

AWD Torque Vectoring

Advanced Vehicle Dynamics (M6X00407)

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4 Abstract

Throughout this report, the unconventionally designed Ferrari FF's mid-front engine, four wheel driver architecture (*Ferrari FF*, 2011) will be investigated. The effects of this powertrain on the car's drivability, in terms of rotational and longitudinal acceleration will be looked at.

4.1 Vehicle Data

Front Axle Mass	641.6 kg
Engine Mass	Ca. 200 kg (<i>Engine weight – a comprehensive list</i> , 2014)
Passenger mass	62 kg (Rettner, 2013) x 2 per vehicle
Rear Transmission Mass	Ca. 40 kg (Assumption)
Rear Axle Mass	874.4 kg
Engine Location	90% front (Assumption)
Passengers Location	50% front (Assumption)
Rear Transmission Location	0% front (Assumption)
Wheelbase	2990 mm
Longitudinal Weight Distribution	47% Front
Front Track	1676 mm
Rear Track	1660 mm
Maximum Torque	683 Nm @ 6000 rpm
Coefficient of Clutch Friction	0.2
Coefficient of Road Friction	0.8

Table 1: Vehicle Data

The Ferrari FF is a four wheel drive GT vehicle with a 6.3 litre V 12 engine (*2015 Ferrari FF Technical Specifications*, 2011). This engine uses a 65° bank angle. The FF was the first car to implement the connectable four-wheel drive system patented by Ferrari (Cimatti and Marawetz, 2010). This novel system would allow for a powertrain system that could operate in both rear-wheel or all-wheel drive. The systems control methods meant that driver input would not be required to determine the driving mode, allowing safety control features to apply power and torque to wheels individually as required. This way the vehicle could maintain the dynamic feel of a rear-wheel drive car, often preferred by users who enjoy driving more aggressively, whilst taking advantage of the increased safety of an all-wheel drive vehicle.

4.2 Torque Vectoring

Torque vectoring is a form of control system used by automotive manufacturers. Its aim is to improve the grip available to the driver, throughout a wide range of manoeuvres, by controlling the application of power to individual wheels (*What is torque vectoring and XDS?*, 2020).

Using traction control systems, cars are able to control the amount of torque that is supplied to the individual wheels of the FF. In an open differential, the maximum longitudinal force that can be applied to all driven wheels, is controlled by the amount of grip on the wheel with the lowest grip. For example on a split μ surface, such as in Figure 1, even on the side with higher grip, only as much force as is available on the low μ surface, can be applied. In a locked differential scenario, all driven wheels spin at the same speed. Whilst this does mean that the wheels with higher grip can apply more force, the wheels on more slippery surfaces are spinning, thereby applying less force, and also reducing their ability to apply lateral loads. The FF's drivetrain allows for the torque on

individual wheels to be controlled independently from one another, by varying the force being applied to compress individual clutch packs. This way, each wheel can apply the maximum amount of torque possible without spinning other wheels. When the system begins to notice that any of the four driven wheels are spinning, torque to that wheel is reduced, preventing excessive slip and maximising acceleration.



Figure 1: Split mu track (Vehicle Dynamics, 2020)

Traction control systems can also assist in maintaining lateral grip. A wheel's grip limit is a combination of the lateral and longitudinal forces it is applying on the vehicle. Reducing the longitudinal force a wheel is exerting, by reducing torque, increases the lateral capacity. Torque vectoring allows the drive train to increase the vehicles yaw moment by reducing the amount of slip each tyre experiences in each manoeuvre. In an understeering scenario more torque can be applied to the outside wheels whilst the inside wheel torque is reduced. By behaving in the opposite direction, the oversteering tendency of a vehicle can be reduced, increasing lateral-dynamic agility. In extreme scenarios, brake intervention, on wheels where a negative torque would help, could further increase the induction of yaw rate.

