# Comparative Analysis of Automotive Powertrain Choices for the Near to Mid-Term Future

by

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Submitted to the Department of Mechanical Engineering on May 26, 2006 in Partial Fulfillment of the Requirements for the Degree of Master of Science in Mechanical Engineering

## ABSTRACT

This thesis attempts a technological assessment of automotive powertrain technologies for the near to mid term future. The powertrain types to be assessed include naturally aspirated gasoline engines, turbocharged gasoline engines, diesel engines, electric hybrids using gasoline engines and advanced transmissions. Advancements in aerodynamics, weight reduction and tire rolling friction are also taken into account.

The basis for the comparison is the potential of these powertrain technologies for reduction of oil consumption and green house gas emissions at the same level of performance as current vehicles.

The fuel consumption and performance of future vehicles was estimated using simple scaling laws and vehicle simulations. The results indicate that the potential for reduction of fuel consumption is significant for all the powertrains examined. More specifically, it seems that the current relative advantage of diesel over gasoline engines in terms of fuel consumption is reduced. Future turbocharged gasoline engines especially, seem to become almost equivalent with diesel engines. Hybrids electric vehicles do maintain a competitive advantage over other powertrains in terms of reduction of fuel consumption. This advantage is however much more pronounced for urban than for highway driving.

Thesis Supervisor: John B. Heywood
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## Acknowledgements

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My eternal gratitude goes to my parents Vasiliki and Panayiotis, my sister Alexandra and the rest of my family for their support throughout my life. Nothing would have been achieved without their love.

The help of my co-workers on this research project Matthew Kroemer, Anup Bandivadekar, Lynette Cheah and Tiffany Groode has been priceless. Our interaction and cooperation on the problems encountered in our research has been ideal.

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## Nomenclature

| | |
|---|---|
| $M_v$ | *Vehicle Mass* |
| $\alpha$ | *Road Slope Angle* |
| V | *Vehicle Velocity* |
| $\delta$ | *Vehicle Mass Increase Factor to account for rotational inertias* |
| $F_t$ | *Tractive Effort* |
| $F_{tr}$ | *Vehicle Resistance Force* |
| $T_r$ | *Rolling Friction Resistance Torque* |
| $F_r$ | *Rolling Friction Resistance Force* |
| $r_w$ | *Wheel Radius* |
| $c_r$ | *Tire Rolling Friction Coefficient* |
| g | *Acceleration of Gravity* |
| $F_w$ | *Aerodynamic Drag Resistance Force* |
| $\rho_{air}$ | *Air density* |
| $c_D$ | *Coefficient of Drag* |
| A | *Vehicle Cross-Sectional Area* |
| $F_g$ | *Grade Resistance Force* |
| $F_{accel}$ | *Acceleration Force* |
| G | *Road Grade in %* |
| $V_w$ | *Wind Relative Velocity with respect to the vehicle* |
| h | *Vehicle Height* |
| W | *Vehicle Width* |
| N | *Engine frequency of Revolution in Hz or rpm(rounds per minute)* |
| T | *Torque in N*m* |
| $\omega$ | *Engine Rotational Speed in rad/s* |
| $\tau_p$ | *Engine Power Stroke Period in s* |
| bmep | *Brake Mean Effective Pressure* |
| imep | *Indicative Mean Effective Pressure* |
| fmep | *Friction Mean Effective Pressure* |
| $\eta_i$ | *Indicative Engine Efficiency* |
| $V_d$ | *Engine Displacement Volume* |
| $\dot{m}_f$ | *Fuel mass Flow rate* |
| LHV | *Fuel Lower Heating Value* |
| $\eta_b$ | *Brake Engine Efficiency* |
| $\eta_m$ | *Mechanical Engine Efficiency* |
| bsfc | *Break Specific Fuel Consumption in g of fuel per kWh of Work out* |
| $W_{curb}$ | *The curb weight of a vehicle-The weight of the complete vehicle without the driver.* |
| $P_{max,eng}$ | *Engine Maximum Power* |
| $W_{glider}$ | *The weight of the vehicle without any powertrain components* |
| S | *Mean Piston Speed* |
| L | *Piston Stroke* |
| $n_c$ | *Number of Cylinders* |

| | |
|---|---|
| B | *Cylinder Bore* |
| R | *Bore to Stroke Ratio* |
| $\dot{m}_{air}$ | *Air mass Flow rate* |
| $\eta_{vol}$ | *Engine Volumetric Efficiency* |
| $\tau$ | *Engine Response Time to Step Load Change* |
| $T_w$ | *Torque at the Wheel* |
| $T_{eng}$ | *Torque Output from the Engine* |
| $i_g$ | *Gear Ratio in the Transmission* |
| $i_{fd}$ | *Gear Ratio of the Final Drive in the Differential.* |
| $\eta_t$: | *Efficiency of the entire driveline (Transmission+ Final Drive)* |
| $i_{tot}$ | *Total Driveline Gear Ratio (Transmission+ Final Drive)* |
| $T_{GEN}$ | *Generator torque* |
| $T_{RING}$ | *Torque at ring gear (not including torque generated by motor)* |
| $T_{MOTOR}$ | *Motor Torque* |
| $\beta$ | *Ratio of Frontal over Rear Braking Force* |
| HR | *The relative size of the Power from the Electric System in a Hybrid over the Power from the Engine* |
| $P_{W,max}$ | *Maximum power required at the wheel* |
| $\sigma$ | *Fraction of the engine's power that goes through the series branch* |
| $P_{eng,max}$ | *Engine maximum power* |
| $\eta_t$ | *Transmission efficiency* |
| $\eta_{mot}$ | *MotorEfficiency* |
| $\eta_{gen}$ | *Generator Efficiency* |
| $P_{battery,,max}$ | *Battery Maximum Power* |
| $RI_{2005}$ | *Relative Fuel Consumption Improvement over 2005* |
| TF | *Average Torque Fraction* |
| $RI_{2030}$ | *Relative Fuel Consumption Improvement over 2030 Naturally Aspirated Gasoline Powertrain* |
| $\bar{i}$ | *Average Gear Ratio Over a Driving Cycle* |

# Abbreviations

| | |
|---|---|
| GHG | *Greenhouse gases* |
| ICE | *Internal Combustion Engine* |
| NA SI | *Naturally Aspirated Spark Ignition (Gasoline) Engines* |
| SI Turbo | *Turbocharged Spark Ignition Engines* |
| Diesel | *Compression Ignition (Diesel) Engines* |
| Hybrid | *Hybrid powertrains using NA SI Engines* |
| mpg | *Miles per gallon of fuel-Fuel Economy* |
| liters/100km | *Liters of gasoline (or the energy equivalent to gasoline) per 100 km of driving-Fuel Consumption* |
| gC/km | *Grams of Carbon)(in Carbon Dioxide) Emitted per km Driven* |
| FTP | *Federal Test Procedure-the standard urban driving cycle in the USA* |

| | |
|---|---|
| HWFET | *Highway Federal Emissions Test-the standard highway driving cycle in the USA* |
| US06 | *An aggressive highway driving cycle used in the USA* |
| unadjusted fuel consumption | *Fuel consumption in liters/100km as they are measured over the FTP and HWFET test procedures* |
| adjusted fuel consumption | *FTP fuel consumption is divided by 0.9, HWFET fuel consumption is divided by 0.78* |
| combined fuel consumption | *55% x (FTP fuel consumption)+45% x (HWFET fuel consumption). Can be adjusted or unadjusted* |
| mph | *Miles per hour-speed* |
| LCA | *Life Cycle Assessment* |
| TTW | *Tank to Wheels* |
| WTT | *Well to Tank* |
| WTW | *Well to Wheels* |
| CTG | *Cradle to Grave* |
| NREL | *National Renewable Energy Laboratory* |
| CAFE | *Corporate Average Fuel Economy* |
| ANL | *Argonne National Laboratory* |
| "Lower Performance Camry" | *Vehicles Equivalent to the 2.5 liter 2005 Camry* |
| "Higher Performance Camry" | *Vehicles Equivalent to the 3.0 liter 2005 Camry* |
| GDI | *Gasoline Direct Injection* |
| PFI | *Port Fuel Injected Engines* |
| lt | *Liters* |
| EGR | *Exhaust Gas Recirculation* |
| CVT | *Continuously Variable Transmission* |
| IVT | *Infinitely Variable Transmission* |
| SAE | *Society of Automotive Engineers* |
| PNGV | *Partnership for a New Generation of Vehicles* |
| OEM | *Original Equipment Manufacturer* |
| Industry Cycle Fuel Consumption | *The Average Fuel Consumption of the unadjusted FTP, unadjusted HWFET and US06* |

# 1. Introduction

## 1.1. Motivation

There are two reasons for examining ways to reduce automotive fuel consumption:

- Reducing oil demand
- Reducing greenhouse gas emissions.

Transportation consumes today about 70% of the worldwide petroleum production. Petroleum demand around the world is projected to increase by 50% by 2030 [1]. Although there is no clear answer to what the effect of this growth will be on petroleum reserves and oil prices, there is definitely a need to examine the options for curbing humanity's oil appetite.

Moreover, transportation today is responsible today for 25% of the world's green house gas (GHG) emissions. The growth in the demand for transportation is such that by 2030, transportation will be the most emitting sector of the economy. Carbon dioxide emissions from transportation scale quite linearly with fuel consumption since most of today's vehicle run on liquid fossil fuels. The increase in GHG's therefore is also about 50% by 2030. The vast majority of the scientific community agrees today that greenhouse gases do have adverse effects on the climate. Any strategy that wishes to mitigate these effects by reducing the increase in GHG emissions would have to include transportation in its portfolio of measures.

Focusing on the U.S.A., about 60% of the petroleum demand for transportation today is attributed to the light duty vehicle fleet. About another 20% is attributed to heavy trucks and rail transportation. The remaining 20% is mostly the consumption of air and sea transportation. [2]

In the U.S.A. the light duty fleet fuel economy average has been constant or slightly decreasing since the early eighties. The sales averaged fuel economy in miles per gallon (mpg) versus model year is presented in Figure 1. Over this time period, automotive technologies witnessed significant improvements. The reason why these improvements were not materialized was that the market moved towards larger, heavier vehicles with higher performance. The sales averaged vehicle weight as well as the 0-60 miles per hour (mph) acceleration times are plotted versus model year in Figure 2. The sales fraction versus model year presented in Figure 3 shows how the increasing fraction of SUV's and vans is partially to blame for the increase in size and weight. The black area in that plot denotes the sales fraction of pickup trucks which has remained about constant and the white area denotes the cars sales fraction which has been declining. The validation of the claim that there have been significant automotive technology advances since the early eighties which weren't materialized in fuel economy is presented in Figure 4. Fuel consumption is plotted in this figure versus weight in lbs for model years 1974 and 2004. It is evident that for the same weight, 2004 models are significantly better. It is also evident how for 2004 models, the sensitivity of fuel consumption on weight has decreased compared with 1974.

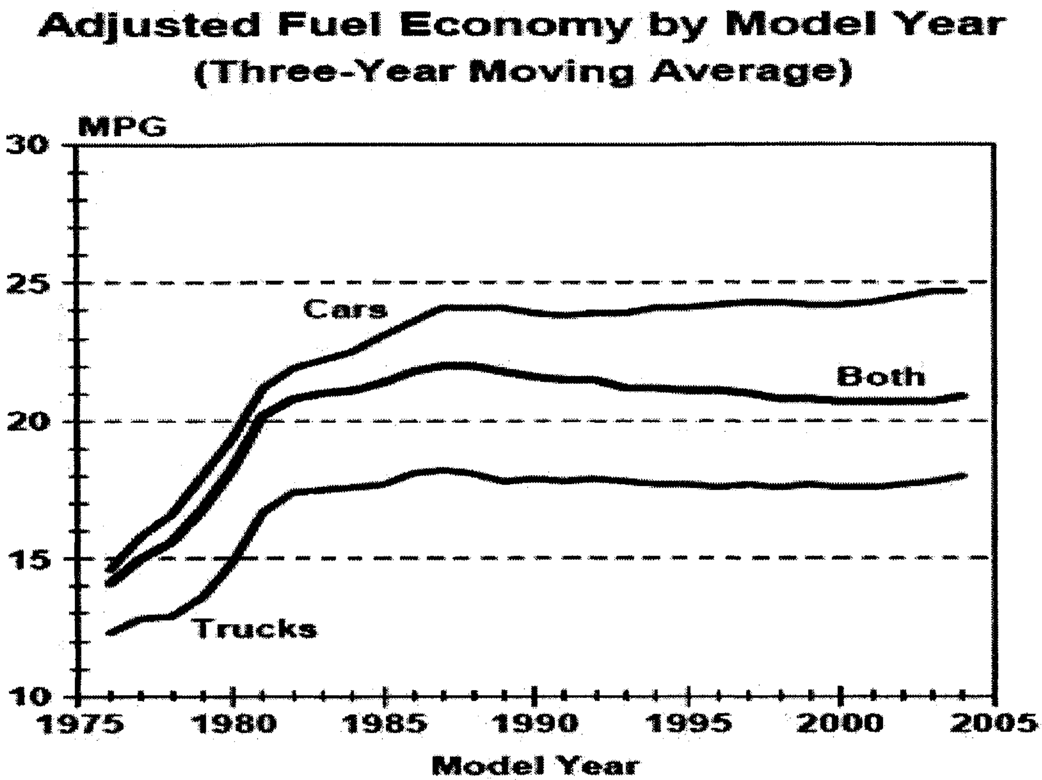



**Figure 1: U.S.A. Light Duty Sales Averaged Fuel Economy versus Model Year .Source: [3]**

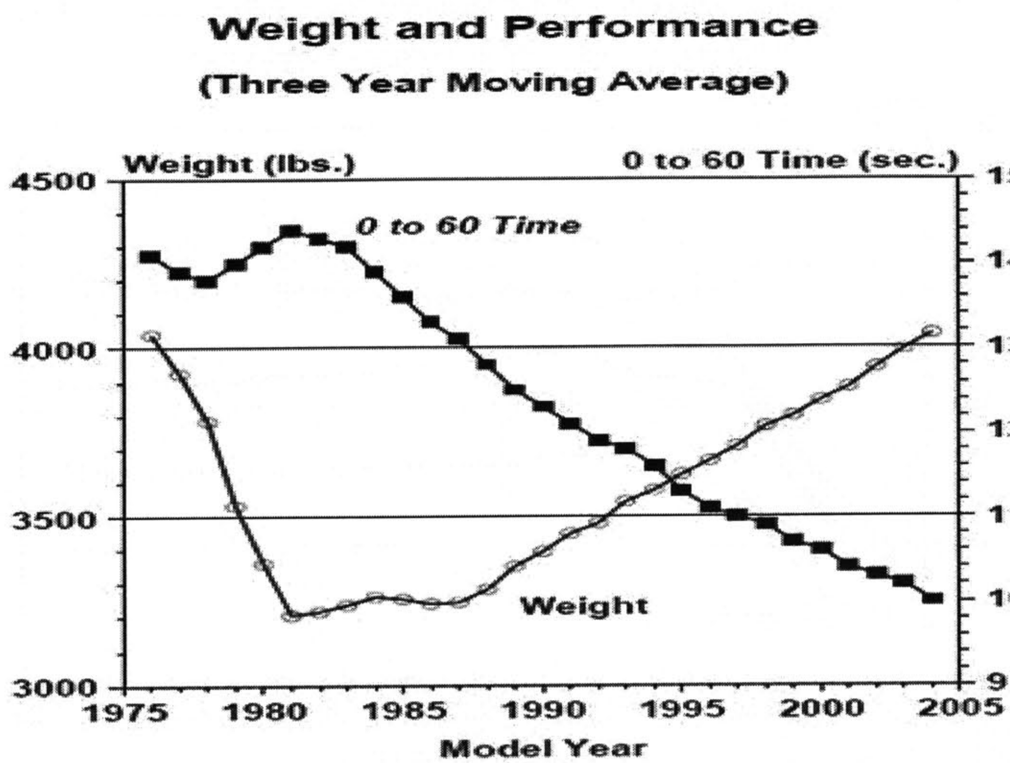



**Figure 2: U.S.A. Light Duty Fleet Weight and Performance versus model year. Source: [3]**

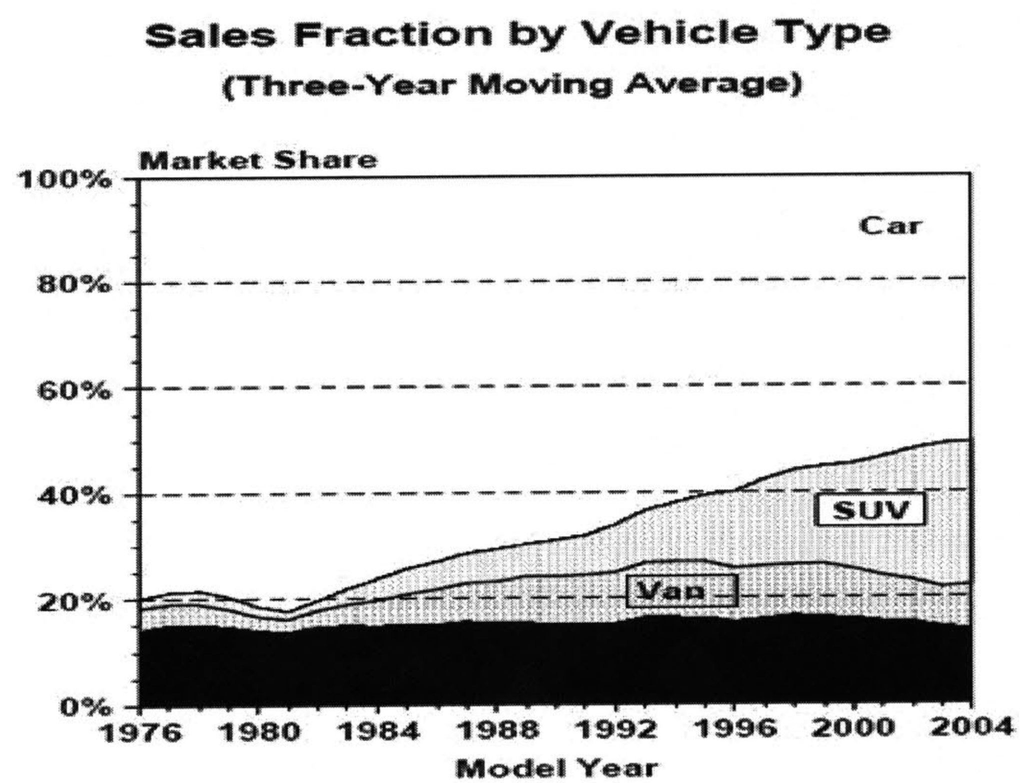



**Figure 3: U.S.A. Light Duty Fleet sales Fractions by Vehicle Type versus Model year. Source: [3]**

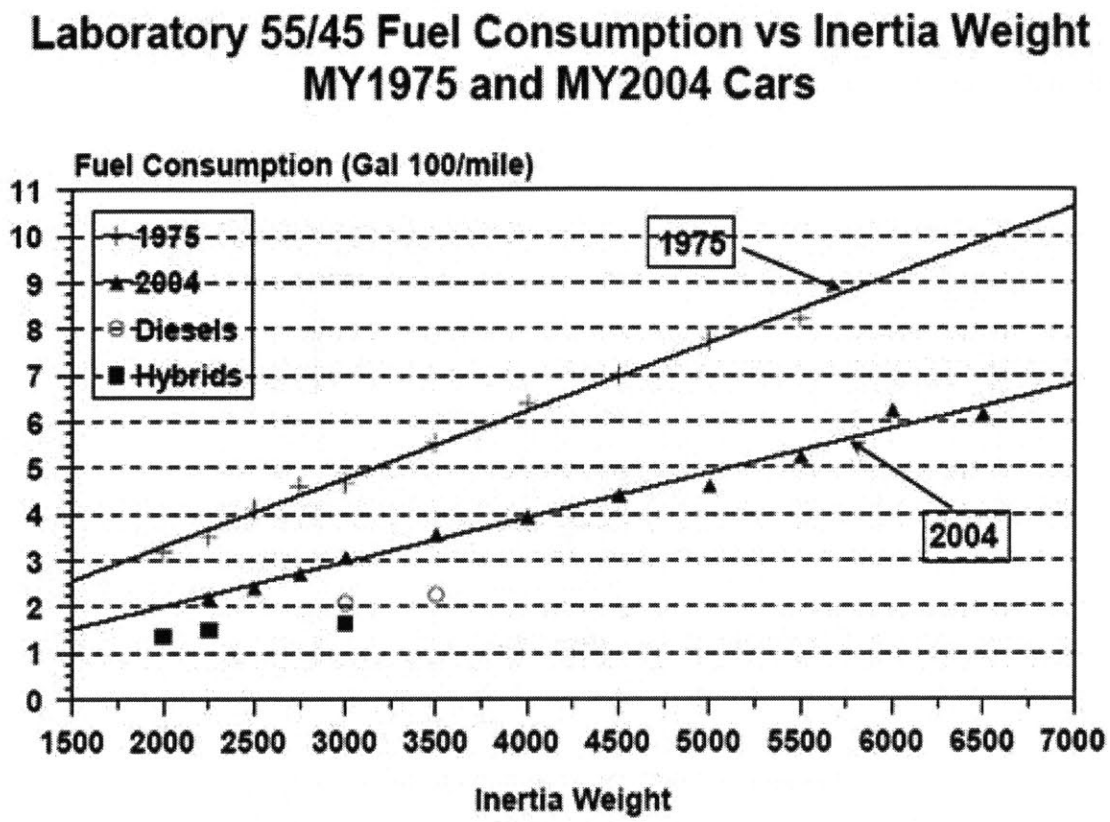



**Figure 4: Fuel Consumption versus weight in lbs for two model years- 1974 and 2004. Source: [3]**

This thesis is a part of a larger research project at the Sloan Automotive laboratory to address ways to reduce fuel consumption. It is evident that technology is only a part of the equation and that policy and behavioral changes are of at least equal importance.

## *1.2. Focus*

This thesis will focus on technological and more specifically powertrain improvements for the near to mid term future in terms of fuel consumption reduction potential. The projections will be used for fleet level calculations of fuel consumption and GHG emissions by other contributors to the research project. More specifically, this thesis will compare powertrain options for the U.S.A. light duty fleet in 2030 Before looking at the technologies examined and the methodology that will be used it is important to define what a powertrain is.

### 1.2.1. What is a Powertrain?

In its most general definition a powertrain consists of all the vehicle subsystems necessary to produce power transform it, and transmit it to the wheels. In the context of internal combustion engine powered vehicles, the powertrain includes an engine, a clutch or hydraulic torque converter to couple it to the gearbox, a gearbox and a final drive-differential which connects the gearbox to the wheels. The purpose of the gearbox is to transform the torque and speed generated from the engine to whatever is needed at the wheel. The final drive includes the differential necessary for the vehicle to take turns without its wheels slipping as well as a final gear ratio supplementing the gearbox. A typical ICE powertrain can be seen in Figure 5.

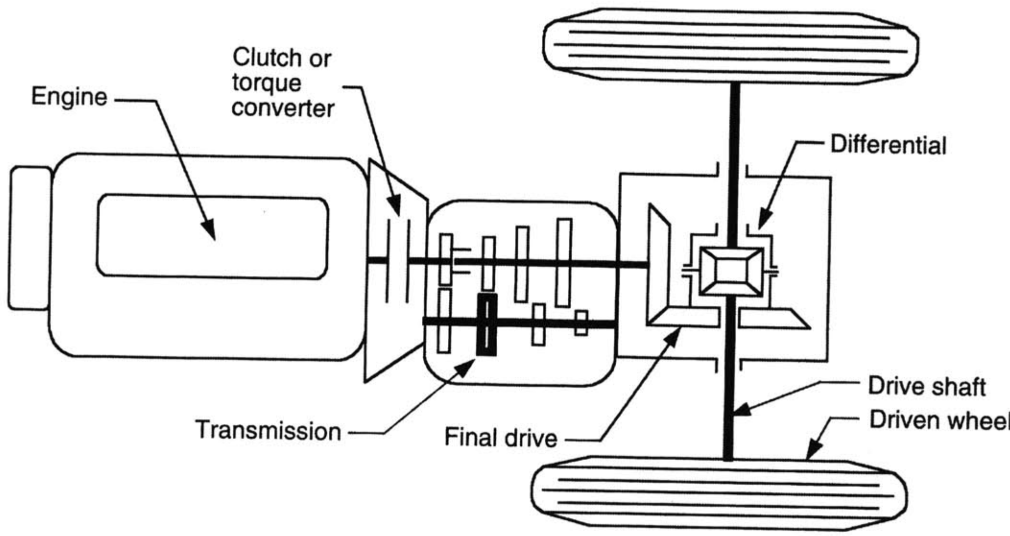



**Figure 5: A Vehicle Powertrain. Adapted from [4]**

### 1.2.2. Which Technologies should be Modeled?

This thesis has a time horizon of 25 years into the future. Currently, the light duty vehicle fleet is almost exclusively dependent on internal combustion engines (ICE) using liquid hydrocarbons as fuels. The fuel infrastructure and industry is also based on the production, transportation and distribution of these fuels. There are good reasons why this combination of powertrain and fuel type has prevailed. Liquid hydrocarbons are extremely energy dense fuels. Internal combustion engines are power dense, relatively efficient for their power density and very technologically mature.

For a significantly different powertrain technology to make a difference in fleet level oil demand and GHG emissions, it must first penetrate the fleet in significant numbers. This process takes a lot of time in the order of ten to twenty years, from the point in time, the vehicle starts being mass produced and is available at the showroom to the time it is a significant fraction of the fleet. Before penetrating the fleet in significant numbers, the new technology must become a significant fraction of all vehicles manufactured. Even before that, the new technology must have become market competitive.

There are new powertrain technologies on the horizon today such as plug-in hybrids, hydrogen fuel cell vehicles and battery electric vehicles. The latter, use electricity to charge their on board batteries as an exclusive source of power. Plug in hybrids are battery electric vehicles also equipped with an ICE, or alternatively phrased, hybrid ICE vehicles with the extra ability to be plugged in the electric grid to recharge their batteries. A plug-in hybrid has therefore two primary energy sources, the hydrocarbon fuel as well as the electric grid. Fuel cell vehicles use on board stored hydrogen to directly generate water vapor and electricity which can be used to propel the vehicle. The time scales required for these technologies to make an impact are significantly larger than those of primarily ICE powered powertrains. This is because there are still technical issues to be resolved before the technology makes it to the marketplace. Additionally, there are significant fuel infrastructure issues especially with hydrogen fuel cell vehicles that would delay fleet penetration. An estimate of the different time scales required for new powertrain technologies to affect US transportation energy use is presented in Figure 6. Although this figure is just an estimate, it does provide an approximation of the order of magnitudes involved. Fuel cells and battery electric vehicles may be viewed as long term solutions while plug-in hybrids as mid to long term solutions. These options will be considered by other contributors to the project.

**Time Scales for New Vehicle Technologies to Affect US Transportation Energy Use**

| | Vehicle Technology | | | |
|---|---|---|---|---|
| Implementation Phase | Gasoline Direct-Injection Spark-Ignition Boosted Downsized Engine | High Speed Direct-Injection Diesel with Particulate Trap, $NO_x$ Catalyst | Gasoline Spark-Ignition Engine/ Battery-Motor Hybrid | Fuel Cell Hybrid Vehicle, Onboard Hydrogen Storage |
| Market competitive vehicle | ~ 5 years | ~ 5 years | ~ 5 years | ~ 15 years |
| Penetration across new vehicle production | ~ 10 years | ~ 15 years | ~ 20 years | ~ 25 years |
| Major fleet penetration | ~ 10 years | ~ 10–15 years | ~ 10–15 years | ~ 20 years |
| Total time required | ~ 20 years | ~ 30 years | ~ 35 years | ~ 55 years |

*This table shows MIT estimates of how long it will take for four new vehicle technologies to be on the road in sufficient numbers to affect total US energy consumption for transportation. In the first phase, the technology must become market competitive in performance, convenience, and cost. In the second, it must become more than 35% of all the new vehicles manufactured. In the third, it must become responsible for more than 35% of total US miles driven. The total times (even allowing for overlap in the phases) demonstrate that new vehicle technology is far from a "quick fix" for America's enormous appetite for transportation energy.*

**Figure 6: Estimated Times for New Powertrain Technologies to Affect US transportation Energy Use, Adapted from [5]**

It is hopefully clear by now that near to mid term powertrain solutions that will significantly affect transportation energy will be based on internal combustion engines. The powertrains that will be modeled in this thesis are:

- Naturally Aspirated Spark Ignition (Gasoline) Engines-NA SI
- Turbocharged Spark Ignition Engines-SI Turbo
- Compression ignition diesel engines-Diesel
- Hybrid powertrains using NA SI engines-Hybrid

## 1.3. Methodology

When addressing energy and environmental issues such as reduction of energy use and GHG emissions, it is important that the scope of the analysis encompasses the whole system. The reduction of emissions by the use of a new fuel in cars for example is of no use if more is emitted during the production of the fuel itself. The methodology for assessing such issues is called a Life Cycle Assessment (LCA) of a product or service. In the context of vehicles the environmental and energy implications of a vehicle can be divided into three parts:

- The "Tank-to-Wheels" (TTW) part. This deals with the amount of fuel/energy used and pollutants emitted by the vehicle when being driven during its lifetime.
- The "Well-to-Tank part" (WTT). This deals with the amount of energy used and pollutants emitted during the extraction, production and distribution of the fuel.
- The first two parts are sometimes called "Well to Wheels-WTW". In order for the lifecycle assessment to be complete, the energy and emissions associated with the production of the vehicle itself need to be accounted for. The approach is then called "Cradle to Grave-CTG". This type of analysis would be needed, for example, to estimate the tradeoff between the fuel saved by an lightweight aluminum vehicle and the large energy required to produce the aluminum.

In this thesis, the focus will be a tank to wheel assessment of different powertrains. Some well to wheels results will be presented using well established well to tank figures from the literature. Moreover, the only emission that will be dealt with is carbon dioxide. Criteria Pollutants like NOx,CO,H/C and particulates will not be investigated. The reason for that choice is first that automotive technology has exhibited great success in adapting to stricter criteria pollutant standards. This is truer for spark-ignition engines and less so for diesel. In the last decade or so, however, diesel engines are also becoming significantly cleaner. More importantly, the goal of this thesis is not to compare different powertrains in terms of criteria pollutants, but to answer the question *"If different powertrains can meet the pollutant standards, what would their impact on fuel consumption and GHG emissions be?"* The implications of stricter criteria pollutant standards on fuel consumption were of course taken into account.

For a powertrain technology to succeed on the marketplace, it is important to at least offer the consumer all the attributes that the competition offers. The criteria consumers use to choose which vehicle to buy range from cost to size and comfort to performance to pure aesthetics. Fuel economy might be one of these criteria, but definitely not the most important one for many consumers. This is however a technical, not a marketing thesis, so the effect of appeal to the consumer was dealt with in the following ways:

- Special care was taken to equalize performance between vehicles using different powertrains. Performance was mainly defined as the 0-60 miles per hour acceleration time, but several performance criteria were used for comparison (40-60mph acceleration, top speed, grade ability, towing capability). Performance as defined by 0-60 times was kept at current levels.
- Size was kept constant at today's levels. The weight reduction assumed was based on technological improvements keeping size constant. Currently, the trend is towards larger and heavier vehicles.
- Exploring the effects of consumer trends in size and performance is an interesting question which was left as future work.

The vehicle platforms chosen for the comparison of different powertrains were:

- The 2005 Toyota Camry
- The 2005 Ford F150

These two vehicles were chosen to represent the two main sub-categories of the light duty fleet; cars and light trucks. They are the best selling family car and pickup truck in the US respectively [6]. Both vehicles come in a variety of trims and engines. The entry level engines (2.5 and 4.2 liters respectively) were chosen as the main vehicle platforms for both models. The reason was that the 0-60 mph acceleration times for these vehicles were close to the fleet average at around 10s. The F150 is very close to the fleet weight average for pickup trucks (2 153kg versus 2155kg). The Camry is also quite close to the average weight for cars (1435kg kg versus 1565kg). A higher performance version using a 3.0 liter engine was also used as a vehicle platform for the Camry to explore the effect of performance. The abbreviation "lower performance Camry" will hence be used for vehicles equivalent to the 2.5 liter 2005 Camry. The abbreviation "higher performance Camry" will be used for vehicles equivalent to the 2005 3.0 liter Camry.

## 1.4. Testing Fuel Consumption

Fuel consumption for vehicles is usually measured over standardized driving cycles. A driving cycle is a driving pattern in which speed is defined for every time instant. Additionally, in some driving cycles the speed selected in the gearbox is also defined for every time instant. In order to actually measure fuel consumption for a vehicle over a driving cycle, the vehicle is placed on a dynamometer that can simulate the resistance forces that the vehicle would actually experience if driving at the speeds defined by the driving cycle.

Alternatively, the fuel consumption of a vehicle can be simulated if the behavior of every component is known to an adequate degree of accuracy. The software used for this purpose in this thesis is called ADVISOR. ADVISOR was developed by the National Renewable Energy Laboratory (NREL) and is now commercially available from AVL. It is a backwards facing calculation including some forward facing loops. What this means is that the simulation starts at the wheel, where the speed and acceleration based on the driving cycle is determined. The speed of the wheel and wheel torque required to overcome resistance forces is subsequently calculated. The requested speed and torque at the wheel is transferred upstream to the gearbox were based on the gear selected it is used to calculate the speed and torque request from the engine. The code is based on the Matlab Simulink environment. A schematic of the signal flow between the subsystems used for the simulation in ADVISOR is presented in Figure 7.

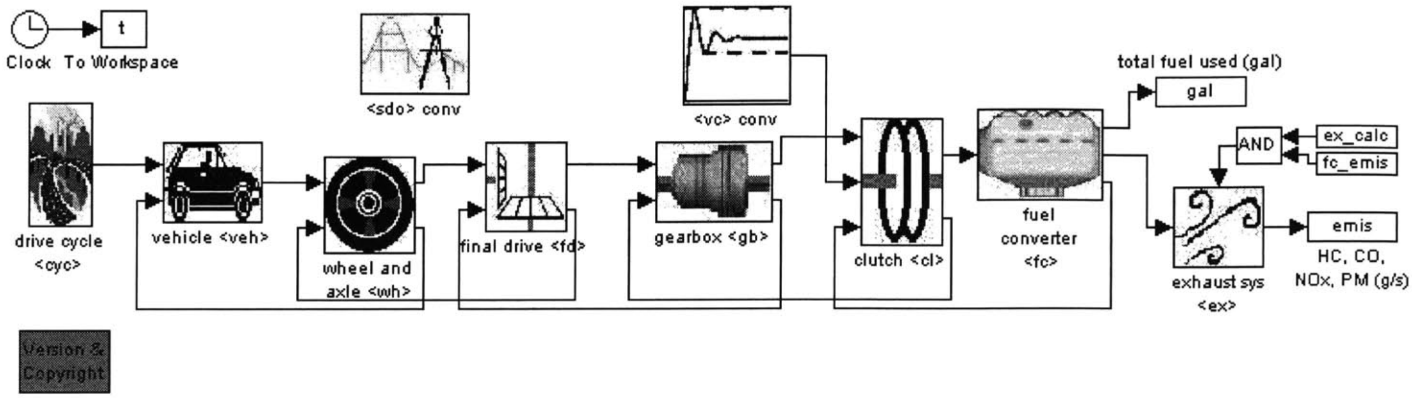



**Figure 7: Schematic of Component Subsystems used in the ADVISOR simulations**

The two standard cycles used in the U.S.A. to test fuel consumption and pollutant emissions are the FTP and the HWFET. The FTP cycle represents urban driving and the HWFET highway driving. The FTP cycle is shown in Figure 8 and the HWFET in Figure 9. To account for real world driving effects like more aggressive driving, weather and use of auxiliaries the FTP fuel consumption is derated by being divided by 0.9, the HWFET fuel consumption is divided by 0.78. The results are known as the *adjusted* fuel consumption. The weighted average in which the FTP is weighted by 0.55 and the HWFET by 0.45 is known as the *combined fuel consumption.* The weights account for the percentage of highway and urban driving for the average vehicle.

The US06 cycle is an alternative cycle which is used in the U.S.A. to represent more aggressive highway driving. The cycle is presented in Figure 10. The statistics for all three cycles used in this thesis are presented in Table 1. Note how the average speeds for the HWFET and US06 are almost the same and significantly higher than that of the FTP. Also note how the rates of acceleration and deceleration are significantly higher than those of the FTP and HWFET which are almost the same.

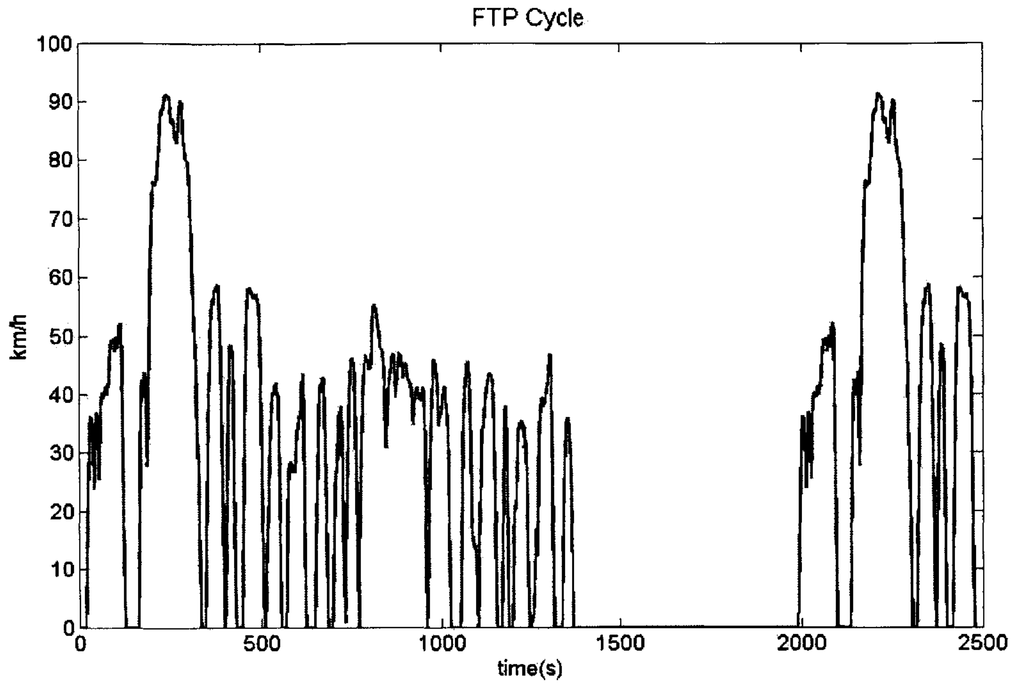



**Figure 8: The FTP Drive Cycle**

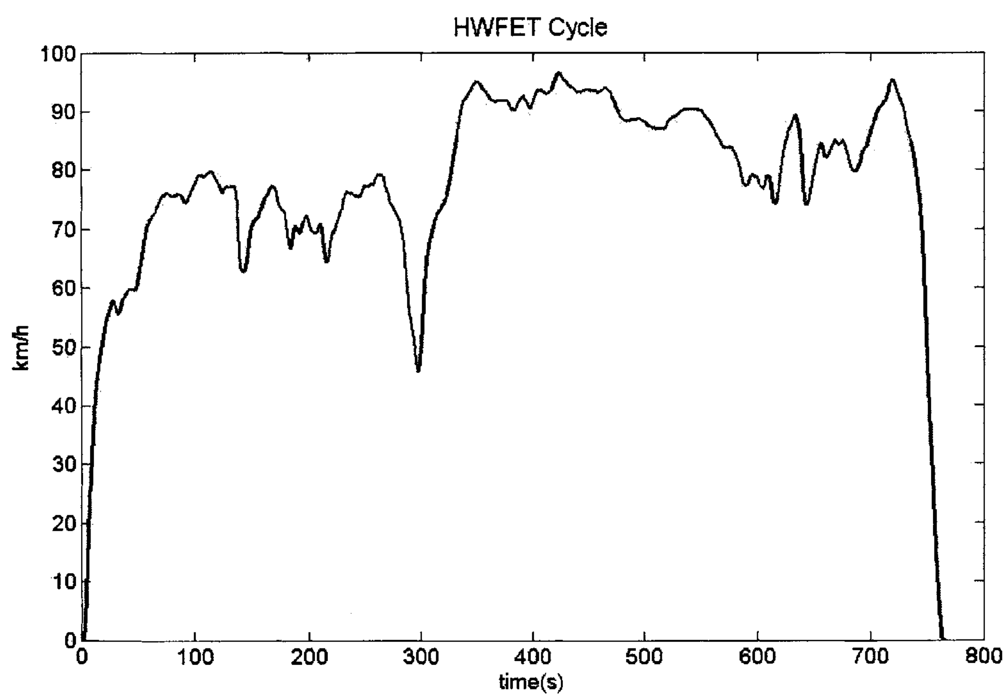



**Figure 9: The HWFET Drive Cycle**

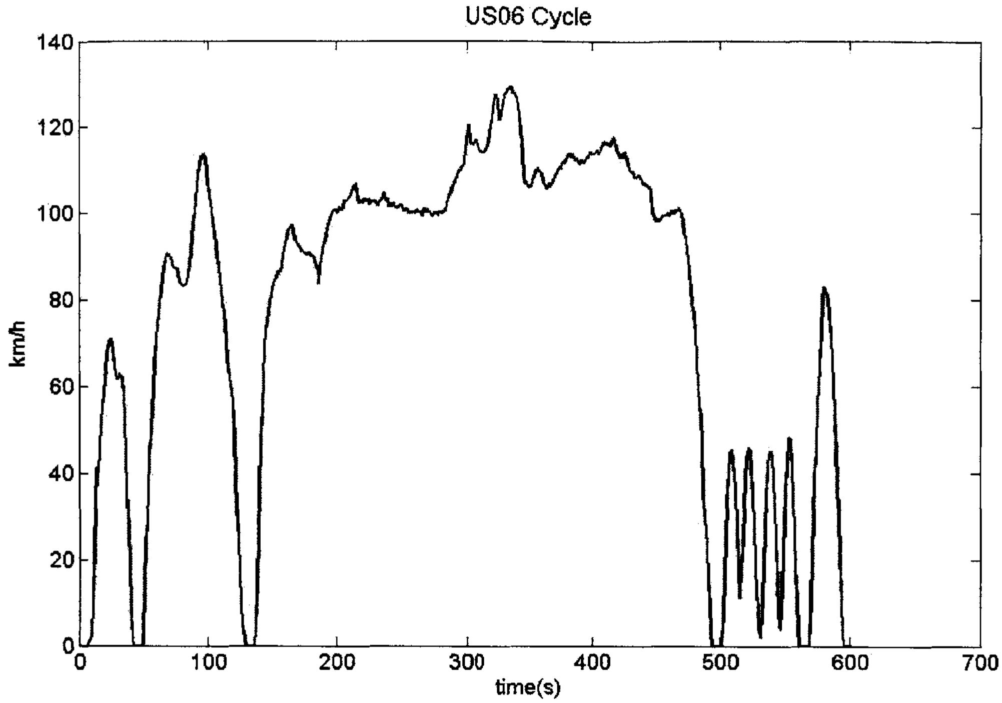



**Figure 10: The US06 Drive Cycle**

**Table 1: Drive Cycle Statistics**

| | US06 | FTP | HWFET |
|---|---|---|---|
| time(s) | 600 | 2477 | 765 |
| distance(km) | 12.89 | 17.77 | 16.51 |
| maximum speed(km/h) | 129.23 | 91.25 | 96.4 |
| average speed(km/h) | 77.2 | 25.82 | 77.58 |
| maximum acceleration (m/s^2) | 3.76 | 1.48 | 1.43 |
| maximum deceleration (m/s^2) | -3.08 | -1.48 | -1.48 |
| average acceleration(m/s^2) | 0.67 | 0.51 | 0.19 |
| average deceleration(m/s^2) | -0.73 | -0.58 | -0.22 |
| idle time (s) | 45 | 360 | 6 |
| no of stops | 5 | 22 | 1 |

## 1.5. Literature Review

There have been numerous technical assessment studies exploring the potential of different automotive technologies. This study is the update of two previous life cycle assessments of future vehicles conducted by the Sloan Automotive Laboratory and the Laboratory for Energy and the Environment at MIT [7] and [8]. The results of this study will be extensively compared with those in [8] in chapter six.

Other important studies assessing costs and fuel economy benefits for different vehicle technologies in the U.S. context are:

- The Argonne National Laboratory (ANL) and General Motors well to wheels study [9]
- The National Research Council study on the effects of Corporate Average Fuel Economy (CAFE) Standards [10].
- The California Air Resources Board study on the effect of proposed regulations to reduce vehicle climate change emissions.[11]
- The Northeast States Center for a Clean Air Future study with AVL[12]
- A couple of interesting technical publications are worth mentioning such as the one by An, DeCicco, and Ross[13] and the one by Greene and DeCicco, [14]
- An important well to wheels study in the European Context is the one by CONCAWE, EUCAR, and ECJRC [15]

## 1.6. Ways to Reduce Automotive Fuel Consumption

Figures 11 and 12 show typical energy flows for a midsize passenger car during urban and highway driving respectively. They were adapted from [16]. From the initial 100% of the energy in the fuel in the fuel tank only about 13-21% ends up at the wheel actually moving the vehicle. The rest is lost as engine and driveline (transmission) losses, wasted while the engine is idling (standby) or powers accessories. The energy that actually ends up at the wheel is used to overcome aerodynamic and rolling resistances and to accelerate the vehicle. The first two parts of the mechanical energy at the wheel are immediately dissipated as heat. The power used to accelerate is also eventually dissipated as heat when the vehicle decelerates through braking.

There are therefore, two ways in which the fuel consumption of a given vehicle may be reduced:

1. Reducing the amount of energy per distance driven required at the wheel.
2. Improving the average efficiency of the powertrain used to produce and deliver this energy at the wheel. Idling reduction may be viewed as another way of improving engine average efficiency.

Reducing the accessory load would be a third way to reduce fuel consumption. The historical trend however has been that this load has been increasing as more and more gadgets are added to vehicles. The size of auxiliary loads depends more on consumer trends and less on technical improvements. The effect of auxiliaries was therefore neglected throughout this study.

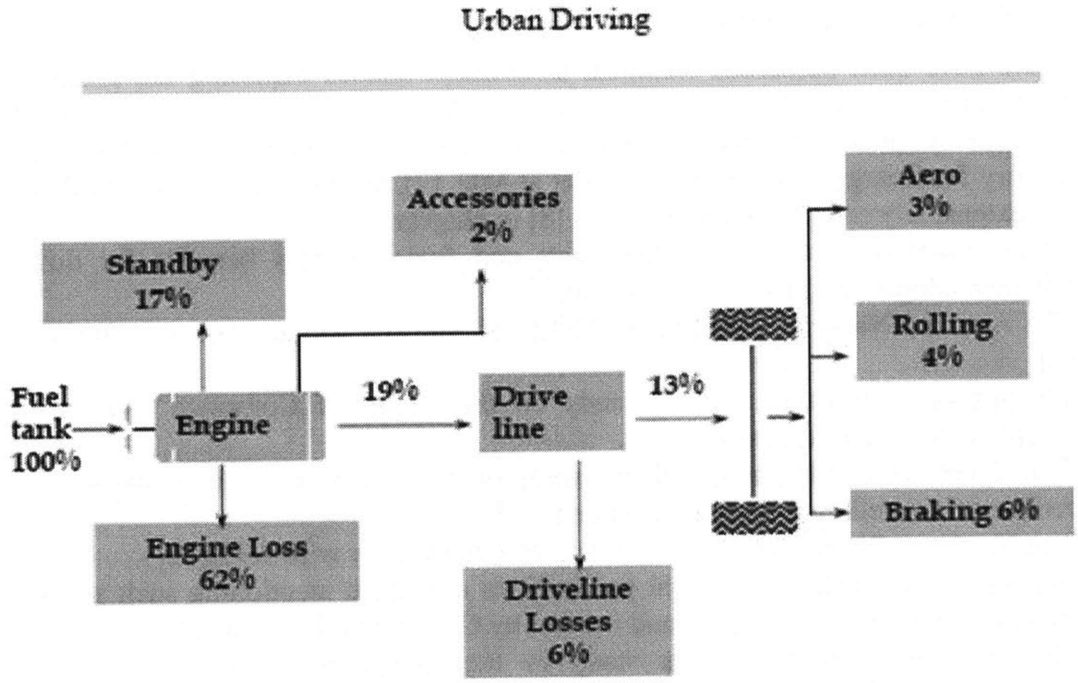



**Figure 11: Example urban driving energy flows for a late-model midsize passenger car.**

**Source: [16]**

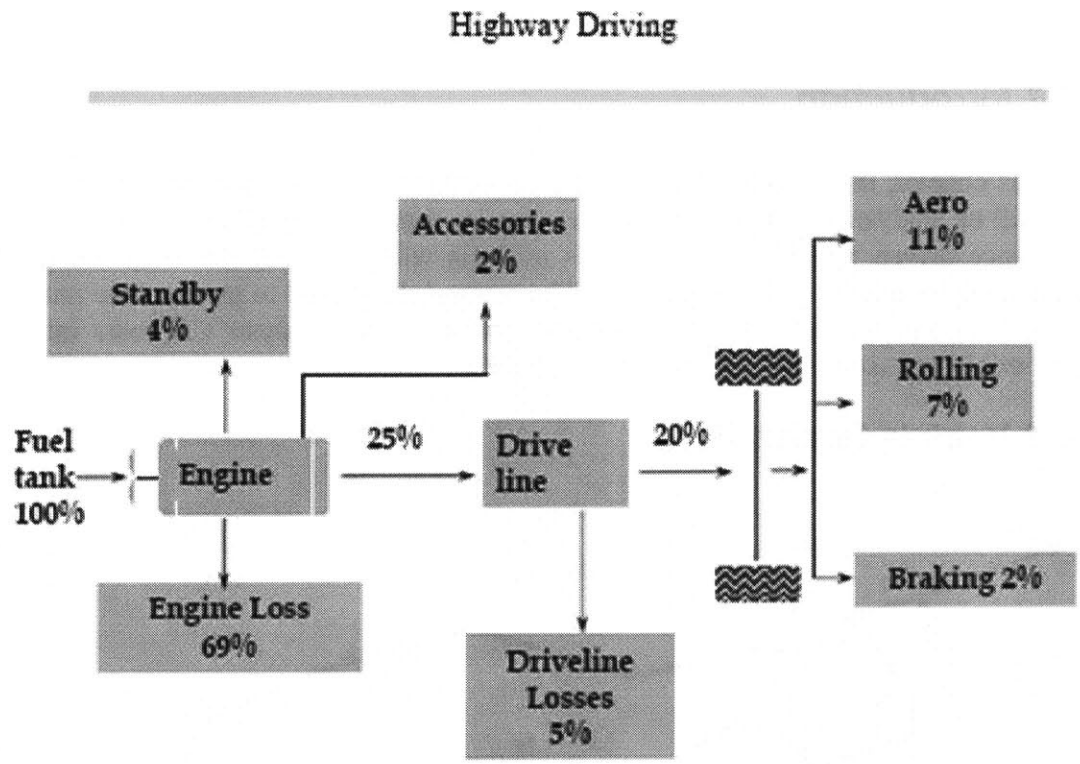



**Figure 12: Example highway driving energy flows for a midsize passenger car. Source: [16]**

## *1.7. Thesis Outline*

This thesis consists of six chapters. The scope and methodology of the analysis are introduced to the reader in chapter one. The potential for non powertrain improvements and the assumptions made are analyzed in chapter two. Predictions for different types of internal combustion engines are made in chapter three. Transmission and integration issues are addressed in chapter four. The future of hybrids is discussed in chapter five. The results of the calculations are presented and interpreted in chapter six.

# 2. Non-Powertrain Improvements

## 2.1. Introduction

In this chapter, the potential for reducing the amount of energy a vehicle needs at the wheel will be analyzed. First, the underlying physics that determine the amount of energy per distance driven will be analyzed. This analysis will point out the areas where the opportunities for reducing that energy are. The methodology used to predict how much of this potential will be realized within the time frame of this analysis (25 years into the future) will be subsequently explained for each area.

## 2.2. Vehicle Resistances

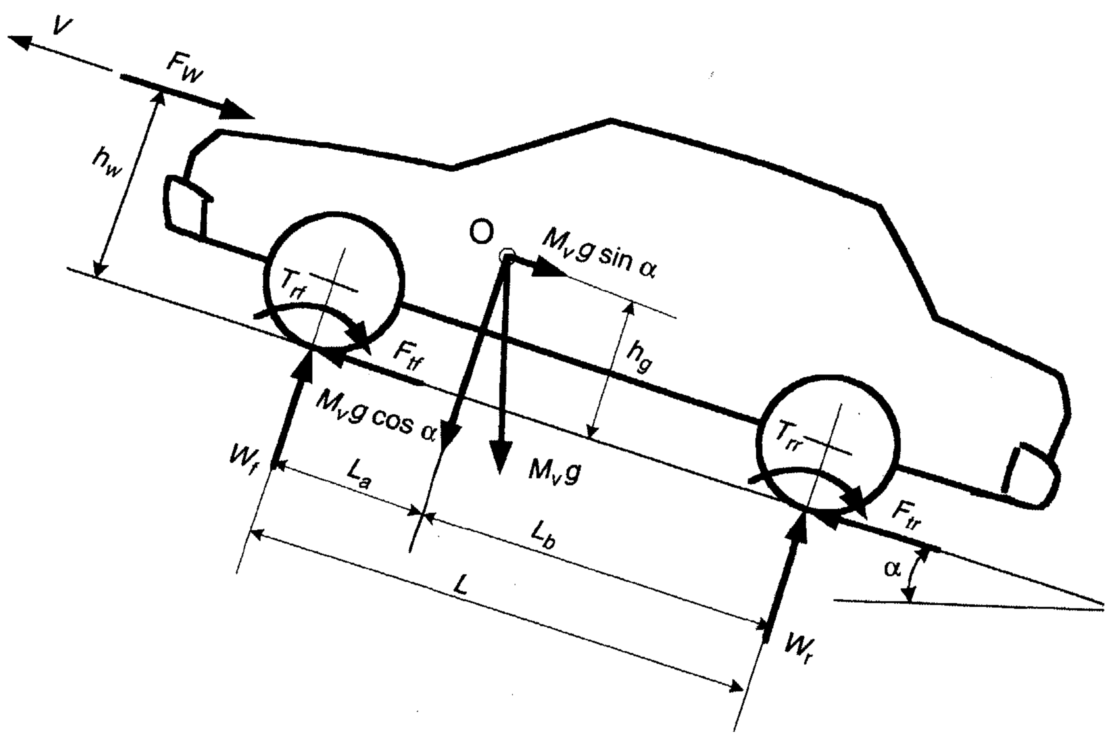



**Figure 13: Vehicle moving at a velocity V up a hill. Adapted from [4]**

Figure 13 shows a vehicle of mass $M_v$ traveling up a hill of slope $\alpha$ at a velocity V. Newton's second law for the vehicle is:

$$\delta * M_v * \frac{dV}{dt} = \sum F_t - \sum F_{tr} \quad (2.1)$$

Where $\Sigma F_t$ is the total tractive effort of the vehicle , $\Sigma F_{tr}$ is the sum of all vehicle resistance forces. $\delta$ is a number larger than one to account for rotating inertias.

As seen on Figure 13, vehicle resistances consist of:

- Rolling resistances: These are presented in the figure in the form of torques $T_{rf}$, and $T_{rr}$ on the frontal and rear wheels respectively. The total rolling resistance torque can be approximated as:

$$T_r = F_r * r_w = c_r * M_v * g * \cos\alpha * r_w \quad (2.2)$$

where $r_w$ is the wheel radius. The coefficient $c_r$ is called the rolling friction coefficient. It depends weakly on vehicle speed but can be assumed constant with good accuracy.

- Aerodynamic force $F_w$ : This is exerted on the center of aerodynamic pressure. It is generally approximated as:

$$F_w = \frac{1}{2} * \rho_{air} * c_D * A * V_w^{\ 2} \quad (2.3)$$

Where $\rho_{air}$ is the density of the surrounding air, $c_D$ is the coefficient of drag which depends on the vehicle shape and may be assumed independent of speed for the purposes of this analysis, A is the cross-sectional area of the vehicle and $V_w$ is the relative velocity of the wind with respect to the moving vehicle. For no wind conditions, it is the same as the vehicle velocity.

- Grade Resistance. This is apparently zero when driving up a flat road. The general expression is:

$$F_g = M_v * g * \sin\alpha \quad (2.4)$$

- Although it is not theoretically a resistance, the acceleration force should be considered one. It is not a resistance because the power used to accelerate the vehicle is not immediately dissipated as heat, as is the case with the other resistances. It does however increase fuel consumption because:
  - The kinetic energy due to the acceleration force is usually lost as heat when the vehicle needs to decelerate through conventional friction braking.
  - Even if the vehicle has regenerative braking capability, this kinetic energy can not be fully recovered as will be explained in chapter four.

  As already explained, the acceleration force, $F_{accel}$ is:

$$F_{accel} = \delta * M_v * \frac{dV}{dt} \quad (2.5)$$

The relative balance of rolling friction and aerodynamic drag is presented versus vehicle speed for the two baseline vehicles used in this study, the 2.5 liter 2005 Camry and the 4.2 liter F-150 in Figure 14 and Figure 15 respectively. It is obvious that rolling friction dominates at low speeds up to 50-60km/h and aerodynamic resistance at higher ones. If a road grade G=tanα is additionally imposed on the vehicle, the total resistance force can be seen in Figure 16.

The instantaneous total resistance for the 2005 2.5 liter Camry over the FTP (urban driving) and HWFET (highway) cycles are presented in Figures 17 and 18.It is obvious that acceleration forces dominate urban driving while aerodynamic forces dominate highway resistances. The required energy *at the wheel* to overcome each resistance can be calculated by integrating the required power over the driving cycle:

$$E = \int_{drivngcycle} Pdt = \int_{drivingcycle} F * udt \quad (2.6)$$

The results from calculating this integral over the two driving cycles are presented in Table 2. Note that in calculating this integral for acceleration, only the negative values of $F_{accel}$ were included in the calculation of the above integral as that is essentially the energy lost through braking.

