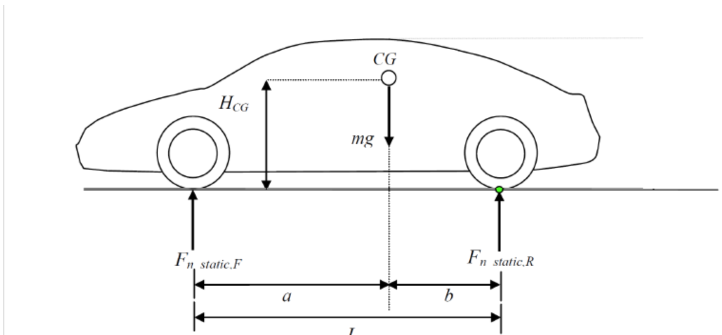
Lateral Dynamics - Roll dynamics and lateral load transfer

Load transfer is a phenomena which has huge influence over the vehicle dynamics, therefore it is a discourse that needs to be handled delicately. Moreover, laod transfer is caused by multiple effects at its definition is not so trival. However, dividing each cause permit to have a general overwview and a precise description, beacause of this each component will be anaylised separatly, then merged together.

Static load transfer

Since the car has a weight, is not a point, and its centre of mass never cross the vertical mid plane (unless very special cars liek the small urban electric microcars) the mass of the car won't be equally distribuited on its body. The more the CoG moves behind, more the weight of the overall car will squeeze the rear tyre, resulting into an higher vertical force. The amount of this difference can be easily calculated through an equilibrium of momentum.

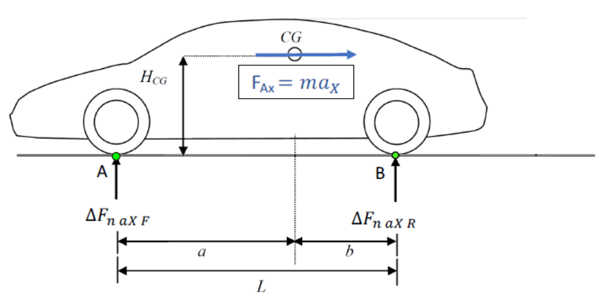


$$\sum_i M_{i,B} = 0 \longrightarrow F_{n\,static\,F} = mg\frac{b}{L} \longrightarrow F_{Z,FLR,st} = F_{Z,FR,st} = \frac{1}{2}mg\frac{b}{L}$$

$$\sum_i M_{i,A} = 0 \longrightarrow F_{n\,static\,R} = mg\frac{a}{L} \longrightarrow F_{Z,RLR,st} = F_{Z,RR,st} = \frac{1}{2}mg\frac{a}{L}$$

Longitudinal acceleration

If the car is performing an acceleration, the inertia force will be applied to the CoG and following the same principle used in the previous section, a certain load trasnfer is get:

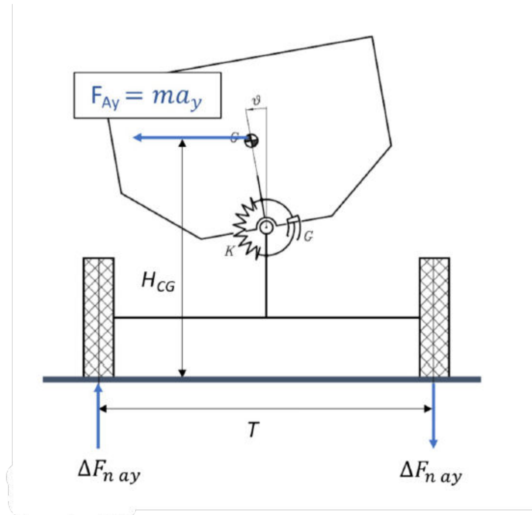


$$\sum_i M_{i,B} = 0 \longrightarrow \Delta F_{n\,aX\,F} = -F_{Ax}\frac{H_{CG}}{L} \longrightarrow \Delta F_{n\,aX\,FL} = \Delta F_{n\,aX\,FR} = -\frac{1}{2}ma_X\frac{H_{CG}}{L}$$

$$\sum_i M_{i,A} = 0 \longrightarrow \Delta F_{n\,aX\,R} = F_{Ax}\frac{H_{CG}}{L} \longrightarrow \Delta F_{n\,aX\,RL} = \Delta F_{n\,aX\,RR} = \frac{1}{2}ma_X\frac{H_{CG}}{L}$$

