Eliminating the Torque Hole: Using a Mild Hybrid EV Architecture to Deliver Better Driveability

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Abstract—**Hybrid vehicle engineering has traditionally and dominantly focused on fuel economy benefits and emissions reductions. Although the transient power delivery benefits of hybrid powertrains are well-understood, these are not a primary focus of the majority of research and development efforts, with some exceptions. Our approach to this problem is to deliver a low-cost, low-tech mild-hybrid powertrain, with unique power delivery features designed to appeal to price-sensitive, but aspirational consumers. The powertrain is a simple post-transmission parallel hybrid configuration. It utilizes a low-powered four-cylinder engine coupled to a four-speed manual transmission through a robotically-actuated clutch. A low-voltage BLDC motor is directly connected to the transmission output shaft, before the final drive. Our research focuses on bringing the benefits of HEV architecture to the world's developing cities, where, it can be confidently argued, local emissions reductions are needed the most. Crucial to the success of this research is the understanding that compared to an equivalent ICE-powered vehicle, an HEV competes at a price disadvantage, no matter how cost-effective the solution is. This disadvantage is amplified in regions of low-middle income, where price sensitivity is greatest. It must, therefore, present better value than an equivalent conventional vehicle if it is to be commercially successful in these particularly price-sensitive markets. We discuss the extent to which control can be used to deliver transient power delivery gains in such a setup, and offer an example powertrain for simulation. To validate the concept, simulation of this research is performed in MATLAB and Simulink. The prototype is based on a generic engine and a BLDC motor. The results mainly focus on the electric drive and comparison of the transient response of drivetrains.**

Keywords—Mild HEV; BLDC; Torque-fill; Simulation; Manual transmission;

I. INTRODUCTION

Society's concern with oil depletion, global warming, fuel economy and more stringent vehicle emissions standards have led many automotive manufacturers to produce alternative energy vehicles, which are more fuel-efficient and environmentally friendly than internal combustion engine (ICE) powered vehicles but do not sacrifice drive comfort or performance. Hybrid Electric Vehicles (HEVs) provide higher fuel efficiency and lower emissions through the combination of the conventional internal combustion engine coupled with electric machines in varying degrees of hybridization [1].

We aim to develop an ultra-low-cost electric hybrid drive system for small vehicles as a proof of concept [2]. Our low-cost drive system is designed primarily for developing markets, due to the correlation these markets have with high atmospheric pollutant levels [3, 4]. Our goal is to develop a hybrid vehicle that has drive characteristics (in particular, shift characteristics) of an automatic vehicle, using low-cost hardware. The hybrid drive system is being developed with the ultimate goal such that in a mass-manufacturing situation, the total extra cost of the system as compared to a traditional automotive powertrain would not exceed 5% of the expense of the base vehicle as manufacture cost for hybridization, which includes the inverter, motor, and battery. This figure is the price point at which the mild hybrid architecture proposed will make immediate business sense from a consumer perspective. From [5], we approximate the cost of a 1.2 kWh battery to be US$60. This cost yields a total system cost of US$450. Our system is designed to be suitable for low-end cars typically sold in developing nations, and would serve both to reduce fossil-fuel dependency in these regions as well as improve air pollution characteristics, which are typically poor owing to urban particulate matter [6]. Because of the price sensitivity of these markets, at costs above the upper limit specified, further justification is necessary. Further investigation is required in the area of cost-reduction to reach and exceed this goal.

The hybrid architecture selected includes an I4 ICE coupled directly to a 4MT. At the output shaft of the MT, a low-voltage BLDC motor is rigidly coupled coaxially, so that it is always spinning at shaft speed. The electric machine is then coupled to the driven wheels through the differential. By lowering the motor voltage as far as practically possible, we limit the cost of associated power electronics and battery [7]. A directly coupled motor also limits the cost of associated mechanical hardware, which includes clutches or gear trains. In recognizing the power density disadvantage of a low-voltage system, we initially develop the system for torque-fill during gear change. By conducting further studies (not discussed herein), we may then optimize the remaining componentry to take full advantage of fuel economy benefits and examine further operating modes, such as ICE-assist, ICE-start, electric only, regen, battery-charge, anti-jerk.