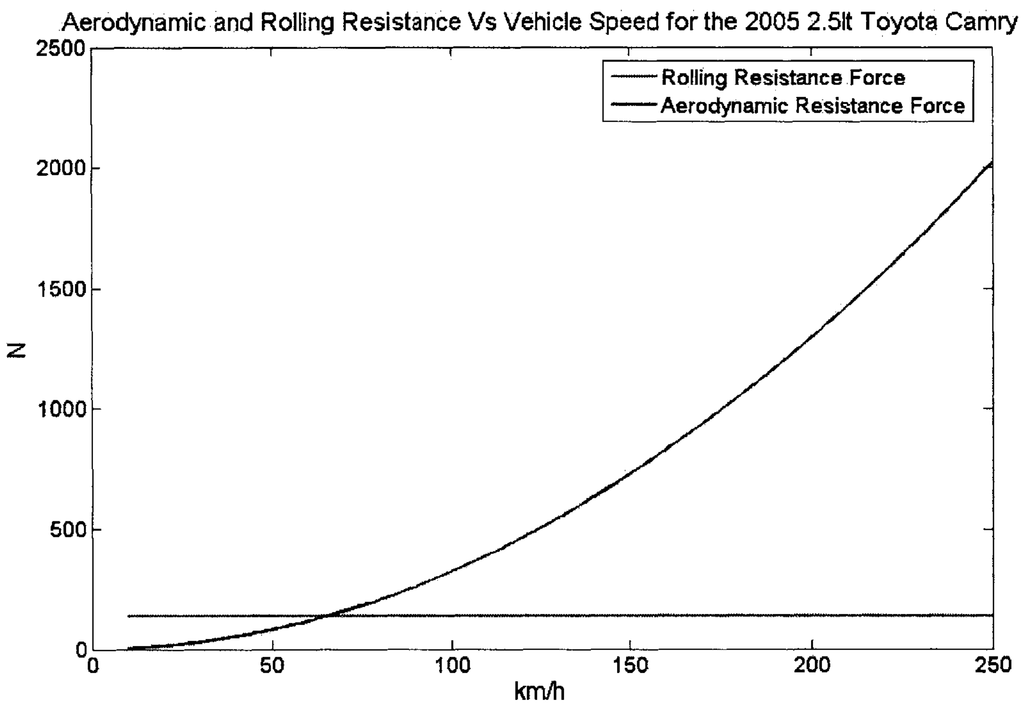



**Figure 14: Aerodynamic and Rolling Resistances versus Vehicle speed**

**For the 2005 2.5 liter Toyota Camry**

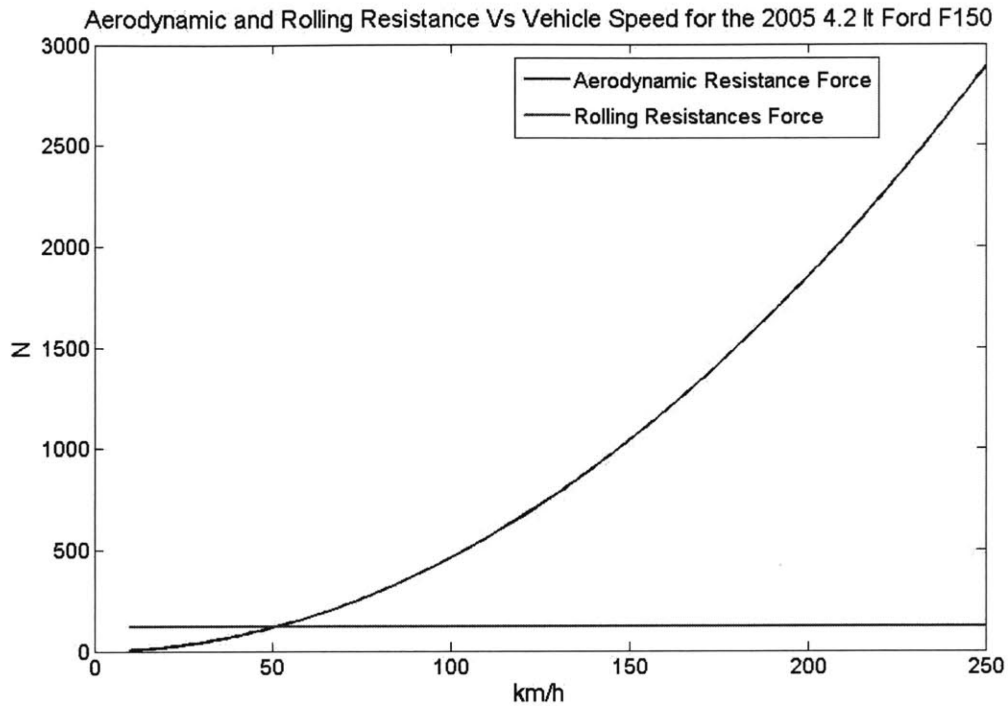



**Figure 15 : Aerodynamic and Rolling Resistances versus Vehicle speed for the 2005 4.2 liter Ford F150**

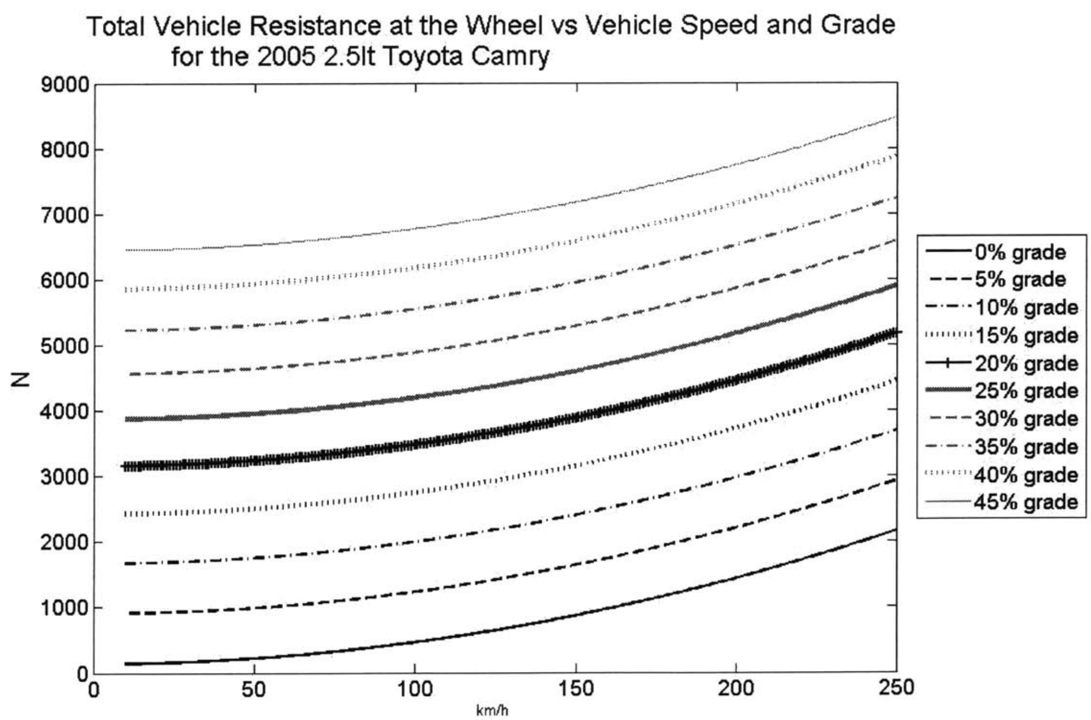



**Figure 16: Vehicle Resistances Including Grade for the 2005 2.5 liter Toyota Camry.**

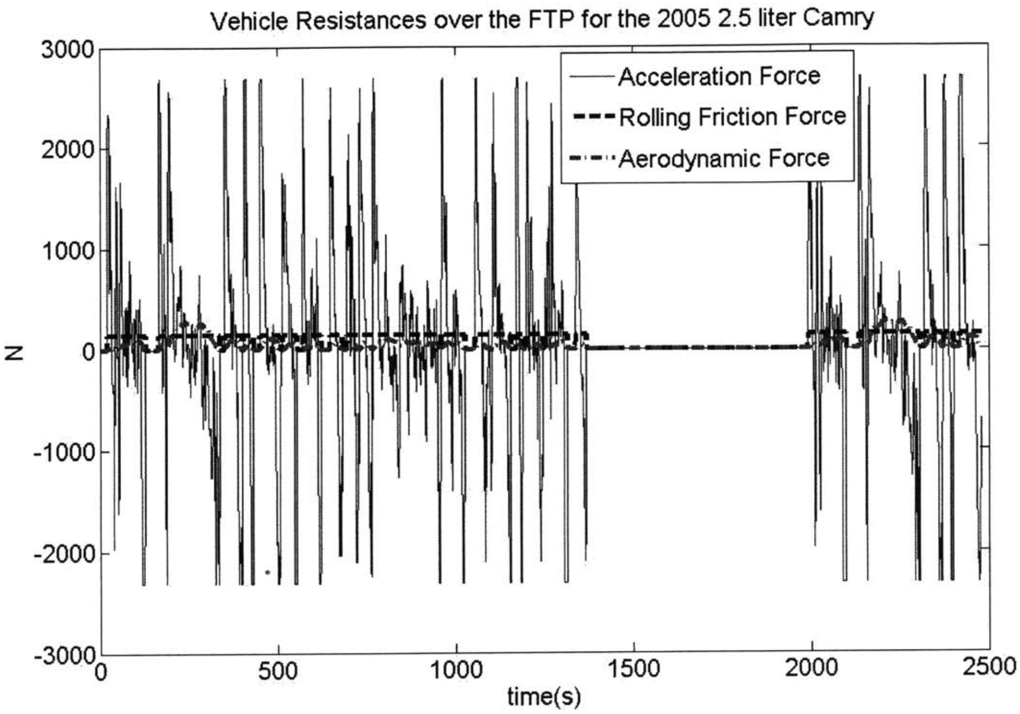



**Figure 17: Vehicle Resistances over the FTP driving Cycle for the 2005 2.5 liter Camry**

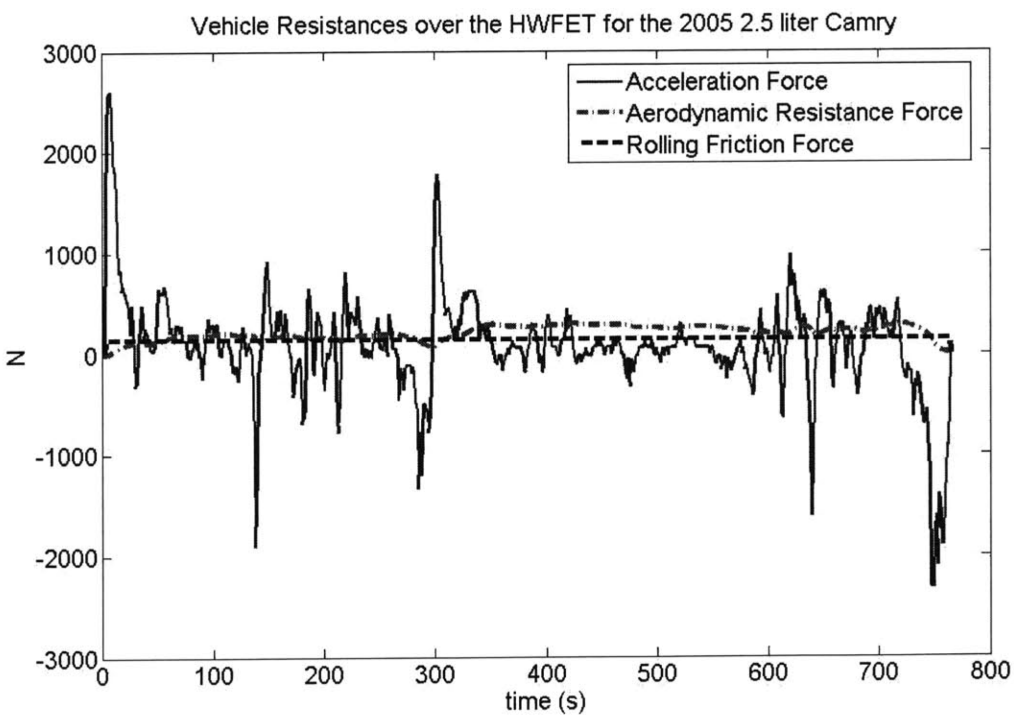



**Figure 18: Vehicle Resistances over the HWFET driving Cycle for the 2005 2.5 liter Camry**

**Table 2: Energy required at the Wheel due to different Resistances**

| | FTP | | HWFET | |
|---|---|---|---|---|
| | MJ | % over Total Energy at the Wheel | MJ | % over Total Energy at the Wheel |
| Energy to overcome Aerodynamic Drag | 1.9 | 21% | 3.6 | 47% |
| Energy to overcome Rolling Friction | 2.5 | 27% | 2.3 | 30% |
| Braking Energy | 4.8 | 52% | 1.8 | 23% |

## 2.3. Improvements in Vehicle Aerodynamics

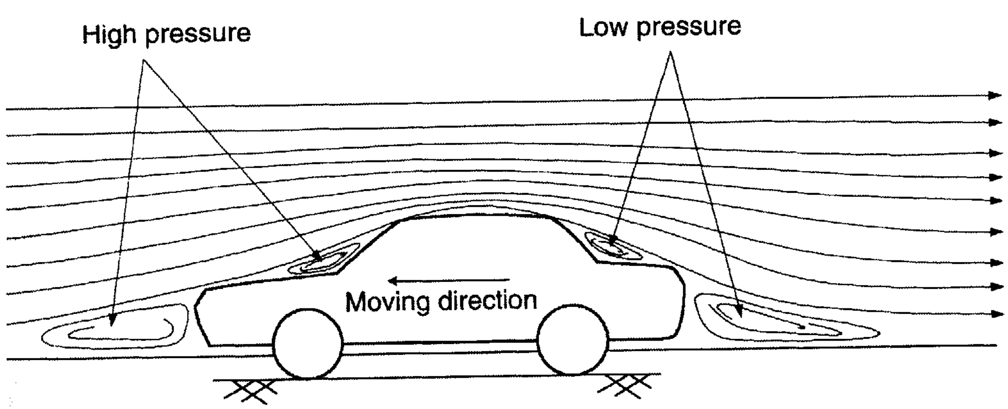



**Figure 19: Streamlines around a Moving Vehicle**

As seen in Figure 19, the flow behind a moving vehicle creates a low-pressure wake. It is mainly this difference of pressures in front of and behind the vehicle created by the flow that causes aerodynamic drag. The effect of size is included in the cross-sectional area A, while the effect of geometry is included in the coefficient of drag, $c_D$ in equation (2.6).

The definition of cross-sectional area is the projected area of a body in the direction of the flow. It is usually approximated as

$$A = C * h * W \quad (2.7)$$

Where h is the overall vehicle height, W is a measure of width and C is a coefficient slightly less than one. [17] suggests a C of 0.81 when using the vehicle overall width as defined in Figure 20. [18] suggests 0.9 when using the tread width which is the distance between the left and right tire. In this study, it will be assumed that the size of future vehicles is the same as that of current ones. As a result, it will be assumed that the cross-sectional area remains the same. This was not the case in [7] where it was assumed that the cross-sectional area was decreased in the future. The correlation between vehicle weight and cross-sectional area is presented in Figure 20 for European cars.

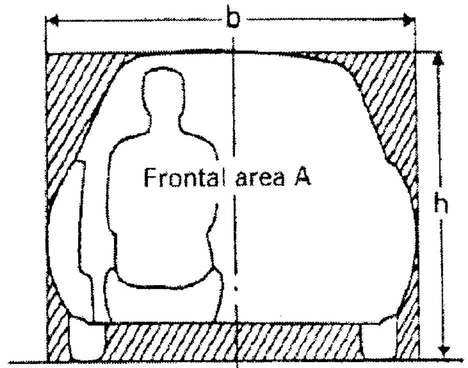
Since cross sectional area is constant, reduction in aerodynamic drag comes exclusively from $c_D$.This reflects improvements in vehicle design based on the principles of aerodynamics. An overview of estimated annual rates of reduction in $c_D$ as well as historical trends for the models of several car manufacturers is presented in Table 3. Plots with the historical evolution of $c_D$ for the models of several automakers and an average for European passenger cars versus experimental models are presented in Figures 21,22 and Figure 23.

It is clear that on the laboratory/prototype level, achieving a vehicle design with a very low $c_D$ is significantly easier than achieving the same cd in a production vehicle. The design limitations of real mass-produced vehicles are much more complicated. The drop shaped vehicle geometry with a cd of 0.14 at the lower left side of Figure 23 has been known since the 30's. A real car with such a design however would be very uncomfortable for the passengers in the back seats. Similarly, lowering the vehicle closer to the road would reduce aerodynamic drag but also the practicality of the vehicle. Although vehicle design with respect to aerodynamics is already quite sophisticated today, for instance some automakers are careful even about the geometry of the undercarriage; there are still some significant improvements in vehicle aerodynamics that should be expected. For example, replacement of mirrors with cameras gives a ~8.5% reduction in $c_D$ as shown in [19].Considering all of the above; a linear 1% per year improvement in $c_D$ was assumed for the Toyota Camry. Extrapolating this over 25 years into the future, the current published [20] $c_D$ of 0.28 for the Camry becomes 0.21, a 25% reduction.

Things are less clear on the assumptions that should be made for the Ford F150. Truck aerodynamics are generally given less attention than those of family cars. Ford doesn't publish official $c_D$ data for the F150, so there was also the issue of what value should be assumed for modeling of the current F150. In the [21] wind tunnel measured data for several trucks were published. The $c_D$ for the 2001 and 2002 F150 varied between 0.48 and 0.55 depending on the trim, year and position of the tailgate. A value of 0.52 was used in the modeling.

On the future evolution of truck aerodynamics, it should initially be observed that pickups have inherently worse geometry. The cargo box leads unavoidably to the creation of a large wake. It could be argued therefore that not a lot can be done to improve truck aerodynamics. On the other hand, however not a lot has been done in the past. Small changes like an improved tailgate design, or a tonneau cover easily achieve -5 to -10% reductions as seen in [21] . For simplicity in this study, the same percentage reduction (25%) was assumed for the Camry and the F150. This led to a $c_D$ of 0.38 for the truck.

a)




| Car Class | Frontal area A $m^2$ |
|---|---|
| mini | 1.8 |
| medium size | 1.9 |
| upper medium size | 2.0 |
| full size | 2.1 |

$$A \approx 0.81 \cdot (b \cdot h)$$

b)

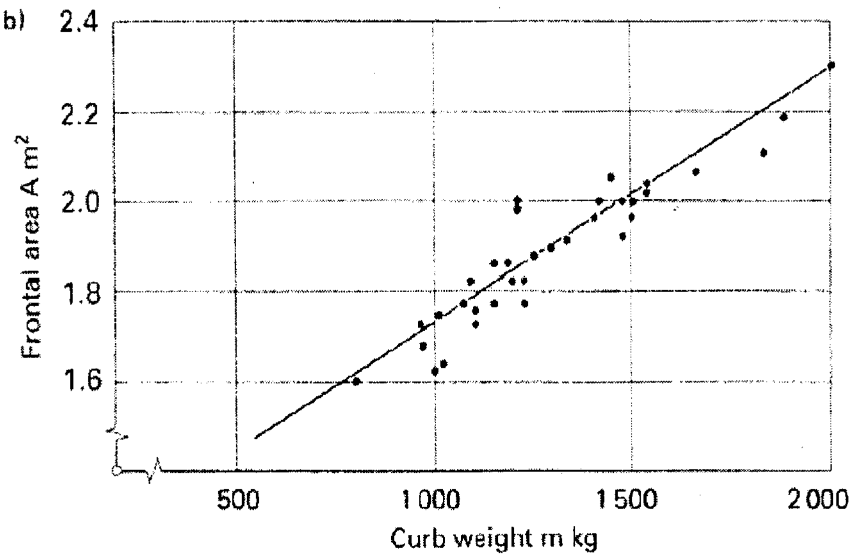



**Figure 20: Cross-Sectional area, definition, approximation and correlation with vehicle size.**

**Source: [17]**

**Table 3: $C_D$ Improvement. Projections and Historical Trends**

| Source | Historically | Estimate for the future | How far ahead? | annually | |
|---|---|---|---|---|---|
| DeCicco, An and Ross [13] | -2.5% per anno | -10% | 10-15 years | 0.80% | |
| On the Road in 2020 [7] | | -18 -33% | 20 years | 0.90% | 1.65% |
| Toyota published for Camry 1996-2005[20] | | | | 0.56% | |
| Opel 1986-90 [17] | | | | 0.36% | |
| Ford 1990-95 [17] | | | | 1.13% | |
| Chrysler 1990-95 [17] | | | | 1.96% | |
| GM 1990-95 [17] | | | | 1.20% | |

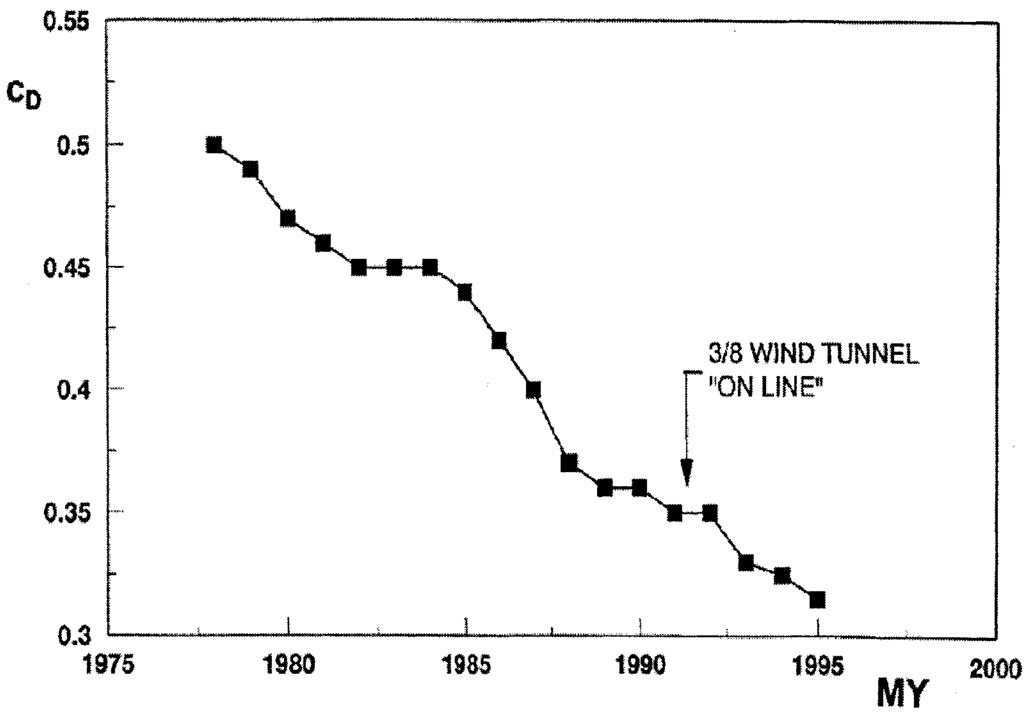



**Figure 21: $C_D$ Evolution for Chrysler Models. Source:[17]**

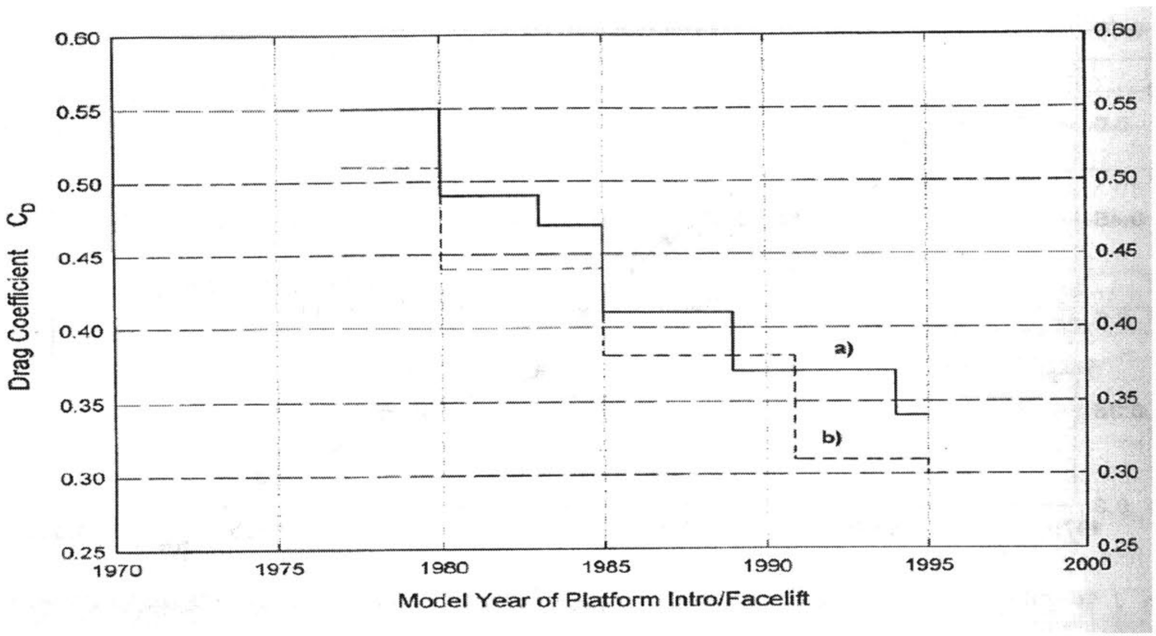



**Figure 22 Coefficient of Drag Evolution for GM Sedans. Source: [17]**

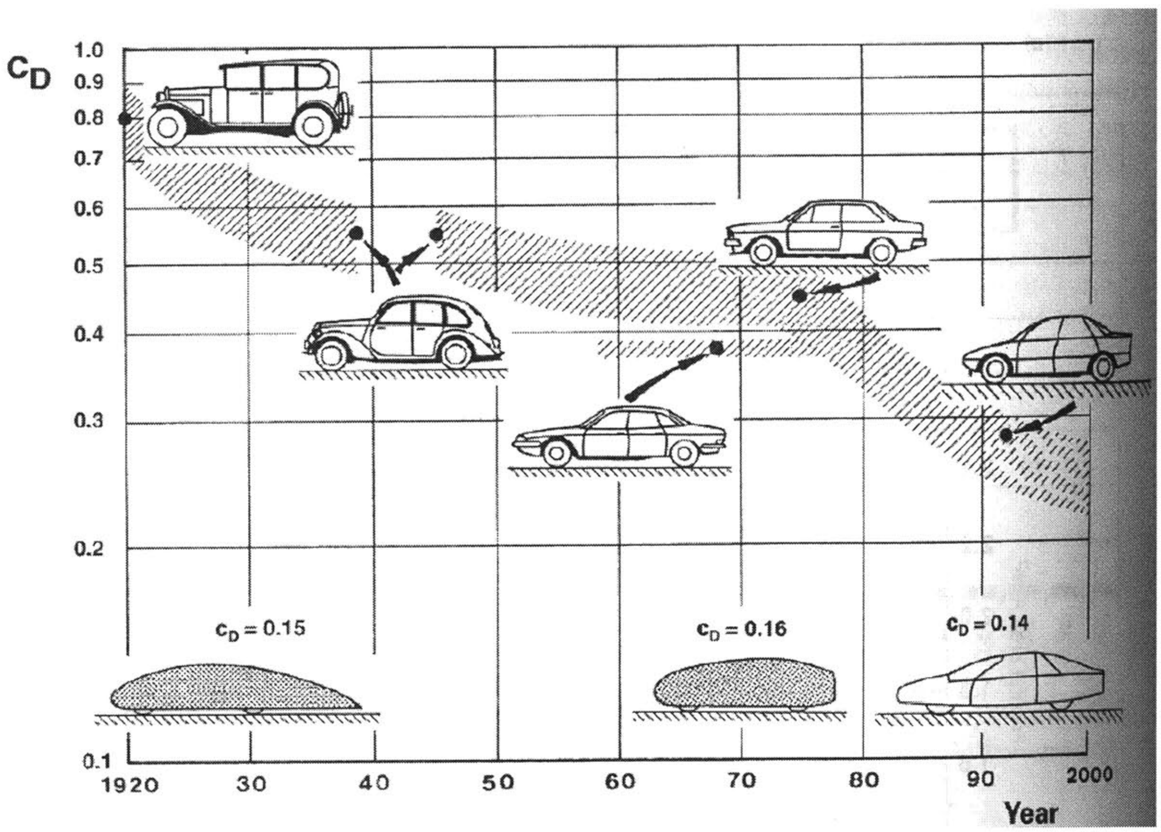



**Figure 23: History of $C_D$ evolution for European passenger Cars compared with experimental prototypes. Source: [17]**

## *2.4.Improvements in Tire Rolling Friction*

Tire rolling friction is caused mainly by hysteresis in the tire rubber due to deformation which causes a deviation from the shape of a perfect circle for the tire. This may be seen in Figure 24. Higher weight loading and lower tire pressures will therefore increase this deformation and thus rolling friction. Simply checking tire pressure more often is the simplest means of reducing friction. Technological improvements in tire materials and design could also significantly reduce rolling friction.

Two divisions of the National Academies—the Transportation Research Board and the Board on Energy and Environmental Systems—have recently issued a special report examining the contribution of tires to vehicle fuel consumption and the prospects for improving tire energy performance without adversely affecting tire life, traction capability, and retail prices. This study, [22] concludes that reducing the average rolling resistance of replacement tires by a magnitude of 10% is technically and economically feasible today.

According to the report, the large majority of new passenger tires, properly inflated, have rolling resistance coefficients ($c_r$) ranging from 0.007 to 0.014, with most having values closer to the average of about 0.01. However, some of the data sets used in this study, included tires currently on the market with rolling friction coefficients as low as 0.005.Furthermore,Table 4 presents estimates from a couple of sources on future evolution of rolling friction coefficient that also conclude that significant reductions in rolling friction are feasible. Taking all of the above into account, the friction coefficients were assumed to be 0.009 and 0.0105 for the current Camry and F150 and 0.006 and 0.007 for their future counterparts respectively.

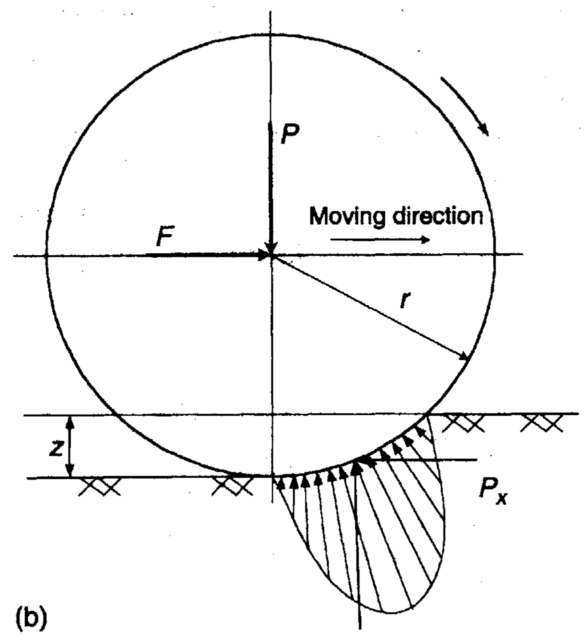



**Figure 24: Tire deflection and rolling resistance**

**Table 4: Estimates for Future Rolling Friction Coefficient Reduction**

| Source | Estimate for the future | How far ahead? | Annually |
|---|---|---|---|
| DeCicco, An and Ross [13] | 15-30% reduction potential in $c_r$-20% assumed | 10-15 years from 2001 | 1.60% |
| On the Road in 2020 [7] | 11-33% | 20 years from 2000 | 1.10% |

## 2.5. Vehicle Weight Reduction.

It was already shown in chapter 1 that vehicle weight has been increasing in an average sense over the last decades in the U.S.A. The technical capability to design lighter weight vehicles while keeping size and safety constant has, as expected, in fact improved over this time. None of that potential was however realized due to consumer trends linked with the increase in average vehicle size as well as passive safety. In this study we will neglect the effect of consumer trends by assuming that size and safety performance are kept constant at today's levels and make reasonable assumptions about the amount of technological weight reduction that can be achieved.

There are 3 main ways to reduce vehicle weight while keeping size constant and offering the same level of passive safety:

- *Advanced materials*: Increased use of advanced materials can lead to significant weight decrease. The most promising materials are advanced steel, aluminum and magnesium. There are mass produced all-aluminum car models already on the market today such as the Audi A2 and A8. Aluminum and magnesium are however significantly more expensive than steel. Additionally, there are issues related with the large amount of electric energy needed for their production if the ultimate goal is reduction of energy demand and carbon emissions. Figure 25 shows the potential for weight reduction on a vehicle curb weight basis resulting from the increased use of advanced materials according to the Freedom Car Project[1] [23]. An even more exotic solution than aluminum that has been proposed is carbon fiber composites. The costs associated with their wide use in cars are however significantly higher than those associated with aluminum. Furthermore, the broader view today is that bringing down composite costs will be a much greater challenge than doing so for any of the other advanced

[1] The Freedom car project is a long term partnership in the United States organized by the Department of Energy (DOE) to bring together people from the automotive industry to set research goals for the car of the future.

materials. An additional drawback of composite materials is that there is no easy way to recycle them.

- *Advanced Designs:* Even without the use of advanced materials advances in design lead to weight reductions. An example of how improved engine design decreases engine weight is presented in [24].
- *Secondary benefits:* Reducing the weight of the vehicle body results in less power required from the powertrain to achieve the same levels of performance. As a result the powertrain components become smaller, achieving further weight reduction.

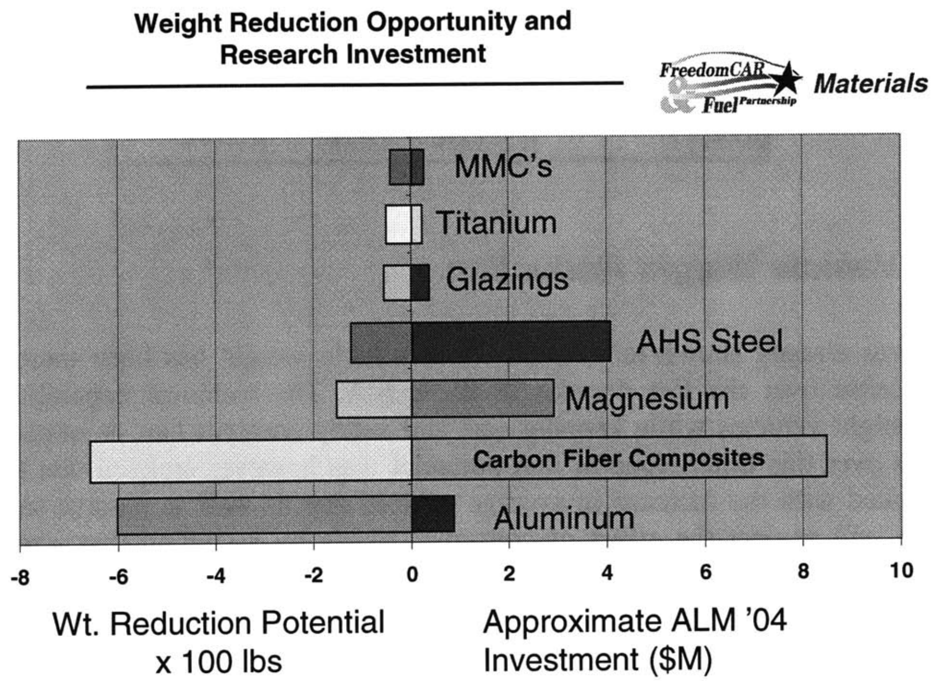



**Figure 25: Weight Reduction Opportunity for Different Materials at the vehicle level.**

A sample of weight reduction predictions for future vehicles in the literature is presented in Table 5. The predictions range from 10-33%. Constant size and safety are usually implied. The analytical prediction of weight reduction for every vehicle component in the "On the Road in 2020"-[7] study is presented in Table 6. There have been studies in the literature however that suggest reductions of 50% by extensive use of carbon fiber composites [25] Due to the aforementioned difficulties associated with widespread use of composites a more conservative 20% reduction in curb weight[2] was assumed for all the future gasoline engine vehicles at constant size and safety. Adjustments were made on this base assumption for different powertrains. The adjustment methodology will be explained further in chapters 3 and 4.

It should finally be noted that the reduction in the energy required for acceleration at the wheel turned out to be less than the 20% assumed in curb weight reduction. This was due to two reasons. The first is that the 20% reduction was assumed for vehicle curb

[2] The curb weight of a vehicle is the total weight of a fully equipped vehicle without the passenger.

weight. For standard vehicle testing however there is an additional mass of 136 kg added to account for the weight of the driver plus luggage. This brings the total (including passenger) weight reduction down to 17-18%. Additionally, the energy required for acceleration depends on rotational inertias of the powertrain components. These account for an additional 3-6% energy required for acceleration. The default rotational inertias in the ADVISOR built in models were used for both current and future vehicles. The rotational inertia of the engine was scaled with engine volume. The other rotational inertias were however left the same. Weight reduction would probably reduce rotational inertias as well; there might be therefore a need to correct for this effect. The effect on vehicle fuel consumption of this additional 3-6% which should have been reduced by around 20% is small.

**Table 5: Samples of Predicted Weight Reduction for Future Vehicles in the Literature**

| **Source** | **Weight Reduction Estimate for the future** | **How far ahead?** | **Notes** |
|---|---|---|---|
| SAE 2001-01-2482,DeCicco,An and Ross | Small Car 0-10% Midsize Car 10%-20% Minivan, Pickup, SUV 20%-33% | 10-15 years from 2001 | |
| Life Cycle Energy Savings Potential from Aluminum-Intensive Vehicles | ~20%(Including Engine Downsizing) | | Slightly old (1995) but Very well Documented Study |
| OTR 2020 | 16-24% | 2020 | |

**Table 6: Estimated Weight Reduction per Component for Future Vehicles with Different Powertrains from the "On the Road in 2020" Study**

| Technology | current | baseline | advanced | advanced | advanced | advanced | advanced | advanced | advanced | advanced | advanced |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Propulsion System | SI ICE | SI ICE | SI ICE | CI ICE | SI Hybrid | CI Hybrid | SI Hybrid | FC Hybrid | FC Hybrid | FC Hybrid | Electric |
| Fuel | gasoline | gasoline | gasoline | diesel | gasoline | diesel | CNG | gasoline | methanol | hydrogen | electr. |
| Transmission | auto | auto-clutch | auto-clutch | auto-clutch | CVT | CVT | CVT | direct | direct | direct | direct |
| Body | 383 | 326 | 249 | 249 | 249 | 249 | 249 | 249 | 249 | 249 | 249 |
| Glazing | 35 | 33 | 33 | 33 | 33 | 33 | 33 | 33 | 33 | 33 | 33 |
| Chassis | 273 | 229 | 216 | 219 | 216 | 219 | 216 | 275 | 25459 | 244 | 243 |
| Propulsion System | 392 | 263 | 252 | 303 | 267 | 303 | 283 | 536 | 475 | 416 | 414 |
| Engine | 164 | 103 | 95 | 149 | 64 | 99 | 67 | 0 | 0 | 0 | 0 |
| Electric Motor | | | | | 19 | 20 | 20 | 73 | 69 | 66 | 66 |
| Fuel Cell System & Reformer | | | | | | | | 351 | 278 | 193 | |
| Battery | 12 | 12 | 12 | 12 | 36 | 37 | 37 | 46 | 43 | 41 | 328 |
| Transmission | 90 | 50 | 50 | 50 | 50 | 50 | 50 | 20 | 20 | 20 | 20 |
| Liquids and Storage | 64 | 45 | 42 | 39 | 34 | 31 | 46 | 33 | 53 | 84 | |
| Other (Accessories, Electronics, etc.) | 62 | 53 | 53 | 53 | 64 | 64 | 64 | 14 | 13 | 12 | |
| Interior & Exterior | 195 | 214 | 214 | 214 | 214 | 214 | 214 | 194 | 194 | 194 | 194 |
| Other | 44 | 44 | 44 | 44 | 44 | 44 | 44 | 44 | 44 | 44 | 44 |
| TOTAL VEHICLE | 1322 | 1108 | 1007 | 1062 | 1023 | 1060 | 1039 | 1330 | 1253 | 1179 | 1176 |

*Not represented is the assumed compensating effect of declining interior mass (seats, trim, etc.) and increasing in body mass for improved crash safety.

# 3. Internal Combustion Engines

## *3.3. Introduction*

It has already been discussed why ICE powertrains burning liquid hydrocarbon fuels are so well established that most likely near and mid-term alternative powertrains will still rely exclusively or primarily on a internal combustion engine. It is therefore essential to carefully estimate their potential for improvement.

More specifically, the U.S. light duty fleet, which is the focus of this study, is almost exclusively powered by naturally aspirated spark-ignition (NA SI, gasoline) engines. Again, there are reasons that account for this. Diesel technology until relatively recently has been significantly dirtier in terms of most criteria pollutants, noisier and still is more expensive. Stricter emissions requirements as well as lower fuel prices haven't allowed for significant penetration of the inherently more fuel efficient diesels in the U.S. light duty fleet as in Europe. Gasoline engines are the real technological baseline in the U.S.

However, although NA SI engines are cheaper and cleaner, they do suffer from one major drawback. Load control in an SI engine is achieved through throttling the air and fuel charge that goes into the cylinders. As a result, SI engines are inherently inefficient at partial load. The essence of the problem arises from the fact that in order to meet the market's ever increasing appetite for performance; engines are greatly oversized compared with their average duty cycle during everyday driving. Consequently, current SI Powertrains end up operating most of the time exactly were they are the most inefficient.

As already explained in chapter one, there are two ways to reduce automotive fuel consumption: Improving the efficiency of the powertrain or reducing the energy demand at the wheel.

Powertrain solutions to the problem that still rely exclusively on an ICE can be divided in two main categories:

1. Engine technologies that exhibit much better partial load efficiencies.
2. Powertrain technologies that prohibit the engine from operating at low loads.

The first category includes advanced diesel engines and HCCI hybrids. Both of these technologies rely on different concepts of combustion and thus eliminate throttling. Diesel engines have experienced several major technological breakthroughs in the last 15-20 years. With the help of higher fuel prices and more lenient emissions requirements in Europe, diesels are now a large fraction of the light duty vehicle fleet. This could also possibly become the case in the U.S. Future advanced diesel engines will be examined thoroughly in this chapter. HCCI is a novel combustion technology that combines the merits of the diesel and Otto cycles. It has the potential of offering diesel level fuel consumption with spark ignition level pollutant emissions. It is still at the research level but it could be promising. HCCI technology will not be investigated further in this thesis. As criteria pollutants are not investigated for this study, the reader may consider HCCI in terms of fuel consumption as a data point in between diesel and advanced gasoline.

The second category includes heavily turbocharged gasoline engines, advanced transmissions and hybrids. Turbocharging enables downsizing the engine displacement volume while keeping power output and thus performance of the vehicle constant. Engine friction roughly scales with piston area. Compared with a naturally aspirated engine of the same displacement volume, the turbocharged engine has about the same friction but higher power output. Its efficiency at the same partial load is therefore higher. Advanced transmissions such as continuously variable transmissions enable more control over the engine operating points. The required torque and speed from the engine may be set almost independently of the required torque at the wheel. The engine can thus operate where it is more efficient. Both of these technologies will be examined thoroughly in this chapter. Hybrid powertrains can also achieve very effective optimization of engine operating points. This is only one of the ways they increase fuel economy. Due to their complexity and special characteristics, they will be analyzed in a separate chapter.

## 3.2. Advanced Naturally Aspirated Gasoline Engines

There are two main questions that need to be answered in order to simulate future engine operation:

1. How will their efficiency improve?
2. How will their volumetric and gravimetric power density improve?

### 3.2.1. Engine Efficiency Map Evolution

#### Methodology

When discussing engine improvements, it is useful to decouple thermodynamic effects from engine friction. More accurate predictions can subsequently be made, depending on what technological improvements are expected in each field.

It is essential, initially, to normalize engine power output by volume. Predictions can thus be made independently of changes in size. The mean effective pressure (mep) is defined as the constant (theoretical) pressure inside the cylinder that would produce the same power output from the same displacement volume, $V_d$ as the real (variable) pressure in an engine. It is essentially engine torque normalized by volume:

$$P = V_d * mep * \tau_p = V_d * mep * \frac{N}{2} = \mathrm{T} * \omega = \mathrm{T} * 2 * \pi * N \Rightarrow$$
$$\Rightarrow mep = \frac{4 * \pi * \mathrm{T}}{V_d} \tag{3.1}$$

where:

N: is the engine frequency of revolution. It is divided by two because a 4 stroke engine fires only every 2 revolutions.

T: the engine torque output in N*m

ω: engine rotational speed

$\tau_P$: The period of the power stroke of the engine

Depending on which definition of engine power is being used, mep can be either brake (bmep) when the actual brake power ($P_b$) measured at the engine output shaft is used ,or indicative based on the power output of the engine if there were no friction.

Indicative power ($P_i$) is usually calculated from in-cylinder pressure measurements. If a friction mep (fmep) is defined in the same manner to account for the torque lost because of friction:

$$imep = bmep + fmep \quad (3.2)$$

It should be noted at this point that the imep definition used in equation (3.2) is known as *gross* indicative mep. This definition assumes that the pumping losses required to pump the charge in and out of the cylinders as well as the losses resulting from driving the auxiliary components needed to run the engine are incorporated in fmep.

If the corresponding efficiencies for indicated and brake power are defined as well as a mechanical efficiency ($\eta_m$) for the friction losses:

$$\eta_i = \frac{P_i * \tau_P}{m_f * LHV} = \frac{imep * V_d}{\dot{m}_f * LHV} \quad (3.3)$$

$$\eta_m = \frac{P_b}{P_i} = \frac{bmep * V_d}{(bmep + fmep) * V_d} \quad (3.4)$$

$$\eta_b = \frac{P_b * \tau_P}{m_f * LHV} = \frac{bmep * V_d}{\dot{m}_f * LHV} = \eta_i * \eta_m = \eta_i * \frac{bmep}{bmep + fmep} \quad (3.5)$$

Using equation (3.5), the metric of interest when calculating fuel consumption (i.e. $\eta_b$) is expressed as a function of indicative efficiency and friction at every load (bmep). Thermodynamic and friction effects can be thus dealt with separately.

It is a fairly good assumption to use a constant $\eta_i$ value across the entire engine map. In reality, of course, it varies slightly both with torque and speed, but this assumption is sufficient for the purposes of this analysis. For details on a detailed prediction of indicative efficiency the reader is referenced to [26]

Additionally, it is known that fmep is mainly a function of rpm. A semi-empirical model that predicts this function given the technical characteristics of the engine (geometry, type of lubricant etc) is described in [27]. This model will be used in this thesis. The model breaks down friction in its components:

- rubbing losses from the crankshaft, reciprocating, and valvetrain components
- auxiliary losses from engine accessories
- pumping losses from the intake and exhaust systems

Given the above, the methodology to predict future engine maps can be summarized as follows:

1. Normalizing maps of the current baseline vehicles using bmep.
2. Decomposing the engine normalized maps into a combination of $\eta_i$ and fmep as a function of rpm.
3. Assuming a reduction in future fmep as a function of rpm and an improvement in $\eta_i$ compared with the current baseline values.

4. The future normalized maps may subsequently be predicted by recombining the $\eta_i$ and fmep as a function of rpm improved by a certain percentage.
5. The future engine's desired power output, displacement volume and new rpm range is determined from the *scaling calculation* described in the next chapter.
6. The future engine efficiency maps are expressed as a function of torque and the new rpm range.

## Decomposing Current Engine Maps

Using the friction model, equations (3.1-3.5) and assuming a value for $\eta_i$, the brake efficiency map of the engines in the current baseline vehicles, the Toyota 2.5 liter Camry, the 3.0 liter Camry and the Ford 4.2 liter F150, may be predicted. In order to do that, the detailed technical characteristics of these engines need to be known. They are presented in Table 7.

**Table 7: Engine Technical Characteristics**

| | Engine | | | |
|---|---|---|---|---|
| | Toyota | Toyota MZFE | Ford | Ford Triton |
| **Total Displacement volume (cc)** | 2357 | 2990 | 4203 | 5408 |
| **maximum Power(hp)** | 160@5600 | 210@5800 | 202@4,350 | 300@5,000 |
| **maximum Torque(N*m)** | 221@5600 | 298@4400 | 260@3,750 | 365@3,750 |
| **Bore(mm)** | 88.392 | 87.376 | 96.774 | 90.2 |
| **Stroke(mm)** | 96.012 | 83.058 | 95.25 | 105.8 |
| **Cylinders** | 4 in Line | V6 | V6 | V8 |
| **Compression Ratio** | 9.6 | 10.5 | 9.2 | 9.8 |
| **Valvetrain** | Double Overhead Camshaft | Double Overhead Camshaft | Double Overhead Camshaft | Double Overhead Camshaft |
| **Follower Type** | Roller | Roller | Roller | Roller |
| **Valvetrain Mechanism** | Direct | Direct | Direct | Direct |
| **No Intake Valves** | 2 | 2 | 2 | 2 |
| **No Exhaust Valves** | 2 | 2 | 1 | 1 |
| **Intake Valve diameter(mm)** | 27.14 | 27.14 | 33.8 | 33.8 |
| **Exhaust valve diameter(mm)** | 26.22 | 26.22 | 37.5 | 37.5 |
| **Valve lift(mm)** | 10 | 10 | 11 | 11 |
| **Oil Type** | 10W30 | 10W30 | 10W30 | 10W30 |
| **Temperature of Oil (Celsius)** | 90 | 90 | 90 | 90 |
| **Idle** | 500 | 500 | 500 | 500 |
| **redline** | 6000 | 6000 | 5000 | 5000 |

Some of these characteristics were found in [28] or provided by Ford. The rest are reasonable estimates.

Using these numbers and the equations from the friction model, the fmep as a function of rpm is estimated. For the case of the 5.4 liter F150 engine this function is presented in Figure 26. The curves for the other engines are similar.

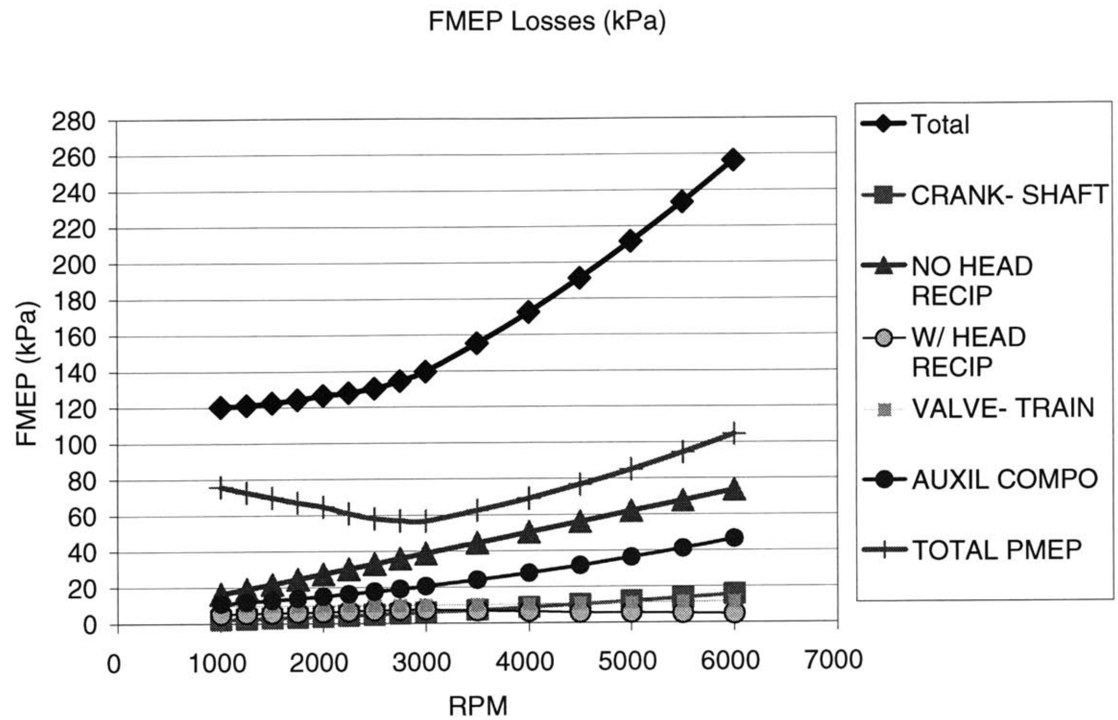



**Figure 26: Fmep losses as a function of rpm. 5.4 liter Ford V8 Triton Engine used in the 2005 F150.**

Using the fmep functions predicted from the model, $\eta_i$ values were chosen to give the best prediction of the engine consumption map. The results were cross checked with the actual values measured for the Ford 5.4 liter engine (kindly provided by Ford) and the 3.0 liter Camry engine was published in [29] .The measured engine maps of the 2.5 liter Camry and the 4.2 liter F150 were not known , it was therefore assumed that the indicative efficiency of the 4.2 liter F150 engine is the same as the 5.4 liter version and that of the Camry 2.5 liter engine is slightly higher then that of the 3.0 liter Camry. The latter was chosen to get the best agreement between simulated and published vehicle fuel economy numbers and is also indicative of the fact that the 2.5 liter engine is newer (2002 design versus 1992 design!!)

The Camry engine is presented in Figure 27. The metric used in this map instead of efficiency, is break specific fuel consumption (bsfc) in grams of fuel over kWh of energy output. Bsfc is inversely proportional to $\eta_b$:

$$bsfc[g / kWh] = \frac{3600}{LHV * \eta_b} \quad (3.6)$$

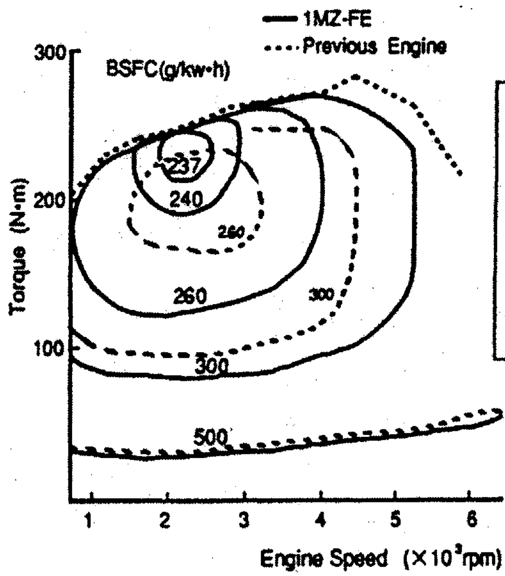



*Fig. 11 BSFC map*

*Table. 1 Engine Specifications*

| | |
|---|---|
| Name | 1MZ-FE |
| Type | Water-cooled, gasoline, 4-cycle |
| Displacement (cc) | 2995 |
| Arrangement & No. of Cylinders | 6-cylinder, V-type |
| Type of Combustion Chamber | Closs-flow, pentroof |
| Valve mechanism | 4-valve, DOHC, Belt & gear drive |
| Fuel system | EFI |
| Bore × Stroke (mm) | 87.5 × 83 |
| Compression ratio | 10.5 : 1 |
| Max. power (Kw/rpm) | 143/5200 |
| Max. torque (Nm/rpm) | 285/4400 |
| Dimensions (L × W × H mm) | 680 × 690 × 695 |

**Figure 27: 3.0 liter Toyota Camry Engine, BSFC Source: [29]**

The values of indicative efficiency that gave the best fit with the measured engine maps/published fuel economy data are presented in Table 8. The engine maps predicted this way are presented in Figures 29 to 32 and the relative error in Figures 33 and 34.

**Table 8: Chosen Indicative Efficiencies to predict Engine Maps**

| Engine | Chosen ind. efficiency |
|---|---|
| Camry 3.0 liter | 39% |
| Camry 2.5 liter | 40% |
| F150 5.4 liter | 40% |
| F150 4.2 liter | 40% |

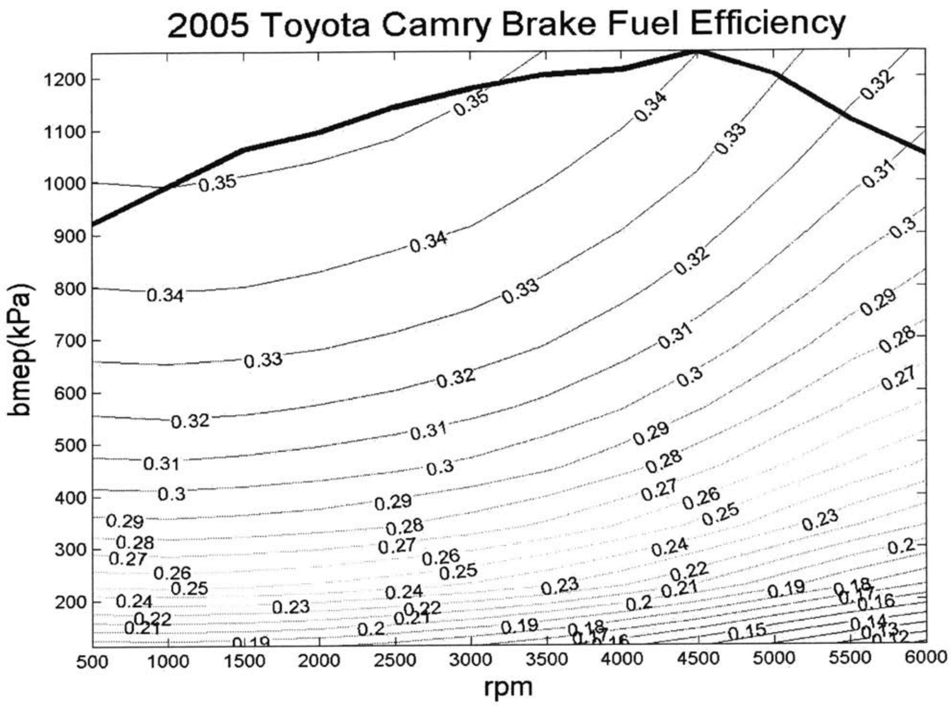



**Figure 28: Normalized Engine Brake Efficiency Map predicted by the friction model.2005 3.0 liter Camry Engine**

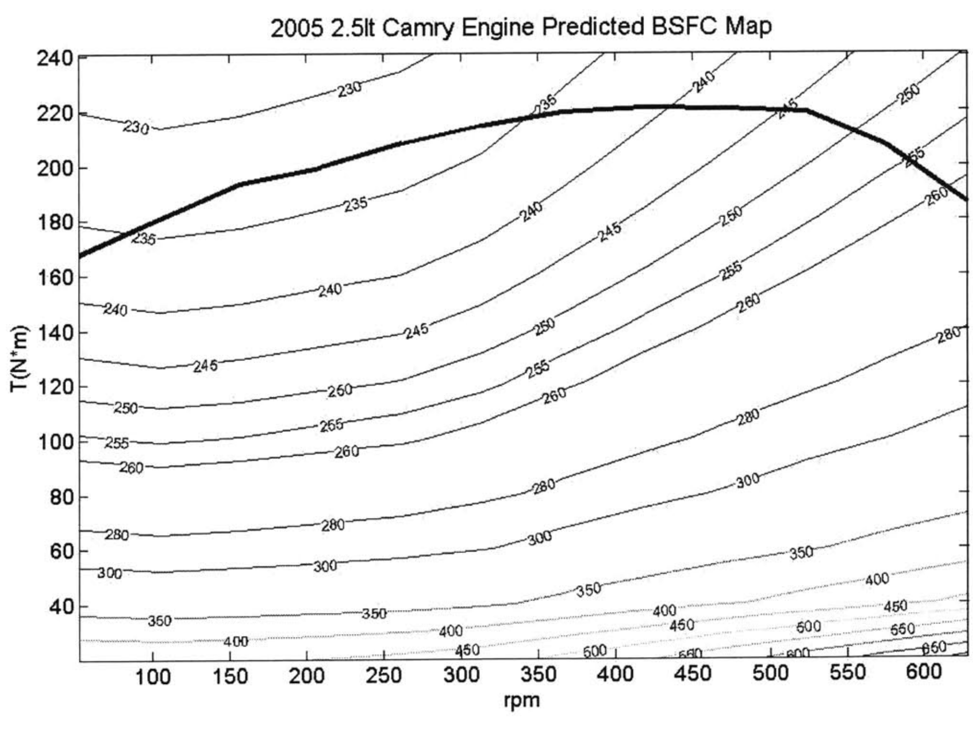



**Figure 29: Normalized Engine Brake Efficiency Map predicted by the friction model. 2005 2.5 liter Camry Engine**

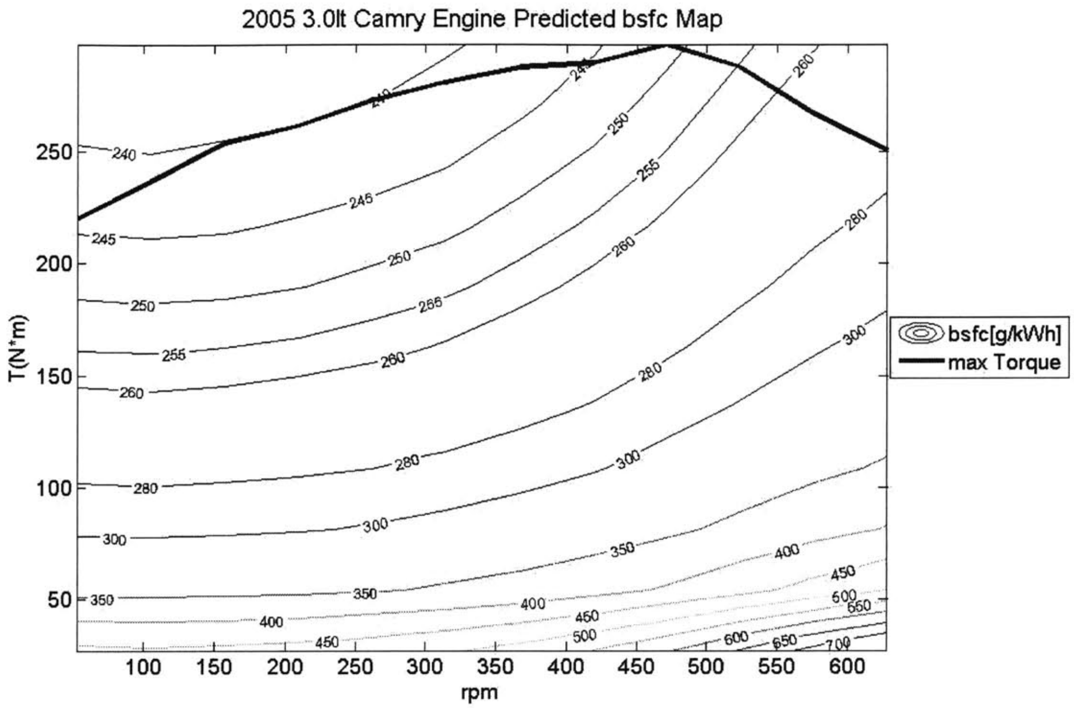



**Figure 30: Brake Specific Fuel Consumption Map predicted by the friction model. 2005 3.0 liter Camry Engine**

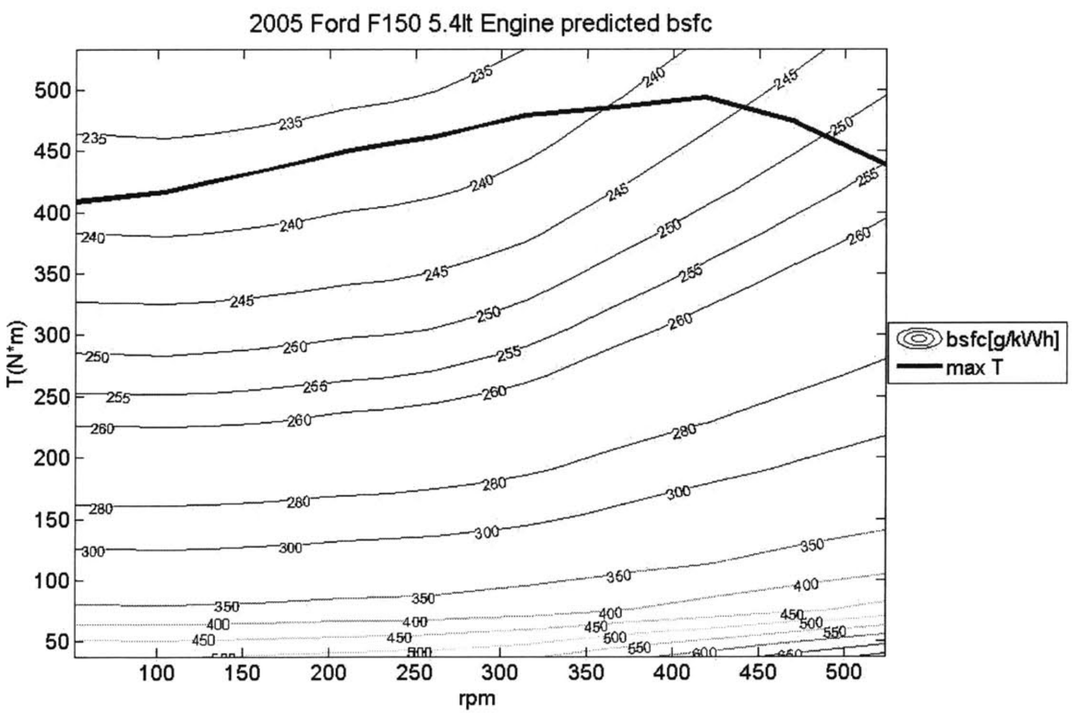



**Figure 31: Brake Specific Fuel Consumption Map predicted by the friction model. 2005 5.4 liter F150 Engine**

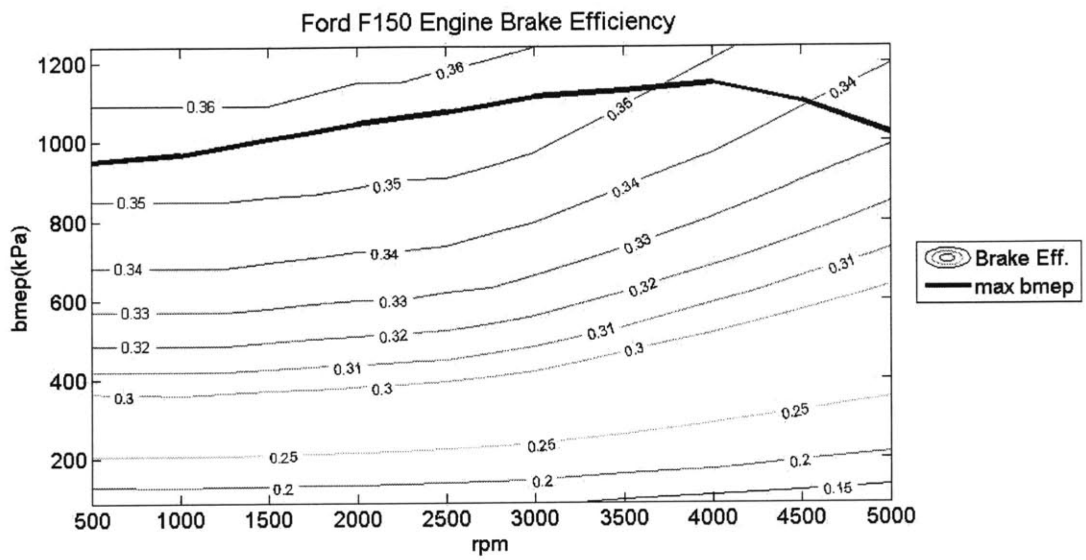



**Figure 32: Normalized Engine Brake Efficiency Map predicted by the friction model.**

**2005 4.2 liter F150 Engine**

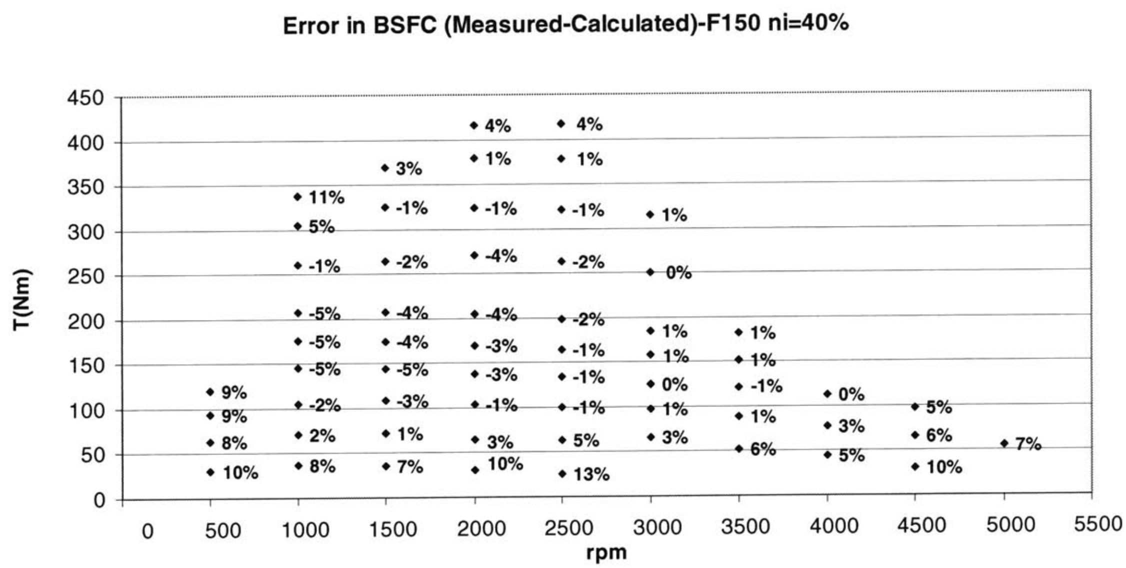



**Figure 33: Relative Error in BSFC between predicted and measured maps.**

**2005 5.4 liter F150 engine.**

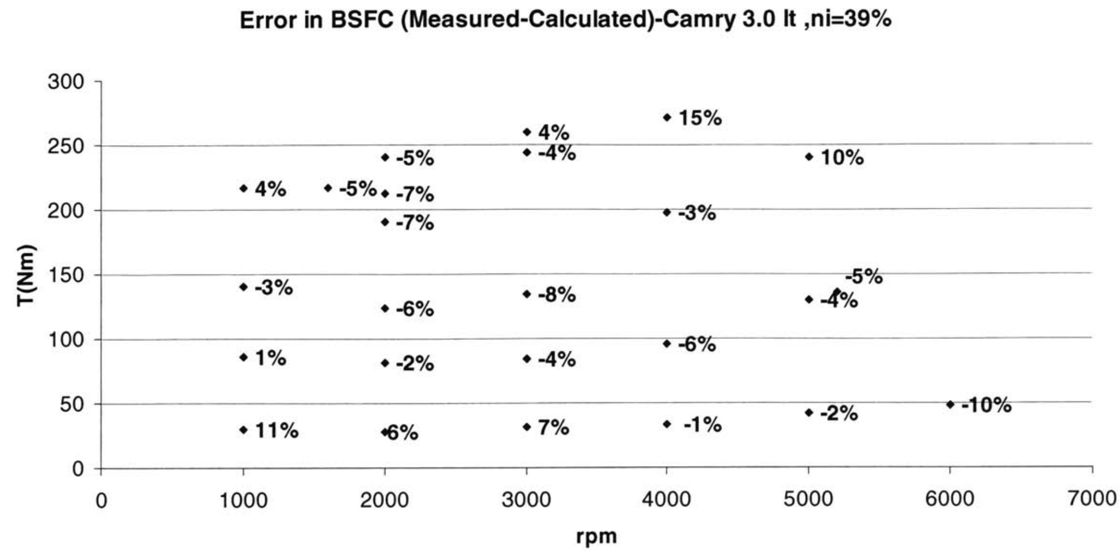



**Figure 34: Relative Error in BSFC between predicted and measured maps. 2005 3.0 liter Camry engine.**

Initially it should be observed, that the shape of the bsfc contours in the measured map (Figure 27) do not really look like the ones in the predicted map (Figure 30). The main reason for this is the modern engine design principle to divert from stoichiometric operation close to the maximum torque curve and switching to a rich fuel to air mixture. This is done in order to increase power output at an efficiency and emissions penalty. The relative error in bsfc between predicted and measured is however small as may be seen in Figures 33Figure 33 and 34. The approximation is more then adequate for the purposes of this analysis. Let it be noted for comparison that in the "On the Road In 2020"-[7] study, both fmep and $\eta_b$ were assumed constant all over the engine map. This is a significantly more sophisticated model which results in appreciably more realistic maps.

