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AWD Torque Vectoring – M6X00415

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20/05/20

1 Contents

1	Contents.....	2
2	Abstract.....	1
3	Base vehicle data	2
4	Equations used	3
5	Ferrari 4RM torque vectoring system.....	4
6	Ferrari FF	6
6.1	Longitudinal centre of gravity position	7
6.2	Moment of inertia (about Z-Z axis)	7
6.3	Inertial conjugate position.....	7
6.4	Dynamic Index	8
6.5	Base vehicle specifications analysis	8
7	Mechanical torque path to front wheels	10
8	Clutch pack analysis	13
8.1	Torque capacity.....	13
8.2	Approximate sizing.....	13
8.3	Axial force application values	13
8.4	Power loss characteristics	14
9	Potential longitudinal acceleration improvements	15
10	Yaw rate increase.....	16
11	Possible control algorithms	19
12	Conclusion	21
13	References.....	22
14	List of plates	23
15	List of tables	23
16	List of figures	23

2 Abstract

The aim of this report is to discuss and analyse the Ferrari 4RM four-wheel drive system, introduced in 2011 with the Ferrari FF (see *Plate 2-1*). The report aims to discuss the effect of the 4RM system on vehicle dynamics, and to discuss potential control strategies. The Ferrari FF included a naturally aspirated V12 engine, producing 486kW at 8000RPM and 683Nm at 6000RPM. The engine produced around 546Nm from 1750RPM, meaning despite being a heavy car, it pulled hard from low in the rev range. This Ferrari used a different design language, utilising four-wheel drive, four-seats in a grand touring style vehicle.



Plate 2-1 - Ferrari FF driving on hard packed snow (Ferrari, n.d.)

3 Base vehicle data

Table 3-1 - Base Vehicle Data (Ferrari, n.d.)

Wheelbase	2.990	m	
Front Track	1.676	m	
Rear Track	1.660	m	
Kerb weight	1880	kg	
Dry weight	1790	kg	
Mass with 90kg driver	1970	kg	
Weight Front	47.00%		
Weight Rear	53.00%		
Mass on front axle	925.9	kg	
Mass on rear axle	1044.1	kg	
Front tyre	245/35 ZR 20		
Rear tyre	295/35 ZR 20		
Front wheel	8.5 J x 20"		
Rear wheel	10.5 J x 20"		
Total displacement	6262	cm3	
Maximum power	486	kW	@8000RPM
Maximum torque	683	Nm	@6000RPM
Maximum RPM	8000	RPM	
Specific output	77	kW / litre	
Compression ratio	12.3 : 1		
Fuel consumption	15.4	l / 100km	
7 speed F1 dual-clutch gearbox			
4RM control			
ABS / EBD			
ESC			

Table 3-1 shows the base vehicle information with data from Ferrari.

4 Equations used

$$\text{Moment of inertia (about Z - Z axis), } I_{ZZ} = mk^2$$

$$\text{Dynamic Index, } DI = \frac{I_{ZZ}}{\text{Mass} \times A \times B}$$

$$\text{Inertial conjugate} = \frac{I_{ZZ}}{\text{Mass} \times A}$$

$$\text{Overall ratio} = \text{Gear ratio} \times \text{Final drive}$$

$$\text{Wheel speed (RPM)} = \frac{\text{Engine RPM}}{\text{Overall ratio}}$$

$$\text{Power accross PTU clutch pack} = T \times \Delta\omega = \frac{T \times \Delta\text{Clutch RPM} \times 2\pi}{60}$$

$$\text{Slip accross PTU clutch pack} = \left[\frac{\text{PTU gearbox RPM}}{\text{Main gearbox RPM}} \times 100 \right] - 100$$

$$\text{Axial force, } W = \frac{3 \times T (r_2^2 - r_1^2)}{2 \times \mu \times n(r_2^3 - r_1^3)}$$

Where outer radius of clutch is r_2 , inner radius of clutch is r_1 , and number of friction surfaces is n

$$\text{Weight transfer} = \frac{m \cdot a \cdot h}{T}$$

$$\text{Moment, } \tau = \Delta\text{Lateral grip} \times \frac{T}{2}$$

$$\text{Yaw rate gain, } \alpha = \frac{\tau}{I_{ZZ}}$$

5 Ferrari 4RM torque vectoring system

Many four-wheel drive vehicles allow different amounts of torque to be sent to the front or rear axles. Commonly, this is done through use of a centre differential. Alongside four-wheel drive systems, torque vectoring is often used to increase the vehicles performance. Torque vectoring aims to enhance dynamic behaviour and stability. Torque vectoring systems allow different amounts of torque to be directed independently to each wheel on the axle. It can be used with a two or four-wheel drive layout. Torque vectoring is beneficial, especially on low grip surfaces, as more power can be sent to the outside wheel when cornering, which will have more grip. This allows for an increased yaw moment, meaning cornering speed can be improved.

‘When cornering, for example, such a system could send more torque to the wheel on the outside of a corner. This extra ‘push’ from the driven outside wheel generates a yaw moment which helps the turn the car into the corner, tightening its line.’

