University of Technology, Sydney

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Manual Transmission with Electric Torque Assist for Torque Hole Compensation

A thesis submitted for the degree of

Master of Engineering (Research)

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Certificate of Original Authorship

I certify that the work in this thesis has not previously been submitted for a degree nor has it been submitted as part of requirements for a degree except as fully acknowledged within the text.

I also certify that the thesis has been written by me. Any help that I have received in my research work and the preparation of the thesis itself has been acknowledged. In addition, I certify that all information sources and literature used are indicated in the thesis.

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Acknowledgement

I would like to express my everlasting gratitude to the people who provided me support in the successful completion of my master study.

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Abstract

The main focus of this research is on the optimization of powertrain of traditional family sedan with manual transmission. The aim of this research is to simulate the transient response transmission torque in a manual transmission with a secondary power resource, electric drive unit, installed on the output shaft of transmission. The current issue that discovered from manual transmission is during the shifting; a torque gap occurs when switching between each gear pair as well as torsional vibration. The torque gap is due to disengagement of clutch when shifting, the torque transferred from engine to transmission gearbox drops to 0, and then after the clutch is engaged, the torque starts increasing and oscillation happens. Such an inconsistency of torque input is the source of power loss and causes decrease in vehicle speed. Torsional vibration is angular vibration of a shaft along its axis of rotation caused by rough torque transmission, and it is considered in this research because it causes jerk or jolt on vehicle body when shifting and it has significant effect on driving comfort or even causes failure if not controlled. Therefore, this research objective is utilizing an electric motor to compensate non-continuous torque transmission. For example, when the clutch is disengaged, motor instead of engine starts operating and produces torque to transmission system. As the results of this alternative, the loss in vehicle's speed is reduced; the torque input to transmission system is more consistent so the torsional deflection of shafts is improved while torsional frequency is not affected, and better driving comfort is achieved.

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1 Introduction

Nowadays one of the most essential environmental issues is increasing levels of health threatening air pollution. Conventional automotive, with internal combustion engine, are one of the causes of environmental degradation. In the conventional internal combustion engine based propulsion system, the energy carrier is gasoline. While vehicles consuming gasoline, chemicals are converted into mechanical energy, but the efficiency of conversion is extremely low, approximately 20% [1]. At high load, even an efficiency of 40% is not acceptable with regard to today's technology [2]. In the meantime of burning petroleum, a vast amount of greenhouse gases, such as water vapor, carbon dioxide and nitrous oxide are released. These greenhouse gases cause a well-known climate problem, which is called global warming, and the following effects of global warming are, for example, rise in temperature and sea level [3].

Beside the state of environmental degeneration, there are some other economic issues stated. For example, as the need for fossil fuel is rising every day, the depletion of fossil fuels becomes the highest priority of governments as a problem of energy resource extraction [4]. "Peak Oil" is the problem that foreshadows depletion of fossil fuel. In Figure 1.1, "Peak Oil" means the time when petroleum extraction reaches a maximum rate, and after that production rate of oil enters terminal decline [5]. If supply cannot satisfy the global demand, the prices of oil will increase to lead further economic recessions, such as the 1973 and 1979 energy crises [6]. As the historical report, this conclusion of depletion by worldwide geologists, physicists, bankers and investors is absolutely true for future projection.

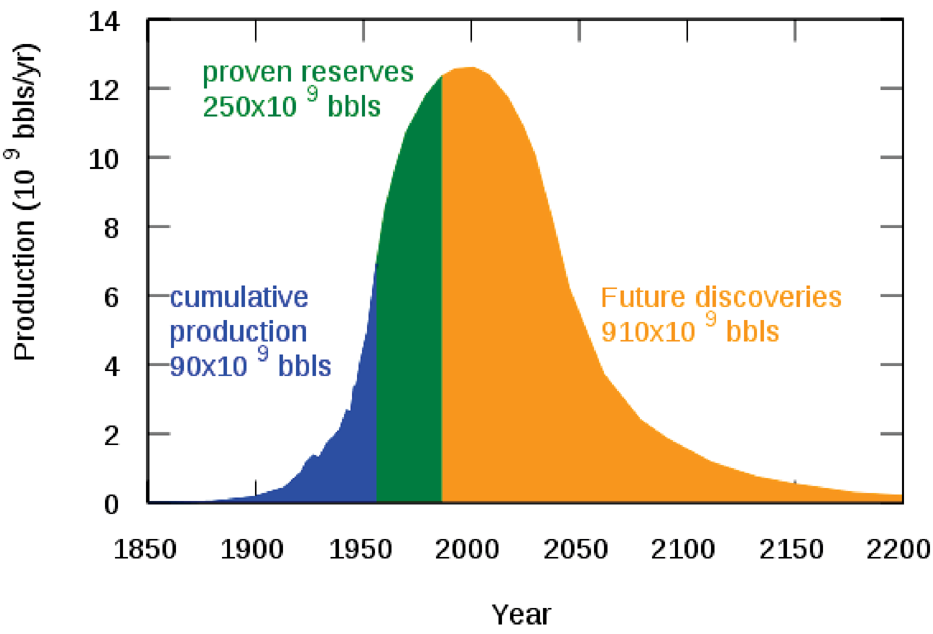

Figure 1.1 Hubbert's Peak Oil Plot [7]

As one of the major elements of the world economy, the transportation industry plays an important role in daily life, which has effects on, but not limited to, the environment impact. An innovation of technology necessitates transition from traditional internal combustion engine powered vehicles to electric vehicles, which is more efficient [8]. In an electric transportation infrastructure, electricity that is consumed can be generated from a wide range of energy resources. For instance, fossil fuels, nuclear power and other renewable sources [9]. Therefore, electric vehicle is significantly more efficient in energy conversion and pollutes less than all other alternatives. However, the primary disadvantage of electric vehicle is the driving range. In Tesla Roadster Review 2010, it has reported that the driving range of electric vehicles is more than 150 miles [10]. So it is concluded that the technology of electric vehicle is not mature to replace petroleum dominant automotive market.

A hybrid electric vehicle has a propulsion system consists of electric motors and conventional internal combustion engine, and it presents an excellent way to reduce consumption through efficiency improvement. In a hybrid electric vehicle, there is more than one reservoir to store energy source. Beside gasoline, the alternative energy resource can be battery, hydrogen fuel cell and hydraulic accumulator. In this research, the main focus will be hybrid electric vehicle, which contains battery and gasoline tank as energy resource to propel the vehicles [11].

A hybrid electric vehicle that combines a conventional internal combustion engine propulsion system with an electric propulsion system presents an excellent way to reduce petroleum consumption through efficiency improvement. A typical HEV will reduce gasoline consumption by about 30% over a comparable conventional vehicle. This number could reach 45% with additional improvement in aerodynamics and engine technology [12]. In principle of improving fuel economy, a HEV is possible to downsize the engine and still fulfill the maximum power requirements of the vehicle; recover some energy during deceleration instead of dissipating it in friction braking; optimize the energy distribution between the prime movers; eliminate the idle fuel consumption by turning off the engine when no power is required and eliminate the clutching losses by engaging the engine only when the speeds match. However, these improvements are counteracted by the fact that HEV is about 10 to 30% heavier than conventional ICE vehicle, and this issue is a vital handicap to HEV design [13].

In this thesis report, the hybridization of a traditional vehicle with electric drive unit is proposed based on the problems of manual transmission introduced in the chapter of background and literature review. In the chapter of modeling, the method of modeling by MATLAB is explained in detail and then simulation results are compared to identify the improvement.

2 Literature review

In this chapter, the main components of powertrain and their working principles are explained, and these are helpful with understanding the powertrain systems. Also, some terminologies are clarified to help understanding problem statements.

2.1.1 Manual Transmission

A manual transmission is commonly used in motor vehicle applications. A brief history shows that manual transmissions preceded automatics by several decades. In 1938, every automotive provided by General Motors had manual transmission. Generally, it utilizes a clutch (dry or wet), which is operated by a foot pedal in an automotive. In the gearbox of manual transmission, it has a finite number of fixed gear ratios and is chosen manually by driver using a shift rod [14].

In modern passenger cars, manual transmission has synchronizer to eliminate the need for double clutching. The synchronizer allows the collar and the gear to make frictional contact before the dog teeth make contact so the collar and gears can synchronize their speeds before the teeth need to engage [15].

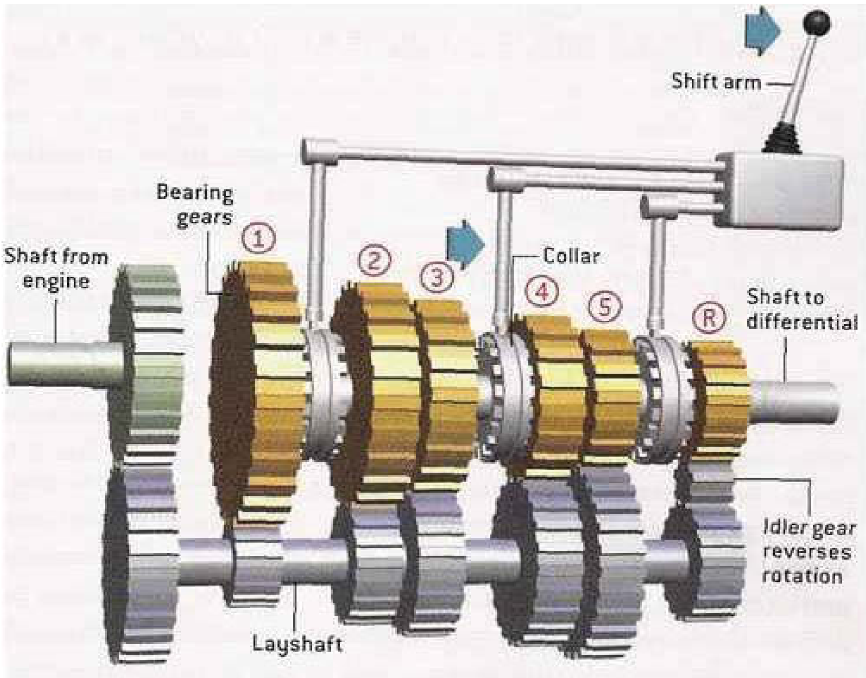

Figure 2.1 Schematic of Manual Transmission [14]

In Figure 2.1, the input shaft on the left is connected to clutch then engine flywheel, and it transmitted power from engine. The layshaft in grey color is an intermediate shaft between engine shaft and the shaft to differential. Layshaft has gears with different sizes, so it can create various gear ratio based on different torque requirements [14]. For a 5-speed manual transmission, there are 6 gear pairs include one reverse gear set on the output shaft with bearing gears. In neutral state, these gears do not rotate with output shaft. The collar is used to synchronize gears onto the shaft. For example, when the clutch is disengaged, one of three collars that is operated manually by driver, slide between each gears to lock one gear on the shaft so a fixed gear ratio is selected to drive the vehicle [14].

2.1.2 Clutch

A clutch in a vehicle is a mechanical device that provides the power transmission from one component to another when engaged. In an automobile, clutches control whether engine power is transmitted to transmission. Usually, clutches are employed in devices with two rotating shafts. One shaft is attached to a power source unit such as engine or motor, and the other shaft is driven to transfer output power. When the clutch is fully engaged, it connects two shafts so they are locked together and rotate at the same speed; when the clutch is slipping, the two shafts may be locked together but spinning at different speeds; when it is disengaged, two shafts are spinning independently at different speeds [16].

The main components of automobile clutch are indicated in Figure 2.2:

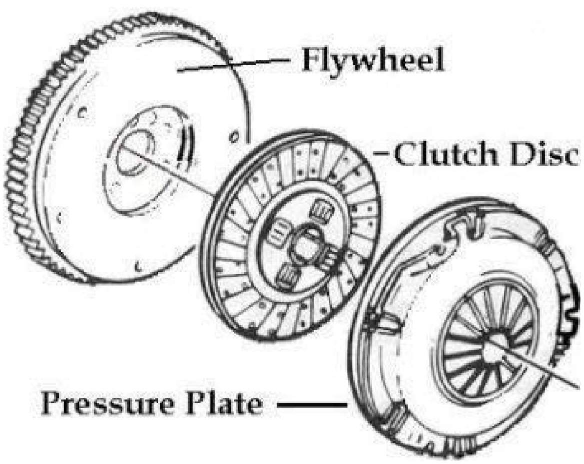

Figure 2.2 Main Components of Clutch [16]

Clutches in a vehicle is directly mounted to the engine flywheel, and this provides a convenient large diameter steel disk that acts as a driving plate. The design of vehicle clutch is based on friction disc depressed together against a flywheel using springs. The friction material of clutch disk varies depending on many considerations for example, dry or wet types [17]. For the heavy-duty trucks and racing cars, they require ceramic clutches that have a higher friction coefficient. When the clutch pedal is depressed, diaphragm of the pressure plate is pulled away from flywheel to disengage. However, when the engine speed is too high, engaging the clutch will cause inordinate clutch plate wear and a harsh, jerky start. Therefore, the control of clutch determines the driving comfort [18].

The default state of the clutch is engaged so that the engine and gearbox are connected until the pedal is depressed. When the clutch is engaged, the engine is at idling and the transmission is in neutral state, the engine will rotate the input shaft of transmission but there is no torque or power transmitted to wheels through gearbox. And this only happens when the vehicle is at idling and gear shifting.

2.1.3 Synchronizer system

As the performances of engine and clutch have been improved constantly, this places increasingly high requirements on manual transmissions and their components. For the synchronization of manual transmission, compact and lightweight products are required, and these components must be satisfactory for minimizing the shift force and improving shifting comfort. In a vehicle with manual transmission, a driver-operated hand lever is used to control the synchromesh mechanism. The synchronization systems align the differing shaft speeds between the constant mesh gear and the shift element located on the shaft in a manual transmission gearbox [15]. The internal of synchronization system is shown in Figure 2.3:

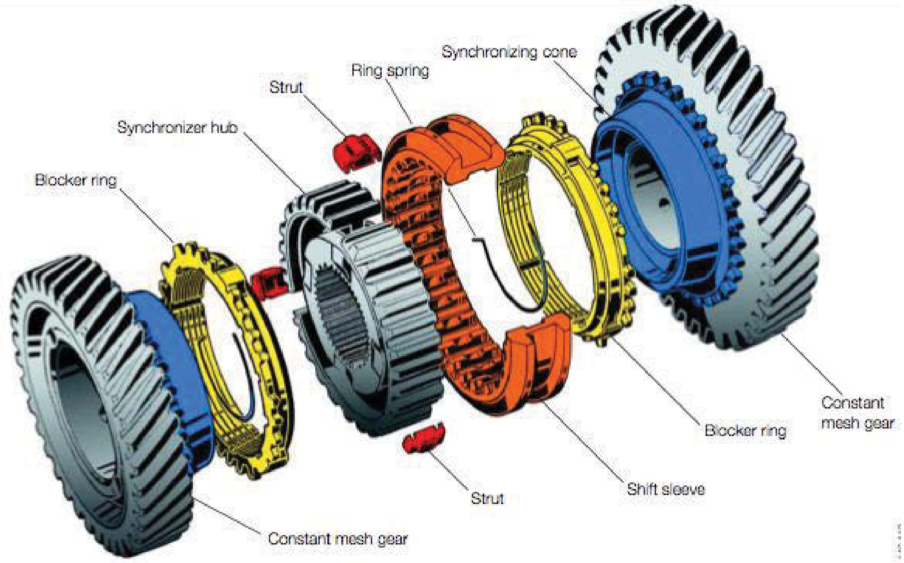

Figure 2.3 Synchronization System [19]

When the synchronization starts, the shift sleeve is moved out of the neutral position and displaced axially toward the constant mesh gear. The chamfered teeth on the sleeve and struts press the blocker ring against the friction cone at the clutch hub of the mesh gear. Due to the frictional torque, the blocker ring immediately rotates with the clearance of the notches in the sleeve support. As the axial displacement force increases, the fully effective frictional torque aligns the differing speeds between mesh gear and the hub so the gear is synchronized. When the equal speeds are acquired, the frictional torque is removed. Since the shift sleeve is frictionally engaged, the teeth of dog clutch will slip into the gaps of blocker ring [19].

During the engagement, the moment of losses, due to splashing, inertia of masses and bearing accelerate or decelerate the constant mesh gear, and the synchronizer has to overcome the momentum of input shaft and clutch to match the new gear ratio. While meshing, the shift sleeve rotates at the same speed as the gear body so that the shift sleeve reaches its final position. The working principle of synchronizer system allows driver to choose gear ratio manually without causing failure to shafts [19].

2.2 Aerodynamic Drag

In fluid dynamics, the force on an object that resists its motion through a liquid or gas is called drag. Drag force is opposite to the relative motion of an object. It is generated by the interaction and contact of a solid body with a fluid not generated by a force field, such as gravitational field. If there is no fluid, there is no drag. Also, drag is generated by the difference in velocity between the solid object and the fluid, so if there is no motion, there is no drag. It makes no difference whether the object moves through a static fluid or whether the fluid moves past a static solid object [20].

Drag is a vector quantity with both a magnitude and a direction; there are many factors affecting the magnitude of the drag. One of the sources of drag is the skin friction between the molecules of the air and the solid surface of the object and it depends on properties of both solid and fluid. For instance, for the solid, a smooth surface can produce less friction than a rough surface; for a fluid, the magnitude of drag depends on the viscosity and compressibility of fluid; and along this solid surface, a boundary layer of low energy is generated and skin friction depends on conditions in the boundary layer [21]. The general equation for calculation of air drag force on objects is:

$$F_{drag} = C_d A_f \rho_{air} \frac{V^2}{2}$$

where C_d *is drag coefficien t*

A_f *is frontal area of object*

ρ_{air} *is air density*

V is the relative velocity of object [22]

From the formula above, it shows evidently that drag increases when the density of fluid, frontal area or speed increases. Also, different shape profile of object can influence coefficient of drag [20].

In automotive engineering, the major concerns of automotive aerodynamics are to reduce drag and undesired lift forces and other causes of instability at high speeds. For racing vehicles, it is important to produce downward aerodynamic forces to improve traction of wheels and thus better cornering ability is acquired. In the formula of drag force, the drag force is proportional to square of velocity magnitude that means frictional force of drag increases rapidly with vehicle speed. Also the net force exerted on vehicle body will be:

$$F_{net} = F_{rolling} - F_{drag}$$

With drag force being applied, the actual force for acceleration will be rolling force by dynamic interaction between tires and road minus drag force. At a specific speed, the net force is equal to 0. There will be no more acceleration on the vehicle. For example,

Figure 2.4 shows the velocity of an example vehicle with 1000kg mass, 300hp power at 5000rpm:

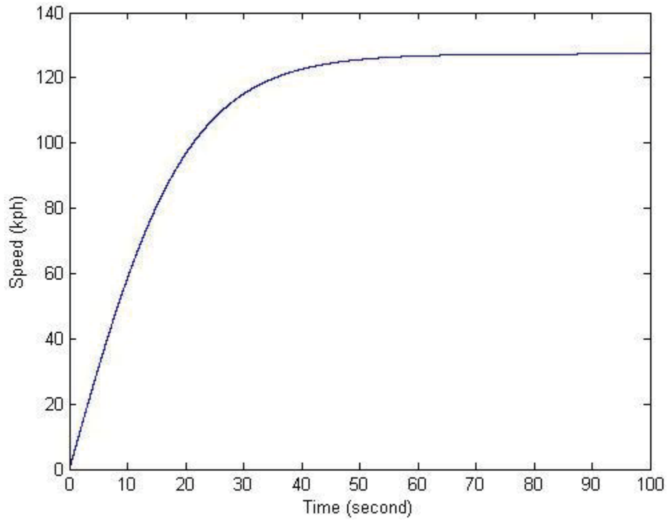

Figure 2.4 Velocity Profile of Example Vehicle

Figure 2.4 indicates an example that once the net force between rolling force and drag force on vehicle body become equal, the vehicle reaches its peak speed. In this case, the peak speed of vehicle is about 125km/h.

2.3 Relationship between transmission torque and RPM

In a manual transmission vehicle, in addition to putting foot firmly on the gas pedal, shifting the gear is also compulsory. But the best way to shift gears such that the top speed is attained in shortest time. Therefore, to get the best acceleration, it is necessary to find the optimal shifting point.

From the chapter of engine working principal, the vehicle acceleration depends on the torque curve. Unfortunately, it is very difficult to obtain precisely the whole torque curve of an engine through normal source. So from the specification of an engine, it normally states the peak torque and power at corresponding rev speed. For example, the engine of Ford Falcon 2000 has peak torque, 357N.m at 3000rpm, and peak power, 157kW at 4900rpm [23].

Gear-shifting strategy aims to reach the top speed in the shortest possible time at a maximum acceleration rate, so this is equivalent to maximizing the torque generated by the engine. The force generated by the engine is equal to torque of several gear ratios divided by tire radius. Since tire radius and final drive ratio is the same for each gear, it turns out that the only thing needs to be considered is the product of torque and the n-th gear ratio, and this is named as transmission torque [24].

More precisely, the optimal shifting point will be at the RPM that gives more transmission torque. From Figure 2.5, the torque curve shows that the peak torque occurs at 3000rpm. The gear changing is done when the car make more torque after shifting than in current gear. To help with demonstrating this, gear ratio is important because it is necessary to know what RPM will drop to after shifting. For example, if the gear ratios of 1st and 2nd gears are 2.78:1 and 1.48:1, then the rpm will drop from 6000rpm to about 3194rpm, the formula used is:

$$RPM_{2nd} = RPM_{1st} \times \left(\frac{gear\ ratio_{2nd}}{gear\ ratio_{1st}} \right) \quad [24]$$

The plot of transmission torque versus engine RPM of Ford Falcon is in Table 2.1

RPM	Engine Torque (N.m)	Transmission Torque of Different Gear Ratio (N.m)			
		2.78	**1.48**	**1.03**	**0.89**
500	213.8	594.4	316.4	220.2	190.3
1000	272.4	757.3	403.2	280.6	242.4
1500	300.2	834.6	444.3	309.2	267.2
2000	312.4	868.5	462.4	321.8	278.0
2500	335	931.3	495.8	345.1	298.2
3000	343.8	955.8	508.8	354.1	306.0
3500	337.2	937.4	499.1	347.3	300.1
4000	320	889.6	473.6	329.6	284.8
4500	312.9	869.9	463.1	322.3	278.5
5000	288.9	803.1	427.6	297.6	257.1
5500	248.4	690.6	367.6	255.9	221.1
6000	198.7	552.4	294.1	204.7	176.8
6500	149	414.2	220.5	153.5	132.6

Table 2.1 Transmission Torque of Different Gear Ratio

The plot of transmission torque is in Figure 2.5:

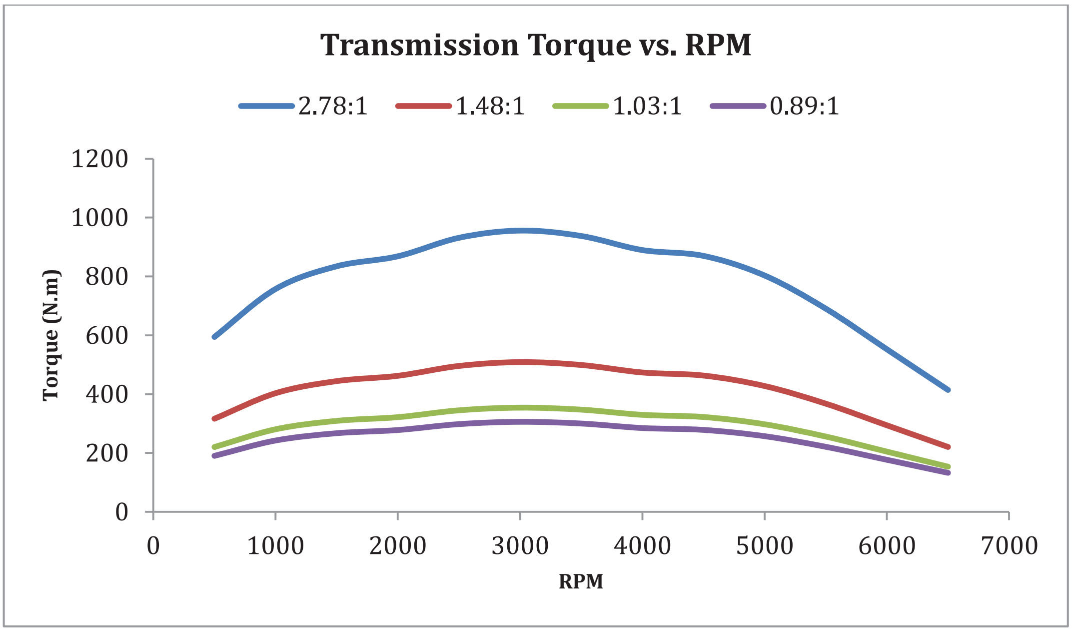

Figure 2.5 Transmission Torque vs. RPM of Ford Falcon 2000

From the example, if gear changing occurs at 6000rpm in the first gear, then the engine rpm will return to 3194rpm, which means transmission torque is decreased from 600N.m to 505N.m. If gear changing occurs at 5000rpm, then the transmission torque drops from 800N.m to 490N.m, the difference in torque is relatively bigger than the first gear, this torque difference may cause jerk on vehicle body. Smooth acceleration can be achieved by obtaining close torque before and after shifting. From the Table 2.1, we can see that the optimal shifting point is between 5500rpm and 6500rpm. If two torques outputs are almost the same, that means the maximum acceleration is achieved. Therefore, the optimal shifting point occurs when the two transmission torques are close before and after gear changing.

2.4 Hybrid Vehicle Technology

2.4.1 History

After Gustave Trouve built the first electric vehicle in 1881, in 1899, hybrid vehicle was introduced at Paris Salon of 1899; it was a tricycle with two rear wheels powered by electric motors. A 0.75 horsepower ICE is coupled to a 1.1kW generator to recharge the batteries [25]. Later in 1903, Camille Janatzy at the Paris Salon presented a parallel hybrid vehicle. The vehicle has a 6 horsepower ICE combined with 14 horsepower electric machine; ICE could either charge the batteries or provide extra power [25]. These early hybrid electric vehicles were designed to improve electric drive range or provide additional power. However, these designs could not rival tremendous improvement of gasoline vehicles after World War I. Evidently, the gasoline engine was improved to have more efficiency with smaller size and higher power density. Beside additional cost of electric motor, environmental impact by lead-acid batteries was another reason to cause extinction of HEV. Also, the control strategy was a barrier to development of HEV. Even during the oil crises in 1973 and 1977, the serious environmental issue did not attract interest of investigation of HEV [26].

Later in 1990s, it was clearly understood that the pure electric vehicles could not accomplish the aim of saving fossil fuel. Ford Motor Corporation started focusing on development of HEV. In U.S, Dodge built the Intrepid ESX 1, 2 and 3. The ESX 1 was a series HEV with a small turbocharged 3-cylinder diesel engine and a battery pack. Two 100 horsepower electric motors were mounted to rear wheels. At the same time, the Ford Prodigy and GM Precept presented a parallel HEV with small turbocharged diesel engines coupled to dry clutch manual transmissions. However, neither of them achieved mass production [27]. The significant commercialization of HEV started by worldwide famous Japanese automaker, Toyota. Toyota Prius sedan revived hybrid electric vehicles' market in 1997. Also, another Japanese automobile manufacturer, Honda released its Insight and Civic Hybrid [28]. Today, these vehicles are obtainable throughout the world because they created an outstanding shape of fuel consumption. As the first commercialized hybrid electric vehicles, Toyota Prius and Honda Insight had significant contribution to settle the problem of vehicle fuel consumption in the modern era.

2.4.2 Degree of Hybridization

Types of hybrid electric vehicles are defined by degree of hybridization, mild hybrid and full hybrid. A mild hybrid is a type of gasoline-electric hybrid that uses internal combustion engine to power the vehicle at all times; an oversized starter motor is incorporated only as a power assist or starter-generator, which means the vehicle cannot be driven solely by electric motor due to its power is too low to propel the vehicle. Mild hybrids save fuel by shutting engine power off under most circumstances such as when the vehicle is stopped, braking or coasting. Then electric motor restarts the engine seamlessly and efficiently. The benefits of mild hybrids are to provide a modest improvement 10 to 15 percent in fuel efficiency because ICE is shut off when stopped.

Since they have much less cost than full hybrid systems, a greater number of drivers are likely to choose mild hybrids [29].

In the drivetrain of mild hybrid, as in Figure 2.6, the electric motor is often mounted between the engine and transmission, taking the place of the torque converter; and it is used as a power booster to provide supplementary propulsion power when accelerating. Therefore, mild hybrids do not have hybrid drivetrain and it provides limited feature utilization. Currently, there are several mild hybrid electric vehicles in the market. For example, Honda Civic Hybrid, Honda Insight, Honda Accord Hybrid, Mercedes-Benz S400 Hybrid.

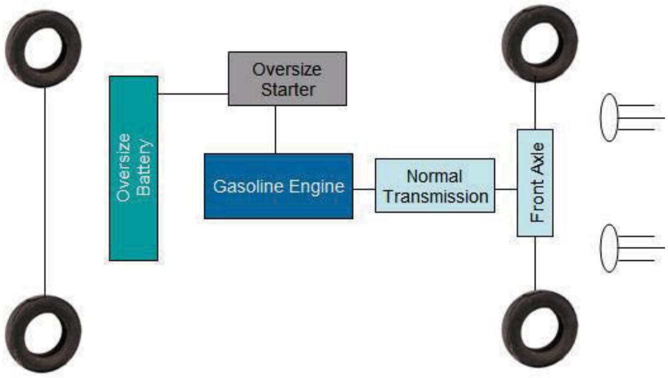

Figure 2.6 Design of GM's Mild Hybrid [30]

Unlike mild hybrid, a full hybrid, some called strong hybrid, has the distinction that can use the electric motor as the only source of propulsion for low-speed, low-acceleration maneuvering, such as in stop-and-go traffic. As soon as additional power is needed, the gasoline engine joins in to provide full power at its most efficient performance speed range. From a point of view of fuel economy, a full hybrid such as Toyota Prius can provide a fuel economy improvement of 60 percent or more. Full hybrid vehicles comprises more sophisticated technology such as using a battery that stores energy generated from gasoline engine or during braking or coasting, from the electric motor. Generally, a full hybrid propulsion system has a gasoline engine as the primary source of power, and an electric motor provides additional power when needed; it has a split power device that allows drivetrain converting between mechanical and electrical power [29].

As a full hybrid can be driven by either mechanical or electrical power, there are several operation options based on state of charge of batteries and power required. For example, in the representative full hybrid vehicle, Toyota Prius has the technology called Hybrid Synergy Drive, and it is a unique drivetrain design with six general operation modes [31]:

1. Electric drive mode
2. Cruising mode
3. Overdrive mode
4. Charging mode
5. EM assist mode
6. Negative power split mode

The elaborate explanation of operation modes is in Section 2.4.4. Each operation mode has its unique function to optimize efficiency of propulsion system.

2.4.3 System Configurations

Base on different operation scheme and requirements, drivetrain of hybrid electric vehicles are classified into three main types:

- Series hybrid
- Parallel hybrid
- Series-parallel, or combined hybrid

In a series propulsion system, there are three main components: one engine, one electric generator and one electric traction motor. The engine acts as an auxiliary power system to extend the electric drive range by using a generator to convert output from engine to electricity that either directly feeds the motor or charges the battery. Refer to this functionality, the series hybrid is so called range-extended electric vehicles (REEV). According to its operation plan, the operation of engine does not concern the power demand of vehicles, thus the engine can have optimal performance with an acceptable efficiency and emission. Another advantage of this series configuration is that the transmission does not require a clutch [29]. For example, the engine is never disengaged since it is mechanically decoupled from the drive axle, shown in Figure 2.7:

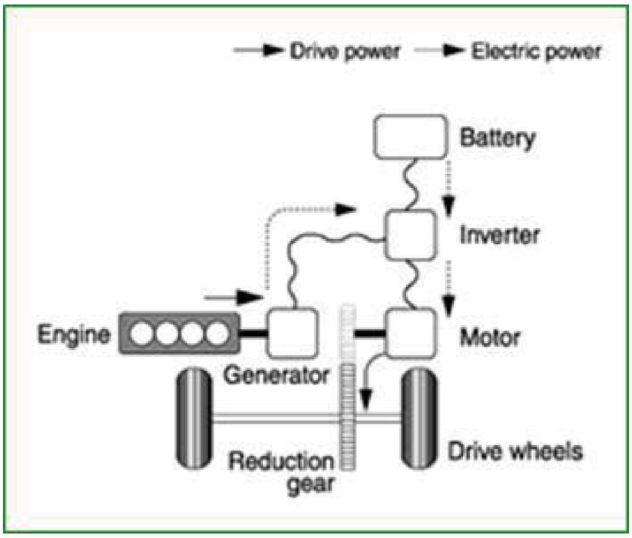

Figure 2.7 Schematic of Series Hybrid Drivetrain [29]

As the traction power from engine is totally converted from electricity, the size of traction motor is required to meet the peak power demand of vehicle, so it may contradict the possibility of downsizing. Therefore, the approaches of series hybrid reducing fuel consumption are:

1. Recover some energy during deceleration instead traditional friction braking;
2. Optimize the energy distribution between primary power resources;
3. Eliminate the idle fuel consumption by turning off the engine in the case of stop-and-go;
4. Accomplish best engine efficiency by operating ICE at its optimal power range.

The cost of these improvements is that the series HEV has the additional weight due to car body reinforcement, electric machines, battery, and this may push the fuel consumption above the value of good ICE dominant vehicles.

If the series hybrid vehicle is considered as a pure electric vehicle with an additional ICE based energy path, a parallel hybrid vehicles are rather an ICE based vehicle with an additional electrical energy path, because in a parallel hybrid electric, both ICE and electric motor can supply traction power individually or in combination, as Figure 2.8. Typically, the ICE is turned off at idle and the electric motor is used to assist acceleration or other high power demand cases. Compare with series hybrid, the advantage of parallel hybrid vehicle is only two machines needed. The electric motor is utilized as a generator to charge the battery from engine or regenerative braking. The benefits of parallel hybrid are similar to series hybrid except since electric motor can provide spare power to compensate extra power demand, it is possible to downsize the engine and still fulfill the maximum power requirements of vehicle [29].

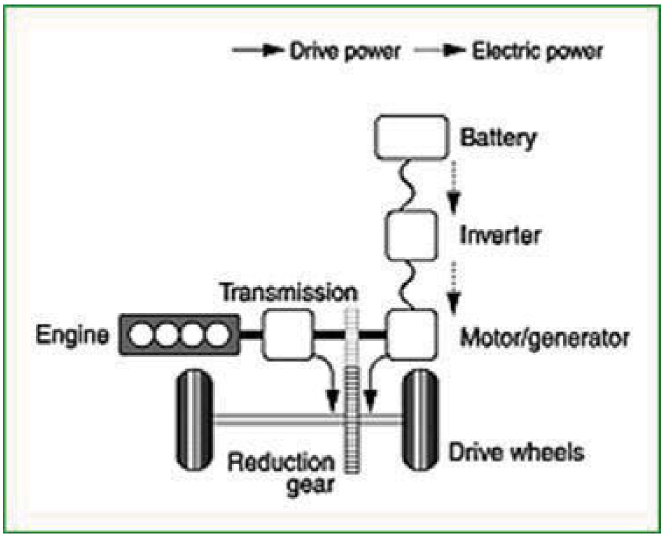

Figure 2.8 Schematic of Parallel Hybrid Drivetrain [29]

A combined hybrid drivetrain is an intermediate between series and parallel hybrids, as in Figure 2.9. In fact, it is mostly a parallel hybrid contains some features of series hybrid. It incorporates power-split device that allows the either mechanical or electrical power paths from the engine to the wheels. In a parallel hybrid configuration, one electric motor absorbs energy as a generator during regenerative braking. The other electric machine acts as a generator when in series configuration; it is used to charge battery while engine is at idling. In this series-parallel drivetrain, a smaller and highly efficient ICE is utilized, so it only operates within a favorable range of rev speed to achieve a higher overall efficiency. The current models with power split systems include Ford, General Motors, Toyota and Nissan [29].