## Future Engine Efficiency Improvements

Some of the expected technological improvements expected in future naturally aspirated spark ignition engines are listed below:

- Friction reduction opportunities: Improved materials and piston ring design, camless valves eliminating valve train friction, synthetic lubricants, electric engine auxiliaries etc.
- Smart cooling systems for reduced heat losses
- Variable valve timing (VVT) and lift at full and part load. These allow both for pumping friction reduction and an increase in power density. VVT is already implemented in a few production vehicles, variable lift in even fewer. There is a large potential both for wider adoption of the technology and its improvement. A

significant improvement of the technology would be enabled through the use of solenoid valves. This would allow for significantly better control of the system.

- Variable length Intake Runners at different rpm: This would also allow for better power densities and possibly lower pumping friction.
- Higher expansion ratio engines for increased efficiency. This concept, also known as the Miller or Atkinson cycle is already implemented in the engine of the Toyota Prius. It allows for higher efficiencies at a power density penalty which could be offset by other improvements in power density.
- Cylinder cut out at lighter loads. Chrysler already implements this, it allows for an engine to operate with less than its full displacement volume at partial loads to improve fuel economy without compromising performance.
- Variable compression ratio. Higher compression ratios lead to higher indicative efficiencies. They are however limited by engine knock at some parts of the map (low rpm, high T). A variable compression ratio mechanism would allow for lower compression ratios at high knock probability operating conditions and higher elsewhere.
- Gasoline Direct-Injection(GDI). This allows for higher compression ratios due to the cooling effect of fuel evaporation which protects against engine knock. Additionally, in a GDI engine there is less throttling. It could also increase power densities and thus reduce the relative importance of friction. One possible issue with GDI engines will possibly be their hydrocarbon emissions.
- Effective lean NOx catalysts; lean engine operation. Lean engine operation allows for a higher charge and exhaust gases ratio of specific heats $\gamma$ and thus higher efficiencies. However SI engines have been traditionally operating at stoichiometric partially because of catalysts being efficient only there. Lean NOx catalysts could change that.
- Engine Controls and integration. These are becoming increasingly more sophisticated.

Taking all of the above into consideration a 25% reduction in fmep and a 7.5% improvement in indicative efficiency will be assumed for future naturally aspirated spark ignition engines. These assumptions are the same as the ones made in [7]. The resulting engine maps in normalized form are presented in Figure 35 and Figure 36 for the Camry and the F150 respectively.

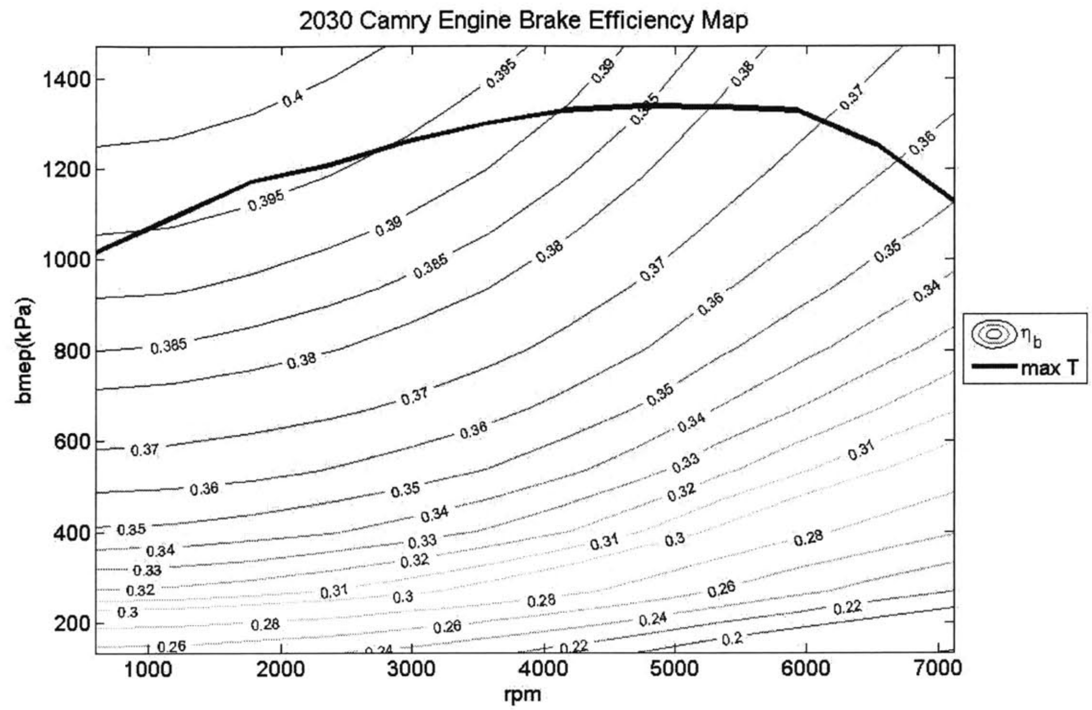



**Figure 35: Future Toyota Camry Brake Fuel Efficiency Map.**

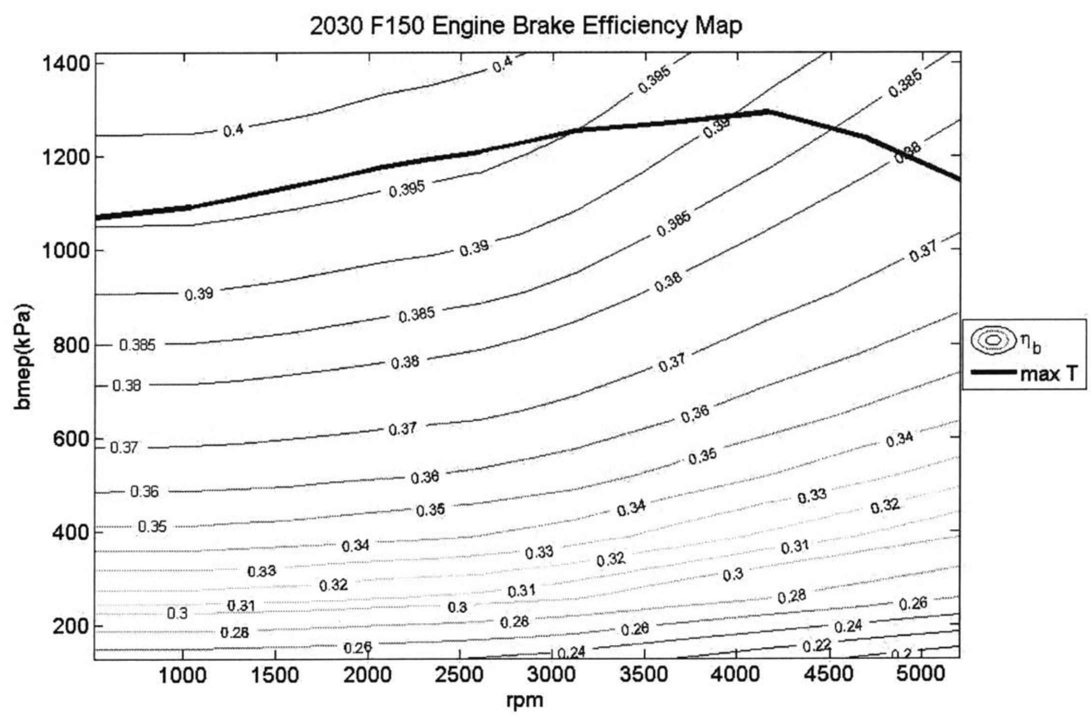



**Figure 36: Future Ford F150 Brake Fuel Efficiency Map.**

### 3.2.2. Power Density Improvements- Engine Sizing

The first step in sizing the engines of the future vehicles is determining the improvement in volumetric and gravimetric power density. The gravimetric density will be used in determining engine weight and subsequently the required power to move the curb weight of the vehicle. The volumetric power density is used in determining engine displacement volume which is needed to run the friction model to determine the future engine efficiency map.

Generally speaking, engine weight data are relatively difficult to come across. Furthermore, there is usually an ambiguity regarding whether the auxiliaries are included or not. For the purpose of this study, however, not a great deal of accuracy is needed as engine weight is only about 10-15% of vehicle weight.

Figure 37 shows the historical evolution of SI engine power density. It appears in a 2000 SAE paper by AVL, [24]. Power density values right before 2000 seem to be around 0.68 kW/kg. If however, the historical evolution presented in this graph were to be linearly extrapolated, the annual rate of increase since the 70's would be ~5%/year.It would seem unrealistic to extrapolate such a high improvement rate 25 years into the future,

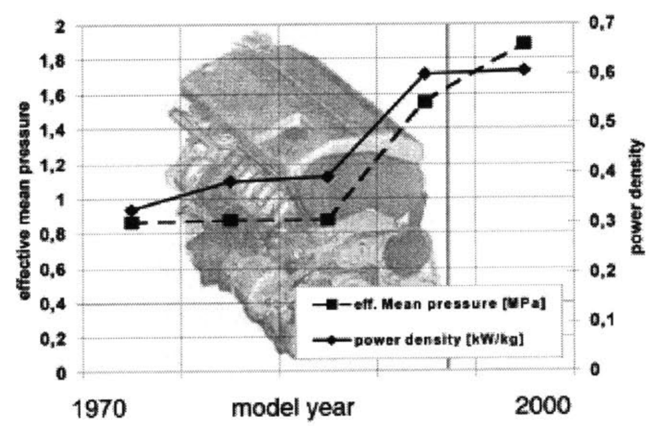



**Figure 37: Historical evolution of engine powertrain density**

Table 9 compares some of the most power dense automotive engines today with the ones in the 2005 baseline vehicles used in this study. It is important to notice that the data for the Camry engine do not correspond to the latest model. Also, it is evident from the Toyota engine numbers that the weight of the auxiliaries is almost as much as that of the engine. Including them or not, can therefore make a huge difference.

For simplicity, the Ford engine (0.74kW/kg) was used as the gravimetric power density baseline for both the F-150 and the Camry engines. Additionally a linear rate of improvement of 1%/ year was assumed which over 25 years gives a 2030 engine power density of 0.925 kW/kg. Engine gravimetric power density improvements that are expected over the next 25 years could be divided into two major categories:

- Engine weight reduction. This could be achieved be achieved with both innovative designs and the extensive use of advanced materials such as aluminum and especially magnesium and new manufacturing techniques. In a 1992 SAE paper [30] by Mitsubishi use of 33% aluminum resulted in a 23% weight reduction for a 2.0liter engine. This is equivalent to a 30% increase in gravimetric power density. Of course, today there engines already engines that use aluminum extensively.

- Engine volumetric density improvements. These include variable valve timing and lift, gasoline direct injection, etc. An increase in volumetric density does not linearly correspond to an increase in gravimetric power density. Displacement volume does go down when the volumetric power density is increased, but the pressures in the cylinder are higher and might require more material to withstand them.

As seen in Table 9, there are engines today that exceed the 0.925kW/kg value assumed for the future; the BMW engine mentioned in Table 9 achieves this by a sophisticated variable valve timing system and extensive magnesium use. The Audi engine uses gasoline direct injection and innovative structural design. In that sense, the assumption of 0.925 kW/kg is conservative. On the other hand, these advanced engines today are already using most of the improvements that could be foreseen for the future. Finally it is important to remember that the assumptions used in this study are supposed to reflect fleet average numbers.

**Table 9: Current Naturally Aspirated Engine Power Densities**

| Maker | Year | Engine Model | Cylinders | Vd(lt) | kW | kg | kW/ kg | kW/ liter | Source | Note |
|---|---|---|---|---|---|---|---|---|---|---|
| Opel | 2005 | Ecotec | I4 | 1.8 | 103 | 118 | 0.87 | 57 | MTZ [31] | |
| Audi | 2005 | | V6 | 3.2 | 191 | 170 | 1.13 | 60 | MTZ [31] | |
| BMW | 2005 | | I6 | 3 | 190 | 161 | 1.18 | 63 | MTZ [31] | |
| Ford | 2005 | Triton | V8 | 5.4 | 224 | 304 | 0.74 | 41 | Ford | |
| Toyota | 1996 | MZFE | V6 | 3 | 157 | | 0.63 | 52 | On the Road in 2020 [7] | without auxiliaries |
| Toyota | 1996 | MZFE | V6 | 3 | 157 | | 0.34 | 52 | | with auxiliaries |

Using the value of 0.925 kW/kg for the future engines and the assumption that overall vehicle curb weight should be reduced by 20% , the required power for future engines can easily be calculated. As already explained, acceleration performance for the future vehicles should be kept constant. In order to achieve that, the ratio of vehicle weight over engine maximum power should remain about constant.

$$\frac{W_{curb}}{P_{\max,eng}} = const = c \Rightarrow \frac{W'_{curb}}{P'_{\max,eng}} = c \Rightarrow P'_{\max,eng} = \frac{0.8 * W_{curb}}{c} \quad (3.7)$$

Where,
$W_{curb}$:is the vehicle curb weight
$P_{max,eng}$ :is the engine maximum power
Prime variables denote future quantities, non-prime are current ones.

However in order to determine the required power for other powertrains the vehicle glider mass is needed. That is the mass of the vehicle without any of the powertrain components needed i.e. engine, engine auxiliaries, transmission, exhaust aftertreatment and additionally in the case of hybrids, motors, controllers and battery pack. In order to calculate that from equation (3.7) it follows that:

$$\begin{aligned} W'_{curb} &= (W'_{glider} + W'_{engine} + W'_{transmission} + W'_{aftertreatment_1}) \Rightarrow \\ W'_{glider} &= 0.8 * W_{curb} - P'_{\max,eng} / 0.925 - W'_{transmission} - W'_{aftertreatment_1} \end{aligned} \quad (3.8)$$

If a reduction of 20% in weight is assumed for the future transmission and aftertreatment due to both technological improvements and engine downsizing, future glider weight may be estimated if the current after treatment and transmission masses are known. These were estimated based on the models and scaling functions in ADVISOR. Details on component masses are provided in the assumptions table in the appendix

The next step after determining the required power for the future engine is to determine its required volume. Using equation (3.1), the maximum engine torque curve as a function of rpm is converted to a $bmep_{max}$ as a function of rpm. Chon and Heywood in [32] concluded by analyzing historical data that $bmep_{max}$ for four valve engines improves linearly by 0.5%/year. Extrapolating this number 25 years into the future, the result is a 12.5% increase in $bmep_{max}$. This was also the assumption made in [7]. If the rpm range is kept the same, this means a 12.5% increase in maximum torque and power per unit volume and the new volume can easily be calculated by diving the required power by the new power per unit volume ratio. This was the methodology used in [7].

However, in order to use the engine efficiency map prediction methodology developed in chapter 3.2.1.1, it is more appropriate to assume that the mean piston speed is the quantity that remains constant between current and future engines instead of rpm. When mean piston speed is constant, charge flow characteristics are the same. Maximum airflow into the cylinder and thus power are limited by choking which depends on flow speed. Mean piston speed is given by

$$S = 2 * N * L \quad (3.9)$$

Where:
S: is the mean piston speed
N: is the frequency of revolution of the crankshaft in Hz
L: is the piston stroke

The engine $bmep_{max}$ as a function of N is thus transformed into a function of S. This is the curve that is multiplied by 112.5% to obtain the future bmep as a function of piston speed curve. This is shown in Figure 38 for the 2005 Camry engine and its future equivalent.

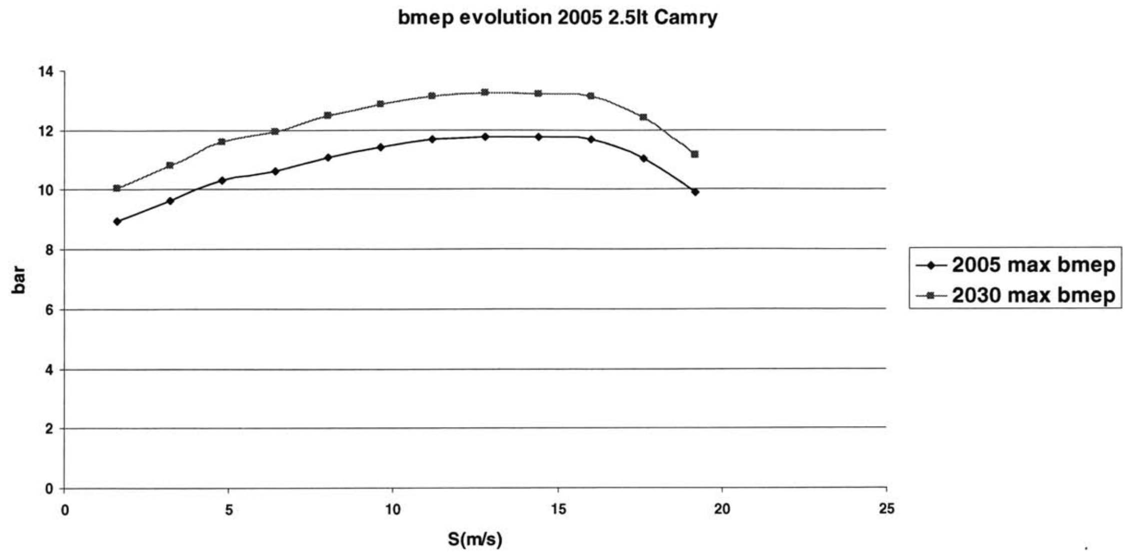



**Figure 38: Bmep Evolution for the case of the 2005 2.5 liter Camry**

If S is kept constant, from equation (3.9) it follows that:

$$N * L = N' * L' \quad (3.10)$$

But engine displacement is given by

$$Vd = n_c * \pi * B * L \quad (3.11)$$

Where:

$n_c$ is the number of cylinders

B is the cylinder bore

Displacement volume determines power output as seen in equation (3.1).

$$P = V_d * bmep_{\max} * \frac{N}{2} = n_c * \pi * B * L * bmep_{\max} * \frac{N}{2} \quad (3.12)$$

Assuming a constant bore to stroke ratio R, using equation (3.10) and bearing in mind that $bmep_{max}$ is a function of N, the expression for power as function of N becomes:

$$\begin{aligned} P'(N') &= n_c' * \pi * \frac{B'^2}{R} * bmep'_{\max}(N') * \frac{N'}{2} \Rightarrow \\ P'(N') &= n_c' * \pi * \frac{B'^2}{R} * bmep'_{\max}\left(\frac{N * L * R}{B'}\right) * \frac{N * L * R}{2 * B'} \end{aligned} \quad (3.13)$$

In other words, when changing engine volume, while keeping a constant R ratio, the engine stroke might change. It will definitely change if the number of cylinders is the same. The assumption used for the number of cylinders is that it is six if $V_d$ is larger than 2.5 liter and 4 for 2.5 liter or less down to the smallest engine used in any of the models (930cc). As a result, in order to keep maximum S constant, engine speed changes. The problem is essentially to determine the required bore so that the maximum of the new power as a function of speed curve is the required .This is an iterative process.

In some of the engines used in the models of this study, the engine went from a six cylinder in the current model to a four cylinder in the future. In those cases, the stroke doesn't change significantly when downsizing and the future engine has a similar maximum speed as the current. When, however the current engine was already a four cylinder, downsizing makes it significantly faster. In the case of the 2005 2.5 liter Camry, it went from a 2.5 liter, 4 cylinder, 6000rpm redline engine with 119kW maximum output , to a 1.4liter with 95.4kW maximum output and a 7100 rpm redline. Torque as a function of rpm for both engines is presented in Figure 39. The new engine speed data are subsequently fed into the (reduced) friction model to predict the new fuel consumption map as already explained.

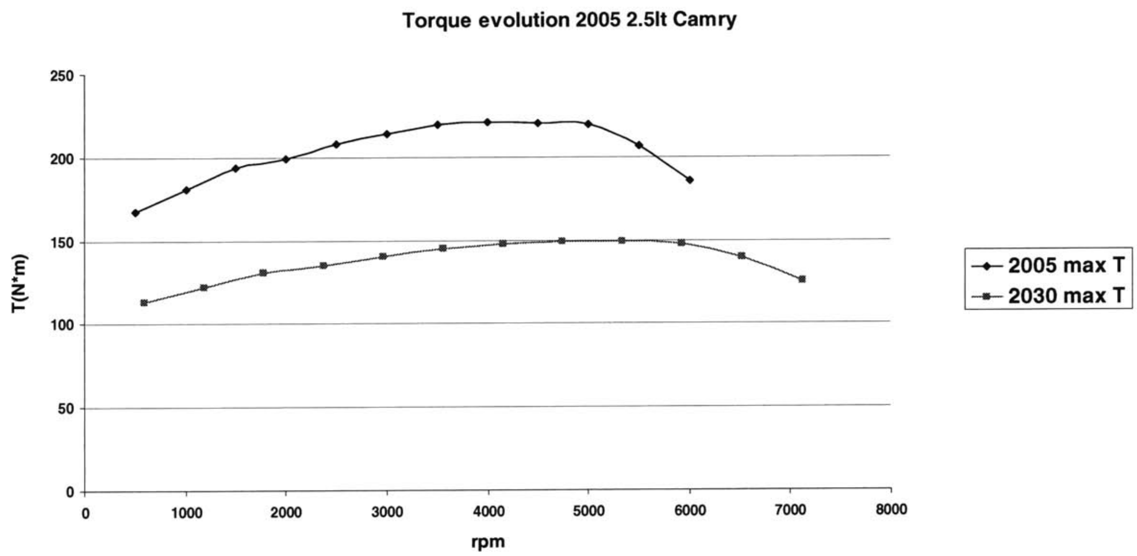



**Figure 39: Torque Evolution for the case of the 2005 2.5 liter Camry**

## 3.3. Advanced Turbocharged Gasoline Engines

When turbocharging a gasoline engine, the power output per unit displaced volume increases. $Bmep_{max}$ goes up significantly. Fmep goes up only slightly as it scales mainly with cylinder surface. The reason it increases slightly is because the mean cylinder pressure goes up. The resulting fuel consumption map has therefore higher partial load efficiencies.

There have been historically 3 main technical limitations to turbocharged engines:

- Engine knock. Average pressures in a boosted engine are higher; they are therefore more likely to knock. As a result, boosting has been historically limited to pressures of only about 0.6-0.8 bar. Additionally, compression ratios are generally lower in a boosted engine, which leads to lower indicated efficiencies.

- Low Engine Torque at low rpm: Vehicle acceleration is therefore worse and larger gear ratios need to be used which in turn leads to lower fuel economy over a driving cycle.
- Turbo Lag: This term is used to describe the slow response of turbocharged engines to transient loads. It deteriorates performance.

There are however, several technologies that could offer solutions to these problems in the foreseeable future.

- Gasoline Direct Injection (GDI): When directly injecting the fuel in the cylinder, its latent heat of vaporization cools the charge down. As a result, it is less likely to knock. This leads to higher boost and compression ratios. [33]
- Variable Geometry Turbines. Variable geometry turbines are already well established for diesel engines. They improve turbocharger efficiency, reduce turbo lag and limit compressor surge which causes low torque at low rpm. Introducing them in SI engines presents several additional issues such as higher exhaust temperatures among others but they can be solved [34, 35].
- Variable Compression Ratio. Being able to actively change the engine compression ratio when operating in different parts of the engine map is currently being investigated [36]. Achieving this on a production engine would mean that the compression ratio is no longer limited by the operating conditions that are the most susceptible to knock
- E-Boosting. Adding a small electrical compressor, or coupling a small electric motor on the turbocharger shaft has been shown to significantly decrease turbolag, increase engine torque at low rpm as well as possibly improving compressor efficiency [37,34]

### 3.3.1. Map Evolution methodology

Given the expected improvements in turbocharged engine technology, the following methodology was used to predict future turbocharged engine specific fuel consumption maps:

- The current engine map was decomposed into a constant indicative efficiency and a fmep as a function of piston speed. Additionally, the engine's volumetric efficiency is obtained as a function of rpm from the maximum torque curve. This is defined as:

$$\eta_{vol} = \frac{2 * \dot{m}_{air}(N)}{V_d * N * \rho_{air,i}} \quad (3.14)$$

Where $V_d$ is the total displaced volume, $m_{air}$ is the air mass flow rate at maximum torque-it can be calculated from the maximum power output, the brake efficiency and the air to fuel ratio.$\rho_{air,i}$ is the density of the air at intake manifold (ambient) conditions.

- The future turbocharged engine sizing calculation is performed to determine the future engine's maximum power, bore, stroke, number of cylinders and rpm range.
- Based on the required torque and rpm output for each point on the map, the corresponding mass of fuel required is calculated based on an assumed brake efficiency. Stoichiometric operation is assumed everywhere.
- The intake pressure and temperature are calculated at every point on the map assuming a constant compressor efficiency of 66% and a volumetric efficiency as a function of piston speed using the volumetric correlation from the current engine .The constant compressor efficiency is used for simplicity instead of using a compressor map.
- Since the intake pressure at every point of the map is now known, the fmep can be calculated using the friction model previously described.
- Pmep and pressure drop across the turbine is calculated assuming for simplicity a constant turbine efficiency of 56%.
- The engine brake efficiency at every point may now be calculated. The initial assumption for brake efficiency is checked. If they differ significantly, the calculation is repeated until they converge.

The methodology was adapted from [38] were the detailed equations used can be found.

Examples of calculated intake pressure (manifold air pressure-MAP), as well as exhaust temperature and pressure maps are presented in Figure 40, 41,42 and 43.These come from the turbocharged engine in the future lower performance Camry. Note that the maximum MAP is only about 1.6 bar. It is also noteworthy that the engine is not turbocharged at low loads. Atmospheric pressures are enough to get the required torque output. These observations might at first seem surprising. $Bmep_{max}$ has increased from current levels as will be explained, but the amount of boost remains at current levels .The phenomenon is due to the increased indicative efficiency and decreased friction that was assumed. Because of the improved engine efficiency, the amount of fuel and thus of air that is required to achieve the same $bmep_{max}$ is less. The amount of boost required to achieve a higher $bmep_{max}$ is about the same. It should finally be mentioned that although $bmep_{max}$ was increased, the compression ratios implied by the indicative efficiency assumptions are the same as those for the naturally aspirated engines. Achieving this even at constant boost levels is not trivial.

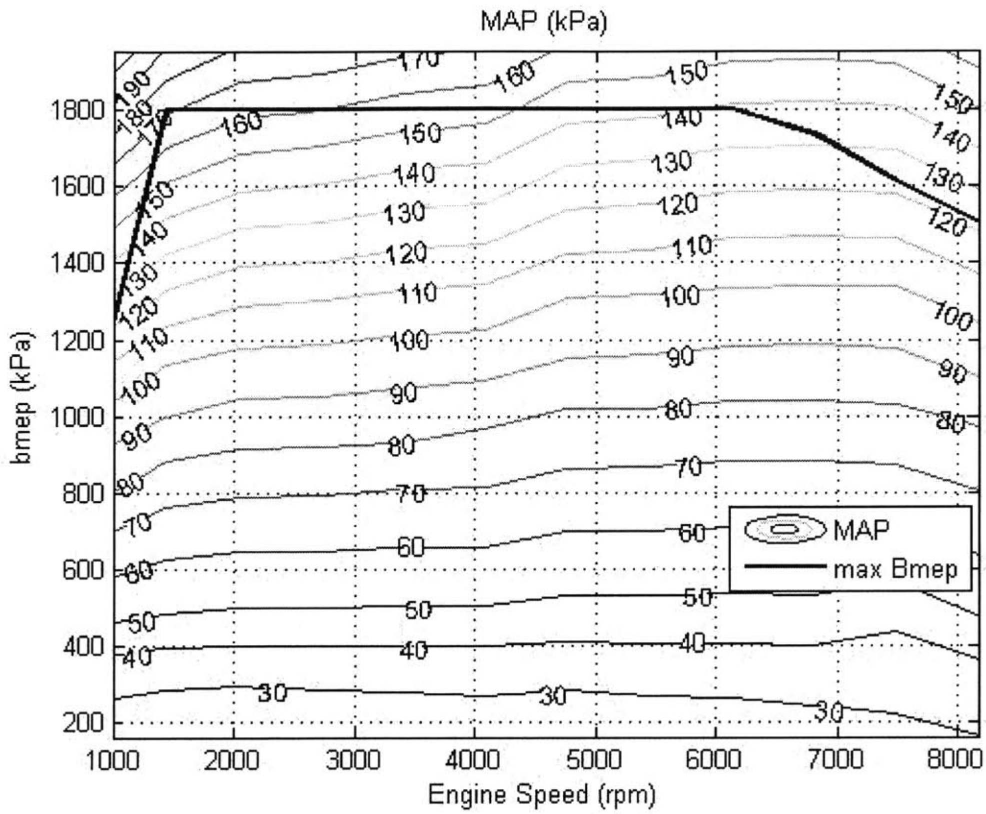



**Figure 40 Manifold Intake Pressure for the Turbocharged Engine of the 2030 Camry 2.5 liter equivalent vehicle**

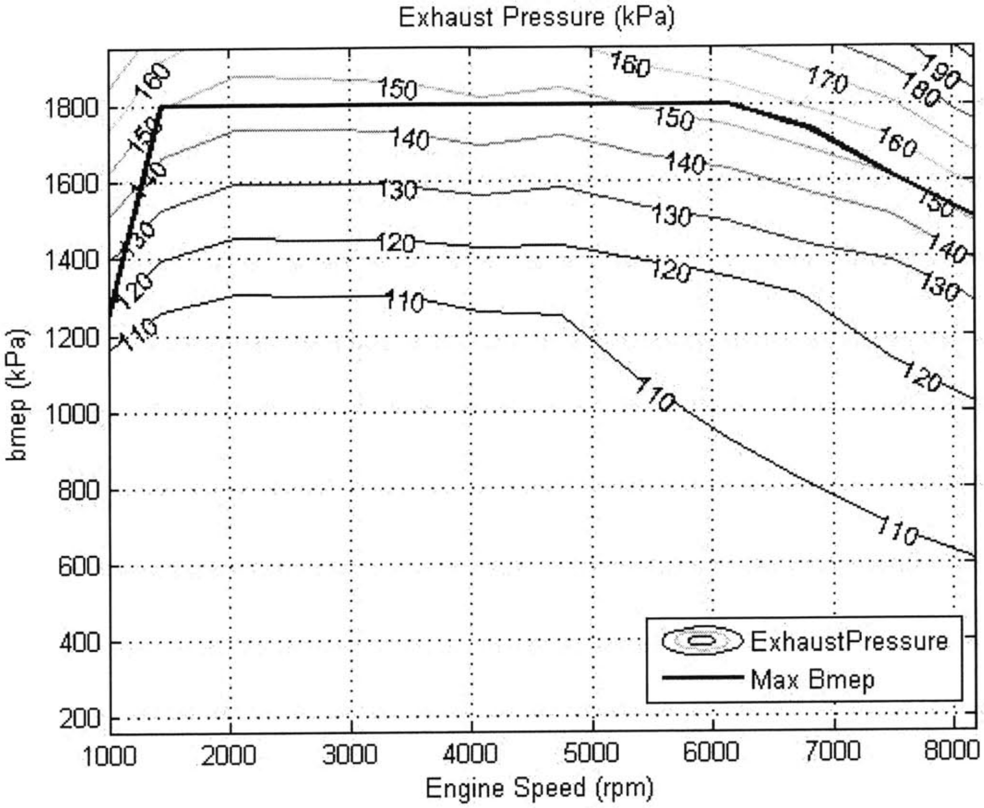



**Figure 41 Exhaust Pressure for the Turbocharged Engine of the 2030 Camry 2.5 liter equivalent vehicle**

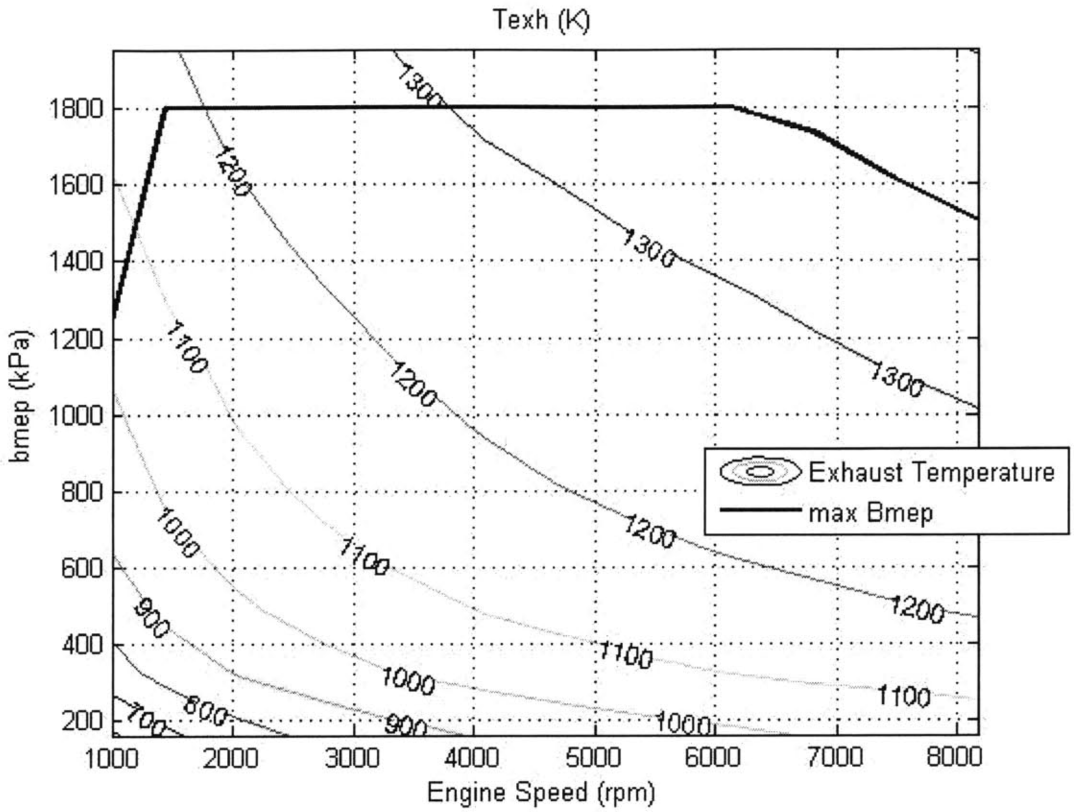



**Figure 42 Exhaust Temperature for the Turbocharged Engine of the 2030 Camry 2.5 liter equivalent vehicle**

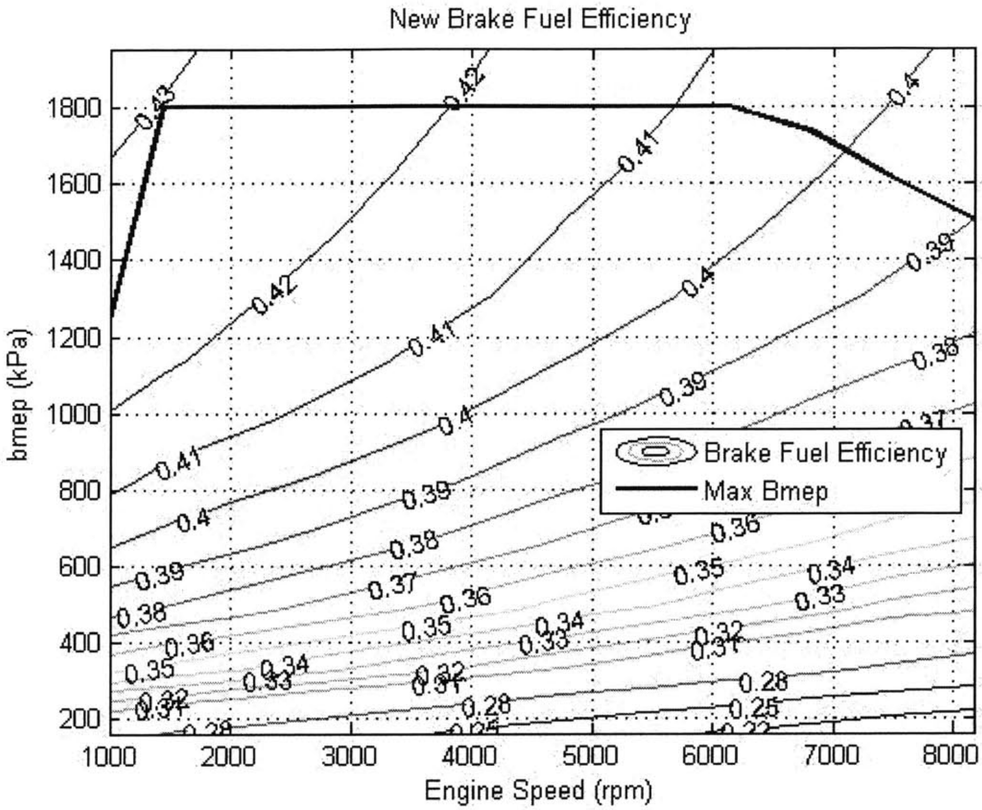



**Figure 43 Brake Fuel Efficiency ($\eta_b$) and maximum bmep (corresponding to maximum torque) for the Turbocharged Engine of the 2030 Camry 2.5 liter equivalent vehicle**

### 3.3.2. Sizing-Scaling methodology

In order to perform the sizing calculation for the turbocharged engine a volumetric power density or $bmep_{max}$ and a gravimetric power density are needed. Additionally, the shape of the maximum torque curve needs to be defined as it is different than that of the naturally aspirated engine.

Current turbocharged engines for passenger vehicles get about 1600 kPa bmep. They are however, usually port-fuel injected (PFI) which forces the designers to choose compression ratios around 8.5-9 to avoid knock. There are experimental engines today however that achieve maximum bmep's of up to 2100 kPa [39]. Gasoline Direct Injected engines can today achieve 1800 kPa bmep at the laboratory level with compression ratios of 10-10.5, comparable that is to naturally aspirated engines [40]. Some of these current advanced turbocharged engines are presented in Table 10. For the purposes of this study a maximum bmep of 1800 kPa was assumed for future turbocharged engines. Since the methodology used to predict fuel consumption maps was based on an assumed value for indicative efficiency, compression ratios for these engines didn't need to be explicitly calculated. However, the assumption of the same indicative efficiencies as in the case of the naturally aspirated engines implies comparable compression ratios.

**Table 10: Current Advanced Turbocharged Engines.**

| Maker | Year | $V_d$ (lt) | kW | kW/lt | Bmep maximum (kPa) | Source | Notes |
|---|---|---|---|---|---|---|---|
| Average US market turbocharged engines | 1999 | | | 67.8 | 1634 | SAE [32] | |
| GM-Europe | 2006 | 1.6 | 132 | 82.5 | | MTZ [31] Issue No: 2006-03 | |
| VW | 2006 | 1.4 | 125 | 89.3 | | [41] | Supercharger+ Turbocharger+ GDI Compression ratio=10 versus 11.5 for the NA SI engine |
| Renault | 2005 | 1.0 | 86 | 82.4 | 2100 | SAE [39] | PFI, Compression ratio=8.5, Experimental Engine |
| FEV | 2006 | 1.8 | | | 1800 | SAE [41], [33] | GDI, Cr=10.5, Experimental engine |

The quantity that needs to be determined next is the value for gravimetric power density. For simplicity, it could be assumed that it scales linearly with bmep. In other words, the power density assumed for future naturally aspirated engines (0.925 kW/kg) can be multiplied by the ratio of future turbocharged maximum bmep over future naturally aspirated bmep (1800kPa/1400kPa) to give a density of 1.19kW/kg. However gravimetric power density in turbocharged engines doesn't exactly scale linearly with bmep. The reason is that turbocharged engine structures need to withstand higher pressures, so they need to be sturdier and thus heavier. Additionally, the turbocharged engine needs more auxiliaries-the turbocharger and intercooler. For these reasons, the scaled power density of 1.19kW/kg was reduced by 10% and the final number used was 1.04 kW/kg. The effect of engine power density on overall vehicle curb weight is small, so these numbers were considered accurate enough.

One final piece of information is needed to perform the sizing calculation-the shape of the maximum torque curve. Traditionally, turbocharged engines have been very limited in the maximum torque they produce at low rpm because of low volumetric flow rates on the exhaust side leading to low turbocharger speeds and thus pressure ratios.

Compressor surge line is also limiting at these flow rates and pressure ratios. In direct injection engines, however, according to [41] there are reasons that allow for significant improvement. Valve timing can be changed to increase the volumetric flow rate through the scavenging effect without worrying about wasting fresh charge as in a PFI engine. Furthermore, the scavenging effect reduces the amount of residual gas and thus the danger of knock, so that spark timing can be advanced thus improving low end torque.

A comparison between naturally aspirated-PFI, boosted-PFI, and boosted GDI maximum engine torques is presented in Figure 44. The figure was adapted from [41]. The GDI boosted engine gets to maximum torque already at 1300 rpm while the PFI needs to get close to 2000. It is also noteworthy how turbocharged engines tend to keep their maximum torque almost constant at mid to high rpm. The shape of the maximum torque curve of the GDI, boosted engine in Figure 44 was used for the future turbocharged engines in this study.

The rest of the sizing calculation is identical to the process described for NA SI engines. The maximum torque curve and displacement was adjusted to the required power output. The engine speed was also changed in order to keep constant piston speeds as explained for the NA SI engines. The only difference is that turbocharged engine maps were not allowed to go below 1000 rpm due to compressor surge limitations. The maximum torque curve used for the 2030 turbo SI lower performance Camry is presented in Figure 45 .Note that the redline is very high- around 8000 rpm as this engine has been downsized significantly to only 930 cc.

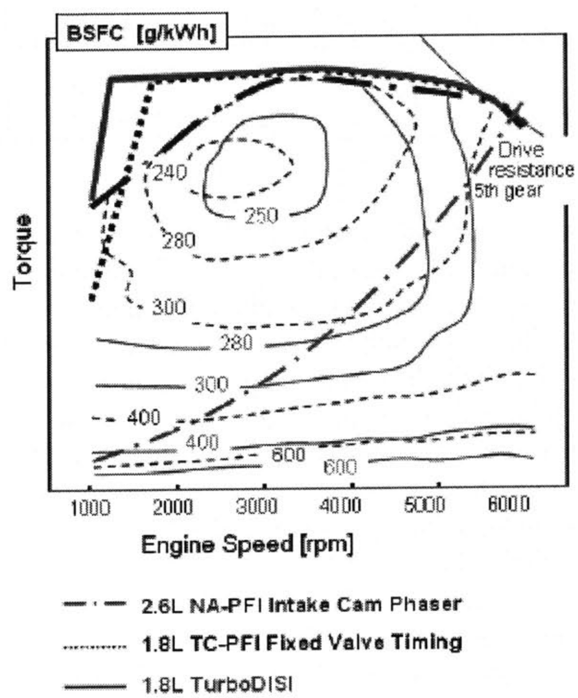



**Figure 44: Maximum Torque curve comparison for NA-PFI, Turbocharged PFI and Turbocharged GDI.**

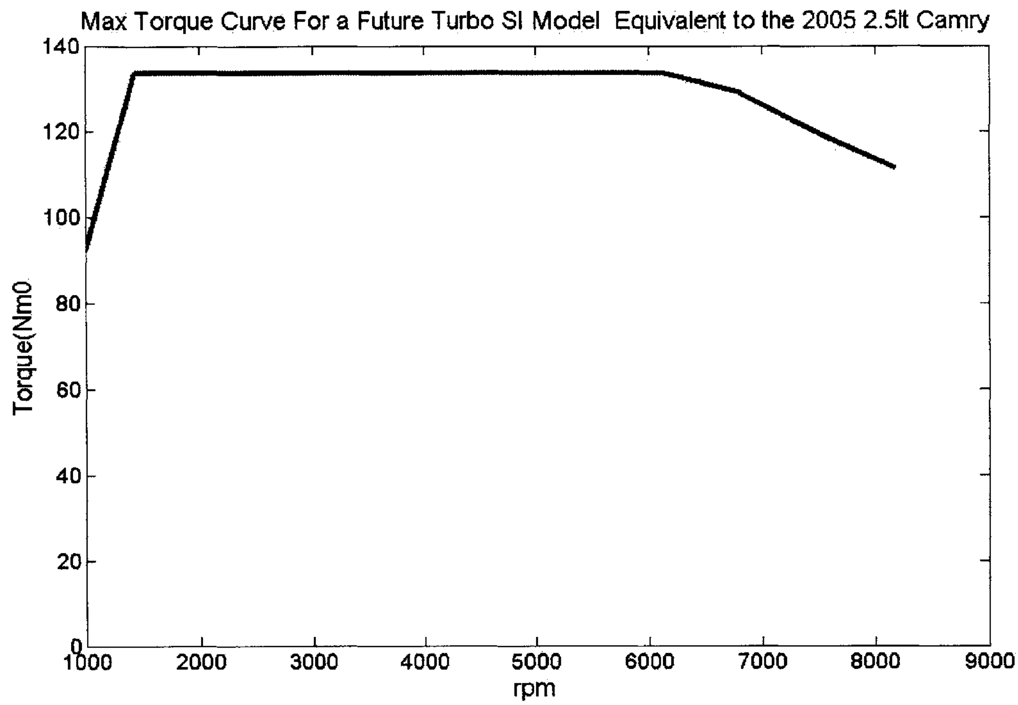



**Figure 45: Maximum T Curve for the 2030 Turbo SI lower performance Toyota Camry**

### 3.3.3. The Effect of Turbo Lag

Turbocharged engines tend to have a sluggish response to very rapid acceleration demands at low rpm. The phenomenon is known as turbo lag. It is usually defined formally using the time $\tau$ it takes the engine to reach 90% of its maximum torque output under highly transient conditions, i.e. when the input torque demand is a step function. A comparison between a NA SI PFI, a turbocharged PFI and a turbocharged GDI engine is presented in Figure 46. It takes the NA SI engine only about 0.5 sec to get to the requested output in comparison with 3-4 seconds for the boosted engines.

The phenomenon is caused due to the behavior of the turbocharger under transient conditions. A centrifugal compressor, like the ones used in turbochargers, operates at very high rpm's. Due to the turbocharger's shaft rotating inertia, it takes the compressor some time to accelerate to the rpm's needed to produce maximum torque. This is especially pronounced at low engine rpm when the exhaust flow rates are low creating thus low accelerating torques on the turbine side of the turbocharger.

There are, however several ways to significantly reduce turbo lag:

- Reducing Turbocharger Inertia through the use of lighter materials. A 10% decrease , decreases $\tau$ by 6%.[39]
- Improved turbocharger bearings.
- Variable Geometry Turbines
- Twincharging: This concept involves supercharging at low speeds and turbocharging at high. Supercharging uses a compressor mechanically driven from the engine. There is therefore no lag, but there is a small fuel economy

penalty. This concept is implemented in the new VW twincharged engine already presented in Table 10.

- Mechanically assisted boosting: Alternatively to twincharging, the supercharger can only assist the turbocharger in the beginning of acceleration until it's up to speed to take over. Then the supercharger is decoupled
- Instead of a mechanically driven assist compressor, an electrically driven one may be used.(E-Boost)
- Two-Stage Turbo charging using two smaller turbochargers instead of a larger one essentially cuts the rotor inertia that needs to be overcome in half.
- Variable Valve Timing. Large valve overlap has been shown to decrease transient times.

The most efficient of the solutions mentioned above are mechanically or electrically assisted boosting. Studies have shown that their use alone without any of the other measures mentioned above, cuts response times $\tau$ to less than 1 sec [37]

In this study, the effect of turbo lag will be neglected. It was assumed that the time frame of the study is long enough for some of the solutions mentioned above to virtually eliminate its effect.

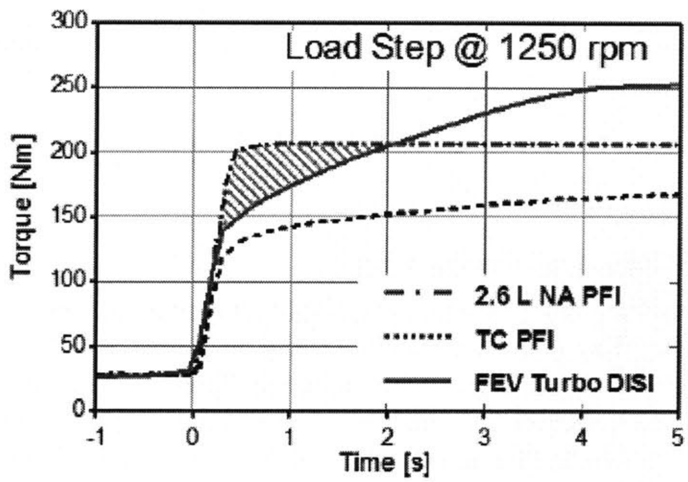



**Figure 46: Response to a step change In requested torque for a NA SI-PFI, a turbo PFI and a turbo GDI engine. Adapted from SAE [41]**

## *3.4.Advanced Diesel Engines*

Diesel engines are a well established, fuel efficient engine technology that has long been a competitive alternative to gasoline engines. Due to their significantly higher efficiency they have traditionally dominated the heavy duty automotive market. Their efficiency advantage stems from the different nature of combustion employed. Diesels use direct injection of fuel in the cylinder, which is subsequently ignited through compression instead of a spark. This is why they are also known as compression ignition engines. The amount of fuel injected is therefore used for controlling the load instead of throttling a well mixed fuel-air mixture as in SI engines. Lack of throttling makes partial load operation significantly more efficient than gasoline engines.

However, due to the nature of compression ignited combustion, diesel engines are dirtier in terms of particulate and NOx emissions and also noisier. The diesel engine witnessed major technological advances over the last twenty years, namely direct versus indirect injection, common rail technology and exhaust particulate filters (traps). These technologies have made diesels considerably cleaner, less noisy and more efficient. These advances, along with policy measures, have lead diesel to significant fractions of new automobile sales in Europe. In the U.S.A., emissions requirements are more stringent and fuel is cheaper. Diesel hasn't therefore penetrated the light duty fleet in significant numbers. There is however a potential for significant fleet penetration if diesels manage to meet emission standards.

### 3.4.1. Map Evolution methodology

As in the case of the SI engines, in order to predict fuel consumption maps for future diesel engines a constant $\eta_i$ and a fmep as a function of rpm will be used. A current diesel engine map was provided by Cummins Diesel. The friction model used for the gasoline engines is in principle also applicable to diesel engines. The constants used however would need to be recalibrated. A Willan's line methodology was alternatively used to extract fmep as a function of rpm. A Willan's line is essentially a plot of torque output at constant engine speed versus fuel flow rate at that speed. If the line is extrapolated to zero torque output, the fuel flow rate obtained represents the power used to overcome friction alone. Fmep can thus be estimated. The technique is illustrated in Figure 47.

In this case, the fuel flow rate and the displacement volume were not known, just the bsfc or $\eta_b$ as a function of rpm and bmep. A variation of Willan's technique was therefore used. Using the inverse of equation (3.5), the following expression is obtained:

$$\frac{1}{\eta_b} = \frac{1}{\eta_i} * (1 + \frac{fmep}{bmep}) \quad (3.15)$$

A constant $\eta_i$ was assumed. Fmep was considered a function of rpm only. The inverse of measured $\eta_b$ as a function of bmep should therefore be a hyperbola for constant speed. As seen in Figure 48 this is true. At the limit bmep$\rightarrow\infty$, the inverse of brake efficiency ($1/\eta_b$) is equal to the inverse of indicative efficiency ($1/\eta_i$). From Figure 48 it can therefore be seen that indicative efficiency varies only weakly with rpm. An average indicative efficiency value can be used to obtain a good fit with the measured data.

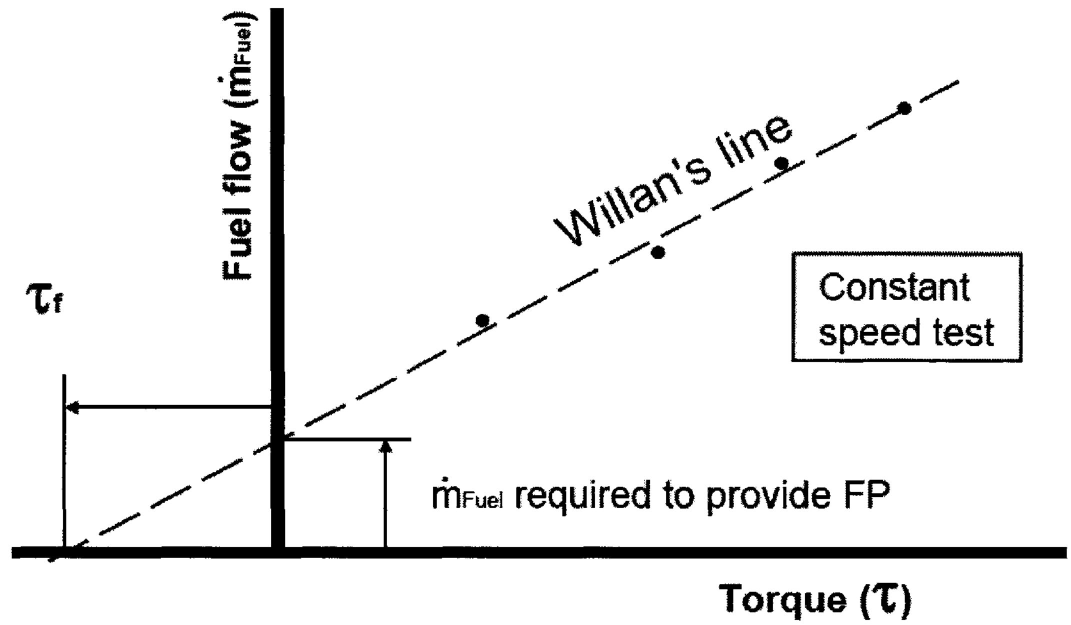



**Figure 47: Willan's Line Technique**

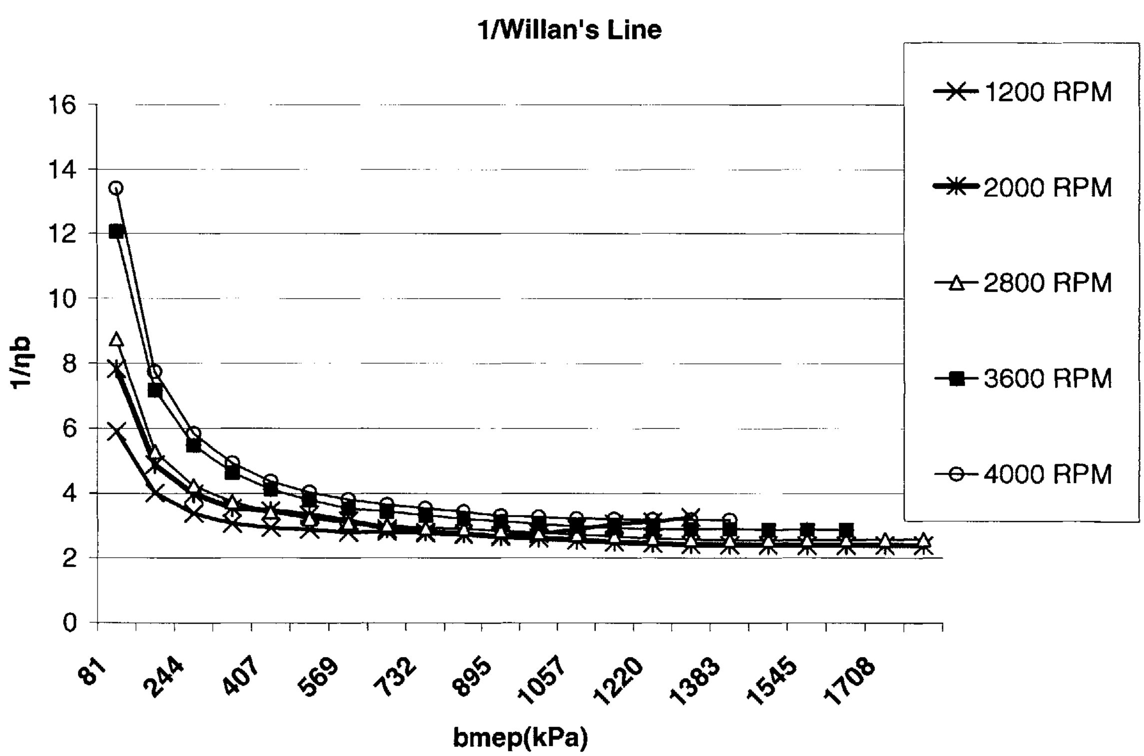



**Figure 48: Validation of the assumption of constant $\eta_i$ and fmep=F(N) only.**

If equation (3.5) is used to back-calculate fmep from the measured values of $\eta_b$, bmep and the assumed constant $\eta_i$, the resulting fmep should only depend on speed and not bmep. However, as seen in Figure 49, this is a valid assumption everywhere except close to the maximum bmep at each rpm, possibly because the constant $\eta_i$ assumption there doesn't hold. At any case, if these lines are extrapolated to zero, values of fmep as a function of rpm are obtained. This function (fmep(N)) was combined with a constant $\eta_i$, to give a good fit to the measured data all across the map. With a value of $\eta_i$ =45% the best fit is obtained with an average relative error of 0.7% and a maximum error of 10%. The resulting map is presented in Figure 50.

