MULTI-SPEED TRANSMISSION FOR COMMERCIAL DELIVERY MEDIUM DUTY PEDVs FINAL SCIENTIFIC/TECHNICAL REPORT

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Acronyms and Abbreviations

Abbreviation	Description
A/C	Air Conditioner
ADB	Advanced Design Bus (Cycle) (a.k.a. Altoona Combined Cycle)
AHP	Analytical Hierarchy Process
ALL, ALT or ADB	Altoona (Combined Cycle made of 3 CBD, 2 ART and 1 CC)
AM	Additive Manufacturing
AMT	Automated Mechanical Transmission
APAC	Asia Pacific
ARB	(Heavy Duty) Air Resource Board (Cycle)
ARRA	American Recovery and Reinvestment Act
ART	Arterial (Cycle) (part of Altoona combined cycle)
BAC	Business Arterial Commuter (Cycle)
BP	Budget Period
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
dB	Decibel
DC	Direct Current
DCT	Dual Clutch Transmission
DMLS	Direct Metal Laser Sintering
DOE	Department of Energy
ESS	Energy Storage System
EV	Electric Vehicle
FHWA	Federal Highway Administration
FD, FDR	Final Drive, Final Drive Ratio

Abbreviation	**Description**
FEA	Finite Element Analysis
FMEA	Failure Modes and Effects Analysis
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
HIL	Hardware in the Loop
HTUF	High-efficiency Truck Users Forum
ICDV	Internal Combustion Drive Vehicle
IPMP	Intellectual Property Management Plan
KI	Kinetic Intensity
LD	Light Duty
MD	Medium Duty
mpgde	Miles per Gallon Diesel Equivalent
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	Rounds Per Minute
RPN	Risk Priority Number
RWD	Rear Wheel Drive

Abbreviation	Description
SLW	Seated Load Weight
SPL	Sound Pressure Level
t	tonne, long or metric ton, 1000 kg
TE	Transmission Error
TRL	Technology Readiness Level
UDDS	Urban Dynamometer Driving Schedule (truck cycle)
WHDC	World Harmonized Drive Cycle
WOT	Wide Open Throttle

1.0 EXECUTIVE SUMMARY

The EV Everywhere Grand Challenge [1] aims 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 addresses the following technical barriers:

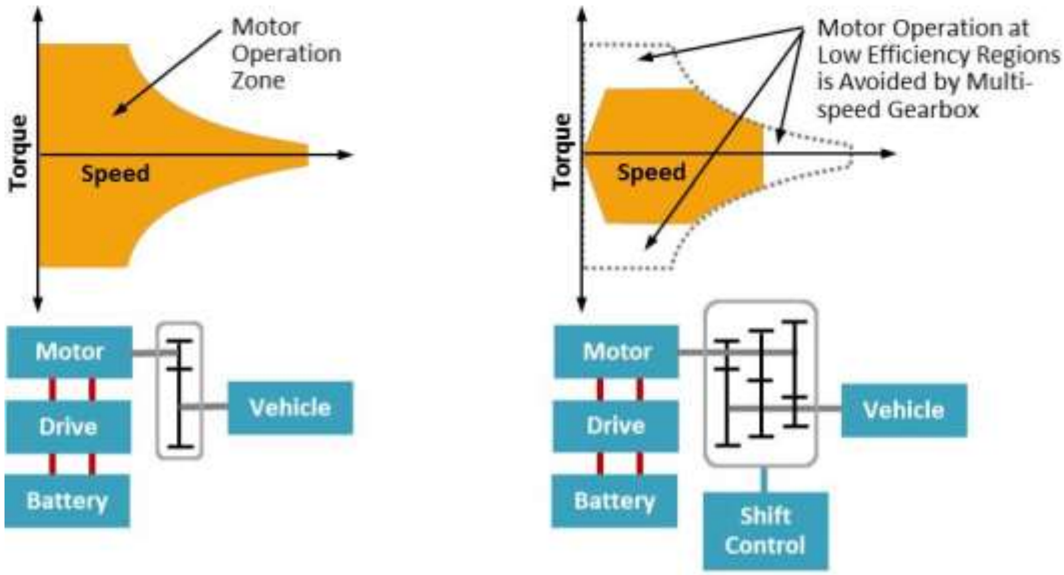
- Performance gaps between electric vehicles (EVs) and ICDVs include range, top speed, acceleration, and gradeability.
- No reliable, affordable, scalable, and low-weight, multi-speed transmissions are available to medium-duty electric vehicle manufacturers on the market.
- The public acceptance of electric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

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 BP1 and for Proterra's BE35 Electric Transit Bus in BP2. The optimization study was extended to heavy-duty PEDVs (GVW up to 36,000 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.



Baseline: Motor with a single fixed ratio

Proposed : Motor with multiple gear ratios (3 shown)

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 On -UDDS On Orange County - OCC On Manhattan - NYC On Altoona-ADB	 20.4 mpgde 19.9 mpgde 17.7 mpgde 20.2 mpgde	 24.8 mpgde 23.9 mpgde 21.2 mpgde 25.0 mpgde
Acceleration time @ SLW 0 to 30 mph 30 to 50 mph	 15.5 s 27.5 s	 < 13 s < 19 s
Gradeability @ GVW 10mph 20 mph	 15% 7%	 >20% >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 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 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.

Project Execution

Eaton assembled an excellent team to develop a multi-speed transmission for medium-duty (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 Budget Period 1 (BP1), the leader in electric transit busses (Proterra) in BP2 and BP3, and critical testing laboratories Oak Ridge National Laboratory (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 budget periods:

- 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 budget period 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-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.

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.

Preliminary Testing at Eaton

Preliminary testing of the prototype transmissions at Eaton included initial shift calibration tests on bench, transmission break-in and efficiency tests on dynamometer (dyno), NVH testing in anechoic chamber, and steady-state speed and torque tests on dyno. Figure 3 and Table 3 show the transmission efficiency test results of three prototype units. The average efficiency of the four-speed EV-AMT is %97 +/- 1.5. Fourth gear is the most efficient gear since it is in direct drive. The transmission efficiency is higher at lower speeds, higher torques and higher gears.

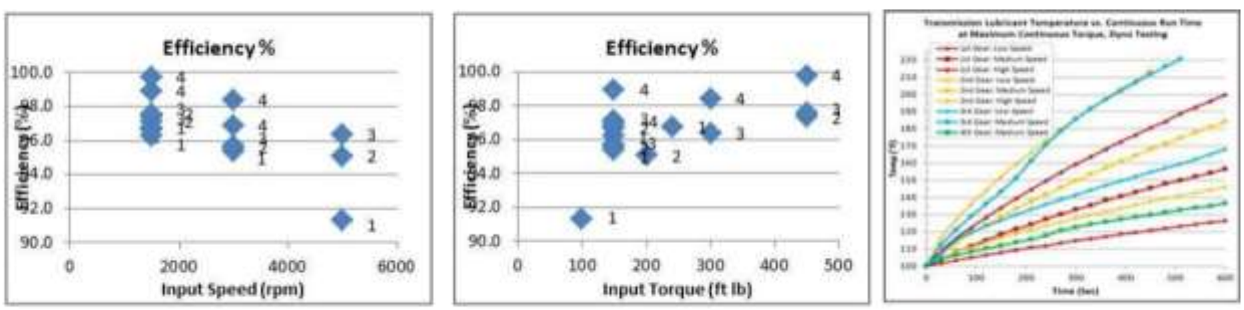


Figure 3. Left: Transmission efficiency vs input speed. Middle: Transmission efficiency vs input torque.
The legends indicate gear positions. Right: Continuous run temperature rise tests. Oil sump temperature stayed below 250 F during the tests.

Steady state temperature tests involved continuously running the transmission at various operating conditions representing the electric transit bus duty cycle. The transmission oil sump temperature stays below 250 F during ten-minute continuous runs. High-speed operations at second and third gears cause the greatest temperature rise—to 220 F, well below the allowable temperature limit of 250 F.

Powertrain HIL Tests at ORNL

Figure 4 shows the integration of powertrain consisting of Eaton four-speed EV AMT and UQM Electric Motor on the HIL test setup at ORNL. Figure 5 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 5. 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 4. Eaton four-speed EV AMT and UQM Electric Motor on the HIL test setup at ORNL

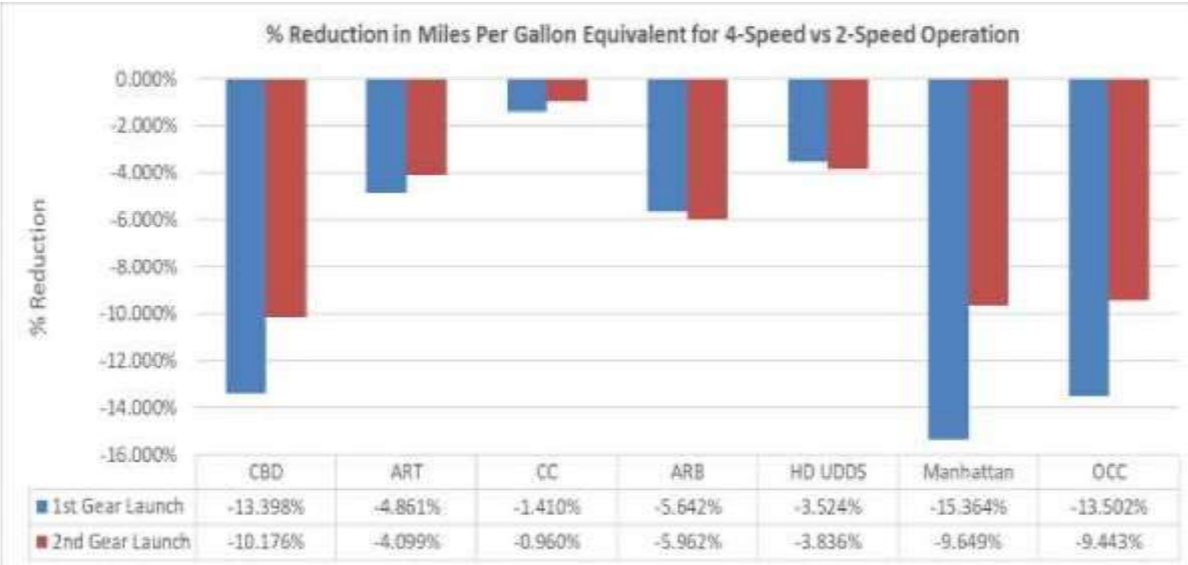


Figure 5. Percent reduction of mpgde between four-speed and two-speed configurations. Ultimately a 1st gear selection for launch gear shows the largest benefit to energy consumptions.

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

Chassis Dyno Tests 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 6**.** 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 6. Vehicle mounted on NREL's heavy-duty chassis dyno with all data acquisition implemented and ready for testing

The efficiency improvements between four-speed transmission and one-speed and two-speed baseline configurations at each drive cycle are shown in Figure 7. 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.

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

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.

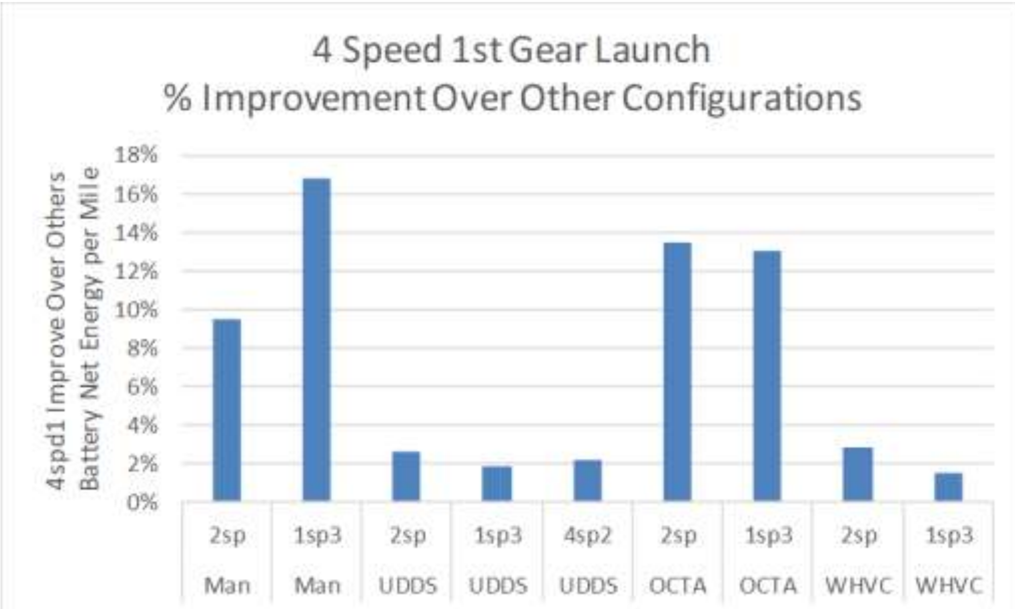


Figure 7. Efficiency improvement verification test results on NREL's HD chassis dyno. This chart does not include the efficiency gains due to the FDR change from 9.8 to 6.2.

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

2.0 PROJECT EXECUTION

The Eaton team conducted the Multi-Speed Transmission for Commercial Delivery Medium Duty PEDVs project in fourteen tasks over three budget periods. The following twelve task subsections detail the project execution and accomplishments by task. Section 3 lists the publications resulting from the team's work. Section 4 lists the references cited in this document.

The team realized the following accomplishments by budget period (BP).

BP1 Accomplishments

- Defined vehicle performance requirements based on operational data and analysis.
- Identified market segments, potential volume projections, scalability, and cost targets for penetration.
- Completed baseline vehicle model development. Integrated and validated component and vehicle models for the baseline vehicle.
- Completed transmission concept selection. Selected Modular Automated Mechanical Transmission with a family of three- and four-speed gears as the winning concept.

BP2 Accomplishments

- Completed extended modeling and simulations of electric transit bus, school bus, refuse and drayage trucks.
- Completed preliminary design layout of the four-speed AMT.
- Developed and validated vehicle model with on-route data of Proterra BE35 electric transit bus.
- Completed modeling and simulations of both the baseline electric transit bus with two-speed transmission and the improved electric bus with four-speed transmission. Predicted improvements in top speed, efficiency, acceleration and gradeability of the electric transit bus to be 50%, 15%, 40% and 30% respectively.
- Completed initial transmission system design.
  - Selected gear ratios for the Proterra BE35 electric transit bus.
  - Completed finite element analysis (FEA) of the transmission housings, rotating components analysis of the shafts, gears and bearings, and the noise, vibration and harshness (NVH) analysis of four-speed transmission.
  - Ensured that the new four-speed transmission design meets the 500,000 miles reliability requirements of the BE35 electric transit bus.
- Initiated prototyping of the new four-speed transmission.
- Completed one display unit of the new four-speed transmission.

BP3 Accomplishments

- Simulations predicted up to 20% increase in system energy efficiency depending on drive cycles, a top speed of greater than70 mph on flat road, 40% faster acceleration and a doubled gradeability with four-speed transmission as compared to the baseline EVs.
- Completed prototyping of six fully functional and one display units of four-speed transmission.
- Integrated prototype units on dynos and tested them at Eaton and ORNL.
- Validated performance gains predicted by the simulations with powertrain in the loop tests at ORNL.
- Integrated one unit on Proterra BE35 Electric Bus and tested at Eaton and NREL.
- Chassis Dyno Tests at NREL verified the simulation results of Eaton team and the HIL test results of ORNL team.
- Documented that performance gaps between MD-EVs and MD-ICDVs were reduced (e.g., range), eliminated (e.g., gradeability and top speed), or reversed in favor of EVs (e.g., acceleration performance).
- Met or exceeded all performance targets. Feedback from the demonstrations to the stakeholders of this project showed enthusiasm and acceptance for the 17,000 kg electric bus with four-speed transmission.
- The new four-speed EV transmission is efficient, reliable, modular, scalable, light weight, small size, and will be affordable. Furthermore, four-speed transmission enables downsizing of motor, battery and final drive, thereby reducing the total system cost.
- Project is completed on time and on budget.

TASK 1. PROJECT MANAGEMENT

Task 1 activities include project management and reporting. All quarterly and annual reports were submitted on time. In addition, annual on-site and off-site project status updates were presented to the DoE. The on-site reviews were carried out at Eaton's Southfield office and Eaton's Marshall Proving Grounds. The yearend reviews were carried out at Eaton's Southfield office and at DoE's Washington, DC office. The annual reviews were presented at the DoE's annual review meeting in Washington, D.C. Monthly web-meetings were held with the DoE project manager to review the progress. Table 5 shows a list of Task 1 Activities with start and finish dates.

Table 5. Task 1 activities

Task Name	Start	Finish
Task 1: Program Management & Planning	10/1/14	10/30/17
Budget Period I	10/1/14	9/30/15
Project team kick-off meeting	10/21/14	10/21/14
Initial Briefing	10/22/14	10/22/14
Research Performance Progress Report Q1	1/22/15	1/22/15
Research Performance Progress Report Q2	4/27/15	4/27/15
Annual Program Merit Review (Year 1)	6/11/15	6/11/15
Research Performance Progress Report Q3	7/30/15	7/30/15
Annual Report (Year 1)	9/18/15	9/18/15
Research Performance Progress Report Q4	10/26/15	10/26/15
End of BP1 Review Meeting	11/5/15	11/5/15
Budget Period II	10/1/15	9/30/16
Research Performance Progress Report Q5	1/29/16	1/29/16
Research Performance Progress Report Q6	4/29/16	4/29/16
Annual Program Merit Review (Year 2)	6/9/16	6/9/16
Research Performance Progress Report Q7	7/29/16	7/29/16
Annual Report (Year 2)	10/10/16	10/10/16
Research Performance Progress Report Q8	10/19/16	10/19/16
End of BP2 Review Meeting	11/2/16	11/2/16
Budget Period III	10/1/16	10/31/17
Research Performance Progress Report Q9	1/23/17	1/23/17
Research Performance Progress Report Q10	4/20/17	4/20/17
Annual Program Merit Review (Year 3)	6/7/17	6/7/17
Research Performance Progress Report Q11	7/27/17	7/27/17
End of BP3 Review Meeting	11/2/17	11/2/17
Annual Report (Year 3)	11/6/17	11/6/17
Final Report	11/30/17	1/29/2018

In BP1 and BP2, Eaton held biweekly project meetings with external partners in addition to its biweekly internal team meetings. The internal and external team meetings were held weekly in BP3. Quarterly review meetings were held with Eaton leadership.

Intellectual property management plan (IPMP) was fully executed between Eaton and its partners. CRADAs with ORNL and NREL were fully executed. Smith Electric was subcontracted in BP1. Subcontract with Smith ended at the end of B1Q4. Eaton selected Proterra as the new EV-OEM partner in BP2. Eaton subcontracted Proterra from BP2-Q3 to the end of project.

Subtask 1.1. Technology Readiness Level

The project advanced the technology readiness level (TRL) of electric vehicle intent transmission from the TRL2, the concept phase, to TRL5, the technology development level, one level at each budget period, as described in Table 6.

Table 6. Technology readiness level targets per budget period

BP	Start TRL	End TRL	Justification
1	2	3	Baseline medium-duty electric vehicle model was developed. Various transmission architecture concepts were generated for the baseline vehicle. The high-level vehicle powertrain models were used to optimize candidate transmission architectures and ratios for concept selection. The detailed driveline design and component dynamics were chosen to meet medium-duty electric vehicle requirements. The best transmission concept was selected by trade off analysis. The electric vehicle performance gains afforded by the selected concept transmission were quantified by the vehicle model simulation runs. The entry criteria for the next higher TRL level is meeting the efficiency, top speed, and acceleration performance targets.
2	3	4	The concept transmission was designed. The prototypes of concept transmission and the controller were built and tested first at Eaton's Vehicle Technology Innovation Laboratories for initial shakedown testing. Then, the transmission was integrated into the hardware in the loop testing cell at the Vehicle Systems Integration Laboratory of Oak Ridge National Laboratory. The general powertrain operation was tested and the interactions validated with virtual vehicle components emulated on a real-time platform. The entry criteria for the next higher TRL, the breadboard transmission, provides efficiency gains and the shift performance targets.
3	4	5	The transmission was fully integrated into a demonstration vehicle. The vehicle was tested and compared with the baseline vehicle performance. Transmission development continued with the vehicle testing to further refine the transmission.

Subtask. 1.2. Risk Management

The team mitigated risks to acceptable levels for project and TRL advancement. Technical risks under consideration were associated with transmission architecture, controller design, gearbox design, cost, electromechanical shifting hardware, vehicle integration, and noise vibration and harshness (NVH) performance at the start of project. The importance of maintaining skilled resources was recognized early on identified as a low-level risk. The key risks and the associated resolution strategies in the beginning of the project are described in Table 7.

The approach we used to identify, analyze, and respond to perceived risks of the project was based on the Risk Priority Number (RPN) methodology. This technique for analyzing the risk associated with potential problems was originally identified as part of a Failure Modes and Effects Analysis (FMEA). It has been incorporated into many risk management processes and found to be highly effective in abating/eliminating/reducing/transferring risk within a project. The standard approach of most risk assessment tools is to prioritize based on the risk levels and then to reduce the risk levels.

The risk management plan was reviewed and updated in team meetings based on the implementation of abatement plans and the emergence of new risks during the execution of project. All but one risk was mitigated during the project. The business case was gradually reduced to "Low" risk at the conclusion of project. The final risk status at the completion of project is shown in Table 8.

Table 7. Risks and mitigations at project start

Risk	Level	Resolution strategy
Peak efficiency region operation at top vehicle speed through **transmission architecture**	High	Use modeling tools, define gear ratios, shift strategy to optimize performance
Gearbox design, efficiency, durability, weight, **cost**	High	Use Design for Six Sigma methodology
Controller design, speed synchronization, communication, resolution between hardware, software, sensors, power electronics and ECU(s)	High	Have the system level requirements and specifications well-defined very early in the project
Speed and efficiency of **electromechanical shifting**	Medium	Concurrent engineering, modeling
Vehicle integration, high voltage/power electromagnetism isolation, brake energy recovery	Medium	Observe EMI design guidelines, enable smooth torque reversal by generator
Shift related driveline **NVH performance** at wide operation conditions	Low	Consider driveline natural frequencies in shift strategy development
Incompatibility/proprietary nature of **component level models and the vehicle model**	Low	Use digitized model outputs in the cases of incompatibility. Share black box model in the cases of proprietary nature
Maintaining skilled resources	Low	Identify key resources. Hold periodic human resource meetings to ensure no issues coming up

Table 8. Risks and mitigations project end

Risk	Level	Resolution strategy
Business case development. Business case does not justify the EV market.	Low	The business case investigation estimates 9K and 7K transmission volumes with wide range of uncertainty for the MD&HD-EV and LD-EV segments by 2023 respectively. The volumes are borderline between low and medium size markets. The recent marketing analysis conducted in APAC region identified sales potential in China due to the subsidies and regulations. The business case risk is lowered from medium-high to medium. The risk is lowered from medium to low because of the successful execution of this project.

TASK 2. IDENTIFICATION OF VEHICLE-LEVEL FUNCTIONAL REQUIREMENTS

Identifying end customer requirements key to successful development of a multi speed transmission for electric vehicles. We identified the vehicle system-level requirements first, and then identified functional requirements for the transmission and controller sub-system (Table 9).

Table 9. Task 2 activities

Task Name	Start	Finish
Task 2: Identification of vehicle level functional requirements	10/1/14	12/31/14
2.1 Baseline vehicle information.	10/1/14	12/31/14
2.2 Medium-duty EV Delivery Drive Cycle definition.	10/1/14	12/31/14

Deliverables: Vehicle Requirements Definition Report

Subtask 2.1 – Baseline Vehicle Information

Smith Electric Vehicles' product offering in the United States spanned medium-duty (Class 4-6), electric vehicles designed for a wide range of applications, including food/beverage, parcel delivery, utility, maintenance, and military transport. Gross vehicle weight ratings for the Smith Newton ranged from 16,500 lb. up to 26,000 lb., with total payload offerings from 7,000 lb to over 16,000 lb. The ladder frame chassis and cab-over design offer a great deal of flexibility for cargo box configurations and powertrain component interfacing. Figure 8 and Figure 9 show the Newton Cab Chassis 22,000 lb GVW (10 t) and Newton Step Van, 16,500 lb GVW (7.5 t), Class-4.

For this project, the 22,000 lb GVW (10 t) Newton Cab Chassis shown in Figure 8 was chosen as the baseline vehicle. The ten tonne GVW Smith Newton demonstrated a 0-30 mph acceleration of 20 seconds and 0-50 mph acceleration of 92 seconds, fully loaded, with top speed of 50 mph as shown in the 6th column of Table 10. The Newton typically demonstrated standardized driving cycle fuel efficiencies of greater than 40 mpgde (miles per gallon diesel equivalent), including 53 mpgde on the Orange County Bus Cycle (OCBC) and 61 mpgde on the Hybrid Truck Users Forum Parcel Delivery Class 4 (HTUF4) cycle. Real-world fuel efficiencies typically vary from 30-38 mpgde.

ORNL modified and distributed a copy of their simulation parameter requirements. This request was supplemented with information required for Eaton to create a secondary vehicle simulation package to support the project. The information request was then formalized and submitted to Smith as a medium for supplying the baseline vehicle information to the project partners as required by Subtask 2.1.

Figure 8. Newton Cab Chassis, 22,000 lb GVW (10 t), Class-6.

Figure 9. Newton Step Van, 16,500 lb GVW (7.5 t), Class-4.

Table 10. Baseline vehicle acceleration test data at various gross vehicle weights.

Test Mass	5000 kg	5000 kg	7500 kg	7500 kg	10000 kg	10000 kg
GVW/Battery Current limit/Power	5T (250A)	5T (350A)	7.5T (250A)	7.5T (350A)	10T (250A)	10T (350A)
0-30 mph [s]	10.56	6.79	14.85	9.67	20.44	13.09
0-50 mph [s]	38.19	20.64	66.38	30.97	91.19	45.75
30-40 mph [s]	9.1	5.32	15.2	7.89	21.39	11.79
0-Vmax mph [s]	76.23	38.64	85.11	49.6	92.09	76.43
Vmax mph [s]	57.01	58.54	52.6	58.47	49.62	57.7

6 runs were performed in each category: times and speed have been averaged

Smith returned the populated simulation parameter requirements document to Eaton with additional baseline vehicle information to support the project. The full information set included:

- Weight parameters
  - Baseline vehicle mass
  - Baseline vehicle gross weight rating (GVW)
  - Dry weight of single speed gearbox is 60 lb.
  - The weight of motor and the single speed gearbox is 244 lb.
- Driveline parameters
  - Existing axle ratio and list of available axles ratios
  - Axle manufacturer and model number
  - Existing single-speed gearbox reduction ratio
  - Description of existing gearbox design

- Wheel parameters
  - Baseline vehicle tire diameter
  - Coastdown test data
  - Wheel inertia
  - Federal braking test results to calculate braking performance
- Vehicle drag parameters
  - Aerodynamic drag coefficient
  - Frontal area of vehicle
- Motor parameters
  - Full motor description
  - Peak power rating
  - Continuous power rating
  - Maximum torque rating
  - Maximum motor speeds
  - Motor inertia
  - Motor efficiency map
  - Motor power and torque profiles
- Battery parameters
  - Type and chemistry
  - Cell manufacturer and model number
  - Battery module specification sheet
  - Nominal voltage
  - Capacity
  - Charge and discharge c-ratings
  - Cell open circuit voltage
  - Maximum cell voltage
  - Minimum cell voltage
  - Total number of cells
  - Battery pack configuration
  - Low-voltage accessory loads
  - High-voltage accessory loads
  - DC-DC converter efficiency
- Drive system overview and powertrain interface schematic
- Drive controller power limiting description

- Baseline vehicle acceleration test data at various gross vehicle weights as in Table 10. As an example, a 10-ton truck accelerates from 0 to 49.6 mph in 92 seconds with a current limit of 250 A on the electric motor.
- Sample high resolution drive cycle with a mix of urban and highway driving to serve as source for model calibration.

Subtask 2.2 – Medium-Duty EV Delivery Drive Cycle definition.

We leveraged NREL’s Drive Cycle Rapid Investigation Visualization and Evaluation (DRIVE) tool and database of Smith EV drive cycle data to identify appropriate standard chassis dynamometer test cycles for evaluating the performance of both the baseline and project vehicles. As part of the DRIVE analysis, NREL performed three additional exploratory analyses to examine individual subsets of data and compare them to the overall vehicle population. First, a drive cycle analysis was performed on the entirety of the Smith vehicle data contained within NREL’s Fleet DNA database. Next, a comparative analysis of different generations of Smith technology was performed, to explore whether identifiable differences appeared in vehicle operation between the first and second generation Smith EV platforms. In addition to performing a comparative analysis of technology vintages, an additional comparative analysis examined operating differences within the Smith dataset based on final vehicle chassis configuration. In this case, the analysis compared the operation of step vans with cab body builds. Finally, to explore whether the Smith EV data was representative of most step vans and cab trucks, the results of the previous three analyses were compared to past NREL fleet evaluation projects and the standard chassis test cycles that were identified and tested.

We identified drive cycles using NREL’s drive tool and a multivariate weighted linear least squares algorithm. The metrics used to identify standard chassis test cycles include average driving speed, maximum driving speed, standard deviation of speed, idle time, stops per mile, and kinetic intensity, with a 5x weighting applied to the kinetic intensity fit calculation.

Drive cycle characterization was performed for each day of Smith Vehicle operation in NREL’s database, with over 200 metrics generated that describe typical vehicle operation.

Figure 10 illustrates the relationship between average driving speed and kinetic intensity which is representative of drive cycle aggressiveness. There is a strong power relationship between average driving speed and driving aggressiveness, and many standard chassis test cycles fit well on the trend line for the Smith dataset. The drive cycles identified for testing sit very closely to the overall trend line, with the NYC Composite cycle being the furthest away.

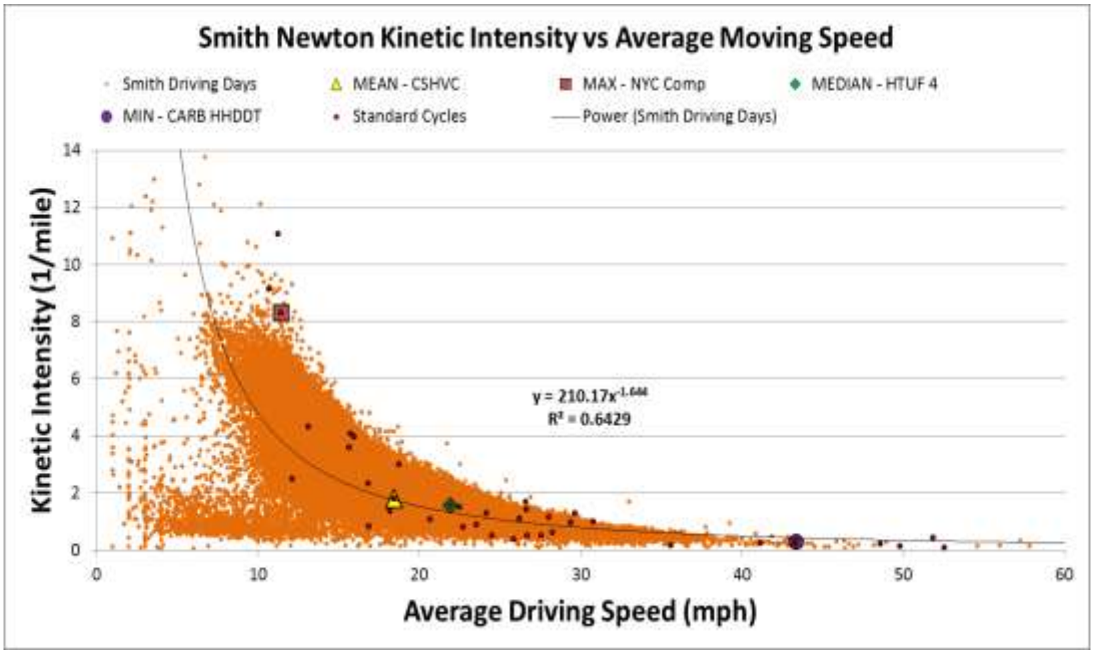


Figure 10. Relationship between average driving speed and kinetic intensity for Smith data

Notice in Figure 11 a strong relationship between stops per mile and average driving speed within the Smith dataset. This correlation confirms an intuitive understanding that average vehicle speed must go down as the number of stops increases, as a result of increased time spent at low speeds while going into and out stops.

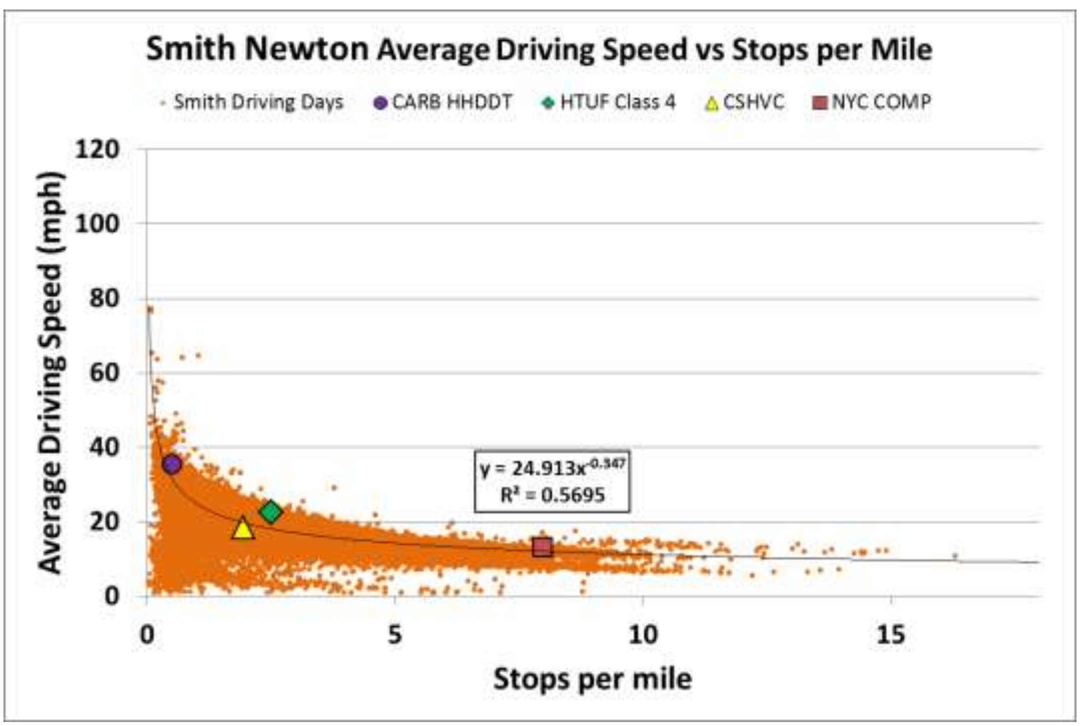


Figure 11. Relationship between stops per mile and average driving speed for Smith data

Figure 12 shows a linear relationship between stops per mile and kinetic intensity for the overall Smith EV population. This relationship suggests that the aggressiveness of the drive cycle is strongly tied to the number of stops the vehicle is required to make. It can also be seen that the selected standard cycles well bracket the overall dataset.

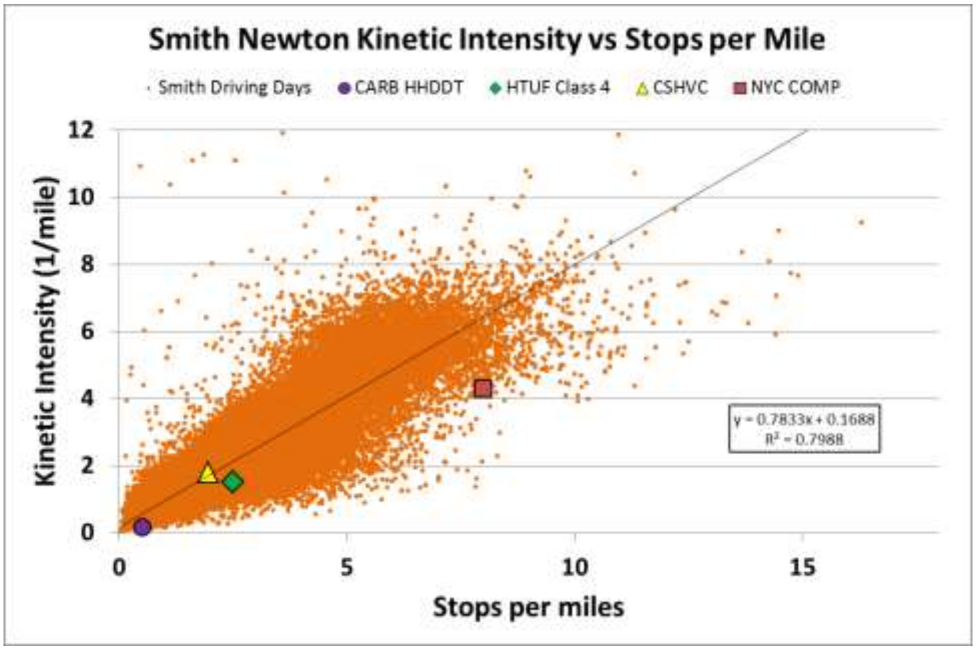


Figure 12. Relationship between stops per mile and kinetic intensity for Smith data

Figure 13 to Figure 16 provide visual examples of the speed-time traces for each of the standard chassis test cycles identified as a result of the analysis of the global Smith EV population.

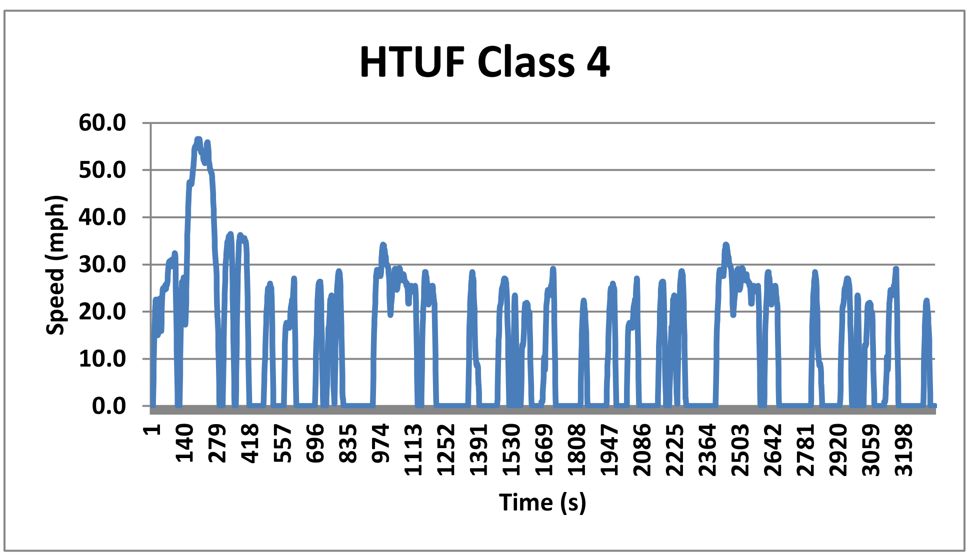


Figure 13. Speed-time trace for HTUF 4 standard chassis test cycle

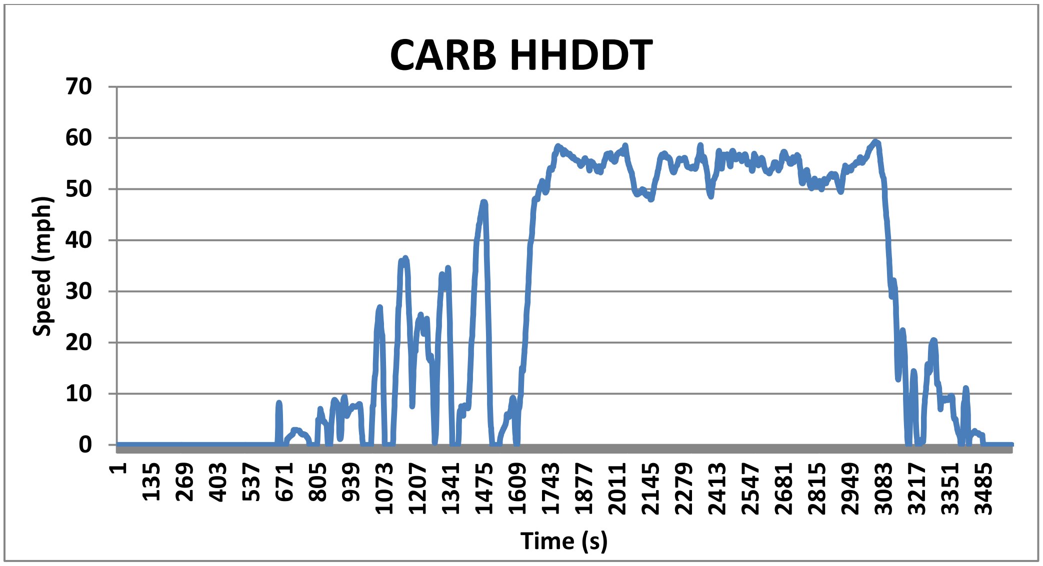


Figure 14. Speed-time trace for CARB HHDDT standard chassis test cycle

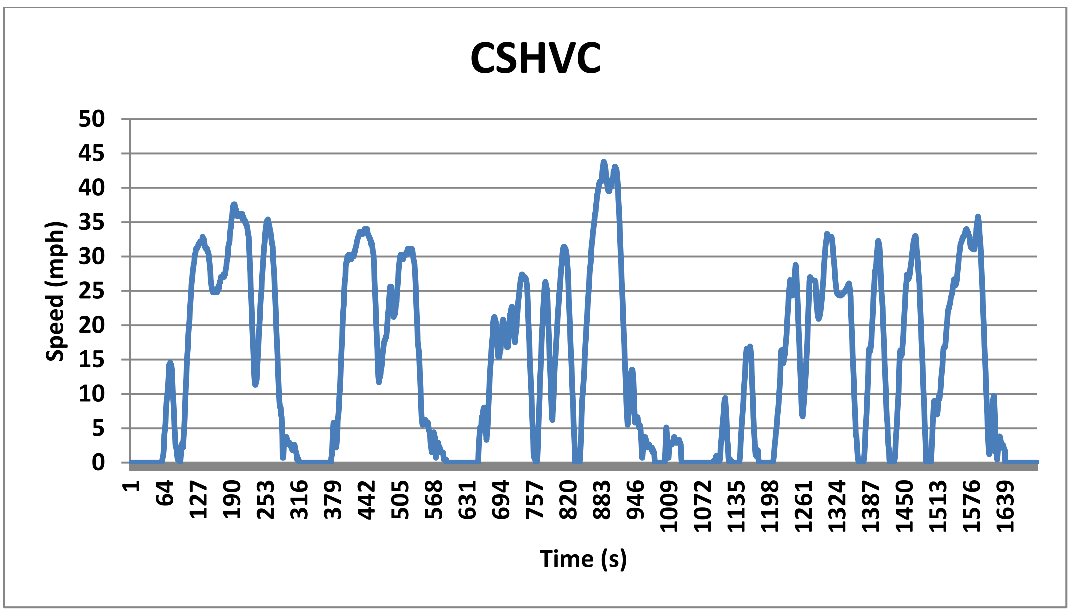


Figure 15. Speed-time trace for CSHVC standard chassis test cycle

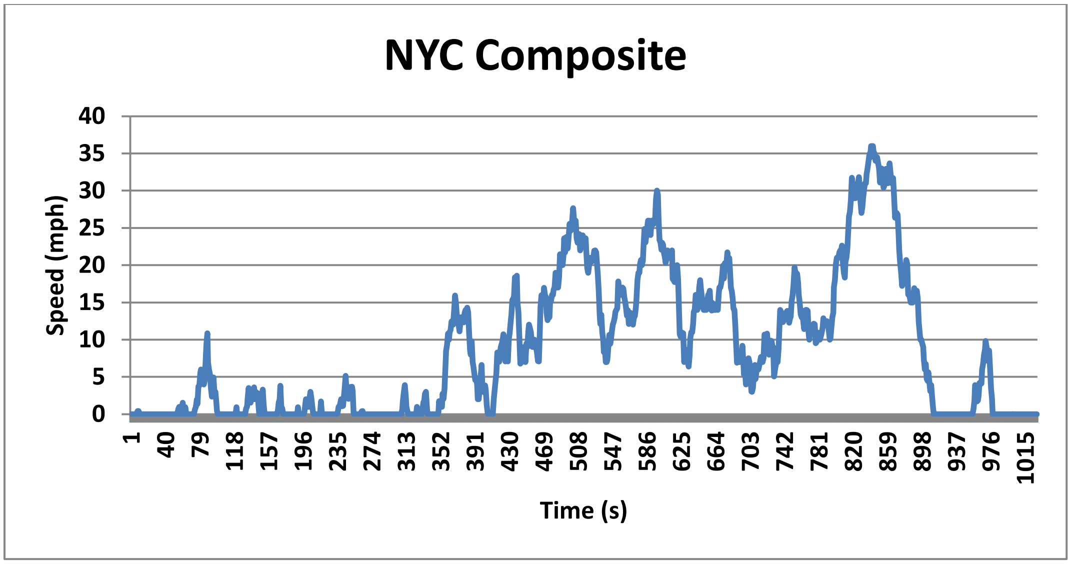


Figure 16. Speed-time trace for NYC Composite standard chassis test cycle

Summary statistics describing the characteristics of each of the standard drive cycles identified as a result of the analysis of the Smith dataset can be found in Table 11.