The literature includes similar architectures to that proposed herein [8-12]. Of these, Baraszu (2002) most closely resembles the architecture proposed herein. However, our proposed architecture is simpler still, by the omission of the motor clutch. Moreover, the primary focus of our research is

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as much on the socio-environmental impact of the technology as it is on the technical development. Our project constraints are derived from the fundamental goal of bringing hybrid technology to developing nations with high mortality rates attributed to environmental pollution.

The literature also includes research into the control of gear shifts in automated manual transmissions [13-15]. The literature mainly focuses on the study of control during the shift for optimization of; comfort, the speed of shift, and fuel economy. It appears there is some novelty in the investigation of a shift control strategy for a post-transmission hybrid architecture such as ours. The novelty arises out of the use of low-voltage, low-power hybridization together with low-cost mechanical hardware, coupled with innovative control methods to deliver powertrain architecture specifically designed for markets that do not have high adoption rates of LEV/PZEV/ZEVs. Our hope is that by identifying this market need, we may begin to see OEMs actively working in this niche to solve the identified socio-environmental problem.

In previous research, our research group has studied a similar architecture, and an initial concept was developed [16]. This paper utilized a flexible body model to study the initial concept. The results of this study have been used as a starting point for whole powertrain integration and vehicle optimization that will be undertaken as part of this broader research project. This paper presents our initial study, which is a simulation, comparison and analysis of two types of drivetrain; a mild hybrid electric powertrain equipped with a manual transmission and output shaft-mounted electric motor, and an otherwise identical conventional ICE/manual transmission powertrain. The comparative powertrains presented are modeled and controlled to reduce the torque hole and improve the driving comfort. Moderately successful results are achieved regarding fuel economy, torque-hole reduction, and acceleration performance, providing a significant improvement over the conventional vehicle. The results suggest further development of the concept would be beneficial.

II. DESIGN CONSIDERATIONS

In developing a drivetrain architecture that is relatively cheap to manufacture, and offers both smooth torque transfer during the gear change, as well as a degree of damping against torque oscillation, a number of options may be considered. The cost-effectiveness must be balanced against the performance criteria required. Principally, there are options related to the type of gearbox, magnitude of hybrid assistance, and the electric motor placement. The placement of the motor also affects the required motor speed and torque characteristics. Table I is presented below, qualitatively assessing the primary options for gearbox type and level of hybrid assistance [17].

Qualitatively, it can be seen that a mild hybrid coupled with an AMT or MT represents the greatest opportunity for improved drivability at the lowest manufacturing cost. Such an architecture calls for a small electric motor at the transmission output, coupled to a controlled power source. Primary input signals to the motor control are: clutch position, ICE torque (calculated from speed and throttle angle), and selected gear. The function of the motor is to eliminate or reduce the torque hole during gear changes by providing a tractive force when the clutch is disengaged, and also, provide damping for torque oscillation, particularly during gear changes and take-off (commonly known as anti-jerk). In another instance, the electric motor may act as a motor or generator in certain driving situations.

TABLE I. GEARBOX TYPE AND LEVEL OF HYBRID ASSISTANCE.

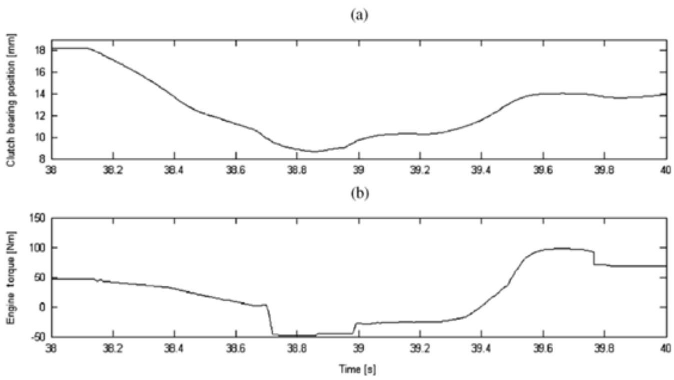
	Sub-mild hybrid (<10% electric assist)	**Mild hybrid (10%<electric assist<20%)**	**Full hybrid (20% or more electric assist)**
Level of hybridization	Limited benefit – torque hole may not be eliminated in many cases, and hybrid assist is not strong	Higher cost but may eliminate torque hole in most cases. Can also be used for anti-jerk	Highest cost – but offers no benefit compared to power split devices and is not economically sound. Packaging of motor and battery difficult
Automatic Transmission (AT)	High manufacturing cost Smooth gear-change characteristics – no need for supplementary torque fill		
Continuously Variable Transmission (CVT)	Cheaper than Automatic but lower torque capacity No discrete gear changes – no need for supplementary torque fill		
Dual Clutch Transmission (DCT)	Relatively high manufacturing cost Constant meshing eliminates torque hole		
Automated Manual Transmission (AMT)	Lower manufacturing cost – clutch and shift hub actuators may be incorporated into hybrid architecture Poorest gear change characteristics Motor must be placed after the first reduction		
Manual Transmission (MT)	Lowest manufacturing cost Motor can only be placed after the first reduction		