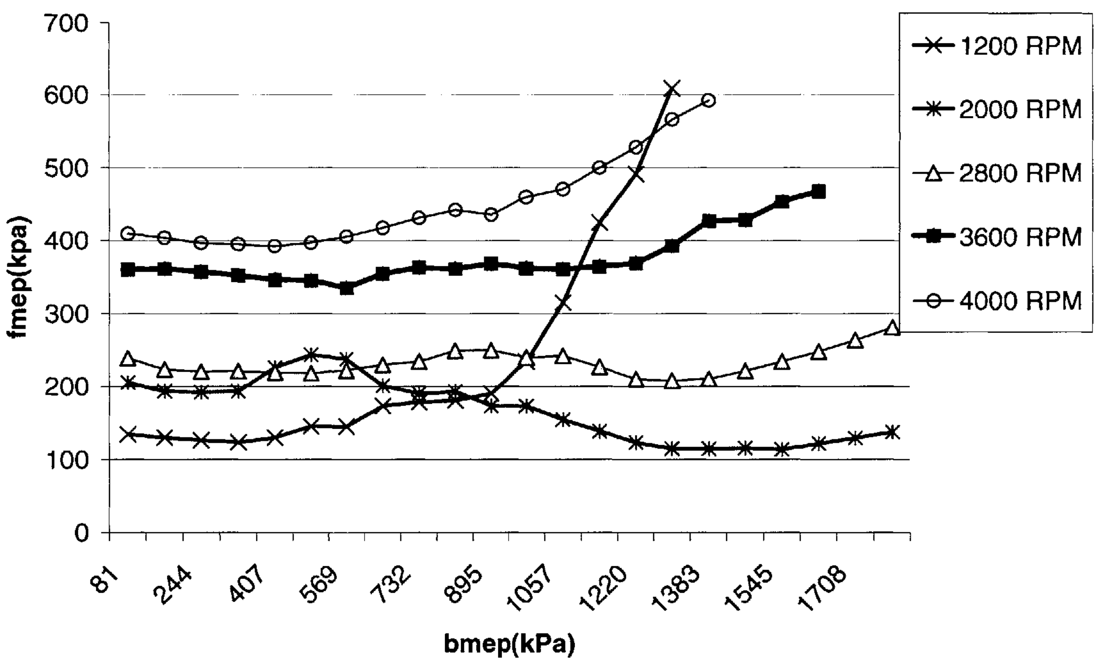



**Figure 49: Dependence of fmep on bmep and speed for the 2005 Cummins Diesel Engine, assuming constant $\eta_i$.**

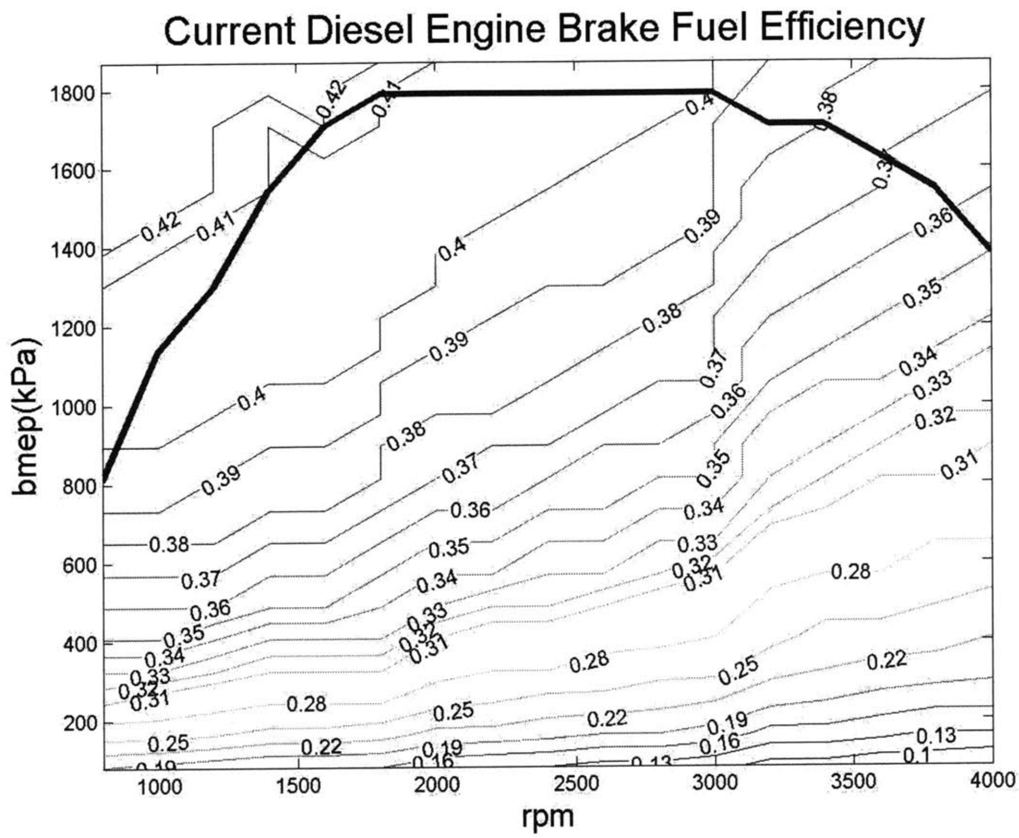



**Figure 50: Current Diesel Engine Map and maximum engine bmep.**

The next step is to estimate the relative improvement over the next 25 years on $\eta_i$ and fmep for diesels. Some of the expected technical advances that were listed as expected for SI engines have already been implemented in modern diesels or irrelevant. For example, in diesels there is no knock limitation, they are already heavily turbocharged with VGT turbines. A variable compression in diesels e.g. wouldn't offer much for the same reason. However, diesels have been improving at rapid rates since the eighties and there several technologies that could still improve them significantly such as:

- Camless Valvetrains for improved valve timing control
- Even higher pressure fuel injectors.
- Improved Thermal and EGR management.

On, the other hand, although diesel emissions have improved significantly, they are still a long way from meeting U.S.A. standards .This may be seen in Figure 51. Currently, diesel engines meet Euro 4 standards with some of them being close to Euro 5(not yet implemented) standards with the use of particulate traps. Meeting Euro 5 standards through the use of traps means that diesels will probably meet the US particulate requirements too. However there will be a small fuel economy penalty due to the trap and its regeneration. Moreover, even at Euro 5 levels, the new US (Tier 2 Bin 5) NOx standards are far from being met. Meeting them will most probably involve some degree of ignition timing retard with the associated fuel consumption penalty. Other tricks to bring diesel emissions down to Tier 2 levels include low temperature combustion with extensive exhaust gas recirculation (EGR), more premixed type combustion, lower air to fuel ratios and NOx aftertreatment. Some of these measures have an impact on efficiency (lower air to fuel ratios) others on power density (EGR). For a detailed treatment of some these solutions the reader is referenced to [42] and [43].

Reducing fmep in diesels will probably be more of a challenge than in gasoline engines. Cylinder pressures in diesels are significantly higher than SI engines and therefore so are loading forces that cause friction. It should be mentioned however that there are studies that suggest that huge fmep reductions could be realized for diesels [43]

In this study, keeping all of the above in mind a reduction of fmep of 15% was assumed along with an increase in $\eta_i$ of 7.5%. These assumptions are in agreement with those used in the two previous studies from our research group [7, 8]. The improvement in indicative efficiency assumed for diesels is the same as the one assumed for gasoline engines. This seems to be contradictory with the fuel consumption penalty that should be expected for diesels to meet emissions standards. The engine map that was provided from Cummins however came from a new prototype operating under heavy emissions regulation. Its indicative efficiency was therefore relatively low (~45% in an average sense). This is why it wasn't penalized further.

Except for the different assumptions used, the process of predicting future diesel engine maps is identical to the one used for gasoline engines. The future diesel map for the 2030 lower performance Camry is presented in Figure 52

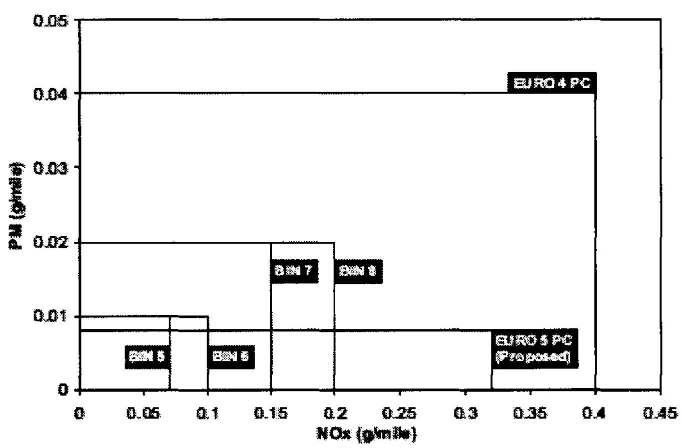



**Figure 51: Comparisons between European and U.S. emissions standards**

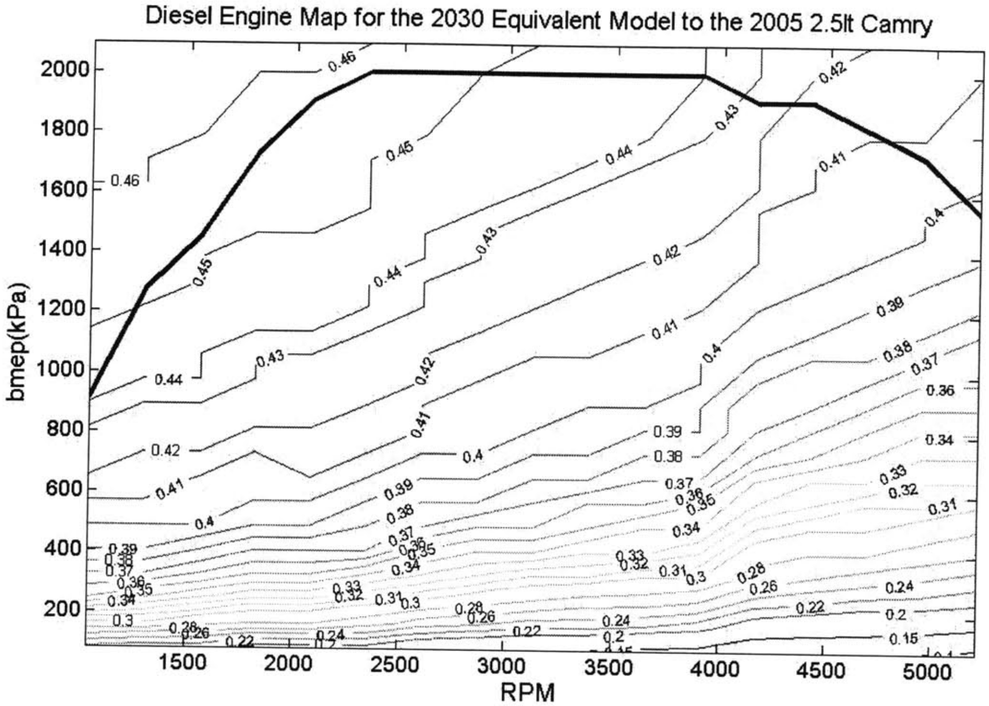



**Figure 52: Diesel Engine map for the 2030 diesel lower performance Camry indexed by bmep**

### 3.4.2. Sizing-Scaling methodology

The sizing calculation for the diesel engines is the same as the one for gasoline engines. The only difference is the values of bmep and gravimetric power density used.

With the introduction of indirect injection, diesel engine volumetric power densities have increased dramatically as seen in Figure 53 adapted from [44]. Diesels are currently at 45-60kW/lt. The current Cummins engine used in this study achieved 54.7kW/lt at a maximum 18 bar bmep. [44] predicts 70kW/l by the end of the decade and mentions that 100kW/l might be possible by 2020. In this study, the goal is not to model the best technologies available but rather a fleet average number. A value of peak bmep of 20 bar was therefore assumed. The corresponding volumetric power density is around 70 kW/lt.

In terms of gravimetric power density, data and projections are scarce. Some of the best current diesel engines achieve densities of 0.7-0.9kW/kg [3] with extensive aluminum and grey cast iron-aluminum alloy use. If these numbers are scaled by the improvement in bmep, the resulting gravimetric power density is the same as what was assumed for the NA SI 2030 engines which was considered too high for diesels. In order to use a number that reflects the future fleet average and not the best future engines, a 0.7 kW/kg power density was assumed. The accuracy of this assumption doesn't influence the results of this study significantly as the engine is only about 10% of the vehicle curb weight.

One last thing about the scaling calculation is that as the engine map used comes from an engine not yet on the market, its displacement volume was not known. The

[3] AUDI FSI V8 4.2lt 240 kW, 255kg, Mercedes V6 3lt 165kW,208kg Source MTZ [31]

displacement volume was needed to scale speed so that piston speed is approximately the same. A $V_d$ of 5.9liters, 8 cylinders and an R ratio of 1.1 were assumed.

The resulting engine map indexed by torque and rpm for the 2030 diesel lower performance Camry is presented in Figure 54. Notice how the redline of the engine is higher than the one of the 2005 engine in Figure 50. The 2030 engine is only 1.33liter.

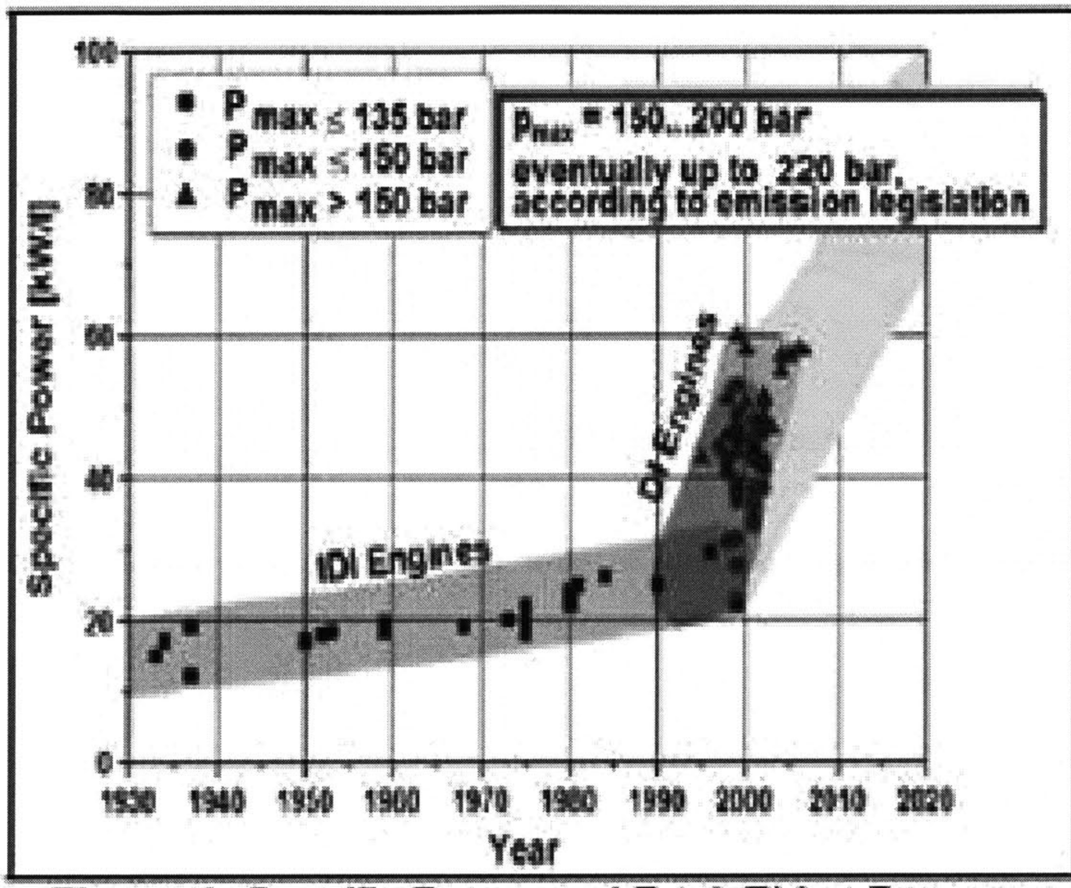


Figure 1: Specific Power and Peak Firing Pressure

**Figure 53: Historical Evolution of Diesel Volumetric Power Density.**

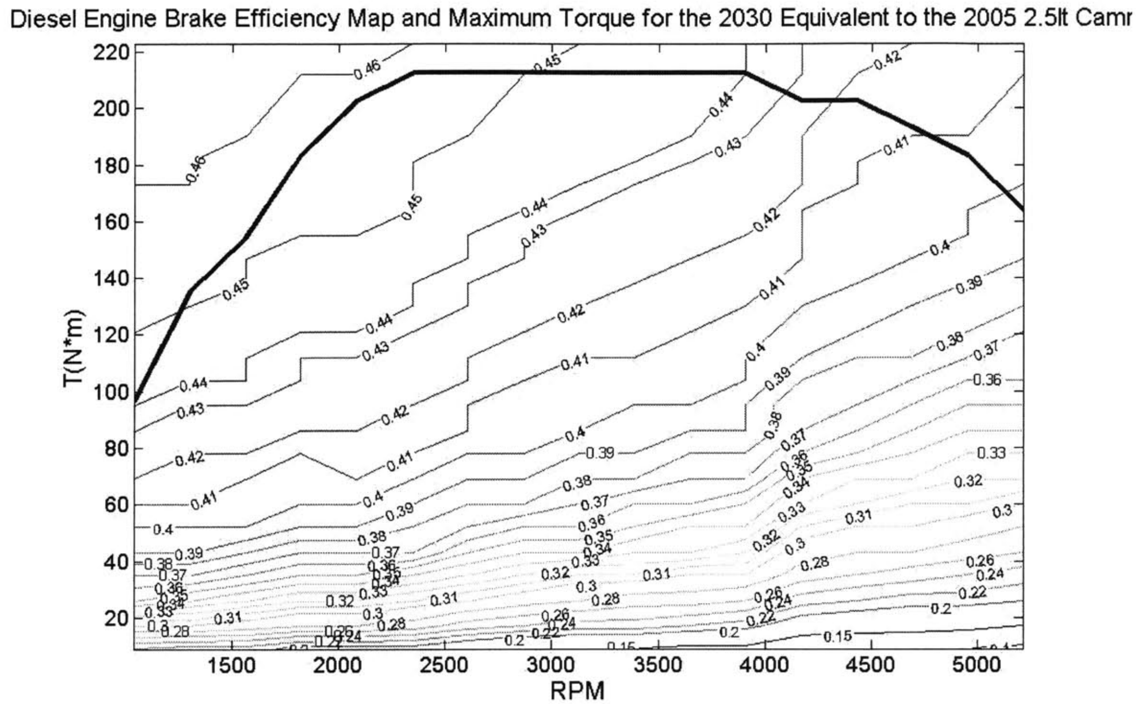



**Figure 54: Diesel Engine map for the 2030 diesel lower performance Camry indexed by torque.**

# 4. Controls-Transmissions for ICE Powertrains

## 4.1. Introduction

A powertrain is made up of more than an engine or a motor. The mechanical power produced by the engine needs to be converted to acquire the characteristics, i.e. torque and rotational speed needed at the wheel. Furthermore, transmission design is the main means to optimize engine operation for fuel economy with a given engine and vehicle. When comparing different powertrains, it is essential that the transmissions chosen are "equivalent". Otherwise, even at the same engine power over vehicle curb weight ratio, vehicles with different powertrains will not offer the same level of performance. Furthermore, their engine operation will not be optimized to the same degree. Fuel economy comparisons will as a result be unfair.

## 4.2. Background-How does a Manual or Automatic Gearbox Work?

Automotive ICE engines operate at speeds higher and torques lower than those requested at the vehicle's wheel to overcome vehicle resistances. A transmission converts the requested torque and speed at the wheel to a torque and speed the engine can actually produce, preferably at high brake efficiency. For a conventional ICE-automatic or manual transmission powertrain as the one presented in Figure 5, the torque on the wheels is related to engine torque by

$$T_w = i_g * i_{fd} * \eta_t * T_{eng} \quad (4.1)$$

Where :
$T_w$ : is the torque at the wheel
$T_{eng}$ : is the torque output of the engine outlet
$i_g$: is the gear ratio of the gear selected in the gearbox at some instant in time.
$i_{fd}$:is the gear ratio of the final drive in the differential.
$\eta_t$:is the efficiency of the entire driveline from right after the engine to the wheels

A gear ratio for any part of the driveline is defined as:

$$i = \frac{N_{in}}{N_{out}} \quad (4.2)$$

The final drive gear ratio ($i_{fd}$) is constant. For simplicity, a total gear ratio may be defined as:

$$i_{tot} = i_g * i_{fd} \quad (4.3)$$

The tractive effort, i.e. the force exerted by the tire on the road is

$$F_t = \frac{T_w}{r_w} = \frac{i_g * i_{fd} * \eta_t * T_{eng}}{r_w} \quad (4.4)$$

Where $r_w$ is the effective wheel radius.
The rotating speed of the driven wheel is

$$N_w = \frac{N_{eng}}{i_g * i_{fd}} \quad (4.5)$$

Vehicle speed (V) is equal to the translational speed of the tire center which assuming no tire slip is:

$$V = \frac{\pi * N_w * r_w}{30} = \frac{\pi * N_{eng} * r_w}{30 * i_g * i_{fd}} \quad (4.6)$$

Where V is in m/s and $N_w$ ,$N_{eng}$ in rpm

In terms of performance of the vehicle, the ideal powerplant would produce its maximum power, constantly over the entire speed range. That is:

$$P = T * \omega = const \Rightarrow \mathrm{T} = \frac{const}{\omega} \quad (4.7)$$

Maximum torque should therefore, ideally, vary hyperbolically with speed. To be exact, for very low speeds, this would lead to very high torques which would cause the tire to slip. At the low speed end therefore, the ideal torque speed is limited to constant torque determined by tire-road adhesion. Then it becomes a hyperbola. The ideal powerplant maximum torque and power curves are presented in. The shape of this curve is very similar to that of a permanent magnet DC electric motor, so electric cars with that type of motor only need a one speed transmission. An ICE maximum torque curve like the ones presented in chapter 3 however, is significantly different, so many gear ratios (ideally infinite) are needed to transform it to something close to the ideal torque curve. The tractive effort versus vehicle speed for an ICE powerplant with a 4 speed manual transmission is presented in Figure 56.For a constant engine operating point, a higher gear speed means a lower gear ratio, so, the maximum tractive force is lower but the tire speed is higher.

The total vehicle resistance for different road grades is also presented in Figure 56.At a given road grade, the vehicle will travel at most at the speed of the intersection of the corresponding resistance curve and one of the tractive effort curves. The powertrain can however operate anywhere below the maximum tractive effort curve. The maximum tractive effort represents wide open throttle-gas pedal to its maximum. Anything below that, at a given gear ratio means the engine is producing less than its maximum torque at a given rpm. For example the vehicle can travel at 70 km/h and 17.6% grade with either the 3rd or 2nd speed. The difference is that the engine will be operating at a lower torque but higher rpm with 3rd versus 2nd gear. It is also obvious that the vehicle couldn't operate at that speed and grade using 1st or 4th gear. Using 1st gear, the required engine speed is higher than the engine redline. Using 4th gear, the maximum engine torque cannot be transformed into enough tractive force to overcome vehicle resistances.

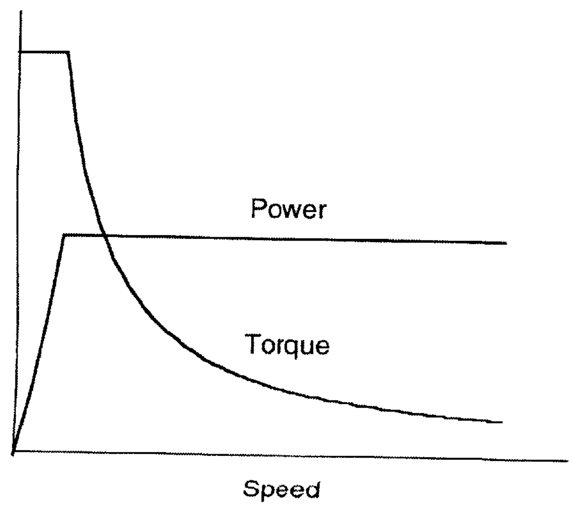



**Figure 55: Ideal Powerplant Torque and Power versus speed Curves. Adapted from [4]**

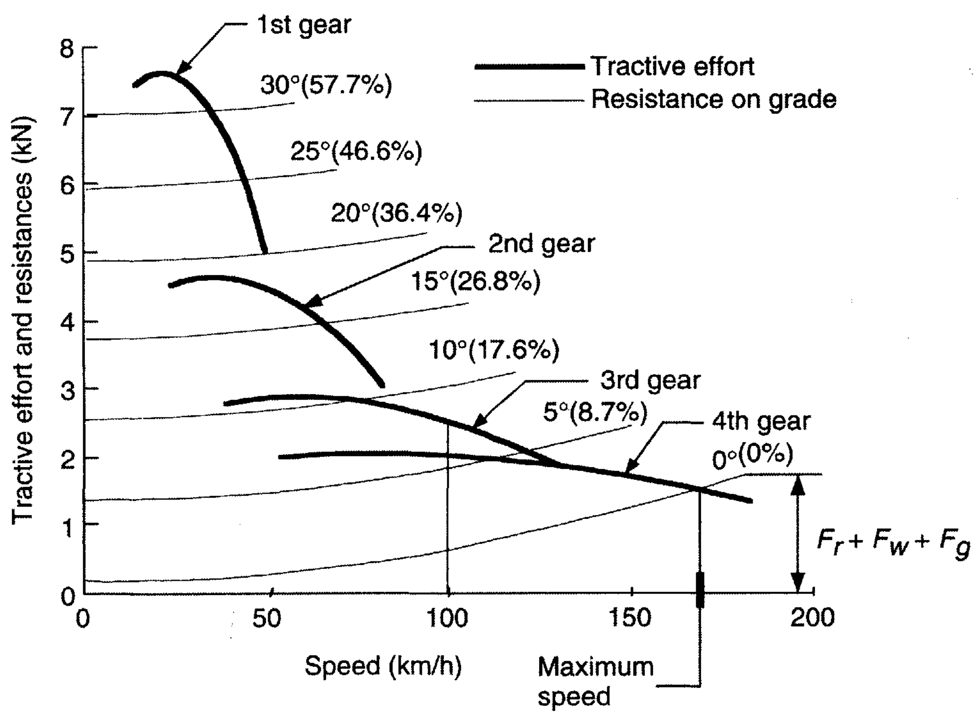



**Figure 56: Tractive Effort for an ICE Powerplant and a 4spd manual Transmission. Adapted from [4]**

For an automatic transmission, things are very similar. There is still a discrete number of gears. The only difference is that there is a hydraulic torque converter instead of a clutch. The converter consists of a pump continuously powered by the engine and a turbine that is powered by the pump, connected to the gearbox. Because of the hydraulic friction, efficiencies for automatic transmissions are slightly lower. More importantly; the torque converter can be used along with the gearbox to transform the engine torque curve further in order to approach the ideal curve even closer. A schematic of a hydraulic torque converter is presented in Figure 57.

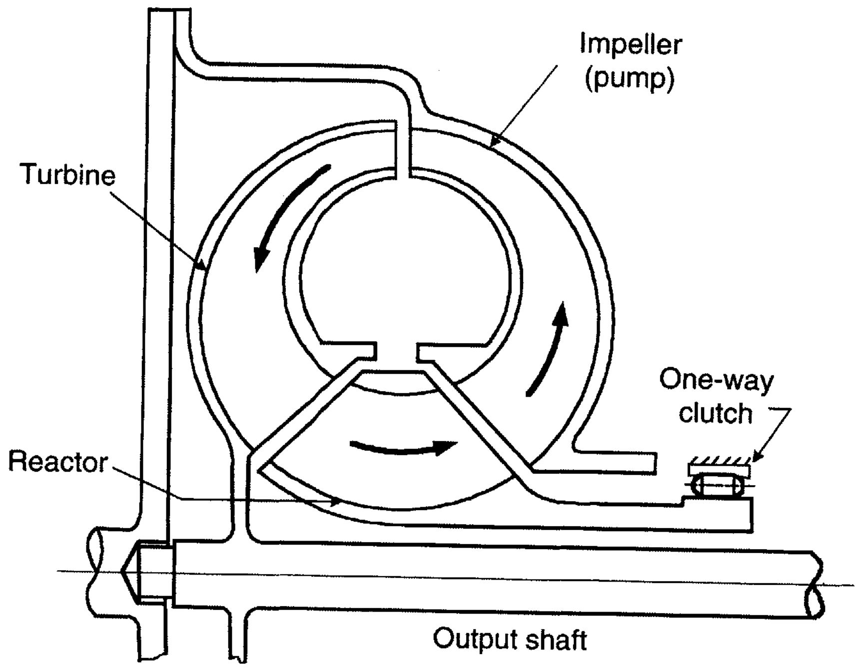



**Figure 57: Schematic of Hydraulic Torque Converter. Adapted from [4]**

For a hydraulic torque converter, the following parameters may be defined:
Speed Ratio:

$$C_{sr} = \frac{\omega_{output}}{\omega_{input}} \quad (4.8)$$

Torque Ratio:

$$C_{tr} = \frac{T_{output}}{T_{input}} \quad (4.9)$$

Efficiency:

$$\eta = \frac{T_{output} * \omega_{output}}{T_{input} * \omega_{input}} = C_{sr} * C_{tr} \quad (4.10)$$

Capacity factor(size factor)

$$K_{tc} = \frac{\omega_{input}}{\sqrt{T_{input}}} \quad (4.11)$$

The Capacity Factor is an indicator of the ability of the converter to absorb or transmit torque. A characteristic map of torque converter $C_{sr}$,$C_{tr}$ and $K_{tc}$ is presented in Figure 58.

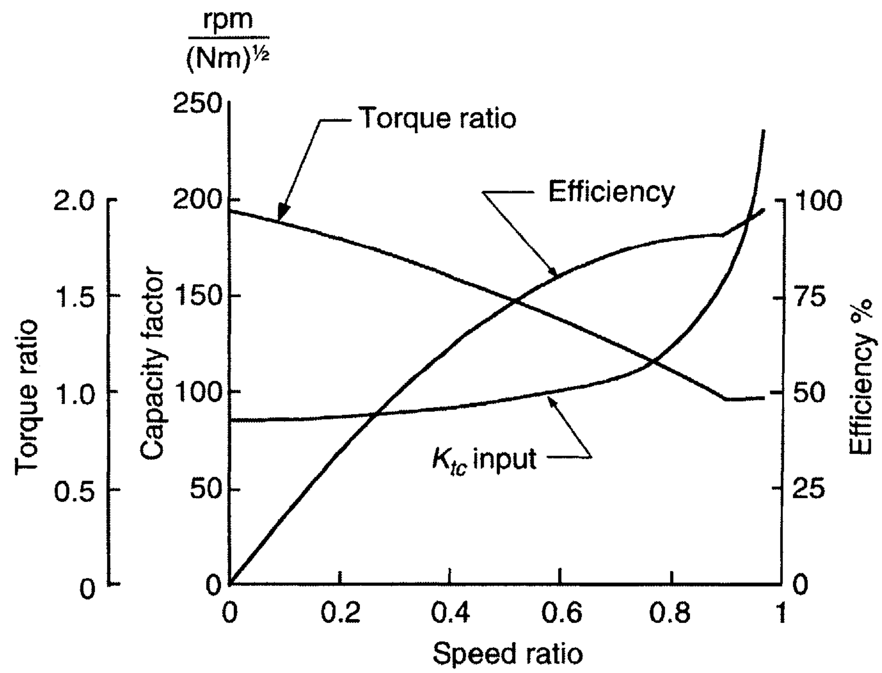



**Figure 58: Characteristics of a Hydraulic Torque Converter. Adapted from [4]**

The tractive force then becomes:

$$F_t = \frac{T_w}{r_w} = \frac{i_g * i_{fd} * \eta_t * C_{tr}(K_{tc}) * T_{eng}}{r_w} = \frac{i_g * i_{fd} * \eta_t * C_{tr}(T_{eng}, \omega_{eng}) * T_{eng}}{r_w} \quad (4.12)$$

$$V = \frac{\pi * N_w * r_w}{30} = \frac{\pi * N_{eng} * C_{sr}(K_{tc}) * r_w}{30 * i_g * i_{fd}} = \frac{\pi * N_{eng} * C_{sr}(T_{eng}, \omega_{eng}) * r_w}{30 * i_g * i_{fd}} \quad (4.13)$$

The hydraulic torque converter acts as an extra, variable gear ratio boosting output torque at lower speeds. The resulting tractive effort curves approach thus the ideal even more. This may be seen in Figure 59.

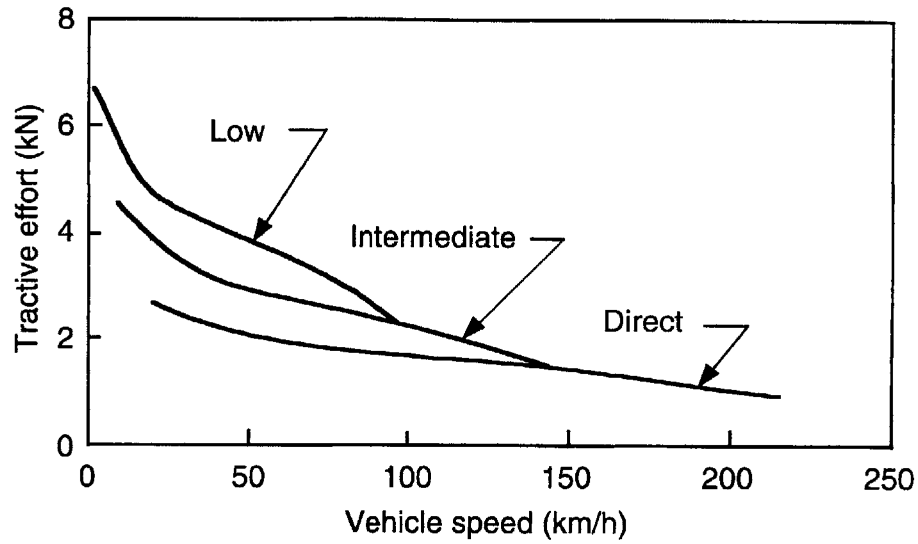



**Figure 59: Tractive Effort for an ICE powerplant and a 3spd Automatic Transmission. Adapted from [4]**

### 4.2.1. Normal Driving Operation

For a given driving pattern, choosing gear ratios is crucial in optimizing fuel consumption. So is, determining, to the extent possible, a gear shifting strategy. In real world driving, the driver can be more or less aggressive in his use of the gas pedal. For a standardized drive cycle, as the ones used for the simulations in this study, the effect of aggressive driving is accounted for in the end using the standard adjustment factors. The operating points of the engine are solely defined by the choice of gear speed at every instant in time as well as the total gear ratio of that speed.

This effect is illustrated in Figure 60. Let's assume that the vehicle is traveling at a constant speed, leading to a constant power demand of 20kW at the wheel to overcome resistances. For simplicity, let's also assume a driveline efficiency of 100% so that the power demand from the engine is also 20kW. The engine operating point for these conditions will lie somewhere on the 20kW constant power hyperbola shown on the graph depending on the gear ratio in the gearbox at that time. A higher total gear ratio means the engine operating point will be towards lower torques and higher speeds. A lower gear ratio will move the operating point in the opposite direction. As optimum efficiencies occur at higher loads, lower gear ratios improve engine efficiency. This of, course comes at a performance penalty as lower gear ratios mean less maximum available torque at the wheel.

Having all this in mind, a gear shifting strategy can be designed. The two lines shown in Figure 60 can be used to specify when gear shifting occurs. So, when the engine operating point reaches the downshift line, the next lower gear is chosen (e.g. from 3rd to 2nd), so the gear ratio is higher, so the operating point moves towards lower torques and higher speeds .Conversely for the upshift line. The engine, remains thus always within the operating window specified by these 2 lines. A plot of the all the operating points during the FTP driving cycle is presented in Figure 61. It is obvious, how all the operating points lie within the "shift window" specified by the two shift lines. Both the gear ratios for every speed and the shifting logic are therefore important in determining engine operating points.

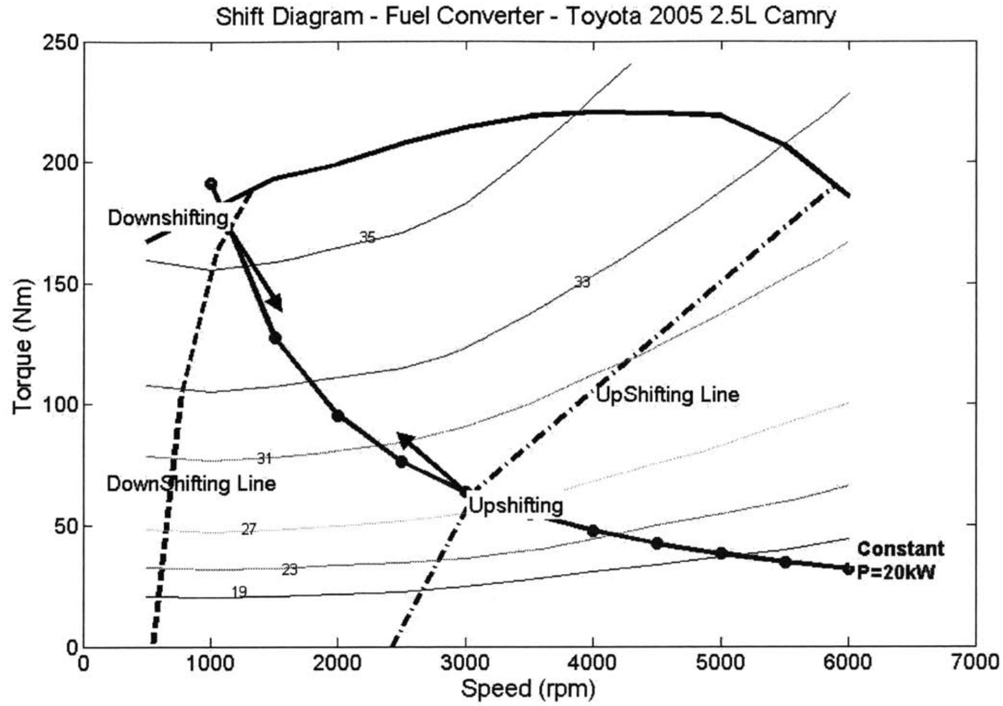



Figure 60: The effect of Gear Shifting on Engine Operating Points

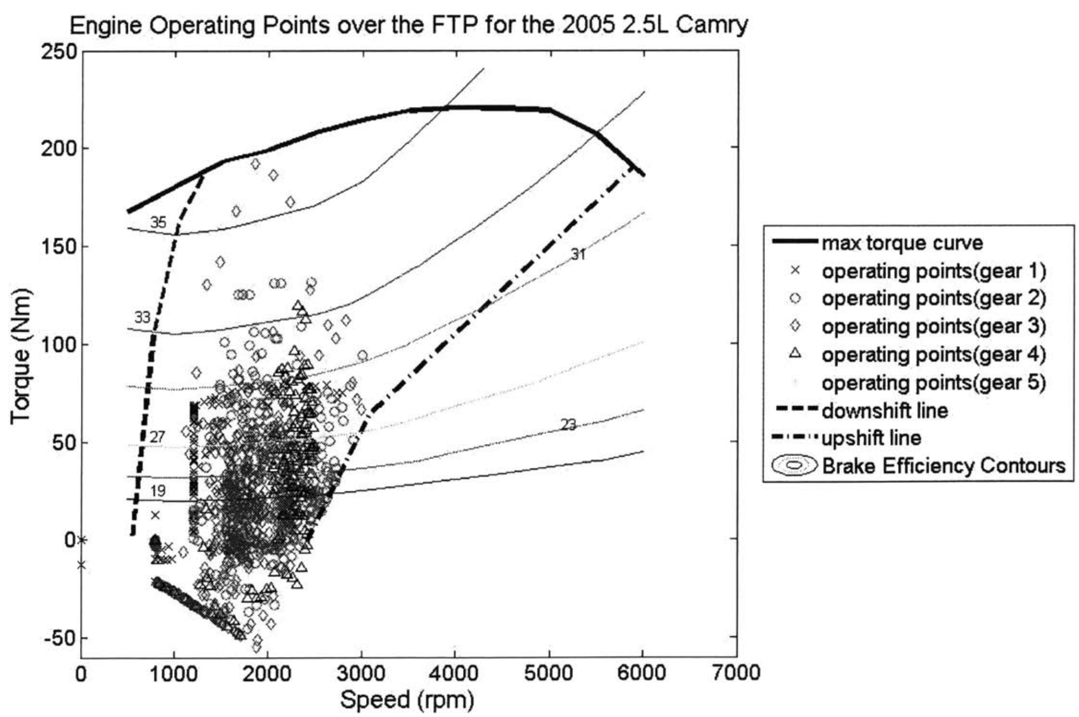



Figure 61: Operating Points for the 2005 2.5 liter Camry during the FTP

Specifying shift lines is a modeling approach. However it is quite close to what a real automatic gearbox does. An automatic gearbox today, has its own controller. That controller can use the engine rotating speed signal from the engine ECU as an input. The engine torque is usually estimated indirectly, from the gas pedal position signal. At any case, the information to allow for "shift window" control logic is available and can be implemented easily. With a manual gearbox, however, the driver shifts whenever he wants. A sensible driver, who listens to the engine while driving, would nevertheless drive in a manner similar to that specified for the automatic gearbox.

It is noteworthy how the "shift window" becomes broader at higher torques. The reason is that the maximum engine power needs to be within the window in order to meet the performance requirements. This however doesn't affect fuel economy significantly since as seen in Figure 61 during everyday driving, such high power outputs are not required from the engine. Narrowing the "shift window" at low loads will significantly improve average engine efficiency as operation will be forced to higher loads. This would however create drivability issues as the number of shifts would increase significantly as would noise and vibration as a result.

### 4.2.2. Performance Tests.

As already explained in chapter 1 there are different performance criteria according to what the vehicle is used for. In this study, the main performance criterion that will be used to equalize current and future vehicles will be the 0-60 miles per hour (mph) acceleration time. Additionally, 40-60 mph acceleration times, maximum speed, as well as grade and towing capabilities will be calculated and compared between the different vehicles.

## Acceleration Performance

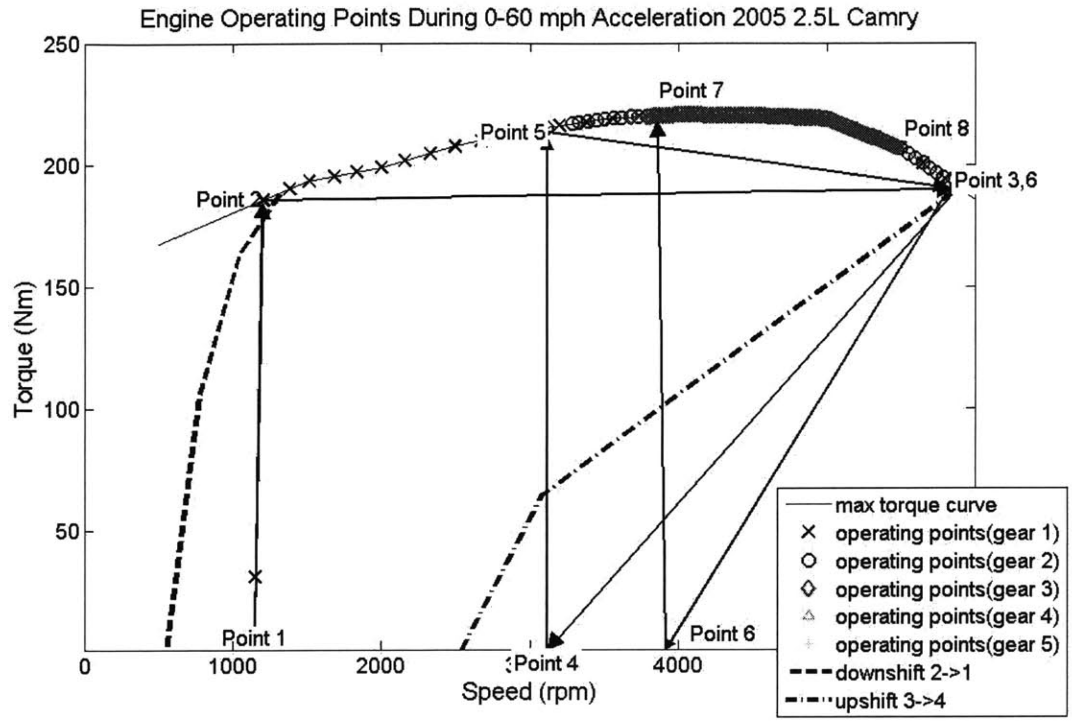



**Figure 62: Engine Operating Points during 0-60mph Acceleration.**

Engine operation during a "hard" 0-60 mph acceleration is presented in Figure 62. The powertrain starts at a low speed and torque using 1$^{st}$ gear. The engine quickly gets to wide open throttle torque .The vehicle and engine subsequently accelerate until the vehicle speed is so high that the engine reaches it's redline. An upshift to 2$^{nd}$ gear brings the engine to a lower speed on the maximum torque curve. This engine speed is set by equation (4.6) using the 2$^{nd}$ gear ratio for $i_g$. For a fraction of a second, while the 1$^{st}$ gear is being disengaged and the 2$^{nd}$ engaged there is no torque transmitted. The engine accelerates next with the 2$^{nd}$ gear until it reaches it's redline. A second upshift to 3$^{rd}$ is called for. The engine accelerates along the maximum torque curve once again. This time, the redline is not reached as 60 mph correspond to a lower engine speed. A 40 to 60 mph acceleration is similar with the difference that the engine starts at an intermediate speed and torque and possibly not using the 1$^{st}$ gear but the 2$^{nd}$ or 3$^{rd}$ instead.

Higher gear ratios will generally lead to better acceleration times. Provided that the tire doesn't slip; higher gear ratios lead to higher torques at the wheel and therefore better acceleration. There is an upper limit however; a higher gear ratio will lead the engine to it's redline faster, i.e. at a lower vehicle speed. Then an upshift will be needed. The vehicle will spent more time using e.g. 3$^{rd}$ instead of 2$^{nd}$ so at a lower gear ratio. Higher gear ratios do therefore generally decrease acceleration times but up to a limit.

## Maximum Speed

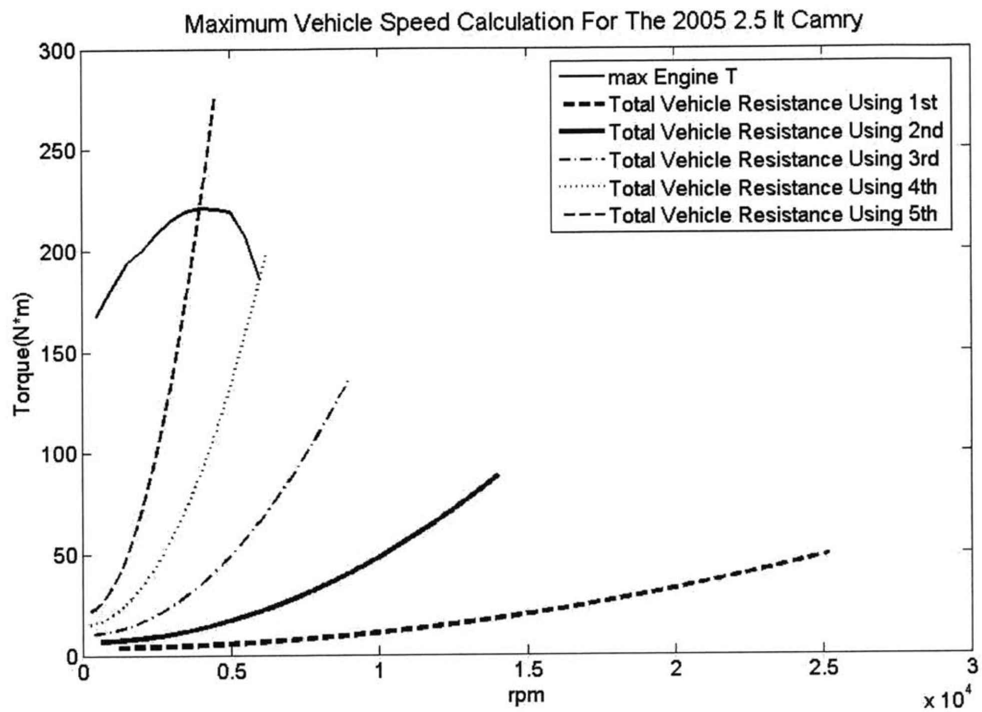



**Figure 63: Maximum Vehicle Speed Calculation for the 2005 2.5 liter Camry**

For given vehicle aerodynamics, maximum vehicle speed depends on both the engine maximum torque curve and the gear ratios in the gearbox. In order to determine the maximum vehicle speed, the engine torque versus speed curve needs to be converted to a tractive force versus vehicle speed and be compared to the total vehicle resistance versus speed. The process was illustrated in Figure 56. Maximum vehicle speed is defined as the intersection of the 0% grade line with one of the tractive effort curves.

Alternatively, the total vehicle resistances curve at 0% grade can be converted into a required engine torque versus engine speed for all the different gear ratios of the transmission. This is presented for the 2005 2.5 liter Camry in Figure 63. All the required torque curves correspond to vehicle speeds 10 to 250km/h. However, with the first 3 gear ratios, the required engine revolutions to achieve high speeds are way off the engine limits. For lower gear ratios, the curve becomes steeper (higher engine torques are required). The vehicle speed can be determined at the intersections of the maximum engine torque curve with the required torque ones from equation (4.6).If there more than one intersections, the one that corresponds to maximum vehicle speed is chosen. A higher gear speed curve will intersect at lower engine rpm's but those are divided by a lower number to find vehicle speed. Maximum speeds are, therefore, as expected, generally achieved with the highest speed in the gearbox.

If the gearbox is optimized for maximum vehicle speed, the intersection of the highest gear speed curve with the maximum engine torque curve should be close to maximum engine power. This is apparently not the case with the Camry as seen in Figure 63. That would require a lower top gear ratio.

## Grade and Towing Capability.

The grade and towing capabilities of a vehicle are not in principle difficult to calculate. The process is essentially the same as calculating maximum vehicle speed. The problem is that this calculation depends strongly on the assumptions used to define these performance tests. The grade and towing tests are not standardized. There are, therefore the following uncertainties:

- What is the speed at which the towing and grade tests are defined?
- What is the grade requirement for the towing test?
- Is downshifting allowed during the two tests?
- An additional uncertainty stems from the fact that during towing, the coefficient of drag of the vehicle is increased. The degree of that increase depends on the geometry of the towed body.

In terms of the speed of the tests, information from different sources in the auto industry suggest 40-65 mph for cars. For trucks it could be lower as suggested by the results for current vehicles. 55mph were used for all the tests in this study. For the grade requirement for towing, it is definitely less than 6% which is the maximum allowed road grade. Different sources suggest either 3% or 6%. Most of the tests in this study were conducted assuming 6%. When it comes to downshifting, there are two main approaches:

- That the towing and grade tests are conducted with the top gear speed. This seems to be closer to some of the results published by the industry.
- That the tests are conducted using the lowest possible gear speed that the vehicle speed of the test allows. This seems closer to real life.

## *4.3.Background-How does a CVT Gearbox Work*

A CVT or continuously variable transmission[4] is a transmission with essentially infinite gear ratios in between its first and top speed. Several ways to implement this concept mechanically have been used. The most common includes two conic pulleys and a belt drive. The 1st pulley is connected to the input shaft, the 2nd to the output. By varying the effective diameter of the pulleys any gear ratio between an upper and a lower limit can be achieved. Any torque and speed demand at the wheel can thus be matched with any engine torque and speed. The tractive effort curve therefore approaches the ideal. Traditionally, CVT gearboxes have found extensive application in low torque powertrains such as ones used in scooters but very limited application in automotive powertrains. The main reason is that the belt drive is very limiting in terms of the maximum torque that can be transmitted. Additionally, their low efficiencies usually negate the benefit from optimizing engine operation. Furthermore they are usually more expensive than conventional gearboxes. They will be examined in this thesis as several new designs are currently looking promising for the future. A picture of the pulley belt assembly of the Nissan CVT system is presented in Figure 64.

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[4] CVT gearboxes are also called sometimes called IVT (Infinitely Variable Transmission).

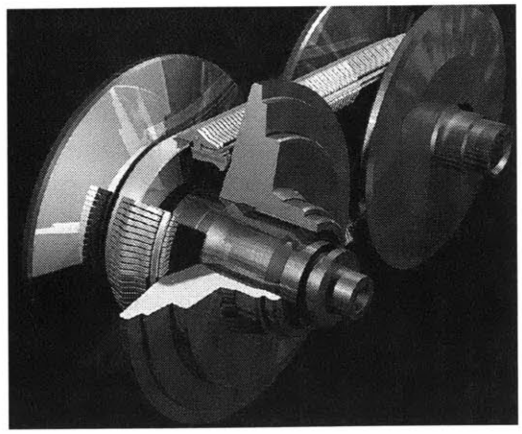

**Figure 64: Pulley-Belt Assembly for the Nissan CVT system**

### 4.3.1. Normal Driving Operation

When using a CVT gearbox, any wheel torque (T) and rotational speed (ω) can be converted into any engine T and ω. Instead of a shift window therefore, a desired engine operation line is specified. This line is obtained by connecting the minimum break specific fuel consumption (bsfc) points for every engine speed. Such an optimum operation line is presented in Figure 65. For every power request from the engine, the desired operating point is the intersection of the optimum operation line with the power hyperbola. Given the speed of the vehicle, the required gear ratio is subsequently selected at the CVT. The engine operating points over the Japanese 10-15 driving cycle for a vehicle using a CVT versus those of the same vehicle using an automatic transmission are compared in Figure 65. It is obvious how the CVT achieves much better optimization of operating points, generally staying close to the optimum line.

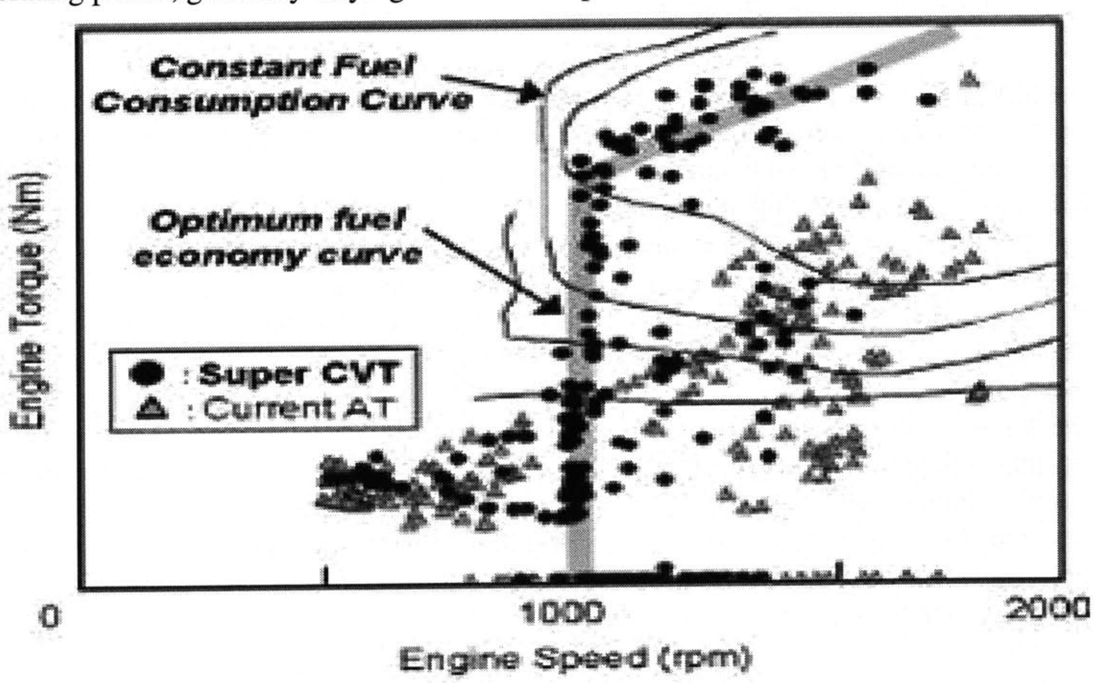


**Figure 65: Comparison of Engine Operating Points when using a CVT and an Automatic Gearbox. Adapted from [45]**

### 4.3.2. Performance Tests.

The engine operating Points during 0-60 mph acceleration for the 2005 2.5 liter Camry using a CVT are presented in Figure 66. The total gear ratio (including final drive) during the acceleration is presented in Figure 67. The phenomena are similar to the acceleration using an automatic gearbox. For comparison, the gear ratio during acceleration for the same vehicle using an automatic transmission is presented in Figure 68. The CVT accelerates using 1$^{st}$ along the maximum torque curve until the point of maximum power, which is close to the redline. There is no discrete down shifting, but instead the gear ratio is reduced gradually as vehicle speed increases to keep the engine at its maximum power. At the same gearbox efficiency, a CVT would result therefore in better acceleration times compared with an automatic or manual. However, the low efficiency of the CVT- about 75% for the best current systems [45] compared with 87% for an automatic negates the benefit from optimized engine operation. The end result is 9.4 seconds 0-60 mph for the automatic, 9.6 sec for the CVT.

In terms of the rest of the criteria (maximum speed, grade, towing) the CVT performs in a similar way compared with the automatic. The benefit from optimized engine operation close to its maximum power is negated from the lower efficiency of the transmission. Performance tests will not be presented for CVT equipped models. The reader can safely assume that they are very close to the equivalent automatic gearbox model.

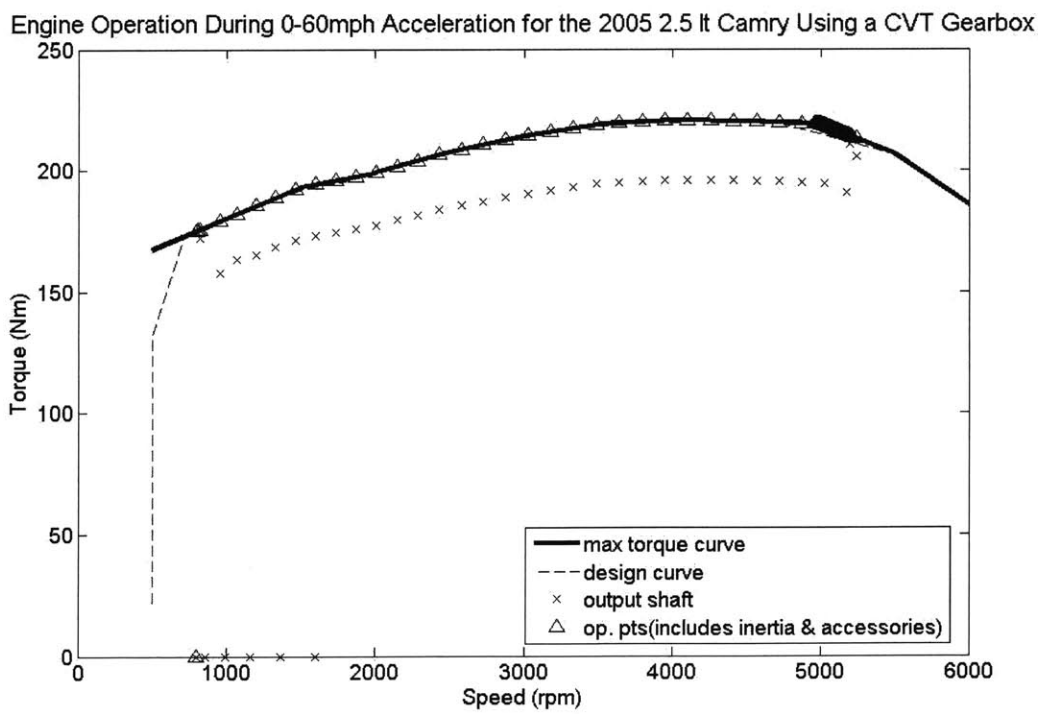


**Figure 66: Engine Operating Points During 0-60 mph Acceleration For the 2005 2.5 liter Camry Using a CVT.**

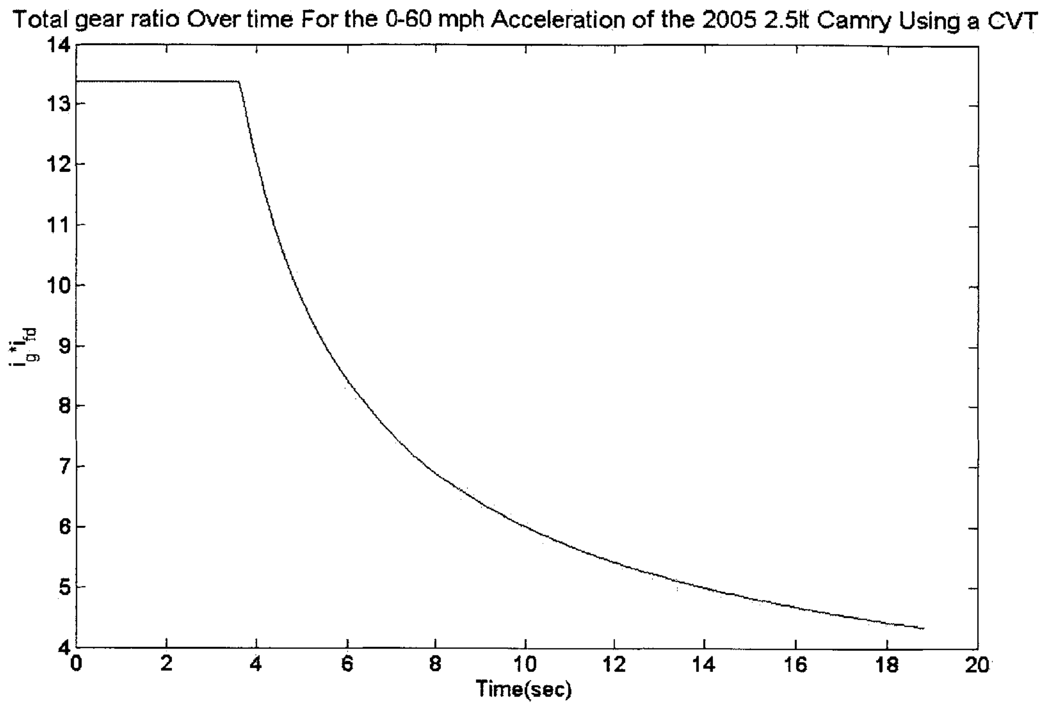


**Figure 67: Total gear ratio during a 0-60 mph acceleration for a CVT**

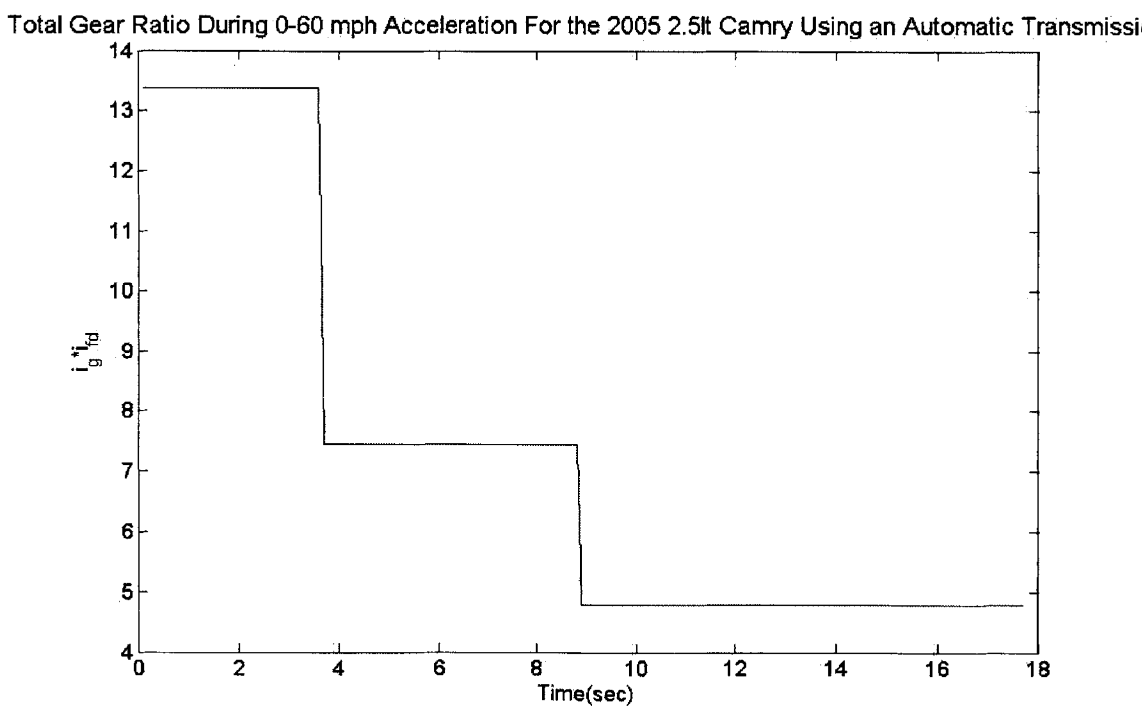


**Figure 68: Total gear ratio during a 0-60 mph acceleration using an Automatic Transmission**

## 4.4. Choosing Gearboxes for 2030 Vehicles

### 4.4.1. Type

The main choice of type of gearbox for the future models will be a 6 speed automatic transmission. Manuals won't be considered as the focus of the study is the U.S.A. fleet where manual transmissions are a very small fraction of the sales. The public is very used to the convenience of an automatic and is not likely to change. Manuals are not significantly different in terms of fuel economy. Theoretically there should be a small gain in fuel economy for manual gearboxes over automatics due to better gearbox efficiency. However:

- In real life driving, the benefit of the higher transmission efficiency can only be realized if the driver shifts in a "sensible" way to optimize fuel economy. With an automatic gearbox the powertrain designer instead of the driver decides when the gearbox will shift. That logic is embedded in the gearbox control software.
- As will be explained, there are several technologies just reaching the mass production level that essentially combine the convenience of an automatic with the efficiency of a manual.