(Kingston, 2018)

Torque vectoring can be achieved using a differential with a clutch pack for each wheel. Often an open differential is used, and the clutches are operated electronically or hydraulically. By varying the slip of the clutches, each individual wheel can have different amounts of torque sent to it.

Torque vectoring can also be achieved using braking. Using an open differential, brakes can be applied to an individual wheel on the axle, forcing more torque to the opposite wheel. The brakes are applied using the cars stability control. During cornering, the brakes are applied to the inside wheel with the least amount of grip. By lightly applying the brakes, there will be more torque on that wheel, forcing more engine torque to the opposite wheel. This generates a yaw moment, and can increase turn-in, reducing understeer. Open differentials sent power to the path of least resistance, which is undesirable as the wheel with least grip will get most torque. Torque vectoring using braking is less expensive than systems using a differential, however, it is commonly less effective.

The 4RM system found in the Ferrari FF uses a Power Transfer Unit (PTU) mounted over the front axle (see *Plate 5-1*). By taking power directly from the crankshaft, torque to the front wheels is varied using two independent carbon-fibre oil-bath wet multi-plate clutch packs. The power is sent to the rear wheels down a torque tube to a transaxle mounted at the rear, which vectors torque to

each individual wheel using Ferrari's E-Differential. According to Ferrari the 4RM control can recognise understeer and oversteer situations and variations in grip.

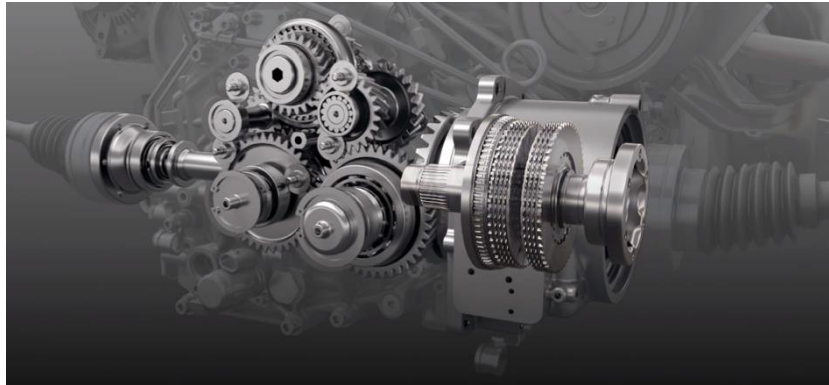


Plate 5-1 - Ferrari Power Transfer Unit (Caroto.gr, 2011)

According to Ferrari;

'Unlike a conventional four-wheel drive system fitted to a front engined car, it allows the retention of the traditional mid-front engine architecture, with rear transaxle connected to the engine by a single driveshaft.'

Ferrari explains how this permits a 50% weight saving over a conventional four-wheel drive system and is less complex. The 4RM systems also lowers the centre of gravity height. The PTU clutches are controlled hydraulically and can be activated very quickly. By working together, the front clutches can limit the amount of torque transferred to the front axle, and individually to each wheel.

6 Ferrari FF



Plate 6-1 - Side view of Ferrari FF (ItzKirb Photography, 2018)

Plate 6-1 shows a side view of the Ferrari FF, the masses acting on the car were simplified to create a lumped mass model, see Figure 6-1.

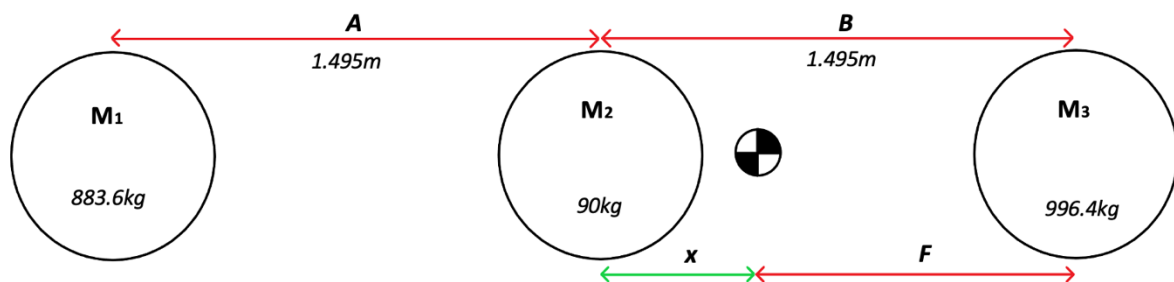


Figure 6-1 - Ferrari FF lumped mass model

In Figure 6-1, M_1 is the mass on the front axle, M_2 is the mass of the driver (90kg was assumed) and M_3 is the mass on the rear axle. It was assumed that the driver is positioned in the centre of the wheelbase. The lumped mass model allows calculation of the longitudinal centre of gravity position and the moment of inertia. By using this lumped mass model, moments could be taken around the centre of gravity to calculate its position.