Lateral acceleration with no roll motion

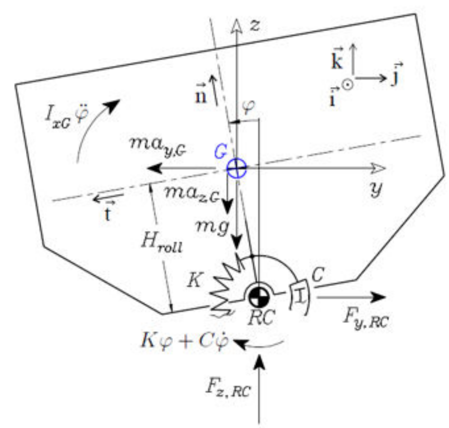
So as the longitudinal acceleration implies a vertical force distribution among the front and rear axles, so does the lateral acceleration for the two sides of the vehicle. However, the second effect is more slighlty more complex due to the roll motion which is one of the DFO that majorly influence the car behaviour. As frist step, this motion won't be considered. Once analysed and understood its dynamic, a complete lateral load transfer with roll motion will be performed.



Since the roll motion is not considered, once again the load transfer contribution can be calculated with a momentum equilibrium: $\Delta F_{n,ay} = ma_y \frac{H_{CG}}{T}$. Noteworthy is how the mass distribution has an effect over the lateral load trasnfer. In fact, the percentage of laod transfer on each axle depends exaclty on the mass distribution. In other words, if the vehicle has a 60:40 mass distribution, the full vehicle can be considered (in this case with no roll motion) as two masses, one equal to the 60% of the overall, the other 40%, which rotates around their CoG indipendently on each other.

Lateral acceleration and roll motion

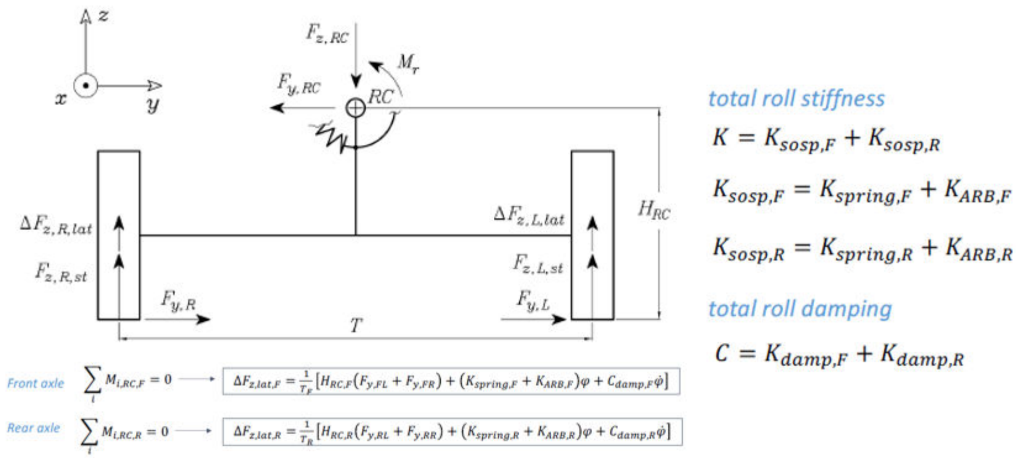
If the roll motion is considered instead, the system is described by a differential equation since there is the contribution of damping.



roll equation for sprung mass

$$\ddot{\varphi} = \frac{mH_{roll}\left(a_{y,RC}\cos\varphi + g\sin\varphi\right) - K\varphi - C\dot{\varphi}}{I_{xG} + mH_{roll}{}^2}$$