As it will be explained in the results chapter, a CVT gearbox at today's level of transmission efficiency offers no significant benefit in fuel economy. Its advantage in optimizing engine operation is cancelled due to higher transmission losses. There are however certain designs at the development stage today that promise significant gains in CVT efficiency. The automotive industry is generally skeptical about the chances of success of these efforts. Furthermore, the cost of a CVT is and probably will be higher than that of an automatic transmission [46]. An efficient CVT option was therefore included in all the calculations in this study, but only as a secondary, less probable scenario.

### 4.4.2. Gear Ratios

As already explained, choosing the gear ratios strongly affects performance and fuel economy. The main constraint in this study is that the all the different future powertrains need to exhibit the same performance, mainly as expressed in the 0-60 mph time. The following semi-empirical methodology was therefore applied in choosing gear ratios. This methodology is a simplified version of what is actually used in the auto industry to select gear ratios for light duty vehicles [46]. Note that as already mentioned, all the vehicles have the same peak engine power over curb weight ratio. In order to equalize performance, the following steps are needed:

1. **Choosing 1st Gear-Equalizing Maximum Tractive Force per Unit Weight:**

The most important parameter in equalizing performance is choosing the 1st gear ratio. Usually, only the first 3 speeds are used in a hard acceleration. The way the 1st gear ratio is chosen for a car is that it has to provide as much tractive force as the tires can put on the road. For the 2005 2.5 liter Camry therefore:

- Curb vehicle weight is 1571 (including passenger).Assuming a 50-50% weight distribution, the weight force on the front axis(this is a front wheel drive car) is:

$$F_W = 50\% \times 1571 \times 9.8 = 7706N$$

- The maximum engine torque is $T_{eng}$= 203 N*m. As maximum engine torque varies with rpm, the average was used for all calculations.
- The total 1st gear ratio is 13.37 and the wheel radius is 0.324m. The maximum tractive force on the driving wheels therefore is:

$$F_{tr} = T_{eng} * i_g * i_{fd} * r_w = 203 * 13.37 * 0.324 = 8377N$$

- The required tire friction coefficient μ for the wheel to actually apply this force on the road without slipping is:

$$\mu = F_{tr}/F_W = 1.09$$

Similarly, for the 3.0 liter Camry:

$$\mu = F_{tr}/F_W = 1.42$$

For the 4.2 liter F150:

$$\mu = F_{tr}/F_W = 1.17$$

The value of μ determines how wide the tires need to be. When selecting gear ratios to design an actual powertrain, a value of μ=1 can be used as a first approximation to select the gear ratio. Light duty vehicle tires don't deviate that much from μ=1. More importantly, μ is a main indicator of performance. Two vehicles that have the same performance should put about the same tractive force per unit weight on the road. They should therefore have similar μ's.

If therefore, choosing the 1st gear ratio for future powertrains results in the same μ required at the tire as their 2005 counterpart, they should have the same performance.

**2. Choosing Gear Span-Equalizing Engine Operating Points:**

Gear span is the number obtained if the 1st gear ratio is divided by the top. The top gear ratio is usually picked for cars in the industry to optimize fuel economy as well as avoid operation at the engine's lug limit. An engine's lug limit is usually located somewhere at low rpm and high torque. Engine operation in that area of the map tends to be avoided as it is very noisy. The criterion used is that at a vehicle speed of 47.5 mph, the engine should be around 1300 rpm using the top gear. Empirically, this criterion has been found to optimize fuel economy for the U.S. standard cycles while avoiding operation at the lug limit.

Using equation (4.6) to back calculate the actual engine operating speed at 47.5 mph vehicle speed for the 2005 vehicles, the results are:

For the 2.5 liter Camry:

$$N_{@47.5mph} = 1493rpm$$

Similarly, for the 3.0 liter Camry:

$$N_{@47.5mph}=1558rpm$$

For the 4.2 liter F150:

$$N_{@47.5mph}=1989rpm$$

A higher rpm number means a higher top gear ratio which means a smaller span. This would deteriorate fuel economy as higher gear ratios result in lower engine torques. It would however probably improve acceleration and definitely better low vehicle speed towing and grade. It would also improve towing and grade if no downshifting were allowed. This is probably why the truck has significantly lower gear span.

Whatever the reasons are for choosing a particular top gear ratio, if the same top gear ratio is chosen for a future SI engine, the operating points over a driving cycle should be similar. However, future engines in this study are generally faster because they are downsized. The top gear ratio was therefore chosen so that the speed at 47.5mph is whatever it was for the equivalent 2005 model scaled by the ratio of future over current redline:

$$N'_{@47.5mph} = N_{@47.5mph} * \frac{N'_{max,eng}}{N_{max,eng}} \quad (4.14)$$

Prime variables indicate future quantities while non-prime ones are current ones.

For future diesel and turbocharged SI engines the choice of $N_{@47.5mph}$ wasn't clear. These engines have different lug limits and different maximum torques for the same power output with a naturally aspirated gasoline. The diesel has higher maximum torque as it is slower, the turbo SI lower because it is faster. The general recommendation of 1300 rpm scaled by $\frac{N'_{max,eng}}{N_{max,eng}}$, where $N_{max,eng}$ is the current diesel or turbo redline was used for all the future diesels and turbocharged SI engines. These engines have an almost constant maximum torque curve, so the choice of a lower $N'_{@47.5mph}$ thus a lower higher gear ratio seemed logical and indeed gave equivalent performance. It could be argued however, quite reasonably, that for the truck a higher top gear ratio should have been used even for the diesel and turbo engines. The effect of that on fuel economy and acceleration would be small. The effect on grade and towing would on the other hand, be significant. Since the definition of these performance criteria was not clear, the value of 1300 rpm was used everywhere for simplicity.

### 3. Choosing Gear Step-Equalizing Engine Operating Points while Maintaining Drivability.

After the 1st and top gear ratios have been chosen, the final task left is determining the intermediate gear ratios. The ratio of two successive gear ratios is called the gear step. Shorter steps generally result in higher fuel economy because engine operation is more accurately controlled. The steps cannot however get too close due to drivability concerns caused by an increase in the number of gear shifts. Heavier duty trucks and race cars usually have smaller gear steps. Finally, it should be mentioned that the gear step for lower gear speeds should generally be higher. In other words, the ratio of the 2nd over first gear ratio should be higher than 3rd over 2nd. This improves drivability significantly in heavy traffic conditions.

The auto industry has over time empirically standardized what the gear steps should be in different types of vehicles. The design space for different types of vehicles is presented in Figure 69. A typical specification for family car vehicles was used for all the vehicles in this study and is presented in Figure 70.

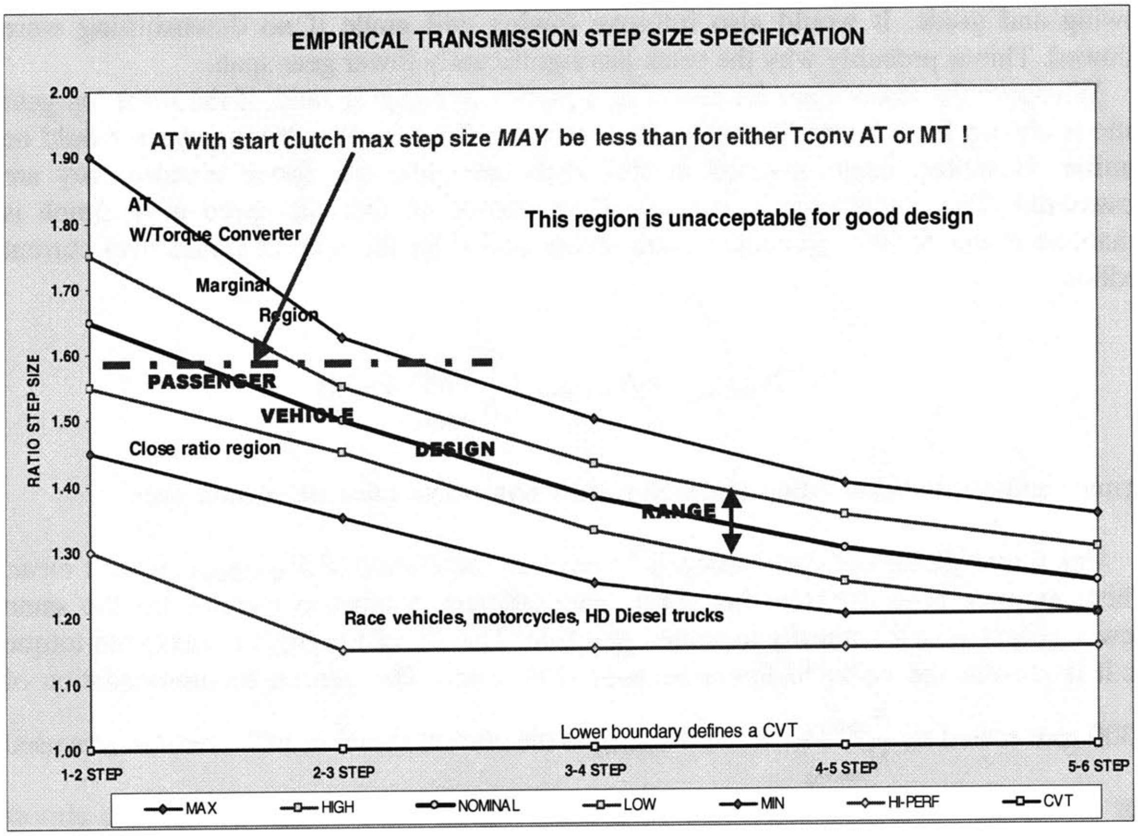


**Figure 69: Empirical gear step size specification [46].**

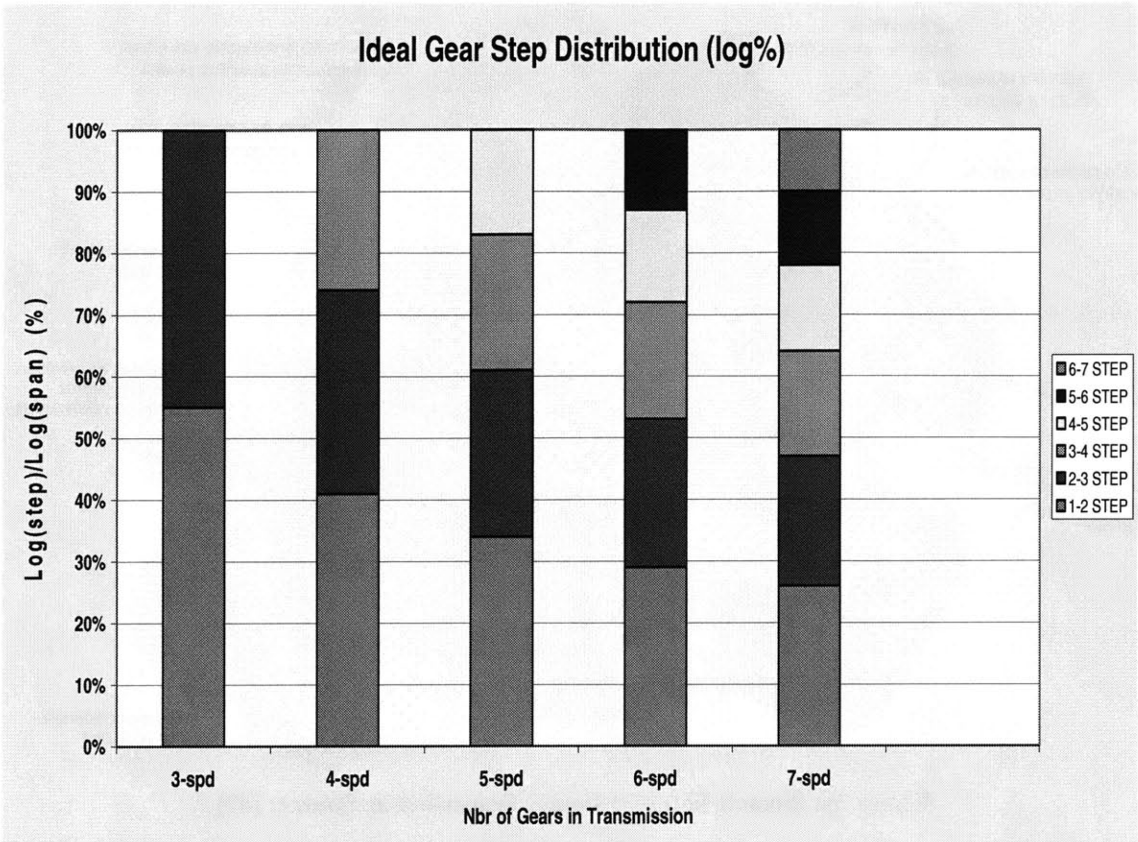


**Figure 70: "Ideal" Empirical Gear Step Sizing for transmissions with different speed numbers [46]**

### 4.4.3. Transmission Efficiencies

There are two main questions that need to be addressed:

- How much will the efficiency of automatic gearboxes improve in the next 25 years?
- How much will the efficiency of CVT gearboxes improve in the next 25 years?

The assumption that will be used to answer the first question is that future automatic gearboxes will reach the efficiency levels of today's manuals. The average efficiency of future gearboxes was assumed to be 93%. There are a couple of automatic transmission systems currently reaching the mass production level that already achieve this level of efficiency. Two of them will be shortly introduced here.

The first system is a robotic manual transmission developed by Renault and used in two of its models (the Renault Traffic and the Renault Master). This system consists of a conventional manual transmission and clutch in which a controller and a hydraulically actuated mechanism do the gear shifting instead of the driver. The system is presented in Figure 71. There is no torque converter, hence no fluid friction losses.

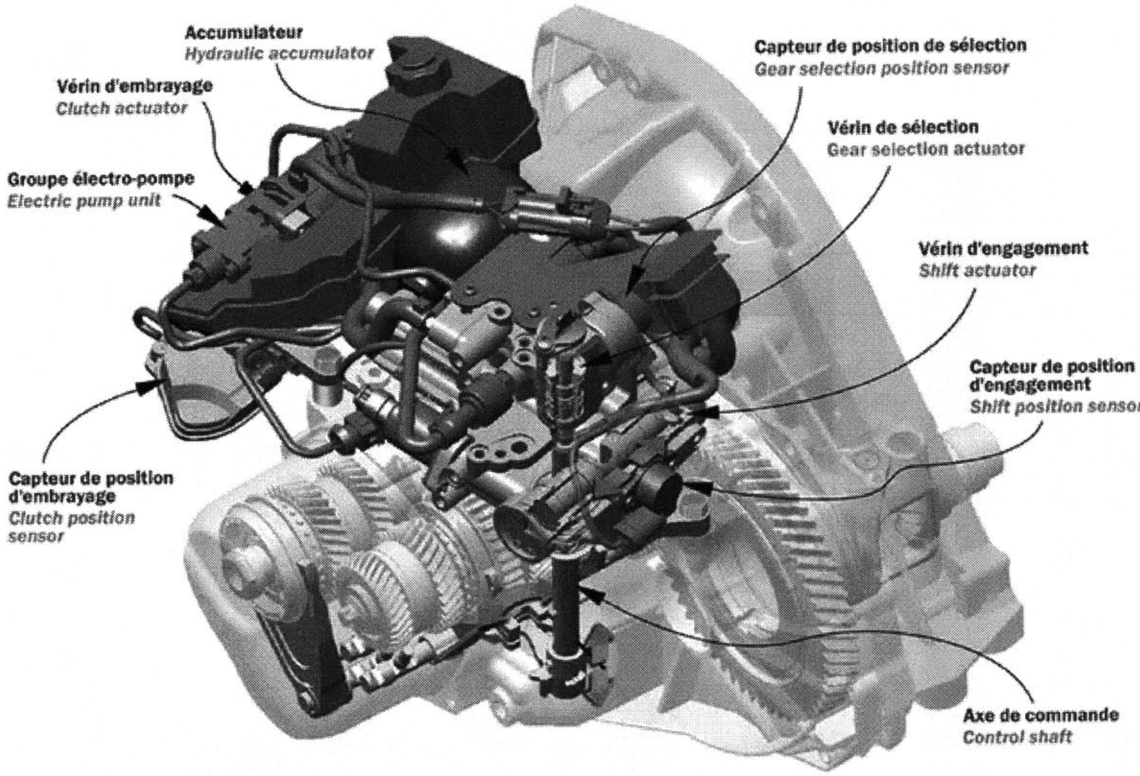


**Figure 71: Renault Robotic Manual Transmission. Source: [47]**

The second system is the dual clutch transmission system already used in high end Audi, VW and Ford models in the European market. The core of the system is presented in Figure 72. It consists of 2 input shafts, 2 hydraulically actuated clutches and gear selectors. One of the clutches controls even gears, the other odd ones. During shifting, the controller smoothly disengages one clutch while engaging the other after the gear selector has already connected the gear to the shaft. Shifting is therefore very smooth. Additionally, although everything is computer controlled, there is no hydraulic torque converter permanently engaged so efficiency is high.

It could be said that with technologies like the robotic manual and the dual clutch system the comfort of an automatic is combined with the efficiency of a manual. The two transmission types thus converge and perhaps the term manual/automatic is more appropriate.

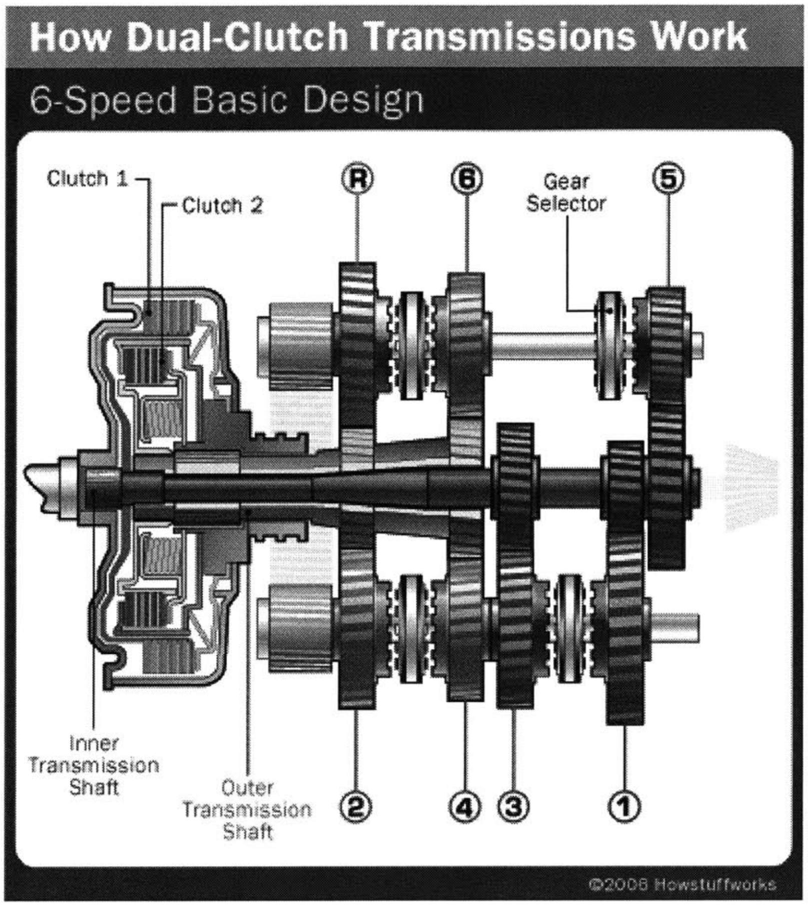


**Figure 72: Dual Clutch Transmission Mechanism. Source [48]**

In terms of future CVT efficiency, a promising idea is the use of toroidal rollers instead of the traditional pulley system. Such a system is developed by Torotrak and documented in several SAE papers [49, 50, 51, 52]

The operating principle of the system is presented in Figure 73 and 74. Changing the angle of the toroidal roller changes the gear ratio in connecting the two shafts. This system has no pulleys, it is therefore not torque limited and the company reports efficiencies around 90% for the variator module. Including the rest of the system efficiencies should be somewhere between 80% and 90%.

There are issues related with the complexity, durability and control of systems like this one and the auto industry is generally skeptical about CVT's including this one. However, in order to explore what the effect of an efficient CVT would be , an efficiency of 87% , which seems reasonable for the Torotrak, was used for future CVT's in this study.

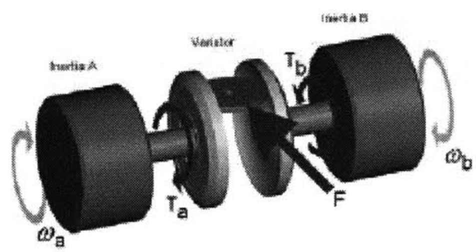

**Figure 73: Toroidal CVT operating principle**

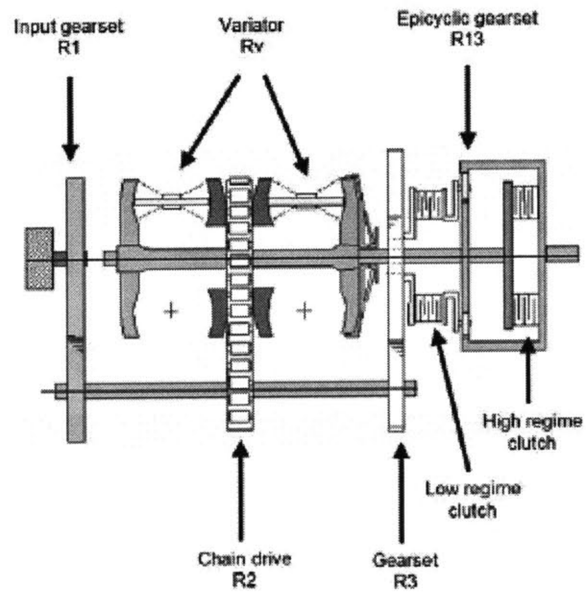


**Figure 74: Toroidal CVT Schematic**

### 4.4.4. Gear Shifting Methodology

Shift scheduling can significantly affect fuel economy. When the shift window is narrowed, the engine is forced to operate at higher loads. Narrowing the shift window however, results in higher noise and vibration. Additionally, there are some hardware limitations such as the transitional performance of the pump in the hydraulic torque converter, accessory speed requirements etc. For these reasons, the default gear shifting window in ADVISOR, presented in Figure 60 will be used throughout this study.

Some new gearbox technologies such as dual-clutch transmissions have a much smoother shift of power during a gear change. The shift window in such a powertrain could potentially be narrowed. In order to show the effect of an advanced transmission that would allow narrowing down the shift window; a second, "narrow", shift window strategy was used in addition to the standard one. The narrow shift window is presented in Figure 75. This shift strategy would be very difficult to implement without running into serious noise and vibration problems. Additionally, a transmission implementing such a strategy would need to have controls intelligent enough to change between using the narrow window for normal driving and a using broader window when high performance and acceleration is required. The purpose of including this alternative shift strategy in the calculations is illustrative in order to show the effect of narrowing the shift window and not actually predicting that such a shift strategy will be possible.

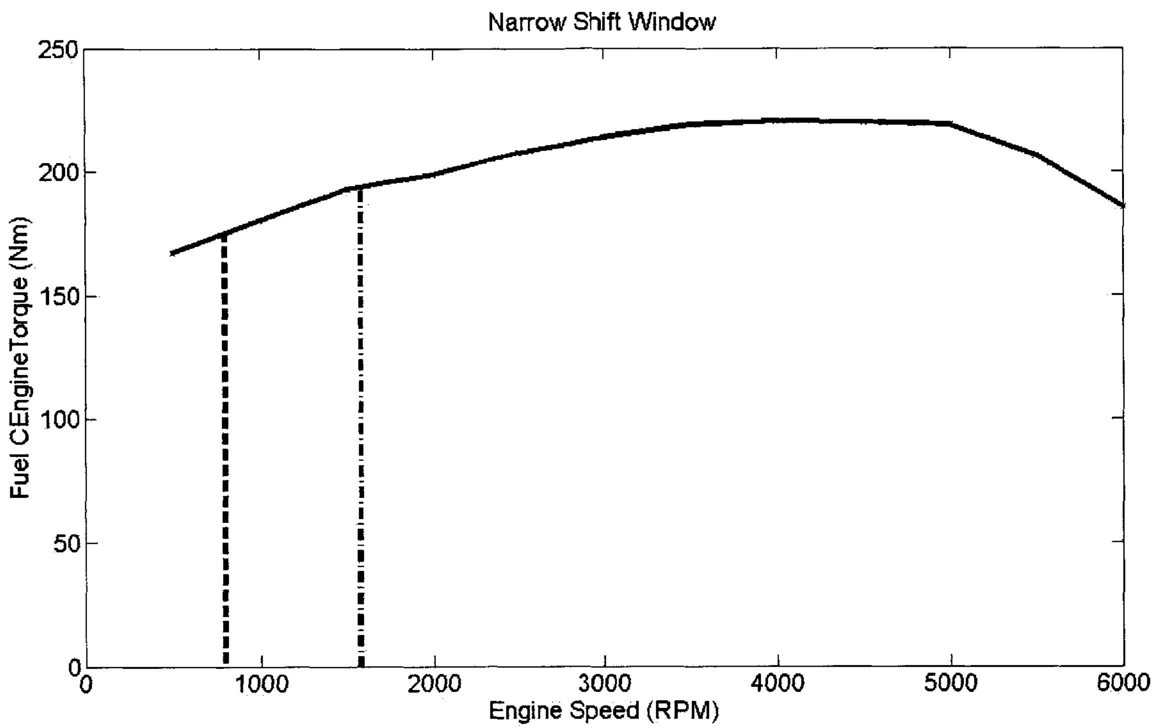


**Figure 75: Narrow shift window to be used in the simulations**

# 5. Hybrid Electric Vehicles

## 5.1. Introduction

In its most general conceptual definition, a hybrid powertrain is the combination of a power source using some fuel and an energy storage medium. The power source could be any form of engine or even a fuel cell and the energy storage medium could be electrochemical (battery, ultracapacitor), mechanical (advanced flywheels) or even hydraulic. The additional flexibility of having an energy storage medium provides much more control over powertrain operation. Fuel consumption but also criteria pollutant emissions are thus lower. The additional components and complexity added do of course come at a cost premium.

In the context of this thesis, spark ignition engines in hybrid powertrains using electrochemical means as the storage medium will be examined. It was explained how in the near to mid-term future, it is very difficult for anything else to significantly penetrate the market other than powertrains employing ICE engines to use hydrocarbon fuels. A hybrid powertrain is probably the most efficient way of doing exactly that. Diesel hybrids were not considered in this study as the additional benefit they provide compared with conventional diesels is relatively small, and their cost premium is significantly higher than that of SI hybrids. The benefit is smaller because one of the main ways in which a hybrid improves fuel economy is that it forces the engine to operate at higher loads where it more efficient. The difference between partial and full load efficiency is however much higher in spark ignition engines than in diesels due to the effect of throttling as explained in chapter three. The relative benefit from hybridizing diesels is therefore much smaller. Electrochemical energy storage will be the only medium considered as the other forms of energy storage mentioned above are not nearly at the same level of maturity.

## 5.2. Background

### 5.2.1. Ways in which an HEV Saves Energy-What is the Potential?

Referring back to the energy flow figures [11, 12] in the first chapter, the ways in which a hybrid improves fuel consumption are:

1. **Eliminating Idling:** When a hybrid vehicle comes to a stop (e.g. at the streetlight) the engine is switched off rather than allowed to idle. The reason a hybrid can achieve that, is the fact that its large motor and batteries ensure an easy and smooth engine start. Actually, what usually happens is that the vehicle starts moving using the electric system only and the engine kicks in only above a vehicle speed. The required electric system size and thus cost to eliminate idling is small and the benefit is quite significant-17% in urban driving according to Figure 11.All hybrids currently on the market can achieve perfect elimination of idling.
2. **Regenerative braking:** A hybrid can use its motor as a generator to recapture some of the vehicle's kinetic energy which is usually wasted as heat in friction breaking. Perfect regeneration would recover about 6% of the fuel energy for urban driving according to Figure 11. To convert this energy to fuel energy, it needs to be divided by the efficiency of the engine and the transmission, i.e.

6%/0.19/0.94=33.6% of the fuel. After this energy has been captured however it needs to be converted to electricity (motor/generator), stored in the battery (charging), released from the battery (discharging) and reconverted to mechanical work at the motor. This number needs to be multiplied therefore by the square of the efficiency of the motor and the battery as well as possibly also the square of the efficiency of the gearbox if the motor is coupled to one. As a result, assuming typical current efficiencies this would be $33.6\% \times 0.88^2 \times 0.86^2 \times 0.93^2 = 16.6\%$ .For practical reasons however, friction braking can never be totally eliminated so at the very best current hybrids recover only a part of this energy.

3. **Reducing the size of the engine.** A hybrid can use its electric system to assist the engine during high load demands such as hard accelerations. A smaller engine can thus be used for the same level of vehicle performance. The engine is consequently operating at higher average loads increasing its efficiency. Additionally, in some hybrids the valve timing of the engine is modified to apply an Atkinson cycle increasing efficiency at a power density penalty. The degree of engine size reduction depends on the relative size of the electric system.
4. **Actively Determining Engine Operating Points.** All hybrids can, in principle, use the electric system at lower loads, so that the engine only operates at very high loads further increasing its average efficiency. In practice not all current hybrids on the market have this capability. All power-split architecture hybrids do so like the Toyota as well as the Ford, and Mercury hybrids. Hybrids using a parallel only architecture currently on the market do not have this capability with the exception to a limited extent of the 2006 Honda Civic. This capability could theoretically provide up to a 20% fuel consumption benefit compared with an equivalent hybrid that doesn't have it as will be seen in the results chapter. Currently, this benefit is less.

The size of the electric system required in order to eliminate idling is relatively small compared with the size of the engine. In order to achieve substantial regenerative braking and engine efficiency improvements however, the maximum power of the electric system has to be at least around 10% of the engine's power. Hybrids with small electric systems that mainly offer fuel consumption benefits through idling elimination are loosely classified as *mild hybrids.* Hybrid vehicles with electric systems large enough to additionally offer significant benefits from engine efficiency improvement and regenerative braking are classified as *full hybrids.* In this thesis only full hybrids will be evaluated.

### 5.2.2. Hybrid Architectures.

From a conceptual point of view, there are three ways in which the components that make up a hybrid powertrain can be arranged:

3. A Series Architecture.
4. A Parallel Architecture
5. A Combination of Series and Parallel

### *Series Architecture*

In a series architecture, the power source (engine/fuel cell) is not directly coupled with the wheels but only through the electric system. In the context of an engine hybrid, this means that all of the energy generated by the engine needs to be converted to electricity. The electricity is subsequently either stored in the battery, or used in the motor to propel the vehicle after going through the power control unit again. There is no direct coupling of the power source with the wheels. During regeneration, the power can also flow in the opposite direction, from the wheel converted to electricity at the motor to the battery to be stored. A typical series architecture including the directions of power flow is presented in Figure 77.

The advantage of a series architecture stems from the fact that all the energy goes through the storage medium. The storage medium can subsequently act as a buffer, essentially decoupling the power demand at the wheel from the power demand at the power source. The power source can as a result operate wherever the system designer chooses. It can be optimized for very low fuel consumption, low emissions or to minimize load following.

The disadvantages of a series architecture are:

- All the energy produced by the engine has to undergo many transformations before it reaches the wheel. Every transformation inflicts losses. The gain from optimizing engine operation versus the loss from not being able to directly power the wheels is only favorable in very urban driving patterns. On the highway, the engine is better off powering the wheels directly.
- The size of the energy storage required to buffer all the energy is very large. The size of the power source could be reduced compared with a non-hybrid. However, energy storage is generally much more expensive than engines.

Due to these disadvantages, none of the light duty ICE hybrids are using a purely series architecture. A series architecture is best suited for fuel cell vehicles since the power produced by the fuel cell is already electric and since it is vital that the fuel cell does as little load following as possible. They might also be suited for certain heavy duty applications such as diesel hybrid buses being used mostly for urban driving if the goal is to minimize emissions.

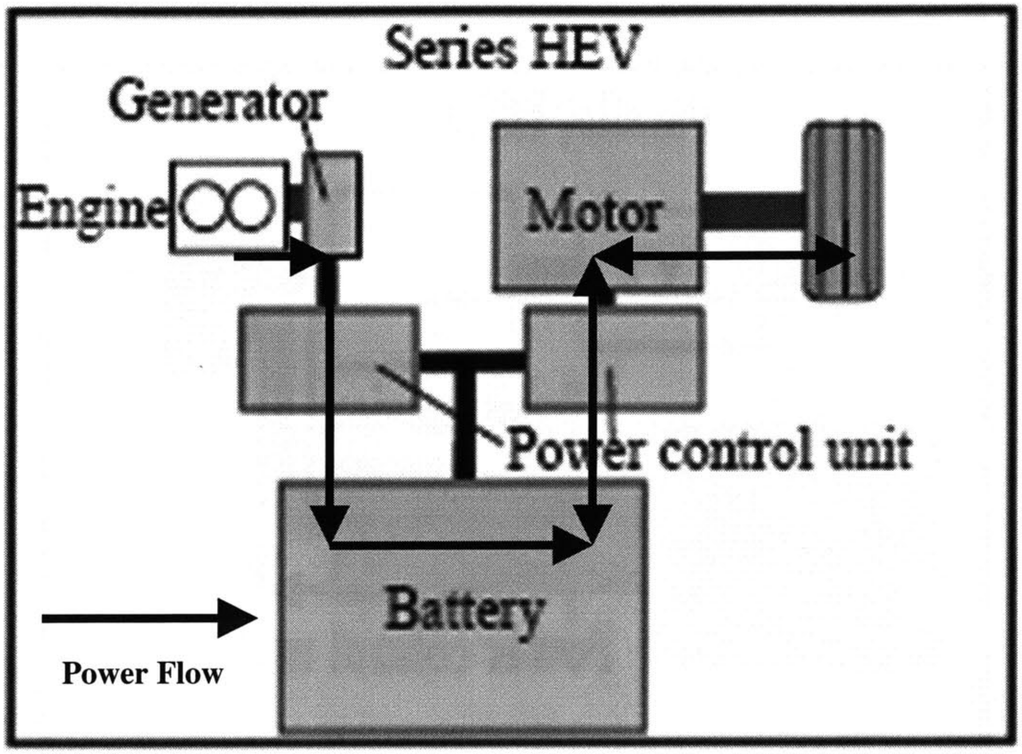


**Figure 76: Power Flow in a Series Hybrid Electric Vehicle**

### *Parallel Architecture*

In a parallel architecture, either the engine or the battery and motor or both can directly propel the vehicle. The engine can additionally recharge the battery through the motor/generator or they can be recharged from the wheels when regenerating. The power flows in a parallel hybrid architecture are presented in Figure 77.

The advantage of a parallel hybrid architecture is that the energy doesn't have to undergo as many conversion steps to get to the wheel. These losses are therefore less. This is particularly important during driving conditions when the engine is operating efficiently such as highway driving. The disadvantage of a parallel architecture is that there is usually less control over engine operating points.

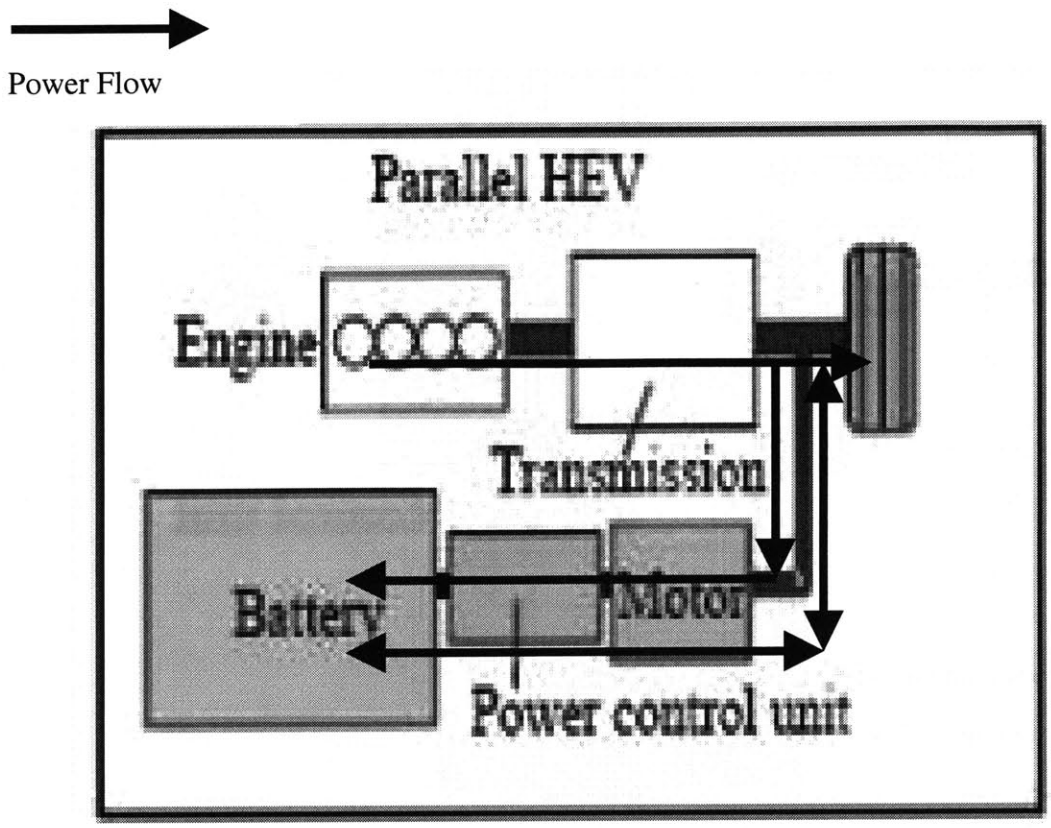


**Figure 77: Power Flow in a Parallel Hybrid Electric Vehicle**

An application of a purely parallel architecture is the Honda hybrid architecture. The powertrain components to implement this architecture can be seen in Figure 78, 79 and 80. The engine and the DC brushless motor (called IMA for Integrated Motor Assist in the figure) are connected on the same shaft and very tightly packed together. The engine and motor are mechanically connected to the transmission through a clutch. The transmission can be either manual, automatic or CVT and is connected to the final drive and wheel. The layout of the powertrain on the vehicle is shown in Figure 79. A photograph of the powertrain is presented in Figure 80. This system is well documented in [53, 54, 55, 56, 57, 58, 59] among others.

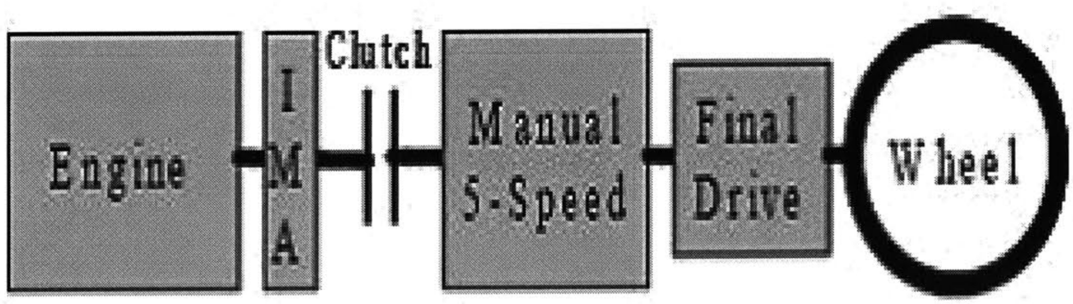


**Figure 78: Honda Hybrid Architecture**

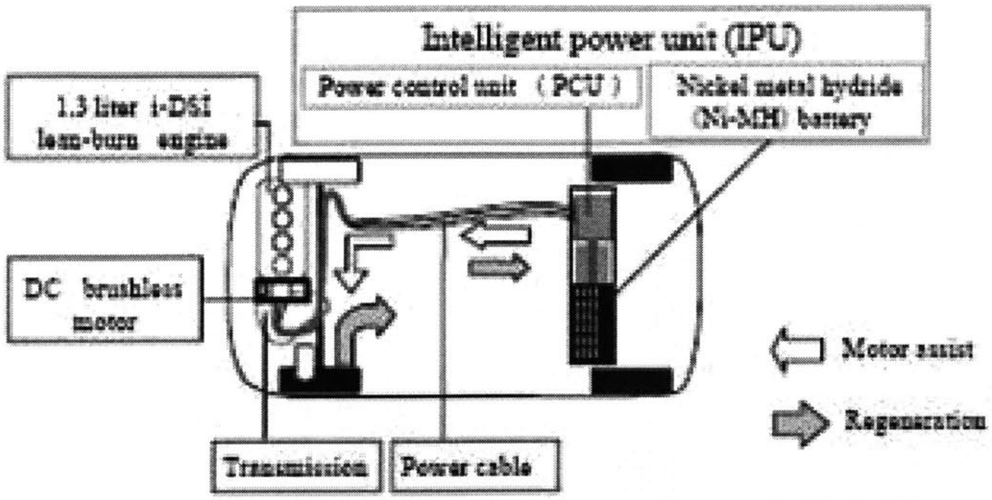


**Figure 79: Honda Hybrid System Layout**

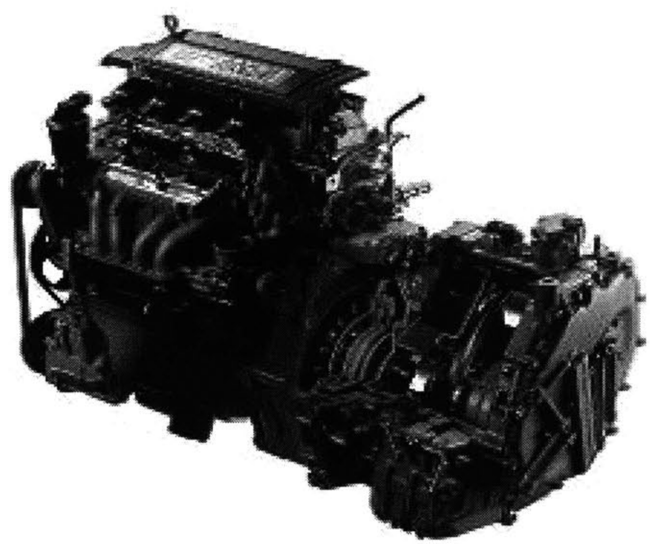

**Figure 80: Photograph of the Honda Hybrid Powertrain.**

### *A Hybrid Architecture Combining Series and Parallel-The Power Split Architecture*

A hybrid architecture can combine series and parallel power pathways to benefit from the advantages of each. This is implemented in the power-split architecture used by Toyota but also Ford and Mercury. A conceptual schematic of this architecture is presented in Figure 81. The power from the engine is split at the transmission into two parts. The first (72%) is used directly to power the wheels. Additionally, a motor coupled on the transmission's output shaft can provide additional torque if needed. In terms of this 72% of the engine's power this is a parallel architecture. The second part (28%) of the engine's power is converted to electricity at the generator and either stored in the battery or used immediately to power the motor. This branch is a series hybrid branch. Regenerative mechanical power from the wheel can be used either in the "motor" or the "generator" to generate electricity. Alternatively, the generator could be used as a motor powered from the battery. These architectures are sometimes referred to as *"power split architectures"*

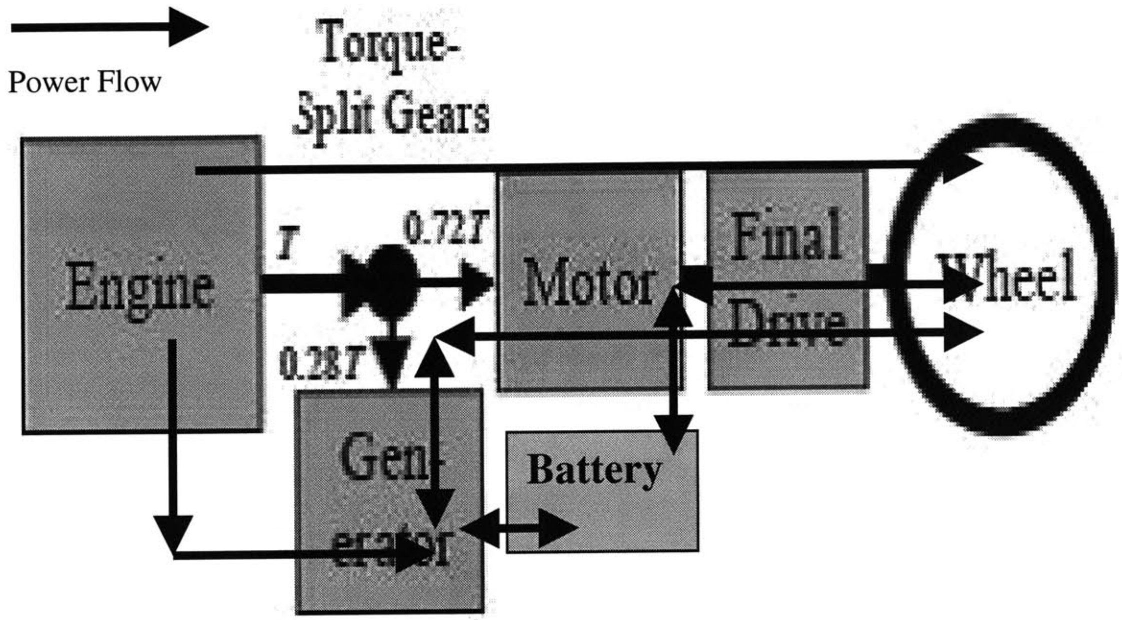


**Figure 81: Conceptual Schematic of a Power-Split Hybrid Architecture.**

To understand the operation of this system its actual mechanical structure needs to be examined. The Toyota Prius transmission/power slit device is presented in Figure 82. It is basically a planetary gearbox in which the engine is connected to the planet gearset, the generator is connected to the sun gear and the motor is connected to the outside ring gear. The motor is additionally coupled to the wheels through the final drive. The speed of the motor is thus directly proportional to vehicle speed. Due to the set up of the planetary gearbox, a linear relationship exists between the speed of the generator and the speed of the engine. The speed of the generator however is controlled by the power electronics. Thus by setting the speed of the generator the engine speed can effectively be determined for a given vehicle speed. This phenomenon is depicted in Figure 83. The speed of the engine, the motor (vehicle) and the generator need to be kept on a straight line in this plot. By choosing generator speed, engine speed is thus determined. By choosing the right generator speed, the engine can be shut off and the vehicle is in all-electric mode. The generator can act as a generator or motor on either side of the axis (positive and negative speed) by changing its connections. Neglecting friction, the relationships for component torques are equations (5.1), (5.2) and (5.3).

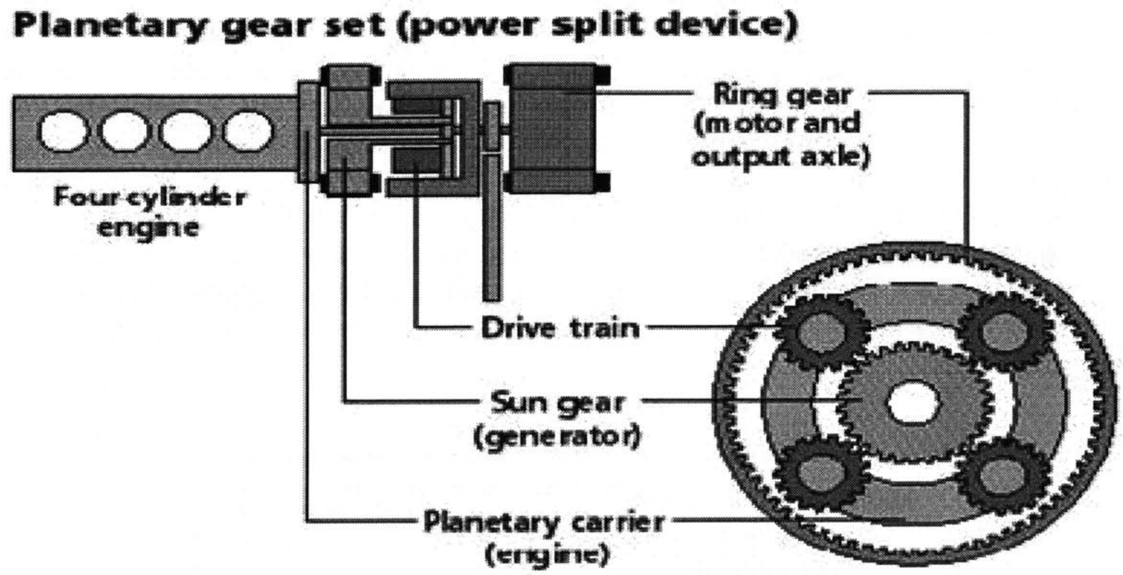


**Figure 82: The Power Split Device of the Toyota Prius**

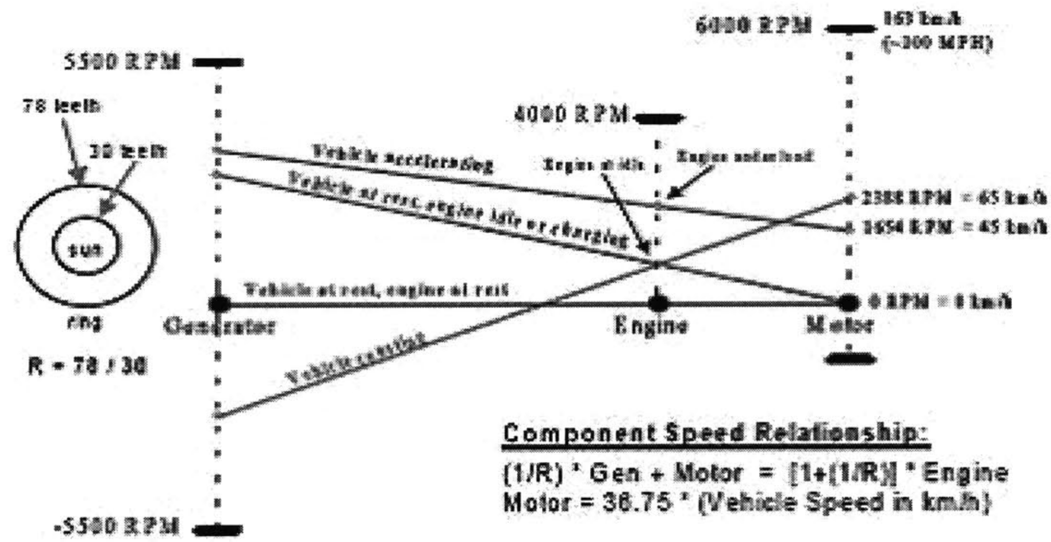


**Figure 83: Relationship of Component Speeds for the Toyota Prius.**

$T_{GEN} = T_{ENG} / (78/30 + 1)$ (5.1)

$T_{GEN}$ – Generator torque

$T_{RING}$ – Torque at ring gear (not including torque generated by motor)

$T_{RING} = 78/30 * T_{GEN}$ (5.2)

$(T_{RING}+T_{MOTOR})*i_{fd}=T_w$ (5.3)

$i_{fd}$-final drive ratio

The engine speed and torque in a power split architecture can thus be set almost independently of wheel speed and torque. This architecture acts therefore as an electronically controlled CVT. The optimum engine operation line to minimize fuel consumption for the engine in the Toyota Prius can be seen in Figure 84. The actual engine operating points over a portion of the FTP driving cycle are very close to this line as seen in Figure 85. The deviation from the optimum operating line is done on purpose as explained in [59]. The control logic is optimized for the whole system and not just the engine. Sometimes, especially at low engine rpm and high loads, a slight deviation from the engine optimum reduces the electric losses and improves the total efficiency of the powertrain.

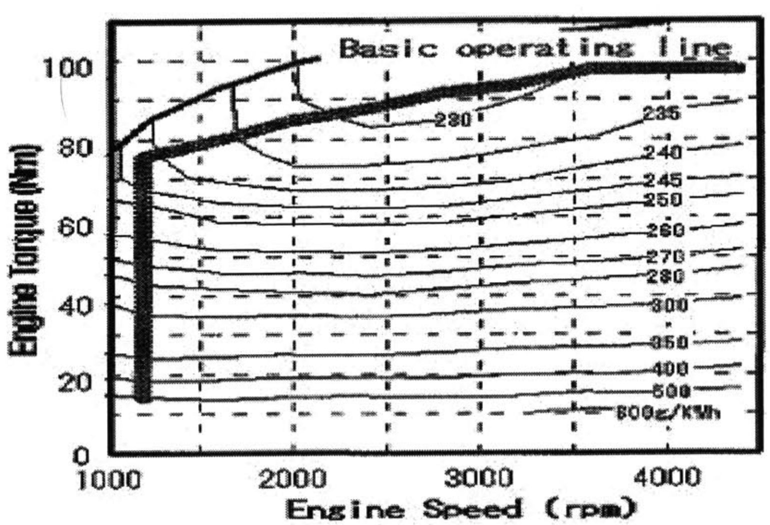


**Figure 84: Optimum Operating Line for the Toyota Prius Engine**

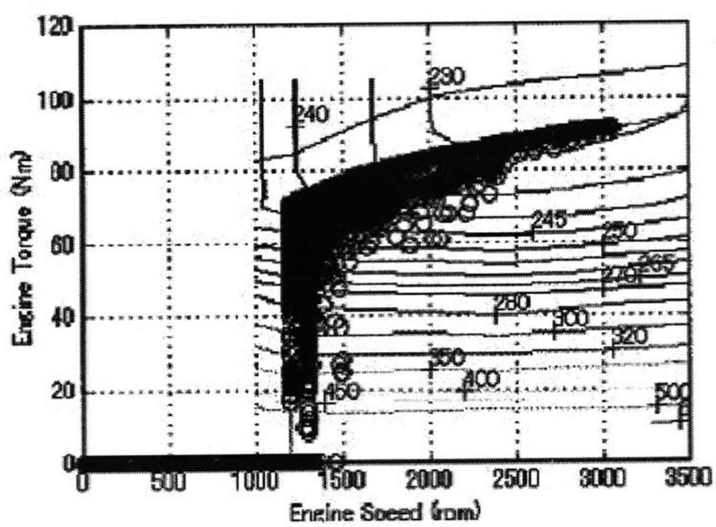


**Figure 85: Actual Operating Points of the Toyota Prius Engine.**

The Toyota power split architecture is documented in [60, 61, 62, 63, 64 ] among others.

Comparing the Power-Split architecture versus the Honda Parallel Architecture some of the conclusions are:

- The power-split architecture achieves very effective optimization of engine operation.
- The Honda architecture does so only in the models equipped with a CVT. Those models are however penalized by the low efficiency of the CVT. The Civic hybrid with a manual transmission gets thus better fuel economy than the CVT version on the highway while the CVT version is better in the city. In the new Civic hybrid (2006) additionally, the vehicle goes all electric below a certain vehicle which is an effort to further actively optimize engine operation.
- The electric-CVT transmission of power split architectures might not have the high mechanical losses of a CVT; it does however have electrical losses for the part of the engine's energy that goes through the series branch.
- Regeneration in the Honda architecture is significantly worse because the engine cannot be decoupled from the motor. The engine needs to be rotated as well during regeneration dissipating a lot of useful work. Honda has actually developed an engine valve deactivation system to reduce this loss. This however, is not a disadvantage of parallel architectures in general it is just a characteristic of the Honda architecture. Recently IVECO, a FIAT group company presented hybrid prototypes using a parallel architecture similar to the Honda one but with an additional clutch between the motor and the engine. Such a system would be equivalent with the power-split architecture in terms of regeneration. The IVECO system may be seen in Figure 86.

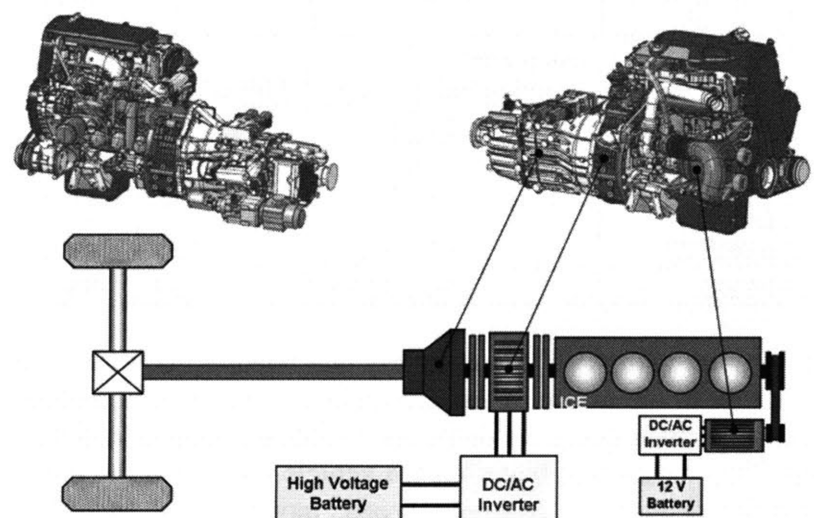


**Figure 86: The IVECO Hybrid System. Source: [41]**

In order to compare the two architectures one can look at the Toyota Prius and the 2006 Honda Civic Hybrid. The two vehicles are almost equivalent in terms of weight and performance; 1304 kg and 10s 0-100 km/h for the Civic, 1311 kg and 10.4 s 0-100km/h for the Prius. The Civic has a fuel consumption of 4.8 liters/100km in the city and 4.6 liters/100km on the highway. The Prius gets 3.9 liters/100km in the city and 4.6 liters/100km on the highway. The Prius thus gets no fuel consumption benefit on the highway, a 19% benefit in the city and an 8.5% benefit in combined fuel consumption. To get this benefit however, the Prius is using 2 larger motors instead of one, about double the battery size and a more complex system. In terms of cost to benefit, complexity as well as packaging the Civic looks better. As it will be seen, the fuel consumption advantage of the Prius can be attributed to its better regeneration and active engine optimization abilities.

## 5.3. ADVISOR Validation

ADVISOR as already mentioned uses a backward facing calculation. This methodology is very accurate for modeling non hybrid powertrains as well as parallel only or series only hybrids. The simulations of current non hybrid vehicles modeled in this study resulted in fuel economies within a 5% margin of the measured numbers as will be seen in the results chapter.

A power split architecture cannot however be modeled accurately by a backward facing simulation as ADVISOR. In ADVISOR there is a model of the first generation Prius built in the software with very accurate component maps as they were the product of extensive measurements conducted by the National Renewable Energy Laboratory (NREL).The simulations results, especially for very urban drive cycles as the U.S FTP or the Japanese 10-15 mode, are however significantly different than measured numbers. A comparison of ADVISOR simulation results for the first generation Prius with measured fuel consumption numbers as published in [59] and [66] is presented in Table 11.

**Table 11 : Measured and Simulation Fuel Consumption Results for the 1st generation Toyota Prius**

| | PRIUS I | | | |
|---|---|---|---|---|
| | measured (unadjusted) | Units | ADVISOR | error |
| 10-15 M | 31 | km/l | 15 | 52% |
| US CITY (FTP) | 58 | mpg | 43.9 | 24% |
| US HW (HWFET) | 58 | mpg | 60.7 | -5% |
| NEDC | 120 | gC/km | 127 | -6% |

In a communication with AVL, the company that releases ADVISOR, this difference was attributed exactly to the nature of the simulation. A power-split architecture is based on a forward facing process using the driver as a feedback loop to determine component operating points. This means that based on the vehicle speed and the position of the gas pedal, the controller determines generator speed, motor torque in order to determine engine speed and torque. The resulting vehicle speed and acceleration is observed by the driver who, in turn, readjusts the position of the gas pedal. In fact, all vehicles even non hybrid ones operate in this manner in real world driving. For non-hybrids powertrains however, the use of a backward facing calculation is accurate enough because for a given vehicle speed acceleration, and gear speed in the gearbox there is only one engine speed and torque. With a reasonably accurate gear shifting logic, the results are thus accurate. Even with a mechanical CVT, the engine speed and torque is unique, the optimum in terms of fuel consumption for that engine power output. CVT's can thus be modeled accurately. Parallel hybrids can also be modeled accurately as the only thing that is different is determining a strategy on when should the motor assist the engine and by how much. If this strategy is well defined, the amount of the power requested at the wheel at a given time instant that comes from the engine can be calculated. Determining the engine operating point still has a unique solution, the minimum fuel consumption for that power.

In a power-split architecture however, because of the series pathway, the optimum solution isn't always the minimum engine fuel consumption point. Minimizing the losses from the electric pathway needs to be accounted for. The control logic in power-split hybrids is based on a system optimizing basis. As this logic is not known, a backward facing calculation cannot model power split architectures accurately.

In order to validate that ADVISOR can model parallel only hybrid architectures; a model of the 2005 Honda Civic hybrid with a manual transmission was set up. The results, which show very good agreement with published numbers, are presented in chapter six.

## 5.4. Choosing an Architecture

Looking at the evolution of hybrid architectures in the future, the following possible pathways can be postulated:

- *A parallel only architecture without active engine optimization.* This could be viewed as the evolution of the Honda architecture using a manual or automatic transmission. A possible modification of decoupling the engine from the motor to increase regeneration could realistically be anticipated. Even Honda is already considering such a modification which could be made possible with the introduction of an additional clutch or by integrating the motor with the transmission [67].
- *A parallel only architecture using a CVT.* This would be the evolution of the Honda architectures using a CVT. With such an architecture, using the motor to actively optimize engine operation isn't required. The CVT takes care of engine optimization though at a high transmission efficiency cost. The success of this pathway depends strongly on the improvement of CVT efficiencies.
- *A Power Split Architecture.* This would be an evolution of the Toyota architecture.
- *An automatic or manual/automatic transmission that actively optimizes engine operation.* This hasn't been implemented yet. There are several technical hurdles to make this idea production acceptable. It could however be an important enabler for hybrid powertrains.

To understand how important improving CVT efficiency is, it should be mentioned that the efficiency gain from optimizing the engine using current CVT's is largely lost due to the low transmission efficiency. A current example is the 2005 Honda Civic Hybrid which comes both with a manual and a CVT transmission. The manual version gets 5.2liters/100km over the FTP and 4.6liters/100km on the highway. The CVT version gets 5.0liters/100km over the FTP, 4.9liters/100km on the highway. For urban driving the

benefit from optimizing the engine is more than the penalty due to transmission efficiency, the CVT version is thus better. On the highway the effect is the opposite.
There are however as already mentioned in chapter four, certain novel CVT technologies that promise to raise the transmission efficiency barrier. Toroidal CVT gearboxes presented in chapter four, could get transmission efficiencies at the level of current automatics [46].Examples of how this technology can be combined with a hybrid architecture can be found in [68,69].A layout of the hybrid powertrain using a CVT (called IVT in this figure) adapted from [68] can be found in Figure 87.