6.1 Longitudinal centre of gravity position

Taking moments about centre of gravity

$$M_3(B - x) = M_2(x) + M_1(A + x)$$

$$\therefore F = 1.495 - 0.08558$$

$$996.4(1.495 - x) = 90x + 883.6(1.495 + x)$$

$$\therefore F = 1.409m$$

$$1489.6 - 996.4x = 90x + 1321.0 + 883.6x$$

$$1489.6 - 1321.0 = 90x + 883.6x + 996.4x$$

Centre of gravity from front = 1.581m

$$168.6 = 1970x$$

$$\therefore x = \frac{168.6}{1970} = 0.08558m$$

6.2 Moment of inertia (about Z-Z axis)

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

$$I_{zz} = M_1(A + x)^2 + M_2(x)^2 + M_3(B - x)^2$$

$$I_{zz} = 883.6(1.495 + 0.0856)^2 + 90(0.0856)^2 + 996.4(1.495 - 0.0856)^2$$

$$I_{zz} = 883.6(1.5806)^2 + 90(0.0856)^2 + 996.4(1.4094)^2$$

$$I_{zz} = 2207.49 + 0.66 + 1979.26$$

$$I_{zz} = 4187.4kgm^2$$

6.3 Inertial conjugate position

$$Inertial\ conjugate = \frac{I_{zz}}{Mass \times A}$$

$$Inertial\ conjugate = \frac{4187.4}{1970 \times 1.581}$$

$$Inertial\ conjugate = 1.344m$$

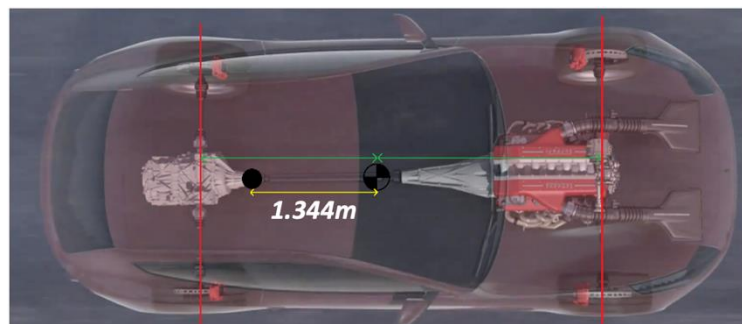


Figure 6-2 - Position of the inertial conjugate

6.4 Dynamic Index

$$DI = \frac{I_{zz}}{Mass \times A \times B}$$

$$DI = \frac{4187.4}{1970 \times 1.581 \times 1.409}$$

$$DI = 0.95$$

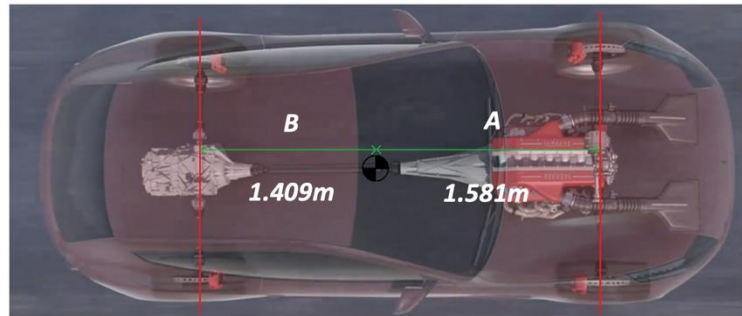


Figure 6-3 - Distances used to calculate the dynamic index

6.5 Base vehicle specifications analysis

Since the Ferrari FF is primarily a rear-wheel drive car, having the longitudinal centre of gravity position further to the rear is sensible. This means that more grip is available at the rear wheels, which have most of the driving force. The centre of gravity position overall is close to the middle of the car. This is beneficial as the slip angles at the front and rear will be similar, resulting in neutral balance whilst steady state cornering. Having the centre of gravity position slightly towards the rear will increase oversteer balance, which in this car will be advantageous as it is more enjoyable for the driver. Since the 4RM can quickly add more driving force to the front wheels, an oversteering balance would be advantageous as increasing driving force to the front wheels could limit oversteer in a moment of near crisis.

The moment of inertia for the Ferrari FF is quite high due to the relatively heavy mass. By using a 50/50 weight split the moment of inertia could be reduced, resulting in a more responsive feeling car which is easier to turn.

The inertial conjugate is positioned in front of the rear axle (see *Figure 6-2*). A radius of gyration of 1.458m was found for the vehicle.

The dynamic index for the Ferrari FF is 0.95 (see *Figure 6-3*). The dynamic index influences how the rear tyres respond during a transient manoeuvre, and vehicle agility for post limit control.

With a dynamic index of below 1, as the car turns in, the rear wheels will move outwards generating a slip angle and generating lateral grip. Any additional steering angle during a corner will result in the

rear wheels building a slip angle, and if they are on the limit and the grip is saturated, this additional slip angle will reduce grip.

When the dynamic index is 1, the inertial conjugate is on the rear axle. Having a dynamic index above 1 means the car will be very stable, but on turn-in can feel sluggish and unresponsive. When the dynamic index is above 1, when the vehicle turns in the rear wheels will move inwards reducing the slip angle and generating lateral grip. Any additional steering angle during a corner will result in a reduction to slip angle of the rear wheels, and if they are on the limit and the grip is saturated this additional slip angle will increase grip.