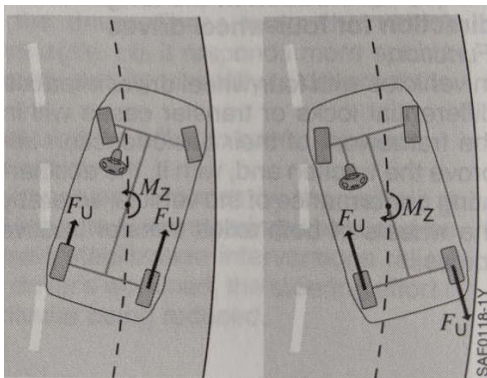


Figure 2: Rear Wheel Drive Torque Vectoring (Bosch, 2018)

5 Ferrari FF

5.1 C of G

Considering the weight distribution of the vehicle lies 47:53 on the front and rear axles, the longitudinal weight distribution can be calculated by finding 53% of the overall wheelbase of the vehicle. This gives the location behind the front axle where the centre of gravity can be found.

$$WB \times \text{Weight Distribution} = WD$$

$$2990\text{mm} \times 53\% = 1584.7\text{ mm}$$

Equation 1: Longitudinal CoG Location

WB – Wheelbase (mm)

WD – Longitudinal CoG Location (mm)

The value of 53% does not include any passengers in the vehicle, however, assuming the passengers are seated centrally in the vehicle, their mass would act 50:50 on each axle and therefore not affect the longitudinal location of the centre of gravity.

5.2 Lumped Mass Model

The lumped mass model shows the longitudinal location of the components in the vehicle that have the most significant mass, as well as the overall centre of gravity of the vehicle.

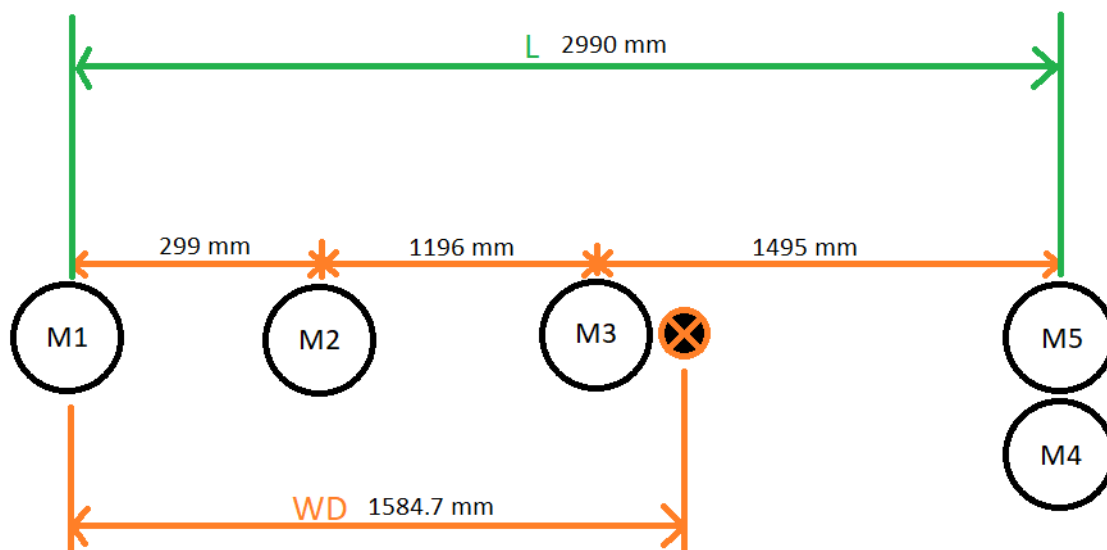


Figure 3: Lumped Mass Model

M1 – Front Axle mass

M2 – Engine mass

M3 – Passengers mass

M4 – Rear Transmission mass

M5 – Rear Axle mass

L – Wheelbase

WD – Longitudinal CoG Location

5.3 Moment of Inertia

The rotational inertia of an object or assembly is the measure of its resistance to rotational acceleration. The more of an objects mass that is close to its centre of gravity, the lower this value will be (*Mass Moment of Inertia Equations*, 2020). For simple geometries this value can be calculated, however, in many cases it is necessary to measure the true value (Jr. and Sturm, 2020).

$$I = \Sigma m(x^2 + y^2)$$

Equation 2: Moment of Inertia (2D)

$$I = \Sigma m(x^2)$$

Equation 3: Moment of Inertia (1D)

m – mass of the part

x – x-axis distance between CoG and the part

y – y-axis distance between CoG and the part

To mathematically calculate the value of rotational inertia, the equations above can be used. Equation 2 can be used to find the rotational inertia of a 2D system, however, considering the near 50:50 distribution of components and mass laterally across the vehicle, Equation 3 can be used instead. The mass of each component along with the distance between the CoG of the component and the CoG of the entire system must be known for this equation to be used.