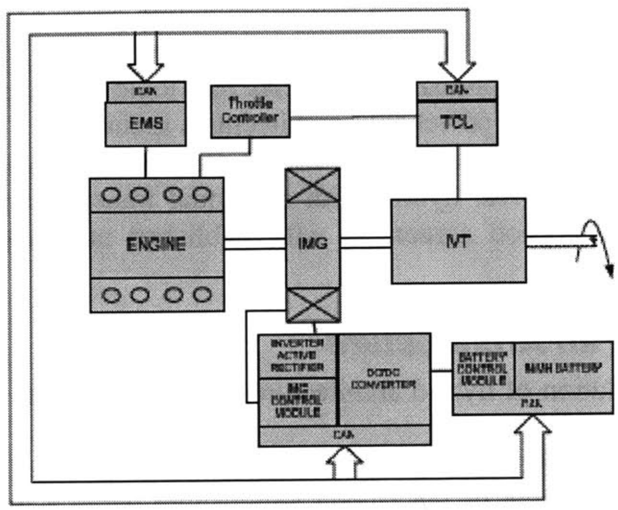


**Figure 87: A CVT gearbox in a parallel hybrid architecture. Adapted from [68]**

Another interesting opportunity for hybrids would be to develop a parallel architecture with an automatic or manual/automatic transmission that can actively optimize engine operation. To understand how such a transmission would work, the way it is modeled in ADVISOR is presented in Figure 88. Either an engine speed or a fraction of the engine's maximum torque across speeds may be set, below which the powertrain goes all electric. This is, of course provided that the battery state of charge (SOC) is above a predetermined level. If the batteries are empty, they are charged from the engine. In the case of the future gasoline engine maps described in chapter three, limiting engine operation to only high speeds doesn't make sense as engine friction and thus bsfc goes up with speed. Specifying an engine off-torque fraction is much more meaningful.

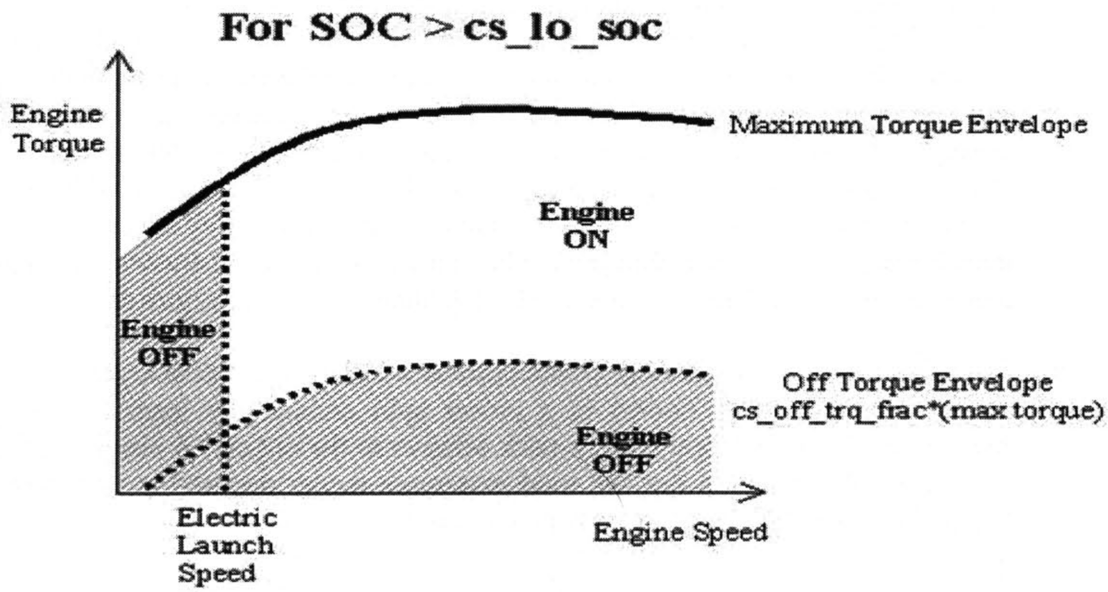


**Figure 88: Modeling an Actively Engine Optimizing Hybrid in ADVISOR**

Actually designing a manual or automatic transmission that can implement the control strategy explained above would be a challenge. First of all, it is critical that the engine can be isolated from the motor. Second, and possibly most difficult of all, the powertrain must be able to smoothly shift from using the engine to using the motor to using both. Additionally this shift must occur very fast for aggressive driving. Perhaps there are some synergies here with the dual clutch transmission concept which can achieve such smooth shifting. Finally, if the engine stays off for long, there might be some issues with emissions if the catalyst cools off. However, there some experts in the industry that believe that the problems could be solved.

A parallel only architecture with either an efficient CVT or with an automatic that can actively optimize engine operation would potentially offer power-split level fuel economy at significantly lower cost. A manual/automatic transmission with optimizing capability would most probably be slightly better than a power split as there would be no unnecessary electric losses from the series pathway. A manual/automatic with optimizing capability or an efficient CVT would probably be cheaper because:

- *There is one less motor:* In order to implement the electric CVT concept two electric motors are needed. A parallel architecture only needs one.
- *The motor in a power split architecture is oversized.* In a power split architecture because of the series pathway, the motor must be sized to meet the power output of the battery plus the output of the generator. In a parallel the motor is sized to meet the power output of the battery alone. Furthermore, from equations (5.1-5.3) and Figure 83 it can be seen that the motor and generator speed and torque affect the gear ratio from the engine to the wheels. The motor and generator need to be sized therefore taking the required gear ratios in mind too. This leads to even higher motor sizes .For example, in the second generation Prius if the motor was sized to meet the power of the series branch (28% x (engine power)=16kW) plus

the maximum battery power (21kW), it would have a maximum power of about 37kW. In fact, its maximum power is 50kW. Actually one of the main changes from the first to the second generation Prius was that the maximum power of the motor was increased from 33kW to 50kW without increasing the power of the battery and while the engine power only increased by 4kW. Additionally, the generator power was increased from 15 to 25kW. The generator additionally became faster. As a result, 0-100km/h acceleration improved by about 20%. This improvement is due to the changes in the effective gear ratios between the engine, motor and wheel and not as much to the 4kW increase in engine power.

- *The battery size is in a parallel architecture is smaller.* It could be argued that the existence of the series branch in a power split architecture leads the system designer to choose a larger battery pack relative to the engine. A simple analysis will show why. Assume that the total maximum power required at the wheel is $P_{w,max}$. For a parallel architecture this is simply:

$$P_{w,\,max} = (P_{eng,\,max} * \eta_t + P_{battery,\,max} * \eta_{mot} * \eta_t \quad (5.4)$$

Where $P_{max,eng}$ is the maximum engine power, $P_{battery,max}$ the maximum battery power and $\eta_{mot}$,$\eta_t$ the efficiencies of the motor and the transmission respectively
For a power split architecture, because of the series pathway:

$$P_{w,max} = \sigma * P_{eng,max} * \eta_t^2 * \eta_{mot} * \eta_{gen} + (1-\sigma) * P_{eng,max} * \eta_t + P_{battery,max} * \eta_{mot} * \eta_t \quad (5.5)$$

Where $\sigma$ is the part of the engine's power that goes through the series pathway and $\eta_{gen}$ is the efficiency of the generator.

Comparing equations (5.4) and (5.5) it is obvious that because of the higher losses of the series branch to get the same $P_{w,max}$, with the same $P_{eng,max}$, a larger battery pack is required for a power split architecture compared with a parallel only.

On the other hand, in order to get the same level of fuel consumption with a parallel only architecture as with the power split, a very sophisticated transmission would be needed. There are people in the industry that believe that a sophisticated transmission might cost more than the over sizing of the main motor and adding an extra one. The cost of the batteries seems however to be generally the most important component of a hybrid's cost premium. Indeed, in an extensive cost study conducted by [70] it can be seen that the cost of one motor in a hybrid is approximately equal to the gain from downsizing the engine. The variable cost of a hybrid comes down to basically the cost of the batteries. Since power split architectures generally lead to higher relative sizes of the electric system and bigger battery packs they should generally expected to be more expensive.

For this study, it was assumed that future hybrids use parallel architectures. Both CVT's and manual/automatic transmissions were modeled with or without active engine optimization. As already explained, power-split architectures couldn't be modeled accurately due to limitations of the simulation. A parallel architecture with a manual transmission and optimization capability should however mimic the fuel economy of a power split. Intuitively, this configuration should be slightly better than the power split on the highway, because the series branch is bypassed and slightly worse in the city because it is probably worse than the power split in optimizing engine operation.

To validate this claim, a numerical experiment was conducted: As explained, an accurate model of the 2005 Civic Hybrid with a manual transmission was developed in ADVISOR. In that model, the engine off torque fraction was set to 40%. This fraction was found to be close to optimum for fuel economy as will be explained. Additionally regeneration characteristics were set to approximately the same as the second generation Prius. The resulting fuel economy was 56 mpg (city-adjusted) and 54mpg (highway-adjusted). The fuel economy of the second generation Prius with almost the same weight and performance is 60 mpg (city-adjusted) and 51mpg (highway-adjusted). This matches well with the predicted results.

For the purposes of this study it will be assumed that the fuel consumption of future hybrids can be estimated from simulations of a parallel architecture using a manual/automatic transmission that offers engine optimization. This can be either viewed as assuming that such powertrains will be developed and enter the fleet or as a data point close to an evolved power split architecture.

## *5.5. Components-How will they evolve?*

### 5.5.1. Engine

The same assumptions that were used for the non hybrid powertrain engines will be used for the hybrid ones. Currently, some hybrid engines use an Atkinson cycle for improved engine efficiency at a power density cost. Future engines were however already assumed to be significantly more efficient and downsized than current ones, so the effect of an Atkinson cycle was considered small.

### 5.5.2. Motor

The standard motor choice for modern hybrid and electric vehicles is DC permanent magnet motors. Power electronics can easily control speed and torque for this type of motors to produce an almost ideal maximum torque versus speed curve. The maximum torque versus speed curve as well as the efficiency map including controller losses are presented for the DC motor in the first generation Prius in Figure 89.

There are two questions that need to be answered in terms of the evolution of motors.

- What will their gravimetric power density in kW/kg be?
- How will their efficiency improve?

The Freedomcar project goals were used to answer these two questions.

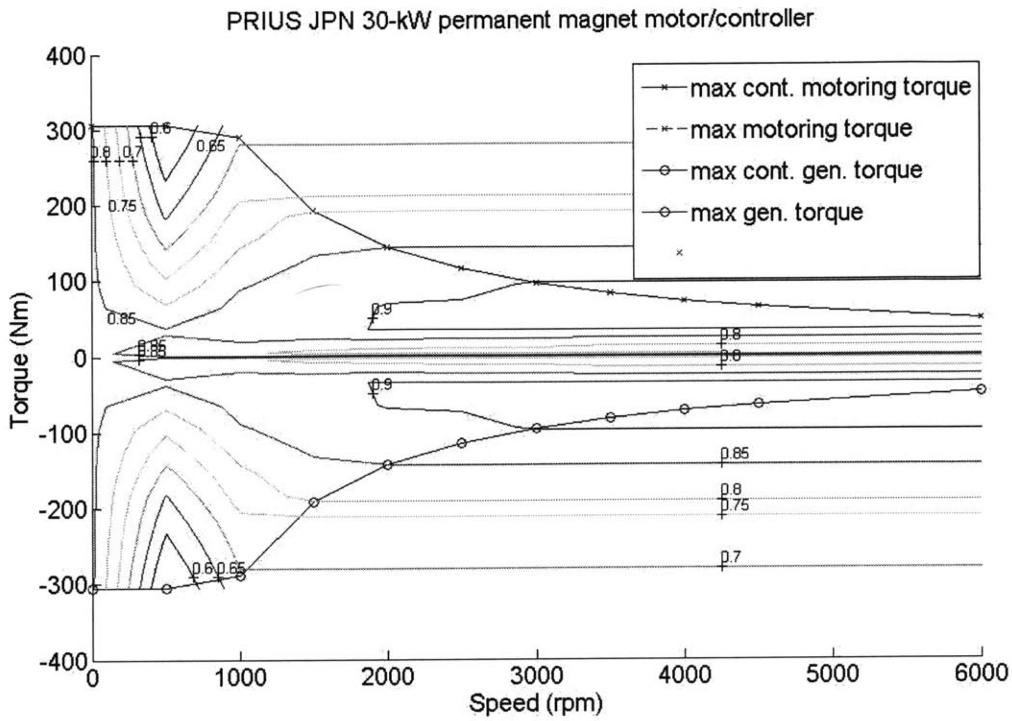


**Figure 89: Motor Map for the 1st generation Toyota Prius**

In terms of power density, the FreedomCar partnership [23] mentions a current (2003) status of 0.95kW/kg for the integrated motor and controller. The 2010 goal is 1.2kW/kg and for 2015 it is 1.3kW/kg. This is equivalent to a linear 4% per year increase between 2003 and 2010, and a 2% between 2010 and 2015. The improvement rate is decreased to about half between 2010 and 2015. Using a simple mindframe as the weight of the motor is not that important for the final results (about 10-20kg), the annual rate of improvement was decreased by another factor of two to 1% per year between 2015 and 2030, leading to a 2030 power density of 1.55kW/kg.

In terms of efficiency, the long term FreedomCar goals are 0.95 for 10-100% of the maximum speed at 20% rated torque and 0.82 for 0-10% of the maximum speed. Again using a simple mind frame, those were the motor efficiencies used independently of torque. An additional constant 0.98 was assumed for the controller again in accordance with FreedomCar long term goals.

### 5.5.3. Electrochemical Energy Storage

The energy storage medium for a hybrid vehicle should have the following characteristics:

- It should provide enough energy storage at a reasonable total added weight. The figure of merit to access this capability is the gravimetric energy density in kWh/kg. The total energy storage capacity required for a midsized hybrid vehicle depends on the amount of electric assist during accelerations and regeneration during decelerations. As will be explained, for a midsized full hybrid vehicle this it is in the order of magnitude of 100Wh.

- The energy storage medium should provide enough power for the electric system. This depends on the relative size of the motor, but is in the order of 10-40kW for a full hybrid midsized vehicle.
- The energy storage medium should exhibit high durability to many charging-discharging cycles.
- Additional requirements are high safety, low volume and low cost.

## Current Status of Hybrid Battery Technology

Currently, the energy storage medium of choice for full hybrids on the market is nickel metal hydride (NiMH) batteries. These batteries offer relatively high energy and power density but most of all they offer excellent durability. Before presenting figures of merit for these batteries it is important to understand some basic battery characteristics:

- The state of charge (SOC) of a battery is the amount of charge left in the battery over the total charge capacity of the battery.
- The depth of discharge (DOD) is the initial minus the final SOC.
- A single *battery cell* consists of an anode, a cathode, an electrolyte to separate them and a simple case. For applications that require more energy or power than that offered by a single cell, cells are connected into *battery modules*. The weight of a module is more than the sum of the weight of the cells as there are extra connectors and casing. Modules are connected into *battery packs* which are complete units including all the necessary auxiliary hardware such as control electronics and possibly cooling equipment.
- The internal resistance for most types of batteries varies significantly with the amount of current being drawn, the battery state of charge and temperature. Therefore so does the power and energy that can be extracted.
- Especially for NiMH batteries, the maximum power that can be drawn when charging the battery is significantly more than that when discharging the battery.

The maximum power for the battery pack of the first generation Toyota Prius is presented versus the net energy that has been removed or added and temperature in Figure 90.The figure has been adapted from [62] which is a paper presenting results from extensive testing of the Prius battery pack by NREL. Notice how the power capability varies with net energy removed corresponding to SOC, temperature as well as whether the pack is being charged or discharged. The peak charging power is slightly less than 30kW. The peak discharging power is about 21kW. The total weight of this pack was according to the same paper 53.5kg. The total energy of the pack is 1.6 kWh. These numbers lead to a power density of 0.4kW/kg for discharging, 0.5kW/kg for charging and an energy density of 29.9Wh/kg. In the 2004 SAE paper by Toyota on the Prius [59] the discharge power of 21kW is quoted as battery power for both the first and second generation Prius, but the values given for power density are 0.88kW/kg and 1.25kW/kg respectively. These numbers are probably based on a cell or module basis excluding some of the battery casing and hardware weight. The second generation Prius pack weighs 45kg [71] and offers about the same power and energy. This brings the total power and energy densities to 0.6kW/kg (charge) 0.46kW/kg discharge and 35.5Wh/kg. Even for the first generation Prius however, the added weight of the batteries at 53.5kg isn't that important to affect fuel economy significantly. It represents a mere 3.8% increase in total vehicle weight.

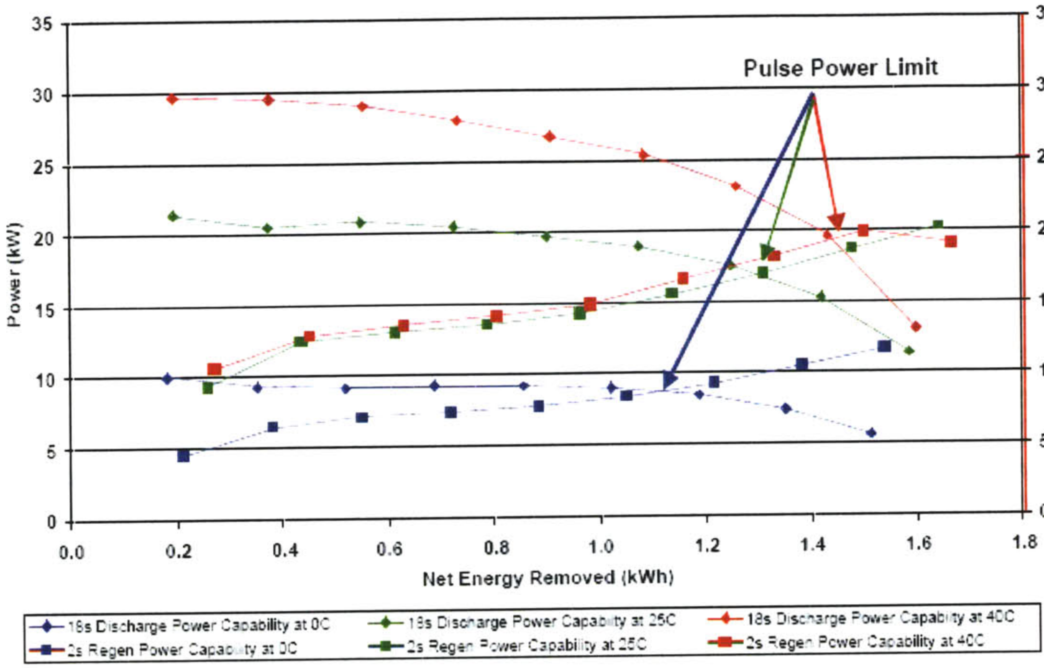


**Figure 90: Power Capability for the Battery Pack of the first generation Toyota Prius versus net energy removed or added to the pack and temperature**

A very important characteristic for hybrid batteries is durability. It is so important because of the very high cost of the battery pack in case it needs replacement. However, batteries have an interesting feature that allows for significant improvement in this area. Their durability is inversely proportional to the allowed depth of discharge. The durability in number of cycles to failure versus allowed DOD for NiMH and conventional lead acid batteries like the ones used in non-hybrid vehicles can be seen in Figure 91. The number of cycles to failure in Figure 91 correspond to laboratory cycling and not real world operation. They are indicative, however, of the high durability of NiMH. This is important because hybrid batteries are sized for power not energy. The pack in the Prius for example has 1.6 kWh in order to get the 21/27kW needed. This is much more than the 100-200Wh needed. In all full hybrids on the market therefore, the DOD is limited to around 25-30% to increase durability. The results seem to be more than satisfactory. Mr. German of Honda [67] mentioned that Honda has seen less than 10 single cell (not whole battery pack) failures in all of the hybrids they have sold!

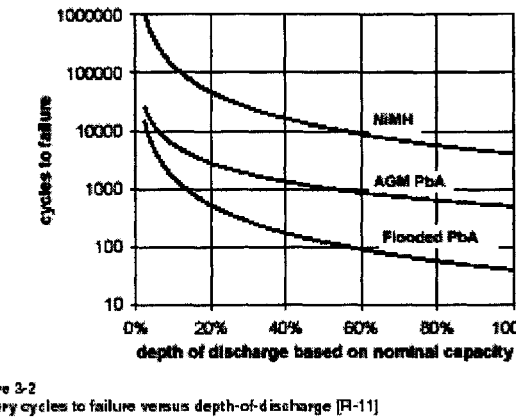


**Figure 91: Battery Durability versus depth of discharge. Source: EPRI report, [72], p3-5**

In order to obtain an estimate about the cost of a NiMH battery pack, the numbers in the EPRI reports [70, 72,] can be used. These reports put current costs for hybrid-type battery pack at around 900$/kWh plus a fixed 700$ for the additional pack hardware. This would bring the total cost of the Prius pack to about 2140$. This corresponds to about two thirds of the total cost premium of a power split architecture hybrid which is estimated at about 3000$. It is very important therefore for widespread adoption of hybrids to reduce the cost of the battery pack. All other features of current battery technology could be improved but are satisfactory at current levels.

## Future Status of Hybrid Energy Storage Technology

A summary of several different studies on the relative cost and performance of future NiMH batteries is presented in Table 12 . These are difficult to precisely compare: the time scale of comparison differs from study to study; it is not always clear whether data is a calculated on a per-cell, per-module, or per-pack basis; etc. The numbers in Table 12 seem to generally refer to costs, specific power, and specific energy on a per-module basis.

**Table 12 NiMH Future Characteristics:**

| Source | Cost | Energy | Power | Time Frame |
|---|---|---|---|---|
| **EPRI[70,72]** | $400/kWHr | 39 WHr/kg | 660 W/kg | 2010 |
| **Anderman [73]** | $650/kWHr | 44 WHr/kg | | 2010 |
| **UCD - Burke, Lipman [74,75]** | $500/kWHr | 48 WHr/kg | 600 W/kg | 2020 |

NiMH batteries have limited potential to reduce costs from present-day values. The cost of the battery is largely a function of material costs, and the price of nickel has been increasing. Moreover, the nature of NiMH manufacturing does not particularly lend itself to huge cost reductions in high-volume production. The other issue with the NiMH chemistry is that it is nearing the limit of its performance improvement.

The general sense seems to be that NiMH cost and performance improvements will reach their limit in the relatively near term. The Anderman report's cost projection is for mid-volume production; EPRI and UCD-Burke both project costs for high-volume production. Using the UCD-Burke study as a middle ground, the long-term cost of the NiMH battery could be estimated at $500/kW-hr.

One of the most competitive alternatives to NiMH technology is Lithium Ion batteries. Lithium ion technologies offer higher power and energy densities, higher efficiencies and their discharge power is not significantly lower than the charge power. Currently however they are more expensive than NiMH and there are several issues with safety and durability that prevent them from entering the market.

Various projections for future cost, energy and power densities are presented in Table 13. The general consensus is that in the longer term, the Li-ion battery has greater potential for reduced costs and a much higher performance ceiling, both in terms of energy and power than does the NiMH battery.

**Table 13: Li-Ion Future Characteristics**

| **Source** | **Cost** | **Energy** | **Power** | **Time Frame** |
|---|---|---|---|---|
| **UCD – Burke, Lippman[74,75]** | $500/kWHr | 74 WHr/kg | 900 W/kg | 2020 |
| **Anderman[73]** | $1000/kWHr | 65 WHr/kg | 2000 W/kg | 2010* *(Still Low Volume)* |
| **ANL* [76]** | $250/kWHr | | | ~2030 *(Distant Future)* |

* - Includes an "optimistic" future price for a pack that retails for $1050 and meets the PNGV[5] performance requirements (40 kg/3 kWHr/30 kW). OEM Price = Retail/1.44.

There are however as already mentioned, a number of issues to be resolved before Li-Ion can replace NiMH on the market –shelf life, cycle life, cost, and safety/abuse tolerance. Based on the FreedomCAR NRC report [23] it seems likely that the abuse tolerance issues and cycle behavior will be dealt with in the relatively near-term (10-15 years); shelf life is much more difficult to ascertain. The key to reducing costs seems to lie in finding less expensive cathode materials and increasing production volume – both as a means of moving along the learning curve, and because the lithium-ion battery lends itself to mass production in a way that NiMH does not. The Anderman lithium-ion cost projection cited above is the estimated cost at the very low production volumes that will be available in 2010. The ANL estimates are based on a very rigorous analysis of potential cost reduction in both cell material costs and manufacturing improvements as production increases. This cost projection assumes that capital investments and R&D expenses have already been recouped. The UC Davis study might be viewed as an early high-volume production estimate – before costs have been recouped.

[5] The PNGV-Partnership for a New Generation of Vehicles program is the Predecessor of the FreedomCar Program

Currently there is no definite consensus among people in the automotive industry whether Li-ion technology will take over or not. Some believe it is inevitable due to the rise in prices of NiMH due to the commodity price of nickel. Others feel that the exceptional durability of NiMH increases its competitiveness significantly.

Another interesting alternative to NiMH batteries in the future might be ultra capacitors. These are electrochemical energy storage devices with operating principles and thus performance characteristics midway between a battery and a capacitor. They have thus very high power density (in the order of 4kW/kg for future systems [74,75] but low energy densities-around 5Wh/kg for future systems[73,74,75]. Their advantages except for the high power density include high efficiency and durability. Additionally ultra capacitors, unlike batteries, can be fully discharged without compromising durability. Their energy density is low but all of it can be used.

The largest hurdle that ultra capacitor technology would need to overcome is their very high cost. Currently ultra capacitors cost about 35$/Wh. They are however a new technology, their costs should come down substantially possibly even down to 6$/Wh in the very long term assuming very large scale production [73]. This is six times the future cost per unit energy for Li-ion using the most pessimistic projection. Even assuming that the ultracapacitor would only need half the total energy capacity compared with a battery, as it can be fully discharged, it would still probably be an expensive solution.

Ultra capacitors may however be very interesting in combination with batteries in hybrid energy storage systems. A smaller ultra capacitor is used in such a system to level the load peaks requested from the battery. The battery is thus sized for energy and the ultra capacitor for power reducing the total system weight and possibly cost. Such a system would also prove advantageous for battery durability. There would be a complexity penalty for such a configuration though. Ulracapacitors may also be interesting storage media for mild hybrids were the energy storage required is not prohibitive.

Finally, it should be mentioned that the energy storage medium in a hybrid vehicle doesn't necessarily have to be electrochemical. Alternative energy storage mediums that have been proposed and are being researched are:

- **Advanced flywheels:** A flywheel stores energy in the form of its rotating inertia. Advances in extremely low friction, possibly magnetic bearings could significantly improve their efficiencies. Coupling the flywheel with a motor/generator in addition to the engine would increase flexibility in power flow control. However, just like ultracapacitors, advanced flywheels are primarily a power storage medium rather than an energy storage medium. They have very high power densities but very low energy densities. Using a flywheel alone in a hybrid wouldn't therefore provide significant total energy storage capacity at a reasonable added weight. They could find some application in mild hybrids or combined with batteries.
- **Hydraulic Systems:** A hydraulic energy storage system uses a reversible compressor-pump/turbine and two accumulators/tanks. To store energy, fluid is compressed and stored in the high pressure tank. To use that energy the high pressure fluid is used in the turbine to produce work. These systems are still very immature and are probably more suited for heavy duty vehicles.

Given all of the above, for this study, the assumptions that will be used for energy storage correspond to advanced lithium ion batteries. The primary assumption to be made is the power density to be used for the battery sizing calculation as that is done based on power rather than energy. A value of 1.1 kW/kg was chosen. The UCD study mentioned in Table 13 projected 0.9kW/kg however [73] talks about current peak power densities around 2kW/kg. Major Li-ion battery manufacturers such as Saft already today offer modules for hybrid vehicles with power densities higher than 2kW/kg and energy densities around 65Wh/kg [77] .These however are module numbers. Pack numbers would probably be a little lower. The power density assumption of 1.1 kW/kg could therefore be interpreted as conservative.

An energy density assumption of 80Wh/kg was used. This may be viewed as rather optimistic but it doesn't really affect the fuel economy results as the pack would have enough energy even at half this energy density even though the allowed DOD was limited to 30% for the simulations.

The bottom line is that neither the power nor the energy density assumptions are that critical for the results in this thesis. This is because the overall weight of the pack is at any case less than 40-50 kg even using current numbers. The sensitivity of vehicle performance and weight to these levels of weight changes are small.

In terms of pack efficiency, this is set in ADVISOR by adjusting the value of the pack internal resistance. The internal resistance for a current SAFT 6Ah Li-ion model built in ADVISOR is already about one third of the internal resistance of the NiMH model built in ADVISOR corresponding to the Prius and Honda Hybrid batteries. The resistance of the Li-Ion model was further decreased by 10% to account for future improvements. The resulting average efficiency over a driving cycle improves from around 82% for a current NiMH to around 92% for the future battery.

### 5.5.4. Regenerative Braking

In order for a vehicle to brake without the danger of its wheels locking up at the same time the ratio of braking force on the front over braking force on the rear wheel must follow a hyperbolic curve. If the vehicle's wheels lock up at different times, directional stability may be lost and accidents caused. It is vital therefore that this curve, called the I-curve is followed to the degree possible. This curve can be derived from vehicle dynamics [4] and can be seen in Figure 92 normalized by vehicle weight. In the absence of regenerative braking, rear wheel braking force is usually set as a fixed ratio $\beta$ of the frontal wheel braking force. The hyperbola is thus approached by a line, the $\beta$-line also presented in Figure 92. Additional corrections can be made if the vehicle is equipped with an ABS system. For a given road-tire traction coefficient $\mu$, the maximum deceleration that the vehicle can experience over the acceleration of gravity g, is equal to $\mu$. For a given maximum acceleration the tractive force on the front and rear axis are related linearly. This is the $\mu$-curve also plotted in Figure 92 for $\mu=0.8$. The ideal, in terms of drivability, ratio of frontal over rear wheel breaking force is the intersection of the $\mu$ line with the I-curve shown on the same graph.

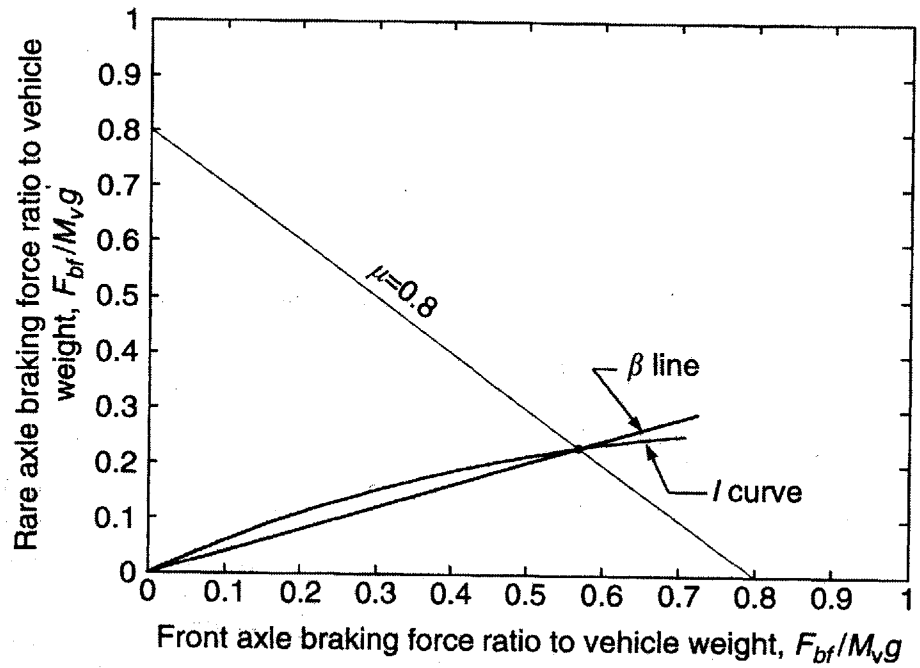


**Figure 92: Ideal and Actual Braking Force Distribution Curves.**

In a vehicle capable of regenerative braking, the motor can only apply a braking force on the driven axle, usually the front. For a given μ value between the tires and the road and a given deceleration rate, there are three different strategies to pick the optimum split between regenerative braking force on the front wheels, friction braking on the front wheels and friction braking on the rear wheels. The optimum is defined by the tradeoff between drivability and safety on one hand and recovering as much regenerative energy as possible on the other

- Series braking for optimized drivability
- Series braking for optimized energy recovery
- Parallel braking

The first strategy is already mature today; the implementation of the second one is still under research. The 3rd one is simpler to implement compared with the other two but compromises both drivability and recovered energy. The details involved in implementing these strategies are rather complicated. For an in-depth analysis the reader is referenced to [4, 78]

The bottom line is that it is generally suggested [4] that for low vehicle deceleration rates, below 0.2g=1.96m/s$^2$ all of the braking force can be supplied by the motor. The maximum deceleration rate over the FTP and HWFET is 1.48 m/s$^2$ .Over the US06 it is 3.08 m/s$^2$ which is still small in terms of g-rate. It would therefore be realistic to assume that all of the braking over these cycles can be done with the motor for these cycles. Toyota in its paper on the second generation Prius [59] presents the graph shown in Figure 93, implying that around 80-90% of the braking in the second generation Prius is done through regeneration. For the Civic hybrid, this plot looks significantly worse because the engine cannot be decoupled from the motor [56].

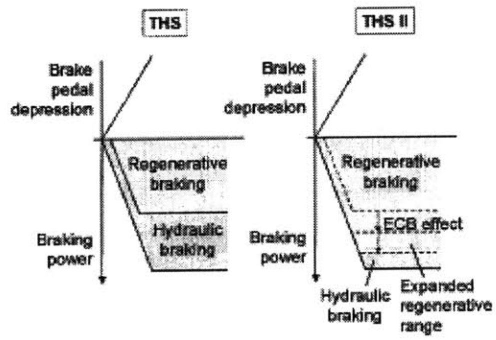


**Figure 93: Prius Regenerative Braking System Efficiency.**

However, there are two additional complications:

- In order for the motor to absorb the regenerative power the braking system can direct to it at a given time instant over the driving cycle, it must be able to produce the required torque at that time instant. This depends on the motor speed at that time. The amount of regeneration depends thus on vehicle speed and gear ratio to the motor.
- The motor and controller have an "inertia' in responding to a required torque input. If for example in the previous time instant the motor was used for power assist and the torque request changes to negative (regeneration) the motor can't immediately start generating electricity.

The parameter that controls how much braking will be done using the friction brakes in ADVISOR is a simple braking force fraction which may or may not depend on vehicle speed. In accordance with the paper on the Prius and because of the low deceleration rates over the drive cycles used, this fraction was set to 90% regeneration, 10% friction braking for all vehicle speeds. Even so, as will be seen in the results chapter, the percentage of friction braking is around 20-30% even when regeneration is not limited by the size of the motor. This is not due to torque limitations from the motor at high speeds; it is mostly attributed to the motor-controller "inertia" when switching from power assist to regeneration. To what degree this phenomenon is simply an artifact of the simulation or it represents an actual limitation in controlling hybrid motors hasn't been investigated. This percentage of additional loss is the same for all vehicles and doesn't affect the trends in the final results.

## 5.6. Controls

As already explained, in the case of a manual/automatic transmission with engine optimizing capability, ADVISOR offers two main alternatives:

- Setting an engine speed below which the powertrain only uses the electric subsystem.
- Setting a fraction of the engine's maximum torque below which the powertrain only uses the electric subsystem.

The first option as explained is not useful for the engine models in this study. The second one will be used instead.

The main question that needs to be answered to implement this control strategy is what the optimum engine off torque is. For a higher engine off torque, the engine efficiency is higher. However more of the energy produced from the energy needs to go through the electric subsystem. This means that more of the energy needs to undergo electric losses. The optimum balance was found to be around 40% for current hybrids. For future hybrids the efficiencies of the electric subsystems are improved, the optimum torque fraction is thus higher. However the higher the engine off torque, the more difficult it will be for the transmission to quickly shift between the engine and the motor as the power source. The engine off torque fraction was thus kept at 40% for all models.

## *5.7. Sizing Methodology*

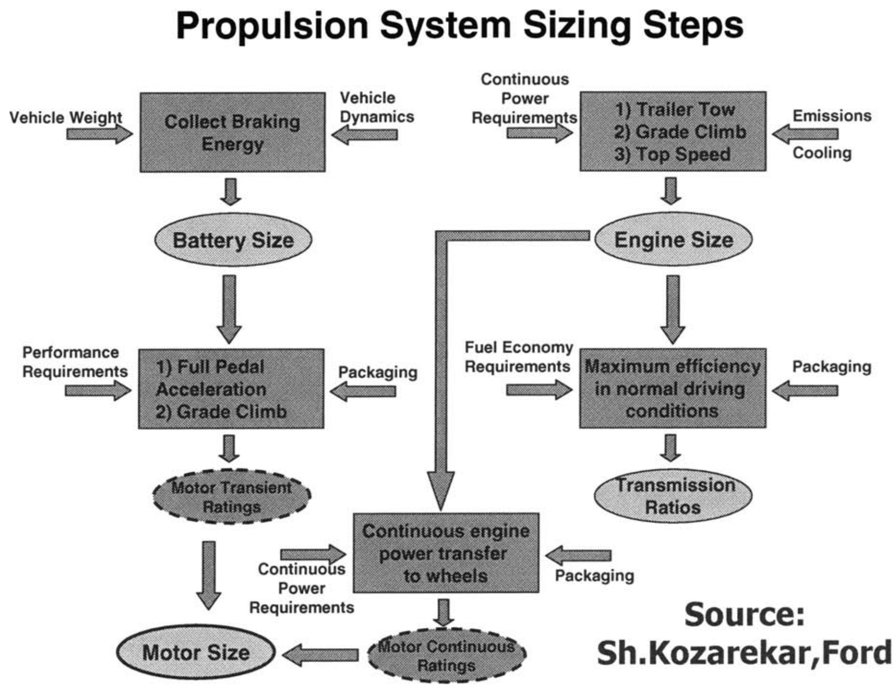


**Figure 94: Hybrid Sizing Methodology Used by the auto industry [79]**

Figure 94 shows the methodology steps used by the industry in order to size different components in hybrids. The engine is sized to meet the minimum towing, grade climbing and top speed specifications set for the vehicle. Towing, grade climbing and top speed are usually defined without specifying a time limit. This means that the vehicle needs to meet these requirements with the engine alone without the electric system. The battery is sized to collect a significant part of the regenerative braking energy. This is usually a power, not an energy limitation. The motor is sized for engine assist during full pedal acceleration. The gear ratios are chosen to optimize engine fuel consumption. For the motor it is important to consider both its constant ratings as well as its transient ones as they might be significantly different.

The methodology used in this thesis will be a simplified version of the industry methodology. More specifically:

- The relative size of the electric system will be treated parametrically. The hybridization ratio HR is defined for this purpose as the maximum motor power over the maximum engine power:

$$HR = \frac{P_{\max,motor}}{P_{\max,engine}} \quad (5.4)$$

As the cost premium of a hybrid scales with the size of the electric system, this was considered necessary to establish how the relative benefit from hybridization scales with the size of the electric system.

- To guarantee the same performance, the same total (engine+motor) power over curb weight ratio was kept the same as in the 2005 vehicles.
- The size of the battery was chosen based on power. The battery should have enough power for the motor:

$$P_{battery} = \frac{P_{\max,motor}}{\eta_{average,motor}} \quad (5.5)$$

$$\frac{W_{curb}}{P_{\max,eng}} = const = c \Rightarrow \frac{W'_{curb}}{P'_{\max,eng} + P'_{\max,motor}} = c \Rightarrow \frac{W'_{curb}}{P'_{\max,eng}(1+HR)} = c \quad (5.6)$$

Where prime variables denote future quantities and non prime current

The curb weight of the hybrid vehicle can be broken down into the weight of the vehicle without the powertrain (glider mass), the weight of the engine, the transmission, the aftertreatment, the motor and controller, the batteries as well as the additional support mass. Usually due to the higher fuel economy of the vehicle the amount of fuel carried on a hybrid vehicle is less to get the same range as a non hybrid vehicle.

$$\begin{aligned} W'_{curb} = (W'_{glider} + W'_{engine} + W'_{transmission} + W'_{aftertreatment} + \\ + W'_{motor} + W'_{batteries} + W'_{\sup port} - \Delta(W_{fuel})) \end{aligned} \quad (5.7)$$

The mass of the additional support minus the change in fuel mass in [7] was 8kg. The same was used here. The glider mass and transmission was assumed to be the same as those of the SI 2030 vehicle. The exhaust aftertreatment system weight was estimated as the weight of the aftertreatment system of the 2030 SI vehicle scaled by engine maximum power. Grouping together all of these known quantities and calling them $W_{rest}$:

$$\frac{W'_{rest} + W'_{engine} + W'_{motor} + W'_{batteries}}{P'_{\max,eng}(1+HR)} = c \quad (5.8)$$

Using the assumptions on engine, motor and battery power densities

$$\frac{W'_{rest} + P'_{\max,eng}/0.925 + HR * P'_{\max,eng}/1.55 * + HR * P'_{\max,eng}/\eta_{motor}/1.1}{P'_{\max,eng}(1+HR)} = c \quad (5.9)$$

The engine power can be calculated from this last expression.

Except having enough power, the sizing should also make sure that the battery has enough energy for the needs of the powertrain. Energy requirements from the battery are:

- The battery pack should have enough energy to power the motor during hard accelerations. 0-100 km/h acceleration takes about 10-12s. A 12 s pulse at the peak power of the motor would give:

$$E_{battery} = t * P_{motor} / \eta_{motor} / \eta_{battery,discharging} \quad (5.10)$$

For 20kW motor and typical efficiencies for the components this equation gives 83 Wh of energy. When sizing the battery to power this motor, 20kW/0.87/1.1kW/kg=18.67kg. At the assumed energy density of 80Wh/kg, this pack has 1494Wh. Even at a state-of charge envelope of 0.3, the available pack energy is 448Wh. An order of magnitude more that is, then what is needed for a 10-20s pulse.

- The battery needs to have enough power so that no regeneration energy is lost due to the battery being full at that time. This is more difficult to estimate without actually running a simulation as it also depends on the driving cycle and hybrid control strategy.

The real reason why battery back energy is not limiting is that the energy flows in and out of the battery during a driving cycle. Accelerations or decelerations during usual driving cycles do not last more than a minute. Moreover, the rates of accelerations and decelerations that last long are low. That is self-evident as it only takes a vehicle 15-20s to get to its top speed. The required electric power is thus seldom equal to the peak electric power of the system. Most importantly, accelerations are usually followed by decelerations and vice versa. The battery state of charge fluctuates therefore around its initial value. The power flow in and out of the battery of the 2005 Civic Hybrid over the FTP is presented in Figure 95 .The state of charge history for the battery of the same vehicle over the FTP is shown in Figure 96. It can be seen that the SOC varies only by about 0.025 which corresponds to about 40Wh.

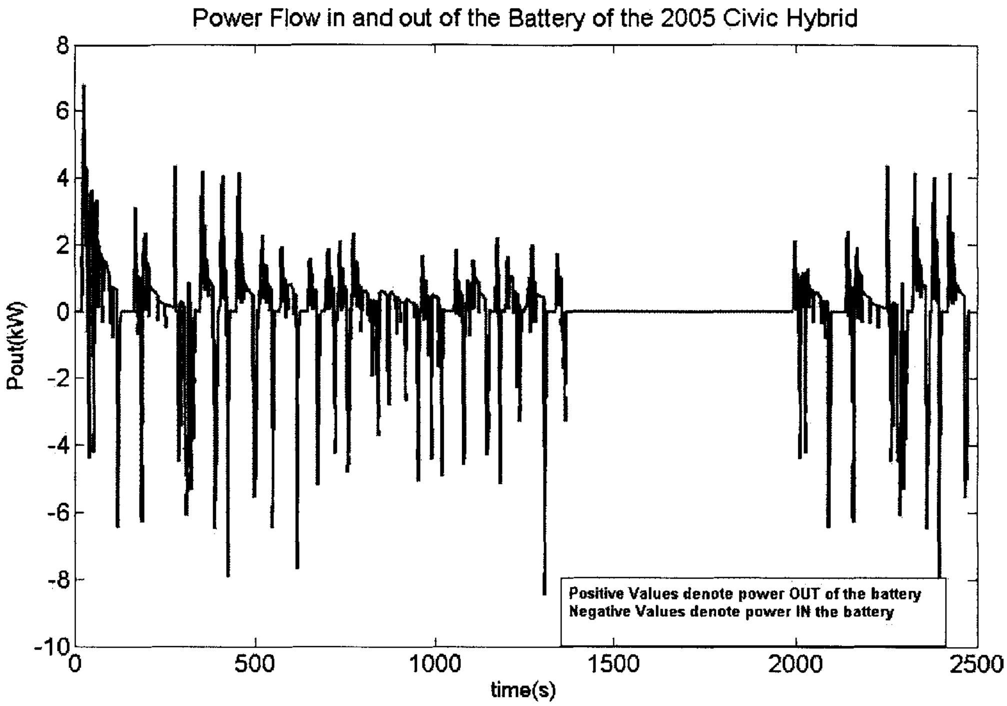


**Figure 95: Power flow out of the battery of the 2005 Civic Hybrid over the FTP**

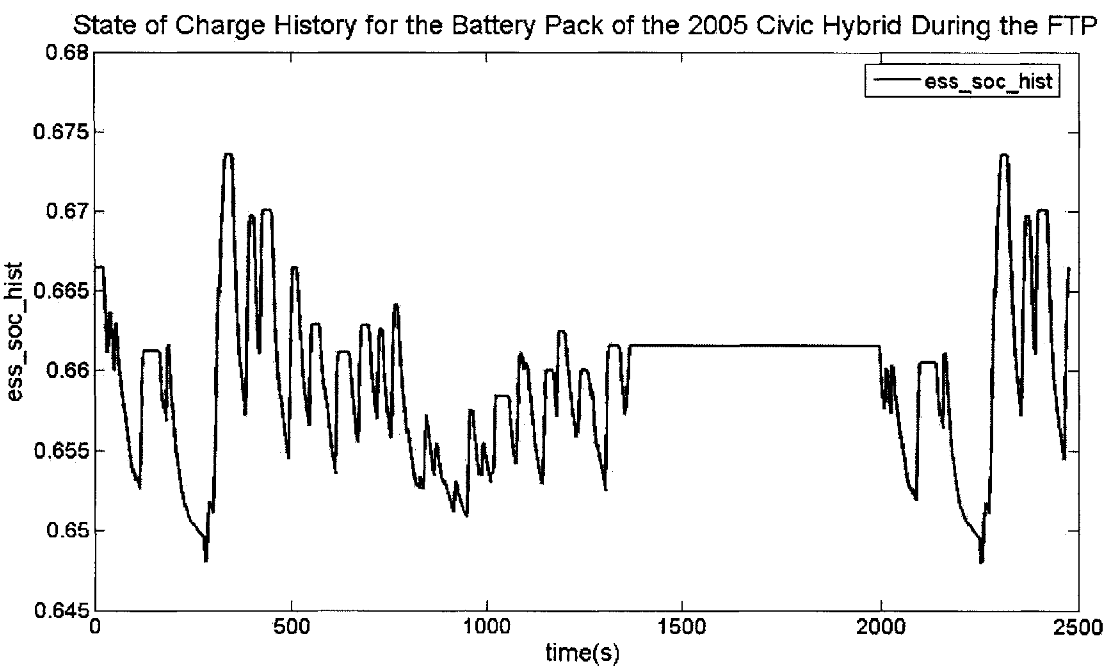


**Figure 96 : State of Charge History over an FTP cycle for the 2005 Civic Hybrid**

## 5.8. The effect of HR

The hybridization ratio will be treated parametrically. It is important, however to understand the tradeoffs involved in varying the relative size of the electric system. The advantages of a larger HR are:

- A smaller engine for the same vehicle and performance. The engine is thus operating at higher loads increasing its efficiency. This benefit is of little value though when the hybrid powertrain offers active engine optimization.
- More regenerative energy can be captured. The minimum size of the electric system to capture all or most of the regenerative energy will be investigated further.
- There is also a minimum size of the electric system to enable active engine optimization.

The disadvantages from a larger HR are:

- The cost of the powertrain goes up.
- The size of the engine is decreased. The towing, grade and top speed performance of the vehicle is therefore lower as these tests are defined or understood as the performance that the vehicle can sustain for a long time period. This means that the hybrid needs to rely on the engine alone after the batteries have been depleted.

The first interesting question to answer is to determine the HR required to implement active engine optimization. It was mentioned that the engine off torque fraction was set to 40% for manual/automatic hybrids with an optimization capability. How much does the motor power need to be to undertake this load? The engine power output for the lower performance 2030 Camry with a naturally aspirated SI engine over the FTP is presented in Figure 97. The continuous, black line represents 40% of this power. It can be seen that its maximum is around 12kW. A 12kW motor should therefore be enough. Since this vehicle has a peak engine power of 95.4kW a 10-15% HR should be enough. The power demand from the engine of the same vehicle over the 40% is shown in Figure 98. 40% of the US06 power curve peaks at 28kW is on the other hand mostly below 20kW. A HR of 20-30% is therefore necessary for the US06.

The second interesting question that can be answered is what the HR should be to capture most of the regenerative energy. A histogram of the deceleration powers required at the wheel is presented in Figure 99. This histogram corresponds to the low performance 2030 NA SI Camry (1297 kg incl. passenger) over the FTP The cumulative energy versus power is also presented in that plot. The same plot for the US06 is presented in Figure 100. If perfect regeneration were possible, a 10kW motor would be needed to capture 80% of the deceleration energy for the FTP and 24kW to capture 100%. For the US06, because of the higher decelerations 22.5kW would be needed to capture 80% of the deceleration energy and 57.4kW to capture all of it. As explained however regenerative braking for a two wheel is less than 100% due to various limitations. 70-80% is more realistic. The motor size that is needed is therefore smaller.

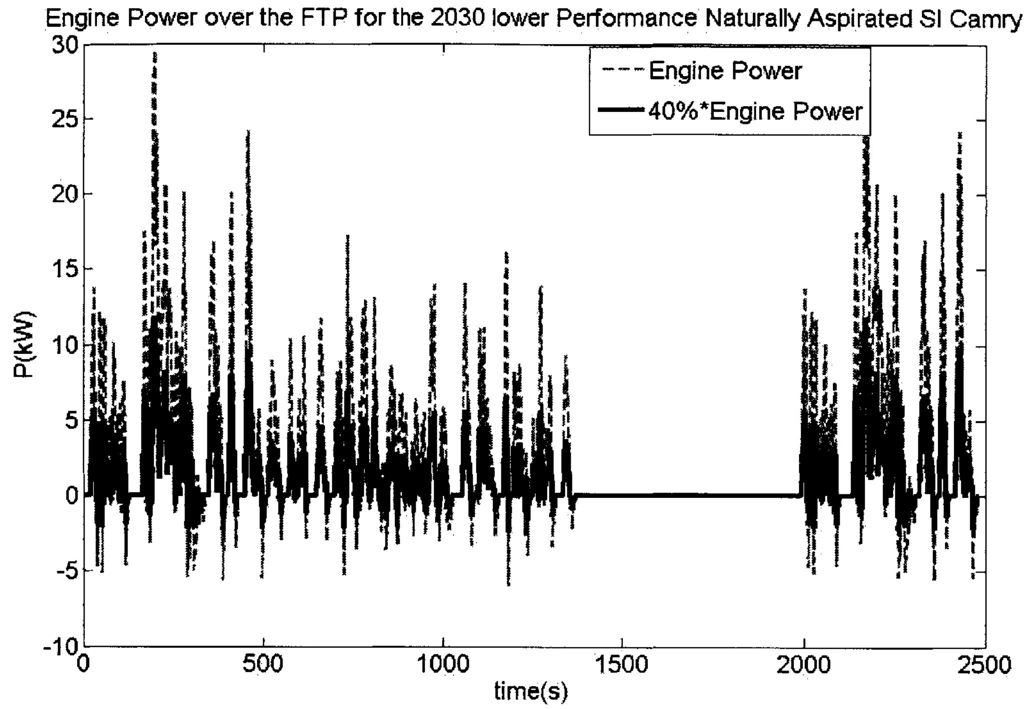


**Figure 97 : Engine Power Output For the lower performance 2030 NA SI Camry over the FTP**

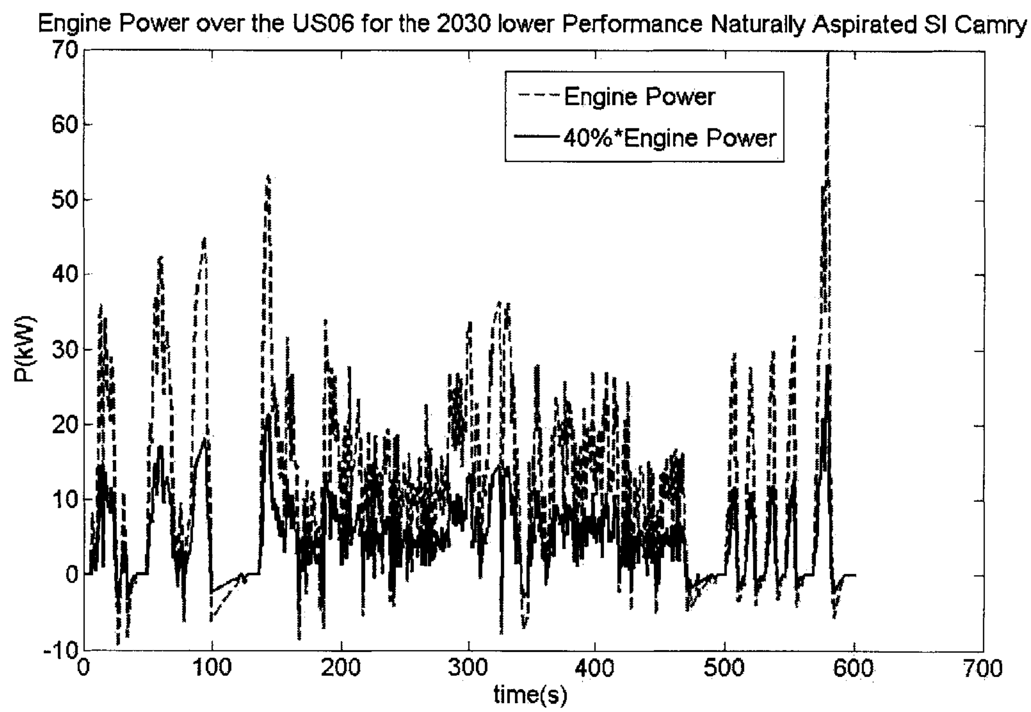


**Figure 98 : Engine Power Output For the lower performance 2030 NA SI Camry over the US06**

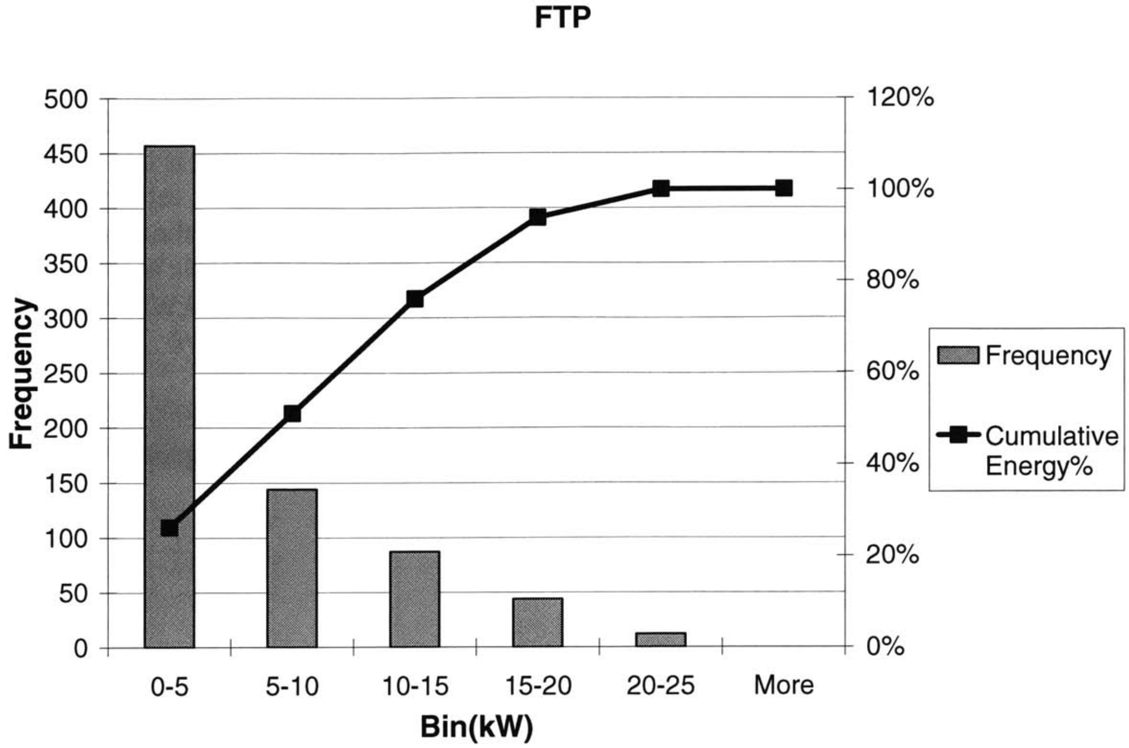


**Figure 99: Histogram of Deceleration Power at the wheel for the lower performance 2030 NA SI Camry over the FTP**

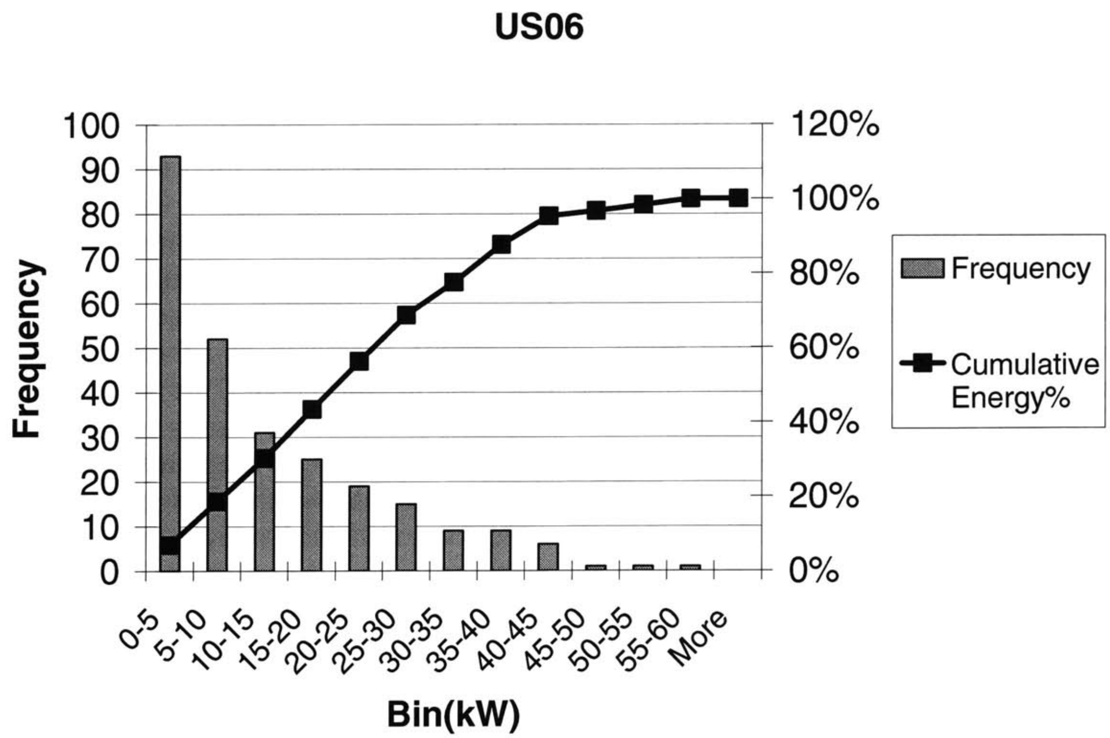


**Figure 100 : Histogram of Deceleration Power at the wheel for the lower performance 2030 NA SI Camry over the US06**

It is finally interesting; in order to choose which HR to model, to look at the HR's in current hybrid vehicles. The 2006 Civic Hybrid has an HR of 18%. Determining the HR for the 2nd generation Prius is slightly complicated. The engine power is 57kW. The motor maximum output is 50kW; however the maximum battery power is 21kW. Both the engine and the motor cannot therefore be at their maximum power at the same time. The maximum system power for the Prius is 82kW [59]. Assuming that the engine is at its maximum power at that point, the electric system contributes 82-57=25kW which still seems unrealistically high as the battery maximum power is only 21kW. Using the battery power as the power of the electric system, the resulting HR is 37%. Assuming a 90% efficiency for the motor, the HR for the Prius is 33%

The HR's which were modeled for this thesis were 10%, 20% and 30%. As will be seen, for future hybrids, the benefit from higher HR's is small compared with the additional cost.

## 5.9. Gear Ratios Selection

The methodology used to select gear ratios for the gearboxes future hybrid vehicles is essentially the same as that used for non-hybrid gearboxes, i.e.:

- The first gear ratio was chosen so that the ratio of tractive force over weight at the wheel ($\mu$) is the same as it was in the 2005 vehicle.
- The top gear ratio was chosen so that at 47.5 mph, the engine is operating at 1300 rpm scaled by its maximum speed.
- The standard industry step ratios were used to determine intermediate gear ratios.

There was however, one additional issue that needed to be resolved to choose gear ratios for a hybrid. For non-hybrids, the maximum engine torque used to calculate the maximum tractive effort is well defined. The maximum engine torque doesn't vary a lot with speed, using an average engine speed is thus appropriate. For hybrids on the other hand, the maximum tractive force is determined by both the maximum engine and motor torque. DC motor maximum torque curves have an almost constant torque at low speeds until they reach a *base speed* above which, torque decreases hyperbolically. When accelerating, the available torque from the motor follows that curve. This can be seen for 0-60 mph acceleration for the 2030 20% HR lower performance Camry in Figure 101. The available tractive effort from the motor varies a lot. If the average maximum motor torque is used to estimate the total tractive effort and choose the first gear ratio, the resulting 0-60 mph time is lower than the time for non hybrid vehicles with the same power to weight ratio. If the maximum motor torque is used, the resulting 0-60 mph time is higher. In order to get the same level of performance a value in between the maximum and the average torque must be used. As the HR increases and the powertrain increasingly relies on the electric system , the motor torque value that gives the same performance as the non-hybrid powertrains moves away from the maximum and closer to the average motor torque. In a relative sense, a higher gear ratio is needed.