Generally, the lower the value of dynamic index, the more responsive and lively the car feels on turn-in, however, an excessively low dynamic index (of below about 0.8) can result in a twitchy unmanageable ride. The Lotus Elan S2 has a dynamic index of 0.90 and BMW M5 of 0.92. The M5 is a comparative car, as it is a powerful sports sedan, which is often used for grand touring. With a dynamic index of 0.95 the Ferrari FF will retain a somewhat responsive feel whilst maintaining stability for less able drivers.

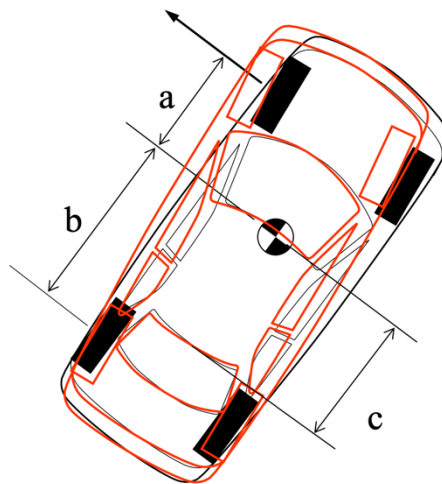


Figure 6-4 - Calculation of dynamic index (Sandford, 2017)

Figure 6-4 shows how if:

$c = b$ then dynamic index = 1

$c < b$ then dynamic index = <1

$c > b$ then dynamic index = >1

7 Mechanical torque path to front wheels

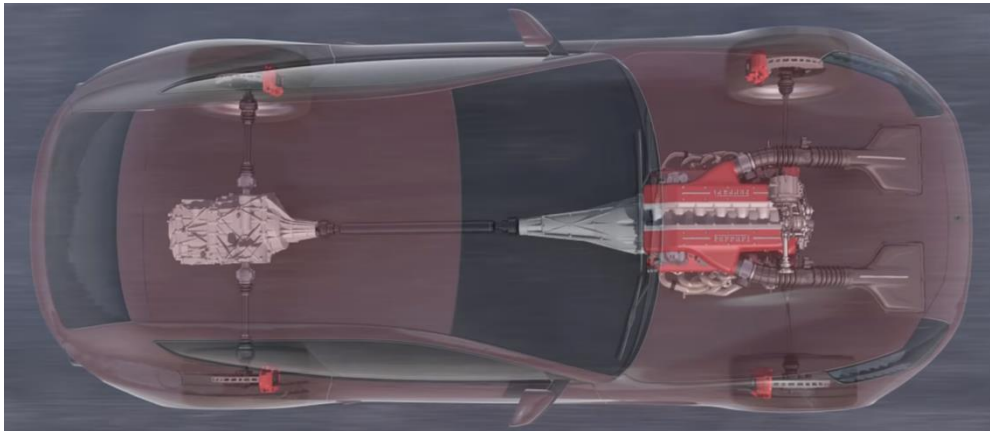


Figure 7-1 - Ferrari FF drivetrain components, including Power Transfer Unit (Caroto.gr, 2011)

Figure 7-1 shows the drivetrain components, including the PTU for the Ferrari FF. Assuming the maximum amount of torque is transferred to the front axle, 136.6Nm, the maximum front wheel speeds were calculated in each gear.

Table 7-1 - Front and rear wheel road speeds in each gear at 1000RPM engine

Gear	Gear Ratio	Final drive	Overall ratio	Rear wheel road speed (RPM)	Front wheel road speed (RPM)
1	3.4	4.38	14.89	67.15	111.01
2	2.18		9.55	104.73	
3	1.63		7.14	140.07	
4	1.29		5.65	176.98	
5	1.03	N/A	4.51	221.66	N/A
6	0.84		3.68	271.80	
7	0.63		2.76	362.40	

@1000RPM engine

@1000RPM engine

Table 7-1 shows the front and rear wheel road speeds with the engine at 1000RPM. Each gear and its gear ratio are shown using data found at caranddriver.com.

The PTU gearbox driving the front wheels, uses its first gear ratio for first and second in the main gearbox and uses second gear ratio for third and fourth in the main gearbox. In gear five, six and seven in the main gearbox, the front wheels are no longer driven. In each gear in the main gearbox, the front wheel speeds can be at any speed up to that shown in Table 7-1 as it uses clutches to vary the wheel speed.

Table 7-2 - Power and slip across the clutch pack at 6000RPM engine when first gear is selected in the main gearbox and in the PTU gearbox

	1st gear front box / 1st gear main box	
1st gear main	402.90	RPM
1st gear front	666.08	RPM
Difference	263.18	RPM
Torque	1230.48	Nm
Power	33912.19	W
Power across clutch pack	33.91	kW
Slip across clutch pack	65.32	%

Table 7-2 shows the power and slip across the clutch pack at 6000RPM engine when first gear is selected in the main gearbox and in the PTU gearbox. A power of 33.91kW is seen across the clutches assuming the left and right wheels share the torque evenly. In this gear configuration, the maximum slip is seen across the clutches, and therefore, the maximum power across the clutch packs.