Table 11. Comparison of selected standard chassis test cycle statistics

Drive Cycle Statistics	CARB HHDDT	CSHVC (CSC)	HTUF 4	NYC Composite
Speed data time duration (h)	1.00	0.47	0.93	0.29
Maximum driving speed (mph)	59.30	43.80	56.59	36.00
Average driving speed (speed > 0, mph)	35.59	18.44	22.49	13.11
Standard deviation of speed (mph)	24.48	13.06	13.92	9.47
Zero speed time (%)	26.89	23.29	46.40	33.14
Total distance traveled (miles)	26.05	6.68	11.17	2.51
Number of stops per mile	0.50	1.95	2.51	7.98
Kinetic intensity (1/mile)	0.17	1.79	1.51	4.30

Examining kinetic intensity, one can see that the NYC Composite cycle is the most aggressive drive cycle, while the CARB HHDDT cycle is the least aggressive in terms of acceleration rates and stop and go behavior.

In an effort to explore any potential variation in vehicle operation between first and second generation Smith EVs in the dataset, additional sub-analyses were performed on data subsets separated between first and second generation builds. The same weighted linear least squares method was applied to each subset to identify representative standard drive cycles reflecting the full range of first and second generation Smith vehicle operation. Upon performing the standard cycle identification process, the same set standard drive cycles were identified for both data subsets (CARB HHDDT, HTUF 4/CSHVC, NYC Composite), confirming that for test cycle selection, both data subsets were identical and matched the overall behavior of the full dataset.

In parallel with exploring the potential for variation in operation between first and second generation Smith EVs, additional interest was expressed in determining whether subsets of vehicle chassis configurations operated under differing conditions. To determine whether vehicle drive cycles differ between vehicle chassis types, an additional series of sub analyses were performed on the Smith data set using subsets of data chosen to reflect cab chassis and step van body configurations.

A major challenge facing large-scale adoption of electric vehicle technology is performance concerns associated with electrified vehicles compared to conventional combustion engine powered vehicles. To evaluate potential deviations that existing Smith EVs may demonstrate in operation from typical conventional vehicles, we performed an additional comparison of standard chassis test cycles identified as a result of past NREL fleet evaluation projects. The cycles being compared against the results generated in this task were identified using the same weighted linear least squares approach mentioned previously. As shown in Table 12, the Smith EV data performs in a very similar capacity to vehicles that have previously been examined. It is interesting to note that the HTUF 4, CARB HHDDT, and NYC Composite cycle have all been identified in the past as characteristic of either step van or cab truck operation.

Table 12. Comparison of standard chassis test cycles with previous projects

	Standard Chassis Test Cycles			
Project	**Low**	**Mean**	**Median**	**High**
UPS MN Step Van	CARB HHDDT	HTUF 4	HTUF 4	NYC Composite
UPS AZ Step Van	CILCC	WVU CITY	WVU CITY	CBD
FedEx CA Step Van	HTUF 4	OC Bus	OC Bus	NY City Cycle
FedEx Straight Truck	CARB HHDDT	HTUF 6	HTUF 6	NYC Composite
Smith Gen 1	CARB HHDDT	CSHVC	HTUF 4	NYC Composite
Smith Gen 2	CARB HHDDT	CSHVC	HTUF 4	NYC Composite
Smith Step Vans	CARB HHDDT	CILCC	HTUF 4	NYC Composite
Smith Cab Bodies	CARB HHDDT	CSHVC	HTUF 4	NYC Composite

Examining the standard chassis cycles selected for this project compared to previous fleet evaluation projects, we noted that the wide range of standard cycles selected during this task reflects the wide range of observed real world operating conditions and applications of Smith Electric Vehicles.

Upon completion of the large-scale analysis of the Smith drive cycle data, individual representative custom drive cycles were generated using NREL's DRIVE tool. These cycles were generated with the goal of matching the overall average drive cycle performance of the entire Smith vehicle population, the step van configuration subset, and the box truck subset. The cycles were generated based on a matching overall drive cycle kinematic metrics, including average driving speed, maximum driving speed, kinetic intensity, standard deviation of speed, and stops per mile. The cycles ignored matching percentage of time spent at zero speed, as the goal of these cycle were to develop simulation and test procedures to measure the performance of the powertrain under dynamic operation. Since idle fueling is minimally affected by powertrain configuration, idle weighting was left out of the design criteria. In addition to focusing the cycle generation on matching the kinematic behavior of the specific vehicle set/subset, the cycles were also developed with a maximum duration of one hour as a goal. This upper ceiling on cycle duration was selected with a look towards chassis testing over the course of the project. Cycles with durations exceeding an hour pose potential hurdles for extended chassis testing due to complications resulting from driver fatigue and dynamometer calibration.

Figure 17 to Figure 19 illustrate the generated drive cycles that can be used for calibration, testing, and simulation along with the identified standard chassis test cycles. The figures show the speed-time traces that are prescribed for each chassis test.

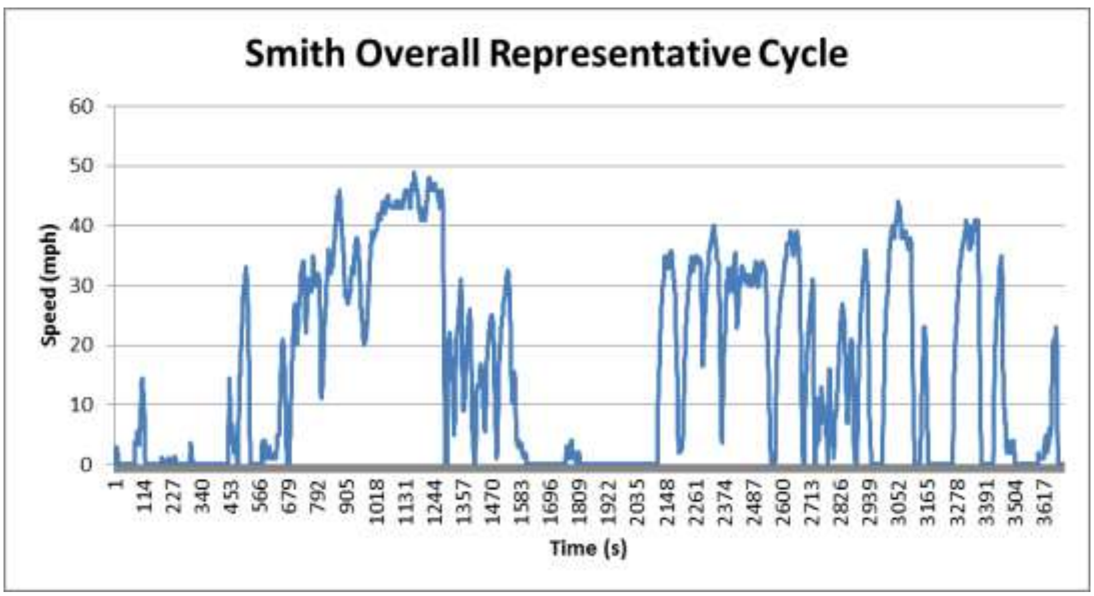


Figure 17. Speed-time trace for Smith overall representative cycle

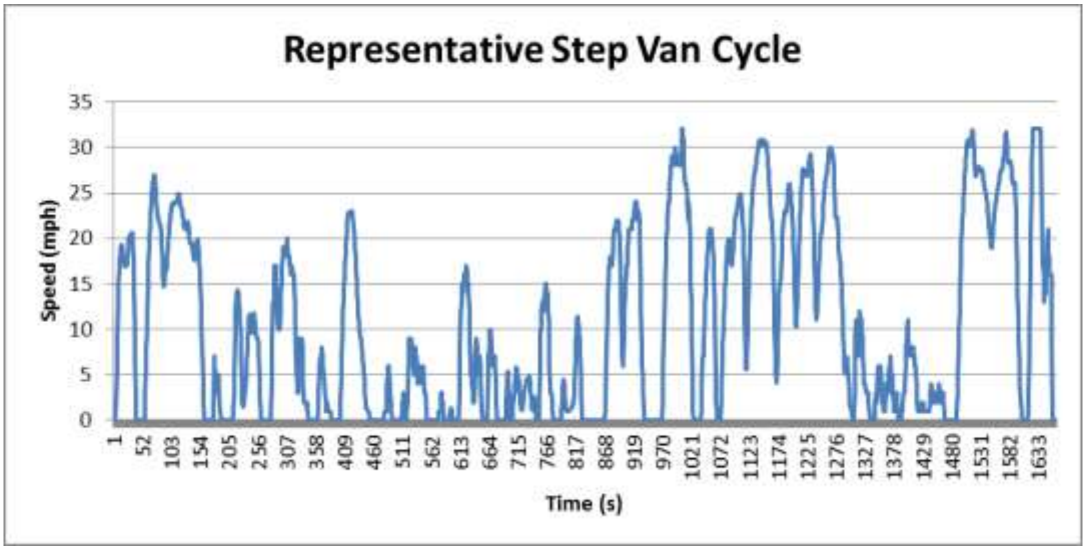


Figure 18. Speed-time trace for representative step van cycle

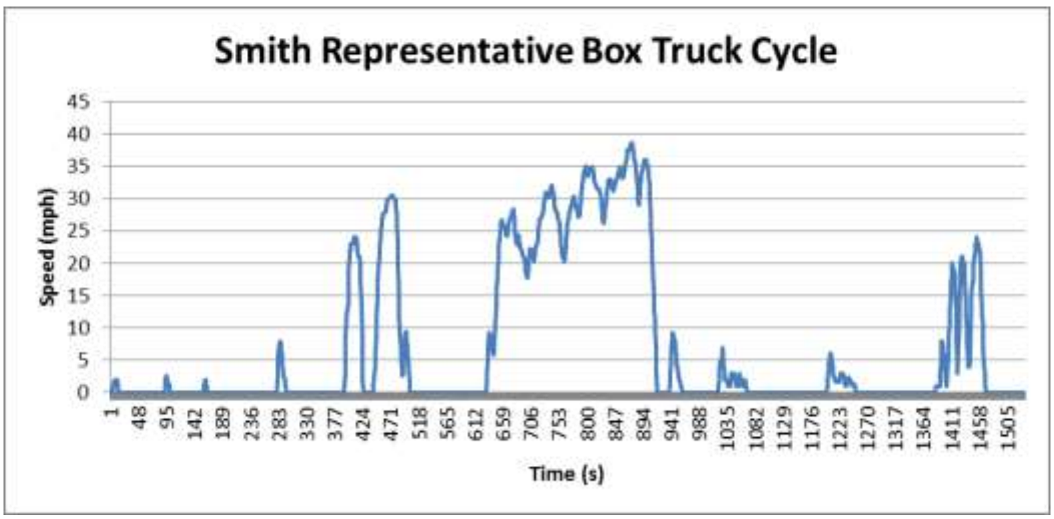


Figure 19. Speed-time trace for Smith representative box truck cycle

To explore the effects of drive cycle behavior on the current energy consumption of Smith Electric vehicles, the results of analysis performed as part of NREL's support of DOE's ARRA-funded projects was leveraged and can be found in Figure 20**.** Figure 21 shows the overall distribution of vehicle energy consumption. This information may prove helpful in evaluating the impact of the energy savings produced as a result of project work.

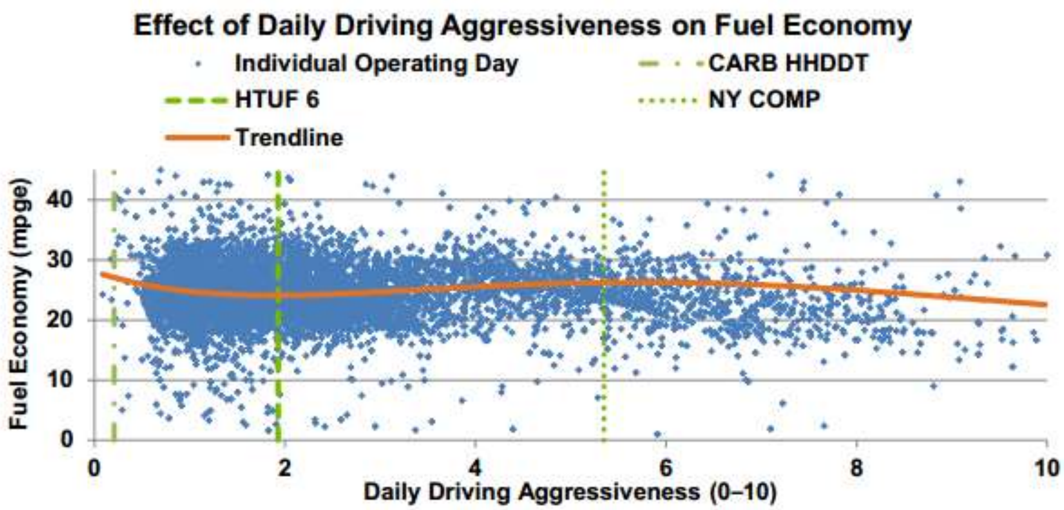


Figure 20. Speed-Time Trace for NYC Composite Standard Chassis Test Cycle

Figure 20 shows that energy consumption is somewhat dependent on drive cycle aggressiveness (kinetic intensity scaled by 2 on this chart); however, given the variation in payload and vehicle weight rating, it is difficult to ascertain a direct correlation between drive cycle and energy consumption.

Figure 21 shows that the average Smith vehicle consumes energy in the range of 1.4-1.5 kWh/mile, with some vehicles consuming more and some consuming less due to a variety of factors of which drive cycle is one.

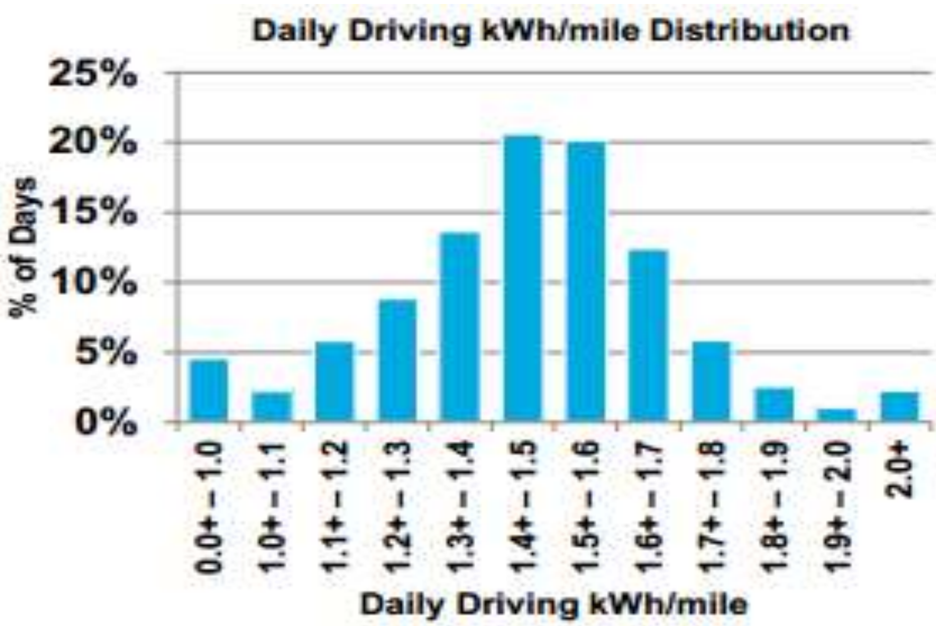


Figure 21. Speed-time trace for NYC Composite standard chassis test cycle

TASK 3. BUSINESS CASE DEVELOPMENT

The business case for multi-speed transmission for electric vehicles is tied to the adoption rate of electric vehicles. To be able to estimate EV market volumes in the next five to ten years, it is important to understand the market drivers affected by zero emissions regulations and government mandates around the world. Table 13 lists the activities covered in this task.

Table 13. Task 3 activities

Task Name	Start	Finish
Task 3: Business case development (Milestone 2)	1/1/15	3/31/15
3.1 – Identify growing market segments and potential volume projections	1/1/15	3/31/15
Electric car market estimates	1/1/15	3/31/15
MD-HD electric truck market estimates	1/1/15	3/31/15
LD electric truck market estimates	1/1/15	3/31/15
Multi speed transmission market estimate for electric trucks	1/1/15	3/31/15
3.2 Identify transmission cost targets for market segments.	1/1/15	3/31/15

Deliverables: Business Case Development Report

Subtask 3.1 – Identify Growing Market Segments and Potential Volume Projections

The business case for electric vehicles can be viewed from two perspectives. One perspective is the growing threat to human health and the environment caused by particulate matter and toxic gas emissions (NO_x, CO_x etc.) from fossil fuel burning vehicles. Only 12% of the big cities meet the Air Quality Guidelines published by the World Health Organization in 2012. Therefore, many densely populated city centers have already been declared low emission zones, and the trend to fight the air pollution is increasing. The other perspective is the total cost of ownership of electric vehicles that must be attractive to the consumers to stimulate far-reaching adoption of electric vehicles beyond the low and zero emission zones.

According to Drew Kodjak [3], a significant gap exists between consumer expectations and actual EV performance. Major disappointments are due to price premium, purchase price, fuel price, recharge time, range gap, and fuel efficiency. When consumer expectations are compared to existing products, market size is limited to 2 to 4% of car buyers in 17 countries surveyed. Electric vehicle adoption rate depends on improvements in range, charge time, convenience of charging, purchase cost, operating cost, and total cost of ownership. Over the coming decade, government policies concerning EV technology will determine adoption rates.

All-electric vehicles are the only zero emission solution addressing the health and environment concerns in the low- and zero- emission zones. Fossil fuel driven vehicle owners must pay a penalty if they don’t meet emission regulations. Hence, the consumers living in the low/zero emission zones will be the first adopters of the electric vehicles.

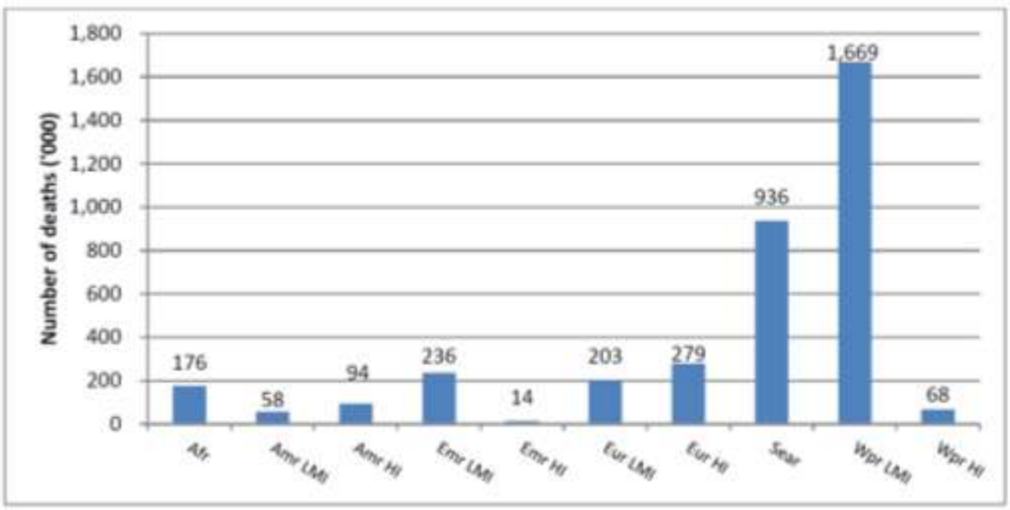
A medium-duty electric truck costs twice as much as a medium-duty diesel truck— that is roughly $120K versus $60K with an incremental difference of $60K, today. The major sources of the price difference are the costs of the battery and the electric motor.

By the year 2020, the payback period is expected to improve significantly in favor of electric trucks. The cost of batteries is expected to decrease significantly from the current cost ($300-$500), Refs. [4,5]. The cost of motors, motor control units, and inverters are also expected to decrease, further reducing the incremental difference between medium-duty electric trucks and diesel trucks. Furthermore, availability of a multi-speed transmission for electric trucks will enable battery savings and electric motor downsizing, further reducing the incremental difference while bringing the performance of an electric truck up to the same level as that of a diesel truck. It is estimated that these advancements will reduce the payback period to close to the 3-5 years typically expected by end users.

One business model for medium-duty electric trucks considers selling the vehicle at a price that excludes the cost of battery. The battery will be owned by the OEM, who will also provide the charging and service centers where batteries will be charged and trucks will be serviced for a fee. This business model eliminates the higher upfront costs to buyers while enabling buyers to make easier purchasing decisions by having direct and reduced comparison of energy and maintenance costs to those of gasoline or diesel trucks. This business model targets a 50% adoption rate in large and small fleets of parcel delivery applications in urban city centers where the daily routes and ranges are well-defined. Far-reaching adoption rates beyond the mega-cities will depend on advancement in the battery technology and development of charging infrastructure that enables longer range travel without range anxiety.

The maintenance costs for electric vehicles are typically much less than that of internal combustion drive vehicles since the powertrain and the driveline components of EVs are much simpler than the engine and the transmission of ICDVs [6]. They have no oil, air filter and sparkplug changes and no catalytic converter or other emissions equipment to maintain. Electric vehicles have an advantage in mechanical simplicity due to the smaller number of moving parts in an electric power train when compared to an internal combustion power train. The current generation Smith Newton includes fewer than 10 moving parts in the electric drivetrain as opposed to nearly 3,000 for a typical internal combustion engine. The result is a preventive maintenance schedule that typically requires less than one hour of labor per vehicle, per year. Component wear is often reduced as well. For example, due to regenerative braking, Smith customers typically experience brake pad intervals three to four times longer than their traditional fleet. Additionally, the two major components of the power train, the electric motor rotor and stator, never physically touch, which eliminates the potential for long-term friction and wear.

WHO estimated that globally, 3.7 million deaths were attributable to ambient air pollution in 2012 as shown in Figure 22. A European Commission-funded website [7] claims that air pollution is responsible for 310,000 premature deaths in Europe each year, more than those caused by road accidents. Air pollution causes lung and heart diseases and especially affects the young and the old. The human health damage caused by air pollution is estimated to cost the European economy between $650 billion to $1.2 trillion in a year. The European Union sets air quality targets that trigger the Low Emission Zones if the measured air quality of cities exceeds the targets. According to the UK-based *Automobile Association Magazine* "low emission zones" were launched or in preparation to launch in more than 200 cities and towns in 10 countries around Europe. The mayor of London proposed London City Center be declared the Ultra-Low Emission Zone, where all taxis and single deck busses must be zero emission at tailpipe by 2020. Worldwide expansion of low-, ultra-low-, and zero- emission zones is expected to drive the sales of electric cars and trucks. Figure 23 shows traffic signs for zero- and low- emission zones.



AAP: Ambient air pollution; Amr: America, Afr: Africa; Emr: Eastern Mediterranean, Sear: South-East Asia, Wpr: Western Pacific; LMI: Low- and middle-income; HI: High-income.

Figure 22. Total deaths ('000) attributable to ambient air pollution in 2012, by region

a b

Figure 23. a) Sign for zero emission zone; b) German sign for low emission zone

The Chinese Government energy policy issued in July 2014 mandates that 30% of new vehicle purchases by government bodies must be powered by alternative energy by 2016. The announcement of the new energy policy and the extension of the government subsidies toward the purchase of electric vehicles are big boosts for the purchase of electric vehicles in China. Figure 24 shows the growth of the heavy-duty electric vehicle market in China through 2019. Figure 25 shows the details of the new energy policy and the subsidies in China.

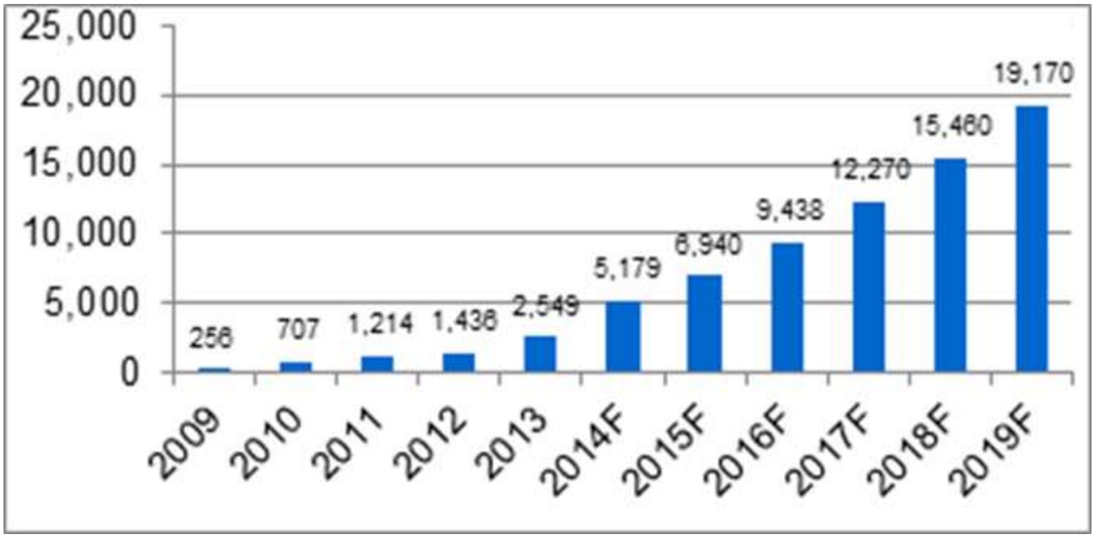


Figure 24. Heavy-duty electric vehicle market forecast for China.

<table>
<tr><th>Central government</th><th colspan="4">Local government</th></tr>
<tr><td rowspan="9">Sales tax (10%) exempted for new energy vehicles in tax free category 1&2 from 1st Sept 2014 to 31st Dec 2017.</td><td colspan="4">Cash subsidy for Large EV bus (RMB) (newly released in 2014)*</td></tr>
<tr><td>Foshan</td><td>500K</td><td>Ningbo</td><td>500K</td></tr>
<tr><td>Wuhan</td><td>500K</td><td>Weifang</td><td>500K</td></tr>
<tr><td>Changsha</td><td>500K</td><td>Taiyuan</td><td>500K</td></tr>
<tr><td>Xiangyang</td><td>500K</td><td>Dalian</td><td>400K</td></tr>
<tr><td>Luzhou</td><td>500K</td><td>Suzhou</td><td>300K</td></tr>
<tr><td>Hangzhou</td><td>500K</td><td>Jiangsu</td><td>200K</td></tr>
<tr><td>Hunan</td><td>500K</td><td>Qingdao</td><td>100K</td></tr>
<tr><td>Xi'an</td><td>500K</td><td></td><td></td></tr>
<tr><td>EV bus charging (or swap battery charging) during valley hours (50% saving compared to normal hour).</td><td colspan="4">A new electricity pricing policy for EV bus implemented in Jiangsu, March 10th (20% cost saving on charging).</td></tr>
<tr><td colspan="5">*(Central + Local government subsidy should not exceed 60% of the bus price)</td></tr>
</table>

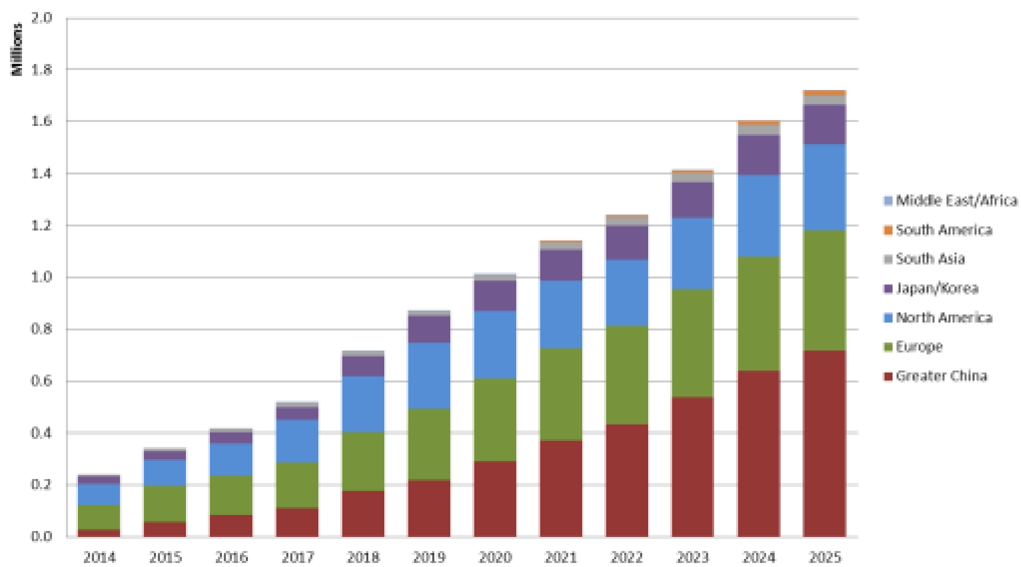
Figure 25. China central and local government new energy policy and subsidy update.
Source: China Ministry of Industry and Information Technology and local government reports.

The state of California is leading the United States on zero emissions efforts. The Ports of Los Angeles implemented a Zero Emission Roadmap in 2011 to achieve zero emissions goods movement in the nation's number one container port. The state also gives incentives to residents and companies to encourage them to buy zero-emission vehicles. California offers as much as $15,000 in rebates for the purchase or lease of new zero-emission and plug-in hybrid light-duty vehicles and up to $120,000 for medium-duty vehicles. According to the California Plug-In Electric Vehicle Collaborative, a result of these incentives is that the state accounts for 40% of plug-in electric vehicles in the US.

Electric car market estimates

Navigant research predicts that the electric car sales in the US will grow from 100,000 in 2013 to 514,000 in 2023 [8,9]. Furthermore, plug in electric vehicles on roads in the United States will surpass 2.7 million by 2023. Global annual sales of plug in electric vehicles will increase from 352,000 in 2014 to 1.8 million (or 2.4% of total market) in 2025, as shown in Figure 26. When conventional hybrids are included, then the EV sales are expected to grow from 2.7 million in 2014 to 6.4 million in 2023. The largest urban markets for the forecast period, according to the report, will be Tokyo, Los Angeles, and Paris, with plug-in electric vehicle sales in 2023 of 49,000, 39,000, and 25,000 vehicles, respectively.

The types of transmissions used in today's plug in electric cars are a single-speed gearbox on Nissan Leaf and Tesla EVs and continuously variable transmissions (CVT) used in GM-Volt and Toyota Prius with two electric motors and a gasoline engine. BMW i8, axle split hybrid car, is the first car that has a two-speed transmission on the electric motor driving the front wheels. BMW i8 also has a six-speed automatic transmission on the gasoline engine driving the rear wheels. The two-speed transmission and the front differential are combined into one package by GKN [10]. The shifting is accomplished by electronically controlled speed synchronization and electromechanical actuation. Multi-speed transmission solutions for electric cars are expected to gain more attention as the electric car market grows. However, OEMs will tend to develop their own multi-speed transmission solutions, as in the cases for GM-Volt and Toyota-Prius because CVT is non-modular and highly integrated and the vehicles use multiple power sources. Pure electric cars such as Tesla are better candidates for third-party modular transmission solutions.



Source: Global Insiders

Figure 26. Global pure electric vehicle production by region.

MD-HD electric truck market estimates

In the United States, commercial truck classification is based on the vehicle's gross vehicle weight rating (GVW) by the Department of Transportation's Federal Highway Administration (FHWA). Medium-duty trucks are Class-4 to 6 and have GVW range of 14K to 26K lb. Heavy-duty trucks are Class-7 to 8 and have GVW range of 26K lb and beyond.

Global sales of medium and heavy-duty trucks reached 2.9 million in 2014, as shown in Figure 27. Heavy-duty truck sales were 1.8 million and medium-duty sales were 1.1 million. Hybrid and plug in electric truck sales were 16,000 units in 2014.

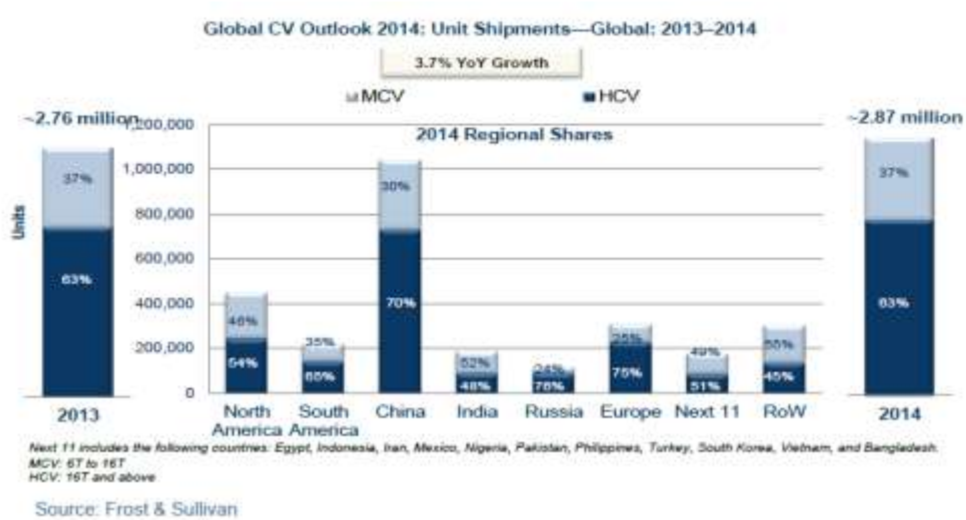


Figure 27. Medium-heavy-duty truck sales (Global), 2013-2014.

Navigant Research predicts that worldwide sales of medium- and heavy-duty commercial, all-electric, hybrid electric, and plug-in hybrid electric vehicles will increase from 16,000 in 2014 to 105,000 in 2020 and 160,000 in 2023 [10]. In the long run, the EV market will be split 50/50 by plug in-EV and hybrid-EV. Hence, we expect a global annual production volume of 80,000 for pure electric MD and HD trucks by 2023. If a quarter of MD and HD electric trucks adopt a multi-speed transmission, the projected volume of multi-speed transmissions for MD and HD electric trucks will be 20,000 units. All units are expected to be rear wheel drive.

The highest sales penetration rate of full electric trucks is expected to be in Japan, with 21.2% by 2023 [11]. Today's hybrid electric busses and trucks use the Eaton 6-speed automated manual transmission adapted with an auto clutch and motor/generator unit as shown in Figure 28. Although this transmission can also be used in pure electric vehicles, there are compelling indications that a pure electric truck may need fewer gears and a simpler transmission controller unit than a six-speed AMT for hybrids, resulting in a lighter, more efficient and more affordable transmission.

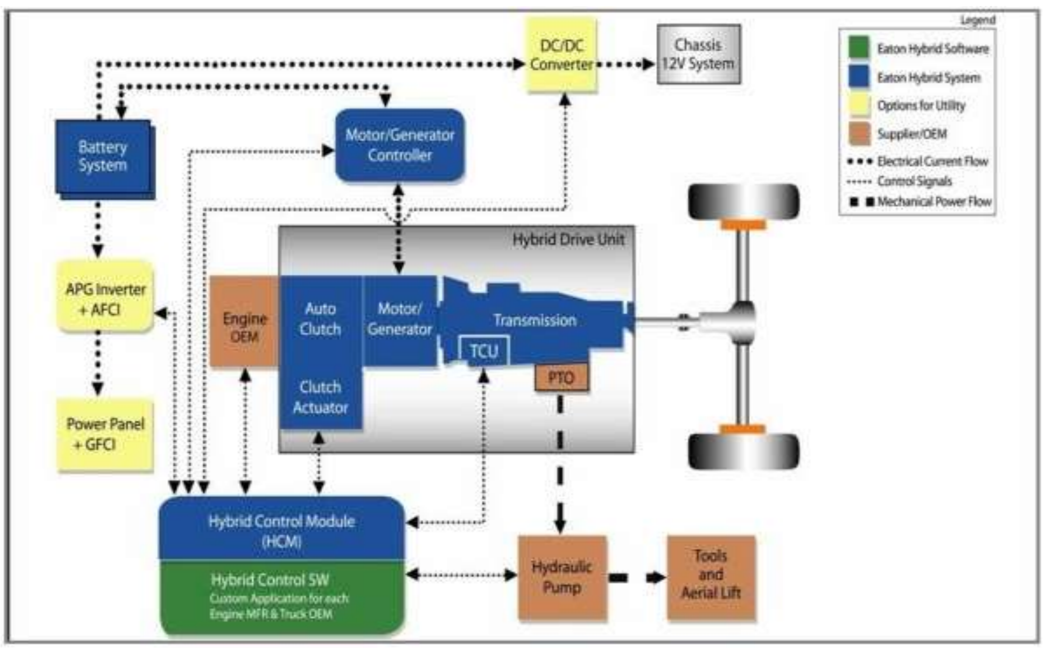


Figure 28. Powertrain system of a MD-Hybrid EV.

LD electric truck market estimates

The light-duty trucks of interest are Class-2 and -3 having a GVW range of 6K to 14K lb. Figure 29 shows hybrid truck programs that are in the production plans of vehicle manufacturers in Far East and North America regions. Assuming electric truck programs will be similar to hybrid programs, we can draw two conclusions: while MD and HD electric truck usage will grow in North America, LD electric truck use will grow in the Far East. Furthermore, the volume of Class-3 electric vehicles will be equivalent to the total volume of Class-5 to -8 electric trucks.

Therefore, it may be reasonable to expect that the multi-speed transmissions market for LD electric trucks will be 20,000 units by 2023, similar that for multi-speed transmissions for MD and HD electric trucks, with a 25% adoption rate. However, unlike MD and HD trucks, some LD trucks are expected to be front-wheel drive (FWD), although the majority will remain rear-wheel drive (RWD). The transmission concepts for RWD and FWD vehicles are likely to be very different. On the other hand, the RWD transmission concept for LD, MD, and HD electric trucks can be fundamentally the same, provided it can be scalable to match the torque demand of a particular class of trucks.

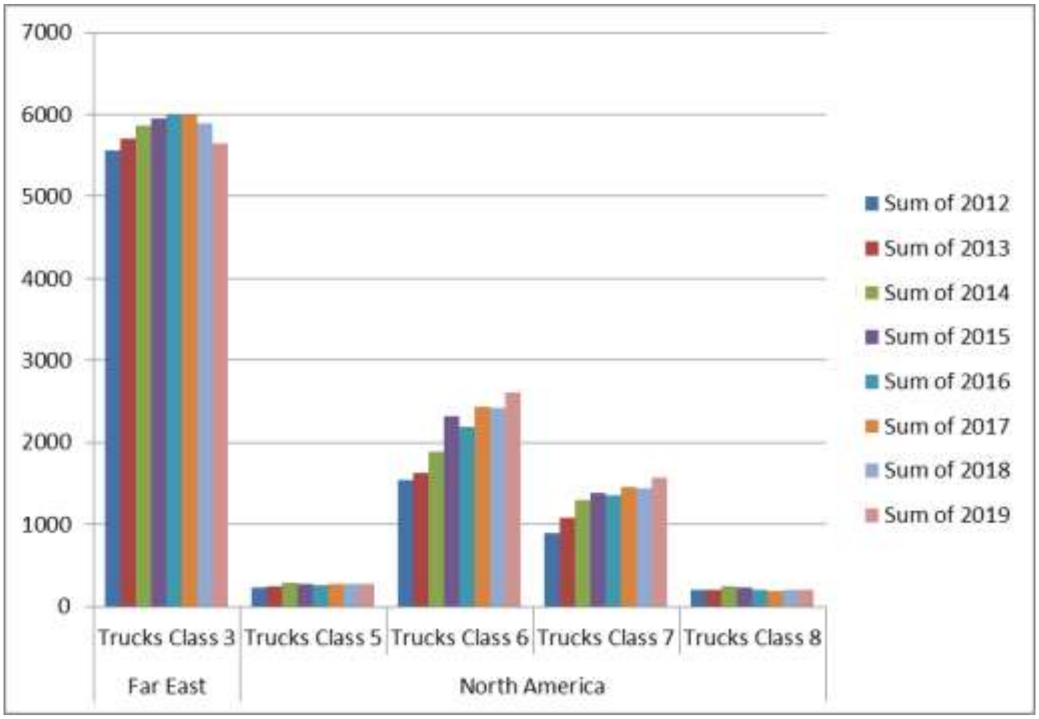


Figure 29. Hybrid truck programs in the product plans of OEMs (source: Eaton sales and marketing team)

Multi-speed transmission market estimate for electric trucks

Based on the electric truck market estimates discussed above, we can generate conservative and aggressive scenarios for multi-speed transmission volumes for electric trucks. In the conservative scenario the electric truck volumes will be halved in anticipation of significant improvement in the green-house gas emissions of ICDVs and the lack of government mandates. Furthermore, the multi-speed transmission adoption rates are anticipated to be between 20% and 40% since some EVs will be contended with a single speed gearbox. The total multi-speed transmission sale volumes are anticipated to average 20K and 16K units by 2023 for the MD&HD and RWD-LD markets respectively, as shown in Figure 30**.** The total volumes are 57.6K and 14.4K units for aggressive and the conservative scenarios.

Similarly, Eaton EV transmission market estimates can be based on two market share scenarios: 25% and 50% of the total market. In the most aggressive scenario Eaton sales will be 28K units and in the most conservative scenario Eaton shares will be 3.5K units as shown in Figure 31. The average Eaton sales will be 9K and 7K for the MD&HD and RWD-LD market segments respectively by 2023.