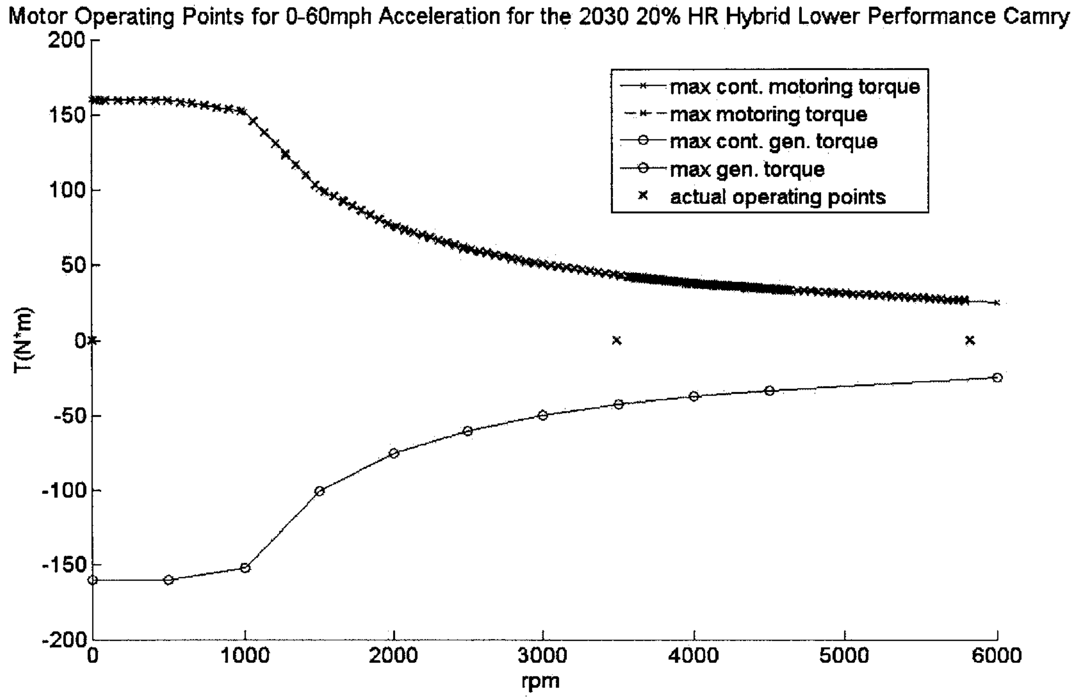


**Figure 101: Motor Operating Points for a 0-60mph acceleration**

# 6. Results

## 6.1. Current models –Validation

In order to validate the simulations, a comparison between simulated and measured combined[6] fuel consumption is presented in Table 14. Simulated and measured 0-60 mph acceleration times are compared in Table 15.It was mentioned in chapter 5 that ADVISOR cannot model power split hybrid architectures accurately. To show that parallel hybrid architectures can be modeled accurately, a 2005 Honda Civic Hybrid was included in these validation simulations. All of the results in Table 14 and Table 15 show good agreement with measured numbers. Note that the fuel consumption values include the standard adjustment[7]. It should be mentioned however, that although fuel consumption measurements are well established, acceleration measurements are not. Fuel consumption measurements are conducted by the EPA in a well standardized manner and results are published at www.fueleconomy.gov and also on automakers webpages. Acceleration numbers are not published by automakers or government agencies; they are rather unofficially measured by car magazines and enthusiasts. The spread of the data is substantial and there is no real way to verify them. The measured acceleration numbers in Table 15 are simply the average of values found on the web. For consistency purposes, all comparisons between future and current models will be made on the basis of simulated values.

[6]As explained in chapter one, the combined fuel consumption is calculated as 45% x (highway fuel consumption)+55% x (city fuel consumption)

[7] As explained in chapter one the standard adjustment is 0.9 for the FTP cycle and 0.78 for the HWFET

**Table 14: Validation of ADVISOR Fuel Consumption Simulations for Current Models**

| | | ***ADVISOR*** | ***EPA 2005*** | |
|---|---|---|---|---|
| ***Model*** | ***Cycle*** | ***(liters/100km-adjusted)*** | ***(liters/100km-adjusted)*** | ***error*** |
| ***Camry 2.5 liter*** | City | 9.9 | 9.8 | -1% |
| | Highway | 7.5 | 6.9 | -8% |
| | Combined | 8.8 | 8.5 | -4% |
| ***Camry 3.0 liter*** | City | 12.5 | 11.8 | -6% |
| | Highway | 8.6 | 8.4 | -2% |
| | Combined | 10.7 | 10.2 | -5% |
| ***F150 4.2 liter*** | City | 14.8 | 15.7 | 6% |
| | Highway | 12.1 | 11.8 | -3% |
| | Combined | 13.6 | 13.9 | 2% |
| ***F150 5.4 liter*** | City | 17.0 | 16.8 | -1% |
| | Highway | 12.3 | 12.4 | 1% |
| | Combined | 14.9 | 14.8 | 0% |
| ***Civic Hybrid Manual*** | City | 5.2 | 5.2 | 0% |
| | Highway | 4.6 | 4.4 | -5% |
| | Combined | 4.9 | 4.8 | -2% |

**Table 15: Validation of ADVISOR Acceleration Simulations for Current Models**

| | ***ADVISOR*** | ***Average Published*** | |
|---|---|---|---|
| ***Model*** | ***0-60 mph (s)*** | ***0-60 mph (s)*** | ***error*** |
| ***Camry 2.5 liter*** | 9.4 | 9.2 | -2% |
| ***Camry 3.0 liter*** | 8.0 | 7.8 | -3% |
| ***F150 4.2 liter*** | 9.8 | 9.6 | -2% |
| ***F150 5.4 liter*** | 8.1 | 8.0 | -2% |
| ***Civic Hybrid Manual*** | 12.3 | 11.4 | -8% |

## 6.2. Tank to Wheels Combined adjusted HW-FTP Fuel Consumption

The predicted fuel consumption for different powertrains is presented in Figure 102 for the future lower performance Camry. These numbers represent the combined city and highway fuel consumption including the standard adjustment. All the diesel numbers in this chapter will be presented in the form of their gasoline equivalent in terms of lower heating value.

The results are presented in the form of a most probable value, denoted by a diamond and a range or error bar. The fuel consumption of the current Camry is also presented for comparison. The most probable value for all non hybrid powertrains represents using an advanced six speed manual/automatic gearbox. The range for these powertrains represents using a very efficient (87%) CVT or a transmission that can significantly narrow down the shift window. These were considered less probable due to the general pessimism in the auto industry about developing an efficient CVT and the large challenges associated with narrowing the shift window. The most probable value for the hybrid represents systems that can actively determine engine operating points. These would include, as explained earlier, either a manual/automatic transmission hybrid, implementing an all electric off-torque envelope in a parallel architecture, or an efficient CVT in a parallel hybrid or a power split architecture. The systems that actively control engine operating points were considered more probable as the power-split architecture is already doing that in current models. The range in the plot represents systems without active engine optimization or with an inefficient CVT. All the hybrids of choice have a hybridization ratio (HR) of 20% for reasons that will be explained later. For the rest of this chapter, the most probable values will be used for all powertrains.

Using the 2030 naturally aspirated SI as a baseline to normalize the most probable combined fuel consumption numbers, the plot in Figure 103 is obtained.

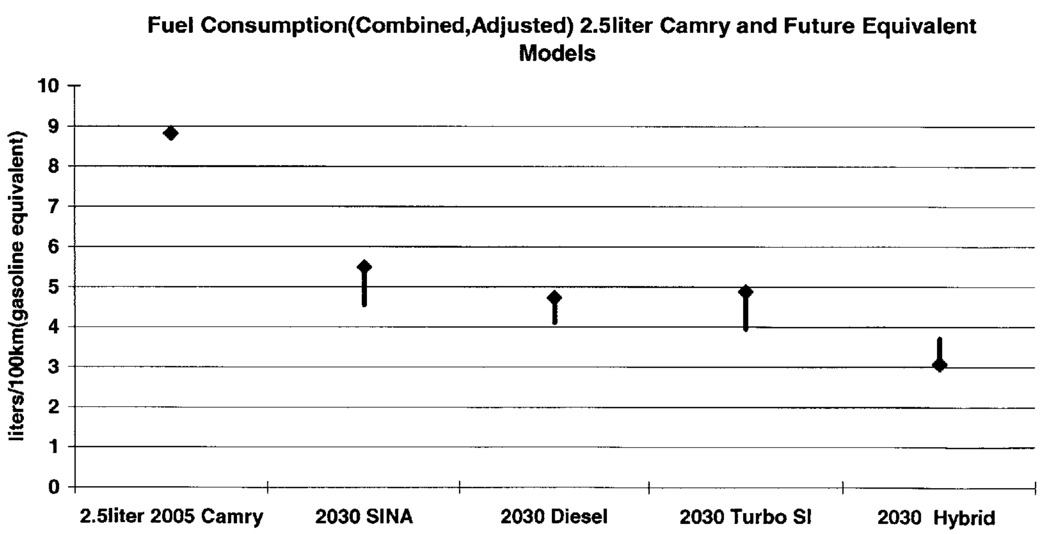


**Figure 102: Predicted Fuel Consumption for Different Powertrain Options for the future lower performance Camry.**

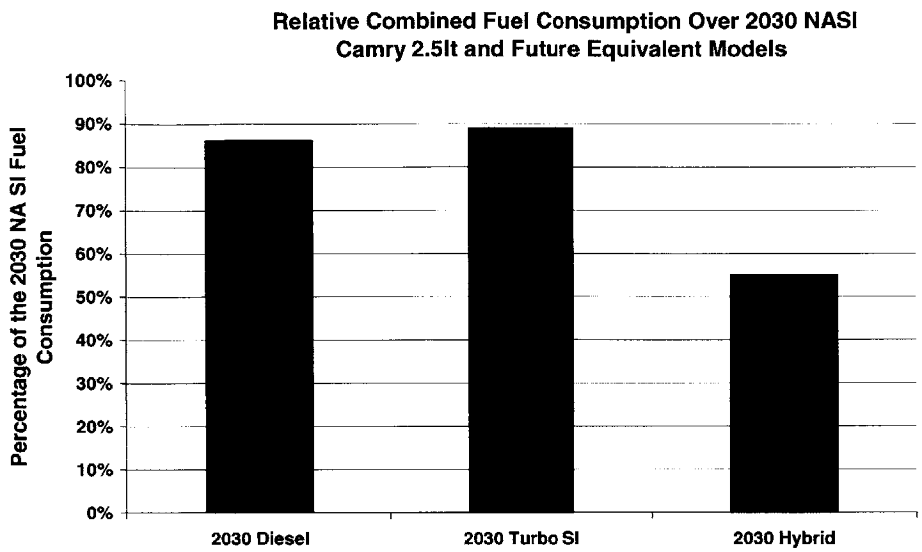


**Figure 103: Relative Combined Fuel Consumption over the 2030 Naturally Aspirated SI.**

The first main conclusions from Figures 103 and 104 is that the current gap in fuel consumption between diesel and SI engines seems to narrow down significantly. The turbocharged SI especially, appears to be close to the diesel. Currently diesels are about 25-30% better in terms of fuel consumption. The continuing downsizing of engines means that SI engines are not forced to operate as much at low bmeps where the diesel has a large competitive advantage. Furthermore, it was assumed that there is more friction reduction potential in SI in percentage terms, than diesel (-25% versus 15%). Although there is considerable confidence that friction reduction in NA SI will be higher than that in diesels, it is less clear what the assumptions for turbo SI engines should be. A sensitivity analysis will therefore be performed on that particular assumption.

The second important conclusion is that hybrids do maintain a significant advantage over conventional powertrains in the future. This is in part due to the assumption of using a manual/automatic transmission that actively optimizes engine operation in order to model a hybrid. Furthermore, it was assumed that this is the most probable scenario. In a sense, it could be argued that if advanced transmissions are the most probable scenario for hybrids, why aren't they the most probable scenario for conventional powertrains? Why was there a higher degree of optimism in assuming an actively-optimizing manual/automatic transmission in hybrids and not a CVT in non hybrids? The answer was of course, that power-split architectures can already optimize engine operation in hybrids today. There is an underlying assumption to this claim, which is that an advanced manual/automatic transmission in a parallel architecture mimics the fuel economy of a power-split architecture. In reality, they should be close.

## 6.3. Performance Results (0-60, 40-60)

The comparison of acceleration times between the current 2.5 liter Camry, 3.0 liter Camry and 4.2 liter F150 and future equivalent models with different powertrains is presented in Figures 104,105 and 106 respectively. 0-60 mph times do not differ by more than 5% from the 2005 models, which validates the sizing and gearbox selection methodology.40-60 times are also very close.

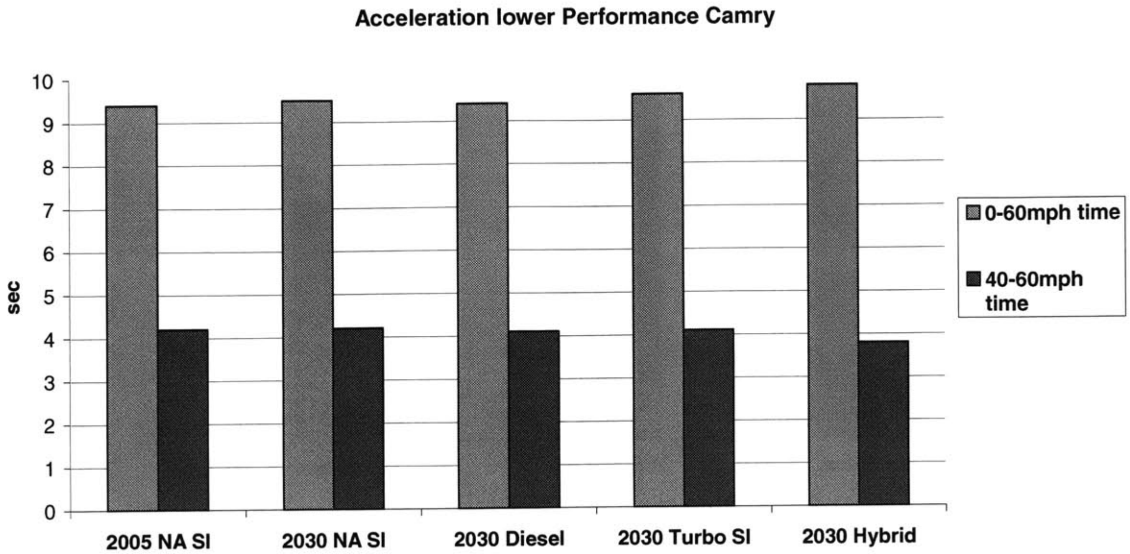


**Figure 104: 0-60 mph Acceleration Times for the Current 2.5 liter Camry and its future equivalent models.**

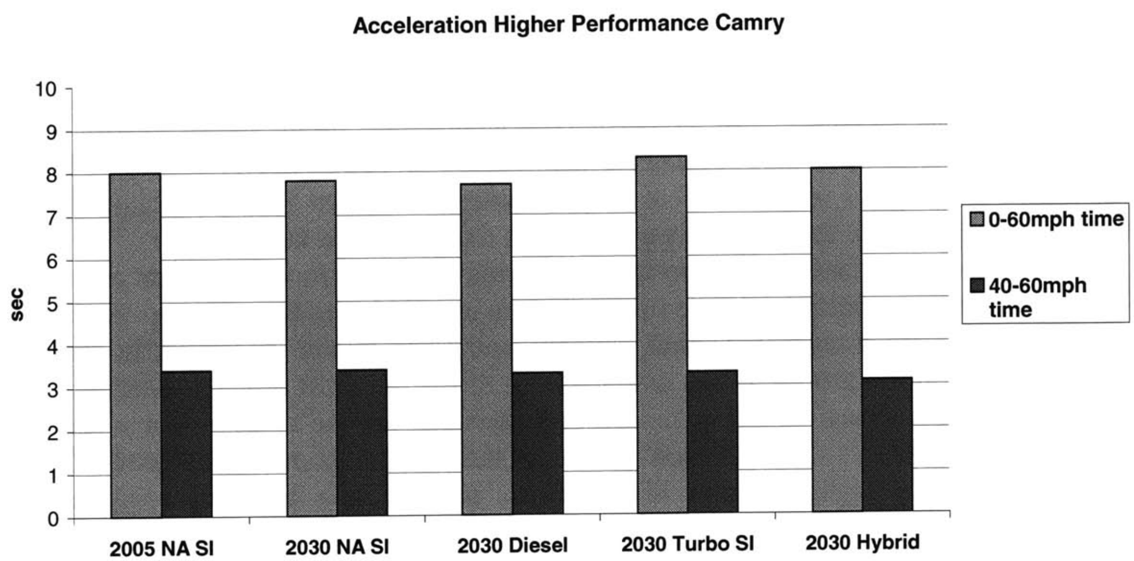


**Figure 105: 0-60 mph Acceleration Times for the Current 3.0 liter Camry and its future equivalent models.**

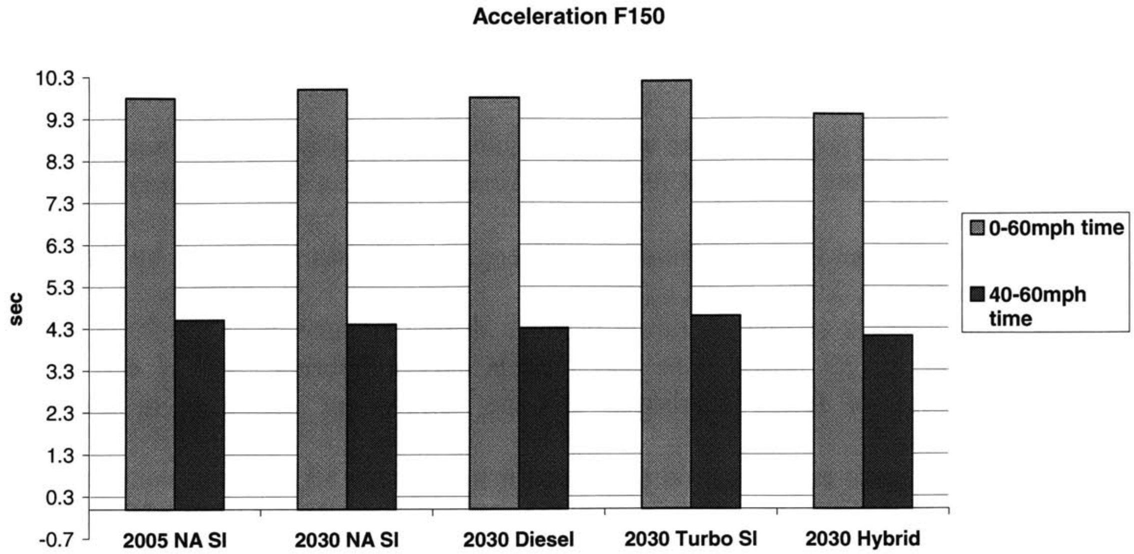


**Figure 106: 0-60 mph Acceleration Times for the Current 4.2 liter F150 and its future equivalent models.**

## 6.4. The Effect of Performance and Vehicle Type

The relative reduction of combined, adjusted fuel consumption for the 2.5 liter Camry, the 3.0 liter Camry and the 4.2 liter F150 and their future equivalent models is presented in Figure 107. The relative reduction for each model is defined as:

$$RI_{2005} = \frac{FuelConsumption2005 - FuelConsumption2030}{FuelConsumption2005} \quad (6.1)$$

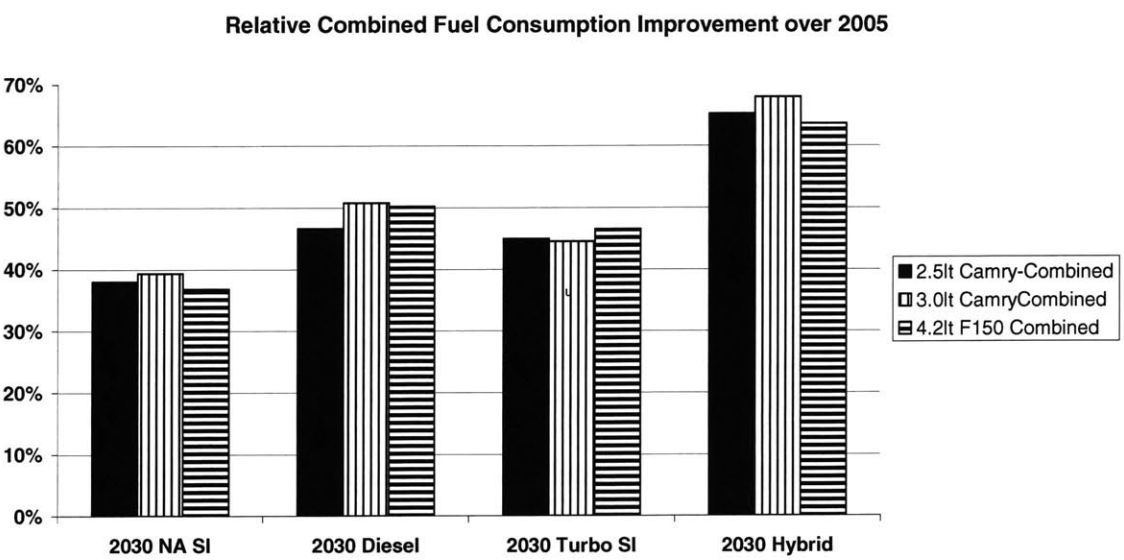


**Figure 107: Relative Fuel Consumption Reduction for Different Future Powertrains for three Vehicle Models**

It may be seen from Figure 107 that the relative reduction of fuel consumption for different models is about the same for the same type of powertrain. This is not surprising because:

- The energy required at the wheel is reduced by roughly the same amount. This is because the individual resistances have been reduced by roughly the same amount.
- The efficiency of the driveline and engine map has improved by the same percent.
- The average engine operating point is about the same as in the 2005 NA SI model for all powertrains and models except hybrids. This is due to the selection of future gearboxes that are "equivalent" to those in the 2005 models.
- The average engine operating point for hybrids for all different vehicle types. This is because the of active engine optimization, elimination of idling and smaller engines.

In order to examine the average engine operating point in future, the average torque fraction over a driving cycle is defined:

$$TF = \frac{\overline{T}_{drivingcycle}}{T_{\max,eng}} \quad (6.2)$$

The numerator is the average engine torque output over the driving cycle. The denominator is the engine maximum torque. As engine maximum torque varies with speed and in a different way for different engine types, the speed averaged maximum engine torque was used.

The average torque fractions over the urban (FTP) cycle for the 2.5 liter Camry, the 3.0 liter Camry, the 4.2 liter F150 and their future counterparts are presented in Figures 108, 109 and 110 respectively. It is obvious that for the non-hybrid powertrains, TF remains almost constant among the different non hybrid powertrains for the same model. The hybrid powertrains manage to significantly increase TF due to active engine optimization. The diesels are operating at slightly lower TF, which probably means that even lower top gear ratios should have been chosen for them, but the difference is small.

The reasons why future non-hybrid powertrains are operating at about the same TF are easily understood with a simple scaling analysis. Average vehicle resistances over a driving cycle were reduced by:

$$\frac{\Sigma\overline{R'}}{\Sigma\overline{R}} \propto k_1 \frac{R'_{accel}}{R_{accel}} + k_2 \frac{R'_{aero}}{R_{aero}} + k_3 \frac{R'_{rolling}}{R_{rolling}} = k_1 \frac{W'}{W} + k_2 \frac{c'_d}{c_d} + k_3 \frac{c'_r * W'}{c_r * W} \quad (6.3)$$

Where W is the curb weight, $c_D$ the coefficient of drag and f the rolling friction coefficient. Prime variables indicate future The k's denote coefficients with values from 0 to 1 that indicate the relative significance of each resistance in a driving cycle, so for the FTP , $k_2$ would be smaller than what it would be for the HWFET. Although, the values of $k_1$, $k_2$, $k_3$ will be different for the Camry and the F150 even for the same drive cycle, the percentage reduction of each resistance was the same. Substituting the values used in this study.

$$\frac{\Sigma \overline{R}'}{\Sigma \overline{R}} \propto k_1 * 0.82 + k_2 * 0.75 + k_3 * 0.55 \Rightarrow 0.55 < \frac{\Sigma \overline{R}'}{\Sigma \overline{R}} < 0.82 \quad (6.4)$$

(Note that 0.82 is used instead of 0.8 for the ratio of weights as the weight of the passenger (136 kg) was not included in the 20% reduction assumed for curb weight).

The ratio of resistances for all the models in this study will therefore be somewhere between 0.55 and 0.82 depending on the relative significance of each resistance. If rolling, aerodynamic and acceleration were weighted equally, the ratio of future over current resistances would be the average of 0.82, 0.75 and 0.55, i.e. 0.69. In urban cycles however, acceleration is more important. On the highway, aerodynamics is the most important resistance. The ratio of resistances is thus closer to 0.82.

In order to keep the same performance, the gearbox was chosen so that the ratio of weight over engine maximum torque was kept constant. This is definitely true for the first gear, but roughly valid for all if we define an average gear ratio $\bar{i}$ over the drive cycle. Hence:

$$Tf = \frac{\overline{T}_{eng}}{T_{\max,eng}} \propto \frac{\overline{T}_{wheel}}{\bar{i} * T_{\max,eng}} \propto \frac{\Sigma \overline{R} * r_w}{\dfrac{\mu * 0.5 * W * r_w}{T_{\max,eng}} * T_{\max,eng}} \propto \frac{\Sigma \overline{R}}{W} \Rightarrow$$

$$\frac{T'f}{Tf} \propto \frac{\dfrac{\Sigma \overline{R}'}{W'}}{\dfrac{\Sigma \overline{R}}{W}} \propto \frac{\Sigma \overline{R}'}{\Sigma \overline{R}} * \frac{W}{W'} \Rightarrow \frac{0.55}{0.82} < \frac{T'f}{Tf} < \frac{0.82}{0.82} \Rightarrow 0.67 < \frac{T'f}{Tf} < 1 \quad (6.5)$$

(Recall from chapter 5 that μ was kept constant).

But the ratio of future over current resistances is closer to 0.82 than 0.55. It will be shown that the ratio of resistances in urban driving for this study is about 0.74. This leads to a ratio of future over current TF of 0.9.

The average torque fraction for all the models are thus roughly the same or decreases very slightly from current to future models. Most of the future models with the exception of hybrids even with different powertrains will, as a result, operate at roughly the same load as the 2005 ones.

The minor differences between vehicle types in Figure 107, are explained when looking at the energy flows of the vehicles. For example, the energy flows for current and future counterparts of the 2.5 liter Camry are presented in Table 16 and Table 17 respectively. Notice how the reduction in acceleration energy at the wheel is less than the 18% used before. This is due to the rotating inertias which were assumed almost constant. The reduction of total energy at the wheel is the same between the lower and higher performance Camry. The 3.0 liter engine however has slightly more of an efficiency gain percentage wise over its 2005 counterpart. This causes the slightly higher percentage reduction over 2005 in fuel consumption for the 3.0 liter.

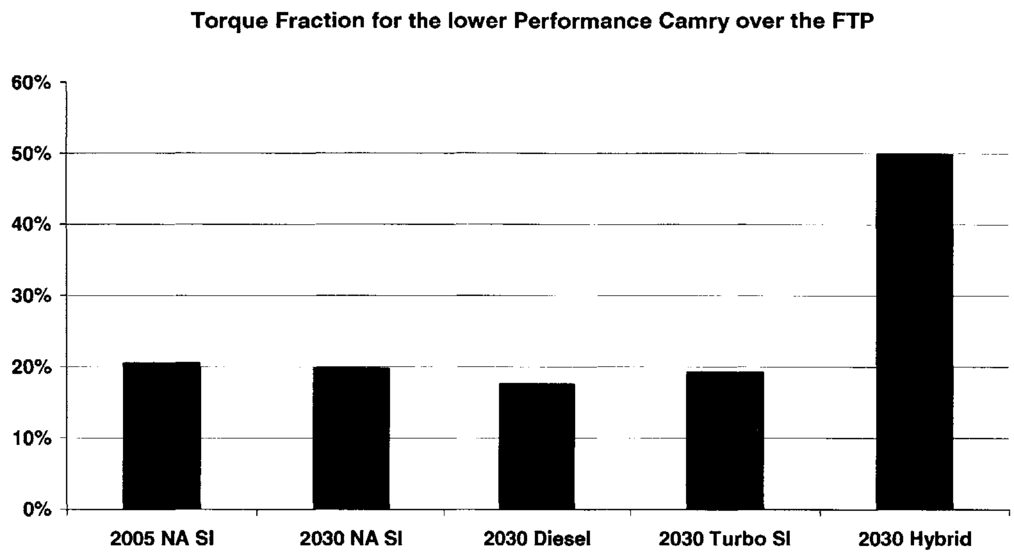


**Figure 108: Torque Fraction over the FTP for the 2005 2.5 liter Camry and its Future Equivalent Models Using Different Powertrains**

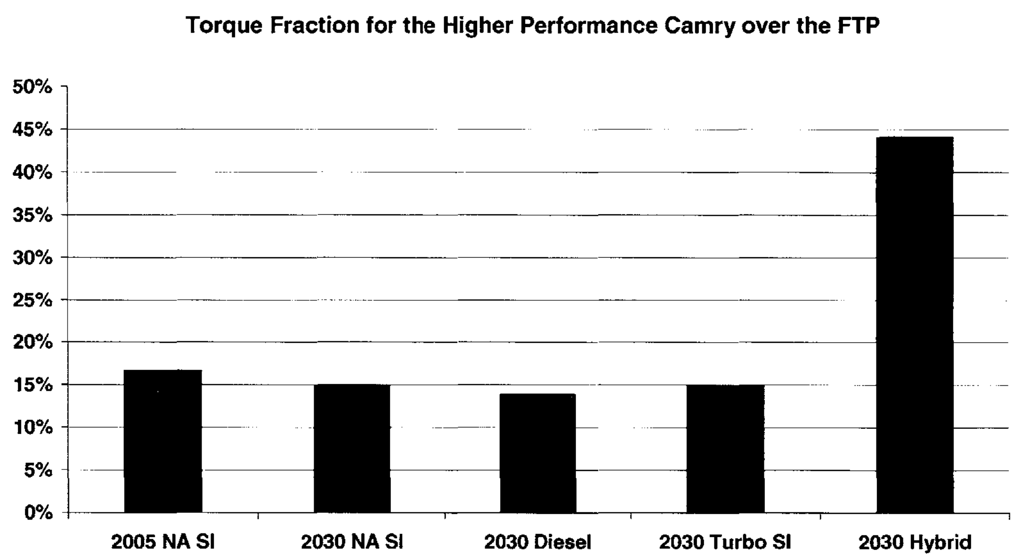


**Figure 109: Torque Fraction over the FTP for the 2005 3.0 liter Camry and its Future Equivalent Models Using Different Powertrains**

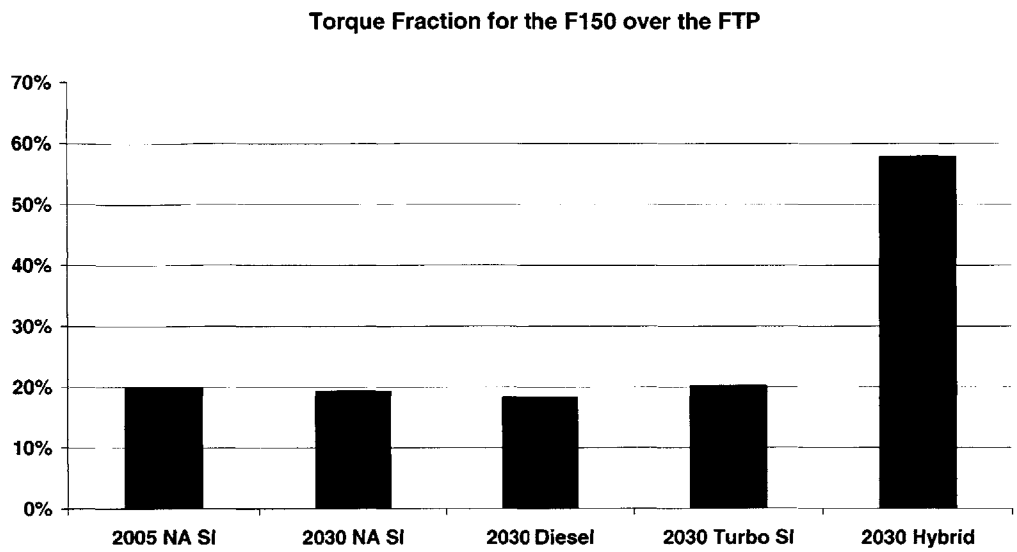


**Figure 110: Torque Fraction over the FTP for the 2005 4.2 liter F150 and its Future Equivalent Models Using Different Powertrains**

**Table 16: Energy Flows for the 2005 2.5 liter Camry and its Future Equivalent SI model for the FTP**

| | 2005 2.5 liter Camry | | | | 2030 Camry | | | | Current over Future Efficiency |
|---|---|---|---|---|---|---|---|---|---|
| | In (kJ) | Out (kJ) | Loss (kJ) | Ave. Eff. | In (kJ) | Out (kJ) | Loss (kJ) | Ave. Eff. | |
| Fuel | - | 49019 | | | - | 31248 | | | |
| Engine | 49019 | 9553 | 39466 | 19% | 31248 | 6883 | 24365 | 22% | 88% |
| Gearbox+ Clutch or Hydraulic Torque Converter+ Final Drive | 9553 | 8326 | 1227 | 87% | 6883 | 6246 | 637 | 91% | 96% |
| Wheel/Axle | 8326 | 7820 | 506 | 94% | 6246 | 5802 | 444 | 93% | 101% |
| Total Energy at the Wheel | 7820 | | 7820 | | 5802 | | 5802 | | |
| | Energy Required At the Wheel | | | | Energy Required At the Wheel | | | | Current over Future Efficiency |
| | | kJ | | % | | kJ | | % | |
| Aerodynamic | | 1898 | | 23% | | 1424 | | 18% | 75% |
| Rolling | | 2459 | | 30% | | 1341 | | 16% | 55% |
| Acceleration | | 3463 | | 43% | | 3037 | | 37% | 88% |
| Total | | 7820 | | | | 5802 | | | 74% |

Table 17: Energy Flows for the 2005 3.0 liter Camry and its Future Equivalent SI model For the FTP

| | 2005 3.0 liter Camry | | | | 2030 Camry | | | | (Current over Future) Efficiency |
|---|---|---|---|---|---|---|---|---|---|
| | In (kJ) | Out (kJ) | Loss (kJ) | Ave. Eff. | In (kJ) | Out (kJ) | Loss (kJ) | Ave. Eff. | |
| Fuel | - | 61379 | | | - | 37604 | | | |
| Engine | 61379 | 9931 | 51448 | 16% | 37604 | 7045 | 30559 | 19% | 86% |
| Gearbox+ Clutch or Hydraulic Torque Converter+ Final Drive | 9931 | 8668 | 1263 | 87% | 7045 | 6453 | 592 | 92% | 95% |
| Wheel/Axle | 8668 | 8135 | 533 | 94% | 6453 | 5999 | 454 | 93% | 101% |
| Total Energy at the Wheel | 8135 | | 8135 | | 5999 | | 5999 | | |
| | Energy Required At the Wheel | | | | Energy Required At the Wheel | | | | Future over Current Energy |
| | | kJ | | % | | kJ | | % | |
| Aerodynamic | | 1898 | | 23% | | 1424 | | 18% | 75% |
| Rolling | | 2533 | | 31% | | 1398 | | 17% | 55% |
| Acceleration | | 3704 | | 46% | | 3177 | | 39% | 86% |
| Total | | 8135 | | | | 5999 | | | 74% |

Figure 107 however only tells half the story. The relative improvement in combined fuel consumption for alternative powertrains over the 2030 naturally aspirated gasoline is presented in Figure 111. The relative improvement over the 2030 naturally aspirated gasoline powertrain is defined as:

$$RI_{2030} = \frac{FuelConsumption2030NASI - FuelConsumption2030otherpowertrain}{FuelConsumption2030NASI} \quad (6.6)$$

In this graph, it can be seen that:

- The differences between different powertrains become more pronounced. When comparing the relative reduction over 2005 vehicles the differences are divided by a large number, they appear therefore small. When divided by 2030 NA SI fuel consumption, the differences between different powertrains are emphasized.
- The diesel-gasoline gap is significantly narrower for the low performance Camry. The gap is wider for the 3.0 liter Camry and the F150.

The differences in Figure 111 can be explained if the following points are considered.

- The relative benefit of using a diesel instead of an SI engine depends on two things:
  - How downsized is the SI engine to be replaced with a diesel in terms of maximum bmep.
  - What is its average TF over a driving cycle?
- The lower performance Camry engine has both a high TF (~ 20% as presented) and a high maximum bmep (11.8bar in 2005, 11.8bar x 108% in 2030). The higher performance Camry has a slightly higher maximum bmep (12.55 bar in 2005, 12.55 x 108% in 2030) but a much lower average TF (15%). Although the F150 has a high TF, its maximum bmep is low (10.56bar in 2005, 10.56 bar x 108% in 2030).The improvement from using the diesel is therefore the least for the low performance Camry.
- For turbocharging a vehicle, the benefit is significantly more when the original engine is less downsized, especially since a constant 18bar bmep was assumed for all turbo SI models. The benefit for the F150 is therefore the most. At very low TF's however, the benefit of a turbocharged engine map compared with a naturally aspirated one begins to diminish. The benefit from turbocharging the high performance Camry therefore is the least.
- The differences in the benefit from hybridization rely mostly on the TF of the original NA SI powertrain to be hybridized. The main difference in relative fuel consumption improvement between different vehicles when being hybridized depends on how efficiently the original engine was operating. A vehicle with a lower TF in its NA SI version will have lower average engine efficiency and will thus benefit more from using a hybrid powertrain .The gain from eliminating idling is about the same for all vehicles. The gain from regeneration might be different in a relative sense between the Camry and the F150 but the differences are small. A hybrid, at least in the way it was defined in this study, manages to operate the engine at almost peak efficiency all the time. Maximum bmep is as a result not as relevant as TF. The most benefit from hybridization can therefore be realized for the high performance Camry because its NA SI version was operating at the lowest TF.

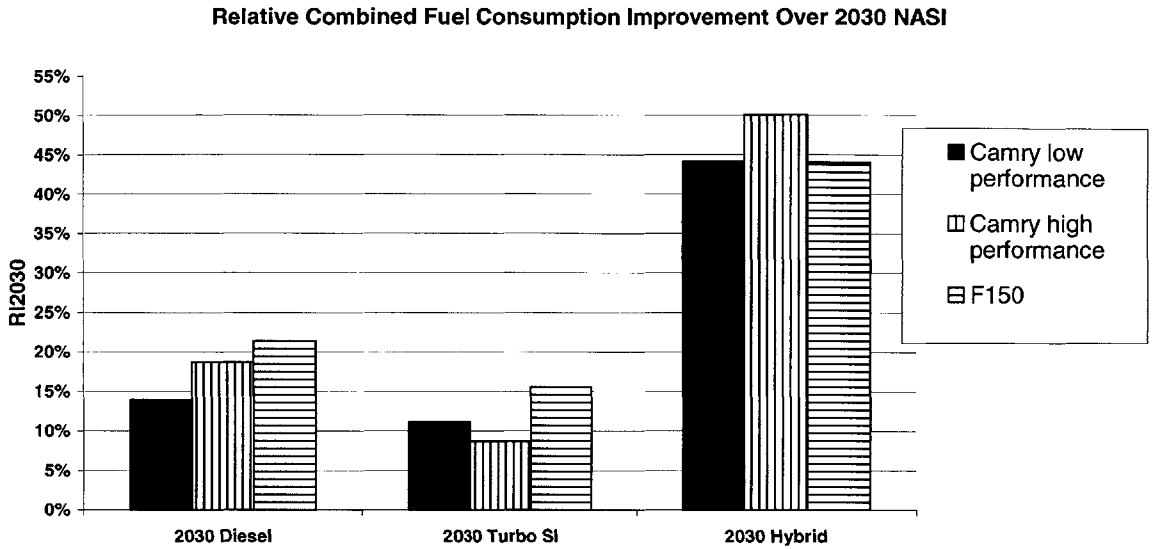


**Figure 111: Relative Improvement in Combined Fuel Consumption for Different Future Powertrains and Models over the 2030 Naturally Aspirated Gasoline**

## *6.5. The Effect of the Driving Cycle*

The effect of different driving cycles on fuel economy is significant. The fuel consumption of the 2005 2.5 liter Camry and its future equivalent models using different powertrains is presented in Figure 112 for a number of different driving cycles. The relative reduction in fuel consumption for different powertrains and vehicles is presented in Figure 113. The fuel consumptions presented are for the following cycles:

- standard city and highway (FTP,HWFET)-unadjusted
- Combined city and highway-unadjusted
- The US06 cycle which represents aggressive highway driving
- The average fuel consumption of FTP-unadjusted, HWFET-unadjusted and US06 called "industry cycle" in the graph which as described in [80] is representative of the driving cycles American car manufacturers use to calibrate their vehicles.

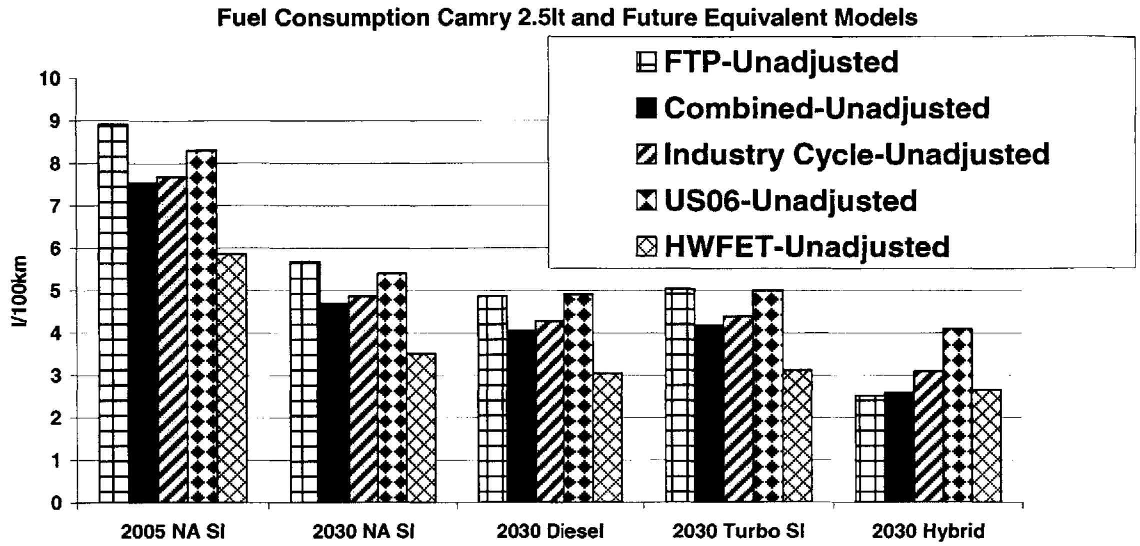


**Figure 112: Predicted Fuel Consumption over Different Drive Cycles for Different Powertrain Options for future models Equivalent to the 2005 2.5 liter Camry.**

The main conclusions from Figure 112,113 and 114 are:

- As expected, all non hybrid powertrains are significantly more fuel efficient on the HWFET cycle than on the FTP. The main reason for this is the lower engine efficiency during the FTP[8]. For the future NA SI equivalent to the 2.5 liter Camry, average engine efficiency in the FTP is 22% when in the HWFET it is 28%. This is can clearly be attributed to the higher average engine torque during the HWFET. TF for the same vehicle over the FTP is 20% and 22% over the HWFET. TF is an averaged number and a small change can cause large differences in average engine efficiency. The HWFET TF's are of course higher because due to the higher vehicle speeds, higher gear speeds (lower gear ratios) are being used.
- The current relative advantage of the diesel in the city doesn't exist in the future. The ratio of fuel consumption for the future diesel over the future NA SI is 86% in the FTP and 87% in the HWFET. This is due to the higher SI downsizing and friction reduction assumed.
- For the hybrid, the FTP and HWFET cycles are almost equivalent. This is mainly due to the active optimization of engine operating points in a hybrid. Average engine efficiency for the hybrid low performance Camry in the FTP is 33%; in the HWFET it is 36%. Furthermore, regenerative breaking almost eliminates the differences in energy required at the wheel.
- Fuel economy over the US06 is significantly worse than the HWFET cycles. This is interesting as the average speed of the two cycles is about the same- 77.6 km/h in the HWFET versus 77.2km/h in the US06. The maximum acceleration and maximum speed however in the US06 are much higher 3.76m/s$^2$ and 129km/h versus 1.43m/s$^2$ and 96km/h. This causes much higher braking losses.

[8] Additionally, the energy required at the wheel during the HWFET is lower than the FTP (278kJ/km versus 326kJ/km )

Additionally, aerodynamic losses in the US06 are higher. In total, the energy required at the wheel is about 486kJ/km for the US06 in the future Camry models versus 278kJ/km for the HWFET. Engine efficiency in the US06 is actually slightly higher due to the higher load-32% for the 2030 NA SI Camry versus 28% for the HWFET but not enough to make up for the difference in required energy.

- Looking at Figure 113 it is evident that the relative advantage of hybrids as presented in Figure 111 decreases significantly for highway(HWFET) and even more so for highway and aggressive driving (US06). This is evident when the HWFET and US06 fuel consumptions normalized by the 2030 NA SI are compared in Figure 114 and 115. This is primarily due to the fact that non hybrid powertrains are as explained, much more efficient over highway than over urban driving.
- The "industry cycle" due to the US06 contribution is slightly less fuel efficient than the unadjusted combined cycle. They are however close.

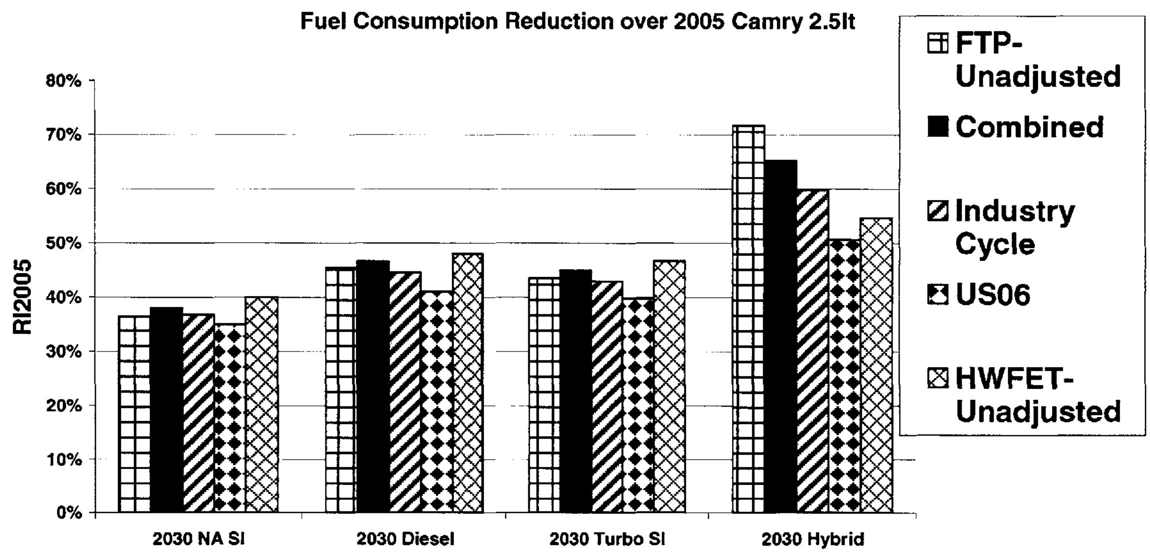


**Figure 113: Relative Fuel Consumption Improvement over the 2005 2.5 liter Camry for Different Drive Cycles and alternative Powertrain Options**

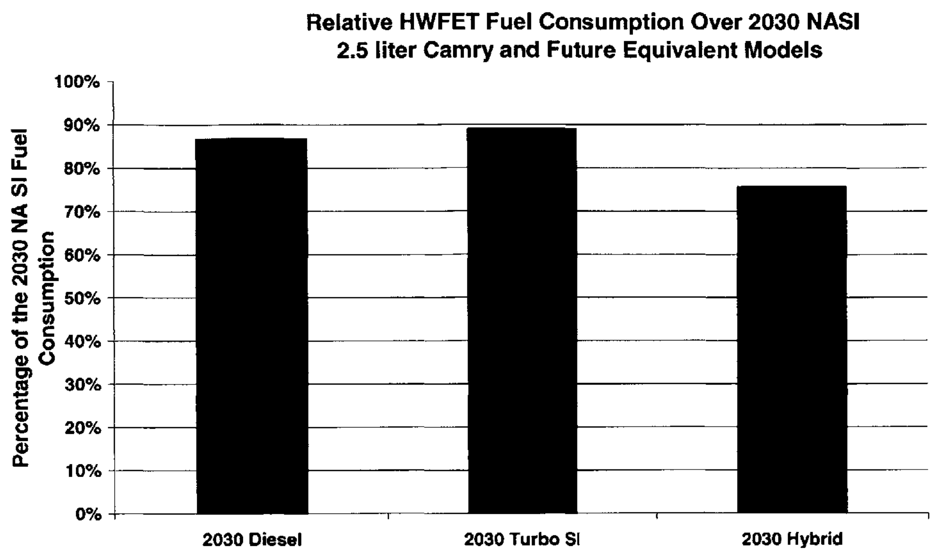


**Figure 114: Relative HWFET Fuel Consumption over the 2030 Naturally Aspirated SI.**

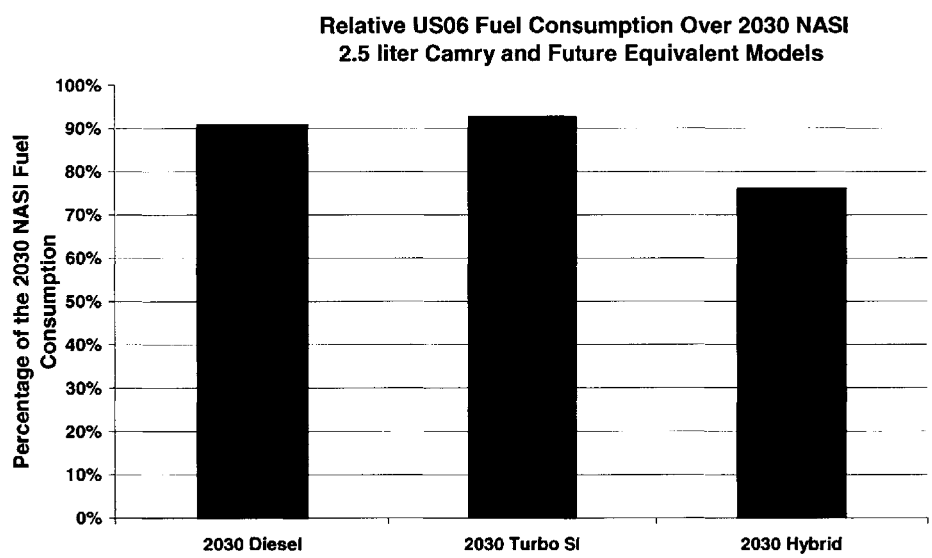


**Figure 115: Relative US06 Fuel Consumption over the 2030 Naturally Aspirated SI.**

In a relative sense, the fuel consumption advantage of hybrids over non hybrid powertrains is even lower over the US06 cycle than over the HWFET cycle as seen in Figure 114 and 115. In order to find an explanation for this result, the energy flows for both driving cycles need to be examined. The energy flows for the lower performance Camry Hybrid are presented in Tables 18 and 19 for the HWFET and US06 cycles. The total energy required is higher in the US06. This however wouldn't decrease the *relative* advantage of hybrids over conventional powertrains if all the efficiencies were the same.

The ratio of average component efficiencies during the HWFET over average efficiencies over the US06 is also presented in Table 19. All of the efficiencies are almost the same except that of the regenerative energy recovered. More energy is lost to friction braking in the US06, although the regenerative braking system was assumed to capture 90% at all speeds. The reason is that the power of the electric system is not enough in the US06 as the rates of deceleration are much higher. This is what causes the decrease in the relative advantage of hybrids versus non hybrids over the US06 versus over the HWFET. As it was seen in chapter 5, the size of the electric system required to capture all of the regenerative energy in the US06 would be very large.

**Table 18: Energy Flows For the 2030 Hybrid Equivalent to the 2005 2.5 liter Camry over the HWFET**

| | 2030 20% HR Hybrid Camry HWFET | | | | | |
|---|---|---|---|---|---|---|
| | In (kJ) | Out (kJ) | Out (kJ) | Loss (kJ) | Stored (kJ) | Ave. Eff. |
| Fuel | - | 14220 | | | | |
| Engine | 14220 | 5111 | | 9109 | | 36% |
| Clutch | 5111 | 5104 | | 7 | | 100% |
| | | From Engine to Gearbox | From Engine to Motor | | | |
| | | 4286 | 163 | | | |
| Motor as generator | 818 | 724 | | -94 | | 89% |
| Battery | 1112 | 1159 | | -83 | -130 | 93% |
| Motor | 1159 | 1055 | | -104 | | 91% |
| Gearbox+ Final Drive | 5341 | 4960 | | 381 | | 93% |
| Wheel/Axle | 4960 | 4600 | | 360 | | 93% |
| Total Energy at the Wheel | 4600 | | | 4600 | | |
| | Regeneration path | | | | | |
| | In (kJ) | Out (kJ) | | Loss (kJ) | | |
| Total Deceleration | - | 663 | | | | - |
| Accel. Recovered (Regen) | 663 | 472 | | 191 (to friction braking) | | 71% |
| Gearbox | 472 | 436 | | 36 | | 92% |
| Motor as Generator | 436 | 388 | | 1257 | | 89% |
| | Energy Required At the Wheel | | | | | |
| | | kJ | | % | | |
| Aerodynamic | | 2678 | | 58% | | |
| Rolling | | 1259 | | 27% | | |
| Acceleration | | 663 | | 14% | | |
| Total | | 4600 | | | | |

**Table 19: Energy Flows For the 2030 Hybrid Equivalent to the 2005 2.5 liter Camry over the US06 as well as their relative magnitude to those over the HWFET**

| 2030 20% HR Hybrid Camry US06 | | | | | | HWFET/ US06 Efficiency |
|---|---|---|---|---|---|---|
| In (kJ) | Out (kJ) | Out (kJ) | Loss (kJ) | Stored (kJ) | Ave. Eff. | |
| - | 17050 | | | | | |
| 17050 | 6167 | | 10883 | | 36% | 99% |
| 6167 | 6130 | | 37 | | 99% | 100% |
| | From Engine to Gearbox | From Engine to Motor | | | | |
| | 5967 | 163 | | | | |
| 163 | 142 | | -21 | | 87% | 102% |
| 1235 | 1219 | | -120 | -104 | 91% | 103% |
| 1219 | 1096 | | -123 | | 90% | 101% |
| 7063 | 6696 | | 367 | | 95% | 98% |
| 6696 | 6303 | | 393 | | 94% | 99% |
| 6303 | | | 6303 | | | |
| **Regeneration path** | | | | | | **HWFET/ US06 Efficiency** |
| In (kJ) | Out (kJ) | | Loss (kJ) | | Ave. Eff. | |
| - | 2188 | | | | | |
| 2188 | 1322 | | 866(to friction braking) | | 60% | 118% |
| 1322 | 1257 | | 1257 | | 95% | 97% |
| 1257 | 1094 | | 1257 | | 87% | |
| **Energy Required At the Wheel** | | | | | | **HWFET/ US06 Energy** |
| | kJ | | % | | | |
| | 3133 | | 68% | | | 117% |
| | 982 | | 21% | | | 78% |
| | 2188 | | 48% | | | 330% |
| | 6303 | | | | | 137% |

## 6.6. Sensitivity to Engine Map Assumptions

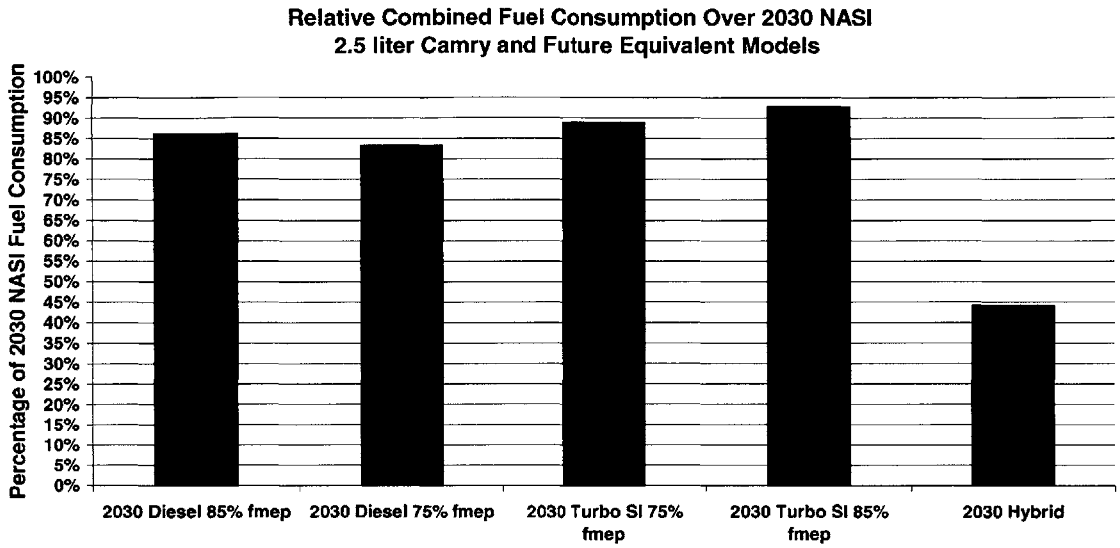


**Figure 116: Relative Combined Fuel Consumption over the 2030 Naturally Aspirated SI. Sensitivity to assumptions about reduction of fmep for the turbo and Diesel**

The sensitivity of the turbo SI and the diesel to the assumptions made is not that significant on a normalized by 2030 NA SI fuel consumption basis as seen in Figure 116. This further reinforces the claim that the diesel-gasoline gap for the low performance Camry narrows because of the effect of downsizing and not as much because of the different friction reduction assumptions used for diesel and gasoline.

## 6.7. Sensitivity to CVT Efficiency and to Shift Window

Different transmission optimization options are presented in Figure 117 for the current 2.5 liter Camry and in Figure 118 for its 2030 NA SI counterpart. Six speed manual/automatic gearboxes using the standard and narrow shift windows defined in chapter five are compared in terms of adjusted, combined fuel consumption with a current CVT with a nominal efficiency of 75% and 87%.

The CVT at 75% peak efficiency yields an 8% benefit compared with the automatic for the 2005 Camry. The CVT equipped vehicle is equivalent to the automatic on the HWFET but better on the FTP. Looking at the energy flows, the average engine efficiency with the standard automatic is 19% over the FTP, 25% over the HWFET. The automatic transmission efficiency is 87%/89% (urban/highway) leading to a total driveline efficiency of 16.5% for the FTP and 22.2% for the highway. With the CVT, engine efficiencies are 27% (urban) and 30% (highway). For a CVT efficiency of 75%, the respective total driveline efficiencies are 20% and 22.2% .It should be noted however that the 75% CVT efficiency used is the upper limit of what exists currently on the market. In reality, the fuel economy achieved by CVT powered vehicles is no better than those of automatic transmissions. This, in addition to the high torque limitation of CVT's and their higher cost has prevented their wide adoption. However, having a CVT with 87% efficiency today would lead to 17% savings in fuel consumption, without any other vehicle improvements.

The benefits in fuel consumption from using a CVT in future vehicles are of equivalent magnitude. About 7% for the 75% efficient CVT and close to 20% for the 87% one compared with the future manual/automatic transmission.

It is also noteworthy that use of a very narrow shift window would bring the manual/automatic powertrain to the level of an 87% CVT. This result is interesting not because a shift window as narrow as the one used here is realistic, but because it provides an upper limit.

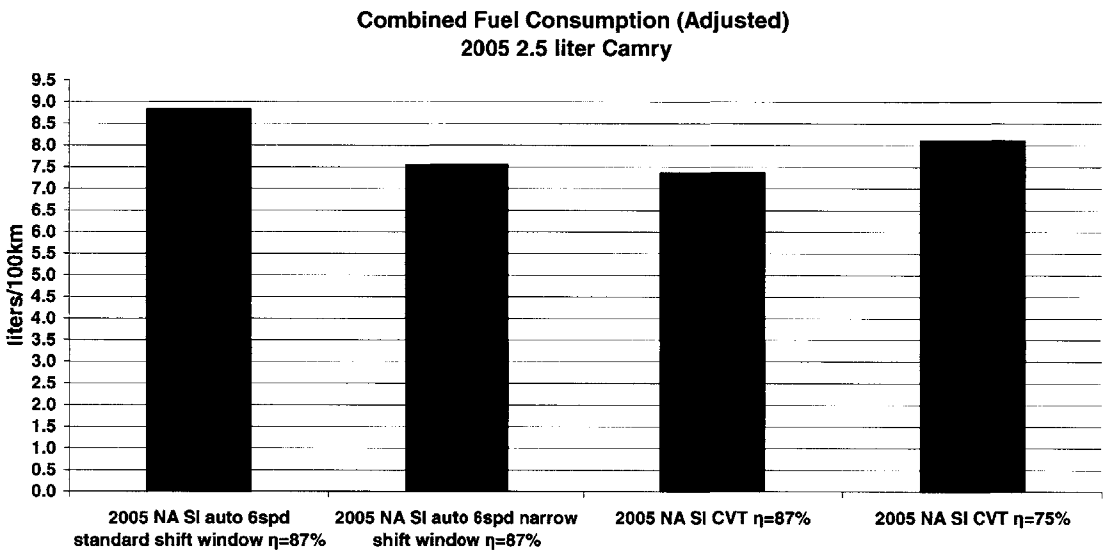


**Figure 117: 2005 2.5 liter Camry Transmission Optimization Options**

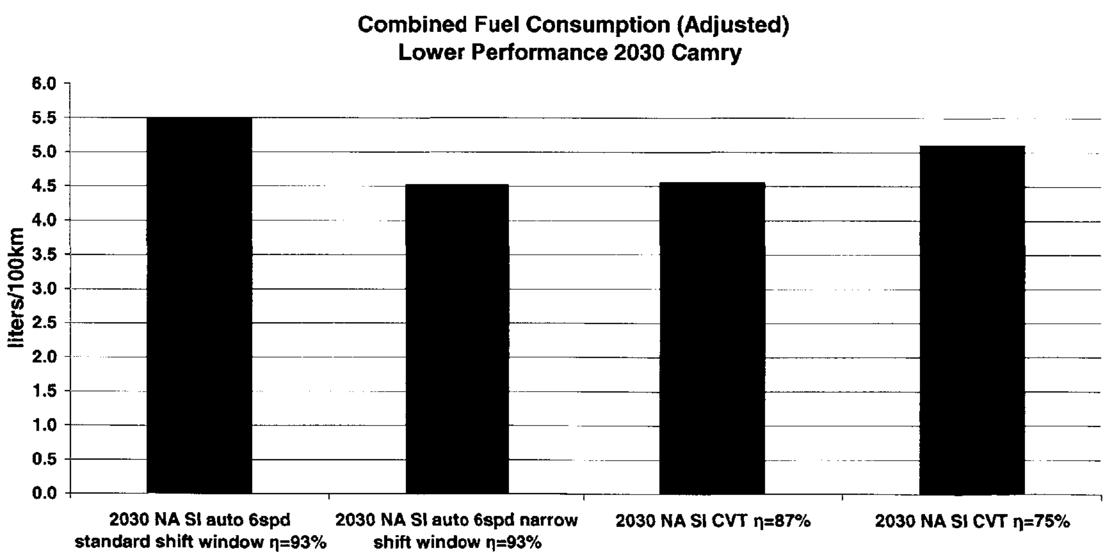


**Figure 118: 2030 Low Performance Camry Transmission Optimization Options**

## 6.8. Sensitivity of Hybrids

It was mentioned before that a hybridization ratio HR of 20% was chosen for the final hybrid results. To explain this choice, the sensitivity of combined adjusted fuel consumption on hybridization ratio is presented in Figure 119. Three different HR ratios were investigated, 10%, 20% and 30%. As explained in chapter five, these are indicative of the hybrid ratios on hybrid models currently on the market. The vehicle platform for the comparison is the future low performance Camry.

