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Advanced Vehicle Dynamics
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AWD Torque Vectoring

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1 Abstract

This report explains and analyses the All-Wheel Drive system Ferrari unveiled in their FF model. The aim is to understand the effects the system has on vehicle dynamics and potential control strategies. In short, Ferrari has created a four-wheel drive car by using a power transfer unit, which incorporates torque vectoring, in the front of a rear wheel-drive car. They unveiled the FF in March of 2011 (Cropley, 2011). The report investigates at the rotational inertia and the dynamic index alongside the torque vectoring clutch pack specification analysing its effect on the dynamics.

2 Base Vehicle Data

For this investigation, many items of data were needed. These were sourced and the essential information was extracted into Table 2-1. The full information set can be seen in Appendix A. Other information was also sourced throughout this project and referenced in their various locations, such as gear ratios in Section 6.

Table 2-1 Base Vehicle Data (Ferrari, 2011)

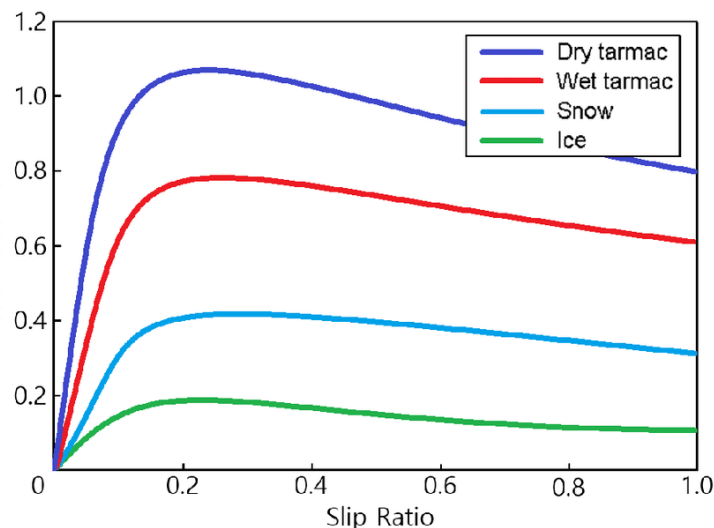
Vehicle Data	
Wheelbase	2.99 m
Track Front	1.676 m
Mass	1880 kg
Distribution (F/R)	47/53
Max Torque	683 Nm
RPM @ Max Torque	6000 RPM
Max Torque to Front	20 %
	136.6 Nm
Max Torque to Each Clutch Pack	10 %
	68.3 Nm

3 Generic Torque Vectoring Systems

Torque vectoring allows more torque to be delivered to one wheel than the other on the same axle, allowing each wheel to have different velocities. This change in wheel speed helps a vehicle's dynamic performance when the yaw rate of the vehicle does not match the input from the driver.

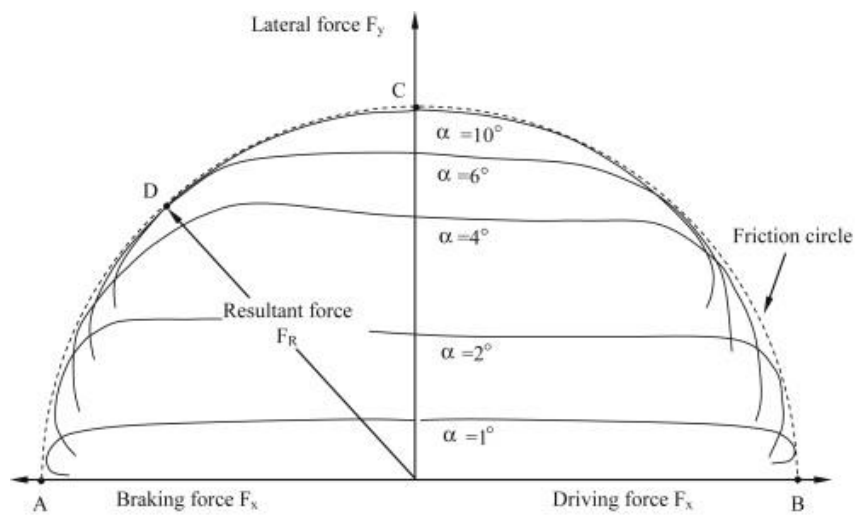
Torque vectoring also allows for the optimal maximum available torque to be given to each wheel even when they are cornering or are on different surfaces. This means the maximum drive force can be applied at all turning radii and multiple coefficients of friction. This allows for improved efficiency of the torque delivery during cornering by improving the difference in torque at each wheel for maximum benefit (Auto Ferrari, 2011).

For a tyre to produce a driving force it must produce a slip ratio. The ratio of slip effects the value of the driving force the tyre produces. This means that the vehicle can monitor and adjust the torque at each wheel to achieve the optimum slip ratio so that the maximum drive force can be produced. This ratio is approximately 15% for all surface conditions, see Graph 3-1.



