ORNL/SPR-2019/1113
CRADA/NFE-15-05451

# Multi-Speed Transmission for Commercial Delivery Medium Duty PEDVs


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**March 8, 2019**

OAK RIDGE NATIONAL LABORATORY
MANAGED BY UT-BATTELLE FOR THE US DEPARTMENT OF ENERGY

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**CRADA/NFE-15-05451**

Energy & Transportation Science Division

# MULTI-SPEED TRANSMISSION FOR COMMERCIAL DELIVERY MEDIUM DUTY PEDVs FINAL REPORT

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# Table of Contents

## I. Acronyms and Abbreviations

| Abbreviation | Description |
|---|---|
| ADB | Advanced Design Bus (Cycle) (a.k.a. Altoona Combined Cycle) |
| ALL, ALT or ADB | Altoona (Combined Cycle made of 3 CBD, 2 ART and 1 CC) |
| A | amps |
| AMT | Automated Mechanical Transmission |
| ARB | (Heavy Duty) Air Resource Board (Cycle) |
| ART | Arterial (Cycle) (part of Altoona combined cycle) |
| BAC | Business Arterial Commuter (Cycle) |
| BP | Budget Period |
| CAN | controller area network |
| CARB | California Air Resource Board (Cycle) |
| CBD | Central Business District (Cycle) (part of Altoona combined cycle) |
| CC | Commuter Cycle (part of Altoona combined cycle) |
| CILCC | Combined International Local and Commuter Cycle |
| CSHVC or CSC | City Suburban Heavy Vehicle Cycle, or City Suburban Cycle |
| CV | Commercial Vehicle |
| CW, CCW | Clockwise, Counter Clockwise |
| DC | Direct Current |
| DCT | Dual Clutch Transmission |
| DOE | Department of Energy |
| ESS | Energy Storage System |
| EV | Electric Vehicle |
| FHWA | Federal Highway Administration |
| FD, FDR | Final Drive, Final Drive Ratio |
| FOA | Funding Opportunity Announcement |
| FTA | Federal Transit Administration |
| FWD | Front Wheel Drive |
| GCW | Gross Curb Weight |
| GVW | Gross Vehicle Weight |
| HD | Heavy Duty |
| HHDDT | Heavy Heavy-Duty Diesel Truck |

| Abbreviation | Description |
|---|---|
| HIL | Hardware in the Loop |
| HTUF | High-efficiency Truck Users Forum |
| ICDV | Internal Combustion Drive Vehicle |
| kg | kilograms |
| kW | kilowatt |
| kWh | kilowatt hour |
| KI | Kinetic Intensity |
| LD | Light Duty |
| MD | Medium Duty |
| mpgde | Miles per Gallon Diesel Equivalent |
| Nm | Newton meter |
| NRE | Non-recurring Engineering |
| NREL | National Renewable Energy Laboratory |
| NVH | Noise Vibration and Harshness |
| NYC-CC | New York City Composite Cycle |
| OC, OCTA and OCBC | Orange County Bus Cycle |
| OEM | Original Equipment Manufacturer |
| ORNL | Oak Ridge National Laboratory |
| PEDV | Plug in Electric Drive Vehicle |
| RPM | Revolutions Per Minute |
| RWD | Rear Wheel Drive |
| SLW | Seated Load Weight |
| t | tonne, long or metric ton, 1000 kg |
| TCU | transmission control unit |
| TE | Transmission Error |
| TRL | Technology Readiness Level |
| UDDS | Urban Dynamometer Driving Schedule (truck cycle) |
| V | volt |
| W | watts |
| WHDC | World Harmonized Drive Cycle |
| WOT | Wide Open Throttle |

## II. Executive Summary

This CRADA between EATON and Oak Ridge National Laboratory (ORNL) was part of the Office of Energy Efficiency and Renewable Energy (EERE), U.S. Department of Energy Vehicle Technologies Office (VTO) FOA DE-FOA-0000991, under Award Number DE-EE0006843. Thus, the following report is a subsection of the larger Award Number DE-EE0006843, covering only the portions that were collaborated on by ORNL and EATON.

The EV Everywhere Grand Challenge [1] aimed at realizing plug-in electric drive vehicles (PEDVs) that meet or exceed the performance of internal combustion drive vehicles (ICDV) based on cost, convenience, and consumer satisfaction. The average range of a PEDV is approximately one-third the range of an ICDV. The project addressed the following technical barriers:

- Performance gaps between electric vehicles (EVs) and ICDVs include range, top speed, acceleration, and gradeability;
- No reliable, affordable, scalable, and low-weight, multi-speed transmissions are available to medium-duty electric vehicle manufacturers on the market; and
- The public acceptance of electric vehicles is low.

The Eaton team proposed to develop a multi-speed transmission that will help close the range gap by increasing the electric powertrain efficiency. The project objective was to develop a new multi-speed gearbox to match the performance characteristics of an electric motor. The gear ratios, shift strategy, cost, and weight would be optimized to provide a commercially feasible solution to meet medium-duty electric vehicle requirements for starting torque, top speed, acceleration, and efficiency. The team expected that customer satisfaction would improve if vehicle acceleration, top speed, and gradeability improved significantly over the baseline vehicle.

### a. Comparison of Accomplishments to Goals and Objectives

The Eaton team successfully completed the three-year project, Multi-speed Transmission for Commercial Delivery Medium Duty PEDVs. The project started with the electric vehicle customer requirements and system analysis. The team identified vehicle-level requirements and specifications and propagated them to the subsystem and component functional requirements.

The team modeled detailed driveline design and component dynamics for the chosen topology suitable for the Smith Newton ™ vehicle in Budget Period (BP) 1 and for Proterra's BE35 Electric Transit Bus in BP2. The optimization study was extended to heavy-duty PEDVs, Gross Vehicle Weight (GVW) up to 36,000 kilograms (kg), in the digital platform only, to gage the proposed technology's scalability and energy savings and to increase deployment and market coverage. The optimized solution includes a multi-speed transmission, a permanent magnet motor, an integrated bidirectional shift strategy, improved regeneration and battery management as shown in **Figure 1**.

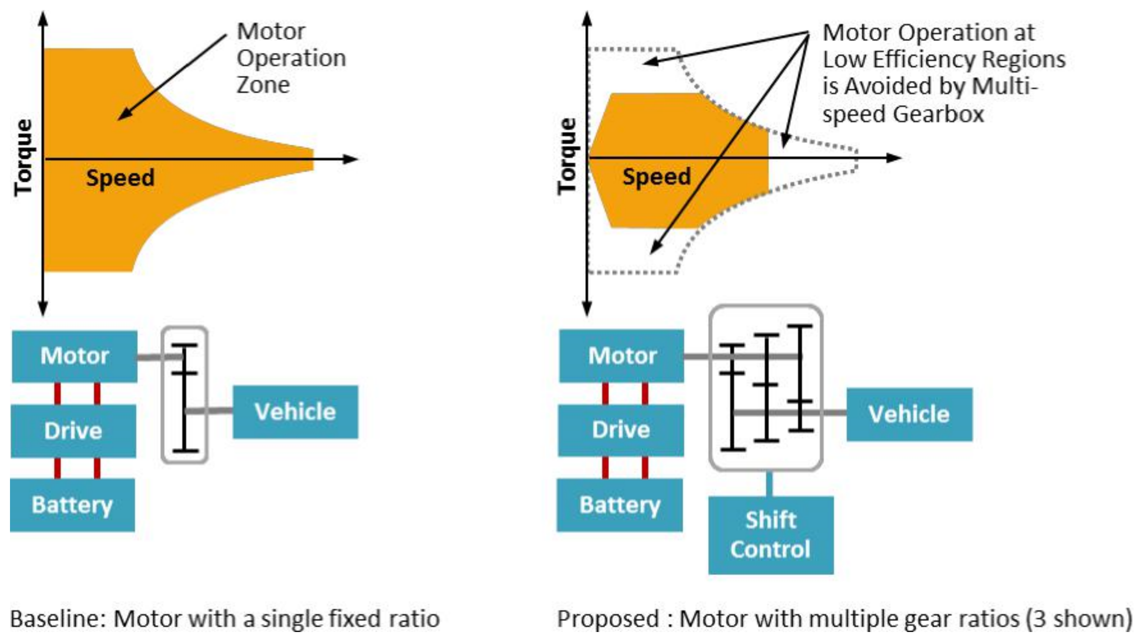



**Figure 1. Baseline and multi-speed EV drivetrain architecture**

The performance improvements targeted—and achieved—by replacing Proterra's current two-speed automated mechanical transmission (AMT) with the new four-speed transmission are listed in **Table 1.**

**Table 1. The baseline and achieved target performance of Proterra BE35 electric bus**

| **Metric** | **Current two-speed AMT (Baseline)** | **New four-speed AMT (Target)** |
|---|---|---|
| Top vehicle speed @ GVW | 53 mph | > 65 mph |
| Vehicle Efficiency @ SLW<br>On -UDDS<br>On Orange County - OCC<br>On Manhattan - NYC<br>On Altoona-ADB | <br>20.4 mpgde<br>19.9 mpgde<br>17.7 mpgde<br>20.2 mpgde | <br>24.8 mpgde<br>23.9 mpgde<br>21.2 mpgde<br>25.0 mpgde |
| Acceleration time @ SLW<br>0 to 30 mph<br>30 to 50 mph | <br>15.5 s<br>27.5 s | <br>< 13 s<br>< 19 s |
| Gradeability @ GVW<br>10mph<br>20 mph | <br>15%<br>7% | <br>>20%<br>>10% |

To achieve these goals, the new transmission incorporated the latest technologies developed by Eaton. Eaton has been advancing vehicle transmission technology by increasing gearbox reliability, automating the shift sequences for ease of operation and precise shifting, developing supervisory control systems to optimize the shift points based on the vehicle duty cycle information for higher efficiency, and developing advanced automated shifting algorithms to reduce torque.

Eaton performed a cost analysis and studied the price targets. The transmission cost structure has two main categories: mechanical hardware and controls. Mechanical hardware costs of a four-speed medium-duty (MD) -EV transmission are less than those of a six-speed MD ICDV-transmission (the cost baseline) because it has two fewer forward gearsets, no reverse gearset, no clutch, and the new shift mechanism is built into the rear transmission housing. However, the controls for a four-speed MD-EV transmission require a new transmission control unit (TCU) and a more sophisticated software for optimization of the traction and regeneration control algorithms, resulting in higher cost controls. Hence, the cost of a MD-EV transmission will be equal to or less than the baseline MD-ICDV transmission.

The price of the MD-EV transmission depends on the market demand. We expect that, in a mature EV market, EV transmission prices will be highly affordable and the payback period will be measured in months rather than in years.

### b. Project Execution

Eaton assembled an excellent team to develop a multi-speed transmission for MD PEDVs. The team included the leading transmission developer for MD and heavy-duty (HD) vehicles (Eaton), the leader in MD PEDVs (Smith Electric) in BP1, the leader in electric transit busses (Proterra) in BP2 and BP3, and critical testing laboratories ORNL and National Renewable Energy Laboratory (NREL).

Eaton designed the transmission hardware and controller, integrated the transmission with the electric motor and inverter, and conducted initial shake down tests. Proterra provided vehicle integration of the powertrain and driveline into 17,000 kg Electric Bus. ORNL performed vehicle level simulations, powertrain loop integration, component testing, and hardware-in-the-loop (HIL) testing. NREL conducted duty cycle analysis, and vehicle coast down and chassis dynamometer (dyno) testing.
The project was conducted in three BPs:

- In BP1: Technology Development, we used high-level vehicle powertrain models to optimize candidate transmission architectures and ratios along with a variety of traction motor characteristics for concept selection. The detailed driveline designs and component dynamics were investigated to meet medium-duty EV requirements.
- In BP2: Technology Development and Prototype Demonstration, we extended the modeling and simulations with multi-speed transmissions to other MD and HD EV platforms. We completed clean sheet design of a compact, lightweight, flexible, and modular, 3 and four-speed transmission. We started development of novel shifting and controls strategies and began procurement of the prototype transmission and the controller hardware.
- In BP3: Technology Integration, Testing, and Demonstration, we completed prototyping the four-speed automated mechanical transmission. The transmission controls system and software development and preliminary gearbox dyno tests were done at Eaton. ORNL conducted integrated powertrain HIL tests. One of the prototype units was fully integrated into a Proterra BE35 demonstration electric bus. We fine-tuned the shift control strategy on the integrated vehicle at Eaton Marshall Proving Grounds. NREL tested the vehicle and validated the performance gains.

Proterra was not included in the initial proposal but joined the team after the first BP when Smith Electric withdrew. At the end of BP1, Eaton decided to rescope the project: instead of working on a breadboard transmission that is an existing six-speed transmission that would be used as a four-speed we decided to do a clean sheet design of a four-speed transmission that is half the weight of breadboard transmission and also significantly smaller in size. Smith Electric could not provide the extra support needed for the change of scope and decided to leave the project.

Smith's departure did not cause any delay in project schedule. In between the partners, the project team worked on the tasks that could be done without the EV original equipment manufacturer (OEM).

Proterra's eagerness to join the team and openness in sharing information helped accelerate the project significantly. While we did need to repeat some of the tasks completed with Smith in BP1, one of the tasks of BP2 was extending the modeling and simulations to other vehicle platforms. So, some tasks would have been repeated, regardless of changes in the team, to gain more knowledge about powertrain needs of EVs in MD and HD application spaces. The rescoped project delivered a clean sheet design and prototypes of a four-speed transmission that went beyond the original deliverables. Furthermore, the new Proterra vehicle platform demonstrated the performance gains by a multi-speed transmission because Proterra was able and willing to change the final drive axle per Eaton's recommendation.

### c. Prototyping

The team built a total of seven prototype transmissions: six units were fully functional, and one unit was a display unit. Figure 2 shows the prototyping of the four-speed transmission in progress.

**Figure 2. Left: Components of four-speed AMT are laid out before the assembly. Right: Eaton four-speed EV-AMT display unit integrated with a UQM Electric Motor.**

### d. Powertrain HIL Testing at ORNL

Figure 3 shows the integration of powertrain consisting of Eaton four-speed EV AMT and UQM Electric Motor on the HIL test setup at ORNL. Figure 4 shows the efficiency gains as compared to the baseline two-speed transmission. Four-speed transmission provides up to 15% efficiency improvement over two-speed baseline with the same final drive ratio (FDR) of 6.2. FDR was the same since the four-speed transmission is operated in two-speed mode to make the comparisons. In the real-life case of the Proterra BE35 Electric Bus, the FDR is 9.8 with two-speed transmission. The small and efficient FDR of 6.2 was enabled by the new four-speed transmission. Final drive with 6.2 ratio is 3% more efficient than the final drive with 9.8 ratio. An efficiency gain of 3% in FDR translates to an additional 6 % efficiency improvement in the drive cycles; the doubling effect comes from the sum of gains in the traction and regeneration modes. Hence, the efficiency gains with respect to the real baseline with two-speed transmission and FDR 9.8 exceeds 20%.

Four-speed transmission can be launched in either first or second gear on most grades. Launching on first gear provides better efficiency gains than launching on second gear, as shown in Figure 4. acceleration from 0 to 50 mph improved 30% with four-speed transmission on HIL as compared to the published test data of baseline EV with two-speed transmission on the Altoona test track [2]. Furthermore, gradeability doubled from 13 mph to 26 mph at 10% grade, with four-speed transmission on HIL as compared to the baseline EV with two-speed on Altoona test track.


**Figure 3. Eaton four-speed EV AMT and UQM Electric Motor on the HIL test setup at ORNL**

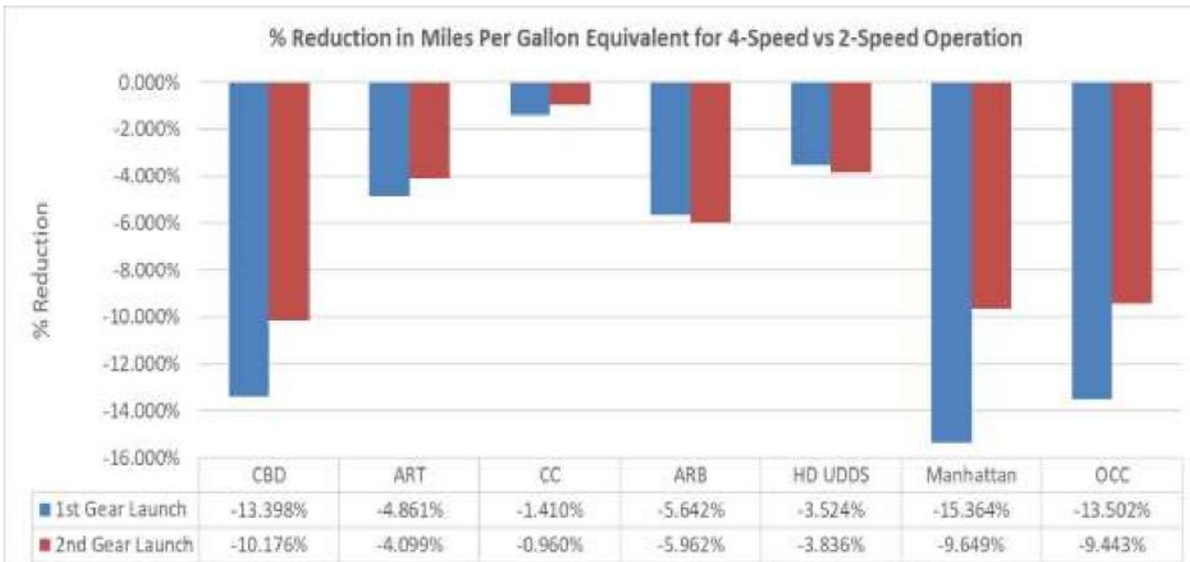



**Figure 4. Percent reduction of mpgde between four-speed and two-speed configurations. Ultimately a 1st gear selection for launch gear shows the largest benefit to energy consumptions.**

**Table 2. Transmission efficiency test results**

| | % Efficiency-Unit#2 | % Efficiency-Unit#3 | % Efficiency-Unit#4 | Average Efficiency (%) |
|---|---|---|---|---|
| 1st gear | 95 | 96 | 94 | 95 |
| 2nd gear | 96 | 97 | 96 | 96 |
| 3rd gear | 97 | 98 | 95 | 96 |
| 4th gear | 99 | 100 | 97 | 99 |
| Overall | 97 | 98 | 96 | 97 |

### e. Chassis Testing at NREL

We tested the Proterra BE35 Electric Bus integrated with four-speed transmission and FDR of 6.2 on NREL's heavy-duty chassis dynamometer as shown in Figure 5**.** Five drive cycles were selected that vary in distance, acceleration rates, number of stops and driving speeds as listed in Table 3. We also performed acceleration rate and gradeability tests for two different payloads.


**Figure 5. Vehicle mounted on NREL's heavy-duty chassis dyno with all data acquisition implemented and ready for testing**

**Table 3. Chassis dynamometer selected drive cycle parameters**

| Cycle | Time (min) | Distance (mile) | Max Sp. (mph) | Avg Sp. (mph) | Avg Driving Sp. (mph) | KI (1/mile) | Stops (#) |
|---|---|---|---|---|---|---|---|
| Manhattan Bus Cycle | 18 | 2.1 | 25 | 7 | 11 | 9.1 | 20 |
| Orange County Bus | 32 | 6.5 | 41 | 12 | 16 | 3.6 | 31 |
| Urban Dynamometer Driving Schedl (UDDS) | 18 | 5.6 | 58 | 19 | 28 | 0.6 | 14 |
| Altoona - Modified Business-Arterial-Commuter (ADB) | 42 | 13.1 | 40 | 19 | 22 | 1.3 | 51 |
| World Harmonized Drive Cycle (WHDC) | 30 | 11.2 | 55 | 22 | 26 | 0.4 | 12 |

The efficiency improvements between four-speed transmission and one-speed and two-speed baseline configurations at each drive cycle are shown in Figure 6. The four-speed transmission was run in one-speed mode in 3rd gear and two-speed mode in first and third gears to emulate the baselines.

We measured up to 17% and 13% efficiency gains with four-speed transmission as compared to one-speed and two-speed baselines depending on the drive cycles. As in the case of HIL testing, the chassis dyno test results do not even include the 6% additional efficiency gains coming from the FDR change from 9.8 to 6.2. Chassis dyno tests verify the earlier predictions made in modeling and simulations as listed in Table 3 that the four-speed transmission provides up to 20% improvement in energy efficiency as compared to the real baseline vehicle depending on the drive cycle.

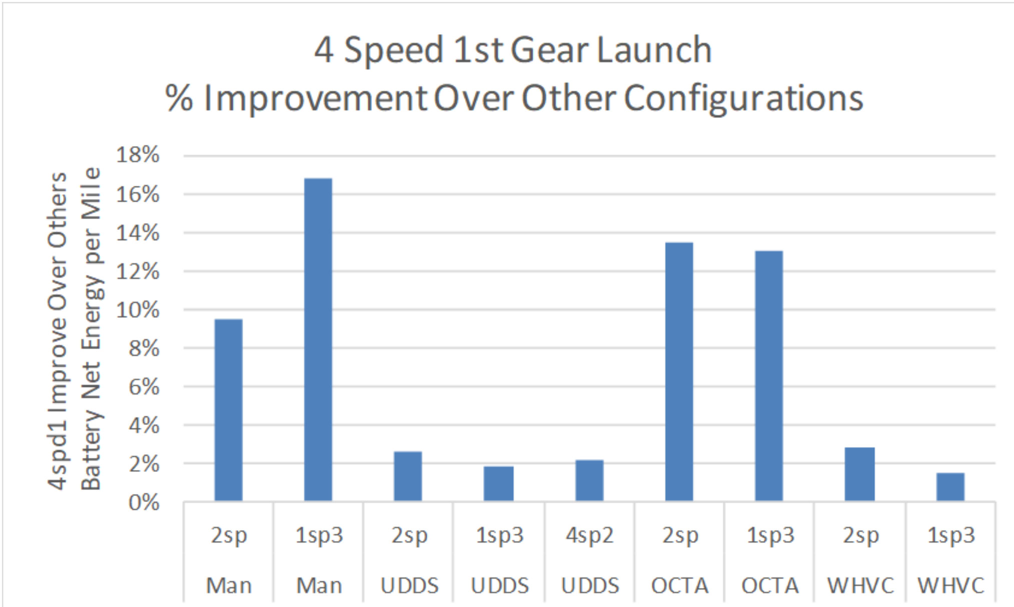



**Figure 6. Efficiency improvement verification test results on NREL's HD chassis dyno. This chart does not include the efficiency gains due to the FDR change from 9.8 to 6.2.**

Table 4 lists the performance of the Proterra BE35 Electric Bus with four-speed transmission and FDR 6.2 powertrain configuration as tested in HIL tests at ORNL and in chassis dyno tests at NREL.

**Table 4. Performance of Proterra BE35 Electric Bus with four-speed transmission and FDR 6.2. Energy consumption and acceleration are tested at SLW (33500 lb). Gradeability is tested at GVW (37530 lb).**

| | Energy consumed (kWh/mile) | | | | Acceleration Time (s) | | Gradeability (%) | |
|---|---|---|---|---|---|---|---|---|
| | UDDS | OCC | NYC | ADB | 0-30 mph | 30-50 mph | 10 mph | 20 mph |
| HIL tests | 1.44 | 1.37 | 1.52 | 1.49 | 12.2 | 17.9 | 22.7 | 11.2 |
| Chassis Dyno | 1.43 | 1.60 | 2.06 | 1.76 | 13.7 | 18.9 | N/A | 11.1 |

## 1. BASELINE VEHICLE MODLE

In Task 4, Eaton and ORNL conducted vehicle modeling activities separately based on the baseline vehicle information. Eaton's MATLAB-Simulink® based vehicle model was aligned with ORNL's vehicle model that was based on the Autonomie® modeling tool. The simulation results from two separate models were comparable, and the deviations were within a few percent.

**Deliverables:** Baseline Vehicle Model Development Report

## 1.1 SUBTASK 1.1 – CREATE VEHICLE LEVEL MODEL

The ORNL team used the Autonomie® vehicle system modeling tool to create the vehicle model architecture for a battery electric MD truck (shown in Figure 7). The battery consists of the following components:

- Energy storage system
- Electric machine and inverter
- Single speed transmission
- Final drive
- Wheels
- Chassis
- DC-DC converter
- Electric accessories loads


**Figure 7. Autonomie representation of battery electric medium-duty truck architecture**

## 1.2 SUBTASK 1.2 – POPULATE COMPONENT MODELS

Each component model was parameterized based on data provided by Smith Electric.

<u>Energy storage system:</u>

- Battery nominal voltage was set to 320 volts (v) based on experimental data, even though battery specifications called for 343V.
- Battery energy capacity was set to 80 kilowatt hours (kWh).
- Internal resistance was calibrated to match experimental data.

<u>Electric machine and inverter:</u>

- The machine's peak power is 150 kilowatt (kW), and its continuous power is 80kW.
- Peak torque is 600 Newton meter (Nm), and continuous torque is 400Nm.
- The combined efficiency map used for the motor and inverter was extracted from the UQM PowerPhase PP145 machine (similar electric motor to the Smith Electric), as there was no detailed experimental data to characterize the efficiency of the Smith Electric machine (see Figure 8 for torque and efficiency map used in the model).
- Motor rotational inertia was estimated to be $0.1 kgm^2$.

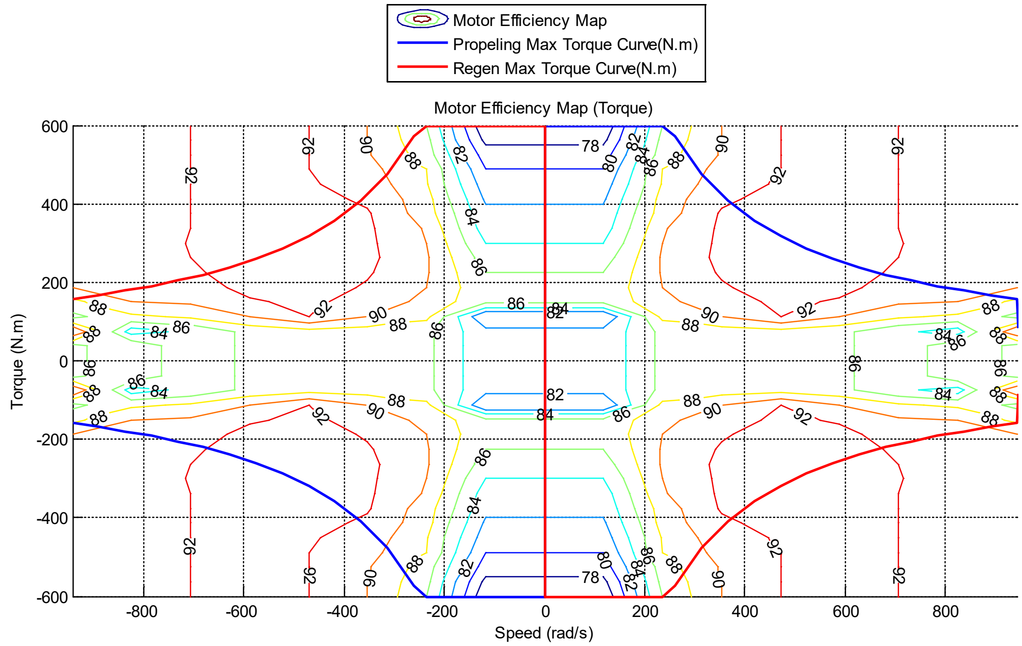



**Figure 8. Electric machine and inverter efficiency map and maximum torque curve**

<u>Single speed transmission:</u>

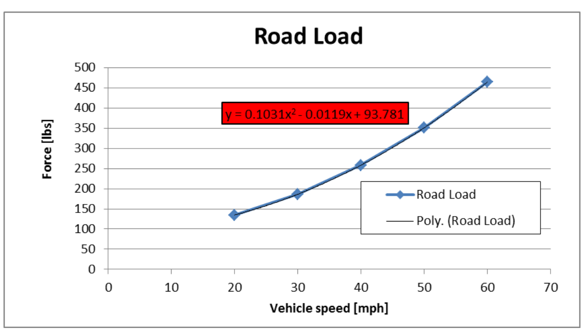
- Ratio of 3.41
- Eaton suggested an efficiency that is speed dependent: from 98% at low speeds down to 86% at 10,000 rounds per minute (rpm)

<u>Final drive:</u>

- Ratio of 4.1
- Eaton suggested an efficiency of 95% for all speed and loads

<u>Wheels:</u>

- Smith Electric provided coast down test data to calculate road load coefficients and rolling resistance as well as aerodynamic drag, shown in Figure 9.
- Rolling resistance: 0.00711
- Tire rolling radius:0.391 m
- Wheel inertia: 1.289 kgm$^2$




| | Formula | Calculated value |
|---|---|---|
| Rolling resistance | A = RollRes*M*g | 0.00711 |
| | | |
| Speed factor | B= RollRes(v)*M*g | 0 |
| | | |
| Aero drag (Cd*A) | C = 0.5*air density*Cd*A | 3.775 |
| Discharge Coefficient (Cd) | | 0.565 |
| Front Area (A) | A=Aero drag/Cd | 6.681 |

**Figure 9. Road load coefficients calculations**

Chassis:

- Based on coast down tests described above, vehicle frontal area was estimated to be 6.68 $m^2$. The discharge coefficient of 0.565 was provided by Smith Electric.
- Empty mass is 4135kg.
- GVW is 9990kg.

DC-DC converter:

- Efficiency of 90% was provided by Smith Electric.

Electric accessories loads:

- Estimated to be 1000 watts (W) based on experimental data.

## 1.3 SUBTASK 1.3 – BASELINE MODEL DEVELOPMENT

Component models and parameters described in Subtask 1.2 were integrated in the vehicle architecture described in Subtask 1.1.

A generic Autonomie Vehicle Supervisory Controller was also implemented and calibrated to coordinate the vehicle electric propulsion drive. Specific features were added such as current limitation set to 240 amps (A) for traction purposes and power limitation during regeneration set to 50kW.

The baseline vehicle also includes a generic Autonomie driver model capable of following speed profiles. Its speed controller parameters were tuned for the MD truck.

Proper operation of the vehicle was verified by operating the vehicle on standard drive cycles such as a Hybrid Truck Users Forum Parcel Delivery Class 4 (HTUF4) and Orange County Bus Cycle (OCBC) cycles to make sure that it can follow those speed profiles.