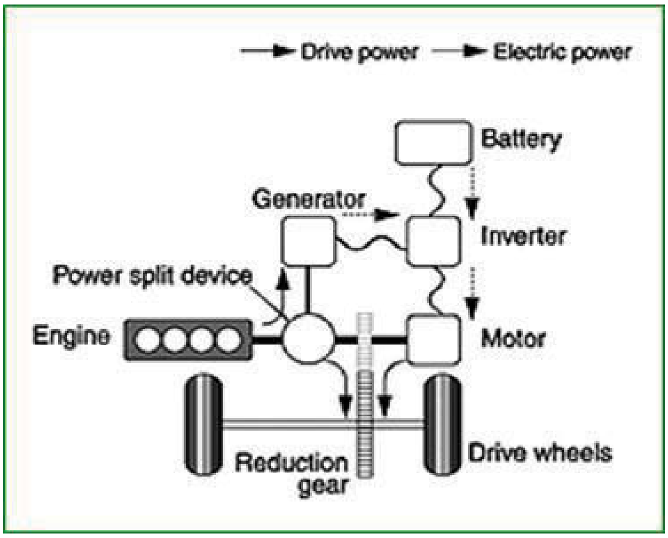

Figure 2.9 Schematic of Series-parallel Hybrid Drivetrain [29]

2.4.4 Examples

In this section, a few current models, such as Honda Insight and Toyota Prius, with good reputation are indicated; and in the past years, lower fuel consumption can evaluate their success.

As a representative mild hybrid electric vehicle, Honda Insight definitely has its essential contribution in reduction of fossil fuel usage. Honda Insight is the first production mild hybrid vehicle featured Integrated Motor Assist (IMA) system [32]. The basic specifications of Honda Insight are shown in Table 2.2:

Engine Type	In-Line 4-Cylinder
Engine Block/Cylinder Head	Aluminum-alloy
Displacement (cc)	1339
Power	73kW@5800rpm
Torque	167N.m@1000-1700rpm
Compression Ratio	10.8:1
Valve Train	8-valve SOHC i-VTEC
Drive Type	Front wheel drive
Electric Motor/ Generator Type	DC Brushless Motor (Permanent Magnet AC Synchronous)
Power	9.7kW@1500rpm
Torque	79N.m@1000rpm

Battery Type	Nickel Metal Hydride
Output	100.8 Volts
Rated Capacity	5.75 Ah
Transmission Type	Continuously Variable Transmission
Gear Ratios	3.172-0.529

Table 2.2 Specifications of Honda Insight [32]

In the drivetrain of Honda Insight, shown in Figure 2.10, an electric motor is directly mounted to the engine's crankshaft between engine and transmission; provide most of the work. At high torque and low revolution speed, electric motor can support acceleration and steady state cruising cases at medium vehicle speeds. During gentle braking and coasting, motor acts as a generator to charge the IMA system. Furthermore, electric motor activates engine as an engine starter and quickly rotating engine to reach idle speed after traffic stop.

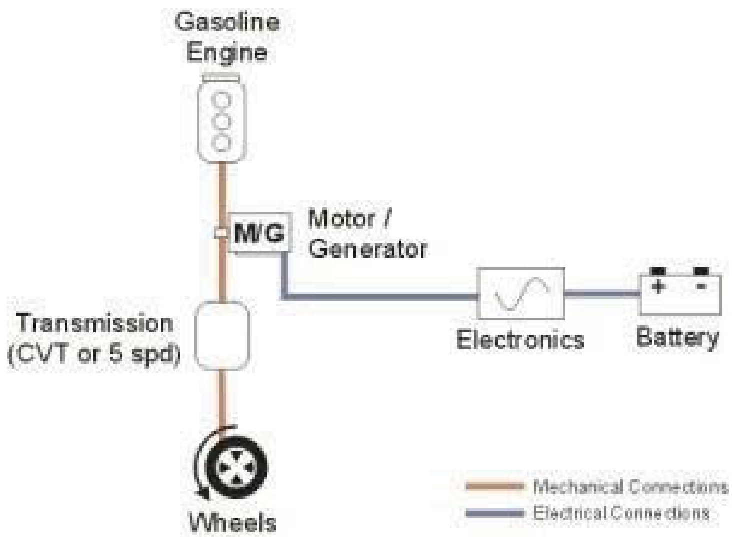

Figure 2.10 Overview of Honda Insight Hybrid Drivetrain [33]

Continuously variable transmission provides infinite ratios to keep engine operating within its optimal range. Forward gear ratios are between 3.172-0.529, the infinite ratio represents that the transmission is electronically controlled to up or downshift and makes engine's output more efficient; hence provide superior performance to those conventional transmissions with fixed gear ratios. Thanks to this advanced configuration, the fuel efficiency rating according to U.S EPA testing methodology is: city 5.9L/100km, highway 5.5L/100km, combined 5.7L/100km [33].

Another representative example of full hybrid with the best fuel efficiency is Toyota Prius; it has demonstrated a cleanest criterion of fuel economy since 1997. As the first mass-produced series-parallel hybrid electric vehicle, its unique design makes it most popular choice and competitive to the other hybrids. The specifications of Toyota Prius are shown in Table 2.3:

Engine Type	In-Line 4-Cylinder
Engine Block/Cylinder Head	Aluminum-alloy
Displacement (cc)	1798
Power	73kW@5200rpm
Torque	142N.m@4000rpm
Compression Ratio	13:1
Valve Train	16-valve DOHC i-VTEC
Drive Type	Front wheel drive
Electric Motor/ Generator Type	Permanent Magnet AC Synchronous
Power	60kW
Torque	207N.m
Battery Type	Nickel Metal Hydride
Output	273.6 Volts
Rated Capacity	6.5 Ah
Transmission Type	Continuously Variable Transmission

Table 2.3 Specifications of Toyota Prius [34]

Compared with Honda Insight, Toyota Prius has similar efficiency advantage that comes from regenerative braking. During regenerative braking, the electric motor operates as a generator to slow the vehicle down and converts some energy of forward motion into electricity to recharge the batteries. Also, the hybrid powertrain of Prius allows ICE managed in its efficient range, and electric motor can provides extra power to meet high power demand [33].

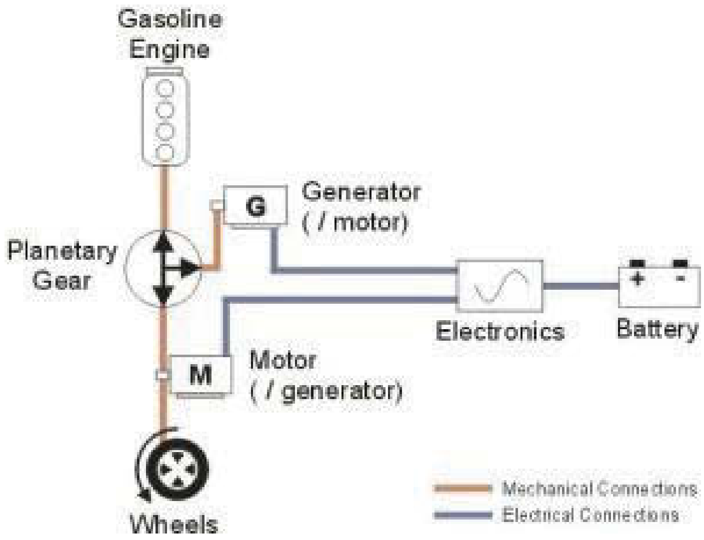

Figure 2.11 Overview of Toyota Prius Drivetrain [33]

Refer to Figure 2.11, the overview of Toyota Prius drivetrain, also called Hybrid Synergy Drive system, has indicated that Prius has two electric machines, both of them can act a generator or motor; and planetary gear set (power split device), shown in Figure 2.12, acts as a continuously variable transmission, and it is one of the smartest design in automotive chronicle.

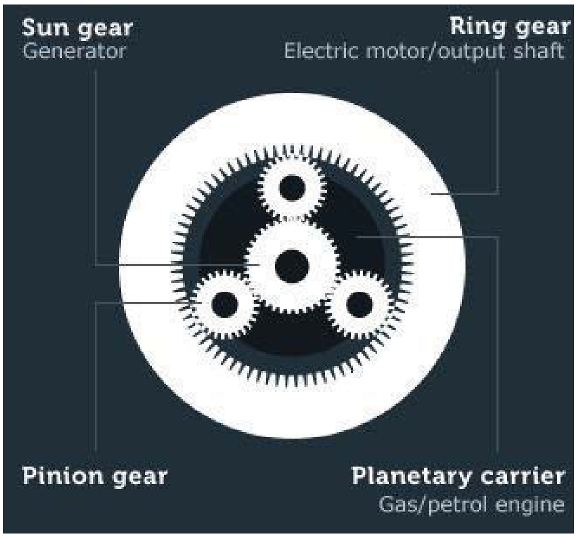

Figure 2.12 Power Split Device [35]

In the power splitting device, one motor generator (G, in Figure 2.11) connects to ring gear and generates electrical power to either recharge battery or supply electrical power to the other motor generator, which is connected to sun gear. Furthermore, this motor generator can regulate generation of electrical power by varying internal resistance and rev speed; also, it serves as the engine starter. The other electric machine (M, in Figure 2.11), which connects to sun gear, drives the vehicle together with ICE to feed energy to the wheels. During the regenerative braking, this electric machine acts as a generator to convert kinetic energy into electrical energy, which is then stored in battery [33].

The electronic control system of Toyota Prius is very complex because there are various phases of operation, and they are as follows [33]:

- Engine start: power is supplied to electric machine G as a starter due to starting engine requires a little amount of power from electric motor G;
- Low gear: when accelerating at low speeds, engine turns more rapidly than wheels but does not develop sufficient torque. Extra engine speed is fed to generator G. The output is fed to motor M, acting as a motor and adding torque to driveshaft;
- High gear: when cruising at high speeds, the engine turns slowly than the wheels but develops more torque than needed. Motor G then runs as a generator to remove the excess engine torque then fed to Motor M as a motor to increase the wheel speed;
- Reverse gear: the computer feeds negative voltage to Motor M, applying negative torque to the wheels;
- Silent operation: at slow speed and moderate torque cruising, the Hybrid Synergy Drive system can drive without running the ICE; electricity is supplied only to Motor G, allowing Motor M to rotate freely;
- Neutral gear: neutral gear is achieved by turning the electric motor off. The planetary gear is stationary; if vehicle wheels are turning, ring gear will rotate, causing the sun gear to rotate as well, while motor M freewheels so no power is dissipated;
- Regenerative braking: by drawing power from motor M and depositing it into the battery pack, then Hybrid Synergy Drive can simulate the deceleration of normal engine braking while saving the power for future boost;
- Engine braking: During braking when the battery is approaching potentially damaging high charge levels, the electronic central system automatically switches to conventional engine braking, drawing power from motor M and shunting it to motor G, speeding the engine with throttle closed to absorb energy and decelerate the vehicle;

- Electric boost: The battery pack provides a reservoir of energy that allows the computer to match the demand on the engine to a predetermined optimal load curve, rather than operating at the torque and speed demanded by the driver and road. The computer manages the energy level stored in the battery, so as to have capacity to absorb extra energy where needed or supply extra energy to boost engine power;
- Battery charging: The Hybrid Synergy Drive can charge its battery without moving the car, by running the engine and extracting electrical power from Motor M.

The advanced and unique technology of Prius allows it to have an optimal fuel efficiency, which is: urban 4.60L/100km, extra urban 4.2L/100km, and combined 4.40L/100km [33].

2.5 Vehicle Power Loss

After over 100 years since the invention, internal combustion engines have become a part of our life. The internal combustion engine takes in air and fuel and burns them to produce mechanical power. However, most of the energy is wasted as heat. That makes engines large radiators. Also, the internal combustion engine is responsible for major power loss of a vehicle power loss. Other resources of power loss are drivetrain loss, idle loss and parasitic losses, shown in Figure 2.13:

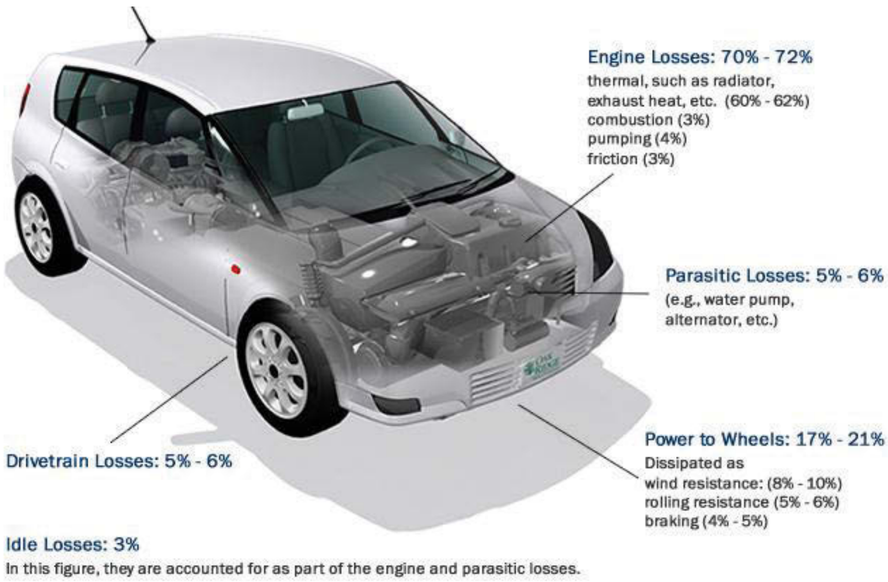

Figure 2.13 Schematic of Vehicle Power Loss [36]

Normally, to measure power losses of a vehicle, a dynamometer, which is an electro-mechanical instrument, is used to measure the torque and power produced by an engine. It applies various loads to an engine and is usually connected to a computer that analyzes and calculates all the aspects of engine operation measured. There are two types of dynamometer: engine dynamometer, which measures engine performance only, and chassis dynamometer, which measures the power from engine through vehicle driving wheels.

Based on Figure 2.13, the result is observed by simulating the vehicle with a city-highway combined test cycle. There is more than 70% of energy loss in the engine. Most of the energy is dissipated as exhaust heat, and other minor losses are due to combustion, pumping and friction. In gasoline-powered vehicles, most of the fuel's energy is lost in the engine, primarily as heat. Smaller amounts of energy are lost through engine friction, pumping air into and out of the engine, and combustion inefficiency. Advanced technologies such as variable valve timing and lift, turbocharging, direct fuel injection, and cylinder deactivation can be used to reduce these losses. Diesel engines have inherently lower losses and are generally one-third more efficient than their gasoline counterparts. Recent advances in diesel technologies and fuels are making diesels more attractive [36].

There is 5% to 6% of energy converted into parasitic losses, and this parasitic loss means power steering. Power steering, the water pump, and other accessories use energy generated by the engine. Fuel economy improvements up to 1% may be achievable with more efficient alternator systems and power steering pumps [36].

The amount of power losses in drivetrain is the same as parasitic losses, which is 5% to 6%. In a vehicle drivetrain, the power loss is most likely because of bearing and friction losses. Energy is lost in the transmission and other parts of the driveline. Technologies such as automated manual transmissions (AMTs), dual-clutch transmission, lock-up transmissions and continuously variable transmissions (CVTs) can reduce these losses.

Idle losses of a vehicle are based on the idling time of the vehicle in city driving, such as stop and go traffic. When vehicle stops at the traffic light, the vehicle is not moving but engine maintains at idling speed. There is 3% of total energy to run the engine and power the water pump, power steering, and other accessories. Integrated starter/generator (ISG) systems, like those used in hybrids, eliminate idling by turning the engine off when the vehicle comes to a stop and restarting it when accelerator is pressed [36].

Compared with other minor losses, losses of power to wheels have higher distribution in total energy wasted. There are 17% to 21% power losses on vehicle wheels. Within these losses, braking losses occurs at any time pressing brakes in a conventional vehicle, energy initially used to overcome inertia and propel the vehicle is lost as heat through friction at the brakes. Less energy is used to move lighter vehicle. So less energy is wasted from braking a lighter vehicle. Using lightweight materials and lighter-weight technologies can reduce weight. Hybrid, plug-in hybrids, and electric vehicles use regenerative braking to recover some braking energy that would otherwise be lost. Wind Resistance occurs when a vehicle expends energy to move air out of the way as it goes down the road – less energy at lower speeds and more as speed increases. This resistance is directly related to the vehicle shape and frontal area. Smoother vehicle shapes have already reduced drag significantly, but further reductions of 20%-30% are possible. Rolling resistance is a resistive force caused by the deformation of a tire as it rolls on a flat surface. New tire designs and materials can reduce rolling resistance. For cars, a 5%-7% reduction in rolling resistance increases fuel efficiency by 1%, but these improvements must be balanced against traction, durability, and noise [36].

2.6 Torque Gap

The manual shift process is usually achieved through the engagement or disengagement of clutch. The entire shift process can be divided into two phases. The torque phase, where the speed ratio remains constant but output torque decreases, and the inertia phase, and the inertia phase, where the speed ratio changes. The shift process is complex and can be influenced by various factors.

Figure 2.14 shows the curves of clutch pressure, output torque and engine speed during a gear change from 1st to 2nd gear. The solid line and dashed line represent well and poorly controlled clutch pressure respectively [37].

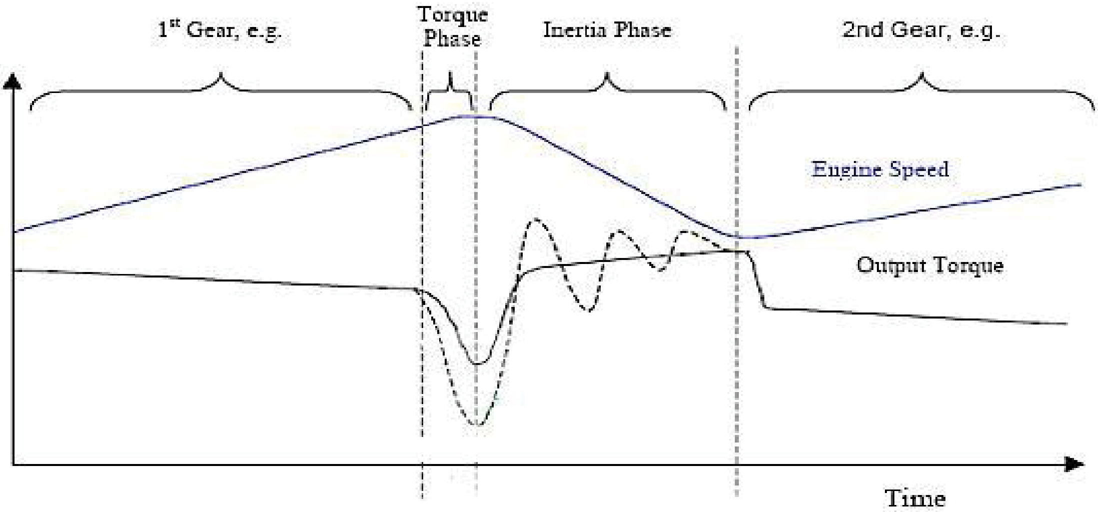

Figure 2.14 Shift Process Analysis [37]

In the beginning of shifting process, the applied pressure to the clutch begins to decrease, but the clutch remains engaged without slipping. During this phase, there is no abrupt change in engine speed. The only difference in this phase is the change of torque; hence it is named torque phase. In the inertia phase, the clutch is kept in slip until the oncoming clutch is completely engaged. Since the two friction components are in slip, the output torque varies sharply, so output torque experiences both a trough period (lower than the torque in the original gear) and a crest period (higher than the torque in the original gear). The trough period is called a torque hole, while the wave crest period is called a torque overshoot, as shown in Figure 2.15. The torque hole is defined by depth and width, where the depth is the difference between minimum torque and the torque in previous gear and the width of the torque hole. The torque overshoot is measured by height that is the distance between the maximum torque and the original value before up-shift [37].

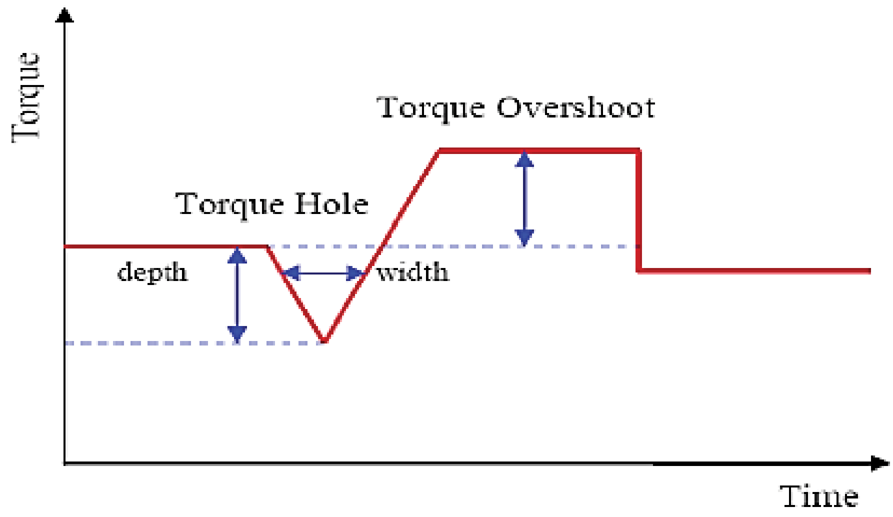

Figure 2.15 Torque Curve in the Up-shift Process [37]

The bigger the torque hole, the larger the decrease of torque in torque phase, which results in a more significant reduction of acceleration. Since the fluctuation of torque transmission causes discomfort for the driver, the expected value of the torque hole should be as shallow and narrow as possible. The torque overshoot denotes the magnitude of output torque, which reflects the shock and vibration of the gearshift.

2.7 Torsional Vibration

Torsional vibration is an angular vibration of a shaft along its axis of rotation caused by rough torque transmission to some degree during startup, shutdown, and continuous operation [38].

Torsional vibration response of rotating machinery components is an essential consideration in defining reliability of a rotary drivetrain where can cause failure if not controlled. In reality, torques generated by internal combustion engine are not smooth, and the driven components do not react to give a smooth response to the torque supply because the components in power transmission systems are not infinitely stiff [39].

Torsional vibrations in shafts may result from following reasons [38]:

- Inertia forces of reciprocating mechanisms, for example, pistons in an internal combustion engine
- Impulsive loads occurring during a normal machine cycle
- Shock loads applied to electrical machineries

- Torques related to gear tooth meshing frequencies, turbine blade passing frequencies.

For machines with massive rotors and flexible shafts, the severity of the torsional oscillations depends on the relationship between the operating speed and excitation frequencies of unsteady torques and the torsional natural frequencies and modal shapes of the shaft system. The accurate prediction of machine torsional frequencies and frequencies of the torsional load fluctuation should not coincide with torsional natural frequencies. Hence, determination of torsional natural frequencies of a dynamic system is very important [39].

3 Problem Statement and Method of Attack

3.1 Problem statement

This research is carried out based on current existing problems of vehicle powertrain with manual transmission. Generally, the problems of manual transmission are torque hole when upshifting, torsional vibration and driving comfort. The causes of these problems will be introduced in literature review.

The research proposal stated that the issues discovered in manual transmission are the torque hole when switching between each gear pair. The Figure 3.1 is the plot of clutch torque versus time of 4-speed manual transmission. The result is extracted from the simulation of original drivetrain in section of Simulation Results. After clutch is disengaged, the clutch torque drops to 0 and this period lasts for approximately 2 seconds; after engagement of clutch, there is an overshoot and oscillation of clutch torque, and this means it will cause the unstable status of transmission system. Then torque return to its desired value.

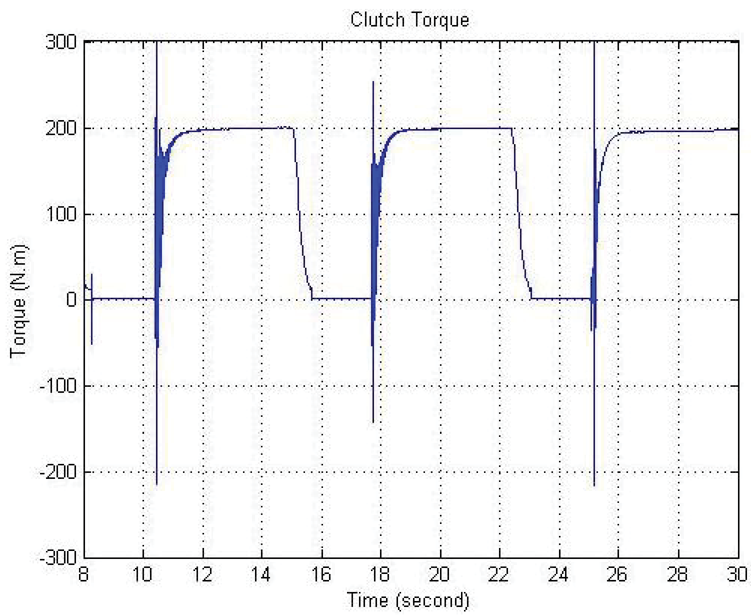

Figure 3.1 Clutch Torque vs. Time

Figure 3.2 indicates that the acceleration of vehicle is fluctuating when shifting. The value of acceleration has both positive and negative magnitudes. And these cause jerk or jolt of vehicle and significantly affect driving comfort.

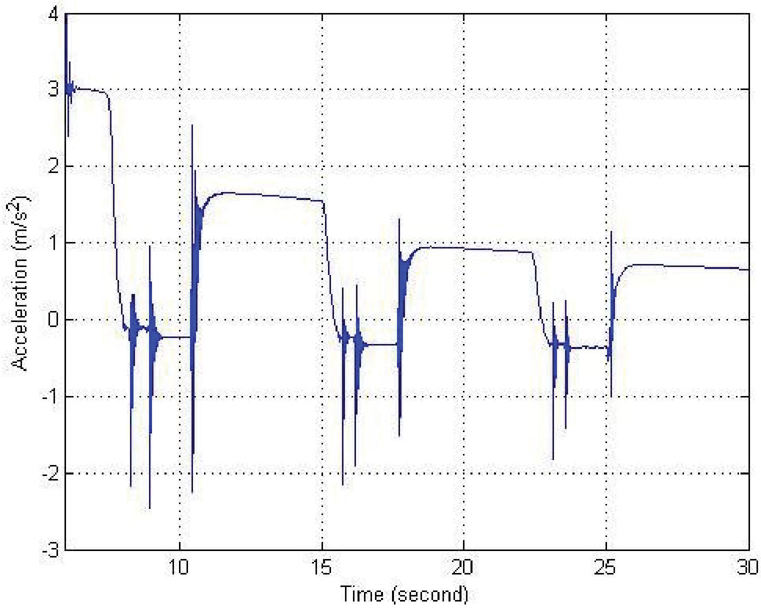

Figure 3.2 Acceleration vs. Time

The negative acceleration due to aerodynamic drag causes loss in vehicle's velocity, the plot of vehicle velocity versus time shown in Figure 3.3:

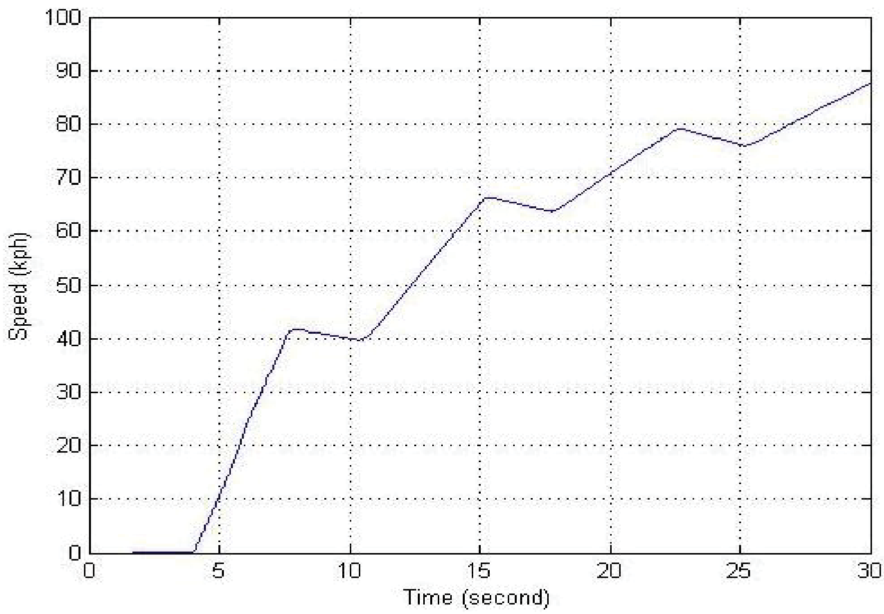

Figure 3.3 Vehicle Speed vs. Time

The vehicle velocity drops by 3km/h during each time of upshifting, and this is one of the sources of vehicle power loss. There are some ideas proposed by other researchers to compensate torque hole. For example, installing flywheel to restore energy and providing torque assist to transmission, or synchronizing an extra pair of gears to keep engine and transmission connected [40, 41].

3.2 Choice of motor

In literature review, there are a few types of motors introduced, and the most suitable one is permanent magnet direct current motor because it is widely used in variable speed drive with variable frequency supplies. The main advantages are the absence of rotor slip power loss and natural ability to supply reactive current.

Figure 3.4 illustrates the performance graph of a DC motor, and it is divided into two zones, the top part is intermittent torque zone, which can provide torque larger than rated torque. However, the motor performance is not stable within this zone because it will cause motor overloaded even failure if without efficient coolant. Below intermittent torque zone, there is a continuous torque zone. Normally, a motor can operate with a reliable performance in continuous torque zone and the torque is constant if the motor speed is below rated speed. However, if motor speed exceeds rated speed, the output torque will decrease as the speed increases.

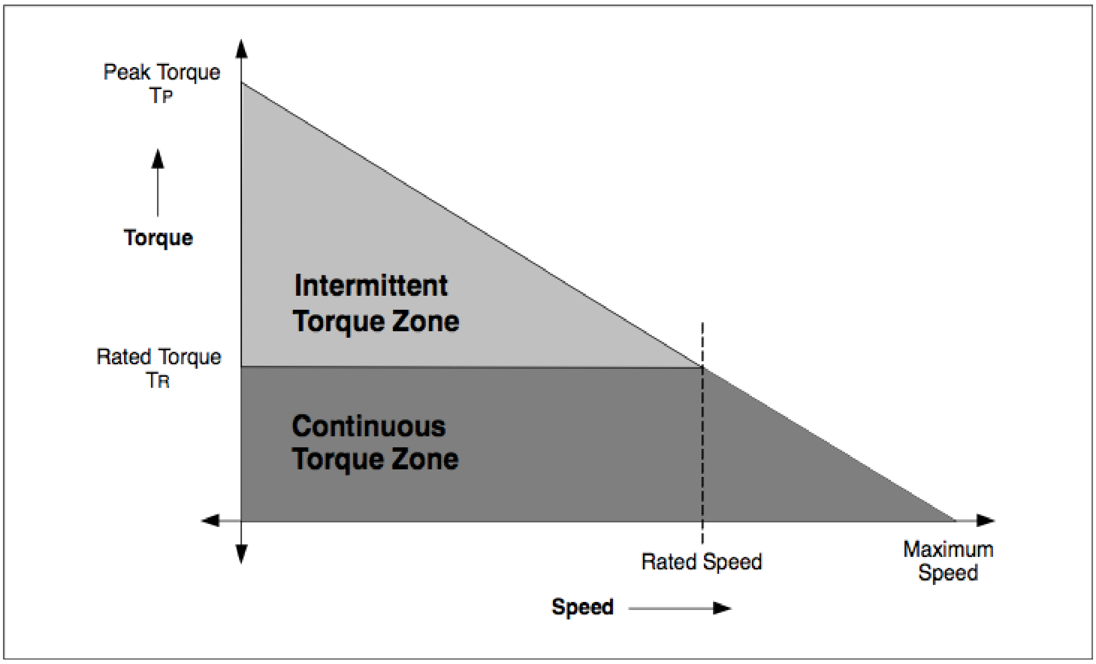

Figure 3.4 Performance Graph of Motor [42]

Based on the characteristic of electric motor, the performance of selected motor should have certain rated torque and rev speed to satisfy the system requirement. Table 3.1 shows the specification of proposed electric motor:

Motor type	DC Brushless Permanent Magnet Motor
Nominal voltage	48V
DC generator output power	4kW
Stall torque	50N.m
No-load speed	3000rpm

Table 3.1 Performance of Proposed Electric Motor

Accompanied with an electric motor, an electric device, which is called power inverter, is installed to change direct current to alternating current.

3.3 Schematic of powertrain

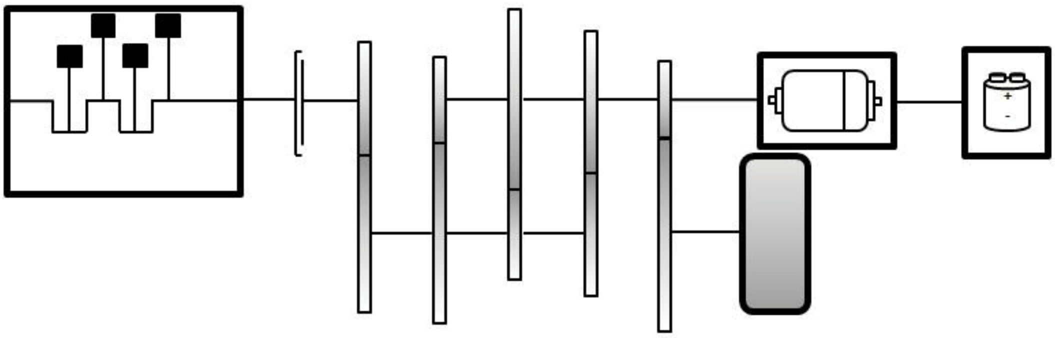

Figure 3.5 Schematic of Proposed Powertrain

From Figure 3.5, the electric devices, such as motor, power inverter and battery are connected with cables, and the motor is mechanically mounted to pinion of final drive gear reduction by a shaft. There is no gear reduction used to transfer torque from motor to final drive gear of transmission. When the motor is not operating, it will be set for free rotating with the transmission output shaft and it will create additional inertia to powertrain. This design is similar to parallel hybrid electric vehicle, both internal combustion engine and electric motor can be power source to propel vehicle. However, this design has lower degree of hybridization because the electric motor only operates when clutch is disengaged. Therefore, electric drive range is extremely short and electric motor with low performance is chosen for considering the cost of this customization.

3.4 Operation modes

The drivetrain has low degree of utilization of electric motor and the main operation mode is to provide torque assist when clutch is disengaged. The normal operation mode is the same as ordinary vehicle with manual transmission.

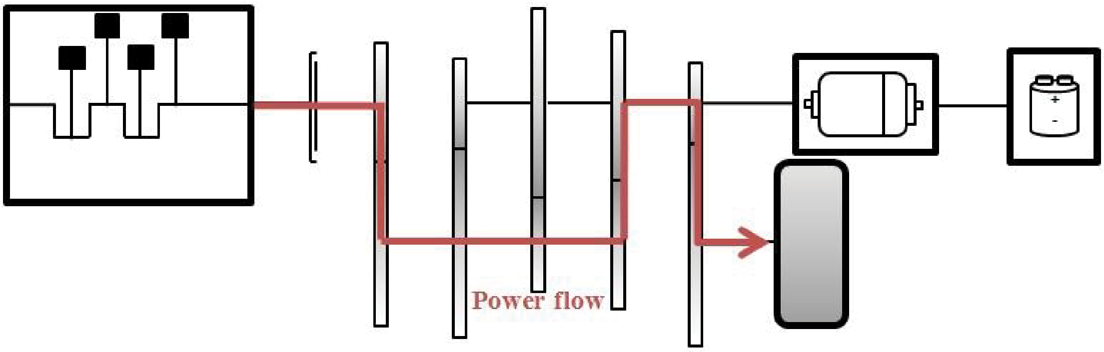

Figure 3.6 Power Flow of Normal Operation Mode

In Figure 3.6, when the clutch is engaged and throttle is opened, engine starts transferring torque and rotational motion to clutch disk, and then this torque is transferred to counter gear, gear box, final drive gear then to the wheels. At this time, the electric motor is at free rotating with output shaft. This only causes extra inertia to drivetrain and there is no energy flow come from battery kit.