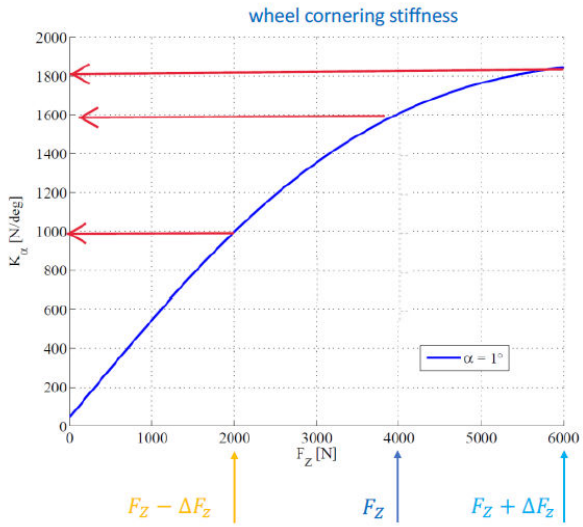
Fundamental in this analysis is that the contribution of stiffness and damping is given by both suspension and axle. In fact, springs and damper provieds contribution of K and C at the same time (to get each contribution knowing the other it is possible to adopt the elastic potential energy principle $\frac{1}{2}kx^2 = \frac{1}{2}k_{damping}[tg(\phi) * l]^2$:



The two load transfer equations permit to get to some conclusions. First of all, the roll dynamics is a consequence of the anti-roll bar stiffness contribution, which is given as the sum of suspensions and axle stiffness. Moreover, the difference of the anti-roll bar stiffness determines the distribution of the total load transfer between the two axles. This means that during roll motion, if one axle is more stiff than the other, it will substain higher vertical forces.

This has huge implications into the vehicle dynamic, since remebering that the vertical force has a direct impact over the lateral cornering stiffness of a vehicle, for a stiffer axle (like a standard vehicle with a single antiroll bar inclusion without any further modification) the vertical laod will be higher but due to the saturation effect, the lateral cornering stiffness will reduce. Having lower cornering stiffness for the same wheel's side slip angle, the tyre will extert lower lateral force, resulting, in the end, into a lateral cornering loss of capacity of that axle.

Putting it in other words, the stiffner and axle the more easily it gets critical and start to slip due to the reduciton of lateral forces developed by it.



Simulation part

This exercise has the aim of studying the effects of some variations over the vehicle handling performances. In particular, the proposed exercise suggest to observe the effect of the antiroll bar in both common scenario and a transient phase. Moreover, also the adoption of damper runned-out will be simulated. Then as a conclusione the design of the stiffness of an antiroll bar for a standard vehicle is proposed.

Further investigation will be perfomed, since it wasy of my interest to evaluate the change of mass distribution first and of the driving wheels traction system then.

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% Run_Lat_Dyn_Model -> Reference script for simulations
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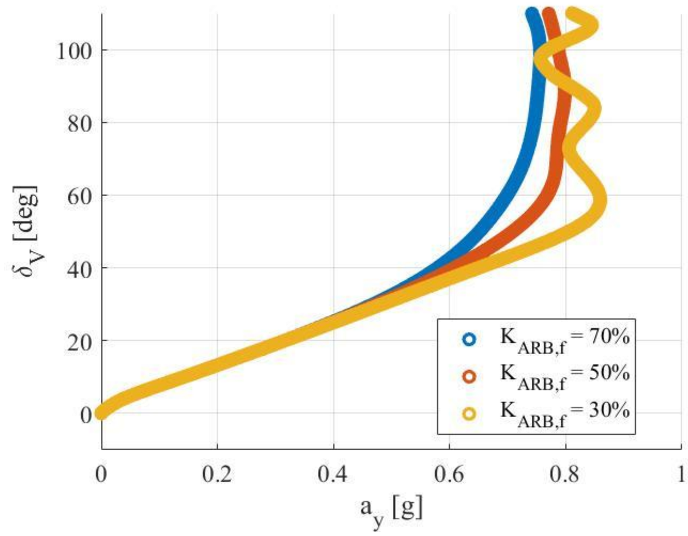
Understeering characteristic – influence of roll bars

- **ramp steer at 100km/h, 5th gear, steering wheel angle gradient 15°/s (the rest all default)**

Since the intresting chart was only one, the simulations and their settings were done indivually and here it will be presented only the resulting chart.

