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Faculty of Engineering and Surveying

An Analysis of Kinetic Energy Recovery Systems and their potential for contemporary Internal Combustion Engine powered vehicles

A dissertation submitted by

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Abstract

The Internal Combustion Engine has played an incomprehensible role in contemporary society ever since its invention. Oil shortages will almost certainly eventually lead towards a search for propulsion from renewable sources, but for the time being there is no sign of any significant alternative for everyday transport. Any product that offers a fuel economy improvement is of benefit to both the individual and the environment. As vehicles speed up, they convert stored energy into kinetic energy. As the mass or velocity increases, the kinetic energy will also increase. It is for this reason that light commercial vehicles on our roads have so much kinetic energy when travelling at speed. The concept of being able to recover this energy when braking is the foundation for regenerative braking or Kinetic Energy Recovery. The energy captured is then stored to be used in the future: in most cases it is converted back into kinetic energy to bring the vehicle back to speed. The technology is particularly effective in drive cycles consisting of frequent stop-start driving.

This project seeks to investigate the feasibility of a mechanical Kinetic Energy Recovery System for implementation via a retrofit on existing light commercial vehicles. In order to be effective, the system must be cost effective and easy to implement. The objective was to design a system able to be fitted to a large number of vehicle platforms and with a reasonable payback period.

A literature review was carried out to discern the most appropriate system for light commercial vehicles. Existing systems were analysed and their benefit was appraised from a retrofit stance. A flywheel system was chosen due to its recent success in F1 and its very high energy density amongst oher factors. A system was designed to be fitted to a representative vehicle, with potential to be fitted to other platforms. The theory of operation, driveline configuration and attachment options were developed. The system was modelled in Creo and a Matlab code was developed to calculate the potential fuel savings under different circumstances using drive cycles.

The dissertation found that the technology was conceptually viable. A vehicle of mass 2680kg with load would save \$0.91 per 100km (6.9% saving). If the vehicle were fully laden, the fuel saving would be \$1.64 per 100km (7.6% saving). The total cost of the system was found to be \$2680. The repayment period ranged from 5-8years to a best case scenario of 3-4 years.

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Nomenclature

ICE – Internal Combustion Engine
GVM – Gross Vehicle Mass
CVT – Constant Variable Transmission
KERS – Kinetic Energy Recovery System
KE – Kinetic Energy
GPE – Gravitational Potential Energy
UDDS – Urban Dynamometer Driving Schedule
RPM – Revs Per Minute
MVB – Metal Variable Belt
PTO – Power Take Off
IRS – Independent Rear Suspension

1 Introduction

Ever since the Industrial Revolution, fossil fuels have been a major source of energy for most of humankind. They have played an incomprehensible role in raising our socio-economic status and power our everyday lives. One prediction is that fossil fuel reserves will be sufficiently depleted by 2042 that shortages will occur (Shahriar Shafiee, 2009). Whether we like it or not, we live in a society dominated by vehicles. We rely on their existence for the transportation of both people and goods. The most common form of energy for powering vehicles remains the Internal Combustion Engine (ICE), as it has ever since its creation. There has been a mounting consensus that electric vehicles will be the future, but there is no efficient means of storing energy that will render the ICE obsolete as of yet, or in the foreseeable future.

Sustainability and environmental concerns are growing as we realise as a society just how dependent we are on fossil fuels. As oil supplies dwindle, fuel prices will continue to increase and the vehicles that are used will need to meet new standards of fuel economy. The world is also becoming more environmentally conscious. The full extent of the detrimental effect of emissions on both the human race and the environment is not fully known, and the more we can do to reduce this impact, the better.

It is clear that any technology that is capable of saving fuel will be of a benefit to the general public. It makes for cheaper running of vehicles, lower emissions and lower consumption of non-renewable energy sources overall.

1.1Energy Losses in Vehicles

A vehicle experiences many energy losses due to the internal engine friction, parasitic losses, drivetrain losses and braking losses. Most of these can be reduced by improvements to components and their subsequent efficiency. However the typical braking process is the only area that cannot be improved in the way of efficiency. A system needs to be devised that harnesses the energy that would otherwise be wasted as heat. This is the only way the process will be made more efficient. Lately there has been a huge improvement in both fuel economy and performance in new commercial vehicles in Australia due to technologies such as common rail diesel and improvements in aerodynamics, yet braking is an area that will always lose an amount of energy proportional to the vehicle weight. As the weights of commercial vehicles have not changed substantially in recent times, the area of regenerative braking has a definite market.

This dissertation focusses on regenerative braking based on mechanical principles; this point is outlined in section 1.5.

Many different figures for braking energy lost in urban environments have been published using different driving cycles and vehicles. For an urban application, small vehicles are considered to waste roughly 10% of combustion energy on braking (US Department of Energy, 2010), whereas the figures for large, commercial vehicles are as high as 30% according to PULKIT GUPTA (2014) and Clegg (1996). The driving cycle utilised for modelling obviously has a large impact on the figure.

1.2Theory of Braking

As vehicles speed up, they convert stored energy into kinetic energy. Kinetic energy is given by:

$$KE = \frac{1}{2}mv^2 \qquad (1)$$

Where $m = mass$

$v = velocity$

It can be seen that as the mass or velocity increases, the kinetic energy will also increase. It is for this reason that vehicles on our roads have so much kinetic energy when travelling at speed. When a conventional vehicle must be stopped, the brake pads are applied to the brake rotors: the friction caused by this slows the vehicle. In this process, kinetic energy is dissipated into heat energy. The heat energy is unusable and is absorbed by the atmosphere, where it is no longer of use. The concept of being able to recover this energy is the foundation for regenerative braking, or Kinetic Energy Recovery (KERS). “A Regenerative Brake is a Mechanism that reduces vehicle speed by converting some of its kinetic energy into some other kind of useful form of energy" (Chibulka, 2009). The energy captured is then stored to be used in the future: in most cases it is converted back into kinetic energy to bring the vehicle back to speed.

1.3Hybrid Vehicles

"A vehicle which contains two such sources of propulsion [commonly] an internal combustion engine (ICE) and an energy storage device) is known as a hybrid system" (Clegg). As such, a vehicle with regenerative braking is often put into the category of hybrid.

Hybrid vehicles are generally broken up into two main varieties: series and parallel. A parallel hybrid will have a supplementary system (regenerative) running in parallel to the original driveline setup. There are many different combinations as a result of the different attachment options. A series hybrid design involves altering the original driveline to implement the supplementary system. This has many advantages, but often costs more to implement and involves much alteration to the original driveline design.

Hybrids also have the added benefit of being able to draw upon stored power whenever it is required. As discussed later in the literature review: in F1 applications, the regenerative braking stored energy was often used principally for overtaking manoeuvres, showing that if implemented correctly, the technology can be used as a 'boost' as well as just an aid to starting from a stationary position. This allows for the downsizing of engines in newer vehicles fitted with such technology, but in the case of a retrofit, gives older vehicles a performance boost to close the engine technology gap.

1.4Electric KERS

The most well established hybrid technology still remains the electric hybrid. Initial research revealed the general trend that electric regenerative braking is effective for small, low energy applications, but not so much for larger applications, where more kinetic energy is available to be recovered. It also appears that the electrical systems are only cost effective due to the fact that the electric motor and batteries already exist in the design of electric hybrid vehicles, making the regenerative side a small investment. On existing internal combustion engines however, research suggests that mechanical systems may be more effective at harnessing the braking energy due to the energy storage capacity and the rate at which energy can be stored and utilised. “The constraints for high performance batteries are high specific energy density, high discharge rate and high number of discharge cycles. At present all three are mutually exclusive” (Clegg, 1996). More recent research also suggests that electrical systems are not effective for applications requiring not just a high energy density, but also a high power density (time rate of energy transfer per unit volume).

A study by Midgley and Cebon (2012), hypothesised that the mechanical systems are likely to be up to 33% smaller and 20% lighter than the closest electrical counterparts and therefore would be a logical selection for heavy goods vehicles. There is also no memory effect as is common with batteries.

1.5Feasibility Requirements for Design

A technology is only successful if it can be feasibly implemented into a relevant market. The Australian Government Department of Resources Energy and Tourism (2012) states that, “the main determinant of suitability within the Australian market appears to be whether sufficient energy can be recovered from braking to offset the disadvantages of carrying the system’s additional weight.” They see particular use in the technology for light commercial vehicles, buses and trucks operating in urban environments. The suitable operating conditions are a drive cycle of frequent stops or highly variable speeds (Clegg, 1996). Scenarios where inertia plays a large part would have significant gain from regenerative systems. This project will involve an economic appraisal to discern the overall cost benefit of such a system.

1.6Aims and Objectives

This dissertation aims to explore the potential role of Kinetic Energy Recovery Systems, also referred to as regenerative braking, in the future, and if possible, develop a cost effective, feasible system for internal combustion engine powered vehicles which can be easily implemented and potentially retro-fitted to existing vehicles.

The end objective of the dissertation was to develop a Kinetic Energy Recovery System for light commercial vehicles that improves fuel efficiencies by a substantial amount, with a low added cost. The literature review will hopefully shed enough light on the topic that an informed decision can be made on an appropriate technology and the requirements for successful implementation. The justification of the system and its potential came in the form of a cost-benefit analysis of fuel-savings from simulation compared to the cost of the system. The specific detailed objectives and their order are listed below.

1) Perform a Literature review of current mechanical KERS technologies: their applications, limitations and successes in the past. As technology advances and the human race looks for alternative energy sources, the future of ICE powered vehicles isn't a certainty. Trends point towards the potential of Electric Vehicles or Hybrid Electric Vehicles if electrical storage technology advances enough to facilitate this. However for the foreseeable future, ICE powered vehicles will still play a prominent role and any potential fuel saving solution will be of use if it is feasible, cost effective and easy to implement. The review should centre on these factors.
2) Choose a technology with the most potential for commercial vehicles in Australia and analyse its feasibility and potential benefit from a retrofit stance. This analysis will include deciding which component of the engine or drivetrain would be most effective to recover energy from and how the system could accommodate different vehicles.
3) Design the chosen system to be retrofitted for the application.
4) Model potential fuel cost savings based on simulation using Matlab software.
5) Assess manufacturing cost of designed system.
6) Using the cost of the system and the potential fuel savings, calculate a repayment period and comment on the overall feasibility of using this technology.

1.7Project Scope

The research undertaken in this project will focus specifically on light commercial vehicles powered by ICEs. There will be limitation on the scope to ensure that the project is both manageable and thorough in the time period.

The engines considered will be petrol or turbo diesel powered and the maximum GVM will be 3500kg. The study investigates regenerative braking and the feasible applications. It does not consider braking optimisation or control systems. The design and optimisation of transmissions (including CVTs) is not considered and assumptions are made as to the functionality of said systems in the design and modelling phase. A representative vehicle is used for the design as an example of the implementation of the design. The system was intended to be suitable over many vehicle platforms and necessary changes between platforms were minimised by relying on common components, aiding other potential retrofits.

1.8Methodology

The project began with a literature review of current mechanical KERS, as stated in the project specifications. Quantitive research methodologies were applied where appropriate data was available. Qualitive methodologies were used to compare the opinions of academics and companies on technologies. To begin the literature review, the theory of braking and the feasibility requirements of new systems were researched. Before the project specifications were completed, it was decided that mechanical systems, as opposed to the more publicised electrical ones, would be the subject of the project. This was researched and discussed further as justification for the decision.

Vehicle specifications were researched for a representative commercial vehicle and relevant constraints and objectives were developed. The rest of the review centred on the analysis of existing systems: both theoretical and practical applications. The EPA and the Australian Government Department of Industry and Science were chosen as starting points as the organisations synthesise a lot of relevant findings on regenerative braking systems and present a solid starting point. A thesis was provided from a previous year on using the ICE as a pneumatic compressor, functioning as a regenerative system and this technology was analysed for further potential.

As with any literature review, the aim was to narrow the scope that was initially very large, to a more refined research methodology, able to cover the topics required to yield results. The different types of technologies, as well as their histories were reported. Through the initial broad research, the major contributors to the technologies were realised. Organisations such as Formula 1, Ford, Eaton and Permodrive were analysed in more detail due to the recent contributions to the technology. Precedence was given to empirical data and applications over analytical data.

The review focussed on energy storage, feasibility, cost-effectiveness and ease of implementation for light commercial vehicles. The vehicle parameters mentioned previously were used to assist in the determination of feasibility and the available energy from braking was used as a benchmark against already available systems. For transparency, two recent studies into the effect of different hybridisation techniques on fuel economy were reviewed and the opinions of the authors taken into consideration.

For the design phase, the project was broken up into subsystems that would form the overall regenerative braking unit. Where possible, quantitive comparisons were used to aid in design decisions, but in the absence of data, decisions were made using logical statements. The location of power harnessing and transmission were decided upon based on the potential for the system being fitted to multiple vehicle platforms. A suitable system was designed using both Matlab and hand calculations. This was then modelled using Creo. Obviously safety will be of paramount performance for this design.

Matlab was then be used for the fuel saving modelling phase. This was based on existing drive cycles that were deemed as a suitable representation of driving conditions. Assumptions, pertaining to the transmission and driveline were made in order to gain results.

The theoretical cost of the system was calculated and a cost benefit analysis was be performed. This determined overall the feasibility of the system and whether or not the technology has a place in contemporary society.

1.9Design Requirements

In order to gauge the suitability and feasibility of Kinetic Energy Recovery Systems, it is vital to have an idea of the design requirements of the system. It must be able to be retrofitted to existing light commercial vehicles as stipulated in the project specification. This is defined as being a commercial vehicle carrier with a gross vehicle mass (GVM) of less than 4.5 tonnes. This category includes vans and utilities. The applications of these vehicles are commonly delivery vehicles and tradesman vehicles. The system chosen must have the capability of storing the required amount of energy that would be lost during braking.

1.10 Energy

In order to calculate the requirements of the system, it is necessary to contemplate the conditions that the system will operate under. The change in velocity will be taken to be 60km/h to 0 km/h. This represents a braking event from the common maximum speed limit in cities to stop. This is realistically the common maximum braking event a light commercial vehicle will experience, as complete highway stops are uncommon. Gravitational Potential Energy (GPE) will also be taken into account, with a gradient of 6% when the braking is taking place. The solution must also be cost effective and be easy to implement and retrofitted to existing vehicles. A vehicle was chosen as a reference. This was a Toyota Hilux 2010 tray back v6. This has a kerb mass of 1680kg and it is assumed that it has a load of 1000kg (Redbook, 2010). This could be a trailer or load on the tray. This leads to a mass of 2680kg. Using equation (1) and the equation for GPE, the available energy can be calculated:

$$KE = \frac{1}{2}mv^2$$

$$KE = \frac{1}{2}(2680)\left(\frac{60}{3.6} - 0\right)^2$$

$$KE = 372.2\ kJ$$

The braking event is considered to last for 5 seconds on a 6% grade. Using kinematics:

$$v = u + at \tag{2}$$

$$\left(\frac{60}{3.6}\right) = 0 + a(5)$$

$$a = 3.33m/s$$

Substituting to find distance:

$$s = ut + \frac{1}{2}at^2 \tag{3}$$

$$s = \left(\frac{60}{3.6}\right)(5) + \frac{1}{2}(-3.33)(5)^2$$

$$s = 41.7m$$

Assuming a grade of 10% for braking event:

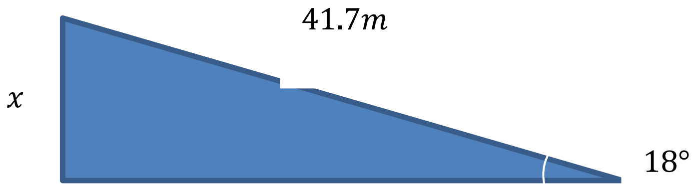

Figure 1 – Elevation case for design constraints

$$x = \sin(18) \times 41.7$$

$$x = 12.89m$$

Calculating GPE:

$$GPE = mgh \qquad (4)$$

$$GPE = 2680 \times 9.81 \times 12.89$$

$$GPE = 338.8kJ$$

Summing GPE and KE:

$$E_{avail} = (338.8 + 372.2)$$

$$E_{avail} = 711\ kJ$$

Therefore the desired system capacity will be the efficiency of the system multiplied by the available energy.

Assuming a 90% mechanical efficiency for the system:

$$E = E_{avail} \times \varepsilon$$

$$E = 711 \times 0.9$$

$$E = 639.39kJ$$

This energy value will be used as the design constraint. It is logical to assume that the energy constraint is based on the deceleration energy available, rather than the energy required by the vehicle. Another requirement of the system is that it must not exceed the GVM when the mass of the new system is added.

1.11 Power

In order to calculate the power requirements of the design, existing drive cycles were used to give a reasonable approximation of values based on decelerations.

1.11.1 Driving Cycles

A driving cycle represents a set of vehicle speed points versus time (Nicolas, 2013). They are generally used to calculate fuel consumption and can be used for generating other data. There are a number of different driving cycles that have been produced and are used for different applications. These will need to be consulted for the modelling stage of the project in order to calculate the potential fuel savings.

The driving cycle chosen was the US Light-duty FTP-75 Urban Dynamometer Driving Schedule (UDDS) or ADR-37 in Australia. It is commonly used for fuel economy testing of light-duty vehicles. It is designed for use in the US, although should prove suitable under Australian conditions as well, as it has been adopted here as a standard as the speeds are comparable to our limits. Both climatically and on driving style, our country should be similar to the US. The basic parameters are listed below:

- Distance travelled: 12.07 km
- Duration: 1369 seconds
- Average speed: 31.5 km/h
- Maximum speed: 91.2 km/h (Transport Policy, 2014)

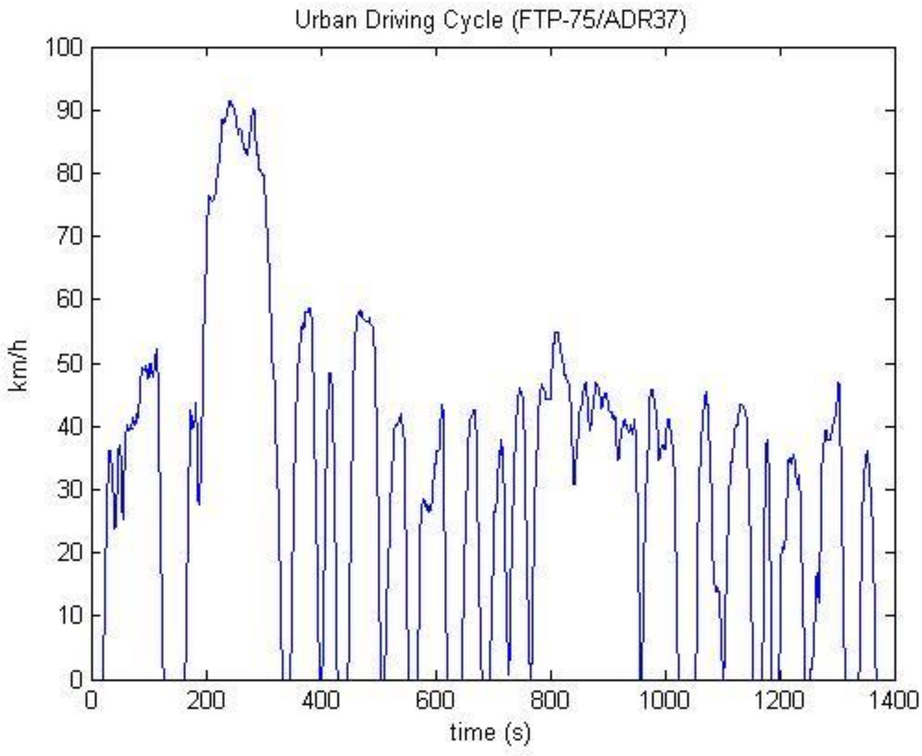

Figure 2 – FTP-75 Driving Cycle

This driving cycle was modelled and plotted in Matlab (See Figure 2). The changes in velocity were then created as a separate matrix and the minimum and average were calculated. The Kinetic energy was then calculated by assuming the vehicle mass and calculating the ΔKE on a per second basis using the maximum deceleration from 60km/h. The results are listed below:

- Maximum deceleration $= -1.47\frac{m}{s^2}$
- Average Deceleration $= -0.58\ \frac{m}{s^2}$
- $\frac{dKE_{max}}{dt} = P_{max} = 50.05kW$
- $\frac{dKE_{avg}}{dt} = P_{avg} = 7.2kW$

As expected, the average power is significantly less than the maximum. These values will be used for the design of the power transmission system. The Change in Kinetic Energy was also plotted for the drive cycle. Anything below the x axis is considered to be a braking event (see Figure 3).

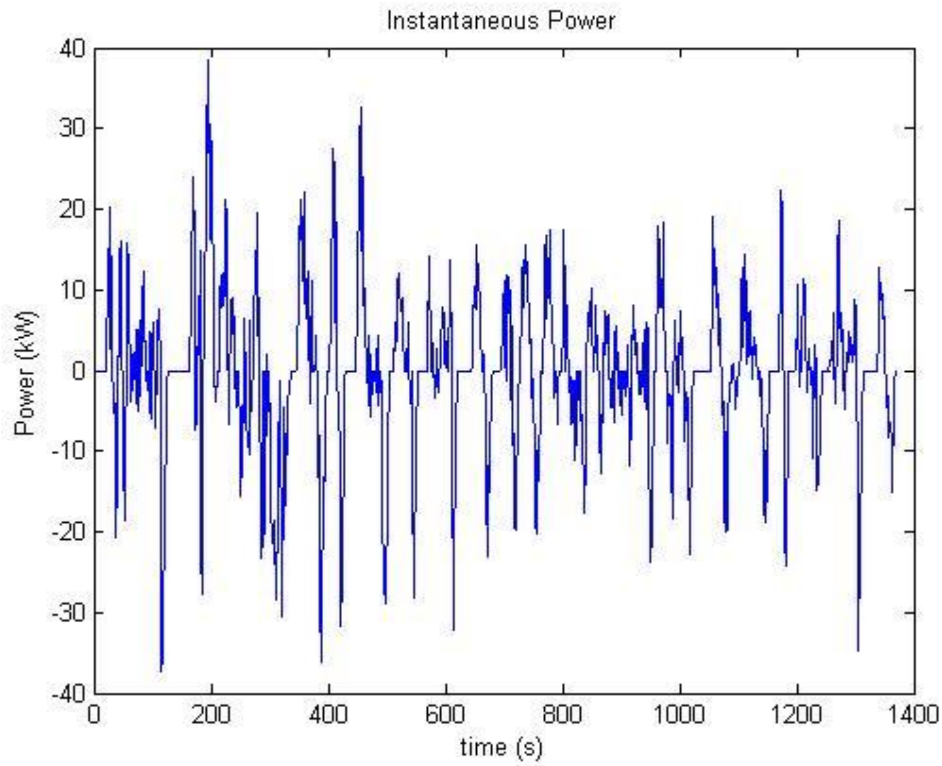

Figure 3 – Change in Kinetic Energy for FTP75 Driving Cycle

2 Mechanical Kinetic Energy Recovery Systems

2.1Mechanical (Flywheel)

2.1.1 Background and History

Flywheels have always been an obvious choice for energy storage as they are able to store large amounts of energy and release this energy very quickly when required. Flywheel Energy Storage revolves around accelerating an inertial mass to a very high speed and effectively transforming kinetic energy to rotational energy (Chibulka, 2009). An early application was the Swiss 'Electrogyro' bus, which ran entirely on energy extracted from a flywheel. This vehicle had a limited range of 6km and relied upon a 1.5t steel flywheel. The eventual downfall was due to safety and efficiency issues after a few years (1980). General Motors also developed a concept utilising engine-flywheel hybrid technology. The project was abandoned as the marginal 13% increase in fuel economy was deemed insufficient to justify the complex drivetrain (Schilke, 1988). A prototype was never built. Many flywheel technologies developed in the 90's involved using electricity to accelerate and decelerate the flywheel, effectively using it as an electrical battery. This process has obvious disadvantages, due to the many energy conversions: each with its own set of inherent losses. Most modern Kinetic Hybrids store kinetic energy by utilising flywheels and a Continuously Variable Transmission (CVT) to transfer energy to the drivetrain (Midgley & Cebon, 2012). The most recent move towards efficient systems that directly store and harness energy from a flywheel could likely be attributed to the influence of Formula 1. In the 2009 Season, the FIA authorised the use of KERS systems on the track (Patel, 2010). This led to the development of systems by several vehicle manufacturers. These recent systems, as well as others, will be analysed in the following sections as they showed potential for a modern application.

2.1.2 Basic Technical Analysis

The energy that a flywheel can store is given by:

$$E_f = \frac{1}{2} I w^2 \tag{5}$$

Where I = Rotational Inertia

ω = Rotational Velocity

It can be seen that either I or ω must be maximised for maximum energy storage. Equation (5) gives the total energy that a flywheel will have when rotation at an angular velocity however it is not feasible to assume that a transmission will be capable of an infinite ratio, which would be required to bring the flywheel to a stationary position and utilise all of the energy. As a result of this, the energy stored needs to be calculated by taking the ΔKE between the ω values. This results in the following equation:

$$E_{stored} = \frac{1}{2} I(\omega_{max}^2 - \omega_{min}^2) \tag{6}$$

Equation (6) can then be rearranged to give a suitable form to calculate the energy delivered:

$$E_{delivered} = \frac{1}{2} I\omega_{max}^2 \left[1 - \frac{\omega_{min}^2}{\omega_{max}^2}\right] \tag{7}$$

James Hanson (2011) and (Thoolen, 1993) state that the most efficient ratio of $\frac{\omega_{max}}{\omega_{min}} = 0.5$. A very low ratio will put high demand on the transmission. The effect of the ratio will be examined in section 3.3.

In order to give the maximum energy, the flywheel must be spun as quickly as the material will allow. Recent advances in materials science have enabled much higher rotational speeds, and therefore higher storage capacity (Midgley & Cebon, 2012). By analysing formulae it can be seen that the flywheel performance is broadly determined by:

- Material Strength
  - This determines the maximum RPM and therefore leads to greater energy storage
- Rotational Speed
  - This determines the energy stored. A higher velocity will give more kinetic energy but greater loads on the flywheel and bearings
- Geometry
  - This determines the specific strength, which determines the specific energy capacity of the system

Being a mechanical system, mechanical losses such as aerodynamic drag exist. Housing the flywheel in a vacuum is a common tactic to reduce this. The high speeds also cause heat and possibly delamination issues. The losses due to the friction of air are known as windage losses. The windage losses increase as the vacuum decreases.

Xin Yang, a researcher at Yinbin University, performed an analysis into the application of flywheels on modern vehicles in 2012. The article agrees with Midgley and Cebon in regard to the high energy storage capacity and goes as far as to say that the capacity is higher than that of typical Hydraulic systems (Yang, 2012).

2.1.2.1 Gyroscopic Effect

An inherent issue with flywheel designs is the gyroscopic effect. "The implication of the conservation of angular momentum is that the angular momentum of the rotor maintains not only its magnitude, but also its direction in space in the absence of external torque" (Nave, 2000). When dealing with moving vehicles this can create significant problems when cornering, as the flywheel will resist a change in its angular momentum vector and attempt to keep the vehicle moving on its original path. The Angular momentum is given by:

$$L = I\omega \quad (8)$$

Where $\omega = Angular\ Velocity$

$I = Moment\ of\ Inertia\ of\ the\ object$

For a solid flywheel:

$$I = kmr^2 \quad (9)$$

Where $k = inertial\ constant\ (shape\ dependent)$

$m = mass$

k is equal to roughly 0.6 for a Flat solid disc (Engineering Toolbox, 2010). The applications of these equations and other k factors will be discussed in section 3.3.

2.1.3 Continuously Variable Transmission (CVT)

This literature review found that a CVT was the most effective means of transmitting power for the majority of flywheel technologies. The concept of a CVT will be explored for the sake of background and context. Generally a gearbox will provide a fixed set of ratios that are used to transform the RPM of the output shaft to an appropriate speed for the axle, in order to make the most effective use of the available energy. This is especially important on flywheel systems, where the output speed is able to vary so considerably. In the case of charging, a CVT must match the speeds of both the flywheel and the driveshaft, and then vary the ratio to initiate energy transfer. A CVT is different in the way that it continually changes the ratio, giving a very large number of available ratios. There are many variations, all with specific characteristics. The toroidal and pulley belt type were the most common types found within the literature.

2.1.3.1 Toroidal Systems

A toroidal system consists of discs with toroidal surfaces coupled with rollers sitting between the discs. The input and output shaft are connected to the discs on each side of the rollers. Each angular position of the rollers gives a certain gear ratio between the two shafts (Figure 4). This design is utilised by Torotrak and other manufacturers. It gives a wide ratio range, is easy to control and often has efficiencies of over 90%.

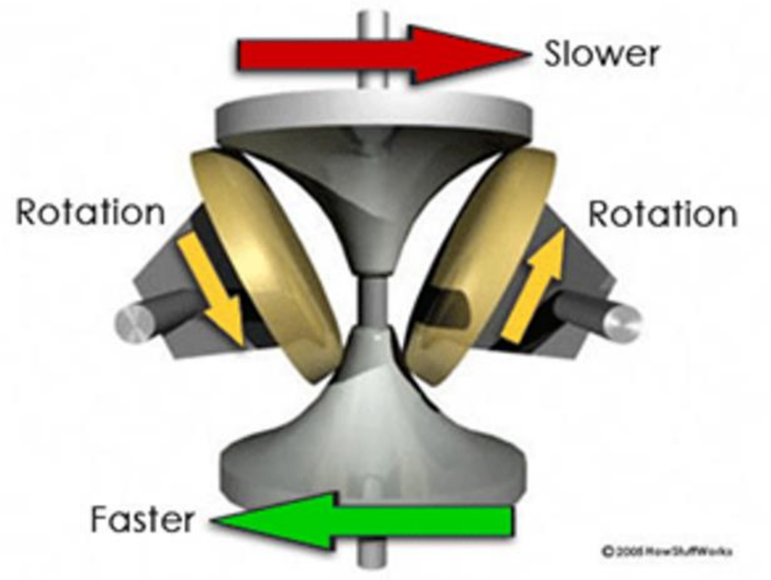

Figure 4 – Toriodal CVT Design (Harris, 2015)

2.1.3.2 Hydrostatic Systems

A hydrostatic system uses variable displacement hydraulic pumps and motors to vary the input to output ratio. Such a system is reliable, controllable and durable, although they are known to be heavy and have poor efficiencies due to the changing of energy forms Pochiraju (2011). They would not be particularly suitable for a flywheel application.

2.1.3.3 Pulley Systems

A variable Diameter Pulley CVT consists of two pulleys connected by a V-belt. The pulley sheave (groove along the edge) is able to be moved on each pulley. This moves the belt up or down on the pulley and varies the effective diameter of each pulley. This enables the creation of infinite gear ratios. The second pulleys diameter will decrease as the first pulleys diameter decreases, in order to maintain belt tension (Pochiraju, 2011). This design is easy to control and simple in design.

2.1.4 Applications

2.1.4.1 Flybrid Systems (F1 Application)

One of the first developments to come out of the variation of the F1 regulations was from a company called Flybrid Systems. They created a unique flywheel and utilised a “Torotrak” Toroidal CVT. The variation of gear ratios is key to the control of energy storage and recovery (Torotrak, 2015). The ratio must work to speed up the flywheel when axle speeds are actually slowing down, and vice versa. The Flybrid flywheel reaches incredible speeds when compared with earlier Flywheel Technologies. The specifications state that it can reach speeds of up to 60000 rpm due to its light weight and reduction of gyroscopic forces. The principle of energy storage remains the same as for other earlier technologies.

