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Torque Vectoring

Combined Advanced Vehicle Dynamics/ACHA6003/6011

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1 Base Vehicle Data

All length values are in mm.

Length	4907
Width	1953
Height	1379
Wheelbase	2990
Front track	1676
Rear track	1660
Kerb weight	1880
Fuel tank capacity (Litres)	91
Front tyre spec	245/35/20
Rear tyre spec	295/35/20
Peak power @8000rpm (kW)	486
Peak torque @6000rpm (Nm)	683
0-60 mph (Sec)	3.7
Top Speed (mph)	208
Mass distribution %Front	47

Table 1-Base Figures

(Ferrari, 2020)

2 What is Torque Vectoring

The grip available for each corner of a vehicle is not constant; thus if the same drive force is delivered to all wheels simultaneously (locked differential), this results in a net moment about the yaw axis of the vehicle. This yaw moment results in a force added to the front axle, effectively increasing the vehicle's yaw rate even if accelerating in a straight line. This results in decreased yaw authority for the driver, creating an unstable and unpredictable driving experience in changeable conditions. This same effect of mismatched wheel rotation velocity is matched when cornering with a locked differential, as the inside corners will rotate the same as the outside. This effectively increases the corner radius more than the desired yaw angle would correspond to. The solution to this is torque vectoring, where the total available torque is distributed to achieve the desired yaw rate at each moment in time, determined from corner radius, velocity, grip available and user settings.

The uneven distribution of torque creates a counter yaw moment, either to balance the induced yaw moment from uneven grip through the vehicle's centerline, or purposely induce a larger yaw moment for tighter corner radii.

Torque vectoring is also a great system to maximise corner grip to its peak slip angle, by reducing or increasing wheel speed and slip angle. This creates a maximise G-G diagram when plotted against a non-torque vectored vehicle. Keeping the wheel correctly rotated provides a more controllable limit for the vehicle under high stress, giving new drivers a better feeling for the limit and reducing the chances of overstepping the traction limit of each tyre. Regarding track performance, torque vectoring is an effective counter-understeer tool, whereby the torque sent to the front and rear axles can be changed, hence increasing or reducing the grip for lateral use rather than longitudinal. This again relates to the maximisation of the traction ellipse.

The mass distribution for a vehicle is essential for performance both laterally and longitudinally. The distribution sets the static loads for each corner, dictating the vertical load available. If the mass distribution is too extreme either way, the torque vectoring system will have to have a larger %torque range for each axle to compensate for the smaller drive force available. The mass distribution dictates the CoG position, which will have large effects on the vehicle's stability, linking to the dynamic index in figure x. The CoG position also dictates the inertial Conjugate, which is the effective point of rotation for the vehicle. This can either increase or decrease rear slip angle with relation to the front, dynamically altering the under/oversteer tendencies under yaw.

3 CoG longitudinal

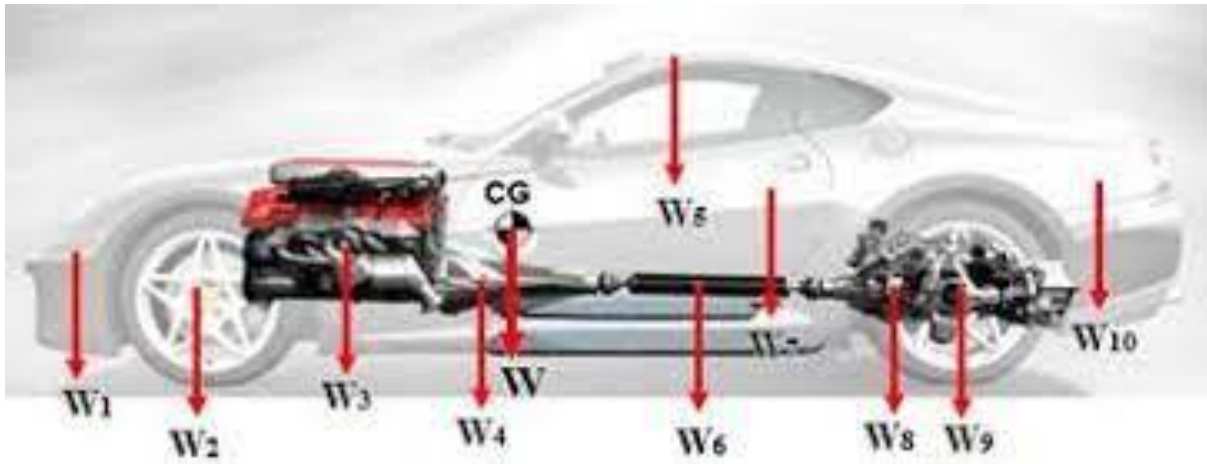


Figure 1-Weight Distribution

(The Car Tech, n.d.)

From table 1, the mass distribution can be applied to finding the corresponding CoG down the longitudinal axis of the Ferrari FF. When a vehicle is flat along a plane, the front and rear forces can be applied as moments about the CoG. If the frontal mass is 47%, this force must act along a longer distance to counter-act the larger rear force for equilibrium to be achieved. Therefore, the rear force must act along a shorter moment arm.