III. TORQUE HOLE IN MANUAL TRANSMISSIONS

Automotive transmission technology today revolves around a few base technologies. These are MT, AT, semi-automatic types such as the DCT and AMT and, the CVT. The manual transmission is the cheapest to manufacture and offers the highest power transfer efficiency, but also faces drawbacks regarding customer acceptance and shift quality. In particular, the traditional manual transmission architecture offers little damping of torque oscillations or step-changes through the driveline (as compared to other types of transmission architecture), and is incapable of transmitting torque during a gear change. These results in a torque hole that is felt by passengers as “stilted” progress as compared to the relatively smooth delivery of automatic and CV transmissions [18].

The main weakness of a manual transmission is that it is not easy to drive as it requires the driver to provide the actuation force for the clutch and gear selector. As a result, gearshifts are sometimes not smooth which causes jerk or jolt on the vehicle body and affects driving comfort for passengers who will notice an interruption of engine torque to the wheels. The manual shift process is achieved through the engagement and disengagement of the clutch. The entire shift process can be divided into three main stages. The first is clutch disengagement, where the driver depresses the clutch pedal, but gear ratio remains constant. The engine is decoupled from the gearbox, so vehicle speed decreases due to internal friction, rolling resistance, and aerodynamic drag. The second is gear synchronization, where the driver selects the next gear for engagement. Finally, clutch engagement, where the clutch pedal is released, engine throttle is opened to apply torque to the transmission. During clutch disengagement, a speed differential is introduced between the clutch friction faces, and the output torque varies sharply, the discontinuity is called torque hole or torque gap. The torque hole is defined by its width, which is the time interval of entire shift process. A larger torque hole results in more significant deceleration of the vehicle. The torque hole is also characterized by torque oscillation at its extreme ends, which reflect the smoothness of clutch disengagement and engagement, related to the shock and vibration of the gearshift [19]. In a traditional manual/ICE powertrain, the depth of the torque hole is equivalent to the torque value immediately before the initiation of the gear shift. Reducing the torque hole introduces a torque value somewhere between zero and the torque value immediately before the gear shift, to smooth the shift process (see Fig. 1) [15].

Fig. 1. Zoom of the upshift from gear 2 to gear 3 with the medium wear of the clutch. (a) Throwout bearing position. (b) Engine torque.



IV. SIMULATION MODEL DEVELOPMENT

In this paper, Simulink and Matlab were used to model a front wheel driveline that has a four-speed manual transmission. The modeling is established based on a typical ICE powertrain with manual transmission, equipped with an electric motor at the transmission output shaft [20]. The application of the lumped parameter method for higher order powertrain models makes use of both powertrain characteristics, in terms of shaft stiffness and rotating inertia, in conjunction with the physical layout to produce representative models for different powertrain configurations. The system is divided into numerous elements to develop the model; these elements are detailed in Fig. 2. Elements for the engine, flywheel and pressure plate, clutch plate, transmission gears, shaft, differential, driveshaft, wheels, and vehicle inertia are included. Assumptions have been applied to reduce the complexity of the powertrain and reduce the computational demand. It is assumed that there is no backlash in the gears, eliminating high stiffness elements in the model. More detailed specific features are shown in Table II.

Fig. 2. A dynamic model for Powertrain System.