Specifications and detailed information are difficult to find due to the patented nature of the new technology. The website states that the flywheel material used is a carbon fibre steel composite. The flywheel is able to be contained in the event of a catastrophic failure which is a significant safety risk in the case of a flywheel. The casing is vacuum sealed, as the flywheel is spinning at such a high angular velocity and air resistance would cripple its storage means. The original design was connected to the transmission of the car on the output side of the gearbox using several fixed ratio gears, a clutch and a CVT, making it a parallel system. In order to comply with the strict F1 regulations, the unit was only able to output 60kW of power and store 400kJ of energy (Patel, 2010). They were able to accomplish this with a final weight of 25kg and a volume of 13 litres. The Flywheel setup can be seen in Figure 5.

Figure 5 – Flybrid Flywheel setup (Patel, 2010)

Torotrak has recently acquired the Flybrid share of the company as well and now is working to make the technology commercially available. The company states that the technology is suitable for use in commercial vehicles and even passenger vehicles. Volvo fitted a sedan with the technology (renamed M-KERS) and claimed it was able to achieve fuel savings of up to 25% (Torotrak, 2015). It was able to utilise a 55kW boost, and the unit is able to store energy for up to 30 minutes effectively. This is difficult to verify as no independent testing has been done. The most expensive part of the design is the CVT transmission, which was thought to be vital to the functionality of the design, although Flybrid now claims that they can utilise only a three gear system effectively with clutch slippage (Nathan, 2011). Using the existing gearbox, many possible ratios are made available.

This particular system offers substantial advantages over its electrical counterpart, which is far more prominent in emerging vehicles. As with most mechanical systems, there is a very high power density and low conversion losses as energy is converted from kinetic to kinetic. Whilst there is a high power density, the energy density is not as high as competing electrical systems (Nathan, 2011). The flywheel itself is quite small in this design. The weight of the system is very low at roughly 60kg (Torotrak, 2015). The company quotes this as being about a third of competing battery systems (Birch, 2013). The service life is advertised as 250000kms.

This technology is of particular interest as Torotrak states that there is the potential for a retrofit, which makes sense given the small volume the systems occupy and the miniscule weight. The system functions as a parallel hybrid, which is ideal from a retrofit stance. The road prototype has had some changes from the initial F1 model. The unit now has the potential to accelerate a vehicle from rest using just the on board unit. The marketed systems are also able to store energy when the vehicle is not braking to optimise the engine efficiency. Other sources have eluded to the potential of flywheel systems to give a "boost" effect similar to the uses in F1. Torotrak quotes a vague repayment period of 5 years, but this is very open to interpretation and there would be many contributing factors.

This technology obviously has a huge potential and it's useful to see some innovation in this area recently. Only time will tell if the Torotrak technology has the potential to be fitted to new vehicles in the future. The technology provides significant information relevant to designing a retrofit.

Gyroscopic Force Reduction

Flybrid systems list the reduction of gyroscopic forces as a definite advantage over earlier technologies. In order to assess the feasibility of these claims, the underlying physics presented in section 2.1.2 was examined. As discussed earlier, the KE is given by:

$$E_f = \frac{1}{2} I \omega^2$$

It can be seen by analysing the formula for angular momentum that there are two ways to lower the Angular momentum, by lowering either the radius, mass or rotational velocity. By minimising the radius, it is obvious that the angular momentum will be reduced by a squared factor, whereas if the angular velocity was reduced, the energy stored would be reduced by a squared term and the momentum would not be decreased by as much. When analysing the Kinetic energy equation, increasing the angular velocity will increase the

Kinetic energy quadratically and will only raise the Angular momentum by the power of 1.

The specifications state that the radius of the flywheel is only 10cm and the mass is 5kg. Therefore by using a lighter, smaller flywheel than earlier models, the design reduces gyroscopic forces, whilst maximising stored energy. Advances in materials technology have enabled this possibility.

The smaller radius coupled with very high angular velocities leads to a solution that has a low gyroscopic effect on the handling of the vehicle, but a high energy storage capacity. This design enabled the engineers to mount the axis horizontally, as it did not have noticeably adverse effects on handling of the vehicle.

2.1.4.2 Ricardo 'Kinergy' (electromechanical composite flywheel)

Ricardo, a European Consultancy has developed an efficient flywheel technology quite different from the standard designs. There is no mechanical coupling or any linkage to the casing. The flywheel is still manufactured from carbon fibre and operates in a vacuum sealed environment. The torque is transmitted directly through its containment wall using a magnetic gearing and coupling system (see Figure 6). This negates the need for a vacuum pump and seal replacements, often considered to be a significant issue for flywheel systems (Ricardo, 2011). The lack of a vacuum pump and vacuum management lowers the maintenance of the design considerably. Other sources have referred to the technology as an "electromechanical composite flywheel". The composite material is embedded with magnetic powder and the magnetic bearings store and transmit the energy. The system is rated to store 960kJ of energy and has been scaled to operate on a prototype bus, the 'Flybus'. The flywheel is coupled with a CVT in this design and operates as a parallel system. In simulation, the Flybus achieved roughly 30% improvements in fuel economy at an estimated cost of $2000 (Ricardo, 2011). They have pricing of smaller units at around $1500 with the CVT. The efficiency of energy storage is quoted at 99.9% for the flywheel. This system has the advantage of electric simplicity of power transmission and high storage and discharge rates. The technology can be used as a very effective battery. "Flywheels are in effect a complementary technology to batteries but in an electric hybrid powertrain they offer direct competition to ultra-capacitors, out-scoring them in terms of cost, volume, weight efficiency and ease of manufacture"

(Ricardo, 2009). They make particular reference to the potential use for delivery vehicles or other commercial applications.

When discussing the potential for a retrofit, Ricardo state that a CVT or a hydraulic displacement unit would be recommended for ease of power delivery. Recent prototypes have utilised the same Torotrak transmission as the Flybrid Flywheel system (Ricardo, 2009). They have also developed units that connect onto the rear of the differential of busses using existing PTOs, which would be ideal for retrofits.

Figure 6 – Ricardo Flywheel Technology (Ricardo, 2011)

2.1.4.3 GKN Hybrid Power Gyrodrive

The GKN Hybrid Power Gyrodrive uses an electric flywheel to store energy. It is another continuation of theories developed by F1. As with most modern designs, the flywheel is manufactured from Carbon-Fibre. It is capable of 36000rpm and operates in a vacuum sealed enclosure. In 2014, it was approved for retrofit fitment to 500 buses (Davies, 2014). The system has the capability to store 1.2MJ of energy. This system does not directly use the kinetic energy stored in the flywheel and instead uses the flywheel more as a battery and converts the kinetic energy to electrical energy and then back to kinetic energy at the wheel using an electric motor. This would lead one to believe that there are conversion inefficiencies that other designs seem to have overcome. Despite this, GKN claims to have achieved a 20% fuel efficiency improvement on buses running a city route.

2.1.5 Summary

Flywheel technologies have improved drastically in the last decade, and the improvements are obvious in the technologies that have been examined. The Flybrid system appears to be safe, compact, light and high in energy density for its size. On a passenger vehicle, fuel savings of 25% were able to be achieved. On a heavier vehicle, the Kinergy technology was able to obtain a 30% increase in fuel economy. The cost of the system was given as $2000. The Kinergy design was able to deliver similar economy figures and costs. The smaller $1500 system is cheap considering the inclusion of a CVT transmission. In terms of mass, it appears that a suitable system would be less than 100kg, which is not substantial given the fact that the vehicles this system is being designed for will likely have a heavy load, making the added weight miniscule in the overall scheme of things.

The typical advantages of flywheel usage are the minimal number of system components, low cost, low weight, high power density and endurance (G. K. GANGWAR1, 2013). The review has confirmed that modern applications are no different. The performance also does not degrade significantly over a lifetime of use, as shown by the 250000km theoretical life of the Flybrid system.

Safety concerns have always been a barrier to implementation of systems. Clegg (1996), outlines some of the safety concern for flywheel based systems. There is the over speed issue, where the rotational forces will cause the diameter of the flywheel to expand due to deformation, examples of damage to the casing, and the worst possible scenario of flywheel disintegration. This occurs if the tensile strength of the material is exceeded. However, this appears to have been addressed in the technologies considered in this review. Ricardo (2009) goes as far as to say that a safety factor of 12 is incorporated into their design. The CVT was originally an obvious cost liability, but Torotrak have trialled the cheaper manual transmission and believe it has potential and Ricardo have lowered the cost of their CVTs considerably in recent times.

From the stance of a retrofit, it can be seen that the flywheel systems are generally attached in a parallel configuration, which is quite useful for a retrofit as major changes to the original driveline are not required. Systems in the past have been attached to the output shaft or the differential, both of which are easily accessible and have room for additional systems on modern commercial vehicles. The system storage capacities presented, even in the case of the limited capacity Formula 1 model are adequate for the light commercial application considered. 400kJ is larger than the 360kJ estimated to be required.

2.2Hydraulic

2.2.1 Background and History

Hydraulic accumulators store energy by compressing a fluid with a hydraulic fluid. Midgley and Cebon (2012) describe the working fluid as being a compressed liquid or gas. The fluid is separated from the gas by a bladder, a piston or a diaphragm. The gas is compressed when fluid is pumped into one side of the accumulator. Two hydraulic accumulators are used: one at high pressure and another at low pressure. The low pressure accumulator has fluid pumped from it to the high pressure accumulator. In order to utilise the stored energy, the pump is used as a motor, transferring the fluid back from the high pressure accumulator to the low pressure accumulator to produce torque. There are many different configurations that have been developed and tested, although they all follow the same basic principle. The technology has been well researched and in recent times, the Ford Motor Company has collaborated with the Environmental Protection Agency (EPA) to build a Hydraulic Hybrid Prototype. From this research came Hydraulic Launch Assist (HLA), developed by Eaton Corporation.

As outlined in section 1.3, there are two different drive-line configurations: Parallel and Series. Both have been shown to give superior performance, improved fuel economy and reduced emissions. The parallel system leaves the original drive-line of the vehicle intact, often allowing the vehicle to operate normally when the hydraulic system is disengaged. It is integrated into either the driveshaft or the differential (EPA, 2015). Weight and size are often reduced as there are less major components (Clegg).

A series hybrid replaces the whole drive-line and leaves no mechanical connection between the engine and the wheels. The hydraulic pressure of the system alone is used to deliver power to the wheels. This configuration holds several advantages for the engine. It can be run at peak efficiency, as there is no direct gear ratio between the engine and the wheels (EPA, 2010). The engine can also be shut off when not needed. This configuration has been shown to give significant improvements in fuel economy.

2.2.1 Basic Technical Analysis

The basic theory of operation will be discussed. The stored energy in an accumulator can be calculated using:

$$E_{acc} = \left[P_{comp} \frac{1 - r_v^{1-\gamma}}{\gamma - 1} - P_{atm}(r_v - 1) \right] v_{comp}$$

$P_{comp} = Pressure\ of\ gas\ in\ compressed\ state$

$r_v = Volumetric\ compression\ ratio$

$\gamma = Adiabatic\ index$

$v_{comp} = Volume\ of\ gas\ in\ compressed\ state$

(Midgley & Cebon, 2012)

This formula assumes adiabatic compression of the gas.

2.2.2 Accumulator Types

There are 3 main accumulator types, each with associated characteristics.

<u>Bladder Accumulator</u>

A bladder accumulator uses a gas filled (often nitrogen) bladder fitted inside a steel pressure vessel. A bladder accumulator is used when high power output is a design requirement (DTA, 2014).

<u>Diaphragm Accumulator</u>

Diaphragm accumulators are useful if the fluid storage capacity is low in the given application. They utilise a rubber diaphragm to separate a fluid and a gas. They are quite similar in functionality to bladder varieties, but can handle higher gas compression ratios (DTA, 2014).

<u>Piston Accumulator</u>

A Piston Accumulator uses a piston to separate the gas from the hydraulic fluid. These systems can handle very high compression ratios and very high flow rates. The disadvantage of this system is the frictional losses that affect the reaction speed of the system (DTA, 2014).

2.2.3 Applications

2.2.3.1 Hydraulic Launch Assist (HLA)

The Hydraulic Launch Assist was a system developed by Eaton Corporation. It is a parallel hybrid system that was commonly fitted to refuse trucks. They claim in this application to be able to capture up to 70% of available kinetic energy. This is used to improve fuel economy by 20-30% and to improve acceleration by up to 20% (Eaton Corporation, 2007). They also claim that the brake life can be improved by up to 4 times. They quote a payback period for large trucks to be within 3 years (Eaton, 2009). This particular system is suitable only for refuse trucks as it is inactive over 40km/h, and the brochure states that the system must be brought to a complete stop to reactivate. A design such as this wouldn't be suitable for faster moving vehicles with less frequent stops. The system weighs 566kg. Interestingly, the Eaton HLA program was discontinued at the end of 2013, stating that hybrids are not serious contenders against the growing usage of CNG (Lockridge, 2013).

Ford modified and included the HLA system on the 2002 F350 concept vehicle. It was reported by Ford to improve fuel economy by 25-35% in an urban environment. Nitrogen gas is used as the working fluid, and it can be pressurised to 5000psi (Arabe, 2002). The system has a specified storage capacity of 380kJ. When the accumulator is at full capacity, it can accelerate the truck to 40kmh without the engine being active. The entire system weighs 204kg and adds $2000 to the cost of the vehicle. Ford made the statement that hydraulics were more suitable to this large vehicle than electrical alternatives due to the requirement of quick energy storage and release of large amounts of energy (Arabe, 2002).

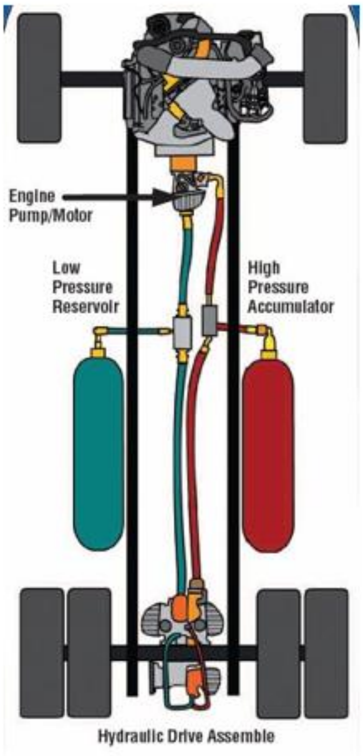

Figure 7 – Eaton's Hydraulic Drivetrain design (Abuelsamid, 2007)

2.2.3.2 UPS Trucks

In 2006, the EPA, working with Eaton and UPS as well as several other partners, developed a hybrid UPS delivery truck. It claimed to have 60-70% better fuel economy and 40% lower greenhouse gas emissions than its standard counterpart (Rodriguez, 2006). The repayment period was estimated at 3 years and offered a $50000 saving over the lifetime of the vehicle.

In terms of design, the truck utilised a series hydraulic configuration. When the accelerator is pressed: the engine drives the hydraulic pump, which creates pressure and in turn drives the hydraulic motor on the rear axle (Abuelsamid, 2007). Once work has been extracted, the pressurised fluid flows back to the reservoir. When the accelerator is lifted, the opposite occurs: The rear axle drives a pump which pressurises the accumulator. The configuration can be seen in Figure 7. The resistance of the pump feels like heavy engine braking. Once the accumulator reaches a certain threshold, the engine can also cut off in traffic or at idle.

2.2.3.3 Other EPA developments

From the early 1990's, the Environmental Protection Agency has researched and published findings on relevant regenerative braking technologies. As seen in other sections, they worked with Eaton to develop both the HLA system and the UPS trucks. They developed an F550 concept truck to show the possibility of a parallel HHV (Hydraulic Hybrid Vehicle) retrofit, and a full series HHV on a Ford Expedition SUV. The SUV was modified to show the potential fuel saving benefits of a full hydraulic system coupled with a diesel engine. It had an initial added cost of $600, and had and was able to capture 55% of available braking energy. They calculated roughly a $4000 - $6000 saving over the lifetime of the vehicle (EPA, 2010).

2.2.3.4 PermoDrive

Following the promising success hydraulic technology had overseas, an Australian company trialled a solution. Permodrive is a company that pioneered the concept of hydraulic regenerative braking on commercial vehicles in Australia. It was marketed as RDS (Regenerative Drive System) and developed in 2000. It was a parallel hybrid system, which made it easy to implement. It was aimed at vehicles with a GVM of less than 5 tonnes. The technology used was not revolutionary, but was complemented by sophisticated control software that made it effective. The accumulator used nitrogen gas as the working fluid (RDS Technologies Pty Ltd, 2015). Generally the pump is connected to the driveline between the gearbox and the differential (see Figure 8). The system was universal apart from the need to match the pump size and gear ratio based on the mass of the vehicle and the drive cycle concerned. There was capability for fitment to any rear wheel drive vehicle and the interface was passive with the vehicle control system. They state that the system concerned is 95% lighter than an electrical system of the same capacity. It has been proven to give a fuel economy improvement of up to 25%.

In 2012, Permodrive went into liquidation and the company was delisted. The CEO blamed the economic state of the country at that point in time and the lack of funds available.

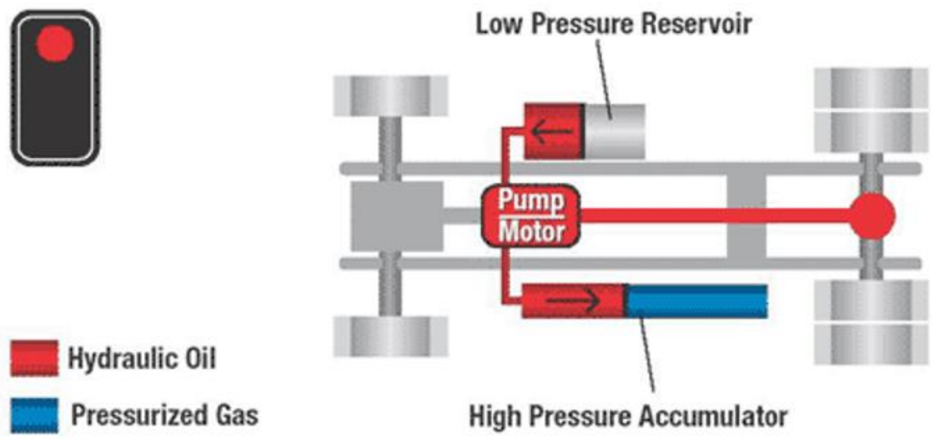

Figure 8 – Permodrive parallel driveline arrangement – braking (RDS Technologies Pty Ltd, 2015)

2.2.4 Summary

After research, it appears that hydraulic systems are mainly suitable for large applications, such as the Eaton Refuse Trucks. The EPA Progress Report of 2004 lists many advantages of a full hydraulic drivetrain:

- The shutting off of the engine when not needed provides significant fuel economy improvement
- There are an infinite number of control systems, meaning there is plenty of room for development
- It has been shown that it is possible to reuse up to 80% of braking energy from EPA data
- The engine can often be downsized die to the peak efficiency operation

The EPA predicts that while that initial application may only be for medium duty vehicles with a high frequency of stop and go (at the time of writing), eventually performance gains without fuel economy trade-offs may prove to be motivation for the small car industry (Jeff Alson, 2004).

On a large commercial vehicle it was shown that fuel economy can be improved by 25-35% in drive cycles with frequent stop-start driving. The Permodrive system also specified an improvement of 30%. The storage capacity of analysed systems was adequate for the considered application. The mass of the system was a definite issue. As the HLA system used was a series hybrid, many modifications were made to the existing driveline, which resulted in a 204kg mass, which is more than twice the weight of competing flywheel systems.

Matheson (2005) performed an analysis into hydraulic hybrid drives and the advantages and problems they present. Several issues were identified:

- The weight of the system is often substantial due to the weight of hydraulic accumulators and other components and the drive is therefore only suited to larger vehicles at this point in time. The weight penalty must not be too great, yet the system must have enough capacity to assist and capture the maximum energy possible to be effective. This was found to be the case as well in the literature reviewed.
- The control strategy is difficult to implement in such a way that the driveability is unimpeded. This will be of paramount importance if the systems are to be widely accepted by the general public. This has been solved somewhat by newer technology since the publishing of this article. Research has been put into control strategies, which has enabled them to be much more effective.
- Leakage from the system that will cause inefficiencies and eventually failure.

The author of Autoblog, Abuelsamid (2007), lists several disadvantages to the system. A light hydraulic setup can only be used to propel the vehicle small distances, due to the limitation of energy storage of a hydraulic system. Also, pertinent to the objectives of this review, the designed system would not be suitable for smaller vehicles due to its size and weight. The potential to design a smaller system still exists and requires further research.

From the perspective of a retrofit, it appears that a parallel system could be feasible, but a series system would add too much weight and require too much modification to be taken on board as a serious solution by the general public.

2.3Pneumatic

A fairly recent topic of interest has been that of pneumatic hybrids. The definition is more general than other technologies due to the flexibility of implementation, but basically defines a system that stores energy using compressed air in a vessel. There has been limited progress when using an electrically powered compressor to store energy in a high pressure vessel, obviously due to the addition of more energy conversion processes, which obviously decreases the efficiency of the process. But there are other systems that have potential as a retrofit and they will be explored below.

2.3.1 Engine as a compressor research

A USQ student of a previous year produced a thesis on the use of an Internal Combustion Engine to recover the Kinetic Energy of a vehicle during braking. The KE would be converted to Potential Energy in the form of a compressed gas in a high pressure vessel. When accelerating, the energy would be converted back to mechanical energy.

When braking, the engine would function as a reciprocating compressor to store energy. This energy could either be used to supercharge the engine or to power the engine in a pneumatic mode. This potential design had the advantage of theoretical minimal modification to the existing ICE system.

It was found that in order to be practical, certain processes had to be removed. The combustion process plays no part in the compression cycle and adds no value. Effectively, for the ICE to be a suitable candidate as a compressor, it must be able to operate in a 2 stroke configuration (Kruger, 2014). It appeared that much more modification to the existing vehicle was required than was initially anticipated.

2.3.2 RegenEBD

A practical design that stems from the 'engine as a compressor' theory is RegenEBD. An EBD is an Engine Braking Device. Most people would be familiar with the functionality of a compression release brake, or a 'Jake Brake' as it is sometimes referred to as. This is a braking system used on large displacement diesel engines. When activated, it opens exhaust valves in the cylinders after the compression cycle, releasing the compressed air in the cylinders and causing a braking effect as the stored pneumatic energy is not returned to the crankshaft (see Figure 9) (Wikipedia, 2015). The technology presented by the Brunel University of London, effectively stores the energy that is otherwise just pumped out of the exhaust system. The functionality is slightly different in the way that is alters the timing of the intake valves as opposed to the exhaust valves(Yan Zhang, 2012). It keeps the intake valves from fully closing, therefore becoming a two stroke compressor where the air is compressed on the upstroke of every crank revolution (Cho-Yu Lee, 2011). The fitment involves retrofitting VVEB (Variable Valve Exhaust Brake) and fitting a sandwich block between the cylinder head and the intake manifold. The simulations were carried out by Lee Cho-Yu and published in an SAE paper. A 6.2% fuel saving was found on the London Bus Route, performance was improved and idling periods were eliminated. A prototype was later developed by the Brunel University and results obtained verify the calculations. The system obtained full pressure in 5 seconds of braking and pressurised an air tank to 10 bar (1000kPa). The system was predicted to be capable of absorbing 27% of the braking energy (Yan Zhang, 2012). It was tested on a city bus in London and obtained a 5-10% improvement in fuel economy. The university states that there is potential for a retrofit and it is a matter of calculating whether or not it would be suitable for smaller vehicles with a smaller displacement.

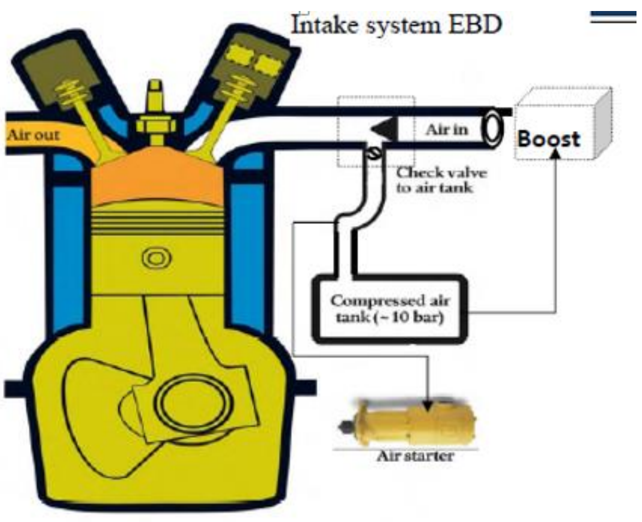

Figure 9 – RegenEBD Configuration (Yan Zhang, 2012)

2.3.3 Summary

There are several advantages presented by this technology. There is no change to the existing driveline or the engine block itself. It has the potential to act as a turbocharger or supercharger or to use the existing engine as a pneumatic compressor. The fitment also appears to be relatively straightforward. The cost was unable to be found, but it is to be assumed that something that requires significant modification to the intake manifold will have significant expense due to the difficulty in mass producing something that must be modified for the multitude of manifold varieties on available vehicles.

The improvement of only 5-10% is minimal compared to other technologies, and smaller displacement engines will obviously produce less air, often at a smaller pressure. This will result in less power being produced by the system. There may be difficulty in reliability and serviceability due to the location of the modifications on such a vital component of the engine.

Based on the review, a retrofit would be possible, yet difficult over multiple vehicle platforms. The student from 2014 who attempted this found the implementation of the design to be quite difficult.

2.4Comparison of Technologies

It is difficult to locate studies that have been done comparatively on all technologies on a common platform. There are a multitude of studies, both empirical and analytical, on individual technologies, although these are very difficult to compare quantitatively as there are so many inconsistent variables. The studies have been performed on different vehicle platforms, with different system capacities and different vehicle characteristics. Two noteworthy comparisons are reviewed below in order to gain a better understanding of which technology would be most suitable and to add transparency to personal opinions formed in the literature review.

2.4.1 Midgley and Cebon (2012)

Midgley and Cebon conducted a holistic study into the available regenerative braking technologies and their characteristics. The study considered hydraulic solutions such as those presented by Eaton and Permodrive, Flywheel solutions such as Kinergy and Flybrid, as well as pneumatic solutions available at the time. The graph in Figure 24 was published by Midgley and Cebon (2012) and shows the results of their study into the different means of energy storage. The multiple points plotted represent multiple systems coupled with different storage devices. The dotted lines are known as “selection lines” and represent the best point in the technology space for a braking event of a specific force. This study finds that the kinetic systems would be heavier than the hydraulic systems per unit power. It can be seen that the highest specific power was given by the hydraulic system. The flywheel systems had considerably higher specific energies, which is in line with other research. The pneumatic system has quite low specific energies and powers.

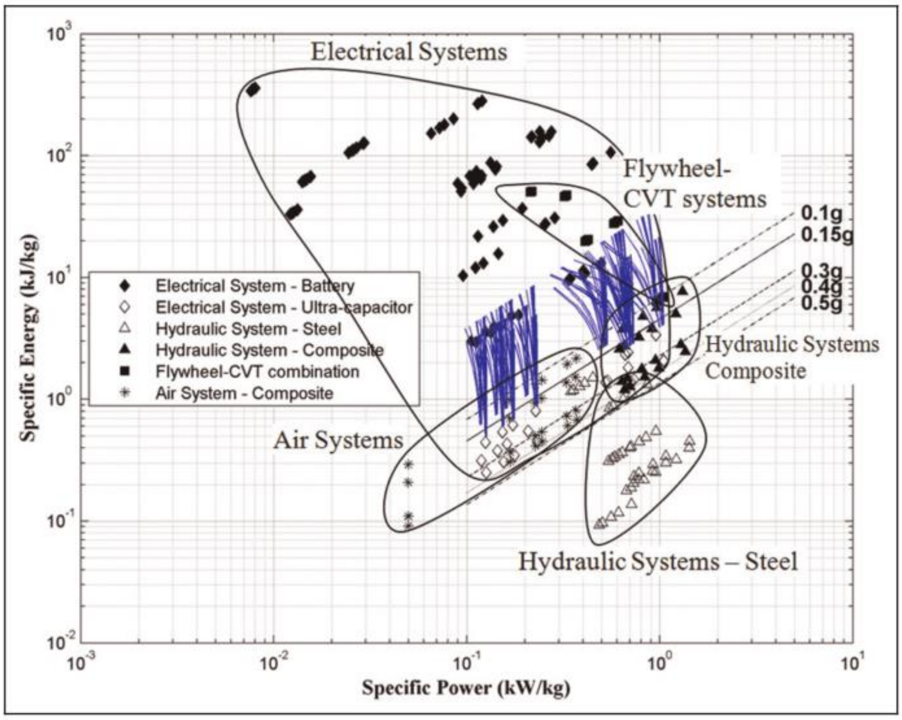

Figure 10 – Specific Power vs. Specific energy for energy storage systems (Midgley & Cebon, 2012)

They find many advantages with hydraulic systems as they have been used before in delivery vehicles and have shown notable success. The wear on components is low and oil leaks are rare with modern systems. In terms of fitment, hydraulic units are able to be manufactured as in-wheel units, meaning no drivetrain alteration. At the time of writing, Midgley and Cebon (2012) found that the available hydraulic solutions were on average 16% lighter than equivalent flywheel units. Variable displacement pumps are best for controllability, but the cheaper and smaller fixed displacement pumps are able to be used with performance trade-offs. The flywheel system has shown success in small scale designs such as the F1 vehicles, whereas the hydraulic systems have only shown potential on trucks. The containment adds weight, but is necessary for safety.

The methodology was critiqued, as the results did not quite line up with findings from practical applications. It was assumed that “the existing commercial devices can be scaled up or down while maintaining the same specific and volumetric energies and specific and volumetric powers" Midgley and Cebon (2012). This may have altered the results as the practical applications of hydraulic systems would have to be scaled down quite considerably for their graph, yet this may not be entirely true in practicality. No gearing was considered for the hydraulic application either, whereas other research suggests that this may be necessary.

2.4.2 Dingel, Ross PhD, Trivic, Cavina, and Rioli (2011) (SAE Paper)

Dingel et al. (2011) performed a study into the impact of different hybridisation techniques on fuel economy over different driving cycles. The standard passenger vehicle compared with three different hybridisation techniques was considered, based upon a 'standard' vehicle platform. The systems were of equal storage capacity and simulations were performed using the VeLoDyn (Vehicle Longitudinal Dynamics Simulation) software package, which runs in a Simulink environment. It offers a straightforward simulation of longitudinal vehicle dynamics with considerations on the driveline. The paper was published in 2011 and considered what the authors believed to be the most applicable hybridisation techniques, which have been covered in the previous sections of literature review.

The vehicle platform used was a mid-sized vehicle with a front wheel drive powertrain. Note that this is not ideal for the aims of this review, but should provide a baseline for comparison. The engine considered is a 1.4 litre turbo petrol engine and the mass of the vehicle was 1470kg (Dingel et al., 2011). VeLoDyn considers the engine, transmission, vehicle and control system models in one environment. The three technologies considered are Flywheel, Hydraulic and Electronic.

The flywheel system was theoretically coupled to the final drive shafts of the vehicle, making it a parallel hybrid configuration. It is coupled using a frictional clutch. The simulation considered the functionality of energy exchange between the engine and the flywheel for optimal power management and the functionality of switching the engine off during idling periods. The braking only occurs on the front end of the vehicle, so in hard braking scenarios some of the braking force will be given by the rear of the vehicle and will be lost as heat energy still. The transmission used was a CVT. The flywheel system adds 40kg of mass to the vehicle.

The Hydraulic Hybrid system considered was also a parallel configuration coupled with the driveshaft. The pump used was an axial piston variable displacement hydraulic pump. The functionality is the same as has been previously covered. The overall mass of the system was 135kg.

