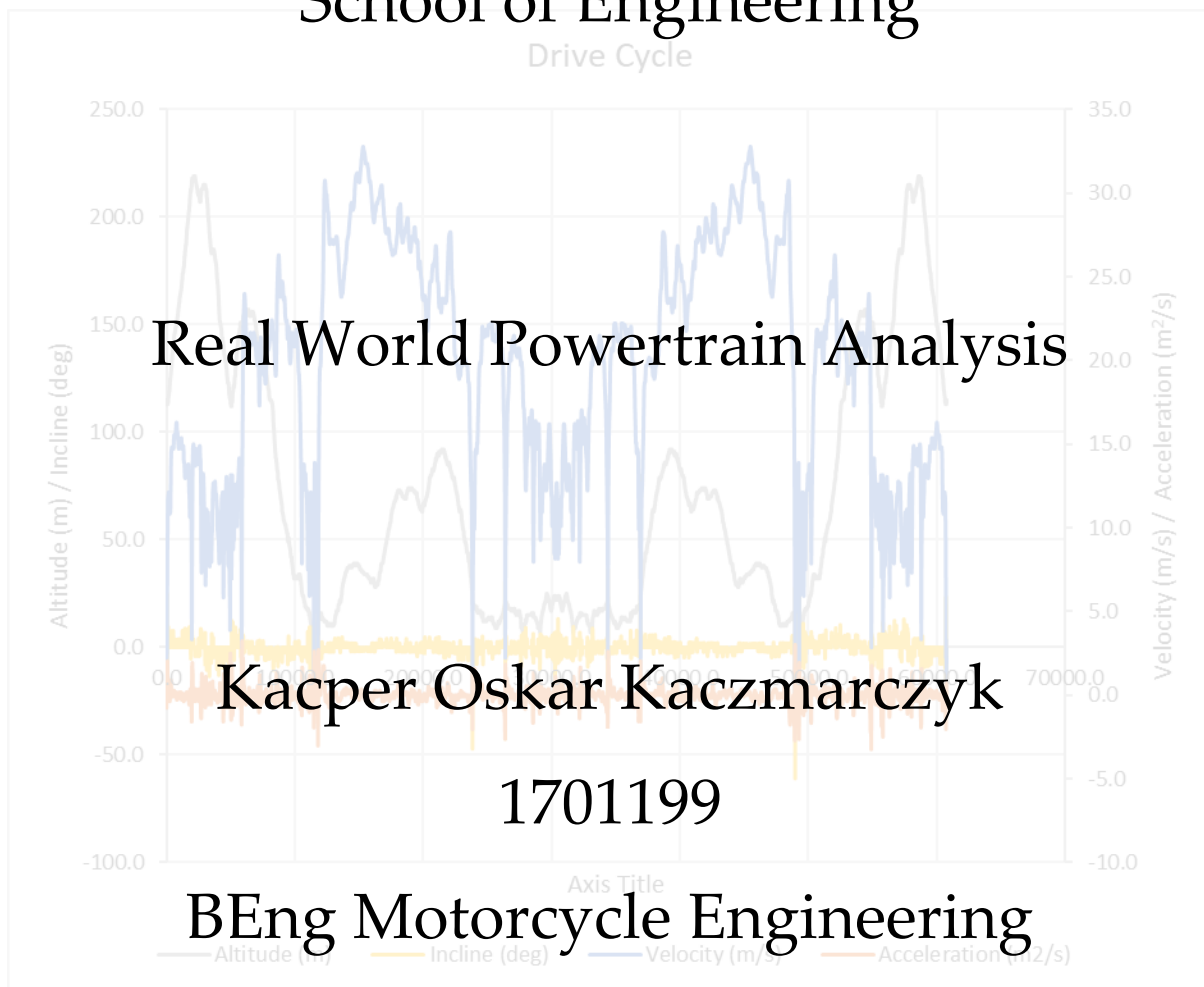


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1 Introduction

This investigation looks into the quality of the recorded data from a real-life drive cycle. The data is critically analysed and expanded to give author enough information to analyse the performance, drivability and efficiency of all three vehicles.

Author created a drive cycle model based on one of the vehicles, which it is then used to assess the impacts of alternative powertrain systems as well as an energy recovery system.

Based on the results of the alternative powertrain analysis, author created a powertrain specification for the year 2028, which was then analysed and compared against, in the drive cycle model.

2 Analysis of Data Quality

After analysing the supplied data, author has decided to disregard all but the following channels:

- Frame Time
- Vehicle Speed Sensor
- GPS Altitude

This was because author didn't have access to a technical manual which detailed where exactly the data for each channel comes from, how it is pre-processed and most importantly, the real accuracy. Therefore, author has decided to derive all other required information, based on the previously mentioned channels, since this improved the reliability and author's confidence in the data.

Author believed this method to be applicable mainly due to the surprisingly high relative resolution of 0.083s, as shown in Figure 2-1.