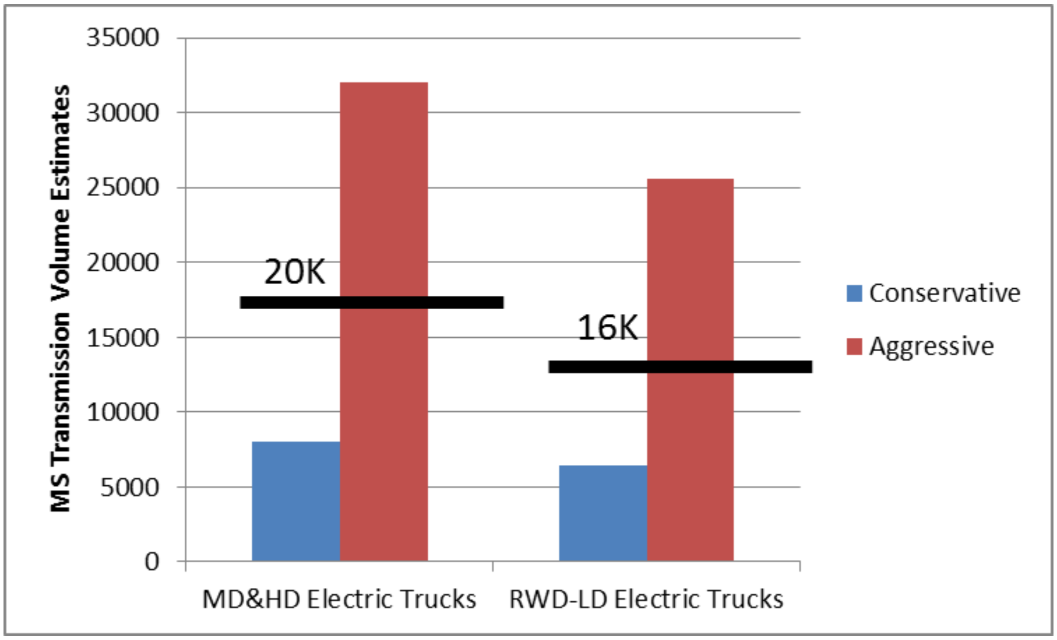


Figure 30. Total multi-speed transmission volume estimates by 2023

The big range of volumes between the conservative and the aggressive scenarios reflects market uncertainty in this very new market place. Even the volumes estimated by the aggressive scenario are modest compared to the volumes typically required to initiate a new development program. However, if the transmission concept is scalable across the vehicle segments and the costing and the pricing of transmission justify a reasonable return on investment then embarking on a new production program may find enough support within the Eaton community.

At this point, it is also appropriate to discuss the impact of a multi-speed transmission on the market penetration of medium-duty electric trucks. We expect that the availability of a multi-speed transmission for electric trucks will enable battery savings and electric motor downsizing, thereby reducing the cost while bringing the performance of an electric truck up to the same level as that of a diesel truck. These advancements are likely to increase the market acceptance of medium-duty electric trucks towards achieving the aggressive scenario.

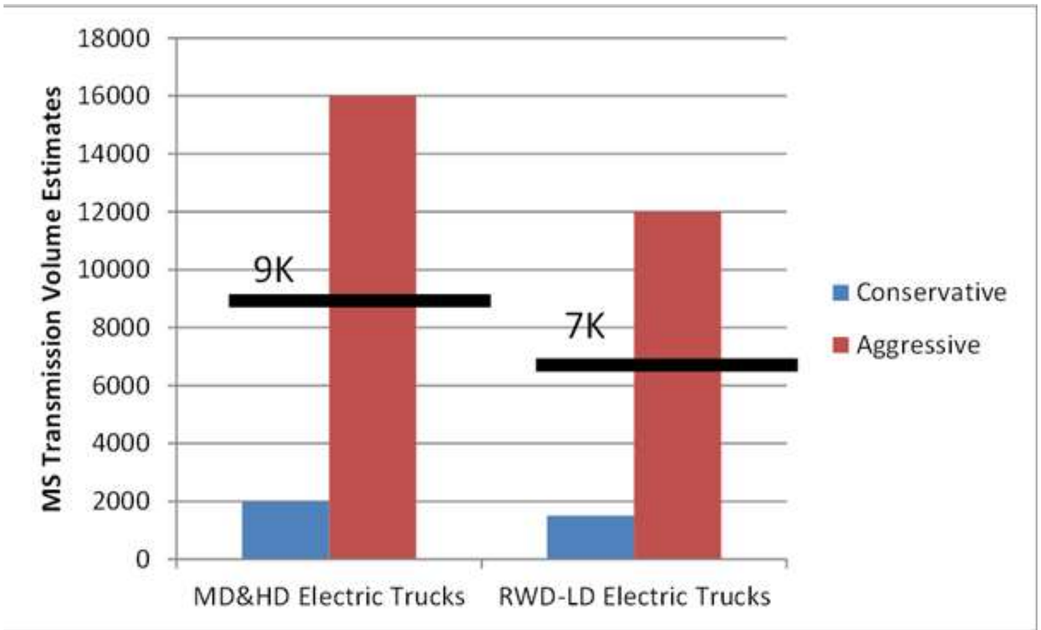


Figure 31. Eaton multi-speed transmission volume estimates by 2023

Subtask 3.2 – Transmission Price Targets for Market Segments

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 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 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.

TASK 4. BASELINE VEHICLE MODEL DEVELOPMENT

In Task 4, Eaton and ORNL conducted vehicle modeling activities separately based on the baseline vehicle information (Table 14). 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.

Table 14. Task 4 Activities

Task Name	Start	Finish
Task 4 – Baseline vehicle model development (Milestone 3)	4/1/15	6/30/15
4.1 – Create vehicle level model architecture	4/1/15	6/30/15
4.2 – Populate component models	4/1/15	6/30/15
4.3 – Baseline model development	4/1/15	6/30/15
4.4 – Validate baseline model	4/1/15	6/30/15

Deliverables: Baseline Vehicle Model Development Report

Subtask 4.1 – Create Vehicle-Level Model Architecture

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 32). 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 32. Autonomie representation of battery electric medium-duty truck architecture

Subtask 4.2 – 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 320V based on experimental data, even though battery specifications called for 343V.
- Battery energy capacity was set to 80kWh.
- Internal resistance was calibrated to match experimental data

<u>Electric machine and inverter</u>

- The machine’s peak power is 150kW, and its continuous power is 80kW.
- Peak torque is 600Nm, 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 33 for torque and efficiency map used in the model).
- Motor rotational inertia was estimated to be 0.1kgm^2.

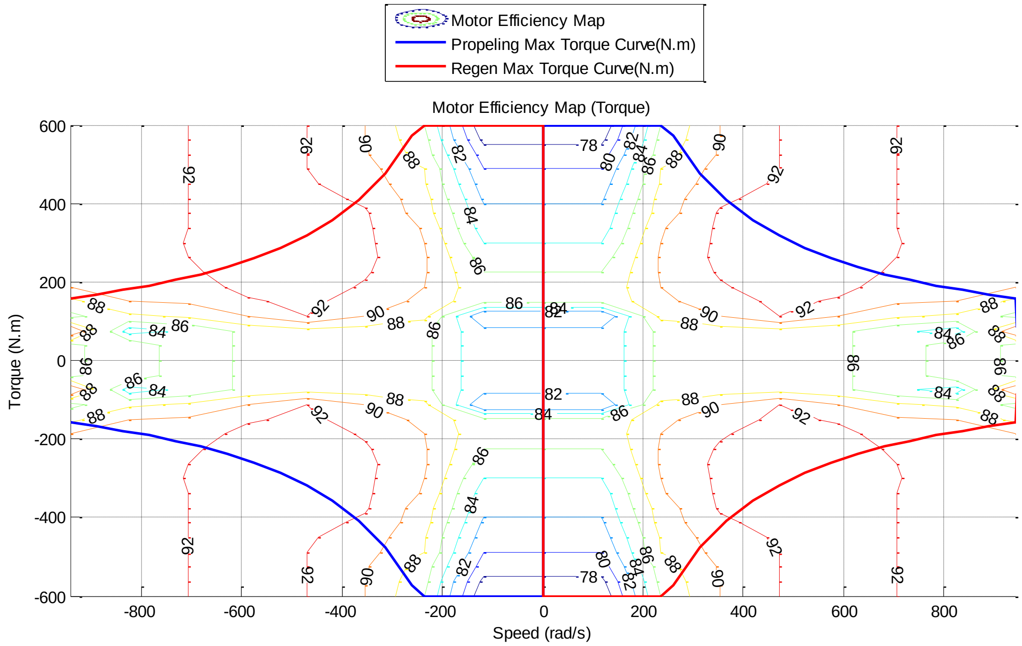


Figure 33. Electric machine and inverter efficiency map and maximum torque curve

Single speed transmission

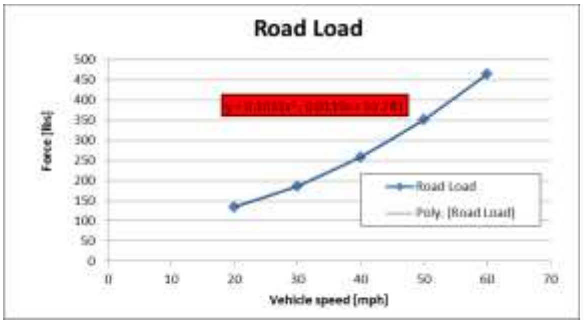
- Ratio of 3.41
- Eaton suggested an efficiency that is speed dependent: from 98% at low speeds down to 86% at 10,000 rpm

Final drive

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

Wheels

- Smith Electric provided coast down test data to calculate road load coefficients and rolling resistance as well as aerodynamic drag, shown in Figure 34.
- 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 34. Road load coefficients calculations

<u>Chassis</u>

- 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 4135 kg
- Gross Vehicle Weight (GVW) is 9990 kg

<u>DC-DC converter</u>

- Efficiency of 90% was provided by Smith Electric.

<u>Electric accessories loads</u>

- Estimated to be 1000 W based on experimental data

Subtask 4.3 – Baseline Model Development

Component models and parameters described in Subtask 4.2 were integrated in the vehicle architecture described in Subtask 4.1.

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 240A 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 medium-duty 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.

Subtask 4.4 – Validate 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 54 mph, 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 35 illustrates an example of model correlation for a wide-open acceleration test performed on a 5000 kg vehicle

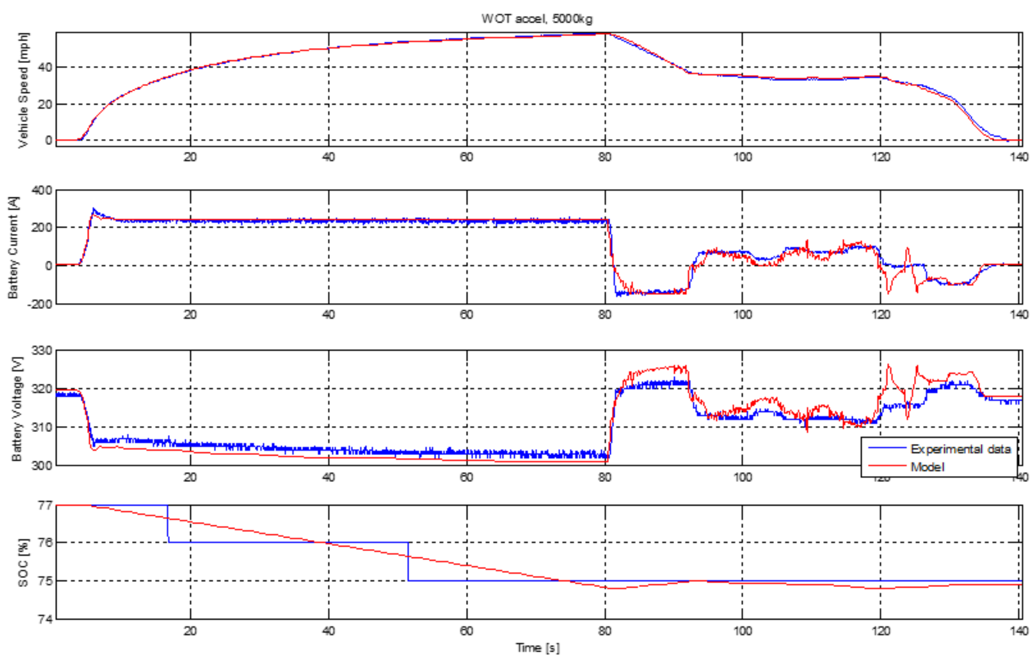


Figure 35. Baseline model correlation with experimental data. Wide open acceleration test performed on a 5000kg vehicle.

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 36)
- Combined International Local and Commuter Cycle (CILCC) (Figure 37)

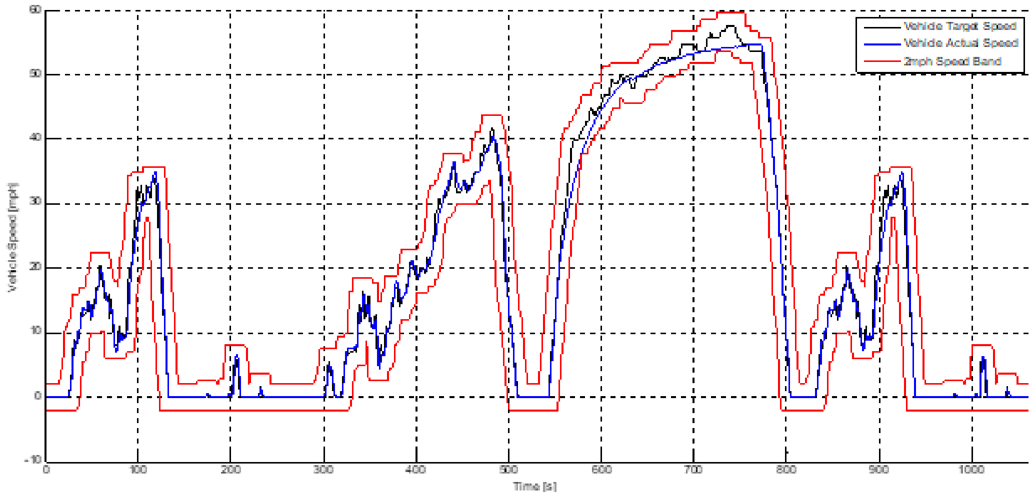


Figure 36. Baseline model on UDDS truck Cycle

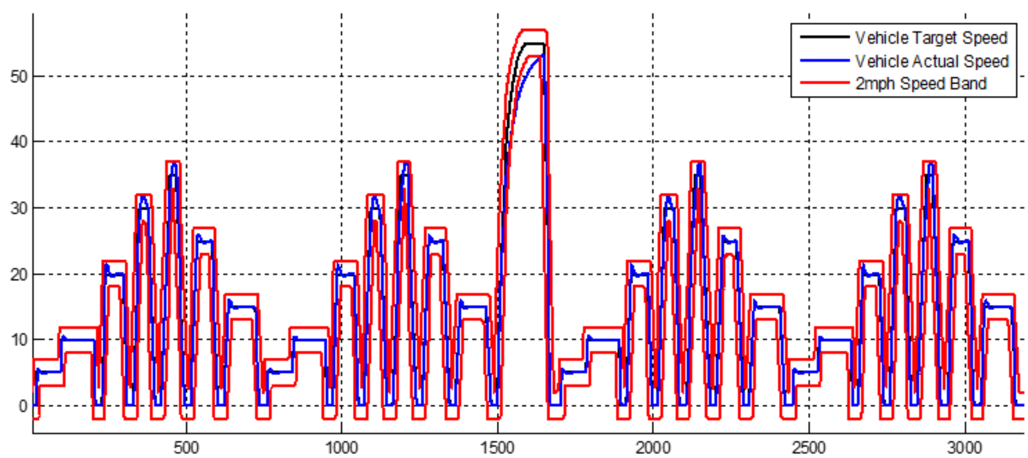


Figure 37. Baseline model on CILC Cycle

The baseline model does not match vehicle energy consumption numbers listed in the FOA, as shown in **Error! Not a valid bookmark self-reference.**. 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 16, 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 15. 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 16. Characteristics of the baseline medium-duty 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 17) to quantify the benefit of a multi-speed transmission because the baseline model has been validated against experimental data.

Table 17. 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 18 lists the energy consumption of the modified baseline single-speed transmission vehicle.

Table 18. 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 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 38.

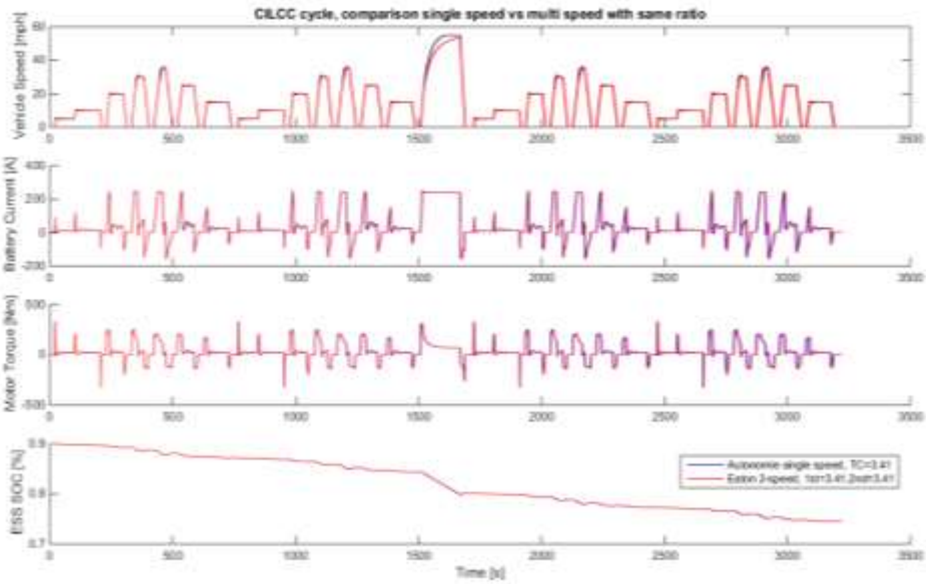


Figure 38. Comparison of single speed (blue) versus multi-speed (red) transmission on vehicle, motor, and energy storage system (ESS) behavior.

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 39 to Figure 43. Both Eaton and ORNL models point at selecting as high ratios as possible.

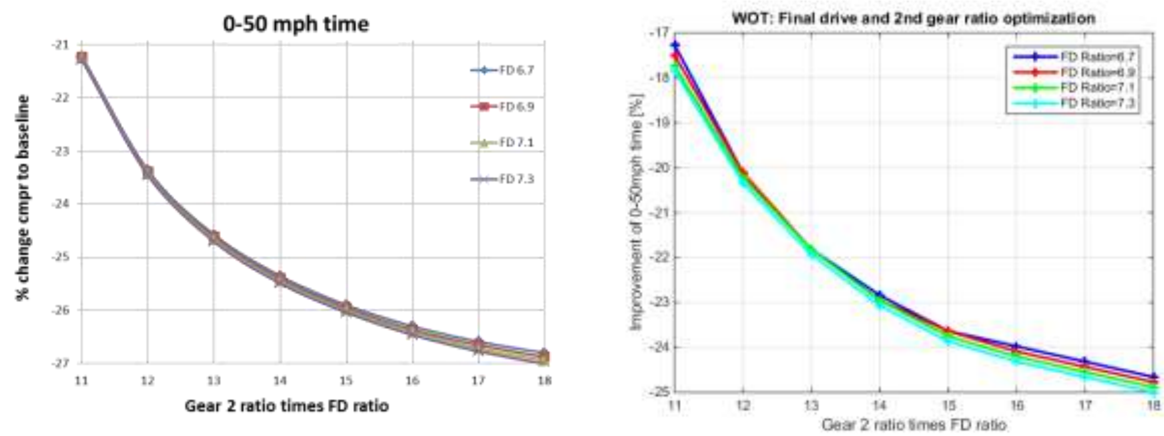


Figure 39. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for 0-50 mph acceleration.

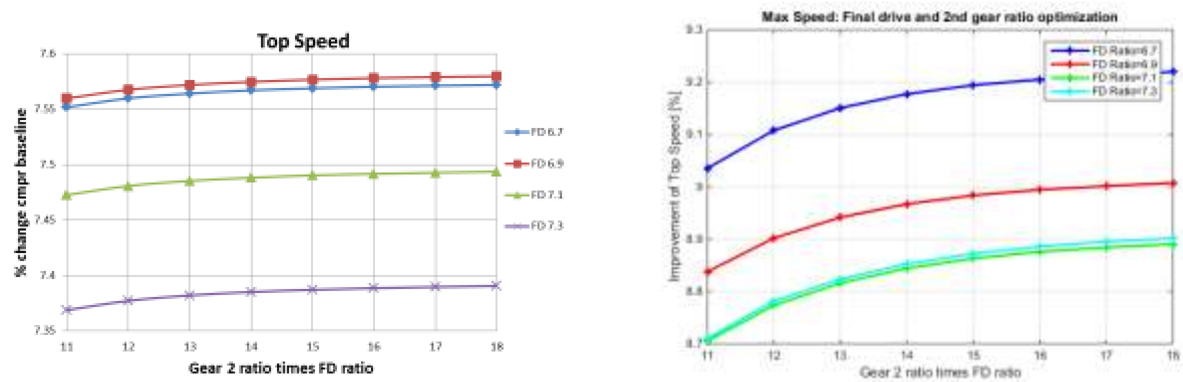


Figure 40. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for top speed.

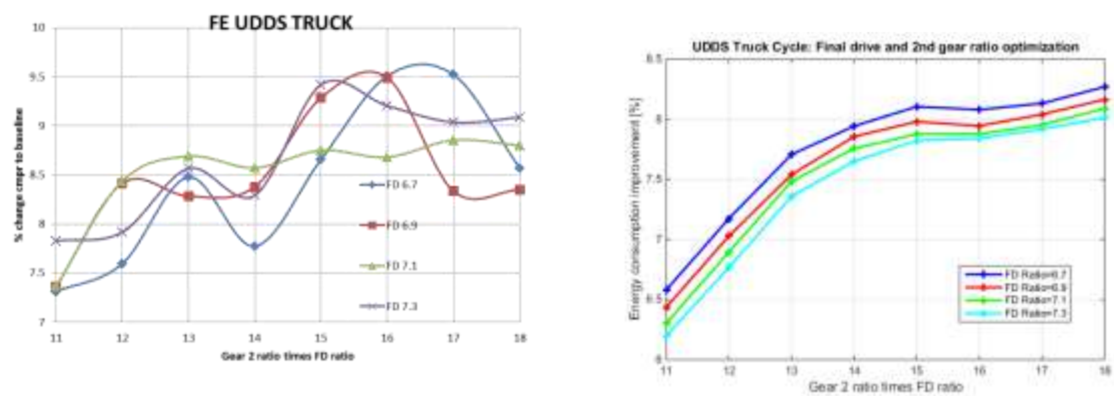


Figure 41. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for energy efficiency improvement on the UDDS drive cycle.

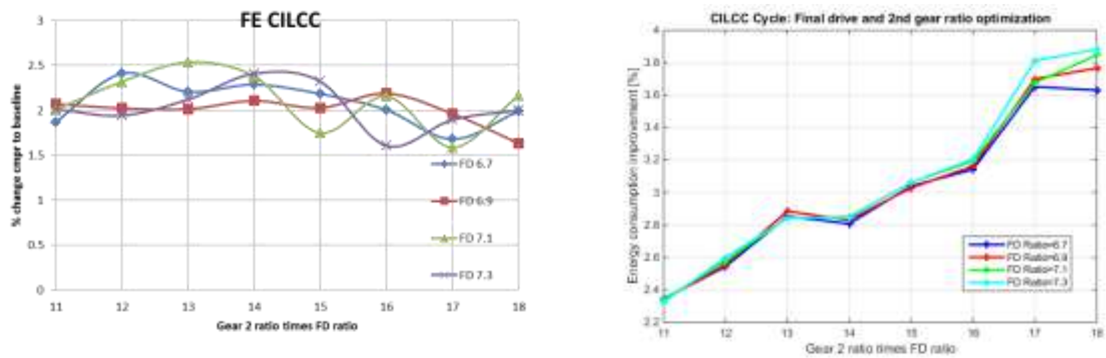


Figure 42. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for energy efficiency improvement on the CILC drive cycle.

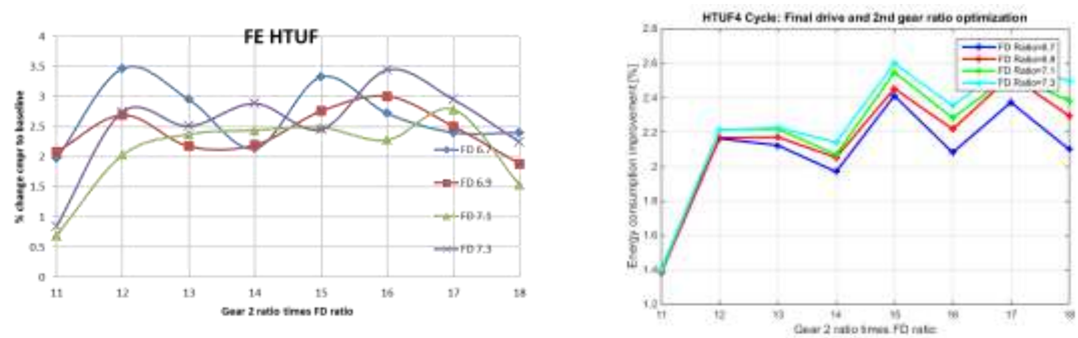


Figure 43. Comparison of Eaton model (left) and ORNL model (right) when optimizing the final drive ratio and the 2nd gear ratio for energy efficiency improvement on the HTUF4 drive cycle.

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 44

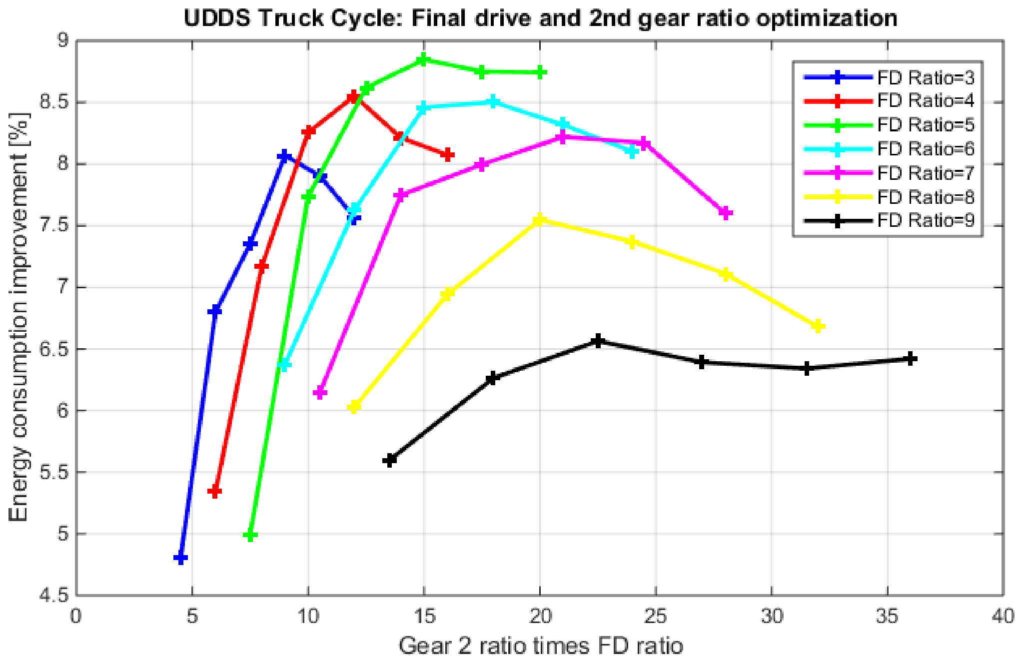


Figure 44. Extended optimization study results for energy efficiency improvement on the UDDS truck drive cycle.

Lower speed and more delivery-type cycles like CILCC and HTUF4 did still better, with the higher final drive ratios, as shown in Figure 45.

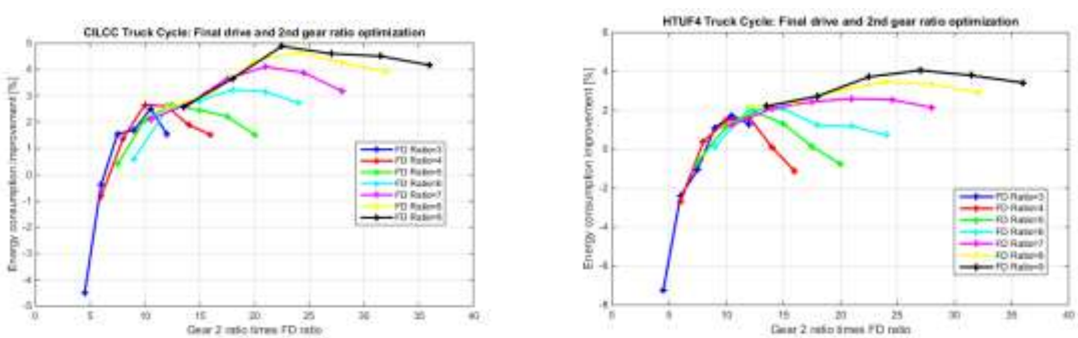


Figure 45. Expended optimization study results for energy efficiency improvement on the CILC and HTUF4 drive cycles.

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 46.

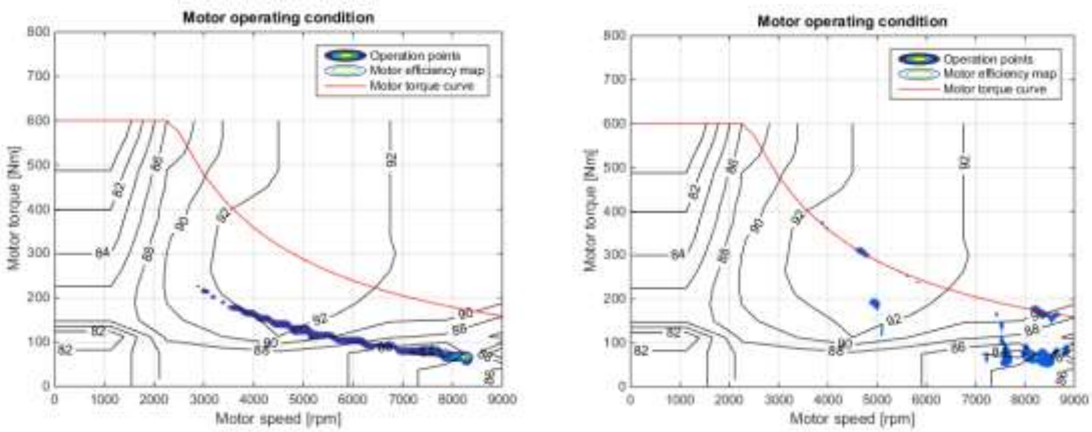


Figure 46. Motor operating condition on same UDDS truck cycle with battery limitation (left plot) and without limitation (right plot)

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 47.

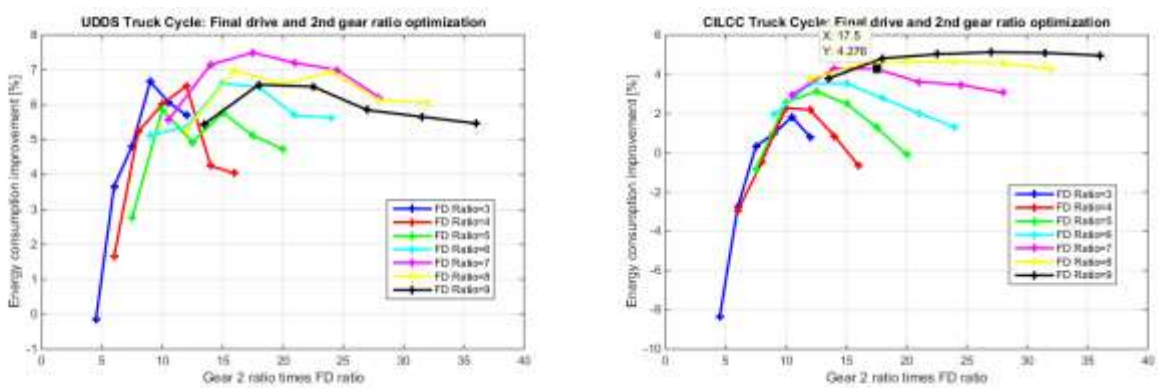


Figure 47. Final drive ratio and the second gear ratio optimization study conducted on modified model without battery restrictions on UDDS truck and CILC cycles

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 48 and won't exceed 5500 rpm with a multi-speed transmission as shown on the right graph of Figure 48.

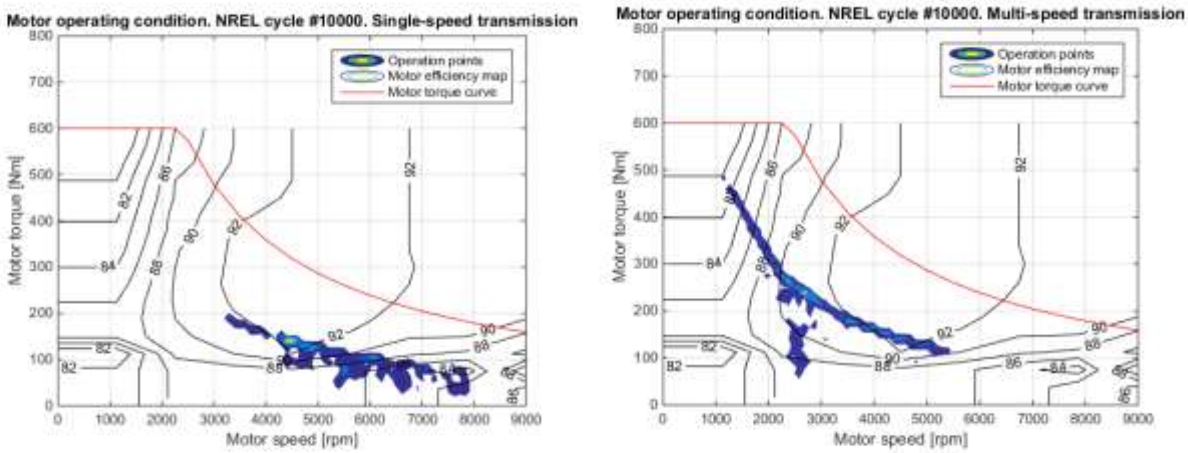


Figure 48. Motor operating conditions with single speed transmission (Left) and multi-speed transmission (Right) on NREL cycle #10000

When the top cycle speed does not exceed 40 mph (like NREL cycle #10028), then motor speed stays below 6000 rpm with a single speed transmission, as shown on the left side graph of Figure 49. 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 49.

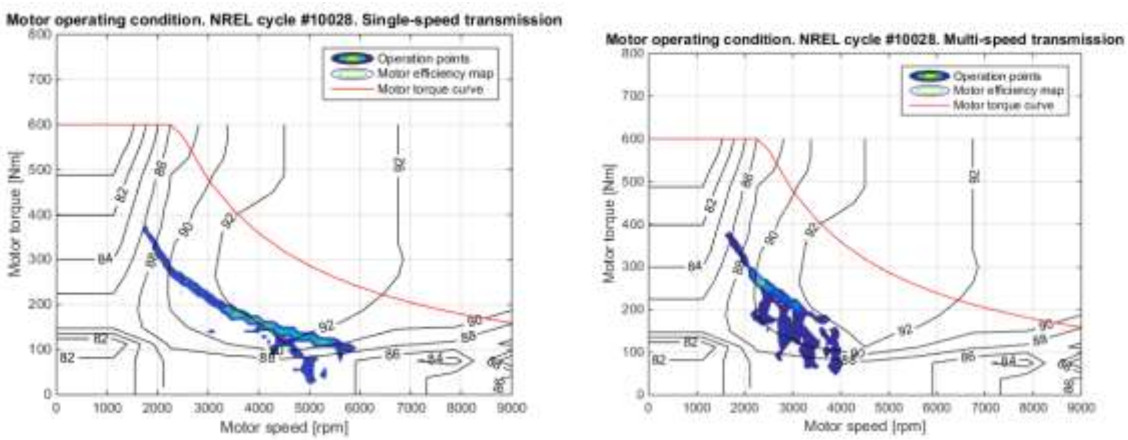


Figure 49. Motor operating conditions with single speed transmission (Left) and multi-speed transmission (Right) on NREL cycle #10028

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 50 shows the efficiency gain for each individual cycle:

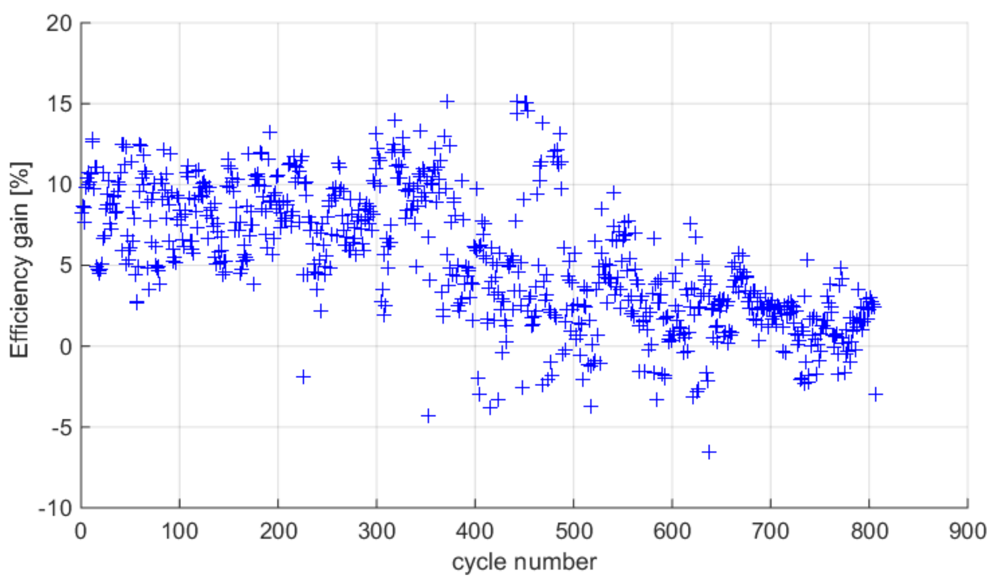


Figure 50. Efficiency gain obtained on each cycle when comparing a three-speed transmission against a single speed transmission

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 51 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.

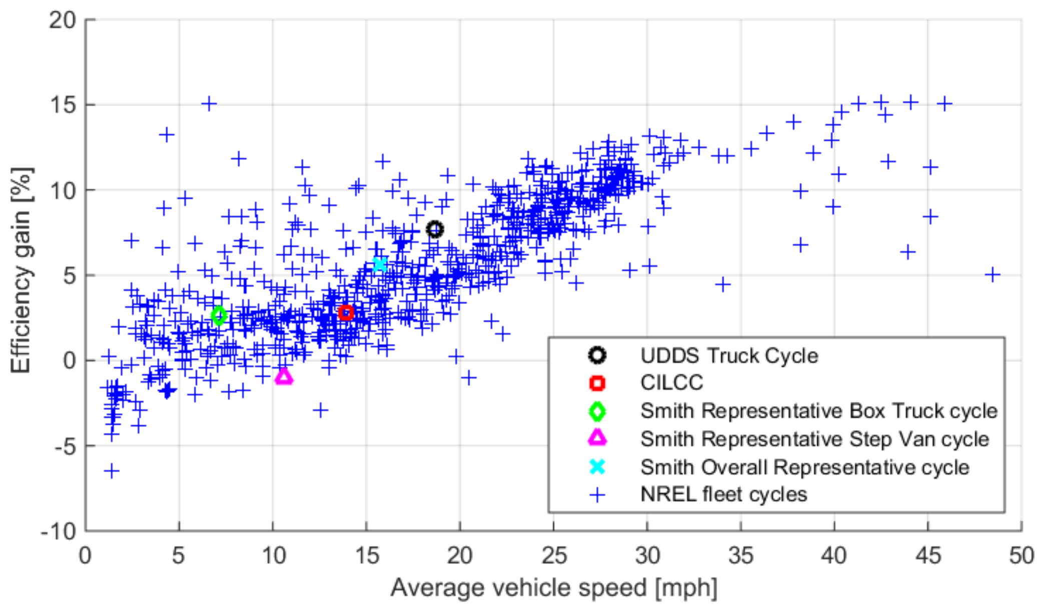


Figure 51. Efficiency gain as a function of vehicle speed for each real-world cycle and standard drive cycles.

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 52.

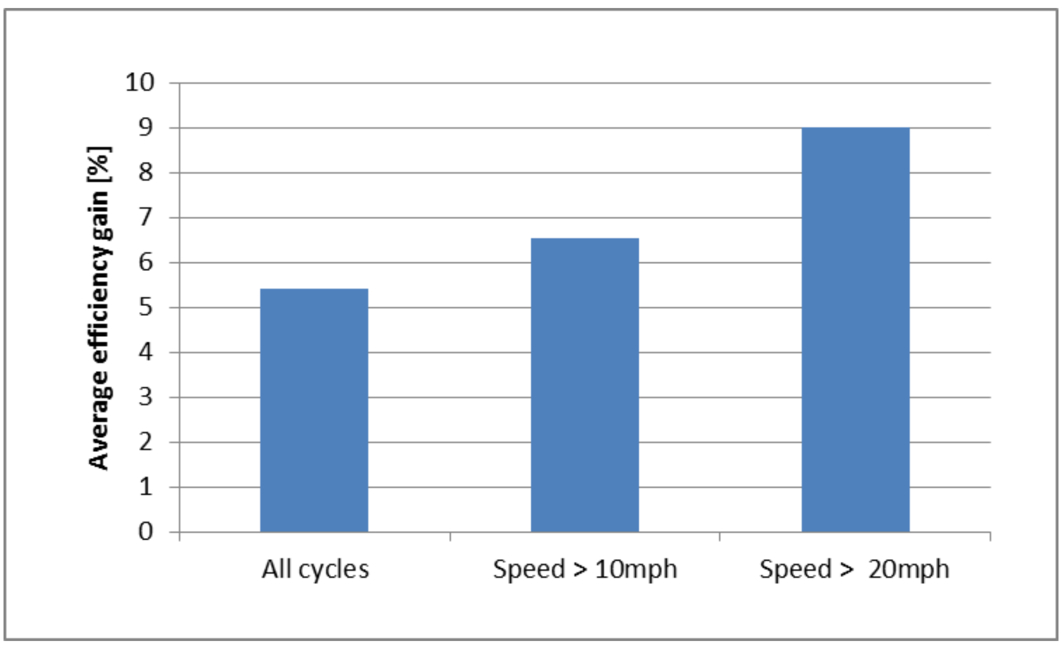


Figure 52. Average efficiency gain for all cycles, average speed above 10 mph and above 20 mph

TASK 5. TRANSMISSION CONCEPT DEVELOPMENT

Identifying end customer requirements is a key aspect in the development of a multi-speed transmission for electric vehicles. We followed the Design for Six Sigma (DFSS) methodology in the development of a multi-speed transmission for electric vehicles (Table 19).

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 Automated Mechanical Transmission (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.

Table 19. Task 5 Activities

Task Name	Start	Finish
Task 5 – Transmission concept development	7/1/15	9/30/15
5.1 – Identify transmission functional requirements	7/1/15	9/30/15
5.2 – Trade off analysis and selection of transmission	7/1/15	9/30/15
5.3 – Gearbox model development	7/1/15	9/30/15
5.4 – Optimization of the number of gears, ratios, and shift strategy	7/1/15	9/30/15

Deliverables: Breadboard Transmission Platform Selection Report

Subtask 5.1. Identify Transmission Functional Requirements

The team used voice of the customer techniques to collect customer wants and needs for the electric vehicle manufacturers who are the customers for electric vehicle transmissions. The wants and needs were organized under performance- and business- related categories and ranked for importance. Then, the functional criteria under each category were identified and ranked for importance. These steps led us towards constructing a house of quality and implementing the quality function deployment.