A Hybrid Control Unit was used in order to control the power split between the ICE and the hybrid for each model. The usable system energy storage capacity was 625kJ for both the hydraulic and flywheel systems.

It can be seen from the fuel economy improvement results (Figure 11) that the Flywheel based system offered the greatest fuel economy improvement. This was due to the high system efficiency and high power capability. The hydraulic system was the second more effective, although it also had the greatest additional weight from the system. The author states that greater fuel economy gains could be achieved by improving the recuperative potential of the systems. Using the simulation on the current drive cycles led to some energy still being absorbed by the mechanical brakes in order to stop in the required time.

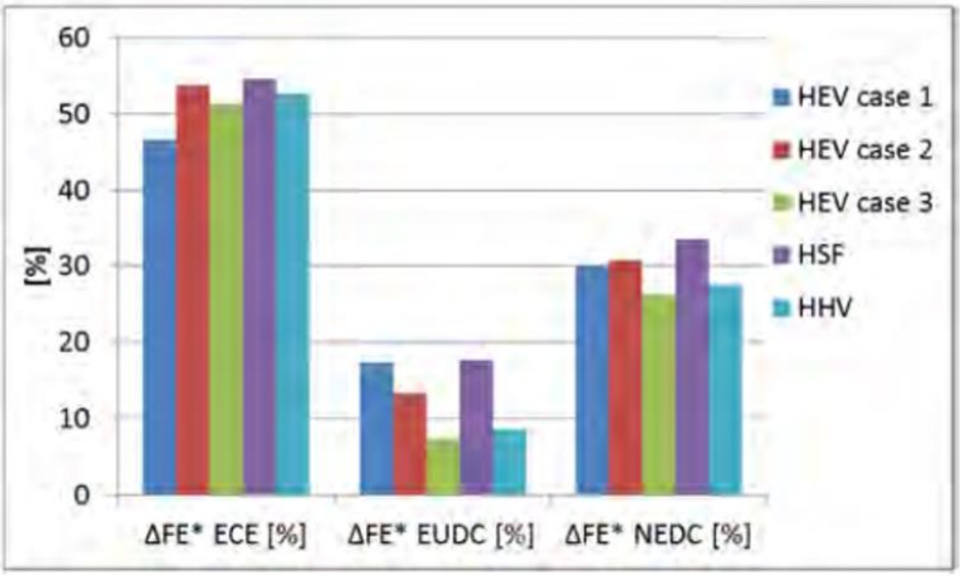

Figure 38. Fuel Economy improvement [%] during NEDC cycle with respect to the base vehicle

Figure 11 - (Dingel et al., 2011)

2.5Evaluation of Technologies

The technologies were assessed on their suitability by comparing their individual characteristics in terms of the criteria established.

Energy Storage

A light commercial vehicle requires a high specific power and specific energy in order to be practical. The study by Midgley and Cebon (2012), found that the specific energy of the flywheel systems is the highest of the systems considered. The hydraulic system trend has the highest specific power. Both technologies have quite similar characteristics, as can be seen in Figure 7. Pneumatic systems have substantially lower specific energy and power. It has been shown that both the hydraulic systems and the flywheel systems are able to store the required amount of energy for the light commercial application considered.

Feasibility

The system must be able to harness sufficient energy to offset the disadvantage of carrying the additional weight. The sources reviewed disagreed on the weight comparison of the hydraulic and flywheel technologies. Midgley and Cebon (2012) found hydraulic systems to be lighter in general, whereas Dingel et al. (2011) found flywheel systems to be lighter. The practical applications reviewed also found flywheel applications to be lighter in general than the hydraulic counterparts.

For the application considered it is a requirement that the designs are reliable and have minimal components. This makes for a cost effective and quick installation, which is required for vehicles that are a source of income. System masses of less than 100kg have been seen for flywheels, which makes them attractive as they do not increase the inertia of the vehicle considerably and will not hinder the amount of mass they are able to legally carry. Mass specifications for parallel hydraulic systems were difficult to locate, but series systems were found to weigh around 200kg.

In terms of reliability, all three technologies appear to fare quite well. The flywheel technology is advertised to have a service life of up to 250 000km and hydraulic and pneumatic technology has shown excellent reliability in heavy vehicle applications overseas. The flywheel technology has the advantage of performance not degrading significantly over the lifetime of its use.

Cost-effectiveness

The study by Dingel et al. (2011) found flywheels to offer the greatest fuel economy improvements. Flywheels have led to fuel economy improvements of 25% in prototypes. This is roughly the same figure obtained by parallel hydraulic systems. It does appear that in order to fully utilise the potential of a hydraulic system, a series configuration must be used, these have shown substantially higher improvements. Pneumatic systems have been shown to give up to a 10% increase in fuel economy, but only on very heavy, large displacement vehicles. The fuel economy improvements were minimal in comparison to other technologies, mostly due to the compression characteristics of engines.

The small flywheel systems have been advertised as costing a minimum of $1500. Hydraulic systems overseas have been estimated to cost a minimum of $2000 by the EPA. The cost of the pneumatic solution is unknown.

Ease of Implementation

It was decided that for ease of retrofitting, it would be favourable to utilise a system that can be installed as a parallel hybrid. For hydraulic systems, the most effective systems were shown to be series configurations, these are more expensive and require extensive modification, which is not the most feasible option for a retrofit. These solutions may be effective if they are implemented from the factory on new vehicles, but making use of the existing driveline is considered to be of upmost importance for a retrofit.

Pneumatic systems require too much modification to the engines of vehicles to be utilised effectively. The installation of sandwich plates and other components on the intake manifold may not be that intensive as far as labour is concerned, but the manufacturing of multiple fitment types for many different vehicles will be cost intensive.

Chosen Technology

After reviewing the available technologies and their applications, it was decided to proceed to the design stage with a flywheel technology. This decision was based on the factors discussed above. Flywheel technology has shown huge potential in recent times, but has not been utilised in the light commercial industry as of yet. They are particularly suitable as they can be utilised effectively as a parallel system, converting kinetic energy to rotational energy. They can be manufactured to be lightweight and compact, making them suitable for a retrofit. The potential fuel economy improvements, long service life and the small design footprint shows definite suitability to the light commercial industry.

3 Design

For the design phase, the flywheel system was broken down into individual areas considered to be integral to the functionality. A large number of components are interdependent and the arrangement of content reflects this. Appropriate further literature review was completed throughout to give a more suitable and adept solution.

3.1 Theory of Operation

The end concept is designed to be simplistic, cheap and as efficient as possible. As such it was decided that the location for harnessing power and the location for transmitting power would be the same. The theory of operation follows and would be managed by a vehicle control system. The system will have three modes:

1. Regenerative Braking: Whilst in this mode, the system will capture vehicle energy and slow down the vehicle. The transmission will be set to match the ratio between the flywheel and the axle. After the ratios have been matched, the flywheel will be connected to the axle by a clutch. When the connection has been established, the gearbox ratio will be varied to lower ratios and this will facilitate energy transfer to the flywheel system. Mechanical brakes will be on standby should more braking energy be required than can be feasibly delivered by the rear axle of the car without losing traction.
2. Acceleration: Whilst in this mode, the system will transfer energy back to the driveshaft to accelerate the vehicle, aiding the engine. Similar to when in regenerative braking mode, the gearbox ratio is set to match the speeds of the flywheel and the axle. The connection is then established and the ratio is varied from low to high in order to transfer energy from the flywheel to the driveshaft. Once the lower limit of rotational velocity is reached for the flywheel or the upper limit of velocity for the vehicle, the flywheel is disconnected from the system. When possible the engine should be disconnected from the driveline when the flywheel system is accelerating. By analysing Figure 12, a representative fuel efficiency map for a diesel engine, it can be seen that the average engine is quite inefficient at low loads. When BMEP (Brake Mean Effective Pressure) is multiplied by RPM, power is the resultant unit. This means that for any point on the graph represents a

particular power output and the contours represent the approximate fuel usage. This would mean that if the engine is still active under light load when the flywheel system is accelerating the vehicle, it would be running quite inefficiently and would work to counteract the net energy gained by utilising the flywheel. If the vehicle were to be accelerating itself without the aid of the flywheel it would be operating at a higher efficiency as shown by the curves. This should be taken into account for control system considerations.

3. Neutral: In this mode, the flywheel is disconnected from the axle by the clutch. This mode is necessary in the event of a failure or when either the flywheel or the axle reach speeds outside of the limit of the transmission or the system capacity is at a maximum. This would also be necessary if vibration or recurring conditions were being imparted to the flywheel that could lead to a failure.

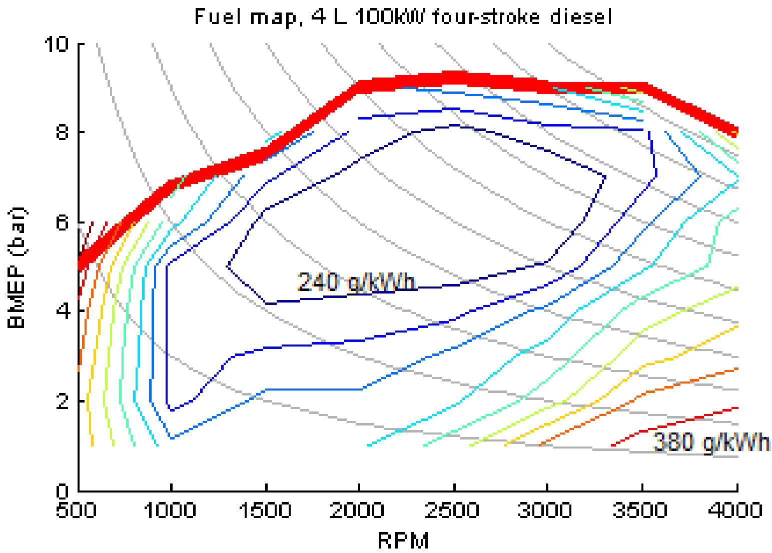

Figure 12 – Representative Fuel Efficiency Map (Griffiths, 2015)

3.2Driveline

3.2.1 Configuration

As a parallel driveline arrangement has been selected, the CVT will have to be supplementary to the existing driveline. There are two basic designs proposed by Dhand and Pullen (2013).

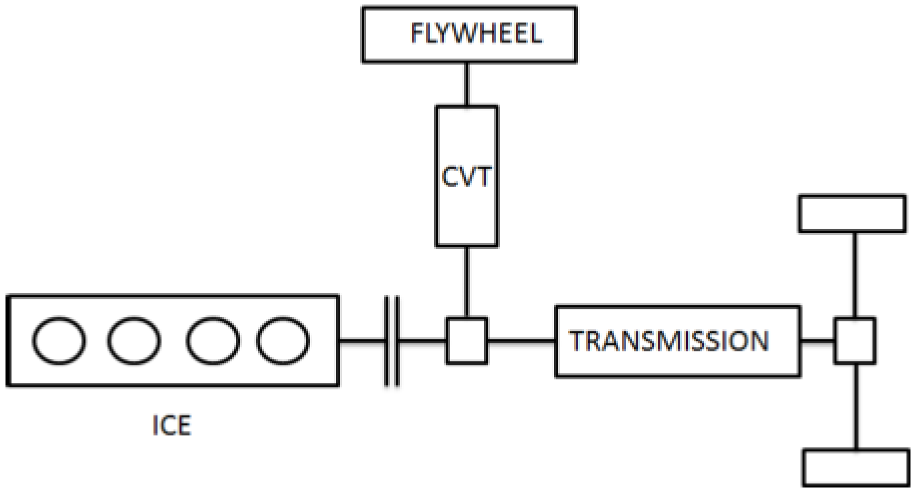

Figure 13 – Driveline configuration 1

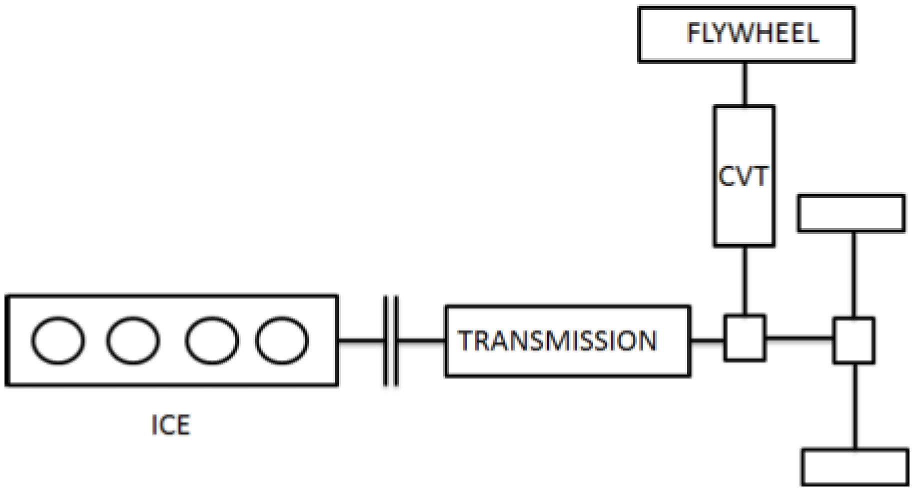

Figure 14 – Driveline Configuration 2

Figure 13 shows the flywheel coupled with the CVT being inserted before the existing transmission, and Figure 14 shows it being inserted after. Both configurations allow the original driveline to remain the same. Figure 14 would be easiest to integrate on an existing system with minimal controls. If the first configuration were utilised, the ratios of the CVT would vary considerably during braking events depending on the particular transmission a vehicle has. It would have benefits as the crankshaft output will run at much higher speeds (closer to that of the flywheel) than a driveshaft. Unfortunately it is very hard to develop a practical design from a retrofit standpoint as any modification would require substantial modification of the gearbox or gearbox location. The control system (for an automatic), or the necessary procedure for braking (in a manual) would be quite difficult to implement. As the gear ratios of the differential are the fairly similar between automatics and manuals, the difference in retrofits should be minimal. The configuration presented in Figure 14 will require a larger overall ratio of gearing in order to reach appropriate flywheel speeds, but will be both easier to implement and easier to control as well.

3.2.2 Braking Control

2014) lists three primary considerations to make in the development of a regenerative braking system.

1. Energy Recovery – The system must recover sufficient energy to be financially and technically viable.
2. Vehicle stability must be considered and must still remain adequate.
3. The system must give the driver a repeatable and predictable outcome when braking demand is required.

Regulations exist in Australia that govern the design of auxiliary braking systems. These will need to be adhered to in order to produce a feasible design.

Whilst the control systems are outside of the scope of this dissertation, it is necessary to research and understand the conditions that the system must operate under in order to have the greatest potential for energy recovery. (Folkson, 2014) believes that the most effective way to do this is with sequentially phased systems. This is achieved by bringing an axle with regenerative systems on it close to the traction saturation point. Then in order to maintain braking performance, if more braking force is required, the braking force should be reduced and put onto the other axle (or wheels). The article states that: “Energy recovery may be maximised at the expense of stability and careful attention must be paid to driver feel to deliver an acceptable level of performance, with respect to regulatory requirements as well as the end user experience” (2014). In order for braking functionality to remain the same, the sum of regenerative and friction braking must remain constant. In the case of lockup, many manufacturers employ the practice of disabling regenerative systems and purely using mechanical systems.

Mounting the flywheel system on the rear end of the vehicle to harness power from the rear wheels will obviously have an effect on the amount of energy that can be collected in a typical braking event, as the front brakes are the main braking force in any event. The general braking distribution is roughly 70/30 front to back. In the case of commercial vehicles, a brake proportioning valve (or electronic system in newer vehicles) controls this ratio and varies it in depending on the load on the rear of the vehicle. It is reasonable to assume that by operating the rear braking system at close to saturation point that much more energy could be harnessed than would otherwise normally be available, but it is not reasonable to assume that all of the energy, especially that in the case of the scenario of a declination could be dissipated without the aid of front brakes. Another efficiency factor was added to account for an optimistic rear distribution:

$$E_{design} = E_{avail} \times 0.8$$

$$E_{design} = 511.51kJ$$

3.2.3 Transmission Design Selection

The transmission is an important part of the design as it initiates and regulates the power transfer from the axle to the flywheel. It must meet several criteria:

- The ratio must be widely variable
- The ratio change must be controllable
- The rated power must match the output of the flywheel
- The transmission must be able to transmit power forwards as well as backwards
- The transmission must be as cheap and as simple as possible

The CVT is often the limiting factor on the upper limit of the flywheel speed, as very high ratios are required to charge it. This also limits the maximum energy that can be stored, as can be seen in previous equations. A major challenge in flywheel design is that as the vehicle speeds up, the flywheel must slow down. This inverse relationship leads to a very large ratio requirement. This allows power transfer. It is the opinion of Pullen and Dhand (2014), that a CVT is a necessity for an effective flywheel system.

The CVT design will determine the speed range for the flywheel. Efficiencies of each technology were found from two different authors on flywheel technologies for comparison. It can be seen that the authors both present figures similar in magnitude to one another.

Table 1 – Typical Transmission Comparison

Transmission Type	**Efficiency (Heath, 2007)**	**Efficiency (Bell, 2011)**
Manual	97%	90-95%
CVT – Rubber Belt	88%	90-95%
CVT – Steel Belt	90%	90-97%
CVT – Toroidal	93%	70-94%

The Belt and Toroidal systems were investigated to assess the gear ratios available from models that have been produced to date.

3.2.3.1 Belt Systems

The initial literature review briefly described the workings of a belt driven CVT. The system is of particular interest as it is very cheap in comparison to other technologies and it very easy to design for. There are two different types: the adjustable centre and fixed centre. The adjustable centre variety is spring loaded, ensuring that tension is kept on the belt at all times. The driven pulley is forced away from the drive pulley and this causes the belt to ride on a decreasing pitch diameter. The opposite occurs when the driven pulley is forced closer to the drive pulley and the pitch diameter increases again.

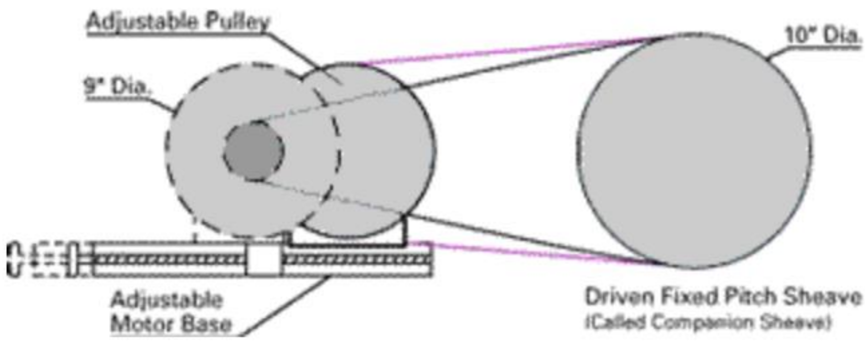

Figure 15 – Adjustable Centre Belt CVT (Speed Selector, 2010)

The fixed centre type works with two adjustable pulleys. The driven pulley is spring loaded, as with the fixed centre variety and the driver pulley pitch diameter is mechanically variable. When the driver pulley pitch diameter is varied, the spring in the driven pulley will then force the belt to a higher pitch as the flanges are forced together. When the reverse happens, the speed of the driven pulley will increase. Both pulleys are a fixed distance away from each other. This will be the most appropriate type for the given application as both the driveshaft and the flywheel shaft will be at set distances apart, which is a requirement. They are also more compact and can offer a higher ratio . These sets are designed to only transmit torque one way, but it will be assumed that torque can also be transmitted from the driven pulley to the driver pulley as the theory should be exactly the same. Hydraulic fluid is used to move pulley sheaves closer together or further apart to change the effective diameter of the pulleys.

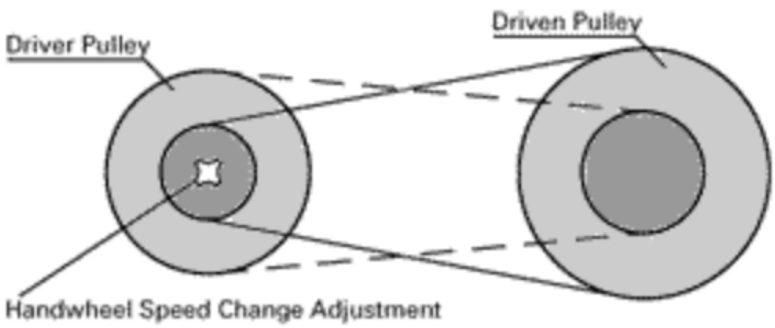

Figure 16 – Fixed Centre Belt CVT (Speed Selector, 2010)

A major drawback of the pulley systems is the efficiency and RPM operating range. A reduction box will be required before the flywheel side pulley in order to slow the rotational velocity down enough to be used with the variable diameter pulley system.

Metal Variable Belt (MVB) CVT

An MVB CVT works on the same principle as a variable belt CVT, using a metal belt instead of a rubber one. The belt is made up of transversely mounted steel plates that are held in place with steel bands. The bands ensure the belt remains together. The plates are only guided by the bands and torque is transmitted by compressing the plates, rather than relying on tension in the band (Automotive, 2013). The design does require transmission fluid in order to function effectively. Normal belts are quite limited by the friction between the surfaces and the strength of the belt. With the MVB CVT, ratios as high as 1:8 are possible and they are more durable than the rubber belt counterparts (Thoolen, 1993). The Nissan Xtronic CVT is a more recent development and shows that the transmissions are capable of transmitting torques typical of a passenger vehicle with ratios as high as 1:5.4. This transmission powers the new Nissan Altima, which Redbook (2015) quotes as having a peak of 183kW. This shows the true potential of the transmission type.

3.2.3.2 Ratio Requirements

To gain ballpark figures of the ratios required for the system, Matlab was used to graph vehicle speed against flywheel speed. See Appendix 8.2.4. The wheel size was assumed to be 195R14 and the flywheel was assumed to operate between 10000 and 20000 RPM. The diff ratio was assumed to be 1:4.3, which is common for Toyota live axle vehicles. The code calculated the required Ratio using:

$$\eta = \frac{\omega_{driveshaft}}{\omega_{flywheel}} \tag{10}$$

The assumption was made that the flywheel operates at the maximum RPM at the minimum vehicle design velocity and the minimum at the maximum design velocity (60km/h).

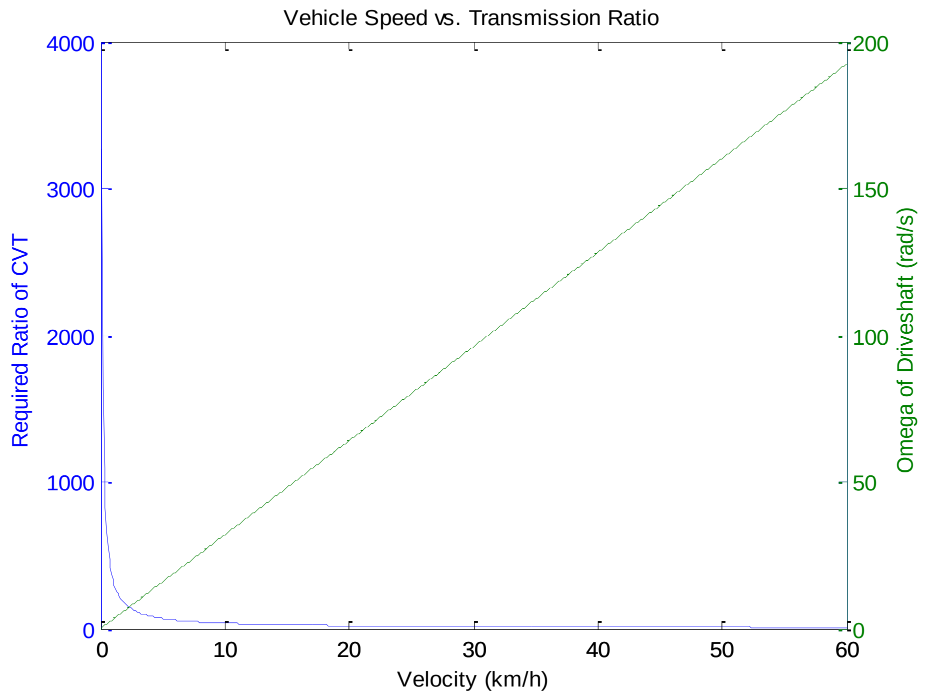

Figure 17 – Vehicle Speed vs. Transmission Ratio for a flywheel operating between 10000 and 20000 RPM

Figure 17 led to three design choices:

1. The regenerative braking cannot be expected to be functional down to 0km/h, as the required CVT ratio asymptotes very sharply. This will increase the cost of the system drastically for minimal gain.
2. There will be substantial gearing required if the tail shaft/differential are going to be used for Power Take Off (PTO). This has also been seen in many examples in the literature. The gearing ratios are going to be higher in this application than others such as the Flybrid system as the driveshaft RPMs are much higher, and therefore much closer to that of the flywheel due to the high operating speeds and the small wheel diameters. It was decided that gearing would be required both before the CVT and after it.

3. The flywheel choice will have to be a lower on the speed spectrum to avoid excessive gearing which would be required to match driveshaft speeds

The power equation can be rearranged to solve for torque, making the assumption that the power transfer remains constant for values of ω:

$$P = T \times \omega \tag{11}$$

Rearranging:

$$T = \frac{P}{\omega} \tag{12}$$

Equation (12) was used to solve for torque at design points. The power requirement to be transmitted to the flywheel will remain the same regardless of the shaft speed, so it can be seen that the higher the rotational velocity, the lower the torque. In order to reduce the torque that the CVT faces, gearing will be used before the CVT to reach higher velocities. Gearing will also be used after to ensure that the CVT is not running at an extremely high RPM. This will cause losses within the CVT. Figure 18 shows the system layout.

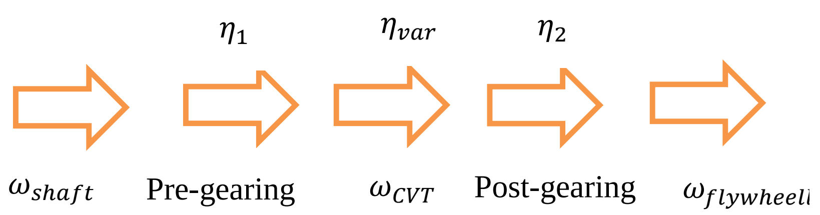

Figure 18 – Design layout of Transmission subsystem

An equation was developed to calculate the speed of the flywheel relative to the speed of the tail shaft using Figure 18:

$$\omega_{flywheel} = \omega_{shaft} \times \eta_1 \times \eta_{var} \times \eta_2 \quad (13)$$

A Matlab script was developed to show the CVT ratios required, depending on the η_1 and η_2 values. η_2 was fixed at a ratio of 1:2.5, as this slows the flywheel ω_{max} down to a reasonable speed for a CVT to work with and minimises the losses due to clutch drag. Note the velocities are taken from section 3.3.4.1. η_1 was then varied and the effects on the required ratio were plotted. See Figure 19 for the graph of the ratio change with respect to η_1. η_2 was kept at a value of 1:2.5 and η_1 was chosen to be 1:2. This leaves the CVT ratio to be from 1:1 to 1:8.5.

Figure 20 shows that this results in the CVT operating in velocities between 576 and 96 rad/s on the vehicle side and 408 and 817 rad/s on the flywheel side. The graph quite clearly shows that as the vehicle velocity is decreasing, the angular velocity of the flywheel is increasing. The torque values obtained will be discussed in later sections.

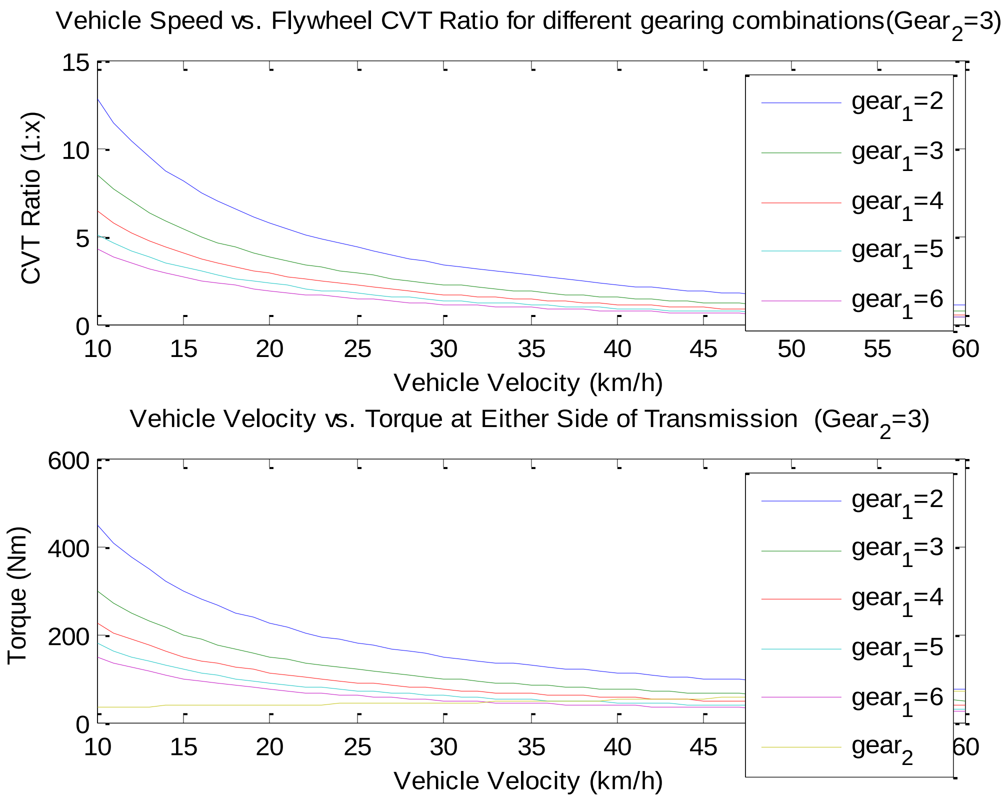

Figure 19 – Gearing and CVT graphs

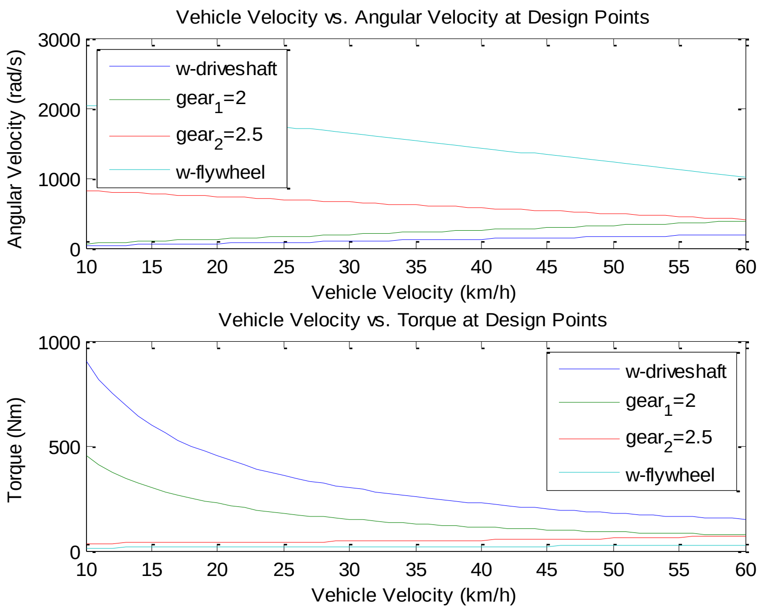

Figure 20 – Gearing and respective Angular Velocity / Torque

3.2.3.3 Gear Choices

There are many different forms of gearing that could be applicable for this application. Due to the high torque, high speed nature, belts are excluded from the analysis as a form of torque transmission. As torque is to be transmitted both forwards and backwards, worm gears have also been excluded. The literature suggests that the most common gears used for similar applications are spur gears. Helical gears are often much quieter and smoother than spur gears, however they are more costly to manufacture and induce shaft torques due to the angled teeth (Juvinall & Marshek, 2006). Other flywheel systems examined earlier were analysed for gearing technology. The Flybrid system by Torotrak that was reviewed in section 2.1.4.1 utilised a set of planetary gears for the large reduction ratio before the clutch. This type of gear train mounts two gears in such a way that the centre of one rotates around the centre of the other. They have concentric axes. The middle gear is referred to as the 'Sun' gear and the outer gear is referred to as the 'Planet' gear. These two are meshed together using a number of 'carriers'. This type of gearing has a very high torque to weight ratio. They are highly efficient and have been found to be in the order of 5% for ratios of up to 5:1 (MEADinfo, 2013). However, after initial analysis, it was revealed that other cheaper gear sets were just as suitable for the application and as a result these were given a priority for selection.