Remember: As in the previous exercise and in general, the ramp sterring is considered as a steady-state manoeuvre, since the variation of the steering angle is slow and constant. The step on the contrary is a sudden change, therefore it is adopted as a transient responce simualtion.



Firstly it is necessary to justify the differences respect to the more complete chart of the slides. Probably the slides adopt further modificiations, but for the aim of this exercise the chart obtained is more than enough.

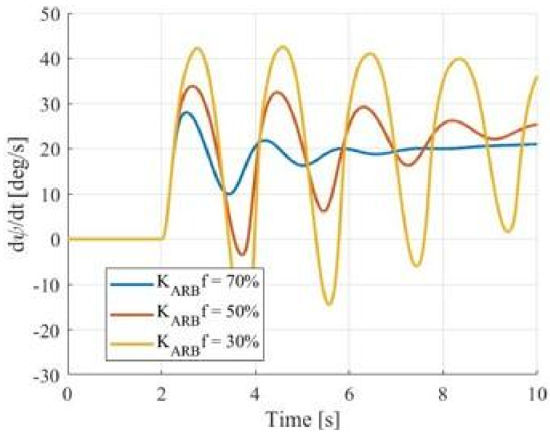
This chart explain how for a given lateral acceleration, the steering wheel change accordingly, suggesting that depending on the antiroll bar stiffness distribution the vehicle behaves differently. In particular, if the contribution of the front axle stiffness decreases (therefore increases the rear one - as shown in the slides), lower value of steering wheel are imposed. This means that in order to react to a certain lateral acceleration having a stiffer rear the driver has to curve less respect to a stiffer front axle vehicle having the same lateral acceleration.

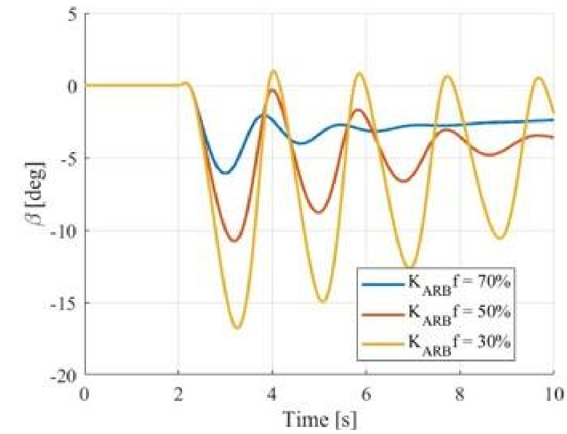
In other words, *increasing the rear axle stiffness has as effect to enhance the oversteering behaviour of a vehicle*. This conclusion is fundamental and critical since it is the one that the professor expect from us. The physical reason has been already explained, but in short the more stiff an axle gets the more it will become critical. Moreover this conclusion has to be observed for all the simulations in this exercise which are related to handling performances.

Coming back to the chart, it can be read also in other way around: for a given steering angle, a stiffer rear axle will imply an higher lateral acceleration. In fact, this vehicle will be more oversteering which means that it will bend more the curve ($= lower \;\; R$). Since $a_y = \frac{V^2}{R}$, the lateral acceleration will increase accordingly.