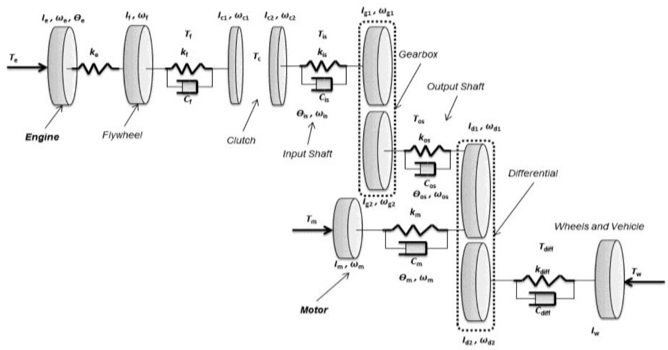
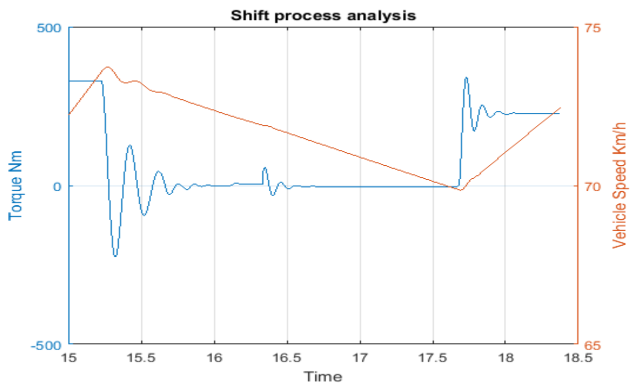


TABLE II. VEHICLE GLOBAL SPECIFICATIONS.

Engine Type	Spark-Ignition	
Cylinder Type	4 cylinders	
Idling speed (rpm)	800	
Maximum Power (kW)	85	
Gear ratio	First	3.581
	Second	2.022
	Third	1.384
	Fourth	1
Vehicle Mass (kg) as hybrid	1200	
Differential ratio	3.1	
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	

Simulation results of this research are mainly focused on the torque hole. The manual shift process is usually achieved through the engagement or disengagement of the clutch. Fig. 3 shows the curves of output torque and speed during a gear change from gear 3 to gear 4.

Fig. 3 Shift process analysis.



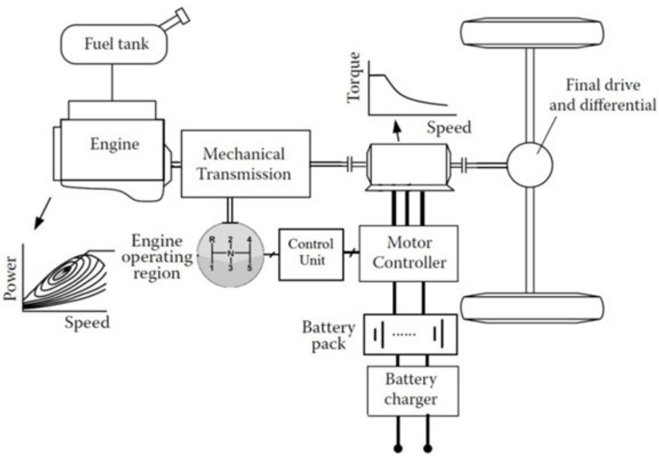
V. MODELING CONSTRAINTS

The mass of the vehicle with traditional ICE is set at 1100 Kg. For the purpose of the hybrid proof-of-concept, a weight penalty of 96 kg was deemed reasonable. The BLDC motor chosen for this study has a mass of 16kg. Estimation of the mass of other componentry included a 40 kg battery, power electronics and controller with a mass of 10 kg, wiring harness including high-power cabling of 10 kg, and housings, powertrain components, and bracketry of 20 kg. Therefore, the nominal mass of the mild hybrid vehicle is 1196kg. The Simulink model was experimentally validated in a laboratory facility at UTS. For the purpose of validation, the electric machine is powered by a grid-connected 157 kW variable DC power supply this is due to safety issues associated with battery storage in the laboratory [21].

VI. MILD HYBRID ARCHITECTURE

Fig. 4 shows the schematic of the drivetrain designed for torque-hole elimination. The electric motor is mounted on the output shaft of the transmission as it is required to provide continuous output torque to the wheels. This solution represents the simplest practical method for providing continuous torque during the gear change. If the electric motor were mounted closer to the engine, a secondary torque path would be required, bypassing the primary drivetrain and adding both cost and complexity. The selected design is therefore characterized as a mild hybrid electric vehicle, consisting of an electric motor of relatively low power output (10 to 20 kW) compared to that of the engine. As the speed of the electric motor is directly proportional to wheel speed, only limited effectiveness may be realized if the electric motor is used to assist the ICE [22]. This limitation is the principal disadvantage of this configuration.