Graph 3-1 (Park, et al., 2015) Drive Force Coefficient against Slip Ratio for multiple μ values

Torque vectoring can also optimise performance by monitoring lateral and longitudinal G forces. This information produces a friction ellipse for the vehicle, see Graph 3-2. The system can then be used to ensure the car remains inside this ellipse under normal conditions as well as under racing conditions the system can ensure the vehicle is as close to being on the limit of the ellipse at all points under acceleration. By controlling the contact patch of the tyre allows the optimum control of the vehicle.



Graph 3-2 (Blundell & Harty, 2015) Friction Ellipse

4 Lumped Mass Model

A lumped mass model is a model in which the car is broken down into a few masses, each with its own centre of gravity. This allows for the rotational inertia and dynamic index to be calculated. The more accurate the model, the more accurate the results of the calculations will be.

The first part of building a lumped mass model is to identify the longitudinal location of the centre of gravity for the entire vehicle, distance a in Figure 4-1.

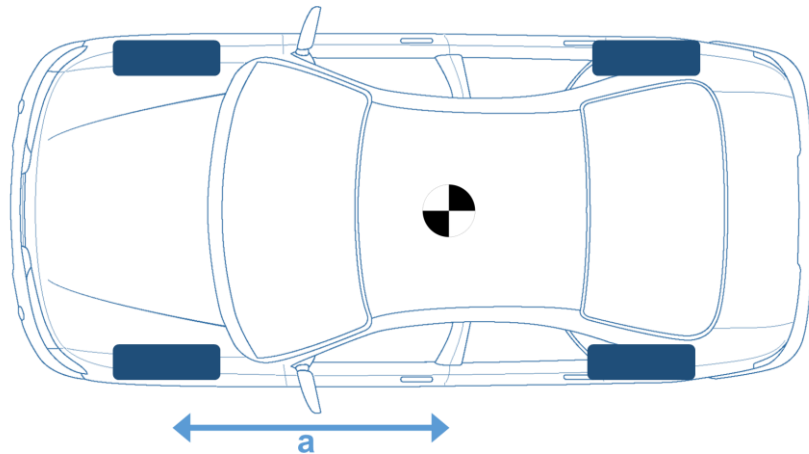


Figure 4-1 Longitudinal Centre of Gravity

The distance from the front axle to the centre of gravity was calculated using Equation 4-1.

$$a = \text{wheelbase} \times \% \text{ of Rear Mass distribution}$$

Equation 4-1 Longitudinal Centre of Gravity

From Table 2-1, the car's wheelbase is 2.99m and the percentage of rear mass distribution is 53%. This means that the distance from the front axle to the centre of gravity, a , is 1.58m, Equation 4-2.

$$a = 2.99 \times 0.53$$

Equation 4-2

To complete the lumped mass model the vehicles total mass had to be split into smaller masses. For this model, the masses considered were the mass of the engine and power transfer unit, PTU, the transaxle and finally a uniform car body mass. The engine and PTU were assumed to be acting just behind the front wheels, the transaxles mass was assumed to be acting directly over the rear wheels and the body was assumed to have half of its mass over the front wheels and the other half over the rear wheels. These mass locations are all depicted in see Figure 4-2.