## 1.4 SUBTASK 1.4 – VALIDATING BASELINE MODEL

Smith Electric provided experimental test data to validate the baseline model described above:

- On-route data was not used because grade was not available, which rendered model matching impossible.
- HTUF4, OCBC and 55mph Cruise cycles were performed on a chassis rolls dynamometer at the University of California, Riverside in 2013. Unfortunately, this data seems questionable because road load coefficients described in the study do not match experimental data collected in that same study. For a cruise speed of 54mph, mechanical power at the wheel is supposed to be 43 kW (based on road load coefficients) but electrical power into the motor at that speed was measured as 40 kW. So, the road load coefficients used for these chassis dynamometer tests might have been much lower than reported; that data was not used to validate the model.
- Acceleration tests performed for several vehicle masses of 5, 7.5, and 10 metric tons were also provided. That data yielded a good match with the model while re-using component specifications and coast down parameters provided by Smith Electric. This data was used as the main validation source for the ORNL model. Figure 10 illustrates an example of model correlation for a wide-open acceleration test performed on a 5000kg vehicle

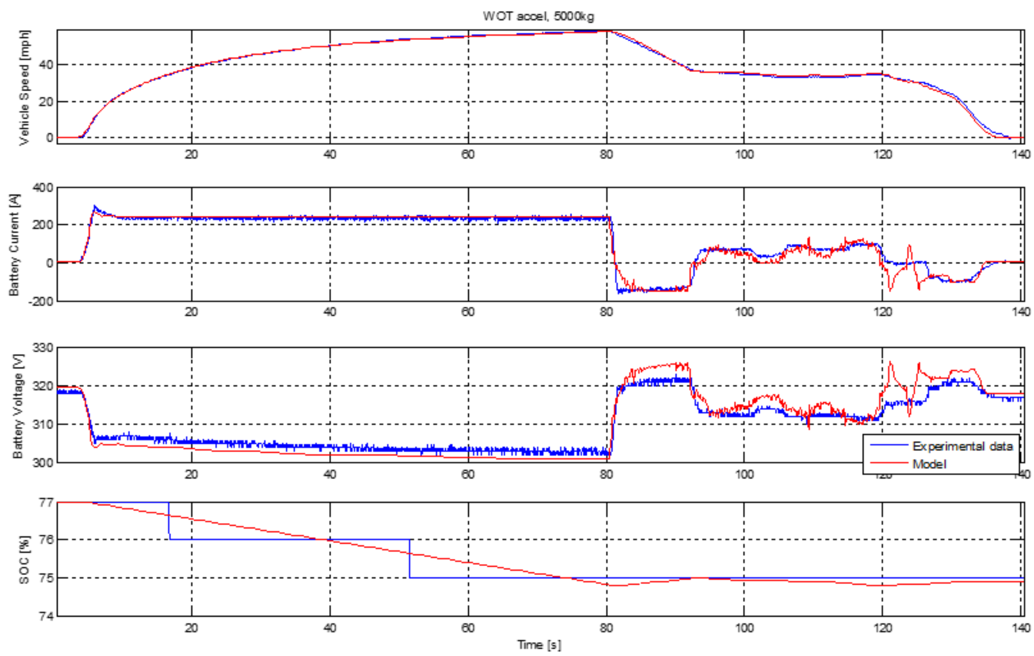



**Figure 10. Baseline model correlation with experimental data. Wide open acceleration test performed on a 5000kg vehicle**

This validated model was used to establish a benchmark for the single speed electric vehicle by exercising it on two drive cycles listed in the FOA:

- Urban Dynamometer Driving Schedule (UDDS truck cycle) (Figure 11)
- Combined International Local and Commuter Cycle (CILCC) (Figure 12)

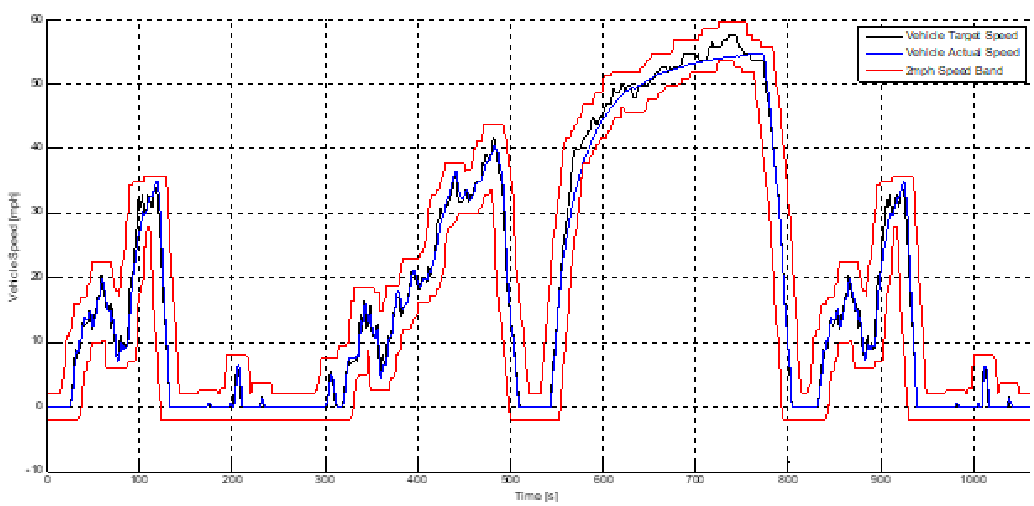



**Figure 11. Baseline model on UDDS truck Cycle**

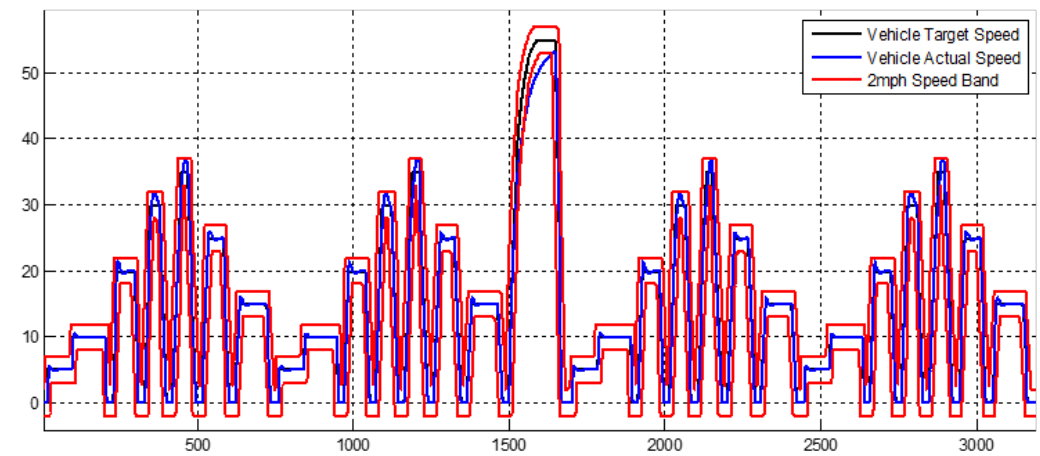



**Figure 12. Baseline model on CILC Cycle**

The baseline model does not match vehicle energy consumption numbers listed in the FOA, as shown in Table 5. The mismatch (10-13%) occurred because vehicle efficiency numbers quoted in the FOA (37 mpgde for UDDS and 50 for CILCC) are based on a high-level simulation study, not on experimental data. They were obtained with vehicle parameters listed in Table 6, which differ from the parameters provided by Smith Electric for this study. For instance, the aerodynamic drag in our baseline model (which is based on actual coast down results) is 41% higher than the FOA parameters. When we used FAO aerodynamic drag in our baseline model, the match improves significantly: down to 1 to 5% error.

**Table 5. FOA model vs Baseline model vehicle energy comparison**

| | FOA model [Wh/mile] | Baseline model [Wh/mile] |
|---|---|---|
| UDDS Truck cycle (10 t) | 1097 | 1242 |
| CILC cycle (10 t) | 816 | 899 |

**Table 6. Characteristics of the baseline MD PEDV described in the FOA**

| Drag , Mass, and Accessory Load Parameters | Traction Motor | Energy Storage | DC-DC Converter | Transmission | Differential & Wheels | Top Speed | Fuel Efficiency |
|---|---|---|---|---|---|---|---|
| Drag coefficient 0.5, Area = 5.33 m2. Weight=10000 kg., Accessory Load = .2 KW | Permanent Magnet, 134 KW peak, 95 KW continuous power | 360 V, 80KWhr | 95% efficiency | Single Speed 2.0 ratio, 93.4% efficient | 3.4 ratio, 97% efficient | 50 mph | 37 MPGdge on UDDS, 50 MPGdge on CILCC |

Therefore, the FOA energy consumption numbers should be superseded with our validated model numbers (listed in Table 7) to quantify the benefit of a multi-speed transmission because the baseline model has been validated against experimental data.

**Table 7. Benchmark energy efficiency for single speed vehicle**

| | Single speed vehicle efficiency [Wh/mile] | Single speed vehicle efficiency [mpgde] |
|---|---|---|
| UDDS Truck cycle (10 t) | 1242 | 30.3 |
| CILC cycle (10 t) | 899 | 41.8 |

### Baseline model update

The baseline model was slightly modified to allow a better comparison with the multi-speed transmission vehicle used for the optimization study:

- A distance compensation feature was added such that the driver covers the same distance regardless of the vehicle configuration. For instance, a more powerful vehicle will better keep up with the speed trace and cover more distance than a sluggish vehicle. It then makes comparing their energy efficiency quantified in Wh per mile more difficult as the distance varies from one vehicle configuration to another. For example, the base vehicle, without the distance compensation feature, covers 0.1 mile (or 2%) less distance than the intended UDDS drive cycle; with it, it matches the cycles within a few yards.
- Driver gains were tuned such that the same gains are used for all driveline configurations during the multi-speed gearbox optimization study and the same driver settings are used on the baseline vehicle, too.
- Vehicle brake size was increased as it was borderline during some HTUF4 cycle decelerations

Table 8 lists the energy consumption of the modified baseline single-speed transmission vehicle.

**Table 8. Energy efficiency for single speed vehicle**

| | Single speed vehicle efficiency [Wh/mile] | Single speed vehicle efficiency [mpgde] |
|---|---|---|
| UDDS truck cycle (10T) | 1268 | 29.6 |
| CILC cycle (10T) | 901 | 41.7 |
| Smith representative box truck cycle (10T) | 1197 | 31.4 |
| Smith representative step van cycle (10T) | 1220 | 30.8 |
| Smith overall representative cycle (10T) | 1180 | 31.9 |
| NREL cycle #10000 | 1077 | 34.9 |
| NREL cycle #10028 | 1083 | 34.7 |

(Note that the mpgde does not include on-board charger efficiency)

### Multi-speed transmission model creation

The model for the multi-speed transmission vehicle was created from the baseline model and removing the single-speed transmission. Eaton supplied ORNL with s-function models and initialization files for its multi-speed transmission and controller. Eaton component models were integrated in place of the single speed transmission model used in ORNL baseline vehicle model. The multispeed model was configured with the same ratios as the single speed transmission to verify that its behavior matches the baseline model. Results are shown in Figure 13.

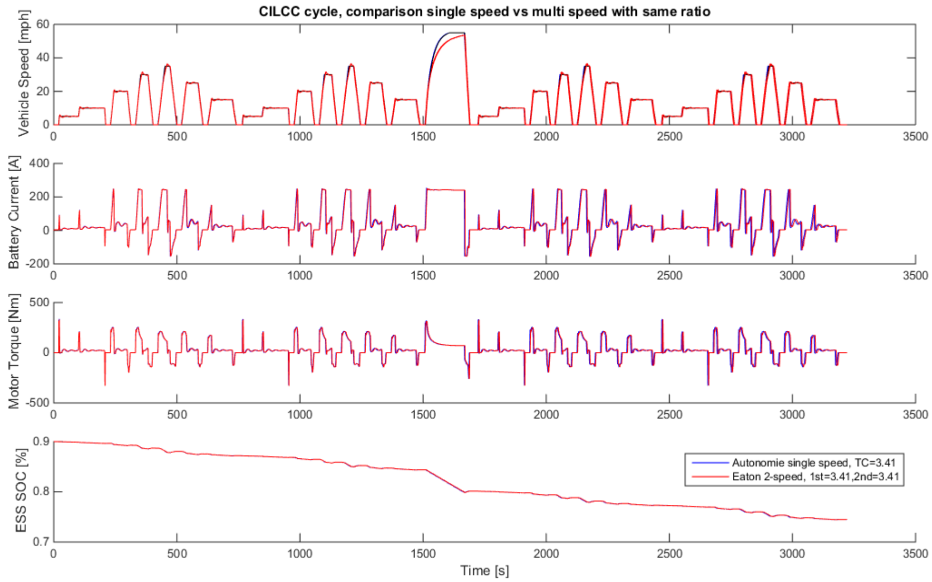



**Figure 13. Comparison of single speed (blue) versus multi-speed (red) transmission on vehicle, motor, and energy storage system (ESS) behavior.**

### Multi-speed transmission optimization

The multi-speed model was exercised to validate the optimization study conducted by Eaton with their vehicle model. The multi-speed transmission has three gears: the first gear is only used on large grades and is not engaged during regular driving, so for all subsequent tests, the vehicle starts in the second gear. The third gear is a direct drive with a 1:1 ratio. Thus, the optimization consists of identifying the best combination of final drive ratio and second gear ratio.

*Acceleration performance and top speed optimization results*

The multi-speed transmission vehicle model was subjected to wide open throttle (WOT) accelerations. Zero to 50 mph acceleration times and the top speed after 200 seconds were recorded for a variety of final drive ratio and the second gear ratio combinations.

Both optimization studies conducted with Eaton and ORNL models exhibit a good match (within 2%) as compared in Figure 14 to Figure 18. Both Eaton and ORNL models point at selecting as high ratios as possible.

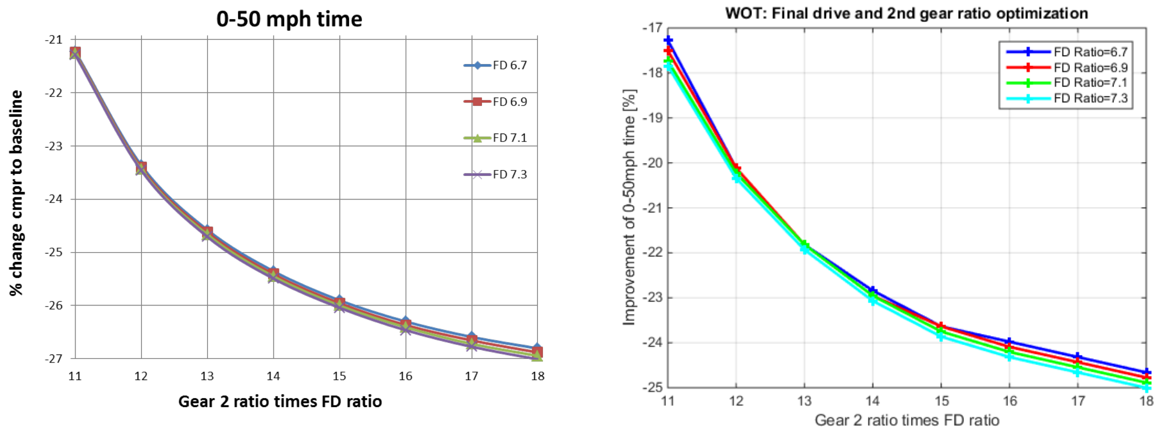



**Figure 14. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for 0-50 mph acceleration.**

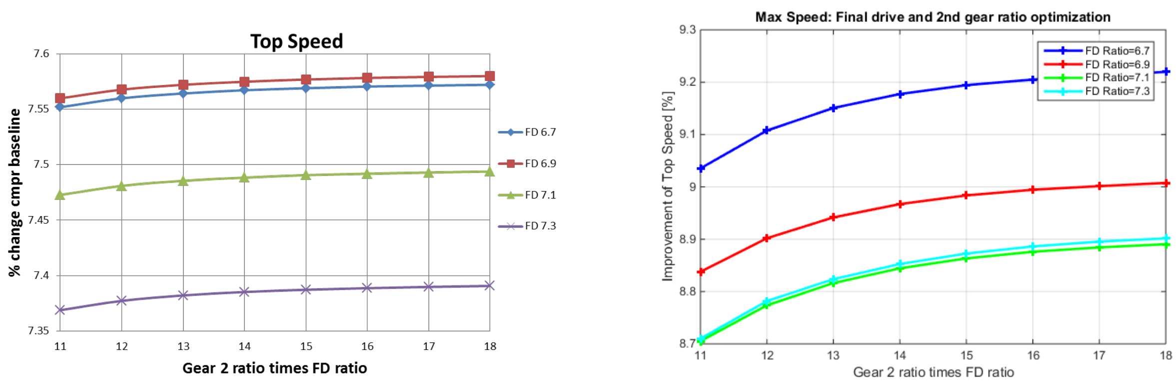



**Figure 15. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for top speed.**

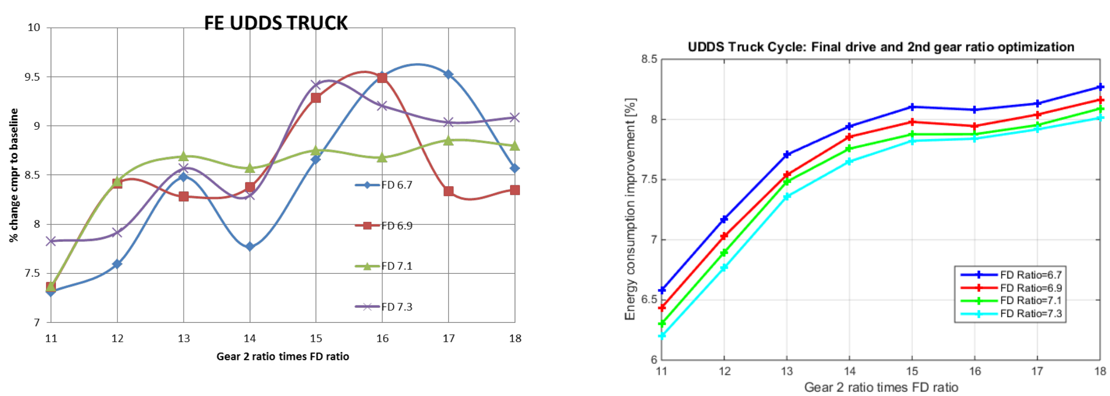



**Figure 16. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for energy efficiency improvement on the UDDS drive cycle.**

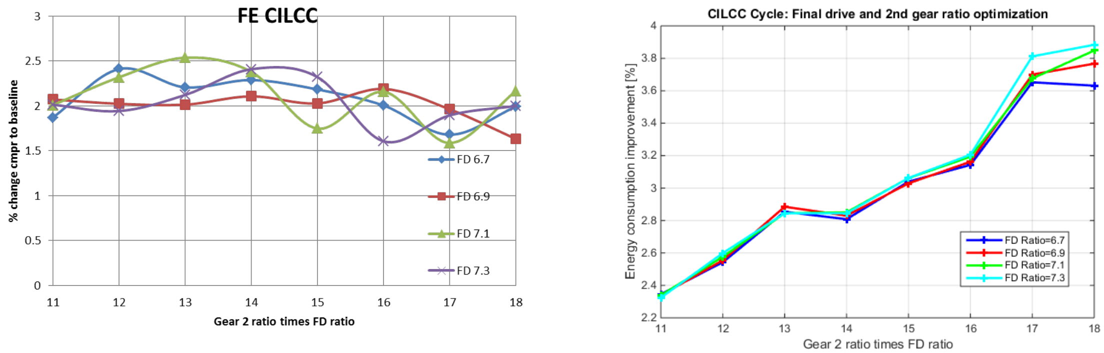



**Figure 17. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2$^{nd}$ gear ratio for energy efficiency improvement on the CILC drive cycle.**

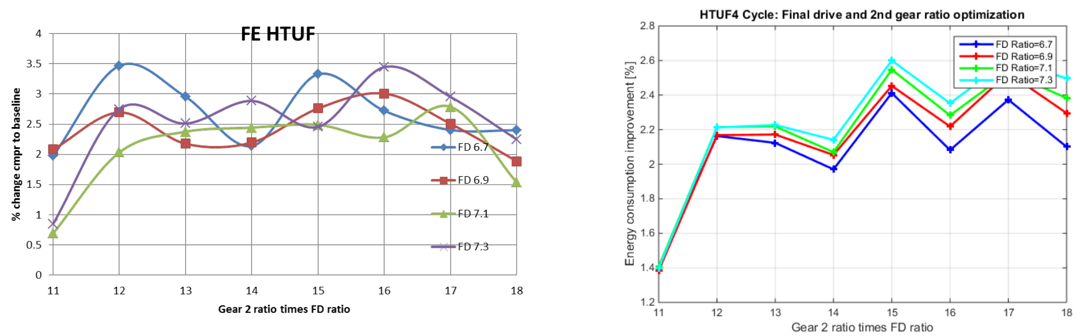



**Figure 18. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2$^{nd}$ gear ratio for energy efficiency improvement on the HTUF4 drive cycle.**

*Energy efficiency optimization results*

The multi-speed transmission vehicle model was subjected to three cycles: UDDS truck cycle, CILCC, and HTUF4. Energy consumption for each cycle was modelled, integrated, and plotted for a variety of final drive ratio and the second gear ratio combinations. The Eaton and ORNL optimization studies showed a good match between where a final drive ratio around 7:1 yields an optimum second gear ratio around 2:1.

*Extended energy efficiency optimization results*

After validating Eaton's optimization study, the ORNL study was extended to a wider range of final drive ratios: from 3:1 to 9:1. The results showed that the lower final drive ratios can result in better energy efficiency. For instance, on a UDDS cycle, a final drive ratio of 5:1 performed better than a final drive ratio of 7:1, as shown in Figure 19**.**

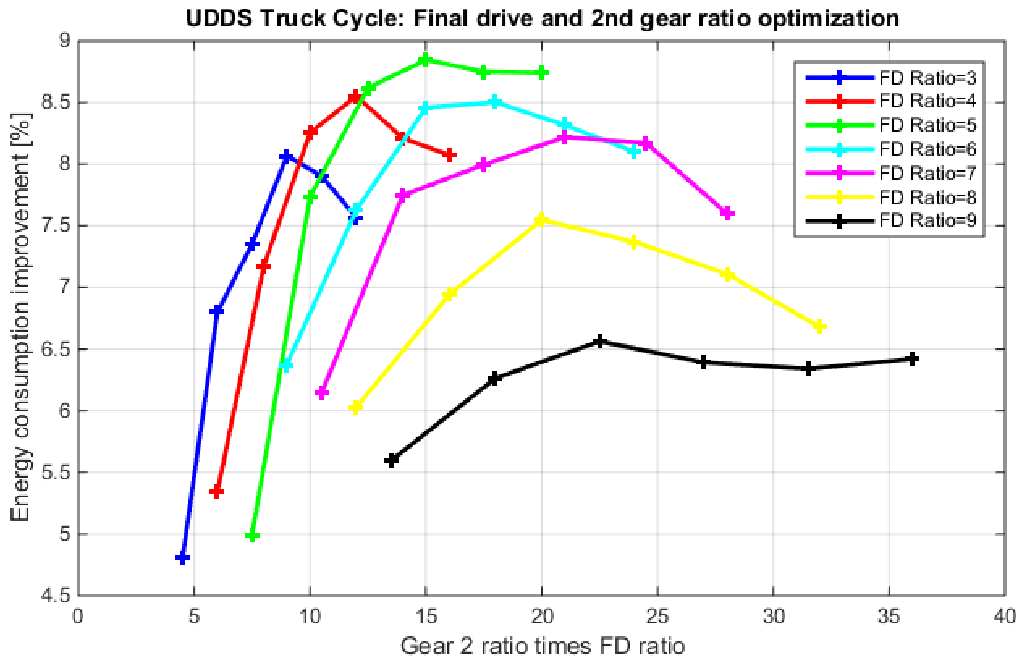



**Figure 19. Extended optimization study results for energy efficiency improvement on the UDDS truck drive cycle.**

Lower speed and more delivery-type cycles like CILCC and HTUF4 did still better, with the higher final drive ratios, as shown in Figure 20.

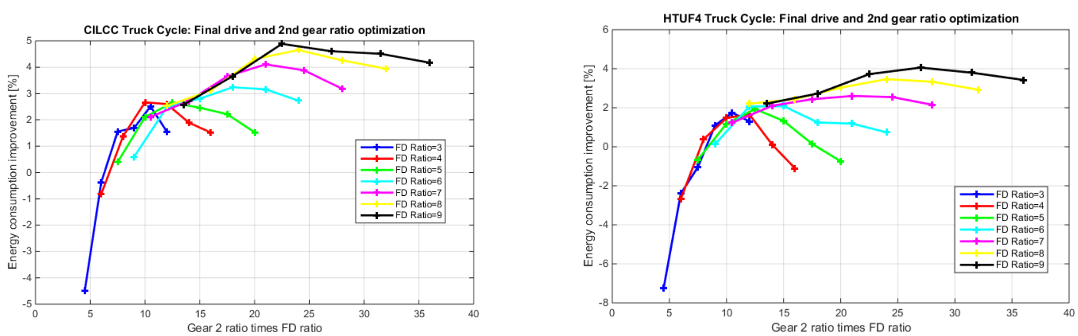



**Figure 20. Expended optimization study results for energy efficiency improvement on the CILC and HTUF4 drive cycles.**

**Effect of battery restrictions on gearbox ratio selection**

Smith Electric enforces a current limitation in its controller such that the electric machine can only be used at about half of its peak power capabilities. This restriction prevents operation in the peak efficiency region of the motor as shown on the left-hand plot of Figure 21.

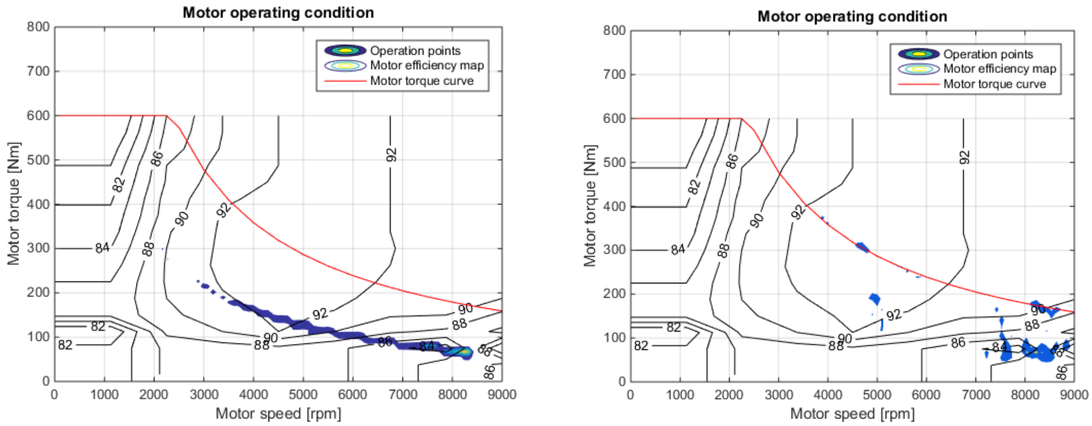



**Figure 21. Motor operating condition on same UDDS truck cycle with battery limitation (left plot) and without limitation (right plot)**

The model was modified to remove that limitation. The optimization study was then re-run to check whether that limitation affects the gearbox ratio selection. Because the baseline vehicle is also improved by removing this limitation, even though the multi-speed transmission vehicle is more efficient without it, the relative improvement is not any better with or without the limitation. The final drive ratio of 7:1 and the second gear ratios of 2:1 or 2.5:1 still yield the best energy efficiency as shown in Figure 22.

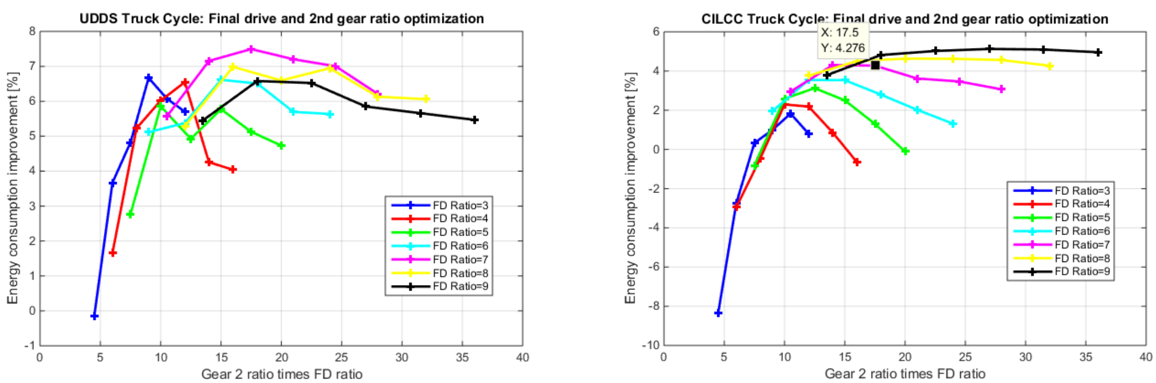



**Figure 22. Final drive ratio and the second gear ratio optimization study conducted on modified model without battery restrictions on UDDS truck and CILC cycles.**

**Effect of multi-speed transmission of motor operating region and energy efficiency**

The multi-speed transmission is especially helpful on cycles that include cruise sections above 40 mph like the NREL cycle #10000 where the motor would spin above 6000 rpm with a single speed transmission, as shown on the left graph of Figure 23 and won't exceed 5500 rpm with a multi-speed transmission as shown on the right graph of Figure 23.