Table 7-3 - Power and slip across the clutch pack at 6000RPM engine when second gear is selected in the main gearbox and first gear in the PTU gearbox

	1st gear front box / 2nd gear main box	
2nd gear main	628.38	RPM
1st gear front	666.08	RPM
Difference	37.70	RPM
Torque	1230.48	Nm
Power	4858.21	W
Power across clutch pack	4.86	kW
Slip across clutch pack	6.00	%

Table 7-3 shows the power and slip across the clutch pack at 6000RPM engine when second gear is selected in the main gearbox and first gear in the PTU gearbox. A power of 4.86kW is seen across the clutches assuming the left and right wheels share the torque evenly.

Table 7-4 - Power and slip across the clutch pack at 6000RPM engine when third gear is selected in the main gearbox and second in the PTU gearbox

2nd gear front box / 3rd gear main box		
3rd gear main	840.41	RPM
2nd gear front	1125.62	RPM
Difference	285.22	RPM
Torque	728.13	Nm
Power	21747.68	W
Power across clutch pack	21.75	kW
Slip across clutch pack	33.94	%

Table 7-4 shows the power and slip across the clutch pack at 6000RPM engine when third gear is selected in the main gearbox and second in the PTU gearbox. A power of 21.75kW is seen across the clutches assuming the left and right wheels share the torque evenly.

Table 7-5 - Power and slip across the clutch pack at 6000RPM engine when fourth gear is selected in the main gearbox and second in the PTU gearbox

2nd gear front box / 4th gear main box		
4th gear main	1061.91	RPM
2nd gear front	1125.62	RPM
Difference	63.71	RPM
Torque	728.13	Nm
Power	4858.21	W
Power across clutch pack	4.86	kW
Slip across clutch pack	6.00	%

Table 7-5 shows the power and slip across the clutch pack at 6000RPM engine when fourth gear is selected in the main gearbox and second in the PTU gearbox. A power of 4.86kW is seen across the clutches assuming the left and right wheels share the torque evenly.

8 Clutch pack analysis

8.1 Torque capacity

The maximum torque entering the clutch packs is 136.6Nm. After it has been multiplied by the PTU's 1st gear ratio, a torque of 1230.5Nm enters the clutch packs. Assuming that there are 3 plates and 6 friction surfaces per clutch pack, then the capacity of each PTU clutch is 205.1Nm.

8.2 Approximate sizing

The PTU unit is relatively small, at barely 170mm long with a weight of 45kg for the entire unit. For each clutch an outer radius of 10mm and an inner radius of 6mm was assumed.

8.3 Axial force application values

According to RB Racing, the coefficient of friction for a wet carbon fibre friction surface is approximately 0.05 - 0.1. Therefore, a coefficient of friction of 0.07 was assumed. The axial force required is approximately 41850N (see *Table 8-1*).

Table 8-1 - Axial force required on the clutch

Clutch torque capacity	205.08	Nm
Number of friction surfaces	6	
Coefficient of friction	0.1	
Inner radius	0.006	m
Outer radius	0.01	m
Axial force	41853.15	N

8.4 Power loss characteristics

Since the PTU is such a small unit, the power loss compared to a standard gearbox is potentially reduced. Inside the PTU, the gears and clutches are bathed in oil, so when the clutches are fully locked there may be less friction in comparison to a standard gearbox. The power to the front wheels follows a shorter path in comparison to the rear wheels, therefore, the power loss could be lower.

9 Potential longitudinal acceleration improvements

Mass of car with driver 1970 kg
 Weight front 47.0%
 Weight rear 53.0%

Vertical load on front axle 9083.1 N
 Vertical load on rear axle 10242.621 N

Assuming all available grip is used

Above is the calculated vertical load on the front and rear axles.

Table 9-1 - Potential longitudinal acceleration improvements on a variety of surfaces

Asphalt ($\mu=1$)	Gravel ($\mu=0.6$)	Hard packed snow ($\mu=0.15$)
Available grip front (N)	Available grip front (N)	Available grip front (N)
9083.1	5449.8	1362.5
Available grip rear (N)	Available grip rear (N)	Available grip rear (N)
10242.6	6145.6	1536.4
RWD max acceleration (m/s^2)	RWD max acceleration (m/s^2)	RWD max acceleration (m/s^2)
5.20	3.12	0.78
4RM max acceleration (m/s^2)	4RM max acceleration (m/s^2)	4RM max acceleration (m/s^2)
9.81	5.89	1.47

Table 9-1 shows how, assuming that all the available grip is used, the 4RM four-wheel drive system has an 88.7% improvement in acceleration when compared to a two-wheel drive system.

10 Yaw rate increase

Cornering G	0.9	<i>*Assumed*</i>
Lateral acceleration	8.83 m/s ²	
Track width front	1.676 m	
Track width rear	1.660 m	
Mass of car with driver	1970 kg	
Weight front	47.0%	
Weight rear	53.0%	
Centre of gravity height	0.6 m	<i>*Assumed*</i>
Corner radius	25 m	<i>*Assumed*</i>
Velocity	14.86 m/s	
Moment of inertia (ZZ)	4187.4 kgm ²	
Steady state yaw rate	10.57 sec/lap	
	34.05 deg/sec	

Lateral slip across one axle must be the same

For steady state, slip angle L = slip angle R

Therefore, lateral grip is shared across the axle in proportion to the vertical loading

Rolling resistance	289.9 N
Aerodynamic resistance	258.5 N
Drive force required	548.4 N
Power	8147.5 W
	10.9 hp

Above are the specifications used for calculation of yaw rate gain at the front and rear of the car.