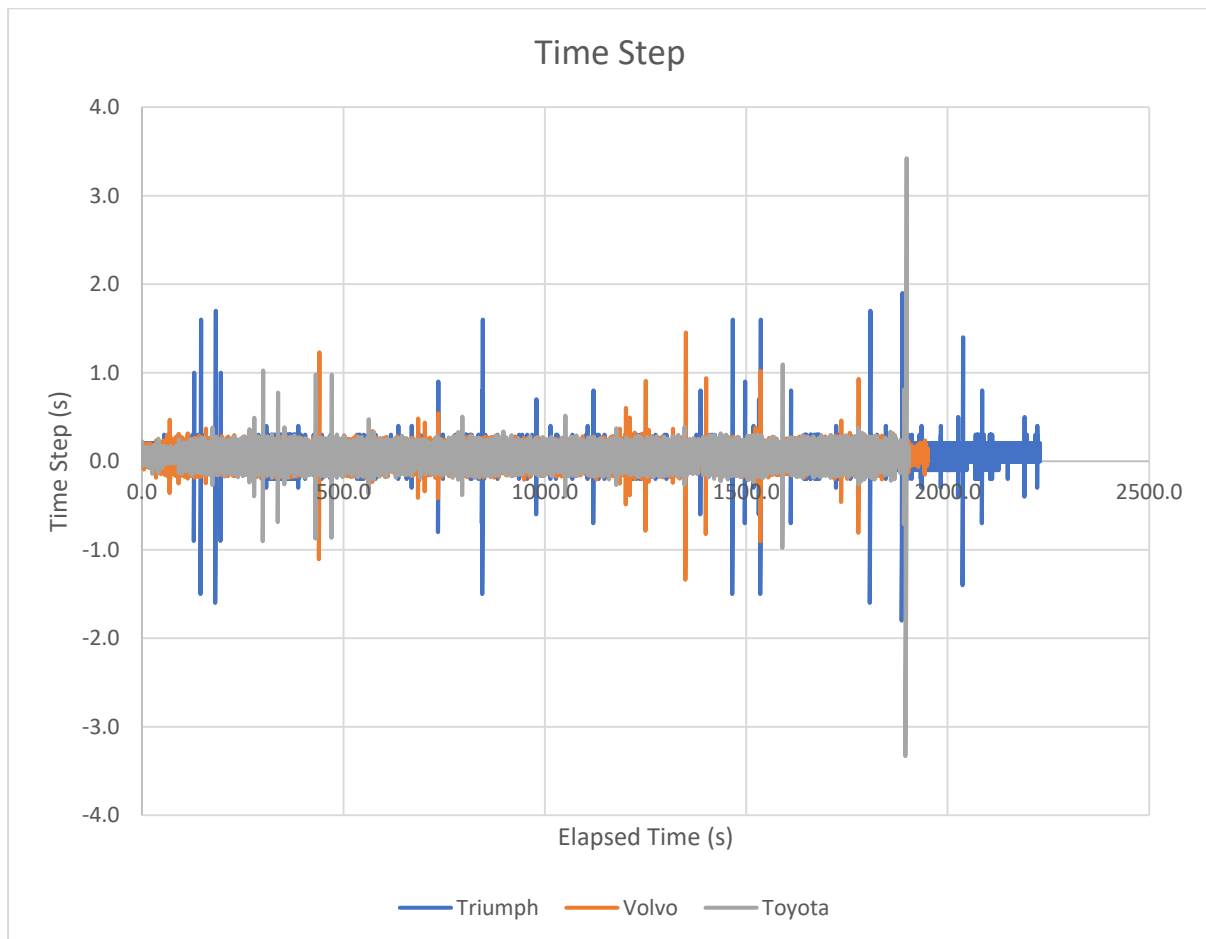


Figure 2-1: Time Step

Author has also decided to filter the data so that the time step became 1.0s, since this improved the validity of the calculated instantaneous values. This was due to the low precision of the speed channel, just ± 1 mph, which when un-filtered, caused extremely high and unrealistic instantaneous acceleration.

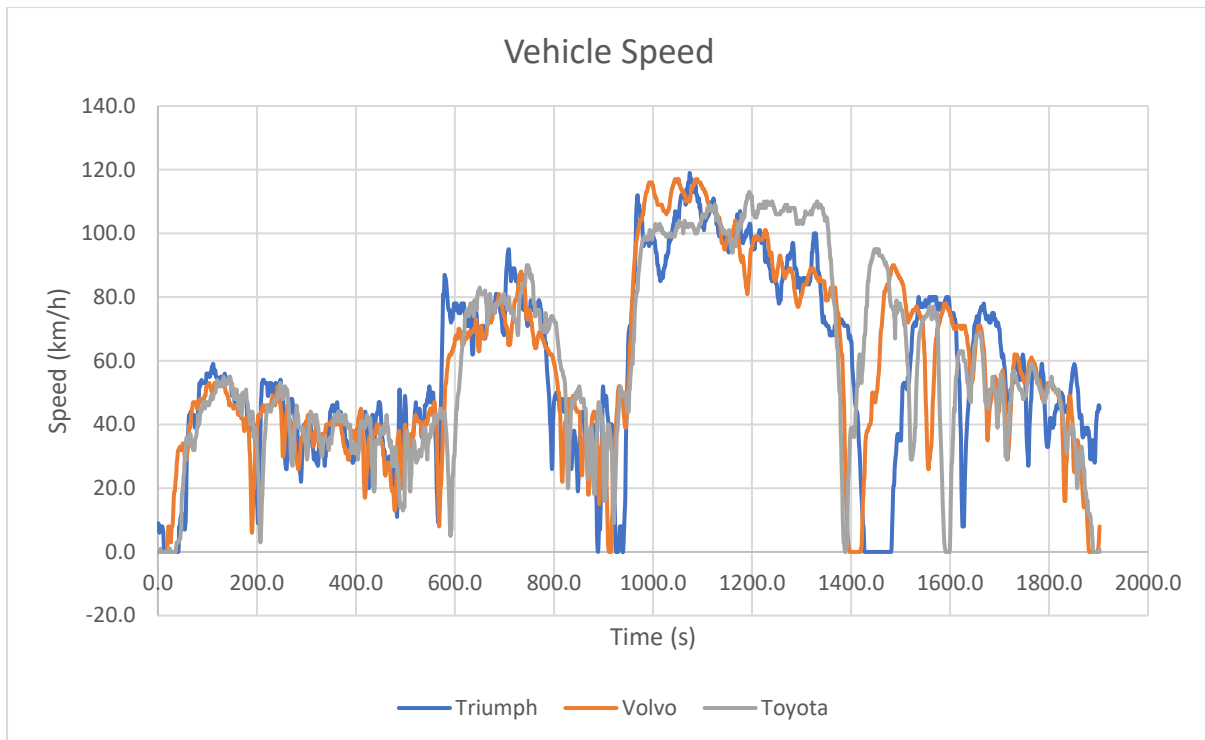


Figure 2-2: Vehicle Speed

As Figure 2-2 shows, data channels for all three vehicles have been phased in respect of each other, so that the speed profiles of all the vehicles match. However, it was impossible to achieve perfect phase, due to the test variability which was introduced due to the real-life aspect of the experiment. It also shows the recorded data to be somewhat viable, due to the sensible magnitudes of 50 km/h, 80 km/h and 110km/h, which correspond to UK's speed limits on urban roads, rural roads and dual carriageways/ motorways respectively.