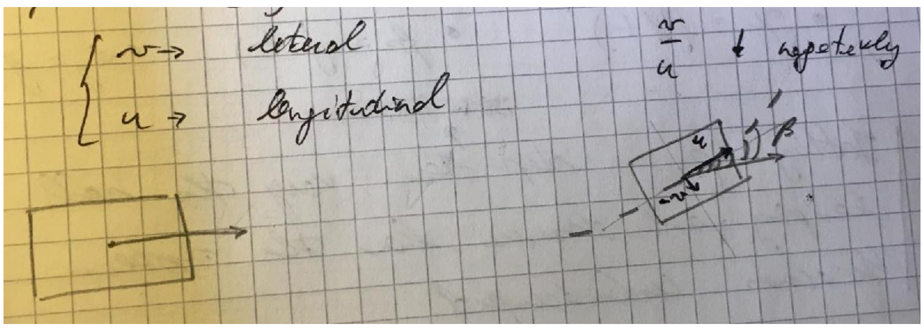
- **step steering 100-100: steering of 100° at 100 km/h (the rest all default)**

Step responces are now evaluated so as the antiroll bar siffness distribution over them.



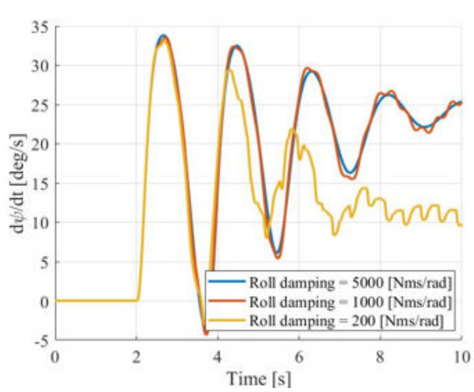


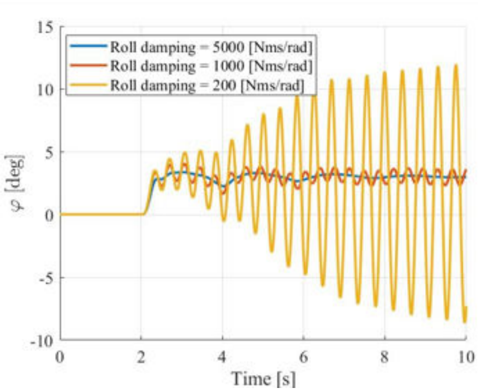
As stated before, for higher values of rear axel stiffness the vehicle tend to become more oversteering, whohc results in higher yaw rate. In fact the vehicle will turn more following a clsoer trajectory, having higher lateral acceleration and therefore also the rotational speed at which it is curving will increase, since yaww rate, lateral acceleration and radius of curvature are quantities all related.



Following the same principle, the vehicle side slip angle confirm the hypothesis done. Since the vehicle is bending at higher lateral acceleration, higher will be its lateral componet (u). Since beta is estimated as $\beta = atan\left(\frac{v}{u}\right)$, higher will be lateral component higher will be the vehicle side slip, which is exaclty what is occurring in the simualtion set. Moreover, the negative sign is due to manoeuvre, since the vehicle was proceeding longitudinally and a suddne step is provoced, V will point toward the outside of the curve but the vehicle is turning according to the curve, resulting then in a negative sign.

- **Transient response sensitivity analysis - effects of worn out shock absorbers: step steer 100-100**





In this part the shock value of damping for the shock abrober is changed. The damping influencec the dynamic behaviour of the vehicle, in partiuclar it damps the oscillations of motions due to the vehicle dynamics. As the logic would tell, higher the damping lower will be the oscillations at the cost of more energy to dissipate for the damper (= higher temperature).

In this simualtions it was supposed that the shock absorber was running out and as consequence the oscillation of motion becomes more severe. But what is the consequence of an oscillatory motion for a vehicle?