Table 2: Component Inertia

Part	Mass (kg)	Distance (m)	Moment (kg.m)
M1	641.6	1.5847	1611.23
M2	200	1.2857	330.6
M3	124	0.0897	0.10
M4	40	1.4053	78.99
M5	714.4	1.4053	1410.85

Using this method, the inertial moment of each individual component around the centre of gravity is calculated. The sum of these results in an overall inertia of 3431.77 kg.m² around the Z-Z or yaw axis for the entire vehicle. This value could be affected by the addition of further passengers, or luggage within the vehicle. For example, assuming two further passengers of a total mass of 60 kg located half a meter behind the front passengers would increase the rotational inertia to 3441.88 kg.m². Furthermore, if it were possible to complete this calculation using the individual mass and distance for every component within the vehicle, a more precise value could be achieved.

5.3.1 Inertial Conjugate

The inertial conjugate, or centre of percussion, is a single point within a geometry that does not experience any translation when a load is applied other than through the centre of gravity (Blundell and Harty, 2015). It can be calculated using the equation below. Assuming the rotational inertia calculated previously with two passengers, the inertial conjugate has a value of 1.18m.

$$c = \frac{I}{m \times a}$$

Equation 4: Inertial Conjugate

m – vehicle mass

a – CoG distance from front axle

5.4 Dynamic Index

Although a low rotational inertia will increase the vehicles ability to turn a corner, making it agile and nimble, if the value is too low it can be considered twitchy and difficult to control. Rotational inertia however, is not the only thing that effects this behaviour. The wheelbase of the vehicle and its mass distribution must also be considered. This is where the dynamic index can be used, as a single value, which can be compared against another vehicle. The equation below can be used to find the dynamic index for the Ferrari FF, resulting in a value of 0.83. The addition of rear passengers increases this value to 0.87.

$$DI = \frac{I}{m \cdot a \cdot b} = \frac{c}{b}$$

m – vehicle mass

a – CoG distance from front axle

b – CoG distance from rear axle

Equation 5: Dynamic Index

5.5 Conclusion

The dynamic index calculated above, is a relatively low value when compared to other road sports cars (Blundell and Harty, 2015). Whilst this does mean that the car won't feel slow to respond to steering inputs, even with the addition of rear passengers, the car has a dynamic index that could result in a car that is difficult to control. Electronic stability control systems could be used to maintain the cars responsiveness, whilst smoothing out erratic driving behaviours, especially in poor driving conditions.

5.6 Torque Path

The unconventional aspect of the FF's powertrain system is to do with the way the front wheels are driven. In a traditional four-wheel drive system, a drive shaft, connects the crank shaft to a transfer case, usually via a fly wheel and a conventional transmission. As a result, the ratio between the front and rear wheels final drive is fixed. Furthermore, the torque path from clutch to wheel is slightly convoluted. To reach the front wheels, the torque first follows the main driveshaft backwards from the engine to the transfer case, before following a secondary drive shaft forwards to the front differential and then to each front wheel.