Performance criteria for multispeed transmission selection

Performance criteria for selecting a transmission concept were identified as top speed, acceleration, range or efficiency, gradeability— that is, top speed on a 2% grade, startability—

the maximum grade at which the vehicle launches and taps into the full power of electric motor in less than 3 seconds, reliability, comfort or NVH performance. We ranked relative importance of performance criteria in pair-wise comparisons using Eaton's Analytical Hierarchy Process (AHP) in a group setting. The team members ranked the importance of performance criteria for Smith Newton Chassis-Cab Drive Vehicle as shown in Table 20. The top five performance criteria were identified: reliability, acceleration, gradeability, startability, and efficiency. The exercise was repeated for a step van vehicle (Class-4), Table 21. For the step van, results were similar, except that the efficiency ranked slightly higher than gradeability and startability and the comfort ranked slightly higher for the step van and for the chassis cab.

Table 20. Ranked performance criteria for Smith Newton chassis-cab drive vehicle (Class-5-6).

Criteria Number	Criteria name	Relative rank (sums up to 1)
1	Reliability	0.36
2	Acceleration	0.18
3	Gradeability	0.125
4	Startability	0.125
5	Range (efficiency)	0.12
6	Top speed	0.06
7	Comfort (NVH etc.)	0.03

Table 21. Ranked performance criteria for Smith Newton step van vehicle (Class-4).

Criteria Number	Criteria name	Relative rank (sums up to 1)
1	Reliability	0.36
2	Acceleration	0.19
4	Range (efficiency)	0.13
3	Gradeability	0.12
5	Startability	0.12
6	Top speed	0.06
7	Comfort (NVH etc.)	0.04

Business criteria for selecting a multispeed transmission

We identified seven business criteria for selecting a multi-speed transmission: enabling motor downsizing, price, weight, ease of maintenance, application commonality, NRE costs, and capital expenses. Application commonality is related to application of the same transmission concept for a wide range of electric truck market segments such as medium, heavy, and light duty, as well as for the sub segments such as chassis cab, step van, school bus, etc. The project team used AHP to rank the relative importance of business criteria. The top four selection criteria were: capital expenses, price, non-recurring engineering costs, and application commonality for the selection of a multi-speed transmission for chassis cab vehicle, as shown in Table 22.

Table 22. Ranked business criteria for Smith Newton chassis-cab drive vehicle (Class-5-6).

Criteria Number	Criteria name	Relative rank (sums up to 1)
1	Capital costs	0.34
2	Price of transmission	0.21
3	NRE Costs	0.15
4	Application commonality	0.13
5	Enable motor downsizing	0.09
6	Ease of maintenance	0.05
7	Weight	0.03

Although AHP provides a forum for discussions and participation by all team members, the ranking is limited to by participants' background and opinion. The results may differ when different people provide rankings. For example, a team that is heavily represented by engineering functions may rank the criteria differently from a team heavily represented by sales and marketing functions. Hence, it would be useful to repeat the ranking exercise with a diverse set of people.

Subtask 5.2 – Trade Off Analysis and Transmission Selection

A two-day design burst event with both internal (Eaton, ORNL) and external (Ricardo, Means Industries, and Illinois Institute of Technology) participants was conducted on July 7 and 8, 2015. Mark Roser, a professional facilitator from Open Innovators led the event. The agenda included panel discussions of the electric vehicle market opportunities, future of electric motor technologies, and the electric truck performance requirements, followed by scenario-led discussions and a concept generation phase. Finally, the attendees challenged, improved, and ranked the concepts. The leading transmission concepts were:

- Eaton automated mechanical transmission (AMT) with 3/4 speed layshaft gear architecture
- Eaton wet dual clutch transmission (DCT) with four-speed layshaft gear architecture
- Eaton dry dual clutch transmission (DCT) with four-speed layshaft gear architecture
- Automated mechanical transmission (AMT) with three-speed planetary gear architecture
- Automatic powershift transmission with three-speed planetary gear architecture

These concepts were then subjected to trade-off analysis based on the leading performance and business criteria identified and discussed above. As a result of the review of decision criteria at the design burst event, we added one more criterion, market share potential. The importance of each criterion was discussed and ranked and their upper and/or lower limits determined.

We then used the trade-off analysis matrix shown in Table 23 to rank each transmission concept for the leading criteria. Eaton AMT with three- or four-speed layshaft gear architecture scored the highest points. The biggest advantages of the Eaton AMT concept are low NRE, low capital costs, high reliability, high efficiency, and better application commonality across market segments and vehicle classes. The first runner up concept was Eaton dry DCT with four-speed layshaft gear architecture. The dry DCT has better comfort and NVH performance than the AMT concept does; however, the importance of comfort and NVH is ranked much lower than the importance of unit cost, NRE cost, capital spending, and reliability— where the Eaton AMT concept has a clear advantage over the DCT concepts.

Table 23. Trade off analysis of medium-duty electric vehicle transmission concepts. Eaton-AMT-Layshaft (three- or four-speed) concept received the highest score.

MD-EV Transmission Concepts	Status Legend Pass = 1.0 Marginal = 0.5 Fail = 0.0 Score	NRE Costs	Application Commonality	Acceleration	Gradeability	Startability	Efficiency	Top speed	Comfort/NVH	Reliability	Use Current Tooling	Packaging and Weight
AMT-Layshaft (3 or 4-speed)	14.3											
DryDCT-Layshaft (4-speed)	11.8											
3-spd AMT-Planetary	8.4											
3-spd Wet-Powershift-Planetary	7.6											
WetDCT-Layshaft (4-speed)	11.4											

Subtask 5.3 – Gearbox Model 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.

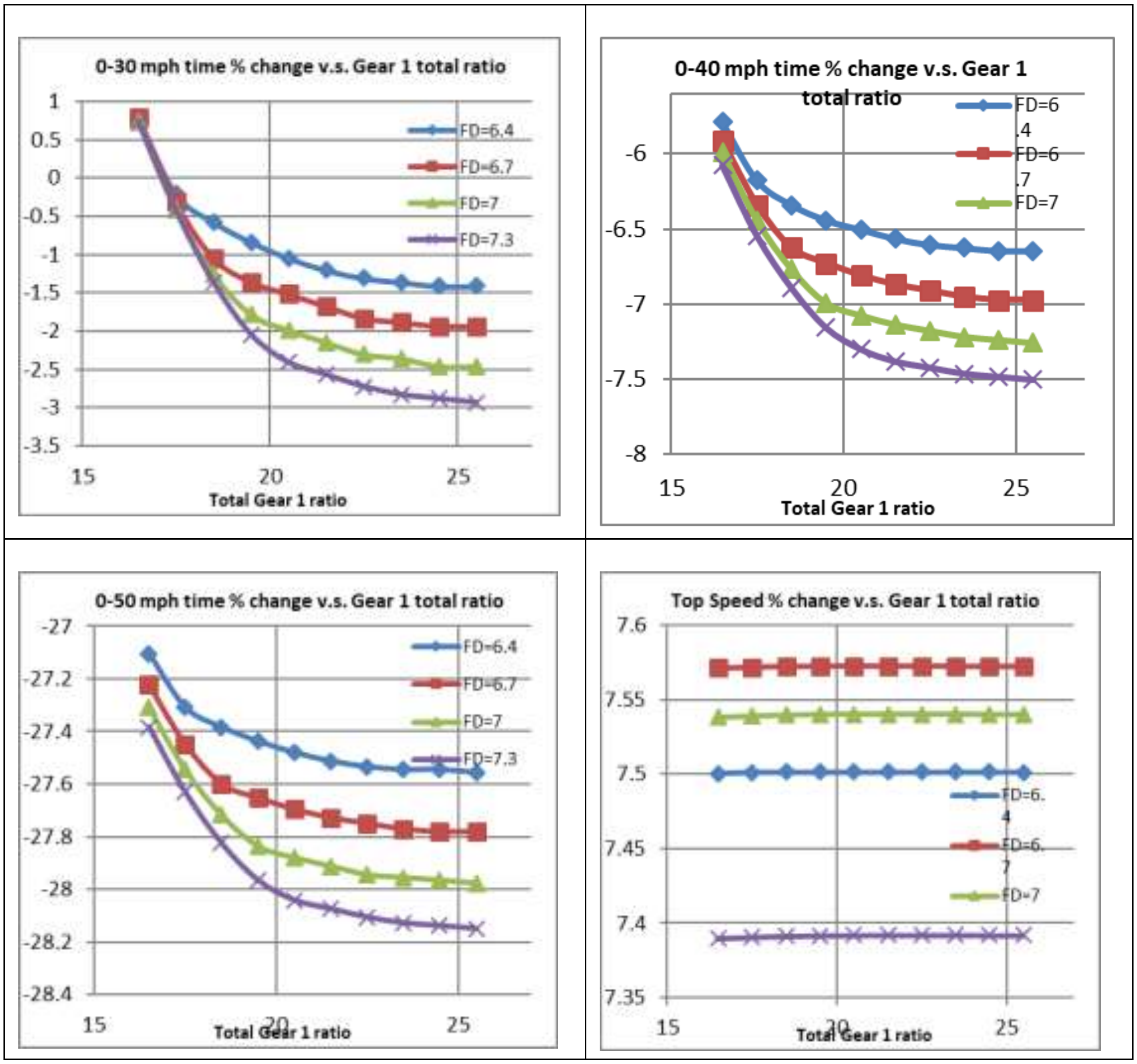


Figure 53 shows initial simulation results of acceleration performance and top speed of Smith-Newton electric truck generated by the validated vehicle model. The x-axes are the total Gear 1 ratio (Gear 1 ratio multiplied by final drive). The y-axes show the percent change of acceleration performance and top speed with two-speed transmission as compared to the single speed baseline. The 0-30 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.

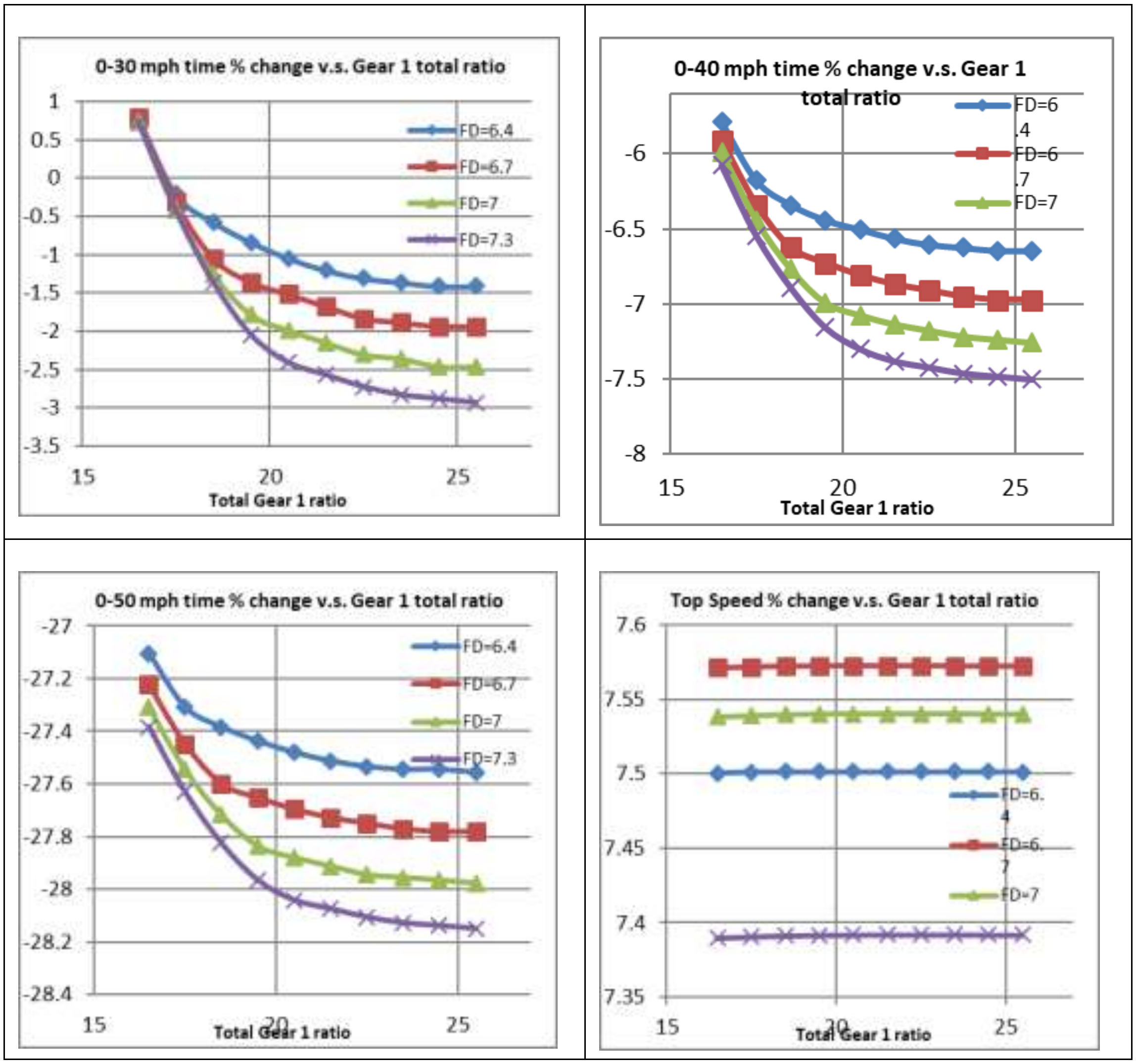


Figure 53. Acceleration and top speed gains of Smith-Newton electric truck with two-speed transmission for different FD and Gear 1 ratio values compared to the single-speed baseline.

Figure 54 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.

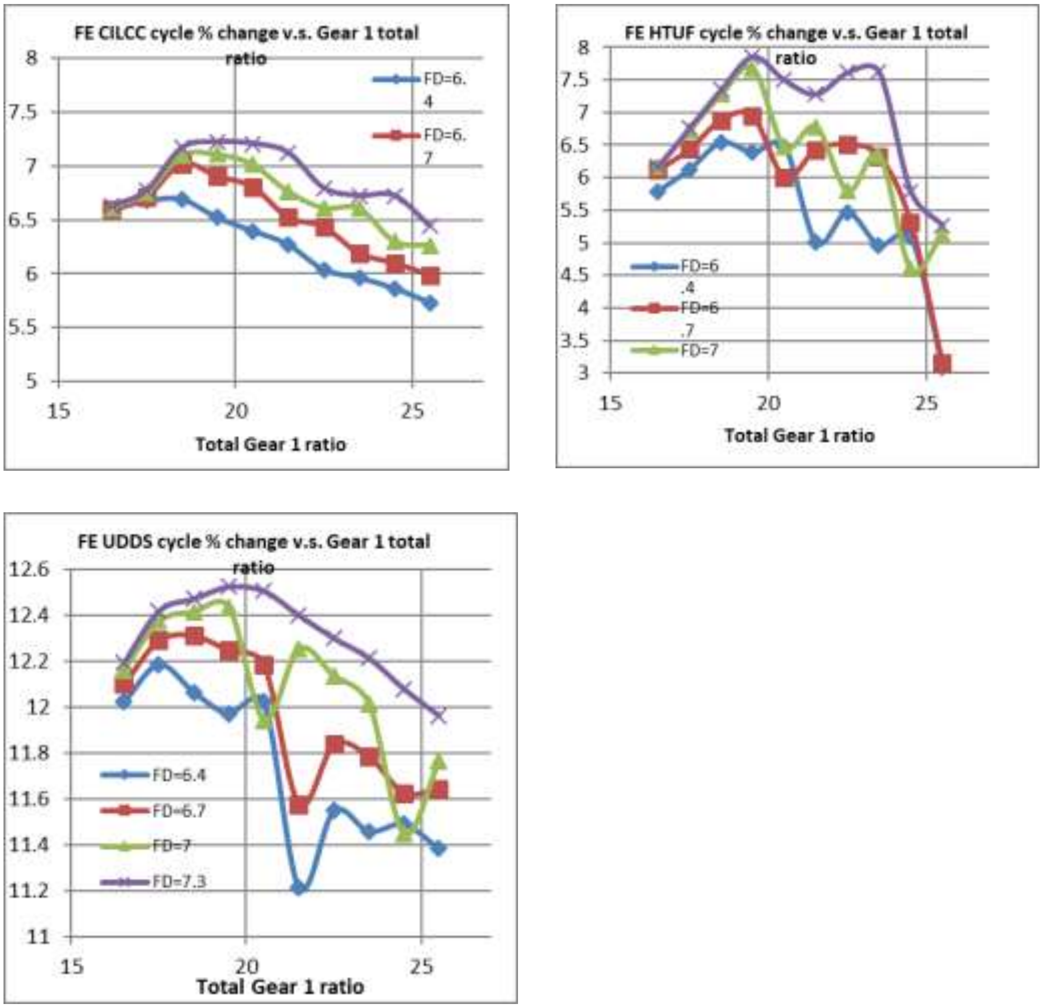


Figure 54. Energy efficiency gains of Smith-Newton electric truck with two-speed transmission for different FD and Gear 1 total ratio values compared to the single speed baseline.

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.

Subtask 5.4 – 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 55 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.

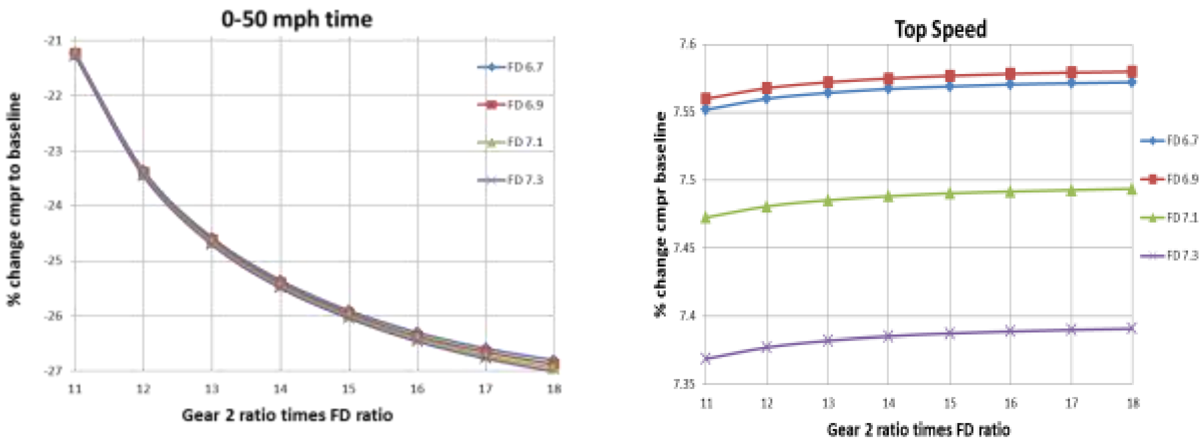


Figure 55. Acceleration and top speed gains of Smith-Newton electric truck with three-speed transmission for different FD and Gear 2 total ratio values compared to single speed baseline.

Figure 56 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 downspeeding 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.

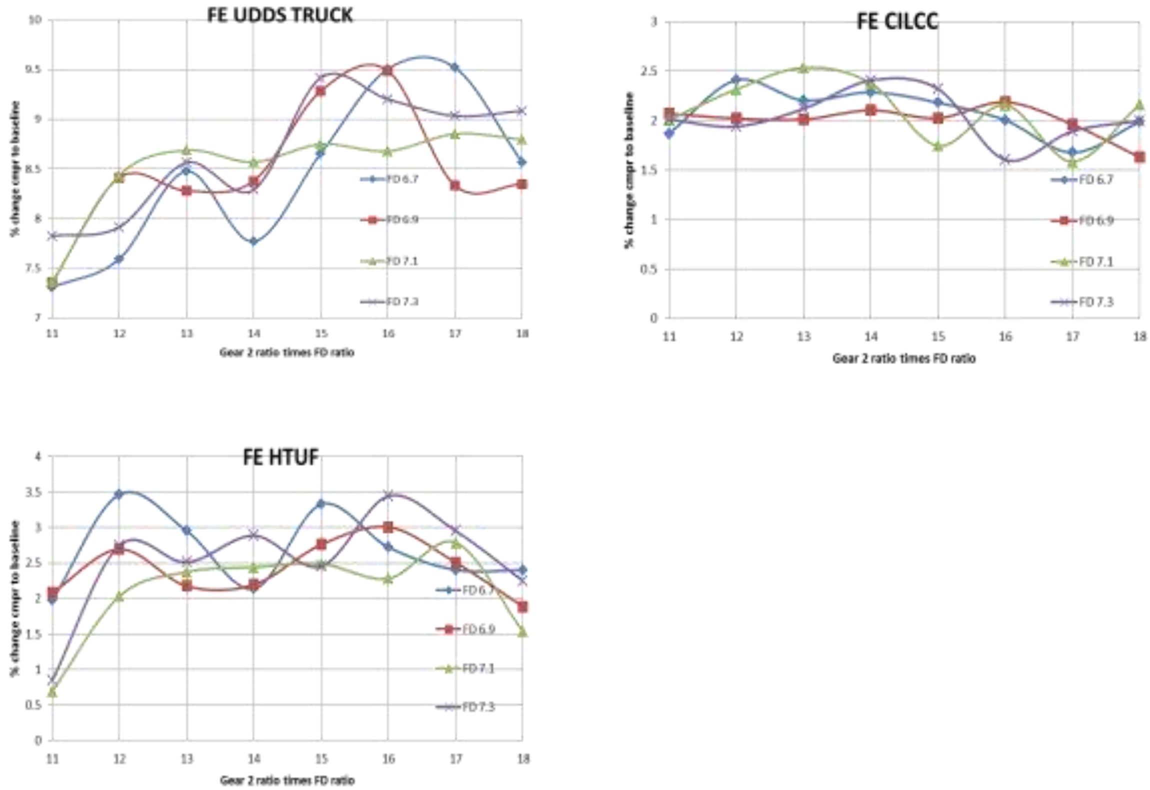


Figure 56. Energy efficiency gains of Smith-Newton electric truck with three-speed transmission for different FD and Gear 2 total ratio values compared to single speed baseline.

Next, we chose the Gear 1 ratio based on startability (launch acceleration) performance on a high-grade road, as shown in Figure 57 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 57. 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 24. 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 24. 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

TASK 6. EXTENDED TRANSMISSION MODELING ACTIVITIES TO OTHER EV POWERTRAIN

We extended the vehicle modeling and simulation studies to other medium-duty, medium-heavy and heavy-duty 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. Table 25 describes the activities included in this task.

Table 25. Task 6 activities

Task Name	Start	Finish
Task 6 – Extend the transmission modeling activities to EV powertrain	10/1/15	12/31/15
6.1 – Electric bus applications	10/1/15	12/31/15
Simulink model	10/1/15	12/31/15
Baseline model validation	10/1/15	12/31/15
Multi-speed transmission vehicle model creation	10/1/15	12/31/15
Acceleration performance and top speed optimization results	10/1/15	12/31/15
Energy efficiency optimization results	10/1/15	12/31/15
Electric transit bus application for China market	10/1/15	12/31/15
Multi-speed transmission study with downsized motor	10/1/15	12/31/15
6.2 – Light and medium-duty vehicle applications	10/1/15	12/31/15
Light-duty vehicle application	10/1/15	12/31/15
Medium-duty step van and school bus applications	10/1/15	12/31/15
6.3 – Refuse and drayage trucks	1/1/16	3/31/16
6.4 – Develop a new vehicle OEM partner	1/1/16	3/31/16
6.5 – Detailed transmission modeling on the electric vehicle selected for the integration of prototype transmission	1/1/16	3/31/16
6.6 – Adaptive downshift control strategy	1/1/16	3/31/16

Deliverables: Expanded Transmission Modeling/Simulation Report

Subtask 6.1 – 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 [13]. The New Flyer bus with GVW of 19,300 kg was fitted with a 200 kWh 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 4 and shown in Figure 32 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 [14]. Those tests consisted of the 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 58 shows the speed profile of tests carried out on the New Flyer XE40 electric bus.

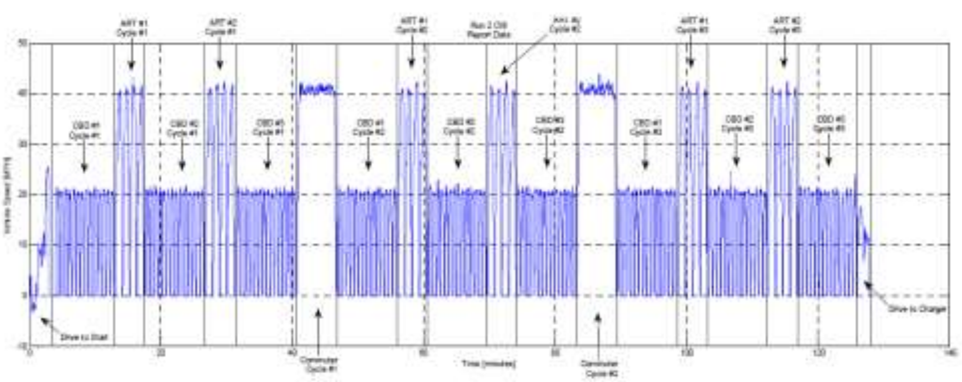


Figure 58. Altoona test profiles

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 59 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.

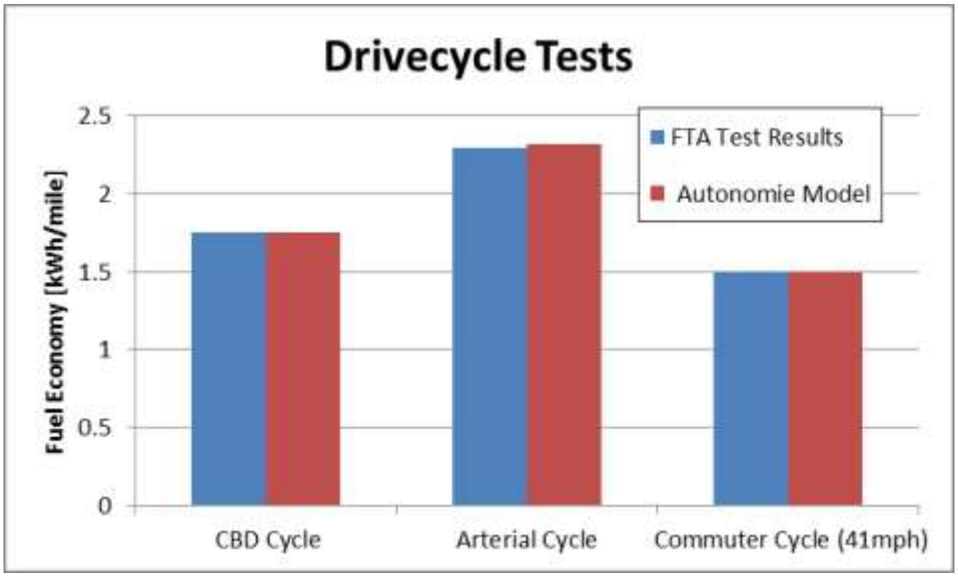


Figure 59. Model validation results for drive cycle tests

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 (labelled “Model”) in Figure 60.

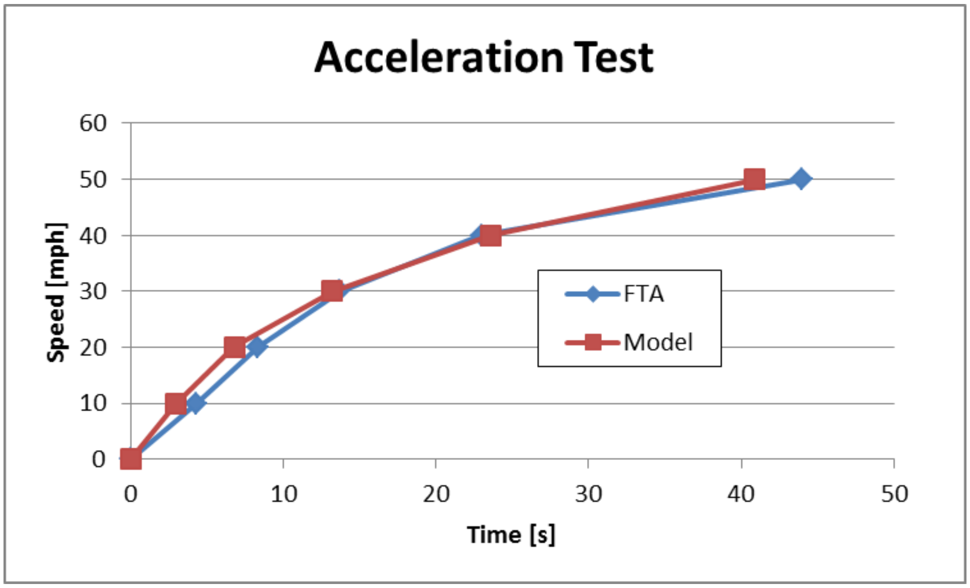


Figure 60. Model validation results for acceleration test

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 medium-duty delivery truck configuration presented in Task 4. 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 10 mph, zero to 50 mph 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 61.

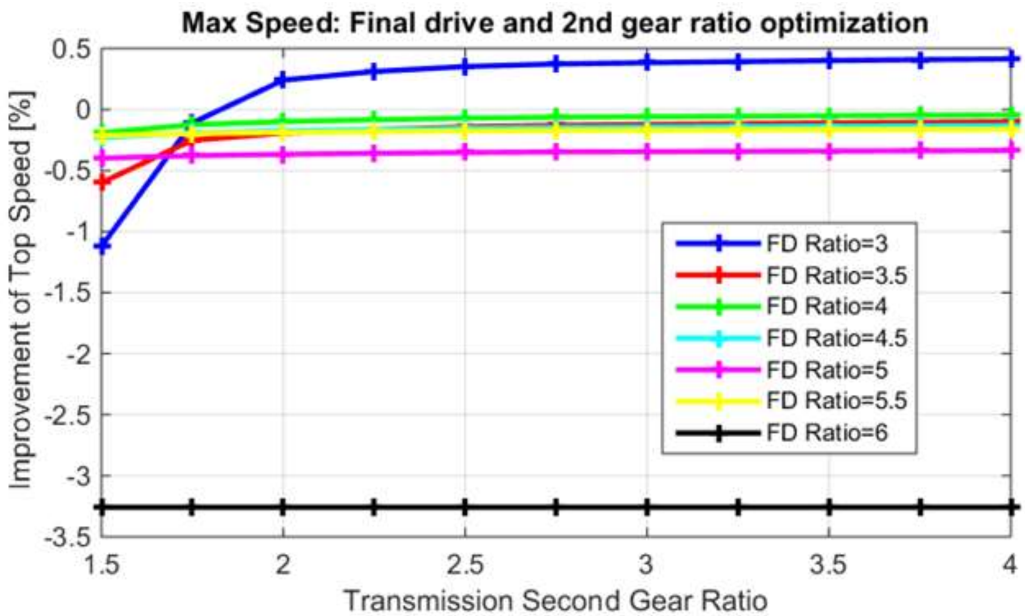


Figure 61. Final drive and second gear ratio effect on vehicle top speed

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 yields the most improvements as shown in Figure 62 and Figure 63.

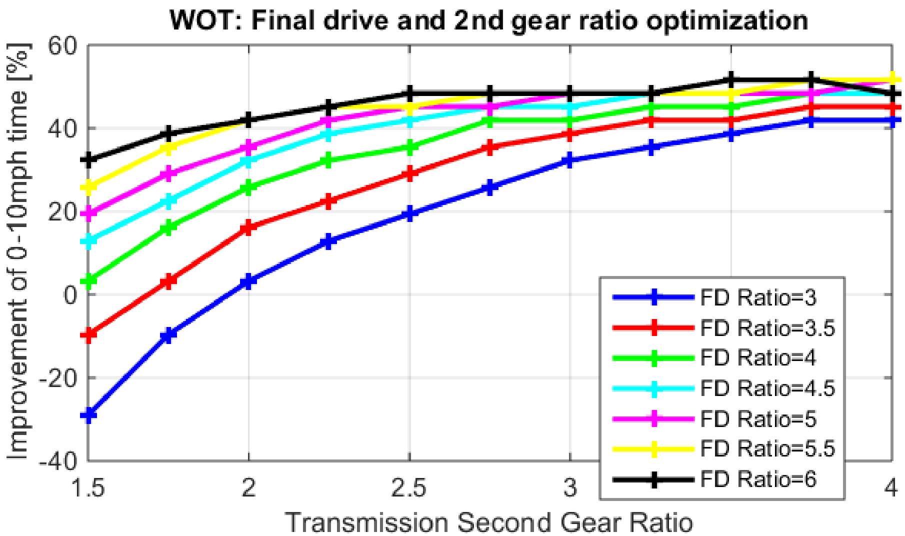


Figure 62. Final drive ratio and the second gear ratio effect on 0-10mph acceleration

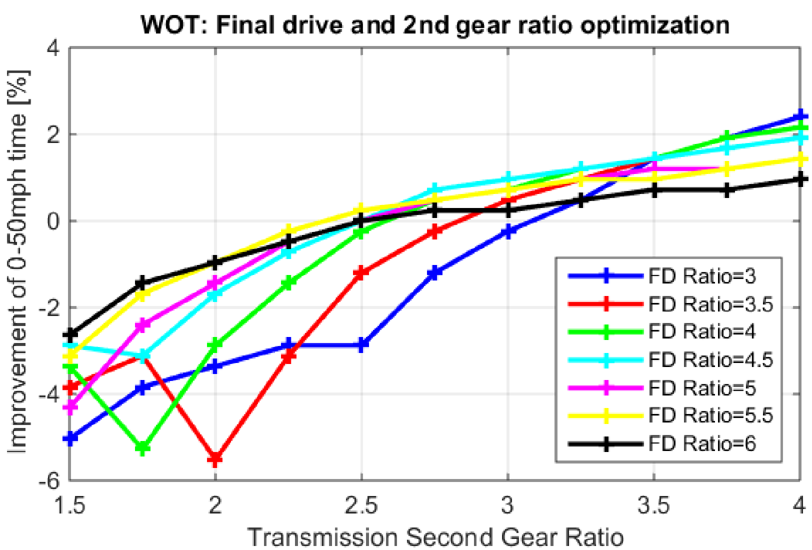


Figure 63. Final drive and second gear ratio effect on 0-50mph acceleration

Energy efficiency optimization results

The multi-speed transmission vehicle model was subjected to the same three cycles used to validate the baseline model: central business district, arterial and commuter cycles. Energy consumption for each cycle was modelled, 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 64 to Figure 66.

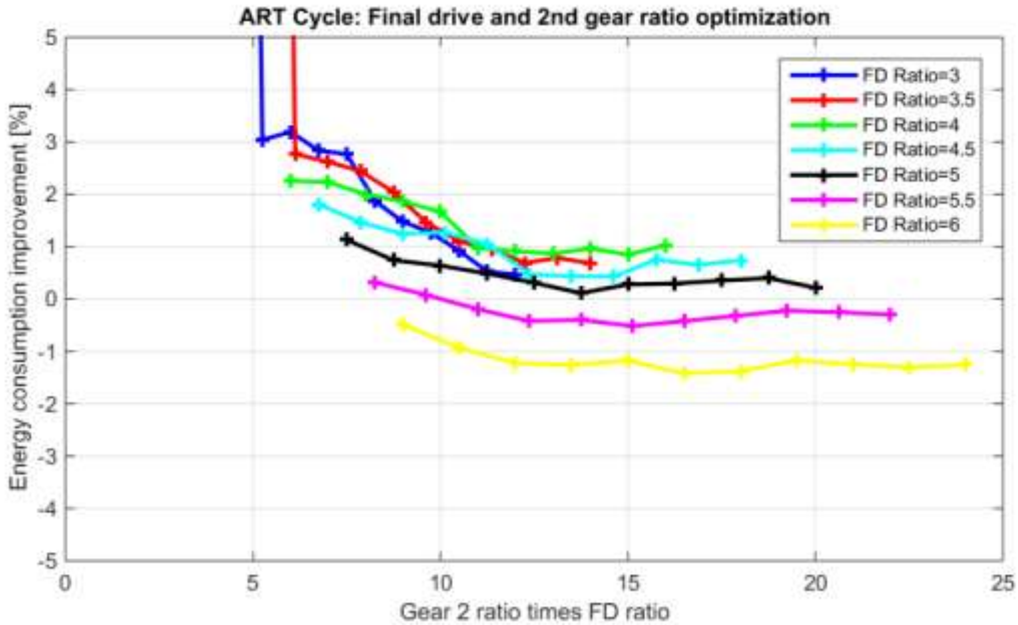


Figure 64. Final drive and second gear ratio effect on energy efficiency improvement of three-speed vehicle on arterial cycle.

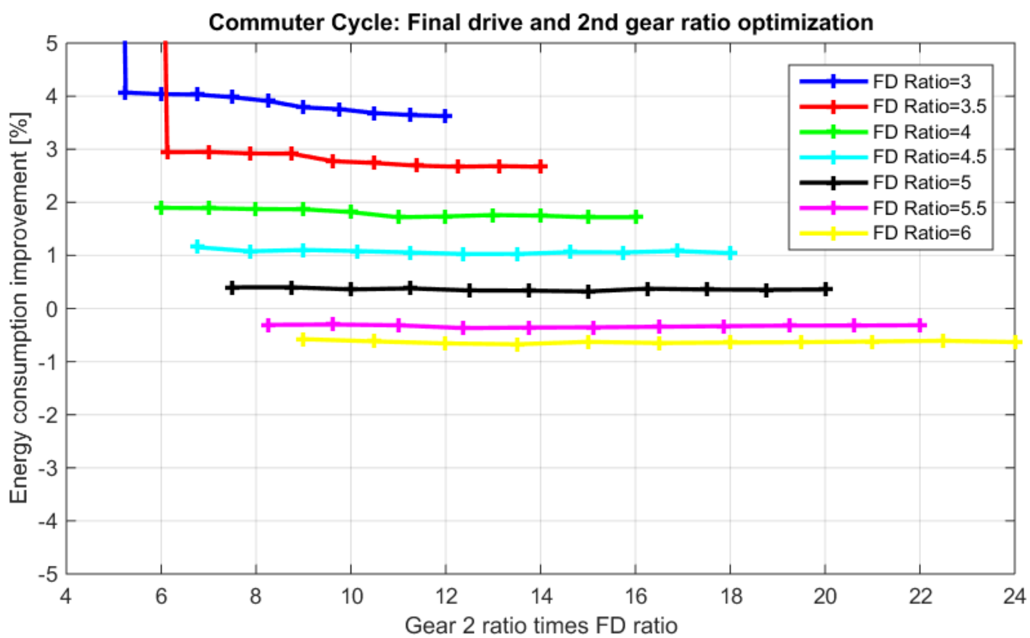


Figure 65. Final drive and second gear ratio effect on energy efficiency improvement of three-speed vehicle on commuter cycle.

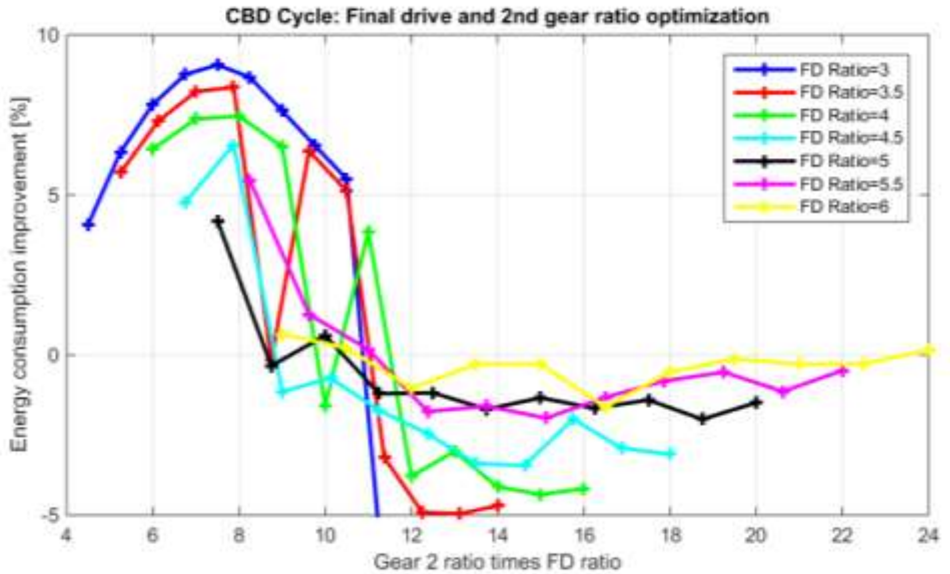


Figure 66. Final drive and second gear ratio effect on energy efficiency improvement of three-speed vehicle on CBD cycle.

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 26 lists the China city bus baseline vehicle parameters used to set up the model. Table 27 lists performance targets for the China market. Table 28 lists the city drive cycles are used to compare the energy efficiency of vehicles configured with and without transmission.

Table 26. 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 27. 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 28. 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 67. 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 68. We found that to meet the top speed requirement, the final drive ratio cannot be larger than 5.8. The 0-30 mph acceleration is in the range of 15 sec, and hence, the acceleration requirement does not impose any design limitation.

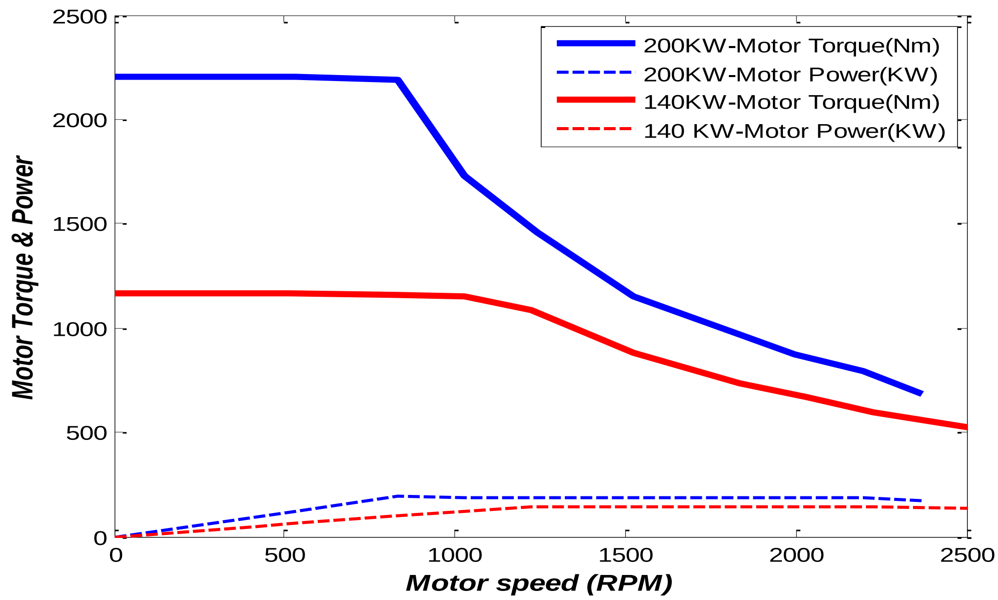


Figure 67. eDrive motor torque and power curves for 200/100KW (DD) & 140/80KW motors.