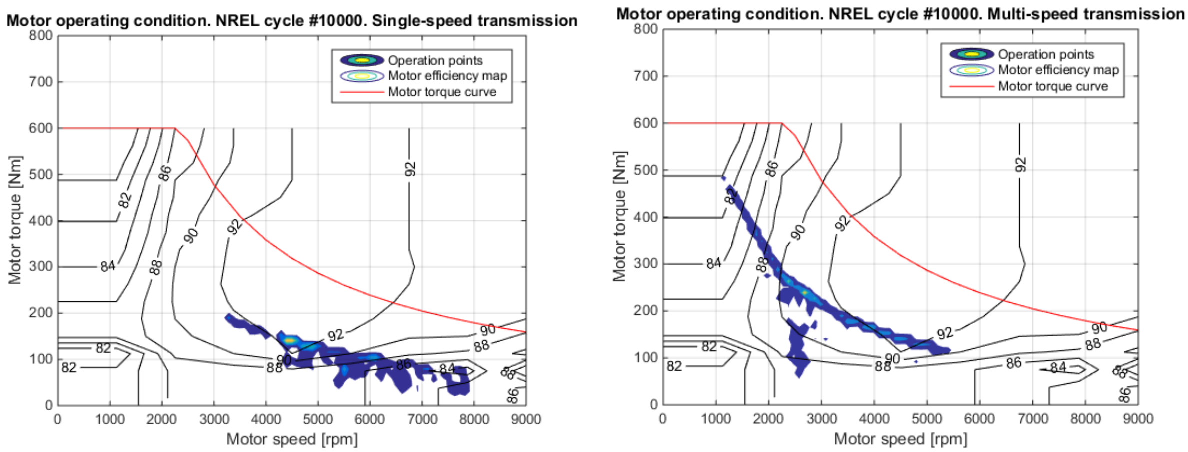



**Figure 23. Motor operating conditions with single speed transmission (Left) and multi-speed transmission (Right) on NREL cycle #10000.**

When the top cycle speed does not exceed 40 mph (like NREL cycle #10028), then the motor speed stays below 6000 rpm with a single speed transmission, as shown on the left side graph of Figure 24. The multi-speed transmission does not provide as much savings, because the motor operates in the same flat efficient part of the motor map, right hand side graph, Figure 24.

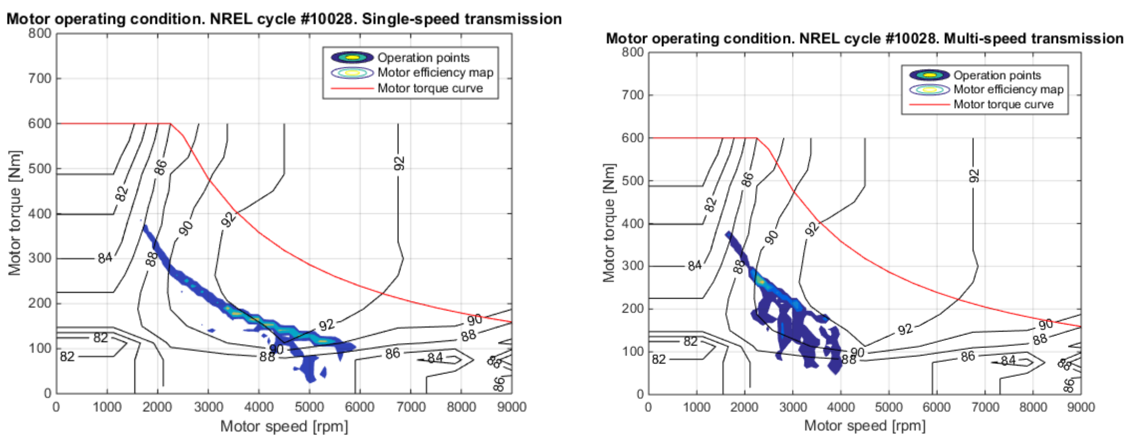



**Figure 24. Motor operating conditions with single speed transmission (Left) and multi-speed transmission (Right) on NREL cycle #10028.**

### MD-EV fleet savings with three-speed transmission

The transmission optimization study mainly used three predetermined cycles (UDDS, CILCC, and HTUF cycles) to identify the best transmission and final drive ratios combination. The simulations were extended to many real-world drive cycles to validate that the original study was representative.
The test vehicle was the Smith Newton™ vehicle, whose model was validated earlier. It runs with a direct drive driveline for the baseline vehicle and a multi-speed transmission for the optimized configuration: final drive ratio of 7:1, three-speed transmission with third gear ratio of 1:1 and second gear ratio of 2:1 The first gear ratio was not used in this study as it engages only on large grades. NREL provided over 800 cycles from their database of real cycles collected on the fleet of Smith Electric vehicles that they had instrumented. Each cycle logs a day's worth of vehicle activities and has at least 10 minutes of driving in that day.

The vehicle speed recorded in each test was used as the speed profile for our validated vehicle models: the baseline vehicle fitted with a single speed gearbox and the three-speed transmission vehicle. Both models were run over the 800 cycles and their respective energy efficiency was calculated. Averaging all cycles, the multi-speed transmission efficiency was 5.4% higher than the baseline vehicle. Figure 25 shows the efficiency gain for each individual cycle:

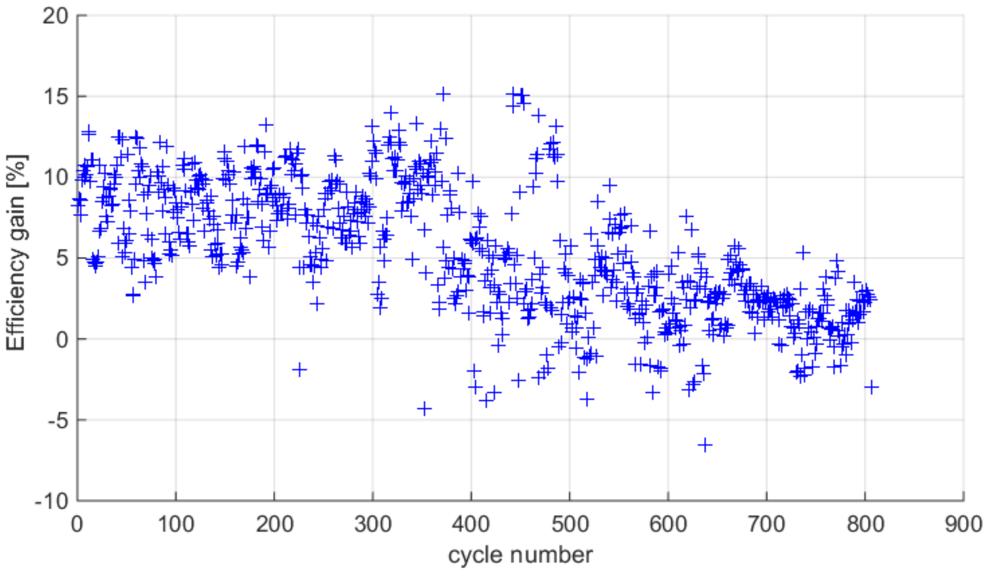



**Figure 25. Efficiency gain obtained on each cycle when comparing a three-speed transmission against a single speed transmission.**

In plotting efficiency gain as a function of average cycle vehicle speed for each cycle, we saw a clear correlation between the real-world fleet data and the standard drive cycles. Figure 26 shows that the higher the speed, the more beneficial the multi-speed transmission becomes. Also, some low-speed cycles do better with a single-speed gearbox, as the efficiency gain is negative. The test cycles used in the original study were run at rather low speed (less than 20mph) and do not show the efficiency gains that can be obtained at higher speeds.

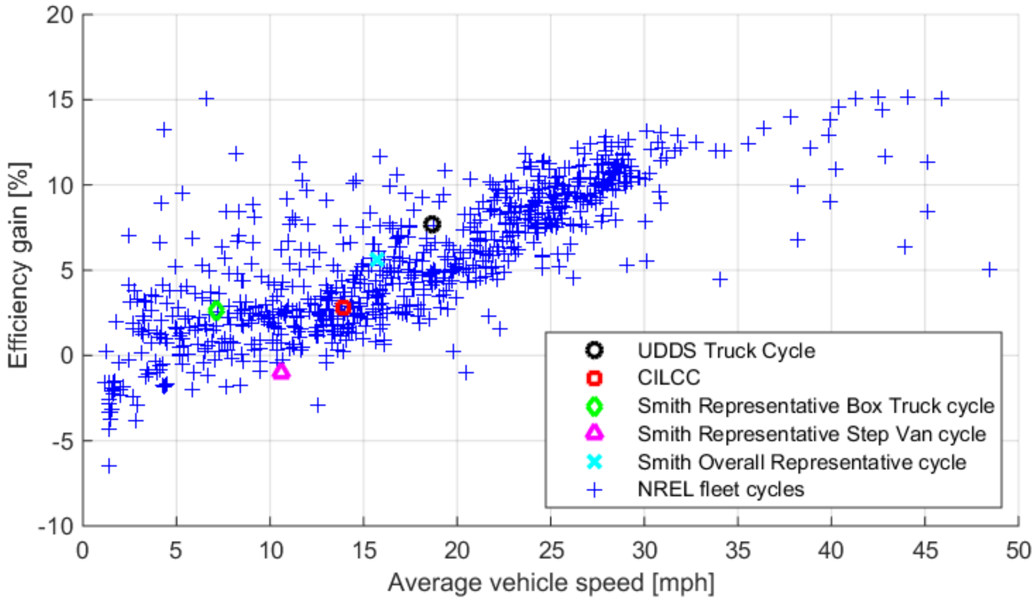



**Figure 26. Efficiency gain as a function of vehicle speed for each real-world cycle and standard drive cycles.**

When removing test cycles with low average speed, the multi-speed transmission efficiency improvement increases to 6.5% above 10 mph and 9% above 20 mph, as shown in Figure 27.

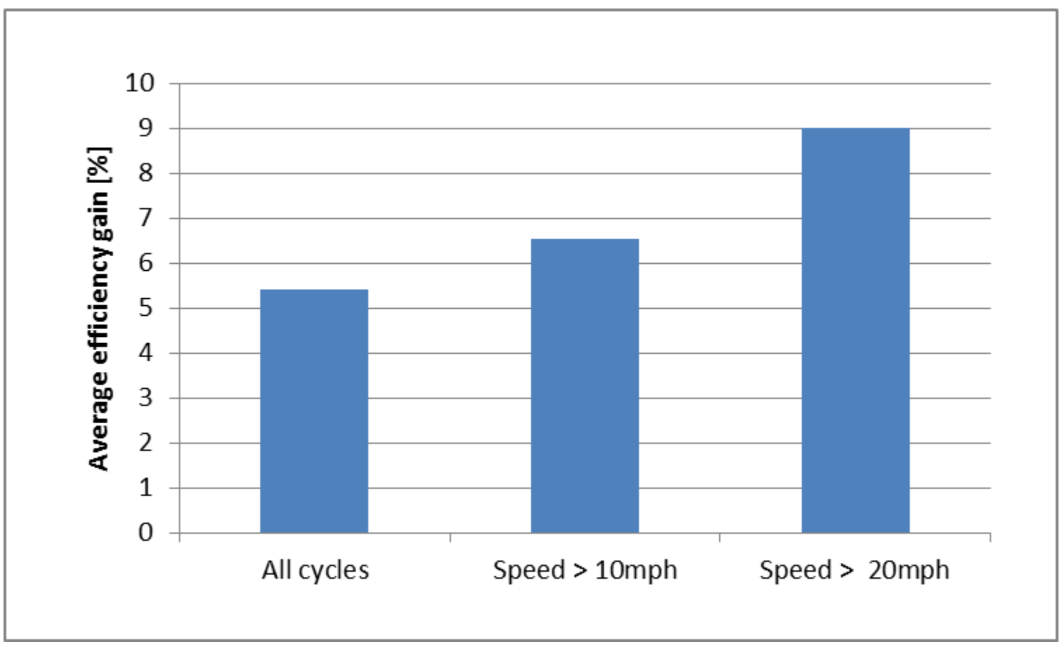



**Figure 27. Average efficiency gain for all cycles, average speed above 10 mph and above 20 mph.**

## 2. MULTI-SPEED GEARBOX MODEL

High-level wants and needs were translated into transmission functional requirements by internal and external project partners, including EV-OEMs, Smith Electric in BP1, Proterra in BP2 and BP3, national laboratories ORNL and NREL, and Eaton as the transmission manufacturer. Transmission concepts were developed to address the functional requirements by a cross-functional team. The leading concepts were subjected to a trade-off analysis based on the requirements. Eaton's AMT family concept with a three- and four-speed layshaft gear architecture was selected as the best concept. The biggest advantages of an AMT layshaft concept are low non-recurring engineering (NRE) cost, low capital costs, high reliability, high efficiency, and better application commonality across the market segments and vehicle classes.

**Deliverables:** Gearbox optimization and selection based on modeling

### 2.1 SUBTASK 2.1 – GEARBOX MODELING DEVELOPMENT

We developed a gearbox model to replace single gear reduction of the baseline model of Smith Newton™ Vehicle. The first gearbox model was based on AMT technology. The gears' efficiency is the same as for the single-speed gearbox used for the baseline, except for the direct drive (1:1 ratio) which is 99.5% efficient. The goal of gearbox optimization is to find the number of gears and ratios in the gearbox along with an appropriate final drive (FD) gear ratio.
We used this analysis method to find the desired ratios:

- The top gear ratio is 1, to achieve the highest transmission efficiency at high vehicle speed representing vehicle highway driving.
- At the top speed, the motor must work at the most efficient point, around 5500 rpm, to increase performance and fuel economy. This gives a rough first estimate of final drive ratio-around 7.
- Given the FD ratio, the lowest gear is chosen to meet the acceleration and gradeability performance.

The shift curve is designed based on the efficiency of the motor. The shift happens at slightly higher rpms of the high efficiency island, when the step size is low enough that the motor rpm after shift is high enough to provide enough power. Otherwise, the shift point happens at higher rpms, calculated to ensure enough motor power availability.

Figure 28 shows initial simulation results of acceleration performance and top speed of Smith Newton™ electric truck generated by the validated vehicle model. The x-axes are the total Gear 1 ratio (Gear 1 ratio multiplied by final drive). The y-axes show the percent change of acceleration performance and top speed with two-speed transmission as compared to the single speed baseline. The 0-30 mph acceleration time for Gear 1 total ratio of 16.5 is slightly worse than that of the baseline with a total ratio of around 14 because it requires half a second torque interrupt for shifting.

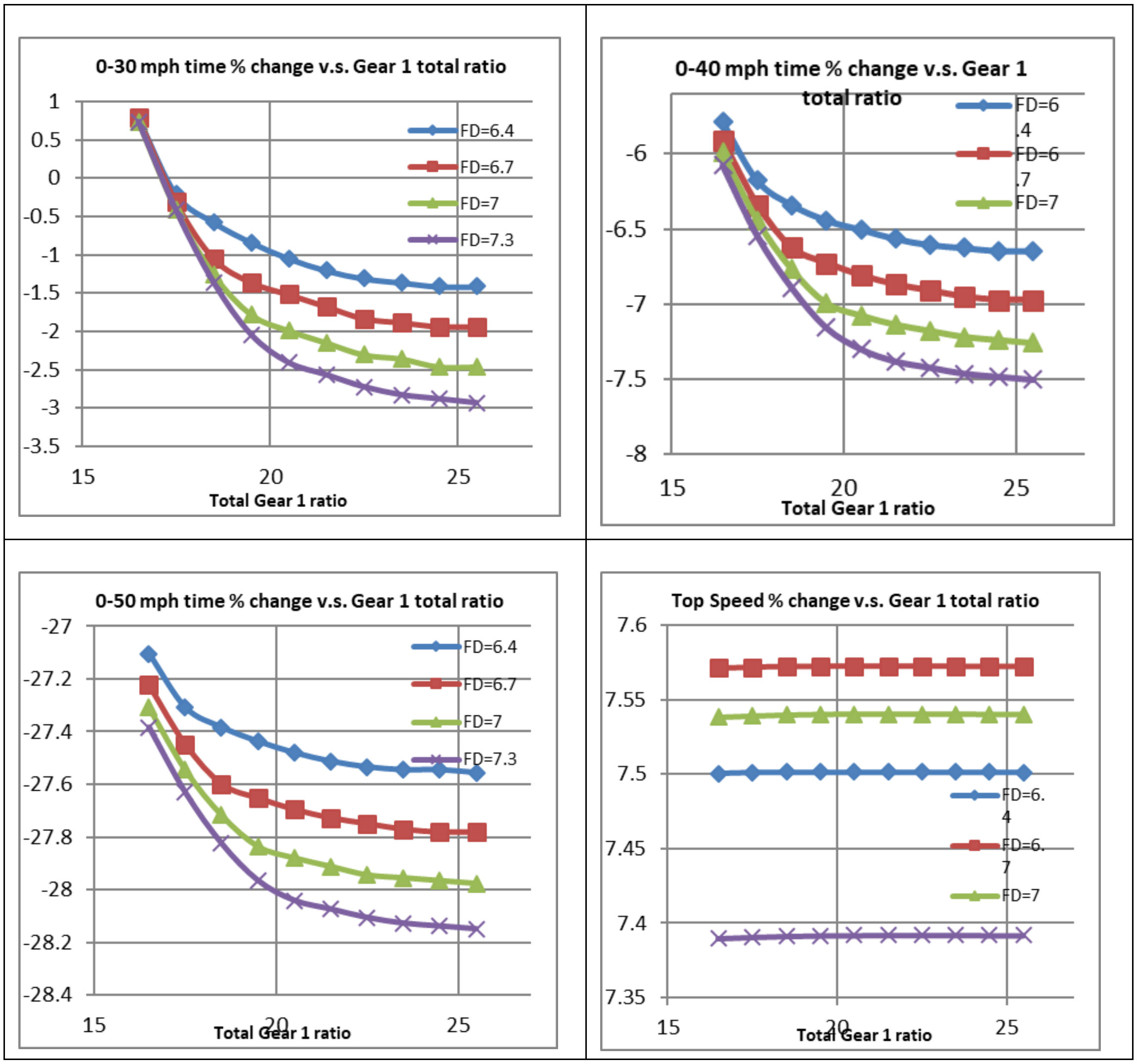



**Figure 28. Acceleration and top speed gains of Smith Newton™ electric truck with two-speed transmission for different FD and Gear 1 ratio values compared to the single-speed baseline.**

Figure 29 shows three drive cycle energy efficiency gains by the Smith Newton™ Electric Truck with two-speed transmission compared to the single-speed baseline. The x-axes are the total Gear 1 ratio (Gear 1 multiplied by final drive). The y-axes show the percent energy efficiency gains of Smith vehicle with two-speed transmission as compared to the baseline vehicle.

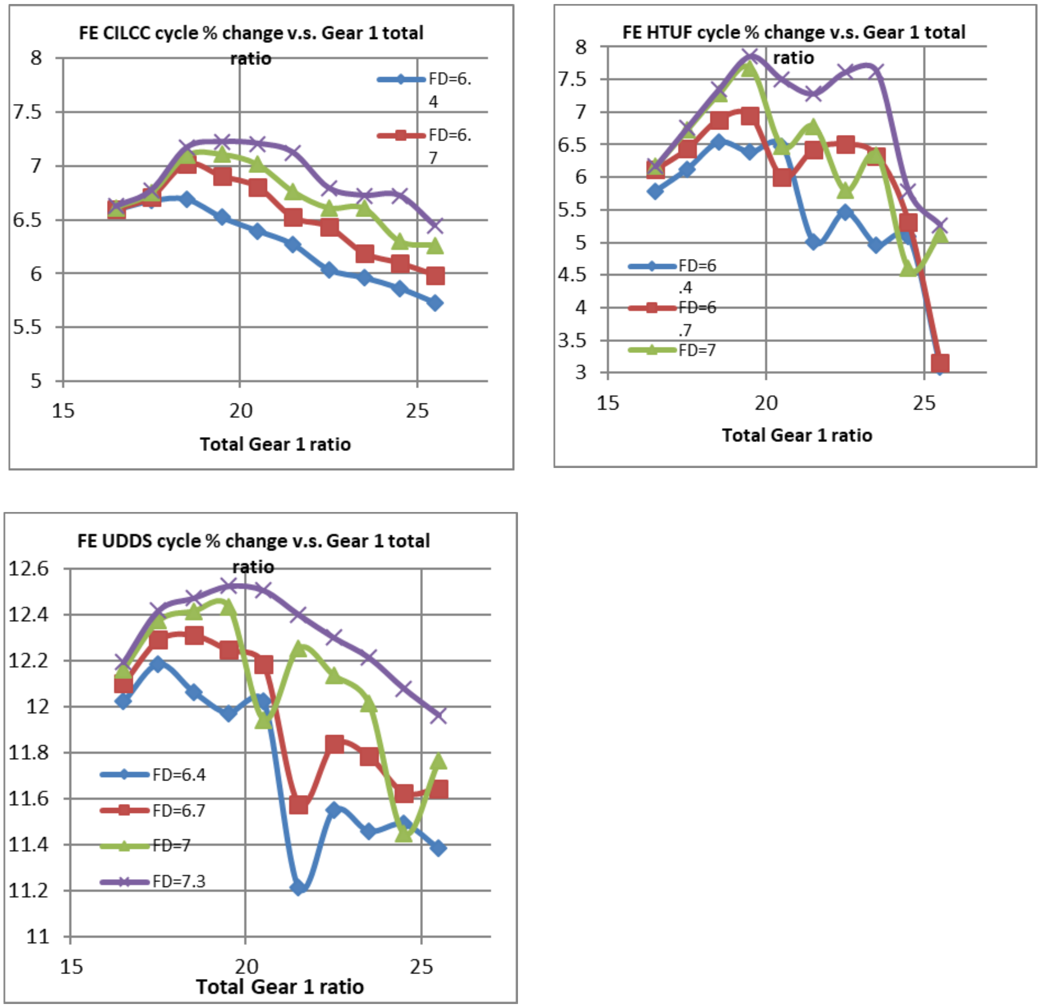



**Figure 29. Energy efficiency gains of Smith Newton™ electric truck with two-speed transmission for different FD and Gear 1 total ratio values compared to the single speed baseline.**

From the initial results, it appears that higher FD ratios and total Gear 1 ratios around 20 gives the best efficiency and performance. Similar studies are reported for three-speed transmission ratios below.

## 2.2 SUBTASK 2.2 - OPTIMIZING THE GEARS, RATIOS, AND SHIFT STRATEGY

We expanded modeling and simulations to study three-speed transmission based on the same assumptions and methodology described above. Gear 1 has a high ratio and is engaged only at grades higher than 15%. In lower grades the vehicle is launched at the second gear. The third gear is direct drive with a ratio of 1.

Figure 30 shows the acceleration performance and top speed improvements with three-speed transmission compared to the single-speed baseline. The x-axes are the total Gear 2 ratio (Gear 2 ratio multiplied by final drive). The y-axes show the percent change of acceleration performance and top speed. The 0-50 mph acceleration is not sensitive to final drive ratio but improves as the Gear 2 total ratio increases. With a high Gear 2 ratio, the shift to Gear 3 happens quickly, and since Gear 3 is more efficient than Gear 2, this means higher power to wheels and better acceleration. The top speed is close to 60 mph with a final drive ratio of 6.9. The improvement of acceleration and top speed over the baseline is achieved by down speeding the motor and driveline, which results in lower losses in the motor and gear.

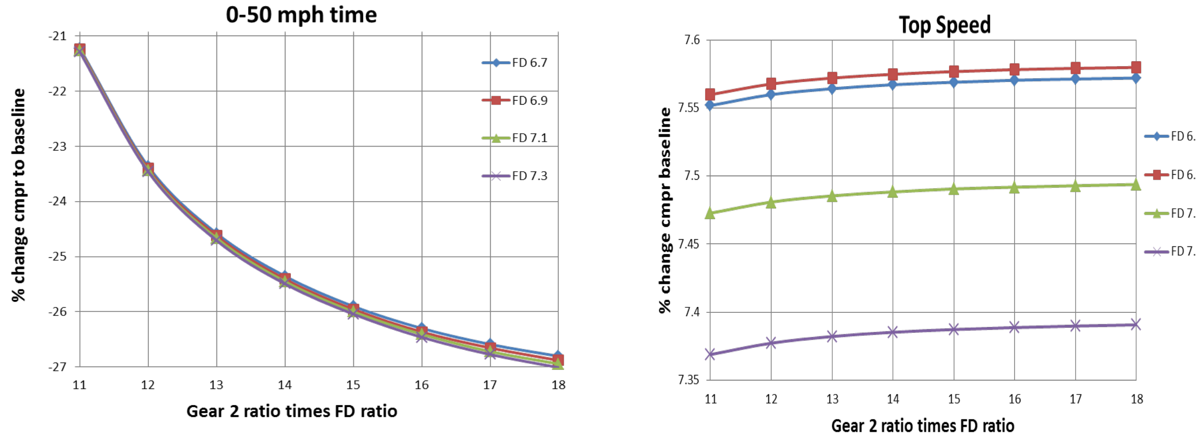



**Figure 30. Acceleration and top speed gains of Smith Newton™ electric truck with three-speed transmission for different FD and Gear 2 total ratio values compared to single speed baseline.**

Figure 31 shows efficiency gains for the Smith Newton™ electric truck with three-speed transmission compared to the single speed baseline for three drive cycles. For HTUF and CILCC cycles, where the vehicle speed is low, the efficiency improvement is 2-3% and insensitive to FD or Gear 2 ratio. However, for the UDDS cycle, since the vehicle speed is higher, the benefit of having the transmission and down-speeding motor and driveline is more pronounced. In this case, the efficiency is more sensitive to FD and Gear 2 ratio choices. To achieve the best top speed and efficiency, the FD=6.9 and Gear 2 total ratio between 15 and 16 results in a Gear 2 ratio between 2.1 to 2.3.

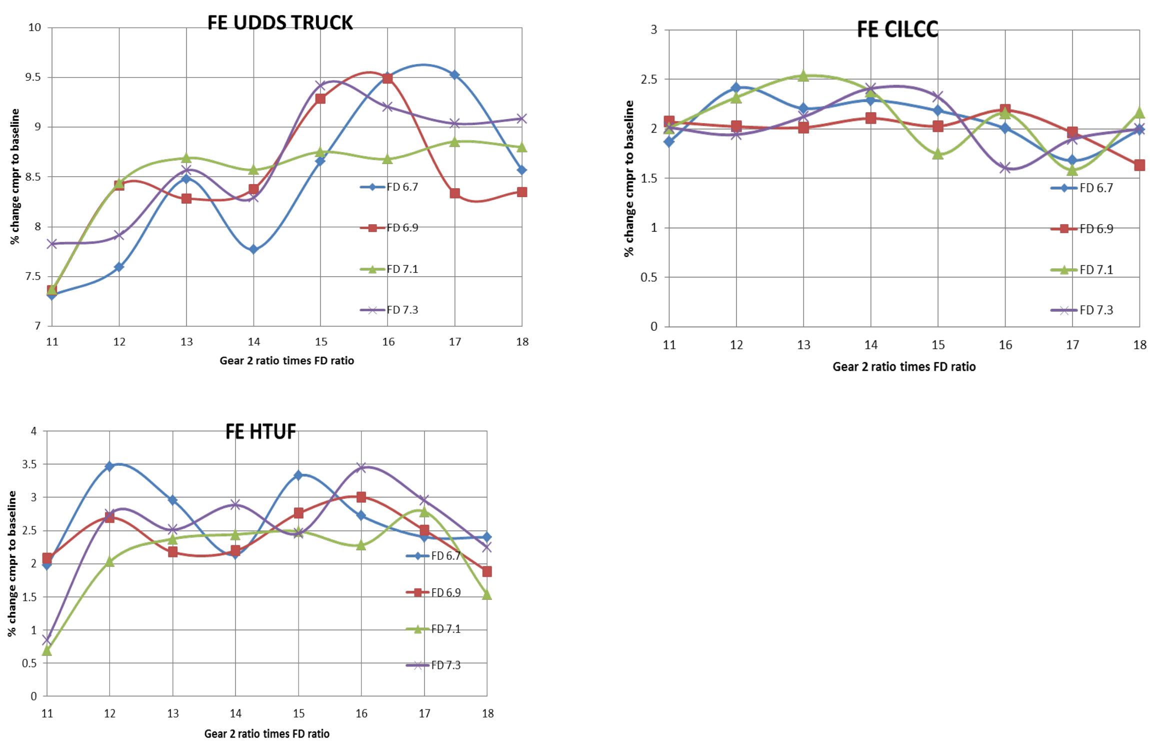
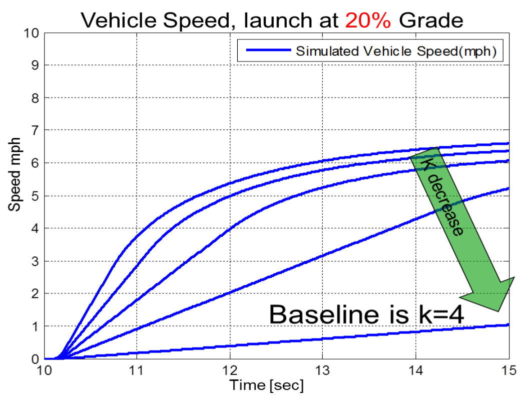



**Figure 31. Energy efficiency gains of Smith Newton™ electric truck with three-speed transmission for different FD and Gear 2 total ratio values compared to single speed baseline.**

Next, we chose the Gear 1 ratio based on startability (launch acceleration) performance on a high-grade road, as shown in Figure 32 for several ratios. With the Gear 1 ratio decrease, the vehicle acceleration is more sluggish because the torque to the wheel is lower. For startability, any ratio greater than 3.5 is acceptable. With a gear ratio of 4.3, the vehicle can launch with acceptable acceleration on grades up to 22.5%, while the baseline capability is limited to 9.8%.




FD=6.9, gear 1 ratio = $FD * 1.2^{K}$

| k | gea1 1 | total |
|---|---|---|
| 4 | 2.07 | 14.31 |
| 5 | 2.49 | 17.17 |
| 6 | 2.99 | 20.60 |
| 7 | 3.58 | 24.72 |
| 8 | 4.30 | 29.67 |

**Figure 32. Vehicle startability (speed launching) on a 20% grade for several Gear 1 ratios.**

The vehicle with three-speed transmission has improved gradeability, that is, it has higher top speed from 0 to 3% grades compared to the baseline vehicle, as shown in Table 9. A multi-speed transmission does not have any impact on the top speed beyond 3% grades, since the electric motor does not operate at the high rpm, lower efficiency region beyond 3% grade on the baseline vehicle in the evaluated drive cycles.

**Table 9. Comparison of top speed at a grade performance of baseline vehicle with single speed gearbox and vehicle with three-speed transmission.**

| Grade % | Top Speed (mph) | |
|---|---|---|
| | Baseline EV with SS gearbox | EV with three-speed transmission |
| 1 | 46.6 | 50.4 |
| 2 | 38.8 | 41.3 |
| 3 | 32.6 | 32.7 |

## 3. EXPAND THE TRANSMISSION MODLING ACTIVITIES TO OTHER EV POWERTRAIN

We extended the vehicle modeling and simulation studies to other MD, medium-heavy and HD market segments to understand the needs of electric trucks from Class 2b to Class 8. We wished to generate a comprehensive database to support the clean sheet design of the three- and four- speed ATM family. The database and design would cover the spectrum of relevant commercial EVs, with duty cycle-driven analysis of efficiency and other performance gains. Modeling included electric motor sizing, and the transmission and the final drive ratio.