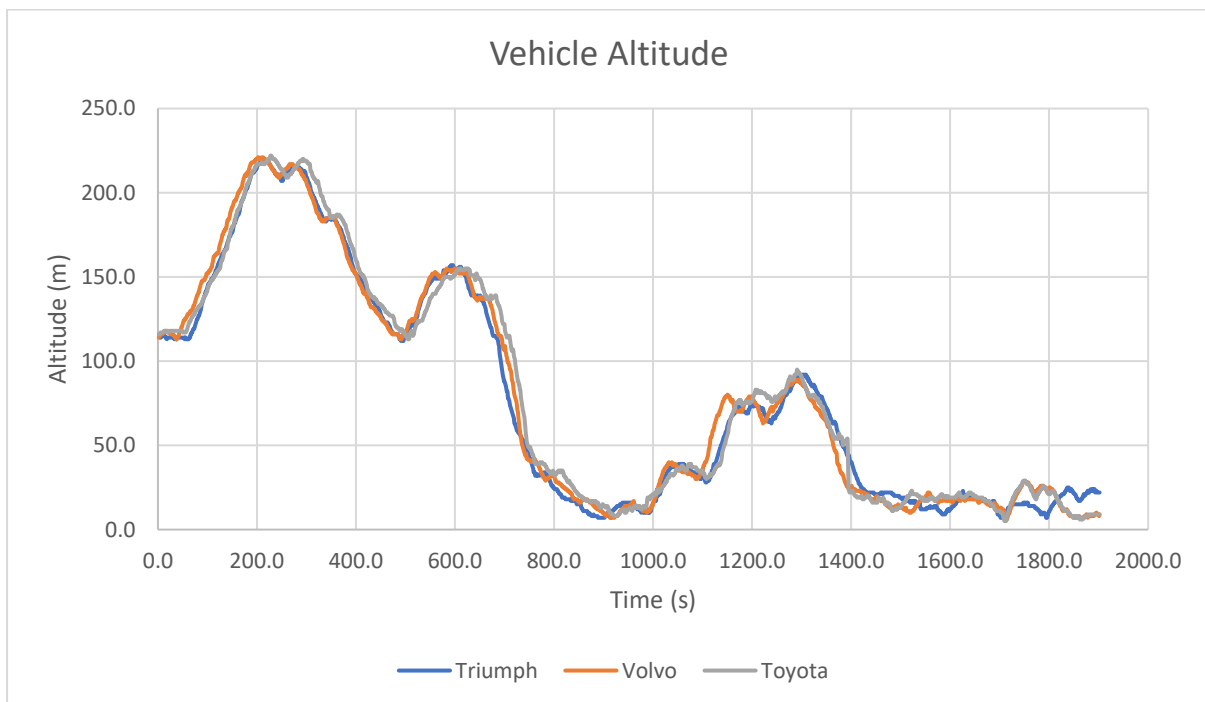


Figure 2-3: Vehicle Altitude

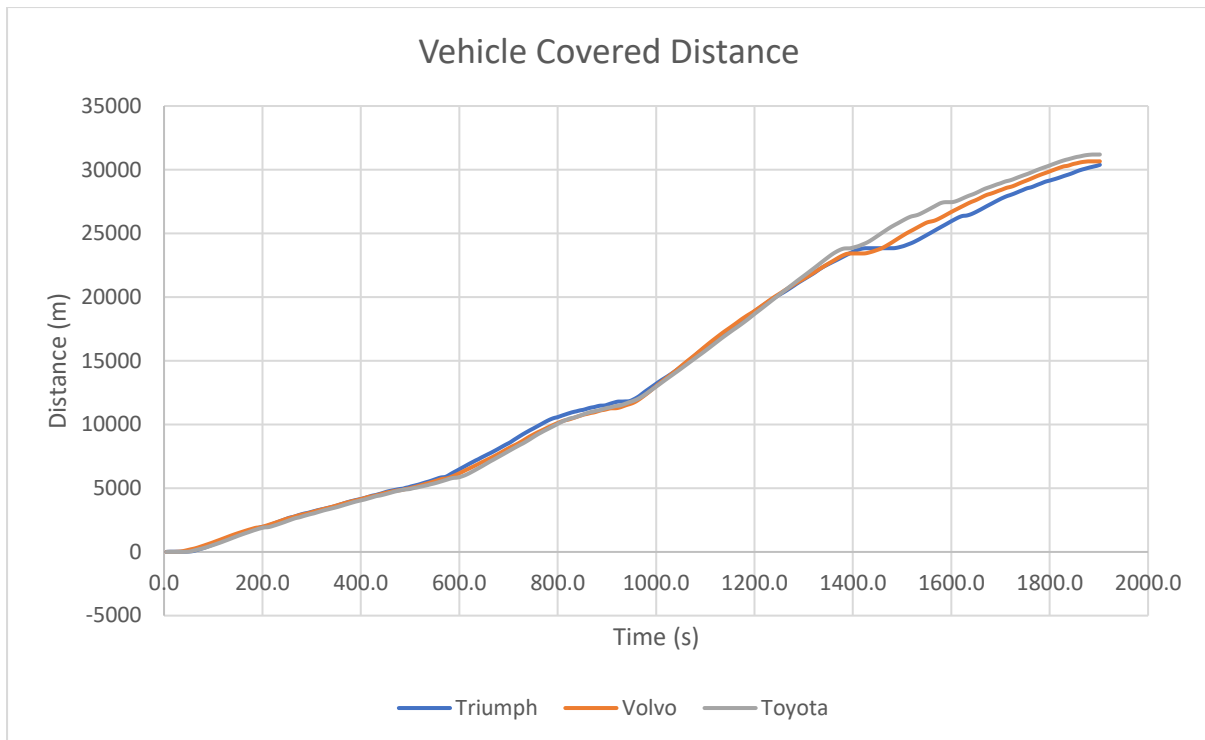


Figure 2-4: Vehicle Covered Distance

Figure 2-3 and Figure 2-4 were also used to assess the quality of the phasing. Interestingly, Figure 2-3 showed Volvo to be significantly out of phase between 1,100s and 1,200s. However, this wasn't displayed by Figure 2-3, as the only 'noticeable' loss of phasing occurred between 1,400s and the end. Due to this, author has also assessed phasing quality in every data set which was being assessed.

The additional data required in Vehicle Analysis section, was vehicle acceleration and covered distance, which were calculated using differentiation and integration respectfully.

Figure 2-4 shows that the averaged covered distance was 30.75km, which author deemed reasonable and therefore accurate, considering the 'advertised' distance of 30km, which was accurate within one significant figure.

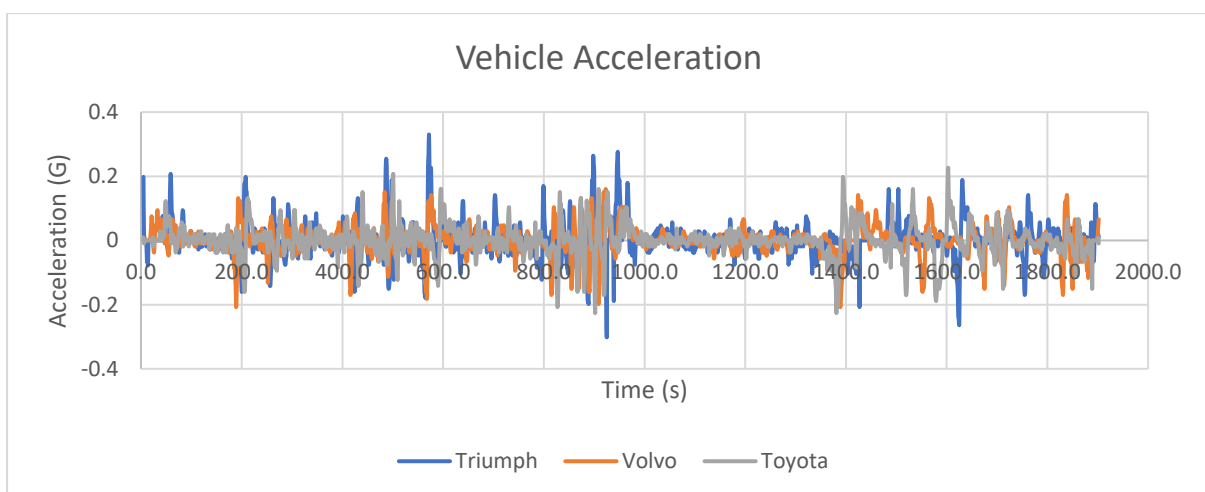


Figure 2-5: Vehicle Acceleration

Figure 2-5 shows, the acceleration and deceleration both rarely exceeded 0.2G, most of which was achieved on the motorcycle, which author believed to be reasonable yet again due to much better power to mass ratio of 0.32 kW/kg as oppose to 0.05kW/kg as well as the tendency of more 'spiritual' riding, which often occurs on bi-wheeled vehicles.

Lastly, since the results for all three vehicles are relatively similar and of reasonable magnitude, author believes the recorded and calculated channels to be 'good enough' for their intent use in sections 3 and 4. However, the equipment basis a lot of its crucial parameters on GPS, of which accuracy and latency compromises the reliability. Other components such as a gyroscope mounted in the phone, which has an obvious packaging consideration that affects the quality of the data, as well as the lack of information about whether or not the OBD II signals have even been read correctly, possess a sever limitation on the usability of the equipment and the recorded data.

3 Vehicle Analysis

3.1 Method

To compare the investigated vehicles, author has created three ‘performance’ indexes. Firstly, author had to calculate the total burned fuel mass, which was done by using Equation 3-1 and assuming an average MPG (miles per gallon) value of 50mpg, for every vehicle, as told by the data provider. Due to this assumption, author decided the calculations of instantaneous indexes to be of little interest, as the fuel consumption throughout the drive cycle stayed constant. The petrol and diesel densities were assumed to be 750 kgm⁻³ and 850 kgm⁻³ respectfully (Engineering ToolBox, 2020).

$$m_f = \frac{\rho_f * l}{MPG * 0.354006 * 10^6}$$

Equation 3-1: Burned Fuel

To define the vehicle performance, author has decided to use the SPD (Specific Performance Delivery), which was calculated using Equation 3-2.

$$SPD = \frac{|a_{+ve}|}{m_f}$$

Equation 3-2: Specific Performance Delivery

Author choose SPD as the ‘performance index’ since it clearly shows how efficient the overall vehicle is at delivering performance (average positive acceleration). SPD is not a dimensionless value, theoretically making it a poor comparative index. However, author believes it to have a ‘dimensionless like’ characteristic, as normally an increase of acceleration would require more fuel. Additionally, author believes that any nonlinearity associated with the index not being dimensionless is effectively taken care off, since data for all vehicles has been conducted on the same drive cycle and the results have been averaged.

