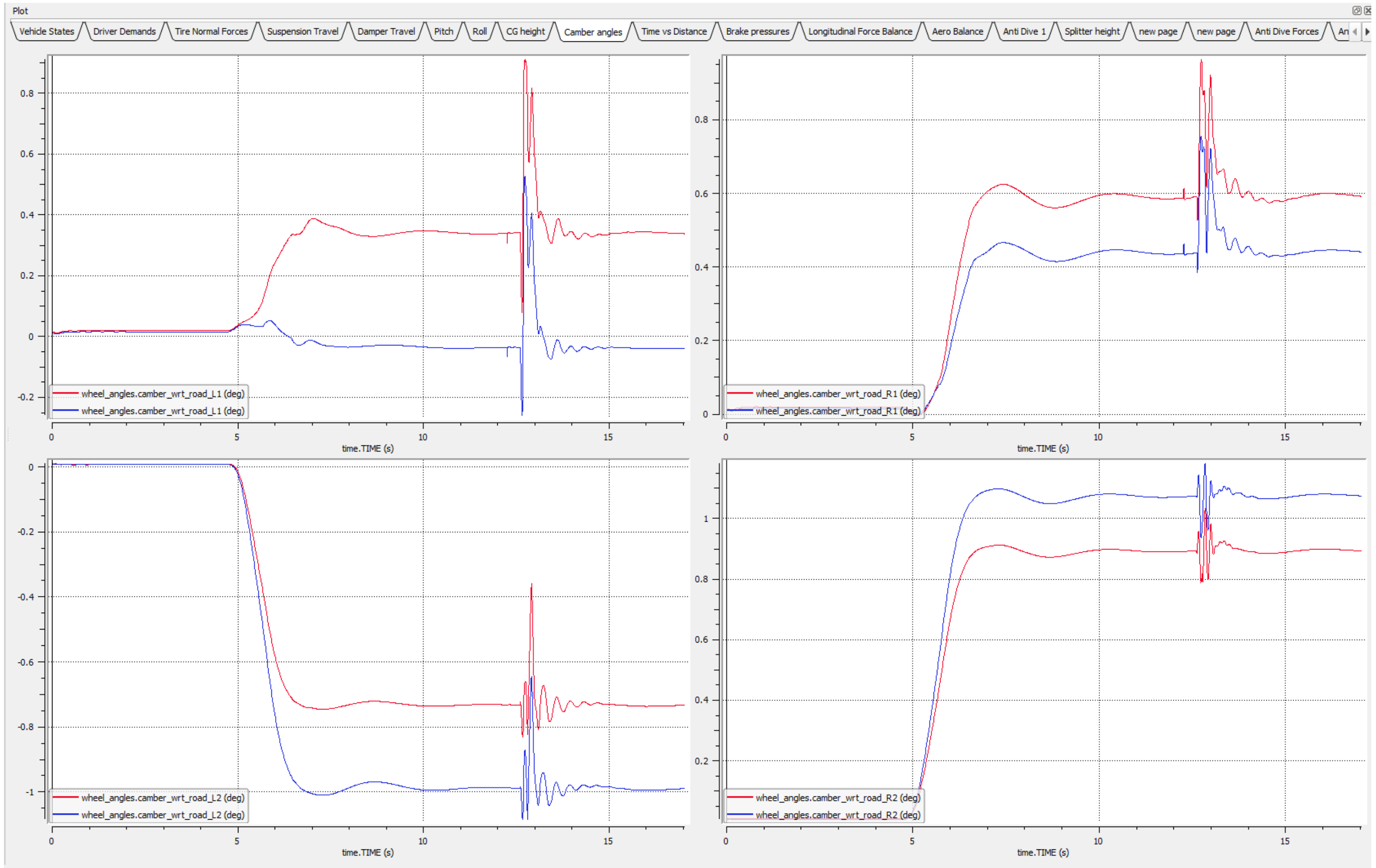
4.2 CRT (4/4): Camber at Ground

The biggest drawback of an high AR percentage is that the recovered camber is smaller. At 100% AR all the roll angle of the body is reflected as a camber loss on the ground.

This will consequently require a greater static camber that is a disadvantage in longitudinal performance.

This seems to be the most important factor to be considered in choosing the AR %.