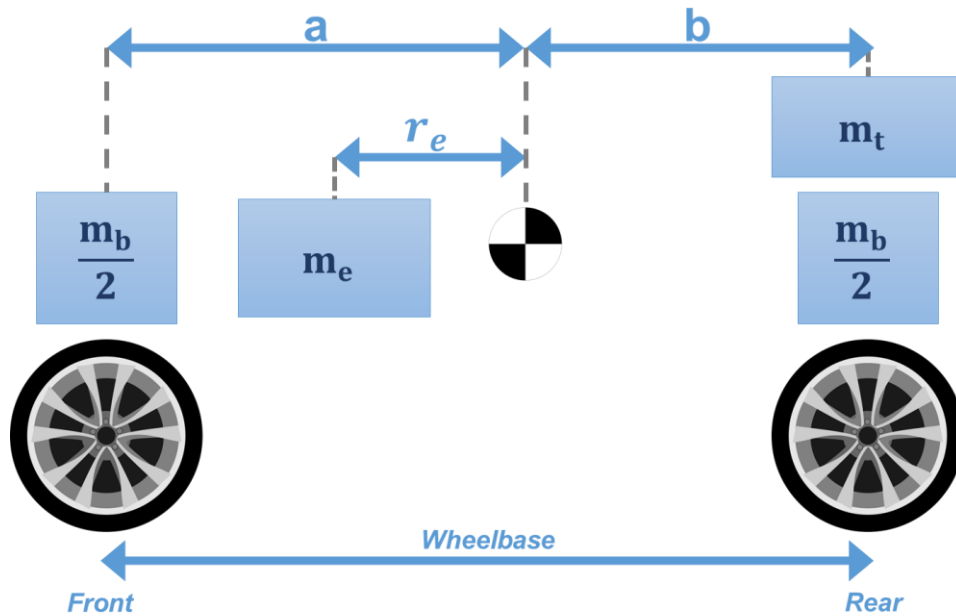


Figure 4-2 Lumped Mass Model

The mass of the engine is 225kg (Wan, 2002) and the PTU is 40kg (Edmunds, 2011). This means the total mass of the engine and PTU is 265kg. The transaxle was assumed to have a mass of 241kg which meant the uniform body will have a mass of 1374kg. This distribution of mass across the entire car must be the same as in Table 2-1, with 47% at the front. From this, moments were taken around the rear wheel to calculate the location of the engine mass, Equation 4-3. This was then recorded with the rest of the data in table 4-1.

$$r_e = \frac{b(m_t + \frac{m_b}{2}) + a(1880 \times 0.47) - a\frac{m_b}{2} - b(1880 \times 0.53)}{m_e}$$

Equation 4-3 Location of Engine

Table 4-1 Lumped Mass Model

Mass (kg)				Distance (m)		
Engine	Transaxle	Body Front	Body Rear	r_e	a	b
265	241	687	687	0.8129	1.585	1.405

4.1 Moment of Inertia

The lumped mass model was used to calculate the moment of inertia of the vehicle. This was calculated using Equation 4-4 for each mass, Table 4-2.

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

Equation 4-4 Moment of Inertia about the ZZ axis

The sum of the inertia of each mass was calculated and this produced the total moment of inertia for the vehicle about the Z-Z axis, 3733.05kgm².

Table 4-2 Moment of Inertia

	Mass (kg)	Distance to CoG (m)	Moment of Inertia ZZ (kgm ²)
Engine	265	0.81294	175.1309326
Transaxle	241	1.4053	475.9432097
Body Front	687	1.5847	1725.2453
Body Rear	687	1.4053	1356.734378
Total			3733.05382

This calculation assumes there are no occupants in the car, however, these masses could affect the calculation. After inspection of where the passengers are located, it is unlikely they would have a noticeable effect on the moment of inertia. This is due to how small the distance from the occupants to the centre of gravity is, the difference is probably less than 50kgm².

4.2 Position of the Inertial Conjugate

The position of the inertial conjugate is also known as the centre of percussion. This is the location at which a body will rotate when a force is applied. For a vehicle, this is the lateral force that is caused by the tyres when a steering angle is applied. This distance from the centre of gravity can be seen in Figure 4-3, c.

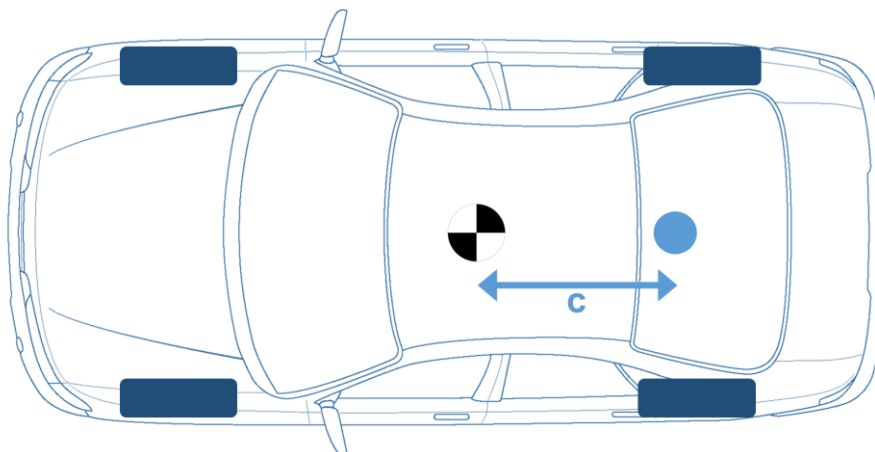


Figure 4-3 Inertial Conjugate

The distance c is calculated using Equation 4-5, where the inertia used is found in Table 4-2, the mass is 1880kg and a is the distance found in section 4, 1.59m. This resulted in a

distance of 1.25m which means that the location of the inertial conjugate is just in front of the rear wheels.

$$c = \frac{I_{zz}}{ma}$$

Equation 4-5 Inertial Conjugate

4.3 Dynamic Index

The dynamic index describes the location of the point of the inertial conjugate in relation to the rear wheels. The value is calculated by dividing the distance of the inertial conjugate to the centre of gravity by the distance of the rear wheels to the centre of gravity. This can be written as equation 4-6.

$$DI = \frac{c}{b} = \frac{I_{zz}}{mab}$$

Equation 4-6 Dynamic Index

The result of this calculation gives a dynamic index, DI, of 0.89 for the Ferrari. This confirms that the point of the inertial conjugate is in front of the rear wheels. A DI of 1 means that the point lies on the rear wheels whereas a DI greater than one indicates it is behind the rear wheels. If it is behind the rear wheels this means the vehicle is very stable but can be reluctant to 'turn in' resulting in the rear wheels moving in when turning reducing the slip angle. Whereas a DI between 0.8 and 1 is stable but is easy to 'turn in' which means the rear wheels move out when turning increasing the slip angle. When the DI is less than 0.8 this causes a very unstable vehicle which will 'turn in' easily but is a challenge to control the balance of the vehicle.