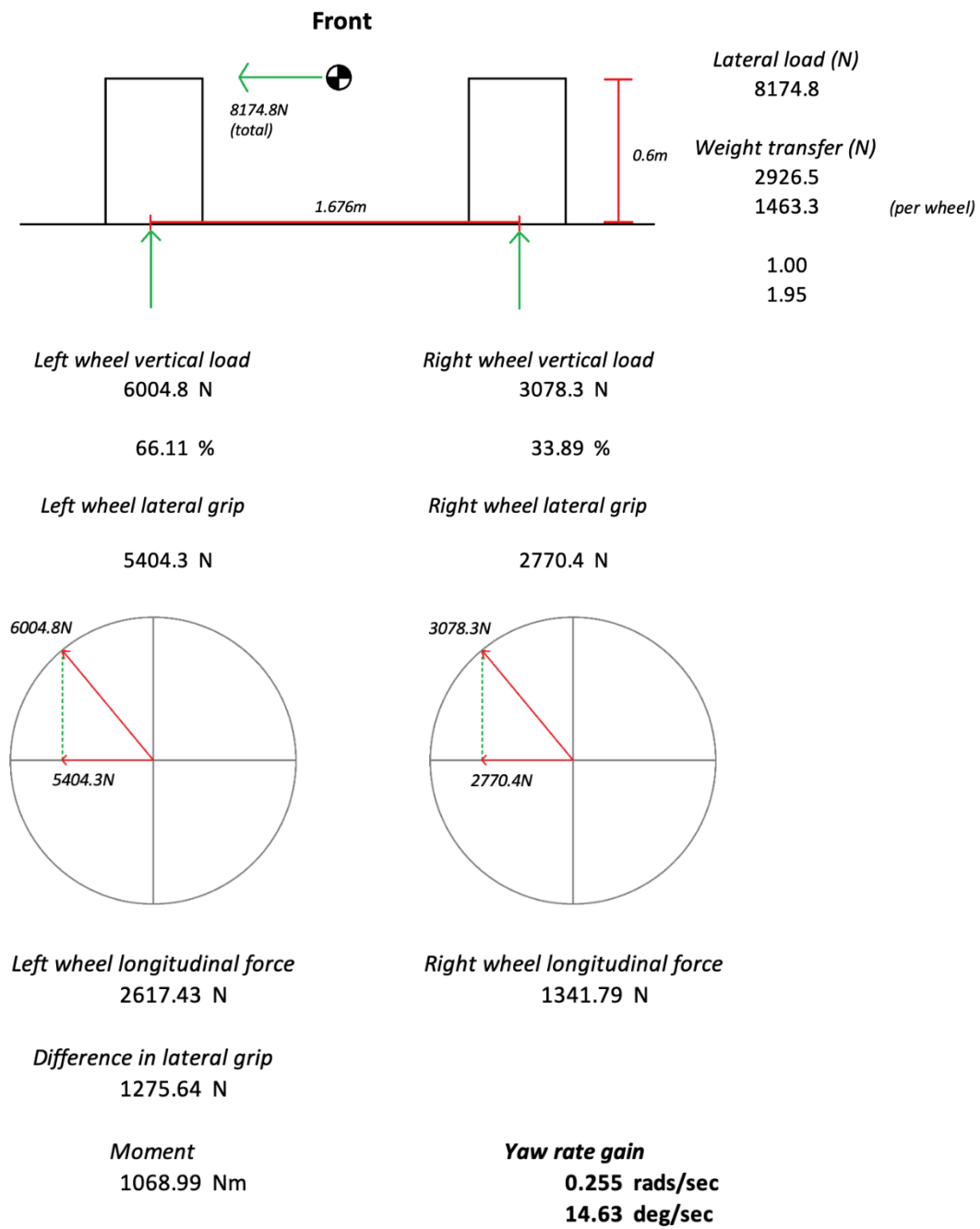


Figure 10-1 – Calculation of yaw rate gain at the front of the car

Figure 10-1 shows the calculation of yaw rate gain at the front of the car. A value of 14.6 deg/sec was obtained.