Fig. 4. System Architecture.



A BLDC is selected as the most appropriate type of electric motor to utilize in this application. Its advantages in HEV and EV applications are well known and include high power density (reducing weight and volume), higher efficiency, low maintenance, and efficient heat dissipation [23-25].

To select an appropriate electric motor, an analysis of the New York City Dynamometer Drive Schedule (NYC Cycle) was conducted [26]. The NYC cycle was selected as the most appropriate representation of the target market for our proposed vehicle architecture. The analysis used a speed trigger to determine the number of gear shift events over the course of the cycle, eliminating superfluous events by excluding shifts made within 5 seconds of another. In cases where speed decreased then increased across a speed trigger over any 5-second window, a superfluous downshift was deleted. Where the speed first increased then decreased across a trigger, then a superfluous upshift would also be deleted.

For the total cycle length of 598 seconds, 42 gearshifts were required. Ignoring downshifts, the average power required at upshift was calculated to be 9.47 kW, with only five upshift events requiring over 10 kW (see Fig. 5). A similar plot can be produced for motor output torque using the target speed at each projected gear change. As the motor is placed after the first reduction ratio, it is independent of this variable, and the output torque is, therefore, a simple function of the target speed and required power. All but four gear change events are found to require less than 130 N.m output torque for torque fill-in. The motor speed and torque characteristics can then be determined by plotting these variables for each gear change event.

Fig. 5. Power required at gear change.

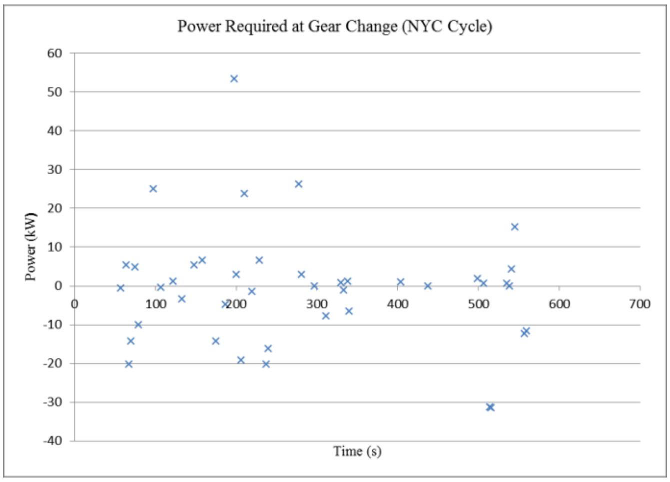
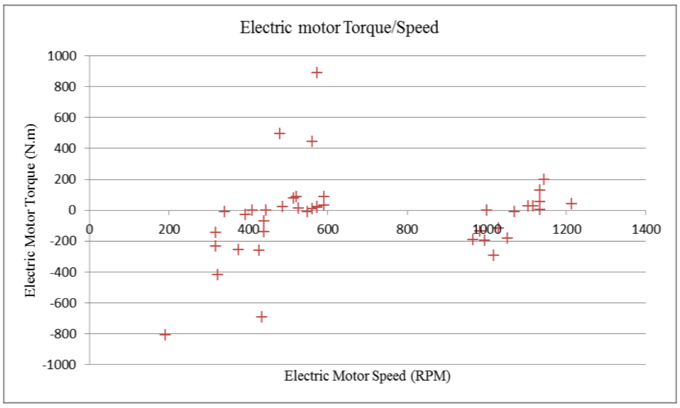


Fig. 6. Electric motor Torque speed.