	Wheelbase %	%Mass	Distance From Front Axle	Distance From Rear Axle	Equilibrium Check
	0.47	883.6	1584.7	1405.3	0
	0.53	996.4	1584.7	1405.3	
Kerb Mass	1880				

Table 2-Calculation of CoG

Table 2 illustrates the calculated moments to solve for the distance of the CoG from the front and rear axle. It is observed that the distance from the front axle is larger than the rear, due to the smaller frontal load requiring a longer moment arm to achieve equilibrium.

	Wheelbase %	%Mass	Distance From Front Axle	Distance From Rear Axle	Equilibrium Check
	0.47	=B2*B5	=(B3*2990)	=B2*2990	=(B2*C3)-(B3*C2)
	0.53	=B5*B3	=(B3*2990)	=B2*2990	
Kerb Mass	1880				

Table 3-Calculation of CoG theory

Table 3 shows the theory of this calculation. The equilibrium check solves for the moments about the CoG which should equate to zero, which it does in table 2.

4 Z-Z Moment of Inertia

The moment of inertia in the Z-Z axis is essentially the vehicle's yaw moment. The higher this inertia value, the larger the resistance to yaw or turning. Since no reliable source for the exact masses and effective radii of these components were found, an informed estimate was made. For instance, the battery radii were taken from the hypotenuse Pythagoras correction, using the known distance of the front axle to CoG and front track with estimation from figure 2 location A.

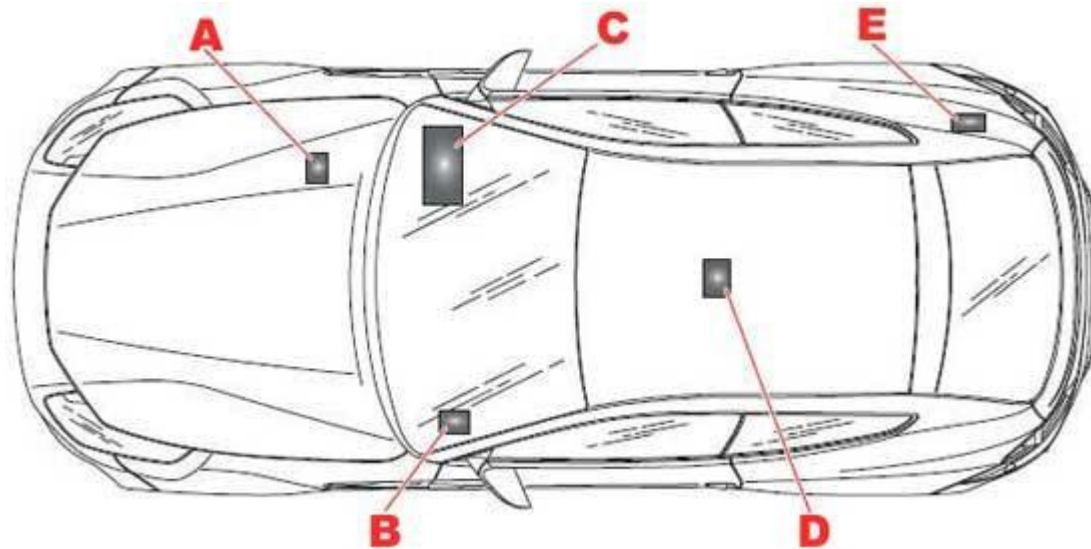


Figure 2-Ferrari FF Fuse Box Schematic

(Shemi, 2016)

It was important to ensure the mass without passengers was kept the same as the kerb mass sourced from Ferrari. This ensures maximum accuracy as one of the two terms in the inertia equation is reliable. The radii can not be certain to be 100% accurate, however the front and rear moments are balanced, so equilibrium is kept continuous. This moment match ensures that the longitudinal direction vectors of the masses from the CoG, when summed together, are correct, assuming the masses are also.

$$I_{zz} = \sum mr^2$$

Equation 1-Yaw inertia

Mass Split			Masses	Distance from CoG	Local Inertia	Moments	Inertia	
1880	Kerb	Engine	250	1.4	490	350	2951.7438	kgm ²
883.6	Front	PDU	60	1.67	167.334	100.2		
996.4	Rear	Gearbox	170	1.69	485.537	287.3		
		Front Passenger	170	0.5	42.5	85		
Frontal Moment	1178	Rear Passenger	170	0.75	95.625	127.5		
Rear Moment	1178	Full Fuel Tank	76.33	1.4	149.6068	106.862		
CoG Distance from Front Axle	1.58	Wheels and Brakes Front	120	1.79	384.492	214.8		
CoG Distance from Rear Axle	1.41	Wheels and Brakes Rear	120	1.65	326.7	198		
		Exhaust	20	2.06	84.872	41.2		
		Front Crash Structure	20	1.8	64.8	36		
		Rear Crash Structure	20	1.9	72.2	38		
		Battery	10	1.54	23.716	15.4		
		Lumped Mass	866.33					
		Front Body/Extra	476.4249	0.79	297.3367801	376.375671		
		Rear Body/Extras	537.2451	0.705	267.0242458	378.7577955		
		Mass without Passengers	1880					
		Mass With Passengers	2220	KG				
		Mass No Passengers	1880	KG				

Figure 3-Calculation of Z-Z Inertia and Balance of Moments

Unfortunately, there is no reference for this Inertia value, as companies usually conceal this information to the public. Methods such as Tri-Filo rig and 3 dimensional rigs can be used to find this resistance inertia. A Peugeot partner has a similar value of 2965kg.m², whereas a Chrysler voyager has 4652 kg.m². The FF's yaw inertia will contribute to an overall index of stability, judging how stable a vehicle is under steady-state dynamics.