Spur gears were chosen for the first gear set on the differential side as they are cheap to manufacture and are simple and reliable at low velocities. Helical gears were selected for the high speed flywheel side as they are quieter and more efficient at high velocities. The torques imparted on shafts and other components should not be as much of a problem due to the lower drive torques encountered at the high speeds.

A Matlab code was developed to determine the feasibility of gear choices for the application through trial and error. The code is in appendix 8.2.4. It used rearranged formulae from Juvinall and Marshek (2006) and iterated to find more accurate values from the relevant tables. The gear choices were:

- SS2-62-J38 (KHK Spur Gears)
- SS2-114-J42 (KHK Spur Gears)
- KHG48RJ30 (KHK Helical Gears)
- KHG138LJ30 (KHK Helical Gears)

3.2.4 Power and Torque Requirements

The transmission must be capable of transferring the energy at the correct rate in order to fulfil its purpose. In order to calculate the power required by the design, both the driving cycles and hand calculations were used. The Matlab script developed for drive cycles calculated the peak negative energy change on a per second basis using the vehicle mass. From section 1.11.1, $P_{max} = 50.05\text{kW}$.

This calculation assumes no change in GPE and that the vehicle continues to decelerate at a constant rate for the entire braking event. When the transmission efficiency and braking bias factors are taken into account:

$$P_{max} = 50.05kW \times 0.8 \times 0.9$$

$$\mathrm{P_{max}} = 36.04\text{kW}$$

As the drive cycle does not account for gradient: for comparison, the E_{avail} was used for a sample braking event using the times taken in the braking cycle. This calculation takes into account GPE. The minimum time for a braking event was taken as 10 seconds as this was stipulated by the drive cycles as being the time taken for a complete stop from 60km/h.

$$P_{max} = \frac{511.11kJ}{10s} \times 0.8 \times 0.9 = 36.80kW$$

The power calculations were very close in magnitude. Using Equation (11) and 36kW, the torque values were calculated for design points. Figure 19 shows the torque at η_2 is quite small in comparison to η_1 with any of the gear ratios listed. The torque of η_1 varies greatly with the variation of the ratio.

3.2.5 Selection

The fixed centre pulley design is capable of a maximum ratio of 4.6:1 and operates effectively at speeds above 1750RPM (183rad/s). The particular design considered was able to transmit 45kW. Preliminary torque calculations revealed that a rubber belt system would not be sufficient. The amount of torque required to be transmitted was well in excess of the force that could be transmitted at the friction surface and was sometimes even larger than the tensile strength of the belt itself. MVB CVTs are relatively hard to come by commercially, meaning one would have to be sized for the purpose. It has been shown in the literature that they are capable of transmitting very high torques and are quite reliable if lubricant is used effectively. Thoolen (1993) finds that by numerical evaluation, they are the most compact system for transmitting large amount of torque. They have been shown to work for ratios up to 1:6. The toroidal transmission is used for most modern high speed composite flywheel designs found in the literature. Automotive (2013) lists nominal ratios of 1:4 with standard designs and was shown to have a ratio of 1:6 on the Flybrid system. They have been shown to be effective for high power applications upwards of 100kW. They are however a very complex transmission design and Automotive (2013) find them to be much more complicated and costly to produce than other types of CVTs.

The belt centre pulley design was eliminated as they are not particularly suitable for high torque applications. High torque and high ratio appear to be mutually exclusive qualities. The toroidal CVT would be suitable for the purpose, but based on the literature, would be more expensive and bulker than other types of transmission. Based on the design requirements, the MVB CVT design was chosen as it is cheaper to produce and has a more suitable design footprint based on the space available. Recent research has suggested that there is potential with respect to flywheel applications.

Despite the fact that the CVT is an integral part of the design, it was established early in the project that the intricate design would be outside of the scope of this dissertation, due to the complexity and time intensive modelling. The transmission will be idealised and rough efficiencies will be used for modelling.

3.2.6 Connection

There many options presented for attachment. Initial design yielded:

- Rear Axle
    - This design involves using a form of power take off from the driveshaft of the vehicle. This should be easily accessible from the underside. The connection would be in the form of a V-belt, chain or possibly a variable diameter pulley CVT.
    - Another option would be to attach a modified centre bearing. Centre bearings are present on the majority of old and modern commercial vehicles to support such long drive shafts from the gearbox to the differential. These could be modified to include a gear to transmit torque. The advantage of this would be that the centre bearing remains stationary all of the time when the vehicle is moving, negating the need for tensioners or other damping systems when the vehicle suspension changes the position of components.
- Front Axle
- Front Wheels
- Back Wheels (In-wheel unit)
    - In general, utilities in Australia still utilise drum brake systems. This rotating drum could be used as a large internal cog to transmit torque. The

outside drum will rotate and the inside drum will remain stationary as it always has. This design will not be applicable for front wheels and will require the functionality of an emergency brake on the shaft.

- Rear Differential
    - This design involves attaching the flywheel PTO to the rear differential by removing the backing plate. The pinion shaft and the backing plate would be modified to include a concentric cog, allowing power to be transmitted to the initial gearing stage of the device. This design will provide reliable traction to the rotating implement as it can be secured using the plate screws (not pictured in Figure 21). Issues may develop due to vibration of the flywheel and the forces that could potentially be imparted upon it due to the lack of dampening between the device and the ground the vehicle is driving on. There is generally some space available in front of the differential and otherwise space would be available behind the differential if the spare tyre were moved to another location. The driveshaft may have to be shortened depending on the space available and supplementary bearings will more likely be needed to counteract the new axial forces that the pinion shaft would support. Vehicles with (Independent Rear Suspension (IRS) would be more suited to this design as the flywheel system would still take advantage of the vehicles dampening from shock absorbers as the axle would not be a live unit.

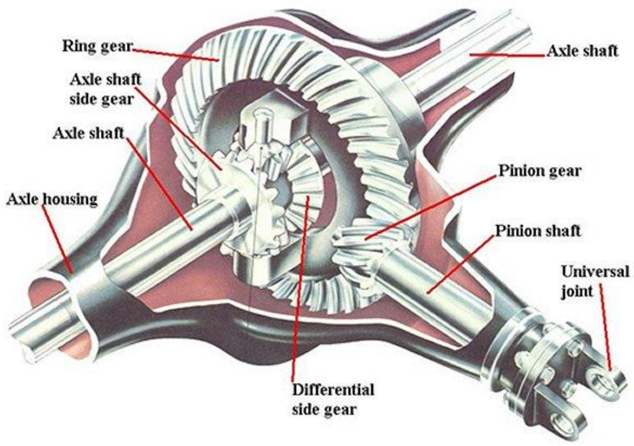

Figure 21 – General Differential Arrangement

The rear axle differential was chosen as it is the most practical for retrofitting and would suit the majority of utilities, with the only changing factor being the pinion shaft and backing plate modification and the possible relocation of the spare tyre. It is necessary to have such a solid PTO as the torque values calculated in section 3.2.4 are of such high magnitude. The flywheel will be mounted with its axis in line with the vehicle driveshaft.

3.3Flywheel

Flywheel designs can be low speed or high speed. Low speed systems are commonly considered to be systems with a maximum RPM of less than 6000. The research yielded that low speed designs are generally used for stationary applications and high speed flywheels are more suitable for vehicle applications. When the same geometry is utilised, the low speed design will have a low moment of inertia and the high speed will have a high moment of inertia. However, as was discussed in section 2.1.4.1, lowering the radius is the most effective means of reducing inertia, allowing a high speed flywheel to be used in dynamic applications. Large mass flywheels are often required to meet power requirements utilising low speed flywheels and this is a design constraint in this application.

Pullen and Dhand (2014) list several points for design of a high speed flywheel for vehicle energy storage:

- The lower the mass of the flywheel the better
- The higher the speed that the flywheel can operate for a given material density, the more energy will be stored for a lighter weight
- Composite materials will be lighter for flywheels, but metallic shafts and discs will be required, which may make attachment difficult due to the differing Young's Modulus between the two. This becomes an issue when the components are stressed
- Steel is often suitable as a material if cost is a major design constraint, providing the energy requirement is not too high. Composites will often be required to meet stress requirements.

Orientation of Flywheel

As has been discussed previously, the ramifications of gyroscopic forces are a major consideration in flywheel design. A gyroscopic torque will result if the flywheel axis is rotated. The torque acts perpendicular to the rotor axis. Pullen and Dhand (2014) find that the vehicle will experience rotating in the yaw axis from cornering, roll axis when cornering and pitch when riding through a crest or dip. Effects of vehicle yaw on the vehicle can be minimised by placing the axis of the flywheel vertically. This will isolate the flywheel from vehicle yaw changes. The authors believe that in most cases, the gyroscopic torques produced by flywheel designs were generally insignificant when compared to other moments on the vehicle. Murphy (1997) agrees with this statement. The ability of the bearings to support the resulting reactive forces when forces are imparted on the flywheel from the vehicle is of greater concern.

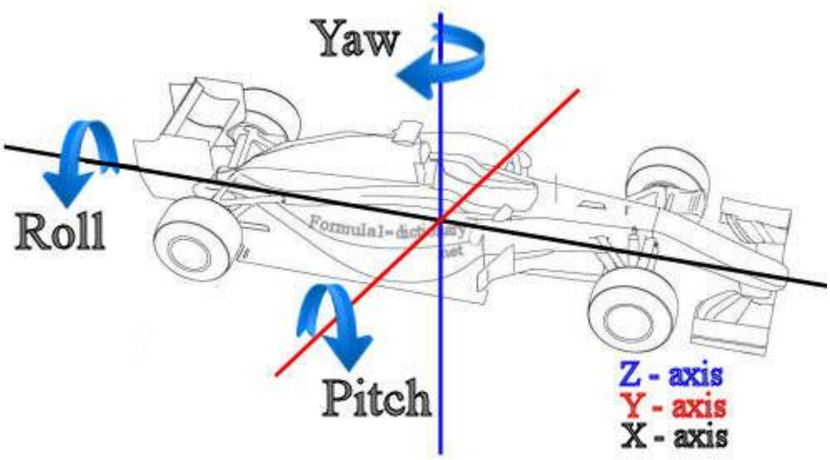

Figure 22– Axes of a vehicle (F1 Dictionary, 2010)

The effects on the bearings of the flywheel are also necessary to analyse when considering the appropriate orientation of the flywheel.

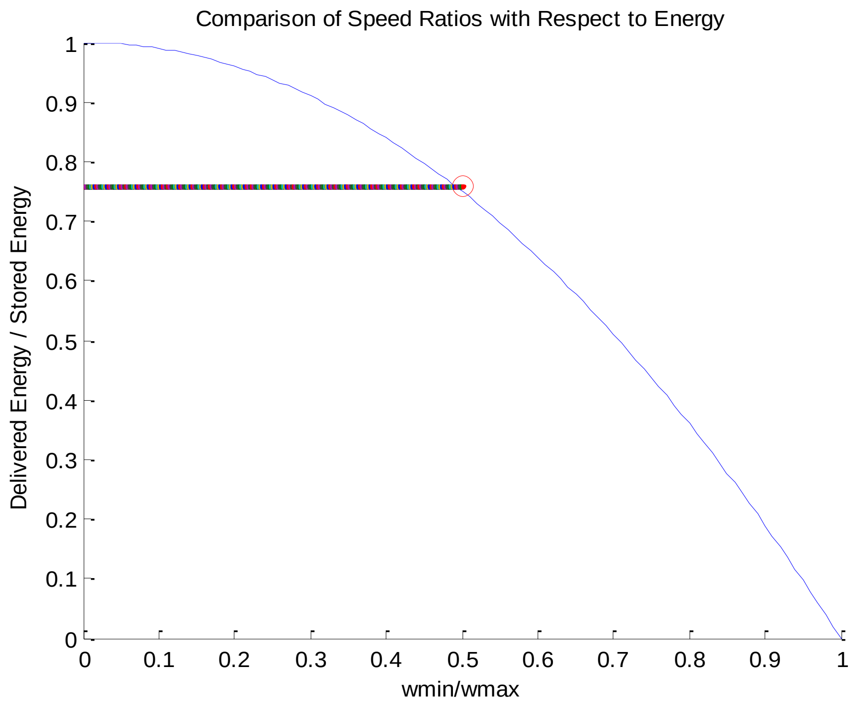

Figure 23 – Comparison of Speed ratios with respect to energy

As was discussed in earlier section, the ratio of $\frac{\omega_{min}}{\omega_{max}}$ will have a direct effect on the efficiency of energy storage. Equation (6) was divided by (7) to give a ratio of delivered energy to energy stored and plotted in Matlab. Due to the squared term of the Kinetic Energy equation, the results will decrease significantly as the ratio is increased. It can be seen that at 0.5, the ratio recommended by most of the literature so far, the energy ratio is 75%.

3.3.1 Materials

The most common materials for flywheel applications still remain isotropic materials. The materials are very dense in comparison to anisotropic materials and as a result specific strength is not very high. Composite materials are generally used for high speeds, due to their high strength and better characteristics under failure (James Hanson, 2011). Table 13 shows that Maraging Steel gives the highest energy density. Isotropic materials have a higher Young's modulus than anisotropic materials, meaning they will deform less under centrifugal force.

The downfall of composite materials is their exorbitant price to manufacture and the limitation of flywheel geometries that are suitable. The maximum K factor possible is limited to 0.5 with known geometries. The most common materials used are epoxy materials reinforced with glass, aramid or carbon fibres. The Flybrid flywheel was an example of a carbon fibre reinforced application. A flywheel of an isotropic material would also generally occupy less space than a composite flywheel of equal energy capacity, even though it would weigh more. The results from excel calculations show that high strength carbon fibre gives the highest energy density.

Failure Analysis

In order to design a flywheel, an understanding of the failure modes and how to prevent them is required. The Radial and Tangential stresses need to be known to predict failing or shattering due to centrifugal forces. Centrifugal forces are the main factor in the case of flywheel failure. From Norton (1998):

$$\sigma_T = \rho\omega^2\left(\frac{3+\vartheta}{8}\right)\left(r_i^2 + r_o^2 + \frac{r_i^2 r_o^2}{r^2} - \frac{1+3\vartheta}{3+\vartheta}r^2\right) \quad (14)$$

$$\sigma_r = \rho\omega^2\left(\frac{3+\vartheta}{8}\right)\left(r_i^2 + r_o^2 - \frac{r_i^2 r_o^2}{r^2} - r^2\right) \quad (15)$$

Where θ_T =Tangential Stress at radius from axis of rotation

θ_r = Radial Stress at radius from axis of rotation

ϑ = Poissons Ratio

Both θ_T and θ_r were plotted in Matlab to find the stress at any radius r from r_i to r_o for a flywheel of radius 250mm. It can be seen that at any given radius within the flywheel, the radial stresses will be higher than the tangential stresses. It can be seen that the maximum stress for each case peaks at $r = r_i$. It can be inferred that if σ_r is higher than the tensile strength for a material the flywheel may fail.

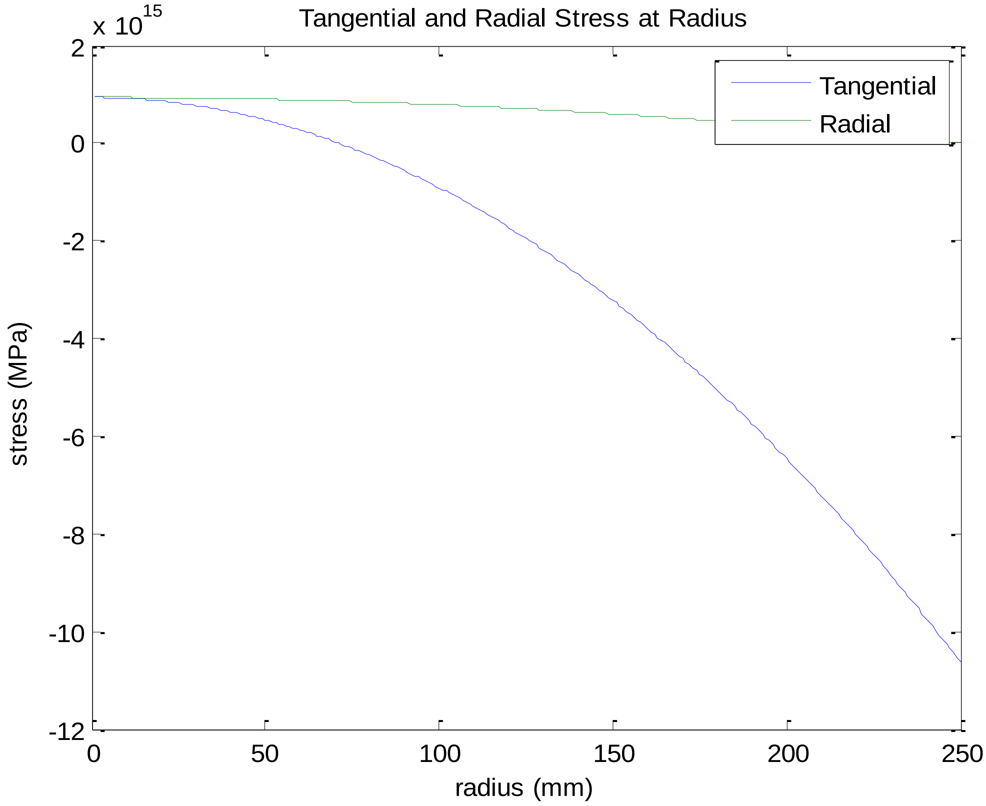

Figure 24 – Tangential and Radial stress with respect to radius

For a particular design, ω_{max} will determine the maximum energy that can be stored without failure:

$$\omega_{max} \propto E_{max}$$

$$\omega_{max} \propto \sigma_{max}$$

The maximum energy that can be stored by a flywheel geometry is given by:

$$E_{max} = KV\sigma \qquad \textbf{(16)}$$

This formula is valid as long as axial symmetry and plane stress can be assumed. When dividing by ρ, Specific energy (e) is given by:

$$e = \frac{E_{max}}{M} = \frac{K\sigma}{\rho} \tag{17}$$

Where K is a shape factor (See Table 2) and V is the material volume of the flywheel. It can be seen by analysing the formula that a higher shape factor would be desirable. The figure shows that some geometries are only suitable for composite materials. This is due to the uniaxial strength characteristics of composites and the biaxial (tangential and radial) strength of isotropic materials. K factors have been derived by various authors and these have a proportional effect on the amount of energy a given flywheel geometry can store. A K factor of 1 is obtained from a "constant stress" or "Laval" geometry. In this case, all material is stressed uniformly in such a way that the radial and tangential components remain equal from the inner radius to the outer radius (Thoolen, 1993). However the optimal geometry is theoretical and the exact shape would be material dependent, so it very difficult to design (Ostergard, 2011). K factors are lower for composites as they only exhibit desirable uniaxial strength characteristics. The maximum K factor obtainable for composites is 0.5.

If a thin or thick hoop geometry is chosen for the flywheel, the rim and hub are generally connected by a circular disc (web) or spokes (arms). Attachment of spokes or discs is quite difficult as they have measurable flexibility under load which can cause instability at high peripheral velocities. Porohit and Sharma (2002) states as a general rules that arms are only used when the diameter of the flywheel is greater than 600mm. A rigid rotor construction is often favourable. This often results in a more inefficient design, but the swept volume is used more effectively, which often results in smaller lighter containments (Porohit & Sharma, 2002). The literature shows that aluminium cylinders are often used for the centre due to their biaxial properties and light weight, which has minimal effect on the overall design.

Table 2 – Shape Factors (Pochiraju, 2011)

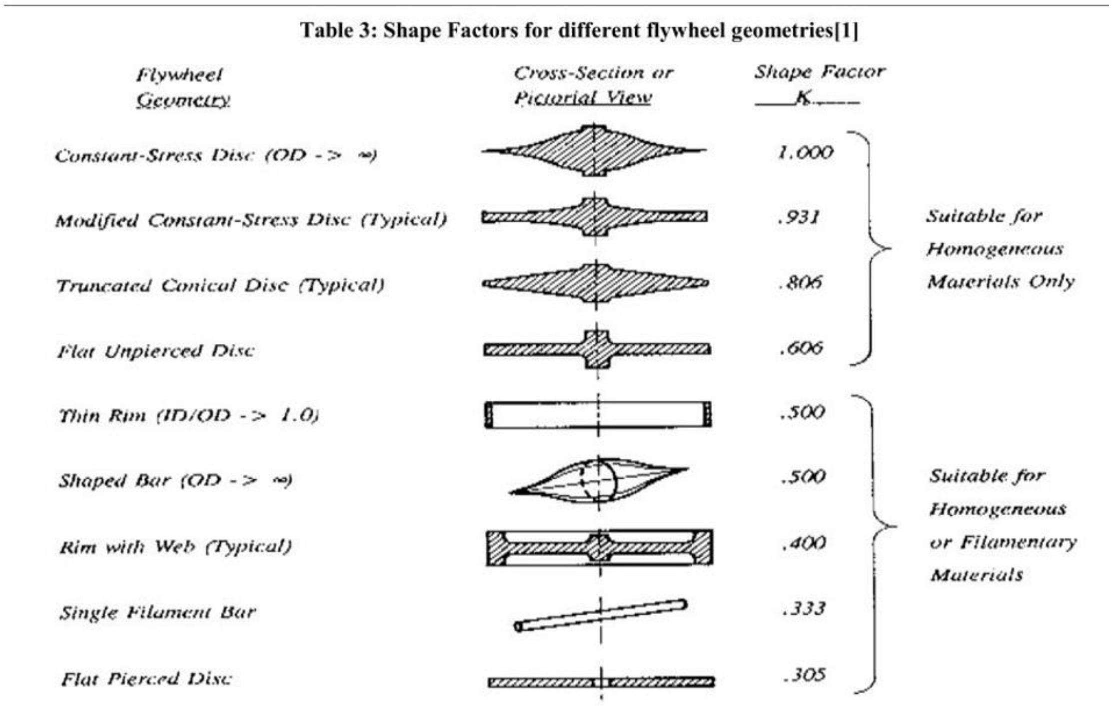

Table 3: Shape Factors for different flywheel geometries[1]

Flywheel Geometry	Cross-Section or Pictorial View	Shape Factor K	
Constant-Stress Disc (OD -> ∞)		1.000	Suitable for Homogeneous Materials Only
Modified Constant-Stress Disc (Typical)		.931	
Truncated Conical Disc (Typical)		.806	
Flat Unpierced Disc		.606	
Thin Rim (ID/OD -> 1.0)		.500	Suitable for Homogeneous or Filamentary Materials
Shaped Bar (OD -> ∞)		.500	
Rim with Web (Typical)		.400	
Single Filament Bar		.333	
Flat Pierced Disc		.305	

3.3.2 Excel Material Modelling

In order to choose the most appropriate material for the flywheel application, several materials that were prominent in the literature were compared for suitability. Specific energy was calculated in excel for a number of different materials. These were selected based on their mention in literature for their suitability for flywheel applications. The maximum tip speed was calculated for each material by manipulating the Kinetic Energy equation and this was used to calculate several dependent variables that can be used for comparison. There were several assumptions that needed to be made for the calculations:

- Based on research from Arslan (2008), the magnitude of shape factors and the ease of manufacturing, a truncated conical disc geometry was decided upon for modelling purposes. This will be easier to manufacture than a constant stress disc and will still have higher K factors than composite geometries. The isotropic disc was modelled as a truncated conical disc (mass and K factor), with the tip speed of a constant stress disc.
- The Anisotropic disc was modelled as a thin rim with mass concentrated at r_o and the mass, tip speed and K factor calculations reflect this

- The fatigue strengths for 10^7 cycles (infinite life) were used for the design stress in order to ensure reliability of the flywheel.

These assumptions were necessary in order to obtain comparable results without using overly time consuming calculations. Much of the literature considered these assumptions to be reasonable for comparative purposes.

3.3.2.1 Anisotropic Materials

The theory applied to the excel spreadsheet is outlined in this section for reference.

Velocity Calculation

When equation (1) for KE is applied to a flywheel, v represents the maximum velocity at any point and this will be at the outermost radial point, as can be inferred from equation (15). Rearranging:

$$KE = \frac{1}{2}m(r\omega)^2$$

For an idealised flywheel with all mass concentrated at R, as is common for composites:

$$\sigma = \rho r^2 \omega^2 \tag{18}$$

$$\frac{\sigma}{\rho} = r^2\omega^2 = v$$

$$\sqrt{\frac{\sigma}{\rho}} = r\omega = v$$

Substituting into (1):

$$KE = \frac{1}{2}m\left(\sqrt{\frac{\sigma}{\rho}}\right)^2 \tag{19}$$

Therefore the approximate tip speed for an idealised flywheel can be calculated using equation (19. Note that this approximation is likely to better represent composite rotors due to the geometries normally considered having a greater mass concentration at the outer rim.

Inertia Calculation

$$I = \frac{1}{2}mr^2$$

Mass calculation

The mass of a composite disc will vary considerably depending on the connecting materials used. A solid cylinder was used as an approximation for the spreadsheet. The volume occupied in the middle would be of a much smaller volume than the approximation accounts for, but the density of the isotropic material would be much higher, hopefully giving a reasonable approximation.

$$M = \pi r^2 w\rho$$

3.3.3 Isotropic Materials

Velocity Calculation

For a constant stress disc, the stress can be modelled using:

$$\sigma = \frac{\rho v^2}{2 \ln\left(\frac{r_o}{\omega}\right)} \tag{20}$$

Rearranging for v:

$$v = \sqrt{\frac{\sigma\left(2 \ln\left(\frac{r_o}{\omega}\right)\right)}{\rho}} \tag{21}$$

In order to make the data comparable with the outer rim design of the composite materials, the equation was rearranged to find the ratio of $\frac{\omega}{r_o}$ that would yield the same peripheral velocity for a constant stress disc:

$$\sqrt{\sigma(2(\ln\left(\frac{r_o}{\omega}\right)} = \sqrt{\frac{\sigma}{\rho}}$$

$$2 \ln\left(\frac{r_o}{\omega}\right) = 1$$

$$\frac{r_o}{\omega} = e^{0.5}$$

$$\frac{\omega}{r_o} = 0.607$$

The ratio of $\frac{\omega}{r_o} = 0.607$ was used for this stage of the comparison. This was used as a starting point for obtaining approximate values. Obviously the optimal value will vary considerably as other variables are altered, but this provides a ball park figure.

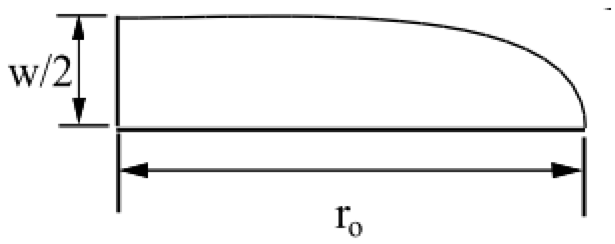

Figure 25 – w/r0

Inertia Calculation

$$I = 1/2mr^2$$

Mass Calculation

The formula used for the mass calculation was:

$$M = \frac{1}{3} \times \rho\pi(r_1^2 + r_1 r_2 + r_2^2) \times 2h$$

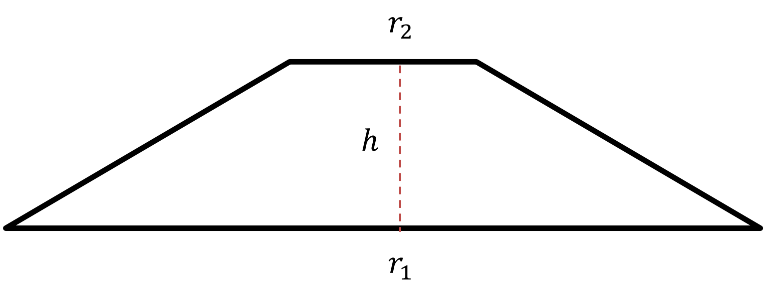

Figure 26 – Cross section of half of a truncated conical disc

3.3.4 Comparison

Using the previously developed ratio of $\frac{\omega_{min}}{\omega_{max}} = 0.5$, and the tip speed maximum, the upper and lower ω values were calculated. These were then substituted into equation (6), accounting for a K reduction factor:

$$E_{stored} = \frac{1}{2} IK(\omega_{max}^2 - \omega_{min}^2)$$

The outer radius was varied, keeping in mind that the largest possible value was 0.15m keeping within the vehicle size constraints. It was found that with the radius equal to 0.12m, two solutions for both isotropic and anisotropic materials remained. These values were compared to the required computed value of 511kJ. Using the containment approximations formulated in section 3.9.1, containment masses were also calculated for comparison. See Table 3 for tabulated results. See Figure 43 and Figure 44 for Excel graphs.

The specifications of the remaining choices are listed in Table 3.

Table 3 – Specifications of Flywheel Material Designs

Material	***Stored Energy (kJ)***	***Flywheel Mass (kg)***	***Containment Mass (kg)***
Maraging Steel	782	15.94	32.7
Titanium	677	9.2	18.4
AISI 4340 Steel	511	16.07	31.89
High Strength Carbon	550	3.173	3.173
High Modulus Carbon	666	3.37	3.37

Using the data in both Table 3 and Table 4, Titanium was ruled out due to its extremely high cost. Maraging Steel was the only other isotropic material capable of meeting the size constraint and once the relative cost had been estimated, it is no longer an attractive option. Carbon Fibre is generally regarded as being one of the more exorbitantly priced materials to only be used when other cheaper materials cannot fulfil the purpose (ASM International, 1998)(See Table 4). Carbon Fibre has been used for most of the flywheels encountered in the literature and the high specific strength and low density make this material exorbitant yet suitable. High Modulus Carbon was also ruled out due to its inferiority to High Strength Carbon.

The chosen material was AISI 4340 steel due to its relative material price and ease of manufacturing. It is a common material and does not store as much energy as the carbon fibre composites, but it will be considerably cheaper and will suit the purpose. Another major design consideration in the favour of the steel was the angular velocity that the composite flywheels are required to run at. The operating velocities may have been suitable on F1 cars and other high speed applications, but for a heavier vehicle, it would be very difficult to design a transmission able to reach speeds upwards of 60000rpm from very low vehicle velocities. This was apparent in the transmission section. The design has the advantage of the multitude of engineering data availability and history and biaxial stress resistance, resulting in more suitable geometries. Carbon Fibre may have had attachment issues when unprecedented loads were imparted upon the flywheel. Design for Anisotropic materials is also far more difficult. Attachment to rims is far more difficult as spokes or discs are required. Maintaining rotor balance and collection are very difficult, especially considering the differing Young’s Modulus between the collection and rim material.

