



UNIVERSITY OF WALES TRINITY SAINT

DAVID

SCHOOL OF ENGINEERING

ADVANCED VEHICLE DYNAMICS

AWD TORQUE VECTORING: FERRARI FF

CRITICAL ANALYSIS

M6X00407

MAXIMILIAN HART

1700625

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Abstract

This report critically analyses the vehicle dynamics and torque vectoring AWD capabilities of the 2011 Ferrari FF. The report finds that the clever geometry of the vehicle makes for unexpectedly good handling characteristics for such a large car, whilst the novel AWD system provides all the advantages of a generic torque vectoring system, with a fraction of the mass and mechanical losses.

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1 Introduction

This report shall critically analyse the vehicle dynamics and torque vectoring systems used within the Ferrari FF. This grand tourer includes a novel all-wheel drive (AWD) torque vectoring system, increasing acceleration and yaw rate capabilities on various surfaces (Ferrari, 2012). Within the analysis, the author shall research how this technology is applied and how the system differs to conventional methods, as well as numerically investigating how performance is enhanced compared to rival vehicles.

1.1 Base Vehicle Data

Table 1 - Base Vehicle Specification

<u>Data</u>	<u>Value</u>
<u>Dimensions and Masses</u>	
Length	4907mm
Width	1953mm
Height	1379mm
Wheelbase	2990mm
Front track	1676mm
Rear track	1660mm
Kerb weight (full fuel tank & fluids)	1880kg
Weight distribution % (front / rear)	47 : 53
<u>Tyres and Rims</u>	
Front tyres	245/35 ZR 20 10.5 J x 20"
Rear tyres	295/35 ZR 20 10.5 J x 20"
<u>Engine</u>	
Engine	6262cc 65° V12
Maximum power	659PS @ 8000rpm
Maximum torque	683Nm @ 6000rpm
<u>Performance</u>	
Maximum speed	208mph / 335kmh / 92.9843ms ⁻¹
Maximum acceleration	0-62mph in 3.7 seconds
<u>Transmission</u>	
Rear 1 st gear ratio	3.40 : 1
Rear 2 nd gear ratio	2.18 : 1
Rear 3 rd gear ratio	1.63 : 1
Rear 4 th gear ratio	1.29 : 1
Rear 5 th gear ratio	1.03 : 1
Rear 6 th gear ratio	0.84 : 1
Rear 7 th gear ratio	0.63 : 1
Final drive ratio	4.83 : 1
Front 1 st gear ratio	6% taller than rear 2 nd gear → 9.008
Front 2 nd gear ratio	6% taller than rear 4 th gear → 5.330

(Ferrari, 2011; DriverSide, 2015)

1.2 Torque Vectoring Systems

Torque vectoring is an effective method of increasing traction when a vehicle is under excessive dynamic load, significantly increasing acceleration and cornering rates across various surfaces by sending more torque to the wheel with most available grip. Under acceleration, this provides a higher tractive force across the axle with most grip, increasing acceleration. When cornering, this provides higher tractive force on the outside wheels, producing a moment about the lateral centre of gravity position, increasing yaw rate, and thus aiding cornering (Yang *et al.*, 2018). These effects will be numerically analysed later in the project.

This difference in torque can be achieved either by sending a higher amount of power to a designated area to increase tractive force, or by braking a designated area to reduce tractive force. Electronic differentials are generally used to deliver more power to desired areas, by mechanically diverting a higher percentage of torque to the appropriate wheel or axle (Kato and Sawase, 2012). Moreover, an independent braking system can be used to brake a specific area of the vehicle to reduce the tractive force. This, again, will be controlled electronically and the system will integrate with a conventional braking system, that allows pressure to be applied to one or two callipers individually (Day, 2014).

The Ferrari FF torque vectoring system consists of a conventional electronic differential at the rear to send torque to either wheel where necessary. The system at the front, however, is unusual in that there is no differential to control torque distribution. Each wheel has a multi-plate clutch pack that constantly slips; each clutch can be engaged independently to control the amount of torque applied to each wheel (Ferrari, 2011). This system will be further explained later in the report. The author found no evidence of a brake actuated torque vectoring system on the vehicle.

2 Vehicle Dynamics Analysis

2.1 Lumped Mass Model

To start to fully analyse the vehicle dynamics, the author produced a lumped mass model, clearly showing where each major component of the vehicle lies along the longitudinal axis. Each lumped mass represents the centre of mass of each component. By taking moments about the centre of gravity and equating them when in equilibrium, the distances could be found. To confirm this, the author opened a top view of the vehicle in photo editing software and measured the distances between each component. By sizing the photo to a certain scale, accurate distances could be calculated. Specific mass data was hard to find, so values were calculated from assumptions and research:

Table 2 - Vehicle Masses

<u>COMPONENT</u>	<u>MASS (KG)</u>	<u>NOTES / SOURCE</u>
Engine and power transfer unit (PTU)	172	An average was taken of various Ferrari 65° V12 masses and increased slightly to account for the PTU (Farmer, 2012).
2 adult front passengers	153.8	The average masses of a British male and female adult were added (BBC, 2010).
2 teenage rear passengers	108.1	The average masses of a British male and female teenager were added (BBC, 2010).
Transaxle	100kg	An average of various 7-speed automatic transaxles were taken (Farmer, 2012).

To account for the mass of the chassis and other components, the remaining mass was taken as acting on the front and rear axle, assuming 53% acts at the rear to account for a full tank of fuel, exhaust system and other driveline components.