Index of stability=Delta(Yaw Moment/Delta(Lat Acceleration)

Equation 2-Index of Stability

A high yaw inertia will result in a low delta moment due to higher resistance to yaw. This will lower the index of stability.

5 Dynamic index

A vehicles dynamic index is the numerical representation of its ability to overcome its rotational inertia. For instance, a formula vehicle should have an excellent ability to turn into a corner compared to a sedan. Therefore, a formula vehicle should have a low dynamic index, represented by:

$$\text{Dynamic Index} = \frac{I_{zz}}{mab}$$

Equation 3-Dynamic Index

Ways of a vehicle to have a low dynamic index are to limit the numerator (Rotational inertia) or decrease mass and/or wheelbase. This is not incorporating aerodynamic effects or suspension dynamics. The dynamic index also relates the stability of a vehicle under braking, as a low dynamic index implies a twitch rear under dynamic situations. The dynamic index is as well performs as a useful tool when inertia matching which will be discussed later. The FF should have a reasonable Dynamic Index as a high range performance vehicle.

The calculated Dynamic Index for the FF is 0.71, which is very low compared to most road cars. However, a low DI is expected from a cruiser, and due to a low expected CoG height, the FF should be relatively stable in transient conditions.

The calculated dynamic index could be low due to the low calculated yaw inertia, which is a rough estimate. The distances are based off top-down views of the vehicle and the masses are educated guesses based off other vehicles of this category. If the yaw inertia was to be calculated, it would be expected to much higher than presented in figure 3. This is one of the primary limitations.

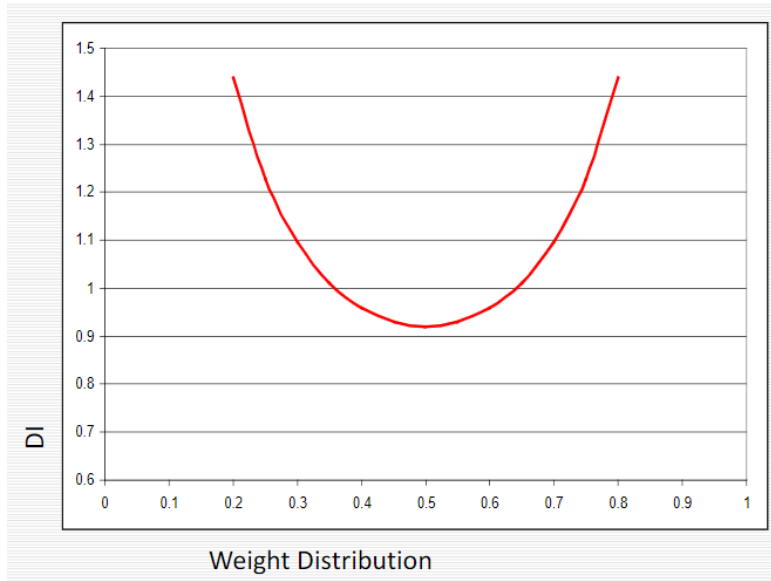


Figure 4-Di against Weight Distribution

Due to the FF's good weight distribution, it is close to its minimum possible Dynamic Index. However, a 50/50 split would result in a lower value, which would increase its responsiveness in transient applications. The dynamic index also significantly impacts the ease of driving due to its effect on yaw acceleration. The vehicle will have a higher limit of yaw acceleration when the dynamic index is lower, due to the low yaw inertia term. This links to the stability index in chapter 6. Although the yaw acceleration is at a higher maximum, the growth of this yaw rate is not affected. In contrast, lowering the wheelbase will increase max yaw acceleration and growth rate. This will make it more difficult for inexperienced drivers to control the vehicle due to non-linear growth of yaw acceleration.