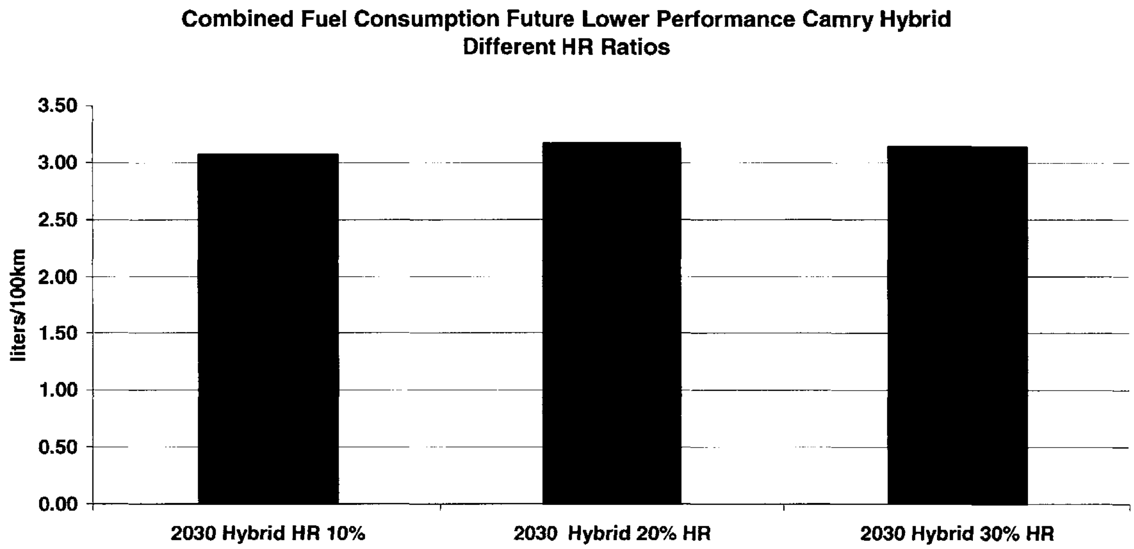


**Figure 119: Sensitivity of Combined Fuel Consumption on Hybridization Ratio HR for the low performance Camry hybrids.**

It is clear from Figure 119, that combined fuel economy is almost independent of the relative size of the electric system. The fuel consumption numbers differ only in a second decimal level. The reasons are:

- The power requirement during the FTP and HWFET cycles is small enough that a 10% hybridization ratio provides enough motor power to optimize engine operation
- Deceleration rates during the FTP and HWFET are low enough that a 10% HR does not significantly limit the amount of kinetic energy can be captured through regeneration.

The engine operating points over the FTP for the 10% HR hybrid low performance Camry are presented in Figure 120. The engine almost never falls under the 40% off-torque limit. A 10% HR ratio results in this case in a 9kW motor which is enough to optimize engine operation. The energy efficiency of the engine in both the 10% HR model and 20% HR models is 33%

Note that the operating points presented in Figure 120 are for a hot start cycle. The electric system won't operate[9] before the engine is heated up as seen for a cold start FTP cycle in Figure 121. This is a strategy applied in all current hybrids so that the engine heats up faster to reduce emissions. It takes the engine about 400 s to heat up.

[9] Except for regeneration.

The regeneration energy captured over the FTP is not effectively limited by the electric system. As already explained in chapter 5, most of the regeneration energy over the FTP is distributed in low powers. To validate that result, the regeneration energy *lost* to friction braking over the FTP is 1085 kJ for the 10% HR hybrid, 694 kJ for the 20% hybrid and 662 kJ for the 30% HR. The total kinetic energy that could be captured (friction + regenerative braking) is about 3000kJ. A 20% HR therefore is enough to capture almost all the regeneration energy that could be captured using the braking system assumptions specified in chapter five.

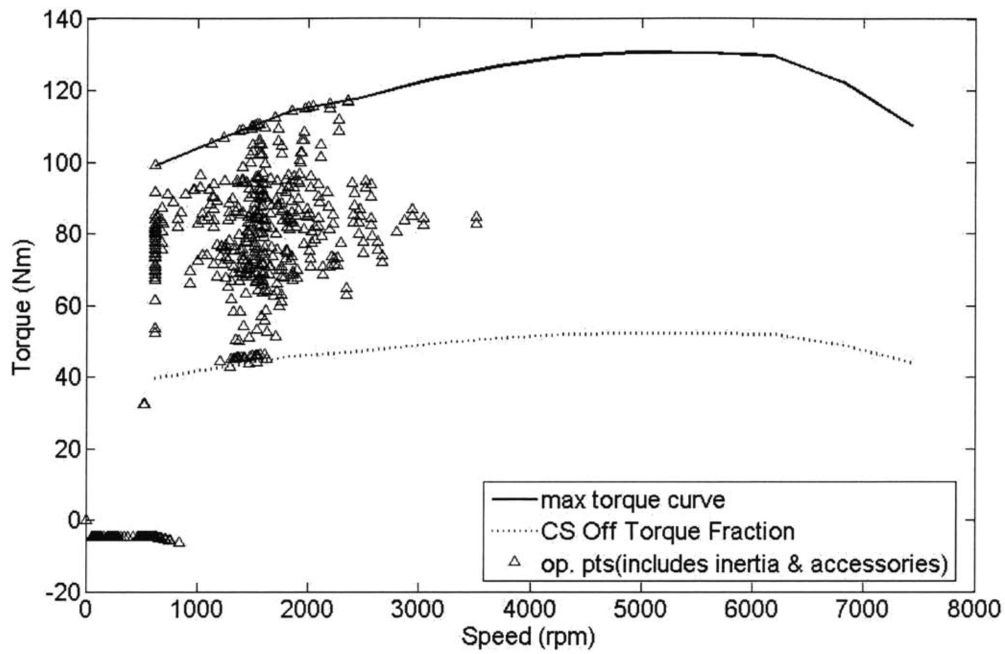


**Figure 120: Engine operating points over the FTP for the 10% HR hybrid low performance Camry. The test was performed under hot start conditions**

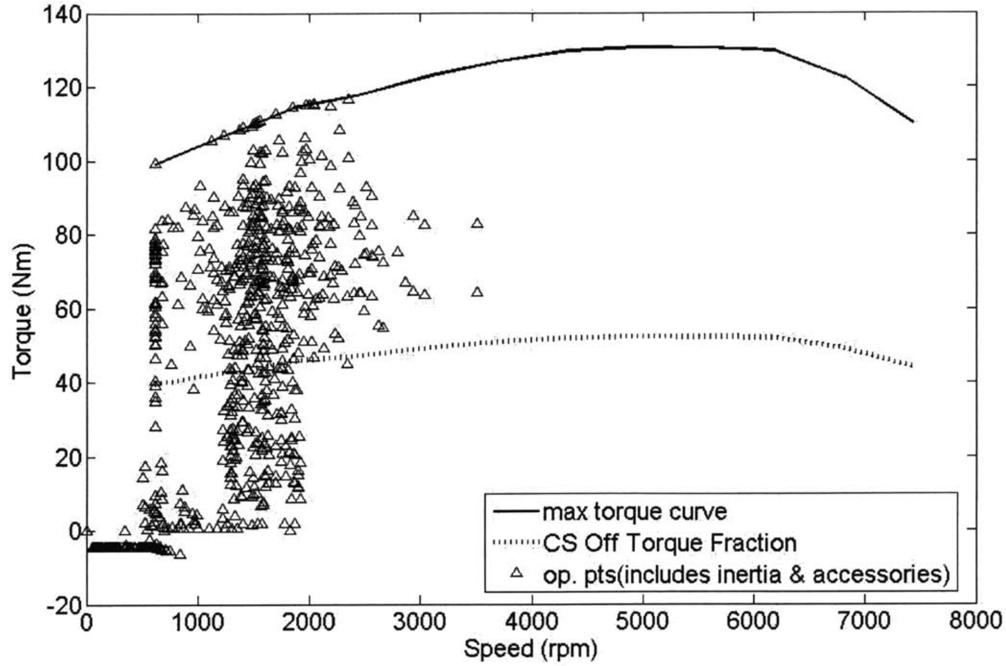


**Figure 121: Engine operating points over the FTP for the 10% HR hybrid low performance Camry. The test was performed under cold start conditions**

The 30% HR hybrid is essentially equivalent with the 20% HR over the combined cycle. In fact, the 30% HR hybrid is slightly worse in the city and slightly better on the highway. Regeneration is equivalent for both vehicles. However, in the city, the engine in the 30% HR hybrid is operating at an average efficiency of 31% versus 33% for the engine in the 20% hybrid. On the highway, the 30% HR hybrid is operating at an engine efficiency of 36.5% when the 20% HR is operating at 35.5% engine efficiency.

It would normally be expected that a higher hybridization ratio would lead to higher engine efficiencies. However, for city driving, this doesn't seem to be the case. Engine efficiency remains almost constant and even falls slightly .This is because the 20% HR hybrid already has enough motor power to perform engine optimization. Additionally, as mentioned in chapter five, higher HR vehicles require slightly higher gear ratios to achieve the same acceleration for the same total power per unit curb weight because they rely on the motor more. As a result, the TF over the FTP actually falls slightly with H. This is seen in Figure 122. Therefore so does the engine efficiency. Over the HWFET, although the gear ratios for the 30% HR hybrid are larger, it spends more time using the $4^{th}$ gear compared with the 20% HR because its faster, more downsized engine upshifts faster. The final conclusion is that in terms of combined fuel economy, the hybridization ratio doesn't play a big role.

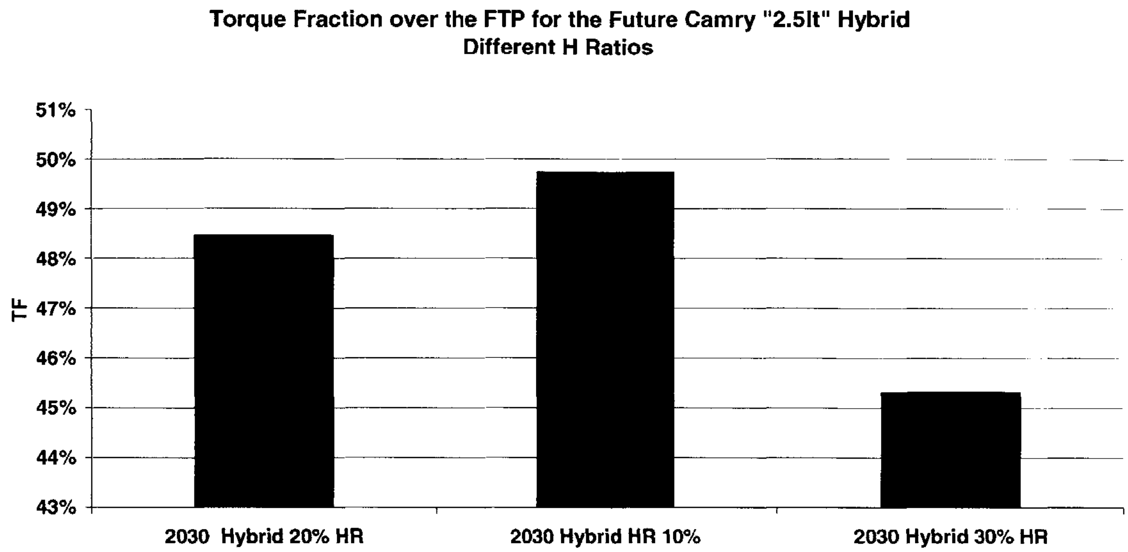


**Figure 122: Torque Fraction TF over the FTP for different HR for hybrid versions of the 2.5 liter Camry**

On the US06 however, the picture is slightly different as seen in Figure 123. There is about a 5% gain when going from a 10% HR to a 20% HR and about 2.5% more when going to a 30% HR. Engine efficiencies are basically the same (36-37%) for all HR due to the high power requirements of this cycle. What varies significantly in this case is the amount of regeneration. Due to the high deceleration powers, the 20% HR hybrid doesn't capture all of the regeneration energy. The energy loss because of friction breaking is 866kJ. For the 30% HR, due to the larger electric system, that number drops by 200kJ.

This saved energy (200kJ) multiplied by 95% x 87% x 90% x 87% x 95%to account for gearbox (twice), generator, battery and motor efficiencies, would save 123kJ at the wheel. To generate this energy at an engine and transmission efficiency of 36% and 93% respectively, 367kJ of fuel energy more would be needed which is exactly the gain in fuel consumption. The differences in US06 fuel economy for different hybridization ratios can therefore clearly be attributed to differences in the captured regeneration power.

The hybridization ratio (HR) of choice for the results presented in this study was 20%. This choice was made on a cost to benefit ratio basis. It was explained that the hybridization ratio in the range of 10-30% doesn't significantly affect combined fuel economy. In terms of US06 driving, going from a 10% HR to 20% offers a 5% benefit in fuel consumption whereas going to 30% HR only offers another 2.5% benefit. Since as explained in chapter 5, the cost premium of a hybrid scales mainly with battery size, both of these changes would cost about the same while the second one only offers half the benefit. It was therefore considered more realistic to choose 20% as the most probable hybridization ratio. It seems that the assumption of a parallel architecture with engine optimizing capabilities as well as the improvement in efficiencies of the electrical subsystem enable lower HR's than the ones currently used in power split architectures.

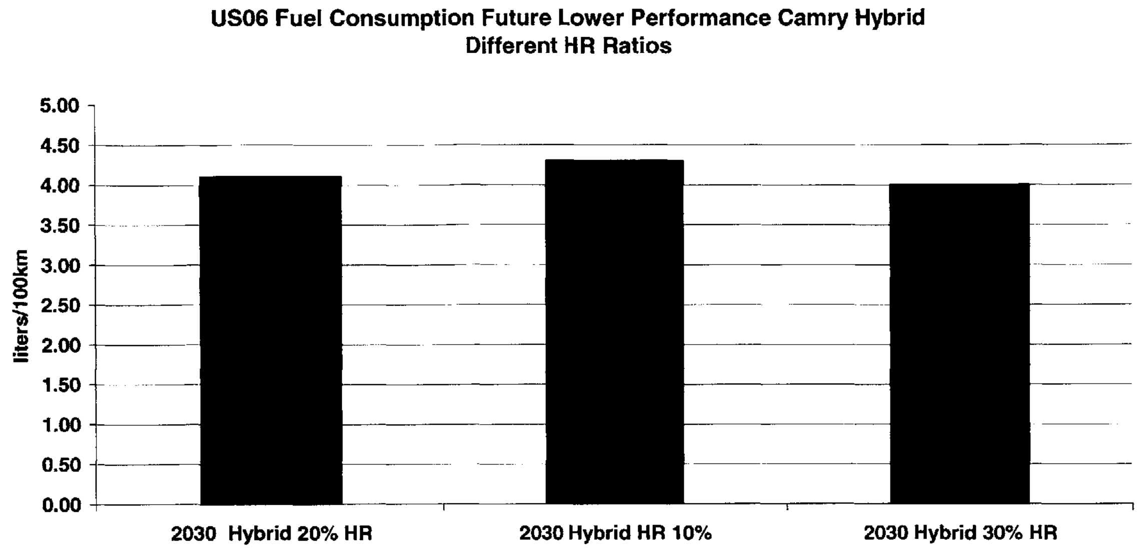


**Figure 123: Sensitivity of US06 Fuel Consumption on Hybridization Ratio H for the low performance Camry hybrids.**

Different transmission optimization options for the 20% HR future low performance Camry hybrid are compared in Figure 124 on the basis of their adjusted combined fuel consumption. The best option is as already discussed is a manual/automatic transmission that offers active engine optimization. Without this optimization ability, the hybrid with manual/automatic transmission has about 20% higher fuel consumption. It should be noted that this vehicle still has a TF of 32% and an engine efficiency of 26% over the FTP which is considerably higher than the 20% TF and 22% engine efficiency for the non-hybrid NA SI. This is due to the smaller engine in the hybrid. The use of a CVT gearbox in the hybrid can optimize the engine even further-83% TF and 38% engine efficiency for the FTP! For the manual/automatic transmission with active optimization, engine efficiency was 33%. For the HWFET, however the gain from further optimizing engine operation is smaller than the loss from a less efficient transmission. Engine efficiencies are 36% for the manual/automatic transmission with optimization, 33% for the conventional manual/automatic and 39% for the CVT's (equal to the engine's peak efficiency!). The end result is that the efficient CVT is in terms of fuel consumption at the levels of the manual/automatic transmission without active engine optimization.

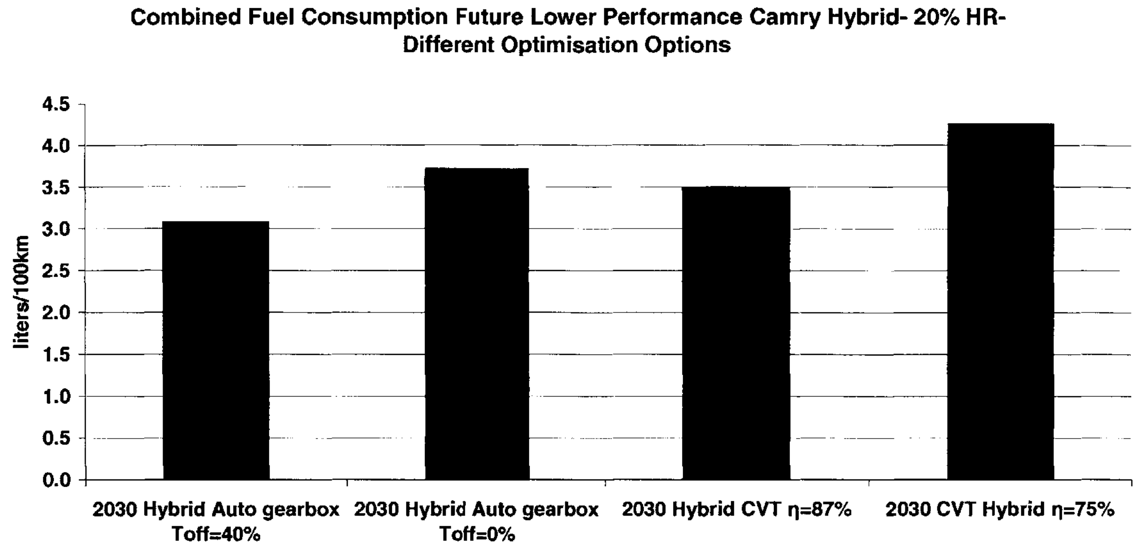


**Figure 124: Adjusted Combined Fuel Economy for Different Transmission Optimization Options for the 20% HR low performance Camry hybrid**

## *6.9. Comparison with the Previous Study.*

In order to compare this study's results with those in the previous study [8], the right metric to use is a relative improvement in fuel consumption over the current or the future SI baseline. Absolute numbers in studies that attempt to look into the future are of less importance than trends. The relative improvement in combined fuel consumption over the current SI baseline for the two studies is presented in Figure 125. "Current" in the context of this study is the 2005 Camry while in the context of [8] it means 2001. Future for this study is 2030; 25 years into the future while for [8] it was about 20 years into the future, i.e. 2020. Also note that as there were two different levels of improvement for the future SI in [8] the comparisons in Figure 125 are using the "advanced" SI numbers from that study as the assumptions in that model were almost the same as the ones used in this thesis.

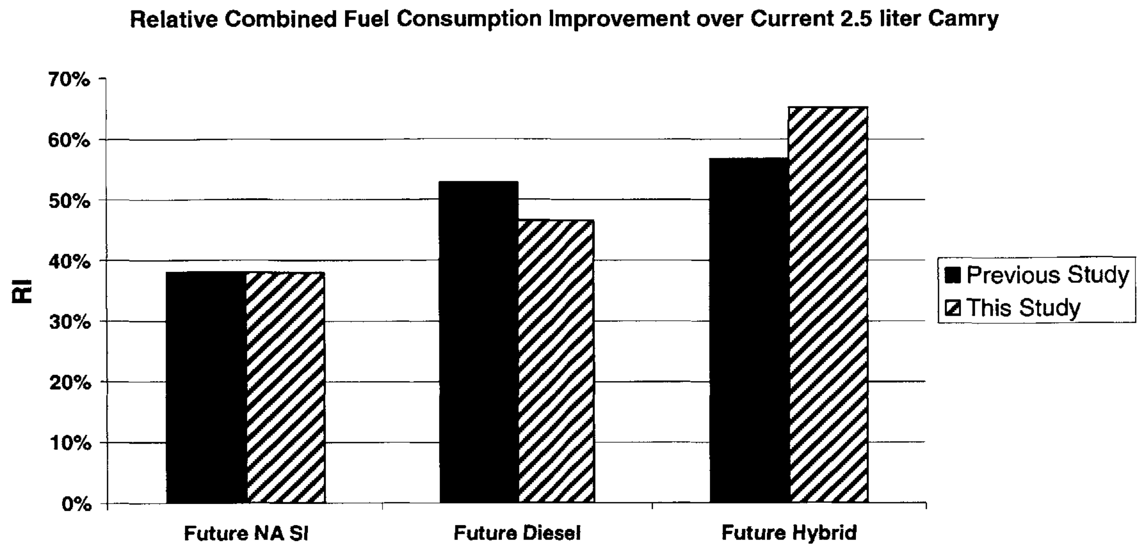


**Figure 125: Comparison of the Results in this Study with "Comparative Assessment of Fuel Cell Cars"-Relative Improvement of Combined Fuel Economy compared with the current 2.5 liter Camry**

The main conclusions from Figure 125 are:

- The relative reduction of fuel consumption by future SI engines is the same
- The relative reduction of fuel consumption by future diesel engines was calculated to be more in the previous study compared with this one.
- The relative reduction of fuel consumption by hybrid SI vehicles was calculated to be less in the previous study compared with this one.

However, it should be noted that although most of the assumptions in [8] were the same as the ones in this study, there were some differences. Some of these differences result in worse fuel consumption, some in better. The combined effect is that there aren't major differences in the end result. More specifically:

- The engines in [8] were improved by the same percentage in $\eta_{i,\ fmep}$ and maximum bmep. The original current SI engine in [8] had a maximum bmep of about 11. In this study it ranged from 10bar (F150) to 12.5 bar.
- A constant fmep value was used throughout the engine map in [8]. In this study instead fmep was a function of rpm. The average fmep for current gasoline was 172 kPa in this study, 165kPa in [8]. The future average fmep in [8] was 124 kPa (a 25% decrease). In this study however, the future average fmep was 147 kPa which is a 15% decrease. The actual fmep as a function of rpm is whatever came out of the friction model for future bore and stroke decreased by 25%. The future engine is however, faster and smaller, which increases fmep, hence the decrease by only 15%. If the future engine was assumed to have the same redline as the current one, the reduction in average fmep would be the same.

- Gear ratios were not optimized in [8] as in this study. The same gear ratios were used for all powertrains while the final drive ratio was roughly adjusted.
- Transmission efficiency was 88% in [8] versus 91-93% in this study .The efficiency in [8] however also included the losses in the clutch or the hydraulic torque converter and wheels, so the two studies are roughly the same with this one being slightly worse.
- Weight reduction in [8] was 22% including passenger weight. In this study it was 20% without including it.
- The reduction in rolling friction was the same.
- The reduction in $c_D$ was the same. However, in [8] the frontal area was also reduced by 10% whereas in this study it was kept constant.
- In this study there was no accessory load except those needed for engine operation. In [8] a 700W load was assumed for current and a 1kW for future vehicles.

If all the differences in assumptions except those for the engine and gear ratios are eliminated in the models used in this study, the final improvement in combined fuel consumption for future over current NA SI is 37.6%, which is less than 1% different than [8]. The difference in fmep decrease is canceled out by the effect of the arbitrary versus optimized gearbox. For the future NA SI engine over the FTP cycle in [8] the average TF is only 8% and over the HWFET it's only 13%.

For the diesel, the difference in relative improvement is significant for the 2.5 liter Camry but it disappears for the 3.0 liter Camry or the F150 (Figure 107). The reason the diesel looks better in [8] is a combination of 2 things:

- The fmep for the current diesel in [8] was not calculated from an actual engine map as is this study. It was back calculated to give a reasonable fuel consumption number for the current diesel. Given the less accurate fmep model used, the result was significantly lower (180kPa) than the average fmep calculated from the diesel map in this study (217kPa). The engine map in [8] is consequently much better at lower loads than the one in this study.
- The comparison with the SI engines was more favorable for the diesel in [8] because all the engines were operating at low average loads due to the gearboxes being selected less carefully. Note that for the diesel in [8], TF =12.3% over the HWFET, 8% over the FTP. Also note that when the comparison between the two studies is performed for the 3.0 liter Camry, the relative improvement in the two studies is almost the same. This is because the 3.0 liter Camry has a lower TF than the 2.5 liter. The relative improvement for diesel is also almost the same in the two studies if the F150 numbers in this study are used. This is because the engine in the F150 has a lower maximum bmep than the one in the Camry.

Hybrids in [8] were modeled assuming a very efficient CVT gearbox (88%) in a parallel only architecture. A power threshold was used below which the vehicle goes all electric. This limit was low (only 2kW) for a 58kW engine, but in a CVT architecture, engine operating points are already optimized. This limit doesn't therefore affect fuel consumption that much. The last difference is that perfect regeneration was assumed in [8] whereas in this study it is almost perfect. The hybrid model used in [8] is roughly equivalent to the efficient (87%) CVT model in this study. If the combined fuel consumption number corresponding to the efficient CVT hybrid model is used in this study the relative improvement drops to 60% which is very close to the relative improvement for the hybrid in [8]. The results therefore would roughly agree. In this study, a parallel architecture with a manual/automatic transmission and engine optimization ability was assumed. This is what makes the hybrids look better in this study However, even if a power split architecture is used in the future instead of what assumed in this study, fuel consumption would probably still be better than a parallel architecture with an efficient CVT.

## 6.10. Well to Tank results

In terms of reduction of petroleum demand, the relevant metric to compare different powertrains is the tank to wheels fuel consumption which is the focus of this thesis. In terms of greenhouse gas (GHG) emissions and total required energy, however, the right metrics are the well to wheels or even better the cradle to grave numbers. Including the fuel side is of some value because there is a difference in the production energy and GHG emissions of gasoline and diesel. For this thesis the cradle to grave numbers were not estimated as no discrete assumptions were made as to what the materials breakdown of the vehicles were.

The comparison for well to wheels versus tank to wheels energy is presented in Figure 126, for GHG emissions in Figure 127 and the assumptions used in Table 20. The well to wheels results are also summarized in Table 20. The assumptions used were adapted from [8] except for the lower heating values which were the default ones used in ADVISOR. The main conclusion from Figure 126 is that the diesel engine looks better when the fuel side is included. The energy required to produce diesel fuel per unit energy in the fuel is significantly less than that of gasoline. In terms of tank to wheels GHG emissions, the diesel looks slightly worse than the tank to wheel energy results as diesel fuel emits slightly more carbon dioxide per unit energy in the fuel burned. If the fuel side is considered as well however, the diesel looks better in terms of GHG emissions as its fuel side emits significantly less GHG.

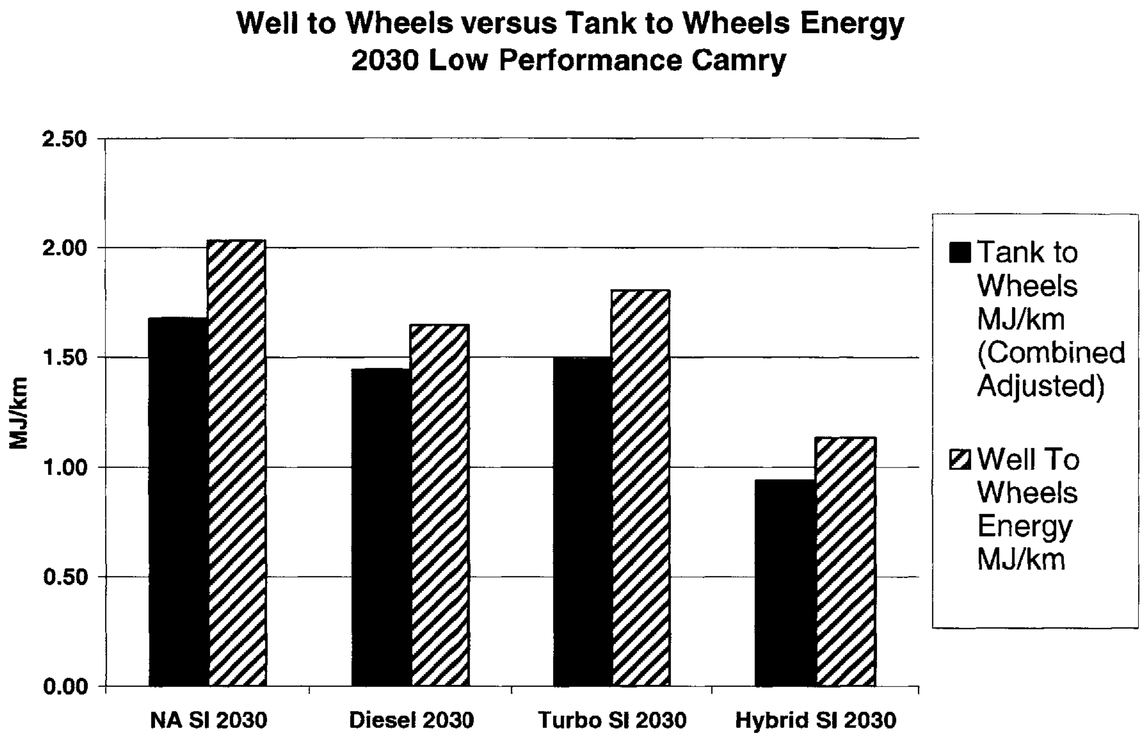


**Figure 126: Well to Wheels versus Tank to Wheels Energy Demand for Different Powertrains in the low Performance Camry.**

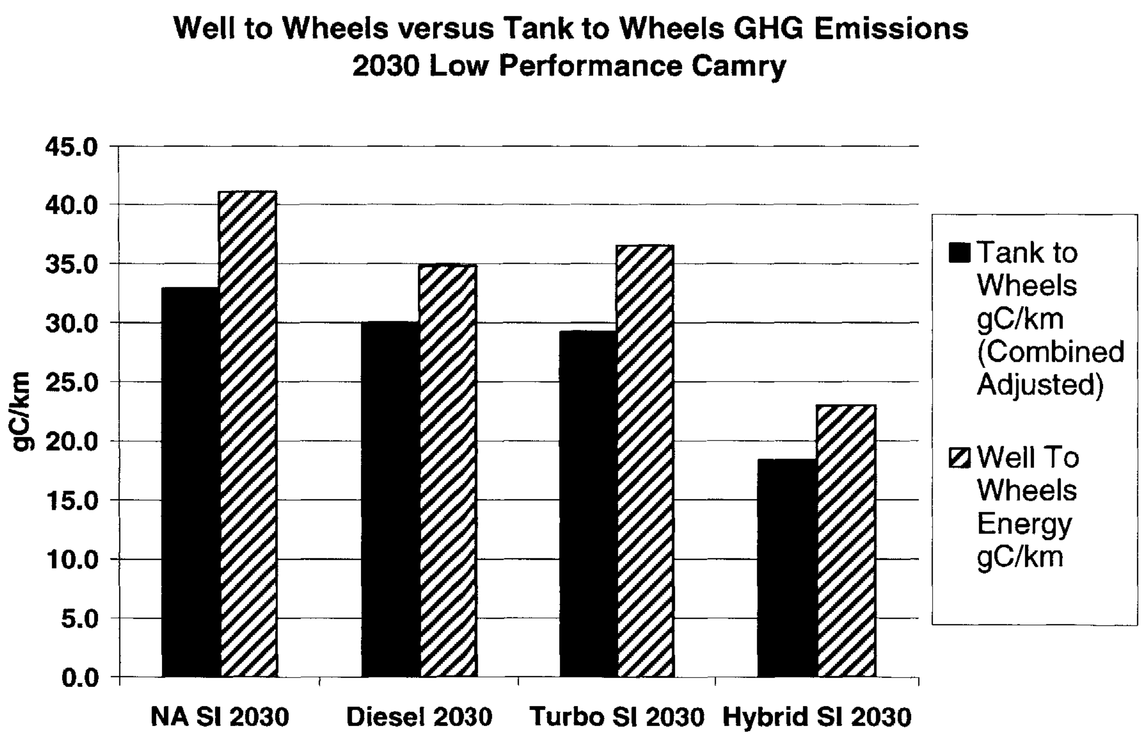


**Figure 127: Well to Wheels versus Tank to Wheels GHG emissions for Different Powertrains in the low Performance Camry.**

Table 20: Assumptions for Wheel to Wheels Calculations and Results

| | Tank to Wheels l/100km Combined Adjusted Gasoline Equivalent | Tank to Wheels MJ/km (Combined Adjusted) | Well to Tank Energy (MJ/km) | Well To Wheels Energy MJ/km | Tank to Wheels gC/km (Combined Adjusted) | Well to Tank gC/km | Well To Wheels Energy gC/km |
|---|---|---|---|---|---|---|---|
| NA SI 2030 | 5.49 | 1.68 | 0.35 | 2.03 | 32.9 | 8.2 | 41.1 |
| Diesel 2030 | 4.725 | 1.44 | 0.20 | 1.65 | 30.0 | 4.8 | 34.8 |
| Turbo SI 2030 | 4.88 | 1.49 | 0.31 | 1.80 | 29.2 | 7.3 | 36.5 |
| Hybrid SI 2030 | 3.07 | 0.94 | 0.20 | 1.14 | 18.4 | 4.6 | 23.0 |
| | | | | | | | |
| **Assumptions** | | | | | | | |
| Gasoline LHV (MJ/kg) | Well to Wheels MJ/MJ fuel in tank-Gasoline | Tank to Wheels Gasoline gC/MJ in tank | Well to Tank Gasoline gC/MJ in tank | | | | |
| 41.472 | 0.21 | 19.6 | 4.9 | | | | |
| Gasoline ρ (kg/lt) | Well to Wheels MJ/MJ fuel in tank-Diesel | Tank to Wheels Diesel gC/MJ in tank | Well to Tank Diesel gC/MJ in tank | | | | |
| 0.737 | 0.14 | 20.8 | 3.3 | | | | |
| Gasoline LHV(MJ/lt) | | | | | | | |
| 30.565 | | | | | | | |

## 6.11. Other Performance Results

Maximum speed in miles per hour is presented in Figure 128 for all powertrains and models. A rough scaling analysis will indicate that top speed depends on the ratio of maximum power over aerodynamic resistances. Additionally, there is the effect of the gearbox as explained in chapter four. In this case, since the power over weight ratio was kept roughly constant, maximum power decreased by roughly 20% for non-hybrid powertrains. Aerodynamic resistance decreased by 33%. It should therefore be expected that maximum speed would increase. This increase however is not large. Let's assume that the gearbox is designed for maximum speed so that the intersection of the total resistances curve and the maximum torque curve described in chapter 4 occurs at maximum engine speed:

$$P_{eng} = P_{aero} = \frac{1}{2}\rho * u^3 * c_d,$$
$$P'_{eng} = P'_{aero} = \frac{1}{2}\rho * u'^3 * c'_d \Rightarrow \tag{6.6}$$
$$\frac{P'_{eng}}{P_{eng}} = \frac{P'_{aero}}{P_{aero}} = \frac{u'^3}{u^3} * \frac{c'_d}{c_d} \Rightarrow 0.8 = \frac{u'^3}{u^3} * 0.66 \Rightarrow \frac{u'}{u} = 106.25\%$$

The intersection however of the resistance versus engine speed curve and the maximum engine torque curve doesn't occur at maximum speed and is largely depended on the gearbox. The generally improvements in maximum speed for future non hybrid powertrains are hence lower than 6%. For hybrids, only the sustained (i.e. without a time limitation) maximum speed was considered. This means that the vehicle was operating on the engine alone. It has therefore even less available power. The decrease however due to the scaling of aerodynamic resistance power with the 3rd power of speed, is again small.

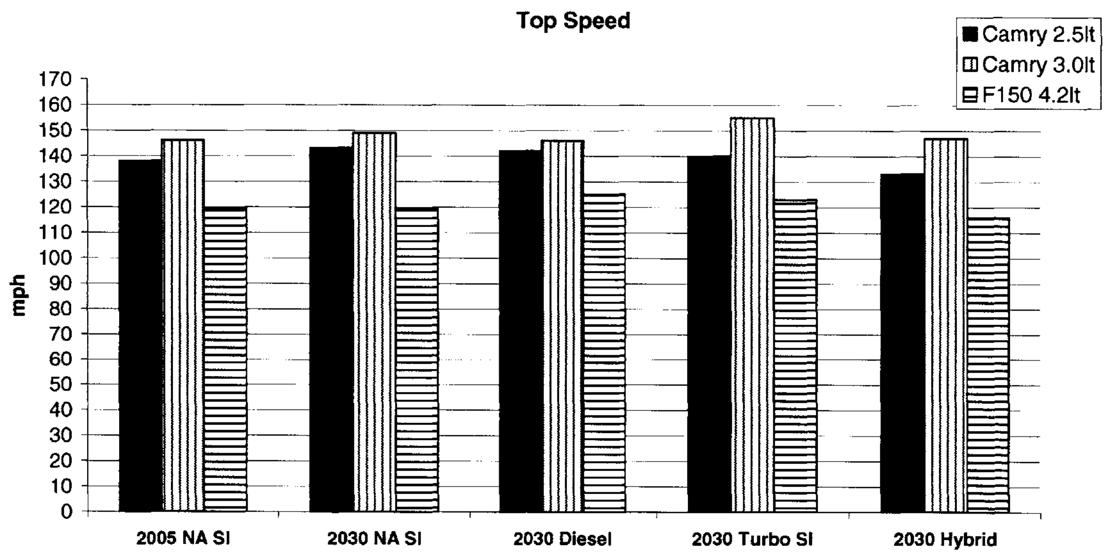


**Figure 128: Top Speed for all Powertrains and Models**

Towing performance was defined as the maximum weight that the vehicle can tow at 55 mph and 6% grade with the lowest possible gear. The coefficient of drag was assumed to remain unchanged by the towed weight. The results may be seen in Figure 129.

Using these assumptions the towing capacity calculated for current vehicles was much higher than the published numbers by the automakers. Towing capacity for the 2.5 liter Camry was estimated at 4210 kg, 6641kg for the 3.0 liter Camry and 3765kg for the F150. The published towing capacity is 907kg for the Camry (independent of engine) and

2585kg for the F150. It is noteworthy however that sometimes, towing and grade ability might be limited by engine cooling problems and not tractive effort.

The industry is reported to use 3% grade and no downshifting allowed, i.e. the test is conducted using only the top gear. A simple calculation shows that for the F150 in order to get a towing capacity of 2585kg at top gear, a grade of 4% would be required according to this definition. For the 2.5 liter Camry at top gear, a 3.8% grade would be needed to limit the towing capability to 907kg. Note that the gearbox was assumed to be at its nominal efficiency for these calculations and that the $c_D$ in real life would become significantly worse. These results are therefore close enough.

It is interesting however, how the huge gear ratios of the F150 are chosen for exactly this type of towing requirement (no downshifting allowed) or low speed towing when allowing for downshift. For example, the F150 at 55mph cannot use the 1st or second gear although the Camry can. This justifies why the 2005 2.5 liter Camry can tow more than the F150 at 55mph when downshifting is allowed, given that their power to curb weight ratio is about the same. At lower vehicle speeds, the F150 would be much better.

In terms of future powertrains compared with the current ones, the expected result is that future vehicles will tow less. Total weight (towed plus curb) that can be pushed up a specific grade basically scales with engine power. For future vehicles engine power is less. Curb weight is also less but for a towing test that allows downshifting the benefit from reduced curb weight is small compared with the large towed weight. This is even truer for hybrids when examing the sustained towing capacity.

The results in Figure 129 verify this conclusion. Future towing capacity is generally smaller than the capacity of the equivalent current model. The exemptions when the towing capacity is the same or even larger than the 2005 model can be attributed either to the future engine being able to use a lower gear speed at 55mph-as is the case with the 2030 NA SI F150 or to slower engines with higher torque output, as is the case with the 2030 diesel F150.

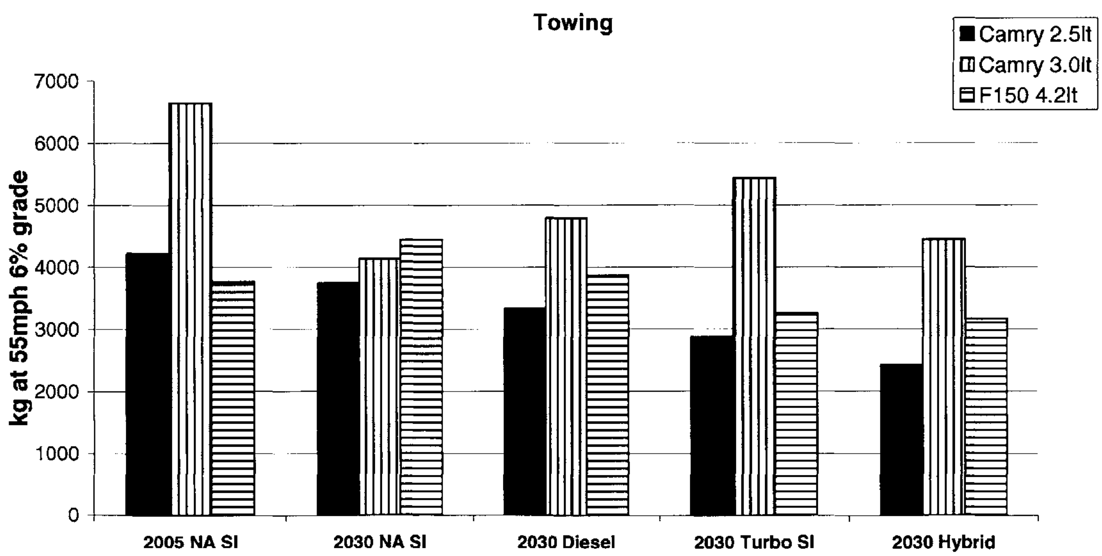


**Figure 129: Maximum Towing Capability at 55mph and 6% grade for all powertrains and models**

Grade capability for all powertrains and models is presented in Figure 130. If all gear ratios were sized to give the same μ value at the tire in future vehicles as in current, grade ability wouldn't vary significantly as it should scale with the power over curb weight ratio which is constant. This is not the case however as although the 1st gear was indeed selected for constant μ, the top gear wasn't. Faster engines with lower torque generally have a disadvantage unless their torque is low enough that they can operate at a lower gear speed, in which case they could be superior to the 2005 powertrain. There is no general conclusion to be made. Perhaps the no downshifting test would be a better criterion for comparison although it doesn't represent real world driving as accurately.

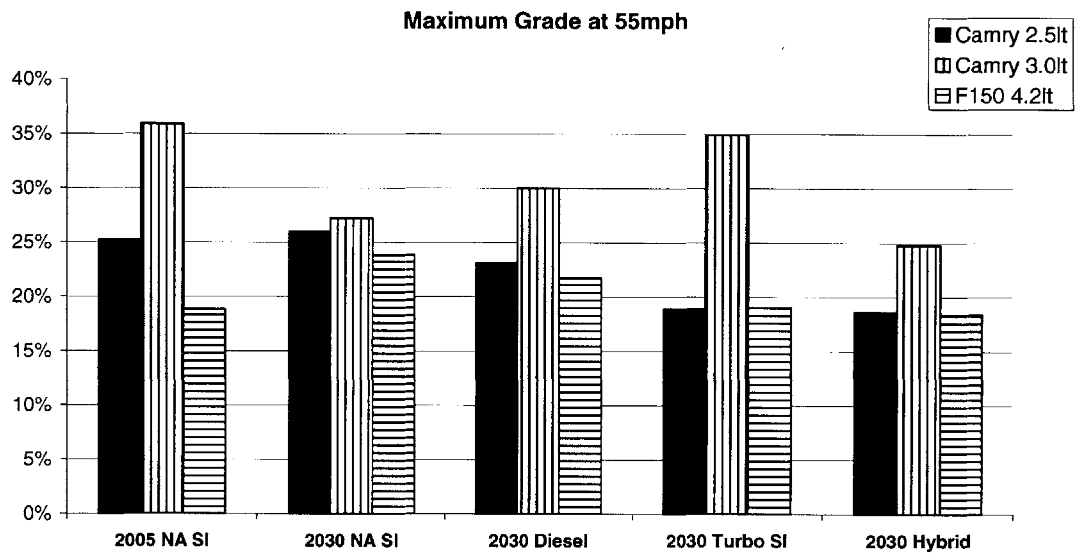


**Figure 130: Maximum Grade Capability at 55mph for all powertrains and models**

## *6.12. Conclusions-Future Work*

The main conclusions of this thesis could be summarized as follows:

- Assuming that vehicles in the near to mid term future will offer the consumer the same level of performance as current vehicles and that their size remains constant, improvements in current automotive technologies do have a significant potential for reducing fuel consumption.
- In order to compare different powertrains on a same performance basis, it is important not only to keep the maximum power over vehicle curb weight ratio constant but also to pick gear ratios so that the transmissions are "equivalent". The powertrain should be viewed as a complete system instead of just an engine.
- The current fuel economy advantage of the diesel engine over naturally aspirated gasoline is reduced in the future. This is largely due to continuing gasoline engine downsizing.
- Turbocharged gasoline engines have the potential of becoming equivalent with low emissions diesel engines.
- Hybrids maintain a competitive advantage over non hybrid powertrains in the future.

- The advantage of hybrids over non hybrid powertrains is significantly more pronounced in urban and "sensible" versus highway and "aggressive" driving patterns.
- When the fuel (well to tank) side is considered as well, diesel engines look better in terms of energy use and GHG emissions than when looking at the vehicle side alone.
- Advanced transmissions and integration/controls are very important in improving both hybrid and non hybrid fuel consumption. Improvements in transmissions and integration/controls could enable powertrain concepts that are significantly different than the current mainstream. Such concepts as powertrains using efficient continuously variable transmissions or manual/automatic transmissions that additionally allow for active engine optimization in a hybrid would significantly benefit future vehicle fuel consumption and possibly also cost (in the case of the hybrids).
- In terms of reducing the cost premium of hybrids, reducing battery costs is important. In terms of battery performance characteristics, improvements would have a small impact on fuel consumption but are not vital for the success of hybrids in the market.

This thesis does not in any way claim to have answered all the questions related to the evolution of vehicle fuel consumption in the future. Some of the most interesting questions to examine as future work include:

- Evaluating the potential of powertrains significantly different than the ones used today. As mentioned in the introduction, there are future powertrain options that do not rely or at least do not rely exclusively on ICE's. The potential of plug-in hybrids for the mid to long term future would be something definitely worth investigating. So is the potential of fuel cell vehicles and all electric vehicles for the longer term.
- In this thesis it was assumed that performance and size remain constant at current levels in the future. The assumption of constant size limits the level of weight reduction that can reasonably be assumed. Historically, this has not been the case. Average vehicle fleet size, weight and performance have all been increasing. This increase has completely negated any fuel consumption benefits that could have been realized because of technical advances. One interesting question therefore would be to try and quantify the effect of weight and performance trends on fuel consumption of future vehicles. How much would future vehicle weight and performance have to increase in order to cancel out future technical advancements in terms of reduction of fuel consumption? Would this increase in weight and performance be reasonable? Thinking in the opposite direction, how much more fuel would be saved if the trends were to be reversed? How much would weight and performance need to be reduced if for example double the reduction in fuel consumption was needed than what technology alone can offer?
- Furthermore, if the effect of weight reduction turns out to be very significant, what is the potential of some of the more drastic weight reduction strategies, such as use of carbon fibers or radically different vehicle designs?

- In this thesis, the effect of accessories such as air conditioning was not explored. Would including it affect any of the conclusions significantly? Would the advantage of hybrids compared with non hybrid powertrains be for example the same if accessories were included?
- The turbocharged SI gasoline map was predicted from the naturally aspirated one. It was not validated versus an existing turbocharged map. Doing so would improve the robustness of the conclusions. Furthermore the assumption of the same indicative efficiency for the naturally aspirated and turbocharged maps implies the same compression ratio. As mentioned in chapter three, currently turbocharged engines have lower compression ratios than naturally aspirated ones although there are some technologies that promise increasing both compression ratio and boost. It is not clear however that highly boosted engines will reach the same level of compression ratios as naturally aspirated ones. The sensitivity of engine indicative efficiency and thus fuel consumption on compression ratio needs to be investigated further.
- Finally, one of the main conclusions of this thesis was that the current relative advantage of diesels decreases in the future. The sensitivity of this conclusion on assumptions about friction reduction was tested and found relatively small. There are however, other sensitivities that should be tested to make this result more robust. How sensitive is it to assumptions about gasoline and diesel downsizing? How sensitive is it to assumptions about the efficiency penalty of emissions control? Moreover, although three different maps were used for current gasoline engines only one was available for diesels. Would the results be any different if a different engine map had been used?

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## Appendix

**Table 21: List of Assumptions for the lower performance Camry**

| Vehicle Parameters | 2005 2.5 liter Camry | 2030 NA SI Camry | 2030 Diesel Camry | 2030 Turbo SI Camry | 2030 Hybrid Camry |
|---|---|---|---|---|---|
| Tire Rolling Friction Coefficient(cr) | 0.009 | 0.006 | 0.006 | 0.006 | 0.006 |
| Coefficient of Drag (cD) | 0.28 | 0.21 | 0.21 | 0.21 | 0.21 |
| Frontal Area m2 | 2.49 | 2.49 | 2.49 | 2.49 | 2.49 |
| Aux Power kW | 0 | 0 | 0 | 0 | 0 |
| Total mass(curb weight+passenger) | 1435+136 | 1148+136 | 1184+136 | 1134+136 | 1162+136 |
| Power to curb weight Ratio(W/kg) | 83.1 | 83.1 | 80.3 | 83.4 | 83 (including motor) |
| **Engine** | | | | | |
| Displacement Volume (cc) | 2357 | 1414 | 1333 | 932 | 1088 |
| Number of Cylinders | 4 | 4 | 4 | 4 | 4 |
| Bore (mm) | 88.4 | 74.6 | 78 | 64.9 | 68 |
| Stroke (mm) | 96 | 81 | 69.8 | 70.5 | 74 |
| Max Bmep (bar) | 11.8 | 13.3 | 20 | 18 | 13.3 |
| Power Density (kW/kg) | 0.74 | 0.925 | 0.715 | 1.03 | 0.925 |
| Max Power (kW) | 119.3 | 95.4 | 95 | 94 | 80.4 |
| indicative efficiency | 0.4 | 0.43 | 0.48 | 0.43 | 0.43 |
| % change in indicative Efficiency | - | 7.50% | 7.50% | 7.50% | 7.50% |
| average fmep(kPa) | 172 | 147 | 194 | 142 | 161 |
| redline(rpm) | 6000 | 7100 | 5211 | 8173 | 7762 |
| weight(kg) | 161 | 103 | 133 | 91 | 87 |
| **Gearbox** | | | | | |
| Type | 5spd auto | 6spd auto | 6spd auto | 6spd auto | 6spd auto |
| Transmission Efficiency | 0.89(peak) | 0.94(peak) | 0.94(peak) | 0.94(peak) | 0.94 |
| 1st Total Gear Ratio | 13.4 | 16.1 | 12.1 | 15.1 | 8.71 |
| Required tire μ with 1st gear | 1.08 | 1.08 | 1.08 | 1.08 | 1.08 |
| Top Gear Ratio | 2.4 | 2.8 | 2.7 | 2.4 | 2.7 |
| Engine rpm at 47.5mph with top gear ratio | 1496 | 1775 | 1693 | 1515 | 1682 |
| weight(kg) | 114(estimated) | 91 | 91 | 91 | 91 |

**Table 21(continued): List of Assumptions for the lower performance Camry**

| Vehicle Parameters | 2005 2.5 liter Camry | 2030 NA SI Camry | 2030 Diesel Camry | 2030 Turbo SI Camry | 2030 Hybrid Camry |
|---|---|---|---|---|---|
| **Motor** | | | | | |
| Pmotor(kW) | - | - | - | - | 16.1 |
| Power Density (kW/kg) | - | - | - | - | 1.55 |
| Efficiency(including electronics) | - | - | - | - | 0.8 for rpm<1/6 max rpm,0.93 else |
| Hybridization Ratio | - | - | - | - | 20% |
| weight(kg) | - | - | - | - | 11.7 |
| **Batteries** | | | | | |
| Average Efficiency over drive cycles | - | - | - | - | ~92% |
| Power Density (kW/kg) | - | - | - | - | 1.1 |
| Energy Density(Wh/kg) | - | - | - | - | 80 |
| weight(kg) | - | - | - | - | 16.8 |
| **Regeneration** | | | | | |
| Friction Braking Fraction | - | - | - | - | 10% |
| **Controls** | | | | | |
| Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy |
| Engine off Torque Fraction | | | | | 40% |

**Table 22: List of Assumptions for the higher performance Camry**

| Vehicle Parameters | 2005 3.0 liter Camry | 2030 NA SI Camry | 2030 Diesel Camry | 2030 Turbo SI Camry | 2030 Hybrid Camry |
|---|---|---|---|---|---|
| Tire Rolling Friction Coefficient(cr) | 0.009 | 0.006 | 0.006 | 0.006 | 0.006 |
| Coefficient of Drag | 0.28 | 0.21 | 0.21 | 0.21 | 0.21 |
| Frontal Area m2 | 2.49 | 2.49 | 2.49 | 2.49 | 2.49 |
| Aux Power kW | 0 | 0 | 0 | 0 | 0 |
| Total mass(curb weight+passenger) | 1515+136 | 1212+136 | 1253+136 | 1192+136 | 1228+136 |
| Power to curb weight Ratio(W/kg) | 103.4 | 103.4 | 98.1 | 103.4 | 103.6 (including motor) |
| **Engine** | | | | | |
| Displacement Volume (cc) | 2988 | 2187 | 1964 | 1640 | 1708 |
| Number of Cylinders | 6 | 4 | 4 | 4 | 4 |
| Bore (mm) | 87.4 | 90.14 | 88.7 | 81.9 | 83 |
| Stroke (mm) | 83 | 85.7 | 79.4 | 77.9 | 78.9 |
| Max Bmep (bar) | 12.5 | 14.1 | 20 | 18 | 14.1 |
| Power Density (kW/kg) | 0.74 | 0.925 | 0.715 | 1.03 | 0.925 |
| Max Power (kW) | 157 | 125 | 123 | 123.3 | 106 |
| indicative efficiency | 0.39 | 0.42 | 0.48 | 0.42 | 0.42 |
| % change in indicative Eff. | - | 7.50% | 7.50% | 7.50% | 7.50% |
| average fmep(kPa) | 180 | 137 | 194 | 136 | 148 |
| redline(rpm) | 6000 | 5816 | 4580 | 6400 | 6316 |
| weight(kg) | 212 | 135 | 194 | 120 | 115 |
| **Gearbox** | | | | | |
| Type | 5spd auto | 6spd auto | 6spd auto | 6spd auto | 6spd auto |
| Transmission Efficiency | 0.89(peak) | 0.94(peak) | 0.94(peak) | 0.94(peak) | 0.94(peak) |
| 1st Total Gear Ratio | 13.94 | 13.85 | 11.3 | 14.42 | 6.53 |
| Required tire μ with 1st gear | 1.42 | 1.42 | 1.42 | 1.42 | 1.42 |
| Top Gear Ratio | 2.5 | 2.4 | 2.4 | 2.2 | 2.2 |
| Engine rpm at 47.5mph with top gear ratio | 1558 | 1561 | 1489 | 1387 | 1368 |
| weight(kg) | 114(estimated) | 91 | 91 | 91 | 91 |

Table 22(continued): List of Assumptions for the higher performance Camry

| Vehicle Parameters | 2005 3.0 liter Camry | 2030 NA SI Camry | 2030 Diesel Camry | 2030 Turbo SI Camry | 2030 Hybrid Camry |
|---|---|---|---|---|---|
| **Motor** | | | | | |
| Pmotor(kW) | - | - | - | - | 21.2 |
| Power Density (kW/kg) | - | - | - | - | 1.55 |
| Efficiency(including electronics) | - | - | - | - | 0.8 for rpm<1/6 max rpm,0.93 else |
| Hybridization Ratio(Pmotor/Peng) | - | - | - | - | 20% |
| weight(kg) | - | - | - | - | 15.44 |
| **Batteries** | | | | | |
| Average Efficiency over drive cycles | - | - | - | - | ~92% |
| Power Density (kW/kg) | - | - | - | - | 1.1 |
| Energy Density(Wh/kg) | - | - | - | - | 80 |
| weight(kg) | - | - | - | - | 22.15 |
| **Regeneration** | | | | | |
| Friction Braking Fraction | - | - | - | - | 10% |
| **Controls** | | | | | |
| Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy |
| Engine off Torque Fraction | | | | | 40% |

Table 23: List of Assumptions for the F150

| Vehicle Parameters | 2005 4.2 liter F150 | 2030 NA SI F150 | 2030 Diesel F150 | 2030 Turbo SI F150 | 2030 Hybrid F150 |
|---|---|---|---|---|---|
| Tire Rolling Friction Coefficient(cr) | 0.0105 | 0.007 | 0.007 | 0.007 | 0.007 |
| Coefficient of Drag | 0.5 | 0.39 | 0.39 | 0.39 | 0.39 |
| Frontal Area m2 | 2.86 | 2.86 | 2.86 | 2.86 | 2.86 |
| Aux Power kW | 0 | 0 | 0 | 0 | 0 |
| Total mass(curb weight+passenger) | 1995+136 | 1596+136 | 1658+136 | 1582+136 | 1615+136 |
| Power to curb weight Ratio(W/kg) | 75.5 | 75.5 | 75.5 | 75.5 | 75.5 |
| **Engine** | | | | | |
| Displacement Volume (cc) | 4192 | 2514 | 2159 | 1792 | 2387 |
| Number of Cylinders | 6 | 6 | 4 | 4 | 4 |
| Bore (mm) | 96.8 | 81.6 | 91.6 | 83.5 | 91.8 |
| Stroke (mm) | 95 | 80.1 | 81.9 | 81.9 | 90.1 |
| Max Bmep (bar) | 10.6 | 11.9 | 20 | 18 | 11.9 |
| Power Density (kW/kg) | 0.74 | 0.925 | 0.715 | 1.03 | 0.925 |
| Max Power (kW) | 150.6 | 120.5 | 124.8 | 119.4 | 101.7 |
| indicative efficiency | 0.4 | 0.43 | 0.48 | 0.43 | 0.43 |
| % change in indicative Eff. | - | 7.50% | 7.50% | 7.50% | 7.50% |
| average fmep(kPa) | 143 | 115 | 194 | 113 | 112 |
| redline(rpm) | 5000 | 5930 | 4438 | 5800 | 5269 |
| weight(kg) | 203.56 | 130 | 175 | 116 | 110 |
| **Gearbox** | | | | | |
| Type | 4spd auto | 6spd auto | 6spd auto | 6spd auto | 6spd auto |
| Transmission Efficiency | 0.89(peak) | 0.94(peak) | 0.94(peak) | 0.94(peak) | 0.94(peak) |
| 1st Total Gear Ratio | 14.6 | 17.6 | 12.83 | 16.8 | 9.56 |
| Required tire μ with 1st gear | 1.17 | 1.17 | 1.17 | 1.17 | 1.17 |
| Top Gear Ratio | 3.73 | 4.42 | 2.7 | 2.8 | 2.3 |
| Engine rpm at 47.5mph with top gear ratio | 1989 | 2359 | 1442 | 1508 | 1370 |
| weight(kg) | 146 | 117 | 117 | 117 | 117 |

**Table 23(continued): List of Assumptions for the F150**

| Vehicle Parameters | 2005 4.2 liter F150 | 2030 NA SI F150 | 2030 Diesel F150 | 2030 Turbo F150 | 2030 Hybrid F150 |
|---|---|---|---|---|---|
| **Motor** | | | | | |
| Pmotor(kW) | - | - | - | - | 20.3 |
| Power Density (kW/kg) | - | - | - | - | 1.55 |
| Efficiency(including electronics) | - | - | - | - | 0.8 for rpm<1/6 max rpm,0.93 else |
| Hybridization Ratio(Pmotor/Peng) | - | - | - | - | 20% |
| weight(kg) | - | - | - | - | 14.8 |
| **Batteries** | | | | | |
| Average Efficiency over drive cycles | - | - | - | - | ~92% |
| Power Density (kW/kg) | - | - | - | - | 1.1 |
| Energy Density(Wh/kg) | - | - | - | - | 80 |
| weight(kg) | - | - | - | - | 21.2 |
| **Regeneration** | | | | | |
| Friction Braking Fraction | - | - | - | - | 10% |
| **Controls** | | | | | |
| Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy | Advisor Gear Shifting Strategy |
| Engine off Torque Fraction | | | | | 40% |

Table 24: Results

| | | Fuel Consumption(liters/100km) | | |
|---|---|---|---|---|
| | | FTP(unadjusted) | HWFET(unadjusted) | US06 |
| Lower Performance Camry | 2005 2.5 liter Camry | 8.9 | 5.9 | 8.3 |
| | 2030 NA SI Lower Performance Camry | 5.7 | 3.5 | 5.4 |
| | 2030 Diesel Lower Performance Camry | 4.9 | 3.0 | 4.9 |
| | 2030 Turbo SI Lower Performance Camry | 5.0 | 3.1 | 5.0 |
| | 2030 Hybrid Lower Performance Camry | 2.5 | 2.7 | 4.1 |
| Higher Performance Camry | 2005 3.0 liter Camry | 11.3 | 6.7 | 9.7 |
| | 2030 NA SI Higher Performance Camry | 6.8 | 4.1 | 6.1 |
| | 2030 Diesel Higher Performance Camry | 5.6 | 3.3 | 5.2 |
| | 2030 Turbo SI Higher Performance Camry | 6.2 | 3.7 | 5.5 |
| | 2030 Hybrid Higher Performance Camry | 2.6 | 2.9 | 4.4 |
| F150 | 2005 4.2 liter F150 | 13.3 | 9.4 | 14.2 |
| | 2030 NA SI F150 | 8.6 | 5.8 | 8.8 |
| | 2030 Diesel F150 | 6.7 | 4.7 | 7.4 |
| | 2030 Turbo SI F150 | 7.1 | 5.1 | 7.7 |
| | 2030 Hybrid F150 | 3.7 | 4.4 | 6.9 |