Anti-Features Study

5. COASTING STEP STEER

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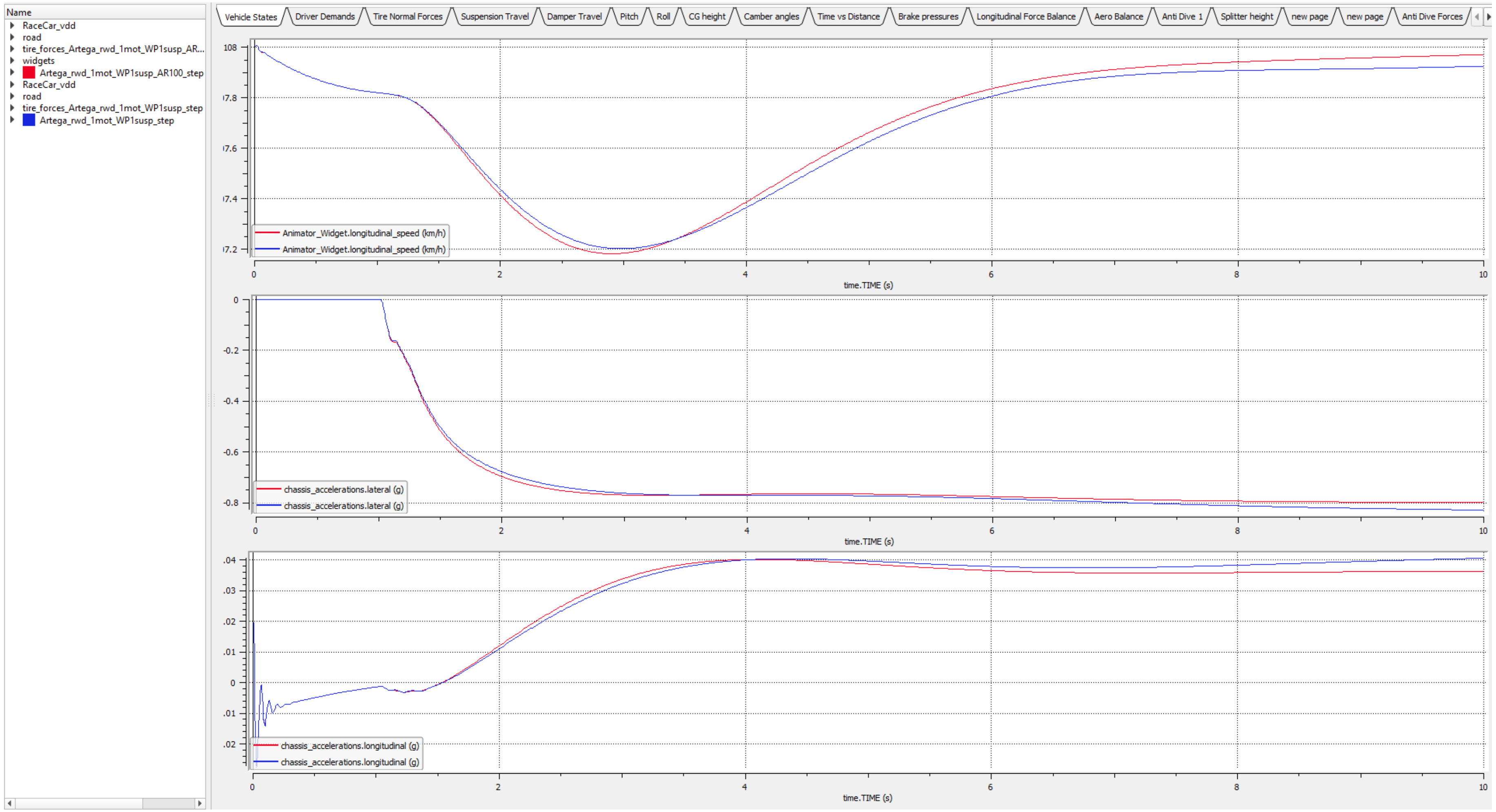
PODIUM ADVANCED TECHNOLOGIES