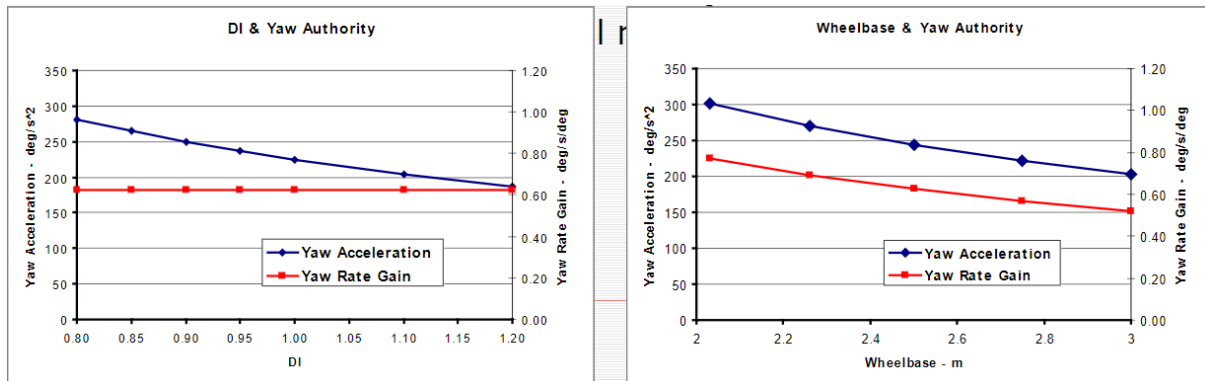


Figure 5-Dynamic index against yaw acceleration

6 Inertial Conjugate

$$Di = \frac{I_j}{\text{Distance from Rear Axle of CoG}}$$

Equation 4-Inertial Conjugate

The inertial Conjugate, or centre of percussion, is the ratio of the dynamic index and the distance of the CoG from the rear axle. It primarily affects additional force on the rear axle, hence larger or smaller slip angles because of additional lateral velocity application if the IC is not on the rear axle. If the Dynamic Index is above One, this implies the inertial Conjugate is in front of the rear axle. Hence, the velocity is applied at this point, with the additional inertia

effects adding to the rear tyre slip angle, and faster response time. If the IC is behind the rear axle, a net decrease in the rear tyre slip angle is observed, with the rotational force counter-acting the centrifugal force seeking to load the rear tyre. This IJ behind the rear axle is ideal for most road cars as it adds additional stability to the rear of the vehicle whilst sacrificing maximum yaw acceleration ability.

$$I_j = 0.71 * 1.41 = 1.001m$$

This IJ will increase rear slip angle under yaw, aiding with turn in.

7 Mechanical torque Path

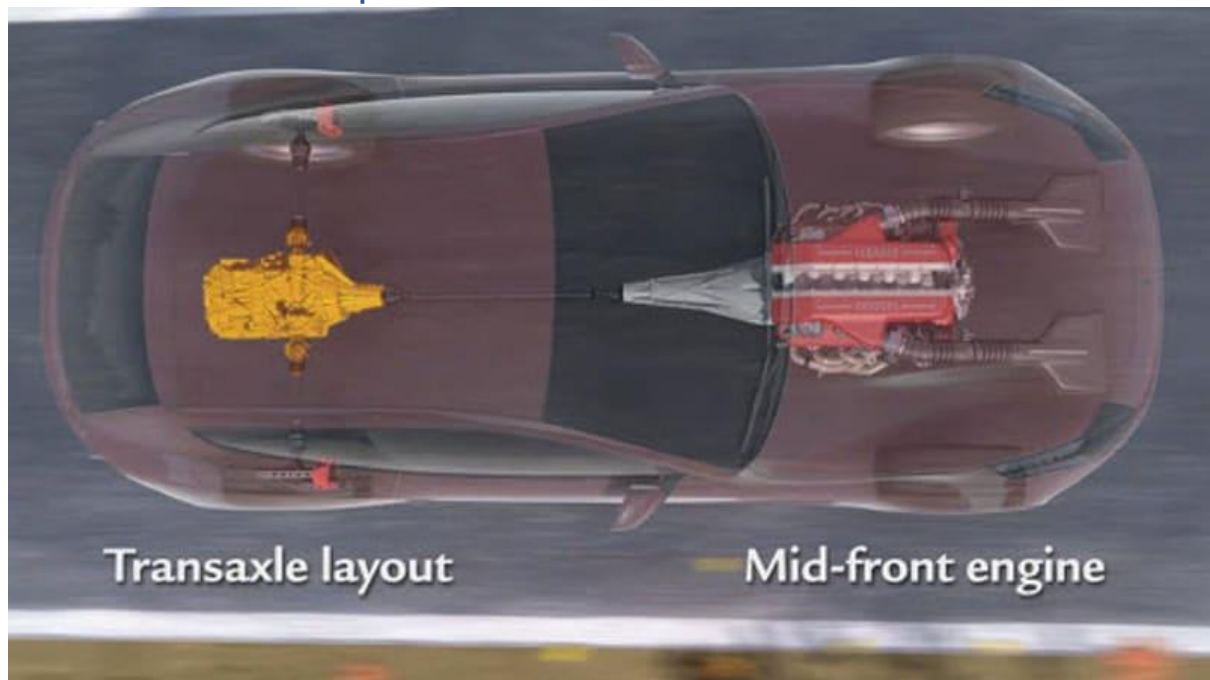


Figure 6-Ferrari FF drivetrain

(Autoblog, 2011)

Figure 6 shows the Ferrari FF drivetrain. The torque is applied through the front axle via the Power Transfer Unit, which uses a conical gear at the nose of the crankshaft. This gear is directly connected to the crank, with the torque transmitted to the two-speed helical gear set in the front gearbox. There lay two electronically controlled wet multiplate clutch packs between the wheel and the gearbox, one for each wheel. These clutch packs are the reason behind the FF's torque vectoring capability, as the slip of the clutch packs manipulates the torque transmitted to each wheel. The rear-wheel power is sent down the prop to an e-diff at the rear. This differential can control the torque supplied to each wheel from 0-100% such as the fronts. However, due to the rearwards weight distribution, the Ferrari system biases 80% of the torque to the rear axle.