FIG. 1 EXTRACT FROM IEE PROCEEDINGS

(vol 127,pt A, no 6 - July 1980)

LARGE SCALE ELECTRICAL ENERGY STORAGE

(by staff of CEGB)

FLYWHEELS - Table 14: Energy Storage and Cost

Material	Design Stress N/mm^2	Density $kg/m^3 \times 10^3$	Useful Energy Whr/kg	Matl. Cost £/kg	Fab. Cost £/kg	Total Cost (Matl + Fab) £/kWhr	Mass of Flywheel for 10MWhr storage $kg \times 10^3$
Carbon fibre/epoxy	750	1.55	51.5	3.6	1.0	89	194
E-glass fibre/epoxy	250	1.9	14.0	0.9	1.0	135	713
S-glass fibre/epoxy	350	1.9	19.6	1.5	1.0	127	509
Kevlar fibre/epoxy	1000	1.40	76.2	4.2	1.0	68	131
Wood (birch)	30	0.55	5.8	0.5	1.0	258	1720
Mild steel	300	7.80	8.2	0.25	2.1	286	1220
Maraging steel	900	8.00	24.0	12.0	3.0	625	417
Titanium alloy	650	4.50	30.8	50.0	5.4	1800	325
For comparison: (All figures are based on 1980 prices)							
Counterstressed mild steel	450	7.8	10.4	0.25	1.03	123	960

These cost figures do not include the containment vessel which for a laminated flywheel is minimal.

Table 4- Comparative Characteristics of Flywheel Materials (West, White, & Loughridge, 2013)

3.3.4.1 Final Design

In order to develop more accurate dimensions for the flywheel and to optimise the design, Matlab was used. The script is in Appendix 8.2.4.

KE vs. $r_o$for different $\omega\backslash r_o$ ratios were calculated and plotted for the truncated conical cylinder. This was used to show the choices available and the overall effect that the ratio has on the energy storage. See Figure 27 for the results. It can be seen that there are several options available to fulfil the requirements of the design. As expected, the larger r_o/ω ratios meant a smaller radius was required in order to meet the energy requirements. The smallest diameter possible was 0.115m with an r/w ratio of 1 and the largest possible was 0.15m with a ratio of 0.2. Due to the fact that a squat design is desirable and that the radius will have a direct impact on the inertial forces the following design specs were put forth:

Table 5 – Properties of Flywheel design

Property	Value
Radius	0.12m
Width	0.072m
Energy Capacity	505kJ
Mass	19.9kg
ω_{max}	2418rad/s

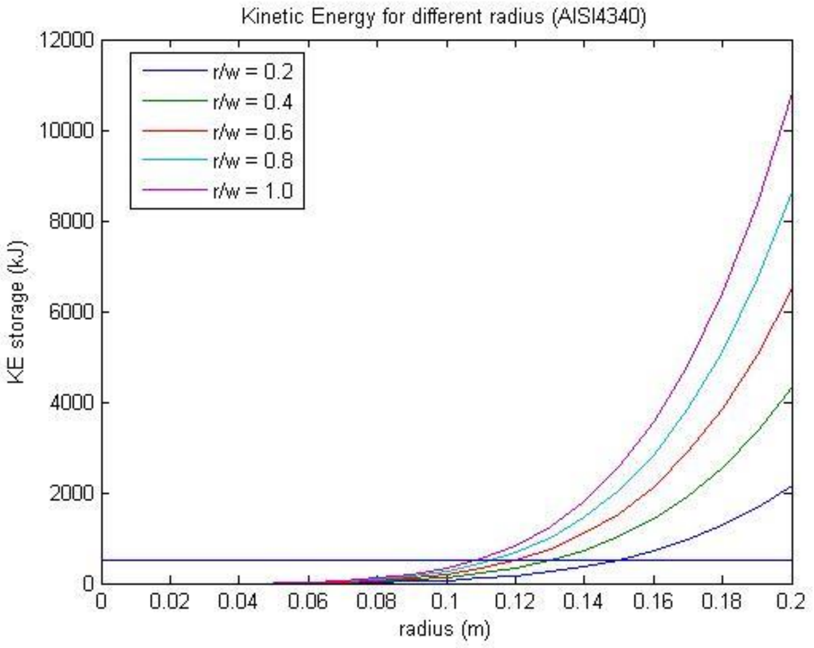

Figure 27 – Isotropic flywheel Kinetic Energy vs. Radius for different r/w ratios

3.4Vacuum Seal

A major design point of the flywheel system is the vacuum seal. This is required to keep the flywheel running efficiently and safely. Under atmospheric conditions, aerodynamic friction would be excessive and would result in both energy loss and overheating. Shaft seals are required to maintain the vacuum and sometimes these are difficult to design at high speeds and high vacuum. A vacuum pump is almost always a necessity.

The temperature of the flywheel in different vacuum environments must be considered. Thoolen (1993) states that for pressure lower than 10^{-1} mbar (1 Pascal), heat transfer by convective form can be neglected. Pressures lower than 10^{-2} mbar (0.1 Pascal) ensure no overheating in any way.

A higher level of vacuum would obviously be preferable and advantageous, however this comes with an exorbitant cost and reliability issues. A level must be chosen that results in a cheap vacuum pump, low heat generation and low windage losses. Thoolen (1993) determined analytically the heat transfer and windage losses for a particular flywheel system under different vacuum conditions. As can be seen in Figure 28, the windage losses increase significantly when the vacuum exceeds 10^{-1} bar. The heat transfer also increases greatly.

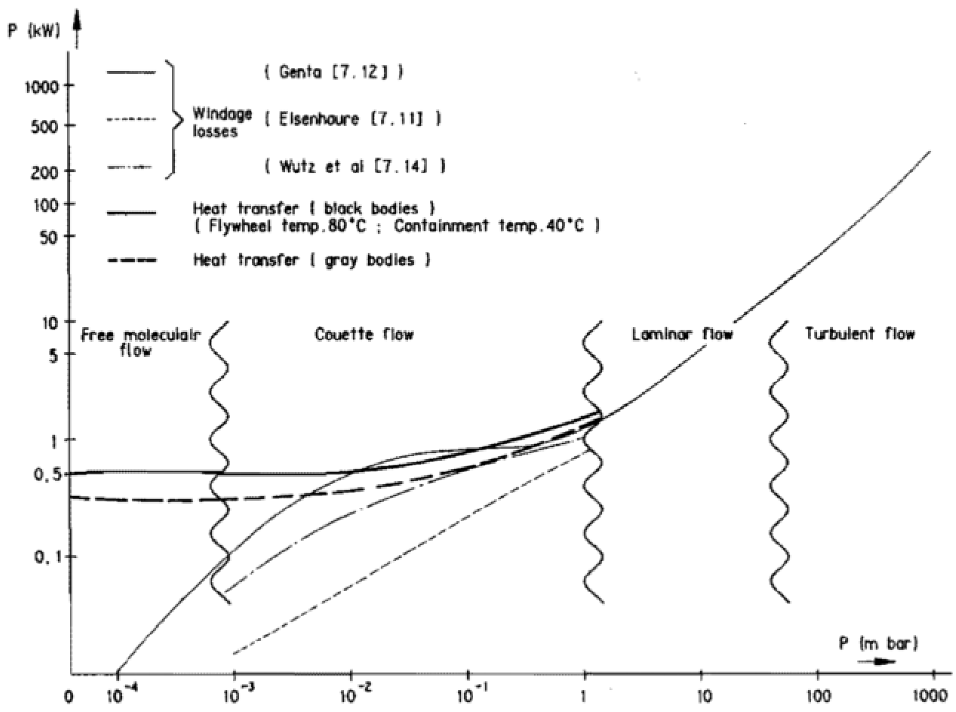

Figure 28 - Windage loss and heat transfer as a function of containment pressure (Thoolen, 1993)

A containment and seal design was proposed by (Hilton, 2010) comprising of 2 shaft seals with an evacuated section in between. This section was filled with oil. The oil caused a hermetical seal of the shaft. The design is aid to be effective upwards of 20 000 RPM. Overheating is greatly reduced compared to the standard seal as the fluid acts to cool the seal and shaft.

A similar design will be used for this system: the lubrication of the bearings will be used to maintain the integrity of the shaft seal on either side. The seals were required to be located before the bearings on the shaft regardless, as running bearings in a vacuum would render the lubrication useless.

Existing vacuum pumps were researched in order to investigate whether or not a suitable unit already exists. Refrigerant vacuum pumps currently exist that are capable of evacuating a system to 5Pa. They have a power usage of a quarter of a Horsepower (187 Watts). Without doing testing on the flywheel unit, there is no feasible way to calculate the required flow rate of the pump or to predict how often the pump will need to operate. Appropriate assumptions were made in the modelling stage to reflect this. The pumps considered operate on 240V, but it was assumed that it would be possible for the unit to be redesigned to run on 12V, or simply run off an inverter. The power supplied would come from the vehicles alternator, as it should already have the capacity to supply that amount of power. The control system would need to ensure that the flywheel does not reach critical speeds before a sufficient vacuum has been achieved when the vehicle is first driven or excessive friction will result in heat issues.

3.5Clutches

As has been mentioned previously, clutches play an important role in the functionality of the system as a whole. There have been attempts to develop theoretically ‘clutchless’ flywheel systems, but so far CVT technology is not adequate enough to facilitate this. Initially the transmission was designed with the idea in mind of having two clutches. The role of the first clutch would be to disconnect the vehicle driveshaft from the flywheel transmission and the second clutch would disconnect the flywheel from the flywheel transmission. This would be advantageous when the flywheel system is not being utilised for either accelerating the vehicle or braking as the energy losses through the transmission would not occur when power transfer is not taking place. This design was decided against after preliminary calculations due to the high cost of a high torque, low RPM clutch. Calculations using Matlab revealed that even a multi-plate clutch would need to be at least 30cm in diameter to be suitable at low speeds (Appendix 8.2.6). The decision was made that although higher drag forces would be induced by having the differential side of the transmission connected at all times, the expenditure, footprint and simplicity advantages outweigh this.

In the final design, the clutch location was placed near the flywheel enclosure (See Figure 34). The role of the clutch is to disconnect the flywheel from the CVT, which is continually powered by the vehicle transmission. Disconnecting the flywheel from the transmission stores energy more effectively as it minimises the mechanical losses affecting the system when it has obtained a velocity and is being used to store energy. The clutch is also used to initially charge the flywheel on the first braking event, using the slipping functionality, as it was established in section 3.2.3.2 that it is not feasible for the transmission ratios to allow the matching of speeds down to a standstill. The clutch location was chosen as opposed to placing the clutch directly off the vehicle driveshaft as the torque is significantly lower after the gearing and the wear to the clutch plates will be significantly less.

There are 3 types of clutches that were considered for this application. The most appropriate clutch will have a low drag coefficient when disengaged, will be long lasting and have appropriate heat dissipation.

Dry Clutch

The functionality of a dry clutch is fairly well known. It uses friction to transmit torque by a gradual connection of two members with a common axis of rotation (Juvinall & Marshek, 2006). Springs force a pressure plate towards a flywheel, which clamps a clutch plate and causes a shaft to rotate. Splines are often used to connect the clutch components to the shaft.

Wet Clutch

The operation and actuation is relatively the same as a dry clutch, if not for the addition of a fluid bath around the clutch. When the clutch is completely locked up, the fluid is squeezed from between the plates, allowing complete traction. This allows for cooling and lubrication of the clutch plates. The wet clutch design generally has provisions to ensure the flow of transmission fluid and to allow for cooling and lubrication. The losses in a wet clutch are due to the viscous drag between the rotating plates that have a relative velocity. The general advantages are:

- High Torque Capacity
- Low wear / long life
- Low drag Torque

Dog Clutch The concept of using a 'dog clutch' was considered as it would significantly reduce frictional losses at high rotational velocities, but was abandoned when the shock loads on the flywheel and other components were considered.

Juvinall and Marshek (2006) was used for the sizing of the clutches to be used. The torque capacity is given by:

$$T = \frac{2}{3}\pi p f (r_o^3 - r_i^3) N \tag{22}$$

Where $p = interface\ pressure$

$f = friction\ coefficient$

$N = Number\ of\ friction\ interfaces$

Woven Metal was chosen as the clutch plate material and a safety factor of 4 was used. This is significantly higher than what is recommended by the text for reliable operation, but this should promote longevity and reliability under the conditions it is operating.

For a maximum torque, $r_i = \sqrt{\frac{1}{3}} r_o$. The equation was then rearranged to solve for r_o given the torque requirements of the clutch.

$$r_o = \sqrt[3]{\frac{T}{\frac{2}{3}(0.805)N\pi p f}} \tag{23}$$

Due to the high speed and high frequency of engagement, it was decided that the clutch should be a wet clutch. The losses and the influence on the clutch subsystem is discussed in section 3.5.1. Research by (VENU, 2013) on the drag of clutches found increasing the size of clutch plates was more effective at reducing drag than to increase the number of plates to suit the same Torque specification. After calculation it was discovered that the dimensions do not vary greatly if the clutch is placed after the gearing. It was originally hypothesised that the increased size of the clutch plates would negate the drag benefit of having lower speeds.

From the options produced by Matlab, this resulted in a choice of a single plate clutch with an outside radius of 6.2cm and an inner radius of 3.6cm.

3.5.1 Losses

In an ideal scenario, when the clutch is disengaged, no torque would be transmitted and no energy would be lost from the flywheel system. In reality, regardless of whether a wet or a dry clutch is used, drag torque will be generated due to the shearing of the respective fluid in between the discs (Shoaib Iqbal, 2013). In the case of a wet clutch system chosen, the viscosity should be minimised in order to reduce drag.

The possibility of enclosing the clutch in the flywheel vacuum or inside a separate vacuum chamber was considered due to the advantage it would hold in term of drag torque resistance. Unfortunately maintaining lubrication within a vacuum is very difficult and the clutch would have to be a dry configuration. In the case of a dry configuration, the main heat transfer medium is convection, and convection relies upon a working fluid to dissipate the heat. The lack of a fluid surrounding the plates would mean that the only effective form of heat dissipation would be radiation to the enclosure, which would not be sufficient to keep the clutch cool enough for regular operation. Due to this, it was decided that the clutch would have to remain outside of the enclosure; a consequence of which is the subsequent drag torque.

Approximations for the power loss of the energy stored can be obtained from equations and available data. Using equation (10), a maximum and minimum power usage due to windage losses can be calculated.

A simple model was proposed by (Kitabayashi, 2003) for the calculation of drag torque for a wet clutch:

$$T = \frac{N \times \mu \times \pi(r_2^2 - r_1^2\,) \times \omega \times r_m^2}{h} \tag{24}$$

Where $N = Number\ of\ Clutch\ Plates$

$\mu = Dynamic\ viscosity\ of\ fluid$

$r_1 = Inner\ Radius\ of\ Clutch$

$r_2 = Outer\ Radius\ of\ Clutch$

$r_m = \frac{(r_1 + r_2)}{2}$

$h = Clearance\ between\ discs$

A clearance was assumed to be 0.2mm, as used by VENU (2013). The viscosity of the fluid was assumed to be 0.0082Ns/m^2 (ATF Dextron III operating at 80°C) (VENU, 2013). ω was assumed to be $\frac{\omega_{max}}{2.5}$ after the gearing stage.

This resulted in a maximum Torque Drag of 118.3 Watts. This will be accounted for in the modelling phase.

Further research also suggests that this estimate of clutch drag torque is also the worst case scenario. Despite the fact that the more viscous medium generally means a direct increase in drag torque, research suggests that for constantly high RPM scenarios, the drag torque is significantly decreased as centrifugal forces diminish the oil film between the discs. This can be clearly seen in Figure 29. This means that the practical results should have even less drag than originally calculated.

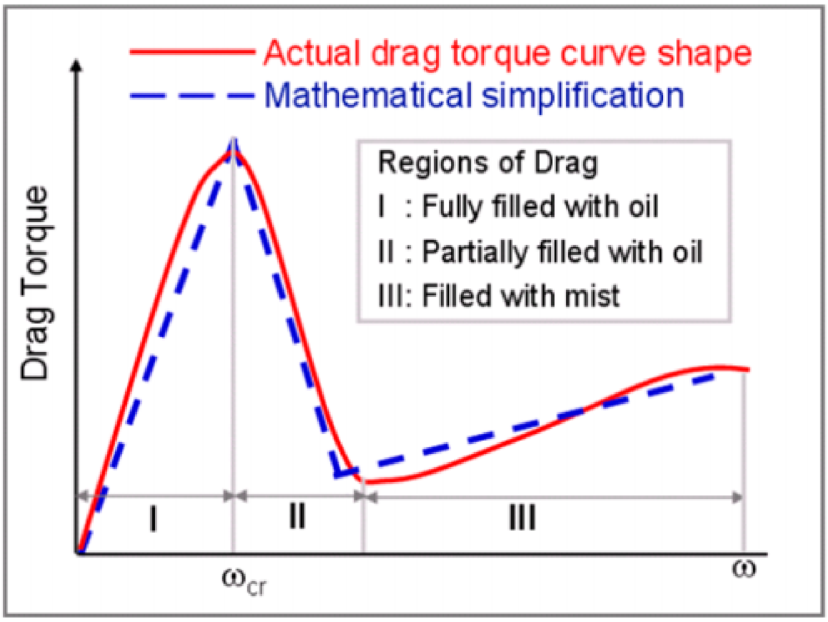

Figure 29 – Generalisation of Drag Torque Curve for wet clutch types (VENU, 2013)

Another major design decision was whether to put the clutch before the gearing or after the gearing on the flywheel side to minimise drag. There was the option to have a larger clutch transmitting a higher torque coupled with gearing losses, spinning slower or a smaller clutch, spinning fat much higher speeds. It was decided that a lower speed clutch should be used due to the lower cost and enhanced reliability.

3.6Keyways

It was necessary to calculate whether or not it was be possible to transmit such large amounts of torque through the proposed driveline. Equations from Juvinall and Marshek (2006) were rearranged to see if keyways of a feasible diameter could be used for the gear on the driveshaft, the point of the largest torque application.

$$T = \frac{\pi d^2 D S_{sy}}{4} \tag{25}$$

$$d = \sqrt{\frac{4T}{\pi D (0.58 S_y)}} \tag{26}$$

The calculations yielded the requirement for a 5mm square shaft on the diff pinion. The other shafts yielded feasible key dimensions.

3.7 Shafts and Bearings

The large angular momentum of the flywheel causes gyroscopic effects that have been discussed previously. The angular momentum reactor points along rotational spin axis. When the orientation of this axis is changed, large torques are generated. This will cause both shaft loads and bearing loads.

A very high vacuum is required for the operation of the flywheel, however lubricant will vaporise if the pressure is too low. The bearings will be kept outside of the flywheel enclosure seal. Pullen and Dhand (2014) state that bearing choice is typically that of a roller bearing due to the requirement for high efficiency, low cost and high capacity. The bearing must be able to support loads due to acceleration shocks and gyroscopic effects. Silicon based oils are often chosen despite their inferior lubrication properties.

Murphy (1997) states that the use of magnetic bearings is becoming more prominent for high speed applications in recent times. They are capable of accommodating much larger mass imbalances than other mechanical bearings, although other sources quote reliability issues.

There are 4 types of loads that flywheel bearings can be subjected to:

1. Shock – Shock events are brief and transient in nature. These could be events such as potholes or rough surfaces. There is generally a peak acceleration of the system as a result.
2. Vibration – Vibration is characterised in a similar way to shock, although the input and response maintain steady amplitudes for a given amount of time (Murphy, 1997).
3. Manoeuvring – Manoeuvring occurs when the vehicle defers from the original vector path that it was taking and alters the vehicle linear momentum. Braking, accelerating and cornering are examples.
4. Rotating Mass Imbalance – The mass imbalance of the flywheel can cause bearing loads under high rotational velocities. The magnitude of the load is proportional to the imbalance.

The SKF catalogue was used to decide on suitable bearings for the application. Self-Aligning ball bearings would be particularly suitable as they are insensitive to angular misalignment and generate less friction than other types of rolling element bearing.

Appendix 8.3 shows the design calculations that were performed to develop the specifications for the shafts of the retrofit.

The flywheel to shaft connection is a particularly important design point due to the high operational speeds and the consequences of imbalance. Possibilities include:

- Interface fit
- Interface fit with a shoulder to reduce stress concentrations in the shaft
- Tapered connection

The interface fit with a shoulder was chosen for its simplicity and prominence in the literature.

Bearing Selections were determined by the relevant shaft diameters, the velocity requirements and the radial and axial forces. Tapered Roller bearings were used to replace the bearings on the pinion shaft. Standard roller bearings were found to be suitable for most applications due to their high limiting speed and suitable axial and radial force support. The bearings and their specifications are listed in Table 6.

The loads on the flywheel bearings were analysed. Using data from (Murphy, 1997), the maximum acceleration a differential will experience due to obstacle collisions with the wheels is 13 gees. Using the mass of the flywheel and shaft subjected to this load, the total bearing load was found to be 1.37kN. The dynamic loading of the self-aligning ball bearings selected was 19kN, far in excess of this load. There are many other load factors that would need to be accounted for in practice, but this provides a ball park figure.

Table 6 – Bearing Selection

Bearing	Designation	Quantity	W	d	D	Limiting Speed
6007	Deep Groove	1	14	35	62	15000
33210	Tapered Roller	2	32	50	90	5000
RMS12	Deep Groove	1	24	38	92.25	11000
61806	Deep Groove	1	7	30	42	22000
61905	Deep Groove	1	9	25	42	22000
6006	Deep Groove	3	13	30	55	17000
6208	Deep Groove	1	18	40	80	11000
1207ETN9	Self-Aligning	2	17	35	72	20000

3.8Lubrication

The gearbox connected to the differential was chosen to have the pinion gear below the gear to ensure that the gears would remain immersed in differential fluid and provide effective lubrication. The bearings and gears in the flywheel side gearbox are also lubricated in an oil bath. See Table 16 for the volumes required. The flywheel bearings are lubricated by an oil jet, as they operate at such high speeds. The CVT is known to require lubrication as well, but the fluid or mode was not determined.

3.9 Safety

Safety has always been a significant factor in flywheel design as there is such potential for ramifications, should the flywheel disintegrate or become dislodged. As was outlined in the initial risk assessment, large safety factors should be used for components to ensure that failure is as improbable as possible. In the case of the containment, safety factors should be high enough that failure can be contained. James Hanson (2011) states that instrumentation should be utilised to detect and prevent failure and that it should be ensured that failure is incremental and that the failure mode results in fragments and not the entire rotor. Pullen and Dhand (2014) disagree with this and believe that the only option is to guarantee containment under all circumstances without relying on instrumentation. Instrumentation should still remain as a preventative measure however. They consider composite flywheels to be a safer option as they disintegrate to dust rather than flying off as chunks of matter.

3.9.1 Containment Design

Containment is of paramount importance when designing flywheel systems. It is an important design point as it is required to not only maintain the vacuum functionality, but to act as a safety containment in the case of a failure. The different flywheel designs will have different failure modes and will subsequently require different forms of containment. Some of these were covered in section 2.1.5. It would be irresponsible to avoid the issue of containment as history has shown on several occasions that severe consequences exist if safety precautions are not implemented. U.S. House of Representatives (1996), state that adequate containment is required if $\sigma_{ult} < 4 \times \sigma_{operating}$. This is the case for the design at hand. Running a flywheel at an RPM much lower than the nominal failure speed will obviously reduce the risk of failure, but will decrease the energy density of the system, which is a design factor of great importance. Composite rotors (specifically carbon fibre) have been shown to break up into much smaller fragments than metallic counterparts when impacting with a solid and often do not require containment of the same magnitude. The often smaller weight often has a direct effect on the containment requirements. However, in the case of vehicle applications, a larger containment would still be necessary when accounting for the possibility of a collision, where the forces would be much higher than a standard internal failure. Thoolen (1993) finds that the best solution is to ensure that the containment is capable of withstanding the worst case failure scenario, in order to ensure safety. The common containment options include:

- Brute force – Designing a containment that is of a magnitude sufficient enough to absorb the maximum energy of the flywheel in the event of a failure
    - Thoolen (1993) finds that containment in the form of a cylindrical shell generally weighs twice that of a flywheel manufactured form isotropic material and half that of an anisotropic one.
- Rotating containment – An outer shell that absorbs energy. This design is essentially a flywheel in a flywheel and is quite complex to design effectively.
- Soft Catch – The method of containment involves designing using a material that spreads out the energy of the impact from fragments. It acts like a net or a sponge in the case of failure.

U.S. House of Representatives (1996) in a report stated that, “With respect to safety, it is not clear that a satisfactory solution has been found to the problem of burst containment. It may be that avoidance of catastrophic burst – rather than burst containment – is necessary for industry and public acceptance.”

The amount of energy imparted to the containment wall is dependent on the size of the fragment and the speed of the flywheel at the time of the incident. It can be seen from Figure 30 that as the angular size of the fragment increases, the imparted force becomes more rotational.

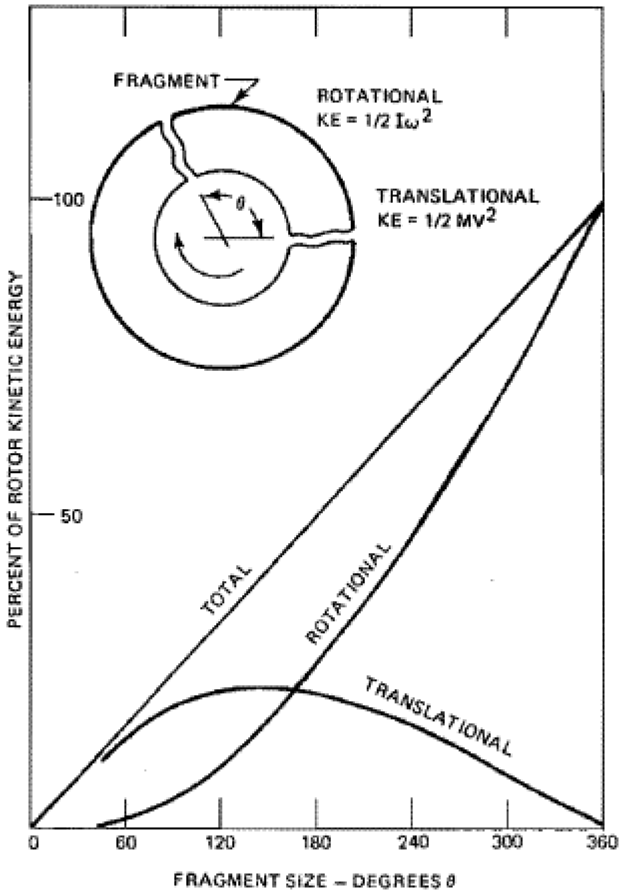

Figure 30 – Rim Fragment Energy (Thoolen, 1993)

The 'Brute Force' method was used for the final design. It relies upon thick ductile steel for containment in the case of a failure. This material was chosen for its superior impact resistance, strength and toughness to other metals. A full analysis of the effectiveness of the containment would be necessary once the nature of failures for this particular flywheel had been established. The general guide by Thoolen (1993) was used. This stated that for an isotropic material, the mass of containment is typically 4 times that of the flywheel. This was verified once the model was completed in Creo.

3.10 Design Drawings

The flywheel system design was modelled in Creo for the representative vehicle. The overall design went through several design stages and the final design effectively wraps around the diff. Figure 31 shows the pictorial view of the system on a representative differential. Figure 32 shows a sectional pictorial view. It was cut along the axis of the shafts to show the internal components. Components not pictured are:

- Vacuum Pump
- Oil Pump
- Attachment to Differential
- CVT
- Bolts
- Oil Pump and attachments
    - The bearings in the flywheel enclosure are oil jet lubricated

Models that were not integral to the design of the system for the differential and clutch were obtained from GrabCAD. The authors were Yondoler (2013) and Jessner (2014) respectively. The Bearing and seal models were obtained from (SKF, 2014b)

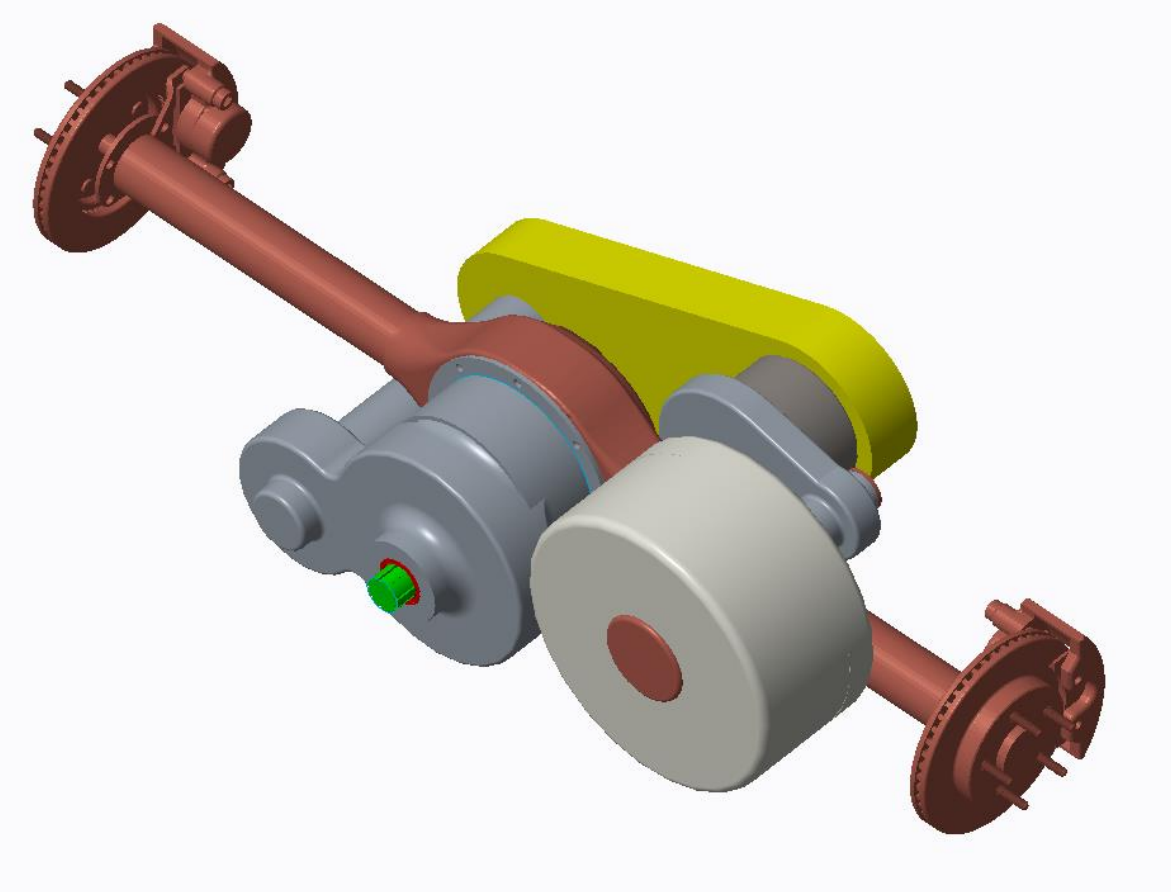

Figure 31 - Pictorial View of System

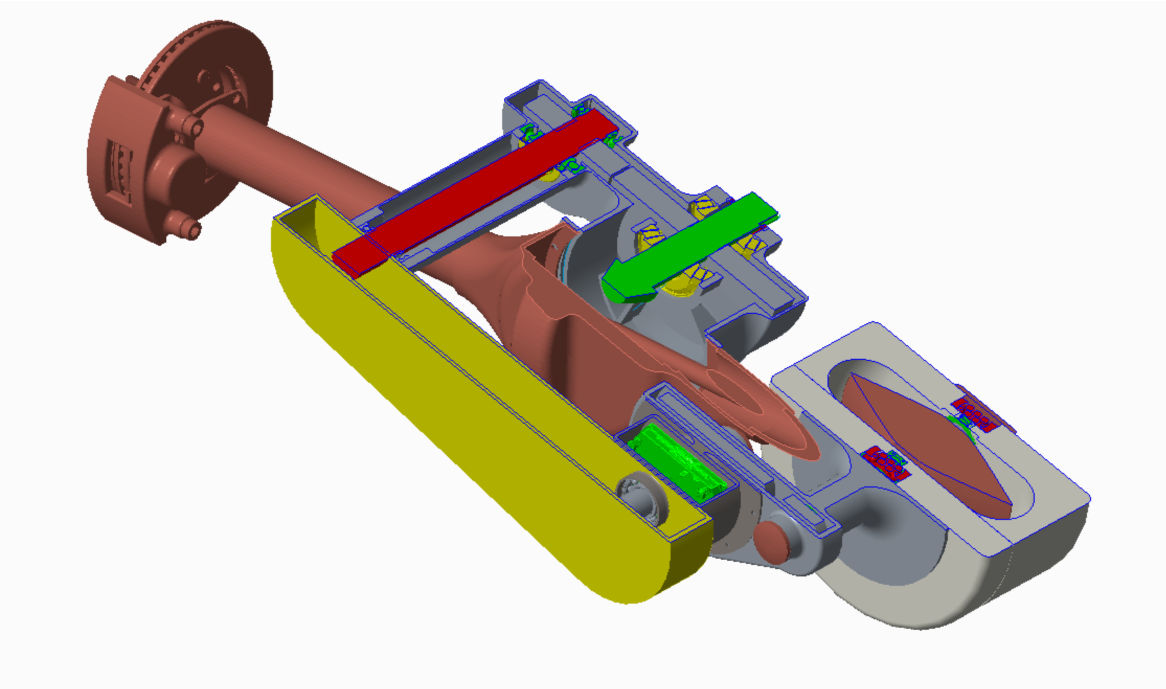

Figure 32- Sectional Pictorial View of System

Figure 33 shows a sectional view to portray the arrangement of components. It can be seen that the pinion shaft of the diff is connected to the first gearing stage, which is connected to the CVT behind the differential. This then connects to the clutch and second gearing stage, before transferring rotation to the flywheel inside the housing. Oil seals are located on the shaft of the first gearbox housing and on both sides of the clutch, second gearbox housing and flywheel housing.