**Deliverables:** Expanded Transmission Modeling/Simulation Report

### 3.1 SUBTASK 3.1 – ELECTRIC BUS APPLICATIONS

To evaluate the benefits of multi-speed transmission for electric transit buses, our first task was to establish the baseline performance of a direct-drive equivalent bus. The joint website between the Federal

Transit Administration (FTA) and Penn State University provided the needed information for the baseline performances of the New Flyer XE40 electric bus based on Altoona Bus Tests [3]. The New Flyer bus with GVW of 19,300 kg was fitted with a 200kWh battery pack and a traction motor directly driving the rear axle. This configuration was well suited for a retrofit with a multi-speed transmission.
The FTA website provides a detailed test report for this bus, including acceleration tests and energy consumption on three drive cycles: central business district, arterial, and commuter. This experimental data allowed us to calibrate our model. The report also lists high-level component specifications for the New Flyer XE40:

Dow Kokam XSYST 7 battery: 200 kW, 650 V
Siemens traction motor: 160 kW, 2500 Nm peak torque
Direct drive
Final drive: ratio :5.667
Michelin XZU / 305/70R/22.5 tire

**Simulink model**

The Simulink model created with Autonomie™ in Task 1 and shown in Figure 7 was re-used for this study, as the electric bus architecture is identical to the truck model. This Simulink model needed to be parameterized with appropriate values to match the New Flyer XE40 components. Most component specifications could be found on the internet. Eaton obtained more detailed information about the NewFlyer electric bus such as aerodynamic drag and rolling resistance. We used this information to refine the baseline model both on transient cycles and acceleration tests provided by the FTA.

**Baseline model validation**

FTA report LTI-BT-R1405 lists average energy consumption for tests performed at Altoona Bus Research and Testing Center [4]. Those tests consisted of the central business district (CBD), arterial and commuter cycles run on an oval track on two different days in both clockwise and counterclockwise direction to minimize the effect of wind and grade. The test bus was instrumented to record distance and electrical energy, among other variables. **Figure 33** shows the speed profile of tests carried out on the New Flyer XE40 electric bus.

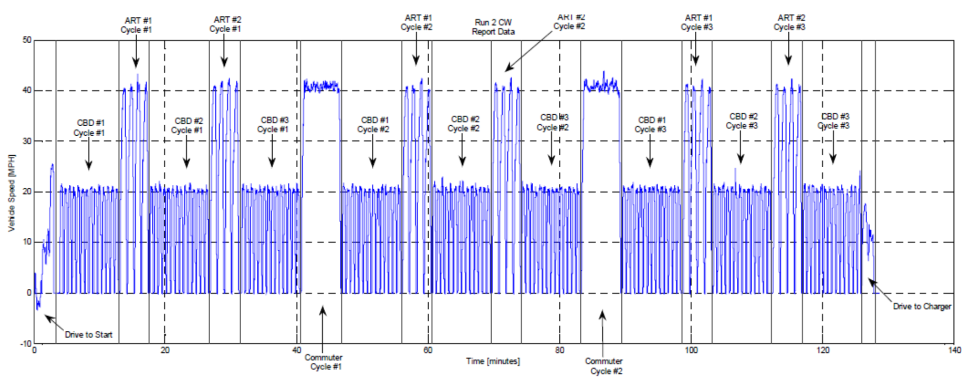



**Figure 33. Altoona test profiles.**

The FTA report compiles average energy use for each test. The Simulink model was tuned to match those experimental results: rolling resistance and aero drag had to be adjusted down slightly to improve the model. **Figure 34** shows energy consumption per mile for the real vehicle tested at Altoona (labelled "FTA") and the Simulink model (labelled "Model"). The error is within 2.6% for all cycles.

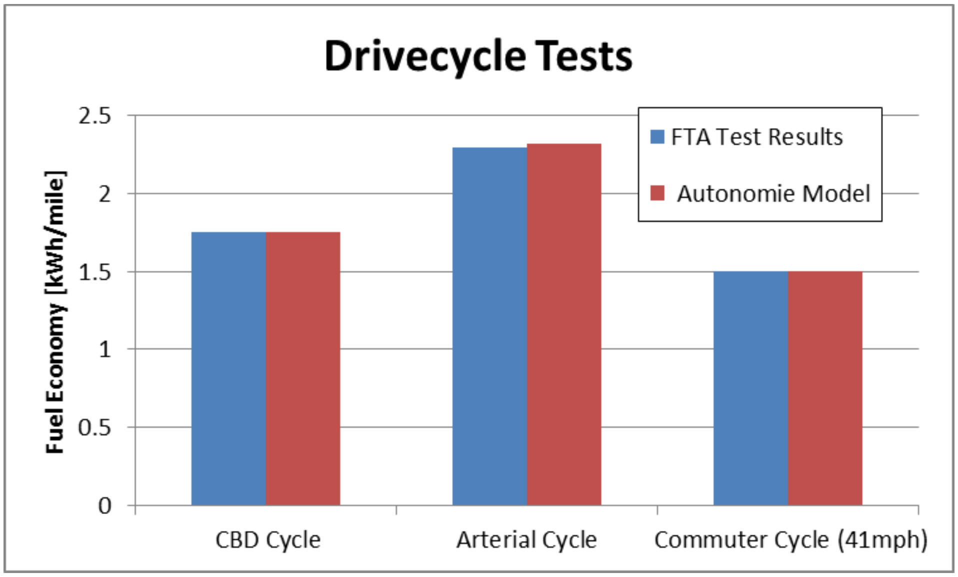



**Figure 34. Model validation results for drive cycle tests.**

The FTA report also provided acceleration test results conducted at Altoona. They consist of wide-open throttle accelerations repeated several times and different directions to remove the effect of varying environmental conditions. The average acceleration profile (labelled "FTA") is compared to the model behavior (labeled "Model") in **Figure 35**.

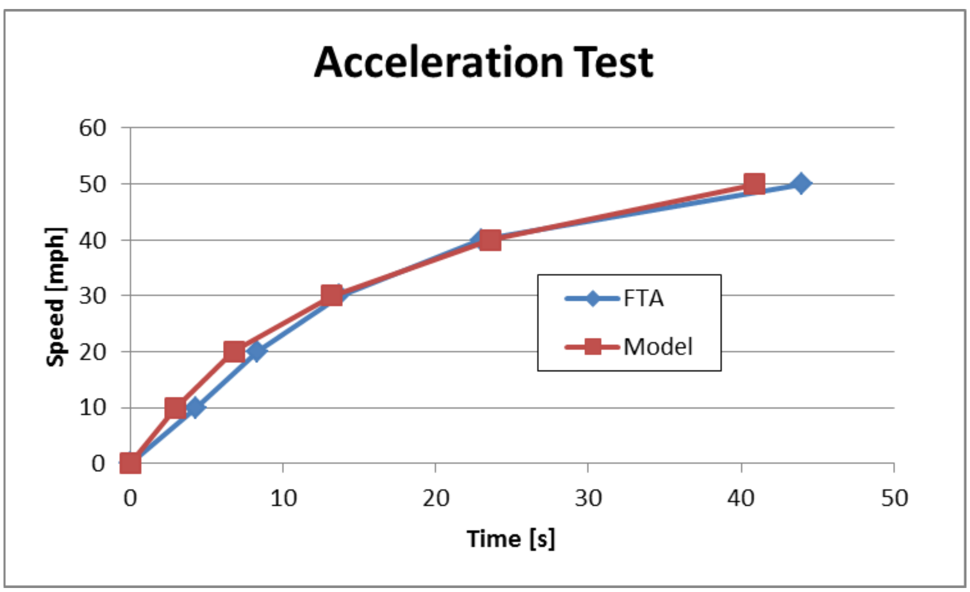



**Figure 35. Model validation results for acceleration test.**

Based on the FTA experimental data, the Simulink model was deemed representative of the real vehicle and was modified to include a multi-speed transmission to quantify the benefits of such a transmission on electric transit buses.

**Multi-speed transmission vehicle model creation**

Following the creation and validation of a baseline model, the model was converted from a direct-drive driveline, which corresponds to the production configuration implemented by New Flyer in their electric bus, to a multi-speed transmission driveline as proposed by this study. Eaton provided the multi-speed transmission and controller models.

The multi-speed model was validated the same way as the MD delivery truck configuration presented in Task 1. We set the multi-speed transmission ratios and efficiencies to the same values as the single-speed transmission and then verified that both models generate identical outputs.

To begin, the multi-speed transmission was defined to model three gears. The first gear is used only on large grades and is not engaged during regular driving, so for all subsequent tests, the vehicle started in second gear. The third gear is a direct drive with a 1:1 ratio.

The reported study assumes that the transmission has three gears, and, except for the transmission, the vehicle was not modified. The original traction motor specified for the direct-drive baseline vehicle is also used with the multispeed transmission.

*Acceleration performance and top speed optimization results*

The multi-speed transmission vehicle model was subjected to wide open throttle (WOT) accelerations. We recorded zero to 10mph, zero to 50mph acceleration times, as well as the top speed after 200 seconds for a variety of final drive ratio and the second gear ratio combinations.
Final drive ratios and the transmission ratios do not affect max speed unless max motor speed is exceeded, because top speed is power limited, not motor speed limited as shown in **Figure 36**.

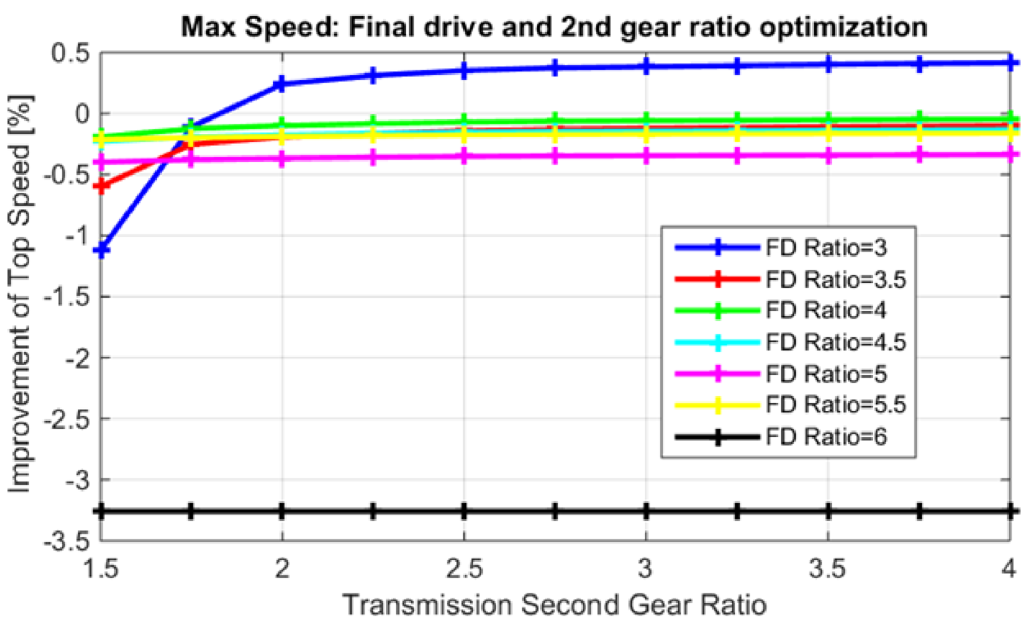



**Figure 36. Final drive and second gear ratio effect on vehicle top speed.**

In this vehicle, a multi-speed transmission mainly improves initial acceleration (0-10mph) thanks to increased torque at low speed. The 0-50mph time effect is smaller because, for higher speeds, acceleration is limited by the flat power curve of the selected motor, regardless of gearings. A motor whose power curve tapers off at high speed would benefit from a multi-speed transmission to allow the motor to keep on operating at a lower speed where power hasn't been derated. High final drive ratios and high second gear ratios yielded the most improvements as shown in **Figure 37** and **Figure 38**.

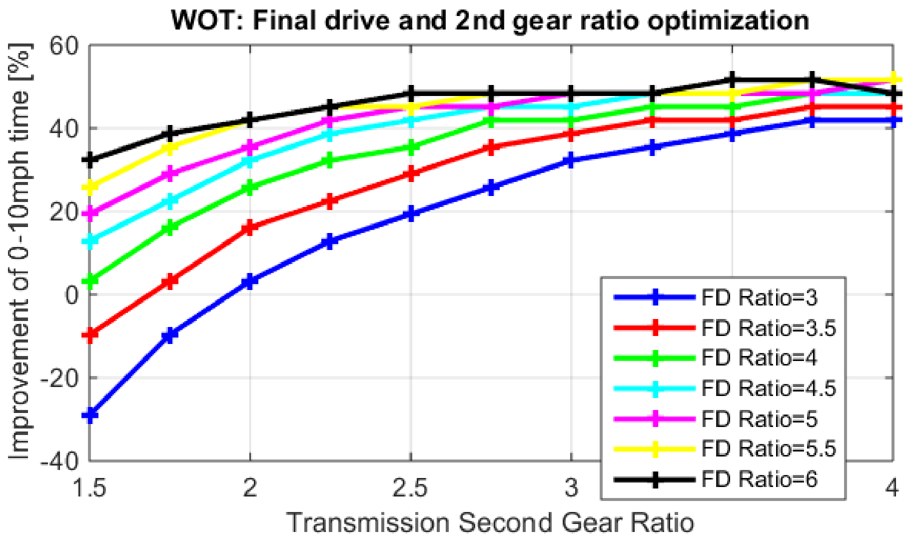



**Figure 37. Final drive ratio and the second gear ratio effect on 0-10mph acceleration.**

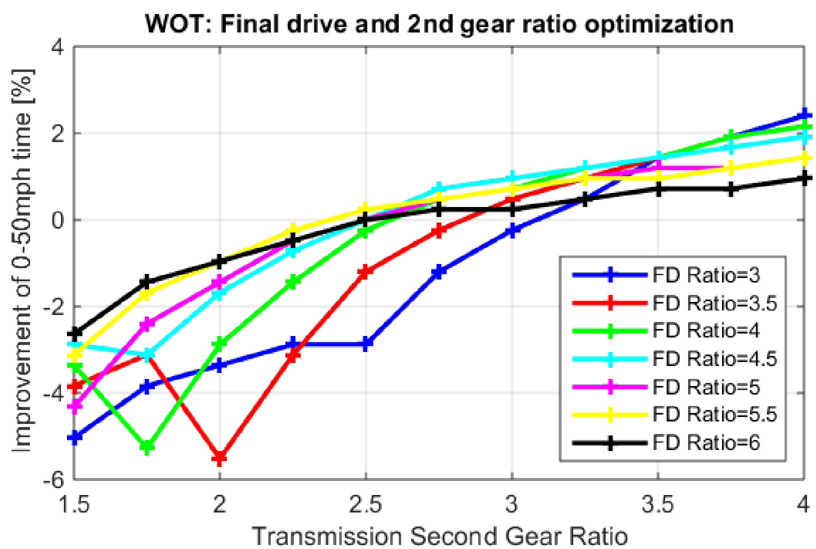



**Figure 38. Final drive and second gear ratio effect on 0-50mph acceleration.**

**Energy efficiency optimization results**

The multi-speed transmission vehicle model was subjected to the same three cycles used to validate the baseline model: CBD, arterial and commuter cycles. Energy consumption for each cycle was modeled, integrated and plotted for a variety of final drive and second gear ratio combinations.
Low final drive ratios (less than 4:1) small second gear ratios (less than 2.5:1) yield the best energy efficiency across the three test cycles. Results are shown in **Figure 39** to **Figure 41**.

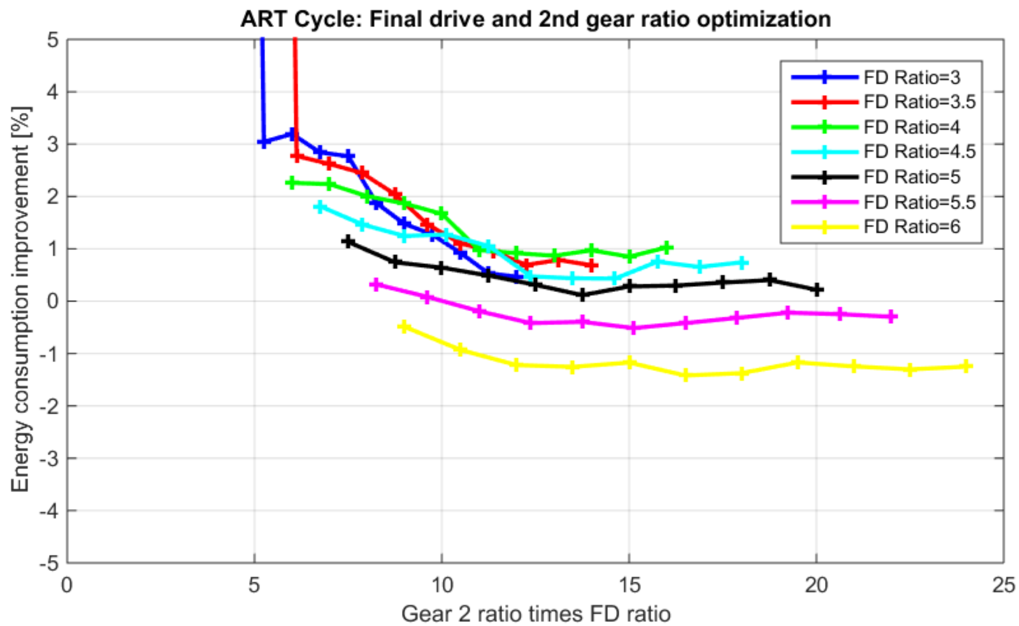



**Figure 39. Final drive and second gear ratio effect on energy efficiency improvement of three-speed vehicle on arterial cycle.**

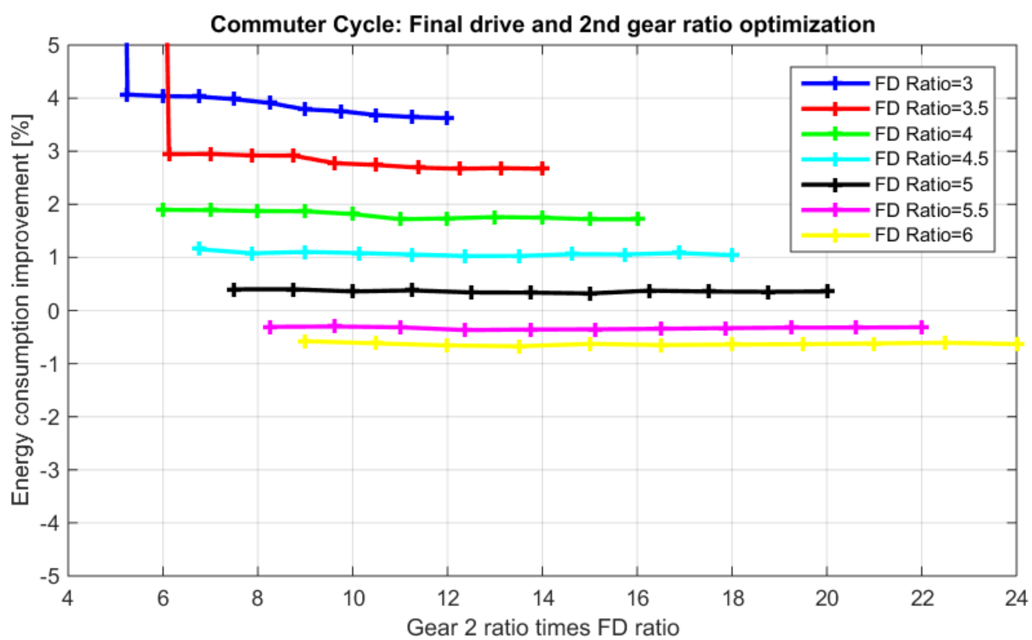



**Figure 40. Final drive and second gear ratio effect on energy efficiency improvement of three-speed vehicle on commuter cycle.**

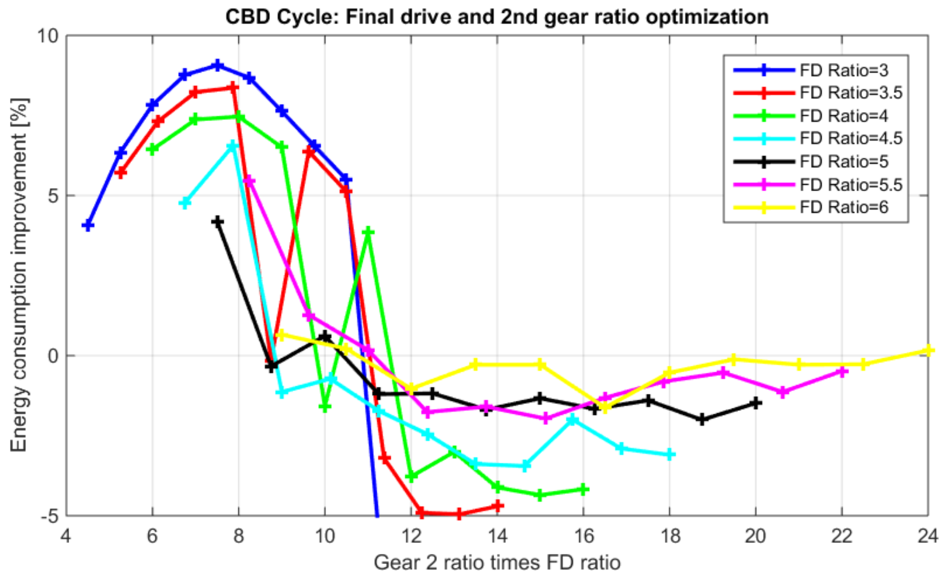



**Figure 41. Final drive and second gear ratio effect on energy efficiency improvement of three-speed vehicle on CBD cycle.**

## Electric transit bus application for China market

Eaton's new market assessment shows that 8-, 10- and 12- meter electric bus segments have been expanding rapidly due to changes in regulations and subsidies in China. We used modeling and simulations to investigate the transmission needs of these three bus segments.
**Table 10** lists the China city bus baseline vehicle parameters used to set up the model. **Table 11** lists performance targets for the China market. **Table 12** lists the city drive cycles are used to compare the energy efficiency of vehicles configured with and without transmission.

**Table 10. Electric city bus vehicle parameters (China)**

| Vehicle Specifications | City Bus Vehicle (12 Meters) | City Bus Vehicle (10 Meters) | City Bus Vehicle (8 Meters) |
|---|---|---|---|
| Vehicle mass (kg) | 18000 | 15600 | 13000 |
| Tire size (m) | 0.507 | 0.493 | 0.512 |
| Aero drag coefficient | 0.565 | 0.565 | 0.65 |
| Frontal area ($m^2$) | 7.5 | 7.0 | 6.5 |
| Rolling road coef. | 0.007 | 0.007 | 0.007 |

**Table 11. 12-meter electric bus performance targets**

| Top speed | >50 mph ( 80kph) |
|---|---|
| 0-30 mph | 25 sec |
| Gradeability | >15% |
| Startability | >8% |

Startability is the maximum grade a vehicle can achieve with at least 1 $m/s^2$ acceleration. The baseline vehicle (direct drive without transmission) meets or exceeds the required performance metrics due to an oversized electric motor.

**Table 12. City drive cycles to compare energy efficiency of vehicles with and without transmission**

| | Chongqing City Bus | China City Bus | San Francisco City Bus | Beijing City Bus |
|---|---|---|---|---|
| Distance (mile) | 4.1 | 3.7 | 8.7 | 4.5 |
| Duration (s) | 1324 | 1314 | 2542 | 1926 |
| Average speed (mph) | 11.2 | 10.1 | 12.3 | 8.5 |
| Max speed (mph) | 29.8 | 37.5 | 28.1 | 29.2 |
| Average grade (%) | -1.3 | 0 | 0.004 | 0 |
| Max grade-up (%) | 3.9 | 0 | 9.0 | 0 |
| Max grade-down (%) | -7.3 | 0 | -10.1 | 0 |

Adding multi-speed transmission enables significant motor downsizing compared with direct drive since high torque is not necessary. Motor downsizing significantly reduces motor cost, weight, and energy waste. Motor efficiency improvement can translate into either battery downsizing (and lowering battery cost) to achieve the same range or range increase with the same size battery; both are very important to the end costumer. Thus, we chose a downsized eDrive motor having almost the same power as the baseline motor and half of its torque capability, as shown in **Figure 42**. We studied efficiency gains from a three-speed and a four-speed transmission and compared the benefits of each. The top speed results as a function of final drive ratio are shown in **Figure 43**. We found that to meet the top speed requirement, the final drive ratio cannot be larger than 5.8. The 0-30mph acceleration is in the range of 15 seconds, and hence, the acceleration requirement does not impose any design limitations.

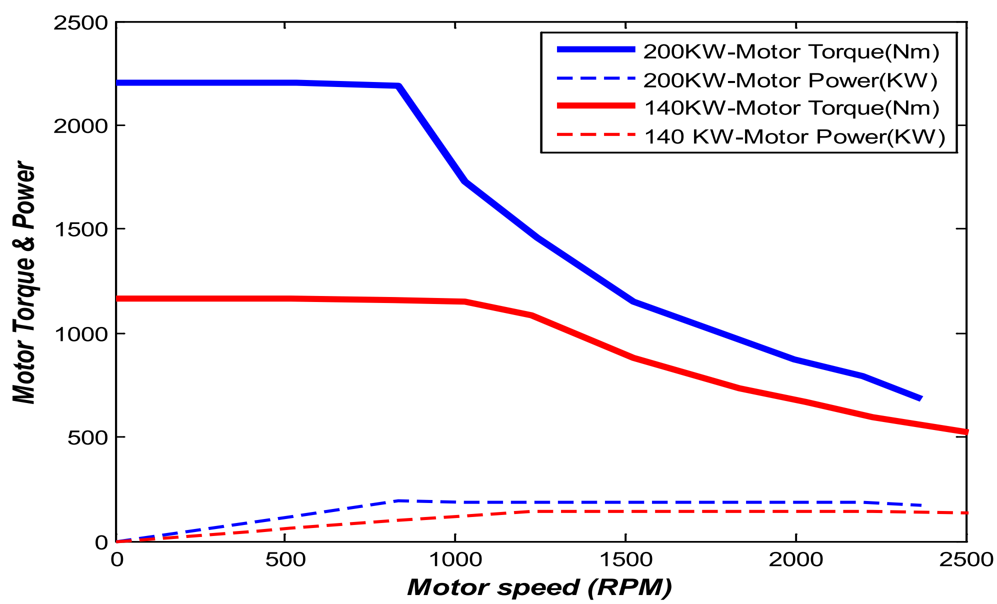



**Figure 42. eDrive motor torque and power curves for 200/100KW & 140/80KW motors.**

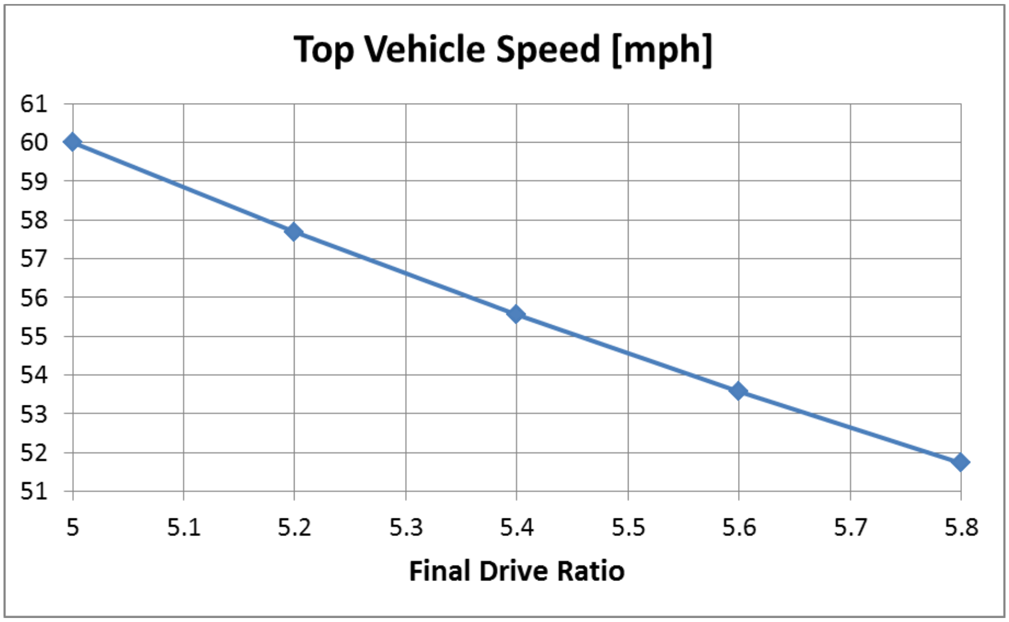



**Figure 43. Top speed of 12 m electric bus as function of final drive ratio.**

*Energy savings with three-speed transmission*

Gear 1 has a high gear ratio and is engaged only at grades higher than 15%. The first gear must meet gradeability and startability requirements. Based on Newton's law, we found that a gear ratio around 5.8 is suitable. In lower grades the vehicle is launched at the second gear. The third gear is direct drive and always 1:1. Hence, the problem is reduced to the determination of the second gear ratio based on the energy consumption optimization. Energy savings were calculated for many combinations of the second gear ratios, final drive ratios, and shift points.

A final drive ratio of 5.8 and second gear ratio of 2.3 yield the best energy savings while meeting the required performance (**Figure 44)**. With these ratio selections, we expect to gain 8% energy saving on average over the baseline for all drive cycles.

Table 13. summarizes energy consumption reduction enabled by the three-speed transmission with optimized second gear ratio of 2.3 and final drive ratio of 5.8 for four drive cycles and three city bus applications (12, 10 and 8 meters)**.** From the results shown in the table, we can expect up to 9.6 % energy savings with a three-speed transmission over the baseline, depending on the drive cycle and the segment size of the city bus application.