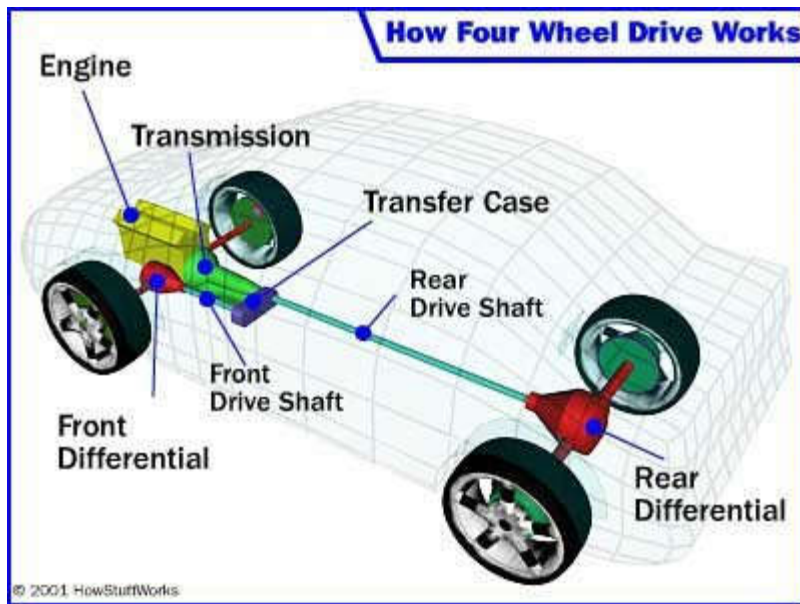


Figure 4: Traditional all-wheel drive (Nice, 2020)

The Ferrari FF's powertrain looks different. Whilst the engine is still connected to a drive shaft towards the rear of the car, this continues all the way to the rear axle where the gearbox and differential for the rear wheels is located. This is like a conventional, rear wheel drive only, system. The front wheels are powered through a secondary gearbox, located on the front of the engine. It is connected directly to the front end of the crankshaft and unlike traditional transmissions, does not have a clutch separating the gearbox from the engine. Two clutch packs, one per wheel, are located after the gearbox and a locked differential.