5.1 CRT (0/4): Vehicle States

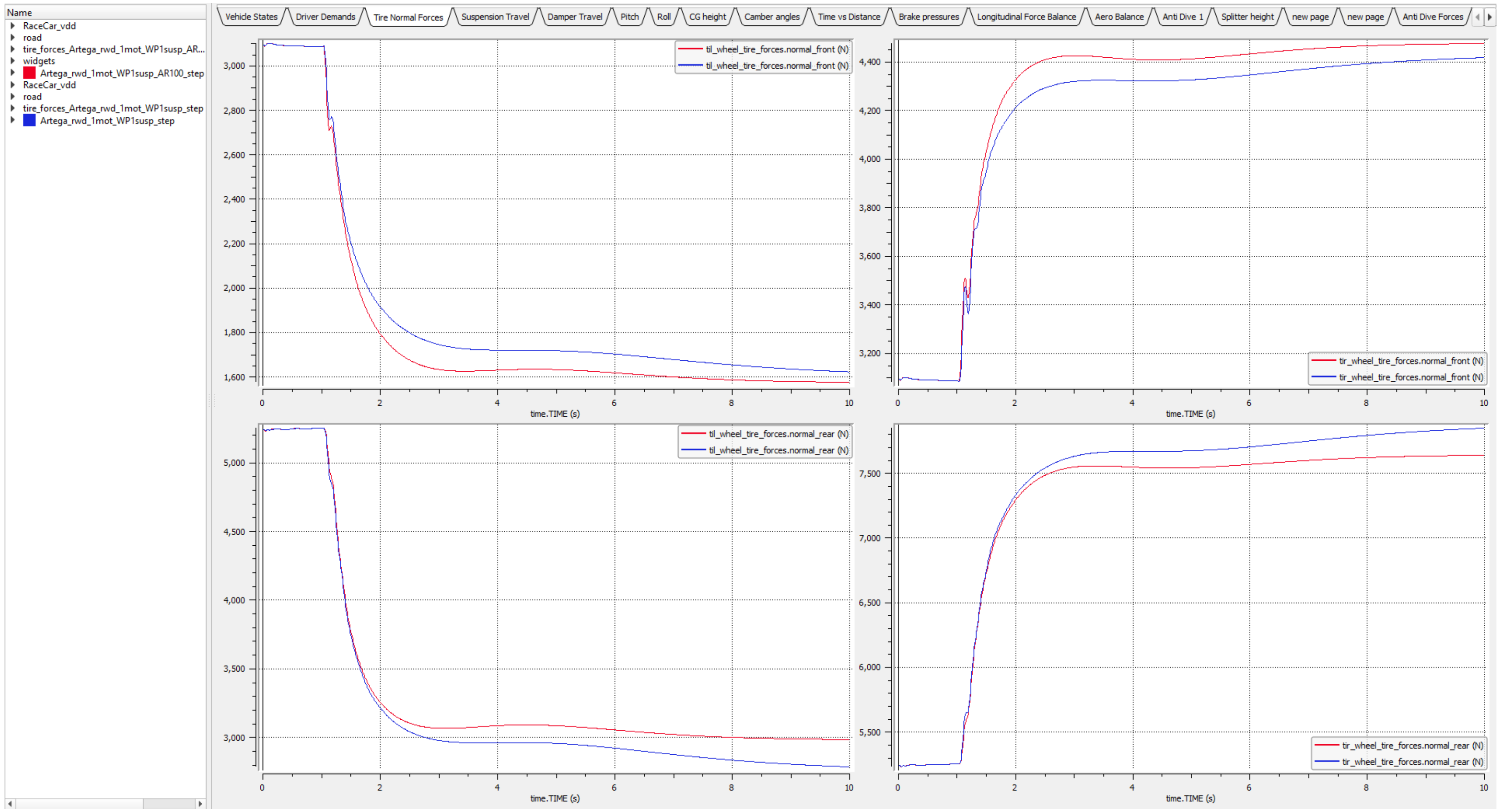


5.1 CRT (1/4): Tire Normal Forces

Since the 2-dof Simulink and CRT models show very similar behaviors, and the last one is presumably more reliable, for step steer simulations only CRT results are presented.

Ground forces have very similar behavior for the two configurations.

The slightly different front/rear ΔF_z distribution is due to different RCH (AR) of the two axles in the WP1 configuration.



5. COASTING STEP STEER

5.1 CRT (2/4): Suspension Jounces

As expected the suspension travels of the higher AR configuration are smaller.

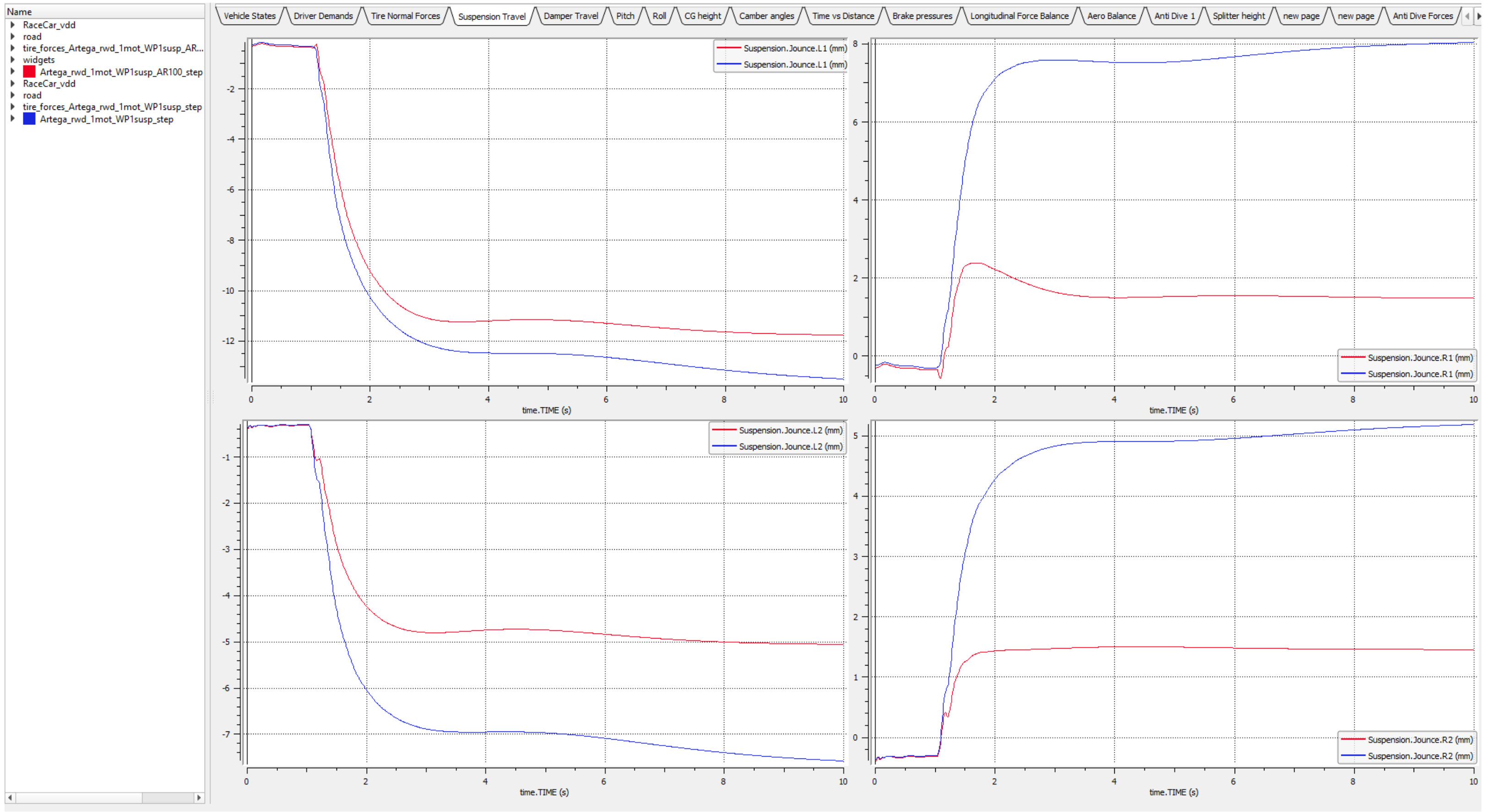
The steady-state lateral acceleration that is reached after the step (18deg) is ~0.8g. For this reason the outer suspension is a little bit compressed.

If the lateral acceleration would increase more it would be about 0mm of jounce at 1g.