**Table 13. Energy savings with three-speed transmission for all dive cycles and all city bus applications**

| Drive Cycle | Energy Savings (%) | | |
|---|---|---|---|
| | 12-Meter Bus | 10-Meter Bus | 8-Meter Bus |
| Beijing | 7.9 | 8.0 | 7.9 |
| China City | 6.0 | 6.8 | 8.2 |
| Chongqing | 8.8 | 9.6 | 9.4 |
| San Francisco | 9.2 | 9.2 | 8.8 |

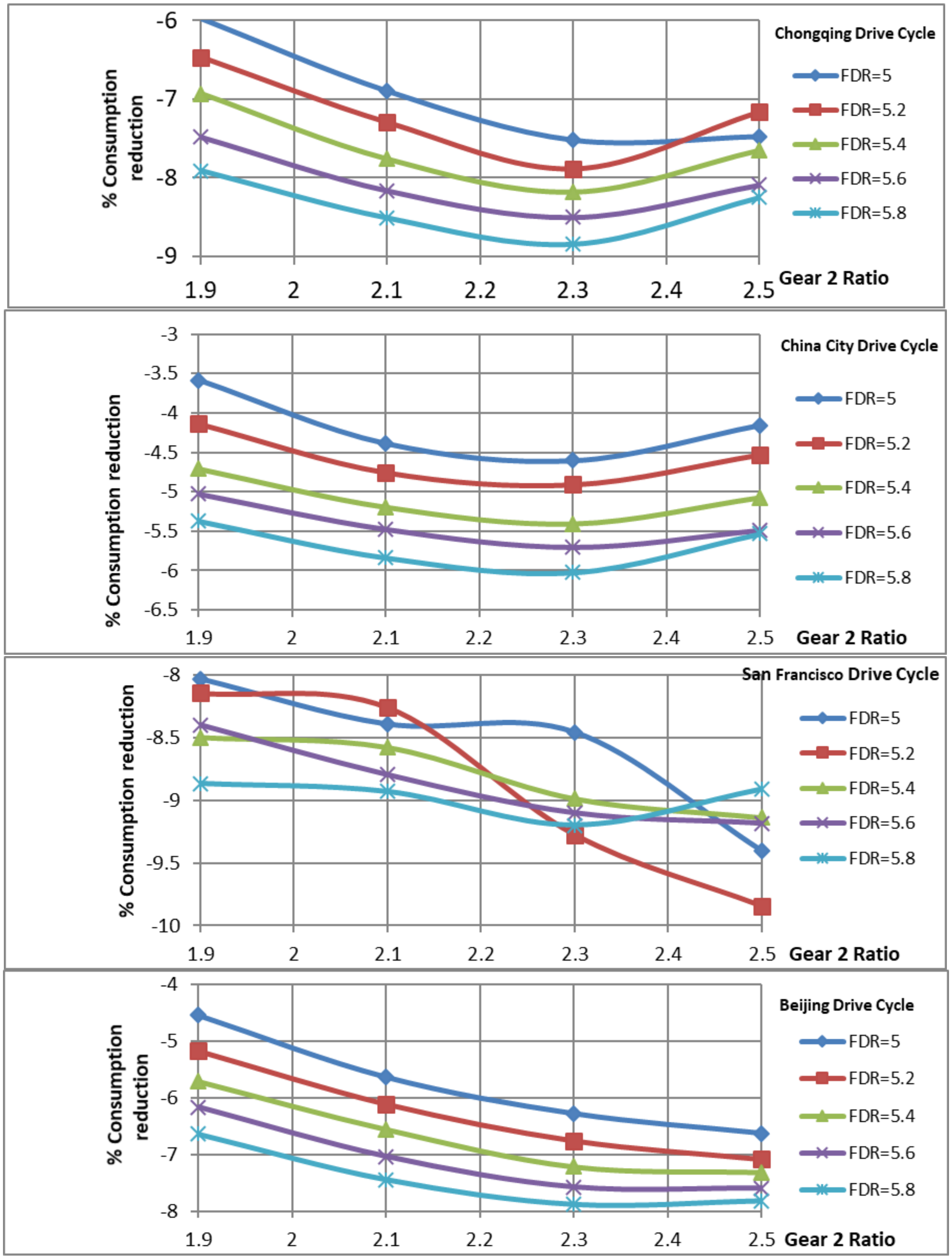



**Figure 44. Energy savings with three-speed transmission and downsized electric motor compared to 12-meter direct drive baseline electric bus.**

*Energy savings with four-speed transmission*

Energy savings with a four-speed transmission were simulated for many combinations of the second and the third gear ratios, final drive ratios, and shift points. The value of the second gear ratio was set at twice the value of the third gear ratio and the fourth gear is always 1:1. Therefore, the third gear and final drive ratio are the only two parameters requiring optimization.

The results for energy savings with a four-speed transmission and downsized electric motor compared to the direct-drive baseline vehicle for a 12-meter city bus application are shown for four drive cycles in **Figure 45**. The final drive ratio of 5.8 and the third gear ratio of 2.1 provide 9.2% average energy consumption reduction over the baseline for all drive cycles while meeting all other performance requirements.

Energy savings with a four-speed transmission having optimized second and third gear ratios and a final drive ratio of 5.8 are listed for four drive cycles and three city bus applications (12, 10 and 8 meters) in **Table 14**.

**Table 14. Energy consumption reduction (%) for four drive cycles and three city bus applications with four-speed gearbox transmission.**

| Drive Cycle | Energy Consumption Reduction (%) | | |
|---|---|---|---|
| | 12 Meter Bus | 10 Meter Bus | 8 Meter Bus |
| Beijing | 9.9 | 9.7 | 9.8 |
| China City | 7.2 | 7.9 | 9.1 |
| Chongqing | 10.3 | 10.9 | 10.7 |
| San Francisco | 9.5 | 9.2 | 8.7 |

From the results listed in **Table 14**, we can expect up to 10.9% energy consumption reduction with four-speed transmission over the baseline, depending on the drive cycle and the segment size of the city bus application.

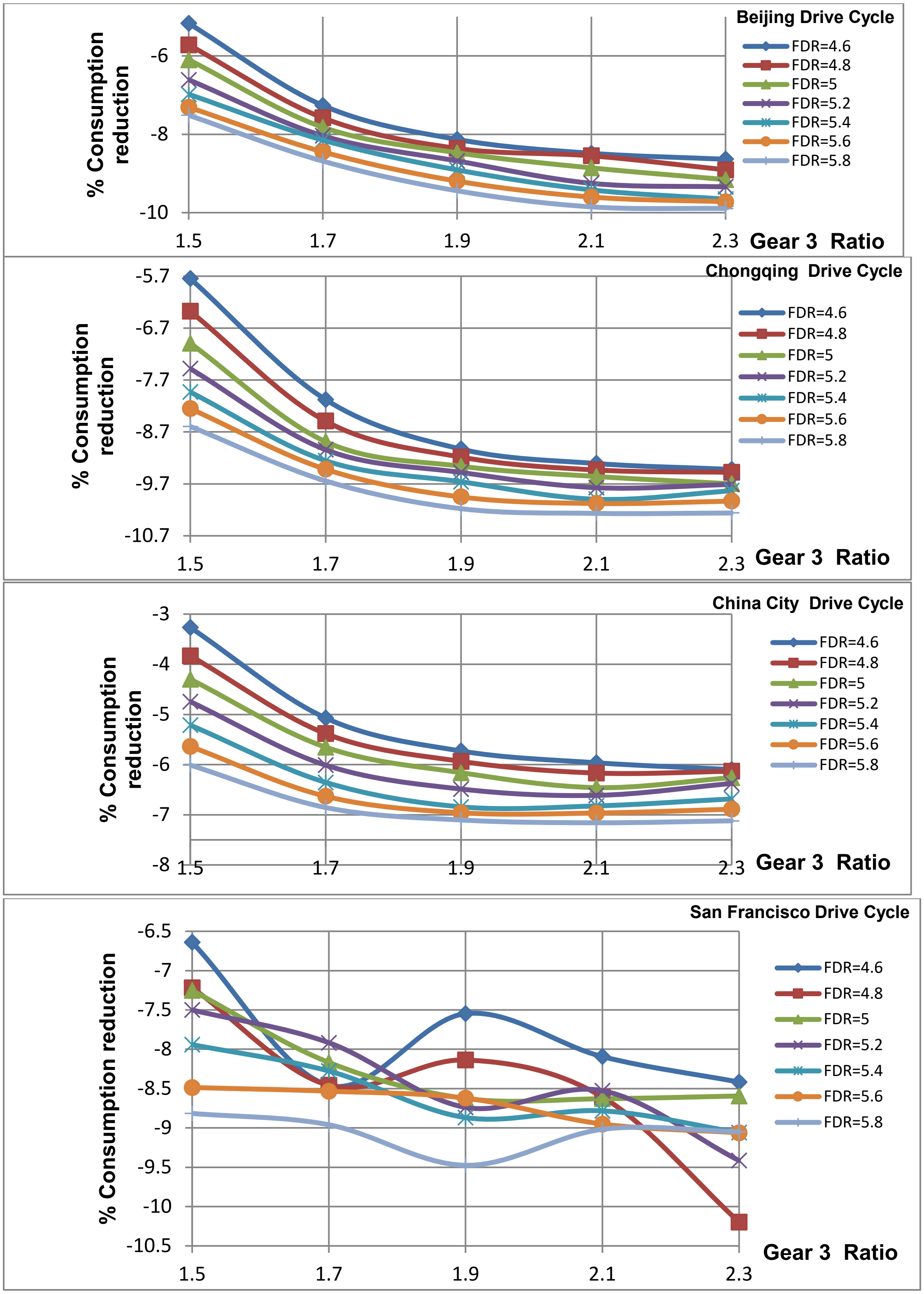



**Figure 45. Energy savings with four-speed transmission and downsized electric motor as compared to 12-meter direct drive baseline electric bus.**

*Energy savings with four-speed over three-speed*

**Table 15** compares energy savings with four-speed transmission to three-speed transmission for all dive cycles and all city bus applications (12-, 10- and 8- meters). From the results listed in **Table 15**, we expect up to 2% more energy savings with four-speed over three-speed transmission, depending on the drive cycle and the segment size of the city bus application.

**Table 15. Energy savings (%) of four-speed compared to three-speed transmission for four drive cycles and three city bus applications (12, 10, 8 meter).**

| Drive Cycle | Energy savings (%) provided by four-speed over three-speed | | |
|---|---|---|---|
| | 12-Meter Bus | 10-Meter Bus | 8-Meter Bus |
| Beijing | 2.0 | 1.7 | 1.8 |
| China City | 1.2 | 1.2 | 0.9 |
| Chongqing | 1.5 | 1.4 | 1.4 |
| San Francisco | 0.8 | 0 | 0.1 |

The ratio spread was kept the same, 5.8, for the above listed comparisons between the three-speed and four-speed transmission. Hence, the ratio spread was set to 5.8 in both three-speed and four-speed simulations in the above comparisons. However, the new four-speed electric vehicle transmission designed in this project is flexible enough to accommodate a gear ratio up to 9.0 for the first gear. Considering this flexibility, the simulations were repeated with a four-speed transmission where the fourth gear is a direct drive with a ratio of 1 and the step size between the gears is 2.1 resulting in second, third, and fourth gear ratios of 2.1, 4.4 and 9.2 respectively. The results listed in **Table 16** shows that an electric vehicle with four-speed transmission provides up to 2.6% better efficiency than an EV with three-speed transmission depending on the drive cycle compared. On the fourth column the four-speed transmission has the same ratio spread as the three-speed case, 5.8. On the 5$^{th}$ column the four-speed transmission has equal ratio steps of 2.1 and a large ratio spread of 9.2. Furthermore, four-speed transmission provides up to 11% improvement in efficiency as compared to a direct drive baseline electric vehicle.

**Table 16. Energy savings for a 12-meter city bus application with three- and four-speed transmissions compared to direct-drive baseline in four drive cycles.**

| Drive Cycle | Baseline | Energy Savings Compared to the Baseline | | | |
|---|---|---|---|---|---|
| | Direct Drive & Large Motor (kWh/km) | Three-speed | Four-speed [Same ratio spread as 3-sp] | Four-speed [Larger ratio spread than 3-sp] | Three-speed vs. four-speed |
| Beijing | 0.61 | 7.9% | 9.9% | 10.5% | 2.6% |
| China City | 0.70 | 6.0% | 7.2% | 7.8% | 1.8% |
| Chongqing | 0.65 | 8.8% | 10.3% | 10.8% | 1.9% |

| | | | | | |
|---|---|---|---|---|---|
| San Francisco | 0.46 | 9.2% | 9.5% | 9.3% | 0.1% |

## 3.2 SUBTASK 3.2 – LIGHT AND MEDIUM-DUTY VEHICLE APPLICATIONS

**Light-duty vehicle application**

We chose the BYD e6 passenger vehicle as the baseline for the light-duty vehicle application. Table 17 lists the publicly available information on the baseline vehicle [5].

**Table 17. Baseline vehicle parameters.**

| | |
|---|---|
| **Mass** | 2380 kg |
| **Top Speed** | 140 kmph/86 mph |
| **Consumption** | 3.1 mile/kWh |
| **Max power** | 121 hp (90 kW) |
| **Max torque** | 450 Nm |

Unfortunately, the final drive ratio of the vehicle was not disclosed. Energy consumption, top speed, and acceleration of the baseline were predicted by simulations, as shown in Figure 46. Based on the simulation results, we chose 5 as the baseline final drive ratio to match the top speed of 86mph.

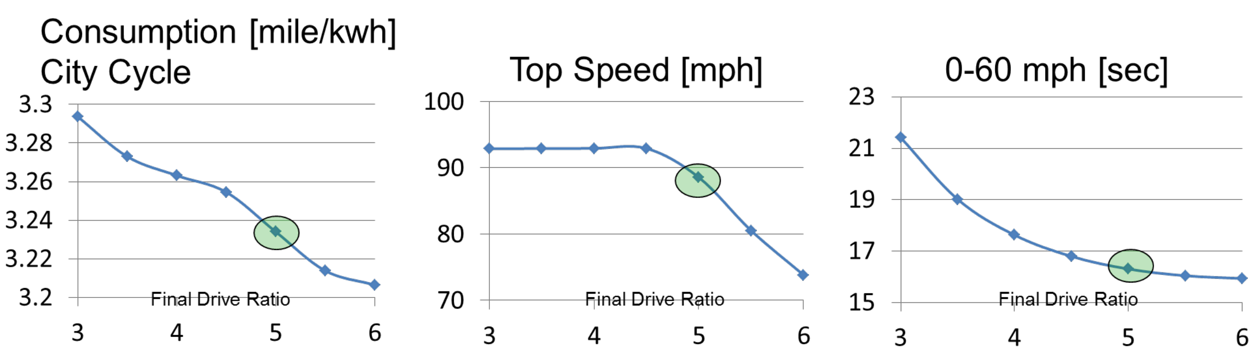



**Figure 46. Energy consumption, top speed, and acceleration of baseline vehicle for final drive.**

The questions we wanted answered were whether motor down-sizing is possible and how much energy saving is possible with a multi-speed transmission, if other baseline performances related to top speed, acceleration, and gradeability are acceptable.

Motor downsizing has multiple benefits, including cost and weight reduction and energy savings. The goal of this simulation was to quantify the latter. The downsized motor has the same power as the baseline (90kW), but half of the baseline torque (225 Nm). The transmission is a two-speed transmission, the first gear ratio needs to be determined and the second gear is direct drive.

Figure 47 shows the performance results for different final drive and first gear ratios with a two-speed transmission. A good choice to balance consumption and performance is a final drive and first gear ratios equal to 3.5 and 2.2 respectively. With these choices, the motor loss is decreased by 45%; however, because of transmission losses, only a portion translates into the overall system efficiency. Nevertheless, the total benefit is still significant — equal to 9% energy savings.

It is important to understand that the acceleration performance is worse than the baseline. Several factors contribute: first, a torque interrupt of about 0.5 second during shifting; second, the equivalent mass of vehicle is higher at high gear ratio because of moving parts, including motor inertia; third, and most important, the shift logic was not tuned for full throttle acceleration performance and there is significant room to improve the acceleration.

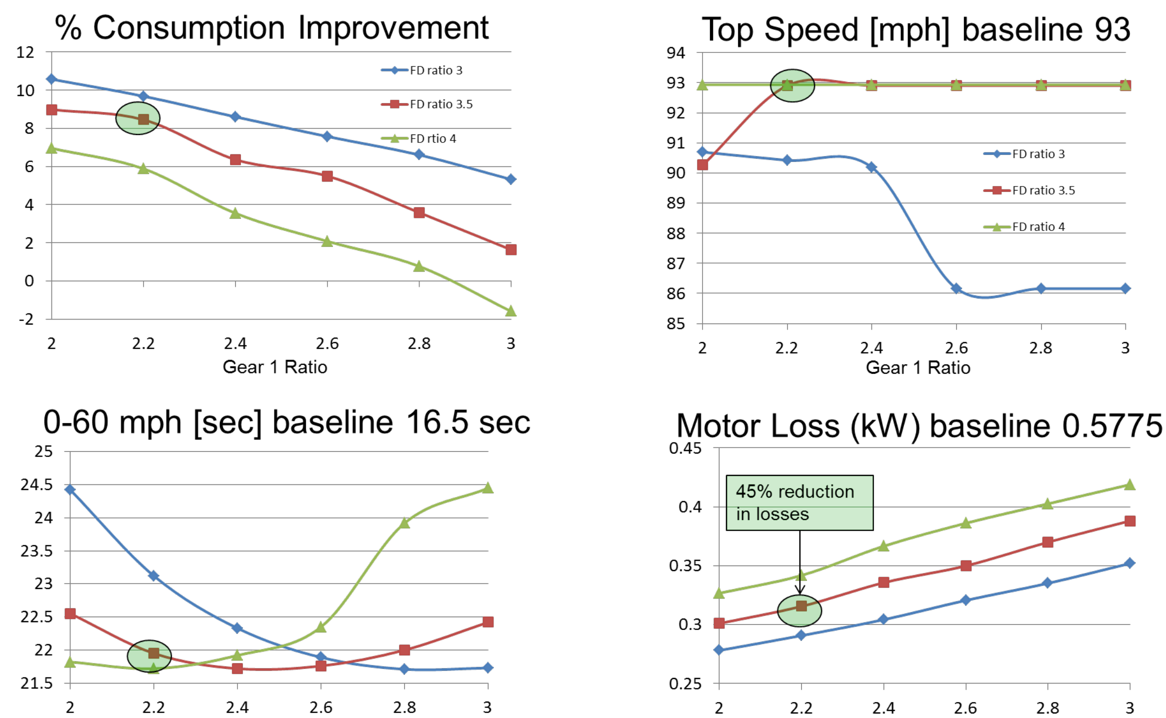



**Figure 47. Energy savings for a city cycle, top speed, acceleration and motor loss with a two-speed transmission and a down-sized motor as function of first gear and final drive ratios.**

**Electric school bus application**

Detailed modeling and simulations were reported on the Smith Electric™ MD Commercial Delivery Truck inTtask 1. The modeling and simulation results on a medium-duty school bus with GVW of 16 ton (36000 pounds [lb]) are presented here. The energy efficiency and vehicle performance with a multi-speed transmission and a downsized motor are compared with the direct drive baseline based on a standard drive cycle for a school bus, as shown in Figure 48. For this application, we used the same vehicle performance targets and motor downsizing considered in the previous section that covered three city bus applications. We used the same assumptions and the methodology as for first gear, described above.


**Figure 48. Standard drive cycle for school bus comparing energy efficiency and vehicle performance with and without multi-speed transmission.**

*Electric school bus energy savings with three-speed and4- speed transmissions*

Energy savings with three-speed transmission are shown in Figure 49 for many combinations of second gear and final drive ratios and shift points. The third gear is a direct drive with 1:1 ratio.
Energy savings with four-speed transmission were also computed for many combinations of second gear, third gear and final drive ratios and shift points. The value of second gear ratio was set to be twice the value of third gear ratio and the fourth gear is always 1:1. The third gear and the final drive ratio are the only two parameters that need to be optimized as illustrated in Figure 50.

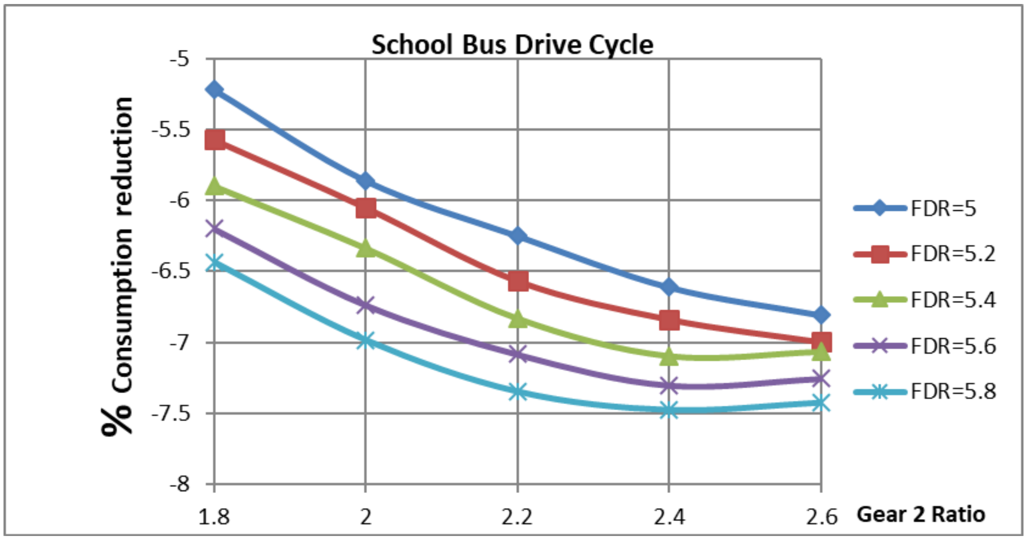



**Figure 49. Energy savings with three-speed transmission and a downsized electric motor as compared to the direct drive baseline electric school bus application.**

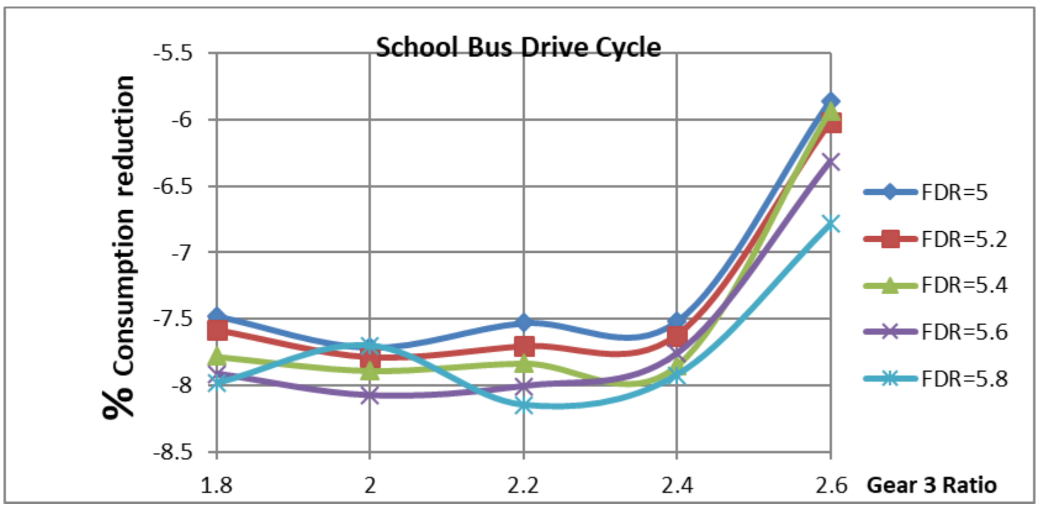



**Figure 50. Energy savings with four-speed transmission and a downsized electric motor as compared to the direct drive baseline electric school bus application.**

Table 18 lists the energy savings with three- and four-speed transmissions with optimized gear ratios and a final drive ratio of 5.8 as compared to the direct drive baseline electric school bus.

**Table 18. Energy savings (%) with three- and four-speed transmissions for electric school bus**

| Drive Cycle | Energy Savings (%) | | |
|---|---|---|---|
| | Three-speed [5.8 2.4 1] | Four-speed [5.8 5.3 2.3 1] | Four-speed over three-speed |
| Standard Cycle | 7.5 | 8.2 | 0.7 |

From the results listed in Table 18, we expect 7.5% and 8.2% energy savings with three-speed and four-speed transmissions over the baseline electric school bus application if the ratio spread is kept the same, 5.8. Hence, the energy saving with four-speed is 0.7% better than with a three-speed transmission. We can also make a case for four-speed transmission with a large ratio spread for drive cycles on large grades. The ratio spread can be increased up to 9 from 5.8 and additional efficiency improvement can be expected with a four-speed transmission compared to a three-speed transmission, as demonstrated earlier in the case of China city bus application.

## 3.3 SUBTASK 3.3 – ELECTRIC REFUSE AND DRAYAGE TRUCK APPLICATIONS

Modeling and simulations of electric vehicle applications were extended to refuse trucks (GVW 27t) and drayage (GVW 36t). The Simulink model was extended to new applications by modifying the vehicle specifications, as listed in Table 19.

**Table 19. Specifications for refuse and drayage electric trucks**

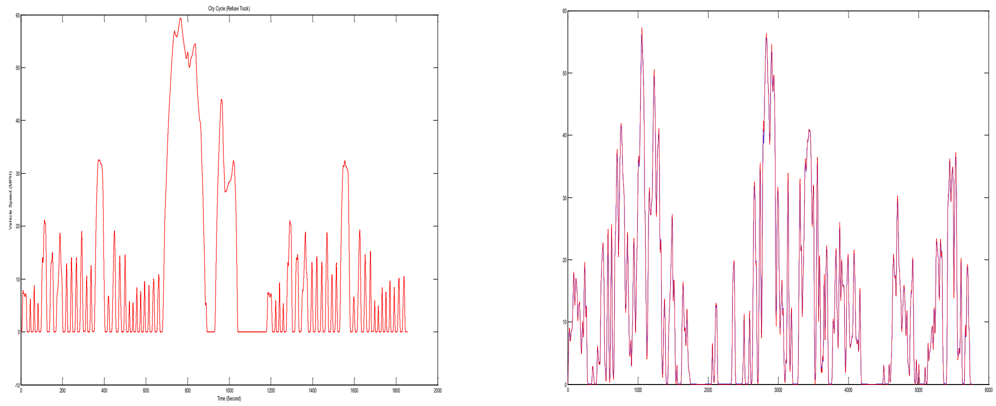
| Vehicle Specifications | Refuse Truck | Drayage Truck |
|---|---|---|
| Vehicle Mass (kg) | 27216 | 36000 |
| Tire Size (m) | 0.52 | 0.512 |
| Aero Drag Coefficient | 0.7 | 0.65 |
| Frontal Area ($m^2$) | 7.1 | 9.5 |
| Road Rolling Coefficient | 0.007 | 0.007 |

**Table 20. Performance targets for refuse and drayage electric trucks**

| | |
|---|---|
| **Top speed** | >60 mph |
| **0-30 mph** | <25 sec |
| **Gradeability** | >15% |
| **Startability** | >8% |

Drayage and refuse trucks are HD vehicles that require big electric motors with higher power/torque capacities than the electric bus applications. We used two electric motors operated in parallel for the direct drive (eDrive-EM 13531303002: 200/100 kW with maximum torque 2203 Nm at maximum speed motor 2500 rpm) and for the multi-speed transmission with two downsized motors (eDrive-RC1244S1314050052: 140/80 kW with maximum torque 1165 Nm at maximum speed motor 2500 rpm).

The vehicle performance targets for electric drayage and refuse trucks are listed in Table 20. The same assumptions and methodology were used as in the electric bus applications in that the first gear has a high ratio and engages only at grades higher than 15%. The vehicle is launched on second gear at lower grades. We compared energy efficiency and vehicle performance with multi-speed transmissions and downsized electric motors with the direct-drive baseline electric vehicles based on the standard drive cycles considered for both vehicles, as shown in Figure 51.



Refuse Drive Cycle Drayage Drive Cycle

**Figure 51. Standard drive cycles with vehicle speed (mph) used to compare energy efficiency and vehicle performance of refuse and drayage trucks configured with and without multi-speed transmission.**

**Electric refuse truck applications**

First gear is calculated at 4.6 to be able to climb the grades higher than 15% for electric refuse truck applications. Furthermore, the final drive ratio cannot be larger than 4.9 to meet the top speed requirement of at least 60 mph and the 0-30 mph acceleration time of less than 25 seconds.

*Electric refuse truck energy savings with three-speed versus 4- speed transmissions*

The energy savings with three-speed transmission were evaluated for electric refuse truck by simulating many combinations of second gear and final drive ratios and shift points with the third gear ratio of 1 as illustrated in Figure 52.

The energy savings with four-speed transmission were evaluated by simulating many combinations of third gear, final drive ratios and shift points. The second gear ratio was set to twice the value of the third gear ratio and the fourth gear was always 1:1, as shown in Figure 53.

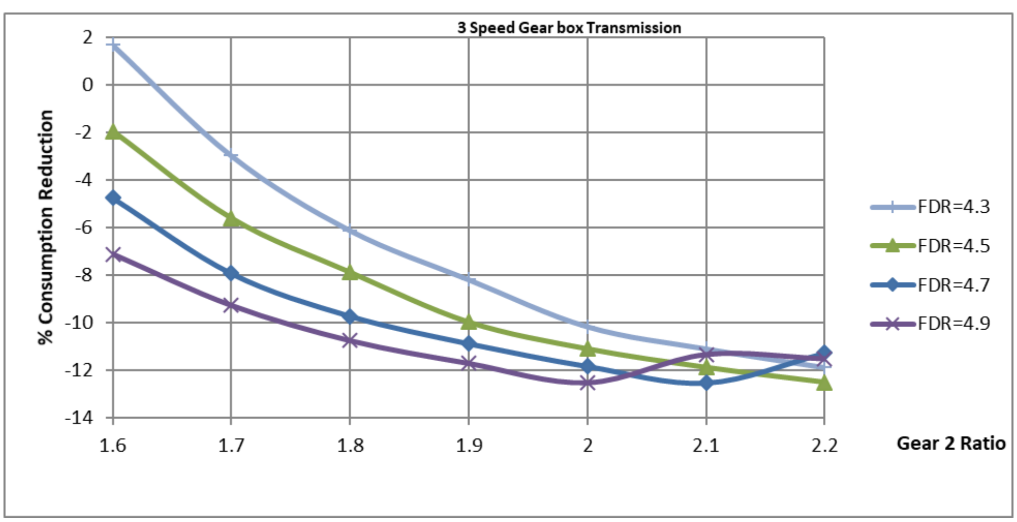



**Figure 52. Energy savings with three-speed transmission and downsized electric motors as compared to the direct drive baseline electric refuse truck application.**

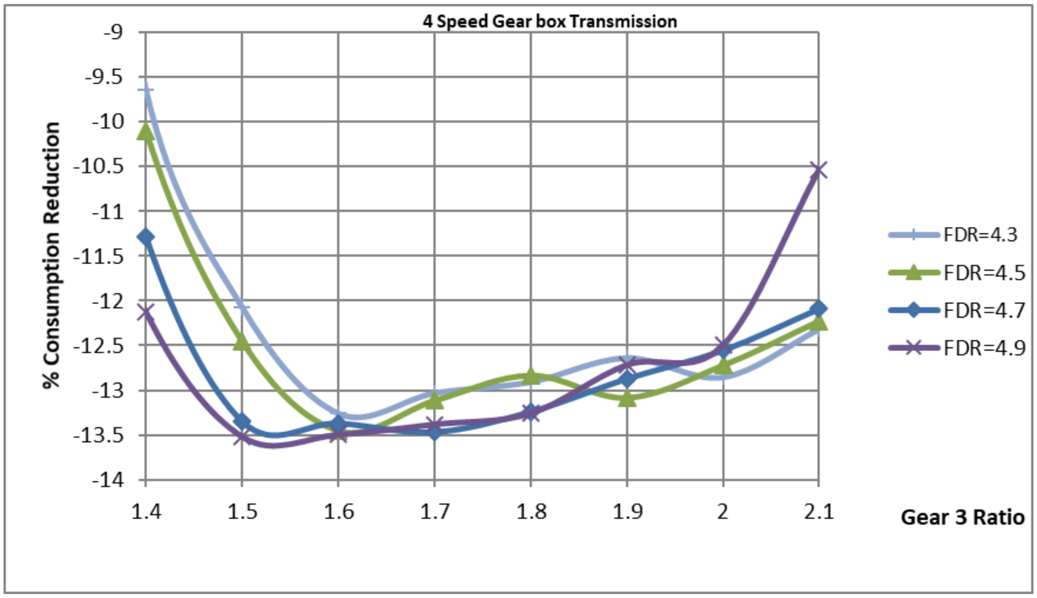



**Figure 53. Energy savings with four-speed transmission and downsized electric motors compared to the direct-drive baseline electric refuse truck application.**

Table 21 lists the energy savings with three- and four-speed transmissions having optimized gear ratios and a final drive ratio of 4.9 compared to the direct-drive baseline electric refuse truck.