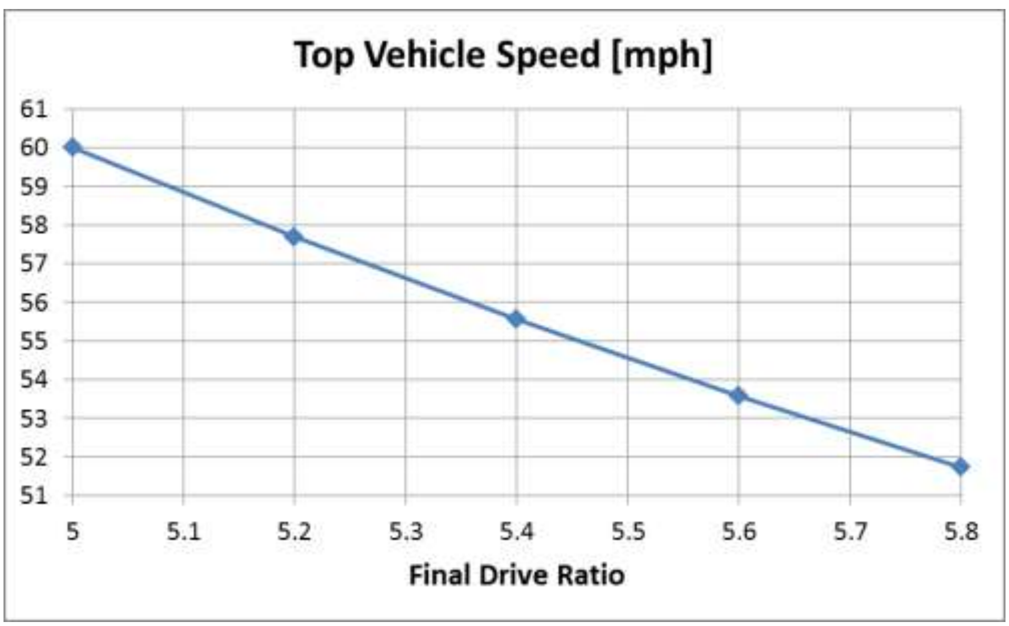


Figure 68. Top speed of 12 m electric bus as function of final drive ratio.

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 69**)**. With these ratio selections, we expect to gain 8% energy saving on average over the baseline for all drive cycles.

Error! Reference source not found. summarizes energy consumption reduction enabled by the hree-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 29. 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

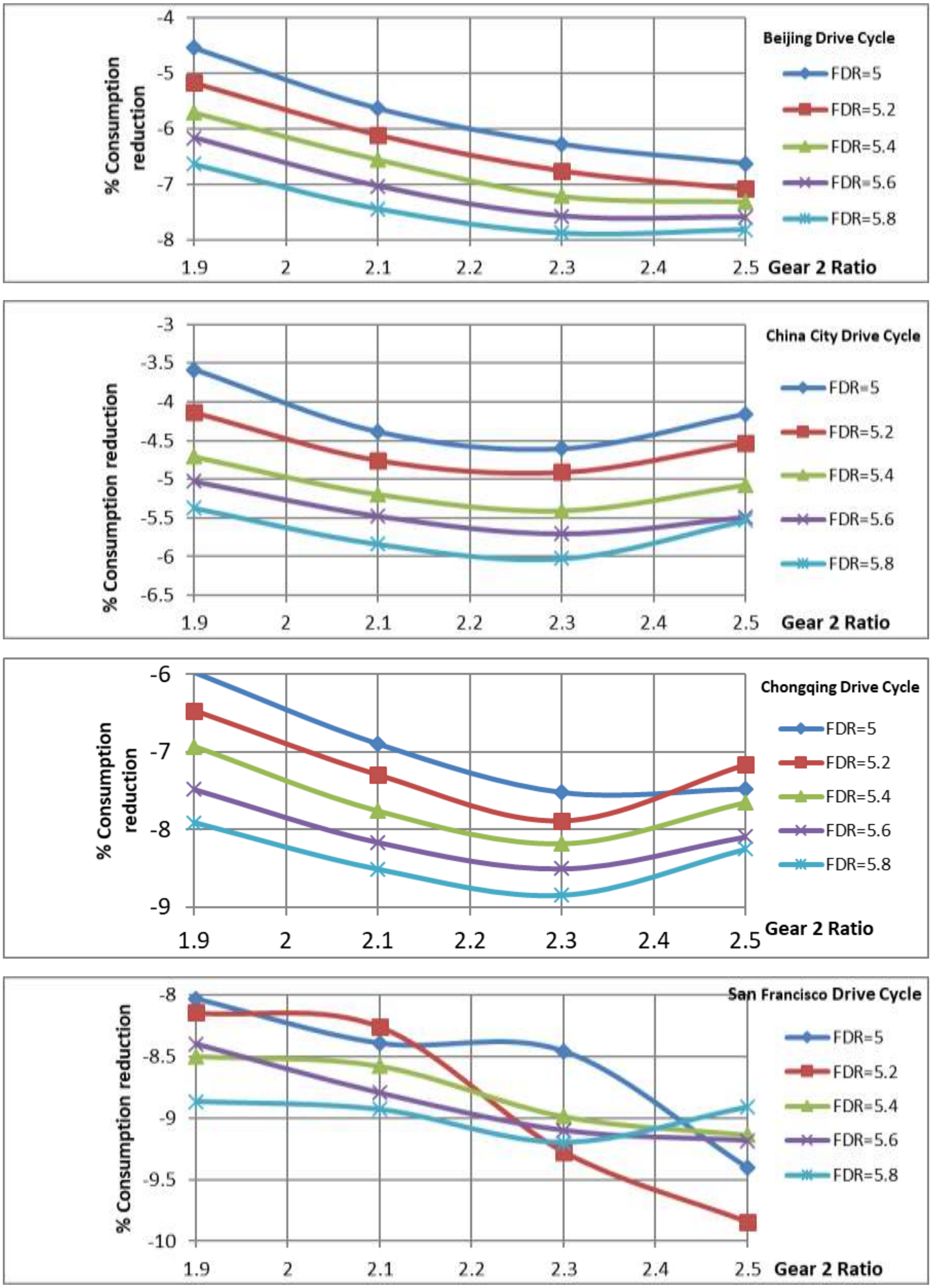


Figure 69. Energy savings with three-speed transmission and downsized electric motor compared to 12-meter direct drive baseline electric bus.

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 70. 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 30.

Table 30. 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 30, 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.

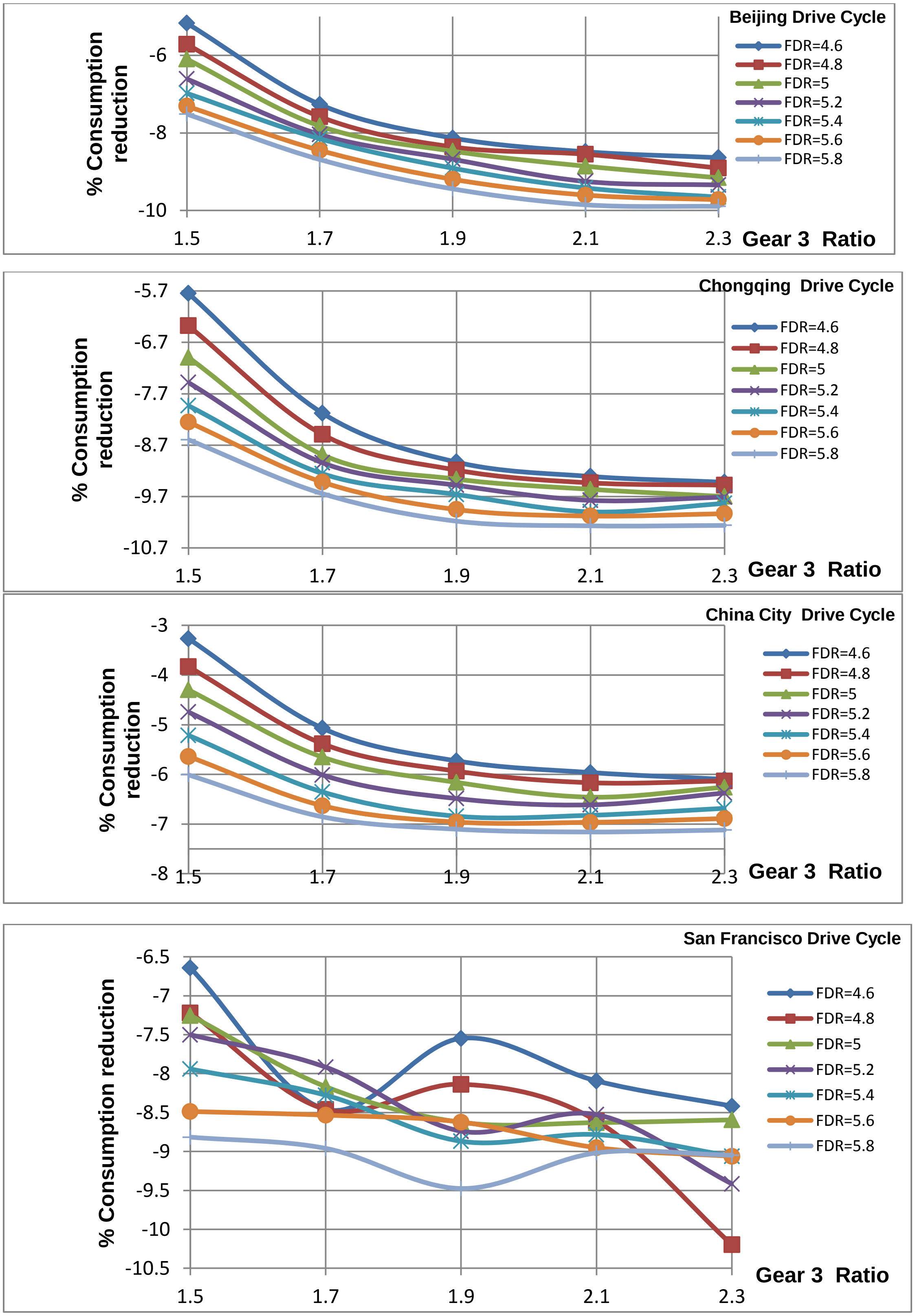


Figure 70. Energy savings with four-speed transmission and downsized electric motor as compared to 12-meter direct drive baseline electric bus.

Energy savings with four-speed over three-speed

Table 31 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 31, 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 31. 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 32 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 5th 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 32. 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%

Subtask 6.2 – 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 33 lists the publicly available information on the baseline vehicle [15].

Table 33. 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 71. Based on the simulation results, we chose 5 as the baseline final drive ratio to match the top speed of 86 mph.

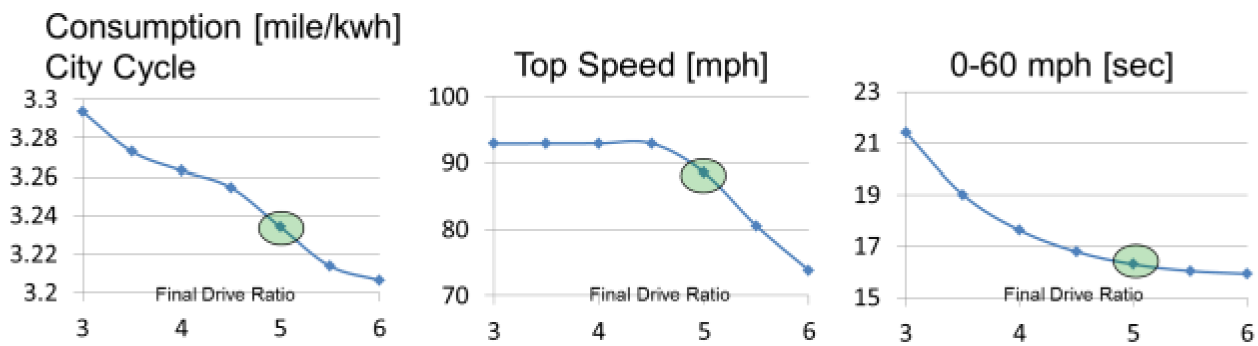


Figure 71. Energy consumption, top speed, and acceleration of baseline vehicle for final drive.

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 (90 kW), 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 72 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.

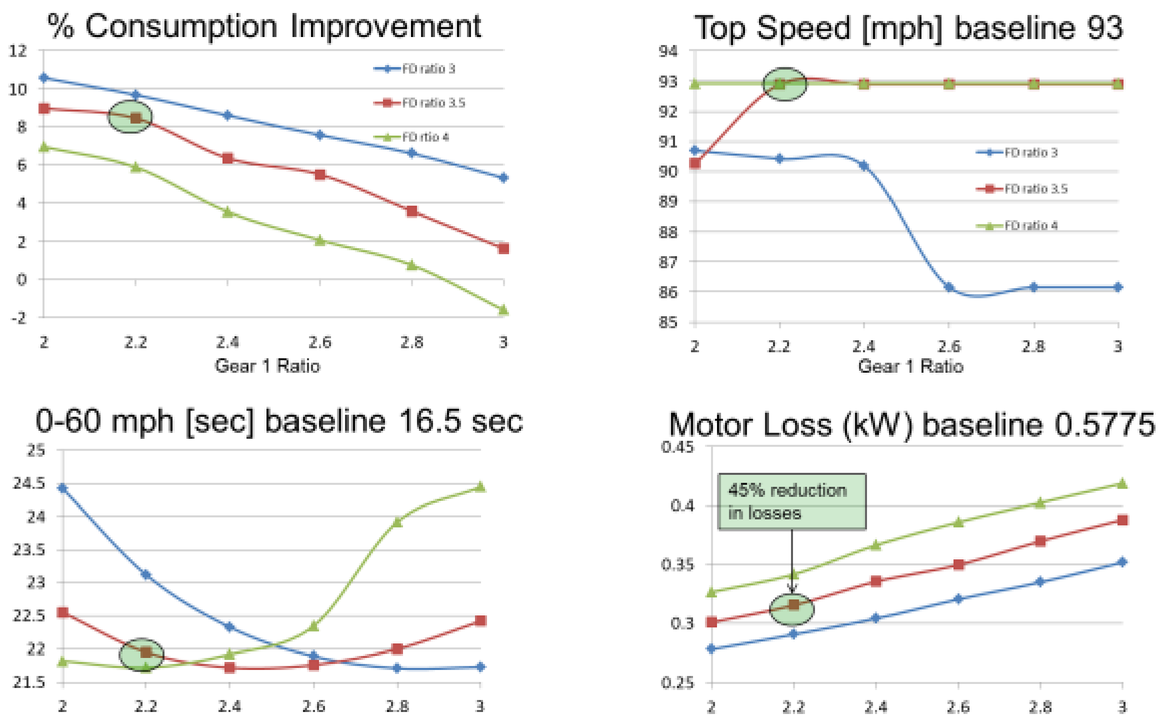
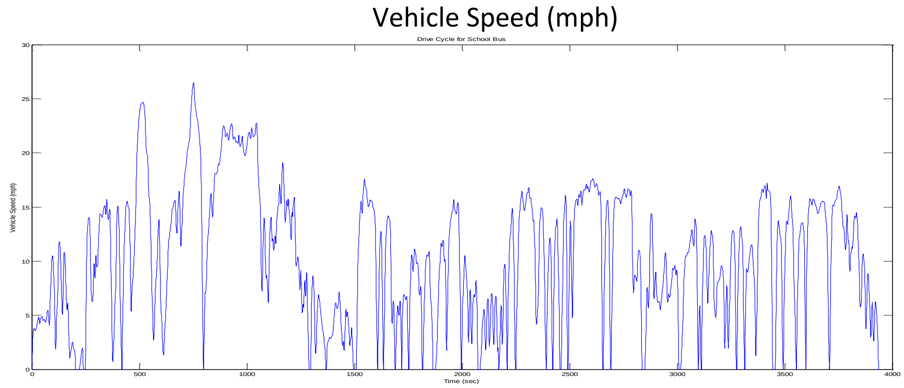


Figure 72. Energy savings for a city cycle, top speed, acceleration and motor loss with a two-speed transmission and a down-sized motor as function of first gear and final drive ratios.

Electric school bus application

Detailed modeling and simulations were reported on the Smith Electric Medium-duty Commercial Delivery Truck in Task 4. The modeling and simulation results on a medium-duty school bus with GVW of 16 ton (36000 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 73. 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.



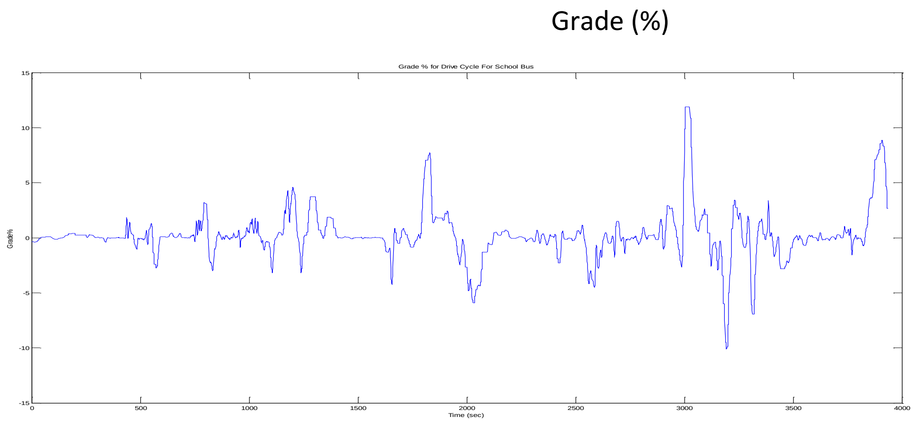


Figure 73. Standard drive cycle for school bus comparing energy efficiency and vehicle performance with and without multi-speed transmission.

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

Energy savings with three-speed transmission are shown in Figure 74 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 75.

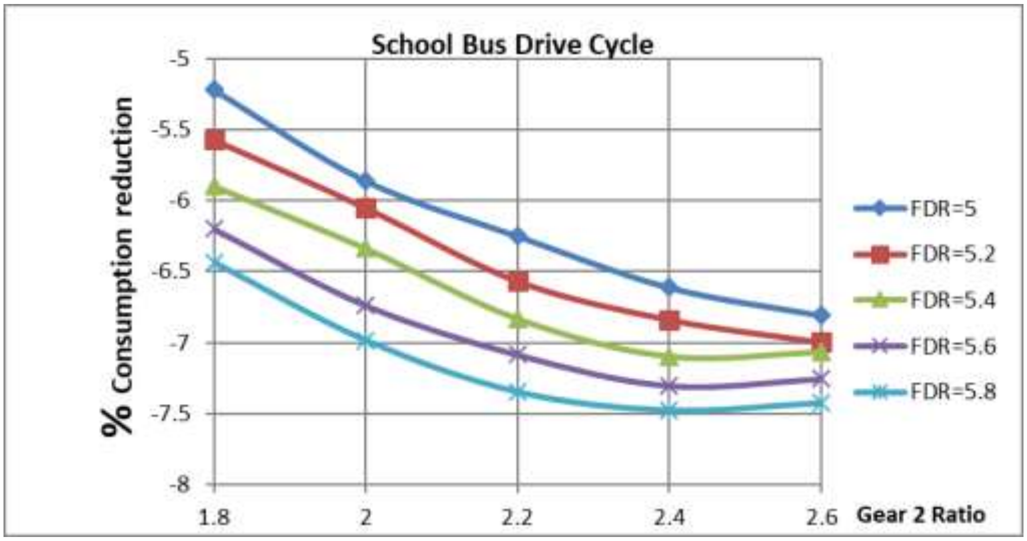


Figure 74. Energy savings with three-speed transmission and a downsized electric motor as compared to the direct drive baseline electric school bus application.

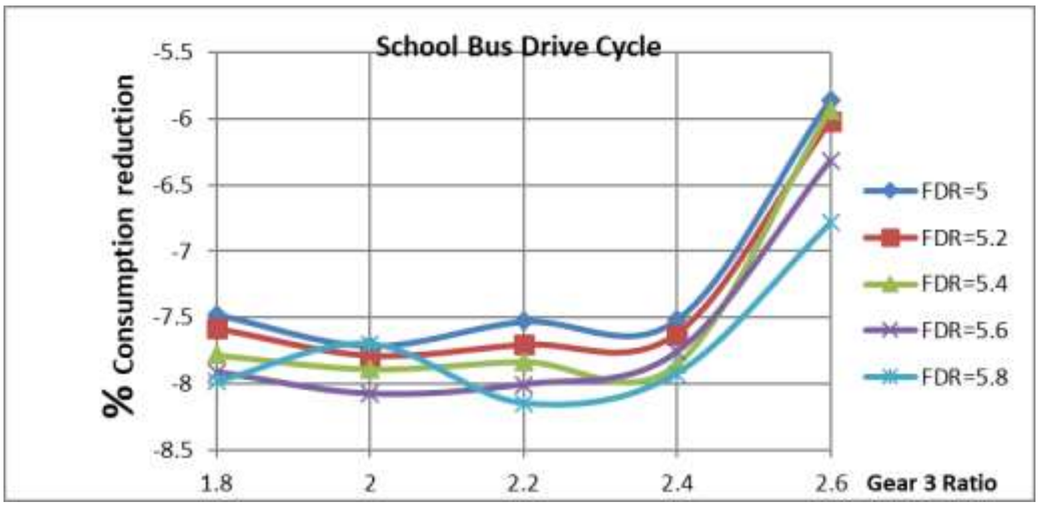


Figure 75. Energy savings with four-speed transmission and a downsized electric motor as compared to the direct drive baseline electric school bus application.

Table 34 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 34. 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 34, 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.

Subtask 6.3 – 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 35.

Table 35. 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

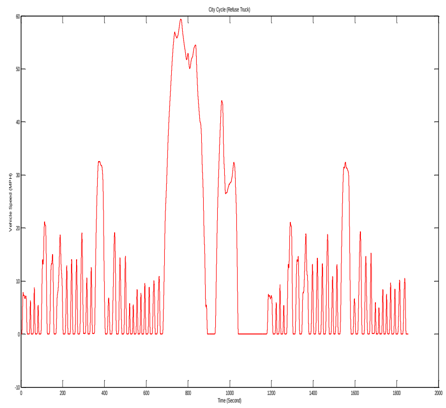
Table 36. Performance targets for refuse and drayage electric trucks.

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

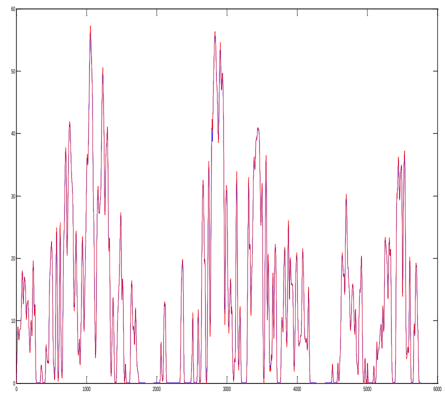
Drayage and refuse trucks are heavy-duty vehicles that require big electric motors with higher power/torque capacities than the electric bus applications. We used two electric motors operated in parallel for the direct drive (eDrive-EM 13531303002: 200/100 KW with maximum torque 2203 Nm at maximum speed 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 36. 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 76.



Refuse Drive Cycle



Drayage Drive Cycle

Figure 76. 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 miles per hour 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 77.

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 78.

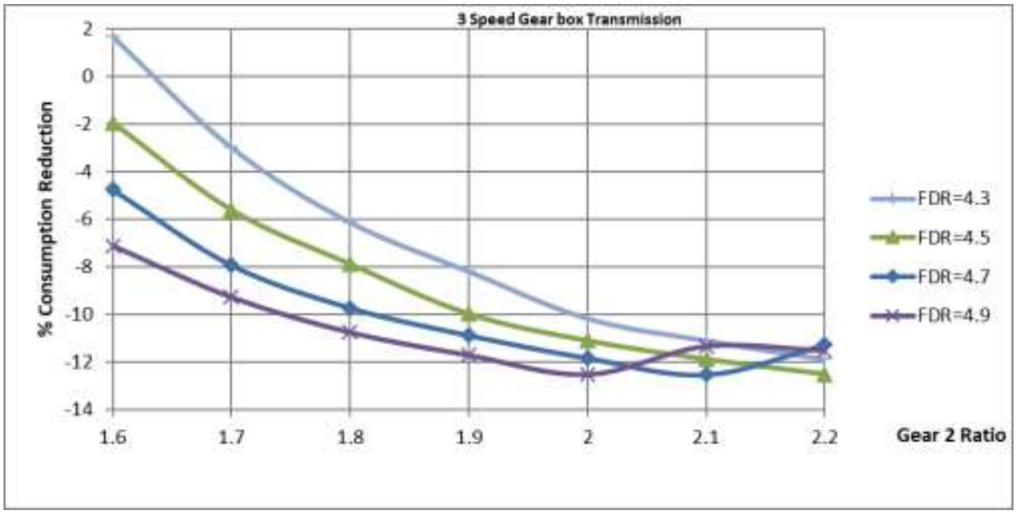


Figure 77. Energy savings with three-speed transmission and downsized electric motors as compared to the direct drive baseline electric refuse truck application.

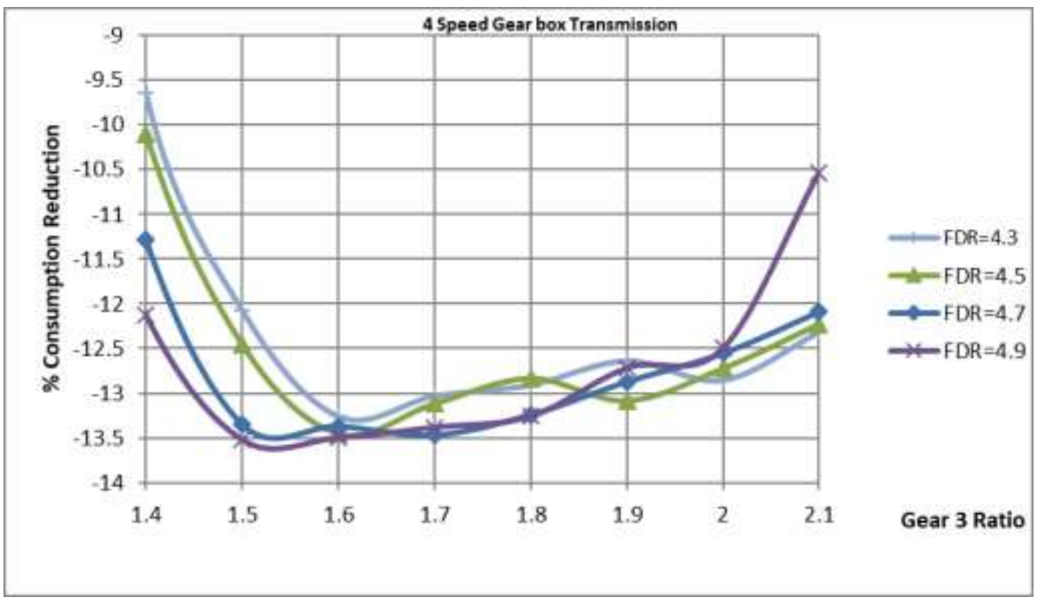


Figure 78. Energy savings with four-speed transmission and downsized electric motors compared to the direct-drive baseline electric refuse truck application.

Table 37 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 37. 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 37. 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 38 and compared to the direct-drive baseline electric refuse truck.

Table 38. 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 sec and met the acceleration requirements as listed in Table 38. 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 79.

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 80. Table 39 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 39. 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 Error! Not a valid bookmark self-reference.. 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 40 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 listed in Error! Not a valid bookmark self-reference.. 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 40. 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

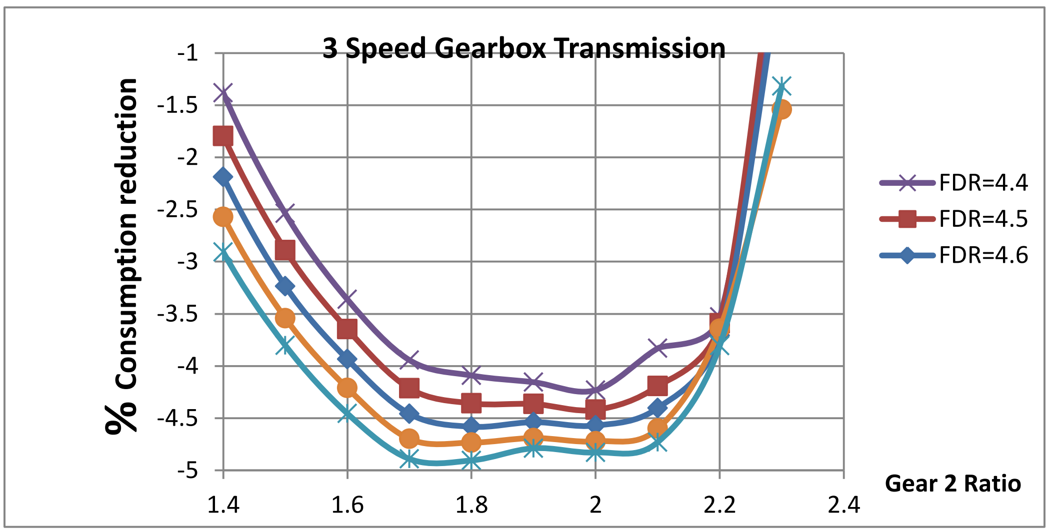


Figure 79. Energy savings for three-speed transmission and downsized electric motors compared to the direct-drive baseline electric drayage truck with large motor.

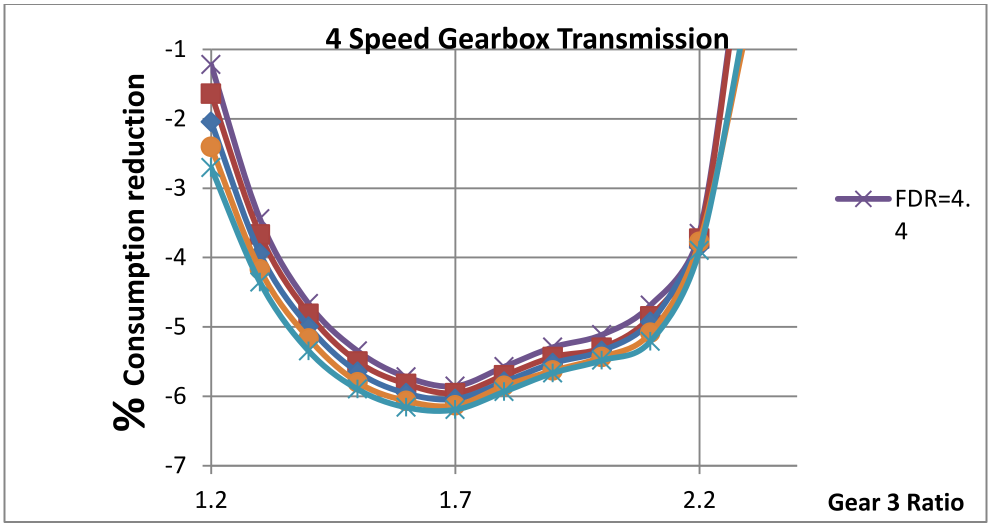


Figure 80. Energy savings for four-speed transmission and downsized electric motors compared to the direct-drive baseline electric drayage truck with large motor.

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 41. 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 medium-duty 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 41. 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%

Subtask 6.4 – 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 Budget Period 1 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 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 81**,** 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/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 42. The range/efficiency, gradeability, top speed, and reliability were identified as the top four performance criteria for the BE35 Electric Transit Bus application.

Table 42. 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

Subtask 6.5 – 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 150 lb per person, this load results in a measured gross vehicle weight of 37,530 lb. 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 81. 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 43.

Table 43. 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		
0 to 30 mph	15.5 s	< 13 s
30 to 50 mph	27.5 s	< 19 s
Gradeability @ GVW		
10mph	15%	>20%
20 mph	7%	> 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 44.

Table 44. 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)	
Gross Curb Weight (GCW)	28200
Seated Load Weight (SLW)	33500
Gross Vehicle Weight (GVW)	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 [16]. 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 45.

Table 45. 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 82.

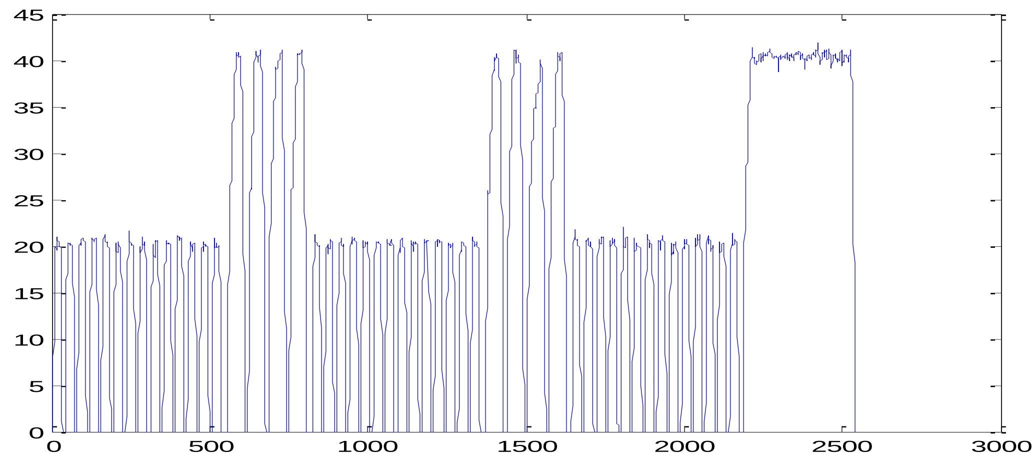


Figure 82. The vehicle speed (mph) for the Altoona ADB drive cycle used to validate the model of baseline EV with two-speed transmission compared to the field tests.

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 46.

Table 46. 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 47 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 47. 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 47. 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 47. 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 48. 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 48 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 48. 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 48 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 83 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).

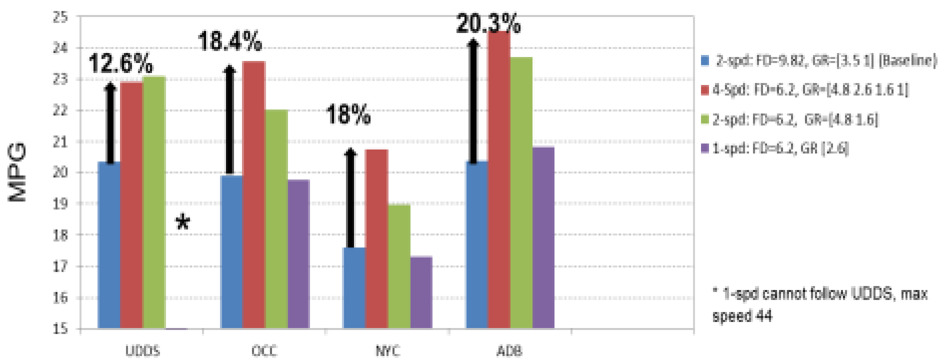


Figure 83. Energy efficiency (mpgde) comparison of the electric bus with different powertrain configurations for various drive cycles.
Four-speed transmission improves the efficiency by 20.3% as compared to the two-speed baseline.

Subtask 6.6 – 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 84 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.

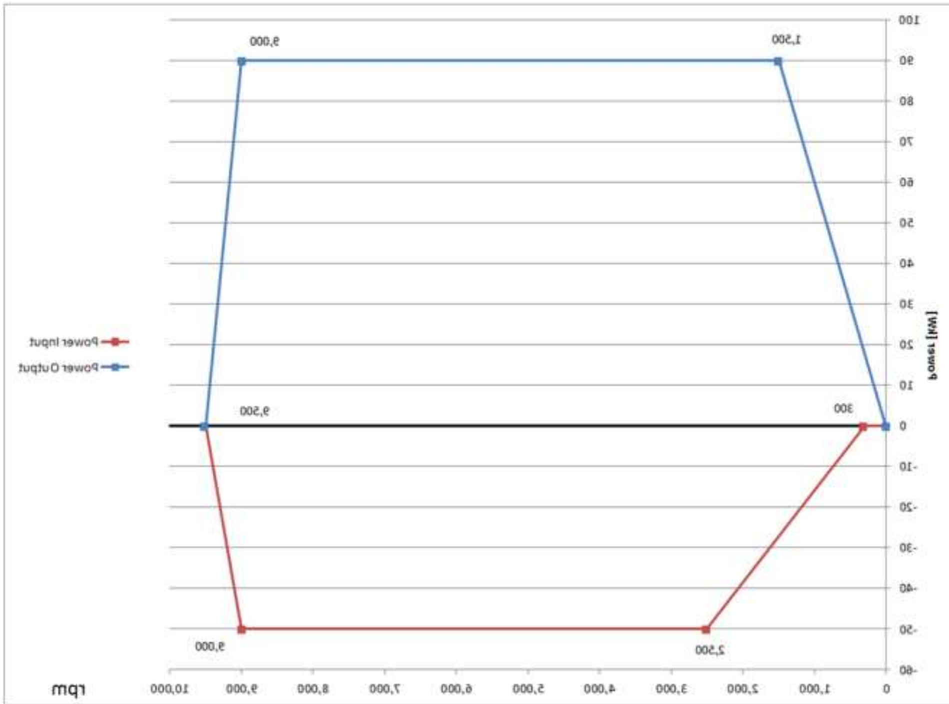


Figure 84. Electric machine power curve based on speed

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 85, 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.

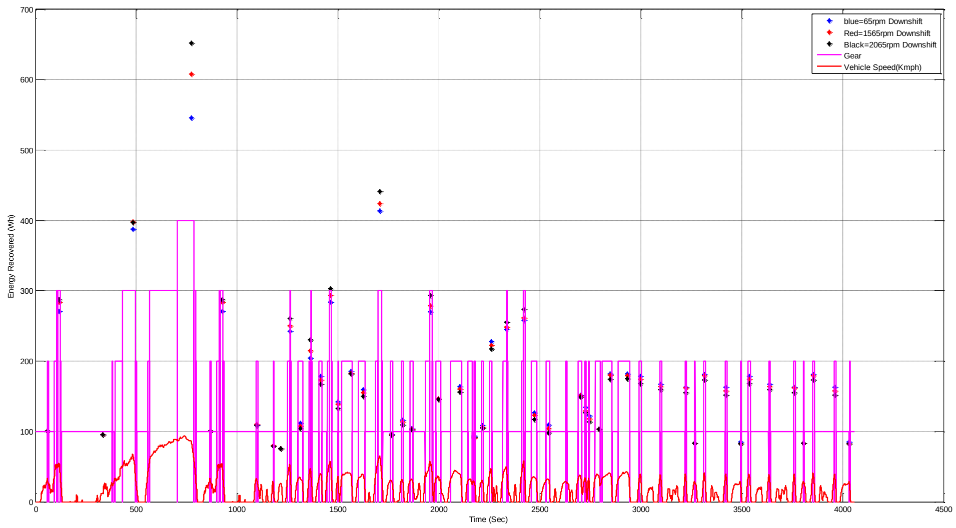


Figure 85. Captured brake energy for combined drive cycle for 3 different downshift strategies.

The goal of 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 85.

Table 49 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 prevents or delays 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 49. 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

TASK 7. TRANSMISSION DESIGN

The transmission design progressed in two stages as shown in Table 50. First, the transmission assembly was conceived and laid out in a model — with the goal of leveraging current product design while preserving an optimum EV transmission platform. The transmission layout design went through multiple reviews and modifications by cross functional teams. In the second stage, the components of transmission system were individually designed, detailed, and be readied for prototyping.

Table 50. Task 7 Activities

Task Name	Start	Finish
Task 7 – Transmission design	1/1/16	6/30/16
7.1 – Transmission layout design	1/1/16	3/31/16
Dry sump versus wet sump churning loss study	1/1/16	3/31/16
Initial additive manufacturing content search	1/1/16	3/31/16
7.2 – Transmission systems design	1/1/16	6/30/16
Four-speed gearset ratio design	1/1/16	6/30/16
Housings design and analysis	1/1/16	6/30/16
Rotating components design and analysis	1/1/16	6/30/16
Shift actuation components design	1/1/16	6/30/16
Lubrication subsystem analyses and design	1/1/16	6/30/16
Stack-up tolerance analysis	1/1/16	6/30/16

Deliverables: Transmission Layout Design Report

Subtask 7.1 Transmission Layout Design

The design team completed the initial 3D design layout of the transmission, maximizing reuse of current production components for speed-to-market and minimizing risk to reliability and durability. Figure 86 shows two different views of the result. A cross-functional Eaton team reviewed the design layout, including several Eaton colleagues who are very experienced with the transmission platform of which components, sub-assemblies, and general design strategy is being leveraged from for this product. Several very good suggestions and observations were made and implemented.

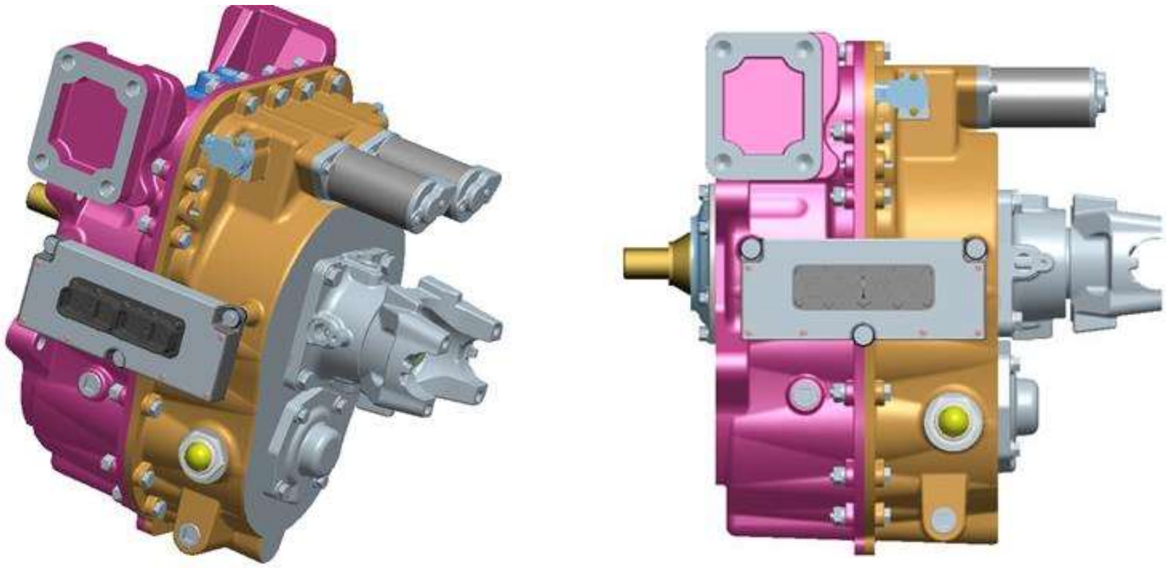


Figure 86. External views of 3- and 4- speed automated mechanical transmission layout.

Dry sump versus wet sump churning loss study

A study was conducted to evaluate the cost benefit of designing the proposed transmission with a dry sump versus a wet sump lubrication system. A dry sump system utilizes an internal pump and piping system in which oil is distributed throughout the transmission by means of oil pressurization. This method is advantageous as it reduces the oil level thereby decreasing oil churning (drag losses) and weight (oil volume). However, the internal pump and piping system add anywhere between $200 to $400 in supplementary material and design cost. A wet sump design simply fills the transmission to a predetermined level and allocates the oil via gear churning; as the gears spin through the oil they create an oil mist which spreads throughout the entire transmission. A wet sump is typically associated with higher parasitic losses thus producing an overall lower efficiency. A simulation was conducted using an internal Eaton program to evaluate the benefits of dry vs wet sump for this design. The churning losses are based on the following parameters:

- Transmission input speed: 1000-5000 RPM
- Oil type: Eaton PS-386 Synthetic Transmission fluid
- Oil characteristics based on operating temperature (Density, viscosity)
- Oil volume: 3.5-6 L
- Oil level: 217.5-0 mm from the centerline of the main shaft

The oil characteristics were evaluated at 100˚C ($\rho = 812\ \frac{kg}{m^3}\ , \nu = 14.8\ cSt$). The churning losses were studied at speeds ranging between 1000-5000 RPM and at an oil level 119 mm from the main shaft centerline (wet sump) and 187.5 mm from centerline (dry sump). A nominal operation point of the electric motor was used to compare the churning power losses:

- 3500 RPM
- Electric Motor Power – 100 kW

Results indicate that at nominal operation only a 0.55% efficiency savings is gained by using a dry sump over a wet sump design. The absolute efficiency losses due to churning were observed to be 0.34% (0.3 kW) and 0.99% (~1 kW) for dry and wet sumps, respectively. Since only half a kilowatt of power is gained by going with a dry sump design at nominal operation it was determined that the added cost of the pump and complexity of the design is not justified for this application. Please note that any speeds lower than 3500 RPM produce even smaller efficiency gains, also since the motor will likely be operating at more than 100 kW the 0.3-1 kW of power loss due to churning will be an even smaller percentage of the total power.