On the front outer suspension it starts showing the inversion phenomena that was discovered for anti-dive.

Differently from anti-dive, it is a very rare maneuver a step of 18deg in 0.1s in normal track driving. While in braking it is normal to apply a strong impulse on the pedal. In any case the unwanted behavior starts showing with very high AR percentage (100% at 1g for a vehicle that can reach 1.3g without aero).

In any case there are not unwanted Fz consequences.



5. COASTING STEP STEER

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5.1 CRT (3/4): Roll Angle and CG Rising

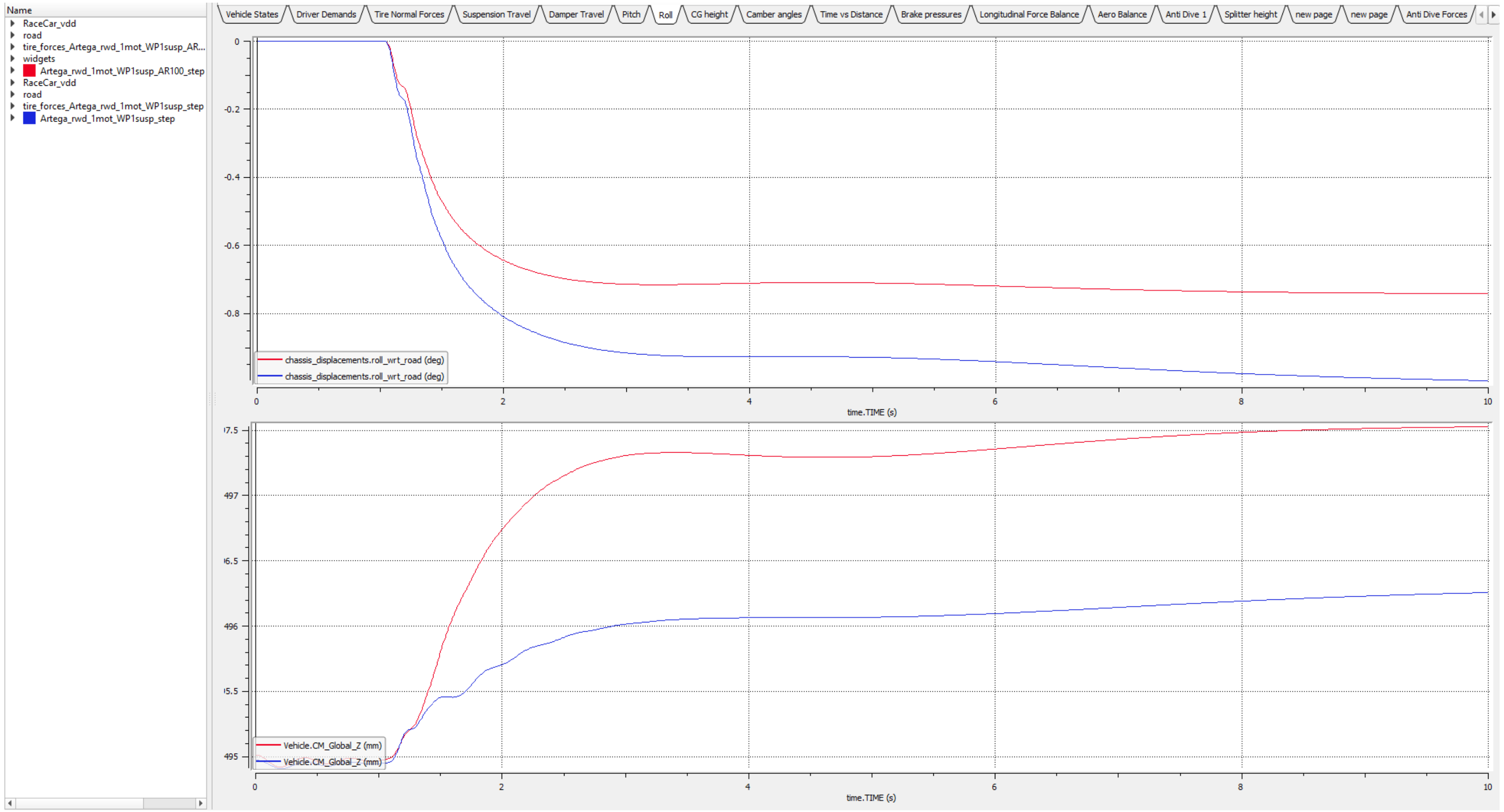
As expected the total roll angle is smaller for the “AR 100” suspension.

The interesting fact to notice is that is not visible any inversion behavior, both in angle and CM height.

For this reason probably the driver will not feel in any way the “strange” jouncing of the outer suspension damper.

This is explained by the fact that the spring extends and than compresses, while the total S-CP distance monotonically decreases. Tire compliance compensates for the suspension movement.