8 Clutch Pack Torque Capacity



Figure 7-Ferrari FF Clutch pack

(AutoGuide, 2011)

From figure 7, an assumption of 4 clutchplates within the multi-plate clutch can be made. Applying the uniform clutch pressure equation, a clutch torque capacity and resultant pressure can be found.

$$T_c = n \frac{2}{3} \mu \frac{r_2^3 - r_1^3}{r_2^2 - r_1^2} F a$$

Equation 5-Clutch Torque Equation

There are some unknowns in this equation. Mu can be assumed to be 0.1 from (Forza, 2011) providing the clutch materials as carbon steel and (D. W. Gibson, 1989) specifying the expected Mu of this setup with wet carbon steel friction application. The number of friction surfaces is assumed to be 8, from figure 7, with unknown inner and outer diameters. These inner and outer radii are set to 75 and 50mm respectively, with ease of change if so needed. As the frontal wheel are driven by 20% max power, hence 20% max torque, the 683Nm can be split into 20% max torque for the front wheels. With a max torque of 136.6Nm transmitted, the torque capacity of the clutch pack must not be lower. The axial force was calculated to be 2690 newtons assuming no slip with the uniform pressure theory.

Mu	No Plates	r1	r2	Normal Force(n)	Total Friction Surfaces	Clutch Area
0.1	4	0.05	0.075	2696.052632	8	0.0098
Max Torque Capacity Uniform Pressure	Clutch pressure	20% Torque				
136.6	0.274756956	136.6				
Max Torque Capacity Uniform Wear	134.8026316					
Nominal Force Required No Slip for 20% Torque	2696.052632					
Power Transmittable Uniform Pressure	85784.8 W					
Power Transmittable Uniform Wear	84656.05263 W					
Power Loss	1128.747368 W					

Table 4-Clutch Pack Calculation

Through the differing calculations of transmittable lower power loss results in a 1.1kW loss in power at max torque @6000rpm. This loss is not much when compared to the base power of 85.8kW available. The maximum clutch pressure is also much less than the max axial pressure of 2.5Mpa (Stockinger, 2021) for these materials.

9 Longitudinal Acceleration Improvement

Mu at different surfaces			Standard Weight Transfer						Total Wheel Loads	
Tarmac	1		197.6829431 kg					2WD	Rear Axel	11713.95
Snow	0.4							4WD	Front Axel	6728.846
Ice	0.1								Rear Axle	11713.95
Accelerations	Tarmac	Snow	Ice							
2WD	6.230826421	2.492331	0.623082642 m/s/s							
4WD	9.81	3.924	0.981 m/s/s							
CoG Height	0.393									
Base Acceleration	0.8 G									

Table 5-Longitudinal Acceleration Improvement

For these calculations, a CoG height must be estimated due to the lack of information by Ferrari for this model. The resources needed to calculate the CoG were also not available. A height of 0.393m was the only dimension found related to a geometry like the Ferrari FF (Secrets, n.d.). The base acceleration is taken from the reported 0-60 time of the FF.

The basis of the calculations is taking longitudinal weight transfer into account for additional load transport between the two axles for a more accurate prediction of the force distribution between rear and front. As seen in table 5, the differing surfaces have a reduction factor, Mu, which will reduce the available tractive effort for the vehicle to use for acceleration. The total wheel loads under 2WD and 4WD are calculated by applying the static weight distribution and adding the load transfer to the rear axle whilst removing the same load from the front.

The observed acceleration decrease between the three surfaces for 2WD is drastic, with the most significant delta being tarmac to snow. Whilst the 2WD model is modest in terms of acceleration on the tarmac, the additional forward thrust produced with the front tyres results in a significant gain in acceleration. This gain could potentially be increased if the maximum torque to the front wheels was shared 60:40 rearwards, to make use of the available front grip more than the current 20% torque. The production of forward motion outweighs the reduced frontal load under acceleration by the otherwise non-useable wheels. The benefits of 4WD are during transient phases, where larger traction forces are required as shown by the performance increase in table 5. However, this model does have limitations, as the load transfer is a function of acceleration, so this should change with every traction surface. However, this is not possible as acceleration is the variable being calculated.