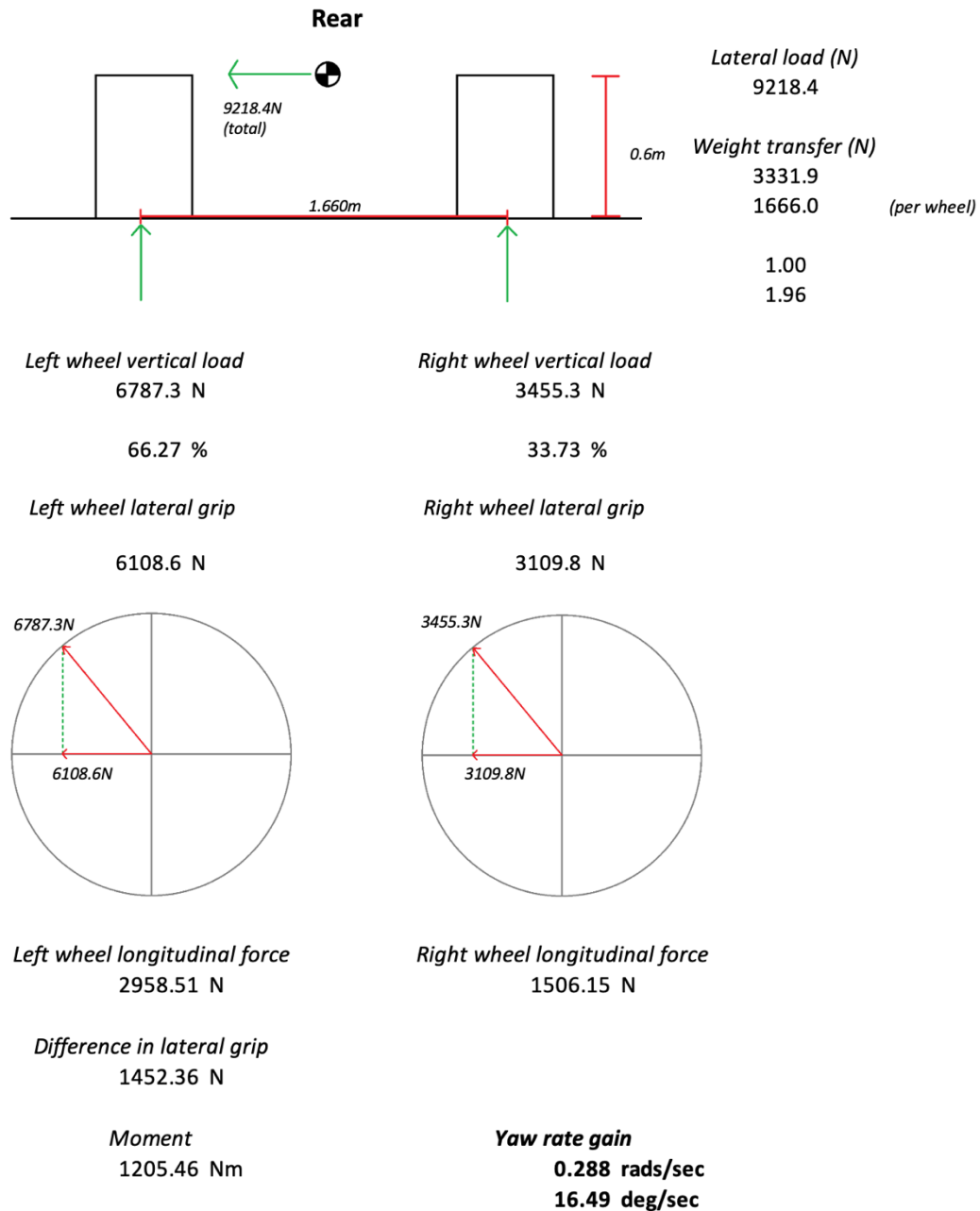


Figure 10-2 - Calculation of yaw rate gain at the rear of the car

Figure 10-2 shows calculation of the yaw rate gain at the rear of the car. A value of 16.5 deg/sec was obtained.

Due to the yaw rate gain, less steering input will be required whilst cornering. The FF already has a quick steering ratio at 11.5 to 1, however, this yaw rate gain will result in a more responsive steering feel. The corner entry and exit speeds will be improved.

11 Possible control algorithms

To effectively distribute the torque to each wheel, the tyre forces must be known. Measuring these directly is very difficult, therefore, approaches to approximate these need to be employed.

'Direct measurement of these forces is not always a solution either, as these forces are very difficult to measure.'
(Parra, et al., 2018)

According to Parra, et al. (2018), the tire loads can be estimated using tyre models such as the linear tyre model, Dugof 's model, or semiempirical models such as the Pacejka's tire model.

An in-vehicle GPS sensor could be used to measure lateral, longitudinal and vertical acceleration. This GPS sensor could estimate tyre loads by approximating the lateral and longitudinal load transfer. This GPS sensor could approximate the yaw rate of the vehicle. Using the yaw rate, the system could sense oversteer and understeer, and correct by generating a yaw moment. To detect wheel slip, wheel speed sensors could be used.

'Acura, for instance, has offered its Super Handling All-Wheel-Drive (SH-AWD) system for several years. It monitors vehicle speed, wheel speed, gear position, steering angle, yaw rate, lateral G forces and other inputs, while automatically adding torque to the outside rear wheel in corners to make the car turn quicker.'

(McCraw, 2009)

Driver inputs must be known, as these could affect the systems response. The throttle position must be taken into consideration as the torque vectoring system would obviously have to stop during no throttle. The brake position must be taken into consideration, as each of the wheels could be engaged during heavy braking and engine braking used to improve deceleration. This could coincide with the ABS and EBD. The steering position must be taken into consideration. If the car started to oversteer and the driver responded by turning the opposite way, then the system would need to detect this. Also yaw rate and steering position would need to be compared, as driver corrections would need to be detected by the system.