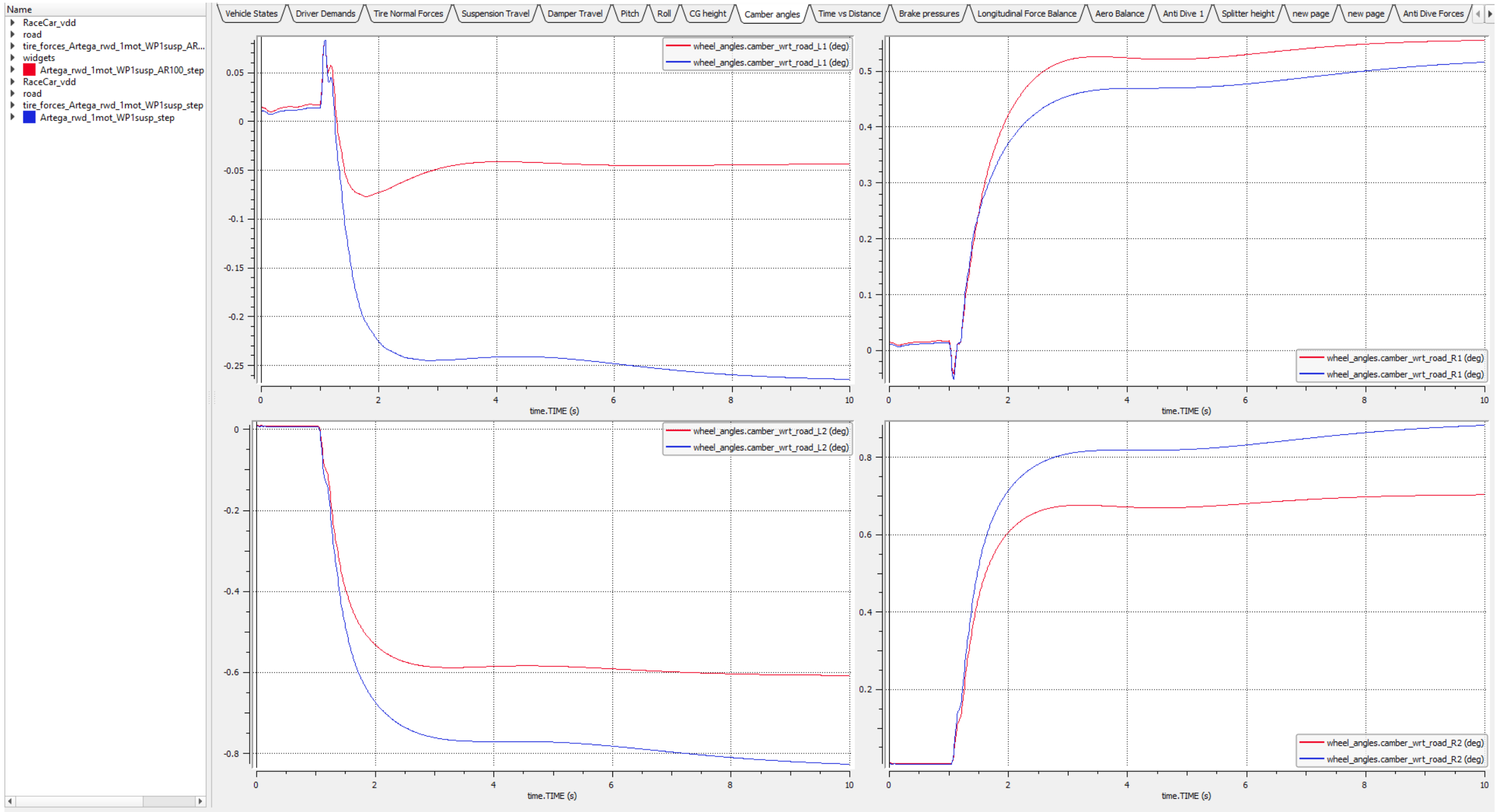
CG rising is again small (<3mm).



4.2 CRT (4/4): Camber at Ground

The camber loss is again equal to the roll angle for the suspension with 100% anti roll.

The nice spot here is that since the suspension compresses before 1g of lateral acceleration is reached, also the camber loss is big only at maximum lateral acceleration.



Anti-Features Study

6. FINAL CONSIDERATIONS

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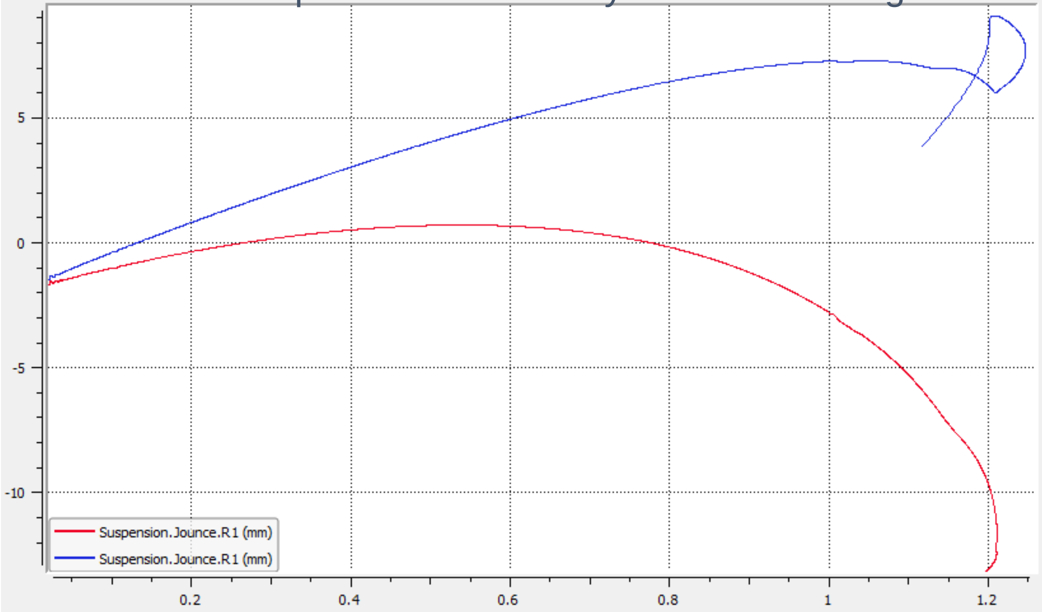
6.1 Trade-off Variables

Higher AR implicates:

- Faster load transfers -> faster response to driver inputs (can also be achieved with ARB)
- Lower roll gradient with same spring stiffness -> Less anti-roll bar needed -> Higher longitudinal grip and kerbs compliance
- Lower camber recovery -> more steer and static camber needed

If kerbs compliance is less important a lower Anti Roll setting with stiffer ARB “forces” the suspension to roll more symmetrically and thus recovering more camber. This allows for reduced static camber but increases ride frequency on the outer suspension. Where the sweet spot is, is difficult to be identified.