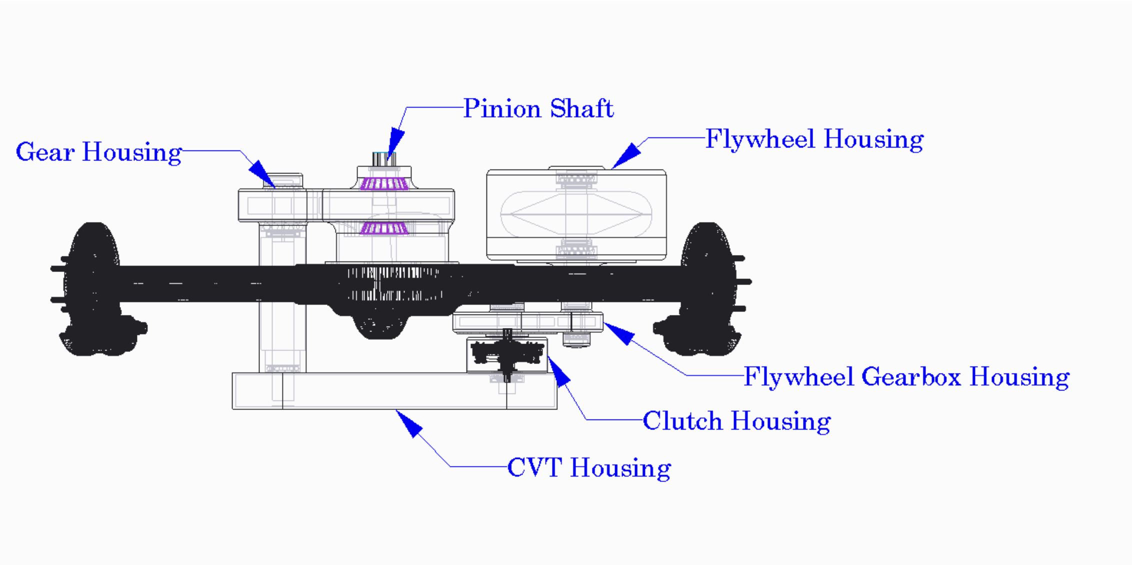

Figure 33 - Sectional View of System on X Axis

Figure 34 shows a sectional view of the flywheel housing. The self-aligning bearings can be seen on either side of the flywheel, behind an oil seal.

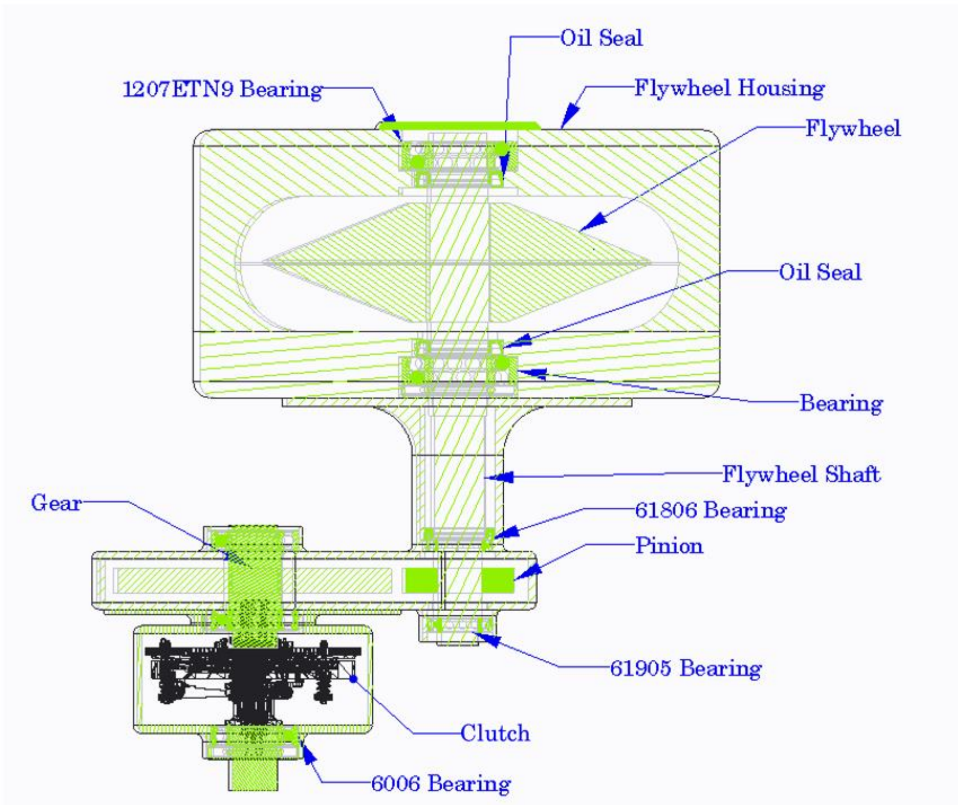

Figure 34 - Sectional View of Flywheel Housing, Gearbox and Clutch

Figure 35 shows the dimensions of the truncated conical flywheel.

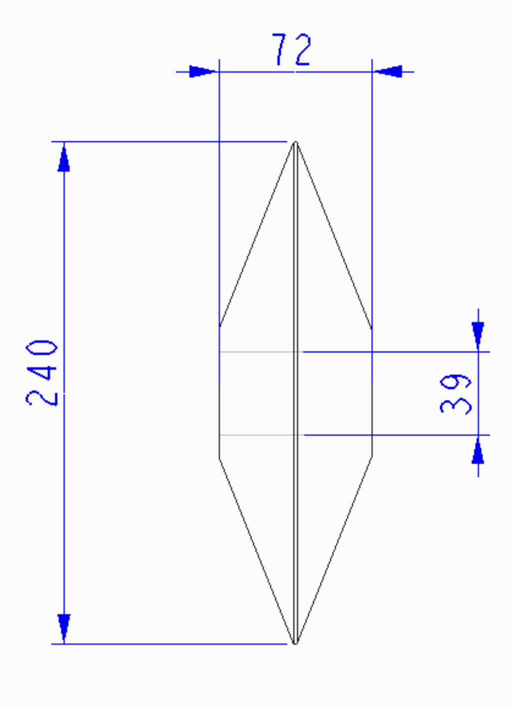

Figure 35 - Flywheel Drawing

Supplementary attachment will be required to the differential, although is not pictured in the assembly drawings as mounting would be very different on a per vehicle basis. The preferred method of attachment would be to the housing on either side of the crown and pinion of the differential. The system was designed to minimise the stress on the axle and the keep the components as close to the housing as possible so as to minimise the forces due to the moments. FEA would need to be used to verify the strength of the system under loading.

3.11 Mass of System

It was established that minimising the mass of the system would be an important facet of the overall design. Not only will additional weight cause more energy to be used by the vehicle in general, but a considerable mass will reduce the maximum load that the vehicle can carry legally.

The mass was calculated using the analysis feature of Creo Parametric. The volume of individual components were found and multiplied by the density of their respective materials. The pinion shaft and the differential itself were removed from the assembly as they are already an integral part of the vehicle regardless of KERS implementation.

The total volume of the system was found to be $1.503 \times 10^7 mm^3$. Taking into account the densities of the materials, this gave a mass of $118kg$. This will be the dry mass of the system. In order to obtain the kerb mass, the mass of fluid to be added to the system was incorporated. The capacities are:

- Diff housing - 5L of 80w90 gear oil
- Clutch Housing – 0.5L of ATF DXIII
- Flywheel Gearing housing – 1 Litre of 80w90 gear oil

$$M_{fluid} = 887\left(\frac{kg}{m^3}\right) \times 6.5$$

$$M_{fluid} = 5.77kg$$

A mass of 20kg was assumed for the CVT. This brings the total mass to 123.8kg.

4 Quantification of Benefit

4.1 Estimated Cost

Obtaining an accurate cost for the system was difficult, as pricing will change so substantially with manufacturing quantity and manufacturing techniques. In order to estimate the cost of the system, available pricing from manufacturers was used, as well as rough costs from literature and comparisons between existing technologies.

4.2 Potential Fuel Savings

Matlab was used to model the potential fuel savings of the system. The script is in Appendix 8.2.3. The design constraints and theory of operation were applied to drive cycle data in order to produce meaningful graphs and statistics. The FTP-75 cycle was the main cycle used for analysis due to its similarity to the actual driving conditions. A model was developed for the original drive cycle to simulate the typical fuel usage for the representative vehicle. Most of the analysis relied upon formulae obtained from The University of British Columbia (2009).

4.2.1 Power requirements of Vehicle

The overall power requirements were developed using the following method:

$$P_{total} = P_{acceleration} + P_{cruising}$$

Where:

$$P_{velocity} = P_{air\ resistance} + P_{rolling\ resistance}$$

$$P_{velocity} = \frac{1}{2}\rho A_{car} C_D d v^2 + d\mu_{RR} m g \qquad (27)$$

$$P_{acceleration} = F \times s$$

$$P_{acceleration} = ma \times s$$

$$P_{required} = \frac{P_{total}}{\epsilon}$$

The fuel consumption was then calculated according to:

$$F = \frac{E_{input}}{\rho_{fuel} * E_{fuel}} \tag{28}$$

When the cumulative sum of the fuel requirements over the whole drive cycle is taken, the fuel requirement over the entire drive cycle was obtained. For comparison to available figures from manufacturers, a L/100km figure was obtained for the specific drive cycle using (29.

$$\frac{L}{100km} = \frac{100km \times 1000m}{distance\ of\ drive\ cycle} \times \sum F \tag{29}$$

Assumptions for the model:

- The Energy density value for fuel is 44MJ/L (Caldirola, 1981)
- The coefficient of rolling resistance was found to be an average of 0.0113 (vejdirektoratet, 2004). This was assumed to be constant over all velocities.
- The drag coefficient was taken to be 0.44 (Wikipedia, 2014). This was for a Toyota light truck.
- The model assumes constant velocity for each second in order to obtain air resistance and rolling resistance values.
- The efficiency of the gasoline engine was taken to be 30% and the common rail diesel engine was taken to be 45%
- The model assumes that the efficiency remains constant for the simulation. This is an idealisation and obviously load and RPM will have an effect on the efficiency in practice.
- Vehicle idle was considered to be negligible for the modelling phase

4.2.2 Regenerative Braking Savings

In order to model the savings of the regenerative braking system, a code was written that stipulates the operation of the flywheel system in relation to the vehicle characteristics and data from the drive cycle at any point in time. See Appendix 8.2.1. A real time storage matrix was used and altered on a per second basis as the drive cycle was run. The constraints were developed and coded. They are shown in a flow diagram in Figure 36.

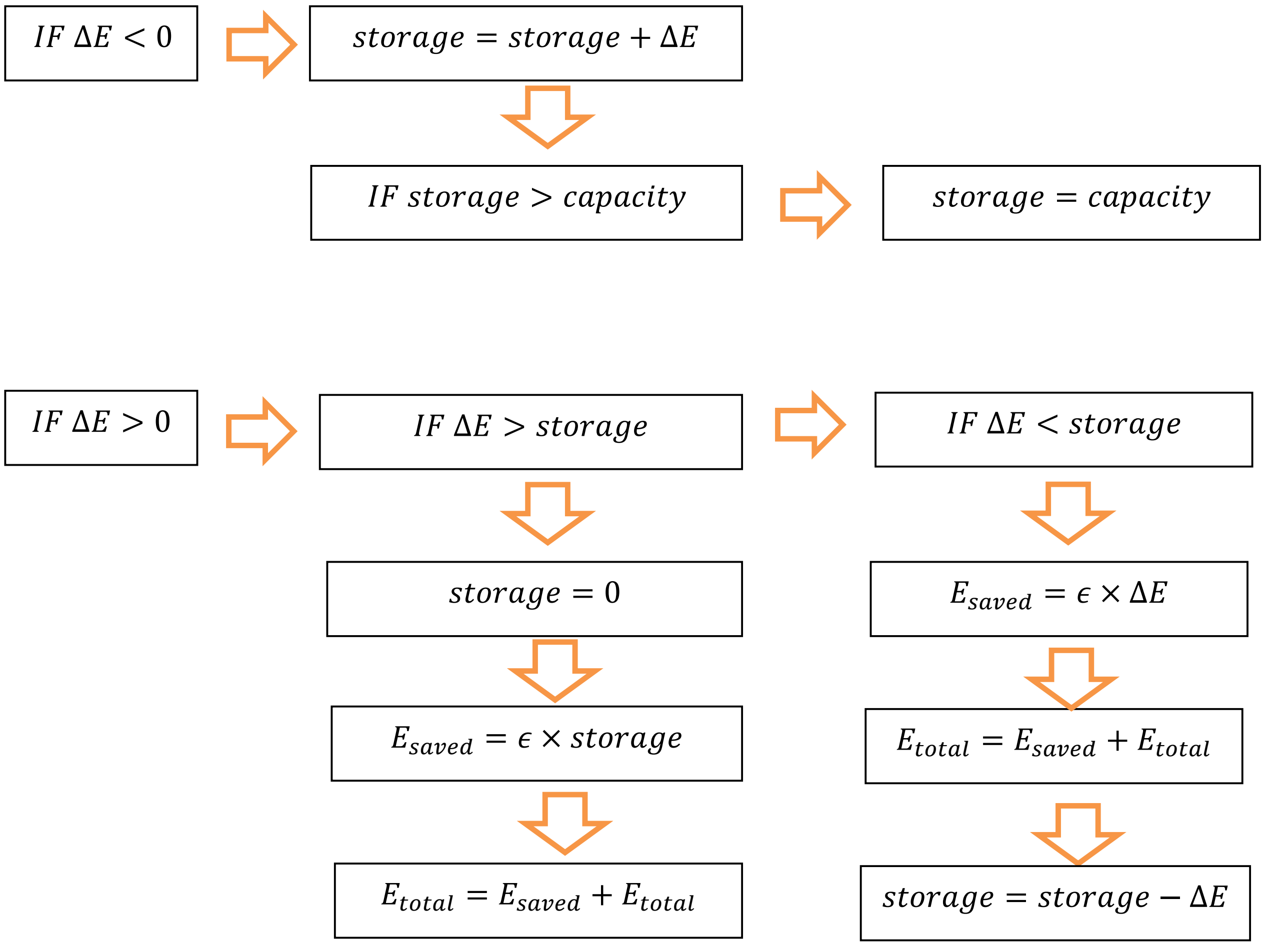

Figure 36 – Flow chart of Matlab system workings

A fuel consumption matrix was constructed by subtracting the energy provided by the system from the original energy requirement figures for each increment of time. By taking the cumulative sum of the fuel consumption for both the original and the regenerative run, a graph was able to be developed to compare the fuel usage of both and assess the fuel savings.

Assumptions for the model:

- Energy is able to be recovered if the vehicle velocity is between 10 and 60 km/h
- Energy is recovered at an efficiency of 80%
- Energy recovered is able to be converted directly back to kinetic energy at an efficiency of 80%
- No atmospheric or frictional losses are considered when the flywheel has been accelerating
- Idling losses are neglected and it is assumed that the vehicle is not using any energy when stationary. It is assumed that newer vehicles have the functionality of turning the engine off when idling for long periods of time.
- The Flywheel will lose energy due to bearing friction, air resistance, seal friction and clutch drag losses. This was found to be 400 Watts. The calculations are seen below

$$P_{loss-total} = P_{clutch} + 4 \times P_{seal} + 2 \times P_{bearing} + P_{vacuum}$$

Clutch Drag losses were calculated in section 3.5.1 as being a maximum of 118.3 Watts. Vacuum losses were calculated in section 3.4 as being approximately 187 Watts.

Bearing and seal losses were calculated using the SKF bearing calculator (SKF, 2014a). The assumption was made that the bearings would be lubricated using an oil jet and that the temperature of the oil would be an average of 60°C. This gave total torque with of 42Nmm per bearing. Seal losses were approximated at 6Nmm.

$$P_{bearing} = t\omega$$

$$P_{bearing} = \frac{42}{1000} \times 2418$$

$$P_{bearing} = 102W$$

$$P_{seal} = \frac{6}{1000} \times 2418$$

$$P_{seal} = 14.5W$$

$$P_{loss-total} = 118.3 + 4(14.5) + 2(102) + 187$$

$$P_{loss-total} = 567.3W$$

This is the worst case scenario and assumes that the flywheel system is at the maximum design RPM for the whole time. This should compensate for the fact that gear drag is not known and would be very difficult to calculate.

The overall efficiency of the system was determined by multiplying together the efficiencies of the different components.

$$\eta_{total} = \eta_{gear1} \times \eta_{CVT} \times \eta_{gear2} \quad (30)$$

Where: $\eta_{gear1} = 0.96$

$\eta_{gear2} = 0.98$

$\eta_{CVT} = 0.9$

η_{total} was rounded down to 80% to compensate for the fact that bearing and seal losses have not been considered.

Statistics from Matlab

Note that all statistics are per drive cycle

Table 7 – Matlab Results (Petrol Engine)

Total available energy (kJ)	5623.79
Recovered energy (kJ)	2549.41
Recovery Rate	45.33%
System Utilisation	53.38%
Fuel Saving (%)	6.59%
Cost Saving per 100km @$1.30/litre	$0.91

It can be seen that although the system has a very high recovery rate of 45.3%, the overall fuel savings are only 6.59%. This is due to the very low overall percentage of fuel energy content that actually makes its way to propelling the vehicle. The energy that the flywheel system is recovering is a small portion of the overall energy usage, which is the reason for the overall fuel savings being so minuscule

It can be seen in Figure 38 that the capacity of the system is never fully utilised in simulation. This is due to GPE being neglected throughout. In practice, changes in elevation would require a larger system such as the one originally designed for. It was assumed that the elevation will lead to fluctuations in the energy stored by the system, meaning that the generous storage capacity would be necessary in practice, but for simulation the average should give relatively the same results over the entire drive cycle. The fact that the system never reaches maximum RPM is also effectively introducing a larger safety factor and will prolong the life of components overall.

Figure 37 shows the Energy Requirements of both the standard vehicle and the vehicle fitted with the flywheel system. It can be seen that the energy requirements are lower overall and points of peak acceleration are especially noticeable. Once again, when the velocity is outside of the range of the transmission, no difference is seen between the original and the altered values.

Figure 38 shows that there are periods when the Energy storage is technically negative. This means that the angular velocity of the system has dropped below ω_{min}. This is especially noticeable in the high speed section of the drive cycle, when the flywheel is isolated from the system as the transmission is not able to match the speed of the driveshaft to the speed of the flywheel. The parasitic losses will continue to use power from the flywheel in these periods. Other than that it can be seen that as the KE of the vehicle decreases, the energy storage increases as expected. It is obvious that the system is most effective for stop start driving, as it was originally designed for.

Figure 41 shows the Power and Cumulative Fuel Consumption over the drive cycle. It can be seen in Figure 40, the plot of cumulative fuel consumption vs. time, that the cumulative fuel consumption with the KERS system steadily deviates downwards from the trend of the standard fuel consumption. The difference is most noticeable for the second half of the cycle, as the stop start driving pattern is more prominent.

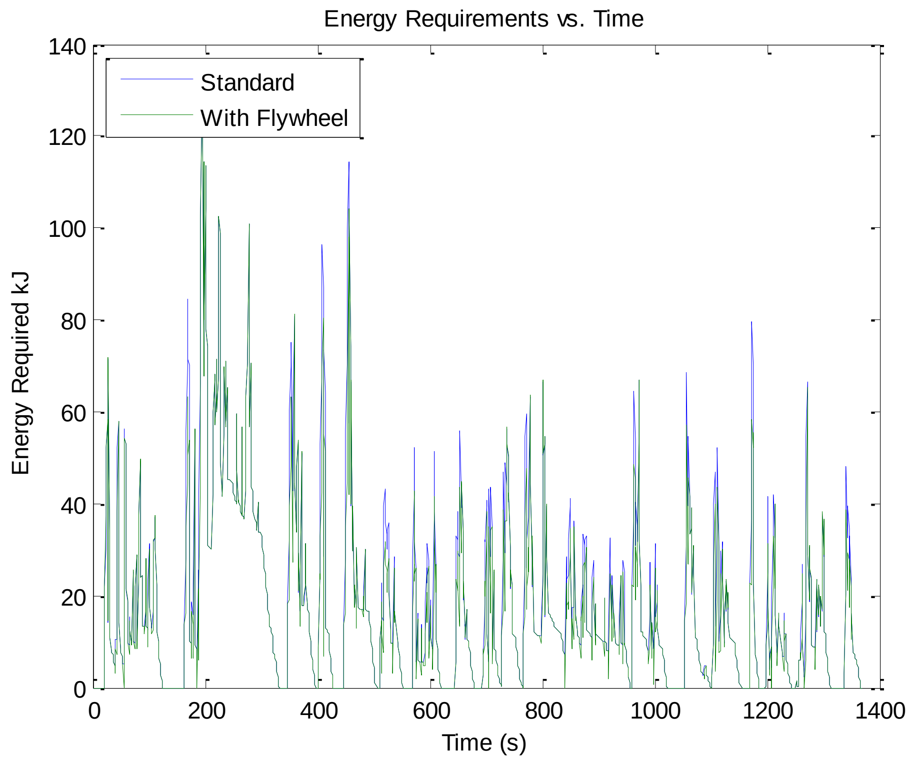

Figure 37 - Energy Requirements vs. Time

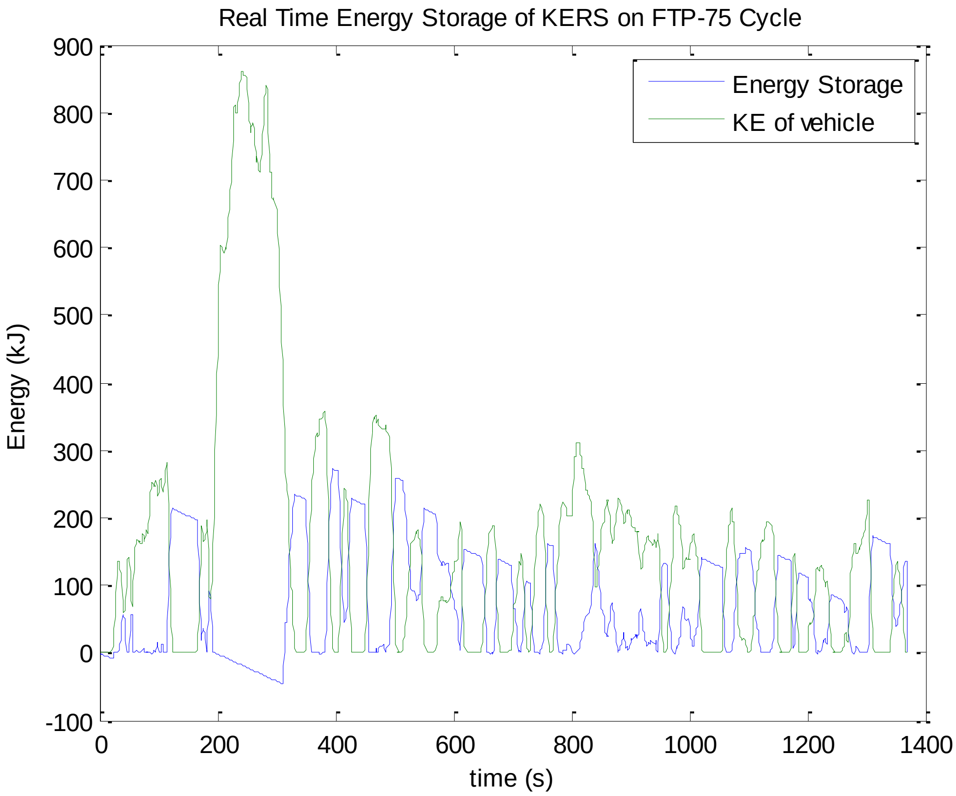

Figure 38 – Real Time Energy Storage of KERS on FTP-75 cycle

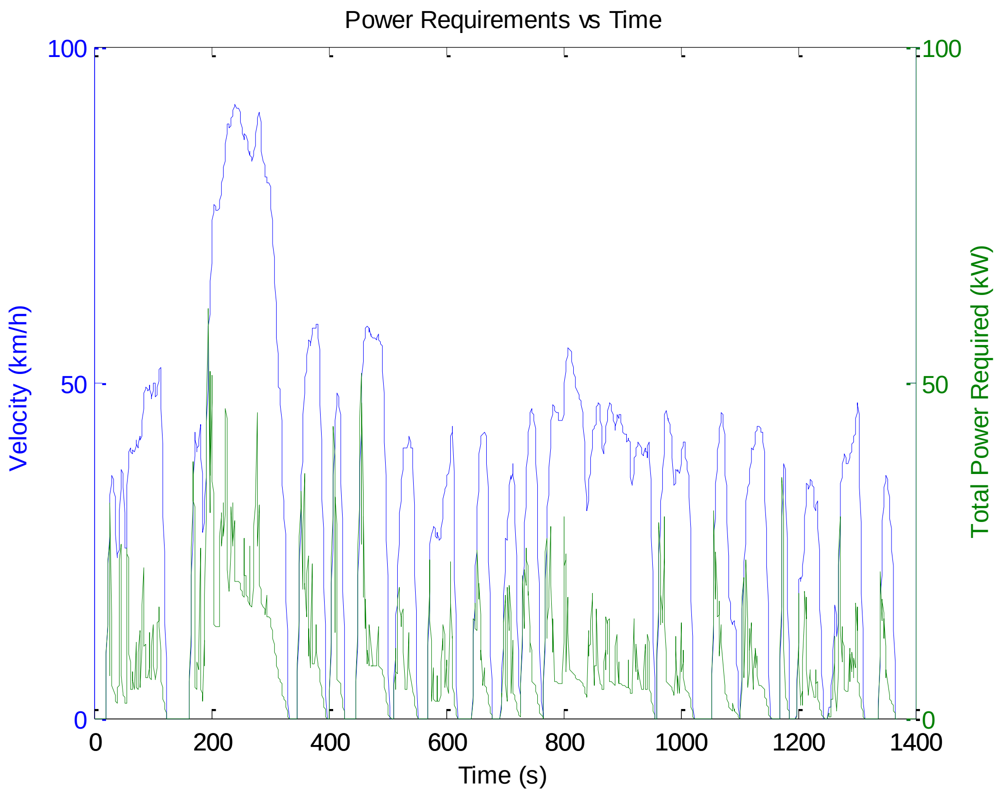

Figure 39 - Power Requirements vs. Time

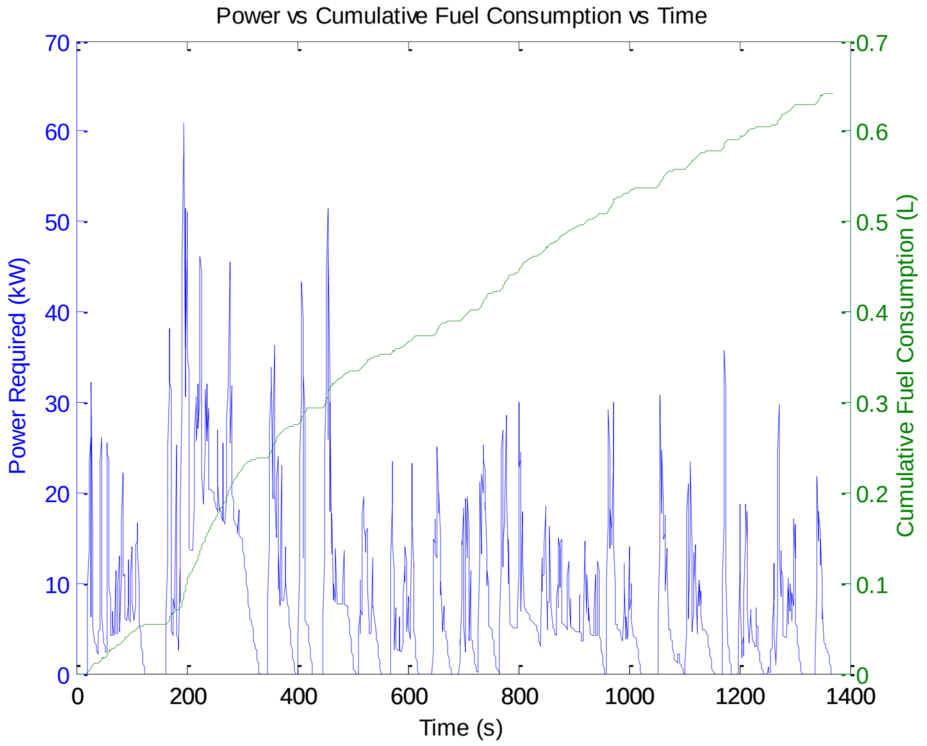

Figure 40 – Power and Fuel Consumption vs. Time

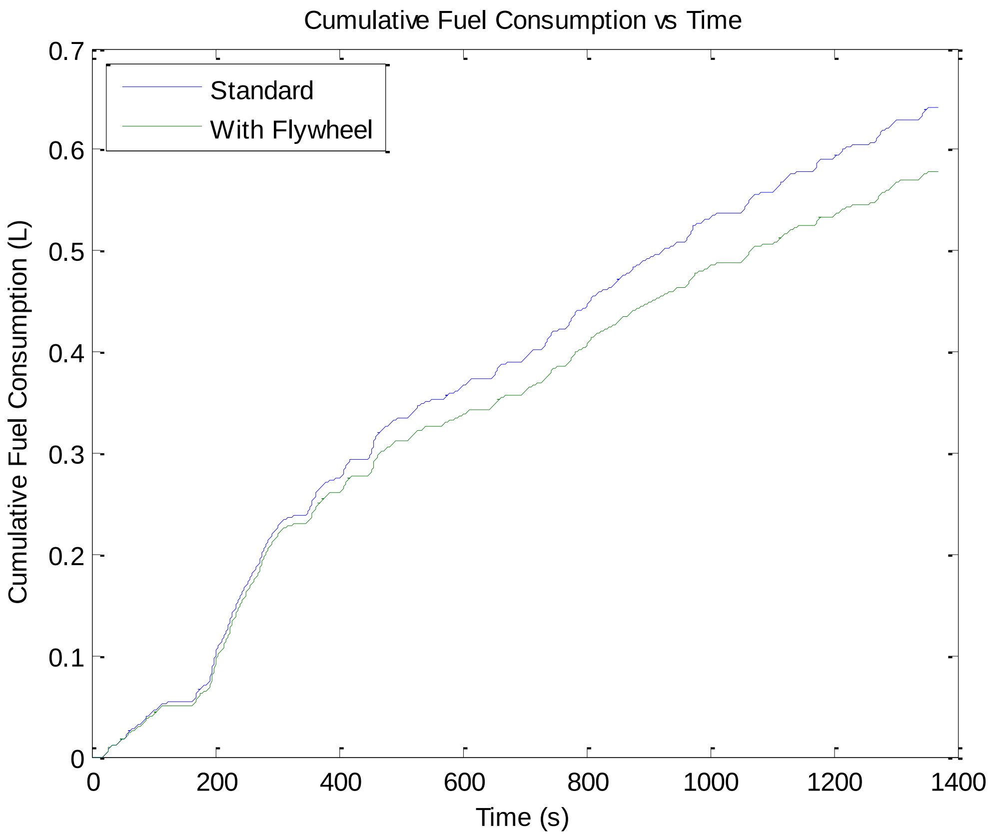

Figure 41 - Cumulative Fuel Consumption over Drive Cycle

4.2.2.1 Diesel Engine

The initial scope of the dissertation was to focus on petrol powered vehicles exclusively, as it was assumed that they could benefit the most from a fuel economy improvement. For comparison, the Matlab script was run for a diesel engine as well. The trends followed that of a petrol vehicle, although the final results were quite different and can be seen in Table 8.