To define vehicle’s drivability, author has used the ratio of average indicated engine power and the peak available power, as shown in Equation 3-3. Author had to use peak break power as peak indicated power wasn’t available, author believes this inconsistency to be of little importance since the index is only used for comparative purposes. Author choose this index since power is directly related to load which clearly shows how much of powertrain ‘potential’ had to be used.

$$\eta_p = \frac{|P|}{P_{max}} = \frac{\frac{m_f * LHV}{t}}{P_{max}}$$

Equation 3-3: Driveability

Lastly, to define vehicle efficiency, author has used the break thermal efficiency as shown in Equation 3-4. Author has decided to use this since it not only considers the thermal efficiency of the powertrain but also the losses associated with the vehicle itself.

$$\eta_t = \frac{(m|a_{+ve}| + 0.5\rho_a C_D A |v|^2 + 0.02mg) * l}{m_f * LHV}$$

Equation 3-4: Break Thermal Efficiency

3.2 Analysis

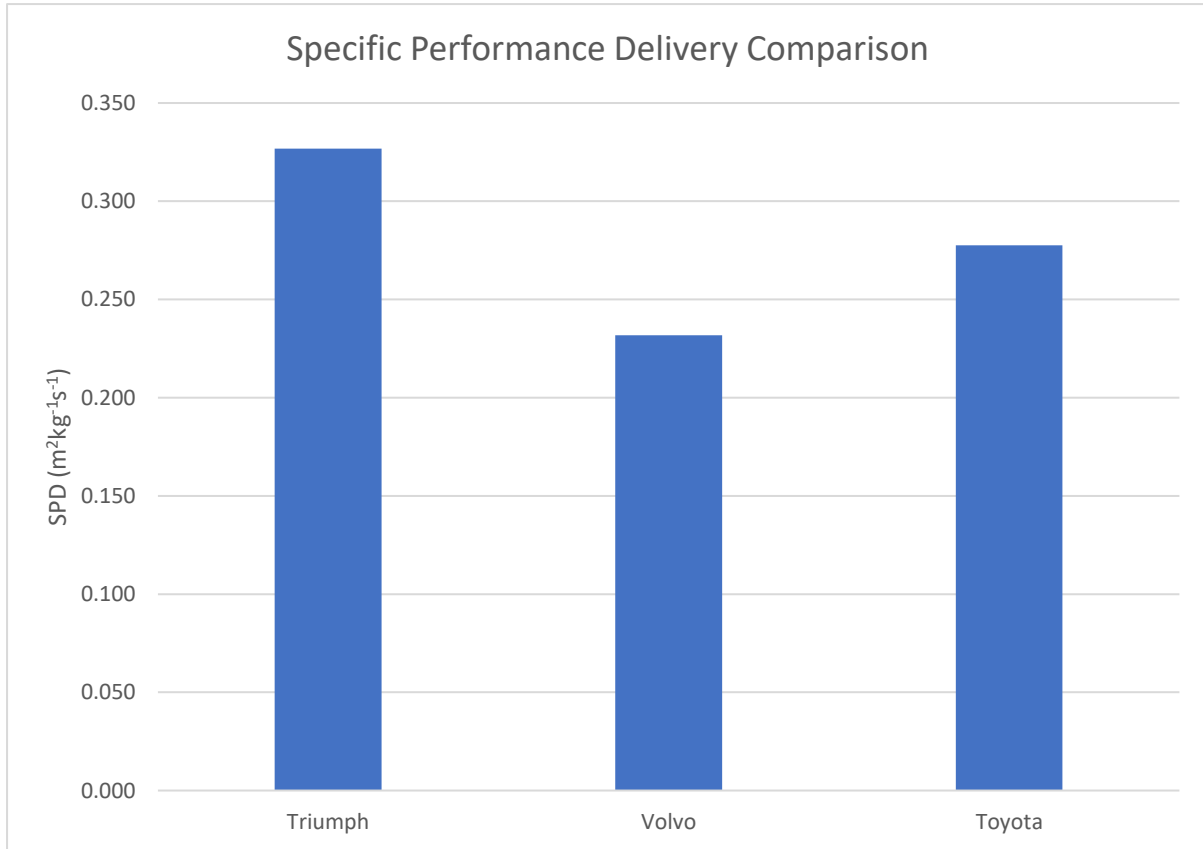


Figure 3-1: Specific Performance Delivery Comparison

Figure 3-1 shows the calculated SPD values for all three vehicles. As implied in the Analysis of Data Quality, the Triumph Tiger 800XC motorcycle had the highest SPD value. This was primarily due to the 17.7% higher average positive acceleration compared to the 2nd highest SPD vehicle that was the Toyota. The reason why Volvo's SPD value was the lowest is due to the initial assumption of 50mpg, as this causes Volvo to consume 13.3% more fuel than Triumph and Toyota, as the diesel density was equally denser than petrol.

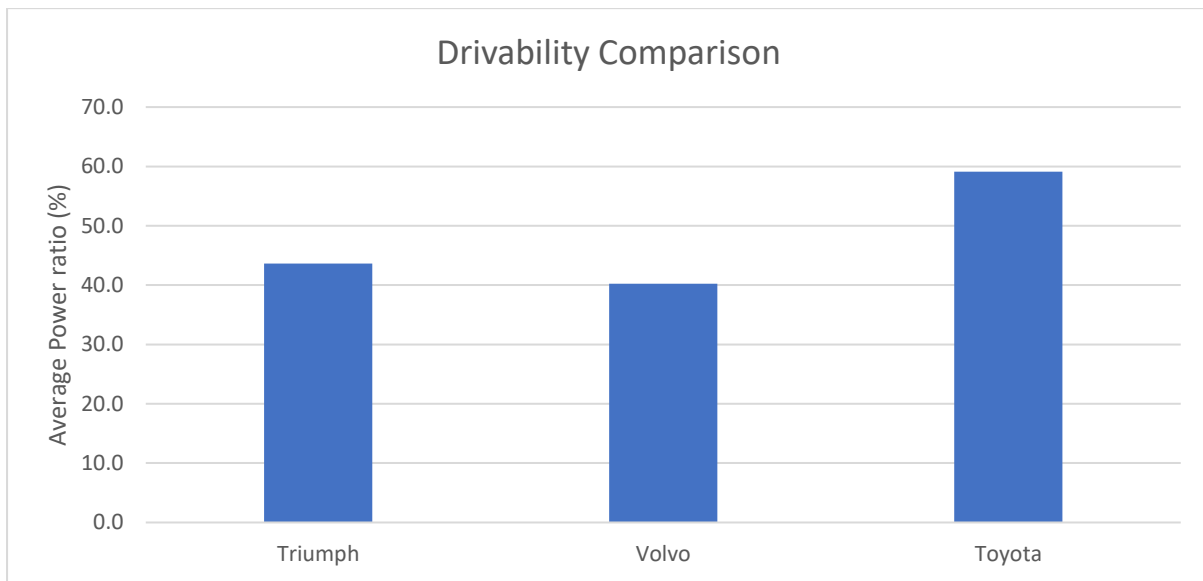


Figure 3-2: Drivability Comparison

Figure 3-2 shows the calculated average power ratios. As expected, the Toyota was the least driveable due to its relatively poor vehicle power density of just 0.041 kWkg^{-1} , compared to 0.055 kWkg^{-1} and 0.322 kWkg^{-1} for Volvo and Triumph respectively. However, author believes this result to be overestimated as Toyota has been drive much more aggressively as previously shown by the SPD. Triumph scored 3.4% more than Volvo. Author believes this primarily to be caused by the fact that motorcycles tend to utilise much higher rpms, especially on motorways, which were a big part of the drive cycle. In comparison, the turbocharged diesel powertrains, like the one found in Volvo, deliver a lot of power at very low rpm (due to superior torque characteristics) and they have much longer gearing, making the Volvo naturally most drivable.