Outer suspension travel vs ay - 100% AR at 1g



6. FINAL CONSIDERATIONS

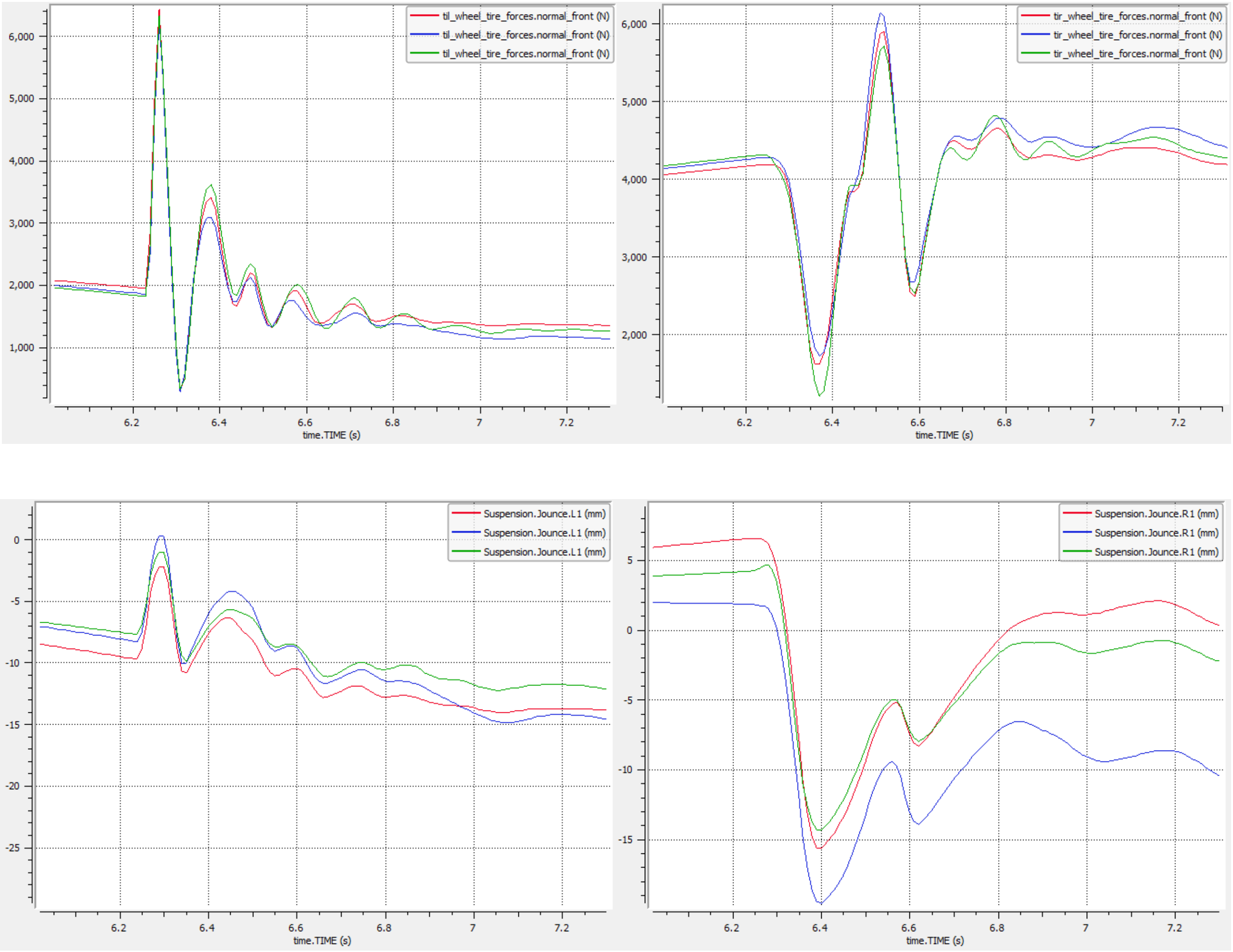
PODIUM ADVANCED TECHNOLOGIES

6.1 Trade-off Variables: Inner kerb compliance

Roll rate:

1 - WP1 no arb:	1.21 deg/g
2 - AR100 no arb:	0.92 deg/g
3 - WP1 arb (balanced):	1.02 deg/g

- Kerb height 50mm (1m chamfer)
- In case number 3, the outer (right) wheel load is more influenced by the inner wheel jumping on the kerb
- Outer suspension jounce is lower (less compliance) in case number 3
- Inner suspension with arb (3) shows higher amplitude load oscillations