4.4 Base Vehicle Specification

The dynamic index of the vehicle means that the car will be stable but still be agile to give it a sports car feel. Due to being able to seat four occupants, the car could have easily had a higher dynamic index. The effect of the dynamic index would have been considered by Ferrari in their design process to ensure the car feels and handles like a Ferrari. They would have investigated locations for the engine and transaxle as well as where the occupants are located.

The mass distribution is also more forwards than other Ferraris at the time, such as the 458 which has a 42% front distribution (Car Folio, 2020). This is due to the increase in mass caused by the PTU. However, it still managed to keep a slightly rear-biased mass distribution like many other Ferraris. The mass of the vehicle is much heavier than the other vehicles of the time such as the F12 which had a mass of 1525kg and the 488 GTB, which was 500kg lighter at 1370kg (Car Folio, 2020).

Most cars have a dynamic index of approximately 0.9-0.92, including the Lotus Elan, the Mazda MX5 and the BMW M5. This could indicate that since most manufacturers of sports vehicles aim for this DI that it defines the best compromise of both agility and controllability of the vehicles.

The mass distribution is related to the dynamic index for constant yaw inertia. The closer the vehicle is to 50% front mass distribution the lower the dynamic index. This is one reason why manufacturers aim for a 50% split in mass for sports cars as this gives the car more agile driving experience.

5 Mechanical Torque Path

5.1 Traditional Four-Wheel Drive

Typically in a four-wheel drive, 4WD, vehicle the engine torque is passed through a gearbox to a transfer box and if 4WD is engaged the torque then travels along a driveshaft to the rear differential and another driveshaft to the front differential. To account for the speed difference front to back the transfer case often has a viscous coupling or a type of geared system. The torque is then split at the differential to each wheel at the front and the same at the rear (Nice, 2001).

This type of system is very heavy causing the vehicle to have a higher rotational inertia. This means that for most sports cars this is not a viable option as the benefits of increase drive force at low velocities is outweighed by the effect the increase in mass causes to the vehicle's performance.

5.2 Ferrari's Four-Wheel Drive

Ferrari has managed to develop a type of four-wheel drive system that is not like the traditional style. The torque from the engine travels through the gearbox and down the driveshaft to the rear transaxle where the torque is split between both rear wheels. This is the same as any conventional front-engine rear-wheel drive car (Wan, 2016). To allow the front wheels to also be driven, torque is taken from the front of the crankshaft and transmitted through a power transfer unit, PTU. This PTU consist of two gear pairs and two clutch packs (Auto Ferrari, 2011). Depending on the main gearbox gear, A PTU gear pair transfer the torque from the engine to each clutch pack which is assigned for each front wheel. Unlike the rear wheels, only 20% of the engine torque is transferred from the engine to the PTU (Edmunds, 2011). So only 10% at each front wheel. The use of clutches also allows the Ferrari to have front-wheel torque vectoring as well as four-wheel drive.

This allowed for a huge saving in mass compared to other four-wheel drive systems due to not having a transfer case and a driveshaft to the front differential. This system is claimed to be half the weight of a conventional four-wheel drive (Auto Ferrari, 2011). This system also allows for a lower centre of gravity as the engine does not need to be high enough for a driveshaft to pass under it. However, to achieve this system the engine had to be mounted behind the front wheels. This would normally cause a loss in occupant space however Ferrari still managed to allow the car to be a four-seater. Mounting the engine behind the front axle is common with Ferrari and some of its rear-wheel drive cars, such as the 550 and F12.

6 Clutch Pack

The clutch pack on each driveshaft consists of a multi-plate clutch. The following calculations into the specification of the clutch pack consider the clutches individually.

To calculate the Torque capacity the gear ratios of the gears in the power transfer unit were calculated. The first gear in the PTU is 6% taller than second gear in the main gearbox and the second gear in the PTU is 6% taller than the main gearbox's fourth gear. The gear ratios were collated into a table the overall ratio was calculated using a final drive ratio of 3.88 (Automobile Catalog, 2011). From these ratios, the PTU gear ratios could be calculated by multiplying second and fourth gears overall ratio by 1.06. This gave the ratio for first and second gear in the PTU, see table 6-1.