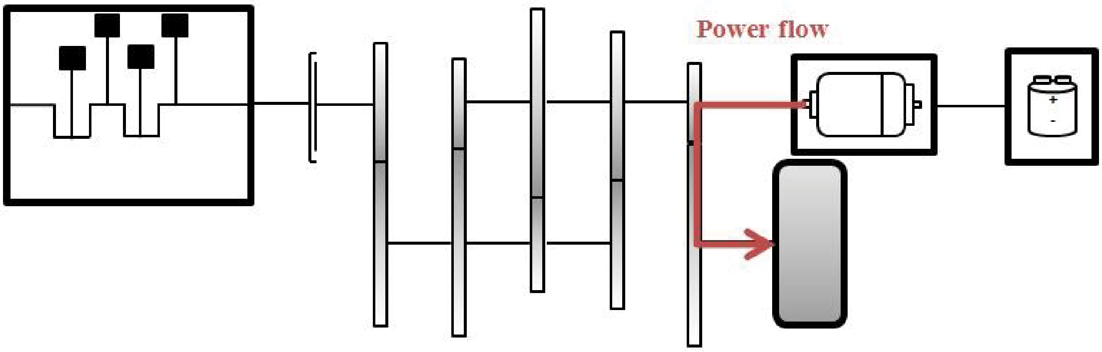

Figure 3.7 Power Flow of Torque Assist Mode

In Figure 3.7, when clutch is disengaged, there is no power flow from engine to gearbox. Electric motor starts operating and torque is transferred from motor to final drive gear and then to the wheels. The motor becomes main power source in torque assist mode to drive the vehicle for approximately 2 seconds. This torque assist mode only lasts until the clutch is engaged after gear is synchronized.

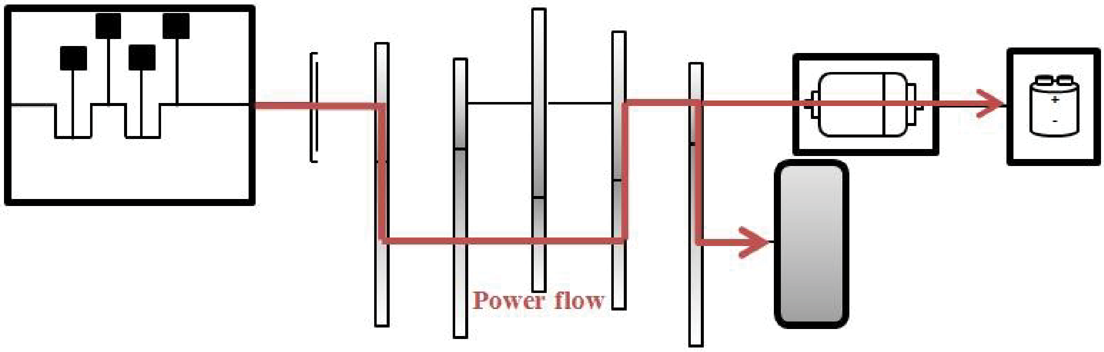

Figure 3.8 Power Flow of Recharge Mode

When battery is at low state of charge, the electrical connection between motor and battery is closed and recharge mode is switched on. Motor acts as a generator to consume mechanical energy from engine and generate electricity to charge battery. This recharge mode requires highly precise control of motor on switching between motor and generator mode because of the short time interval. Therefore, engine alternator will charge the battery if necessary.

4 Modeling

The driveline of a vehicle is a fundamental part and it is modeled depending on the purpose. The aim of modeling is to figure out the improvement by electric drive unit by comparing with traditional manual transmission driveline. The models are combinations of rotating inertias connected by damped shaft flexibilities. The generalized Newton's second law is used to derive the models.

In order to explain the oscillations in the transmission speed, the main observation will be focusing on low gears. The reason for this is that the lower the gear is, the higher the torque transferred in the drive shaft is, and deflection on the shaft is bigger. This means the shaft torsion is higher for lower gear ratios, and hereby also the oscillation problems.

4.1 Basic driveline equations

4.1.1 Original system

In the original systems, the driveline is assumed as a dynamic system with 5 degrees of freedom. The modeling is established based on following basic driveline equations:

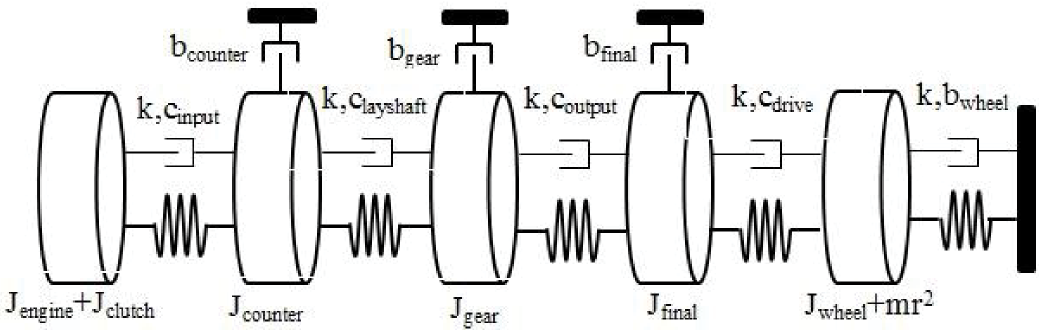

Figure 4.1 Original Systems of Driveline with inertias, stiffness and damping

Engine: The output torque of the engine is characterized by the driving torque T_{engine} resulting from the combustion, the internal friction from the engine $T_{f.engine}$, and the external load from the clutch T_{clutch}.

$$(J_{engine} + J_{clutch})\ddot{\theta}_{engine} = T_{engine} - T_{f,engine} - T_{clutch}$$

Clutch: a friction clutch consists of a clutch disk connecting the flywheel of the engine and the transmission input shaft.

When clutch is engaged:

$$T_{clutch} = T_{input}$$

When clutch is disengaged:

$$T_{clutch} = 0$$

Input shaft: the input shaft is modeled as a damped torsional flexibility, having stiffness k_{input}, and internal damping c_{input}.

$$T_{input} = T_{clutch}$$
$$= k_{input}(\theta_{clutch} - i_{counter}\theta_{counter}) + c_{input}(\dot{\theta}_{clutch} - i_{counter}\dot{\theta}_{counter})$$

Counter gear: counter gear is a gear pair, with a gear ratio $i_{counter}$, connecting input shaft and layshaft of transmission. The inertia of counter gear is $J_{counter}$. The viscous damping coefficient is $b_{counter}$.

$$J_{counter}\ddot{\theta}_{counter} = i_{counter}T_{input} - b_{counter}\dot{\theta}_{counter} - T_{layshaft}$$

Layshaft: same as the input shaft, layshaft has a stiffness of $k_{layshaft}$, and damping coefficient of $c_{layshaft}$.

$$\begin{aligned} T_{layshaft} &= T_{gear} \\ &= k_{layshaft}(\theta_{counter} - i_{gear}\theta_{gear}) + c_{layshaft}(\dot{\theta}_{counter} - i_{gear}\dot{\theta}_{gear}) \end{aligned}$$

Transmission gear: in a manual transmission, there are 4 or 5 pairs of gear reductions available for choosing. Assuming inertia of selected gear is J_{gear}, and viscous damping coefficient is b_{gear}. Also, gear ratio i_{gear} is variable due to gear shifting.

$$J_{gear}\ddot{\theta}_{gear} = i_{gear}T_{gear} - b_{gear}\dot{\theta}_{gear} - T_{output}$$

Output shaft: output shaft has a stiffness of k_{output}, and internal damping of c_{output}.

$$T_{output} = T_{final} = k_{output}(\theta_{gear} - \theta_{final}) + c_{output}(\dot{\theta}_{gear} - \dot{\theta}_{final})$$

Final drive: final drive is a gear pair, with a fixed gear ratio i_{final}, connecting output shaft and drive shaft of transmission. Assuming inertia of selected gear is J_{final}, and viscous damping coefficient is b_{final}.

$$J_{final}\ddot{\theta}_{final} = i_{final}T_{final} - b_{final}\dot{\theta}_{final} - T_{drive}$$

Drive shaft: drive shaft is a shaft connecting final drive and driving wheel; it has stiffness of k_{drive}, and internal damping of c_{drive}.

$$T_{drive} = T_{wheel} = k_{drive}(\theta_{final} - \theta_{wheel}) + c_{drive}(\dot{\theta}_{final} - \dot{\theta}_{wheel})$$

Vehicle and wheels: a vehicle with mass m, and radius of wheel is r_{wheel}, inertia of wheel is J_{wheel}. The friction force exerting on the tires are such as air drag, rolling resistance, gravitational force.

$$\begin{aligned}&(J_{wheel} + mr_{wheel}^2)\ddot{\theta}_{wheel}\\ &= T_{wheel} - b_{wheel}\dot{\theta}_{wheel} - 0.5c_{drag}A_{frontal}\rho_{air}r_{wheel}^3\dot{\theta}_{wheel}^2\\ &- r_{wheel}m(c_{r1} + c_{r2}r_{wheel}\dot{\theta}_{wheel}) - r_{wheel}mg\sin(\alpha)\end{aligned}$$

4.1.2 Design system

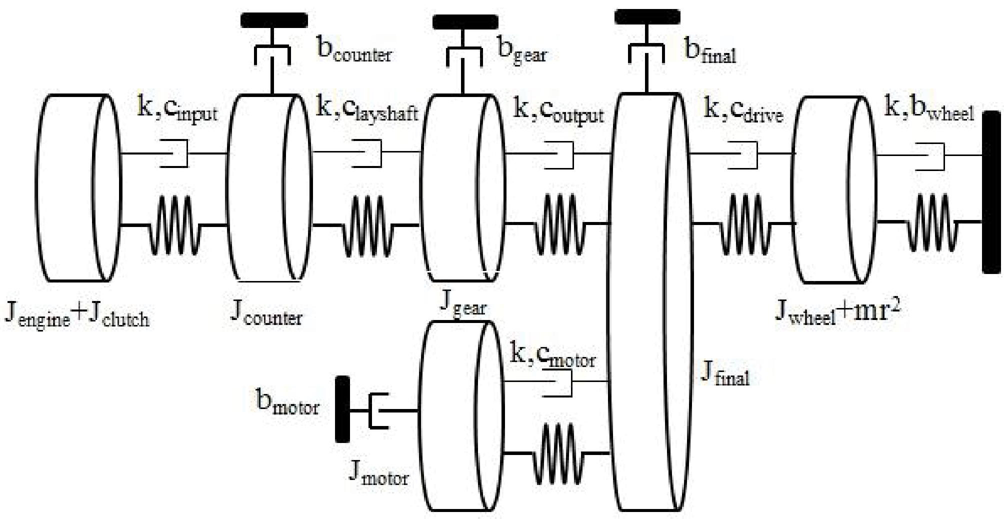

Figure 4.2 Design Systems of Driveline with inertias, stiffness and damping

In Chapter 3.3, Schematic of powertrain, the motor is mounted to pinion of final gear reduction by a mechanical shaft. Therefore, the design system contains 6 degrees of freedom. The equations of engine, counter gears and transmission gear are the same as original model. However, the equations of final gears are varied due to new component is added in.

Motor shaft: motor shaft is a shaft connecting final drive and electric motor; it has stiffness of k_{motor}, and internal damping of c_{motor}.

$$T_{motor,shaft} = T_{motor\rightarrow final}$$
$$= k_{motor}\left(\theta_{motor} - i_{final}\theta_{final}\right) + c_{motor}\left(\dot{\theta}_{motor} - i_{final}\dot{\theta}_{final}\right)$$

The electric drive unit is connected to the final drive gear and transfer driving torque by the motor shaft. Therefore, the equation becomes:

$$J_{final}\ddot{\theta}_{final} = T_{motor,shaft} - b_{final}\dot{\theta}_{final} - i_{final}T_{output} - i_{final}T_{motor\rightarrow final}$$

4.2 Equation of Motion

The equation of motion of the system is expressed as:

$$J\ddot{\theta} + C\dot{\theta} + K\theta = T$$

Where J, C and K are matrices assembled from corresponding. The equation of motion can be used for free vibration and force vibration analysis.

In the original drivetrain, when the clutch is engaged, the matrices of J, C, K and T are:

$$\begin{bmatrix} J_{\mathrm{engine}} + J_{\mathrm{clutch}} & & & & \\ & J_{\mathrm{counter}} & & & \\ & & J_{\mathrm{gear}} & & \\ & & & J_{\mathrm{final}} & \\ & & & & J_{\mathrm{wheel}} + m r_{\mathrm{wheel}}^2 \end{bmatrix}_{5\times 5}$$

$$\begin{bmatrix} C_{\mathrm{input}} & -i_{counter} C_{\mathrm{input}} & & & \\ -i_{counter} C_{\mathrm{input}} & i_{counter}^2 C_{\mathrm{input}} + C_{layshaft} + b_{\mathrm{counter}} & -i_{gear} C_{layshaft} & & \\ & -i_{gear} C_{layshaft} & i_{gear}^2 C_{\mathrm{layshaft}} + C_{output} + b_{\mathrm{gear}} & -i_{final} C_{output} & \\ & & -i_{final} C_{output} & i_{final}^2 C_{\mathrm{output}} + C_{drive} + b_{final} & -C_{drive} \\ & & & -C_{drive} & C_{\mathrm{drive}} + b_{\mathrm{wheel}} \end{bmatrix}_{5\times 5}$$

$$\begin{bmatrix} k_{\mathrm{input}} & -i_{counter} k_{\mathrm{input}} & & & \\ -i_{counter} k_{\mathrm{input}} & i_{counter}^2 k_{\mathrm{input}} + k_{\mathrm{layshaft}} & -i_{gear} k_{\mathrm{layshaft}} & & \\ & -i_{gear} k_{\mathrm{layshaft}} & i_{gear}^2 k_{\mathrm{layshaft}} + k_{output} & -i_{final} k_{\mathrm{output}} & \\ & & -i_{final} k_{\mathrm{output}} & i_{final}^2 k_{\mathrm{output}} + k_{drive} & -k_{drive} \\ & & & -k_{drive} & k_{\mathrm{drive}} + k_{\mathrm{wheel}} \end{bmatrix}_{5\times 5}$$

$$\begin{bmatrix} T_{\mathrm{engine}} - T_{\mathrm{f,engine}} \\ 0 \\ 0 \\ 0 \\ -r_{\mathrm{wheel}} F_{\mathrm{wheel}} \end{bmatrix}_{5\times 1}$$

When the electric motor is attached to the transmission by a shaft, matrices of J, T, K and C become:

$$\begin{bmatrix} J_{\mathrm{engine}} + J_{\mathrm{clutch}} & & & & & \\ & J_{\mathrm{counter}} & & & & \\ & & J_{\mathrm{gear}} & & & \\ & & & J_{\mathrm{final}} & & \\ & & & & J_{motor} & \\ & & & & & J_{\mathrm{wheel}} + \mathrm{m}r_{\mathrm{wheel}}^2 \end{bmatrix}_{6\times6}$$

$$\begin{bmatrix} C_{\mathrm{input}} & -i_{counter}C_{\mathrm{input}} & & & & \\ -i_{counter}C_{\mathrm{input}} & i_{counter}^2 C_{\mathrm{input}} + C_{layshaft} + b_{\mathrm{counter}} & -i_{gear}C_{layshaft} & & & \\ & -i_{gear}C_{layshaft} & i_{gear}^2 C_{\mathrm{layshaft}} + C_{output} + b_{\mathrm{gear}} & -i_{final}C_{output} & & \\ & & -i_{final}C_{output} & i_{final}^2 C_{\mathrm{output}} + C_{drive} + b_{final} + i_{final}^2 C_{motor} & -i_{final}C_{motor} & -C_{drive} \\ & & & -i_{final}C_{motor} & C_{motor} + b_{motor} & \\ & & & -C_{drive} & & C_{\mathrm{drive}} + b_{\mathrm{wheel}} \end{bmatrix}_{6\times6}$$

$$\begin{bmatrix} k_{\mathrm{input}} & -i_{counter}k_{\mathrm{input}} & & & & \\ -i_{counter}k_{\mathrm{input}} & i_{counter}^2 k_{\mathrm{input}} + k_{\mathrm{layshaft}} & -i_{gear}k_{\mathrm{layshaft}} & & & \\ & -i_{gear}k_{\mathrm{layshaft}} & i_{gear}^2 k_{\mathrm{layshaft}} + k_{output} & -i_{final}k_{\mathrm{output}} & & \\ & & -i_{final}k_{\mathrm{output}} & i_{final}^2 k_{\mathrm{output}} + k_{drive} + i_{final}^2 k_{motor} & -i_{final}k_{motor} & -k_{drive} \\ & & & -i_{final}k_{motor} & k_{motor} & \\ & & & -k_{drive} & & k_{\mathrm{drive}} + k_{\mathrm{wheel}} \end{bmatrix}_{6\times6}$$

$$\begin{bmatrix} T_{\mathrm{engine}} - T_{\mathrm{f,engine}} \\ 0 \\ 0 \\ 0 \\ 0 \\ -r_{\mathrm{wheel}}F_{\mathrm{wheel}} \end{bmatrix}_{6\times1}$$

The free vibration analysis, including undamped or damped vibrations, is used to determine the free vibration response of powertrain system. For undamped free vibration, it requires the system has no damping effect but stiffness. The expression of undamped free vibration of system is:

$$M\ddot{x} + Kx = 0 \text{ [43]}$$

where M is mass and K is stiffness

To determine natural frequencies and modal shapes, the acceleration and displacement vectors are defined as:

$$x = X\cos(\omega t + \phi)$$

$$\ddot{x} = -X\omega^2\cos(\omega t + \phi) \text{ [43]}$$

Substituting these two equations into matrix model, then it becomes

$$M(-X\omega^2\cos(\omega t + \phi)) + KX\cos(\omega t + \phi) = 0 \text{ [43]}$$

Assuming displacement, X and harmonic component, $\cos(\omega t + \phi)$ are not equal to zero, the equation then becomes

$$-M\omega^2 + K = 0 \text{ [43]}$$

The eigenvalues λ of matrix equation are the squared natural frequencies of the system. And the eigenvectors corresponding to each eigenvalue represent the modal shapes of the system.

Using state space method to determine natural frequency of a damped system, by taking inverse of the mass matrix, the unforced equation of motion is

$$\ddot{x} + M^{-1}C\dot{x} + M^{-1}Kx = 0 \text{ [43]}$$

If the state vectors are chosen as:

$$x = \begin{Bmatrix} x \\ \dot{x} \end{Bmatrix}, and\ \dot{x} = \begin{Bmatrix} \dot{x} \\ \ddot{x} \end{Bmatrix}$$

The state matrix is defined as:

$$A = \begin{bmatrix} 0 & I \\ -M^{-1}K & -M^{-1}C \end{bmatrix}$$

Where 0 represents a zero matrix with size of n by n, and n is the degrees of freedom, and I is the identity matrix. The eigenvalues of the state matrix are represented in pairs with both real and imaginary components, such as $\lambda = a \pm ib$, and the eigenvectors represent modal shape corresponding to a particular natural frequency. The natural frequency is the absolute value of eigenvalues, which is:

$$\omega_n = \sqrt{a^2 + b^2} \text{ [43]}$$

The damping ratio is then:

$$\xi = \frac{|a|}{\omega_n} \text{ [43]}$$

Damped natural frequency is:

$$\omega_d = \omega_n\sqrt{1 - \xi^2} \text{ [43]}$$

For the original drivetrain model, the natural frequencies of each gear ratio are

Mode	1st gear	2nd gear	3rd gear	4th gear
1	1.6305	2.1904	2.4256	2.5433
2	5.2243	5.9583	6.6058	7.1220
3	35.8867	32.6606	29.4420	27.0796
4	73.6990	52.8670	47.7728	45.9459
5	162.6349	162.5925	162.5835	162.5799

Table 4.1 Natural Frequencies of Original Drivetrain (Hz)

Mode	1st gear	2nd gear	3rd gear	4th gear
1	1.6287	2.1856	2.4189	2.5355
2	5.2160	5.9540	6.6036	7.1210
3	35.8683	32.6116	29.3816	27.0199
4	72.6761	52.7347	47.7250	45.9249
5	83.6688	82.7936	82.7240	82.7001
6	241.9938	241.9896	241.9887	241.9883

Table 4.2 Natural Frequencies of Design Drivetrain (Hz)

Compared with original drivetrain, the design system has one more degree of freedom so there is one more state in natural frequency; the high frequency response is higher than original drivetrain. For lower frequency response, there is no significant difference. Therefore, it can be concluded that inserting an electric motor with additional inertia does not have significant effect on low frequency response.

4.3 MATLAB control scheme

The modeling of this research is completely by MATLAB/Simulink, the flowchart of operation scheme is shown in Figure 4.3:

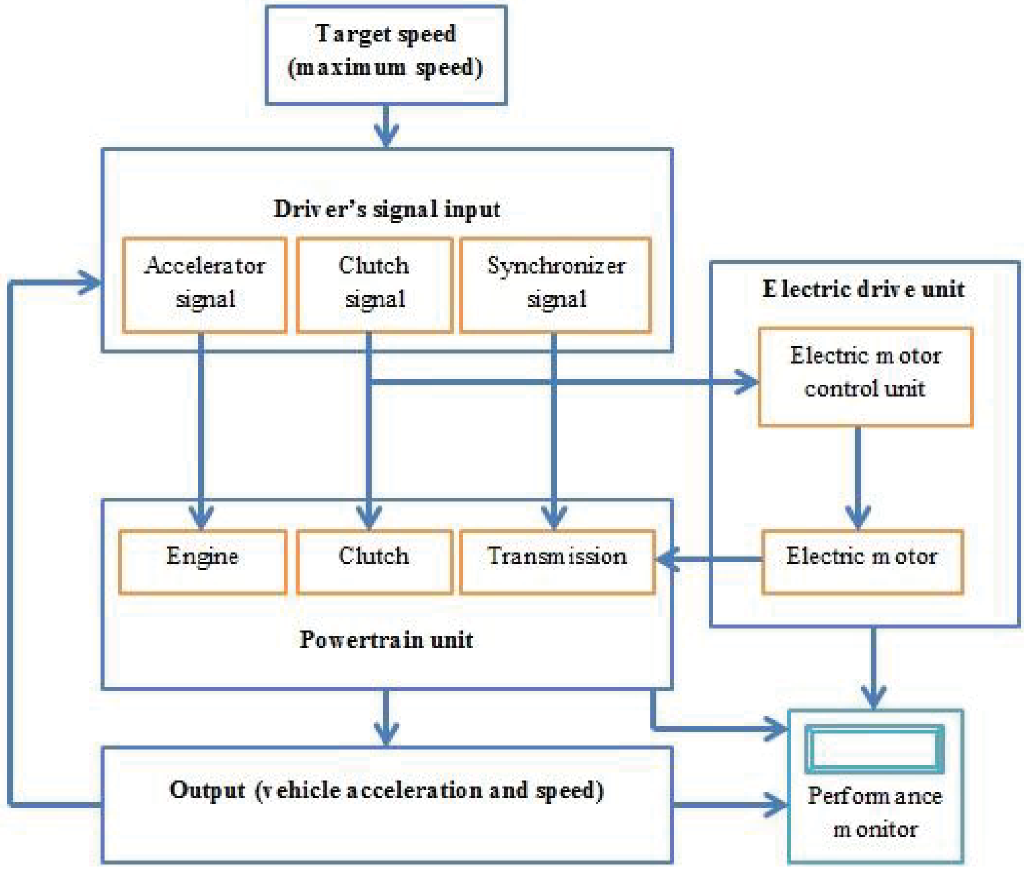

Figure 4.3 Flowchart of Operation Scheme

Firstly, the simulation is conducted based on target speed of vehicle. In this case, the target speed means to reach the top speed with throttle fully opened. The driver signals, such as throttle, clutch and synchronizer signals, initiate the simulation. Accelerator, clutch and synchronizer signals are sent to engine, clutch and transmission block in powertrain unit. The clutch signal is also connected to electric motor control unit. This clutch signal controls the electrical switch between motor and battery. Once the clutch signal exceeds its threshold value, the motor will synchronize the speed of output shaft and provide corresponding output torque based on its performance map. The output torque is applied to output shaft of transmission block and this affects the performance of transmission. The outputs of powertrain unit are vehicle acceleration and speed. These values are fed back to driver's input signal for deciding when to shift to the next gear. Performance monitor, which is made of some scopes, is used to monitor the performance of components in powertrain unit, electric drive unit and vehicle body.

The simulation results of this research are extracted from performance monitor for comparison and discussion.

4.4 Simulink Model

The simulation model is built with SIMULINK/MATLAB for algorithm. This SIMULINK model can derive the numerical solution of manual transmission with electric torque assist and thus to compare the result with original manual transmission. In this section of thesis report, the SIMULINK model is divided into blocks and the methods of building each model block are explained.

4.4.1 Driver Inputs

A single signal builder block is used to make the driver inputs. The signal builder block allows to create interchangeable groups of piecewise linear signal sources and use them in a model. The signal builder block outputs a virtual nonhierarchical bus, scalar or array of real signals of type double.

The input signals arrangement is illustrated in Figure 4.4:

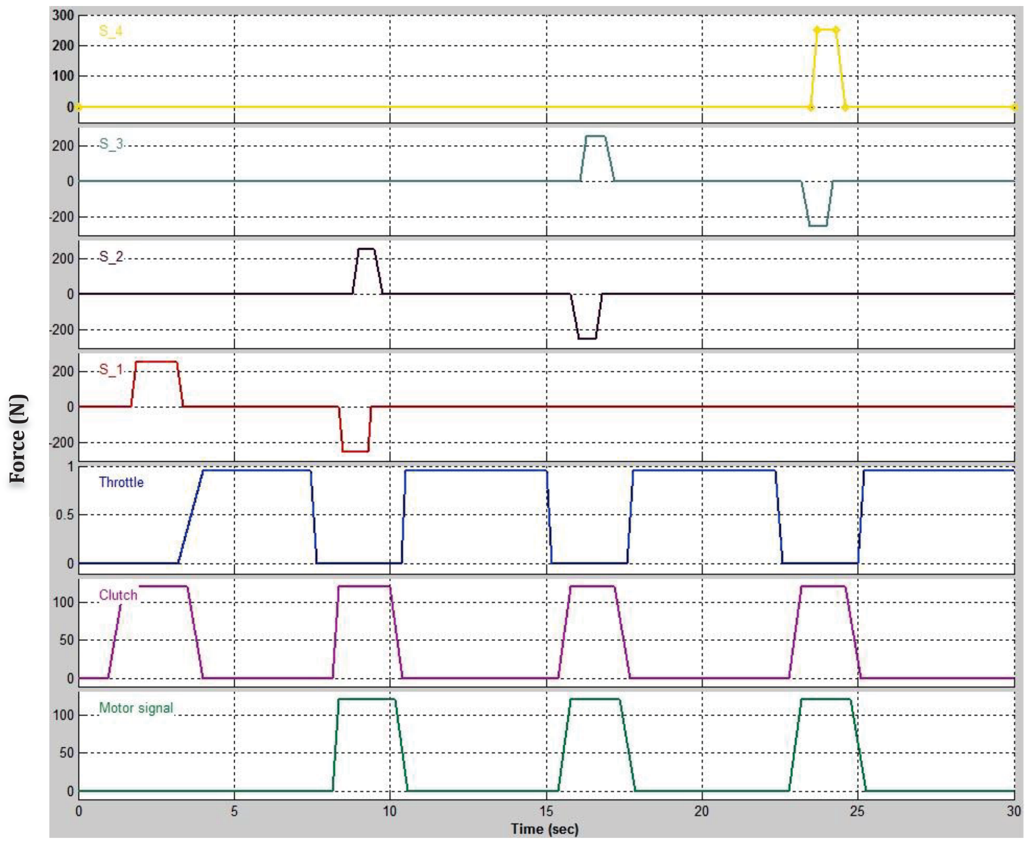

Figure 4.4 Drive Input Signal

In Figure 4.4, the first four signals, S_1, S_2, S_3, S_4, represent input signals to the synchronizers in transmission box. At the beginning, the clutch signal starts from 0, and then increases to 150N, this simulates that the driver depresses the clutch pedal by 150mm, and thus the primary is disengaged. At this time, the throttle stays at 0, which means driver has not depressed accelerator pedal. The gear selection signal S_1 is 0 while the clutch pedal is depressed, and then signal S_1 increases to 250N within 0.15 second, this represents that the driver applies force on the shift rod to push the synchronizer ring and cone into the hub, and then locks the mesh gear onto shaft. For the first gear, the motor signal stays at 0 because electric motor is not required to operate when vehicle is at stationary for the safety concern. After the shift signal S_1 returns to 0, the gear shift is successfully switched to 1st gear, the throttle signal starts increasing to 0.95, which stands for 95% of throttle opened, and at the same time of throttle increasing, the clutch signal is gradually decreased to 0 to until it is finally engaged.

At 7.5 seconds, the throttle drops to 0 for upshifting to the next gear. The clutch signal increases to 150N again to disengage the clutch. When the clutch is fully disengaged, the shift signal S_1 is decreased to -250N to pull synchronizer ring and cone away from hub and unlock mesh gear. At the same time, shift signal S_2 is increased to 250N to lock the 2nd gear onto the output shaft. Differing from the 1st gear, the motor signal starts operating to switch on electric motor while shifting to 2nd gear, this motor signal is generated by control system based on clutch signal operated manually by driver. It synchronizes clutch signal but the threshold value is 1N, so when the positive signal exceeds 1N, the switch in electric drive unit is closed, and motor starts operating to provide torque assist. The motor transfers torque to output shaft and this only lasts until the clutch is fully engaged. For the 2nd, 3rd and 4th gears, motor and clutch signals are exactly the same. In the shifting schedule, the driver upshifts to 1st gear at 3 seconds, to 2nd gear at 8 seconds, to 3rd gear at 16 seconds and to 4th gear at 23 seconds. Also, throttle is opened to 95% between each time of upshifting.

The total simulation time is 30 seconds, and it simulates an ordinary driver driving a vehicle with manual transmission starts from rest and shift from neutral state to 4th gear.

The motor signal synchronizes clutch signal to operate the add-on electric motor for filling in torque hole.

4.4.2 Engine model

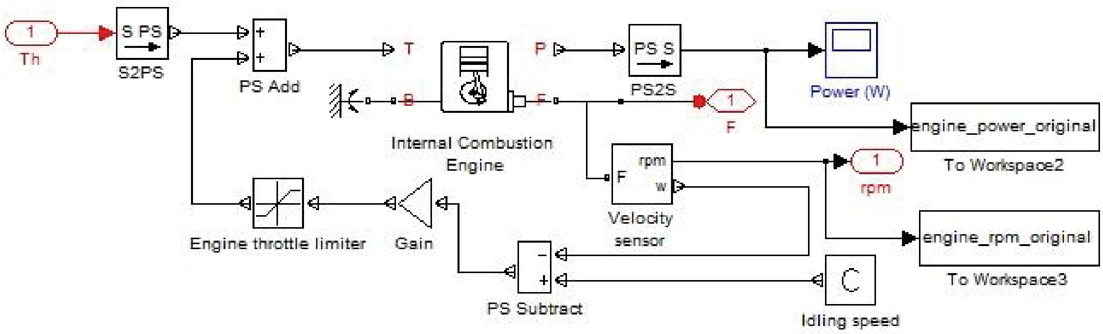

Figure 4.5 Schematic of Engine Model

The engine model has characteristics of normal spark-ignition internal combustion engine. One of the input ports T is connected to driver signal block; this signal is a physical signal and it must lie between 0 and 1, if it is above 1 or below 0, it is interpreted as 1 or 0. The signal controls engine to generate torque with respect to its specified rev speed. However, it does not include efficiency of a normal combustion engine. The output port P represents physical output data of engine power, it is used to monitor the power of engine, in this simulation, and it has no other use.

The vector of revolution speed values of engine is an independent variable, and the unit is revolution per minute (rpm). In this vector, the lower limit of speed vector represents idling speed. If the revolution speed of engine falls below the idling speed, the engine torque is blended to zero. If the speed exceeds the maximum engine speed, the simulation will stop as a simulation error. The torque vector, with a unit of newton.meter (N.m), is a dependent variable based on revolution speed. For the torque-RPM function, linear interpolation method is implemented between discrete relative speed values.

The inertia and initial speed are also specified in the engine model, they are constant of 0.4 kg.m^2 and 500 rpm, respectively. The rotational inertia is due to the crankshaft and a small portion of engine power is consumed to overcome this inertia.

The connections ports F and B are mechanical rotational conserving ports associated with the engine crankshaft and engine block, respectively. Connection port B is linked to a mechanical rotation reference that maintains at 0rpm until simulation is complete. Connection port F is connected to clutch unit and it transfers rotation motion from internal combustion to clutch flywheel.

In the engine model block, the velocity sensor is added; it observes engine speed and imports the data to MATLAB workplace. Also, the engine speed signal is used to control engine throttle. In the configuration, to avoid simulation error, if the engine speed is lower than idling speed, the throttle will be adjusted to higher to satisfy the speed requirement.

In the engine block, the engine model is defined as a function of power P(ω). The function has a maximum power for a given engine speed ω. Based on a normalized throttle input signal, the power (kW) is delivered as

$$Power(\omega) = \frac{T(\omega)\times\omega}{9{,}515}, where\ 9{,}515\ is\ a\ unit\ conversion\ constant\ [44]$$

The power of engine is to show the output performance of an engine. Usually, a dynamometer is implemented to place a load on the engine and measure the amount of power that the engine can produce against the load. As power is force multiplied by speed, power increases with rotational speed up and passes the point of maximum torque. However, at higher rotational speed, the engine performance starts to be limited by the amount of air that it can be drawn in and the torque then decreases more rapidly than the rotational speed increased, and therefore power also decreases [45]. The maximum acceleration is obtained by having maximum propulsive force at the wheels. At a given speed, maximum acceleration is obtained by having the engine operates at maximum power [46]. An engine performance curve of Ford Falcon 2000 is shown in Figure 4.6:

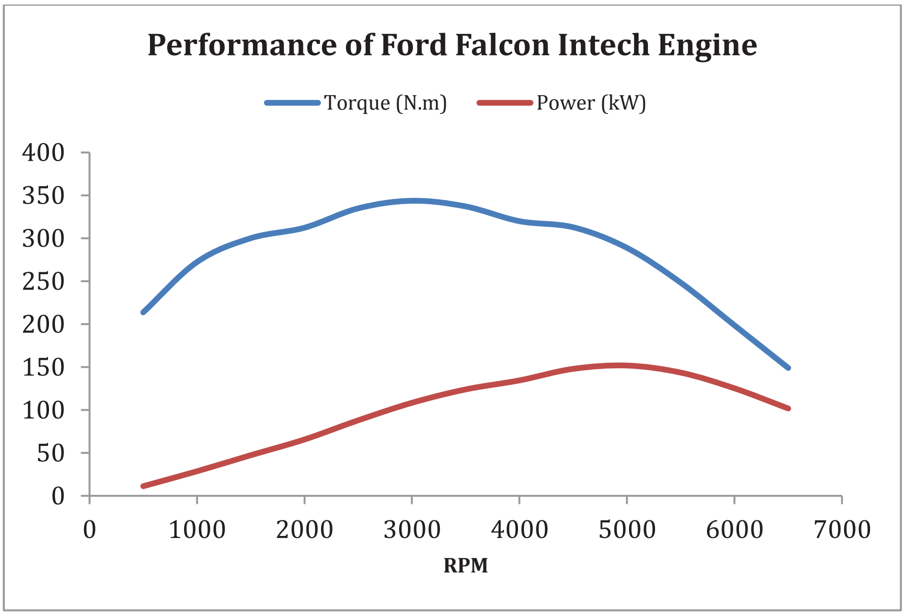

Figure 4.6 Engine Performance Curve of Ford Falcon Intech 2000

4.4.3 Clutch Unit

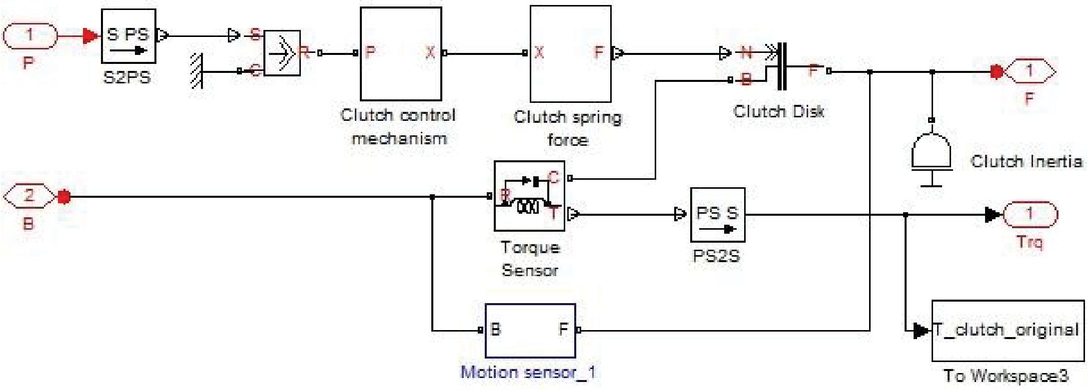

Figure 4.7 Clutch Model

The clutch block is modeled as an ordinary mechanical device in every vehicle with manual transmission; the clutch pedal, clutch mechanism and friction disk are built. The connection ports of clutch block consist of two input ports and two output ports. The connection ports B and F refer to mechanical rotational conserving ports associated with the engine crankshaft and input shaft, respectively. Connection port B connects clutch flywheel and engine crankshaft; it has the same rotational speed as the engine and there is no rotational damping and stiffness between clutch hub and engine. Connection port F is connected to input shaft of gearbox unit that provides rotational damping and stiffness.