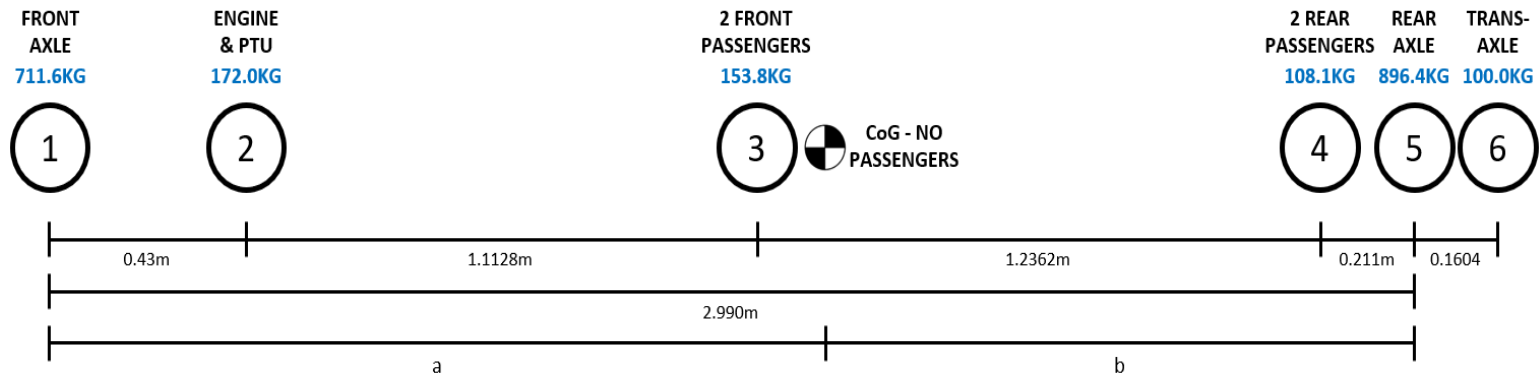


Figure 1 - Lumped Mass Model

2.2 Longitudinal Centre of Gravity (CoG) position

The longitudinal position of the centre of gravity is defined by lengths a and b shown in figure one. These change depending on how many passengers are in the vehicle and were calculated by finding the percentage of mass before and after the standard CoG position and multiplying the wheelbase by this ratio.

Table 3 - Longitudinal CoG Positions

PASSENGERS	FRONT / REAR %	LENGTH A (m)	LENGTH B (m)	TOTAL MASS (KG)
None	47 / 53	1.5847	1.4053	1880.0
Front	51 / 49	1.4651	1.5249	2033.8
Front & rear	48.5 / 51.5	1.5399	1.4501	2141.9

Table three shows that passengers even out the weight distribution even more. This due to the design, with the front passengers sitting almost directly on the centre of gravity, and the rear passengers situated to try to even out the moments from front masses when taken about the centre of the car (Adams, 1992).

2.3 Moment of Inertia About Z Axis

A vehicle's moment of inertia is its resistance to angular acceleration when rotating (Atkinson, 1968). Because of this, a vehicle with a larger moment of inertia will generally experience worse handling, being prone to understeer and feel sluggish (Adams, 1992).

The moment of inertia is calculated from the sum of the products of each mass in figure one and the square of their distance from the centre of gravity:

$$I_{YY} = \sum (m \cdot x^2)$$

Equation 1 - Moment of Inertia about Y Axis

2.3.1 No Passengers

$$I_{YY} = (711.6)(1.5847)^2 + (172)(1.1547)^2 + (896.4)(1.4654)^2 + (100)(1.5658)^2 \\ = \mathbf{4032.052 \text{ KGm}^2}$$

2.3.2 Front Passengers

$$I_{YY} = (711.6)(1.4651)^2 + (172)(1.0351)^2 + (153.8)(0.0777)^2 + (896.4)(1.5249)^2 \\ + (100)(6854)^2 = \mathbf{4081.151 \text{ KGm}^2}$$

2.3.3 Front and Rear Passengers

$$I_{YY} = (711.6)(1.5399)^2 + (172)(1.1099)^2 + (153.8)(0.0029)^2 + (108.1)(1.2362)^2 \\ + (896.4)(1.4501)^2 + (100)(1.6106)^2 = \mathbf{4209.640 \text{ KGm}^2}$$

It is clear that there is only a 4% increase in moment of inertia when the car has full occupancy, this is again down to clever design and mass distribution.

Table 4 - Experimental Moment of Inertia Examples - No Passengers

VEHICLE	MOMENT OF INERTIA I_{YY} (KGm ²)
Ford Focus Hatchback (2006)	2170.1
Peugeot Partner Van (2005)	2965.4
Chrysler Gran Voyager (2004)	4651.5

(Engineering Solutions, 2010)

These results, when compared to other examples can be taken as valid. The Focus and Partner, both expected to have somewhat snappy handling have considerably lower moments of inertia, whereas the Gran Voyager, expected to have sluggish, cumbersome handling shows a much higher value. The results for the Ferrari FF confirm that this is a large car, however it is designed with performance handling in mind.

2.4 Inertial Conjugate

As a vehicle corners, linear and rotational acceleration will be experienced at different points. A point behind the centre of gravity will experience zero linear acceleration. This is the vehicle's pivot point, or inertial conjugate (Blundell and Harty, 2014). The author calculated the inertial conjugate's distance behind the centre of gravity:

$$\text{Inertial Conjugate } (C) = \frac{I_{zz}}{m \cdot a}$$

Equation 2 - Inertial Conjugate

2.4.1 No Passengers

$$C = \frac{4032.052}{(1880)(1.5847)} = \mathbf{1.353m}$$

2.4.2 Front Passengers

$$C = \frac{4081.151}{(2033.8)(1.4651)} = \mathbf{1.369m}$$

2.4.3 Front and Rear Passengers

$$C = \frac{4209.640}{(2141.9)(1.5399)} = \mathbf{1.276m}$$

These values can be best used to analyse vehicle dynamics by calculating the vehicle's dynamic index. This is a key stage of the inertia match theory, which derives these numerical components to establish how a vehicle will corner.

2.5 Dynamic Index

The dynamic index (DI) relates the position of the inertial conjugate to the position of the rear axle. When the DI is less than one, the inertial conjugate is in front of the rear axle. When the car turns into a corner, the rear tyres will move into the turn; this makes the slip angle build against the cornering tractive effort, making the vehicle more cumbersome and susceptible to understeer. When the DI is greater than one, the opposite is true, making the slip angle build with the cornering tractive effort, making the vehicle more twitchy and susceptible to oversteer (Bastin, 2014).

$$\text{Dynamic Index } (DI) = \frac{I_{zz}}{m \cdot a \cdot b}$$

Equation 3 - Dynamic Index

2.5.1 No Passengers

$$DI = \frac{4032.052}{(1880)(1.5847)(1.4053)} = \mathbf{0.963}$$

2.5.2 Front Passengers

$$DI = \frac{4081.151}{(2033.8)(1.4651)(1.5249)} = \mathbf{0.898}$$

2.5.3 Front and Rear Passengers

$$DI = \frac{4209.640}{(2141.9)(1.5399)(1.4502)} = \mathbf{0.880}$$

There is often disagreement as to what an optimal DI is. Some suggest that the lowest value possible is best, and it is up to driver ability to control it (Milliken and Milliken, 1995). However, it is more commonly considered that a well-balanced car may have a DI of between 0.90 and 0.93 (Blundell and Harty, 2014). The Ferrari FF shows a very good balance overall, with the possibility of slightly more twitchy handling when loaded with passengers and luggage, especially rear passengers. Overall the DI values are very close to the generally accepted values of a well-balanced vehicle, which is a fantastic design achievement for a large car.