Table 6-1 Road Wheel Speeds at Max Torque

Gear	Ratio	Overall Ratio	Rear Road Wheel Speed @ Max Torque (RPM)
1	3.397	13.18036	455.2227708
2	2.186	8.48168	707.4070231
3	1.626	6.30888	951.0404382
4	1.285	4.9858	1203.417706
5	1.029	3.99252	1502.810255
6	0.839	3.25532	1843.136773
7	0.634	2.45992	2439.103711
<i>front gearbox</i>			
1		8.001584906	749.8514445
2		4.703584906	1275.622769

The rear-wheel speed and front wheel speed were calculated when the vehicle exhibits maximum torque. Equation 6-1 was used and the engine RPM at maximum torque is 6000RPM.

$$Wheel\ RPM = \frac{Engine\ RPM}{Overall\ Ratio}$$

Equation 6-1 Wheel angular velocity

To calculate the Torque at each clutch pack the amount of torque that is transmitted from the engine to the front wheels must be calculated. As mentioned before, 10% of the engine torque is transmitted to each wheel. Since the engine has a peak torque of 683NM only 68.3Nm is transferred from the engine towards each wheel. However, this is not the torque at each clutch pack as the torque is increased as it passes through the gear ratios. The

torque at each clutch pack when maximum engine torque occurs was calculated using equation 6-2, see table 6-2.

$$T_{clutch} = T_{from\ engine} \times Overall\ Ratio$$

Equation 6-2 Torque across Clutch Pack

The power dissipated across the clutch pack was then calculated. Since power is torque multiplied by the angular velocity, the maximum power dissipated would be the maximum difference in angular velocity of the input and output shaft of the clutch pack. The input shaft is the speed calculated at maximum torque in table 6-1. Whereas the output shaft speed is the actual wheel speed which is the same as the rear speed. The maximum difference occurs when the PTU and main gearbox are in first gear. Also when the PTU is in second gear the maximum difference is when the main gearbox is in third. The maximum power dissipated across each pack is 16.86kW in first gear, see table 6-2.

Table 6-2 Torque and Power Dissipated Across each Clutch Pack

Gear	Difference in Angular Velocity @ 6000rpm		Torque at each Clutch Pack	Power Dissipated across Each Clutch Pack	
1	294.628674 RPM	30.85344256 rad/sec	546.5082491 Nm	16861.6609 W	16.8616609 kW
2	324.58233 RPM	33.99018216 rad/sec	321.2548491 Nm	10919.5108 W	10.9195108 kW

The Maximum torque capacity of each clutch pack is the maximum torque that the clutch pack will receive and this is 546.51Nm. An approximate axial load can be calculated from the peak torque capacity using Equation 6-3.

$$F_{axial} = \frac{T}{n \mu r_{mean}}$$

Equation 6-3 Axial Load on the Clutch (Thomas, 2020)

The axial force calculated was 6831N. This was assuming the mean radius was 0.2m, the number of clutch plates was 4 and the coefficient of friction was 0.1.

7 Longitudinal Acceleration

To investigate the potential longitudinal acceleration gains, by having this four-wheel drive system fitted, the acceleration of just rear-wheel drive needs to be calculated first. The maximum rear drive force across both wheels was calculated using Equation 7-1, where the radius of the rear wheel is 0.357m. This gave a maximum drive force of 25216N, this means that the limiting factor in acceleration will most likely always be the available grip.

$$F_{\max drive} = \frac{T_{\max}}{r_{\text{wheel}}}$$

Equation 7-1 Max Drive Force across an axle

The actual drive force was calculated at a variety of surface conditions to see how the acceleration improvement is effected across multiple surfaces. Equation 7-2 was used to calculate the rear drive force.

$$F_{\max grip} = \mu F_{\text{normal}} = \mu \times (1880 \times 0.53 \times 9.81)$$

Equation 7-2 Max Force available due to grip rear

The acceleration for each surface condition was calculated using Equation 7-3.

$$a = \frac{F}{m}$$

Equation 7-3 Acceleration

To investigate the improvement the same calculations needed to be carried out but to include the front-wheel drive force. The maximum front drive force was calculated using Equation 7-1 where the torque was twice the clutch torque capacity calculated in section 6 and the wheel radius was 0.341m. This force was 3205N, which meant that this force would be a likely limiting factor at higher μ surfaces than the grip available. The available force was calculated using Equation 7-4 and compared to the maximum drive force, see Appendix B. If the available force was less than the driving force it was used in the calculation however when grip was not a limiting factor the maximum drive force was used.

$$F_{\max grip} = \mu F_{\text{normal}} = \mu \times (1880 \times 0.47 \times 9.81)$$

Equation 7-4 Max Force available due to grip front

The front and rear drive forces were then added together, and Equation 7-3 was used to calculate the new acceleration. The difference in acceleration was investigated to see how much quicker the vehicle could accelerate with the four-wheel drive system, see Table 7-1.