Based on the NYC cycle analysis, the Motenergy ME0913 (12 kW/30 kW pk.) motor is selected as a suitable motor for initial testing. Although its peak torque is somewhat lower than ideal, it is the closest off-the-shelf solution to meet our hardware needs. Bench-testing yields characteristic curves, which are included in the Simulink program. The system is programmed to function in two operation modes. In the engine drive mode, the clutch is engaged, and the throttle is opened. Rotational motion is transferred from the engine to the clutch disk, counter gear, output shaft, final drive gear and then to the wheels. At this time, the motor is free-rotating with the output shaft. It induces a small amount of power loss as it is permanently connected. The electrical connection between motor and battery is switched off. In the torque-fill mode, when the clutch is disengaged, no power flows from the engine to the gearbox. The BLDC motor, which is already spinning at output shaft speed, is energized. Torque is transferred from the motor to the final drive gear and then wheels. The motor becomes the main power source in the torque-fill mode, providing the sole motive power source for approximately 2 seconds. The torque-fill mode is active until the next gear is synchronized, and the clutch is re-engaged [15].

VII. SIMULATION RESULTS AND ANALYSIS

Both a conventional and a torque-fill drivetrain are modeled for comparison. The results mainly focus on the mild hybrid design and compare the transient response of drivetrains. Figure 3 represents the output shaft torque of the conventional and torque-fill drivetrain when upshifting from 3rd to 4th gear. In each upshift event, there are three discrete torque oscillation responses. Disengaging the clutch causes the first torque excitation. When the clutch is decoupled, the engine and flywheel inertia are decoupled from the transmission. This sudden change in inertia causes excitation of torque response. Synchronizing gears causes a second, smaller oscillation. When the previous gear is desynchronized, and the next gear is locked on the output shaft, this causes a variation in shaft speed due to different gear ratios. The third oscillation occurs when the clutch is re-engaged. A torque overshoot can occur due to different rotational speeds between flywheel and clutch disk [27]. In Fig. 7, the torque excitations on the output shaft are clearly illustrated. The torque profiles of original drivetrain and the torque-fill drivetrain are compared. When the system is operating in torque fill-in mode, it is shown that the torque hole is reduced, as well as a marked reduction in the oscillatory peak, by approximately 50 N.m. In Fig. 8 shows the actual vehicle data for half-shaft torque (with torque fill during shifts). Good correlation in both frequency and magnitude are clear [28]. Fig. 9 shows the velocity of the vehicle during an acceleration event 0-100 km/h. The acceleration time is reduced by approximately 2.5 seconds using the torque-fill drivetrain, and the deceleration at each gear shift is reduced markedly.

VIII. FUEL ECONOMY BENEFIT

Although the focus of the torque-fill system is not primarily a fuel-economy benefit, by reducing the time taken to achieve cruising speed, a fuel economy benefit may be realized. A comparison of the torque-fill drivetrain and the conventional ICE-only powertrain was conducted to investigate the effect on fuel economy. Table III shows that a minor benefit is realized when using the torque-fill drivetrain, showing a saving of approximately 4.16% fuel consumption over the conventional powertrain.

TABLE III. FUEL CONSUMPTION SUMMARY.

Vehicle Model	Fuel Consumption L	Distance Traveled km/L	Fuel Economy L/100 km
Conventional	0.03593	11.86	8.432
Mild HEV	0.03627	12.37	8.081
Difference		0.51	0.351

Fig. 7. Simulated Torque profile.

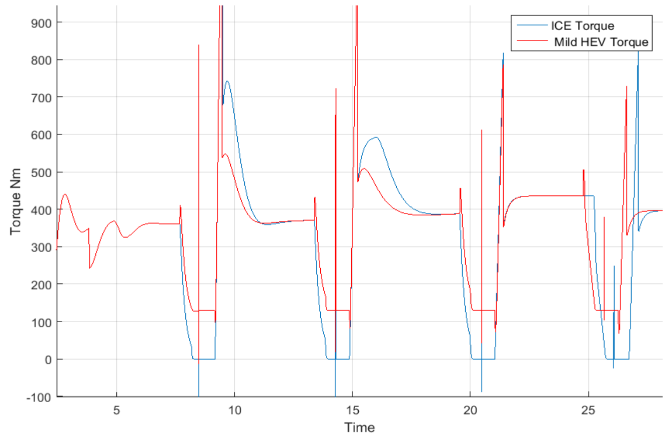


Fig. 8 Actual Half-shaft torque with Fill-in [8].