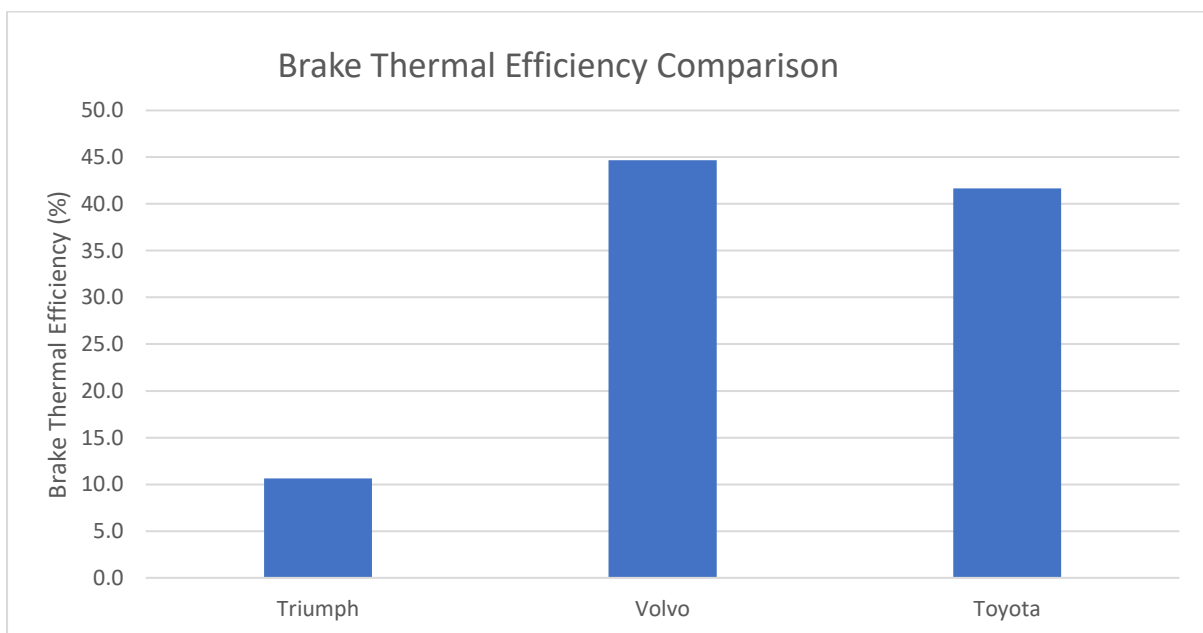


Figure 3-3: Brake Thermal Efficiency Comparison

Figure 3-3 shows the calculated mean brake thermal efficiency. In real life modern turbocharged diesel engines have the highest thermal efficiency out of any production ICE (Internal Combustion Engine). This is primarily due to its diesel cycle which is limited by work input as oppose to fuel, compression ignition and much higher compression ratios. However, in author's model, Volvo's diesel ICE showed to have the highest thermal efficiency primarily due to much heavier vehicle mass, as this increased the work done the vehicle had to do to achieve the assumed 50mpg. Toyota has scored only 3.0% lower, which was primarily due to the same reasons as for Volvo. Triumph on the other hand has achieved astonishing 34.1% lower thermal efficiency than Volvo. Author believes this value to be too low for it to be real. However, it does show how incredibly inefficient the motorcycle engine is in the investigated drive cycle, compared to modern car petrol and diesel counterparts.

3.3 Conclusion

Author believes the trends shown in Vehicle Analysis to be correct, since they match common sense and author's knowledge/ experience as well as the limited data. However, the initial assumption of constant 50mpg for all vehicles has shown to be an enormous limitation. This is because: it prevents a reasonable calculation of instantaneous indexes and it dictates the magnitudes of all results. Due to this, author believes that any conclusion made beyond the shown trends lack too much evidence to be factual.

4 Drive Cycle Simulation

Author has decided to further investigate the Triumph motorcycle as it has achieved 'the same' MPG as other vehicles, while being extremely inefficient, suggesting a big potential for improvements.

4.1 Method

To create the drive cycle, author has taken the time, distance, speed, acceleration and altitude data of the Triumph motorcycle, from Vehicle Analysis. To simulate the vehicle travelling back to the point of origin, author has mirrored the same data which was then added at the point of original destination. Making the total journey distance 60.8km.

The road incline has been calculated using Equation 4-1.