Table 7-1 Longitudinal Acceleration

Mu	RWD		AWD			Improvement
	Rear Drive Force (N)	Acceleration (m/s/s)	Front Drive Force (N)	Total Drive Force (N)	Acceleration (m/s/s)	Acceleration (m/s/s)
0.2	1954.9368	1.03986	1733.6232	3688.56	1.962	0.92214
0.6	5864.8104	3.11958	3205.326974	9070.137374	4.824541156	1.704961156
0.9	8797.2156	4.67937	3205.326974	12002.54257	6.384331156	1.704961156

The coefficient of friction values, μ , that were selected were 0.2, 0.6 and 0.9. These values represent road surfaces of snow, wet asphalt and dry asphalt (Engineering ToolBox, 2011).

The front wheels were only grip limited for the first road surface whereas the other surface they were power limited. The improvement in acceleration was significant for all of the surfaces however it was more significant at low coefficients of friction. In snow, the car has the potential to accelerate nearly twice as fast with the four-wheel drive system. This means the system has huge benefits in conditions where there is minimal grip available.

8 Yaw Rate Moment

The potential yaw rate increase that the system could offer would occur when one clutch pack is engaged to give maximum torque to one front wheel, as demonstrated with the green arrow in Figure 8-1.

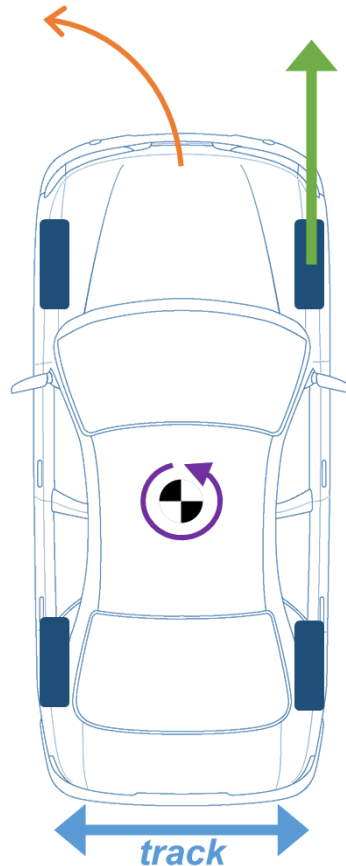


Figure 8-1 Yaw Rate Diagram

To calculate the yaw rate moment, the purple arrow in Figure 8-1, the maximum drive force at one wheel is calculated using Equation 8-1. Where the torque is the clutch pack capacity and r is the front wheel radius, 0.341m.

$$F = \frac{T}{r}$$

Equation 8-1 Max Drive Force for 1 front wheel

The maximum drive force, 1602.7N, is used in Equation 8-2 to calculate the yaw rate moment increase.

$$Yaw\ Rate\ Moment = F \times \frac{Track}{2}$$

Equation 8-2 Yaw Rate Moment

The potential increase in yaw rate moment for the car with torque vectoring is 1343Nm. The increase in rotational acceleration is then calculated using Equation 8-3, where the torque is the yaw rate moment and I_{zz} is the vehicles rotational inertia calculated in Section 4.1, 3733kgm².

$$\alpha = \frac{T}{I_{zz}}$$

Equation 8-3 Rotational Acceleration

9 Torque Vectoring effect on Vehicle Dynamics

Torque vectoring affects many dynamic responses of a vehicle. For this to happen the vehicle must contain many sensors to understand the vehicles 'live' behaviour. Some of the types of sensors used can be seen in Table 9-1.

Table 9-1 Sensors

Each Wheel Speed	Handwheel Angle
Vehicle Speed	Brake Application
Rotational Yaw Rate	Throttle Request
Acceleration in the X axis	Driving Mode (sport/comfort)
Acceleration in the Y axis	Tyre Slip angle
Acceleration in the Z axis	Wheel Loads

By monitoring the car's behaviour, the system can manipulate the yaw damping when the car is accelerating. This will in turn, improve the vehicle handling response. The sensors will also be able to deduce the maximum grip available and ensure that under full acceleration the car is utilising the available grip on both left and right front wheels. This allows the car to use its available torque as efficiently as possible. This not only improves straight-line performances but also means when a vehicle is travelling around the corner more torque can be given to the outside wheel as it will be rotating faster. The system will also help when a vehicle carries out on-limit manoeuvres, which would normally cause instability. This could be reduced if the vehicle reduced torque to the outside front wheel if the cars yaw rate is much larger than the requested yaw rate from the driver. This would help in allowing the car to carry on in a straight path without spinning out of control.

The system can also work alongside other vehicle dynamic control systems such as anti-lock brake systems, ABS. The system will consider which control systems are of higher safety and hierarchy so will prioritise those systems if more than one system is trying to work independently at the same time, such as prioritising ABS over torque vectoring. For example, the clutches of the torque vectoring system can either lock so the front wheels travel at the same speed or they can open completely to allow both wheels to be controlled independently, which is what would be needed when the ABS is effecting the vehicle dynamics.