The problem of having oscillations for a vehicle suring its curve is that not only all the quantities will have an harmonic tendency, also the forces that the tyre develops on the ground will have the same trend, since they are directly related. Oscillating forces at tyre-ground contact implies a traction capacity of the vehicle very unstable since it is changing too rapidly respect to the lateral dynamics quantities frequency, resulting then into a loss of drivability and unsafety. This is well explained from the trajectory draw by the low-damepd vehicle: just having worn out dampers implies a change of location of *32.57 meters!*

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sqrt((133.25-105.364)^2+(133.57-116.74)^2); % location read on the trajectory chart
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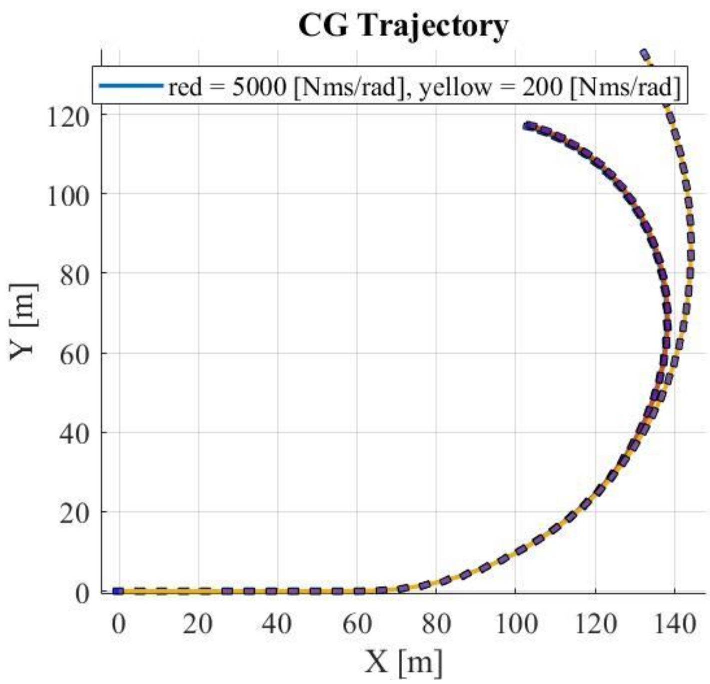


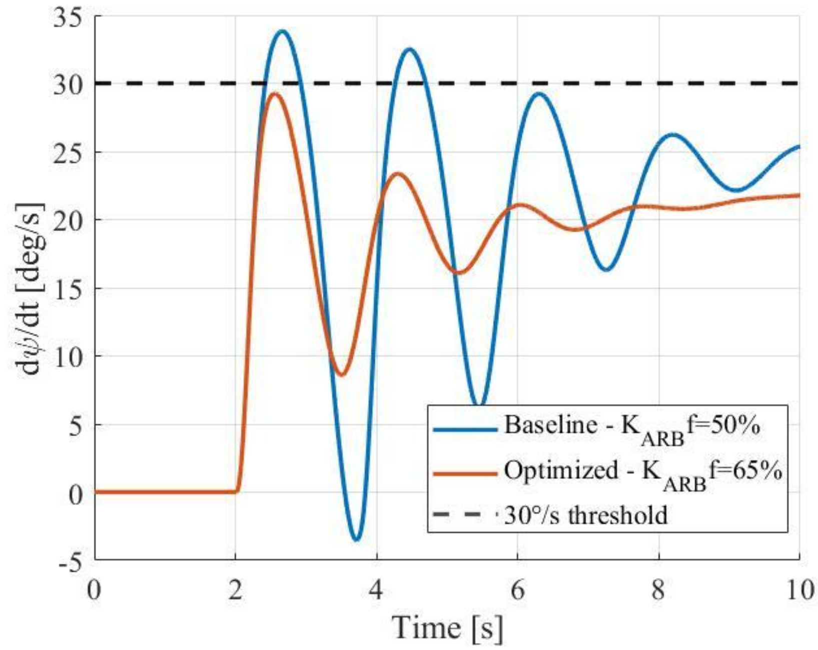
- **Design of roll bars: steer step of 100° at 100km/h: maximum sideslip angle is 8°; yaw rate peak does not exceed 30°/s.**