Initial additive manufacturing content search

The front and rear transmission housings, adapter plates and the shift yokes were identified as potential candidates for additive manufacturing. The prime manufacturing methods for the housings and the shift yokes are aluminum casting and ductile iron casting respectively. In parallel with the prime manufacturing paths the design of these parts were reevaluated from the additive manufacturability perspective. By having a prime path and a parallel path we avoided placing an unproven technology on the critical path of the project and risking the deliverables. Ultimately one or two of the prototype transmissions can be built with the additively manufactured parts while several other prototypes can be built with the traditional manufacturing processes.

The front and rear housing were identified as candidates for prototyping through additive, but the size of these components was a big challenge. Current popular additive manufacturing machines did not have the size capability to print them. Therefore, housings and adapter plates were ruled out as candidates for the AM technology implementation due to the size and the cost of these components. On the other hand, the shift-bar assembly presented itself as a good opportunity for the implementation of AM technology. The shift-bar assembly that was originally seven components could be manufactured as one component using the AM technology. Although this activity was not on the critical path it taught important learnings on the feasibility, cost, timing and performance (durability, weight etc.) of the AM made components in series production scenarios.

Subtask 7.2 - Transmission Systems Design

At this point, the transmission system design was nearly complete. The input shaft and the motor adapter had to wait for the operating agreements with the new EV-OEM partner to be completed so the Proterra could supply the motor design information to complete these two components.

Four-speed gearset ratio design

For the initial prototypes, a set of ratios for a four-speed transmission were selected, designed, and detailed, and readied for fabrication. These ratios were chosen after much analysis and simulation of optimum ratios for the weight class of proposed MD vehicles. The gear ratios were finalized after Proterra was selected as EV-OEM partner. To ensure durability and efficiency, and minimize risk and development cost and time, proven gear geometry was taken from production Eaton gears. These four gear ratios vary from 4.82:1 to 1.00:1 – offering ample launch ability on the steepest expected grades while still employing cruising ratios that keep the motor in its highest efficiency ranges as much as feasible.

The flexible and modular design concept of the Eaton four-speed automated mechanical transmission enabled the project team to select six potential input gearset ratios with the headset ratio choices of 1.55 or 1.88 as listed in Table 51. Availability of three choices of final drive ratio 6.2, 7.4, and 9.8 increased the number of potential solutions to 36. However, the final drive ratios of 7.4 and 9.8 were eliminated quickly due to the low drive axle efficiency at these ratios. Furthermore, the headset ratio of 1.55 was selected to get a torque rating that is close to the UQM maximum motor torque and achieve our target for the gradeability requirements. Hence, six viable input gearset ratios (Options 1 to 6) with the final drive ratio of 6.2 were evaluated further for the Proterra EV Model BE35 bus application to be equipped with the new four-speed transmission.

Table 51. Gearset options with the headset ratios of 1.55 and 1.88 for the four-speed transmission

Headset / Gear Ratio / Existing Gear / New Gear / Direct Drive

Option	1.55	Gear Ratio	Step	Option	1.88	Gear Ratio	Step
Opt. 1	4.00	2.58		Option 7	4.00	2.13	
	2.50	1.61	1.60		2.50	1.33	1.60
	1.60	1.03	1.56		1.60	0.85	1.56
	1.00	0.65	1.60		1.00	0.53	1.60
Opt. 2	4.26	2.75		Option 8	5.17	2.75	
	2.76	1.78	1.54		3.16	1.68	1.64
	1.64	1.06	1.68		1.79	0.95	1.77
	1.00	0.65	1.64		1.00	0.53	1.79
Opt. 3	4.45	2.87		Option 9	5.40	2.87	
	2.76	1.78	1.61		3.16	1.68	1.71
	1.64	1.06	1.68		1.75	0.93	1.81
	1.00	0.65	1.64		1.00	0.53	1.75
Opt. 4	4.82	3.11		Option 10	5.85	3.11	
	2.76	1.78	1.75		3.16	1.68	1.85
	1.64	1.06	1.68		1.79	0.95	1.77
	1.00	0.65	1.64		1.00	0.53	1.79
Opt. 5	6.76	4.36		Option 11	8.20	4.36	
	3.57	2.30	1.90		4.08	2.17	2.01
	1.94	1.25	1.84		1.99	1.06	2.05
	1.00	0.65	1.94		1.00	0.53	1.99
Opt. 6	7.29	4.70		Option 12	8.84	4.70	
	3.80	2.45	1.92		4.32	2.30	2.04
	1.94	1.25	1.96		2.07	1.10	2.09
	1.00	0.65	1.94		1.00	0.53	2.07

We found that the gearset options that provide the best vehicle acceleration time and energy efficiency are Options 4 and 5 with FDR 6.2 based on the simulations. Gearset Options 4 and 5 were further evaluated for three drive cycles, shown in Figure 87, for the final decision. Option 4 has smaller ratio steps (avg. 1.7) and smaller gear ratios (4.8, 2.8, 1.6, 1) than Option 5 (avg. 1.9 ratio step and 6.8, 3.6, 1.9, 1 gear ratios). The three drive cycles, UDDS, OCC, and NYC cover a wide range of city bus driving with the top speed exceeding 60 mph. The four-speed transmission can be launched in first or second gear. In the following paragraphs the three-gear setup means that the vehicle is normally launched in second gear at grades lower than 20% and the first gear is engaged only at grades higher than 20% due to the concern of shift comfort. In the four-gear setup the vehicle is always launched in first gear regardless of the grade.

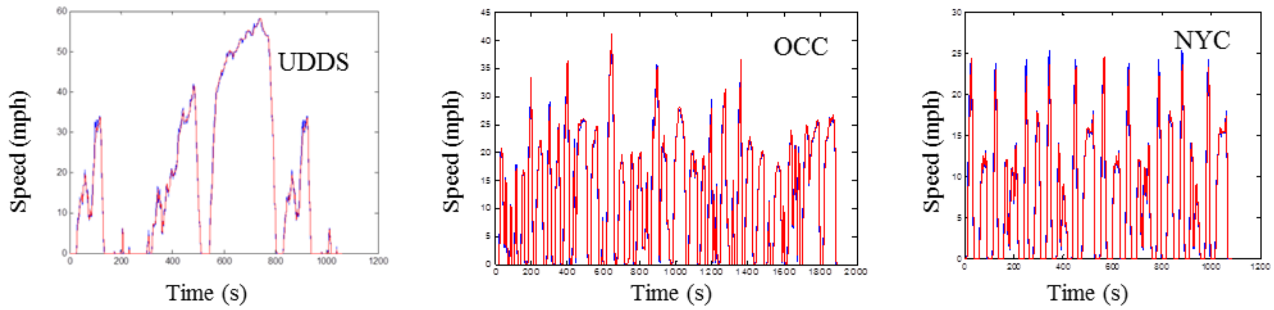


Figure 87. Drive cycles with vehicle speed versus time comparing energy efficiency and vehicle performance with various gearset options.
Left UDDS, middle OCC, right NYC.

The simulation results for energy consumption improvement with a four-speed transmission and FDR 6.2 over the baseline two-speed transmission (3.5, 1) and FDR 9.8 are presented in Figure 88 for three drive cycles. The four-speed transmission is evaluated further for the gearset Options 4 and 5 using either three-gear setup, where the four-speed transmission is launched in second gear, or four-gear setup, where the four-speed transmission is launched in first gear (Figure 89**)**. The launch in first gear provides 2 to 14% better efficiency than the launch in second gear, depending on the drive cycle. NYC seizes the biggest benefit of launching in first gear due to the frequent stop and go operation of the bus in this cycle. Option-5 with a four-gear setup shows a little better improvement over Option-4 with four-gear setup, except for the NYC drive cycle.

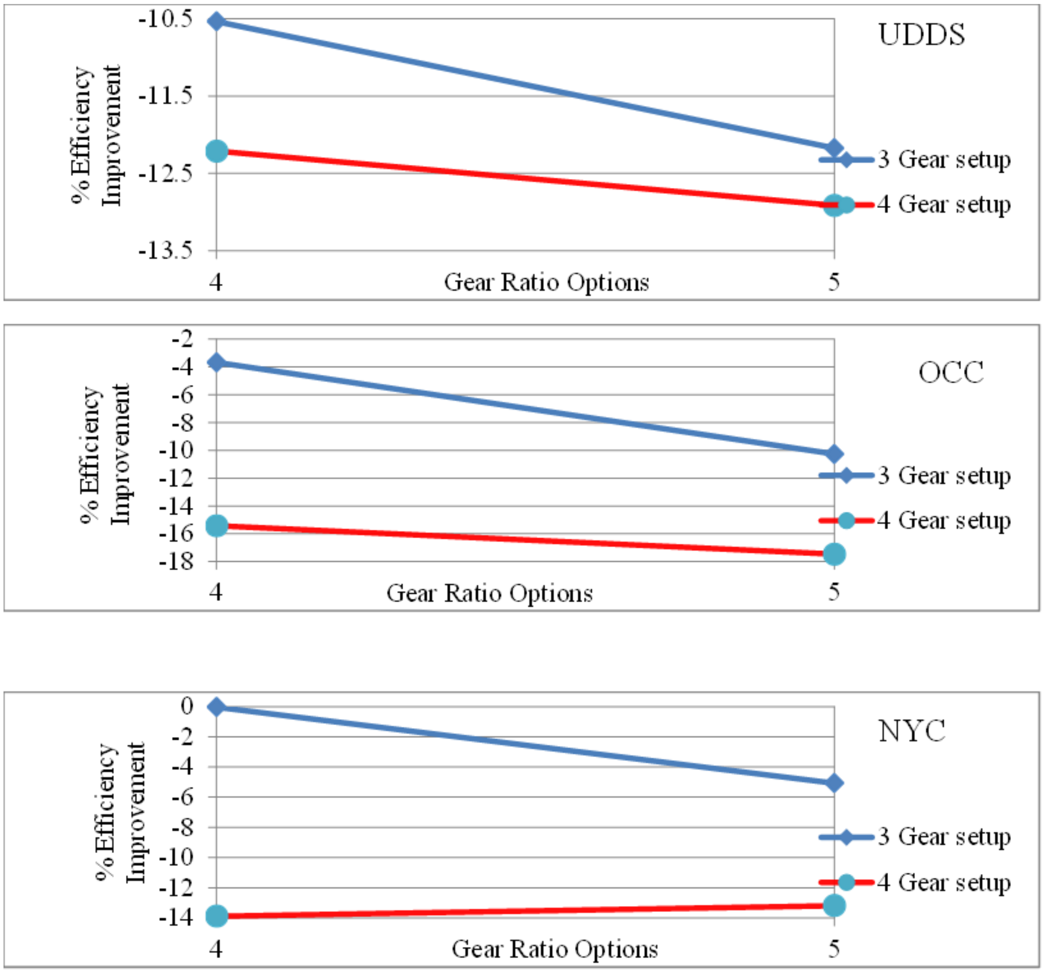


Figure 88. Simulation results for Proterra EV Model BE35 bus for three drive cycles. The graphs show energy efficiency improvement (% of kWh/mile) with four-speed over the baseline two-speed transmission. The four-speed transmission is evaluated with 1st gear launch (four-gear setup) and second gear launch (three-gear setup) for gearset Options 4 and 5 listed in Table 51.

To finalize the gearset selection between the options 4 and 5 using the 4 Gear setup, the vehicle acceleration performances at full throttle were also compared. The total wheel torque versus the vehicle speed curves with gearset options 4 and 5 are shown in Figure 89 for full throttle accelerations. The torque step size between the first and the second gears is an important factor affecting the driver and passenger perception of shift quality. It is clear from Figure 89 that the torque step size during the shift from first to second gears is larger with Option 5 compared to Option 4. Hence, Option 4 provides a better shift comfort than Option 5 does.

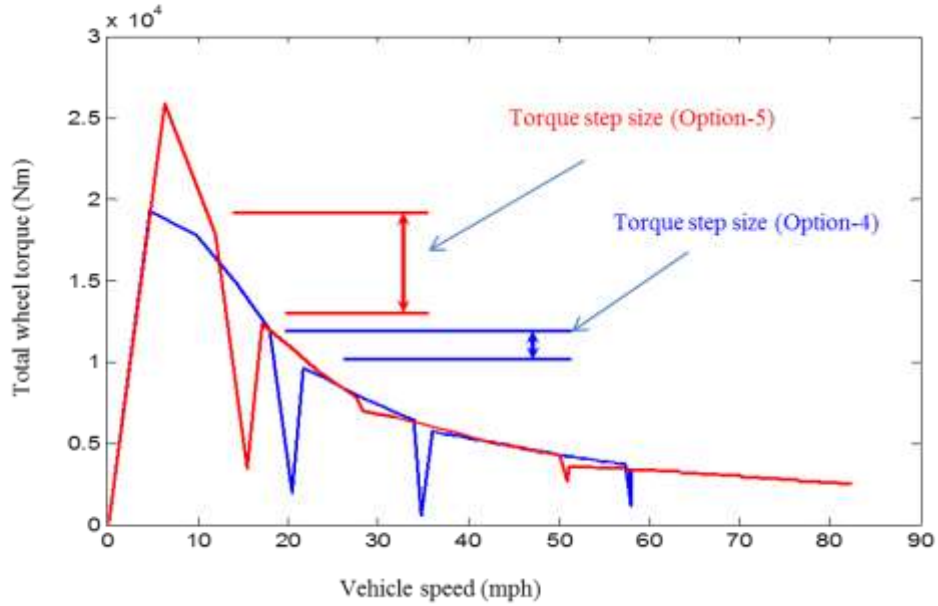


Figure 89. Total wheel torque versus vehicle speed curves of four-speed transmission with gearset options 4 and 5.
Simulation results are for 1st gear launch and full throttle acceleration.

The large starting torque with Option 5 is close to the wheel spin torque and would cause a rough driving feel. Therefore, the motor torque must be limited to improve the driver comfort for this case. However, limiting the motor torque may lower the re-generator power efficiency. Therefore, we selected gearset Option 4 for the new four-speed automated mechanical transmission to be integrated on Proterra BE35 electric bus for the vehicle validation tests. Finally, Table 52 lists all the performance metrics results obtained from simulations including the improvements with the new four-speed transmission with gearset Option 4 and first gear launch compared to the baseline two-speed transmission for Proterra BE35 electric bus application. The Altoona combined drive cycle was included in the simulation evaluations since this cycle was also used for the HIL tests of the actual powertrain hardware and the vehicle validation tests of the four-speed transmission by ORNL and NREL. In the simulations listed in Table 52**,** the baseline vehicle with two-speed transmission has limitations on the motor as induced by the vehicle OEM in contrast to Table 48 where the baseline simulations with two-speed transmission has no limitations. On the other hand, the vehicle with four-speed transmission is evaluated without limitations on the motor, since there will be either no need for the limitations or the limitations will be reduced significantly due to the four-speed transmission.

Table 52 indicates that four-speed transmission improves top speed, gradeability at 20 mph and acceleration at 30-50 mph by 53%, 43%, and 31% respectively. In addition to these significant performance gains, four-speed transmission enables 16 to 21% efficiency gains in four different drive cycles. These simulation results indicate that the four-speed transmission meets or exceeds the original performance targets set in the beginning of project, as listed in Table 1.

Table 52 also indicates that the difference between the vehicle efficiency gain and the motor/generator efficiency gains are due to enabling the change in FDR from 9.8 to 6.2 in the vehicle drive axle with four-speed transmission. Hence, about half of the efficiency gains can be attributable to the FDR change. Furthermore, the biggest advantage of four-speed transmission over the baseline two-speed comes from enabling greater energy regeneration during braking.

Table 52. Proterra BE35 electric bus performance simulation comparison between four-speed and the baseline two-speed transmissions.
The four-speed transmission has the gearset option-4, launched in 1st gear, and integrated with FDR 6.2. The baseline two-speed transmission has the gear ratios of 3.53 and 1, and integrated with FDR 9.8.

<table>
<tr><th colspan="2">Metric</th><th colspan="2">Two-speed</th><th colspan="2">Four-speed</th><th colspan="2">% Benefits four-speed vs two-speed</th></tr>
<tr><td>1</td><td><u>Top vehicle speed</u> @ GVW</td><td colspan="2">53 mph</td><td colspan="2">81 mph > 65 mph</td><td colspan="2">52.8%</td></tr>
<tr><td rowspan="5">2</td><td><u>Vehicle Efficiency @ SLW</u></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td></tr>
<tr><td>On -UDDS</td><td colspan="2">20.4 mpgde</td><td colspan="2">23.2 mpgde</td><td colspan="2">15.7%</td></tr>
<tr><td>On Orange County - OCC</td><td colspan="2">19.9 mpgde</td><td colspan="2">23.5 mpgde</td><td colspan="2">18.3%</td></tr>
<tr><td>On Manhattan - NYC</td><td colspan="2">17.7 mpgde</td><td colspan="2">20.5 mpgde</td><td colspan="2">16.1%</td></tr>
<tr><td>On Altoona-ADB</td><td colspan="2">20.2 mpgde</td><td colspan="2">24.4 mpgde</td><td colspan="2">20.8%</td></tr>
<tr><td rowspan="3">3</td><td><u>Acceleration time @ SLW</u></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td></tr>
<tr><td>0 to 30 mph</td><td colspan="2">15.5 s</td><td colspan="2">13 s</td><td colspan="2">12.9%</td></tr>
<tr><td>30 to 50 mph</td><td colspan="2">27.5 s</td><td colspan="2">19 s</td><td colspan="2">30.9%</td></tr>
<tr><td rowspan="5">4</td><td><u>Motor&Generator Efficiency</u></td><td>Motor Efficiency</td><td>Gen. Efficiency</td><td>Motor Efficiency</td><td>Gen. Efficiency</td><td>Motor Efficiency</td><td>Gen. Efficiency</td></tr>
<tr><td>On -UDDS</td><td>92.0%</td><td>87.6%</td><td>92.2%</td><td>89.6%</td><td>0.2%</td><td>2.2%</td></tr>
<tr><td>On Orange County-OCC</td><td>90.2%</td><td>83.9%</td><td>90.9%</td><td>89.0%</td><td>0.7%</td><td>6.0%</td></tr>
<tr><td>On Manhattan- NYC</td><td>89.4%</td><td>83.7%</td><td>90.2%</td><td>89.3%</td><td>0.9%</td><td>6.7%</td></tr>
<tr><td>On Altoona -ADB</td><td>91.1%</td><td>82.3%</td><td>91.1%</td><td>89.6 %</td><td>0.0%</td><td>11.6%</td></tr>
<tr><td rowspan="3">5</td><td><u>Gradeability @ GVW</u></td><td colspan="2"></td><td colspan="2"></td><td colspan="2"></td></tr>
<tr><td>- 10 MPH</td><td colspan="2">15 %</td><td colspan="2">20%</td><td colspan="2">33.3%</td></tr>
<tr><td>- 20 MPH</td><td colspan="2">7.0%</td><td colspan="2">10%</td><td colspan="2">42.9%</td></tr>
</table>

Housings design and analysis

The design of the front and rear main gearbox enclosures of the four-speed transmission were completed with the aid of finite element analysis (FEA). The aluminum cast housings underwent two full cycles of FEA. The design improvements on the front and rear housings were implemented to eliminate the deficiencies indicated by the FEA before the 3D CAD models and 2D details could be completed. Figure 90 shows the equivalent stresses under the maximum input torque (1300 Nm) conditions in the front (left) and rear (right) housings.

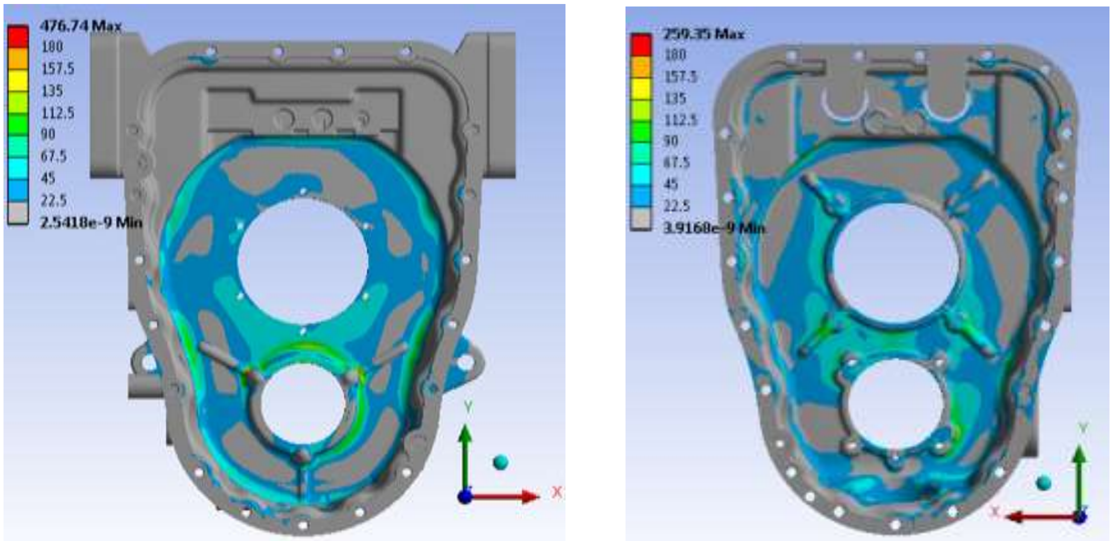


Figure 90. Equivalent stress contours in the front (left) and rear (right) housings under 1300 Nm input torque.

Since the housings are long-lead items, we placed an order for seven sets of housings with a fabrication vendor after getting several quotes to make the parts. However, the vendor was requested to only cast and machine one set of housings, and to hold back on the other six sets, so that the housings could be inspected and installed with the mating components to ensure the design was correct — these were very complicated components that mate with nearly all of the major bearings, covers, housings, shift bars, shift motors, sensors, and many other key components. Once the designs were checked and all fits verified, the other six sets of housings were released for fabrication.

Rotating components design and analysis

Rotating components, gears, shafts, bearings etc., were designed and analyzed for stress-strain, thermal and key life considerations. Rotating components analysis were conducted using the MASTA software based on the 500,000 miles required transmission life and checked against the requirements of the vehicle duty cycles based on the all combined cycles of UDDS, OCC, and NYC as described in Task 6. The rotating components analyses indicated that gear bending and contact stresses were within acceptable limits and the gear bending reliability and the contact reliability exceeded 99% for the combined duty cycle. Furthermore, all bearings met the target life with a reliability of 83% that exceeded the reliability target of 80% as shown in Figure 91. Support bearing contact stresses were well below allowable material limit. A couple of needle bearings had higher contact stresses mainly due to the braking/regenerating conditions at maximum motor torque ratings. The torque ratings during the braking/regenerating conditions are limited by the transmission controls in order not to exceed the design limitations of the needle bearings.

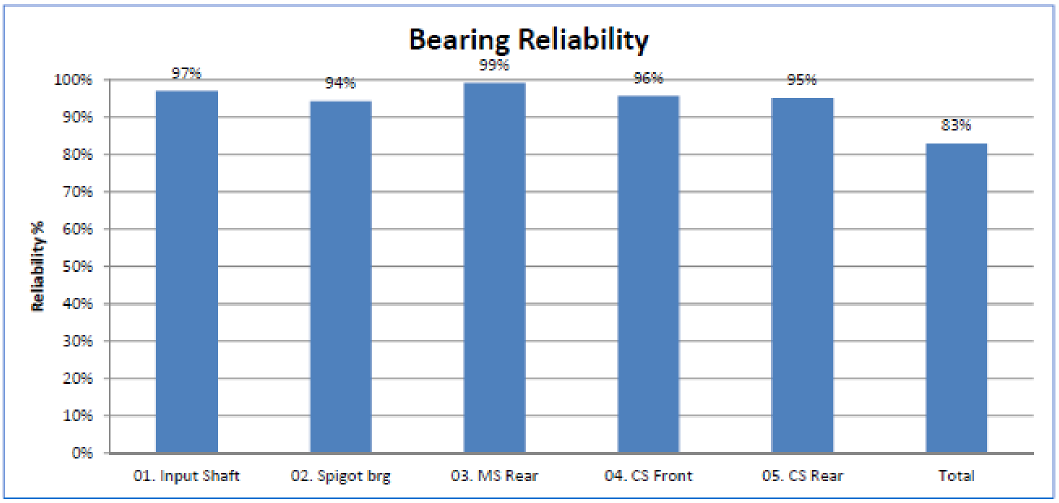


Figure 91. Four-speed transmission bearing reliability analysis results

We also evaluated gear whine noise performance of four-speed transmission. The simulation was based on the latest transmission design. The CAD models of the housing and internal gear sets were imported into rotation system dynamic models to accurately predict the gear whine noise due to transmission error (TE). The TE analysis and gear tooth micro-geometry optimization was performed with LDP program. The bearing stiffness analysis was performed with MASTA software. The system dynamic analysis was performed with ANSYS software. The vibro-acoustics of the transmission was performed with LMS software. The typical operating conditions for gear noise analysis are listed in Table 53.

Motor peak power of 220 kW, motor continuous power of 135 kW and the final drive ratio of 6.2 were used in the analysis. A typical power flow schematic is shown in Figure 92 for the vehicle at 50 mph in third gear. The gear tooth profile optimization results are summarized in Table 54.

Table 53. The working conditions for gear noise analysis

Items	2nd Gear	3rd Gear
Vehicle speed (mile per hour)	42	50
Vehicle speed (m/s)	19	2
Wheel Radius (m)	0.49	0.49
Wheel RPM	368	438
Axle Ratio	6.2	6.2
Output Gear RPM	2284	3718
Number of Teeth Headset MS	29	29
Number of Teeth Headset CS	45	45
Number of Teeth Speedset Gear MS	47	35
Number of Teeth Speedset gear CS	28	33
Output Speedset Gear RPM	2284	2718
Counter shaft Speedset Gear RPM	3833	2883
Counter shaft Headset RPM	3833	2883
Input Gear headset RPM	5948	4474
Motor Output Power (kW) - Peak	220	220
Motor Output Torque (Nm)	353	470
Motor Output Power (kW) - Continuous	135	135
Motor Output Torque (Nm)	217	288

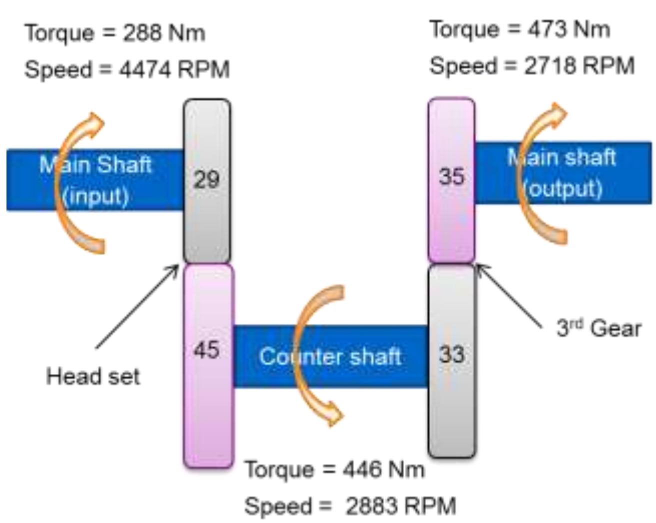


Figure 92. Typical power flow schematic.

Table 54. The gear tooth profile optimization results.

Vehicle Speed (mph)	Gear Pair	Motor Output RPM	Motor Power Type	Motor Output Torque (Nm)	Input Torque to Gear pair (Nm)	Average Gear Mesh Stiffness (N/m)	TE 1st harmonic (micron)	TE 2nd harmonic (micron)
42	Headset	5947	Continuous	217	217	328*10^6	0.95	0.23
			Peak	353	353	382*10^6	0.87	0.06
	2nd	5947	Continuous	217	336	409*10^6	1.79	0.39
			Peak	353	547	479*10^6	1.75	0.11
50	Headset	4474	Continuous	288	288	358*10^6	0.91	0.12
			Peak	470	470	420*10^6	0.66	0.18
	3rd	4474	Continuous	288	447	318*10^6	1.23	0.22
			Peak	470	730	374*10^6	2	0.55

The gear tooth profile crown and lead crown have significant impact on the TE values. An optimized gearset can reduce the TE values by 25% to 50%, which helps to reduce noise generation by 3 to 6 dB. For the example, a lead crown of 10 microns and tip relief of 18 microns were proposed for the headset gear. The TE values are reduced by half compared to baseline. The proposed micro-geometry modifications did not impact the contact and bending stress during the optimization and have the least impact on the manufacturing process. Further investigations can be carried with more emphasis on the practical design and manufacturing considerations.

The dynamic models of the gear, shaft, bearing, and housing were used to identify major vibration modes of the system and potential design changes. Major structure vibration modes and natural frequencies were obtained from the simulations. The overall noise radiation of the transmission is shown in Figure 93.

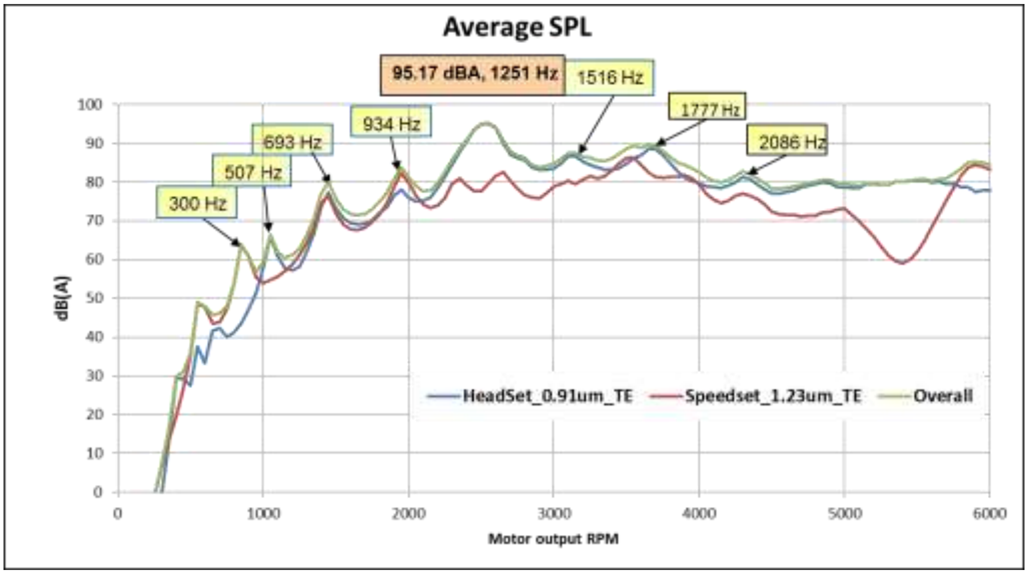
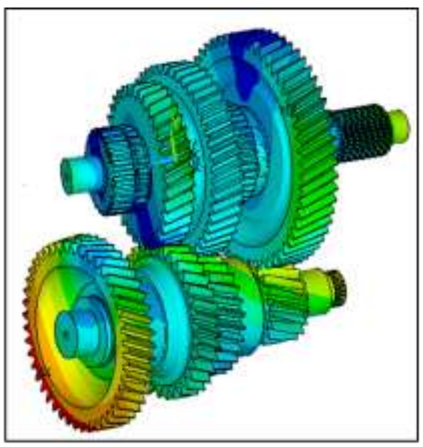


Figure 93. Average sound pressure level (SPL) as function of motor output speed.

As an example, the noise peak at the 1251 Hz is associated with the gear and housing vibration modes as shown in Figure 94.



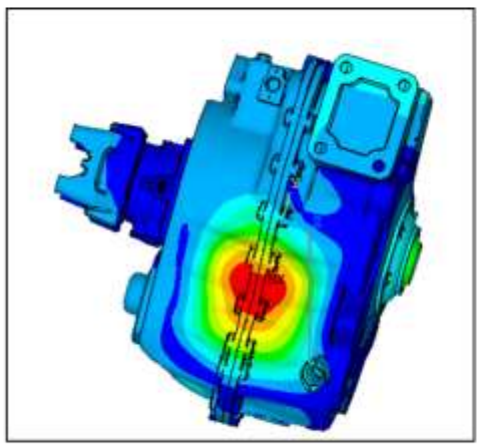


Figure 94. 1253 Hz – Countershaft bending and housing side panel breathing mode

The NVH modeling and simulation help engineering team to understand potential noise and vibration issues and their root causes. These simulations also help to evaluate the effectiveness of various design and manufacturing changes without prototyping, thus significantly reduce development time and cost.

Shift actuation components design

We completed shift bar, shift fork, shift actuation block, and related hardware designs and readied them for fabrication. The FEA analysis on the shift forks was also completed – showing acceptable deflection and stress characteristics. Several actuation components and sub-assemblies were taken directly from Eaton's production X-Y shifter assembly and were designed into the new transmission. These production components included electric motors, ball-screw assemblies, position sensors and related linkage, bearings, and bushings. The new transmission employed these production components in a manner very similar to the Eaton's production transmissions—minimizing risk and development time.

Lubrication subsystem analyses and design

As mentioned above, we selected splash lubrication as opposed to dry sump lubrication. Splash lubrication minimized development time and risk and still provided good efficiency —due to the low-speed, low-churn countershaft design, as well as the newly-released Eaton PS-386 synthetic transmission lubricant that has low viscosity at operating temperatures and provides optimum cooling and lubrication for the life of the gearbox – eliminating the need for oil changes and used oil disposal. Using proven lube level determination methods, we chose a lubricant level and volume that ensures adequate lubrication while minimizing oil usage and churning loss.

Stack-up tolerance analysis

Stack-up tolerance analysis of final design assembly was conducted and finalized before the critical internal components were ordered for fabrication. However, since much of the design philosophy was adopted from Eaton's current six-speed transmission product family, the need for extensive stackup analysis was minimal, as the transmission architecture was set up to be very tolerant of part dimension variations.

TASK 8. PROTOTYPE FABRICATION

After completing transmission systems design, prototyping activities were performed as outlined in Table 55. One set of prototype front and rear housings were made using rapid-prototyped castings. These housings were made to check for fit of existing production shafts and bearings as well as to have a mockup housing to check overall size and fit. After the dimensional checks and fits additional housings were cast from these castings.

Table 55. Task 8 Activities

Task Name	Start	Finish
Task 8 Prototype fabrication	7/1/16	9/30/16
8.1 – Prototype build, inspections, dimensional checks and fits	7/1/16	9/30/16
8.2 – Shift-bar assembly for additive manufacturing	10/1/16	12/31/16

Deliverables: Transmission Systems Design, first Stage of Transmission Prototyping Report, Transmission Prototyping Report

Subtask 8.1 – Prototype Build, Inspections, Dimensional Checks and Fits

One display unit was prepared as shown in Figure 95. The display unit has all the exterior components but not all the interior components. The unit was displayed at the IAA Commercial Vehicles Trade Fair in Hannover on September 22-29, 2016. The display unit was also used to assemble, inspect, and check for the electrical harnesses of the control box of the new transmission.

Figure 95. The display unit of the new four-speed EV Transmission.

Purchase orders for the prototype transmission components were issued and all parts were received. Figure 96 shows the transmission parts as received. Dimensional checks were completed. Several parts needed rework and taken care of as in any new product development. Nothing major affected or delayed the project timeline. Six fully functional prototypes were completed with all internal and external components, electronics, wiring harnesses, and controllers. Two prototypes were given to Proterra for vehicle integration. Another unit was sent to ORNL for integration and testing in the PIL test laboratory. The rest of prototypes were used to generate more test data and to raise the commercial awareness on the new four-speed EV-AMT. Pictures of the fully functional prototype units are shown in Figure 97.

Figure 96. Transmission components as laid out at the assembly workshop.

Figure 97. Three fully functional prototypes of the new four-speed EV transmission.

Subtask 8.2 – Shift-bar Assembly for Additive Manufacturing

The feasibility of additive manufacturing of various parts and sub-assemblies was discussed in Subtask 7.1 where shift bar assembly was chosen for the implementation. The shift bar assembly shown in Figure 98 was zoned as an apt choice to demonstrate part integration through additive manufacturing.

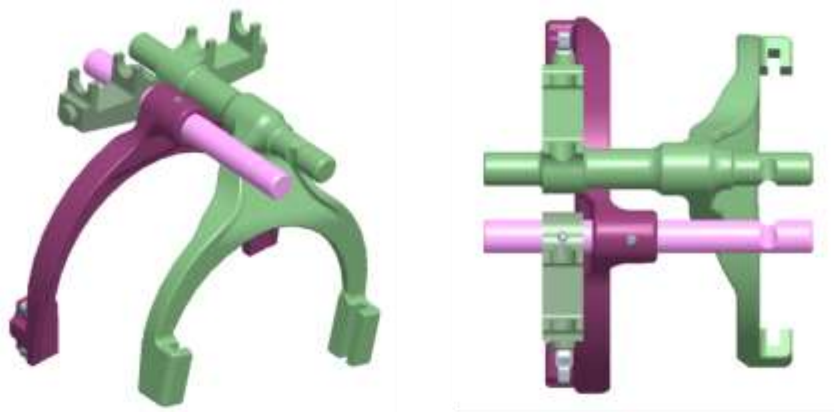


Figure 98. Two shift bar assemblies.
The six-component shift-bar assembly (the colored parts) can be reduced to a single component by AM (the light green one).

The conventional shift bar mechanism is an assembly of multiple parts requiring fastening joints. A methodology was established to analyze the shift bar mechanism as a single integrated product and a baseline assembly was drafted from the existing group of parts that make up the shift bar.

Once the parts integration was complete, we performed a durability simulation to ensure that the integrated part will satisfy all the stress and deformation requirements. The workflow of simulation methodology is shown in Figure 99.

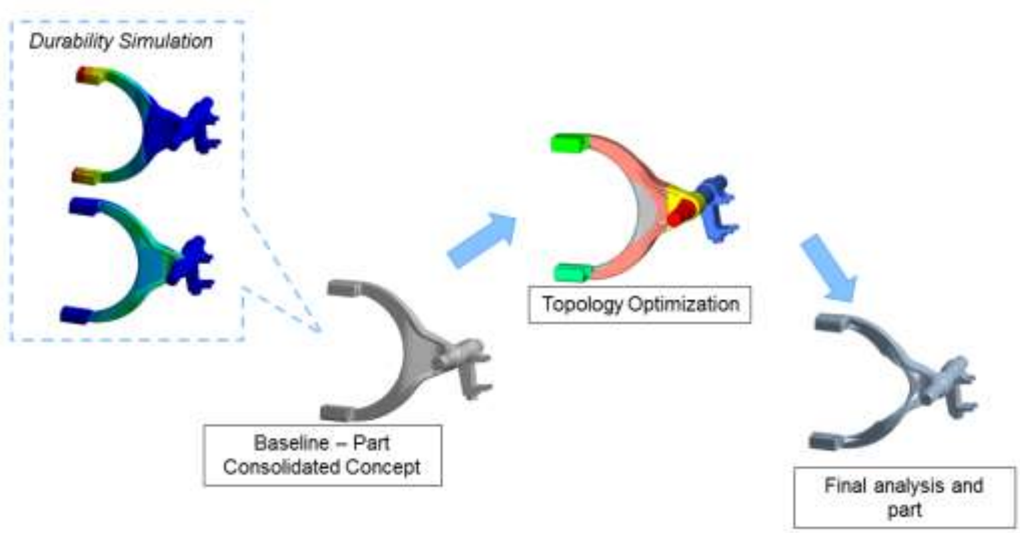


Figure 99. Workflow of simulation methodology to additively design the shift bar mechanism.

On successful stress validation, the part was taken to topology optimization for further mass reduction (and material required for 3D printing reduced) was performed. In this part, 8% reduction in mass was achieved by topology optimization. The optimized part was then analyzed again for durability (stress and fatigue) to ensure that the part will perform satisfactorily under the given loading and boundary conditions and all the stress and deformation requirements as shown in Figure 100. Furthermore, process simulation was performed on the part to ensure that it can be manufactured using DMLS technology without any manufacturability/quality issues. Key highlights of the workflow include:

- Successful part integration for the shift bar assembly
- Reduction in weight of the part by 10%
- Reduction in failure modes in the part due to elimination of multiple joining mechanisms
- Demonstration of concept to final part through detailed analytical models

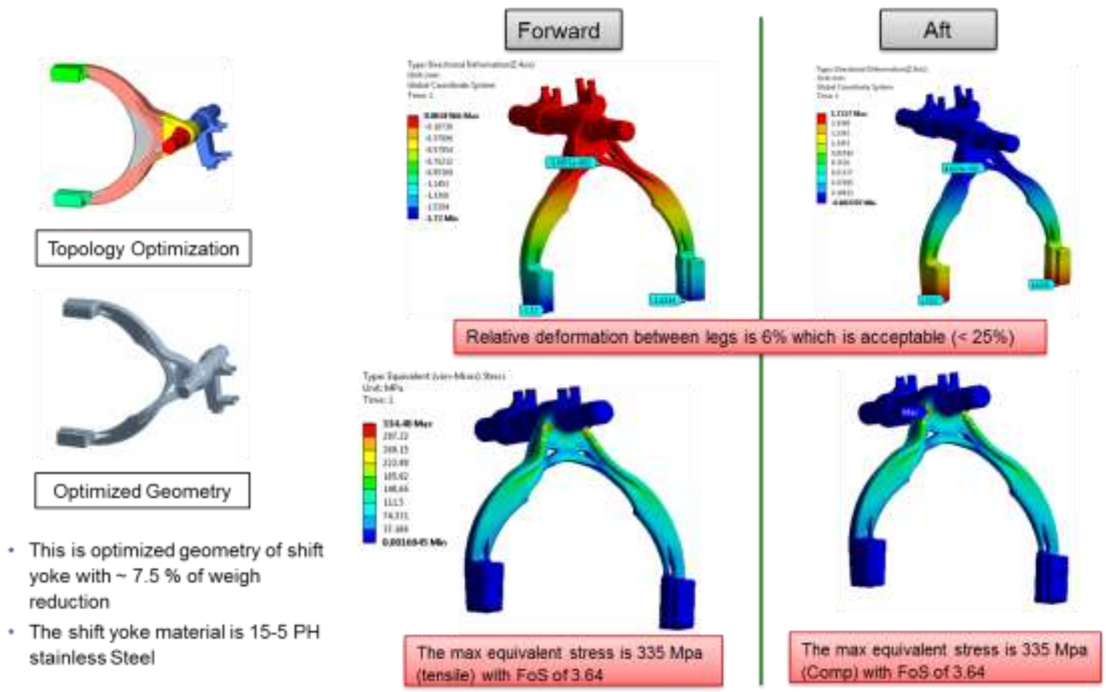


Figure 100. Stress-strain simulations of 3D-printed shiftbar assembly.
The 3D-printed part can operate safely for the specified stress and deformation requirements and provides 8% weight reduction.

The shiftbar subassembly was 3D-printed from 15-5-PH stainless steel powders by DMLS technology. Figure 101 shows the newly printed shiftbar assembly on the base plate and the support structure.

Figure 101. 3D-printed shiftbar assembly on the base plate and the support structure.

Figure 102 shows the 3D-printed shiftbar assembly separated from the support structure.

Figure 102. 3D-printed shiftbar assembly separated from the support structure.
Top left: 3D printed integrated shiftbar assembly after rough finish; Bottom left: after final finish. Top right: Two of 7-piece traditional shiftbar assemblies assembled together. Bottom right: 3D one-piece shiftbar assembly.