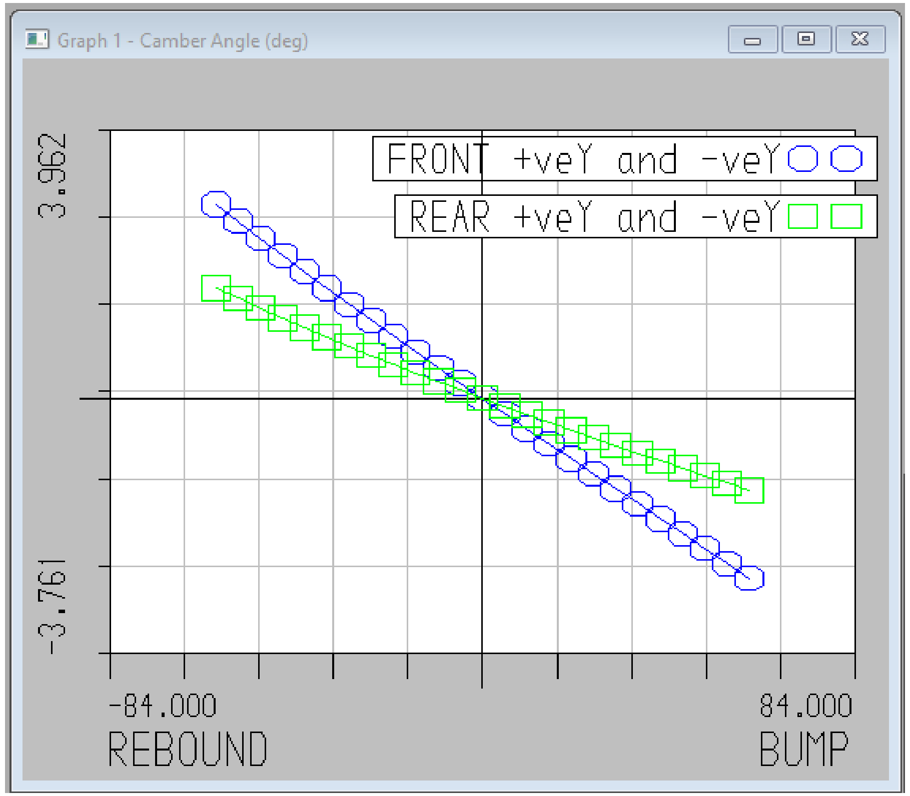
6.2 Camber observations

- Since the front suspension can recover camber also in steer, it may be opportunistic designing for a lower camber recovery in jounce at front than rear. This way, in braking, the front wheels are more parallel to the ground without adversely affect cornering.
- If the suspension has been designed with 100% AR at peak lateral acceleration, the camber recovery in this condition is null, as a consequence the amount of camber change in bump is very much less important for the outer wheel. Still on the inner it has some importance (high camber loss in extension can compensate for high static camber).
- Also at rear, if high kingpin/caster angles are used, may be investigated if a little bump steer may be used to recover some camber with very small steer angles.
- Lap simulations may be useful to define the static camber setup that maximizes lateral performance (or longitudinal). Finding the best compromise can also be done (requires some computational time). Finding best compromise requires the correct camber sensitivity of the Pacejka model (nevertheless camber coefficients are usually always included in .TIR files and we have them in 2015 Dunlop ones).