2.6 Dynamic Base Vehicle Specification

As has been discussed in detail, the Ferrari FF has an expectedly large moment of inertia compared to other performance vehicles, but not compared to vehicles that offer as much space, comfort, and practicality. Because of this, the dynamic index values are slightly lower than would be expected in other performance vehicles, but again much higher than would be expected in vehicles offering as much practicality. The numerical analysis clearly shows that Ferrari have put lots of thought into the geometry of the vehicle, with each component placed to keep weight distribution as even as possible. Despite this, the size of the vehicle cannot be ignored and does reduce dynamic performance somewhat; any detrimental effects to handling this may have can be greatly reduced by the torque vectoring system, as analysed in the following section of the project.

3 Torque Vectoring Analysis

3.1 Mechanical Torque Path

In a performance vehicle driving all four wheels, torque is usually sent to through a drive shaft to a transaxle to power the rear wheels. The torque is then put through a gearbox and sent back along a parallel, counter-rotating drive shaft (Bir A Gill *et al.*, 2018). This is heavy and produces significant mechanical losses. This type of system is seen on the Nissan GTR.

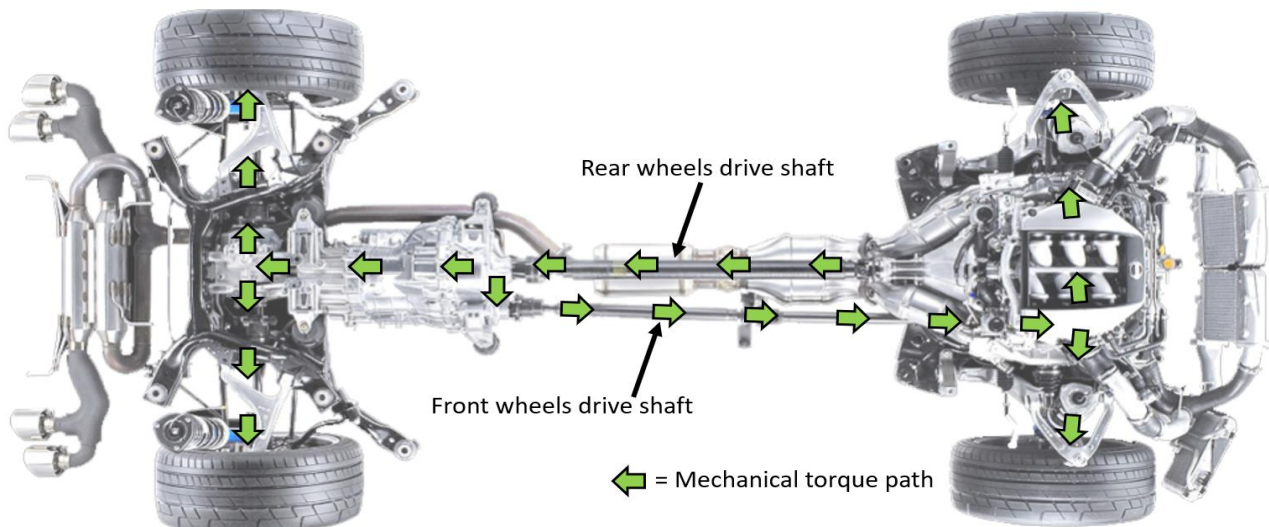


Figure 2 - Nissan GTR Mechanical Torque Path (Nissan, 2015)

The Ferrari FF sends torque to the front directly from the crankshaft. A high speed bezel connects this to a small, front mounted gearbox (PTU), which distributes torque to each axle. No differential is needed as a multi-plate wet clutch pack sits on each axle to transfer drive (Ferrari, 2011). There is a significant overspeed from the PTU to the road wheels, so the clutches constantly slip to equalise this. As the front wheels are not always driven, Ferrari state the clutches will not overheat (Autocar, 2011). The clutch packs are individually electronically controlled to enable torque vectoring, achieving a torque differential across the axle. Despite the need for extra clutches, Ferrari say this reduces both weight and mechanical losses.

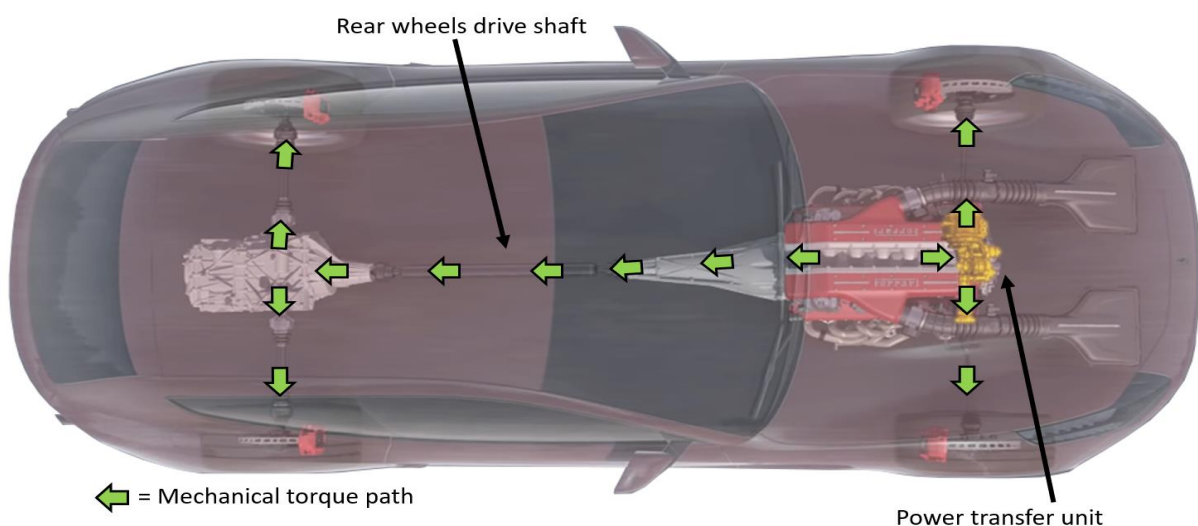


Figure 3 - Ferrari FF Mechanical Torque Path (Ferrari, 2011)