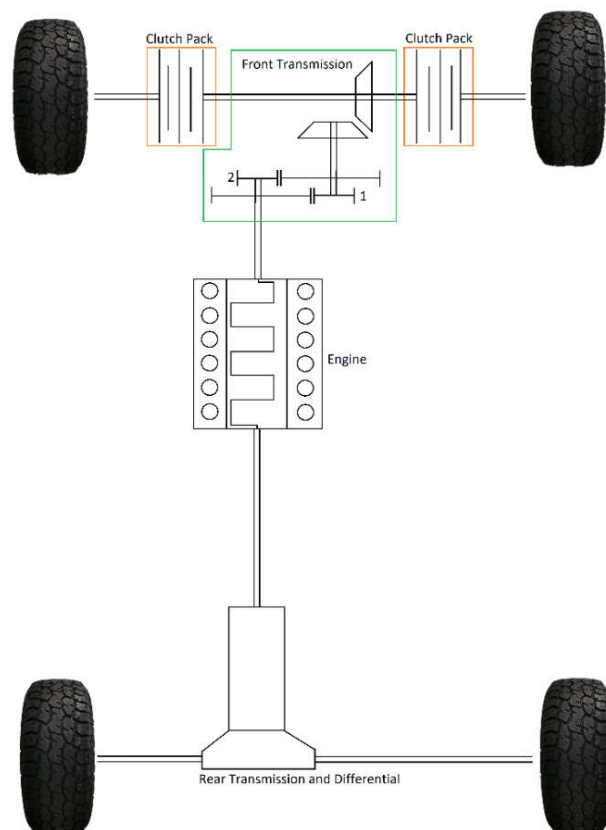


Figure 5: Ferrari FF Torque Path

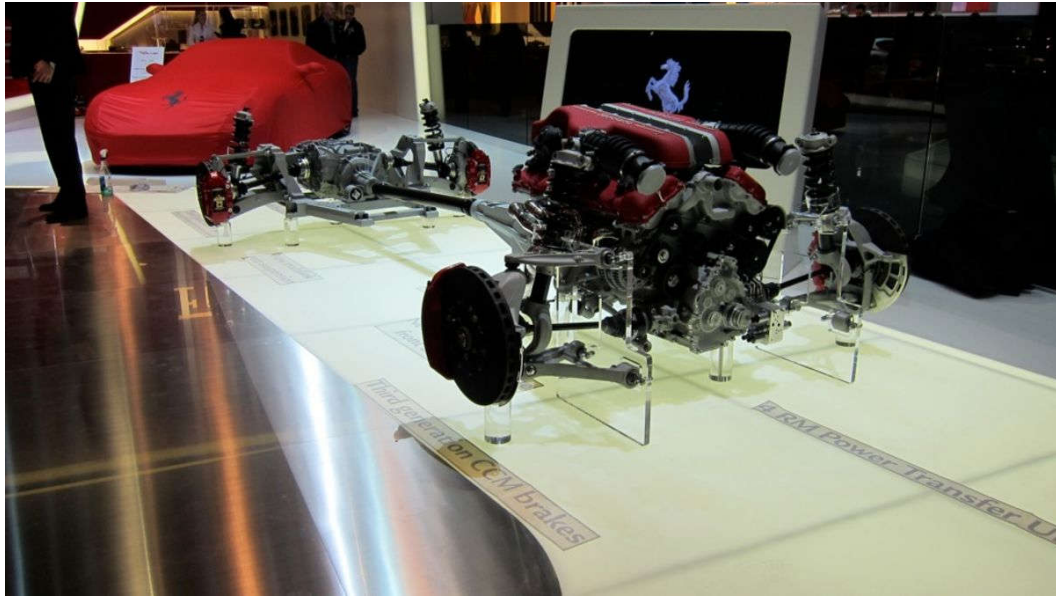


Figure 6: Ferrari FF Powertrain (Wert, 2011)

5.6.1 Clutch Pack Capacity

To work out the true capacity of the clutch packs used in the FF, several unknowns would need to be taken into account. The equation below could be used to calculate the capacity if the clamping force and coefficient of friction between the clutches was known.

$$\tau = \frac{F \times R \times n \times \mu}{2}$$

Equation 6: Clutch Pack Torque capacity (Balzer et al., 1999)

τ – Clutch pack torque capacity (Nm)

F – Clamping force (N)

R – Radius of gyration of the friction surface (m)

n – The number of friction surfaces

μ - Coefficient of friction between surfaces

Since this information is not available, the value can be estimated based on other known values. The engine is known to produce its maximum torque of 683 Nm at 6000 rpm. Considering the 10% per front wheel torque split, and the known front gear ratios of 9.11:1.06 or 8.59 in first gear and 4.97:1.06 or 4.69 in second gear, the maximum torque being applied by the engine can be calculated. The highest front wheel torque of 587 Nm will occur in first gear. As a minimum, this is what each front clutch pack must be capable of handling to transmit the maximum engine torque.

When creating a specification for the clutch pack, engineers must consider this value, as well as the intended lifetime of the clutch pack. In order to cope with the difference in wheel speed between the front and rear wheels, this clutch pack will constantly need to be able to slip in order to prevent the contact patch with the road from being the point where slip occurs. To allow this slip to occur and so that torque vectoring can be applied to these wheels individually, the pressure acting on the clutch packs can be controlled to increase or reduce the torque they transmit. This allows the system to take away torque from individual wheels as necessary.

Due to the difference in gear ratio between the front and rear wheels, to maintain a constant contact patch between the wheels and the road, the front clutch must always be slipping. This results in a loss of power that can be calculated using Equation 7. Here the torque is the maximum torque applied by the clutch pack, which previously was found to be 587 Nm. The rotational velocity is the difference in rotational velocity between the two ends of the clutch pack. To determine this value, the engine RPM, 6000 rpm at maximum power, is multiplied by the rear final drive, 6.56 in third gear. This gives a rear wheel speed of 914.63 rpm. Since front and rear wheel diameters are the same, the output of the clutch pack must be rotating at the same speed. To find the input velocity of the clutch pack, the engine RPM is multiplied by the final drive for the front gear ratio, 4.69. This results in a rotational velocity of 1279.32 rpm. The difference of 364.68 rpm, or 38.19 rad/s results in a power loss of 22.42 kW.