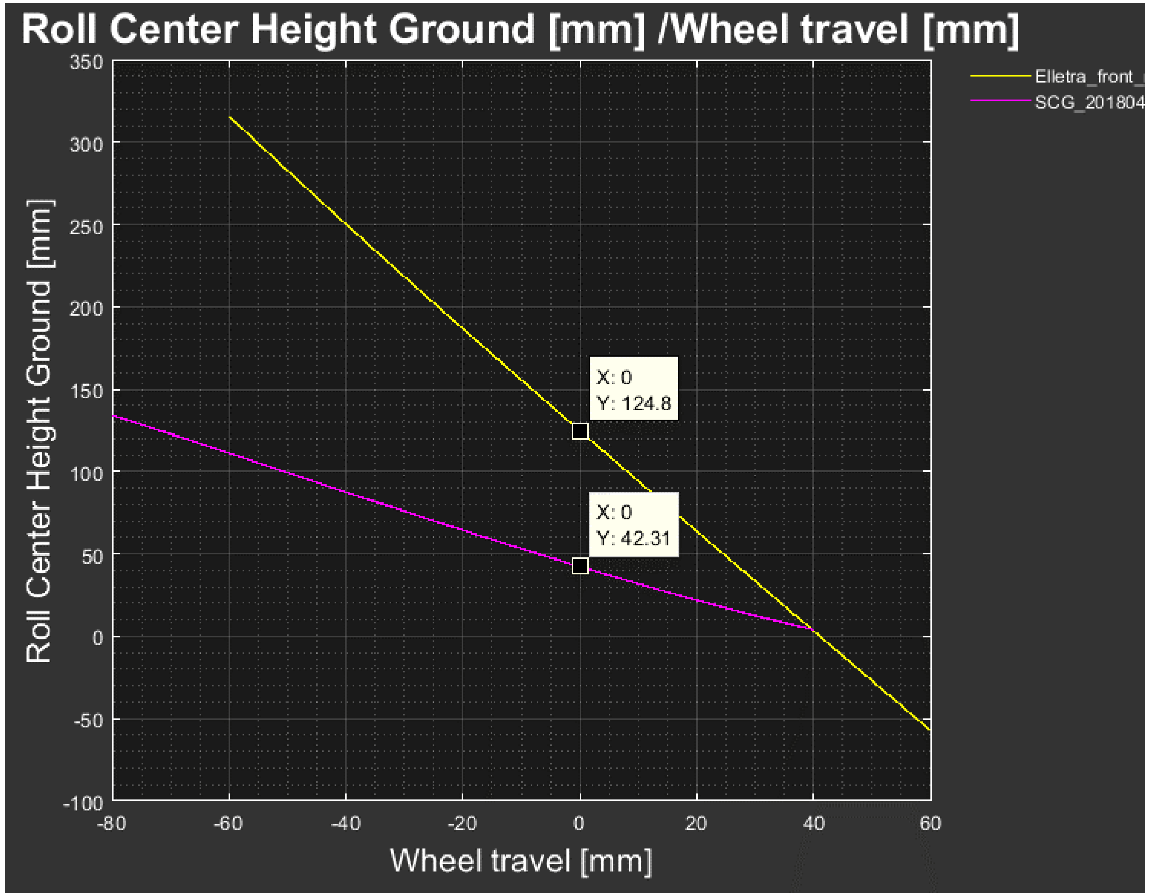
6. FINAL CONSIDERATIONS

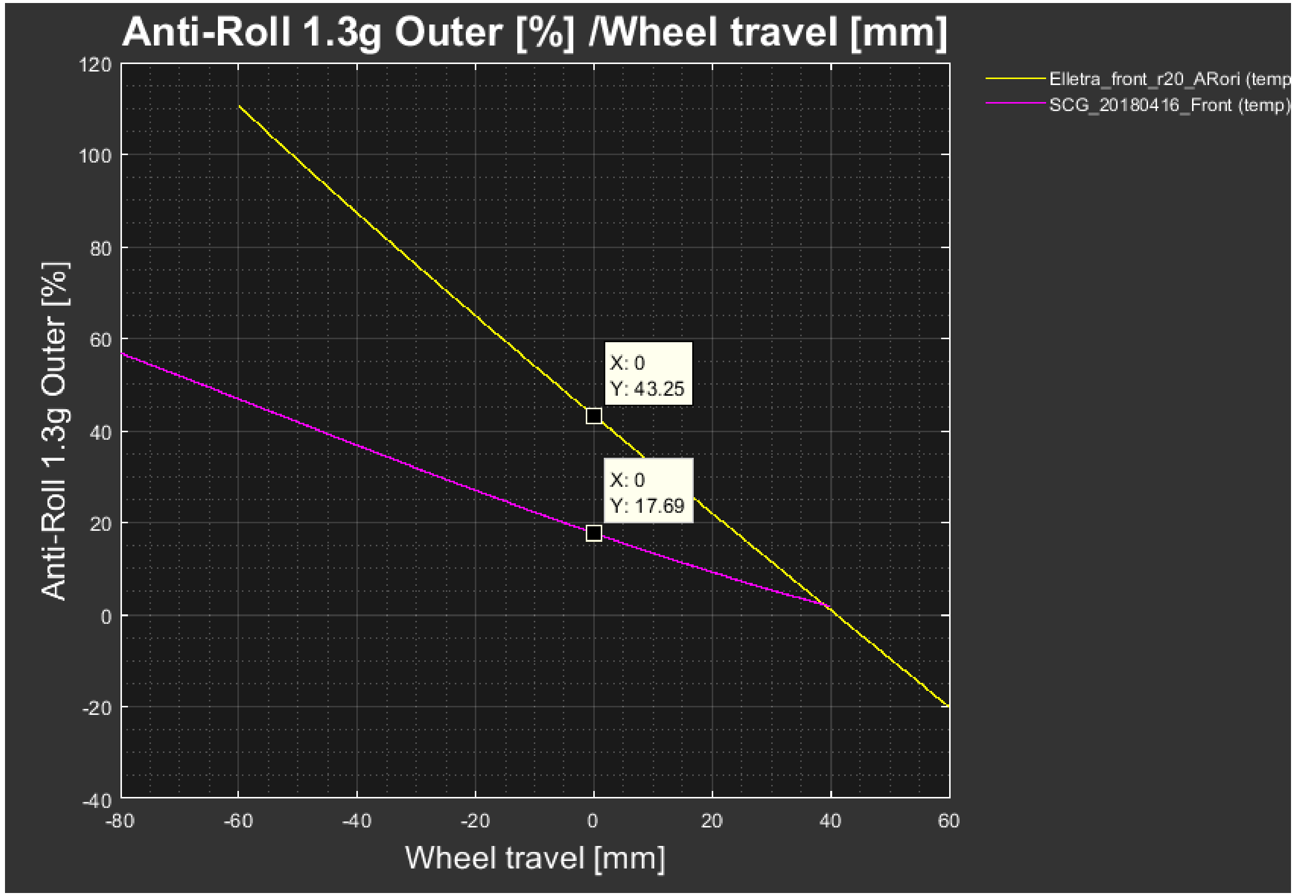
6.3 Artega WP1 suspension considerations

Camber change in jounce: at front may be reduced in comparison to the rear one or the rear one may be increased if possible



6.4 Anti-Roll SCG vs Artega WP1





6. FINAL CONSIDERATIONS

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6.5 Open points

- Why in literature is often suggest higher RC at rear axle?

  Found consequences of higher RC at rear are:

  - higher load transfer at rear due to different roll stiffnesses -> moving balance towards oversteering

  - if rear ride frequency is higher than front the situation is even more evidenced (similar tracks hypothesis)

- More?

- Develop SKCT with math functions for calculation of:

  - Jounce vs ay (inner and outer)
  - Roll angle vs ay
  - CG rising vs ay
  - Camber at ground vs ay (inner and outer)
  - 2-dof ride frequency with and without ARB

6. FINAL CONSIDERATIONS

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6.6 Design Guidelines Proposal

- Set target roll gradient (deg/g)

- Check how much anti-roll is necessary
  (if different ride-frequency between F/R or very different tracks check roll stiffness! And adjust RCH/AR to have a neutral base setup)

- Check how much camber is recovered and how much static one is needed

- If too much static camber (criteria for too much? Common sense?) reduce anti-roll and use an ARB
  (benefit of ARB: not only the outer suspension moves more per-se, but also the ARB forces the suspensions to move more symmetrically)

- …

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