3.2 Front Axle Clutch Analysis

3.2.1 Maximum Torque Capacity

The maximum torque capacity of the front clutches should be the maximum value the system can send to them from the engine. This allows full performance potential to pass through them, whilst acting as a mechanical fuse, ensuring any torque spikes cannot damage the driveshaft or wheels.

$$\text{Maximum engine torque} = 683\text{Nm @ 6000RPM}$$

$$\text{Torque is split } 20 : 80 \text{ (front : rear)}$$

$$\therefore \text{Torque at each front wheel} = 683 \cdot 0.1 = 68.3\text{Nm}$$

Considering:

$$\text{Maximum clutch torque capacity} = 68.3 \cdot \text{Gear Ratio}$$

$$\text{Road wheel angular velocity} = \frac{\text{Engine RPM}}{\text{Gear ratio}}$$

Equation 4 - Clutch Torque Capacity Equations

Table 5 - Clutch Maximum Torque Capacity Calculations

	GEAR RATIOS		FINAL DRIVE	OVERALL	ROAD WHEEL SPEED @ 6k RPM (RPM)	MAXIMUM TORQUE THROUGH FRONT CLUTCH PACK (Nm)
REAR BOX	1	3.400	4.380	14.892	402.901	-
	2	2.180	4.380	9.548	628.378	-
	3	1.630	4.380	7.139	840.407	-
	4	1.290	4.380	5.650	1061.909	-
	5	1.030	4.380	4.511	1329.964	-
	6	0.840	4.380	3.679	1630.789	-
	7	0.630	4.380	2.759	2174.386	-
FRONT BOX	1	9.008	-	9.008	666.080	615.241
	2	5.330	-	5.330	1125.624	364.065

From table five, it is clear that the maximum torque capacity of the clutch is **615.241Nm**. This is in first gear as it has the highest ratio, thus producing the highest instantaneous torque.

3.2.2 Power Loss Across Clutch Packs

Considering:

$$Power = Torque \cdot Angular\ velocity$$

$$\therefore Power\ Loss = Torque\ through\ clutch\ pack \cdot \Delta Wheel\ Speeds$$

Equation 5 - Clutch Pack Power Loss Equations

The following calculations can be made using data from table five:

Table 6 - Clutch Pack Power Loss Calculations

GEAR COMBINATION		Δ WHEEL SPEED (RPM)	POWER LOSS ACROSS EACH FRONT CLUTCH (Kw)	POWER LOSS ACROSS EACH FRONT CLUTCH (bhp)	POWER LOSS ACROSS FRONT AXLE (Kw)	POWER LOSS ACROSS FRONT AXLE (bhp)
FRONT	REAR					
1	1	263.179	10.034	13.455	20.067	26.911
1	2	37.703	1.437	1.928	2.875	3.855
2	3	285.217	10.874	14.582	21.748	29.164
2	4	63.715	2.429	3.257	4.858	6.515

Table six shows that the maximum power loss is experienced when the rear 3rd and front 2nd gears are selected. Overall, a maximum of **21.758Kw (29.164bhp)** is lost the clutch. This is only and 4.55% of the maximum power output of the engine. It is generally considered that more conventional AWD system run at losses of up to 25% across the entire drivetrain, so losing this small amount at the front is a significant improvement (Pennycott *et al.*, 2014).

3.2.3 Clutch Sizing and Required Axial Force

To fully analyse the clutches on the front axle, the author used the constant pressure and constant wear theories to find required clutch sizes and axial forces to transmit the required torque. The clutch packs consist of two 3 plate, wet clutches (Ferrari, 2011). Because of this, the author took the coefficient of friction (μ) to be 0.07 (Holgerson, 1997). Moreover, as a starting point for sizing, the author found that a Ferrari 360 main clutch has an outer diameter (d2) of 250mm and an inner diameter (d1) of 70mm (Farmer, 2012) so axial force (W) could be calculated.

$$Constant\ pressure\ torque = \frac{2}{3} \cdot \mu \cdot W \cdot \frac{r_2^3 - r_1^3}{r_2^2 - r_1^2}$$

Equation 6 - Constant Pressure Theory

$$Constant\ wear\ torque = \frac{\mu \cdot W \cdot (r_2 + r_1)}{2}$$

Equation 7 - Constant Pressure Theory

CLUTCH DATA						
OUTSIDE RADIUS r2 (m)	INSIDE RADIUS r1 (m)	FRICTION COEFFICIENT μ	No. OF PLATES	MAX TORQUE ON EACH PLATE FACE (Nm)	CONSTANT PRESSURE AXIAL FORCE REQUIRED (KN)	CONSTANT WEAR AXIAL FORCE REQUIRED (KN)
0.125	0.070	0.070	6.000	51.270	7.318	7.512
0.094	0.053	0.070	6.000	51.270	9.757	10.016

By dividing the maximum torque through each clutch pack by 12, the torque acting on each face was found, from here calculations to find required axial force could be made. Starting with the clutch size of the Ferrari 360, a maximum required axial force of **7.512KN** once worn was calculated. The author considered that due to being on the axle, the clutches used for the front wheels are likely to be considerably smaller than a conventional clutch. The author re-calculated the axial force at a clutch 75% of the size, finding **10.016KN** maximum required force once worn.

These values are rather high but thought to not be unreasonable. The axial force is provided by a hydraulic actuator, constantly varying it to provide different levels of torque to the wheels. This fully hydraulic system does not rely on springs like some clutches, allowing a much larger force to be applied (Cimatti and Morawetz, 2010).

Wet clutches were used to reduce frictional losses, but also to aid cooling. The oil is required to transfer the heat produced by the 10Kw lost by each clutch. Due to the constantly slipping nature of the clutches, lots of heat will be dissipated through the oil so cooling is essential.