10 Summary

Overall, the four-wheel drive system on the Ferrari FF is a design which has a niche market. The system only benefits the car under 'sporting' driving conditions. This means if the car is only to be used as a 'show' car the system is unbeneficial however, very expensive.

However, for a sporting driver or a driver which lives in an area prone to snow this vehicle's system will be beneficial as it can drastically improve low-velocity performance. However, at high velocities, the system is not useful at all as once the car is above fourth gear the system is disengaged.

Initially, the idea was desirable as in 2011 Ferrari sold 250 cars and then the following year, they sold 480 cars. However, this was their peak sales and in 2013 sales dropped to 208 and by 2018 only 3 cars were sold (Demandt, 2019). This demonstrates that there was a buyer initially for the car however once the gimmick of the four-wheel drive system wore off not many people were interested in buying them. Initially, it was believed to feel nimble and have an excellent ride (Bovingdon, 2011) however in more recent years it was said to have a 'raw' driving feel for a GT car (AUTOCAR, 2019). However, from the initial verdict of the four-wheel drive system was negative, as it was said to be clumsy (Bovingdon, 2011). This is due to the car system improving traction in a corner sometimes yet others it reduces drive which reduces the yaw rate and the car starts to understeer. This unpredictability was not favoured from the start. Since the four-wheel drive system is quite a big selling point for the vehicle and yet it was the main component to be criticised. It can be deduced that the system's algorithms will need a lot of work before the system will be accepted as sports car enhancement. However, for the average daily car which is not being driven sportingly, its torque vectoring could be a benefit to the safety and controllability of a non-Ferrari vehicle.

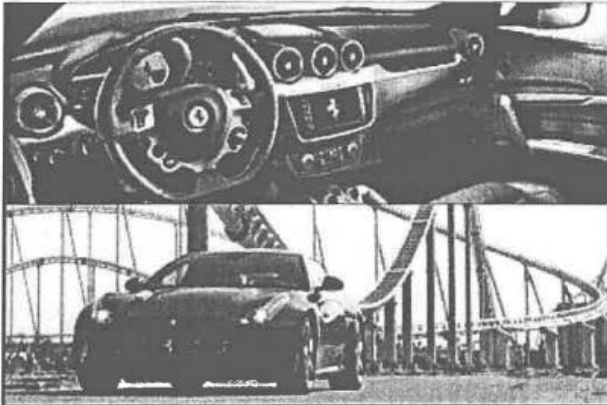
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Appendices

Appendix A – Vehicle Specification (Ferrari, 2011)

Technical specifications



Dimensions and Weight

Length

193.2 in (4907 mm)

Width

76.9 in (1953 mm)

Height

54.3 in (1379 mm)

Wheelbase

117.7 in (2990 mm)

Front track

66.0 in (1676 mm)

Rear track

65.4 in (1660 mm)

Kerb weight*

4145 lb (1880 kg)

Dry weight*

3946 lb (1790 kg)

Weight distribution

47% front - 53% rear

Boot capacity

15.9 cu ft (450 l)

Fuel tank capacity

24 US gal - 20 UK gal (91 litri)

Tyres and rims

Front

245/35 ZR 20 8.5 J x 20"

Rear

295/35 ZR 20 10.5 J x 20"

Rear Winter

285/35 ZR20 10.5 J x 20"

Brakes

Front

15.7 x 1.5 in (398 mm x 38 mm)

Rear

14.2 x 1.26 in (360 mm x 32 mm)

Engine

Type

V12 - 65°

Total displacement

382.13 cu in (6262 cc)

Bore and Stroke

3.70 x 2.96 in (94 mm x 75,2 mm)

Maximum Power**

486 kW - 660 CV at 8.000 rpm

Maximum Torque

683 Nm - 504 lbft(70 kg)m at 6.000 rpm

Specific Output

77 kW/litre - 105 CV/litre

Maximum revs per minute

8000 rpm

Compression ratio

12.3:1

Performances

Maximum speed

208 mph (335 km/h)

0-62 mph (0-100 km/h)

3.7 sec

0-124 mph (0-200 km/h)

11 sec

0-62 mph (0-100 km/h)

114.8 ft (35 m)

Dry weight/power ratio

2.7 kg/CV

Fuel consumption / CO2 emissions (ECE+EUDC)***

Fuel consumption (european market version)

15.4 l/100km

CO2 Emission (european market version)

360 g/km

Transmission and gearbox

4RM

7-speed F1 Dual-Clutch gearbox

Electronic controls

4RM control (Integration of F1-Trac, Ediff and PTU controls)

ABS/EBD

ESC

Suspensions

SCM3 (third generation of magnetorheological suspension control)