**Table 21. Energy savings (%) with three- and four-speed transmissions for electric refuse truck application**

| Drive Cycle | Energy Savings (%) | | |
|---|---|---|---|
| | Three-speed [4.6 2.0 1] | Four-speed [4.6 2.3 1.5 1] | Four-speed over three-speed |
| Standard cycle | 12.5 | 13.5 | 1.0 |

We expected 12.5% and 13.5% energy savings with the optimized three-speed and four-speed transmissions over the direct-drive baseline electric refuse truck application when the ratio spread is kept the same— 4.6, as listed in Table 21. Hence, the four-speed is 1.0% better than the three-speed transmission. We could also make a case for a four-speed transmission with a large ratio spread for drive cycles on large grades. Additional efficiency improvement can be expected from a four-speed transmission over a three-speed transmission if the ratio spread is increased to 9 from 4.6.

*Electric refuse truck acceleration with three-speed versus four-speed transmissions*

The vehicle accelerations times with three- and four-speed transmissions with the optimized gear ratios and a final drive ratio of 4.9 are listed in Table 22 and compared to the direct-drive baseline electric refuse truck.

**Table 22. Vehicle accelerations time for refuse electric truck application with 3- and 4 -speed transmissions compared to direct-drive baseline electric refuse truck**

| | Vehicle Acceleration Time in seconds | | |
|---|---|---|---|
| | **Direct drive baseline electric refuse truck** | **Three-speed [4.6 2.0 1]** | **Four-speed [4.6 2.3 1.5 1]** |
| 0-10 mph | 4.5 | 3.8 | 3.4 |
| 0-20 mph | 7.9 | 7.4 | 6.9 |
| 0-30 mph | 12.2 | 13.6 | 12.9 |

For all electric driveline configurations, 0-30 mph vehicle acceleration times were less than 25 seconds and met the acceleration requirements as listed in Table 22. Acceleration times at 0-10 mph and 0-20 mph are reduced for both three- and four-speed transmissions over the direct-drive baseline electric refuse truck. However, the 0-30 mph acceleration time for three- and four-speed transmissions is little higher than that of the direct-drive baseline.

**Electric drayage truck applications**

The first gear ratio for electric drayage truck applications is calculated at 3.2 to be able to climb grades higher than 15%. Furthermore, the final drive ratio cannot be larger than 4.8 to meet the top speed requirement of at least 60 miles per hour with 0-30 mph acceleration time less than 25 seconds.

*Electric drayage truck energy savings with three- and four- speed transmissions*

We evaluated energy savings with three-speed gearbox transmissions by simulating many combinations of second gear and final drive ratios and shift points with a third gear set to 1:1 direct drive, as illustrated in Figure 54.

The energy savings of electric drayage truck with four-speed transmission were evaluated by simulating many combinations of third gear and final drive ratios and shift points with the second gear ratio set at twice the value of the third gear ratio, while the fourth gear is always 1:1, as shown in Figure 55. Table 23 lists the energy savings (%) for three- and four-speed transmissions with optimized gear ratios and a final drive ratio of 4.8 compared to the direct-drive baseline electric drayage truck.

**Table 23. Energy savings (%) for optimized three-speed and four-speed transmissions drayage electric truck applications**

| Drive Cycle | Energy Savings (%) | | |
|---|---|---|---|
| | **Three-speed [3.2 1.8 1]** | **Four-speed [3.2 2.9 1.7 1]** | **Four-speed over three-speed** |
| Standard Cycle | 4.9 | 6.2 | 1.3 |

When the final ratio spread is kept at 3.2, we expect 4.9% and 6.2% energy savings for three-speed and four-speed transmissions over the direct-drive baseline electric drayage truck. Hence, the efficiency with four-speed transmission is 1.3% better than for a three-speed transmission.

*Electric drayage truck acceleration with three- and four-speed transmissions*

The vehicle accelerations time for three- and four-speed transmissions with optimized gear ratios and a final drive ratio of 4.8 are listed in All electric driveline configurations delivered 0-30 mph vehicle acceleration times less than 25 sec and met the acceleration requirements listed in Table 24. The 0-10 mph and 0-20 mph vehicle acceleration times were slightly reduced with four-speed transmission compared to the direct-drive baseline electric vehicle. However, the 0-30 mph acceleration times for three- and four-speed transmissions were higher than for the direct drive baseline drayage electric truck. Table 24 and compared to the direct-drive baseline. All electric driveline configurations delivered 0-30 mph vehicle acceleration times less than 25 sec and met the acceleration requirements. The 0-10 mph and 0-20 mph vehicle acceleration times were slightly reduced with four-speed transmission compared to the direct-drive baseline electric vehicle. However, the 0-30 mph acceleration times for three- and four-speed transmissions were higher than for the direct drive baseline drayage electric truck.

**Table 24. Vehicle acceleration times for three- and four- speed transmissions compared to the direct-drive baseline electric drayage truck application**

| | Vehicle Acceleration in Seconds | | |
|---|---|---|---|
| | **Direct drive baseline drayage electric truck** | **with three-speed [3.2 1.8 1]** | **with four-speed [3.2 2.9 1.7 1]** |
| 0-10 mph | 5.1 | 6.0 | 4.2 |
| 0-20 mph | 9.3 | 12.1 | 9.1 |
| 0-30 mph | 15.2 | 20.1 | 17.0 |

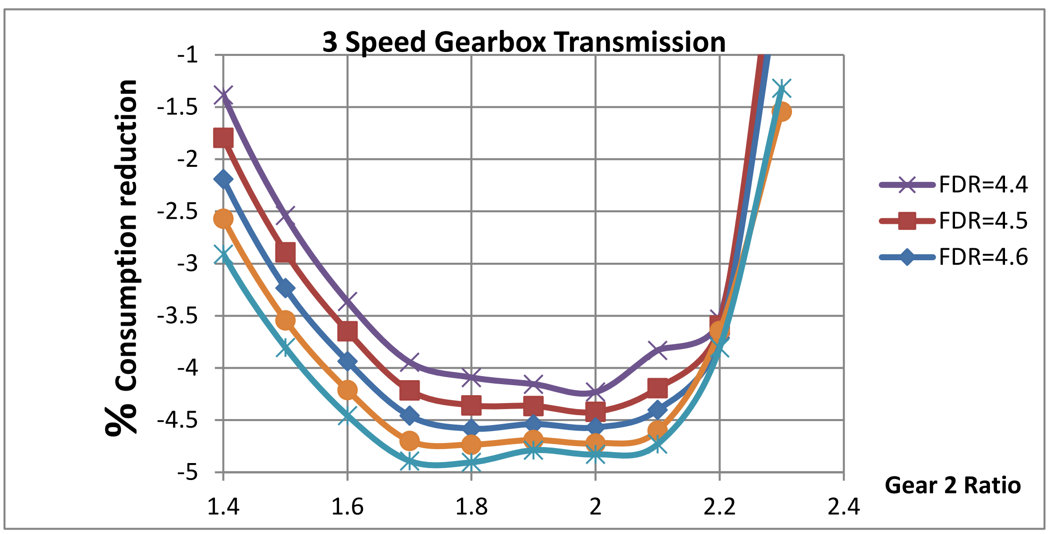



**Figure 54. Energy savings for three-speed transmission and downsized electric motors compared to the direct-drive baseline electric drayage truck with large motor.**

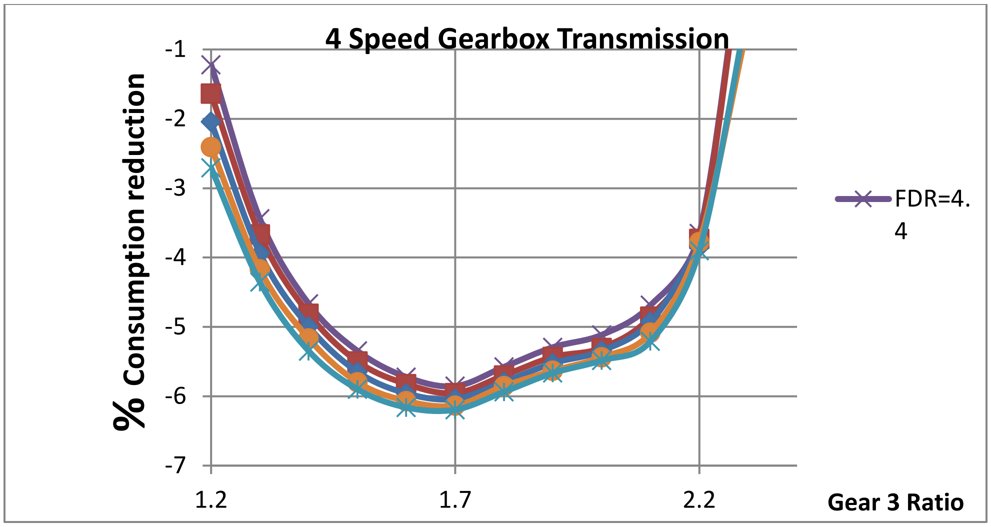



**Figure 55. Energy savings for four-speed transmission and downsized electric motors compared to the direct-drive baseline electric drayage truck with large motor.**

**Summary of extended modeling and simulations**

We investigated a wide range of EV applications to understand how many gears are desirable for each application and whether any application requires steps beyond a four-speed transmission. The modeling and simulations clearly showed that all electric vehicle applications can expect efficiency gains with multi-speed transmissions, as summarized in Table 25. Light-duty EV applications such as sedans, vans, and light-duty trucks can expect 9% efficiency gains with a two-speed transmission while three or more gears cannot be justified for these applications.

Most MD applications such as step van, cargo truck, or commercial delivery truck need a three-speed transmission to increase top speed, acceleration, and gradeability to gain 8% efficiency. Bus applications and heavy-duty trucks such as refuse, drayage, or yard trucks can expect up to 14% efficiency gains, depending on the application. A four-speed transmission provides up to 3% better efficiency over a three-speed transmission, but a five-speed transmission for these applications cannot be justified over a four-speed by the efficiency gains alone. The main benefit of a multi-speed transmission for drayage applications is to enable much faster acceleration and higher gradeability while still providing 6% efficiency gain.

Eaton’s modular multi-speed AMT family concept addresses the transmission needs of all EV applications by offering great flexibility in the number of gears and the gear ratio selections.

**Table 25. Multi speed transmissions and associated efficiency gains for all EV applications**

| Application | GVW (t) | Class | Recommended EV Trans. | Expected Efficiency gain |
|---|---|---|---|---|
| Passenger Car | 3 | LD | 2 speed | 9% |
| Step Van (Smith) | 10 | MD | 3 speed | 8% |
| School Bus | 16 | MHD | 4 speed | 8 % |
| City Bus | 18 | MHD | 4 speed | 11% |
| Refuse | 27 | HD | 4 speed | 14% |
| Drayage | 36 | HD | 4 speed | 6% |

### 3.4 SUBTASK 3.4 – DEVELOP A NEW VEHICLE OEM PARTNER TO SUPPORT AND IMPLEMENT TESTING PLAN

To find a new electric vehicle OEM partner after the departure of Smith Electric™ at the end of BP1, the team prepared a long list of electric vehicle manufacturers. After multiple reviews we selected five companies for interview. In the interviews, we sought answers to the following questions:

- Eaton has secured funding from the government to deliver a next generation EV transmission prototype. An integration partner would consist of selecting an electric motor, inverter, and battery supplier then working with each of the suppliers to ensure the respective components meet Eaton's communication specifications. Would you have any interest in becoming our integration partner on the program?
- Can you share any information regarding your EV strategy?
- Who currently supplies your EV system?
- Do you make a MD-EV Delivery Truck? If not, do you have a path to MD-EV?
- Can you provide an electric vehicle for Eaton transmission integration? Can you meet the technical requirements?
- Can you meet the project timing and budget requirements?

We performed a trade-off analysis on the EV-OEM partnership candidates based on the feedback from the interviews and identified the top two candidates. We signed two-way mutual non-disclosure agreements (NDAs) and discussed high-level work plans with each candidate company. In May 2016, the Eaton team visited both EV-OEM partner candidates to evaluate their resources and facilities up close.

Eaton decided to partner with Proterra for the vehicle integration of a new MD-EV transmission. Eaton and Proterra agreed on a statement of project objectives (SOPO), signed a contract, and presented the contract to the DoE for approval. Proterra offered the BE35 Electric Transit Bus, shown in Figure 56**,** as the baseline vehicle to integrate the multi-speed transmission and to implement the testing plan. The baseline vehicle has a two-speed Eaton transmission, which provides the necessary gradeability and acceleration performances. However, it requires an inefficient final drive with a high gear ratio, 9.8, and is not optimized for energy efficiency, size, and weight. Nevertheless, the two-speed transmission provides much better performance than the direct-drive and single-speed transmission alternatives that were the original project baselines.

Upon the selection of a new EV-OEM partner, we revisited the performance criteria evaluation and ranking with Proterra engineers. The team identified the top performance criteria as top speed, acceleration, range/efficiency, reliability, comfort/Noise Vibration and Harshness (NVH), gradeability, and torque disturbance. We then ranked the performance criteria for their importance to Proterra and the BE35 Electric Transit Bus application using the analytical hierarchy process shown in Table 26. The range/efficiency, gradeability, top speed, and reliability were identified as the top four performance criteria for the BE35 Electric Transit Bus application.

**Table 26. Ranking of importance of performance criteria for Proterra, BE35 Electric Bus**

| Criteria Number | Criteria name | Relative rank (sums up to 1) |
|---|---|---|
| 1 | Range (efficiency) | 0.40 |
| 2 | Gradeability | 0.18 |
| 3 | Top speed | 0.16 |
| 4 | Reliability | 0.10 |
| 5 | Acceleration | 0.06 |
| 6 | Torque disturbance | 0.06 |
| 7 | Comfort (NVH etc.) | 0.05 |

## 3.5 SUBTASK 3.5 – DETAILED TRANSMISSION MODELING ON THE ELECTRIC VEHICLE SELECTED FOR THE INTEGRATION OF PROTOTYPE TRANSMISSION

The BE35 electric bus has seating for 34 passengers, including the driver. Free floor space accommodates 27 standing passengers, resulting in a potential load of 61 persons. At 150lb per person, this load results in a measured gross vehicle weight of 37,530lbs. This vehicle is equipped with the two-speed Eaton transmission model EEV-7202 with gear ratios of 3.53 and 1. The final drive ratio of the baseline vehicle is 9.8. The power to Proterra BE35 electric bus is provided by a Proterra Fast Fill charging unit. Power is then supplied via Proterra model Terra Volt batteries (LTO-AltairNano-60AH) to a UQM Technologies Inc. PP220+, electric drive motor coupled to a two-speed transmission. The UQM P220+ is a permanent magnet synchronous motor with a peak/continuous power of 220/135 kW and maximum torque of 700 Nm at maximum motor speed of 6000 rpm.


**Figure 56. Proterra BE35 Electric Transit Bus equipped with two-speed Eaton transmission and a Lift-U-Model Lu-11-12-05 fold-out handicap ramp.**

The vehicle performance targets for the BE35 electric bus with the four-speed transmission as compared to the current two-speed transmission are listed in Table 27.

**Table 27. Proterra BE35 electric bus performance with baseline two-speed transmission and target performance with four-speed transmission**

| Metric | Current 2 speed AMT (Baseline) | New 4 speed AMT (Target) |
|---|---|---|
| Top vehicle speed @ GVW | 53 mph | > 65 mph |
| Acceleration time @ SLW<br>0 to 30 mph<br>30 to 50 mph | <br>15.5 s<br>27.5 s | <br>< 13 s<br>< 19 s |
| Gradeability @ GVW<br>10mph<br>20 mph | <br>15%<br>7% | <br>>20%<br>> 10% |

Proterra provided the BE35 electric transit bus vehicle specifications to create the model of baseline vehicle and run simulations on the MATLAB®/Simulink platform. We updated and calibrated the MATLAB/Simulink model created by Eaton team and used for previous EV applications based on the vehicle parameters and the test results provided by Proterra. Some of the important vehicle parameters and values are listed in Table 28.

**Table 28. Proterra BE35 Electric Transit Bus vehicle specification**

| | |
|---|---|
| **Tire Size (m)** | 0.9736 |
| **Aero Drag Coefficient** | 0.65 |
| **Frontal Area (m2)** | 6.7 |
| **Road Rolling Coefficient** | 0.00532 |
| **Weight (lb)**<br>**Gross Curb Weight (GCW)**<br>**Seated Load Weight (SLW)**<br>**Gross Vehicle Weight (GVW)** | <br>28200<br>33500<br>37530 |
| **Size (H" x L" x W")** | 132" x 426.5" x 103 |

**Validation of performance modeling results of baseline EV with two-speed AMT**

We used publicly-available field test results of BE35 Electric Bus to validate Eaton's MATLAB/Simulink EV model based on the vehicle acceleration performance [4]. The field tests consisted of three runs in both the clockwise and counterclockwise directions on a test track. Velocity versus time data obtained for each run and results were averaged to minimize test variability that might be introduced by wind or other external factors. The test was performed up to a maximum speed of 50 mph. From these tests, we calculated gradeability and the average acceleration time from 0 to 50 mph. The vehicle acceleration performance results obtained from the simulation of baseline EV with two-speed transmission are compared with the BE35 Electric Bus field test results as listed in Table 29.

**Table 29. Baseline model validation based on vehicle acceleration time for Proterra BE35 Electric Bus**

| Vehicle Speed (mph) | Average Acceleration Time (s) | | | |
|---|---|---|---|---|
| | Field Test Results | | | Simulation Results |
| | CCW Direction | CW Direction | Average | |
| 10 | 3.2 | 3.0 | 3.1 | 2.6 |
| 20 | 9.2 | 7.4 | 8.3 | 8.3 |
| 30 | 15.9 | 15.0 | 15.5 | 15.8 |
| 40 | 26.9 | 25.1 | 26.0 | 26.5 |
| 50 | 46.1 | 39.8 | 43.0 | 43.9 |

Acceleration performance simulation results of the baseline EV with two-speed transmission correlated very well with the actual vehicle acceleration performance results obtained from the field tests. Hence, the model is deemed to be very close to represent the real vehicle. However, we found a slight error for the first 10 mph acceleration which could be due to the restriction of motor torque at low speed that was not implemented in the baseline model.

**Validation of energy efficiency modeling results of baseline EV with two-speed AMT**

We also validated the energy efficiency and power losses of the baseline model with two-speed transmission with the Altoona ADB field test results. The Altoona ADB field test is a set of cycles for a certain number of miles in a fixed time and has three CBD phases, two arterial phases, and one commuter phase run in this order: CBD, arterial, CBD, arterial, CBD, and COMM. The CBD phase covers approximately two miles with seven stops per mile and a top speed of 20 mph; the arterial phase is approximately two miles with two stops per mile and a top speed of 40 mph; and the COMM phase is approximately four miles with one stop and a maximum speed of 40 mph; see Figure 57.

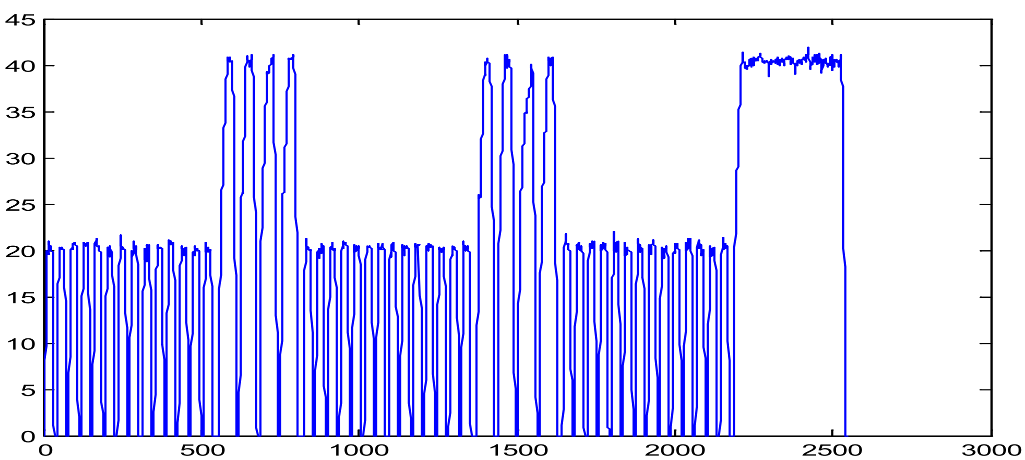



**Figure 57. The vehicle speed (mph) for the Altoona ADB drive cycle used to validate the model of baseline EV with two-speed transmission compared to the field tests.**

The simulation results of the energy efficiency obtained from the baseline model with two-speed transmission compared to the field test results are shown in Table 30.

**Table 30. Energy efficiency and power loss comparison of the model prediction versus field test results of baseline Proterra BE35 Electric Bus with two-speed transmission**

| Energy/losses | Altoona ADB Field Test | Model Test | Errors (%) |
|---|---|---|---|
| Energy efficiency (kWh/mile) | 1.8 | 1.9 | 3.0 |
| Overall Energy Consumption (kWh) | 48.5 | 49.5 | 1.8 |
| Electrical Regenerator Energy (kWh) | 13.6 | 13.9 | 2.7 |

Energy efficiency simulation results of the baseline EV with two-speed transmission correlated very well with the actual vehicle efficiency results obtained from the field tests. Hence, the model was deemed close enough to represent the real vehicle. The first row of Table 31 represents the real test data reported by Proterra for the two-speed baseline transmission and 9.8 final drive ratio. The top speed tested was 53 mph although the top speed capability of the vehicle is higher than 53 mph. Proterra limits the motor torque, especially at high speed   due to limitation of the baseline driveline and negative efficiency impact. The second row represents the simulation data after the artificial motor torque limit is removed.

Simulation results for Proterra with four-speed transmission are also presented in Table 31. The mpgde improvement is in 14% to 21% range depending on the cycle. Note that the vehicle mpgde for Baseline-1 and -2 are quite similar for all the cycles because the drive cycles do not have aggressive acceleration, deceleration, and high top speed; hence, for both baselines, most motor operating points are similar. The limits introduced by Proterra target to curb the real-world drivers with more aggressive acceleration and brake usage. We expected that four-speed transmission would ease limitations on the motor usage, promising percentage performance gains in between the improvements over Baseline-1 and Baseline-2 as listed in the bottom two rows of Table 31. Both baselines have a 3.5 step ratio, which makes it challenging to accelerate and upshift on the moderate grades (7% grade) due to gear hunting. For a four-speed transmission, the step ratio is in the rage of 1.6, and the vehicle can upshift easily on grades up to 10%.

**Table 31. Powertrain analysis for BE35 Electric Bus two-speed baselines with FDR 9.8 vs. four-speed AMT with FDR 6.2. Baseline is evaluated with and without limitations on the motor**

| | Electric Bus with: | Top Speed @ GVW (mph) | Acceleration Time @ SLW (s) | | Gradeability @ GVW (%) | | Vehicle Efficiency @ SLW (mpgde) | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0-30 mph | 30-50 mph | 10 mph | 20 mph | UDDS | OCC | NYC | ADB |
| Baseline 1 | Two-speed w limitations | 53 | 15.6 | 27.5 | 15 | 7 | 20.4 | 19.9 | 17.8 | 20.2 |
| Baseline 2 | Two-speed w/o limitation | 70 | 11.7 | 17.7 | 24 | 7.5 | 20.3 | 19.9 | 17.6 | 20.4 |
| Simulation | Four-speed | 81 | 9.7 | 16.6 | 24 | 10 | 23.2 | 23.5 | 20.5 | 24.4 |
| % Impr. over baseline 1 | | 52.8 | 37.4 | 39.6 | 33.3 | 42.9 | 13.7 | 18.1 | 15.8 | 20.8 |
| % Impr. over baseline 2 | | 15.7 | 17.1 | 6.1 | 0 | 33 | 14 | 18 | 16.4 | 19.7 |

The baseline modeling and simulations studies reported above considered the final drive ratio of 9.8 as in the current baseline Proterra Electric Busses with two-speed transmission. The same Proterra BE35 electric bus integrated with the new-four-speed transmission is also used to emulate the single-speed and two-speed baseline conditions for the efficiency and performance testing at NREL. The test vehicle integrated with four-speed transmission will have a final drive ratio of 6.2. Therefore, the baseline efficiency and performance predictions by modeling and simulations are extended to the single-speed and two-speed transmission cases with 6.2 final drive ratio as shown in Table 32. The extended modeling and simulation study reveal how much efficiency improvement is afforded by the four-speed transmission and how much by the change of final drive ratio.

Comparisons of the improvements over Baselines-2 and -3 show that the improvements over Baseline-3 is much smaller than the improvements over Baseline-2, the real baseline. Hence, the validation tests at NREL did not reflect the improvements over the real baseline. Table 32 indicates how much additional performance gains were expected on top of the performance gains observed at the validation tests NREL, if the real baseline vehicle with FDR 9.8 had been tested.

**Table 32. Powertrain analysis for Proterra-BE35 Electric Bus two-speed baselines with FDR of 9.8 and 6.2 and one-speed baseline with FDR 6.2 versus four-speed AMT with FDR 6.2 without limitations on the motor**

| | Electric Bus with: | Top Speed @ GVW (mph) | Acceleration Time @ SLW (s) | | Gradeability @ GVW (%) | | Vehicle Efficiency @ SLW (mpgde) | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0-30 mph | 30-50 mph | 10 mph | 20 mph | UDDS | OCC | NYC | ADB |
| **Baseline 2** | Two-speed w FDR 9.8 | 70 | 11.7 | 17.7 | 19.5 | 7.5 | 20.3 | 19.9 | 17.6 | 20.4 |
| **Baseline 3** | Two-speed w FDR 6.2 | 68 | 10.8 | 16.4 | 19.5 | 9.0 | 23.1 | 22.0 | 19.0 | 23.7 |
| **Baseline 4** | One-speed w FDR 6.2 | 44 | 11.7 | Does not reach 50 | 12.0 | 9.0 | na | 19.8 | 17.3 | 20.8 |
| **Simulation** | Four-speed w FDR 6.2 | 81 | 9.7 | 16.6 | 19.5 | 9.8 | 22.9 | 23.6 | 20.8 | 24.5 |
| % Impr. 4-spd over B2 | | 15.7 | 17.1 | 6.1 | 0 | 31 | 12.8 | 18.6 | 18.2 | 20.0 |
| % Impr. 4-spd over B3 | | 19.1 | 10.2 | -1.2 | 0 | 9 | -0.9 | 7.2 | 9.4 | 3.3 |
| % Impr. 4-spd over B4 | | 84.1 | 17.1 | enabler | 62.5 | 9 | enabler | 19.1 | 20.2 | 17.8 |

For example, the four-speed transmission provides 18% and 16.4% improvements in efficiency compared to the baseline two-speed transmission in OCC and NYC cycles. However, more than half of the improvement (11.2% and 8.5% for OCC and NYC resp.) is attributable to enabling the selection of 6.2 final drive ratio with the four-speed transmission. Hence, the efficiency gains measured at NREL validation tests had to be multiplied by 2 to estimate the efficiency gains compared to the real baseline vehicle with two-speed transmission and FDR of 9.8.

Table 32 also clearly shows the advantage of four-speed transmission over the Baseline-4, a single speed gearbox or a direct drive electric bus. Four-speed transmission not only provides significant improvements in top speed, gradeability, and efficiency but also is an enabler for 30-50 mph acceleration and UDDS drive cycle that are off-limits to an electric bus with a single speed gearbox. An electric bus with one-speed transmission has a very low top speed, 44 mph, and cannot climb a grade higher than 12% at 10 mph while the EV bus with four-speed transmission has top speed of 81 mph and climbs up to 24% grade at 10 mph. Furthermore, an EV bus with one-speed transmission cannot even follow the UDDS route while an EV bus with four-speed transmission does not have any problems following any of the standard routes. Figure 58 compares the energy efficiency of one-speed, two-speed, and four-speed powertrain configurations in terms of miles per gallon, diesel equivalent for various duty cycles. The hypothetical two-speed transmission and a final drive ratio of 6.2 (green bar) performs better than the baseline two-speed transmission and the final drive ratio of 9.8 (blue bar) but worse than the four-speed transmission and a final drive ratio of 6.2 (orange bar).