$$\vartheta = \tan^{-1} \left(\frac{x_2 - x_1}{l_2 - l_1} \right)$$

Equation 4-1: Road Incline

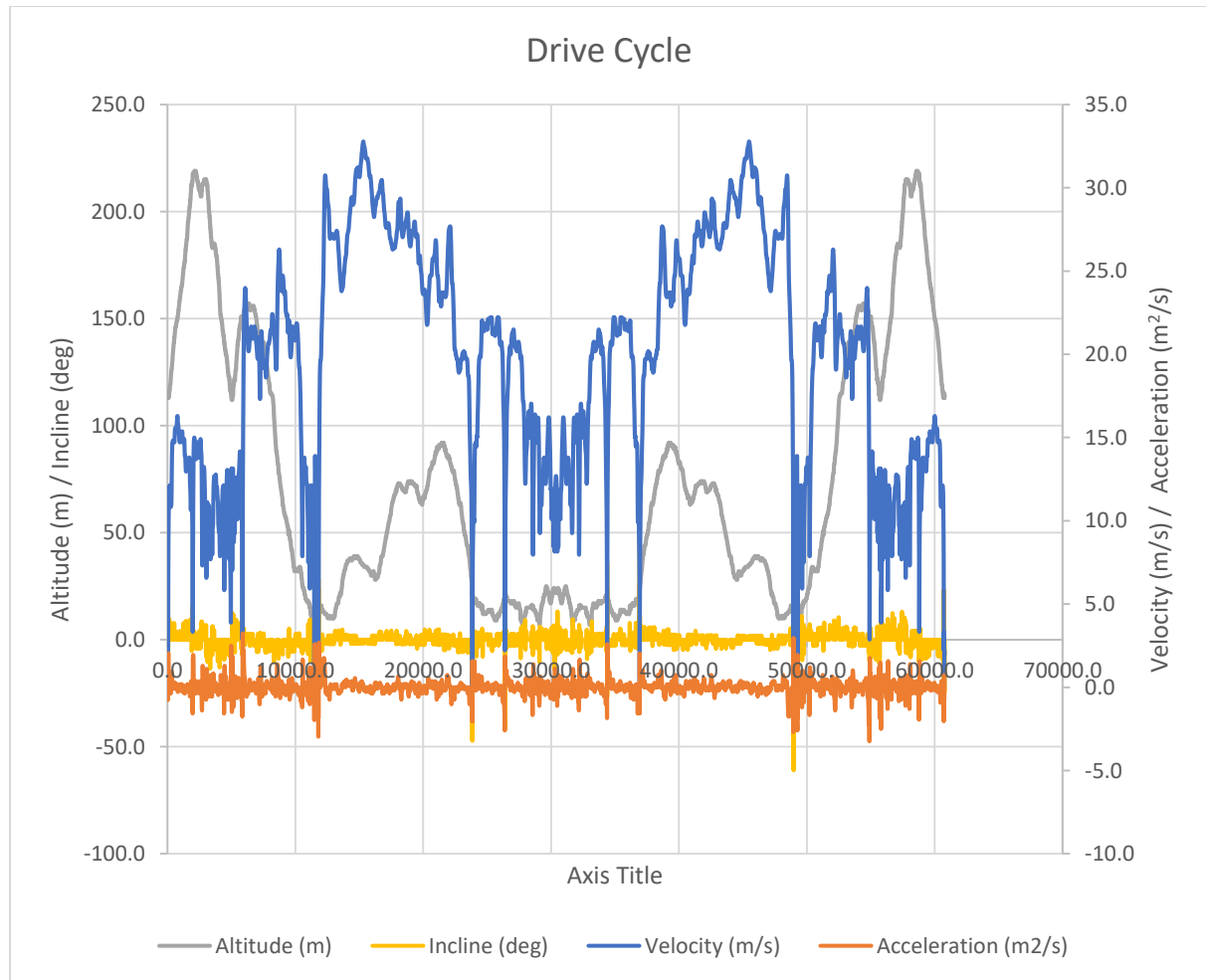


Figure 4-1: Drive Cycle

Figure 4-1 shows the drive cycle. Naturally, the speeds and therefore accelerations and distances, wouldn't be the same when travelling both directions. However, author believes this to have a relatively small effect and more realistic determination of speeds on the back journey is outside the scope of this investigation.

$$P_{inst.} = \frac{WD}{t} = \frac{\sum F_{inst.} * (l_2 - l_1)}{t}$$

Equation 4-2: Instantaneous Power

Due to the relation shown in Equation 4-2, author had to calculate the sum of all relevant forces applied to the vehicle throughout the drive cycle, as shown in Equation 4-3.

$$\sum F_{inst.} = (F_D + F_A + F_R + F_G)_{inst.}$$

Equation 4-3: Sum of Instantaneous Forces

Where all constituents are shown in Equation 4-4, Equation 4-5, Equation 4-6 and Equation 4-7:

$$F_{D_{inst.}} = (m_{vehicle} + m_{reduced} + m_{rider}) * a_{inst.}$$

Equation 4-4: Instantaneous Driving Force

$$F_{A_{inst.}} = 0.5 * \rho_a * C_D * A * v_{inst.}^2$$

Equation 4-5: Instantaneous Air Resistance

$$F_{R_{inst.}} = (m_{vehicle} + m_{reduced} + m_{rider}) * g * \cos \vartheta_{inst.}$$

Equation 4-6: Instantaneous Rolling Resistance

$$F_{G_{inst.}} = (m_{vehicle} + m_{reduced} + m_{rider}) * g * \cos(90 - \vartheta_{inst.})$$

Equation 4-7: Instantaneous Gravitational Force

To separate the driving from braking forces, Equation 4-8 and Equation 4-9 have been used respectfully.

$$\sum F_{inst.} \xrightarrow{IF+ve} F_{Driving_{inst.}}$$

Equation 4-8: Gross Instantaneous Driving Force

$$\sum F_{inst.} \xrightarrow{IF-ve} F_{Braking_{inst.}}$$

Equation 4-9: Gross Instantaneous Braking Force

To simulate braking energy recovery systems such as KERS, author has used Equation 4-10 if the result wasn't above the specified maximum storable energy.

$$E_{KERS_{inst.2}} = \left(WD_{Braking_{inst.2}} * \eta_{KERS} + E_{KERS_{inst.1}} \right) - \left(WD_{Gross\ Driving_{inst.1}} - WD_{Net\ Driving_{inst.2}} \right)$$

Equation 4-10: Recovered Braking Energy

4.2 Validation

Firstly, author has 'validated' the model by simulating the original Triumph motorcycle with the assumed values shown in Table 4-1 and Table 4-2.

Table 4-1: Original Vehicle Specification

Vehicle	
Rolling Chassis Mass (kg)	137.52
Vehicle Mass (kg)	214.50
Mean Reduced Mass (kg)	300.00
Rider Mass (kg)	90.00
Energy Recovery Systems Mass (kg)	0.00
C _D	0.88
Frontal Area (m ²)	0.46
Mean Braking energy Recovery Efficiency (%)	0.00
Total Braking Energy Recovery Storable Energy (MJ)	0.00

Table 4-2: Petrol Engine Specification

Petrol Engine	
LHV (MJ/kg)	44.00 (Engineering ToolBox, 2020)
Peak Power (kW)	69.10
Fuel Capacity (l)	18.90
Fuel Density (kg/m ³)	750.00 (Engineering ToolBox, 2020)
Energy Capacity (MJ)	623.70
Engine Mass (kg)	57.80 (redw123, 2017)
Fuel Mass (kg)	14.18

Fuel Tank Mass (kg)	5.00
Mean Reduced Mass (kg)	300.00
Mean Engine Thermal Efficiency (%)	13.15
Specific CO ₂	3.06

Author assumed the rider mass, C_D , tank and mean reduced masses based on experience and the frontal area is that of previously measured 2009 Yamaha YZF-R6, due to lack of data.

Table 4-3: Original Simulation Outputs

Initial Outputs	
Total Driving Energy (kJ)	14898.5
Total Braking Energy (kJ)	-1540.0
Powertrain Energy Output (kJ)	113296.4
Total Burned Fuel (kg)	2.6
Petrol Equivalent MPG	50.0
Average Driving Power (kW)	3.9
Average Indicated Engine Power (kW)	29.8
Produced CO ₂ (kg)	7.9
Used Energy %	18.17

Table 4-3 shows the simulation output of the original vehicle. The 50mpg and 2.6kg burn fuel (double the amount due to doubled distance) has been achieved by adjusting the mean thermal efficiency value, which increased by 3.1% compared to the result shown in Vehicle Analysis. Author believes this discrepancy to be caused by the additional considerations which were incorporated into the drive cycle model. Considering these factors, author believes that drive cycle model cannot be disproven with the supplied data.

4.3 Analysis

4.3.1 Kinetic Energy Recovery System

An example of KERS working in the drive cycle model is shown in Appendix 1.

Table 4-4: KERS specification

KERS	
Mean Efficiency (%)	60.0 (theWPTformula, 2014)
Storable Energy (MJ)	0.04
Energy Density (MJ/kg)	0.016 (theWPTformula, 2014)
KERS Mass (kg)	2.5

Table 4-4 shows the chosen specifications of the KERS (Kinetic Energy Recovery System). Author has applied the F1 KERS technology from 2014, which isn't most efficient. However, it is much more realistic to implement older F1 technology.

The storable energy of 0.04MJ has been chosen since it yielded the highest MPG value, making it the optimal compromise between the availability of KERS power and the additional mass of the system.

The addition of KERS resulted in additional 3.04MPG, which equalled to fuel saving of 5.7%, while the vehicle mass only increased by 1.2%. Author believes the results to be respectable as the fuel saving was 4.75 times larger than the vehicle mass increase.

However, author doesn't believe this system to be optimal for a motorcycle. This is primarily because of the relatively low mass, which causes the braking forces to not be as high as in a bigger vehicles, such as a car. For this particular simulation, the total braking energy was only 10.0% of the total energy required to complete the driving cycle. Furthermore, the flywheel in KERS would apply gyroscopic forces on the motorcycle, which could be detrimental to the handling. Lastly, since packaging is a concern on traditional motorcycles, author believes that packaging of KERS would prove to be challenging at best and it is likely to require some compromise in motorcycle design.