The connection port P represents the input signal from driver input signal block, the clutch signal is used to control the engagement status. The unit of clutch signal indicates how much the driver displaces the clutch pedal. In the clutch control mechanism, it has three lever systems: foot lever, withdrawal lever and release lever. One end of foot lever has a pedal stop that sets up a final position of foot lever, in this case the maximum available displacement is 170mm. Also, a spring is attached to the foot lever for returning foot lever to its original position. The other end of foot lever is connected to the withdrawal lever that is attached to a mass withdrawal bushing with a viscous damping of 100N/(m/s). This can reduce the backlash due to vibration of engaging clutch. The release lever is connected to the pressure plate that is free to slide when clutch is disengaged; it causes displacement of pressure plate with a mass of 4kg. Then this value of displacement is transferred to clutch spring force block.

In the clutch spring for block, the clutch normal force F is determined by the clutch spring compression. When the spring is compressed, the force is determined by spring constant k. Port X corresponds to the clutch plate translational motion. The measurement P is initially zero, this corresponds to the spring being fully compressed and the clutch fully engaged. Depressing the clutch pedal reduces the clutch spring compression and hence the clutch force. When the clutch plate displacement is less than -3mm, the clutch force is zero, and hence it is fully disengaged.

The clutch spring force is transferred from clutch spring force block to clutch disk block. The clutch disk is modeled as a loaded-contact rotational friction model with an effective torque radius of 300mm. It represents a model of a loaded-contact rotational friction that transmits torque between two rotating shafts. The friction is applied when the normal force presented at the physical signal port N exceeds the threshold force that is 10N in this model. For the shafts to lock, the relative bass-follower speed must be less than the velocity tolerance, 0.001rad/s, and the transmitted torque must be less than the static friction limit, 0.5. If the torque transmitted does not exceed the static friction limit, the shafts will remain locked. The clutch inertia is set to be 0.005 kg.m^2.

The default state of the clutch is engaged and that means the connection between engine and transmission gearbox is always on unless the pedal is depressed. If the engine is at idling and the transmission is in neutral state, the engine spins the input shaft of the transmission but there is no power transmitted to wheels. And the driver only sends clutch signal when vehicle is at idling and gear shifting.

4.4.4 Electric Drive Unit

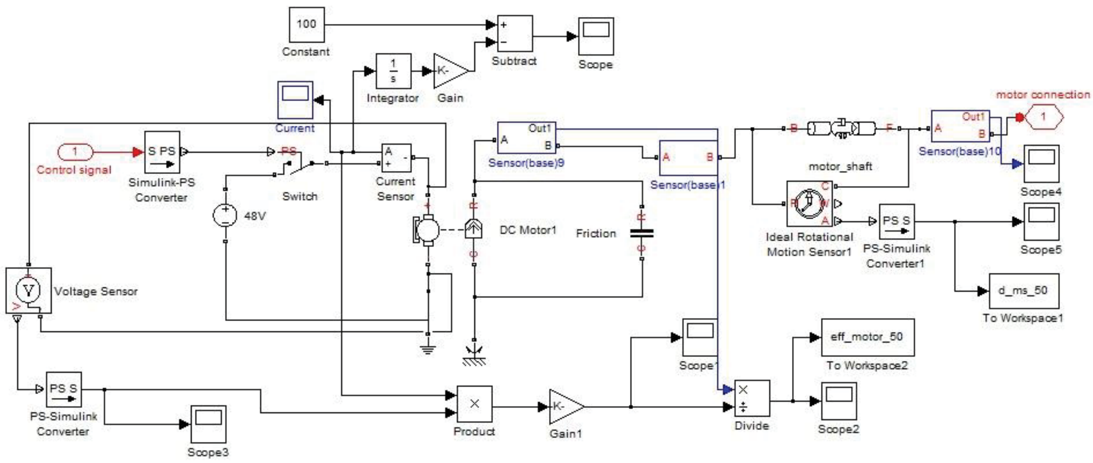

Figure 4.8 Electric Drive Unit Model

Electric drive unit consists of a battery, electric switch and a DC motor. Beside the mechanical connection, it also has an electric circuit because the power source is an electric source.

The DC motor block represents the electrical and torque characteristics of a DC motor using equivalent circuit model in Figure 4.9:

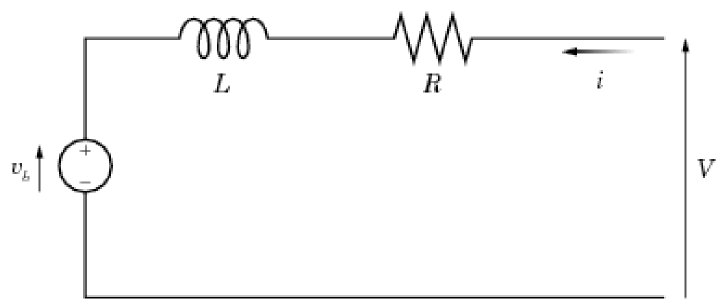

Figure 4.9 Circuit Model of DC Motor Block [47]

The torque-speed characteristic for the DC motor block is related to Figure 4.8. The model parameterization is set to by stall torque and no-load speed, the block solves for the equivalent circuit parameters by:

$$T = \frac{k_t}{R}(V - k_v\omega) - J\dot{\omega} - \lambda\omega, \ [47]$$

$$where\ J\ is\ motor\ inertia, \lambda\ is\ damping, k_t\ is\ torque\ constant$$

The motor block assumes that no electromagnetic energy is lost, and hence the back-electromagnetic force and torque constants have the same numerical value when in SI units. Motor parameters can either be specified directly, or derived from no-load speed and stall torque. When positive current flows from the electrical + to – ports, these two ports are connected to electric circuit with battery and switch. A positive torque acts from the mechanical C to R ports. Altering the sign of the back- electromagnetic force or torque constants can change motor torque direction. The mechanical conserving port C is connected to a mechanical rotational reference because the motor is not required to transfer power from one component to another. The port R is connected to output shaft of transmission gearbox. It transfers the rotational motion and torque to output shaft while shifting. The rotor inertia is 0.025kg.m^2, damping is 1e-8N.m/(rad/s) and initial speed is 0rpm. Also, an external friction block is added into mechanical connection to simulate mechanical friction generated by motor. The parameters have 3e-2N.m of breakaway friction torque and 3e-2N.m of Coulomb friction torque.

In the circuit of electric drive unit, there is a switch to open or close the circuit. The switch block models a switch controlled by an external physical signal. If the external physical signal is greater than the value specified in the threshold parameter, and then the switch is closed, otherwise the switch is open. The closed resistance is defined by parameter R, which is set to be 0.00001ohm, and parameter G, which is 1e-8/ohm, defines open conductance, so in this case, the resistance of switch is neglected. The threshold of switch is 1. It is based on the motor signal in the driver inputs block. Once the motor signal exceeds 1, the switch will close and motor starts operating.

The battery of this circuit is modeled as a DC voltage source block; it represents an ideal voltage source that has specified voltage at its output regardless of the current flowing through the source. The output voltage of this voltage source is set to be 48V. There are only two connection ports on voltage block, connections + and – are conserving electrical ports corresponding to the positive and negative terminals of the voltage source. The current flows from positive to negative and in this case it is monitored by current sensor of the circuit. The circuit is connected to an integrator block to integrate the current flow to get the energy consumed, and then subtract from initial capacity of battery to get the instant capacity of battery.

There is a mechanical shaft that connects motor and pinion of final drive mesh gear. The shaft provides stiffness and damping that affect torque transmission, the dimensions and other characteristics are indicated in Table 4.3.

4.4.5 Gear Box

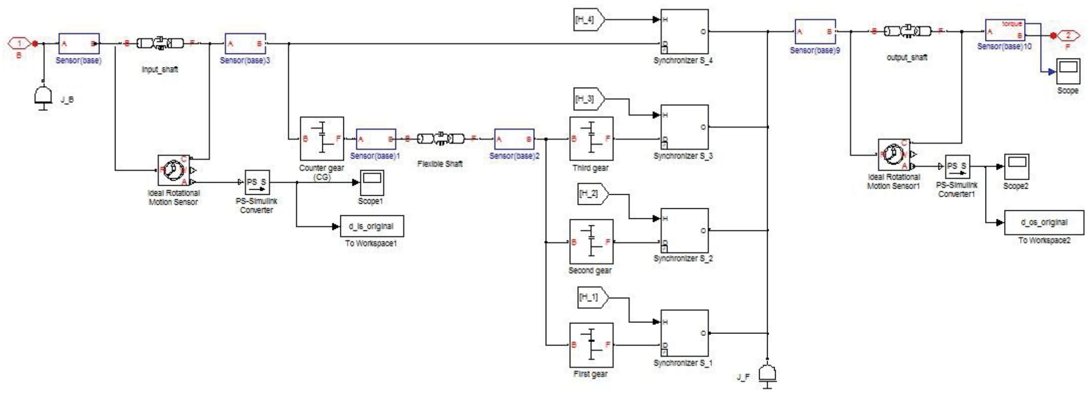

Figure 4.10 Gearbox Model

Several different SIMULINK blocks are used to model the gearbox model. Mechanical rotational inertia blocks can simulate the inertia of gearbox. It is a mechanical rotational conserving port with one end connected to the gearbox system. The block positive direction is from its port to the reference point. And this means that the inertia torque is positive if the inertia is accelerated in the positive direction. There are two rotational inertia blocks implemented in this system, the inertias added into systems are $0.026 kg.m^2$ and $0.034 kg.m^2$.

The shafts in gearbox system are modeled as flexible shafts. A flexible shaft block can simulate torsional dynamics of a flexible shaft. The shaft is assumed as an approximate lumped parameter model consisting of finite number of segments. The block is parameterized by material properties. The shaft is specified by its size and continuum properties. The shafts have parameters of length, diameter, density, shear modulus and damping ratio. In this driveline system, there are five solid shafts: input shaft, layshaft, output shaft, drive shaft and motor shaft. Their parameters are indicated in Table 4.3.

	Length (m)	Inner Diameter (m)	Outer Diameter (m)	Material Density (kg/m^3)	Shear Modulus (Pa)
Input shaft	0.2	0.015	0.030	7.8e+3	7.93e+10
Layshaft	0.75	0.015	0.030	7.8e+3	7.93e+10
Output shaft	0.45	0.015	0.030	7.8e+3	7.93e+10
Drive shaft	0.75	0.05	0.075	7.8e+3	7.93e+10
Motor shaft	0.2	0.015	0.030	7.8e+3	7.93e+10

Table 4.3 Parameters of Shafts

Based on above information, the shaft stiffness and inertia can be computed by following relationships:

$$J_P = (\pi/32)(D^4 - d^4)$$

$$m = (\pi/4)(D^4 - d^4)\rho L$$

$$J = (m/8)(D^4 + d^4) = \rho L \cdot J_P$$

$$k = J_p \cdot G/L \quad [48]$$

A gear reduction block models the mesh spur or helical gear in transmission. The simple gear block constrains two connected driveline axes. The connection ports B and F represent rotational conserving ports of base and follower to rotate in the opposite direction with a fixed gear ratio as specified. As one of the important parameters of mesh gear, gear ratio represents the ratio of rotational speeds of gear and pinion. For example:

$$g_{FB} = r_F/r_B = N_F/N_B \quad [49]$$

The most complicated block is the synchronizer block, it consists dog clutch and cone clutch and some other sensor blocks. The cone clutch is a friction clutch that transfers torque between two components by coupling them with friction. The cone clutch block has a friction clutch based on conical surfaces in contact with kinetic friction and static friction. The rotational conserving ports B and F connect shaft and gear. The shift signal is sent signal to control the clutch by applying normal force on the friction surfaces in contact. The parameters of cone clutch are in Table 4.4:

Contact maximum diameter	60mm
Contact minimum diameter	50mm
Cone half angle	15 degrees
Kinetic friction coefficient	0.5
Static friction coefficient	0.52

Table 4.4 Parameters of Cone Clutch

The cone clutch is used to model a synchromesh that equalizes the speeds of two shafts prior to engage a dog clutch. A dog clutch will be locked after the cone clutch is engaged. A dog clutch has two components, one is fixed to spinning driveshaft; the other one is mounted freely on the same shaft. The clutch engagement is measured by shift linkage position. The motion of clutch is measured as the velocity of ring relative to hub. To engage the clutch, the ring must be translated towards the hub until the teeth mesh. Connection S is a mechanical translational conserving port representing the ring handle. A normal force is applied to the engage the clutch. The port X is linked to driver inputs signal and control the shift linkage position. When signal is 0, the clutch is fully disengaged. As the shifting signal increases, the dog clutch will engage and start transmitting torque from mesh gear to drive shaft. In the parameters set-up of the dog clutch, the limit of the clutch teeth mean radius is 50mm.

Number of teeth	6
Rotational backlash	20 degrees
Clutch teeth mean radius	50mm
Torsional stiffness	1e+5
Torsional damping	2000N.m/(rad/s)
Tooth height	10mm

Table 4.5 Parameters of Dog Clutch

After the dog clutch is fully engaged, the gear is locked onto shaft and there is no speed difference. One pair of gear reductions is used to transmit torque from engine to vehicle wheels.

4.4.6 Final Drive Unit

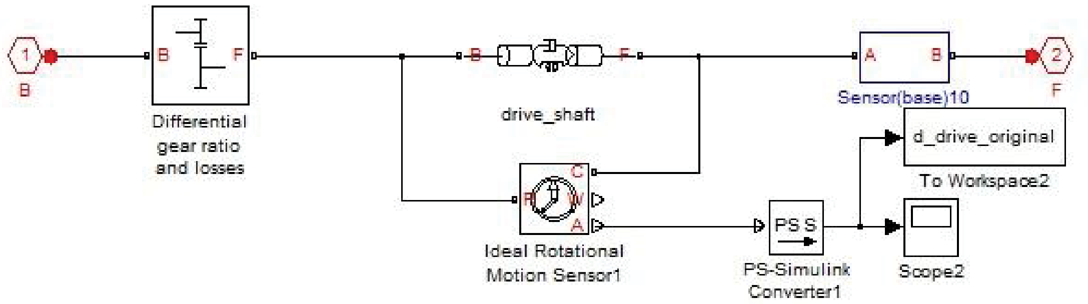

Figure 4.11 Final Drive Unit

The final drive unit simply consists of a differential gear ratio and a flexible shaft, which is named as drive shaft. The differential gear block is modeled by a simple gear block of SIMULINK, its ratio between base and follower gears is 3.1:1, with constant efficiency of 0.86. The dimension and mechanical properties of drive shaft are indicated in Table 4.3. The port B is connected to gearbox block and port F is linked to vehicle body block.

4.4.7 Vehicle Body

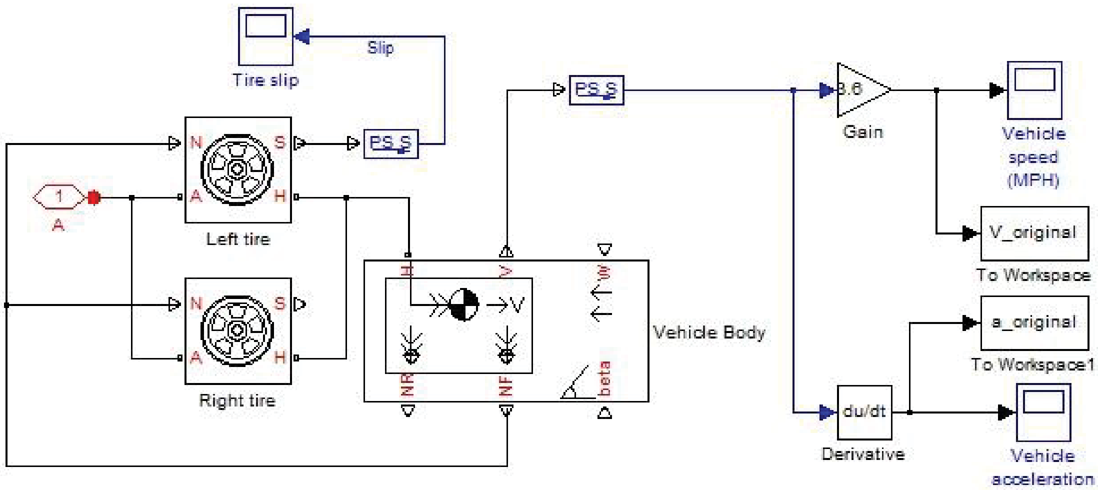

Figure 4.12 Vehicle Body Model

The vehicle body model simulates longitudinal dynamics of a vehicle body with rotational power transmitting to vehicle wheels. The vehicle wheel is modeled as a tire block with type dynamics formula. It represents the longitudinal behavior of a tire characterized by peak longitudinal force and corresponding slip with the effects of tire inertia, stiffness, and damping. The connection port A is a mechanical rotational conserving port connected to the drive shaft of driveline. The port H is a mechanical translational conserving port for the wheel hub through which the thrust developed by tire is applied to the vehicle. Connection N is a physical signal input port that applies the positive and downward normal force on the tires. Port S is a physical signal output port that records the tire slip; the signal will be transferred to a scope that demonstrates tire slip numerically.

The tire model is parameterized by characteristics such as peak force and corresponding slip. In the parameters of tire model, the rated vertical load is set as 3000N; peak longitudinal force at rated load is 3500N; slip at peak force at rated load is 10%; rolling radius is 0.4m and inertia is 0.8kg.m^2.

The vehicle body block is a two-axle vehicle body model with longitudinal dynamics and motion and adjustable mass, geometry and drag properties. The model has two equally size wheels connected to the front wheel port. The block accounts for body mass, aerodynamic drag, road incline, and weight distribution between axles due to acceleration and road profile. The vehicle does not have vertical displacement relative to ground level. The connection port H represents mechanical translational conserving port associated with the horizontal motion of the vehicle body. The resulting traction motion developed by tires could be connected to this port. Connections V, NF are physical signal output for vehicle velocity and front normal positive downward wheel force. Port W and beta are physical signal inputs for headwind speed and road inclination angle; in this case, there is no wind speed and inclination angle considered. The parameters of vehicle model are in Table 4.6:

Mass (kg)	1200
Distance from CG to front axle (m)	1.4
Distance from CG to back axle (m)	1.6
CG height (m)	0.5
Frontal area (m^2)	3
Drag coefficient	0.4

Table 4.6 Parameters of Vehicle Body

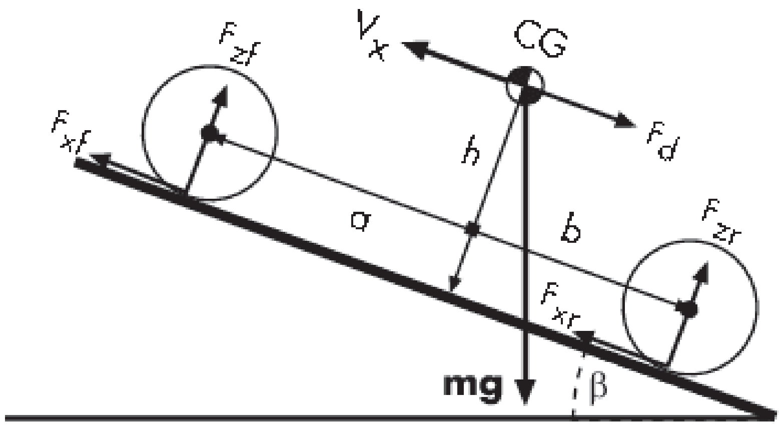

Figure 4.13 Vehicle Dynamics and Motion [50]

Figure 4.13 illustrates the vehicle dynamics and motion. The net forces and torques acting on vehicle body can determine the vehicle motion. For example, the longitudinal tire forces push the vehicle forward or backward. Depending on the vehicle parameters, the normal forces are computed by following equations:

$$ma_x = F_x - F_d - mg \cdot \sin\beta$$

$$F_x = n\left(F_{xf} + F_{xr}\right)$$

$$F_d = \frac{1}{2} C_d \rho A (V_x - V_W)^2 \cdot sign(V_x - V_W)$$

$$F_{zf} = \frac{-h(F_d + mg\sin\beta + ma_x) + b \cdot mg\cos\beta}{n(a+b)}$$

$$F_{zr} = \frac{+h(F_d + mg\sin\beta + ma_x) + a \cdot mg\cos\beta}{n(a+b)} \quad [50]$$

The limitation of this vehicle body model is that it does not simulate vertical displacement. It only simulates longitudinal dynamics that is parallel to ground and oriented along the direction of motion.

5 Simulation Results

5.1 Simulation Results of Original Driveline

The driveline of a conventional manual transmission is modeled for comparing with designed model. The original manual transmission model is similar to the design but without electric drive units.

5.1.1 Engine and Clutch

The shift schedule is in Figure 5.1. Compared with Figure 4.4, they have the same shift schedule and clutching time but the input signal in Figure 5.1 does not have electric drive control signal.

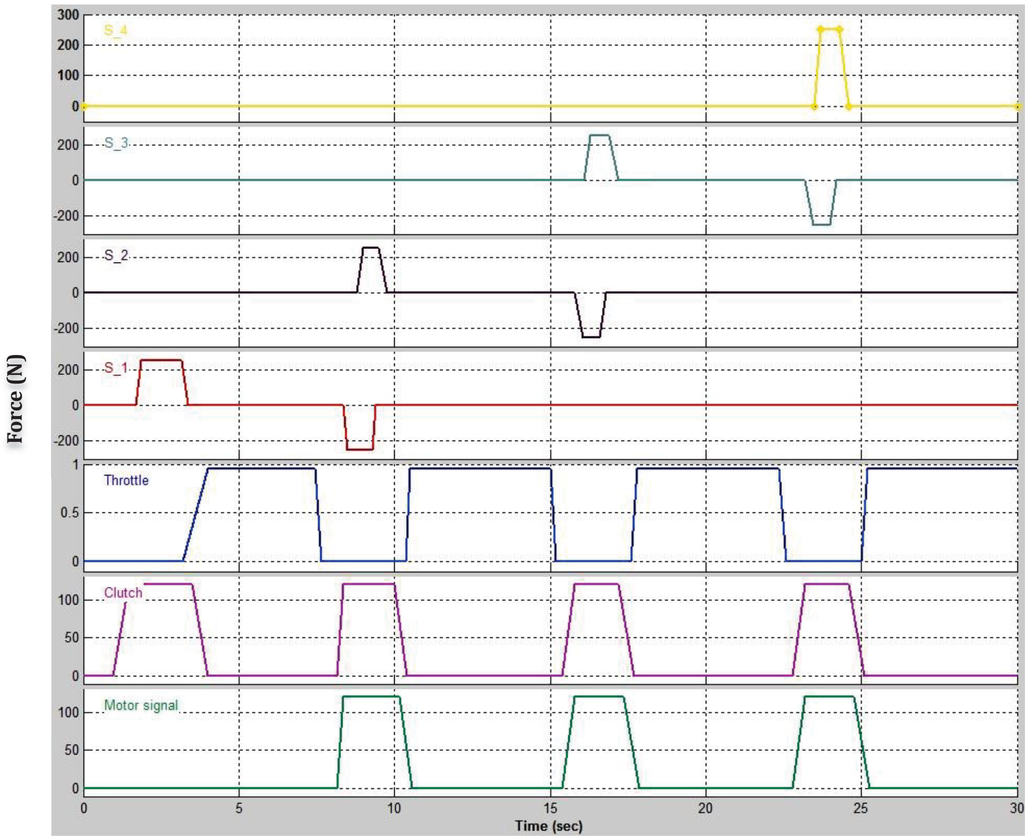

Figure 5.1 Shift Schedule of Original Driveline

The time span for each time of upshifting is at below:

1st to 2nd: 8.15 to 10.4 seconds

2nd to 3rd: 15.4 to 17.7 seconds

3rd to 4th: 22.8 to 25.1 seconds

The time span is measured based on clutch input signal, when the magnitude of clutch signal starts increasing, the driver starts upshifting. At the point where clutch signal returns to 0, the upshifting is completed.

The output of simulation results represent by plots of torque and revolution speed of shafts.

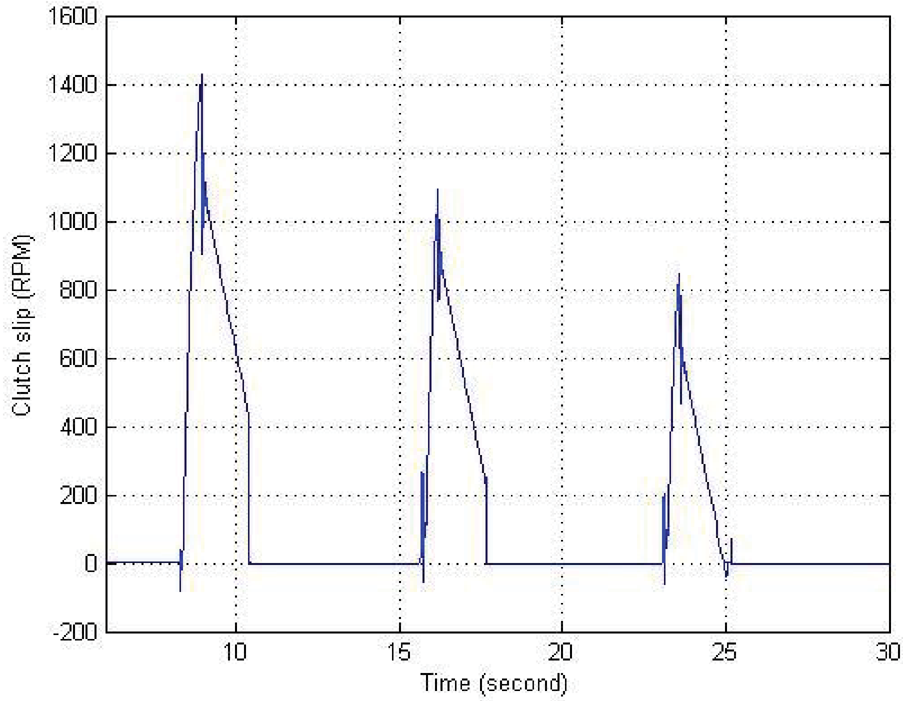

Figure 5.2 Clutch Slip versus Time

Figure 5.2 represents the slip of clutch; it illustrates the speed difference between base and follower of the clutch, where $\omega_{slip} = \omega_{base} - \omega_{follower}$. The clutch slip is below 2000 rpm. At 95% throttle, when clutch is disengaged, clutch slip is increased. Magnitude of clutch slip depends on timing of throttle and clutch engagement. When the value of clutch slip drops to zero, for example between 12 and 15 seconds and that means clutch is fully engaged by its static friction. The clutch slip rises again when the next shift starts. When upshifting from 1st to 2nd gear, clutch slip increases to 1,427 rpm. This value decreases to about 1,092 rpm while shifting to 2nd and 3rd gears. Clutch slip for upshifting from 3rd to 4th gear is 843 rpm. Therefore, this plot shows that as the transmission is switched to higher gear, the clutch slip is decreased.

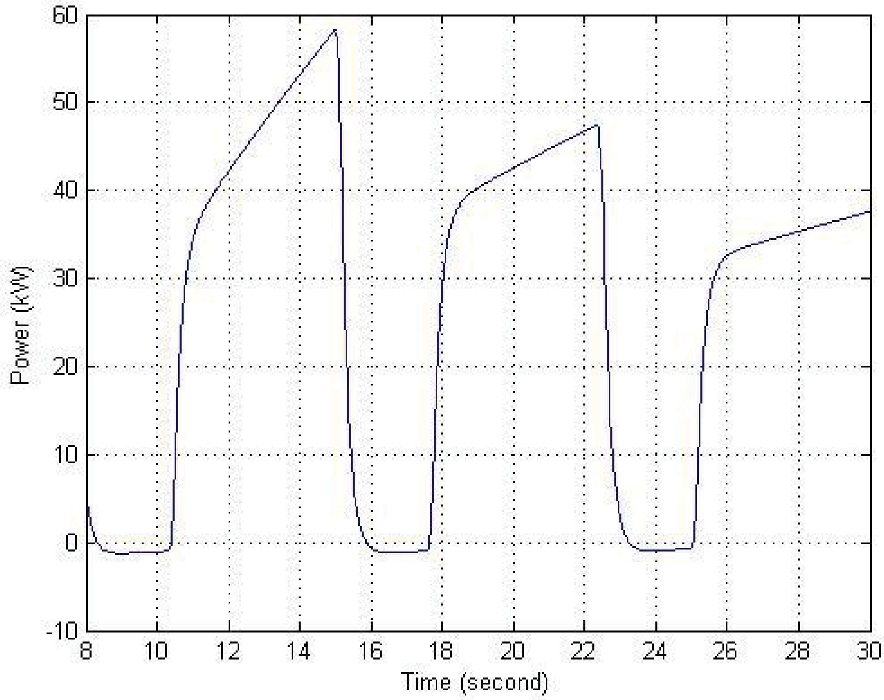

Figure 5.3 Engine Power versus Time

Figure 5.3 shows the output power of the engine. During gear shifting, the power drops to below 0 when throttle is fully released and there are some power losses by inertia of system and internal friction. At higher gear, there is less output power transferred from internal combustion engine based on the engine performance curve in Figure 4.6. After the peak torque value, engine power output decreases rapidly as the throttle is released.

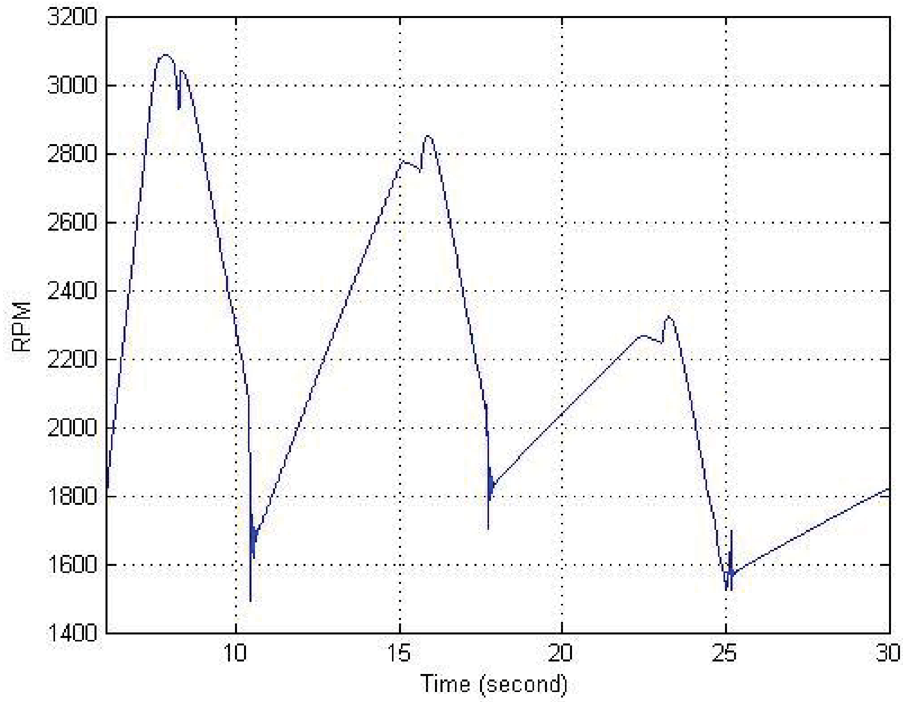

Figure 5.4 Engine RPM versus Time

Figure 5.4 represents the engine speed. It starts from 500 rpm that is set as idling speed, so the engine speed will never drop below this idling speed. The maximum revolution speed reaches 3,100 rpm for 1st gear. When the throttle is opened, the engine speed will increase, and if the throttle is released, engine speed is decreased. From the plot, the magnitude of engine speed is either increasing or decreasing, and there is no constant zone because the vehicle is either accelerating or braking through the simulation unless the power of aerodynamic drag and the output power of vehicle are balanced.

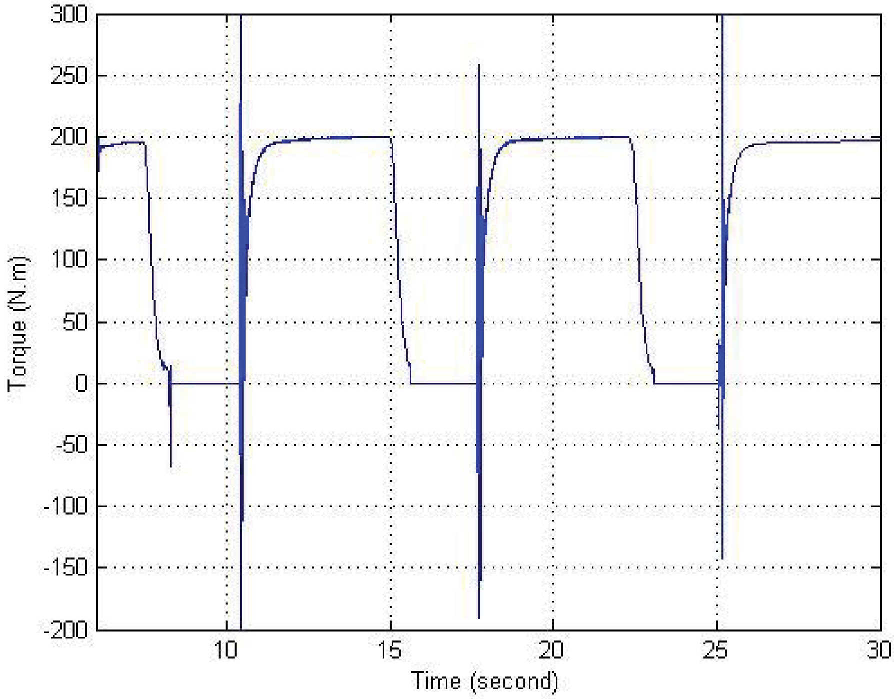

Figure 5.5 Clutch torque versus Time

Figure 5.5 represents the torque transferred from the clutch. The clutch torque is inconsistent and torque holes are illustrated. When clutch is disengaged, the torque drops to 0. When clutch is engaged, torque transferred start increasing immediately, and there is an overshoot in transient response of torque. In manual transmission, the clutch cannot be engaged smoothly with perfect desired rotational speed. Therefore, vibration happens when the clutch is engaged. In other types of transmission, such as automatic, automated and dual clutch transmission, this torque fluctuation is efficiently improved. In manual transmission, it can be improved by depressing clutch pedal more smoothly and slowly. Also, the engine speed and transmission gearbox output speed are influenced as well, because the flywheel of clutch is attached on the engine and the clutch disk are connected to input shaft of gearbox. If the difference between engine speed and gearbox speed is too big, the vibration on the clutch will be more serious when it is being engaged. If this difference is very small, then this torque oscillation can be reduced. In reality, only experienced driver can conduct clutch engagement and gear shifting smoothly, and this can increase lifetime of clutch and decrease the rate of wear of clutch disk.