3.3 Longitudinal Acceleration Improvements from Torque Vectoring

The Ferrari FF is predominantly rear wheel drive. The front wheels only assist tractive effort when loss of grip at the rear wheels affects acceleration. For the purpose of these calculations, the vehicle will have maximum torque sent to all wheels at all times.

It must be considered that, in practice, the vehicle acceleration is limited by surface grip, and therefore tractive force:

$$Force = Mass \cdot Acceleration$$

$$\therefore \text{Maximum tractive force} = \text{Vehicle mass} \cdot g$$

Taking the following values for the rear wheels:

$$\text{Final drive in 1st gear} = 14.892 : 1$$

$$\text{Maximum torque per wheel @ 6000rpm} = 683 \cdot 0.4 \cdot 14.892 = 4068.49\text{Nm}$$

Considering a 20" tyre and therefore assuming the rolling radius to be 0.254m,

$$\therefore \text{Maximum tractive force} = \frac{4068.490}{0.254} = 16017.69\text{Nm per wheel}$$

Taking the following values for the front wheels:

$$\text{1st gear ratio} = 9.008 : 1$$

$$\text{Maximum torque per wheel @ 6000rpm} = 683 \cdot 0.1 \cdot 9.008 = 615.246\text{Nm}$$

Considering a 20" tyre and therefore assuming the rolling radius to be 0.254m,

$$\therefore \text{Maximum tractive force} = \frac{615.246}{0.254} = 2422.210\text{Nm per wheel}$$

Taking the maximum tractive forces front and rear, and reapplying $F = M \cdot a$,

Table 7 - Vehicle Acceleration Data

FRONT TRACTIVE FORCE (N)	REAR TRACTIVE FORCE (N)	MASS F PASSENGERS (kg)	MASS F & R PASSENGERS (kg)
2422.210	16017.694	2033.800	2141.900

Table 8 - Vehicle Acceleration Calculations (Waluś, 2016)

SURFACE	μ	FORCE (N)	ACCELERATION (m/s ²)	
			F PASSENGERS	F & R PASSENGERS
DRY ASPHALT	0.700	25815.866	12.693	12.053
WET ASPHALT	0.400	14751.923	7.253	6.887
PACKED GRAVEL	0.550	20283.895	9.973	9.470
LOOSE GRAVEL	0.450	16595.914	8.160	7.748
PACKED SNOW	0.300	11063.942	5.440	5.165
LOOSE SNOW	0.100	3687.981	1.813	1.722
ICE	0.060	2212.788	1.088	1.033

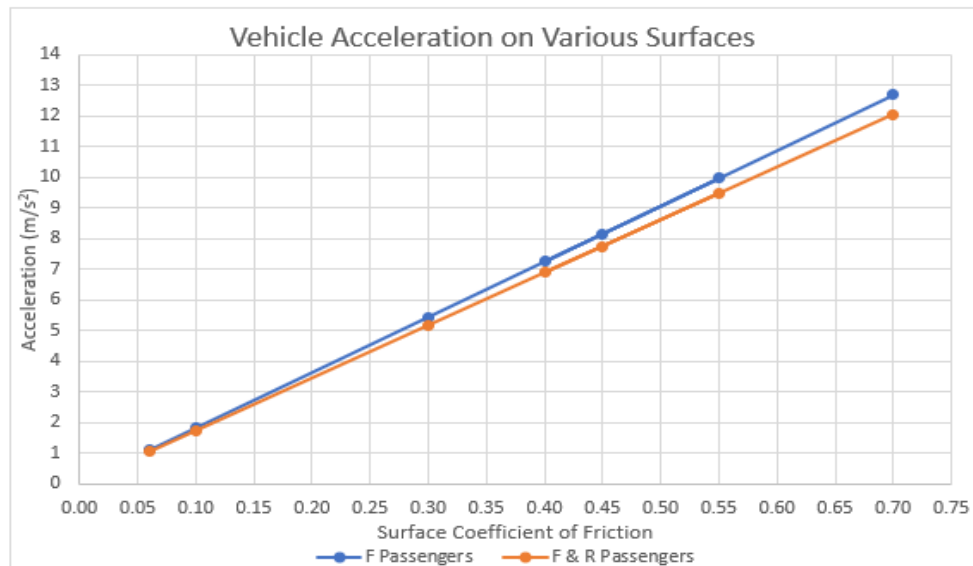


Figure 4 - Longitudinal Accelerations on Various Surfaces

Table eight shows the theoretical maximum acceleration considering all wheels are driving at maximum torque. This is only a theoretical representation of acceleration, as in reality the achievable figures would be considerably lower. Firstly, the car is not at 6000rpm for the full acceleration, moreover, as stated, the maximum available tractive force is significantly lower than the tractive forces calculated above. Moreover, the author presumes that the torque value stated is from the flywheel, thus available torque would be less once mechanical losses have been factored in.

It should be mentioned that this analysis also only considers constant μ surfaces. If one side of the vehicle is on a more slippery surface, acceleration would be compromised. Due to the torque vectoring abilities within the FF, the vehicle could send more torque to the side with most grip for fastest acceleration. This, however, would produce a moment across the car, steering it back to the low μ surface. A yaw rate sensor in the vehicle could sense this and again use torque vectoring to correct this, or the driver could counter-steer (Gillespie, 1992).

To validate these findings, a simple calculation can be made knowing that the Ferrari FF accelerates to 62mph in 3.7 seconds:

$$62\text{mph} = 27.77\text{ms}^{-1}$$

$$\text{Average acceleration} = \frac{\Delta\text{Velocity}}{\Delta\text{Time}} = \frac{27.7}{3.7} = 7.49\text{ms}^{-2}$$

The calculated value for acceleration on asphalt is almost 40% larger than this value, however due to the assumptions stated above, this is an acceptably close result, thus proving the calculations are trustworthy.

3.4 Yaw Rate Improvements from Torque Vectoring

As previously discussed, creating a torque differential across an axle can assist in cornering.

The increase in yaw rate can be calculated as follows:

$$\text{Difference in tractive force} = \text{Outside force} - \text{Inside force}$$

$$\text{Moment produced} = \Delta \text{Force} \cdot \frac{1}{2} \text{Track}$$

$$\text{Yaw rate} = \frac{\text{Moment produced}}{\text{Moment of inertia}}$$

Equation 10 - Yaw Rate Increase Equations

Once again, these calculations are theoretical, to understand the assistive capabilities of torque vectoring. The maximum tractive force of $F = M \cdot a$ would reduce yaw rate in a real-world application. These calculations are carried out for 3rd gear rear and 2nd gear front, as this is a realistic situation where torque vectoring would be utilised.