TASK 9. TRANSMISSION SHAKEDOWN TESTING

Transmission shakedown testing involves preliminary gearbox testing in each gear ratio on an Eaton dynamometer for proper function, lube, and heat control. Table 56 lists the transmission shakedown testing activities. We completed prototype four-speed transmission verification and shakedown testing. The prototype transmissions were spin-tested and checked for proper operation at all speeds and gears. We also performed the break-in procedure under load on all prototype units. Then, the prototype units were tested in all gear ratios at all speeds and all ranges of torque and power expected in normal operation.

Table 56. Task 9 activities

Task Name	Start	Finish
Task 9 Transmission Shakedown Testing	1/1/17	3/31/17
9.1 – Transmission controller development	1/1/17	3/31/17
9.2 – Motor and transmission integration	1/1/17	3/31/17
9.3 – Initial testing on the prototype unit on dyno	1/1/17	3/31/17

Deliverables: Transmission Shakedown Testing Report

Subtask 9.1 – Transmission Controller Development

The transmission control unit (TCU) for the four-speed transmission was designed and prototyped using a current production ECU and new software for the newly developed YY shifting mechanism. The YY-shifter is actuated by two DC motors: Y1 and Y2. The Y1 motor controls the position of the sliding clutch via shift mechanism for the first and second gear, and the same as Y2 motor for the third and fourth gear. Each motor operates independently and not simultaneously.

The control software that operates the YY-shifter was developed using the foundation (with some modifications) of the XY-shifter, another Eaton's shifter currently in production. The YY-shifter performed the automated shift of any gear from the first to fourth gear on the bench and dyno tests. The software integration of the YY-shifter and the transmission into vehicle supervisory level were also completed.

Initial transmission testing and calibration was performed during bench testing at Eaton in Southfield, as shown in Figure 103. This calibration, performed on the six assembled transmissions, established values to define when the shifters are in neutral, initially engaged, or fully engaged in gear. The calibration was confirmed through observation of input motor speed resulting in a corresponding speed on the output shaft, while carrying some torque.

Figure 103. Transmission calibration and controls bench test setup

Using this data to confirm when the correct gear was engaged, transmission testing occurred on the dynamometer in Southfield, confirming correct transference of speed and torque at multiple conditions across all four gears.

Subtask 9.2 – Motor and Transmission Integration

The proper fit and function of the motor/transmission mechanical interface was checked by mating the two together and measuring all critical interface dimensions and fits.

The selection and procurement of motor to transmission input shaft, companion flange from transmission to driveshaft were completed. Some spline clean up and lapping were needed between the motor input shaft and the input head gear. The mounting of the prototype units on the anechoic test bench was completed.

Subtask 9.3 – Initial Prototype Unit Testing on the Dynamometer

The prototype four-speed transmissions were broken-in under load using a proprietary Eaton transmission break in test procedure on the bench dyno. Then, the prototype transmissions were spin-tested and checked for proper operation at all speeds and gear by driving the transmission input with the dyno electric motor and absorbing the output shaft power using the dyno absorber unit.

The prototype transmissions were also subjected to NVH tests, efficiency tests, and steady state operation temperature rise tests. In all tests, the units were tested in all gear ratios at all speeds and all ranges of torque and power expected in normal operation. Transmission oil temperature was monitored; the ideal oil fill level was also determined through these tests.

NVH tests

Three prototype four-speed transmission units were tested for NVH on the dyno in the anechoic chamber at Eaton. Six pressure transducers were placed around the transmission for noise level measurements. Two accelerometers are attached to the bottom and side surfaces of the transmission to measure the vibration. The test rig and instrumentation are shown in Figure 104.

Figure 104. Transmission NVH setup

Figure 105 shows the noise levels of Unit 2, 3 and 4 at half load at Gear 1. The three units show consistent noise levels and are well below the team target level of 82 dBA. The tests results are summarized at different loads and speeds.

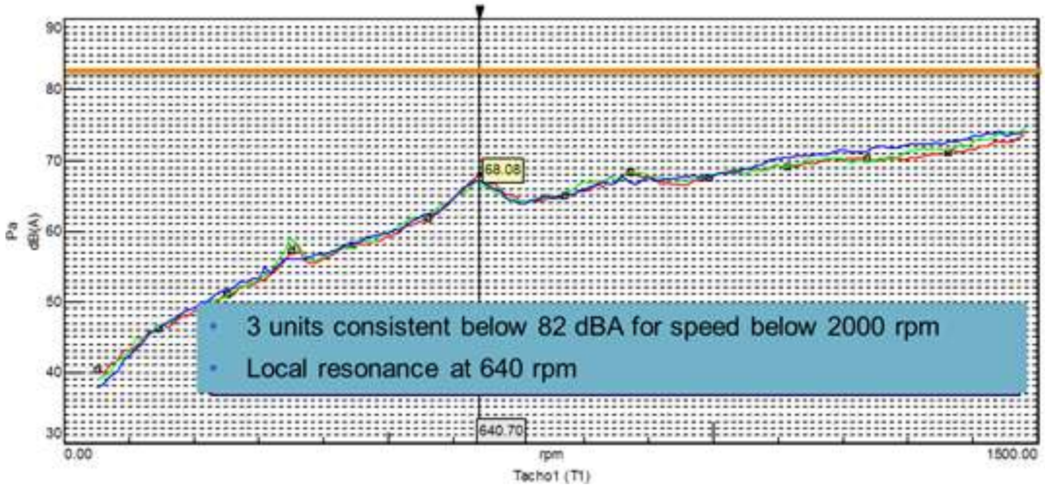


Figure 105. Noise levels of 3 units at half torque in first gear

Figure 106 shows the three units at full load and first gear. Again, the units have consistent noise level and all are well below the internal target level of 82 dBA.

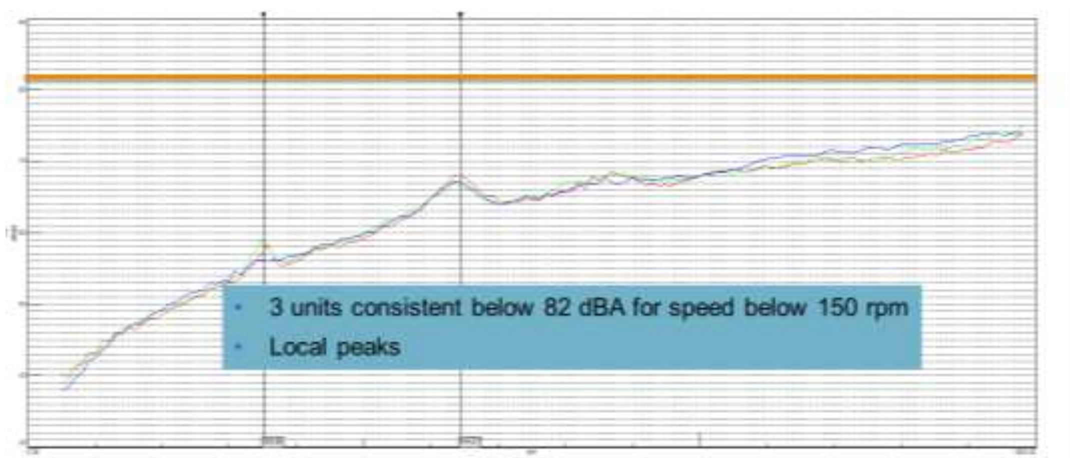


Figure 106. Noise levels of 3 units at full torque in first gear

Figure 107 shows the noise level of three units at half load at second gear. The three units are all below 82 dBA.

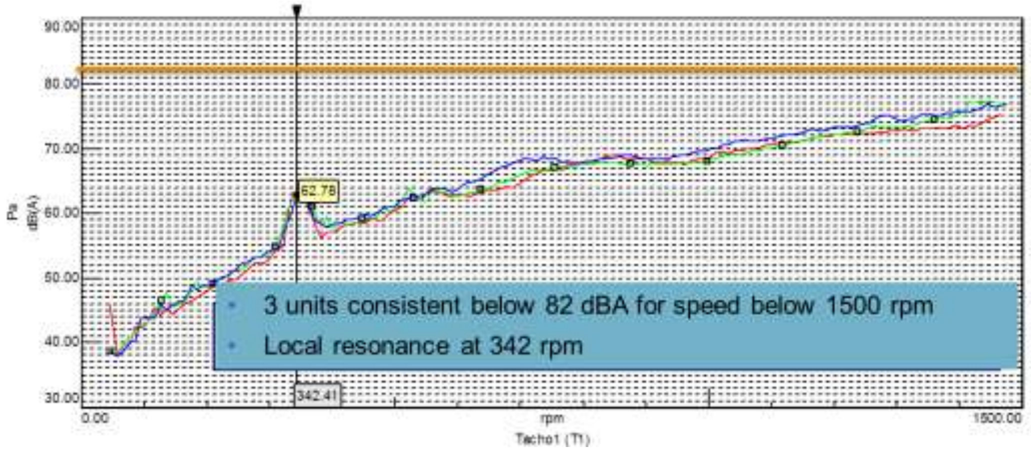


Figure 107. Noise levels of 3 units at half torque in second gear

Figure 108 shows the noise level of three units at full load at second gear. All three units are approaching the target of 82 dBA at high speed.

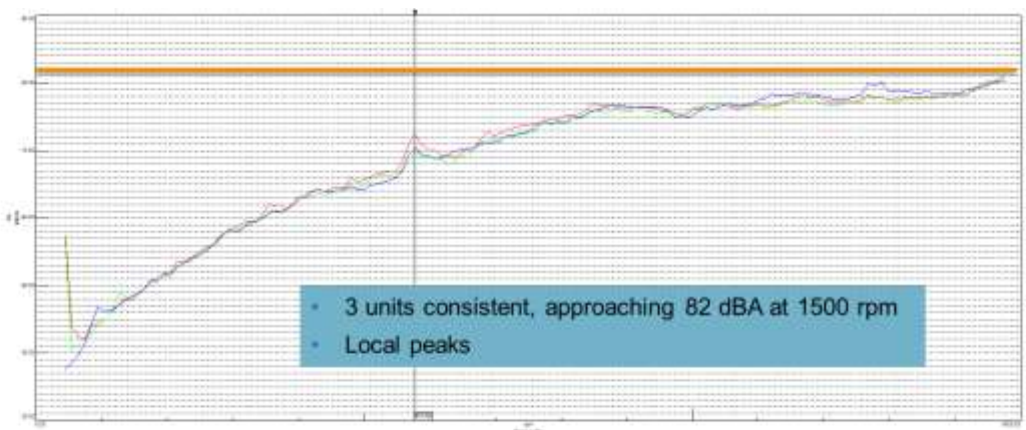


Figure 108. Noise levels of 3 units at full torque in second gear

Figure 109 shows the three units at half load in third gear. The units exceeded the noise level of 82 dBA when the speed is above 1600 rpm.

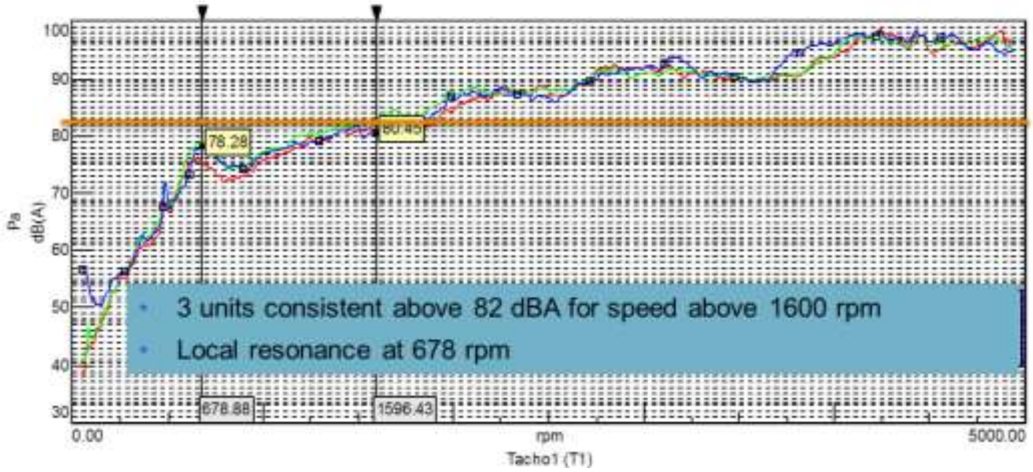


Figure 109. Noise level of 3 units at half torque in third gear

The noise data was further analyzed to identify root causes to noise performance. The order contribution curves in Figure 110 shows that the headset gear first order noise is the main noise contributor for transmissions.

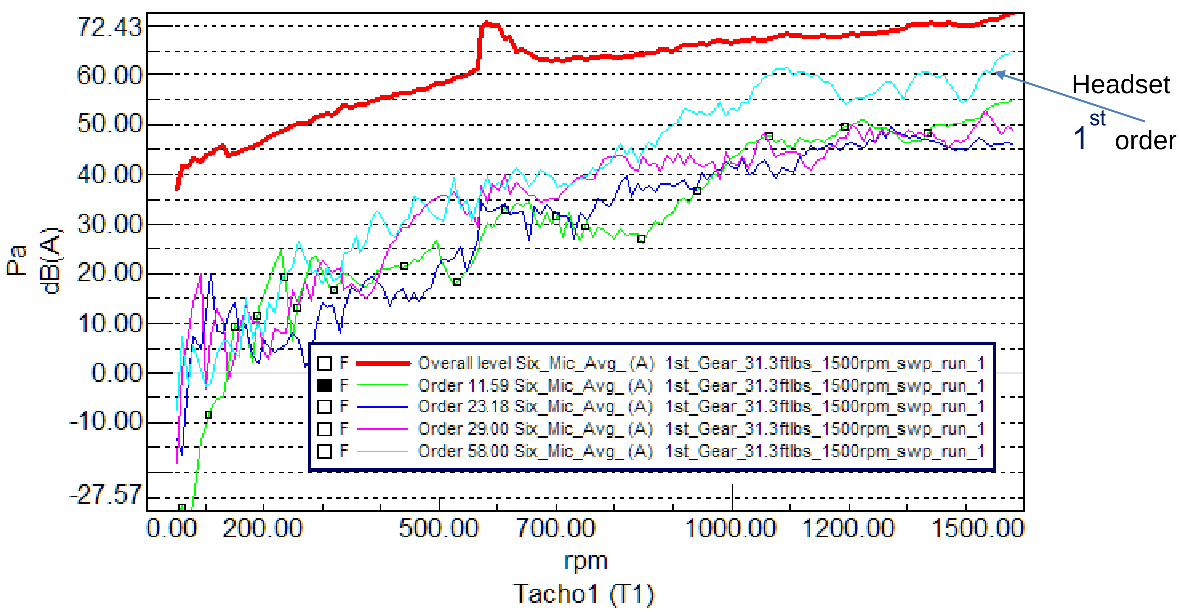


Figure 110. Noise level contributions

A possible reason of the headset gear set being the main noise contributor is that the gear set is an off-the-shelf design for internal combustion engine vehicles. The macro-geometry and surface finishing for EV applications need to be optimized.

Another investigation of the noise root cause is shown in Figure 111. Many side bands show along the main gear meshing order. These sidebands are typically caused by inconsistent meshing of gear due to assembly and tooth finishing variations.

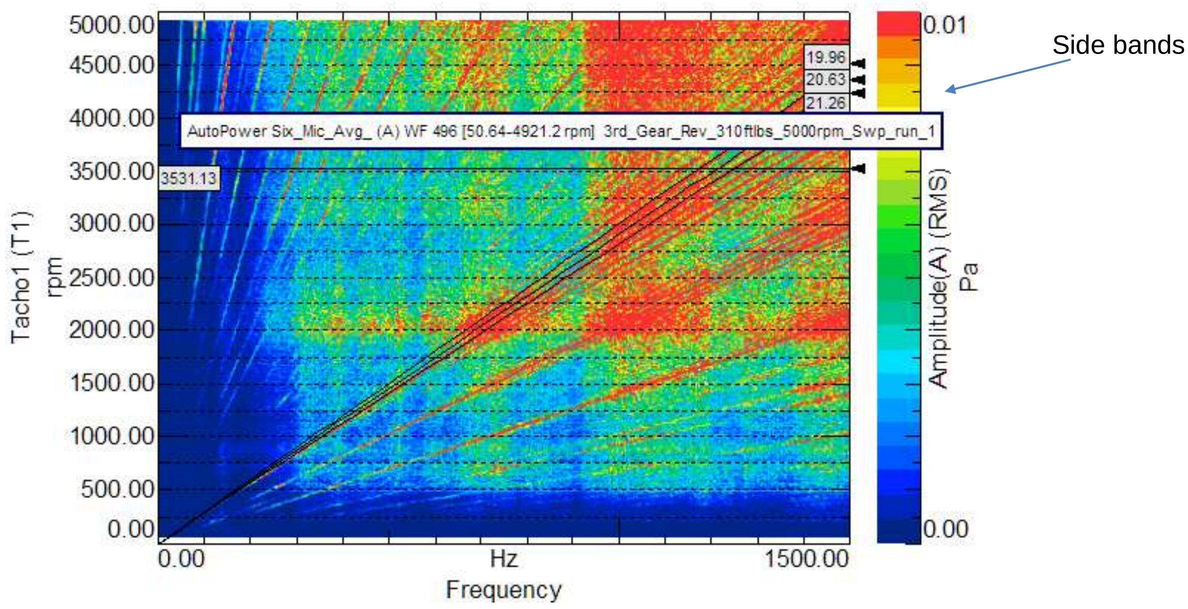


Figure 111. Noise waterfall analysis

Overall, the tested units show consistent noise performance. The units are at higher noise than Eaton internal noise target when the speed is above 2000 rpm. The head gear set is identified as the main noise contributor. Sidebands are noticed in the units. These noise tests and root cause analyses provide suggestions for design, manufacturing, and assembly improvement for better noise performance.

Transmission efficiency tests

We conducted a simulation using an internal Eaton program, Transmission Descriptive Language (TDL), to evaluate the gearbox efficiency for the four-speed EV transmission. Table 57 lists TDL's transmission efficiency predictions. The average efficiency with a wet sump was estimated to be 97.3% by simulation. Efficiencies are higher at higher loads and at direct drive.

Table 57. Transmission efficiency predictions by modeling software TDL

Gear	Ratio	% Efficiency at 100 kW, 3500 rpm	% Efficiency at 50 kW, 3500 rpm
1	4.8	97.5	96.5
2	2.6	97.5	96.5
3	1.6	97.5	96.5
4	1	99.0	98.0

The dedicated efficiency test dyno qualified for running the Standard § 1037.565 Transmission Efficiency Test was not available due to a maintenance shut-down. Therefore, the preliminary gearbox efficiency tests were carried out on the anechoic chamber dyno as well. The torque cell calibration ranges of anechoic dyno do not meet the calibration requirements of the standard test procedure therefore the efficiency test results are not official but only indicative. The anechoic dyno torque readings have large variation at low torque values such as those experienced on the fourth gear.

Prototype Unit#2, Unit#3, and Unit#4 transmissions were filled with oil and tested for gearbox efficiency under 16 different operating conditions: four conditions on each of four gears. Each condition was tested three times in a randomized test run. The oil fill amount was 4.8 l. Table 57 summarizes the average efficiency values at each gear as well as the overall average efficiency of the transmission. The overall efficiency value of %96.6 with a standard deviation of 1.1% is within two standard deviations of the targeted value of 98%. Furthermore, the measured efficiency is very close to the predicted average efficiency of 97.3% by simulation. The efficiency of EV-AMT is much better than the efficiency of a single-speed gearbox (93.4%) or a typical automatic transmission (~90%). Four-speed AMT efficiency can be improved 0.5 to 1% further using a dry sump lubrication technology as discussed in Subtask 7.1. However, the additional cost of adding a dry sump lubrication unit cannot be justified at this time.

Table 58. Transmission efficiency values measured on the NVH bench dyno on three units of prototype four-speed transmissions.

	% Efficiency-Unit#2	% Efficiency-Unit#3	% Efficiency-Unit#4	% Average Efficiency	StDev (%)
1st gear	94.9	95.8	94.2	95.0	0.80
2nd gear	96.2	97.0	95.5	96.2	0.77
3rd gear	96.7	97.5	95.2	96.4	1.19
4th gear	98.5	100.3	97.1	98.6	1.62
Overall	96.6	97.7	95.5	96.6	1.09

Transmission temperature rise tests in steady state operations.

We investigated the steady state temperature rise and the performance of the transmission without oil cooling with an anechoic test dyno. The cooler was disconnected, and a fan placed at the door to the dyno chamber to help simulate wind speed and air convection from driving. The tests were run with multiple scenarios for 10 minutes each, a conservatively long time for the electric motor to be running in a steady state, and then waited for the transmission oil to cool down to 100 F, a hot ambient temperature, before starting the next test. The results of the testing, shown in Figure 112**,** were very promising:

- As expected, testing in fourth gear (direct drive) did not generate much heat, even at maximum torque, and the oil reached only 141 F after 10 minutes.
- Even in second and third gears, with lots of oil churn through the gear meshes, the temperature reached 191 F after 10 minutes. This compares favorably with Eaton production transmissions. Considering the rare likelihood that the motor would run at its maximum continuous torque for ten minutes during any drive cycle, especially in a lower

gear, the team has no concerns that the transmission oil would near its maximum operating temperature of 250 F. Furthermore, real-world ambient cooling should be more effective than the fan in the dyno chamber.

Lastly, a reverse scenario (conditions would be the same for a braking/regenerative scenario) was run in first gear to discover any unique heat generation concerns running in the opposite direction. The very low temperature ramp (19 F in 10 minutes) confirms no heat issues from regeneration or reverse.

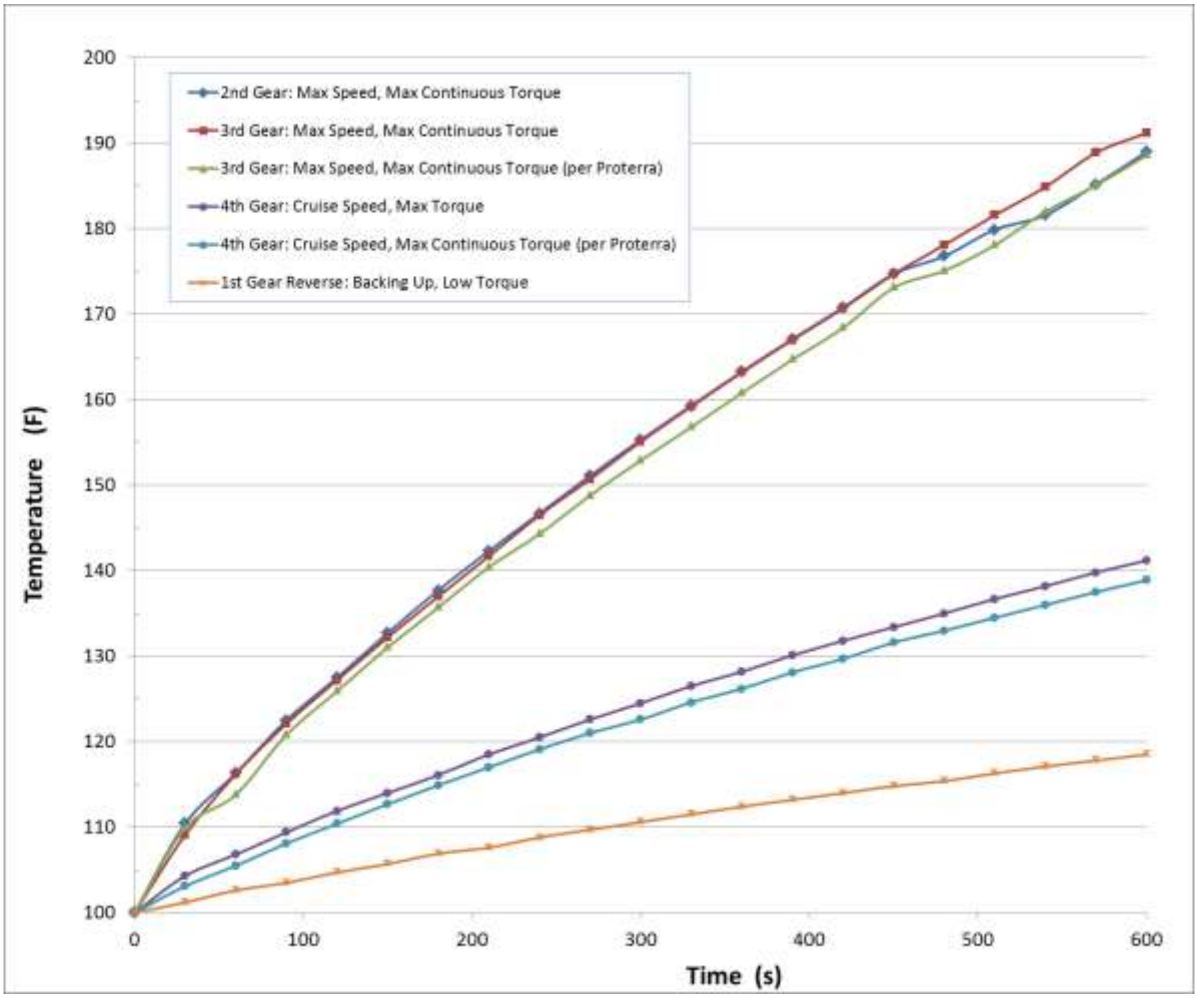


Figure 112. Transmission lubricant temperature versus continuous run time in steady state operation dyno tests.

TASK 10. POWERTRAIN INTEGRATION AND INSTALLATION FOR 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. Table 59 lists the activities accomplished at ORNL.

Table 59. Task 10 activities

Task Name	Start	Finish
Task 10. Powertrain integration and installation for hardware-in-the-loop (HIL)	4/1/17	9/30/17
10.1 – Mechanical installation of motor, MCU, transmission, TCU at the ORNL-VSI Lab	4/1/17	6/30/17
10.2 – HIL environment initialization and debugging	4/1/17	6/30/17
10.3 – Preliminary functional verification testing	4/1/17	6/30/17

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

Subtask 10.1 – Mechanical Installation of Motor, MCU, Transmission, TCU at the ORNL-VSI Lab

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 113. 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 114. 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.

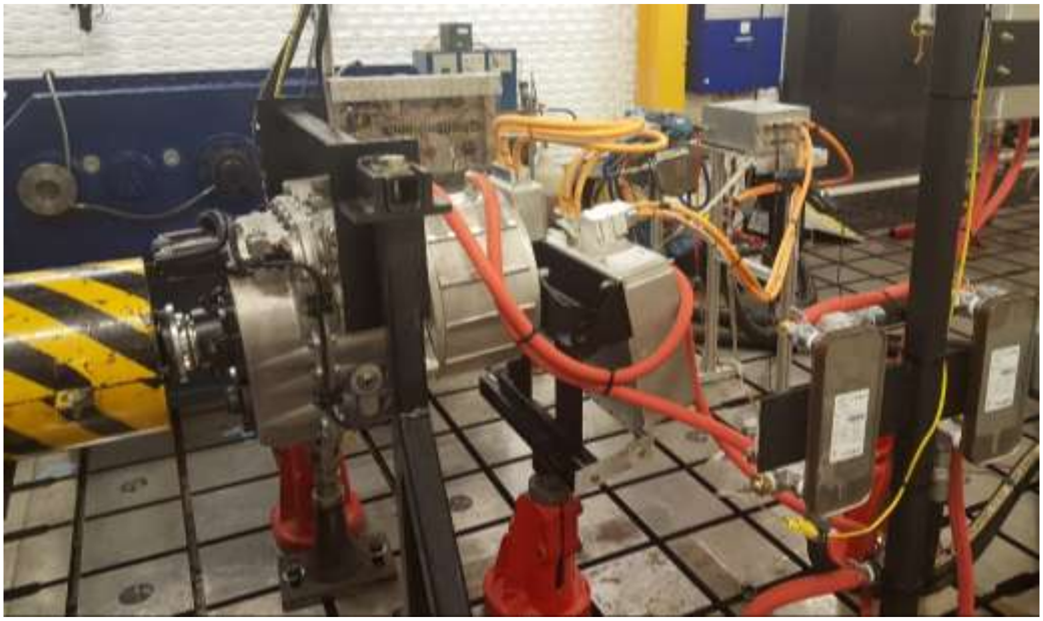


Figure 113. Electric powertrain system fully installed at ORNL's VSI Powertrain Test Facility

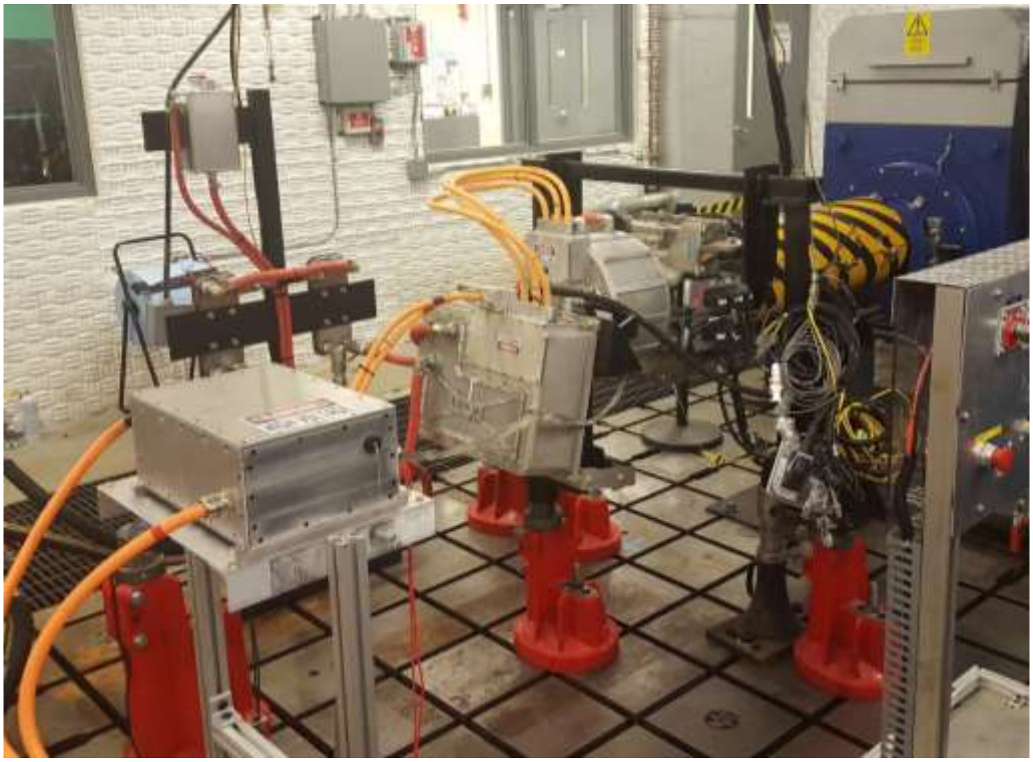


Figure 114. The high voltage equipment installed at ORNL's VSI Powertrain Facility.

Subtask 10.2 – HIL Environment Initialization and Debugging.

The vehicle model used in Tasks 3 and 4 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 115), 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 3 and 4 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 116).

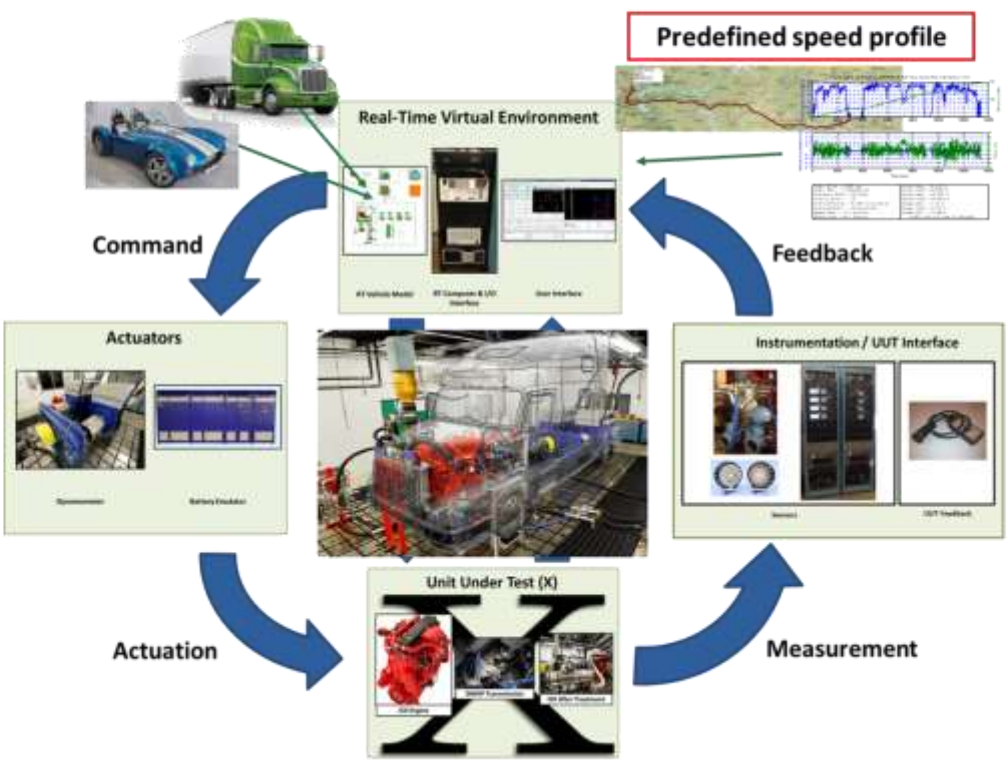


Figure 115. HIL flowchart

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

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.

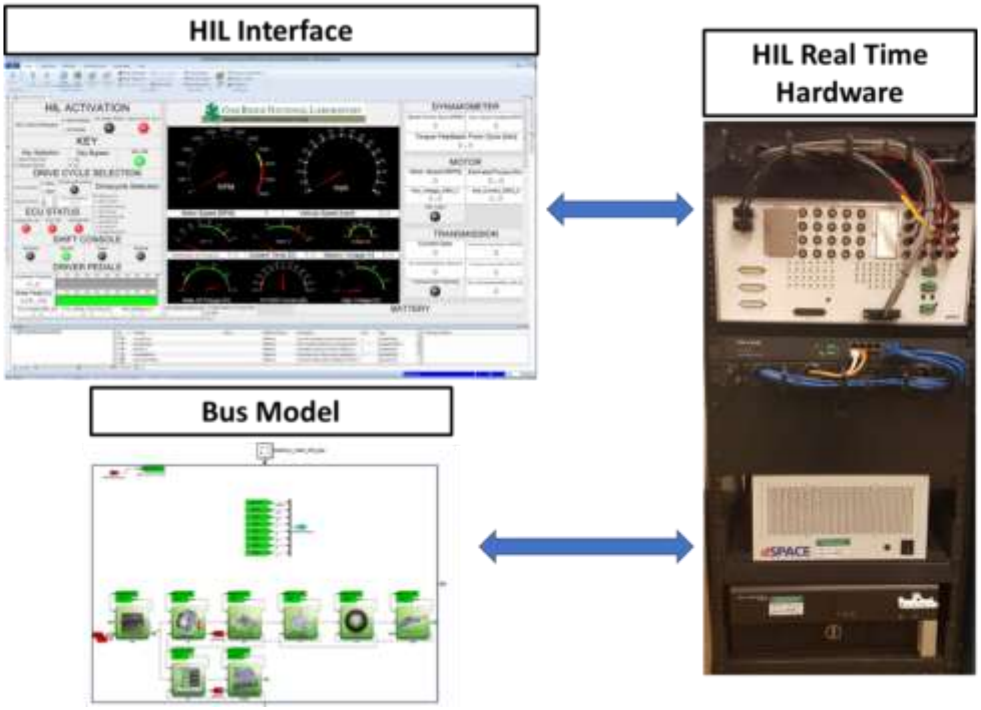


Figure 116. HIL layout and integration for powertrain testing

Subtask 10.3 – Preliminary Functional Verification Testing

As a sanity 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 117 shows one of the comparison graphs with vehicle and motor values.

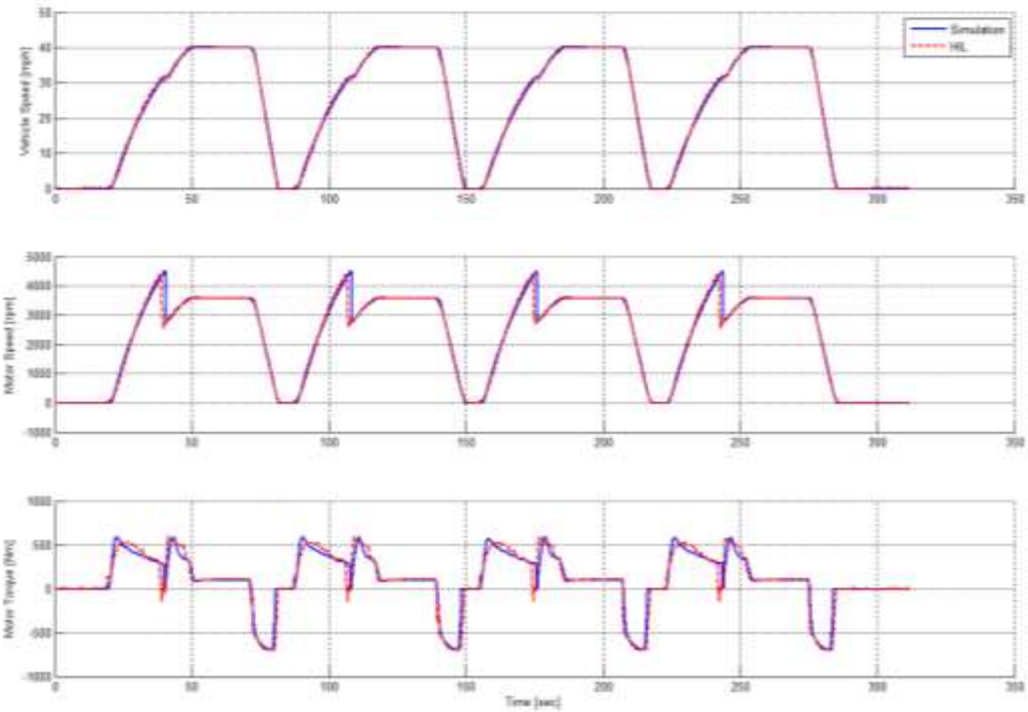


Figure 117. Comparing simulation results (blue) to HIL tests on the real hardware (red). From top to bottom; Vehicle Speed, Motor Speed, Motor Torque

TASK 11. 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, as listed in Table 60.

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.

Table 60. Task 11 Activities

Task Name	Start	Finish
Task 11. Electric powertrain HIL testing	7/1/17	9/30/17
11.1 Shift strategy validation and tuning	7/1/17	9/30/17
11.2 Steady state HIL tests	7/1/17	9/30/17
11.3 Acceleration and gradeability testing	7/1/17	9/30/17
11.4 Transient HIL tests with drive cycles	7/1/17	9/30/17

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

Subtask 11.1 – 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.

Subtask 11.2 – 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 118.

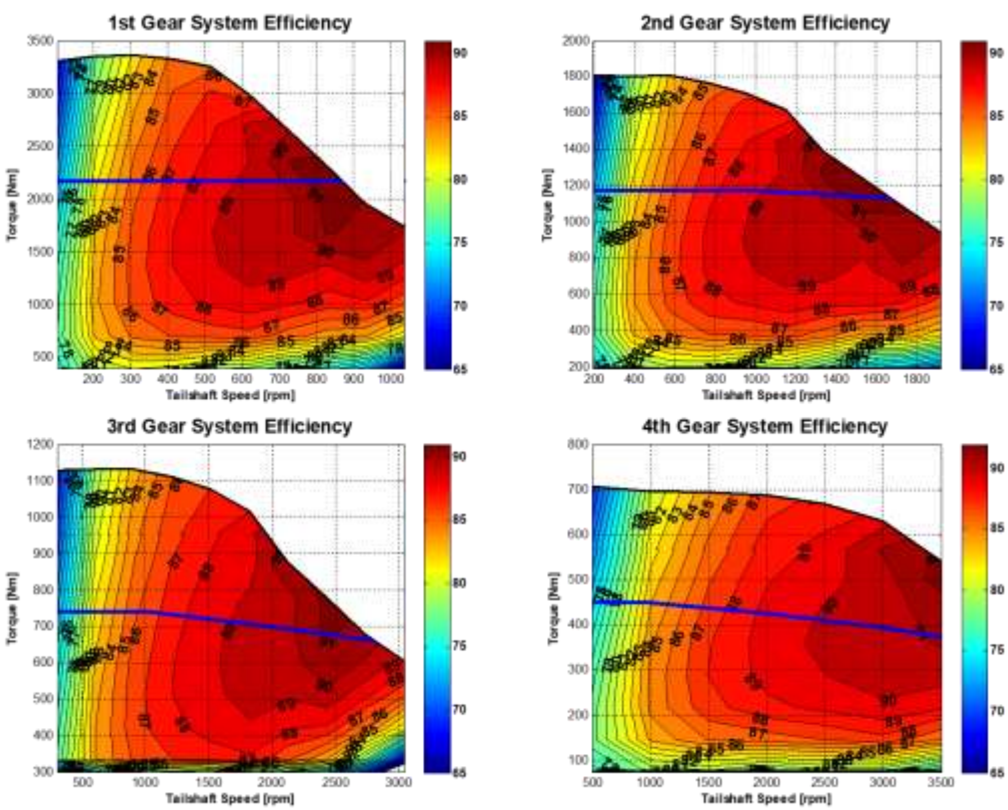


Figure 118. System efficiency maps for each powertrain gear.
Blue line denotes the constant power curve of the UQM motor at that given gear.

Subtask 11.3 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 61 to Table 64 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 61. 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 61 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 62. Maximum achieved vehicle speed on a grade at SLW

	HIL Gradeability Results at SLW 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 63. Maximum achieved vehicle speed on a grade at GVW

	HIL Gradeability Results at GVW 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 64. Maximum achievable grade at 10mph and 20mph @ GVW

	HIL Gradeability Results 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 62 to Table 64 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.

Subtask 11.4 – 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 65. 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)

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 66. 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 65. 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

Table 66. 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 67. 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 67. 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 119 to Figure 121. 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 122). 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.