4.3.2 Fully Electric

Table 4-5: Fully Electric Powertrain Specifications

Full Electric	
Energy Density (MJ/kg)	0.84 (Hatton, 2020)
Energy Capacity (MJ)	81.02
Energy Mass (kg)	97.03
Motor Power Density (kW/kg)	5.1 (Manen, 2014)
Peak Power (kW)	80.25
Motor Mass (kg)	15.74
Motor Efficiency (%)	91.00 (McLaren, 2018)
Mean Reduced Mass (kg)	50.00

Table 4-5 shows the specification of the fully electric powertrain. Author has decided to use the Formula E technology, since it is arguably the best in the world, not just in terms of power and efficiency but also the reliability. Author has assumed the reduced mass. The energy capacity as well as peak power, were both adjusted so that the used energy % and power to mass ratio were the same as in the original vehicle.

This has resulted in a total vehicle mass of 250.28kg, which is an 16.7% increase. Due to this, the peak power had to be raised to 81.02kW from original 69.10kW. However, due to thermal efficiency, which is nearly 7 times larger, the required energy capacity was just 81.02MJ, which is 87.0% less than the original 623.7MJ.

The energy saving is a clear advantage of the electric powertrain, especially since in real life, the best of the new ICEs achieve thermal efficiency about a half of the electric counterpart. However, due to extremely poor power density of the current 'best' battery technology, the motorcycle has become very heavy in its class. This problem cannot be mitigated by power alone as done in this investigation, since heavier motorcycle will experience higher forces and as a result, the actual motorcycle would most likely become even heavier to achieve safety factors. Furthermore, even if the final product would have the same power to mass ratio as original, the additional mass would have an extremely negative impact on the handling.

Therefore, author believes the electric powertrain is a competitive proposition for luxury, lightweight, city focused, commuting style motorcycles, as the negatives of this powertrain, such as additional mass or cost, won't have as big of an effect. However, company developing such machine would have to develop a convenient and much faster charging method than what is currently available, due to the 'commuting' aspect.

4.3.3 Hydrogen Internal Combustion Engine

Table 4-6: Hydrogen Powertrain Specification

Hydrogen Engine	
LHV (MJ/kg)	120.90
Peak Power (kW)	75.40
Energy Capacity (MJ)	255.16
Engine Mass (kg)	57.80
Fuel Mass (kg)	2.11
Tank to Fuel Mass Ratio	17.50
Fuel Tank Mass (kg)	36.93 (Kane, 2014)
Mean Reduced Mass (kg)	300.00
Mean Engine Thermal Efficiency (%)	33.00 (Barbir, 2013)
Specific CO ₂	0.00

Table 4-6 shows the specification for the hydrogen ICE powertrain. Just as in Fully Electric chapter, the peak power and energy capacity were both adjusted to match the original used energy % and power to mass ratio. Author has assumed the engine related masses to be the same as the original and that the tank to fuel mass ratio of Toyota Mirai to be scalable and applicable.

Due to the relatively heavy fuel tank, the total vehicle mass increased by 9.3%, to 234.4kg, requiring 75.4kW to match the original power to mass ratio. However, due to 20% more efficient engine, the energy capacity was decreased by impressive 59.1%.

Just as with the electric powertrain, the clear advantage of hydrogen ICEs is their efficiency. However, the ratio of energy capacity decrease and the vehicle mass increase of the hydrogen system is 6.35, which is 22.0% higher than the investigated electric powertrain. This means that the hydrogen system yields better energy savings and lower mass increase associated with the system. Additionally, author believes that the proposed hydrogen powertrain system would cost fraction of the proposed fully electric powertrain, since it comes from Formula E,

whereas the hydrogen technology applied in this investigation comes from production vehicles. Another advantage is that it is much quicker to refuel hydrogen tank than it is to charge a battery. Despite the clear advantages that hydrogen system has over fully electric, the hydrogen powertrain still retains the same main 'eco' characteristic that of its electric counterpart, which is the fact that the vehicle itself does not produce any carbon-related emissions. Furthermore, author believes that by employing modern technology, a motorcycle hydrogen engine could be designed to have practically zero NO_x emission as well, making it the obvious choice of the powertrain of the future.

5 Determination of the Vehicle Specification

5.1 Specification

Table 5-1: 2028 Powertrain Specifications

2028 Powertrain Specifications	
Vehicle Type	Motorcycle
Engine:	4 stroke
Layout:	Inline-3
Displacement (cc)	449
Fuel Type	Hydrogen
Valvetrain	Desmodromic 4V per cylinder
Fuel Control	2 Direct Injectors
Combustion System	Turbulent Jet Ignition
Charging	Single High Response Turbocharger
Compression Ratio	15:1
Thermal Efficiency (%)	55
Peak Power (kW/PS)	105 / 143 @ 10,000rpm
Peak Torque (Nm)	100 @ 7,000rpm
Peak RPM	14000
Wet Mass (kg)	215

Table 5-1 shows the author's proposition for the 2028 powertrain. Author has decided to downsize the engine from 800cc to 449cc, this was done so that the engine becomes physically smaller, creating space for the turbocharger. It also makes the bores smaller, which aids combustion rate, as the flame front will reach the cylinder walls quicker with the smaller bore. Small capacity also causes a decrease of fuel consumption at idle and slow traffic.

Author has decided to keep the inline-3 for two reasons: 1) Triumph has created loyal customers and their inline-3 is the epicentre of that, as well as brand's signature for the 'modern front'. 2) combined with low capacity, the stroke will become very small, even if the engine is designed to be under square to enhance the combustion rate even further. This will allow the engine to reach higher rpms safer, due to decreased mean piston speeds and therefore forces acting on crucial components such as the crank, connecting rods and pistons.

Naturally, the fuel used will be hydrogen, as it has already been discovered in Hydrogen Internal Combustion Engine chapter, it is the optimal technology which company such as Triumph should pursue. It allows ICE to become truly 'emissions free' (in a local sense), which is extremely important in today's society, as people are becoming more environmentally conscious. Author believes strongly in hydrogen combustion because of its appeal to both major customer types, one of which is looking for adrenaline and sound, the other is looking for eco-friendly machine. Hydrogen is also part of the strategy of achieving a very high thermal efficiency of 55%. On average, hydrogen ICE is about 20% more efficient than their petrol counterpart (Barbir, 2013). This is mainly because of 2 factors: 1) flame temperature which is about 200°C cooler than petrol (Bent Sørensen, 2018), causing the ratio of specific heats to increase, which results in higher efficiency. 2) the maximum flame velocity of

hydrogen is about 8 times higher than of petrol, resulting in almost instantaneous and therefore ideal combustion (Bent Sørensen, 2018).

Author has also decided to attempt to incorporate advanced valvetrain system which would reduce frictional losses. Advanced low friction valvetrain solutions do exist, for example Freevalve in the Koenigsegg Gemera. However, author believes that such solutions are too novel for company like Triumph, especially since author is already proposing a novel combustion system. Therefore, author has decided to incorporate Ducati's Desmodromic valvetrain, as the patent has expired and it has already been proven to be reliable, manufacturable and affordable in a 'mass produced' motorcycle.

Another strategy for obtaining high thermal efficiency is the incorporation of a small and responsive turbocharger, as it will extract work from the wasted heat. However, the turbocharger will have to be very small, both on the turbine and compressor side, to aid the controllability of the power, as it is much more crucial than in other turbocharged vehicles such as cars. It is very likely that Triumph will have to work very closely with turbocharger experts such as Garrett, in order to create a turbocharger which will deliver satisfactory response for a motorcycle. Author believes this to be possible since such collaboration is extremely common in the automotive world of cars.

The turbocharger is also the primal reason behind very high peak power and torque of 105kW and 100Nm respectively, from just 449cc. Author believes that boost pressure of about 1.10 Bar will be required to achieve this. The unheard-of combination of high boost pressure and compression ratio of 15:1 will ensure that the power, torque and efficiency targets are met. Author believes this combination to be possible due to 2 reasons: 1) hydrogen has extremely high anti knock properties and it requires higher temperatures for spontaneous ignition (Bent Sørensen, 2018). 2) hydrogen can combust in very wide range of mixtures, including extremely lean, which enhances the anti-knocking ability even further (Bent Sørensen, 2018).