Assumptions

- Energy density=48MJ/kg
- $\rho_{diesel} = 832L/m^3$
- Thermal Efficiency of Common Rail Direct Injection Diesel Engine = 0.45 (The Japan Society of Mechanical Engineers, 2008; U.S. Department of Energy, 2003)

Table 8 – Matlab Results (Diesel Engine)

Total available energy (kJ)	5623.79
Recovered energy (kJ)	2549.41
Recovery Rate (%)	45.33
System Utilisation (%)	53.39
Fuel Saving (%)	9.86
Cost Saving per 100km @$1.35/litre	$0.71

Diesel engines are far more efficient and this will greatly improve the effectiveness of the system as less energy is wasted as heat and more goes towards actually accelerating the vehicle to begin with. This energy is recoverable and therefore the overall efficiency of the flywheel system is greater. As the vehicle uses less fuel to begin with though, the cost saving under the same circumstances is still less than that of a petrol.

4.2.2.1 Other Scenarios

The Matlab code was run for other driving scenarios to quantify the benefit of the system under different conditions. The urban driving cycle was used with the maximum GVM of the vehicle (4500kg). The results from this scenario are shown in Table 9 for a petrol engine and Table 10 for diesel.

Table 9 – Statistics from full GVM scenario for petrol engine

Total available energy (kJ)	9442.93
Recovered energy (kJ)	4563.46
Recovery Rate (%)	48.33
System Utilisation (%)	90.48
Fuel Saving (%)	7.6
Cost Saving per 100km @$1.30/litre	$1.64

Table 10 - Statistics from full GVM scenario for diesel engine

Total available energy (kJ)	9442.93
Recovered energy (kJ)	4563.46
Recovery Rate (%)	48.33
System Utilisation (%)	90.48
Fuel Saving (%)	11.35
Cost Saving per 100km @$1.35/litre	$1.32

5 Discussion

5.1Cost Benefit Analysis

A cost benefit analysis was required in order to quantify the benefit of the system and develop an appropriate payback period. A parts list was developed in excel from the Creo model and was used to sum the costs of each individual component. The task of calculating a cost is not a simple one, as costs vary quite largely with production quantities and order quantities. There is also often a large disparity between price quotes from manufacturers. The quote developed was for a 'once off' system and obviously large production numbers would make the system considerably cheaper to produce. See Table 16 for the cost breakdown.

The bearing part numbers used were from the SKF catalogue, and pricing was available through (NationSkander California Corp., 2015). The gears used in the analysis were from the KHK gearing catalogue (KHK Gearing, 2012) and pricing was obtained for each of the gears.

The current price of the materials required for the rest of the machined or cast components was obtained from CustomPartNet (2009). The prices are shown in Table 11.

Table 11 – Material Cost

Material	Cost/lb ($US)	Cost/kg ($US)	Cost/m^3 ($US)	Cost/m^3 ($AU)
1040 Steel	0.36	0.163296	4542.895	6583.905
1018 Steel	0.34	0.154224	4290.512	6218.133
1095 Steel	1.02	0.462672	12871.54	18654.4
Iron 80-55-06	1.08	0.489888	13628.68	19751.72

Estimations for the cost of manufacturing are difficult to calculate accurately as there are tooling costs, overheads and other factors that are unable to de accurately predicted without obtaining an official quote from a machinist. The overall figure will vary significantly with the production quantity. CustomPartNet (2009) uses quite detailed information about the manufacturing process and tooling to develop rough quotes. This was used to calculate the cost per shaft.

The various housings for components were given rough labour time allocation varying with complexity and size to give representative costs. All housings need to be cast and machined to fine tolerances for bearing and seal fitment. Rough costs were added to the spreadsheet.

As has been discussed already, there is no particular transmission commercially available able to fulfil the required role in this system. Comparisons were drawn between existing vehicle CVTs and the required CVT. The Nissan REOF10A transmission used in some recent vehicle models can be obtained for around $1300. The torque requirements for the regenerative braking system are similar. Pre and post gearing have already been accounted for in the cost analysis, but there is the high probability that another source of gearing will be required within the CVT system to reach the incredibly high ratio requirements.

There is a severe lack of data available for the manufacturing cost of flywheels from various materials. Assumptions were made based on the available literature. The flywheel cost was obtained using data from a study by West et al. (2013). They concluded that the fabrication cost of a flywheel from mild steel was 4.5 times that of its material cost. This gave a fabrication cost of $76.50.

As can be seen in Table 16, the overall cost of the system is $2850.

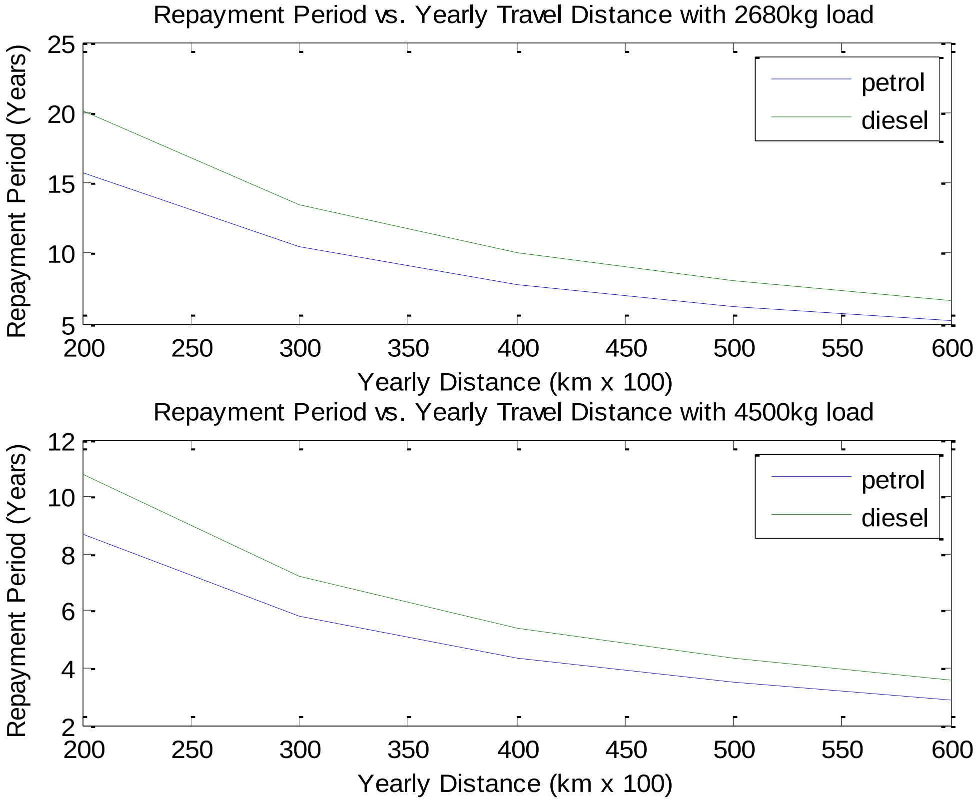

Figure 42 – Repayment period for different scenarios

It can be seen that under the original design circumstances, if 40000km were travelled per year (770km per week), the repayment period would be 8 years. If 60000km were travelled per year (1150km per week), the period would be 5 and a half years. Looking at the fully laden theoretical case, the repayment period for 40000km a year would be 4 and a half years and for 60000 it would be just over 3 years.

It can be seen that the system shows potential for high yearly travel distances and high load cases, as the literature initially suggested. It can be seen in Table 9, that the system only becomes fully utilised on level terrain, when the vehicle is fully laden. The system would not be suitable for light vehicles that do not cover much distance in a year.

5.2 Limitations of Analysis

5.2.1 CVT Functionality

The assumption was made that it will be possible for a CVT to have a range from 8.5:1 to 1:1. Upon further research it was discovered that at this point in time, this is only possible if a 2 speed gearbox is utilised in series with the CVT. This can be done using planetary gears to give a relatively cheap solution. Currently MVB CVTs commercially available are only capable of roughly 6:1. The addition of another stage of variable gearing would ensure that the possible ratio is twice that of the original system. Ideally, a transmission would be developed that does not require this extra stage of gearing.

5.2.2 Dynamic Analysis

The dynamic analysis of the vehicle was quite limited due to time and scope constraints. Accurate braking saturation points were not calculated and it was instead assumed to be a simple factor for calculations. More detailed vehicle specifications would lead to a more accurate dynamic analysis. It was very difficult to develop an accurate capacity requirement as the gradient of road is not taken into account for drive cycles. The assumption that a larger system capacity is required than the drive cycle itself predicts is necessary in practice. The literature made the general statement that flywheel gyroscopic effects have been shown not to effect vehicle handling adversely, and this was taken as a true. However more analysis is necessary to determine if this is the definitely the case. Before any content of the dissertation was taken any further, a more accurate dynamic analysis would be necessary. Dynamic analysis of the flywheel is also required in order to verify the integrity of the containment solution. The shafts and flywheel must remain perfectly balanced or vibrations will cause serious issues for the system.

The shock that the differential experiences would be transferred directly to the flywheel system. In depth analysis would be required to determine the forces the differential experiences and the effect that this has on the components and their lifespan. Currently the analysis just made an assumption based on data from Murphy (1997), that 13 gees is the maximum acceleration experienced by a differential on a similar under everyday use in testing.

5.2.3 Vehicle Assumptions

In the Matlab simulation of the fuel savings using drive cycles, many assumptions were made. These were necessary due to time constraints to gain reasonable results. The efficiencies of engines, drag coefficients and rolling friction coefficients change with respect to many variables that are not known at this stage. The drag torque losses of the flywheel and flywheel gearing were also largely due to assumptions. Accurate drag losses would require testing to give an accurate result.

5.3Retrofitting

The design presented has components positioned in such a way that they make use of space that would otherwise go to waste around the differential. The point of attachment must be the differential as any other part of the vehicle would not move in the same way and power take off would not be possible. By effectively ‘wrapping’ the components around the diff, the moments have been reduced and the system should balance out more effectively than if all the components had have been placed on one side.

The situation modelled here was for a representative vehicle and the assumption was made that it would be suitable across multiple platforms with similar chassis, differential and tailshaft setups. Some changes may have to be made to components such as the differential pinion shaft, the driveshaft length and the flywheel shaft length to fit different vehicle platforms. In some cases on utilities, the spare tyre may have to be moved back further or to another location to facilitate the addition of the CVT.

The drawings assume a vehicle with a solid rear differential. Vehicles with IRS would need a different layout of components, but the main design should still hold providing the components are able to be arranged in such a way that they do not impeded the motion of each driveshaft’s movement due to suspension. Vehicles with IRS should also benefit from lower shock loads, due to the diff being stationary with respect to the chassis.

5.4 Barriers to Implementation

As with any new design in amongst a field of tried and tested established design, there are significant barriers to implementation. The economic state has a definite impact on the success of a design. If the public are unwilling to invest due to economic factors, such as was the case for the Permodrive system, there will be no market for the product. People need to be willing to accept a high initial outlay in order to see cost benefits in the long run. The safety issues surrounding flywheel systems could also be an issue. There have been cases overseas of injuries caused by flywheels piercing containments and this creates a negative perception surrounding the product. The potential catastrophic consequences can be enough reason for people to steer clear of the technology. The limited space around the differential could also be an obstacle on some vehicles.

5.5Comparison to existing technologies

When developing a technology it makes sense to compare to existing technologies as benchmarks. In comparison to existing flywheel technologies, the Volvo and Torotrak system for sedans was developed at a cost of $2000 and it was quoted as having a 5 year repayment period and fuel savings of 25%. After modelling the fuel savings of the system it is apparent that in order to achieve these figures, the regenerative system must be active over a much wider range of vehicle velocities. The drag of the Volvo system could also have been much lower. Even then, it is hard to comprehend how the manufacturer's efficiency figures could be so high. It is clear that if these figures from manufacturers are reliable, that a scaled version of these systems would probably be more suitable for the light commercial application than the product developed here. Light commercial vehicles, with their substantially higher mass than a sedan, should have gained much more benefit from the technology than smaller vehicles.

The Eaton HLA system had a much smaller capacity of 380kJ and had a quoted fuel economy improvement of 25%, a cost of $2000 and a mass of 204kg. The system developed is lighter and has a higher capacity, but does not boast such high economy improvements overall.

Theoretically the developed system is still more effective than the pneumatic system, which had a fuel economy improvement of 5-10%, was quite expensive and only on very large vehicles such as trucks.

It is worth noting that the costs to the large companies would be significantly less for similar components due to buying power and manufacturing capabilities and this will influence the comparison between results.

Once comparisons are mode between electrical and flywheel systems, it becomes very easy to see how electrical systems could be more effective if the battery technology improved due to the much lower weight and cost of power transfer components.

5.6 Improvements

The system developed is a basic design with high safety factors. The design could be optimised and lighter materials could be used to make a more optimal design. The CVT is the main determinant of the velocities that can be used to capture braking energy and advances in this technology would lead to more energy being captured and the possibility of the technology being used for highway cycles as well would be presented. This would enable braking from higher speeds and for the system to still remain functional at speeds below 10km/h. If control systems and fitment options permitted attachment between the vehicle engine and the gearbox, the ratio requirement would be significantly less, less supplementary gears would be required and the range of effective velocities would be higher. The concept of a flywheel enclosure that does not require a vacuum pump, such as the one presented by Ricardo shows great potential and would reduce power losses due to the vacuum pump, seals and gearing. These parasitic losses cause a lot of energy to be wasted.

6 Conclusions

6.1 Findings

The dissertation aimed to analyse the potential role of Kinetic Energy Recovery Systems and their potential for contemporary Internal Combustion Engine powered vehicles. A system was to be developed aiming to be easily implemented and cost effective. The flywheel system in its current state weighs approximately 123.8kg. The dissertation found that a petrol powered vehicle of mass 2680kg mass would save $0.91 per 100km (6.9% saving). If the vehicle were fully laden, the fuel saving would be $1.64 per 100km (7.6% saving). The total cost of the system was found to be $2680. The repayment period ranged from 5-8years to a best case scenario of 3-4 years. When the flywheel is charged, it is capable of a 36kW boost and delivers torque straight to the wheels.

6.2 Significance of Findings

The dissertation found that a flywheel based regenerative braking solution for light commercial vehicles is a commercially viable technology. The final solution was able to be fitted to multiple vehicle platforms and with more research, the system could be made lighter and more efficient. The models presented are a representation of the potential fitment to vehicles. The calculations for the system were used to show that it is possible to design the required components for a functioning system. More research would be required before the model is able to be produced as a prototype. The Matlab scripts used for the drive cycle modelling generate figures that show how the system is utilised and the system could be optimised for different drive cycles.

6.3 Overall Feasibility of System as a retrofit

The dissertation established that the technology itself definitely has potential, however it is the opinion of the author that more appropriate systems appear to exist, which could be scaled for the purpose. Whilst this research presents a solution in the light commercial sector that doesn't presently exist, it does appear that the results obtained were not as effective as those obtained from manufacturers aiming for the smaller vehicle market. The retrofit options developed, whilst suitable for multiple platforms, will require modification of existing vehicle parts and production costs would be high given the fact that some parts would be completely different dimensions per vehicle. There is potential in the fitment option as space was able to be used that would otherwise have been wasted. Some vehicles would not have to relocate any existing vehicle hardware and this is important from a retrofit stance. Having the system as a parallel system still makes fitment much easier than if it was a series system and this is still deemed to be an appropriate decision. As it is, the system would still be suitable for delivery vehicles or other heavy vehicles subject to large amounts of frequent stop-start driving.

6.4Further Work

There are several areas for further work on this project.

- The quantification of the environmental benefits of using the design
- The in depth design of the transmission system used for the flywheel. This is a major design point and due to time constraints, assumptions based on the specifications and capabilities of available units were made to quantify the value of the system.
- Retrofit designs for other types of vehicles
- Refining of the Matlab scripts for simulation of the system to ensure a more accurate result
    - Air Resistance in flywheel enclosure
    - Effect of idle on system effectiveness
    - Incorporating the front/rear bias/saturation point of braking system
- Finite Element Analysis on the system to ensure strength and mitigate failures
- Life Cycle Analysis of the components to ensure longevity of the system

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8 Appendices

8.1Project Specification

FOR: **Steven CARLIN**

Topic: An analysis of Kinetic Energy Recovery Systems (KERS) and their potential for contemporary ICE powered vehicles

Supervisor: Ray Malpress

Project Aim: To explore the potential role of regenerative braking in the future and if possible, develop a feasible system for ICE powered vehicles able to be easily implemented and cost effective.

Programme: (18/03/15)

1) Literature review of current mechanical KERS technologies: their applications, limitations and successes in the past. As technology advances and the human race looks for alternative energy sources, the future of ICE powered vehicles isn't a certainty. Trends point towards the potential of Electric Vehicles or Hybrid Electric Vehicles if electrical storage technology advances enough to facilitate this. However for the foreseeable future, ICE powered vehicles will still play a prominent role and any potential fuel saving solution will be of use if it is feasible, cost effective and easy to implement. The review should centre on these factors.
2) Choose a technology with the most potential for commercial vehicles in Australia and analyse its feasibility and potential benefit from a retrofit stance. Mechanical KERS are generally limited in their use due to the added weight of most systems. Vehicles often with a large payload (and therefore more kinetic energy), stopping and starting frequently, offset this drawback and are therefore more appropriately paired with the technology. These types of vehicles are not as suited to current electronic KERS as electrical systems are not as capable of storing high levels of energy in short periods of time, leaving a window for these technologies. This analysis will include deciding which component of the engine or drivetrain would be most effective to recover energy from and how the system could accommodate different vehicles.
3) Design the chosen system to be retrofitted for the application.
4) Model potential fuel cost savings based on simulation using Matlab software.
5) Assess manufacturing cost of designed system.
6) Using the cost of the system and the potential fuel savings, calculate a repayment period and comment on the overall feasibility of using this technology.

As Time permits:

7) Research and report on potential barriers to implementation and report on overall potential compared to other existing technologies.
8) Quantify the environmental benefits of using the designed unit using Matlab.

Table 12 - (Thoolen, 1993) – Properties of Anisotropic Materials

Material data[1a] / Uni-directional composites (60% fibre vol.)	Mass density (kg/m^3)	Young's modulus (GPa)		Tensile strength (MPa)		Allowable stress[3] (MPa)		Energy Density (//;K=0.5)	
		//[2]	⊥[2]	//	⊥	//	⊥	volume (Wh/1)	mass (Wh/kg)
E-glass	2000	45	10	1100	35	220	10	30	15
High modulus aramid	1400	75	5.5	1400	27	700	8	97	69
High strength carbon	1550	130	10	1400 2000[1b]	50	1100 1600	15	222	143
High modulus carbon	1650	220	6	1100 1728[1b]	40	850 1380	12	192	116

[1a] According to Nijhof (1983), [4.9]; [1b] According to Courtaulds (1989), [4.10].
[2] // = longitudinal; ⊥ = tranverse.
[3] Calculated from tensile strength: // using fatigue reduction factor for 10^7 cycles according to [4.11]; ⊥ using creep reduction factor (0.3), according to [4.6] and [4.7].

Table 13 - (Thoolen, 1993) – Properties of Isotropic Materials

Material data / Materials	Mass density (kg/m^3)	Young's modulus (GPa)	Tensile strength (MPa)	Fatigue strength (10^7 cycles) (MPa)	Energy density (K=1)	
					volume (Wh/1)	mass (Wh/kg)
Ferro 360	7800	210	360	120	33.3	4.3
Maraging steel 18 Ni-400	8000	189.6	2337	765	212.5	26.5
Wrought steel 26 NiCrMoV 14 5	7800	210	950	550	152.8	19.6
Titanium 6AL-6V	4500	115	1186	662	183.9	40.8
Aluminium 7075-T6	2800	71.8	529	173	48	17.2

Table 14 – Comparison of flywheel energy storage (Ter-Gazarian, 1994)

Table 5.2 Comparison of flywheel energy storage

Material	Design stress 10^6N/m	Density 10^3 kg/m^3	Useful energy 10^3 J/kg	Mass of the flywheel 10^3 kg	Relative cost for material and fabrication p.u./J
Wood birch	30	0·55	21	1720	1·0
Mild steel	300	7·80	29·5	1220	1·11
E-glass 60% fibre/epoxy	250	1·90	50·4	713	0·523
S-glass 60% fibre/epoxy	350	1·90	70·5	509	0·492
Maraging steel	900	8·00	86·4	417	2·18
Titanium alloy	650	4·50	110·8	325	6·98
Carbon 60% fibre/epoxy	750	1·55	185·4	194	0·34
Kevlar 60% fibre/epoxy	1000	1·4	274·3	131	0·26

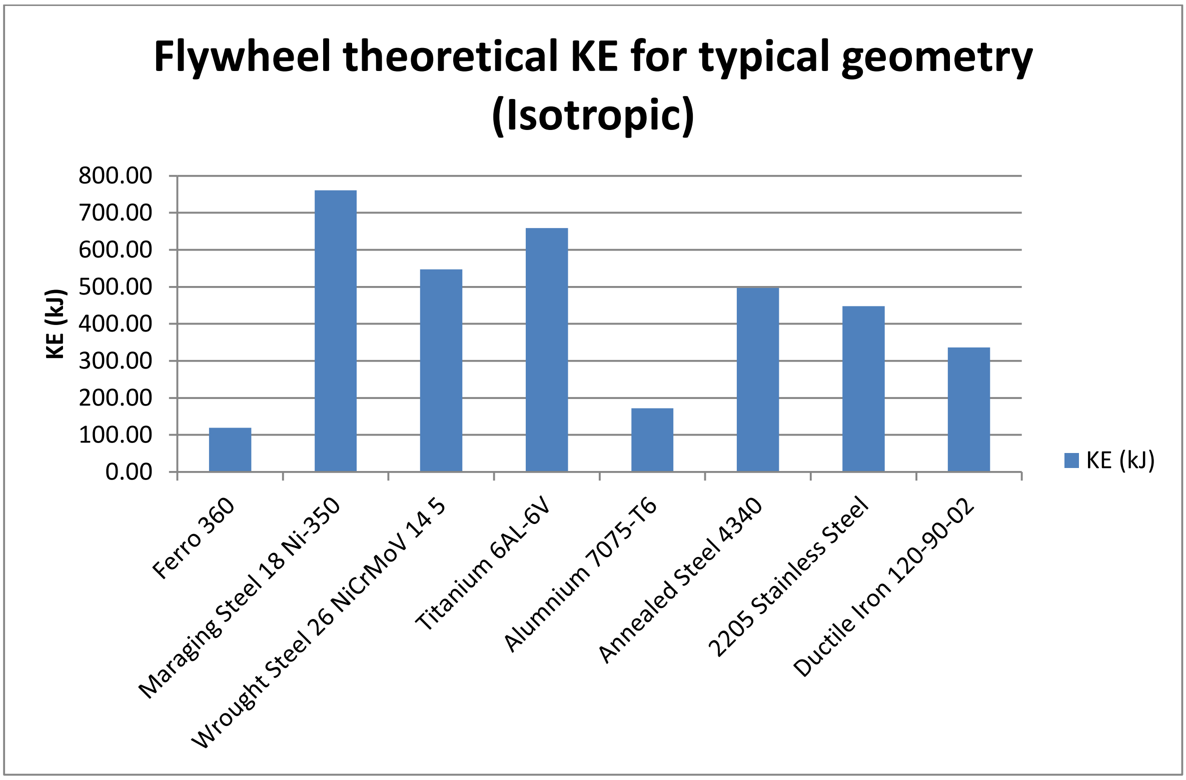


Figure 43 – Flywheel theoretical KE for typical geometry (Isotropic Materials)

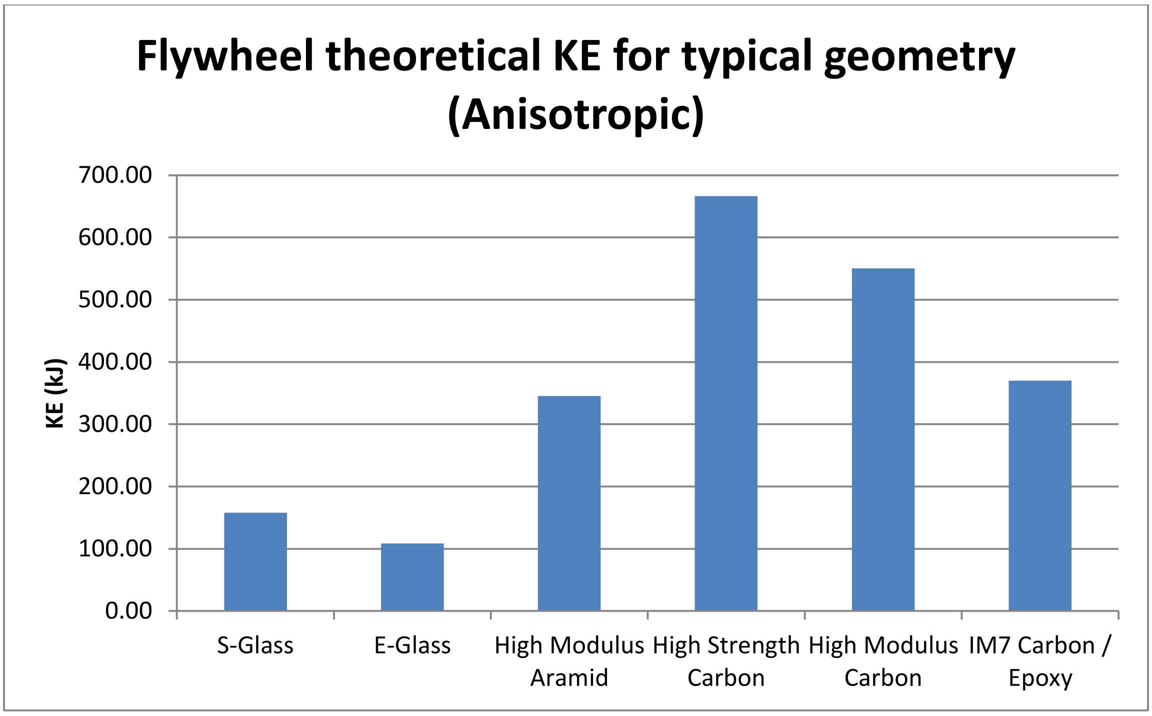


Figure 44 – Flywheel theoretical KE for typical geometry (Anisotropic Materials)

Table 15 – Comparative cost of metal in 2005 (Beardmore, 2012)

Material	Density	Cost/tonne	Relative	Cost /m^3	Relative
	kg/m3	�/tonne	�/tonne	�/m^3	�/m^3
Carbon Steel	7820	550	1	4301	1,0
Alloy Steels	7820	830	1,51	6490,6	1,5
Cast Iron	7225	830	1,51	5996,75	1,4
Stainless Steel	7780	4450	8,1	34 621	8,0
Aluminium/alloys	2700	2220	4,0	5994	1,4
Copper /Alloys	8900	5550	10,1	49 395	11,5
Zinc alloys	7100	2220	4,0	15 762	3,7
Magnesium /alloys	1800	4000	7,3	7200	1,8
Titanium /alloys	4500	17 000	30,9	76 500	17,4
Nickel alloys	8900	18 000	32,7	160 200	36,8

Table 16 – Cost Analysis

Component	Volume (mm^3)	Material	Cost m^3	Manufac. Cost	Quant.	Item Cost	Total	Source
Diff Housing	1.49E+06	1018 Steel	6218.1	100	1		$109.3	
6007 Bearing					2	4.5	$9.0	VBX
Pinion Shaft	3.17E+05	1040 Cold Rolled	6583.9	20	1		$22.1	
Gear 1 Shaft	4.72E+05	1040 Cold Rolled	4542.9	15	1		$17.1	
Flywheel	1.34E+06	4340 Steel	6583.9	76.5	1		$85.3	
40558 Shaft Seal					1	10	$10.0	VBX
Pinion 1					1	99.85	$99.85	RS Components
Gear 1					1	154.55	$143.55	RS Components
6208 Bearing					1	9.0	$9.0	VBX
Bearing Cap	5.50E+03	1018 Steel	6218.1	10	3		$30.1	
33210 Bearing					2	17.9	$35.8	VBX
RMS12 Bearing					1	13.8	$13.8	VBX
Flywheel Housing	5.97E+06	Ductile Iron	19751.7	40	1		$157.9	
33457 Seal					1	20	$20.0	VBX
375510 Seal					2	15	$30.0	VBX
61806 Bearing					1	9.0	$9.0	VBX
30528 Seal					2	10	$20.0	VBX
Flywheel Housing Pt 2	3.06E+06	Ductile Iron	19751.7	20	1		$80.4	
61905 Bearing					1	6.9	$6.9	VBX
6006 Bearing					3	9.7	$29.0	VBX
Flywheel Shaft	2.91E+05	1095 Normalised Steel	18654.4	15	1		$20.4	
Flywheel Pinion					1	48.8	$48.8	RS Components
Flywheel Gear					1	90	$90.0	RS Components
Second Gearbox Housing	7.44E+05	1018 Steel	6218.1	50	1		$54.6	
Single Plate Clutch					1	105	$105.0	
Clutch Enclosure	2.73E+05	1018 Steel	6218.1	50	1		$51.7	

Flywheel Gearbox Shaft 2	6.45E+04	1040 Cold Rolled	6583.9	13	1		$13.4	
1207ETN9 Bearing					2	13.8	$27.6	VBX
RLS13 Bearing					1	34.5	$34.5	VBX
33457 Seal					1	15	$15.0	VBX
80W90 Gear oil					6	10	$60.0	Repco
ATF DIII					1	10	$10.0	Repco
CVT					1	1300	$1,300.0	Alibaba
Vacuum Pump					1	80	$80.0	Ebay
							$0.0	
Summation					46		$2,849.2	