5.1.2 Input Shaft

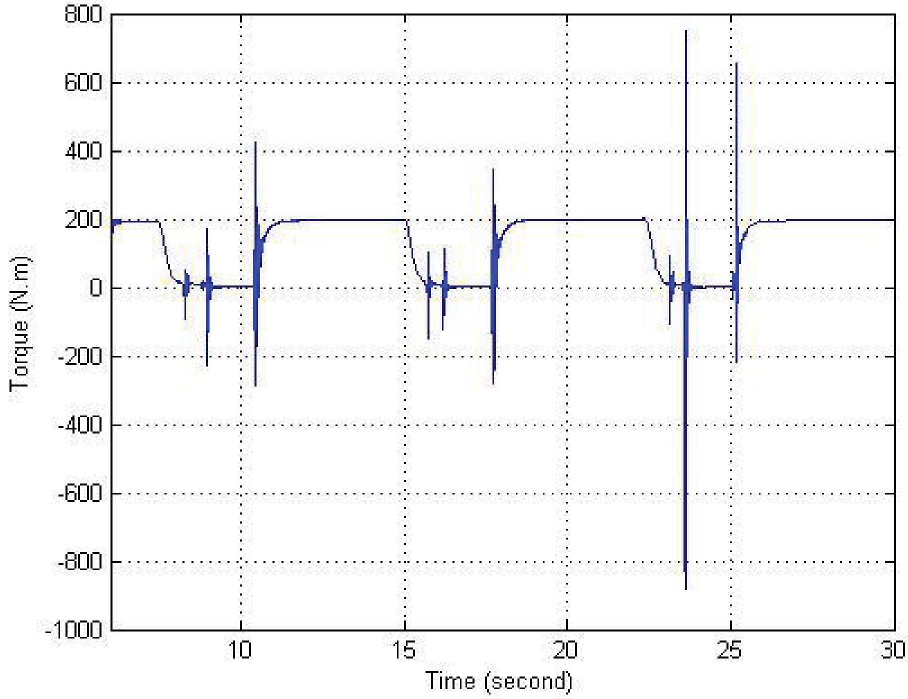

Figure 5.6 Input Shaft Torque versus Time

Figure 5.6 shows the plot of input shaft torque versus time, it represents the torque applied on input shaft. The oscillation of torque value is very easy to identify. For upshifting from 1st to 2nd gear, the fluctuation occurs at 8.3 seconds. The excitation happens when the clutch is disengaged. Then there is another overshoot when synchronizing gears. The settling time of transient response is very short due to stiffness and damping of driveline shafts. The design operation mode states that the electric motor does not operate when transmission is upshifted from neutral to 1st gear. Therefore, the system response from t=0 to 6 seconds will not be focused in further analysis. For shifting from 3rd to 4th gear, synchronizing gear causes the highest torque overshoot because there is a rough oscillation due to inappropriate shaft speed for gear synchronization. The magnitude of torque overshoot is uncontrollable because shifting is totally based on manual operation by the driver.

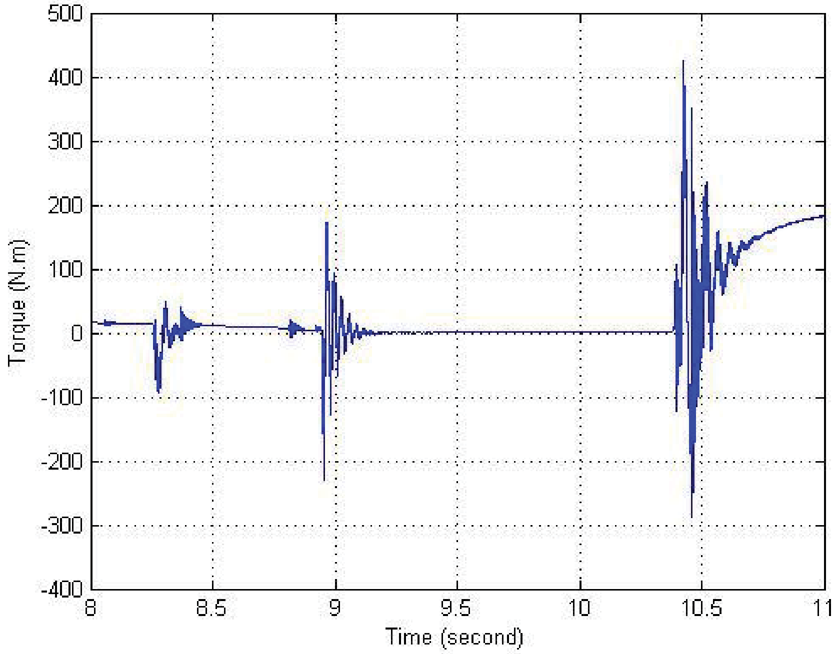

Figure 5.7 Torque Response (1st to 2nd gear) versus Time

Figure 5.7 shows the plot torque response when shifting from 1^{st} to 2^{nd} gear, and this time span is from 8.2 to 11 second. There are three times of excitation in torque response of input shaft. The first oscillation occurs at t=8.25 second, when the clutch is disengaged, there is abrupt decrease in inertia of driveline because the engine and clutch flywheel are decoupled from transmission gearbox. At t=8.94 second, the 1^{st} gear is desynchronized from output shaft and 2^{nd} gear is synchronized with output shaft; the inertia and revolution speed of transmission are changed due to different gear ratio chosen. Therefore, the second time of excitation happens. When the clutch is engaged at t=10.4 second, the clutch is engaged and torque starts being transferred from engine to transmission. There is a big difference between input shaft and engine speeds, and this is represented in Figure 5.2 Clutch Slip versus Time. Therefore, the third time of torque overshoot happens. The same explanation can apply to the other gear ratios.

In Figure 5.6, from t=6 to 30 second, there are 9 times of excitation in torque response because of the change in inertia and revolution speed of transmission. After transient response of driveline, the steady torque from engine is transferred through transmission box. The maximum magnitude of torque overshoot reaches 880 N.m when shifting from 3^{rd} to 4^{th} gear.

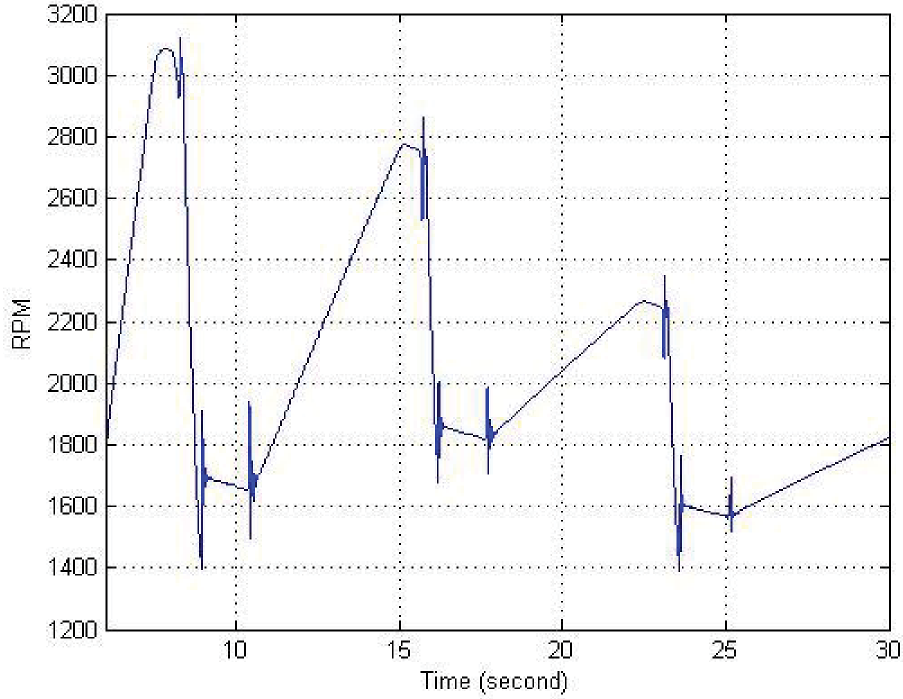

Figure 5.8 Input Shaft Speed versus Time

Figure 5.8 shows the rotational speed of input shaft. Based on the formula:

$$\omega_{clutch} = \omega_{intput\ shaft} \times i_{counter\ gear\ ratio}$$

However, the rotational speed is not exactly proportional to follower in transient simulation because of the effect of damping and stiffness. There are some oscillations in revolution speed at for example t=9 and 11 second.

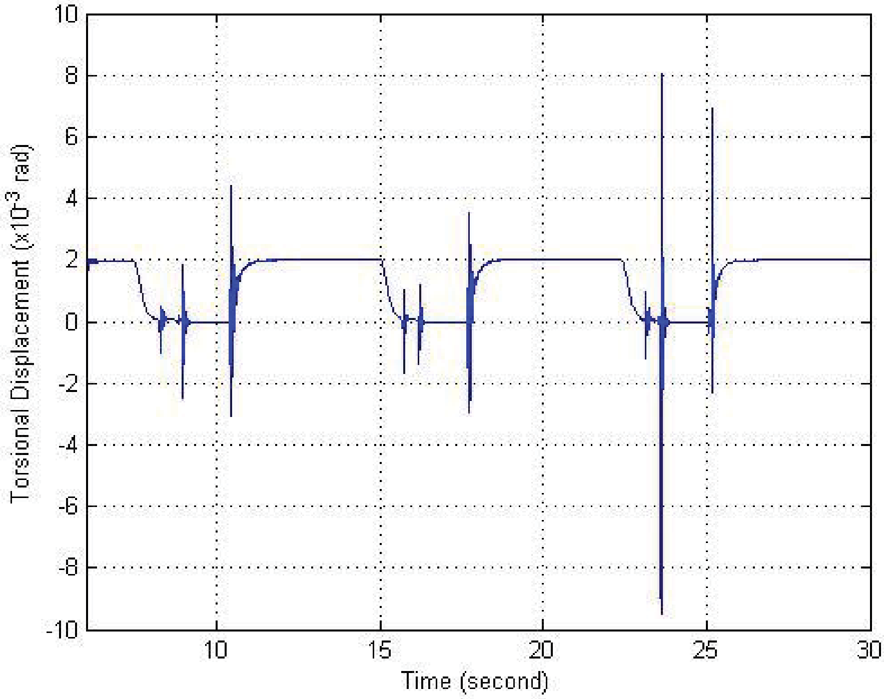

Figure 5.9 Torsional deflection of Input Shaft

When the torque is transferred to input shaft, the torque twists the shaft about its axis of rotation. Figure 5.9 is the plot of torsional deflection of input shaft versus time. The torsional deflection of input shaft is proportional to torque applied on the shaft. So the bigger the torque applies, the more torsional deflection on the shaft. The maximum torsional deflection is 9.493×10^{-3} rad, and it occurs when shifting from 3rd to 4th gear.

		Clutch Disengagement	Synchronization of gears	Clutch Engagement
Torque overshoot (N.m)	1st to 2nd gear	93.49	230.54	424.84
	2nd to 3rd gear	152.44	125.41	343.53
	3rd to 4th gear	106.99	880.24	656.79
Torsional deflection ($\times 10^{-3}$ rad)	1st to 2nd gear	1.028	2.508	4.432
	2nd to 3rd gear	1.684	1.377	3.564
	3rd to 4th gear	1.188	4.494	6.930

Table 5.1 Data of Transient Response of Original Input Shaft

In order to compare transient response of original driveline with the proposed driveline that has an electric drive unit installed on the output shaft, the numerical result of transient response is recorded in Table 5.1. It is not necessary to record the response for shifting from neutral to 1st gear because the electric drive does not operate when shifting from neutral to 1st gear.

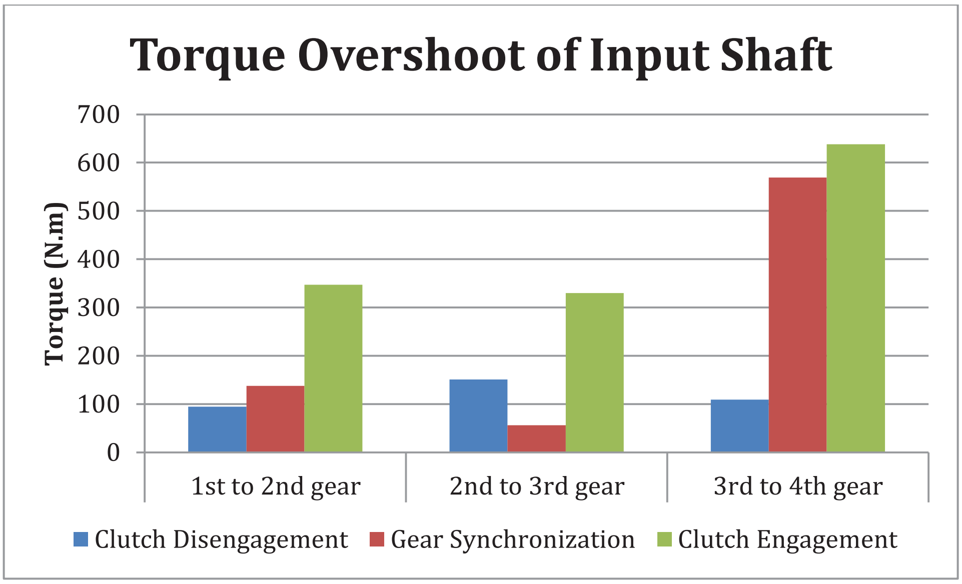

Figure 5.10 Torque Overshoot Comparison of Input Shaft (original)

Figure 5.10 indicates the comparison of torque overshoot value of the three stages of upshifting. Engaging the clutch gives the highest torque overshoot value, and disengaging clutch and synchronizing gears give relatively lower overshoot values. In these three stages, upshifting to higher gear has more torque overshoot applied on the input shaft.

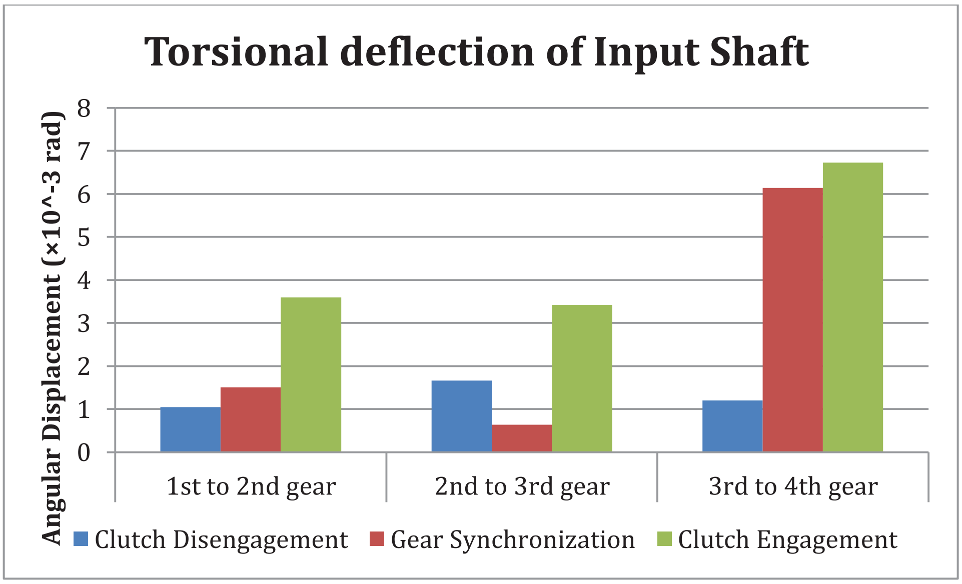

Figure 5.11 Torsional deflection Comparison of Input Shaft (original)

Figure 5.11 shows the comparison of torsional deflection of input shaft. The highest torsional deflection and torque overshoot happen by engaging the clutch when upshifting from 3rd to 4th gear. Also, synchronizing gear gives relatively higher torque overshoot and torsional deflection. Compared to upshifting from 1st to 2nd gear, upshifting from 2nd to 3rd gear does not have higher torsional deflection and torque overshoot. Therefore, the above two figures shows that upshifting does not absolutely cause higher torque overshoot or torsional deflection. The shaft speed when shifting and appropriate clutch timing also affect transient response of system.

5.1.3 Output Shaft

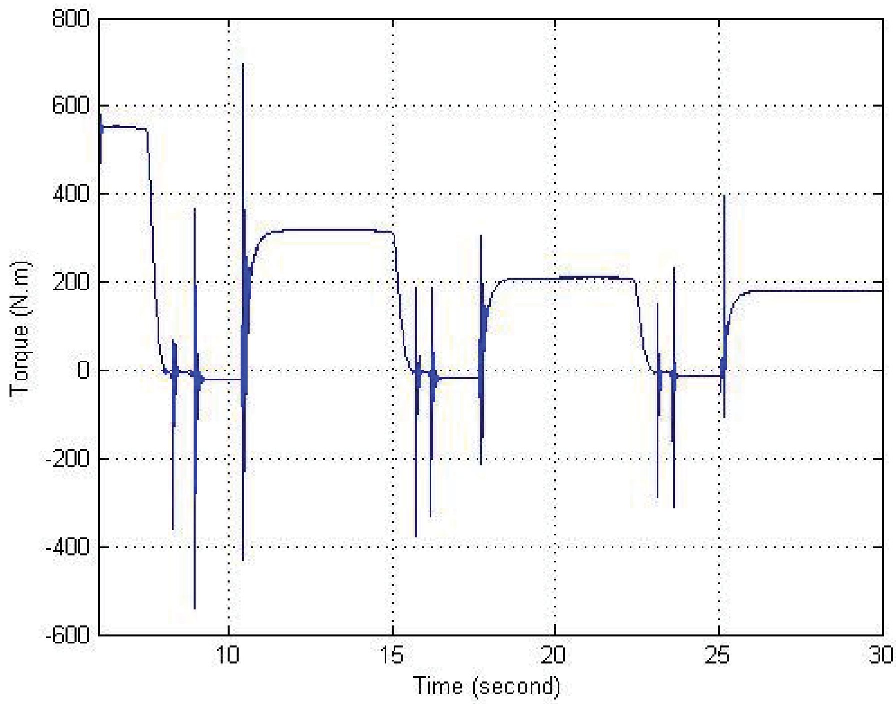

Figure 5.12 Output Shaft Torque versus Time (original)

Figure 5.12 shows the torque applied on the output shaft of driveline. From its general appearance, it seems similar to the plot of input shaft torque. However, the steady-state and overshoot values of torque are bigger than input shaft torque because there are several gear ratios between input shaft and output shaft. Also, the torque fluctuation happens more delayed than as on input shaft.

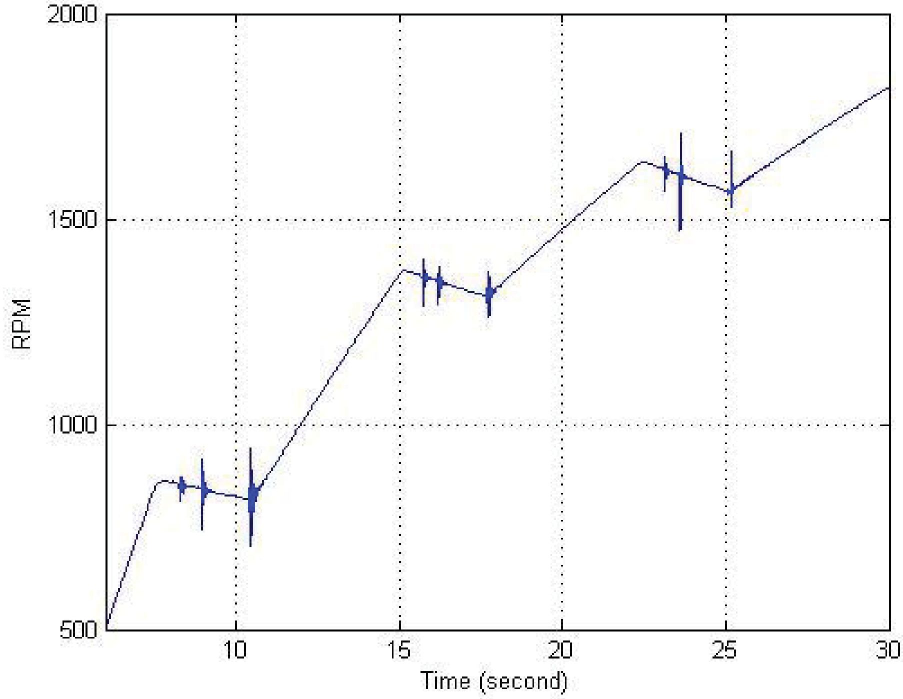

Figure 5.13 Output Shaft Speed versus Time

Compared with Figure 5.8, rotational speed of output shaft in Figure 5.13 is different. Rotational speed of output shaft is increasing through the simulation because the gear ratio is varied by shifting. Negative gradient in shaft speed represents that there is no output torque from engine because the clutch is disengaged, and shaft speed slows down because of internal friction, inertias, aerodynamic drag and other power losses.

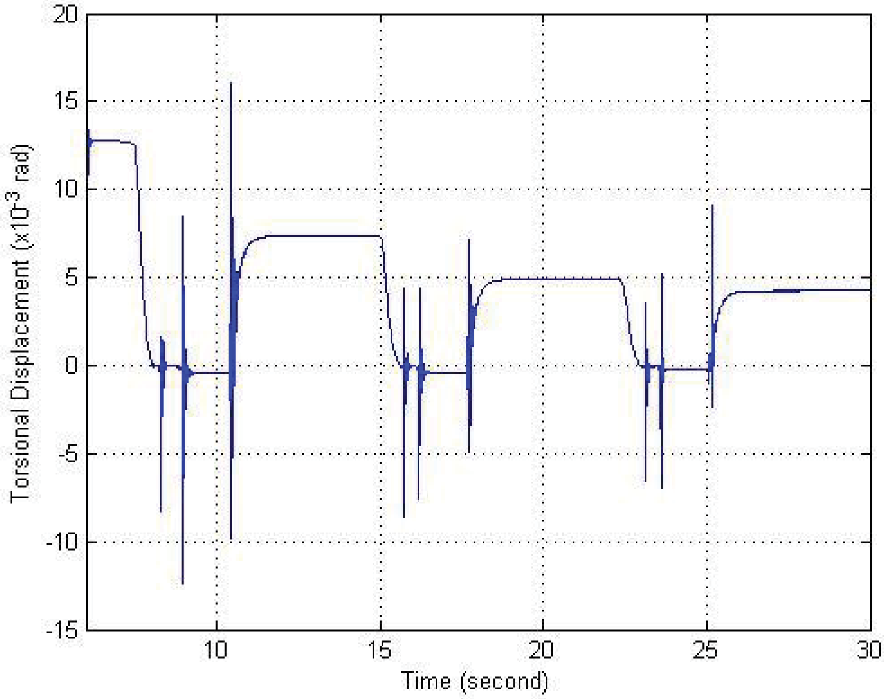

Figure 5.14 Torsional deflection of Output Shaft

Figure 5.14 shows that torsional deflection of output shaft is bigger than input shaft in Figure 5.9. Table 5.2 shows the response of output shaft:

		Clutch Disengagement	Synchronization of gears	Clutch Engagement
Torque overshoot (N.m)	1st to 2nd gear	361.89	541.67	697.03
	2nd to 3rd gear	377.97	330.67	308.67
	3rd to 4th gear	286.68	312.46	398.22
Torsional deflection ($\times 10^{-3}$ rad)	1st to 2nd gear	8.281	12.405	16.062
	2nd to 3rd gear	8.631	7.539	7.162
	3rd to 4th gear	6.520	7.010	9.147

Table 5.2 Data of Transient Response of Original Output Shaft

On the output shaft, the maximum torque magnitude and torsional deflection are 697.03 N.m and 16.062×10^{-3} rad respectively.

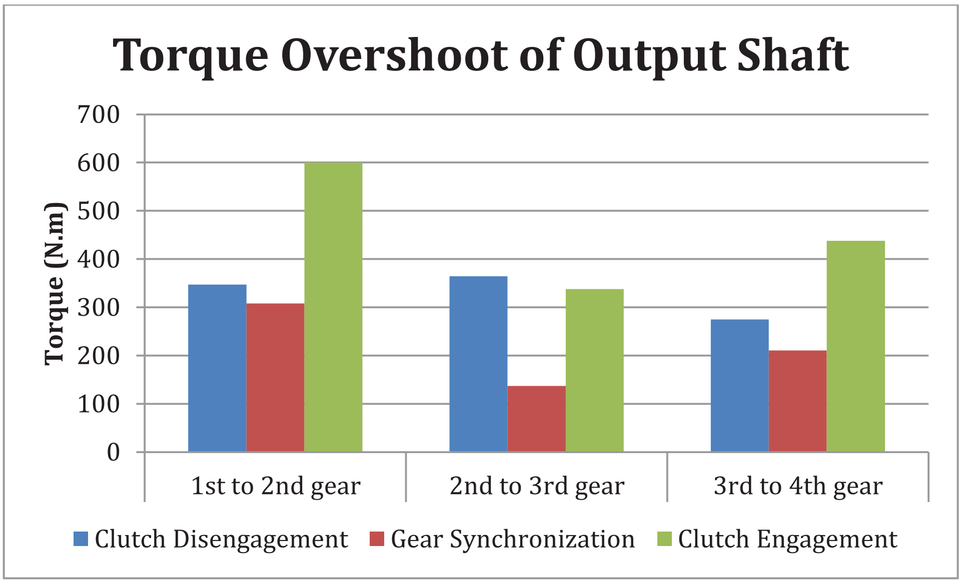

Figure 5.15 Torque Overshoot Comparison of Output Shaft (original)

Figure 5.15 indicates the torque overshoot value of output shaft. When shifting to higher gears, the torque overshoot and torsional deflection become relatively smaller. When the gear ratio is smaller for higher gear, torque transferred from engine is reduced.

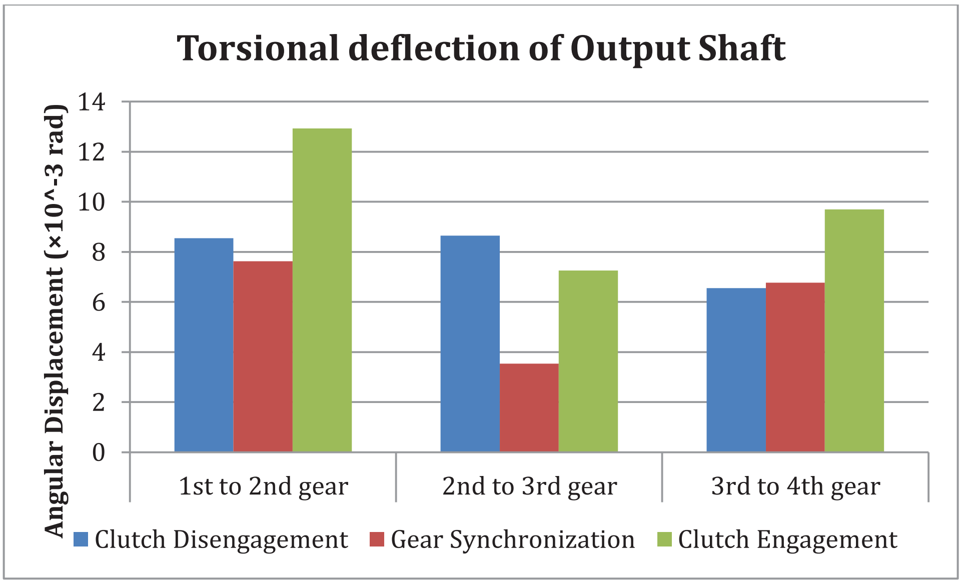

Figure 5.16 Torsional deflection Comparison of Output Shaft (original)

Figure 5.16 shows the torsional deflection of the output shaft. The highest torsional deflection happens by engaging the clutch when upshifting from 1st to 2nd gear. Therefore, the lower gear ratio can reduce torsional deflection.

5.1.4 Drive Shaft

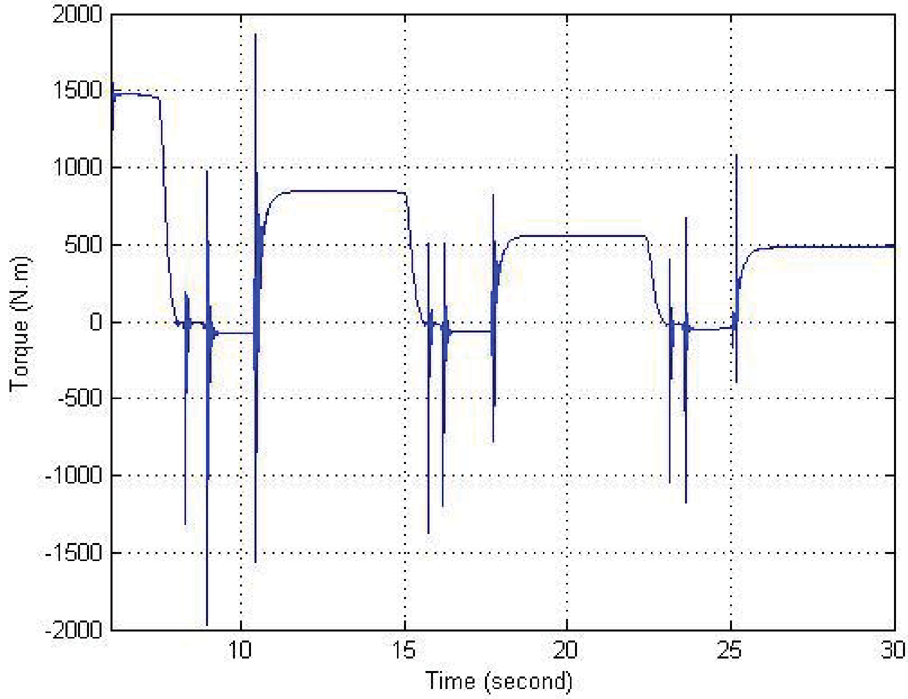

Figure 5.17 Drive Shaft Torque versus Time

Figure 5.17 indicates the torque applied on the drive shaft. Its profile is similar to Figure 5.6 and Figure 5.12 but torque is amplified by bigger gear ratio of final drive. The peak torque is more than 1,800 N.m.

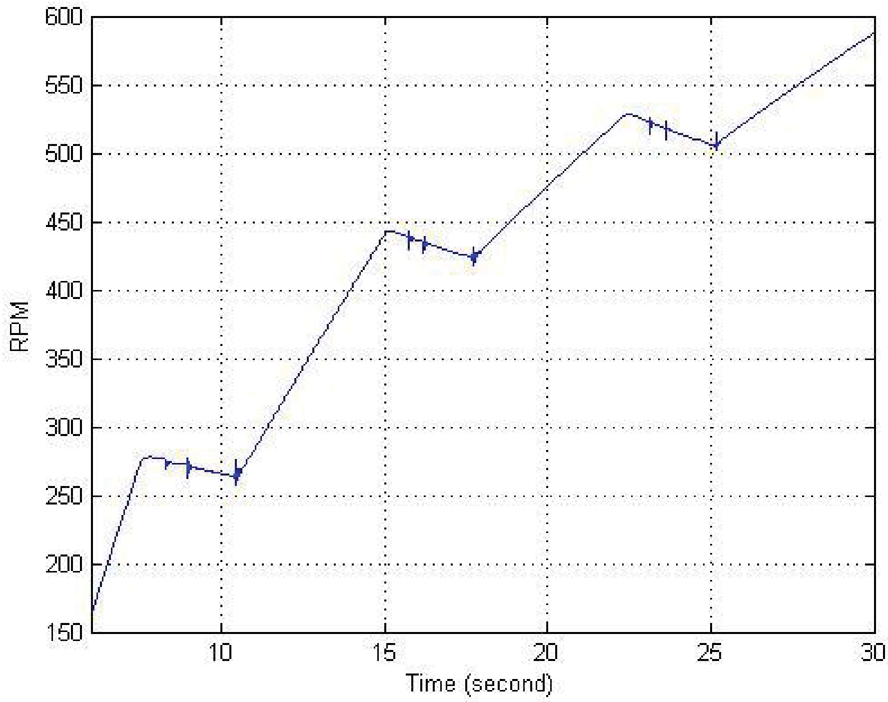

Figure 5.18 Drive Shaft Speed versus Time

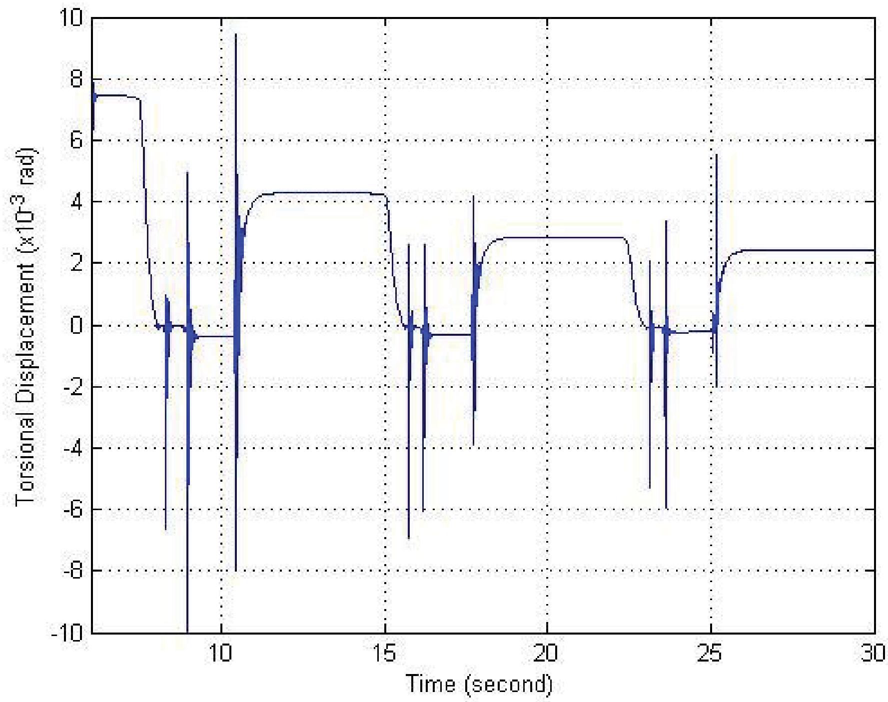

Figure 5.19 Torsional deflection of Drive Shaft

In Figure 5.18, compared with Figure 5.13, the rotational speed is reduced because there is a final gear ratio between output shaft and drive shaft. The formula is:

$$\omega_{output\ shaft} = \omega_{drive\ shaft} \times i_{final\ gear\ ratio}$$

		Clutch Disengagement	Synchronization of gears	Clutch Engagement
Torque overshoot (N.m)	1^{st} to 2^{nd} gear	1313.05	1965.52	1866.99
	2^{nd} to 3^{rd} gear	1370.43	1201.37	821.96
	3^{rd} to 4^{th} gear	1041.42	1172.95	1086.85
Torsional deflection ($\times10^{-3}$ rad)	1^{st} to 2^{nd} gear	6.642	9.949	9.454
	2^{nd} to 3^{rd} gear	6.939	6.082	4.163
	3^{rd} to 4^{th} gear	5.272	5.938	5.511

Table 5.3 Data of Transient Response of Drive Shaft

The peak torque and torsional deflection are 1965.52 N.m and 9.949×10^{-3} rad. After three pairs of gear reduction, the torque and torsional deflection are multiplied.