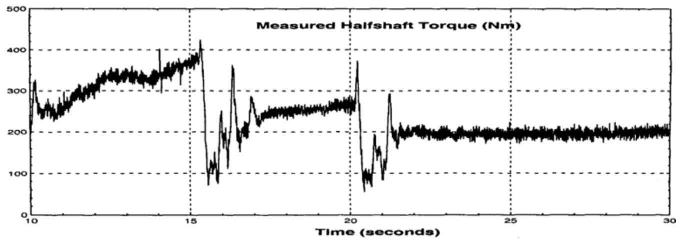
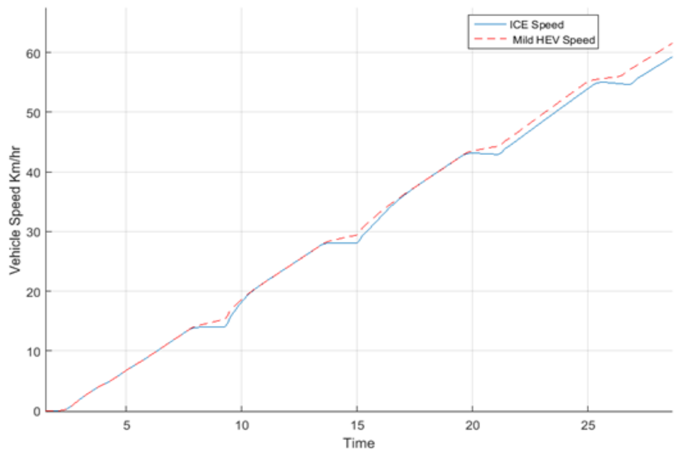


Fig. 9. Wheel Speed as a function of time.



It is clear that a simple mild-hybrid design as proposed would likely yield worthwhile benefits in both fuel economy and driveability. Such a design is favoured for its simplicity, requiring lower development time, as well as its manufacturing cost, which opens the possibility of increasing market penetration of low-emissions vehicles in price-sensitive markets. The investigation presented has shown that for the purpose suggested, an electric motor generating approximately 130 N.m and 16 kW would be ideal. Nonetheless, an electric motor with lower capacity also yields gains.

The advantages of the proposed design are that a single electric machine may be used in many modes, filling torque-hole during gear shifts, anti-jerk control, regeneration, ICE assist, and electric vehicle mode at low speeds to enable ICE stop-start. When coupled with a clutch and gearshift actuator, the operation of the powertrain will be fully automatic, requiring no driver intervention and achieving superior driveability measures. The NYC drive cycle was used to specify the motor characteristics. These characteristics were programmed into Simulink, and initial simulations have focused on shift quality and fuel economy, using a 30-second acceleration test as a benchmark. The simulation shows that improvements of up to 36% are possible in acceleration performance while a small fuel economy benefit is also noticeable. Further simulations will be conducted using the full NYC cycle and several benchmarking tests to develop further the torque-fill controller, as well as introduce the modes described above. However, the initial simulations clearly show that there are gains to be made.

Finally, the cost of manufacturing and assembling the system must be fully investigated, as the retail cost of components bears no resemblance to the manufacturing cost. At a minimum, the system would require additions to a typical vehicle consisting of; a motor housed in the transmission, a controller/converter, and battery. These components must be carefully selected to minimize the total cost as well as maximize value and benefit to the end user.

IX. Conclusion

The torque-fill drivetrain is not designed to be a full-featured hybrid vehicle. It is intended for price-sensitive markets dominated by vehicles equipped with manual transmissions. These markets typically have a high number of aspirational new-vehicle buyers and are also often geographically located in cities with a high level of air pollution. A low-cost hybrid vehicle utilizing a driveline as proposed could be successful in such a market.

The torque-fill drivetrain can be used equally successfully with automated manual, and traditional manual gearboxes, and the limited motor power and duty cycle limits the size and cost of other system components, such as batteries and converters. Due to the intermittent operation, it is also possible to safely operate the components beyond their rated continuous output and yield greater benefit.

As shown, a 12 kW torque-fill drivetrain will improve vehicle drivability and comfort by reducing torque holes during gear shifts. Because the vehicle reaches its cruising speed quicker, it is possible to yield a small but measurable improvement in fuel economy. The greater economy may be realized at the same hardware cost by including operating modes to assist the internal combustion engine or regenerate electricity from the front axle.

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