The torque vectoring systems must work alongside the Electronic Stability Control (ESC) and Anti-lock Braking System (ABS) / Electronic Brakeforce Distribution (EBD). The car must know how to use torque vectoring and ESC in conjunction. For example, in a corner the outside wheel could have

more torque sent to it, and the inside wheel could have the brake applied using ESC to create a larger yaw moment.

During on and off power, the vehicle would need to sense for wheel slip as this could induce kangaroo effect, a rocking about the centre of gravity. To limit this effect power could be added gradually as throttle is applied. This method is employed in the Ferrari FF, where power is applied progressively to the front wheels. During straight line accelerations the torque vectoring differentials would need to behave like a standard open differential until wheel slip is detected.

12 Conclusion

In conclusion, torque vectoring can improve a vehicles response especially during cornering. The 4RM system allowed Ferrari to maintain a front-mid engine layout keeping the normal dynamics of a Ferrari. In all weather conditions, the 4RM system clearly increases performance.

'The FF first introduced the 4RM system integrated with the E-Diff, F1-Trac, SCM and ESC to guarantee superb drivability in all weather conditions...'

(Ferrari, n.d.)

The increased safety aspect removed from the Ferrari experience. The 4RM system was not popular with everyone, as it did not compare to most other Ferraris.

'From an emotive point of view the addition of all-wheel drive has eroded a little of the interaction often expected with a Ferrari.'

(Auto Car, n.d.)

The handling was more focused at safety and driveability compared to sheer driving pleasure, reaching out to a wider customer base.

13 References

Auto Car, n.d. *Ferrari FF*. [Online]

Available at: <https://www.autocar.co.uk/car-review/ferrari/ff-2011-2016/ride>

Car And Driver, n.d. *Ferrari FF Features And Specs*. [Online]

Available at: <https://www.caranddriver.com/ferrari/ff/specs>

Caroto.gr, 2011. *Ferrari FF 4RM System*. [Online]

Available at: <https://www.youtube.com/watch?v=IPHAtT8BQXs&t=26s>

Ferrari, n.d. [Online]

Available at: https://auto.ferrari.com/en_EN/wp-content/themes/mobile.ferrari-gt/model-templates/innovation-all.php?id=23652

Ferrari, n.d. *Ferrari FF*. [Online]

Available at: <https://www.ferrari.com/en-EN/auto/ff>

Ferrari, n.d. *Photos*. [Online]

Available at: <https://www.ferrari.com/en-EN/auto/ff>

ItzKirb Photography, 2018. [Online]

Available at: <https://pixels.com/featured/9-ferrari-ff-print-itzkirb-photography.html>

Kingston, L., 2018. *What is torque vectoring? PH Explains*. [Online]

Available at: <https://www.pistonheads.com/news/features/what-is-torque-vectoring-ph-explains/38379>

McCraw, J., 2009. *Torque Vectoring: The Hyper-Smart, Fuel-Efficient Future of All-Wheel Drive*.

[Online]

Available at: <https://www.popularmechanics.com/cars/a2178/4225886/>

Parra, A., Zubizarreta, A., Pérez, J. & Dendaluze, M., 2018. *Intelligent Torque Vectoring Approach for Electric Vehicles with Per-Wheel Motors*, Bilbao: Wiley.

RB Racing, n.d. *Clutch Technology & Clutch Torque Calculator*. [Online]

Available at: <https://www.rbracing-rsr.com/clutchtech.html>

Sandford, M., 2017. *Past Paper*. [Online].

14 List of plates

Plate 2-1 - Ferrari FF driving on hard packed snow (Ferrari, n.d.)	1
Plate 5-1 - Ferrari Power Transfer Unit (Caroto.gr, 2011)	5
Plate 6-1 - Side view of Ferrari FF (ItzKirb Photography, 2018)	6

15 List of tables

Table 3-1 - Base Vehicle Data (Ferrari, n.d.)	2
Table 7-1 - Front and rear wheel road speeds in each gear at 1000RPM engine	10
Table 7-2 - Power and slip across the clutch pack at 6000RPM engine when first gear is selected in the main gearbox and in the PTU gearbox	11
Table 7-3 - Power and slip across the clutch pack at 6000RPM engine when second gear is selected in the main gearbox and first gear in the PTU gearbox.....	11
Table 7-4 - Power and slip across the clutch pack at 6000RPM engine when third gear is selected in the main gearbox and second in the PTU gearbox	12
Table 7-5 - Power and slip across the clutch pack at 6000RPM engine when fourth gear is selected in the main gearbox and second in the PTU gearbox	12
Table 8-1 - Axial force required on the clutch.....	13
Table 9-1 - Potential longitudinal acceleration improvements on a variety of surfaces.....	15

16 List of figures

Figure 6-1 - Ferrari FF lumped mass model	6
Figure 6-2 - Position of the inertial conjugate	7
Figure 6-3 - Distances used to calculate the dynamic index.....	8
Figure 6-4 - Calculation of dynamic index (Sandford, 2017).....	9
Figure 7-1 - Ferrari FF drivetrain components, including Power Transfer Unit (Caroto.gr, 2011)	10
Figure 10-1 – Calculation of yaw rate gain at the front of the car	17
Figure 10-2 - Calculation of yaw rate gain at the rear of the car	18