Note

*

European market version

**

Engine power is expressed in kW, in accordance with the International System of Units (SI) and in CV for reasons of homogeneity. The horse power (hp) can be calculated as follows: 1

Combined cycle with HELE system (ECE+EUDC)

Appendix B – Front Drive Force ‘IF’ Statement

$$= IF(((1880 \times 0.47 \times 9.81) \times \mu) > F_{\max drive}, F_{\max drive}, ((1880 \times 0.47 \times 9.81) \times \mu))$$

Appendix C – Excel Spreadsheet of Calculations

Data Sheet

	A	B	C	D
1				
2		Vehicle Data		
3		Wheelbase	2.99	m
4		Track Front	1.676	m
5		Mass	1880	kg
6		Distribution (F/R)	47/53	
7		Max Torque	683	Nm
8		RPM @ Max Torque	6000	RPM
9		Max Torque to Front	20	%
10			=C7*(C9/100)	Nm
11		Max Torque to Each Clutch Pack	10	%
12			=C7*(C11/100)	Nm
13				
14				
15				
16				
17				
18				
19				
20				
21				

Lumped Mass Model Sheet

	A	B	C	D	E
1					
2		Lumped Mass Model			
3			Mass (kg)	Distance to CoG (m)	Moment of Inertia ZZ (kgm²)
4		Engine	265	0.81294	=C4*D4^2
5		Transaxle	241	1.4053	=C5*D5^2
6		Body Front	687	1.5847	=C6*D6^2
7		Body Rear	687	1.4053	=C7*D7^2
8				Total	=SUM(E4:E7)
9					
10		CoG Long			
11		Distance a (m)	=Data!C3*0.53		
12		Distance b (m)	=Data!C3-C11		
13					
14		Inertial Conjugate, c (m)	=E8/(Data!C5*C11)		
15		Distance Infront of Rear Axle (m)	=C12-C14		
16					
17		DI	=E8/(Data!C5*C11*C12)		
18					
19					
20					
21					
22					

Torque Vectoring sheet 1 of 2

	A	B	C	D	E	F	G	H	I	J	K	L
1												
2		Gear	Ratio	Overall Ratio	Rear Road Wheel Speed @ Max Torque (rpm)							
3		1	3.397	=C3*3.88	=Data!\$C\$8/D3							
4		2	2.186	=C4*3.88	=Data!\$C\$8/D4							
5		3	1.626	=C5*3.88	=Data!\$C\$8/D5							
6		4	1.285	=C6*3.88	=Data!\$C\$8/D6							
7		5	1.029	=C7*3.88	=Data!\$C\$8/D7							
8		6	0.839	=C8*3.88	=Data!\$C\$8/D8							
9		7	0.634	=C9*3.88	=Data!\$C\$8/D9							
10		front gearbox										
11		1		=D4/1.06	=Data!\$C\$8/D11							
12		2		=D6/1.06	=Data!\$C\$8/D12							
13												
14												
15		Gear	Difference in Angular Velocity @ 6000rpm		Torque at each Clutch Pack		Power Dissipated across Each Clutch Pack					
16		1	=E11-E3 RPM	= (C16/60)*2*PI() rad/sec	=Data!C12*D11 Nm	=G16*E16 W	=I16/1000 kW					
17		2	=E12-E5 RPM	= (C17/60)*2*PI() rad/sec	=Data!C12*D12 Nm	=G17*E17 W	=I17/1000 kW					
18												

Torque Vectoring sheet 2 of 2

	Q	R	S	T	U	V	W	X
2								
3								
4			RWD		AWD		Improvement	
5		Mu	Rear Drive Force (N)	Acceleration (m/s/s)	Front Drive Force (N)	Total Drive Force (N)	Acceleration (m/s/s)	Acceleration (m/s/s)
6		0.2	= (1880*0.53*9.81)*R5	=S5/1880	=F(((1880*0.47*9.81)*R5)>\$P\$4,\$P\$4,(1880*0.47*9.81)*R5)	=S5+U5	=V5/1880	=W5-T5
7		0.6	= (1880*0.53*9.81)*R6	=S6/1880	=F(((1880*0.47*9.81)*R6)>\$P\$4,\$P\$4,(1880*0.47*9.81)*R6)	=S6+U6	=V6/1880	=W6-T6
8		0.9	= (1880*0.53*9.81)*R7	=S7/1880	=F(((1880*0.47*9.81)*R7)>\$P\$4,\$P\$4,(1880*0.47*9.81)*R7)	=S7+U7	=V7/1880	=W7-T7
9								
10			Max Torque at One Front Wheel (Nm)	Max Force Difference (N)	Yaw Rate Moment (Nm)	Rotational Acceleration Increase (rad/s^2)		
11			=D20	=S11/0.341	=T11*(1.676/2)	=U11/Lumped Mass Model!E8		
12								
13								
14								
15								
16								
17								
18								
19								
20								
21								
22								