3.4.1 Analysis of Increasing Torque to Outside Wheels

Table 9 - Acceleration Torque Vectoring Data

FRONT TRACTIVE FORCE (N)	REAR TRACTIVE FORCE (N)	MOMENT ARM (1/2 TRACK) (m)
2866.449	15357.282	0.838

Table 10 - Acceleration Torque Vectoring Calculations

μ		0.700				FRONT PASSENGERS		FRONT & REAR PASSENGERS	
OUTSIDE WHEEL TORQUE %	INSIDE WHEEL TORQUE %	Δ TRACTIVE FORCE (N)	MOMENT PRODUCED (Nm)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)
50	50	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
60	40	2551.322	2138.008	0.524	30.016	0.508	29.100		
70	30	5102.645	4276.016	1.048	60.032	1.016	58.199		
80	20	7653.967	6414.024	1.572	90.047	1.524	87.299		
90	10	10205.289	8552.032	2.095	120.063	2.032	116.398		
100	0	12756.612	10690.041	2.619	150.079	2.539	145.498		

μ		0.500				FRONT PASSENGERS		FRONT & REAR PASSENGERS	
OUTSIDE WHEEL TORQUE %	INSIDE WHEEL TORQUE %	Δ TRACTIVE FORCE (N)	MOMENT PRODUCED (Nm)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)
50	50	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
60	40	1822.373	1527.149	0.374	21.440	0.363	20.785		
70	30	3644.746	3054.297	0.748	42.880	0.726	41.571		
80	20	5467.119	4581.446	1.123	64.320	1.088	62.356		
90	10	7289.492	6108.595	1.497	85.759	1.451	83.142		
100	0	9111.866	7635.743	1.871	107.199	1.814	103.927		

μ		0.100				FRONT PASSENGERS		FRONT & REAR PASSENGERS	
OUTSIDE WHEEL TORQUE %	INSIDE WHEEL TORQUE %	Δ TRACTIVE FORCE (N)	MOMENT PRODUCED (Nm)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)
50	50	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
60	40	364.475	305.430	0.075	4.288	0.073	4.157		
70	30	728.949	610.859	0.150	8.576	0.145	8.314		
80	20	1093.424	916.289	0.225	12.864	0.218	12.471		
90	10	1457.898	1221.719	0.299	17.152	0.290	16.628		
100	0	1822.373	1527.149	0.374	21.440	0.363	20.785		

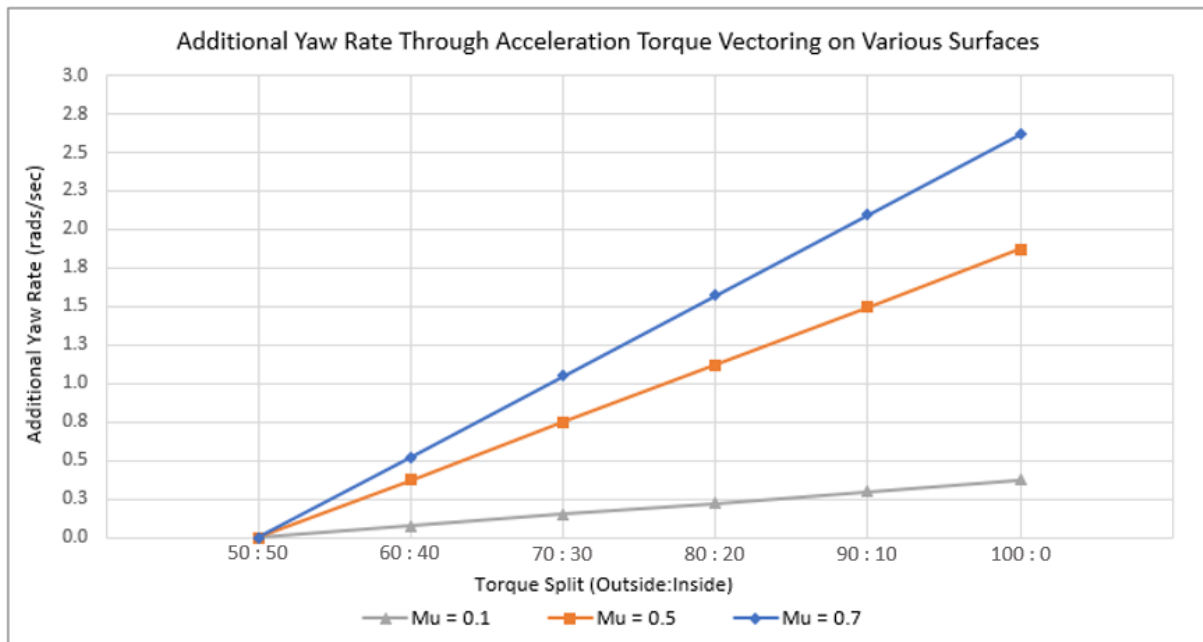


Figure 5 - Acceleration Torque Vectoring on Various Surfaces

Figure five shows a substantial yaw rate increase on every surface by increasing torque to the outside wheels in a corner, proving that accelerative torque vectoring is an efficient way of increasing cornering ability.