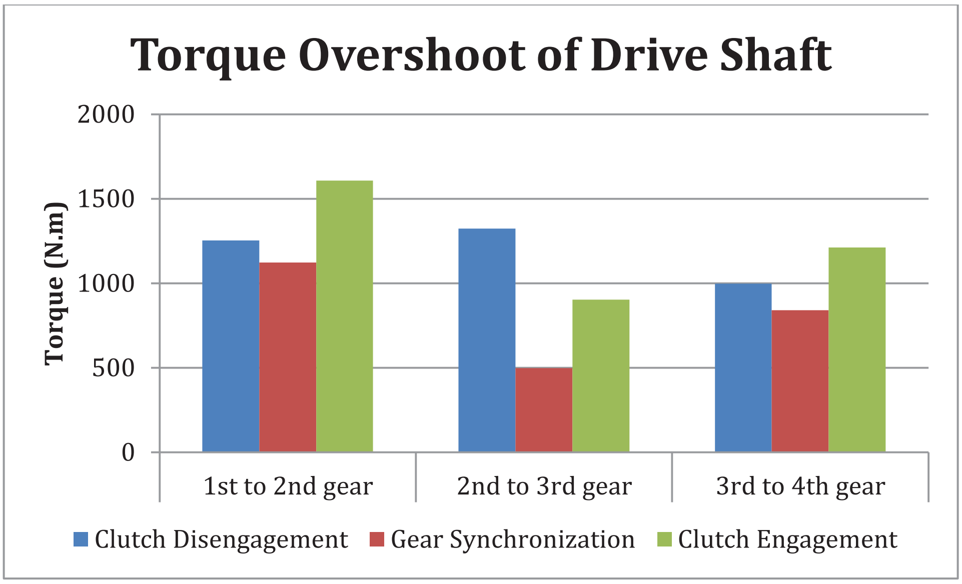

Figure 5.20 Torque Overshoot Comparison of Drive Shaft (original)

Figure 5.20 shows the torque overshoot value of drive shaft. The maximum torque overshoot value happens when the gear is synchronized by upshifting from 1^{st} to 2^{nd} gear. Same as response of other shafts, torque overshoot is only dependent on gear ratio. Shaft speeds when shifting the gears also affect this value.

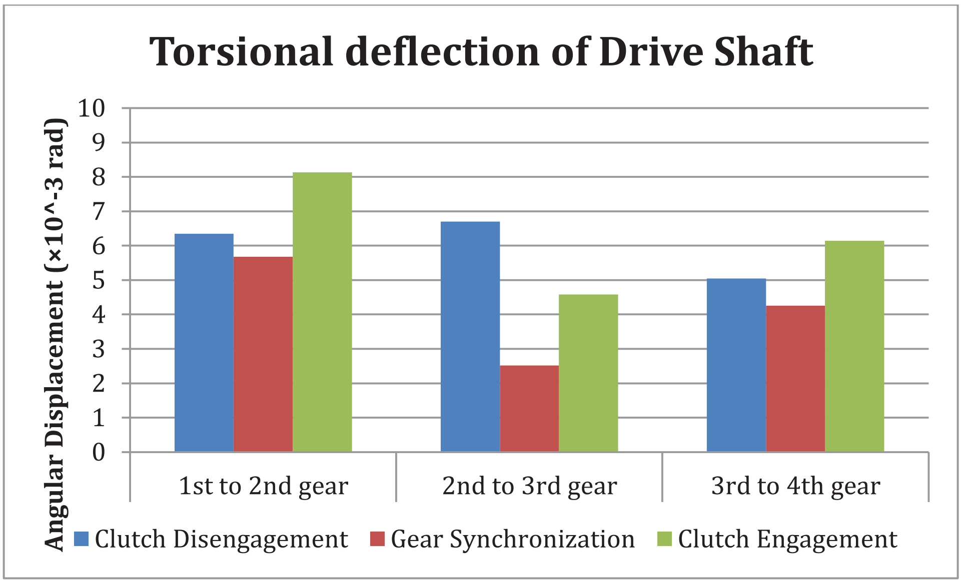

Figure 5.21 Torsional deflection Comparison of Drive Shaft (original)

Figure 5.21 shows that upshifting from 3rd to 4th gear causes relatively lower torsional deflection because there are less torque applied the shaft. Upshifting from 2nd to 3rd gear causes the most torsional deflection. The maximum torsional deflection is 9.95×10^{-3} radian.

5.1.5 Vehicle Body

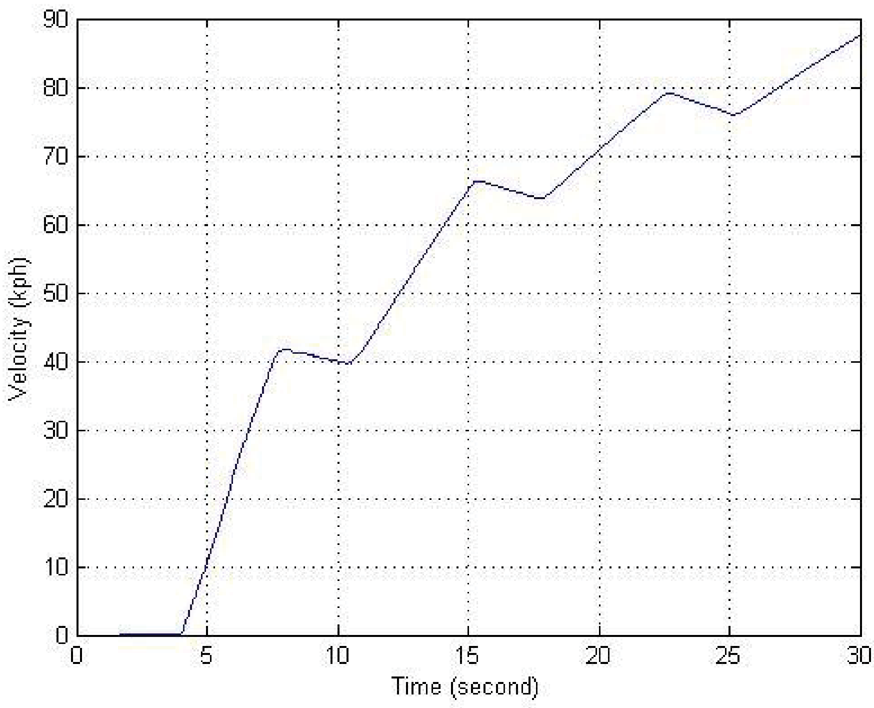

Figure 5.22 Velocity versus Time

Figure 5.22 represents the velocity of vehicle, the positive slope in the plot represents the vehicle is accelerating, and negative slope represents that the vehicle is braking when shifting. There are also fluctuations in velocity plot but they are not easy to identify because of the scale of vertical axis. When the throttle is opened, engine starts operating to transfer torque output to driveline, and this is converted to longitudinal motion of vehicle body. Therefore, the vehicle is accelerating. When the driver is shifting, the clutch is disengaged to decouple the engine from transmission. But losses by friction of components, inertias and aerodynamic drag still exist when transmission is operating. Negative torque is applied to the transmission to cause it slowing down. Therefore, the velocity of vehicle decreases.

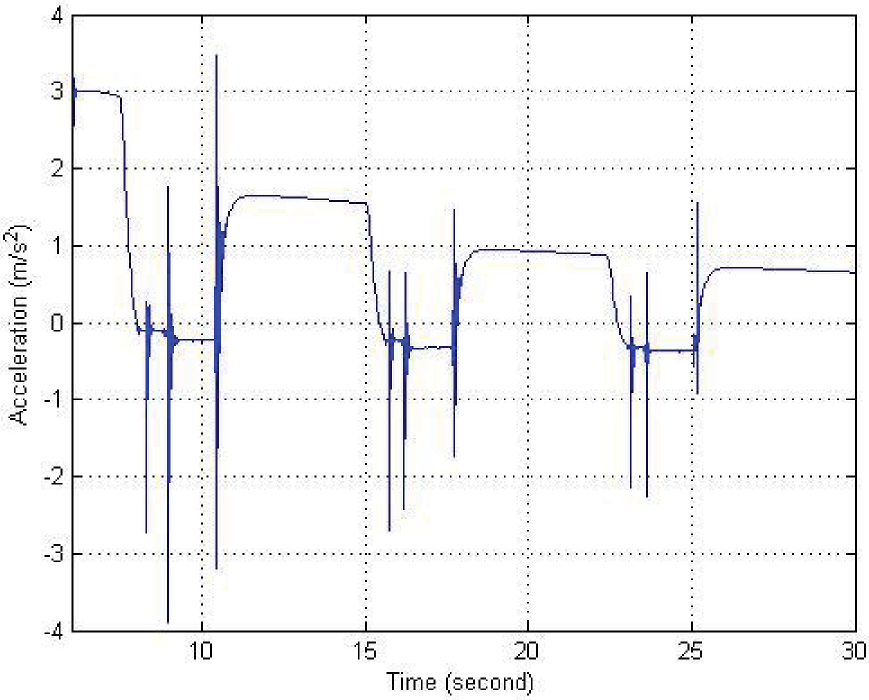

Figure 5.23 Acceleration versus Time

Figure 5.23 shows the acceleration of vehicle and it represents jerk or jolt of vehicle body. Fluctuations in acceleration happen because of torque overshoot existing on the shafts, so there is also negative value in acceleration oscillation. This causes vehicle body vibrating in longitudinal direction. However, the vibration does not last for long time and does not have obvious effect on the velocity of vehicle, and the driver may not have evident feeling on this variation in acceleration.

The steady-state acceleration values for 4 different gear ratios are 3 m/s^2, 1.6 m/s^2, 0.9 m/s^2 and 0.7 m/s^2 respectively. Based on the time dependent simulation, acceleration is the derivative of velocity, so negative acceleration value means the negative slope in velocity plot.

5.2 Simulation Results of Designed Driveline

The method of modeling and shifting schedule has been indicated in Chapter 4.4. The mechanical driveline of design is similar to original driveline, but it has additional electric drive unit that consists of an electric motor and battery. The total time of simulation is 30 seconds and simulation results are indicated in following sections.

5.2.1 Engine and Clutch

The shift schedule of design is in Figure 4.4. Compared with the input signal in Figure 5.1, it has electric drive control signal. The control of electric motor depends on clutch pedal and it has been briefly explained in Section 4.4.1.

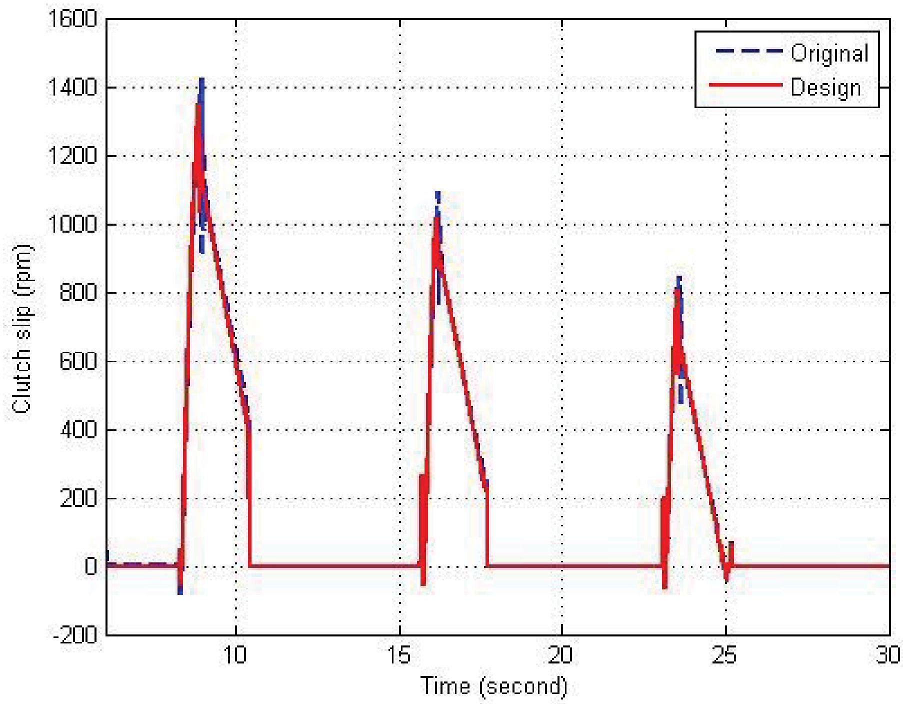

Figure 5.24 Clutch slip versus Time (design)

Figure 5.24 is the plot of clutch slip of clutch; same as clutch slip plot of original driveline, it also indicates the speed difference between flywheel and friction disk of the clutch. When clutch is fully engaged by its static friction, the value of clutch slip drops to 0. As the transmission is switched to higher gear, the clutch slip is decreased. Compared with original driveline, electric motor does not cause significant effect on clutch slip. In Figure 5.24, there are three times of slipping, the peak values of original driveline are approximately 100 rpm higher than designed. This is because the motor accelerates the input shaft and friction disk of clutch, so the speed difference of flywheel and friction disk is reduced.

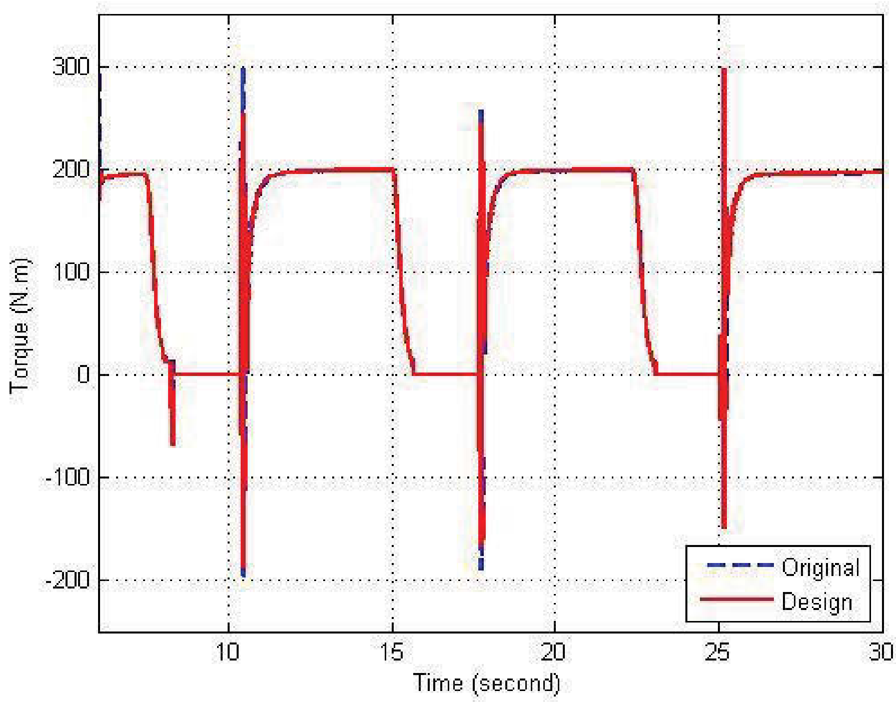

Figure 5.25 Clutch torque versus Time (design)

Figure 5.25 represents the torque transferred on the clutch. The torque transmission of clutch is still inconsistent. The plot of clutch torque is similar to original driveline. However, the torque overshoot is clearly improved. Since the motor can accelerate the input shaft to reduce the clutch slip, engagement of clutch becomes smoother. So the torque overshoots of designed driveline are lower than the original model.

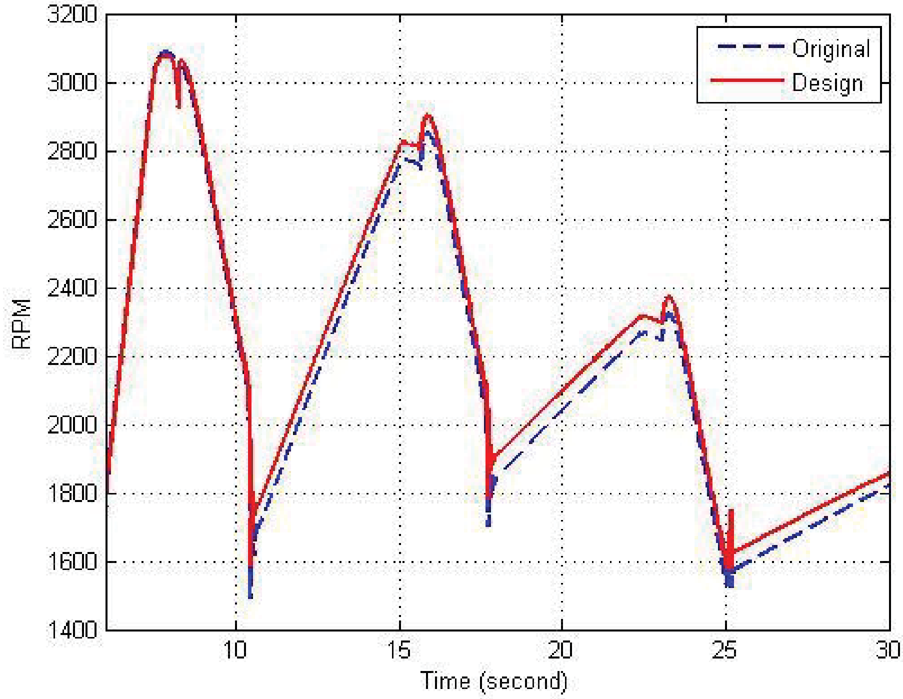

Figure 5.26 Engine RPM versus Time (design)

In Figure 5.26, the idling speed is still 500 rpm for the new driveline. Engine speed of designed driveline is higher than original driveline because the motor accelerates the speed of transmission system, so engine speed is higher when clutch is engaged.

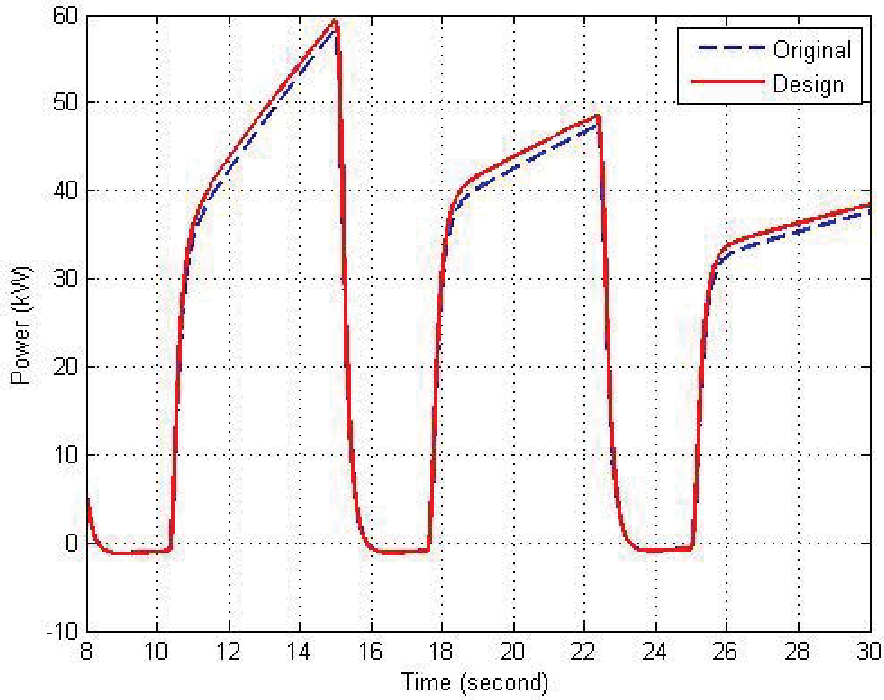

Figure 5.27 Engine Power versus Time (design)

Figure 5.27 shows the output power of engine of designed driveline, there is still negative power output when throttle is released because engine is decoupled from transmission. Since electric motor can accelerate the engine speed, power of engine is also increased with its speed.

5.2.2 Input Shaft

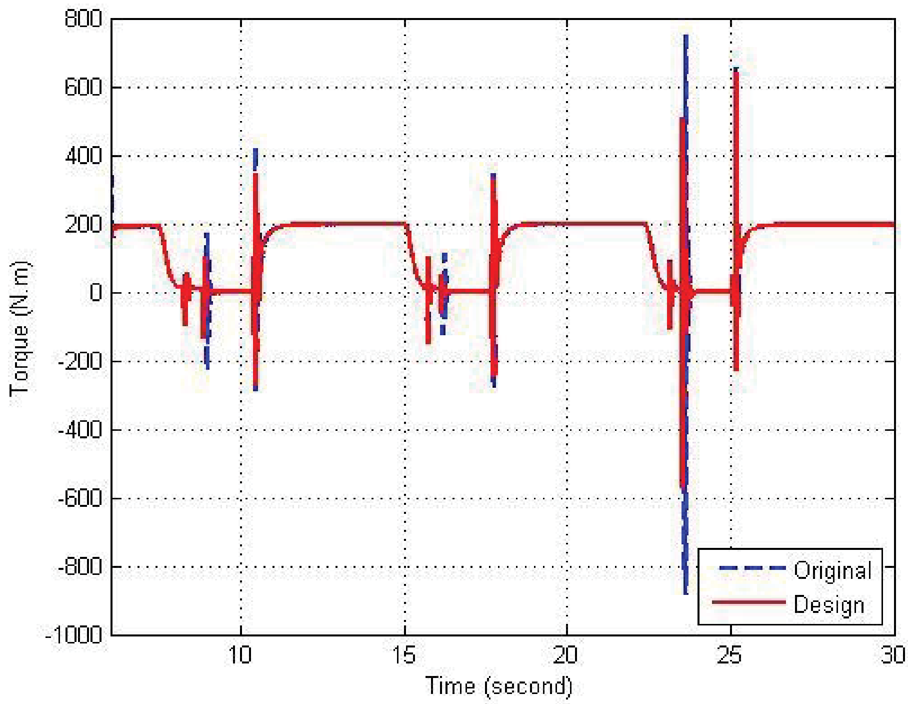

Figure 5.28 Input Shaft Torque versus Time (design)

Figure 5.28 shows the plot of input shaft torque versus time, it represents the torque applied on input shaft. The torque fluctuations occur at each time of clutching and synchronizing gear. The torque holes on input shaft still exist in Figure 5.28. A small amount of torque is transferred from electric motor to input shaft for torque hole compensation. The torque transferred to input shaft is not remarkable. However, the torque overshoots on the input shaft are evidently reduced. For a clear view, torque response of the three times of upshifting are in following figures:

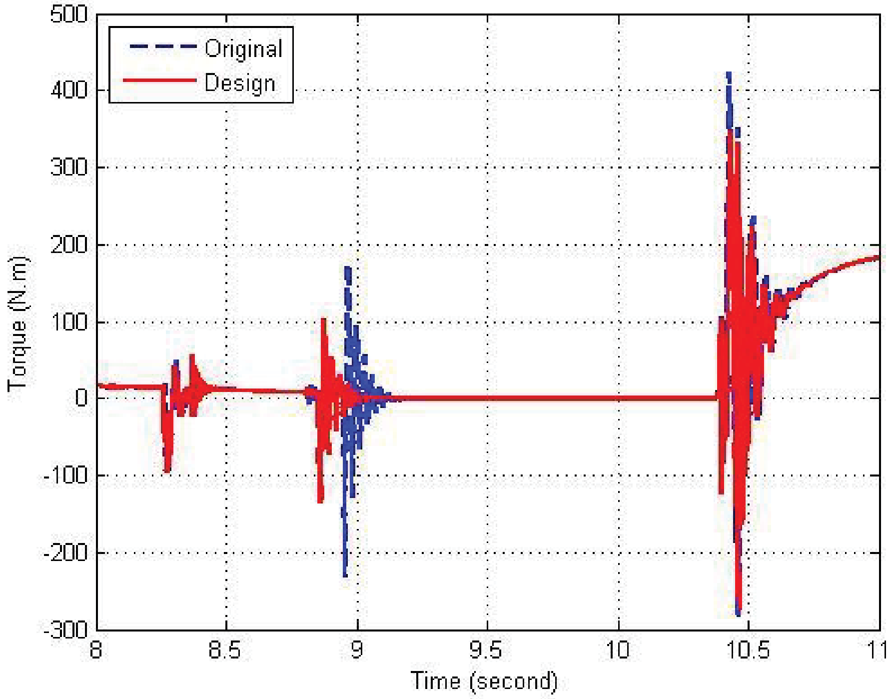

Figure 5.29 Torque Response (1st to 2nd gear) versus Time

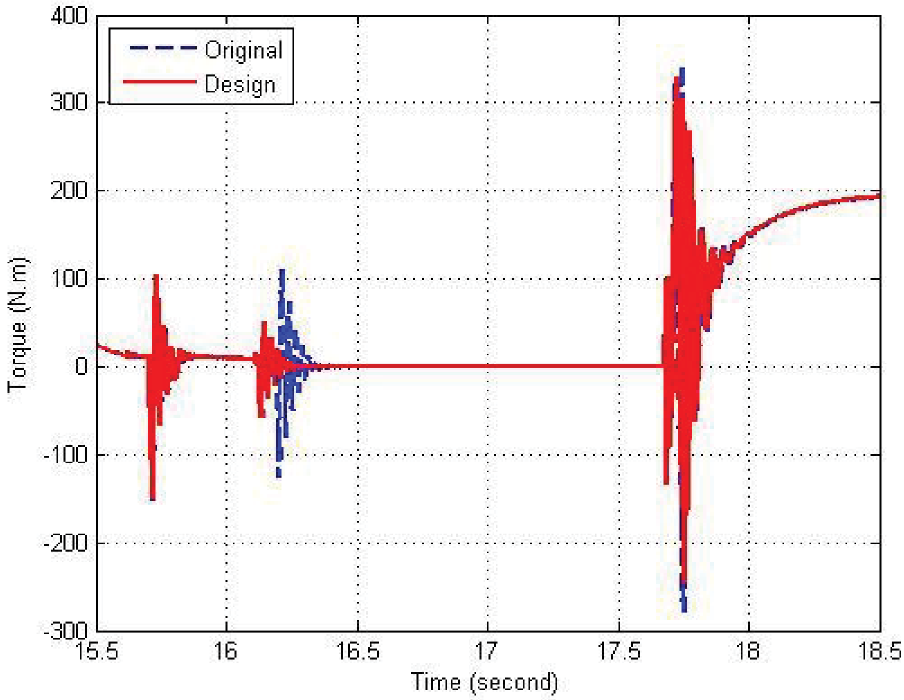

Figure 5.30 Torque Response (2nd to 3rd gear) versus Time

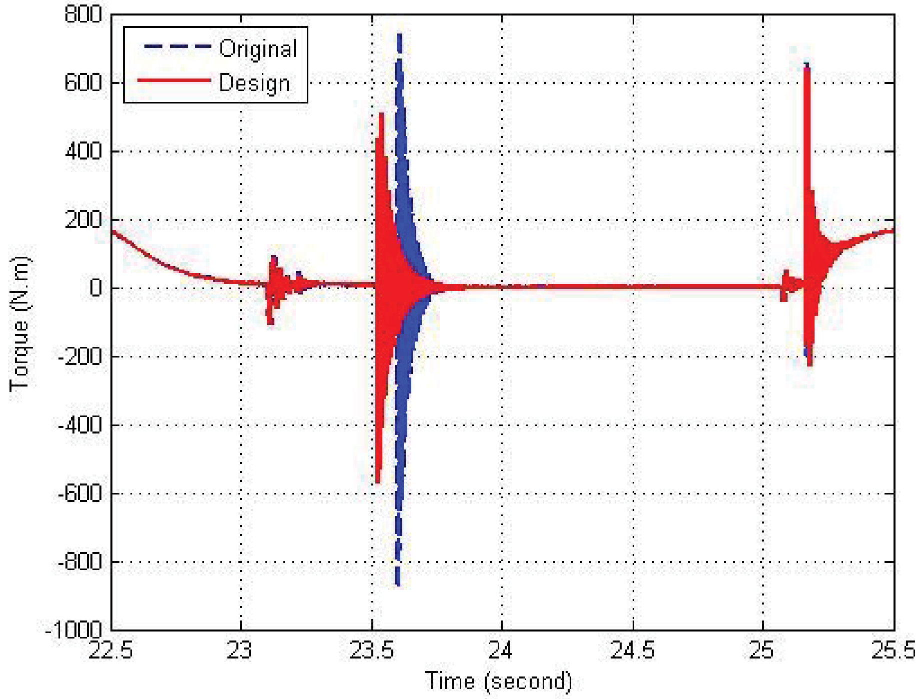

Figure 5.31 Torque Response (3rd to 4th gear) versus Time

In above figures, it is extremely easy to identify the difference between the original and designed drivetrains' data. The torque response of designed drivetrain has lower torque overshoot. The overshoots of designed drivetrain occur earlier than original drivetrain when synchronizing gears because accelerating shaft when synchronizing gear can increase rate of successful cone clutch and dog clutch engagement. Therefore, response of gear synchronization happens earlier than original drivetrain. The numerical results will be compared in later chapter.

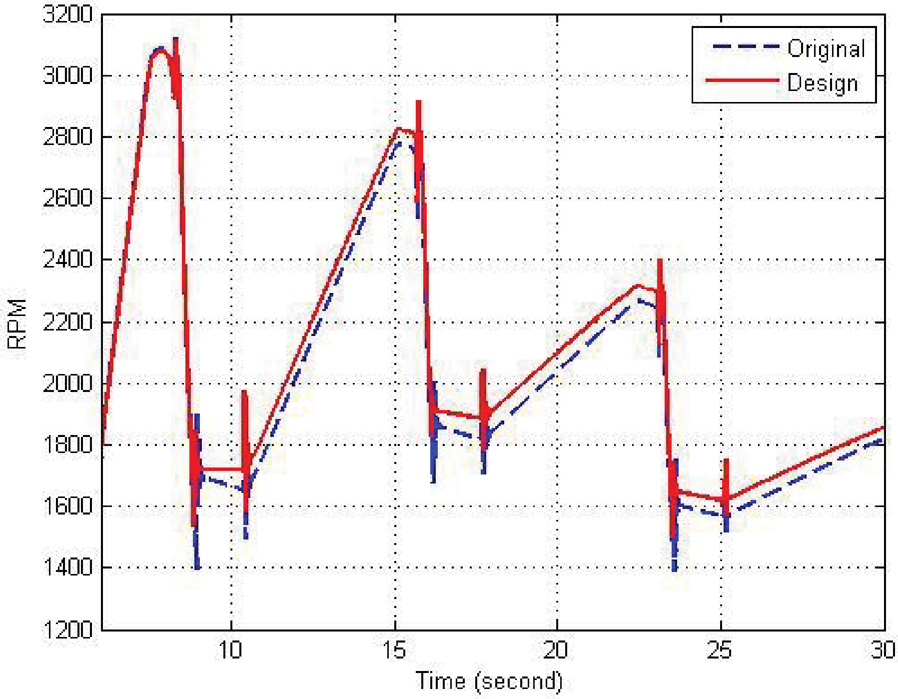

Figure 5.32 Input Shaft Speed versus Time (design)

Figure 5.32 shows the rotational speed of input shaft of designed driveline. The designed drivetrain has higher input shaft speed than the original drivetrain. In the figure, the negative gradient of shaft speed is reduced, and this can conclude that electric motor has recovered a little mount of shaft speed loss.

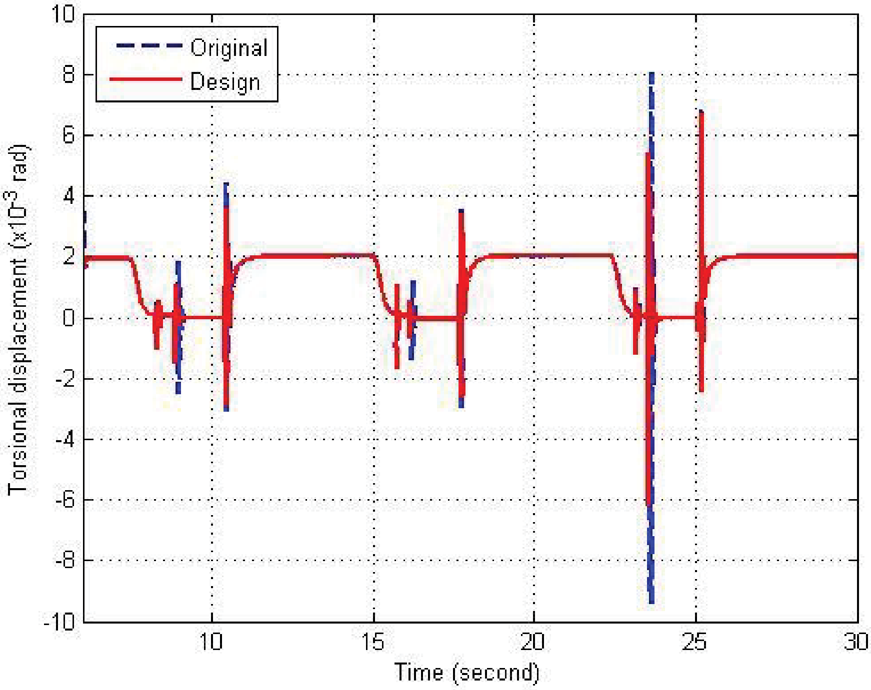

Figure 5.33 Torsional deflection of Input Shaft (design)

Figure 5.33 shows the plot of torsional deflection of input shaft. It has similar appearance as plot of applied torque. If the torque applied on the shaft is reduced, torsional deflection of input shaft is definitely reduced.

Unit: $\times 10^{-3}$ rad	Original	Design	Difference
1st to 2nd gear:			
Clutch Disengagement:	1.028	1.049	**0.021↑**
Gear Synchronization:	2.508	1.483	**1.025↓**
Clutch Engagement:	4.432	3.592	**0.840↓**
2nd to 3rd gear:			
Clutch Disengagement:	1.684	1.659	**0.025↓**
Gear Synchronization:	1.377	0.659	**0.718↓**
Clutch Engagement:	3.564	3.423	**0.141↓**
3rd to 4th gear:			
Clutch Disengagement:	1.188	1.206	**0.018↑**
Gear Synchronization:	9.494	6.186	**3.308↓**
Clutch Engagement:	6.930	6.718	**0.212↓**

Table 5.4 Torsional deflection Comparison (Input Shaft)

Unit: N.m	Original	Design	Difference
1st to 2nd gear:			
Clutch Disengagement:	93.49	95.50	**2.01↑**
Gear Synchronization:	230.54	135.24	**95.30↓**
Clutch Engagement:	424.84	346.10	**78.74↓**
2nd to 3rd gear:			
Clutch Disengagement:	152.44	150.04	**2.40↓**
Gear Synchronization:	125.41	58.32	**67.09↓**
Clutch Engagement:	343.53	330.22	**13.31↓**
3rd to 4th gear:			
Clutch Disengagement:	106.99	108.81	**1.82↑**
Gear Synchronization:	880.24	572.65	**307.59↓**
Clutch Engagement:	656.79	637.29	**19.50↓**

Table 5.5 Torque Overshoot Comparison (Input Shaft)

Table 5.4 and Table 5.5 indicate that torque overshoot and torsional deflection on the input shafts are reduced by electric drive unit. However, there is a slight increase in torque overshoot and torsional deflection by disengaging clutch. Since the values of these increases are negligible, it can be concluded that the electric drive unit has improved the transient response of input shaft except for disengaging clutch.