The next reason why author believes 55% thermally efficient engine is possible, is the incorporation of Turbulent Jet Ignition. This system works by separating the main combustion chamber with nozzles, from which plasma propagates as a result of combustion in the pre-chamber. The plasma streams then create many very high temperature ignition sources in the main chamber, which cause very fast combustion. Due to the high ignition energy, this system also allows for combustion of much leaner mixtures than what was previously possible, which would allow extreme reduction of NO_x, making the ICE practically emit zero emissions. Author is unsure whether or not this system will be required, as the hydrogen promises these combustion characteristics on its own. Therefore, author proposes to incorporate this system as a failsafe, in case the target thermal efficiency cannot be obtained with the original strategies.

Author realises the proposal to have very ambitious targets. However, author believes that those targets are achievable since all of the incorporated technologies have already been proven to work in real life. Additionally, author believes that due to the current environmental and social challenges, companies such as Triumph must progress at much higher pace than what they are used to, otherwise they won't be able to compete in the 2028 market.

5.2 Analysis

Table 5-2: 2028 Powertrain Specification

2028 Engine	
LHV (MJ/kg)	120.90
Peak Power (kW)	105.00
Energy Capacity (MJ)	283.21
Engine Mass (kg)	40.00
Fuel Mass (kg)	2.34
Tank to Fuel Mass Ratio	15.00
Fuel Tank Mass (kg)	35.14
Mean Reduced Mass (kg)	300.00
Initial Engine Thermal Efficiency (%)	15.00
Thermal Eff. Incr. due to turbocharging and DI (%)	20.00
Thermal Eff. Incr. due to TJI (%)	10.00
Thermal Eff. Incr. due to Hydrogen (%)	10.00
Final Thermal Efficiency (%)	55.00
Specific CO ₂	0.00

Table 5-2 shows the powertrain specification of the 2028 proposal engine. Author has assumed that the engine mass will be reduced by 30.8%, due to the downsizing. The energy capacity has been adjusted so that the vehicle mass becomes 215kg and tank to fuel mass ratio has been assumed to be improved by 14.3%.

Since author believes the MPG to not be the optimal fuel efficiency index for hydrogen, due to inconsistent fuel tank pressures. Author has calculated the specific range, which is the distance vehicle can travel on 1kg of fuel.

The original vehicle has scored 23.6 kmkg⁻¹, whereas the initial hydrogen engine and the 2028 engine scored 158.5kmkg⁻¹ and 271.0kmkg⁻¹, respectively. Making the range on single fuel tank/ charge, of the original vehicle, fully electric as well as the initial hydrogen powertrain to be 334.5km, whereas the 2028 powertrain can cover 634.8km, which is 1.9 times longer than the original.

6 Conclusion

Despite the data being very unreliable and therefore limiting, author believes that a lot of useful information has been extracted, all of which correlated with author's expectations and experience.

Author believes that the model created for the purpose of this investigation to be a good starting point, as it has proven to calculate reasonable and logical results, all of which author was able to explain with rudimentary physics.

The investigation of alternative powertrain systems and energy recovery system had not only shown the potential advantages and limitations of each system, but also the versatility and applicability of the previously created drive cycle model. The investigation has proven to be extremely useful in the determination of the 2028 powertrain system, as it has allowed the author to calculate relative indexes which gave magnitude to all of the considerations, allowing author to create an educated decision which was backed up with values.

Lastly, the 2028 powertrain proposal is a very big challenge for Triumph. However, author believes the challenge to be realistic, given the current technology as well as the growing dangers in social as well as environmental aspects.

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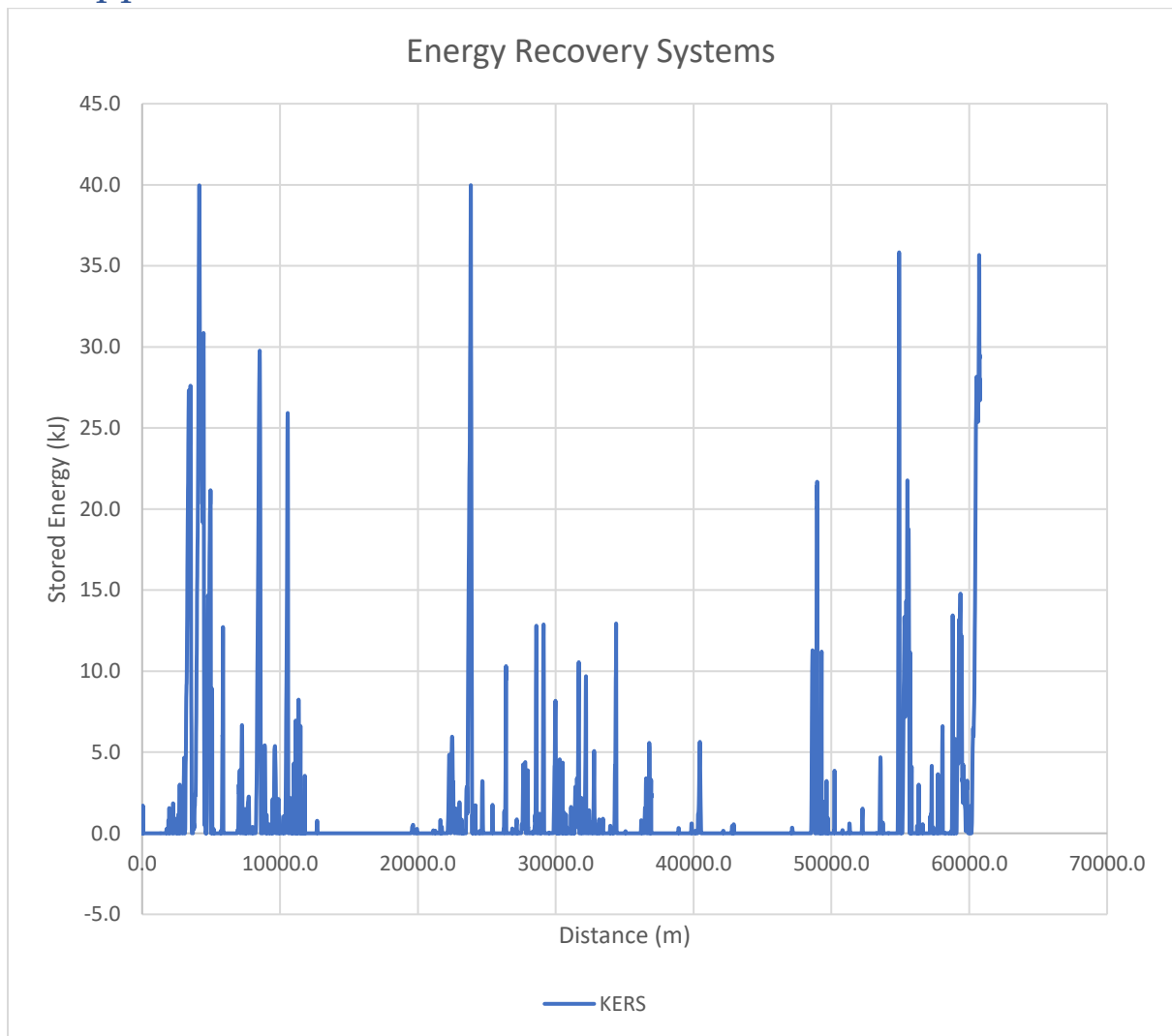
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8 Appendix



Appendix 1: Stored Recovered Energy - original vehicle