$$Power = Torque \times Rotational Velocity$$

Equation 7: Clutch pack power loss

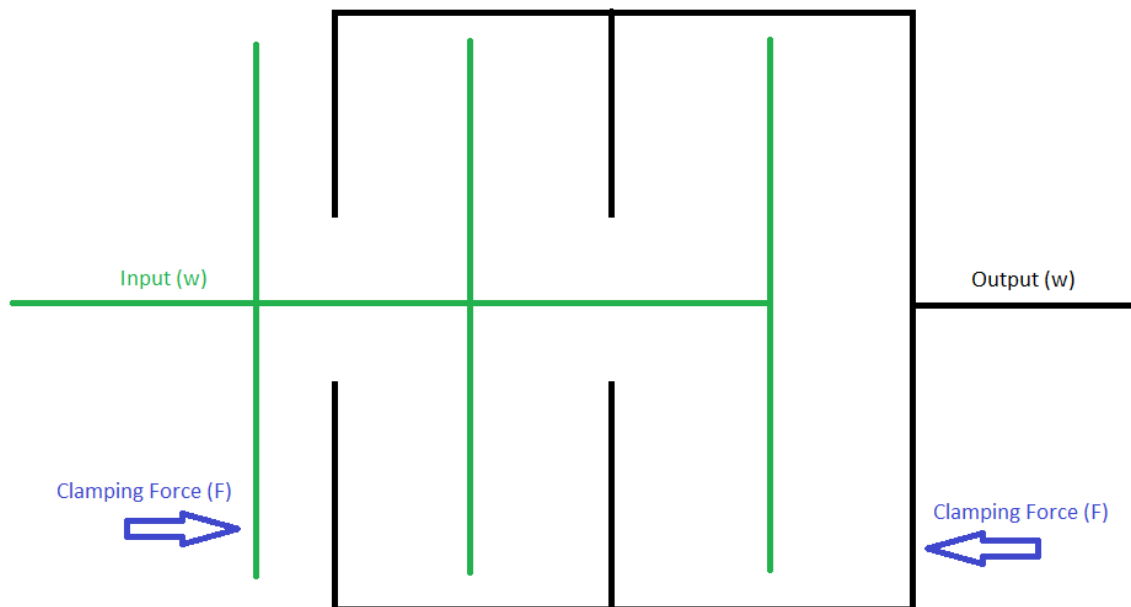


Figure 7: Ferrari FF Front Clutch Pack

Considering the torque capacity, the powertrain needs to be able to transmit, it is possible to estimate what the dimensions of the clutches are likely to be. Considering the application being considered, the clutches are likely to be smaller than conventional clutches, as packaging is likely to be an issue. To compensate, multiple clutch faces can be used. Furthermore, although the clutches need to be able to consistently transmit the required torque, they are designed to serve as a mechanical fuse, as slipping the clutch is a better outcome than breaking another component. There is a trade-off between this feature, and component lifespan. Considering this information, and the fact that each clutch pack has 3 clutch plates (Cimatti and Marawetz, 2010), an estimate of the necessary force to clamp the clutch pack together can be made by rearranging Equation 6.

$$F = \frac{2\tau}{R \times n \times \mu}$$

Equation 8: Clutch pack clamping force

Assuming 3, 20mm diameter clutches per clutch pack, each with 5mm wide friction material on either side and a coefficient of friction of 0.2 results in a clamping force of 55.9 kN. Whilst compared to a traditional clutch pack, where the clamping force is as a result of springs compressing the clutch plates, this is a very high force, for the FF, where a hydraulic system is used to compress the clutches (Cimatti and Marawetz, 2010), this value is reasonable.

5.6.2 Longitudinal Acceleration

In order to calculate the maximum longitudinal acceleration, two areas must be considered in order to determine which one is the limiting factor. The first area is the maximum force that can be applied, due to the coefficient of friction between the tyres and the road. If the engine can apply a higher force than this value, that would simply cause the wheels to spin as opposed to transferring the additional force to the road. Furthermore, spinning, or slipping wheels often have a lower coefficient of friction the road, than when there is good traction.

To determine the maximum force due to friction, the equation below can be used. This shows what the maximum longitudinal force (F) is that the tyres can exert on the road, given the vehicle weight (R) and the coefficient of friction between the tyres and the road (μ).

$$F = \mu \cdot R$$

Equation 9: Force due to Coefficient of Friction

Considering a kerb weight of 1880 kg, a 47% forward weight distribution and two evenly distributed 62 kg passengers, the load on each tyre would be 9276.34 N at the front and 10382.9 N at the rear. Considering an average coefficient of friction of 0.8 in good weather and road conditions (Zhao *et al.*, 2017), this means a total longitudinal force of 15727.39 N can be used to accelerate the car.