10 Yaw Rate Increase

Yaw Moment Calculator		
Hand to Road Wheel Ratio	15	
Mass	1880	KG
Yaw Inertia	2951.74383	KGM^2
Torque Split %Left	0.5	RATIO
Acceleration	1	M/S
Mu Left	0.1	
Mu Right	1	
Driving Force	1880	N
Left Wheel Driving Force	94	N
Right Wheel Driving Force	940	N
CoG from Rear Axle	1.405	M
Moment Force	846	N
Torque at CoG	1188.63	NM
Rotational Acceleration	0.40268738	RADS/S
Force on Front	750.066259	N
Stiffness	953.7	N/DEG
Degrees at the Wheels	0.7864803	DEG
Degrees at the Steering Wheel	11.7972044	DEG

Table 6-Yaw Rate Calculator

Table 6 shows the torque effect of differing levels of grip from one side to another. This is assuming rear-wheel drive only with the driving force calculated by the acceleration and mass. Table 6 clearly shows the large reduction in driving force on the left side of the vehicle due to the reduced Mu value, even with 50% torque vectoring. This difference in drive force creates a moment about the CoG, which in force creates a torque acting on the front wheels. This torque is acting like a linear force at the front wheels with the distance of the front axle to the CoG. The additional yaw force can be transformed into a steering angle related to a tyre stiffness. Given a hand to road wheel ratio, the effect the driver will feel can be calculated as an 11.8 degree turn to the left from the above simulation.

Yaw Moment Calculator		
Hand to Road Wheel Ratio	15	
Mass	1880	KG
Yaw Inertia	2951.74383	KGM^2
Torque Split %Left	0.9	RATIO
Acceleration	0.9	M/S
Mu Left	0.1	
Mu Right	0.9	
Driving Force	1692	N
Left Wheel Driving Force	152.28	N
Right Wheel Driving Force	152.28	N
CoG from Rear Axle	1.405	M
Moment Force	0	N
Torque at CoG	0	NM
Rotational Acceleration	0	RADS/S
Force on Front	0	N
Stiffness	867	N/DEG
Degrees at the Wheels	0	DEG
Degrees at the Steering Wheel	0	DEG

Table 7-Torque Vector Example

Table 7 shows how torque vectoring can be used for different grip level surfaces. The above simulation shows the left wheel having substantially less grip available to the right side. However, the %torque delivered to each wheel has been manipulated to share the available torque to have the same driving force on each wheel. This results in a net zero yaw moment; hence the vehicle will be stable and driving forwards with no angular acceleration. The FF's ability to transfer 0-100% torque gives it a considerable advantage in dramatic scenarios like above. If only 0-60% torque could be shared, there would still be a net torque acting on the vehicle.

11 Control Algorithms

For a vehicle to be able to apply this technology, all aspects of the vehicle's systems must be able to refer to, transfer and read data from each system. This results in the requirement for sensors within the vehicle to be read by the onboard computers. When taking torque vectoring into account, yaw is significant to measure from chapter 12; therefore, there must be a reading for the raw rate for the e-diff to make torque correction for each wheel to balance the drive forces. This raw rate sensor will work via gyroscopic effects and accelerometers about the vehicles pivot axis to calculate the net difference of slip angle to optimise grip for its 5 main settings.

For optimum traction, the slip angle must be optimised for each scenario, thus, the wheel speed needs to be monitored via a wheel speed sensor, to compare to vehicle speed using GPS and calculate the longitudinal slip ratio. These wheel speed sensors are also critical for the function of ABS. They primary work around electromagnets and the Hall effect (Apec, 2021).

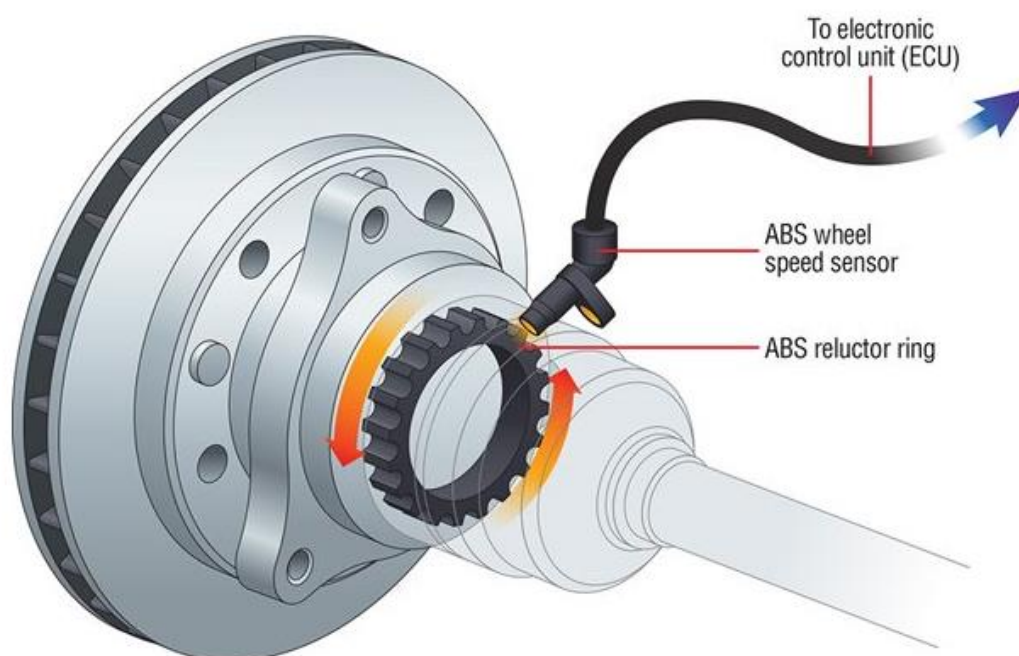


Figure 8-Hall Sensor

(Apec, 2021)