3.4.2 Analysis of Reducing Torque to Inside Wheels

As mentioned, another form of torque vectoring is to apply brakes individually to reduce the tractive force available at the inside wheels. This system is not implemented on the Ferrari FF; however, these calculations prove it would be an effective system to use. The maximum braking force available would be the maximum tractive effort, $F = \mu(M \cdot a)$. Considering the outside wheels stay at 50% available torque:

Table 11 - Braking Torque Vectoring Data

OUTSIDE DRIVE FORCE (N)	MOMENT ARM (1/2 TRACK) (m)
18223.731	0.838

Table 12 - Braking Force Data

MAXIMUM BRAKING FORCE		
μ	FRONT PASSENGERS	FRONT AND REAR PASSENGERS
0.700	13966.100	14708.430
0.500	9973.790	10506.020
0.100	1995.160	2101.200

Table 13 - Braking Torque Vectoring Calculations

μ								FRONT PASSENGERS		FRONT & REAR PASSENGERS	
OUTSIDE WHEEL TRACTIVE FORCE	INSIDE WHEEL BRAKING %	F PASSENGERS INSIDE WHEEL BRAKING FORCE (-N)	F & R PASSENGERS INSIDE WHEEL BRAKING FORCE (-N)	F PASSENGERS Δ TRACTIVE FORCE (N)	F & R PASSENGERS Δ TRACTIVE FORCE (N)	F PASSENGERS MOMENT PRODUCED (Nm)	F & R PASSENGERS MOMENT PRODUCED (Nm)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)
9112	10	1396.610	1470.843	10508.476	10582.709	8806.102	8868.310	2.158	123.630	2.107	120.703
9112	20	2793.220	2941.686	11905.086	12053.552	9976.462	10100.876	2.445	140.061	2.399	137.479
9112	30	4189.830	4412.529	13301.696	13524.395	11146.821	11333.443	2.731	156.492	2.692	154.255
9112	40	5586.440	5883.372	14698.306	14995.238	12317.180	12566.009	3.018	172.922	2.985	171.031
9112	50	6983.050	7354.215	16094.916	16466.081	13487.539	13798.575	3.305	189.353	3.278	187.807

μ								FRONT PASSENGERS		FRONT & REAR PASSENGERS	
OUTSIDE WHEEL TRACTIVE FORCE	INSIDE WHEEL BRAKING %	F PASSENGERS INSIDE WHEEL BRAKING FORCE (-N)	F & R PASSENGERS INSIDE WHEEL BRAKING FORCE (-N)	F PASSENGERS Δ TRACTIVE FORCE (N)	F & R PASSENGERS Δ TRACTIVE FORCE (N)	F PASSENGERS MOMENT PRODUCED (Nm)	F & R PASSENGERS MOMENT PRODUCED (Nm)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)
9112	10	997.379	1050.602	10109.245	10162.468	8471.547	8516.148	2.076	118.933	2.023	115.910
9112	20	1994.758	2101.204	11106.624	11213.070	9307.350	9396.552	2.281	130.667	2.232	127.893
9112	30	2992.137	3151.806	12104.003	12263.672	10143.154	10276.957	2.485	142.401	2.441	139.876
9112	40	3989.516	4202.408	13101.382	13314.274	10978.958	11157.361	2.690	154.135	2.650	151.859
9112	50	4986.895	5253.010	14098.761	14364.876	11814.761	12037.766	2.895	165.869	2.860	163.841

μ								FRONT PASSENGERS		FRONT & REAR PASSENGERS	
OUTSIDE WHEEL TRACTIVE FORCE	INSIDE WHEEL BRAKING %	F PASSENGERS INSIDE WHEEL BRAKING FORCE (-N)	F & R PASSENGERS INSIDE WHEEL BRAKING FORCE (-N)	F PASSENGERS Δ TRACTIVE FORCE (N)	F & R PASSENGERS Δ TRACTIVE FORCE (N)	F PASSENGERS MOMENT PRODUCED (Nm)	F & R PASSENGERS MOMENT PRODUCED (Nm)	YAW RATE (rads/sec)	YAW RATE (degs/sec)	YAW RATE (rads/sec)	YAW RATE (degs/sec)
9112	10	199.516	210.120	9311.382	9321.986	7802.938	7811.824	1.912	109.546	1.856	106.324
9112	20	399.032	420.240	9510.898	9532.106	7970.132	7987.904	1.953	111.894	1.898	108.720
9112	30	598.548	630.360	9710.414	9742.226	8137.327	8163.985	1.994	114.241	1.939	111.117
9112	40	798.064	840.480	9909.930	9952.346	8304.521	8340.066	2.035	116.588	1.981	113.513
9112	50	997.580	1050.600	10109.446	10162.466	8471.715	8516.146	2.076	118.935	2.023	115.910

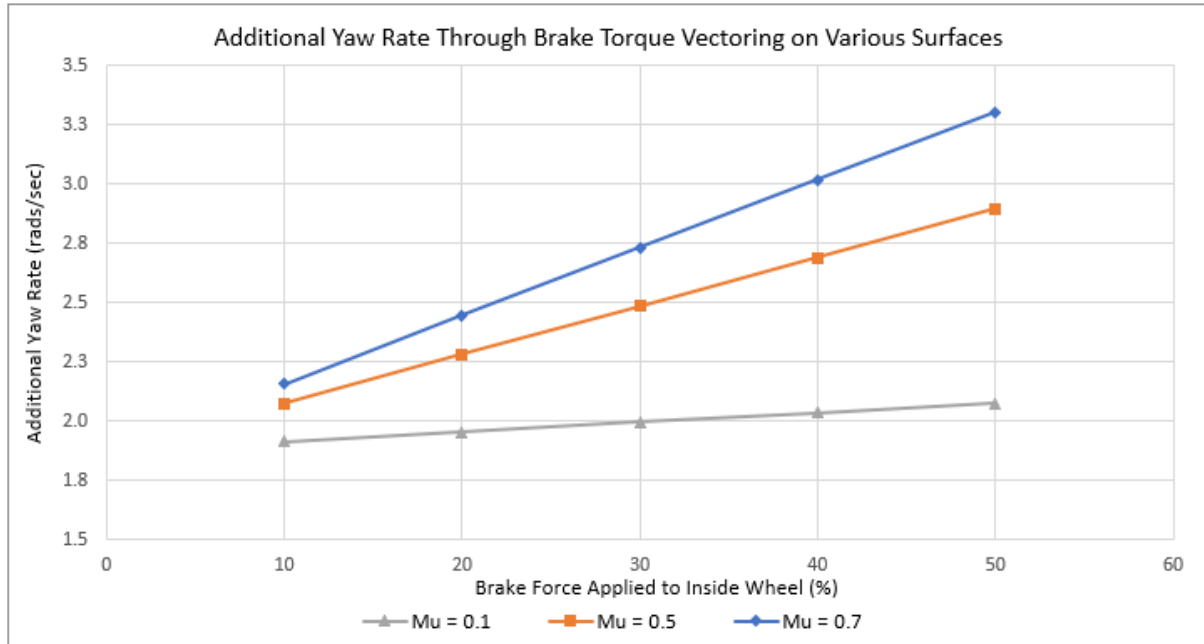


Figure 6 - Braking Torque Vectoring on Various Surfaces

Figure six shows a substantial yaw rate increase on every surface by decreasing torque to the inside wheels in a corner, proving that braking torque vectoring is an efficient way of increasing cornering ability. It is generally accepted that at most 50% brake should be applied to ensure the vehicle stays controllable (Day, 2014).