To determine the maximum acceleration the engine is capable of producing in the vehicle, the 20:80 torque split front to rear, as well as the 50:50 torque split to either side and the maximum engine torque of 683 Nm, must all be considered. Considering the maximum engine torque of 683 Nm, the 80% torque split towards the rear and the final drive ratio of 13.17 in first gear, each rear wheel can produce a maximum torque of 3598.044 Nm. Considering a wheel diameter of 0.508m (20") for the standard tyre and rim combination, this results in a maximum lateral force of 14165.53 N per rear wheel. Furthermore, considering the 20% torque split towards the front and the final drive front ratio of 8.59 in first gear, each front wheel can produce a maximum torque of 586.697 Nm. With the same wheel radius as the rear, this results in a lateral force of 2309.83 N. As a result, the maximum force, all four wheels can produce, is 32950.72 N.

$$F = m a$$

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

Equation 10: Newton's second law of motion

Because the engine is able to overcome the maximum force due to friction, the coefficient of friction is the weak point in the system. Once this force is saturated, maximum acceleration has been achieved. With a total mass of 2004 kg including two passengers, the maximum friction force of 15727.39N results in a maximum acceleration of 7.85 m/s². In poor conditions however, the coefficient of friction can drop, reducing the tyres capacity to accelerate the car.

5.6.3 Yaw Rate

As discussed previously, by intelligently applying or reducing the torque to individual wheels, the yawing capacity of the Ferrari can be increased, reducing over- and understeer. By multiplying the maximum force a wheel can apply to the road by half of the axles track, the torque induced around the centre of gravity can be calculated.

$$\text{Front Torque} = (9276.34 \times 0.8) \left(\frac{1.676}{2} \right) = 6218.86 \text{ Nm}$$

Equation 11: Front torque

$$\text{Rear Torque} = (10382.9 \times 0.8) \left(\frac{1.66}{2} \right) = 6894.25 \text{ Nm}$$

Equation 12: Rear torque

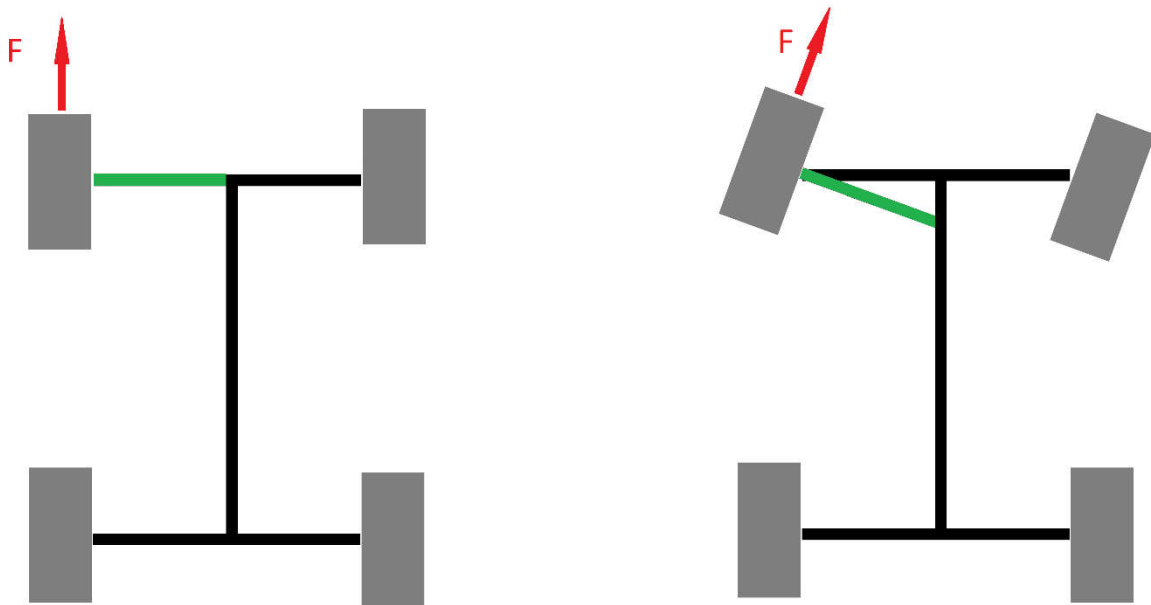
$$\text{Torque} = \text{Inertia} \times \text{Yaw Acceleration}$$

Equation 13: Torque, Inertia and Yaw relationship

Equation 13 shows the relationship between torque, inertia and yaw acceleration. By re-arranging it and using the values for torque and inertia calculated previously, the overall effect torque vectoring has on a vehicles lateral dynamic can be found.