The individual normal vehicle loads are critical for calculating available drive force at each wheel; hence a load sensor must be used at each wheel. There are different types of load sensors, such as suspension compression sensors relating to the known stiffness and damping values to calculate load. However, a load cell is primarily used for more accurate and faster dynamic calculations (Abdolmaleki, 2018).

Torque transducers are used to measure the torsional input to a shaft into an electrical output based on shaft characteristics. These torque sensors are essential to compare the demanded torque at each wheel to the torque applied; this ensures the smallest error margins are met when torque match each corner (CVEL, 2018).

With a calculated angular acceleration from the yaw sensor, and a wheel rate to handwheel rate value, the effective steering wheel rate can be found, corresponding to net lock needed to be corrected by the torque vectoring unit. A steering wheel angle sensor can be used along the steering rack to measure the rotation of the shaft, hence comparison between vehicle yaw rate and steering wheel rotation.

Inertial measurement units can be used to predict the yaw inertia of the vehicle better, hence its dynamic index. With a known D_i , the various drive forces can be balanced by the e-diff to correct for instabilities under braking and share the braking forces better for corners with more available grip.

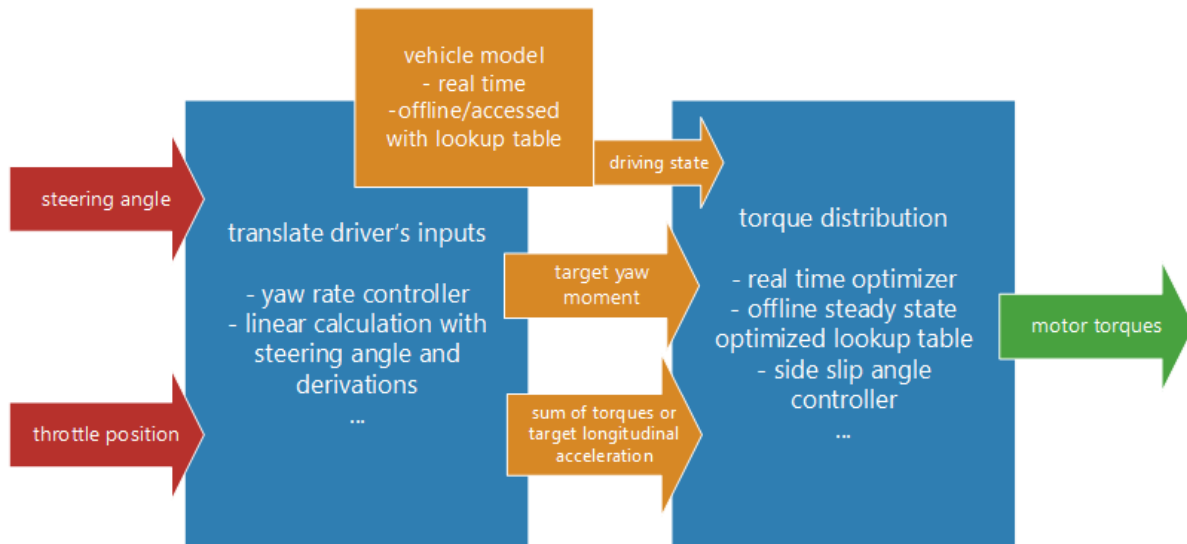


Figure 9-Torque Vector Schematic

Figure 9 shows typical calculations required for an accurate torque distribution, with additional sensors like throttle position sensors, brake position sensors, and brake pressure sensors (required for ABS).

All these sensors with multiple ECU's should provide a smooth and accurate torque distribution system for optimum yaw angle. Many tests have conducted comparisons with and without torque vectoring, such as (Andrea Mangia, 2021).