5.2.3 Output Shaft

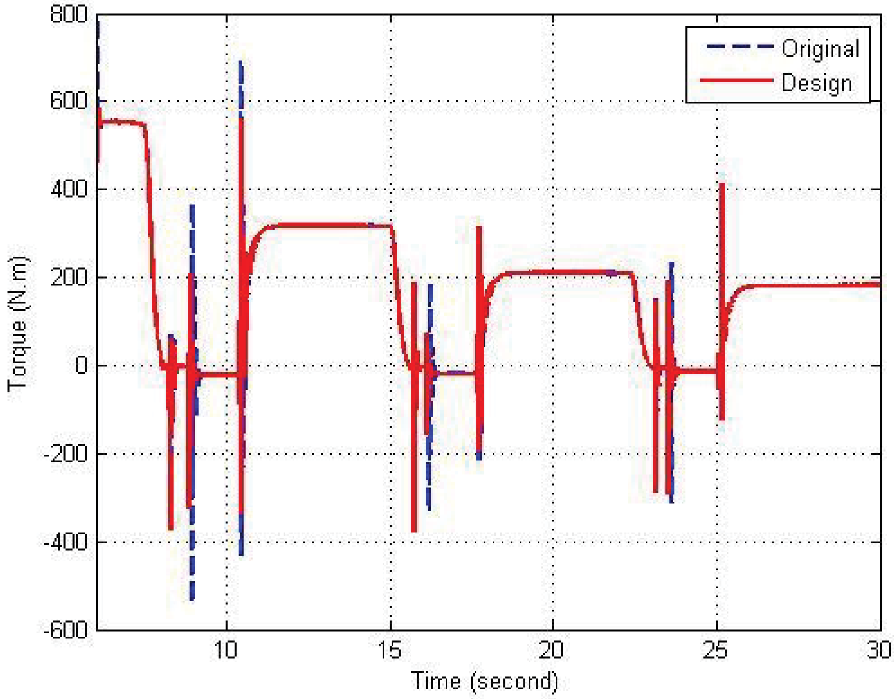

Figure 5.34 Output Shaft Torque versus Time (design)

Figure 5.34 shows the torque applied on the output shaft of designed driveline. It clearly indicates the torque transferred from electric motor for torque hole compensation. The torque from motor is more evident because the motor is directly mounted on the output shaft and synchronization of gears does not cause intermittence to torque transferred. The torque difference in torque when upshifting is the amount of torque transferred from electric motor.

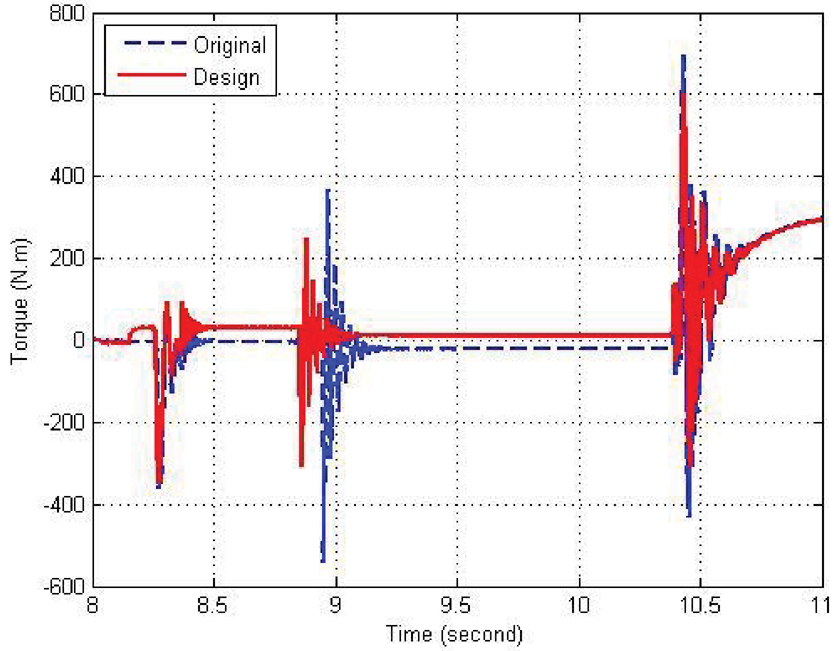

Figure 5.35 Torque Response (1st to 2nd gear) versus Time (design)

Figure 5.35 focuses on the torque response of shifting from 1st to 2nd gear. The rise in torque between disengaging and engaging clutch is from electric drive unit. Before the clutch is fully disengaged, motor transfers relatively constant torque till t=8.15 second. At t=9.1 seconds, the gear synchronization is completed, there is a decrease in torque because the inertia of transmission is changed as a result of different gear ratio.

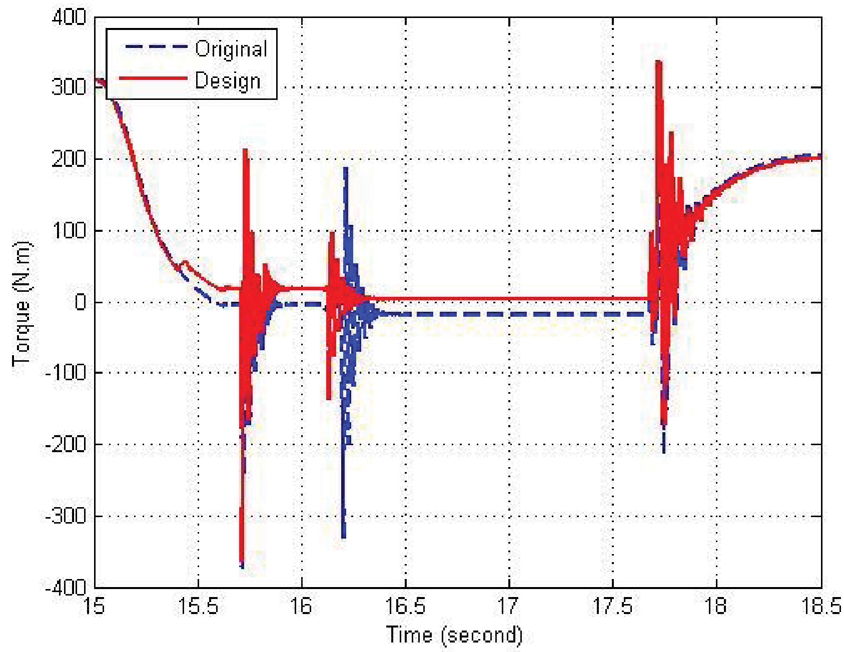

Figure 5.36 Torque Response (1st to 2nd gear) versus Time (design)

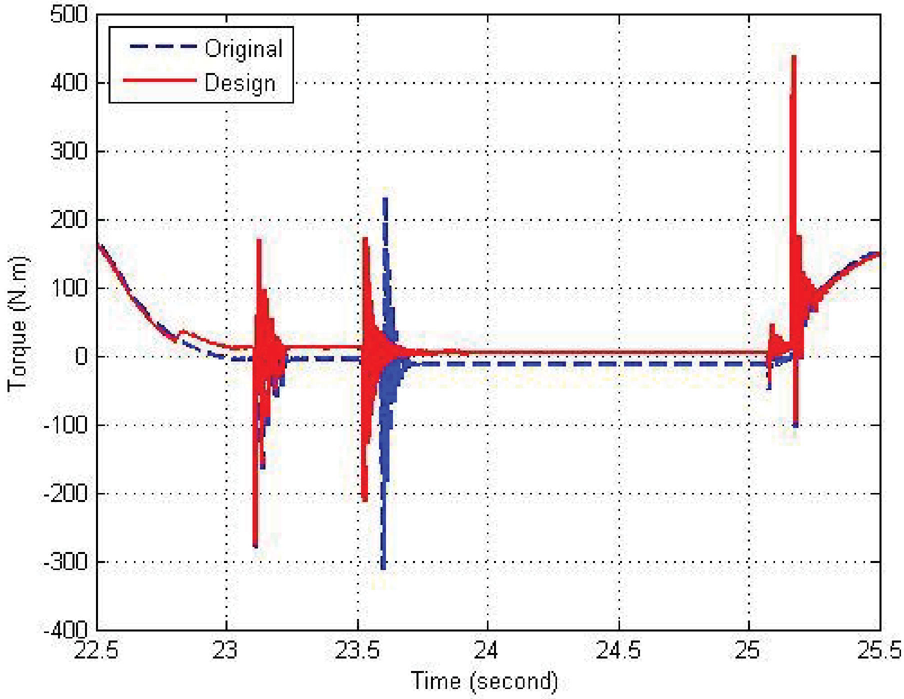

Figure 5.37 Torque Response (1st to 2nd gear) versus Time (design)

Compared with response of shifting from 1st to 2nd gear, the motor torque is reduced at higher gear because at higher rotational speed, motor has lower output torque. Therefore, the torque difference becomes smaller.

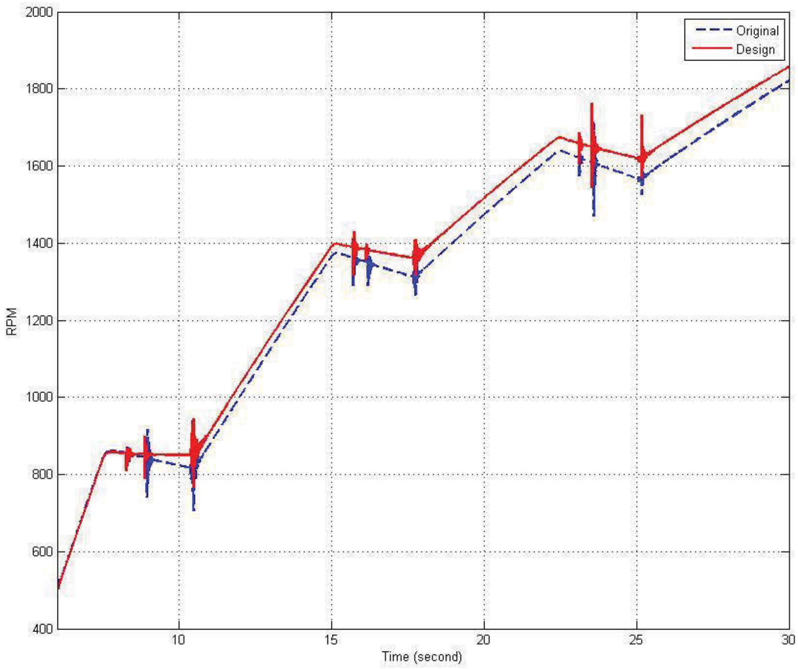

Figure 5.38 Output Shaft Speed versus Time (design)

In Figure 5.38, the slope of rotational speed between 7.5 and 11 seconds is about 0, which means the electric motor keeps the speed of shaft constant and eliminates the speed loss at this stage. However, when upshifting to 3rd and 4th gear, the slopes of rotational speed are still negative, because at higher rotational speed, there is less output torque transferred from electric motor. The motor cannot provide enough output power to keep rotational speed to maintain speed of output shaft.

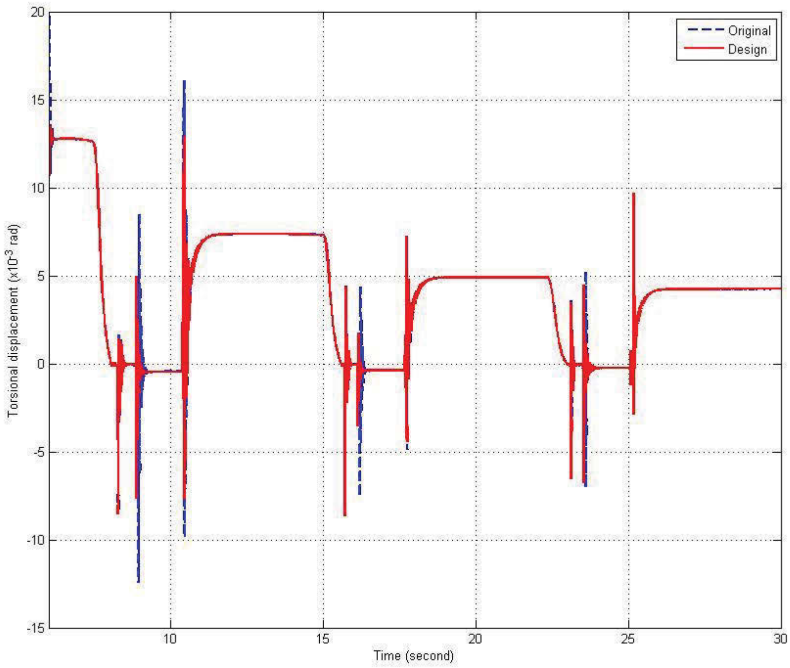

Figure 5.39 Torsional deflection of Output Shaft (design)

Figure 5.39 shows that the torsional deflection of output shaft is bigger than input shaft. Torsional deflection of output shaft is also clearly diminished. The numerical comparisons are shown in Table 5.6 and Table 5.7.

Unit: $\times 10^{-3}$ rad	Original	Design	Difference
1st to 2nd gear:			
Clutch Disengagement:	8.281	8.538	**0.257↑**
Gear Synchronization:	12.405	7.448	**4.957↓**
Clutch Engagement:	16.062	12.937	**3.125↓**
2nd to 3rd gear:			
Clutch Disengagement:	8.631	8.639	**0.008↑**
Gear Synchronization:	7.539	3.561	**3.978↓**
Clutch Engagement:	7.162	7.258	**0.096↑**
3rd to 4th gear:			
Clutch Disengagement:	6.520	6.548	**0.028↑**
Gear Synchronization:	7.010	6.837	**0.173↓**
Clutch Engagement:	9.147	9.710	**0.563↑**

Table 5.6 Torsional deflection Comparison (Output Shaft)

Unit: N.m	Original	Design	Difference
1st to 2nd gear:			
Clutch Disengagement:	361.89	371.98	**10.09↑**
Gear Synchronization:	541.67	325.57	**216.10↓**
Clutch Engagement:	697.03	560.65	**136.38↓**
2nd to 3rd gear:			
Clutch Disengagement:	377.97	378.74	**0.77↑**
Gear Synchronization:	330.67	157.13	**173.54↓**
Clutch Engagement:	308.67	313.12	**4.45↑**
3rd to 4th gear:			
Clutch Disengagement:	286.68	287.61	**0.93↑**
Gear Synchronization:	312.46	290.39	**22.07↓**
Clutch Engagement:	398.22	412.79	**14.57↑**

Table 5.7 Torque Overshoot Comparison (Output Shaft)

Above tables show that the torque overshoots and torsional deflection of gear synchronization are efficiently improved at some stages. However, it seems that there is negative effect on the response when engaging and disengaging the clutch; the torque overshoot and torsional deflection are increased but not very distinct.

5.2.4 Drive Shaft

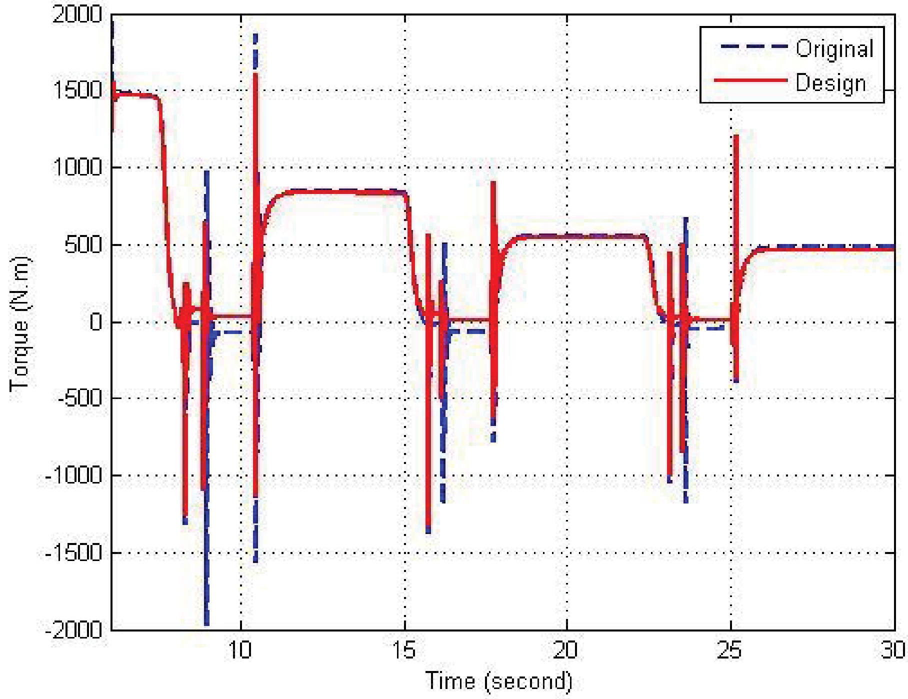

Figure 5.40 Drive Shaft Torque versus Time (design)

Drive shaft torque in Figure 5.40 looks similar to Figure 5.34 but the magnitude torque is greatly amplified because the gear ratio of final drive is 3.1:1.

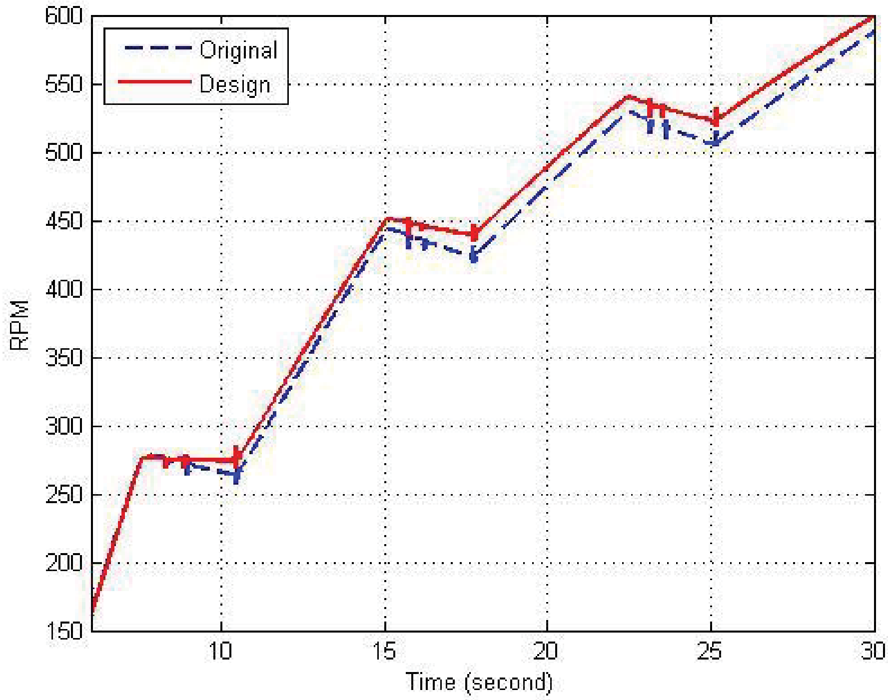

Figure 5.41 Drive Shaft Speed versus Time (design)

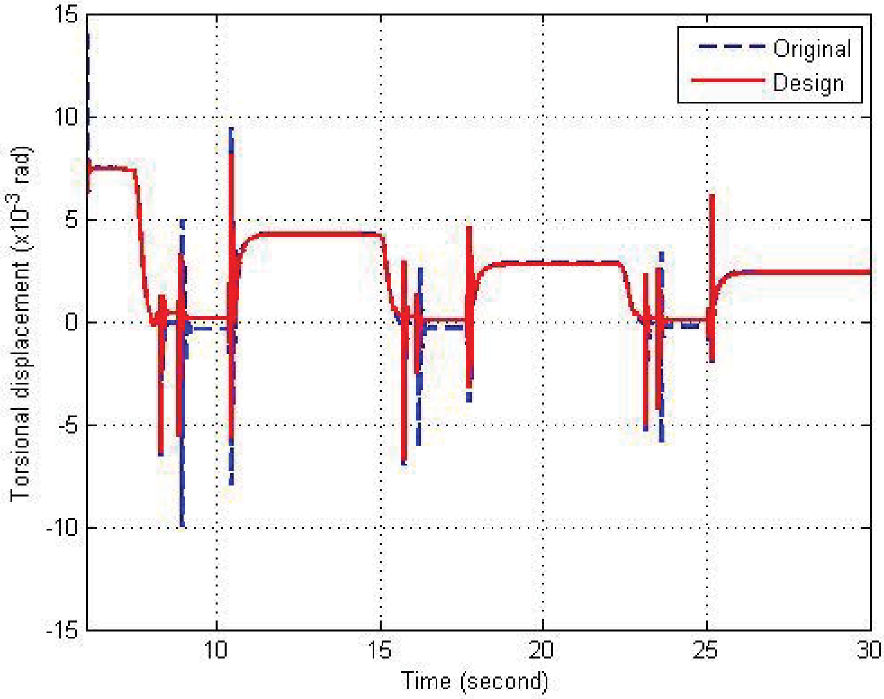

Figure 5.42 Torsional deflection of Drive Shaft (design)

Figure 5.42 represents the torsional deflection of drive shaft. The peak torsional deflection and frequency are recorded in Table 5.8 and Table 5.9:

Unit: $\times 10^{-3}$ rad	Original	Design	Difference
1st to 2nd gear:			
Clutch Disengagement:	6.642	6.349	**0.293↓**
Gear Synchronization:	9.949	5.544	**4.405↓**
Clutch Engagement:	9.454	8.132	**1.322↓**
2nd to 3rd gear:			
Clutch Disengagement:	6.939	6.705	**0.234↓**
Gear Synchronization:	6.082	2.518	**3.564↓**
Clutch Engagement:	4.163	4.576	**0.413↑**
3rd to 4th gear:			
Clutch Disengagement:	5.282	5.046	**0.236↓**
Gear Synchronization:	5.938	4.295	**1.643↓**
Clutch Engagement:	5.511	6.145	**0.634↑**

Table 5.8 Torsional deflection Comparison (Drive Shaft)

Unit: N.m	Original	Design	Difference
1^{st} to 2^{nd} gear:			
Clutch Disengagement:	1313.05	1254.96	**58.09↓**
Gear Synchronization:	1965.52	1095.46	**870.06↓**
Clutch Engagement:	1866.99	1606.30	**260.69↓**
2^{nd} to 3^{rd} gear:			
Clutch Disengagement:	1370.43	1324.20	**46.23↓**
Gear Synchronization:	1201.37	497.78	**703.59↓**
Clutch Engagement:	821.96	902.82	**80.86↑**
3^{rd} to 4^{th} gear:			
Clutch Disengagement:	1041.42	996.65	**44.77↓**
Gear Synchronization:	1172.95	848.30	**324.65↓**
Clutch Engagement:	1086.85	1211.70	**124.85↑**

Table 5.9 Torque Overshoot Comparison (Drive Shaft)

Based on above tables, the response of drive shaft is effectively improved. For example, the biggest reduction in torque overshoot is above 800 N.m. However, the effect on the response of engaging clutch is opposite, the torque overshoot and torsional deflection are increased by about 10 percent. Therefore, electric drive unit can only improve performance of drive shaft when disengaging clutch and synchronizing gears.

5.2.5 Electric Motor

The electric motor provides additional torque for torque hole compensation. The motor only operates when clutch pedal is depressed by driver. The electric control of motor is the same as clutch input signal except for shifting from neutral to 1^{st} gear.

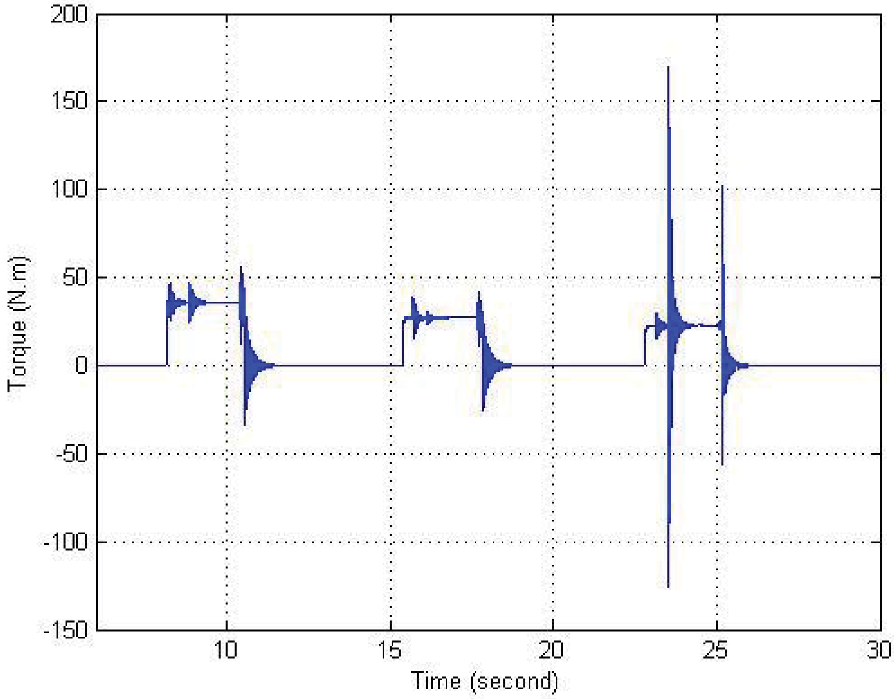

Figure 5.43 Motor Torque versus Time (design)

Figure 5.43 shows the output torque of electric motor, when internal combustion engine is coupled to transmission gearbox, the motor torque maintains at zero. The steady-state values between each time of clutching are actual torque output of motor. The output torque values are 36 N.m for upshifting to 2nd gear, 27 N.m for upshifting to 3rd gear, 23 N.m for upshifting to 4th gear.

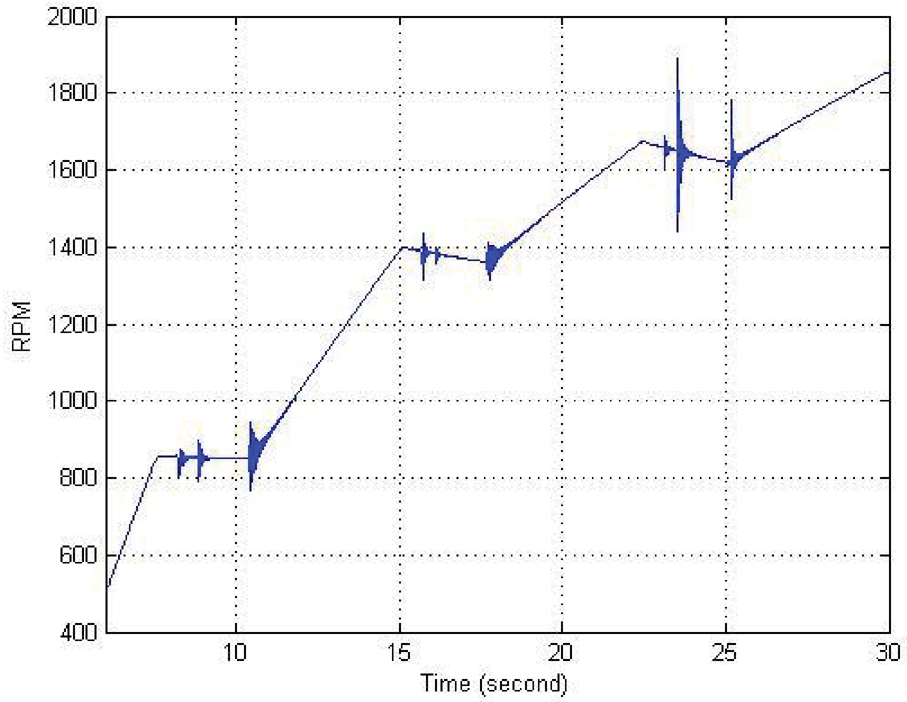

Figure 5.44 Motor RPM versus Time (design)

Figure 5.44 shows the rotational speed of electric motor. Since the motor is always attached to the output shaft, the speed of motor is the same as output shaft speed, as in Figure 5.38. Based on the target speed, the motor can provide corresponding torque.

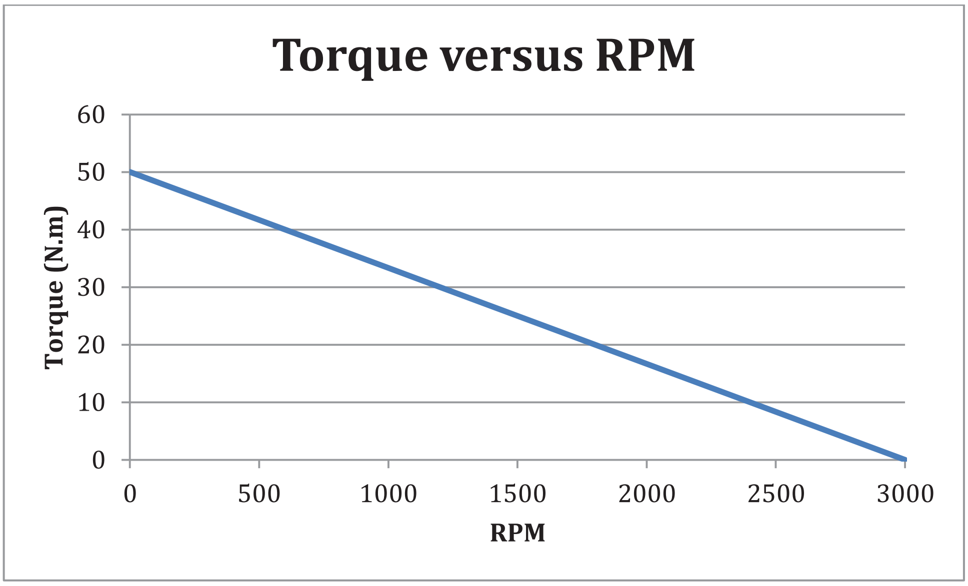

Figure 5.45 Motor Torque versus RPM

Figure 5.45 shows the performance plot of electric motor, the intersection point on vertical axis represents stall torque, and the intersection point on horizontal axis represents no-load speed. Based on the target speed, corresponding torque can be determined from the performance plot. For example, if the target speed is 1000 RPM, output torque of motor is 33 N.m.

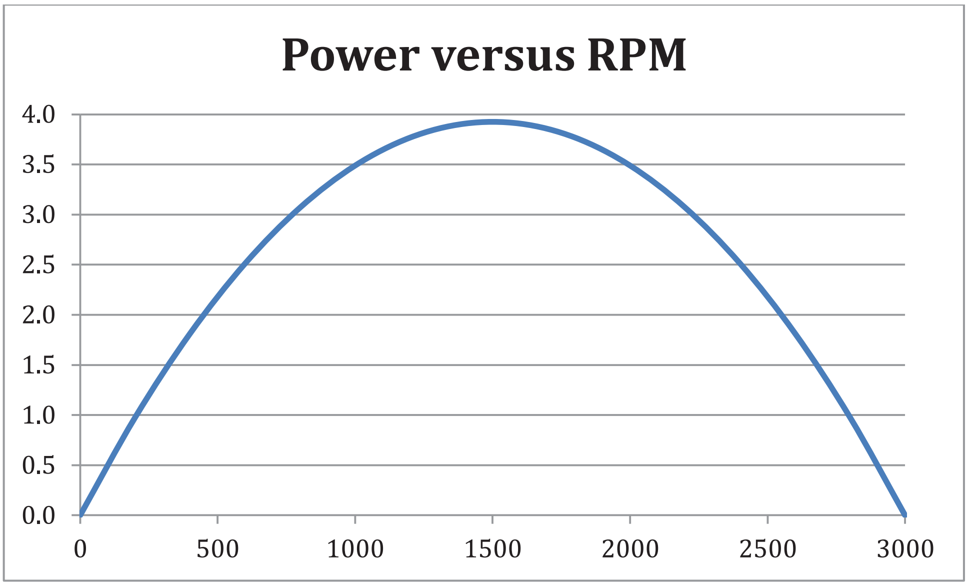

Figure 5.46 Motor Power versus RPM

Figure 5.46 shows the output power of the motor. The power of motor is dependent on the rotational speed. The formula that is used to calculate power is:

$$power(kW) = \frac{RPM \times Torque(N \cdot m)}{9{,}554}, where\ 9{,}554\ is\ conversion\ constant\ [44]$$

For example, at 1000 RPM, output power of motor is 3.5 kW. From the plot, the maximum power is 3.9 kW but this only occurs at 1500 RPM. Before point of peak power, as the rotational speed increases, output power increases. After peak power, the output power will decrease as rotational speed increases.

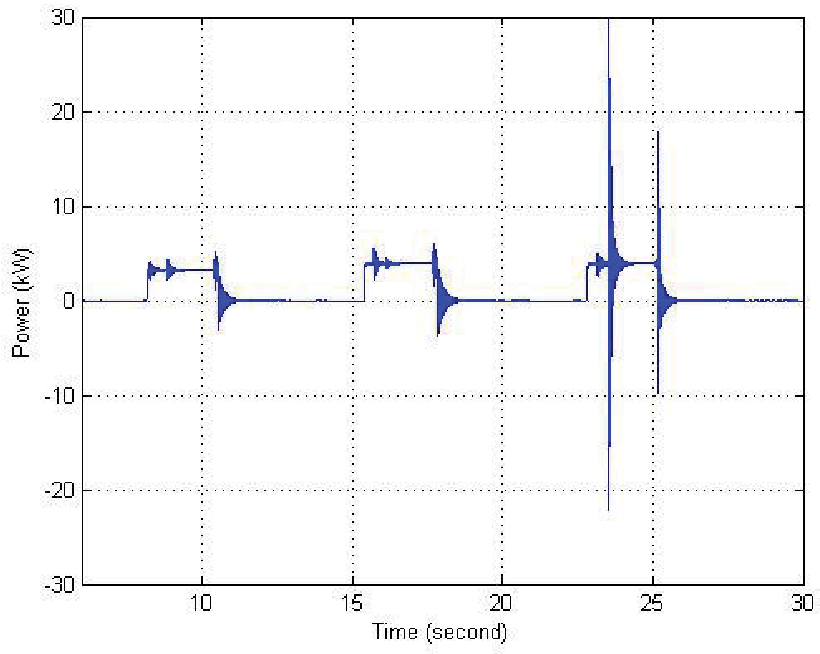

Figure 5.47 Motor Power versus Time

Figure 5.47 illustrates the motor power of motor. In Figure 5.43, there are some oscillations in motor torque, and that is the reason of oscillations in output power. The output power values are 3.2 kW for upshifting from 1st to 2nd gear, 3.9 kW for upshifting from 2nd to 3rd gear, 3.8 kW for upshifting from 3rd to 4th gear.

5.2.6 Vehicle Body

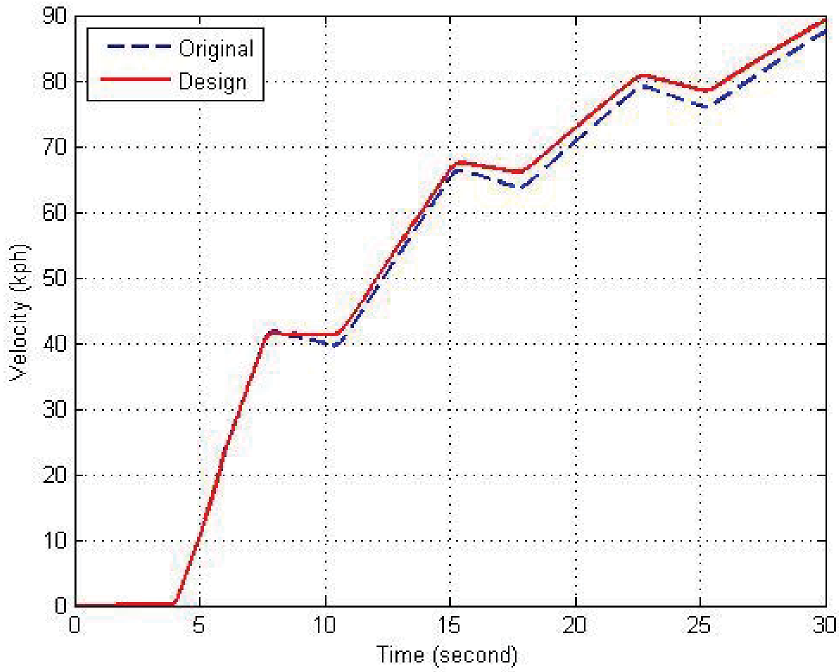

Figure 5.48 Velocity versus Time (design)

Figure 5.48 represents the velocity of vehicle, the positive slope and negative slope have been explained in Chapter 5.1.5. The velocity of vehicle is affected. We can see that the negative slope from 8 to 11 second has been eliminated. However, since the motor torque at higher rotational speed is small, the decrease in vehicle speed at higher speed still exists. Compared with original drivetrain, the acceleration time is improved. 0 to 100 km/h acceleration time is improved by 1.324 seconds.