Using solely torque vectoring to apply a maximum force on the outside wheels of the car, and completely removing the force applied on the inside wheels, produces a yaw acceleration of 3.82 rads/s^2 . However, if braking could be used to apply a braking force, equal to the maximum the friction between the tyres and road will allow, the yaw acceleration would increase to 7.64 rads/s^2 .

These values for yaw acceleration assume a straight hand wheel. Turning the wheel would itself cause a change in yaw rate. However, when the hand wheel is turned, the lever arm, between the contact patch at the front tires and the centre line of the car, is increased, reducing the torque the front wheels apply in the yawing direction.



Equation 14: Yawing lever arm

5.7 Vehicle Control

Dynamic stability control in vehicles can use several different approaches in order to give the driver the safest and most fun driving experience possible. As well as controlling the torque at each wheel, the system can control individual wheel braking. This way the system can ensure the car behaves in a predictable manner.

To apply control methods for accelerating and cornering as discussed throughout this investigation, it is important for the car to have as much information about what is happening as possible. To interpret the driver's inputs, a hand wheel angle sensor, accelerator pedal sensor and brake pedal sensors are required. These help in determining the intentions of the driver. For sub-limit driving, wheel speed sensors help to determine which wheels have the correct amount of grip and which are spinning. This way the system can determine which wheels require more or less torque to be applied. For post limit driving, a combination of GPS and gyroscopic sensors help determine the cars trajectory and angle of motion. This information can be used to determine the best strategy for torque application. Even in sub-limit driving, this information can help to determine slip angles, predicting when post limit events are imminent so that they can be avoided.

A closed loop control algorithm, allows the traction control system to control the front clutch pack force, depending on the torque required. As the figure below shows, the difference in front and rear wheel speed E , is used to determine the required front wheel torque. In block **27** this value is converted to a required oil pressure P_{rif} which after comparison between the current oil pressure measured through a sensor, results in a change in oil pressure request signal which is sent to the hydraulic actuator **22**. The PID helps control this signal, increasing responsiveness and stability of the system.

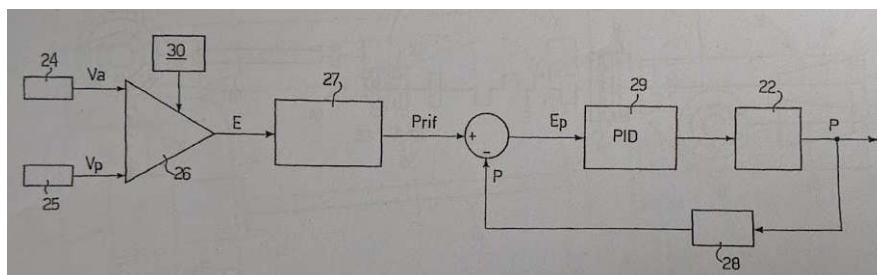


Figure 8: Front Clutch Pack control algorithm (Cimatti and Marawetz, 2010)

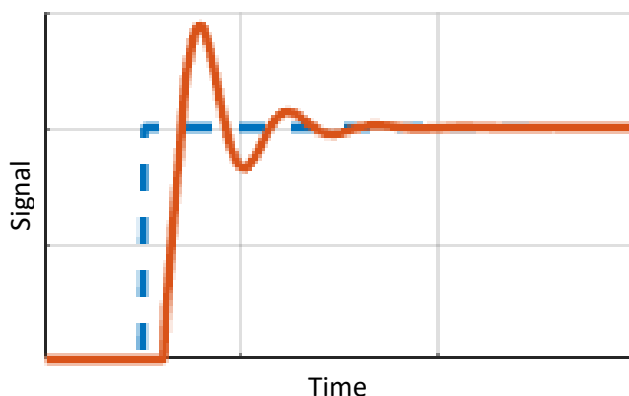


Figure 9: PID Control (PID control – PID-tuner.com, 2020)

6 Summary

The Ferrari FF's novel approach to the four-wheel drive powertrain has significant advantages over its competitors. The control this system gives the stability control system makes for a safe driving experience whilst letting the driver enjoy the vehicles full potential. The expected drawbacks usually expected with a heavier vehicle seem to have little effect, leaving the car feeling nimble and agile, requiring the onboard systems to smooth out an otherwise twitchy driving experience. Overall, the car should feel responsive and fun to drive, with excellent lateral driving dynamics and acceleration.

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