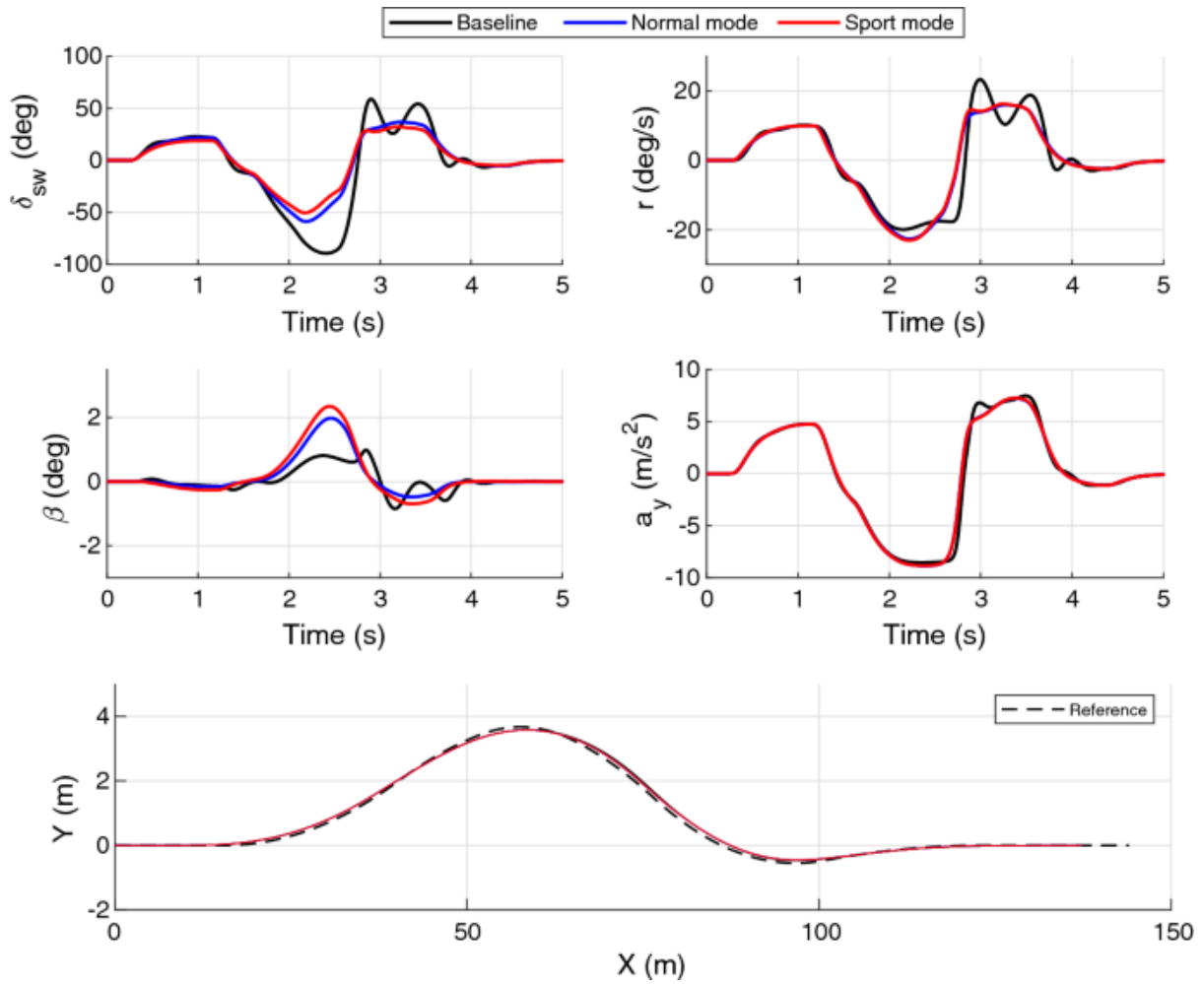


Figure 10-Torque Vectoring effects on yaw

Figure 10 shows the various variables during a double lane change. As expected, the steering wheel rate is reduced with torque vectoring and is more predictable without the need for a correction like the baseline. The yaw rate, r , is also more stable, with a higher max rate achieved with substantially less steering lock. It appears the baseline turned the wheel past the peak slip angle, thus adding more angle reduced available lateral grip. The yaw angle is also increased and smoother than baseline, with angular acceleration relatively similar to smoother peaks. This shows that torque vectoring can be used for driver aid as well as pure grip maximising.

12 Conclusion

To conclude, the FF's dynamic characteristics of the low dynamic index and a well placed inertial conjugate results in a good balance between a lively rear under load but one which is stable around flowing corners. It is a vehicle with a significant performance capability in the right hands, with the 4WD and torque vectoring system being programable for given scenarios such as smooth ride or maximum grip. The torque-vectoring ability makes this Ferrari very controllable under changeable conditions, with the ability to distribute torque to each wheel to benefit performance for the desired way rate. As seen in chapter 10, with the resultant yaw moment added under un-even grip levels, the FF's torque vectoring ability allows it to create a counter-moment about the yaw axis to provide an equal and opposite frontal force, meaning no correction is required on the wheel. This car was the pioneer of a new range of 4WD torque vectoring vehicles and inspired many other auto manufacturers to follow suit and develop systems to improve driver aid and add performance when possible. The FF seems to incorporate a low dynamic index and long wheelbase to provide both a snappy rear end but controllable under large corner radii; this appears to be the aim of the FF, which can be pushed if wanted but performs excellently under cruising conditions. The 50% mass reduction of the PTU also helps with the mass reduction, and the placement of components over each axle will reduce the distance to the CoG, hence lowering yaw inertia. The mass distribution encourages greater traction with the 80:20 torque split to the rear to use the more significant traction force available at the rear, reducing 0-60 times and increasing performance. Overall, the FF provides a solid chassis with good handling characteristics, which provides the driver with predictable handling with minimal extra steering wheel movements.

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