To validate these results, the author considered the vehicle to be travelling around a 25m radius corner with lateral acceleration of 1.1g:

$$1.1g = 10.79ms^{-2}$$

$$a = \frac{v^2}{r} \therefore v = \sqrt{10.79 \cdot 25} = 16.42ms^{-1}$$

$$\text{Corner circumference} = 2\pi \cdot 25 = 157.1m$$

$$\text{Time} = \frac{\text{Distance}}{\text{Speed}} = \frac{157.1}{16.42} = 9.56 \text{ seconds per lap}$$

$$\therefore \text{yaw rate} = \frac{360}{9.56} = 37.64 \text{ degrees per second}$$

In this situation, the author considered the following:

$$\text{Road wheel angle} = \frac{\text{Wheelbase}}{\text{Corner radius}} = \frac{2.99}{25} = 0.1196 \text{ radians} = 6.85^\circ$$

The Ferrari FF has a steering ratio of 11.5 : 1 (DriverSide, 2015),

$$\therefore \text{hand wheel angle} = 6.85 \cdot 11.5 = 78.78^\circ$$

Therefore, in this situation the vehicle has a yaw rate of 37.64°/s. Taking realistic values of a 60 : 40 torque split, and just 10% braking force, with μ of 0.7 and two front passengers, the yaw rate can be increased by 30.016°/s and 123.63°/s, respectively. This means acceleration torque vectoring can add almost double yaw rate, braking torque vectoring can increase it by over four times, both keeping the hand wheel angle at 78.78°. This shows how torque vectoring can be a fantastic driver aid, with the driver not required to give any input.

Further investigation into this area could consist of analysing weight transfer as the vehicle corners. Load is transferred to the outside wheels, increasing the maximum tractive force available, thus increasing the efficiency of torque vectoring. Considering a steady state section of the aforementioned corner with two front passenger and assuming a CoG height of 0.32m:

$$WT = \frac{m \cdot a \cdot h}{t} = \frac{2033.8 \cdot 10.79 \cdot 0.32}{1.67} = 4204.97N$$

This will add an extra 4kN of tractive force on the outside wheels, significantly increasing the effects of acceleration torque vectoring.

3.5 Torque Vectoring Control Algorithms and Sensors

The torque vectoring system at the front axle can be effective even when the vehicle is not on power. The clutches can be engaged evenly to slow the vehicle, or the clutch of the inside wheel could be engaged to slow the wheel, increasing yaw rate as in section 3.4.2 (Ivanov *et al.*, 2012). This process could be used as a safety feature also; the clutches could be engaged when the car is thought to be out of control, to try and reduce yaw rate; this would be working as a yaw damper. To achieve this, multiple inputs from sensors need to be recorded and analysed to understand exactly how the vehicle is behaving:

Figure 7 - Sensors Required to Control a Torque Vectoring System (RoadSafetyFacts, 2020)

<u>REQUIRED SENSOR</u>	<u>FUNCTION</u>
Wheel speed	Generic wheel speed sensor, magnet attached to each driveshaft with sensor to count pulse frequencies, converting to angular velocity. Useful to have on every corner to control wheel speed differentials across the torque vectoring system.
Vehicle speed	Due to potential for unmatched wheel speeds, a GPS speed sensor would be optimal for a consistent, accurate reading.
Yaw rate	Micromechanical yaw rate sensor to feed back to vehicle the current yaw rate and how severe the turn is. There should be a pre-defined yaw rate at which the car is deemed out of control.
X - Y - Z acceleration	Accelerometers along each plan to relate to each other and analyse traction available.
Hand wheel angle	Sensor measuring frequency of marker to create digital signal. Using steering ratio this can calculate requested road wheel angle.
Brake force request	Sensor to manage braking force requested by driver.
Drive force request	Sensor to manage throttle percentage requested by driver.

Along any straight road or corner, these parameters need to be constantly monitored to make best use of the AWD system for two main reasons; to ensure optimal vehicle balance and to ensure safety of the driver and occupants. Accelerometers and yaw rate sensors can measure vehicle balance and combine with the sensors measuring the input the driver has requested, to feed back to the torque vectoring system to help keep the vehicle within the limit of grip. Wheel speed sensors can monitor torque distribution and change it accordingly (Jalali *et al.*, 2013). As mentioned, the yaw rate sensor should have a pre-defined limit at which the car is deemed out of control to initiate ABS and ESP systems to reduce acceleration and yaw rate to regain control and ensure driver safety (NHTSA, 2020). These processes must be treated with appropriate prioritisation. The car cannot be allowed to go too far outside the limit before the stability aids kick in. Driving modes can be selected to dictate the pre-defined yaw rate limit, from driver aids helping at the first sign of danger, to the driver being completely alone with the car.

4 Evaluation and Conclusion

This report has found that the geometric design of the Ferrari FF is fantastic at helping the large, spacious, luxury vehicle to handle like a much smaller performance car. The near 50/50 weight distribution and clever positioning of passengers ensures that the vehicle is practical but fast in any situation.

Any shortfalls in handling due to the size of the vehicle is countered to some extent by the advanced torque vectoring system within the vehicle. This report has found that the ability to send torque to specific areas enables huge handling improvements. The way this is achieved through the novel 4RM AWD system significantly reduces mechanical losses, whilst keeping mass to a minimum, and reducing the height of the centre of gravity.

The Ferrari FF, in the author's opinion, was an engineering masterpiece that the world was not ready for. Fast, yet practical and luxurious cars had never been created in such a refined way before. The demographic most likely to buy this type of vehicle do not care for the minor vehicle dynamic improvements that come from such clever engineering, they look only for comfort and luxury. It is because of this, sports utility vehicles (SUVs) are currently so popular. Nowadays, if a consumer wants a fast, well balanced performance car, they buy a thoroughbred supercar. If they want practicality and luxury, they buy an SUV. The Ferrari FF was the engineering marvel that fills the gap in the middle, that nobody wanted.

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