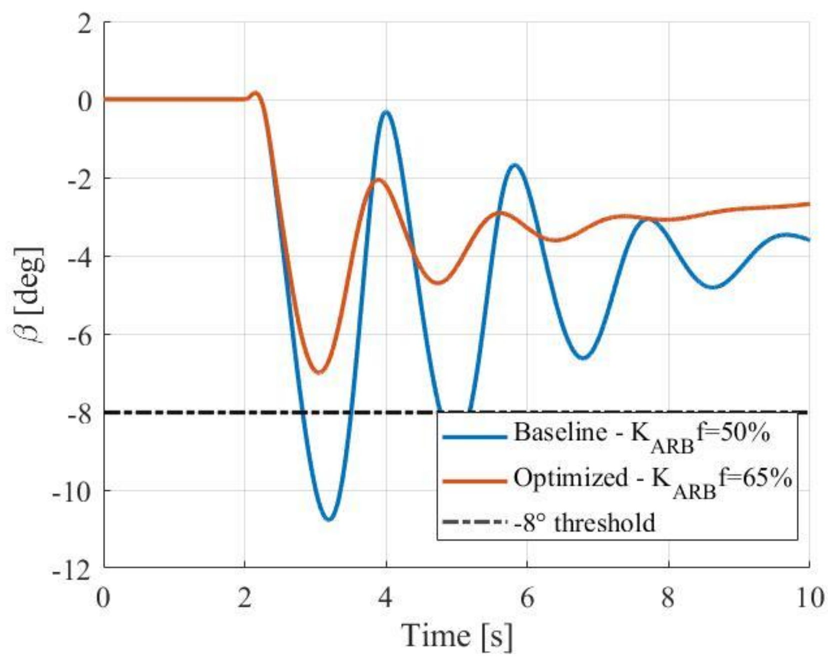
The design requirements given represent resonable values for the quantities involved. Therefore the design procedure can be implemented also in real scenarios. In order to find the ratio optimal, a trial and error precedure was adopted since it was the most rapid considring the fast setting up and the low time for each simualitons.

However a certain criteria was followed along the procedure. Starting from the baseline (all default values), it was observed how the beta and yaw rate were higher than the request. Therefore the vehicle was too much oversteering. In order to improve the understeering behaviour, the frontal axle should gain some stiffness.

Fro this principle some simualtion were run until a certain value reached the solution required. A welocomed solutioncould be to run an iterative for cycle with a flag that progressively increase the stiffness.

For sake of comprensibility and clearness, only the baseline quantities and the optimal one are presented.



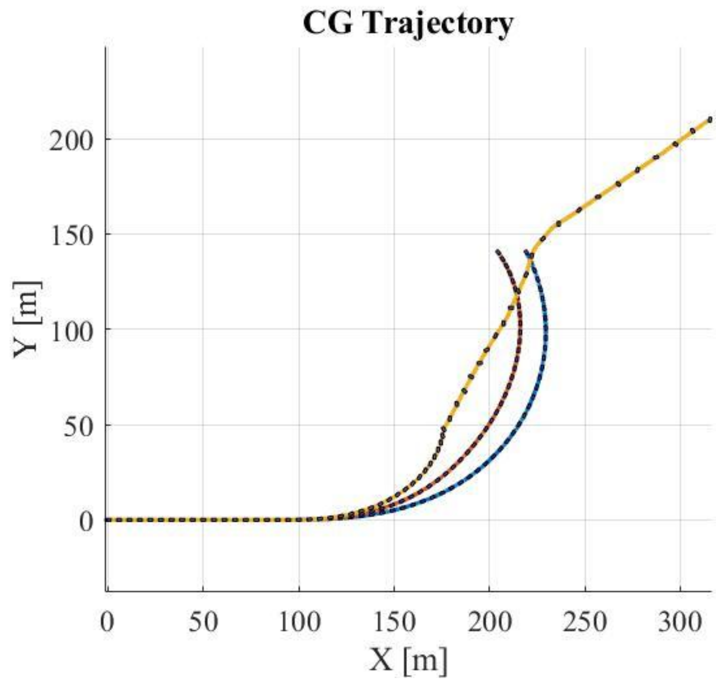


Vehicle side slip angles (betas) affect the entire dimanic of the vehicle. In real-world contexts, a Ferrari might go a maximum of 7° while a simple Panda will go no further than 3°. So it makes design sense to look for lower betas. Tha same goes for the raw rate, taking into consideration that a low velocity slalom does not exceed values above 1.5 °/s.