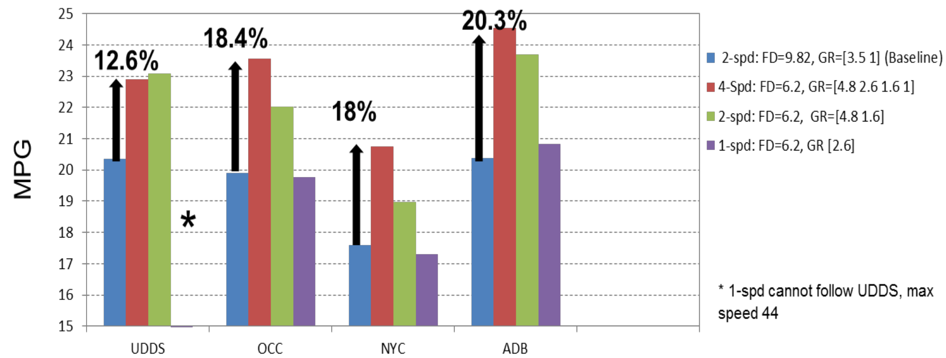



**Figure 58. Energy efficiency (mpgde) comparison of the electric bus with different powertrain configurations for various drive cycles. Four-speed transmission improves the efficiency by 20.3% as compared to the two-speed baseline.**

## 3.6 SUBTASK 3.6 – ADAPTIVE DOWNSHIFT CONTROL STRATEGY

We developed an adaptive downshift control strategy to maximize energy recovery during vehicle braking as part of the supervisory controls strategy. In a conventional vehicle, the mechanical friction brakes convert the vehicle kinetic energy into heat during braking. In electric vehicles with regenerative braking systems, some portion of kinetic energy can be recovered by operating the electric machine in the generator mode and storing the regenerated electric energy in the battery. The electric motor can then use the recovered energy and improve electric vehicle energy efficiency.

As the vehicle speed decreases during a braking event, motor speed decreases as well, and at some point, downshift must happen. The downshifting prevents the electric motor from operating at low efficiency and below the base-speed where the power is limited. For the electric machine shown in Figure 59 electric machine power curve is based on speed; when the electric machine operates as a generator, the base speed is 2500 rpm and the maximum power that can potentially be captured monotonically decreases to zero.

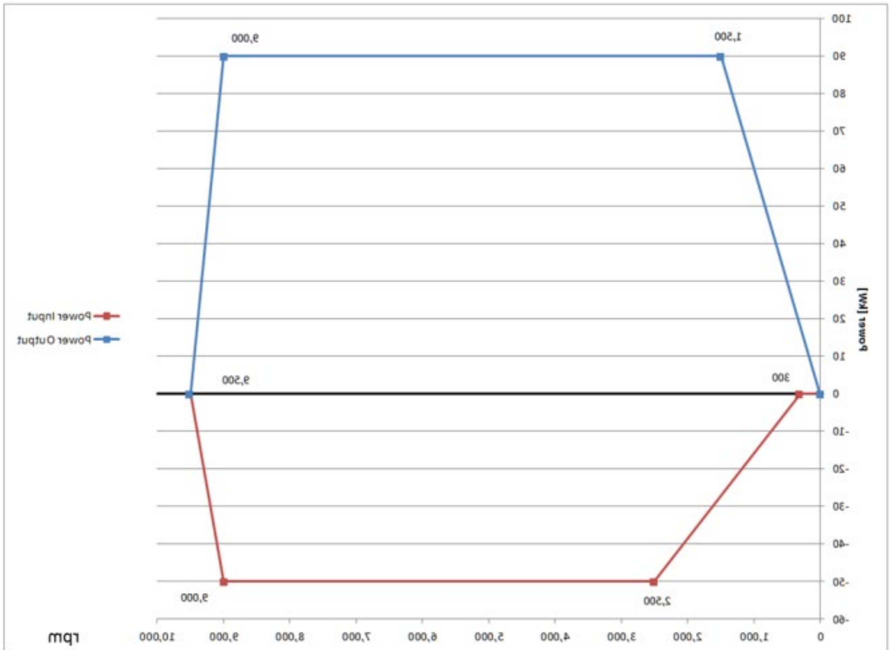


**Figure 59. Electric machine power curve based on speed.**

Thus, we need to prevent operation below the base speed to obtain the highest regenerative braking energy capture; this goal is achieved by downshifting. However, during downshifting, the electric machine is disconnected from the driveline and the regenerative breaking energy recovery opportunity is lost for this period. This tradeoff is captured in Figure 60, which shows captured brake energy for each braking event and for different downshift strategies, assuming downshift would take 0.9 second. We expect that downshifting at 2065 rpm, which is close to the base speed, would result in the best energy capture; however, due to the regen interrupt during shifting, there are times when a 65 rpm downshift (well below the base speed, which essentially means holding the gear) is better for any braking event below 2500 rpm.

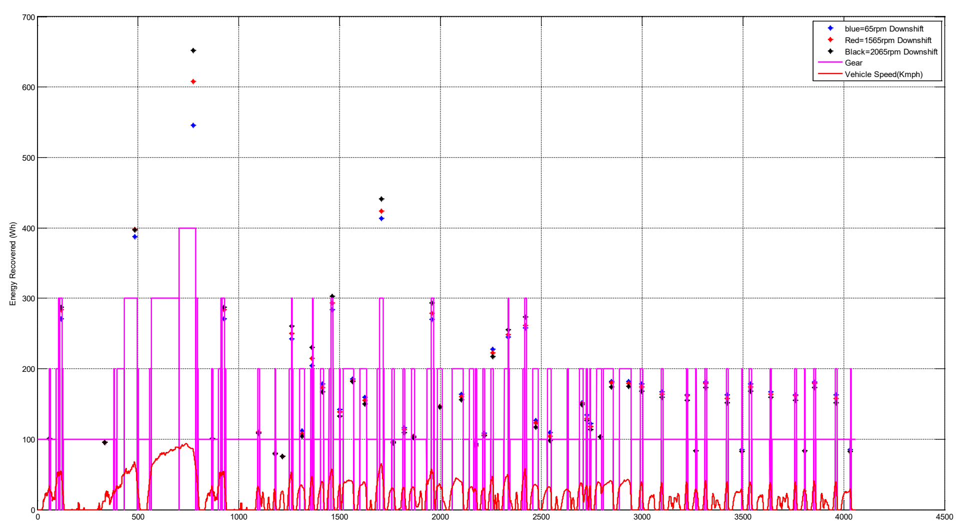



**Figure 60. Captured brake energy for combined drive cycle for 3 different downshift strategies.**

The goal of the adaptive downshift strategy is to identify the best downshift speed as a function of vehicle speed, brake duration, energy, gear number, and so on. The simplest strategy is to downshift at 2065 rpm when the vehicle is in third or fourth gear and at 65 rpm when the vehicle is in first or second gear during the brake. With this strategy, the highest braking energies can be captured, as shown in Figure 60.

Table 33 lists the overall energy savings (% kWh/mile) for all driving cycles with the new four-speed transmission over the existing two-speed baseline transmission using 0.9 s and 0.5 s gear shift delay times. The real delay time is a function of the final control strategy as well as transmission mechanical design, but it is in the range considered above. The table shows that when the shifting delay is 0.9, the adaptive strategy shows more improvement, since it prevented or delayed downshifts due to their severe effect on capturing braking energy. For all combined cycles, up to 0.95% additional efficiency gain can be achieved with a smart downshifting strategy.

**Table 33. Energy savings (%) with two gear shift times (0.9 s & 0.5 s) used for Proterra BE35 electric bus application with the new four-speed transmission with and without adaptive control**

| Gear Shift Time (s) | Drive Cycle | Top Speed (mph) | % energy savings | | |
|---|---|---|---|---|---|
| | | | Without Adaptive Control | With Adaptive Control | Adaptive versus no Adaptive Control |
| 0.9 | UDD | 58 | 12.24 | 12.26 | + 0.02 |
| | OCC | 41 | 15.25 | 16.62 | + 1.48 |
| | NYC | 25 | 13.67 | 15.58 | + 1.91 |
| 0.5 | UDD | 58 | 12.77 | 12.72 | +0.05 |
| | OCC | 41 | 16.79 | 17.19 | +0.39 |
| | NYC | 25 | 14.86 | 15.51 | +0.65 |

## 4. Powertrain Integrations and Installation for Hardware-in-the-Loop (HIL)

Eaton provided one of the prototype transmission units to ORNL for integration and installation in the HIL testing setup. Eaton provided the four-speed transmission, wiring harnesses, electronic controls unit, and the controls software. Proterra provided the electric motor, the inverter, and the motor control unit to ORNL for the HIL powertrain testing.

**Deliverables:** Integrated powertrain HIL testing and initial vehicle shake-down testing report

### 4.1 SUBTASK 4.1 – MECHANICAL INSTALLATION OF MOTOR, MCU, TRANSMISSION, TCU AT THE ORNL-VSI LABORATORY

ORNL adapted the prototype four-speed transmission unit to be operated in ORNL's test facility. ORNL received the following components from Eaton and Proterra:

- Four-speed gear box
- UQM electric machine and inverter
- Transmission control unit (TCU)
- Motor control unit (MCU)
- Electric vehicle control module (EVCM)
- EMP coolant pump

At ORNL's Vehicle System Integration (VSI) Laboratory Powertrain test facility the integration of the mechanical system into laboratory began by designing and constructing a frame with the same dimensions and rubber isolators found in the Proterra test vehicle. A driveshaft was fabricated specifically for lab testing but retained the test vehicle length for proper inertia emulation, as seen in Figure 61. The system was then mated with the VSI Lab's low inertia AC dynamometer. After the mechanical installation was designed and fabricated, the electrical and cooling installations were tackled in parallel. The cooling system consisted of two heat exchangers running in parallel, each with an electrically controlled throttling valve to control temperatures consistently. The high voltage system was wired to the VSI Lab's 400 kW Battery Emulator. Between the inverter and battery emulator, a 600-amp DC slow fuse protected both the powertrain and the battery emulator from possible failures, as seen in Figure 62. The DC side of the electrical system was then instrumented with a Hioki Power Analyzer at the battery emulator and the leads into the inverter for proper energy consumption measurements.

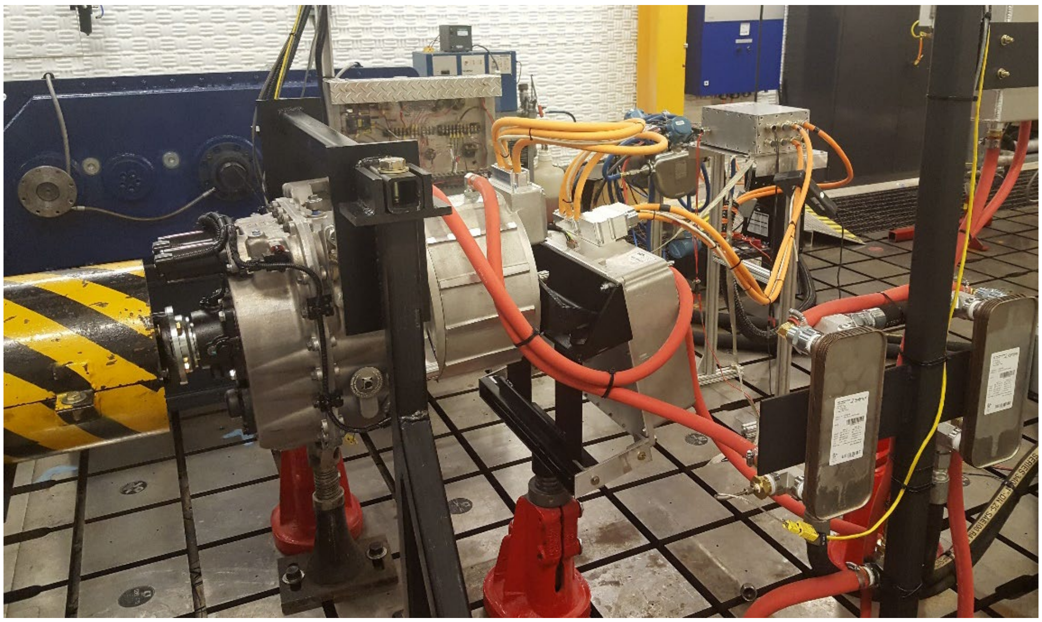


**Figure 61. Electric powertrain system fully installed at ORNL's VSI Powertrain Test Facility.**

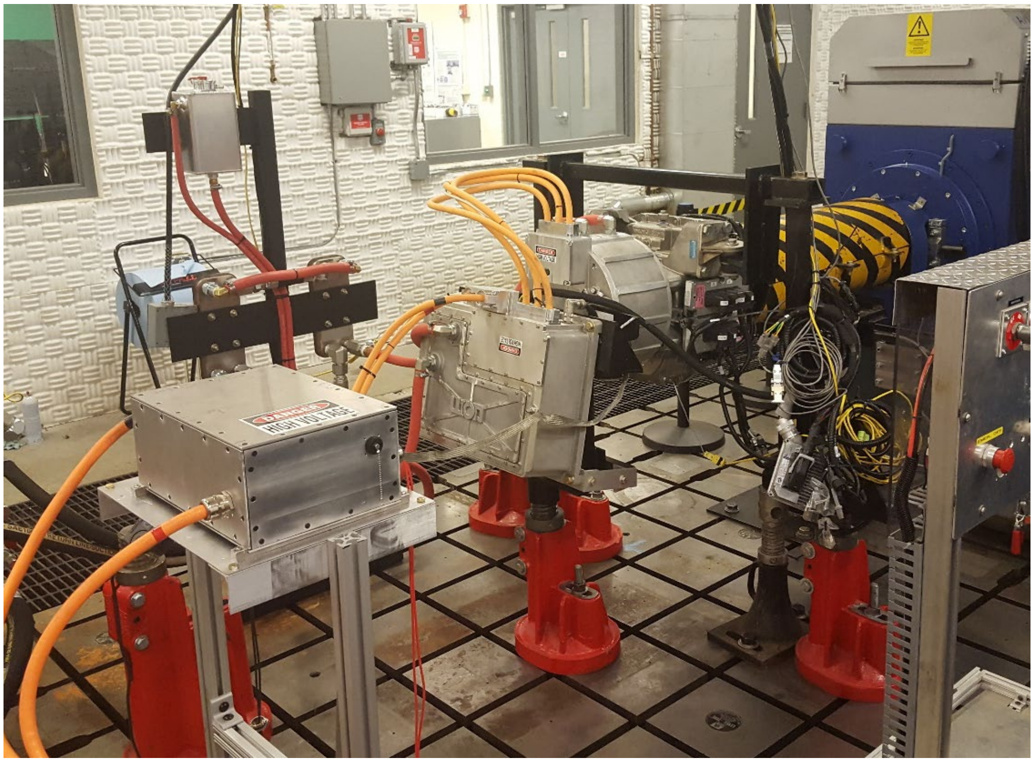



**Figure 62. The high voltage equipment installed at ORNL's VSI Powertrain Facility.**

## 4.2 SUBTASK 4.2 – HIL ENVIRONMENT INITIALIZATION AND DEBUGGING

The vehicle model used in Tasks 1 and 3 was integrated into the HIL platform and performed as expected with the powertrain in the test cell. This setup allowed ORNL to test the powertrain as if it were on the road by emulating the entire vehicle minus the powertrain.

For HIL development and testing, a portion of the vehicle was modeled in real-time and the rest of the components were real and under operation in loop with the actuators like the battery emulator and dynos. This system was controlled based on a physics-based model (Figure 63), so the operation is very representative of on-road operation. For this project, the components listed above were real and in-the-loop, while the rest of the Proterra bus was modeled in Tasks 2 and 3 based on values provided by Eaton and Proterra. Once the framework for the HIL setup was built, ORNL built the HIL interface and automation for testing to allow quick and repeatable testing iterations (Figure 64).

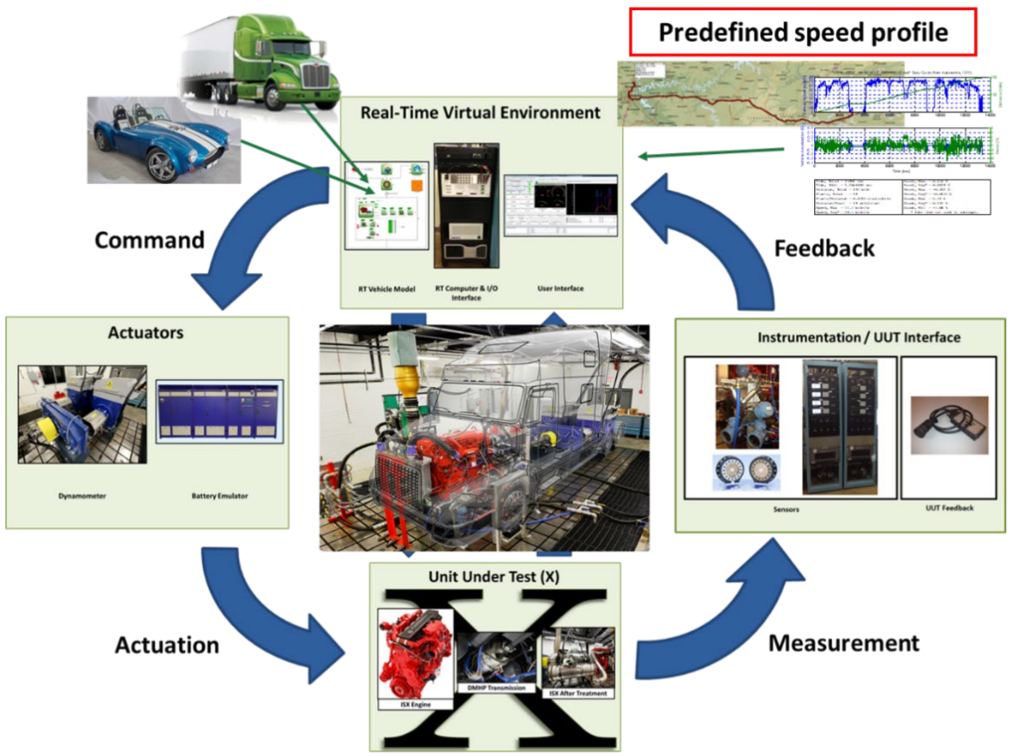



**Figure 63. HIL flowchart.**

Eaton and Proterra were on site for the commissioning and validation of proper operation in the test cell. All parties performed the necessary checks and fits for the start of testing. System integration was the primary task during the initial period at ORNL. Signals were properly exchanged between the Eaton EVCM and the Proterra ECU. The Eaton transmission was able to request a behavior from the motor (torque request), command the motor to generate that request (torque), and properly monitor the signals from the motor and vehicle, some through vehicle controller area network (CAN), and some through proprietary CAN (PNL), directly between the motor and transmission. Once all parties were confident in proper realistic vehicle operation, the team moved on to verification of model vs HIL testing.

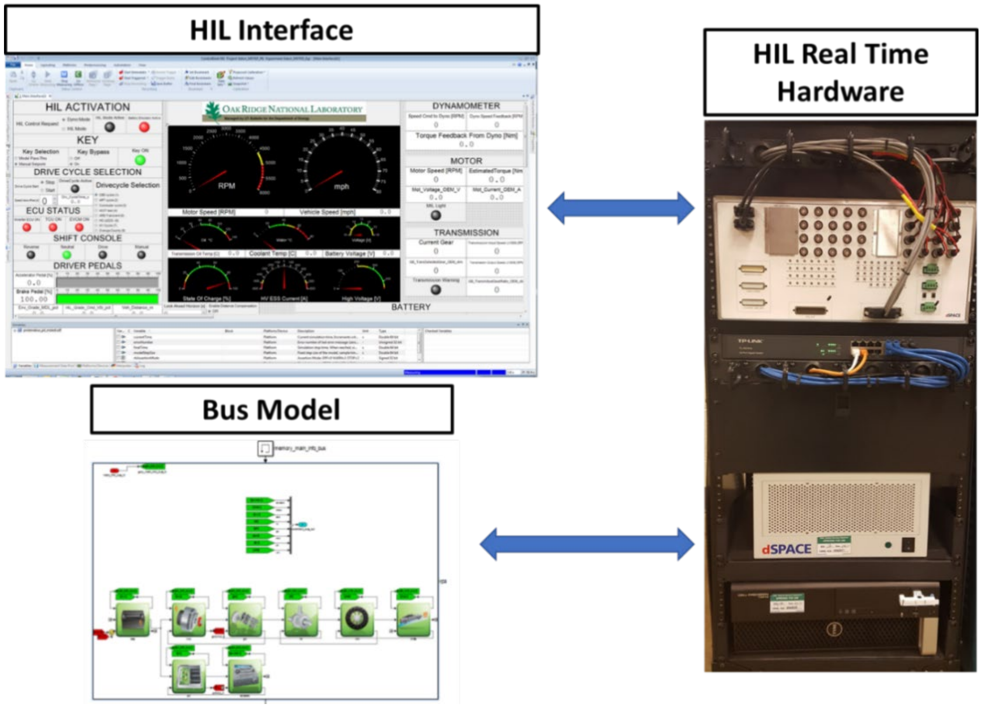



**Figure 64. HIL layout and integration for powertrain testing**

## 4.3 SUBTASK 4.3 – PRELIMINARY FUNCTIONAL VERIFICATION TESTING

As a check for the HIL environment and testing, ORNL compared the signal traces of vehicle speed, motor speed, and motor torque with that of the model simulation. With the hardware installed and integrated into the test cell, and the HIL interface up and running with the vehicle model, the setup was exercised over basic drive cycles to check functionality and verify that the system properly emulates the real vehicle when utilizing the real-time platform.  This helped identify any issues with the HIL environment and validate the model. Figure 65 shows one of the comparison graphs with vehicle and motor values.

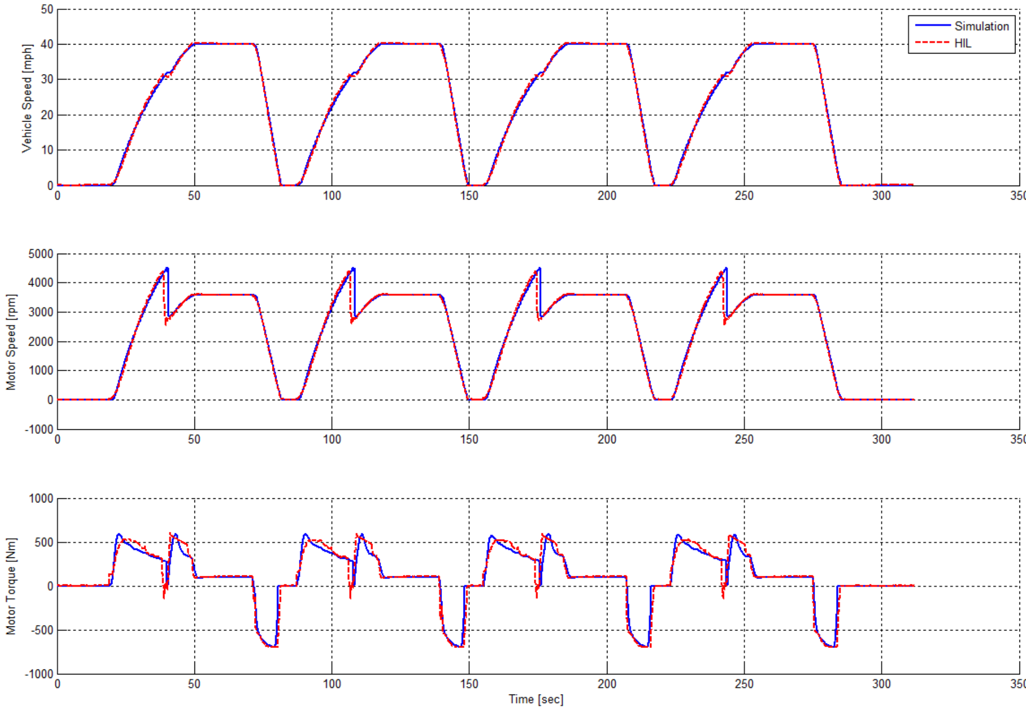



**Figure 65. Comparing simulation results (blue) to HIL tests on the real hardware (red).**

## 5. Integrated Powertrain HIL Testing

Eaton provided one prototype of the new four-speed transmission with the transmission controller unit (TCU), and Proterra provided the electric motor, inverter, and EV controller to ORNL for the HIL testing activities. The final drive ratio (FDR) was set to 6.2 on the HIL test setup to emulate the new final drive axle used with the four-speed transmission as implemented on the Proterra BE-35 Electric Bus. The original baseline Proterra BE-35 Electric Bus with the original Eaton two-speed transmission and the original drive axle had a final drive ratio of 9.8 that has 3% lower efficiency on average than the efficiency of a drive with 6.2 ratio. Any improvement in the efficiency of final drive results in even greater improvement in the efficiency of drive cycles for electric vehicles due to the added effect of the regenerative braking energy on the motor efficiency, literally doubling the efficiency gain to 6%.

Changes in controls algorithm make it possible to operate the four-speed transmission in two-speed and one-speed modes. The baseline tests used the four-speed transmission in single gear mode on third gear and in two-speed mode on first and third gears for the duration of tests. This setup enables comparison of energy efficiency of each mode of operation. However, excluding the impact of the change of final drive ratio enabled by the four-speed transmission meant that the emulated baseline tests do not capture the entire energy savings between the new driveline configuration of four-speed transmission and FDR-6.2 and the real baseline configuration of two-speed transmission and FDR-9.8. Hence, the real energy efficiency gains are 5% to 6% higher than the values measured at the HIL tests and described below. The real baseline vehicle configuration with two-speed transmission and FDR 9.8 was tested on Altoona test track in 2013 [2]. The team attempted to compare the HIL test results of four-speed transmission and FDR 6.2 powertrain configuration to the Altoona test track results of the real baseline powertrain configuration as described at the end of this task.

**Deliverables:** Integrated powertrain HIL testing and initial vehicle shake-down testing report

## 5.1 SUBTASK 5.1 – SHIFT STRATEGY VALIDATION AND TUNING

Initial shift strategies and tuning of the shifts of four-speed transmission were done on the test bench at Eaton-Southfield. After integrating the four-speed transmission on Proterra BE35 electric bus, Eaton refined the shift strategies and tuning at its proving grounds in Marshall, Michigan. Eaton provided the baseline shift strategies and tuning as they were developed and updated to ORNL. In return, ORNL provided steady state maps and transient data from the HIL testing environment for Eaton to further optimize shifts.

## 5.2 SUBTASK 5.2 – STEADY STATE HIL TESTS

Steady state HIL tests were conducted at ORNL. The test data were processed and reported to Eaton. The full powertrain system was mapped by taking each gear in manual selection mode from minimum motor speed (500 rpm) to maximum motor speed (5000 rpm). The motor was then operated from 0% torque to full torque in 10% increments. This operation allowed ORNL to measure the total electrical energy input and mechanical energy output to calculate total system efficiency maps that are shown in Figure 66.

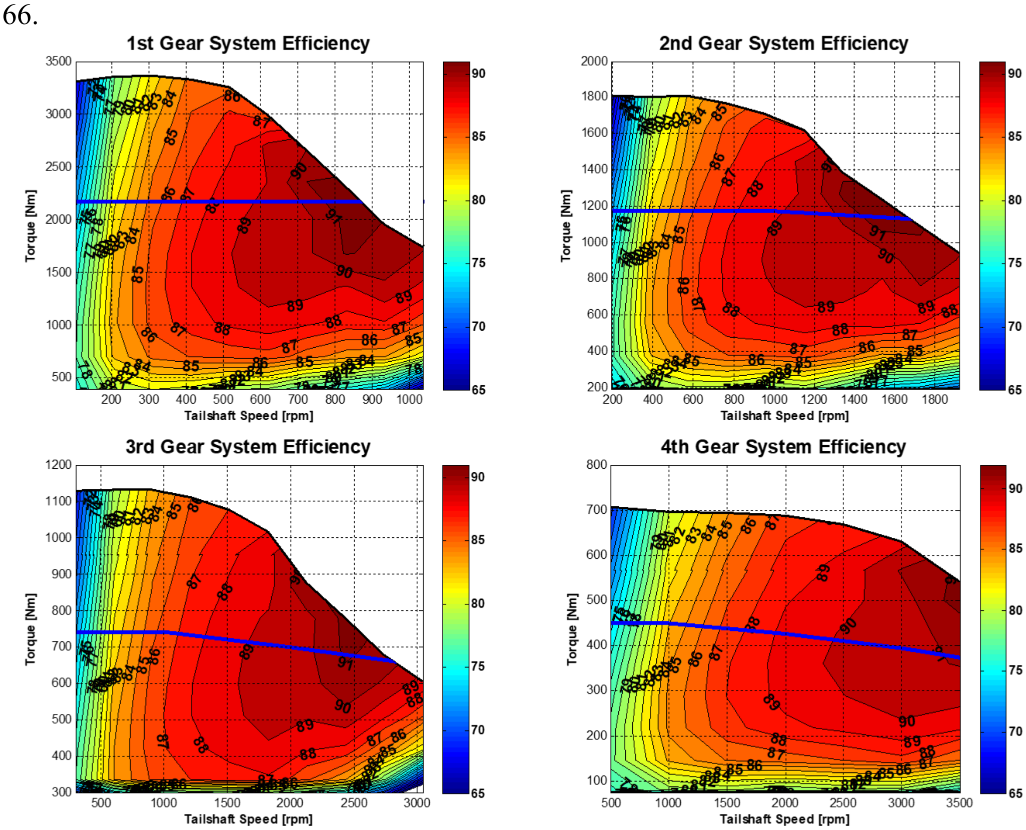



**Figure 66. System efficiency maps for each powertrain gear.**
**Blue line denotes the constant power curve of the UQM motor at that given gear.**

## 5.3 SUBTASK 5.3 – ACCELERATION AND GRADEABILITY TESTING

Before testing with drive cycles could begin, ORNL tested acceleration and gradeability. An important part of this project was to demonstrate better acceleration and gradeability metrics than a conventional electric powertrain. **Table 34** to **Table 37** show the results of the vehicle both at seated load weight (SLW) of 33,500 lb. and gross vehicle weight (GVW) of 37,530 lb.

**Table 34. 0-50 mph acceleration results at seated load weight**

| | HIL Acceleration Time (s) at SLW | | |
|---|---|---|---|
| **Operating Mode** | **0-30 mph** | **30-50 mph** | **0-50 mph** |
| Four-speed, 2nd gear launch | 12.2 | 17.9 | 30.1 |
| Two-speed | 18.4 | 16.3 | 34.7 |
| % Improvement | 33.8% | -9.8% | 13.3% |

While **Table 34** shows a decrease in 30-50 mph acceleration, the system overall is much improved over the Proterra two-speed variant emulated by the system's two-speed mode. It showed a 33.8% improvement of 0-30 mph, which is very important for a city bus application. A shift at around 30-40 mph explains the loss of the 30-50 mph acceleration time.