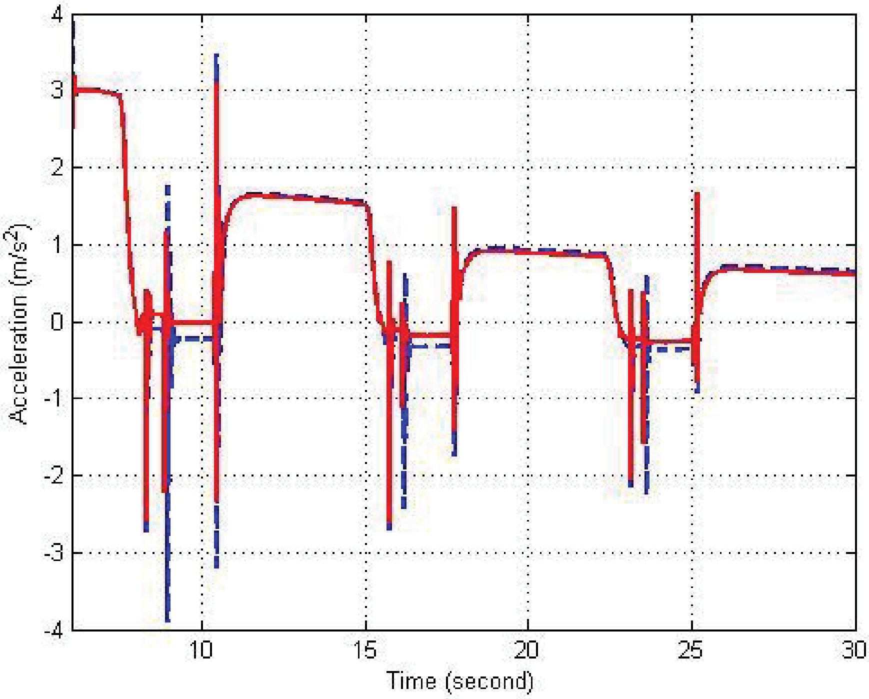

Figure 5.49 Acceleration versus Time (design)

Figure 5.49 shows the acceleration of vehicle with jerk effect on the vehicle body. Negative steady-state value represents power loss by aerodynamic drag and friction loss and the motor is not powerful enough to overcome these power losses at higher speed. The acceleration when shifting from 1st to 2nd gear is zero, which means there is no speed loss between this time span. However, as the speed of vehicle increases, aerodynamic drag increases to cause more friction force; and motor cannot provide enough torque to overcome the negative force exerted on the vehicle body.

6 Discussion

In this chapter, the simulations results of original driveline and designed driveline are compared and discussed. Also, other investigation such as altering performance of motor is attempted for achieving better improvement on manual transmission.

6.1 Results Discussion

Based on the simulation results and comparison in Chapter 5, the change in transient response of drivetrain is extracted to Table 6.1 and Table 6.2:

Unit: N.m	Input Shaft	Output Shaft	Drive Shaft
1st to 2nd gear:			
Clutch Disengagement:	**2.01↑**	**10.09↑**	**58.09↓**
Gear Synchronization:	**95.30↓**	**216.10↓**	**870.06↓**
Clutch Engagement:	**78.74↓**	**3.15↑**	**2.86↑**
2nd to 3rd gear:			
Clutch Disengagement:	**2.40↓**	**0.77↑**	**46.23↓**
Gear Synchronization:	**67.09↓**	**173.54↓**	**703.59↓**
Clutch Engagement:	**13.31↓**	**4.45↑**	**80.86↑**
3rd to 4th gear:			
Clutch Disengagement:	**1.82↑**	**0.93↑**	**44.77↓**
Gear Synchronization:	**307.59↓**	**22.07↓**	**324.65↓**
Clutch Engagement:	**19.50↓**	**14.57↑**	**124.85↑**

Table 6.1 Improvement of Torque Overshoot

Based on Table 6.1, the improvement of torsional deflection is presented. The green number represents positive effect that causes reduction in torque overshoot; the red number represents the negative effect on the shafts by torque hole compensation, and the torque overshoot is increased. For the input shaft, the negative result happens when disengaging clutch and the maximum increase is 2.01 N.m when shifting from 1st to 2nd gear. Compared with the other change in torque overshoot, these increased values are neglectable. For the other stages of shifting, the torque overshoot on the input shaft is moderate. For example, the biggest reduction is about 307.59 N.m when shifting from 3rd to 4th gear.

For the output shaft, applying additional torque on the output shaft does not have impactful improvement. For instance, the torque overshoots on output shaft increases when disengaging and engaging clutch. The biggest increase is about 14.57 N.m. The decrease in torque when synchronizing gears is very obvious. For example, the torque overshoot when upshifting from 1st to 2nd gear is improved by 216.1 N.m.

For the drive shaft, there is more torque overshoot reduced by electric motor. For example, the maximum reduction is about 870 N.m when synchronizing gears. However, there is also increase when engaging the clutch. The torque overshoot is increased by 124.85 N.m.

Unit: $\times 10^{-3}$ rad	Input Shaft	Output Shaft	Drive Shaft
1st to 2nd gear:			
Clutch Disengagement:	**0.021↑**	**0.257↑**	**0.293↓**
Gear Synchronization:	**1.025↓**	**4.957↓**	**4.405↓**
Clutch Engagement:	**0.840↓**	**3.125↓**	**1.322↓**
2nd to 3rd gear:			
Clutch Disengagement:	**0.025↓**	**0.008↑**	**0.234↓**
Gear Synchronization:	**0.718↓**	**3.978↓**	**3.564↓**
Clutch Engagement:	**0.141↓**	**0.096↑**	**0.413↑**
3rd to 4th gear:			
Clutch Disengagement:	**0.018↑**	**0.028↑**	**0.236↓**
Gear Synchronization:	**3.308↓**	**0.173↓**	**1.643↓**
Clutch Engagement:	**0.212↓**	**0.563↑**	**0.634↑**

Table 6.2 Improvement of Torsional deflection

For the torsional deflection of the shafts, the values of torsional deflection are related to torque overshoot. The higher torque applied, the bigger torsional deflection exists on the shaft. Therefore, if torque overshoot increases, torsional deflection also increases. The increase in torsional deflection of drive shaft is 0.634×10^{-3} rad. The biggest reduction is 4.957×10^{-3} rad. This means torque hole compensation effectively reduces the risk of shaft failure.

6.2 Alteration of Motor Performance

The motor utilized in the design has a maximum output power of 4kW. The simulation of designed drivetrain also concerns about increasing motors output power. For example, the power of motor is increased from 4kW to 32kW with increment of 4kW. The simulation results are indicated as figures of torque overshoot, torsional deflection and other system responses.

6.2.1 Engine and Clutch

The simulation results of each performance are combined for comparison:

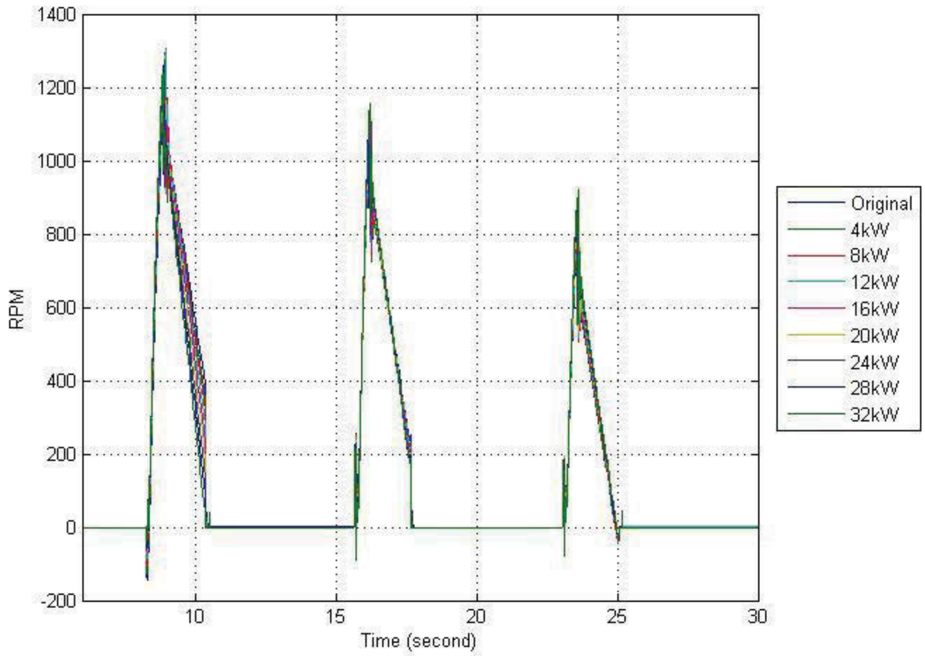

Figure 6.1 Clutch slip versus Time (Combined)

In Figure 6.1, the clutch slip for motors with different output power are illustrated, the difference between each alteration is very tiny, there seems no essential effect from torque hole compensation.

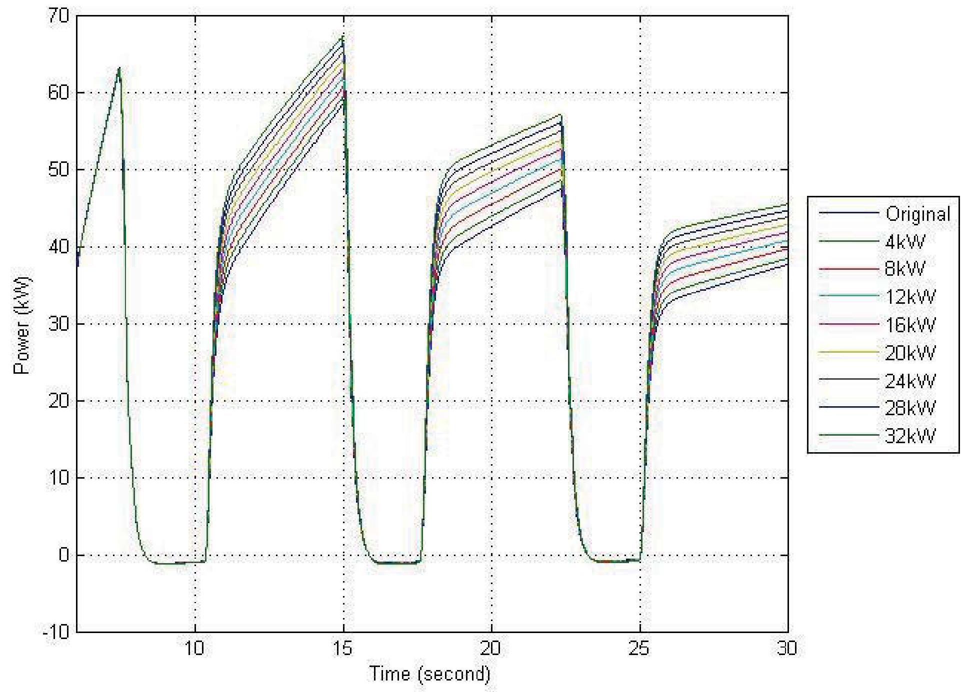

Figure 6.2 Engine power versus Time (Combined)

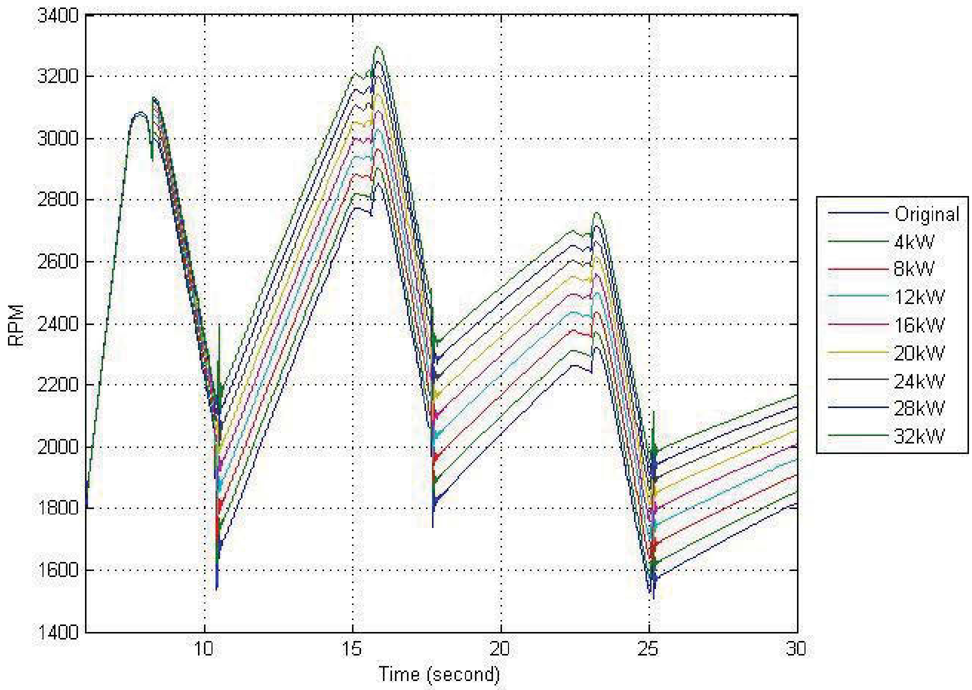

Figure 6.3 Engine RPM versus Time (Combined)

In Figure 6.2, the engine power increases as motor power increases, because when engine is decoupled from transmission, motor accelerates the entire drivetrain instead of engine. When engine is reconnected to transmission with higher rotational speed, shown in Figure 6.3, more power can be transferred from engine, so the engine will be operating within its more efficient performance range. Therefore, electric motor can help engine operating with higher efficiency.

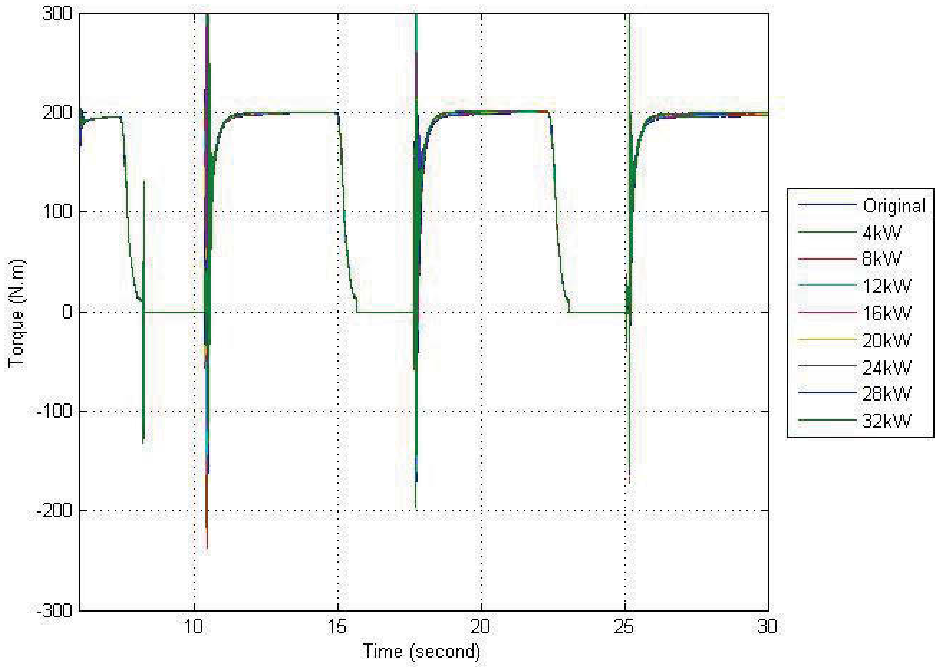

Figure 6.4 Clutch Torque versus Time (combined)

Figure 6.4 shows the effect on clutch torque by motors with different output power. The effect has been explained in Chapter 5.2.1, accelerating input shaft to higher speed can make the clutch engagement smoother because the speed difference between flywheel and friction disk is reduced. Therefore, higher motor power can reduce the wear of clutch and hence extend the lifetime of clutch.

6.2.2 Electric Motor

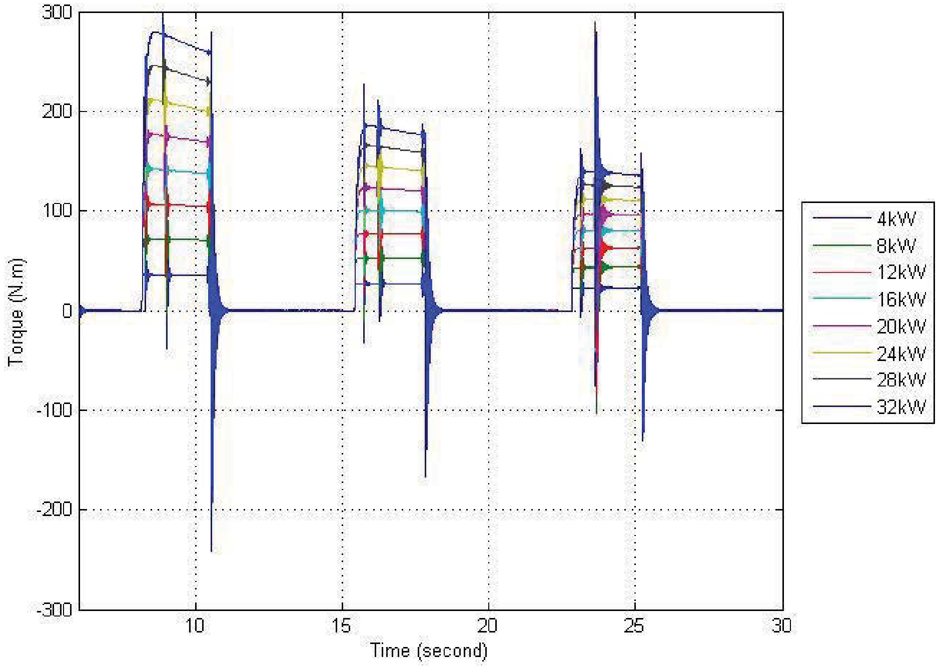

Figure 6.5 Motor Torque versus Time (combined)

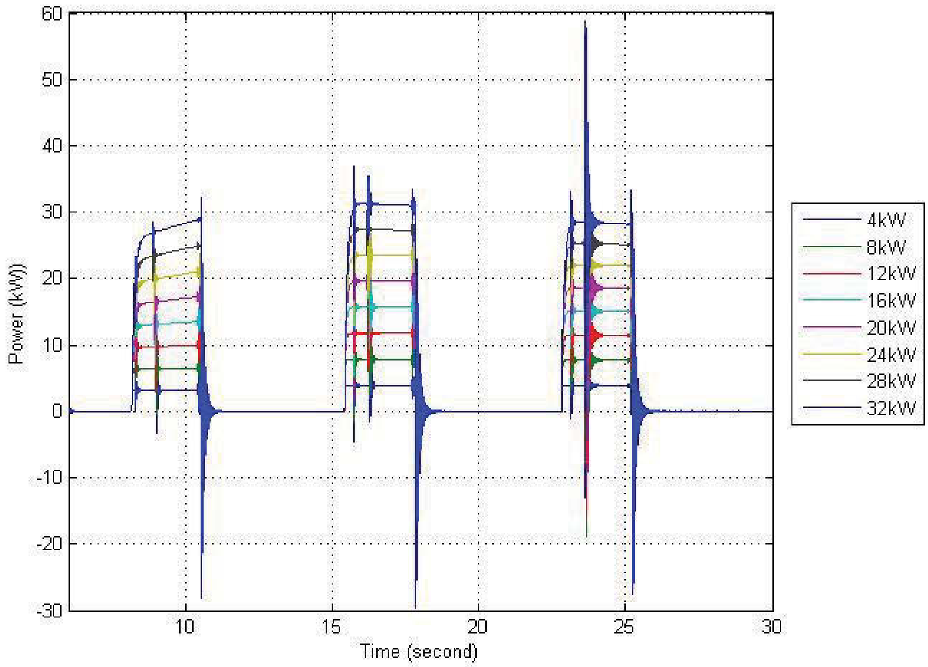

Figure 6.6 Motor Power versus Time (combined)

The output torque and power of electric motor are illustrated in Figure 6.5 and Figure 6.6. The motor torque and power are relatively constant, and the oscillations in torque and power of motor are transferred from output shaft of transmission system because motor is connected to output shaft with a mechanical shaft with 20cm length. Based on the formula in Chapter 5.2.5, the motor with higher power can provide more torque. Beside this, there is nothing particular to be discussed.

6.2.3 Transmission Shafts

Similar to the other figures, the responses of shafts are plotted based on the data exported from SIMULINK. For example, Figure 6.7 and Figure 6.8 indicate the torsional deflection and torque of drive shaft.

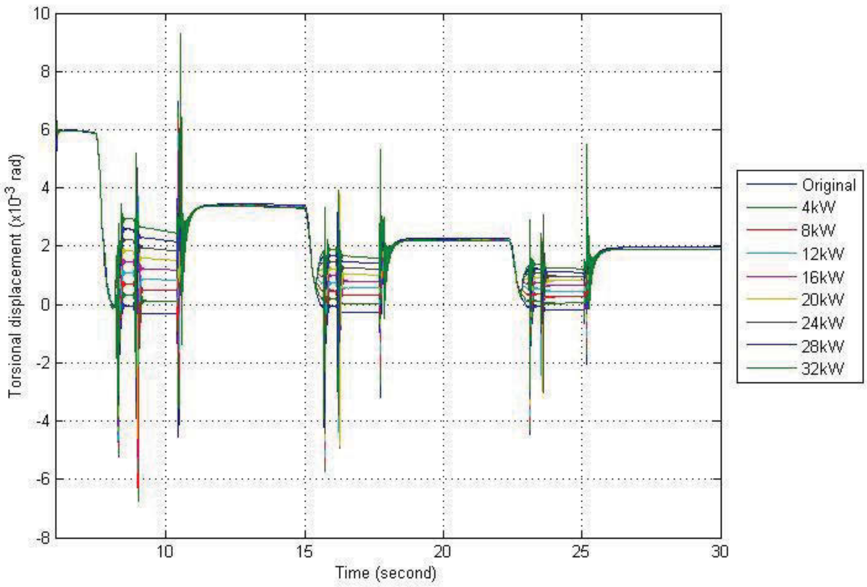

Figure 6.7 Drive Shaft Torsional deflection versus Time (combined)

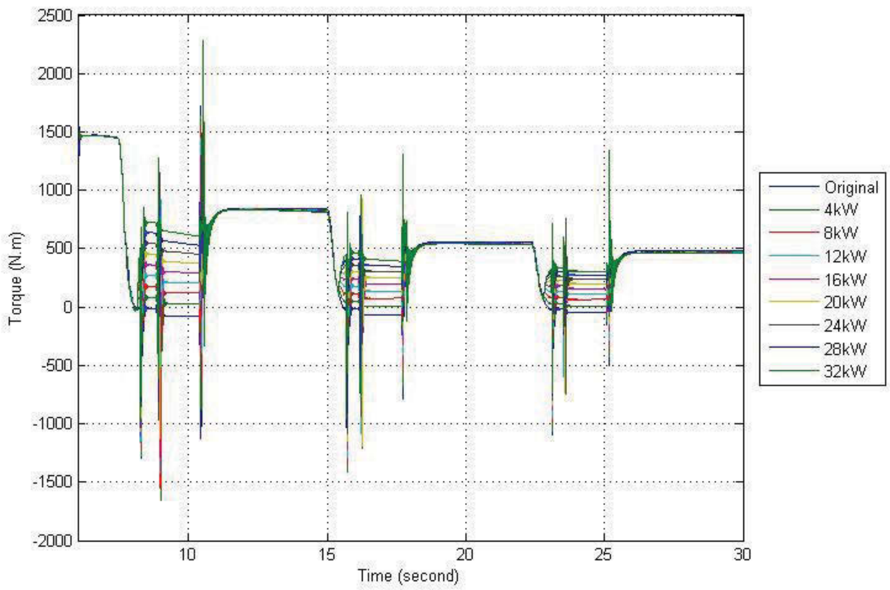

Figure 6.8 Drive Shaft Torque versus Time (combined)

The maximum values of torque overshoots and torsional deflection cannot be clearly identified from the figures. Therefore, the peak magnitude of torsional deflection and torque overshoot are extracted from the figure and the trendline is plotted in Microsoft Excel.

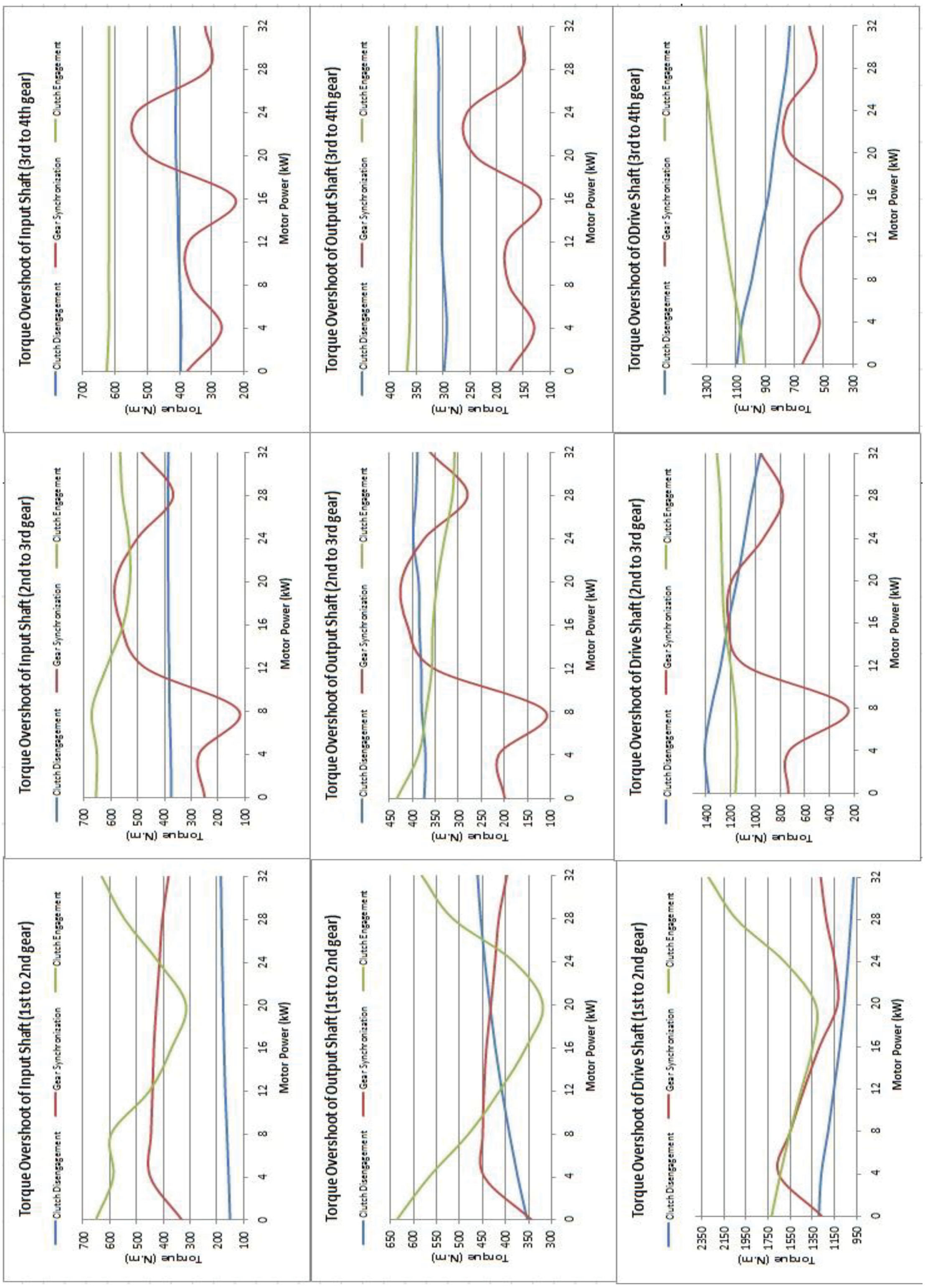

Figure 6.9 Torque Response of Shaft versus Motor Power

In Figure 6.9, the torque overshoot on each shaft versus motor power is plotted. The intersection point at power equal to 0 means the response of original drivetrain without electric motor. For all of the shafts, the responses due to engaging and disengaging clutch are linearly proportional to motor power but the response due to gear synchronization seems to be erratic because the synchronizer has a critical speed to achieve smooth synchronization and upshifting process is done manually by driver, so this may be considered as an uncontrollable or unpredictable factor to the simulation. Therefore, it cannot be used to determine the best option of motor power.

The torque overshoots on the input and output shafts increase linearly as motor power increases while the torque overshoot on drive shaft decreases dramatically. This shows that increasing motor power causes regression on the responses of input and output shafts. However, when the clutch is being engaged, the responses on the shafts are totally opposite to the responses when disengaging the clutch, which means the torque overshoot on the drive shaft increases while torque overshoots of the other two shaft decrease.

The summary of effect by increasing motor is indicated in Table 6.3

	Clutch Disengagement	Clutch Engagement
Input shaft	Unimproved	Improved
Output shaft	Unimproved	Improved
Drive shaft	Improved	Unimproved

Table 6.3 Effect on shafts by increasing motor power

The best option for choosing a motor is based on the cost and benefit to the drivetrain system. In this motor performance analysis, increasing motor power can greatly reduce the torque overshoot and torsional deflection on the drive shaft. Compared with this, the increase in torque overshoot of input and output shaft is inappreciable. However, cost of motor is also one of the concerns when choosing motor, the motor with higher performance has higher cost, but the effect is notable. The optimal choice is totally decided by the investment.

6.2.4 Vehicle Body

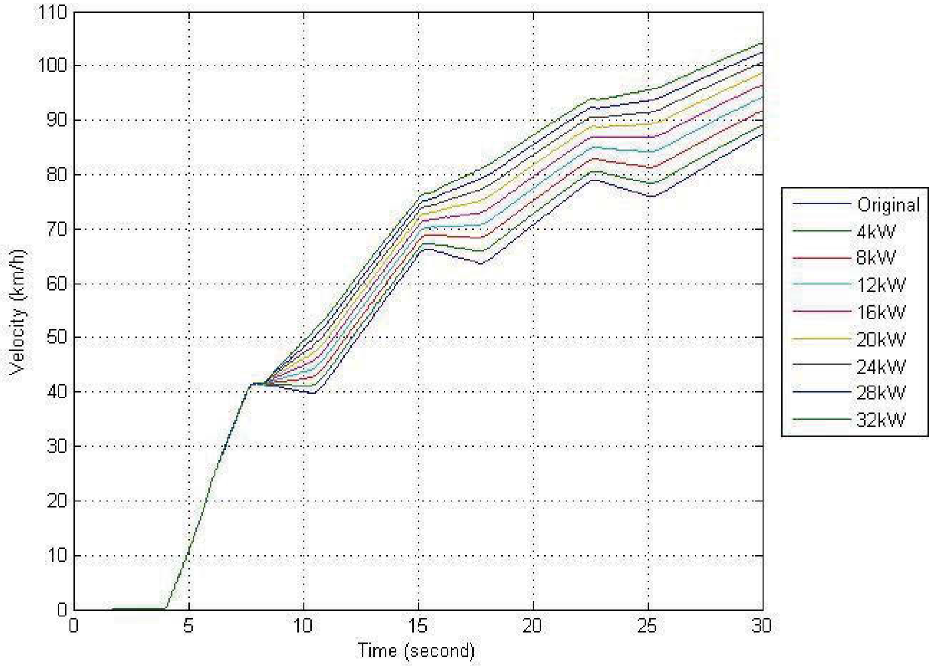

Figure 6.10 Velocity versus Time (Combined)

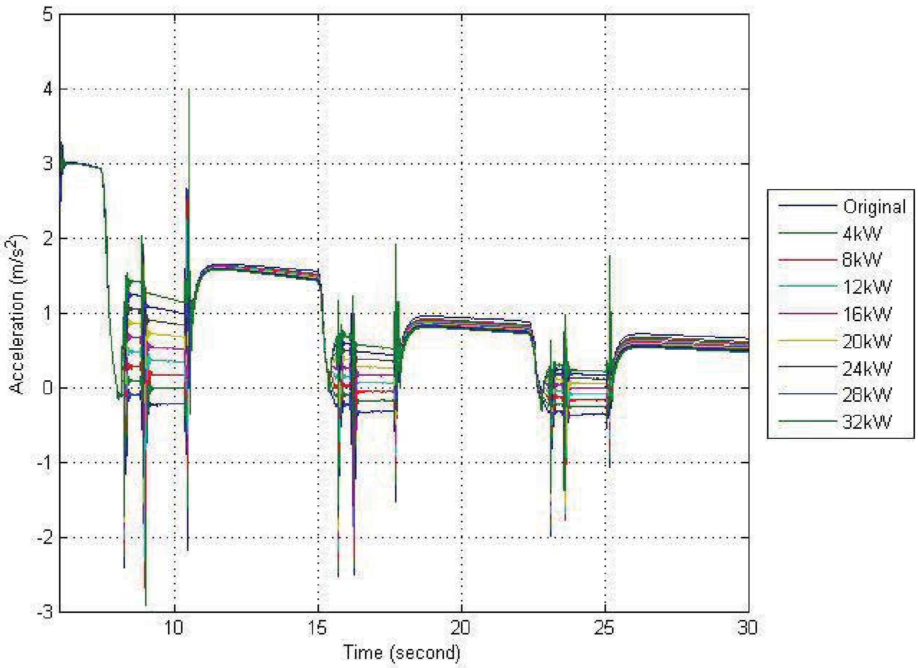

Figure 6.11 Acceleration versus Time (Combined)

Figure 6.10 and Figure 6.11 show that the acceleration of vehicle during upshifting is improved by electric drive unit. If the torque from transmission system is more consistent, the acceleration rate of vehicle body is more constant. In Figure 6.11, the plot of 32kW motor shows that it has the highest acceleration rate. In Figure 6.10, the negative gradient in velocity can be totally eliminated by a 16kW motor. To determine the jerk on vehicle body, the overshoot of acceleration is an essential factor that directly affects driving comfort. Figure 6.11 shows that motor with higher power can improve the fluctuation in acceleration when disengaging clutch and synchronizing gears. However, based on the figure of drive shaft torque, more torque transferred cause negative effect when engaging the clutch. Therefore, higher motor output can cause worsen jerk effect when engaging the clutch and deteriorate the driving comfort.

7 Conclusion

In conclusion, the powertrain of traditional family sedan with manual transmission can be optimized by installing an electric drive unit to output shaft. Since the vehicle is using more than one power sources, we can say that the vehicle is hybridized. The overall configuration of components is similar to a parallel electric hybrid vehicle because the electric drive unit and conventional drivetrain unit are connected in parallel. The drive modes of this hybridized vehicle contain both internal combustion engine drive mode and electric drive mode. However, electric drive range is extremely short because electric motor only operates in the period of upshifting.

The simulation results show the agreement with the purpose of this thesis research. With a 4kW electric motor, the transient response of drivetrain system, such as torque overshoot and torsional deflection, is improved. For example, the maximum torque overshoot reduction on the driveshaft is more than 800N.m. This result is premising because if the torsional deflection of shaft can be minimized, then the dimension of shafts can be downsized for saving the material cost. However, there is also negative effect caused by this torque hole compensation method. For instance, the torque overshoot and torsional deflection of input shaft are slightly increased when disengaging the clutch, but this may not be a serious issue because the raise in these values are not significant. Also, increasing motor power can worsen this result.

The speed loss due to internal friction and aerodynamic drag is partially reduced. From simulation result of vehicle velocity, if the motor power is increased to 16kW, there will be no negative acceleration on vehicle body, which means the negative forces by aerodynamic drag and internal friction are overpowered. Based on Section 5.2.6, since the torque overshoot on the shafts is reduced, the fluctuation of vehicle longitudinal acceleration is improved, and this could improve driving comfort and 0 to 100kph acceleration time.

The further improvement of this design is to apply the regenerative braking, so more energy can be recovered from decelerating and used in accelerating the vehicle. Also, the torque assist mode like a mild hybrid electric vehicle can be considered to support internal combustion engines operating in its efficient range.

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