8.2 Matlab Scripts

8.2.1 Driving Cycle Design Validation

```
%% Driving Cycle Analysis
% Steven Carlin
% 0061033029
% This script uses drive cycles to simulate the benefit of a Flywheel
% Regenerative Braking System implimented on a commercial vehicle
% Change efficiency, fuel and fuel to alter fuel type

clc
clear all
close all

% input driving cycle - URBAN FTP-75
data1=dlmread('Urban_Dynamometer_Driving_Schedule.txt')
urbancycle=data1(:,2)';
t=data1(:,1)';

% input highway cycle
data2=dlmread('Highway_Fuel_Economy_Driving_Schedule.txt')
highwaycycle=data2(:,2)';
t2=data2(:,1)';

% convert drive cycle to km/h
urbancyclekm=urbancycle*1.609;
highwaycyclekm=highwaycycle*1.609;

% plot driving cycle 1
q=1
figure(q)
q=q+1
plot(t,urbancyclekm), xlabel 'time (s)', ylabel 'km/h', title 'Urban
Driving Cycle (FTP-75/ADR37)'
figure(q)
q=q+1

% plot driving cycle 2
plot(t2,highwaycyclekm), xlabel 'seconds', ylabel 'km/h', title
'Highway Driving Cycle'

% convert to m/s
urbancyclems=0.44704*urbancycle;
highwaycyclems=0.44704*highwaycycle;

% generate acceleration matrix by calculating difference between
velocities
differenceurbanms=[0,diff(urbancyclems)];
differencehighwayms=[0,diff(highwaycyclems)];

% maximum desceleration in m/s
maxdesc_urban=min(differenceurbanms);
maxdesc_highway=min(differencehighwayms);

% Average desceleration
P=differenceurbanms(differenceurbanms<0);
avgdesc=mean(P);
```

```
% Power Calculations - AVG
m=2680;
KE3=0.5*m*(60/3.6)^2;
KE4=0.5*m*((60-avgdesc))^2;
avgpower=(KE3-KE4)/1;


%% Theoretical Kinetic Energy Calculations

KEfactor=0.5*m/1000;
KE=(KEfactor.*((urbancyclems).^2));

% Plot Kinetic Energy for Urban Cycle
figure(q)
q=q+1;
plot(t,KE), xlabel 'time (s)', ylabel 'Kinetic Energy (kJ)', title
'Urban Driving Cycle Kinetic Energy'

% Seperate Braking events
KEchange=[1,diff(KE)];
for pp=1:length(KEchange)
    if differenceurbanms(pp)<0
        % Treat the change in energy as +ve as energy is to be
recovered
        KEchangefinal(pp)=-KEchange(pp);
        % Allocate negative entries to a seperate matrix
        KEchangenegative(pp)=KEchange(pp);

    end
end

% Plot instantaneous Power
figure(q)
q=q+1;
plot(t,KEchange), xlabel 'time (s)', title 'Instantaneous Power',
ylabel 'Power (kW)'

% Calculate Max Power Available from matrix
maxpower=min(KEchange);

%% Work Calculations
% Vehicle Variables
Urr=0.011;                  % coefficicent of rolling friction
g=9.81;                     % gravity (m/s^2)
d=urbancyclems./1;          % distance travelled (m)
a=differenceurbanms;        % vehicle acceleration
Cd=0.40;                    % drag coefficient    %Toyota Truck
(Wikipedia)
A=1.76*1.8;                 % vehicle cross-section (m^2)
(Redbook,2015)
rho_air=1.27;               % density of air (kg/m^3)
%vefficiency=0.30;          % gasoline - see references
vefficiency=0.45            % common rail diesel - see references


% Velocity
% Work to overcome air resistance
w_ar1=d.*urbancyclems.^2.*((1/2)*rho_air*A*Cd);
```

```
% Work to overcome rolling resistance
w_rr=d.*(Urr*m*g);

% Work done to cruise
work_v=w_ar1+w_rr;

% Acceleration
% set negative accelerations to zero
differenceurbanms(differenceurbanms<0)=0;

% Work to accelerate     f=ma
w_acc=differenceurbanms.*(m*d);

% Total Work
work_a=w_acc;

% Power
% Energy Input = Work/vefficiency
energy_input=(work_v+work_a)/vefficiency;

% Power
Power_velocity=work_v*10^-3;
Power_acc=work_a*10^-3;
totalpower=Power_velocity+Power_acc;

% Fuel Consumption
% Calculate Fuel usage
%fuel=1000*(energy_input/(737*44e6));     %gasoline
fuel=1000*(energy_input/(832*48e6));     %diesel

% Calculate Cumulate Sum of fuel usage
cumfuel=cumsum(fuel);


%% plots
figure(q)
q=q+1;

% acceleration and velocity power Comparison
[y,line1,line2]=plotyy(t,Power_velocity,t,Power_acc);
title('Power Requirements vs Time')
xlabel('Time (s)')
ylabel(y(1), 'Power to maintain velocity (kW)')
ylabel(y(2), 'Power to accelerate vehicle (kW)')

% Total Power Requirements vs. Time
figure(q)
q=q+1;
[y,line1,line2]=plotyy(t,urbancyclekm,t,totalpower);
title('Power Requirements vs Time')
xlabel('Time (s)')
ylabel(y(1),'Velocity (km/h)')
ylabel(y(2),'Total Power Required (kW)')

% Cumulative Fuel Consumption
figure(q)
q=q+1;
```

```
[y,line1,line2]=plotyy(t,totalpower,t,cumfuel);
title('Power and Cumulative Fuel Consumption vs Time')
xlabel('Time (s)')
ylabel(y(1),'Power Required (kW)')
ylabel(y(2),'Cumulative Fuel Consumption (L)')


%% Kinetic Energy Recovery Simulation
efficiency=0.8;       %Efficiency of Power Transmission
capacity = 511;       %kJ
flywheel_losses=0.4   %kW

KEavailable=KE;
% initiate storage and energy used matricies
storage=0;
eused=0;

for k=1:length(KEavailable)-1
    % check within speed range of the gearbox
    if urbancyclekm(k)> 10 && urbancyclekm(k)<60
        % can incorporate another if statement for desceleration
limits of
        % design
        % desceleration event
        if (KEavailable(k+1)-KEavailable(k))<0
            % add energy to storage
            storage=storage+abs((KEavailable(k+1)-
KEavailable(k)))*efficiency;
               e_saved(k)=0;
            % apply cap on capacity of system
            if storage>capacity
                storage=capacity;

            end
            % acceleration event - utilise storage energy
    elseif  (KEavailable(k+1)-KEavailable(k))>0
            % if storage is greater than the KE change - use available
            % energy
            if (KEavailable(k+1)-KEavailable(k))<storage
                storage=storage-(KEavailable(k+1)-KEavailable(k));
                e_saved(k)=(efficiency*(KEavailable(k+1)-
KEavailable(k)));    % incorporate efficiency
                eused=eused+e_saved(k);
                %store energy saving in matrix
                % if storage is less than the KE change - only use
remaining
                % energy available
                else
                    e_saved(k)=(efficiency*storage);        %
incorporate efficiency
                    eused=eused+e_saved(k);
                    storage=0
        end
        end
    else        e_saved(k)=0;
    end
    %incorporate flywheel system losses
    storage=storage-flywheel_losses
    % keep matrix values
        storagematrix(k)=storage;
```

```
end

% Corrections for forward difference method
storagematrix(1370)=0;
e_saved(1370)=0;      %e_saved is in J

%% Comparison of data
% subtract flywheel energy from vehicle energy requirements
energy_input1=energy_input-(e_saved*10^3);
energy_input1(energy_input1<0)=0

% Fuel Consumption
% Calculate Fuel usage
%fuel1=1000*(energy_input1/(737*44e6));     %gasoline
fuel1=1000*(energy_input1/(832*48e6));     %diesel

% Cumulate Sum of fuel usage
cumfuel1=cumsum(fuel1);

% Calculate total distance travelled in drive cycle
total_distance=sum(d);   %(m)

% Calculate fuel usage using pro rata distance value
fuel_use_100km_standard=(100e3/total_distance)*max(cumfuel);
fuel_use_100km_flywheel=(100e3/total_distance)*max(cumfuel1);

% saving per 100km
saving=fuel_use_100km_standard-fuel_use_100km_flywheel;
saving_percent=saving/fuel_use_100km_standard*100;

% Cumulative Fuel Consumption Comparison
figure(q)
q=q+1;
plot(t,cumfuel,t,cumfuel1)
title('Cumulative Fuel Consumption vs Time')
xlabel('Time (s)')
ylabel('Cumulative Fuel Consumption (L)')
legend('Standard','With Flywheel' },'Location','northwest')

% calculate statistics
sumenergy=abs(sum(KEchangenegative));
recoveryrate=eused/sumenergy*100;
maxutilisation=max(storagematrix)/capacity*100;
fuelprice=1.40    %dollars/l
saving100km=saving*fuelprice
% plot storage usage
figure(q)
q=q+1;
plot(1:length(storagematrix),storagematrix,1:length(storagematrix),KE)
xlabel 'time (s)', title 'Real Time Energy Storage of KERS on FTP-75
Cycle', ylabel 'Energy (kJ)',  legend('Energy Storage','KE of
vehicle'},'Location','northeast')
clc

% Plot real time energy usage vs time
figure(q)
q=q+1;
plot(t,energy_input/1000,t,energy_input1/1000)
```

```
title('Energy Requirements vs. Time')
xlabel('Time (s)')
ylabel('Energy Required kJ')
legend('Standard','With Flywheel'},'Location','northwest')

% Write simulation results to the workspace
disp(['==================================SYSTEM
STATISTICS====================================='])
disp(['The vehicle mass is ',num2str(m),' kg'])
disp(['The system efficiency is ',num2str(efficiency),' and the
capacity is ',num2str(capacity),'kJ'])
disp(['The total amount of energy to be recovered is
',num2str(sumenergy),'kJ and recovered amount is
',num2str(eused),'kJ'])
disp(['This results in a recovery rate of ',num2str(recoveryrate),'%
of available Kinetic Energy'])
disp(['The maximum system utilisation is
',num2str(maxutilisation),'%'])
disp(['The system saves ',num2str(saving),' Litres/100km'])
disp(['This is a saving of ',num2str(saving_percent),' %'])
disp(['This equates to $',num2str(saving100km),' per 100km'])
```

8.2.2 Transmission Design

```
%% Transmission Calculations
% Steven Carlin
% 0061033029
% This code is used to model the flywheel transmission and give
relevant
% data about the angular velocity and Torque at design points

clc
clear all
close all

k=1
for k=1:5
wheelspeed=[60:-1:10]; %km/h      % DESCELERATING
diffratio=4.3;          %for respresentative vehicle
wheelsize=195;
rimsize=14*25.4;
Diameter=(2*wheelsize+rimsize)/10/100;     %m

w_driveshaft=((wheelspeed*0.2778*2*pi)*diffratio/(Diameter*pi))
%rad/s
interval=length(wheelspeed);

w_max=2043;         %rad/s
w_min=w_max/2;      %rad/s

%set increment for w_flywheel
w_flywheel=linspace(w_min,w_max,interval);

% pregearing and postgearing
step1=1+k*1
x=[3:1:6*1]
step2=2.5
```

```
% Divide w_eta_1 by w_eta_2 to get range of ratios between each end of
CVT
ratio(k,:)=(w_flywheel./step1)./w_driveshaft./step2;

reqratiorange(k)=max(ratio(k,:))
end

%plot results
figure(1)
subplot(2,1,1)
%[Ax,h1,h2]=plotyy(wheelspeed,ratio,wheelspeed,w_driveshaft);
plot(wheelspeed,ratio(1,:),wheelspeed,ratio(2,:),wheelspeed,ratio(3,:)
,wheelspeed,ratio(4,:),wheelspeed,ratio(5,:)), ylabel 'CVT Ratio
(1:x)'
title('Vehicle Speed vs. Flywheel CVT Ratio for different gearing
combinations(Gear_2=2.5)')
xlabel('Vehicle Velocity (km/h)')
legend('gear_1=2', 'gear_1=3', 'gear_1=4', 'gear_1=5',
'gear_1=6','gear_2=2.5'},'Location','northeast')   %% CHANGE BACK
FJKFDSAJKSDJKFADSPL

torquemultiplier=step1*step2*reqratiorange;
k=k+1;

RPM1min=min(w_driveshaft*step1)/(2*pi/60)
RPM1max=max(w_driveshaft*step1)/(2*pi/60)
RPM2min=min(w_flywheel/step2)/(2*pi/60)
RPM2max=max(w_flywheel/step2)/(2*pi/60)

%% Torque Plot

P=50000*0.8*0.9 %(Watts)
for k=1:5
    step1=1+k*1;
gear1(k,:)=w_driveshaft*step1;
gear2=w_flywheel/step2;
Torque1(k,:)=P./gear1(k,:);
Torque2=P./gear2;
k=k+1;
end

subplot(2,1,2)
plot(wheelspeed,Torque1(1,:),wheelspeed,Torque1(2,:),wheelspeed,Torque
1(3,:),wheelspeed,Torque1(4,:),wheelspeed,Torque1(5,:),wheelspeed,Torq
ue2), title 'Vehicle Velocity vs. Torque at Either Side of
Transmission  (Gear_2=2.5)', xlabel 'Vehicle Velocity (km/h)', ylabel
'Torque (Nm)'
legend('gear_1=2', 'gear_1=3', 'gear_1=4', 'gear_1=5',
'gear_1=6','gear_2'},'Location','northeast')

%% Velocity Plot for different Locations
figure(2)


subplot(2,1,1)
plot(wheelspeed,w_driveshaft,wheelspeed,gear1(1,:),wheelspeed,gear2,wh
eelspeed,w_flywheel), title 'Vehicle Velocity vs. Angular Velocity at
```

```
Design Points', xlabel 'Vehicle Velocity (km/h)', ylabel 'Angular
Velocity (rad/s)'
legend('w-driveshaft', 'gear_1=2', 'gear_2=2.5', 'w-
flywheel'},'Location','northwest')


%plot of torque at design points
subplot(2,1,2)
Torque1=P./w_driveshaft;
Torque2=P./gear1(1,:);
Torque3= P./gear2;
Torque4=P./w_flywheel;
plot(wheelspeed,Torque1,wheelspeed,Torque2,wheelspeed,Torque3,wheelspe
ed,Torque4), title 'Vehicle Velocity vs. Torque at Design Points',
xlabel 'Vehicle Velocity (km/h)', ylabel 'Torque (Nm)'
legend('w-driveshaft', 'gear_1=2', 'gear_2=2.5', 'w-
flywheel'},'Location','northeast')
```

8.2.3 Flywheel Variable Optimisation

```
%% flywheel optimisation graphs
% Steven Carlin
% 0061033029
% See excel for material information
% selected AISI4340 Alloy Steel
% Mass calulations taken from Thoolen

clc
clear all
close all
% Fixed Variables
rho = 7800;     %kg/m^3
K = 0.806;      %Truncated conical disc
sigma = 500;   %MPa
specificenergy =(K.*sigma)/rho;
Safety = 1.4
sigma=sigma/Safety
tipspeed = sqrt((sigma*10^6)/rho)


% Define radius variable
r_o = [0.05:0.01:0.2];    %(m)

%define widths
width(1,:)=r_o.*0.2;
width(2,:)=r_o.*0.4;
width(3,:)=r_o.*0.6;
width(4,:)=r_o.*0.8;
width(5,:)=r_o.*1.0;

% Calculate w_max
wmax = tipspeed/(2*pi/60);  %rad/s
% wmin is half of wmax
wmin = 0.5*wmax;
% mass - will be different per width
% assuming r2 is 0.15*r1
for jj=[1:5]
r1=r_o
```

```
r2=0.15*r_o
h=width(jj,:)
m(jj,:)=(1/3).*pi.*(r1.^2+(r1.*r2)+r2.^2).*2.*h.*rho

end

%Inertia
for l=[1:5]
I(l,:)=m(l,:).*r_o.^2
end

%Kinetic Energy Calculations
%Shape 1
for kk=[1:length(r_o)]
KE(1,kk,:) = (0.5*K*I(1,kk)*wmax^2*(1-(wmin^2/wmax^2)))/1000
KE(2,kk,:) = (0.5*K*I(2,kk)*wmax^2*(1-(wmin^2/wmax^2)))/1000
KE(3,kk,:) = (0.5*K*I(3,kk)*wmax^2*(1-(wmin^2/wmax^2)))/1000
KE(4,kk,:) = (0.5*K*I(4,kk)*wmax^2*(1-(wmin^2/wmax^2)))/1000
KE(5,kk,:) = (0.5*K*I(5,kk)*wmax^2*(1-(wmin^2/wmax^2)))/1000

end

%plot results
%x - w/r     y - KE
plot(r_o , KE(1,:,:), r_o,KE(2,:,:), r_o,KE(3,:,:), r_o,KE(4,:,:),
r_o,KE(5,:,:))
xlabel 'Radius (m)', ylabel 'KE Storage (kJ)', title 'Kinetic Energy
for Different Radius (AISI4340)', legend('r/w = 0.2', 'r/w = 0.4','r/w
= 0.6','r/w = 0.8','r/w = 1.0'},'Location','Best')


% plot guideline
hold on
x=[0,max(r_o)]
y=[511,511]
line(x,y)
```

8.2.4 Gear Calculations

```
%% Gear Stress Calculations
% Steven Carlin
% 0061033029
% This scripis used to calculate the required gear size through trial
and
% error
% The contact ratio and Reliability are also checked

clc
clear all
close all

% Pinion Gear
Torque=500;   %Nm     %change for each combination

SF=1.0;
Kv=1.8 ;       %p644
```

```
Ko=1.5;         %p645
Km=1.3 ;        %p645
fi=20;       %pressure angle
R=2;
I= ((sind(fi)*cosd(fi))/2)*R/(R+1);
Cp=2300;  %table 15.4a
sfccli=225000;    %p400  - 10^7 cycles

%input
prompt = 'What is the face width in mm?';
b = input(prompt)/25.4;      %inches
prompt = 'How many teeth?';
N=input(prompt);        %teeth
prompt = 'What is the pitch diameter in mm?';
dp=input(prompt)/25.4;     %inches
P=pi*dp/N;              %teeth per inch
Ftconstant=(Torque/((dp*25.4/1000)/2))*0.224;    %lb


LHS=Cp*((((Ftconstant))*SF*Kv*Ko*Km)/((b*dp*I)))^(1/2);
RHS=sfccli;

%figures for reiteration
V=(300*dp*60)/(12*2);

%Check Fatigue Strength      %j from table
prompt = 'What is the geometry factor,j?';
j=input(prompt);
Cl=1.0;   %bending loads
Cg=1.0;    %P>5
Cs=0.897;   %figure 8.13
kr=0.814;    %99% reliability
kt=1.0;
kms=1.0;
P=N/dp;
Sn=(((Ftconstant*P*P)/b*j)*Kv*Ko*Km)/Cl*Cg*Cs*kr*kt*kms;


%Check Contact Ratio
db=dp*cosd(fi);
pb=pi*db/N;
rp=dp/2;
rg=rp*R;
c=rp+rg;
a=1/P;
rap=rp+a;
rag=rg+a;
rbp=rp*cosd(fi);
rbg=rg*cosd(fi);

CR=((rap^2-rbp^2)^(1/2)+(rag^2-rbg^2)^(1/2)-c*sind(fi))/pb;

%maximum addendum cirle radius
ramax=(rg+c^2*sind(fi^2))^2

%display results
disp(['===================================GEAR
RESULTS===================================='])
if LHS>RHS
```

```
    disp(['LHS needs to be smaller'])
end
disp(['Sn is equal to ',num2str(Sn),' and the contact Ratio is
',num2str(CR),' '])
```

8.2.5 Preliminary Design Calculations

```
%% Thesis Design Calculations
% STEVEN CARLIN
% Tangential Stress vs. Radial Stress

clc
clear all
close all

v=6
omega=2000
r_i=0
r_o=250
ro=1000
r=[1:r_o]

sigmaT=ro*omega^2*(3+v/8)*(r_i^2+r_o.^2+(r_i^2*r_o.^2./r.^2)-
((1+3*v)/3+v)*r.^2)

sigmaR=ro*omega^2*(3+v/8)*(r_i^2+r_o.^2-(r_i^2*r_o.^2./r.^2)-r.^2)

%plot results
figure(1)
plot(r,sigmaT,r,sigmaR), xlabel 'radius (mm)', ylabel 'stress (MPa)',
title 'Tangential and Radial Stress at Radius',
legend('Tangential','Radial')

% Energy Storage Graph
wmax=1
ratio=[0:0.01:1]
wmin=wmax.*ratio
delivered=0.5*(wmax.^2-wmin.^2)
stored=0.5*(wmax.^2)
result=delivered/stored

%plot results
xx=0.5
yy=result(xx/0.01)
figure(2)
hold on
plot(ratio,result), xlabel 'wmin/wmax', ylabel 'Delivered Energy /
Stored Energy', title 'Comparison of Speed Ratios with Respect to
Energy'
plot(xx,yy,'ro','markersize',10)
xxx=[0:0.0001:xx]
yyy(length(xxx))=yy
hold on
plot(xxx,yy)
```

8.2.6 Clutch Design

```
%% Clutch Design
% Steven Carlin
% 0061033029
% This script gives the required dimension of clutch plates dependent
on
% number of friction surfaces (N).  It also calculates the clutch drag
for
% the chosen design of clutch

clear all
close all
clc

% Define Variables - Woven Clutch Material
f = 0.09*0.75  ;     %wet avg
p= 1552e3;           %kPa
N=1:8;               %Friction Surfaces

T=28.2;              %Nm @ flywheel
r_o=((T./((2/3).*pi.*p.*f.*N))./0.805).^(1/3);     %m
r_i=0.58*r_o;%m

%plot results
plot(N,r_o,N,r_i), title 'Number of Friction Surfaces and Respective
Dimensions to Meet Torque Requirement', xlabel 'Number of Friction
Surfaces', ylabel 'Radius (m)', legend('r_o','r_i'} Shaf)

%Check formula
Torque=2/3.*pi.*p.*f.*((r_o).^3-(r_i).^3).*N;




%% Clutch Drag calculations
%calculates drag torque for specific clutch
%
r_2=r_o(4);
r_1=r_i(4);
r_m=(r_2+r_1)/2;
N=2;
h=0.2e-3; %spacing between clutch plates (m)
mu=0.0082;    %Ns/m^2
omega=7803.7 %RPM
T_drag=(N.*mu.*pi.*(r_2^2-r_1^2).*omega.\r_m)\h
P_drag_max=T_drag*omega     %Watts


%% Shaft Torque Calculations - Keyway
% Assume Carburising Steel x grade
% Torque will be higher as RPM lower
% Calculate for worst case scenario
% Diff Keyway location
T=449;
Sy=530/2;     %MPa
D=0.035;     %m
d=((4*T)/(pi*0.58*Sy*D))^(1/2)/4;
```

```
L=1.8*D;
```

8.2.7 Repayment Period

```
%% Repayment Period
% Steven Carlin
% U1033029
clc
clear all
close all


cost=2850
savingpetrol=0.91 %$/100km
savingdiesel=0.71
savingpetrol1=1.64 %$/100km
savingdiesel1=1.32

km=[200:100:600]

periodpetrol=cost./(savingpetrol.*(km))
perioddiesel=cost./(savingdiesel.*(km))

periodpetrol1=cost./(savingpetrol1.*(km))
perioddiesel1=cost./(savingdiesel1.*(km))
subplot(2,1,1)
plot(km,periodpetrol,km,perioddiesel)
xlabel('Yearly Distance (km x 100)'), ylabel('Repayment Period
(Years)'), title('Repayment Period vs. Yearly Travel Distance with
2680kg load')
legend({'petrol','diesel'})

subplot(2,1,2)
plot(km,periodpetrol1,km,perioddiesel1)
xlabel('Yearly Distance (km x 100)'), ylabel('Repayment Period
(Years)'), title('Repayment Period vs. Yearly Travel Distance with
4500kg load')
legend({'petrol','diesel'})
```

8.3 Shaft and Bearing Calculations

<u>Shaft 1</u>

Try 1040 cold rolled steel $\sigma = 530MPa$

Torsion should be the main factor

$$D^3 = \frac{12000K_s}{F_r}T_e$$

$$K_s = 1.8$$

$$T_e = 1.15\sqrt{M_q^2 + 0.75T_q^2}$$

$$= 1.15\sqrt{0.75(449)^2}$$

$$= 7447$$

$$F_r = 0.45s_u$$

$$= 235s_u MPa$$

$$D = 34.3mm$$

$K = 1.5$ (Figure 7)

$K_s = 1.3$ (Figure1)

Recalculating:

$$D^3 = \frac{12000(1.3)(1.5)x446}{238.5}$$

$$D = 35.27mm$$

$$\frac{D_1}{D} = \frac{38}{36} = 1.1$$

$\Delta = 0.1$ (Figure 3)

$$Z = \frac{R}{D} + \Delta$$

$$= \frac{1}{36} + 0.1$$

$$= 0.13$$

$\therefore K = 1.4$ (Figure 4)

Considering press fit for bearings:

Use K8/K6 fit

K=1.6 (figures 5)

K=1.6+0.2=1.4

=1.88

Formula for power applied, torque reversals:

$$D^3 = \frac{10_s^{4f} K_s K}{Fr} \sqrt{\left(m_q + \frac{P_d D}{8000}\right)^2 + \frac{3}{4} T_z^2}$$

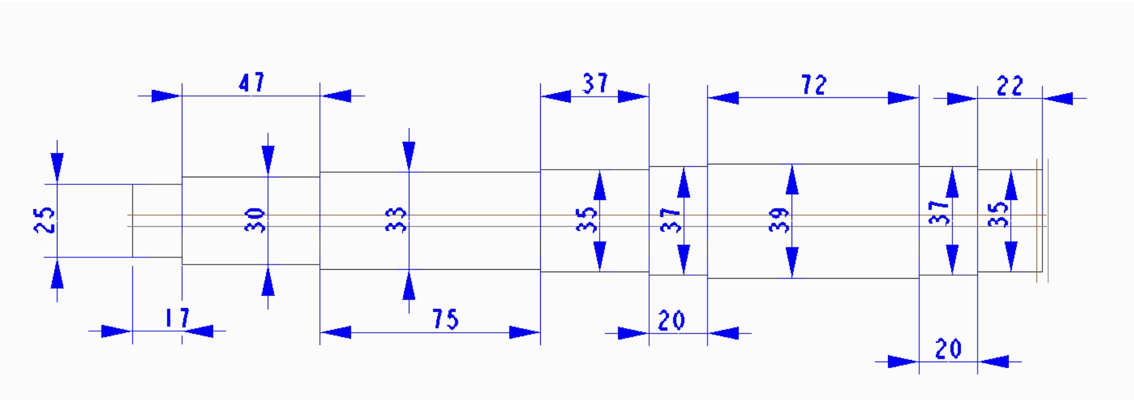
$$D^3 = \frac{10^4(1.2)(1.88)(1.4)}{238.5} \sqrt{0 + 0 + \frac{3}{4}(499)^2}$$

$$D^3 = 132.427\sqrt{151200.75}$$

D=37.2mm Round to 38mm

A third iteration yielded no change in diameter.

35mm should be satisfactory for the step down as the bearing does not support torque.



<u>Bearings</u>

Conditions:

ω=576rad/2= 5500RPM

$F = 0.6m \times 449Nm = 270N$

Radial loads will be due to gear torque on the shaft. Thrust loads will be only due to vehicle dynamic influence.

Select 6007 bearing

d=35mm

c=16.8kN

w=15000RPM

Select 6208 bearing for second bearing

D=44mm

c=32.5kN

w= 11000RPM

<u>Shaft 2</u>

Trial diameter

Conditions:

w=2034 rod/s

t_q=29N_n

$$D^3 = \frac{12000K_s}{F_r}TE$$

Where

$$K_s = 1.8$$

$$T_e = 1.15\sqrt{M_q^2 + 0.75\,T_q^2}$$

$$F = Mass\ of\ flywheel\ x\ acceleration$$

$$= 19.9x9.81x13$$

F=2527.85 m_q=127Nm

$$TE = 1.15\sqrt{127^2 + 0.75(29^2)}$$

$$= 148.76$$

$$D^3 = \frac{12000 k_s T_e}{F_r}$$

$$D^3 = \frac{12000(1.8)}{(\frac{891.5x0.45}{2})}$$

D=24.41mm

Stress Concentrations

- Press fit (flywheel)
- Shoulder x2 (1mm)
- Press Fit (Bearings)

K_s=1.2 (Figure 1) – size

$$\frac{D}{D} = \frac{24}{22} = 1.1$$

(Figure 3) shoulder

$\Delta = 0.10$

$$Z = \frac{R}{D} + \Delta$$

$$= \frac{1}{24} + 0.10$$

= 0.125

$$\therefore k_1 = 1.7$$

$$K_z = 3.1$$

(Figure 6) Press

$$K_t = k_1 + 0.2\, k_2$$

=3.1 +0.2(1.7)

K_t=3.44

$$P^3 = \frac{10^4(4)(1.2)(3.44)}{0.45\ (891.5)} \sqrt{237^2 + 0 + \frac{3}{4}(29^2)}$$

D=37.63mm ≈38mm

K_s = 1.35 $\Delta = 0.13$

$$\frac{D_1}{D} = \frac{38}{26} = 1.05$$

$$Z = \frac{1}{38} + \Delta = 0.16$$

$\therefore K = 1.6$

For press → k=3.1

K_E=3.1+0.2(1.6)

=3.42

$$D^3 = \frac{10^4(4)(3.42)(1.35)}{0.45(891.5)} \sqrt{(12)2 + \frac{3}{4}(29^2)}$$

D=39mm

<u>Bearings</u>- Self aligning

w=2048 rad/s

f= 1270N

d=35

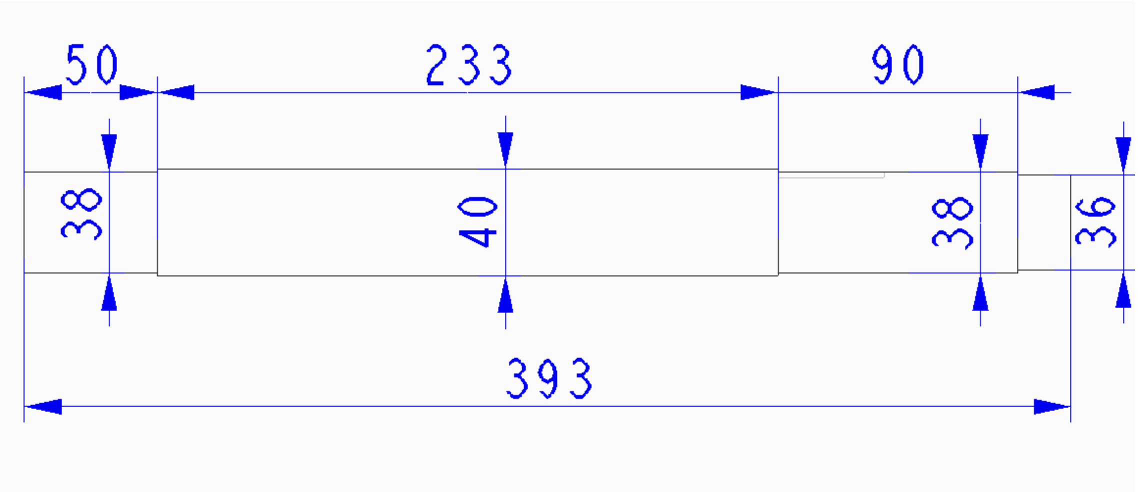
Chosen 1207 ETN9

c=19kw

d=35mm

p=72mm

b=17mm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