**Table 35. Maximum achieved vehicle speed on a grade at SLW**

| | HIL Gradeability Results at SLW<br>Max speed (mph) at grades | |
|---|---|---|
| **Operating Mode** | **10% Grade** | **20% Grade** |
| Four-speed | 25.9 | 13.9 |
| Two-speed | 18.4 | 13.9 |
| % Improvement | 40.8% | 0.0% |

**Table 36. Maximum achieved vehicle speed on a grade at GVW**

| | HIL Gradeability Results at GVW<br>Max speed (mph) at grades | |
|---|---|---|
| **Operating Mode** | **10% Grade** | **20% Grade** |
| Four-speed | 23.1 | 12.4 |
| Two-speed | 18.4 | 12.4 |
| % Improvement | 25.5% | 0.0% |

**Table 37. Maximum achievable grade at 10mph and 20mph at GVW**

| | HIL Gradeability Results<br>Max grade (%) at speeds | |
|---|---|---|
| **Operating Mode** | **at 10 mph** | **at 20 mph** |
| Four-speed | 22.7 | 11.2 |
| Two-speed | 22.7 | 7.3 |
| % Improvement | 0.0% | 53.4% |

**Table 35** to **Table 37** show improvement across the board under different operating conditions. The two-speed mode operation and four-speed mode operation achieve the same results at points that both modes share the same first gear ratio. Ultimately, the multi-speed transmission has up to a 53.4% gradeability improvement at higher speeds thanks to the addition of two gears, as compared to the baseline two-speed transmission.

## 5.4 SUBTASK 5.4 – TRANSIENT HIL TESTS WITH DRIVE CYCLES

To compare modeling and simulation, HIL, and chassis vehicle testing, we created a common test matrix to test operating modes in a variety drive cycles, as listed in **Table 38**. The drive cycles related to the transit bus applications included:

- Central Business District cycle (CBD)
- Arterial cycle (ART)
- Commuter Cycle (CC)
- Heavy-duty Air Resource Board cycle (ARB)
- Heavy-duty Urban Dynamometer Drive cycle (HD UDDS)
- Manhattan Bus Cycle
- Orange County Cycle - Bus (OCC Bus)

**Table 38. Test matrix for simulation, HIL, and vehicle chassis testing. Single speed configuration cannot follow the UDDS cycle**

| Operating Modes (Gears) | CBD | ART | Commuter | ARB | HD UDDS | Manhattan Bus | OCC Bus |
|---|---|---|---|---|---|---|---|
| Four-speed (Start in 2nd) | x | x | x | x | x | x | x |
| Four-speed (Start in 1st) | x | x | x | x | x | x | x |
| Two-speed (1st and 3rd) | x | x | x | x | x | x | x |
| Single Speed (3rd Gear) | x | x | x | x | | x | x |
| Single Speed w/ Dist Comp (3rd) | x | x | x | x | | x | x |

The test matrix covers multiple operating modes to explore energy consumption improvements in one- and two-speed operating modes over four-speed operation. Each combination was run at least three times in the VSI Lab, after which the coefficient of variation (COV) was calculated and reported with the energy consumption information on **Table 39**. When the COV of energy consumption either in Wh/mile or mpgde was higher than 0.25%, the tests were run twice more (5 in total) to try to identify outliers.

**Table 39. Testing statistics for the Orange County Bus Cycle**

| | | Actual Distance | Energy Consumption | | | Pos Energy | Neg Energy | Mech Energy |
|---|---|---|---|---|---|---|---|---|
| **Test File** | **Cycle** | **Miles** | **kWh** | **Wh/Mile** | **mpgde** | **kWh** | **kWh** | **kWh** |
| 20170829_019 | OCC | 6.58 | 8.99 | 1366.1 | 27.52 | 15.20 | -6.21 | 5.82 |
| 20170829_020 | OCC | 6.58 | 8.99 | 1364.9 | 27.55 | 15.21 | -6.21 | 5.82 |
| 20170831_001 | OCC | 6.58 | 9.00 | 1366.4 | 27.52 | 15.22 | -6.23 | 5.80 |
| | Average | 6.58 | 8.99 | 1365.78 | 27.53 | 15.21 | -6.22 | 5.81 |
| | COV | 0.01% | 0.05% | 0.06% | 0.06% | 0.09% | -0.17% | 0.14% |

After all vehicle configurations were run, we determined the % reduction in mpgde and overall reduction in energy consumed from one operating mode to another. Each test was broken-down to determine what variable has the largest impact or variation from one operating mode to another. Test averages for all modes of operation were broken down into energy consumption and then further broken into the positive electrical (tractive) energy, negative electrical (regen) energy and mechanical energy output of the tail shaft of the transmission as shown in **Table 40**. This process allows for insight into possible future powertrain improvements or shift strategies/system optimizations that could further enhance the efficiency of the system and its operation. A four-speed transmission launching on first gear provides 14.4% and 15.4% less energy consumption in mpgde compared to one-speed and two-speed modes respectively. An additional 5 to 6% efficiency gain must be added to these efficiency gains due to the change of FDR from 9.8 to 6.2 on the real baseline vehicle.

**Table 40. Example of cycle break downs for each test cycle**

| | | Actual Distance | Energy Consumption | | | Pos Elec Energy | Neg Elec Energy | Mech Energy Output |
|---|---|---|---|---|---|---|---|---|
| **Test File** | **Cycle** | **miles** | **kWh** | **Wh/mile** | **mpgde** | **kWh** | **kWh** | **kWh** |
| One-speed mode | Manhattan | 2.07 | 3.61 | 1741.0 | 21.6 | 5.72 | -2.11 | 1.84 |
| Two-speed mode | Manhattan | 2.06 | 3.61 | 1754.7 | 21.4 | 5.65 | -2.04 | 2.29 |
| Four-speed mode 1st gear start | Manhattan | 2.06 | 3.14 | 1521.0 | 24.7 | 5.72 | -2.59 | 1.79 |
| 4- speed mode 2nd gear start | Manhattan | 2.06 | 3.30 | 1600.3 | 23.5 | 5.65 | -2.34 | 1.98 |
| 4spd1 improve over 1spd3 | % Diff | -0.48% | 13.0% | 12.6% | 14.4% | 0% | 22.7% | -2.7% |
| 4spd1 improve over 2spd | % Diff | -0.10% | 13.3% | 13.3% | 15.4% | -1.3% | 27.1% | -21.8% |
| 4spd2 improve over 1spd3 | % Diff | -0.48% | 8.6% | 8.1% | 8.8% | 1.2% | 10.9% | 7.6% |
| 4spd2 improve over 2spd | % Diff | -0.17% | 8.6% | 8.8% | 9.7% | 0.1% | 15.1% | -13.8% |
| Two-speed mode vs one-speed mode | % Diff | 0.74% | 0.05% | 0.8% | -0.8% | 1.3% | 3.6% | 19.7% |

Other testing performed in parallel to the tests above, was to examine the differences between the ORNL HIL/simulation and the Eaton simulations to discover where they deviate from each other. The team examined the behaviors in two ways. The first method was to analyze the signal traces of the simulated variables vs recorded, as seen in **Figure 67** to **Figure 69**. We could then determine where the model misses some of the realistic or dynamic behaviors that are found in actual hardware interactions with each system component or the vehicle environment. With the second method, we examined heat maps of areas of operation for the powertrain (**Figure 70**). This examination helped the team to quickly tell if the model and real HIL tested powertrain were operating in the same regions. It also made it possible to tell whether system efficiencies are different in the model vs the real powertrain.

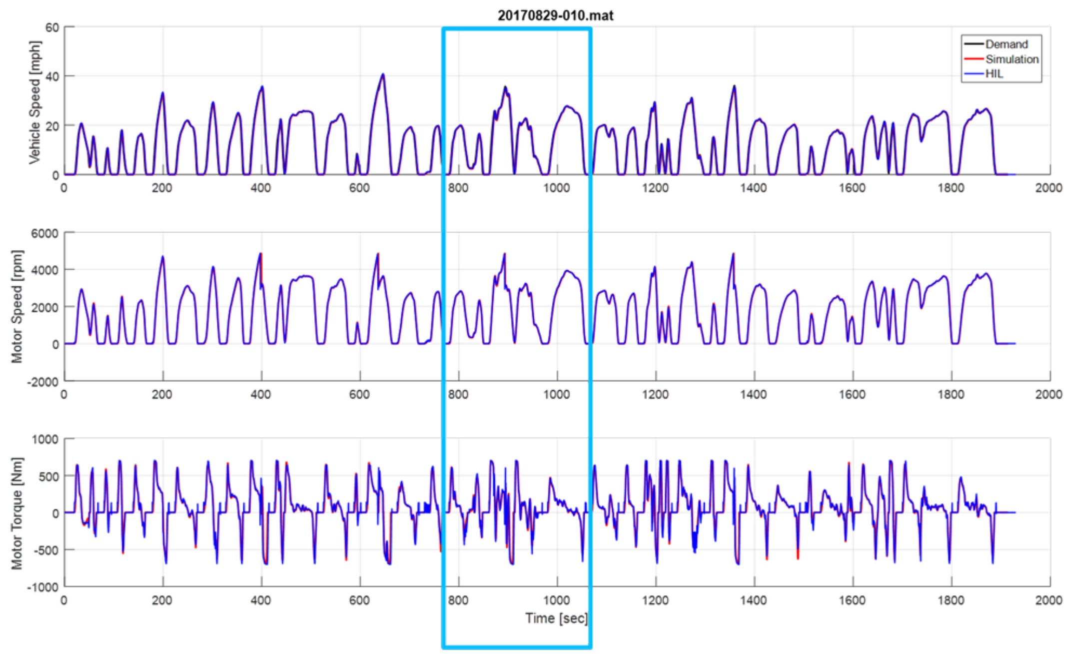



**Figure 67. A single Orange County Bus Cycle run on the HIL system vs the simulated vehicle.**

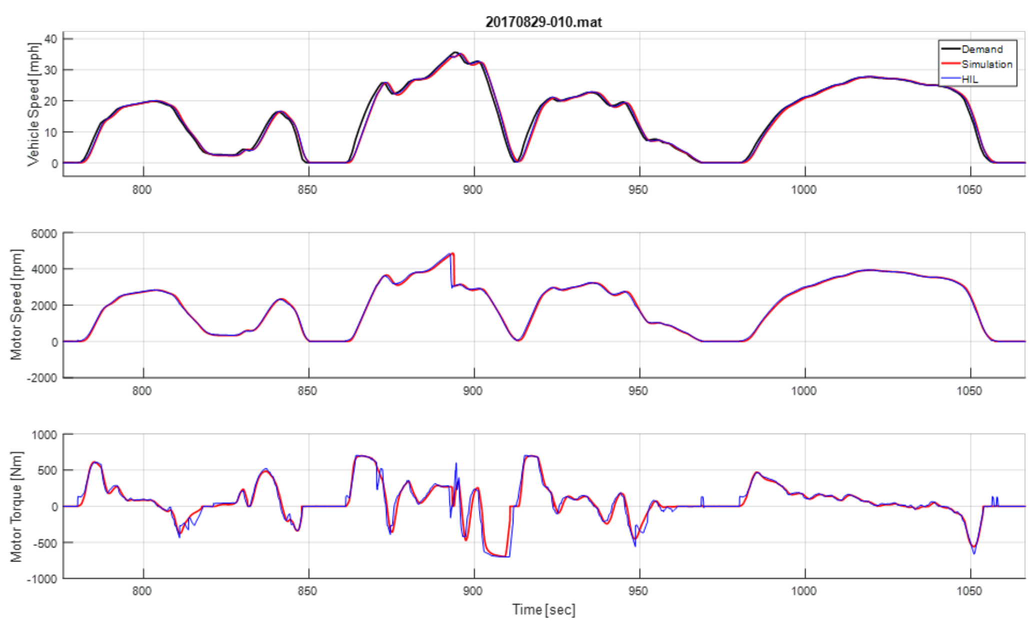



**Figure 68. Enlargement of boxed area showing motor values for HIL system vs simulated vehicle.**

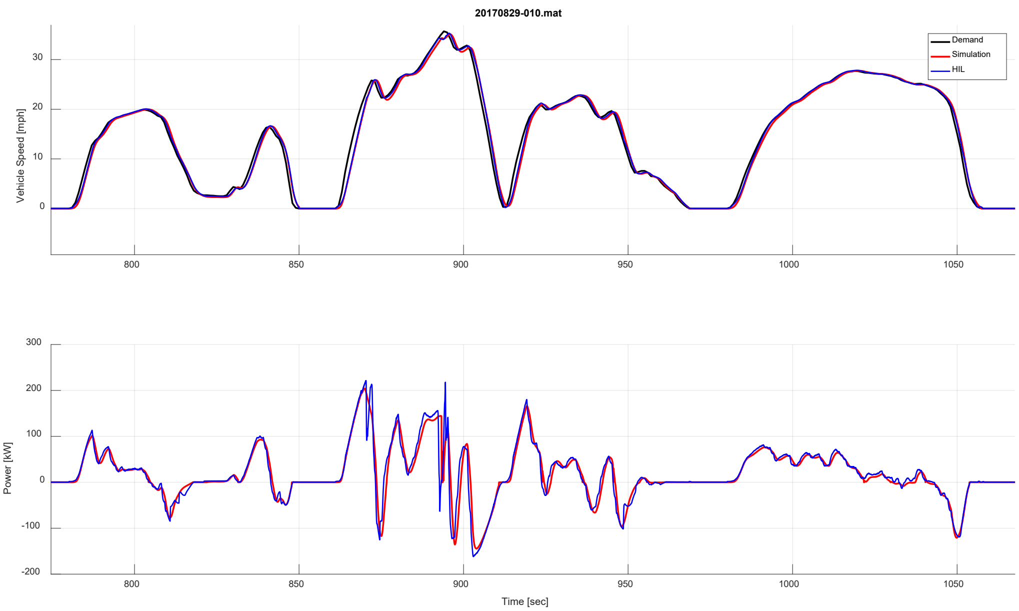



**Figure 69. Enlargement of boxed area showing electric power for HIL system vs simulated vehicle.**

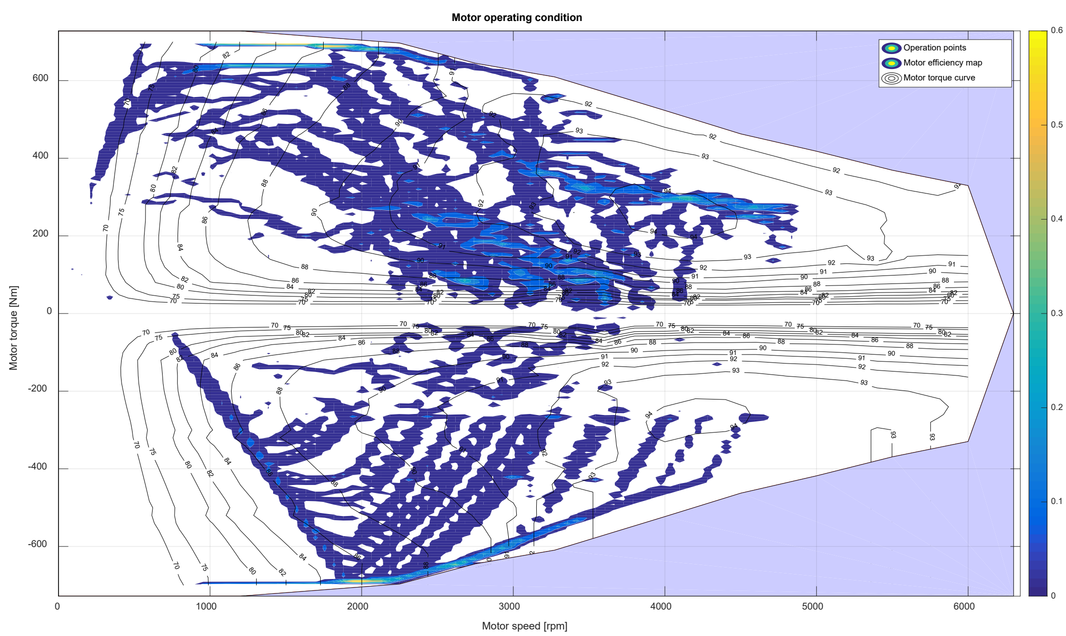



**Figure 70. Operating region heat map of the HIL test run on the Orange County Bus Cycle.**

Once the test matrix was complete, ORNL compiled results with a comparison using the two-speed mode as a baseline, which shares similar overall ratios with the current Proterra two-speed powertrain from which the vehicle model values were pulled. The four-speed operation was compared with the baseline while looking at using both the first gear and second gear for launch. Ultimately, the first gear launch showed the largest benefits for miles-per-gallon equivalency and energy consumption in Wh/mile, as shown in **Figure 71** and **Figure 72**. The largest benefits for the four-speed unit was in the two real world cycles, the Manhattan Bus Cycle and the Orange County Bus Cycle, showing a 15.4% and 13.4% reduction in mpgde respectively. When comparing the HIL test results in **Figure 71** to the simulation results listed in Table 31, simulation results (Baseline 3 in Table 31) underestimate the efficiency gains significantly compared to the HIL tests.

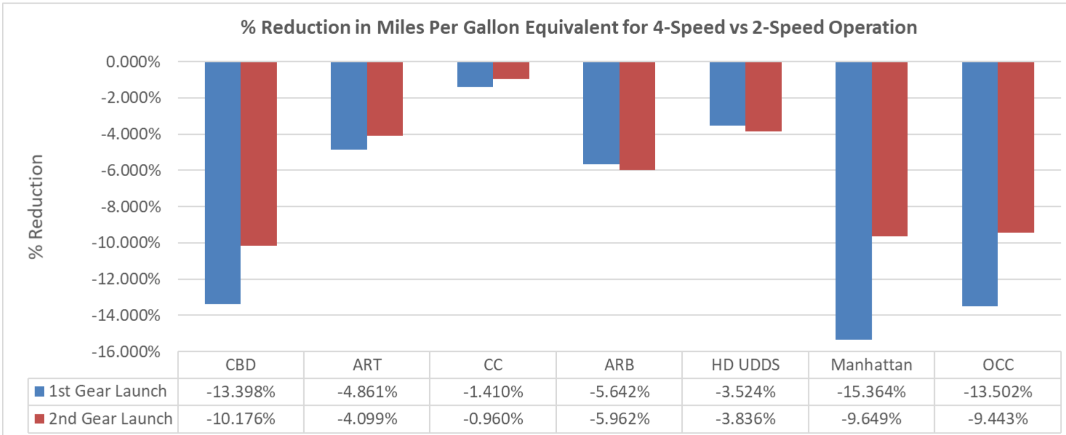



**Figure 71. Percent reduction of mpgde for 2- speed vs 4 speed operation. Ultimately a 1st gear selection for launch gear shows the largest benefit to energy consumption.**

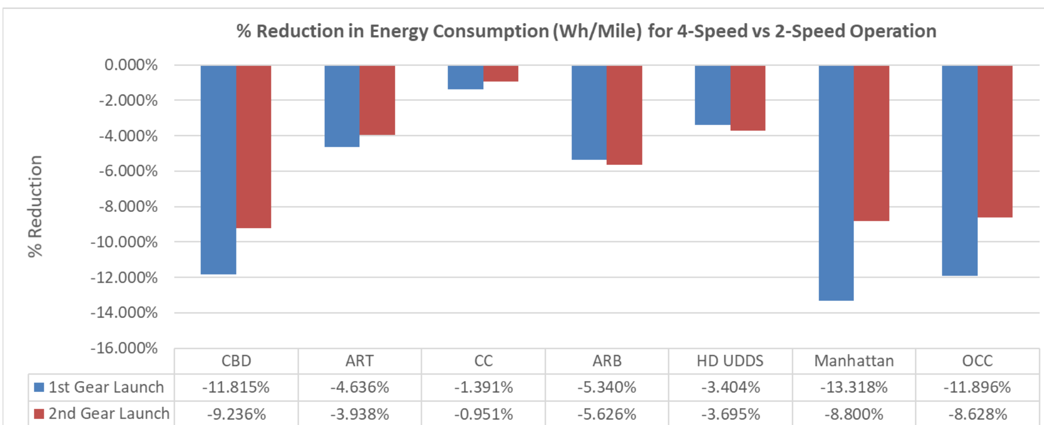



**Figure 72. Percent reduction of energy consumption in Wh/mile for 2- speed vs 4 speed operation. Ultimately a 1st gear selection for launch gear shows the largest benefit to energy consumptions.**

The Advanced Design Bus (ADB) drive cycle is one of the most important drive cycles for evaluating performance of transit busses. The ADB drive cycle was conducted on the Altoona test track under the auspices of FTA. The Altoona ADB field test is a set of cycles for a certain number of miles in a fixed time and has three CBD phases, two arterial phases, and one commuter phase run in this order: CBD, arterial, CBD, arterial, CBD, and COMM. Test runs are repeated both clockwise and counter-clockwise until bus operation is limited by range, i.e., the bus is no longer able to maintain the specified speed. The real baseline vehicle with two-speed transmission and FDR 9.8 had been tested on Altoona Test Track in the past, making it possible to compare total benefits of the new powertrain configuration with four-speed transmission and FDR 6.2. **Table 41** to **Table 43** list the HIL test results of individual phases of the ADB drive cycles.

**Table 41. HIL experimental results for the Central Business District (CBD) drive cycle**

| | Distance | Energy Consumption | | |
|---|---|---|---|---|
| **Test Mode** | **[miles]** | **[kWh]** | **[Wh/mile]** | **[mpgde]** |
| Two-speed mode | 2.03 | 3.56 | 1755.04 | 21.42 |
| Four-speed mode 1st gear start | 2.03 | 3.14 | 1547.67 | 24.29 |
| Four-speed mode 2nd gear start | 2.02 | 3.23 | 1592.94 | 23.60 |
| One-speed mode (3rd Gear) | 2.02 | 3.49 | 1728.85 | 21.75 |
| 4spd1 improve over 2spd | 0.2% | 11.6% | 11.8% | 13.4% |
| 4spd2 improve over 2spd | 0.0% | 9.3% | 9.2% | 10.2% |
| 2spd improve over 1spd3 | 0.3% | 1.8% | 1.5% | 1.5% |

**Table 42. HIL experimental results for the Arterial (ART) drive cycle**

| | Distance | Energy Consumption | | |
|---|---|---|---|---|
| **Test File** | **[miles]** | **[kWh]** | **[Wh/mile]** | **[mpgde]** |
| Two-speed mode | 2.01 | 3.97 | 1975.12 | 19.04 |
| Four-speed mode 1st gear start | 2.01 | 3.79 | 1883.56 | 19.96 |
| Four-speed mode 2nd gear start | 2.02 | 3.83 | 1897.35 | 19.82 |
| One-speed mode (3rd Gear) | 2.01 | 4.05 | 2021.33 | 18.60 |
| 4spd1 improve over 2spd | 0.2% | 4.5% | 4.6% | 4.9% |
| 4spd2 improve over 2spd | 0.4% | 3.6% | 3.9% | 4.1% |
| 2spd improve over 1spd3 | 0.2% | 2.1% | 2.3% | 2.3% |

**Table 43. HIL experimental results for the Commuter (CC) drive cycle**

| | Distance | Energy Consumption | | |
|---|---|---|---|---|
| **Test File** | **[miles]** | **[kWh]** | **[Wh/mile]** | **[mpgde]** |
| Two-speed mode | 3.69 | 4.41 | 1194.60 | 31.47 |
| Four-speed mode 1st gear start | 3.69 | 4.35 | 1177.99 | 31.92 |
| Four-speed mode 2nd gear start | 3.69 | 4.37 | 1183.24 | 31.78 |
| One-speed mode (3rd Gear) | 3.69 | 4.40 | 1191.98 | 31.54 |
| 4spd1 improve over 2spd | 0.0% | 1.4% | 1.4% | 1.4% |
| 4spd2 improve over 2spd | 0.0% | 0.9% | 1.0% | 1.0% |
| 2spd improve over 1spd3 | 0.1% | 0.3% | 0.2% | 0.2% |

As illustrated in Table 41 to Table 43, four-speed transmission and FDR 6.2 provides, 13.4%, 4.9% and 1.4% efficiency improvements over two-speed transmissions and FDR 6.2 in terms of mpgde improvement in CBD, ART, and CC cycles respectively. The efficiency improvement for the overall ADB drive cycle is 7.8%, not including the efficiency gains from the FDR change in the HIL tests.

**Table 44** shows the energy consumption and performance comparison of the real Altoona test track results of the Proterra BE35 baseline vehicle with two-speed transmission and FDR 9.8 and four-speed transmission used in two-speed mode and FDR 6.2 in HIL Altoona tests, as well as four-speed transmission and FDR 6.2 in HIL Altoona tests. The Altoona track results of the real baseline vehicle enabled us to gauge both the total efficiency gain and the proportion of efficiency gains attributable to the increasing the number of gears from two to four and to the FDR change from 9.8 to 6.2.

**Table 44. Comparisons of energy consumption in mpgde**

**1st row: The real Altoona test track results of Proterra BE35 Baseline vehicle with two-speed transmission and FDR 9.8.**
**2nd row: four-speed transmission used in two-speed mode and FDR 6.2 in HIL Altoona tests. 3rd row: four-speed transmission and FDR 6.2 in HIL Altoona tests**

| | CBD | ART | COM | Overall ADB |
|---|---|---|---|---|
| On Track, two-speed, FDR 9.8, mpgde | 20.6 | 16.9 | 28.1 | 21.5 |
| On HIL, two-speed, FRD 6.2, mpgde | 21.4 | 19.0 | 31.5 | 23.4 |
| On HIL, four-speed, FRD 6.2, mpgde | 24.3 | 20.0 | 31.9 | 25.1 |
| % Improved due to FDR 6.2 over 9.8 (HIL Tests to Track Tests) | 3.9 | 12.4 | 12.1 | 8.7 |
| % Improved due to AMT 4sp over 2sp (HIL tests to HIL tests) | 14.1 | 5.9 | 1.4 | 7.8 |
| % Improved HIL-4sp-FDR6.2 over BE35-2sp-FDR9.8 | 18.0 | 18.3 | 13.5 | 16.5 |

The efficiency improvement from the two-speed to four-speed change is 7.8%, as shown in the sixth row of Table 44. An additional 8.7% efficiency improvement is attributable to the FDR change from 9.8 to 6.2, as shown in the fourth row. Finally, the overall efficiency gain between the real baseline and the new powertrain configuration is 16.5%, as shown in the last row. Hence, the comparison of the HIL tests that did not include the original FDR and the real baseline tests on the Altoona test track that included the original FDR of 9.8 clearly shows that the total efficiency gain afforded by the four-speed transmission, which also enables the most efficient FDR, is more than doubled as measured in the ORNL HIL tests and the NREL chassis dyno tests.

When comparing the Altoona test results to the simulation results listed in Table 45., one can tell that the modeling and simulations overestimate the efficiency gains by 5% compared to the test results in ADB drive cycle.

Table 45 shows the performance comparison of the real Altoona test track results of Proterra BE35 Baseline vehicle with two-speed transmission and FDR 9.8 and four-speed transmission used in two-speed mode and FDR 6.2 in HIL Altoona tests as well as four-speed transmission and FDR 6.2 in HIL Altoona tests. The last row of Table 45 shows that 0 to 50 mph acceleration is improved by 30% and the gradeability is doubled with the new powertrain of the four-speed transmission and FDR 6.2 compared to the real baseline powertrain configuration of two-speed transmission and FDR 9.8. The fifth row of Table 45 clearly shows that HIL tests underestimate performance gains in acceleration and gradeability because the HIL tests could not include the impact of FDR. The same conclusion is also true for the chassis dyno tests of the BE35 electric bus at NREL, where only FDR 6.2 could be tested because only one vehicle was available for validation testing with four-speed transmission. Like the HIL testing, the two-speed transmission was emulated by using the four-speed transmission in two-speed mode in chassis dyno tests.

**Table 45. Comparisons of performance metrics.**
**1st row: The real Altoona test track results of Proterra BE35 Baseline vehicle with two-speed transmission and FDR 9.8. second row: four-speed transmission used in two-speed mode and FDR 6.2 in HIL Altoona tests. 3rd row: four-speed transmission and FDR 6.2 in HIL Altoona tests**

| | 0-30 mph | 30-50 mph | Max grade at 13.9 mph | Max speed at 10% grade |
|---|---|---|---|---|
| On Track, two-speed, FDR 9.8, mpgde | 15.5 | 27.5 | 9.9% | 13.3 mph |
| On HIL, two-speed, FRD 6.2, mpgde | 14.3 | 16.3 | 20% | 18.4 mph |
| On HIL, four-speed, FRD 6.2, mpgde | 12.2 | 17.9 | 20% | 25.9 mph |
| % Improved due to FDR 6.2 over 9.8 (HIL tests to Track tests) | 8 | 41 | 0 | 38 |
| % Improved due to AMT 4sp over 4sp (HIL tests to HIL tests) | 13 | -6 | 102 | 57 |
| % Improved HIL-4sp-FDR6.2 over BE35-2sp-FDR9.8 | 21 | 35 | 102 | 95 |

## III. Commercialization Possibilities and Plans for Future Collaboration

This project as a whole was a huge success, both satisfying DOE sponsors as well as industry partners. The prototype was so successful that is has joined EATON's product line as can be found at https://www.eaton.com/us/en-us/catalog/emobility/4-speed-ev.html. With the project complete, ORNL and Eaton are having open discussions on using this platform for future work with the new Energy Efficient Mobility Systems (EEMS) Lab Call 5A award at ORNL.

## IV. Publications

Publications, conference papers, or other public releases of results:

1. Chavdar, B., 2015-DOE-AMR PowerPoint Presentation at Arlington, Washington, on June 11, 2015. Project ID-vss161, Multi-Speed Transmission for Commercial Delivery Medium Duty Plug-In Electric Drive Vehicles.
2. Chavdar, B., FY 2015 Vehicle Systems Annual Progress Report (FY2016 VS APR), September 2015.
3. Chavdar, B., 2016-DOE-AMR PowerPoint Presentation at Arlington, Washington, on June 9, 2016. Project ID-vss161, Multi-Speed Transmission for Commercial Delivery Medium Duty Plug-In Electric Drive Vehicles
4. Chavdar, B., FY 2016 Vehicle Systems Annual Progress Report (FY2016 VS APR), October 2016.
5. Chavdar, B., FY 2016 Vehicle Systems Annual Progress Report (FY2017 VS APR), October 2017

## V. References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5. BYD America.e6 Pure Electric Car. http://www.byd.com/la/auto/e6.html

## VI. List of Figures

## VII. List of Tables