- **Mass distribution effect - ramp steer**

The definition of under/oversteering is by no means trivial. In the previous exercise it was taken as a geometric consideration for simplicity of the model. If a 40:60 mass is set but the pressure of the pneumatics is asymmetric, for example, having the rear end more unloaded the vehicle might even become understeering. Thus, the assumption that a mass distribution implies behaviour is true, but only in these simplified models.

For sure the mass is one of the major factor over the vehicle dynamics since it is a factor for a lot of properties so as its very high values respect to the other parameters. Due to this, in fact, the simualtion selected for this part has been chosen as a ramp instead of a step. This is due to the fact that for a step steering a 40:60 will be so unstable that there is a numerical error which will block the simulation.



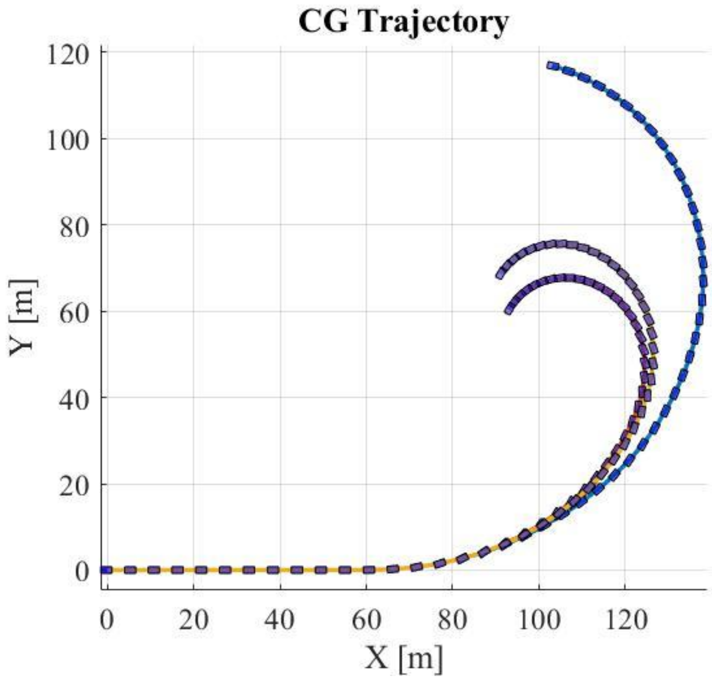
<u>Legend - mass distribution: Red = 60:40; Blu = 50:50; Yellow = 40:60</u>

As expected, to have more weight on the rear implies more vertical load and therefore the rear axle becomes critical. Moreover, the mass distribution impact so much the behaviour that a 40:60 becomes unstable and spins. This is a claer demonstration of the considerations done in all the lateral dynamics exercises.

- **Driving system influence - step steering**

The traction system as well has some influence over the vehicle behaviour, even if less than the mass distribution, which is by any doubt one of the major ones (not by chance a step steer simualtion can be perfomed with a RWD, a 40:60 mass distribution not). However some effects of the DW selection are present. This is due to the fact that the driving wheel will exert higher longitudinal forces, considering the elliptical model, for trye-groudn forces, the capability to develop lateral forces reduces for higher longitudinal forces. This transaltes into less cornering capability of the axle with higher longitudinal forces, which in few word can be stated as: the driving wheel axle will be more critical than the non driving.

WIth this assumption, it is expected that a RWD will travel a closer trajectory, having a more an oversteering behaviour, meanwhile a FWD following the same principle will exert more an understeering behaviour. Regarding 4WD the situation becomes a bit more complex since it is not possible to say at priori which of the two effects will prevale. However it is resonable to think that the AWD will put itselft in between the RWD and the FWD considering the cornering capability.



<u>Legend - mass distribution: Red = FWD Blu = RWD Yellow = AWD</u>

As hypothesized, the FWD has higher radius of cornering, the AWD is in between and the RWD has the lower radius of cornering.