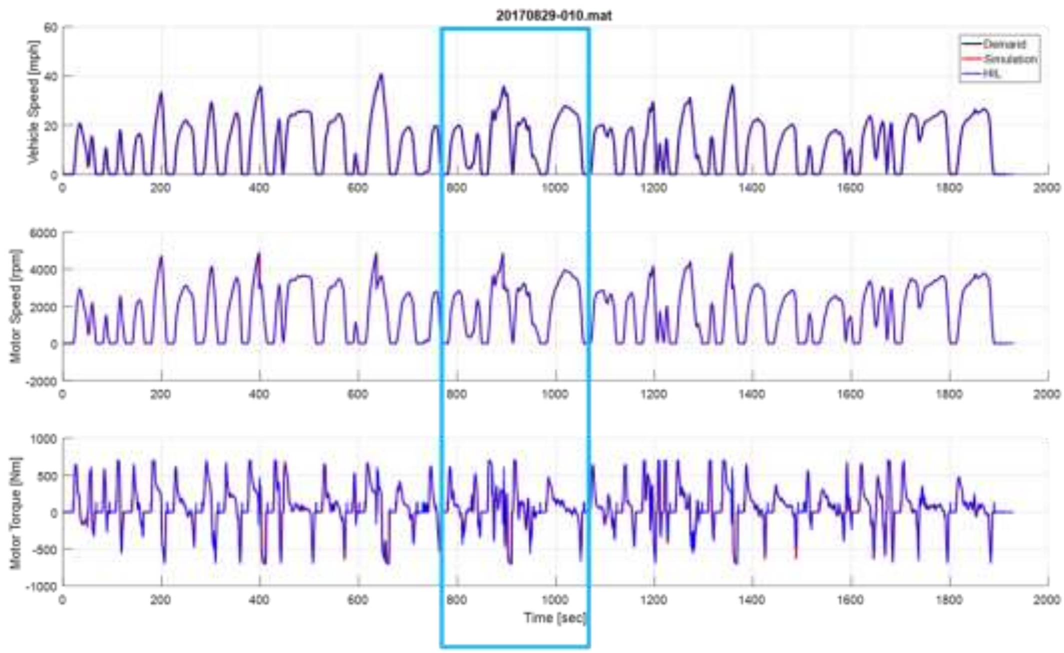


Figure 119. A single Orange County Bus Cycle run on the HIL system vs the simulated vehicle

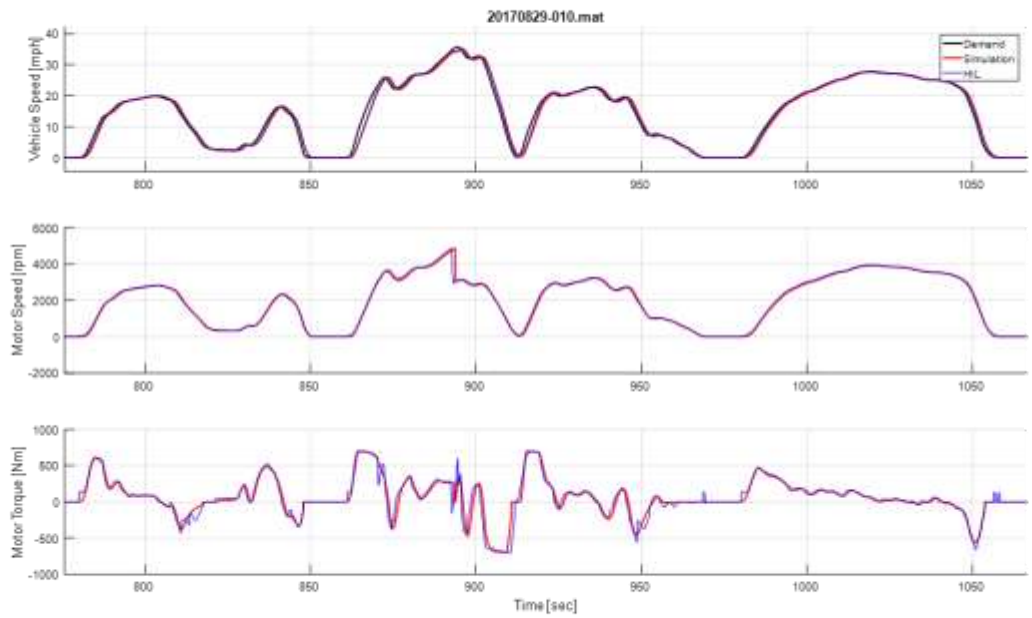


Figure 120. Enlargement of boxed area showing motor values for HIL system vs simulated vehicle

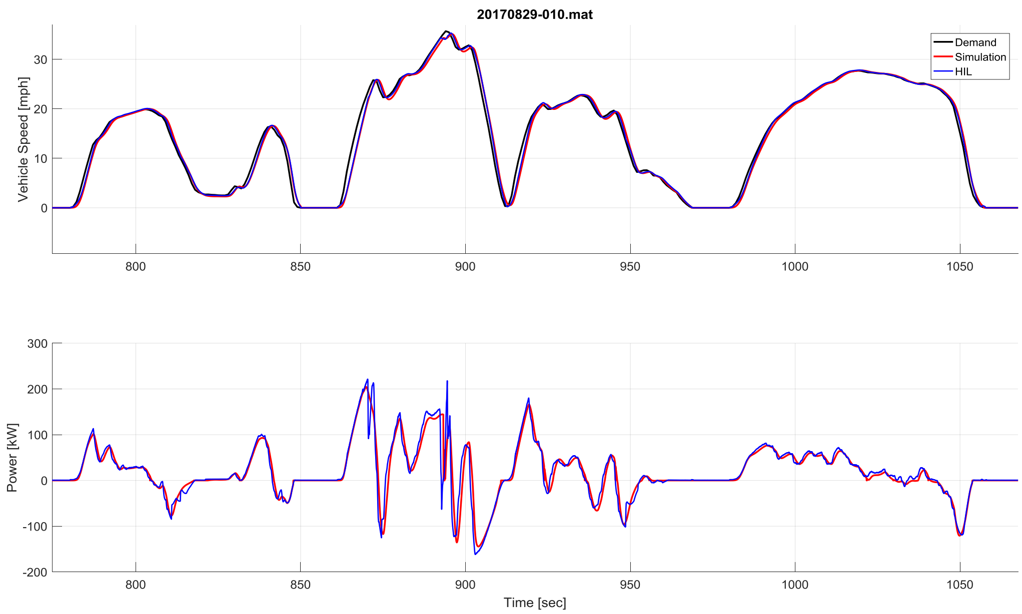


Figure 121. Enlargement of boxed area showing electric power for HIL system vs simulated vehicle

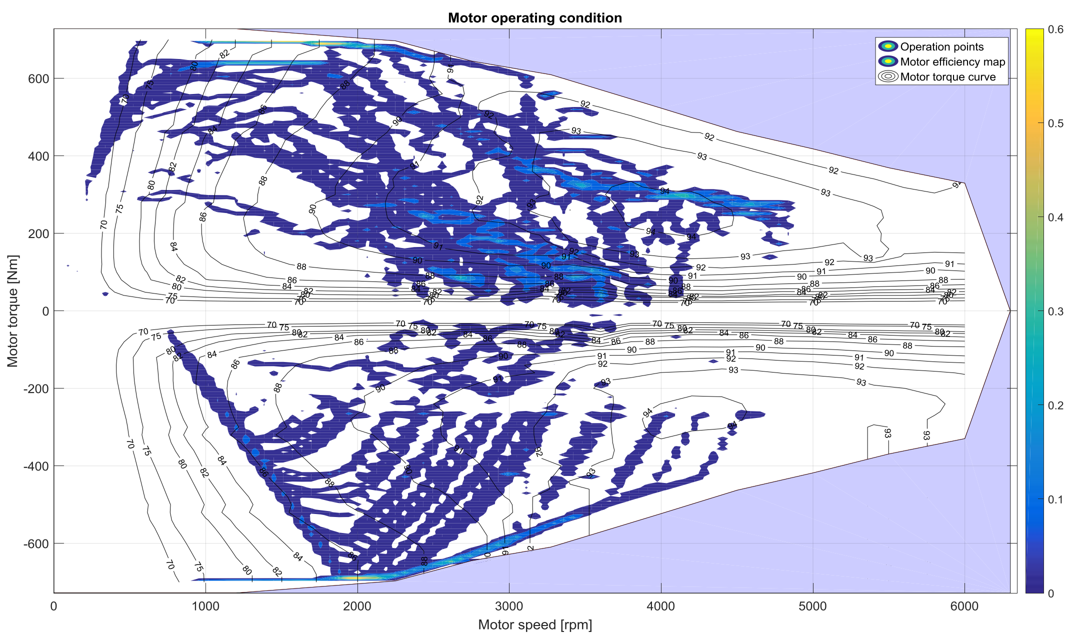
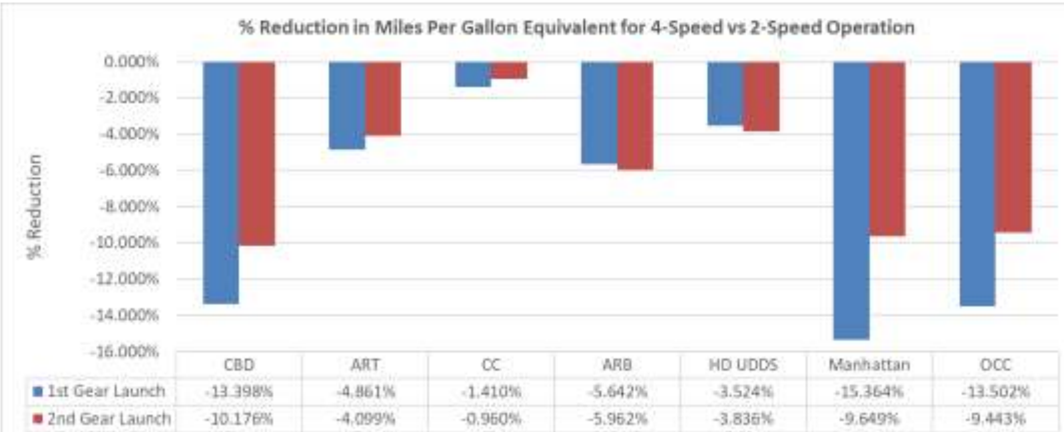


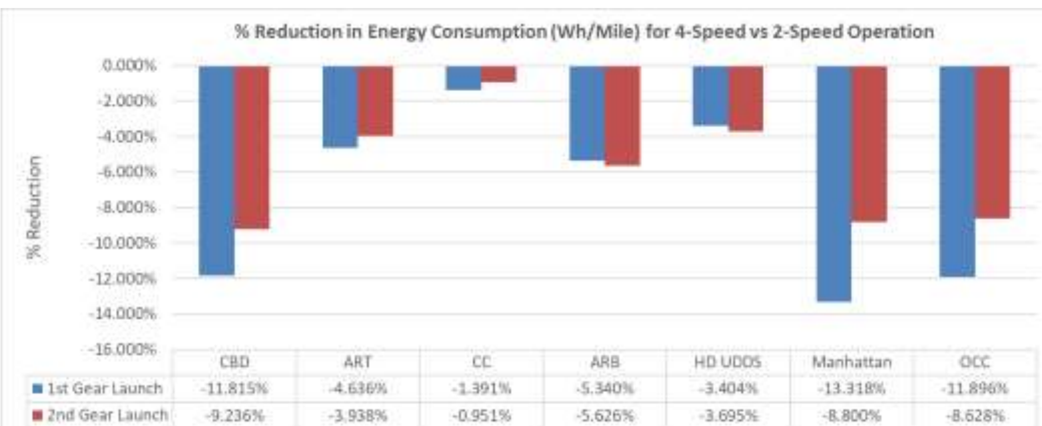
Figure 122. Operating region heat map of the HIL test run on the Orange County Bus Cycle

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 123 and Figure 124. 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 123 to the simulation results listed in Table 47, simulation results (Baseline 3 in Table 47) underestimate the efficiency gains significantly compared to the HIL tests.



	CBD	ART	CC	ARB	HD UDDS	Manhattan	OCC
1st Gear Launch	-13.398%	-4.861%	-1.410%	-5.642%	-3.524%	-15.364%	-13.502%
2nd Gear Launch	-10.176%	-4.099%	-0.960%	-5.962%	-3.836%	-9.649%	-9.443%

Figure 123. 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.



	CBD	ART	CC	ARB	HD UDDS	Manhattan	OCC
1st Gear Launch	-11.815%	-4.636%	-1.391%	-5.340%	-3.404%	-13.318%	-11.896%
2nd Gear Launch	-9.236%	-3.938%	-0.951%	-5.626%	-3.695%	-8.800%	-8.628%

Figure 124. 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 Federal Transit Administration (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 68 to Table 70 list the HIL test results of individual phases of the ADB drive cycles.

Table 68. 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 69. 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 70. 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 68 to Table 70, 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 71 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 71. 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 71. 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 52, one can tell that the modeling and simulations overestimate the efficiency gains by 5% compared to the test results in ADB drive cycle.

Table 72 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 72 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 72 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 72. 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

TASK 12. BASELINE AND DEMONSTRATION VEHICLE INTEGRATION AND BUILD

Eaton provided two prototype units of four-speed transmission together with wiring harnesses, transmission control unit, and a display unit indicating the motor rpm and transmission gear to Proterra for vehicle integration. The Task 12 activities are shown in Table 73.

Table 73. Task 12 activities

Task Name	Start	Finish
12.0 - Vehicle powertrain integration at Proterra	4/10/17	6/1/17
12.1 - Traction motor/gearbox mounting	4/10/17	6/1/17
12.2 - Wiring harness installation	4/10/17	6/1/17

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

After receiving the four-speed transmission from Eaton, the Proterra build team modified the rear ProDrive cage to accept the new powertrain system. The team used the same motor mounting brackets. The transmission brackets were designed for the four-speed transmission and the cage was modified to accept the new mounts. Because the new transmission was much shorter than the baseline, a new driveshaft was required. Figure 125 highlights the mechanical modifications.

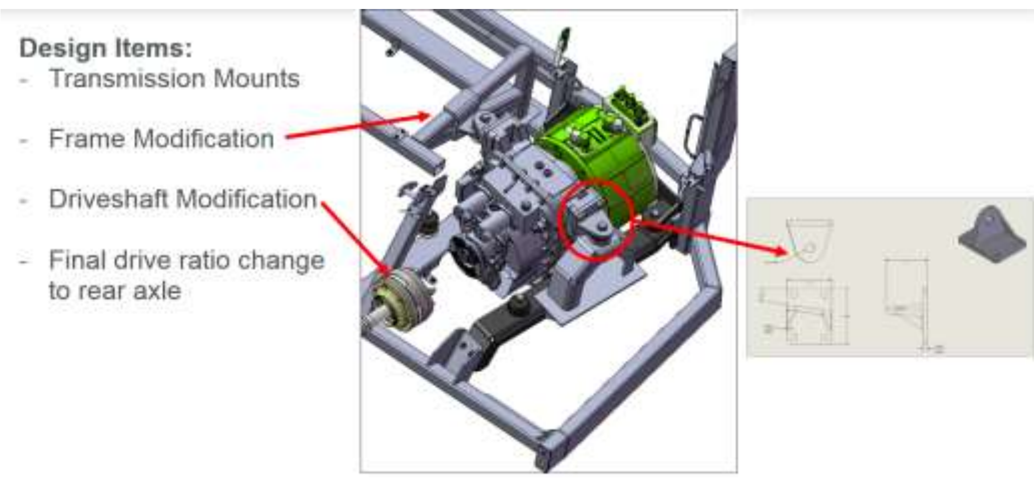


Figure 125. Mechanical modifications to mount the transmission on ProDrive cage

Subtask 12.1 – Traction Motor and Gearbox Mounting

Figure 126 shows how Proterra built up the rear ProDrive cage. The picture on the left shows the existing motor brackets.

Figure 126. Integration of four-speed transmission on ProDrive cage

Subtask 12.2 – Wiring Harness Installation

The vehicle side low voltage harnesses were modified to accept the four-speed transmission harness. This harness provides power, ground, and CAN communication for the transmission and ECUs. Figure **127** highlights the CAN communication interface between the vehicle and transmission. The Eaton team traveled to Proterra for initial vehicle system calibration to ensure that the bus could safely be loaded and unloaded from the transport trailer. The vehicle executed successful shifts from neutral to first gear, and from neutral to reverse (which had not yet been tested at ORNL). The team also implemented an appropriate level of "Creep"/Urge to Move, and added more functionality in reverse to mimic behavior of the existing Proterra buses. An additional CAN display unit was installed to support testing of the transmission modes for one-, two-, and four-speed operation.

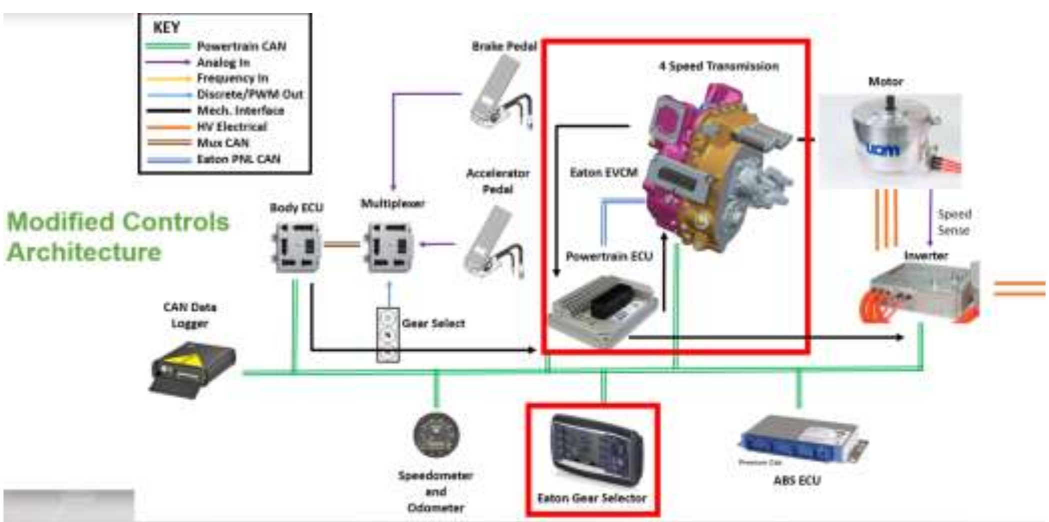


Figure 127**. CAN communication interface between the vehicle and transmission**

TASK 13. INITIAL VEHICLE SHAKE-DOWN TESTING

After Proterra integrated the four-speed transmission on the Proterra BE35 electric bus in Greenville, South Carolina, the vehicle was delivered to Eaton at the Marshall, Michigan proving ground. Further refinements to shifter control strategy, tuning, and shift quality assessment were conducted. Initial vehicle shakedown testing was performed on the integrated bus as outlined in Table 74.

Table 74. Task 13 activities.

Task Name	Start	Finish
13.0 – Initial vehicle shakedown testing at Eaton	6/2/17	8/28/17
13.1 – In-vehicle shift control strategy, shift tuning, and shift quality assessment.	6/2/17	8/28/17
13.2 – Vehicle NVH testing	6/2/17	8/28/17

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

Subtask 13.1 In-Vehicle Shift Control Strategy, Shift Tuning, and Shift Quality Assessment

Eaton programmed a ProEmion display unit that easily displays the current gear of the transmission, as shown in Error! Reference source not found..

Figure 128. Dashboard display unit shows transmission mode, gear engaged, motor rpm, and transmission oil temperature.

The display unit improved driver functionality to allow control of:

- Driving mode: four-speed (first through fourth gears), two-speed (first and third gear only), manual mode
- Launch gear for four-speed mode: first- or second- gear

The display now shows the current gear (0 for neutral), motor RPM, and trans. oil temperature (°C).

The bulk of the controls activity occurred at Eaton's Marshall Proving Grounds. Torque requests during the shift process were refined to reduce the ramp down and recovery times, leading to an average total shift time of about 1.2 seconds. The torque ramp ups and ramp downs were modified for each shift based on feel (drivability), resulting in the smoothest shift possible while still maintaining an acceptable shift time. The actual torque interrupt time was 0.3 seconds or less during the shifts. Gear shifts and corresponding motor shift points are listed in Table 75.

Table 75. Transmission shifts and corresponding motor shift points

Shift	Gear Ratios	Upshift (RPM)	Post-Upshift (RPM)	Downshift (RPM)	Post-Downshift (RPM)
1st-2nd	4.828 : 2.605	4800	2590	150	278
2nd-3rd	2.605 : 1.646	4800	3033	2000	3165
3rd-4th	1.646 : 1.000	4800	2916	2000	3292

The transmission is limited to 5000 RPM input shaft speed, so a late shift of 4800 RPM for all shifts allows the motor to operate at higher efficiency for as long as possible. Downshift points of 2000 RPM ensure no "hunting" – the post-upshift speeds are far enough from 2000 RPM that slight vehicle decelerations won't cause a downshift, but the motor operating range during regen is in a more efficient zone than if a single gear was maintained down to zero speed.

The Eaton team, based on a history of transmission development, suspected that launching the bus in second gear would be preferred from a drivability standpoint, and this proved to be true. Figure 129 shows that launching in first yields a slight improvement in acceleration to 20 mph, but launching in second yields faster times to 30 mph and to the start of the second-to-third shift. The overall bus acceleration also feels smoother to the passengers and driver without the first-to-second shift. In production, Proterra, or even a specific fleet, could decide whether to launch in first or second, perhaps based on load weight, route, and efficiency.

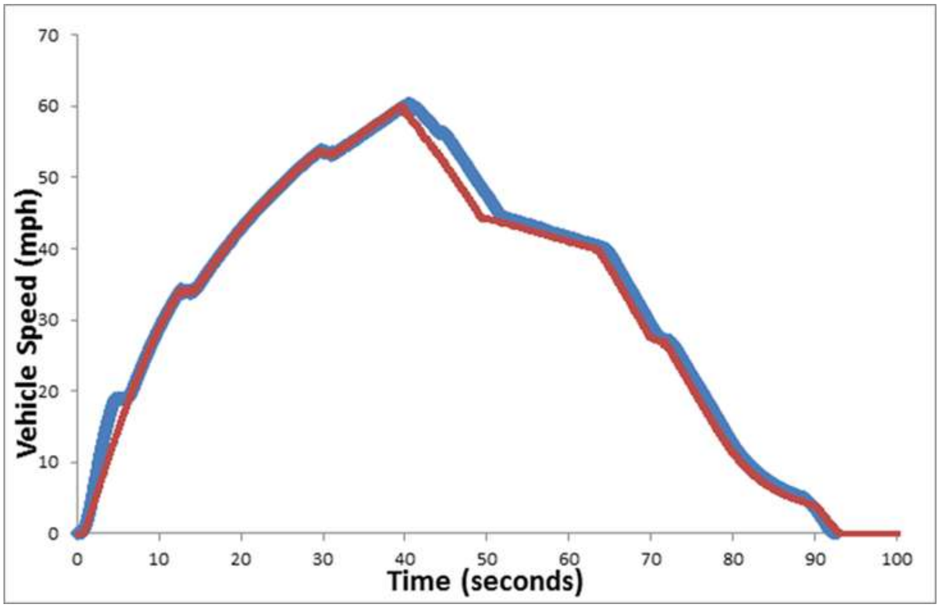


Figure 129. Vehicle speed vs. time for unloaded bus launching in 1st (blue) and 2nd gear (red)

Gradeability was another key target, with the longer paved 15% and 8% grade areas available for testing at Marshall Proving Grounds. The unloaded bus could handle both grades at speed, maintaining 30 miles an hour up the length of the grade in both second and third gears. However, for launching from rest in the middle of the grade, first gear allowed a faster launch and improved acceleration up the grade. The transmission was subsequently programmed to use the combination of grade sensor and vehicle weight to determine the best gear to launch from a grade. For our unloaded testing in Marshall, the transmission downshifted to first to launch from the 15% grade and from (or maintain) second gear to launch from the 8% grade.

Lastly, the team determined the right amount of regeneration, during both coast and braking events. Torque commands were calibrated for each of gear, and the torque ramps were smoothed to eliminate some of the initial jerks due to the switch from a positive-to-negative torque request. Higher regeneration is commanded when the driver presses the brake than when the bus is coasting (no accelerator or brake).

Subtask 13.2 – Vehicle NVH Testing

The vehicle level noise test of the four-speed transmission was conducted at Eaton's Marshall Proving Grounds. The Proterra BE-35 Electric Bus integrated with four-speed transmission was instrumented to measure the noise and vibration under various driving conditions. Three runs were conducted for each drive condition:

- First gear launch and coast down at flat ground, and track
- Second gear launch and coast down at flat ground, and track
- Third gear constant speed coasting

- First gear launch at 15% grade
- Second gear launch at 8% grade
- First gear at reverse

Figure 130 shows the drive's right ear microphone set. An accelerometer was also instrumented at the driver's seat track to measure the vibration at the driver's location.

Figure 130. Vehicle noise measurement setup at driver seat

Figure 131 shows the microphone at the rear passenger seat.

Figure 131. Vehicle noise measurement setup at passenger's location

Four accelerometers are also instrumented at the four drive unit's mounts. Figure 132 shows the locations of the two of the four accelerometer locations.

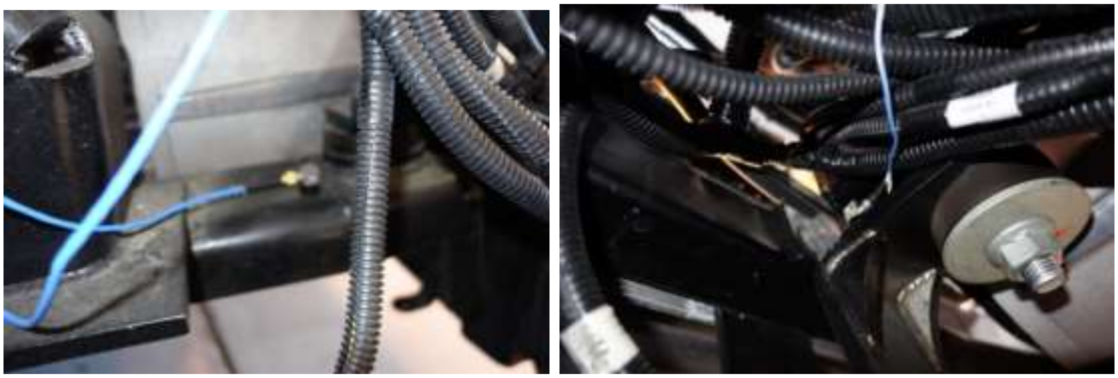


Figure 132. Vehicle vibration measurements locations

A speed sensor was instrumented at the input shaft of the transmission. All order-based analyses are based on this speed signal. The order numbers for each gear set at different transmission gear position are listed in Table 76.

Table 76. Gear set noise orders for transmissions at different gear position

Position	# TEETH	Trans first order	Trans second order	Axle first order	Axle second	Portal first	Portal second
M/S 1st Gear	56						
C/S 1st Gear	18	11.6	23.2	3.7	7.5	2.2	4.5
M/S 2nd Gear	47						
C/S 2nd Gear	28	18.0	36.1	6.9	13.8	4.1	8.3
M/S 3rd Gear	35						
C/S 3rd Gear	33	21.3	42.5	10.9	21.9	6.6	13.1
C/S Drive	45						
Input	29	29	58				
Direct drive (4th)				18	36	10.8	21.6

Figure 133 shows the noise levels at the driver and passenger sides as a function of motor speed in first gear. The figure shows that noise at the rear passenger location is higher than that at driver's location, as the passenger location is close to the powertrain. The differences can be as high as 5 dBA. All noise levels are well below 80 dBA and acceptable.

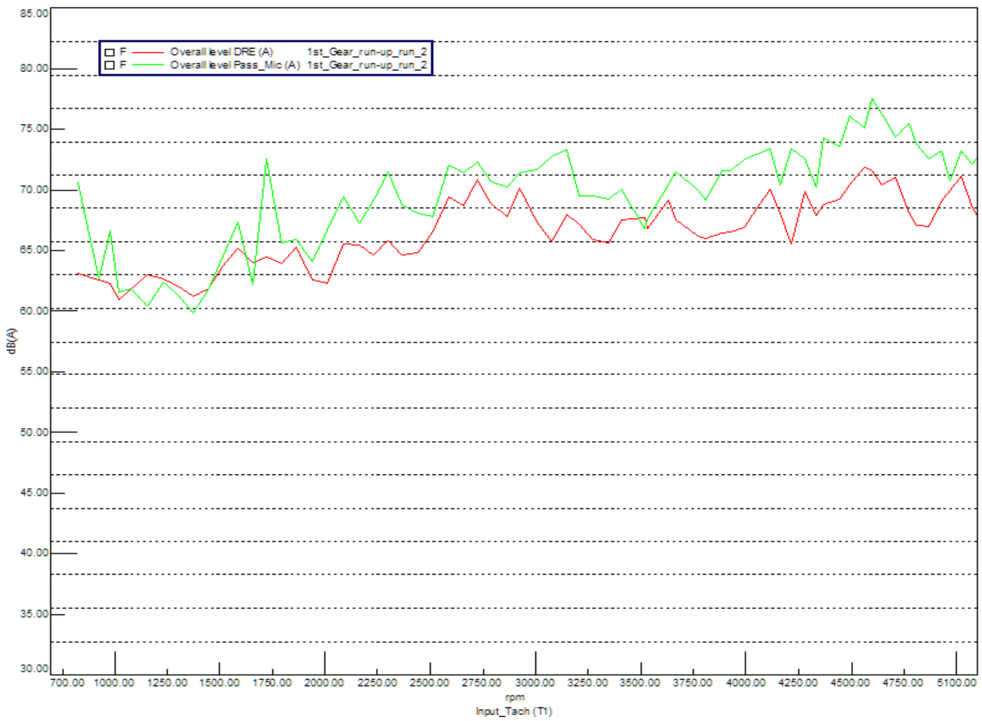


Figure 133. Noise levels inside vehicle at first gear launching

Figure 134 shows the driver's noise level vs the driver seat's vibration level at first gear. We concluded that the drive's noise is not directly correlated with the driver seat vibration. Thus, the noise is largely air-borne noise from the drivetrain at the rear of the vehicle.

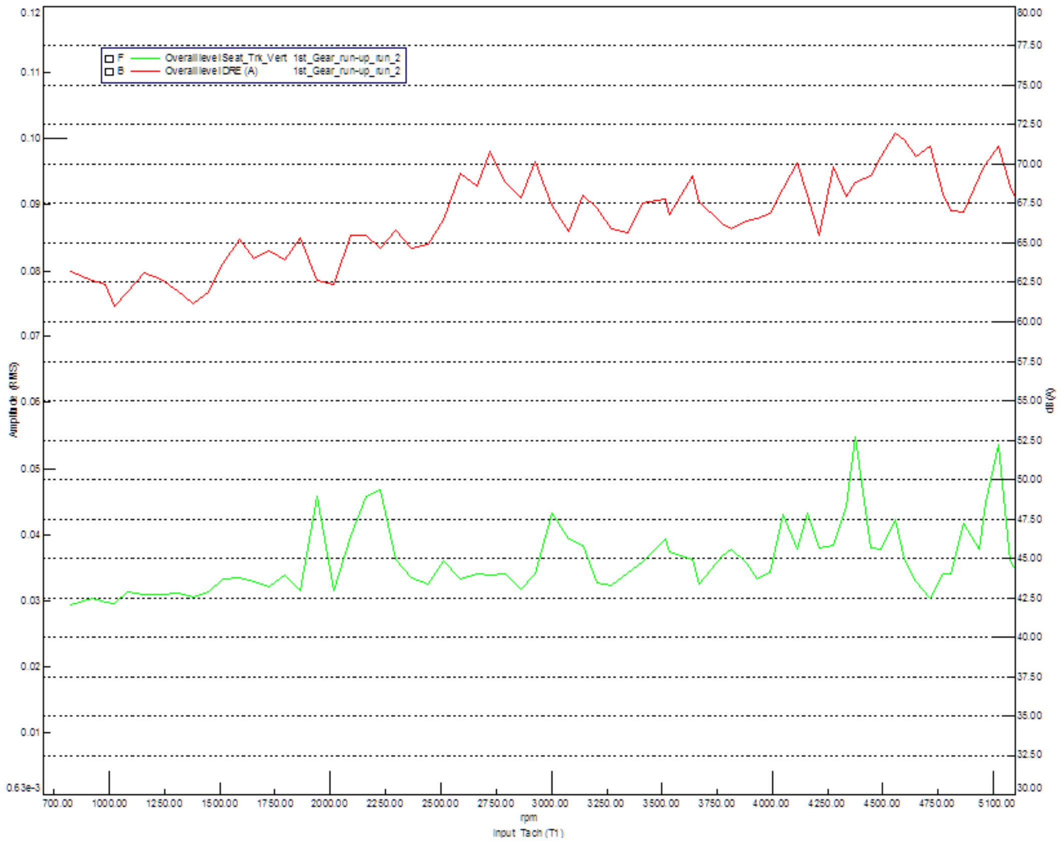


Figure 134. Noise and vibration level at driver's location

A noise contribution investigation ranked the contributions of all the gear sets inside the drivetrain. Figure 135 shows all the gear set noise contribution during first gear launch. The two sets of rear axle gears are the first two noise contributors, the third contributor is the headset gear set of the transmission.

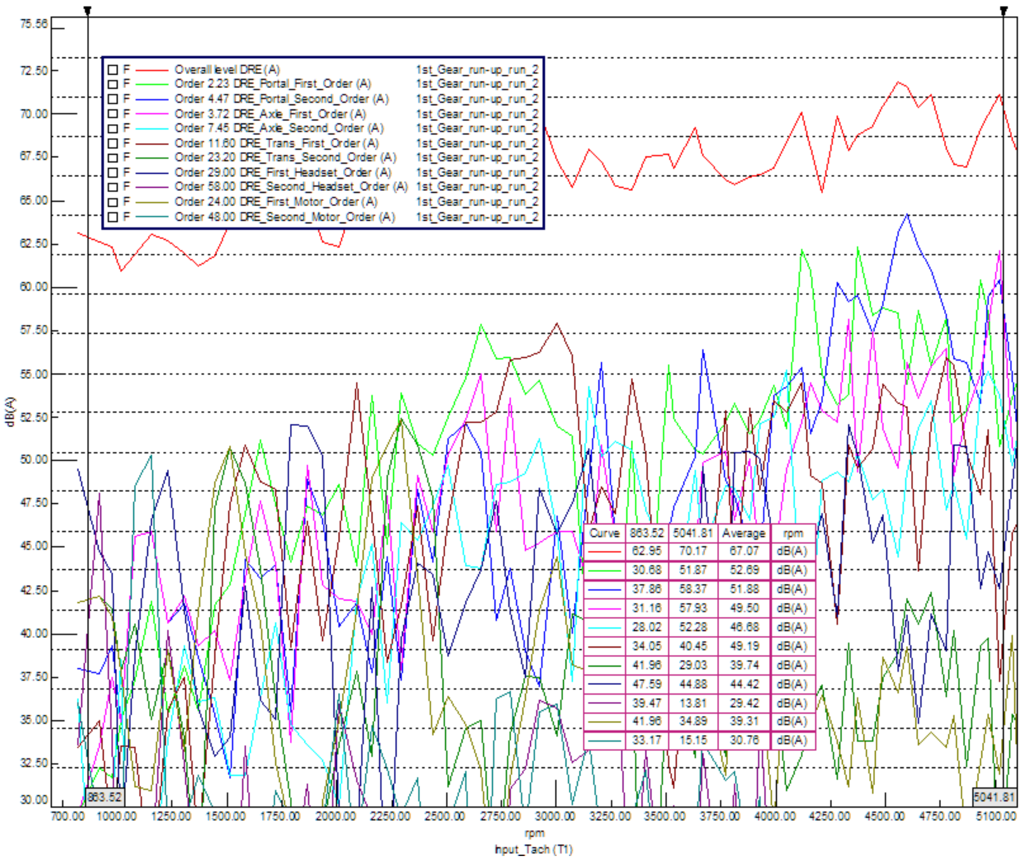


Figure 135. Noise contribution ranking

In summary, the noise, vibration and harshness tests of the electric bus with four-speed transmission were conducted at Eaton's Marshall Proving Ground. The bus demonstrated consistent NVH performances during the tests. The in-cab noise is mainly air-borne from the powertrain. The major noise contributor is the rear axle.

TASK 14. INTEGRATED VEHICLE TESTING

The Proterra BE-35 Electric Bus with four-speed transmission was delivered to NREL in Golden, CO. The vehicle was tested and compared against baseline configurations emulating single- or two-speed transmissions on a heavy-duty chassis dynamometer at NREL. Table 77 lists the integrated vehicle testing activities.

Table 77. Task 14 activities

Task Name	Start	Finish
14 - Integrated vehicle testing	8/28/17	10/31/17
14.1 - Vehicle instrumentation and preparation for testing	8/28/17	9/13/17
14.2 - Vehicle coast down testing on track	8/28/17	9/13/17
14.3 – Vehicle transient drive cycles testing on chassis dyno	9/13/17	10/31/17
14.4 - Vehicle gradeability testing on chassis dyno	9/13/17	10/31/17
14.5 - Vehicle acceleration testing on chassis dyno	9/13/17	10/31/17

Deliverables: Vehicle demonstration testing report.

Subtask 14.1- Vehicle Instrumentation and Preparation for Testing

The primary purpose of the work NREL performed was to evaluate the effects of a multi-speed transmission on the electric vehicle efficiency by performing vehicle dynamometer studies over a variety of drive cycles and conditions. NREL also performed vehicle coastdowns. The EV was the full battery-electric transit bus provided by Proterra which was equipped with the prototype four-speed Eaton transmission. ORNL conducted HIL testing prior to vehicle testing at NREL. The vehicle was evaluated to compare the EV’s energy efficiency operating in one-, two-, or four-speed transmission arrangements. Different transmission arrangements were achieved through electronic control. The transmission could be operated as a four-speed with a regular take-off in first or second gear (practically a three-speed), a single speed using the third gear, or a two-speed using the first and third gear.

Test Vehicle

The vehicle used in this study was a modified 35 ft. 2014 Proterra ECORIDE BE35 equipped with a prototype Eaton four-speed automatically operating gearbox.

Vehicle Model: Proterra ECORIDE BE35

Model Year: 2014

VIN: 1M9TG16J3ES826059

Seating capacity: 31

Odometer reading: 10964 miles (upon arrival and coast down testing)

Vehicle Test Weight: 33 500 lb (SLW for drive cycle and acceleration testing)
37 530 lb (GVW for acceleration and gradeability testing)

Heavy-duty chassis dynamometer

Vehicle efficiency experiments were performed using the heavy-duty chassis dynamometer at the NREL ReFUEL lab, as shown in Figure 136. The chassis dynamometer can simulate transient loads on heavy-duty vehicles of up to 80,000 lb. GVW at speeds up to 60 mph.

Figure 136. Vehicle on the chassis dynamometer

The dynamometer is an in-ground installation with 40-inch diameter rolls protruding above the surface to interface with the vehicle tires, see Error! Reference source not found.. The base nertia of the dynamometer rotating components simulates 31,000 lb. of road load force. A direct current (DC) motor (380hp absorption/360hp motoring capacity) supplements dyno forces to simulate the vehicle inertia/force to a range of 8,000 to 80,000 lb.

To assure road load simulation accuracy and consistency, the dynamometer was subjected to continual quality control procedures and checks. With the vehicle lifted off the rolls, an automated dynamometer warm-up procedure was performed daily prior to drive cycle operations until the dyno temperatures and parasitic forces stabilized. A parasitic losses function eliminated day-to-day parasitic force changes. After each test run, a loaded coastdown procedure ensured vehicle stability and dynamometer parasitic losses and accurate road load simulation throughout the day. A 20-minute soak period was used between each test. Each drive cycle and transmission configuration was tested for three hot runs to ensure data repeatability. The transmission configurations were tested in an alternating pattern to account for daily drift in the datasets for all configurations. Prior to the first hot run, a conditioning run using the same drive cycle ensured the dynamometer and vehicle temperatures were stabilized and the dynamometer loading was set at stable conditions.

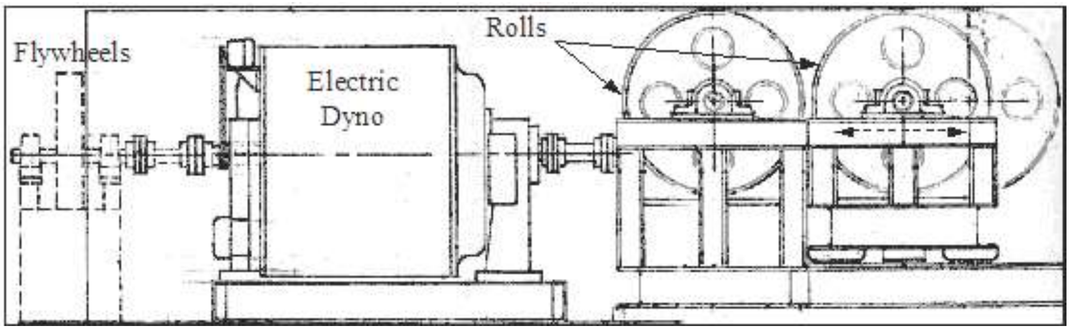


Figure 137. Schematic of ReFUEL's in-ground heavy-duty chassis dynamometer

The vehicle was secured to the dynamometer with the drive axle over the rolls. The vehicle was driven by NREL staff following a prescribed speed trace on a monitor. A small fan forced cooling air under the rear of the vehicle to cool the transmission and the axle to replicate real-world driving operations in which air would flow under the vehicle and provide cooling. Various vehicle parameters, including vehicle speed, electric motor and transmission operation, and other component temperatures were monitored and logged by a data acquisition (DAQ) system.

Data acquisition and instrumentation

All data were recorded at 1 Hz. Multiple chassis dynamometer measurements were recorded, and the vehicle was instrumented to record operating conditions. Testers measured the ambient temperature around the bus in the test bay with thermocouples along with the surface temperatures of the gearbox and the axle.

Vehicle energy supply/consumption

Vehicle battery storage was recharged during the soak period between each drive cycle run by a three-phase, 50kW generator and an Eaton-manufactured fast charger supplied by Proterra, as shown in Figure 138. Vehicle energy to and from the motor was measured with instrumentation already installed on the motor/batteries and reported over the CAN bus.

Figure 138. Generator and a fast charger

Subtask 14.2 - Vehicle Coast Down Testing on Track

Vehicle coastdowns were performed on a private service road at the Front Range Airport in Watkins, Colorado, as shown in Figure 139. The road north-south and was surveyed for elevation changes and length. The usable section of the track is approximately 1.9 km long (1.2 miles). The southern end of the track has an average grade of 0.4% (Figure 140), which meets requirements of the SAE J2263 coastdown standard. Forces due to the grade of the test track were corrected during data processing. Due to track length limitations, the coastdown runs were split into two passes (~60-40, and ~40-0 mph). Five repeats were performed in both directions for both speed ranges (a total of 20 passes). The coastdown test weight (TW) was 33,500 lb. The resulting coefficients are listed in Table 78. The road-load force equation is given in [1].

Figure 139. Vehicle coasting at the Front Range Airport

Table 78. Road-load coefficients

a	294.81	lb
b	0.7176	lb/mph
c	0.0872	lb/mph^2

Where:

$$\boldsymbol{Road\ Load\ Force\ (lbs) = c * speed_{mph}^{2} + b * speed_{mph} + a} \qquad \textbf{(1)}$$

These coefficients were used for the chassis dynamometer drive cycle evaluations. A plot of the road-load forces as a function of speed is shown in Figure 141.

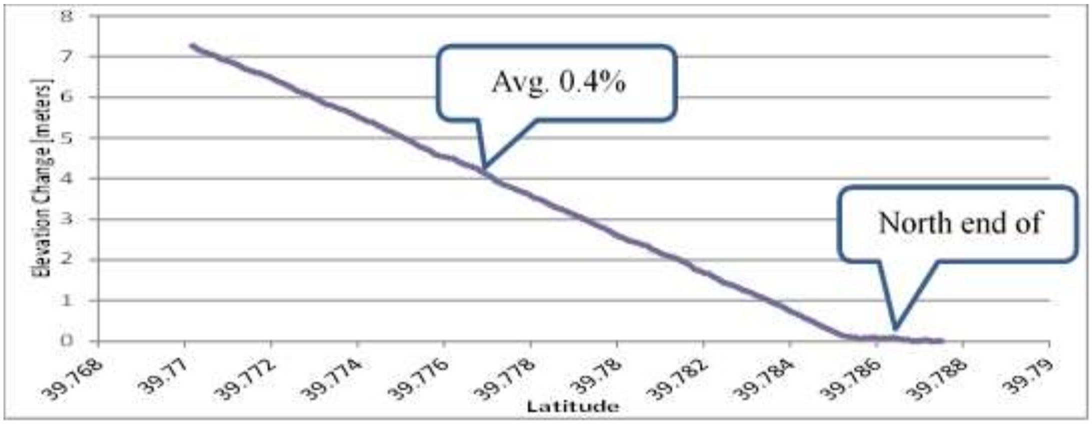


Figure 140. Grade profile of the test track.

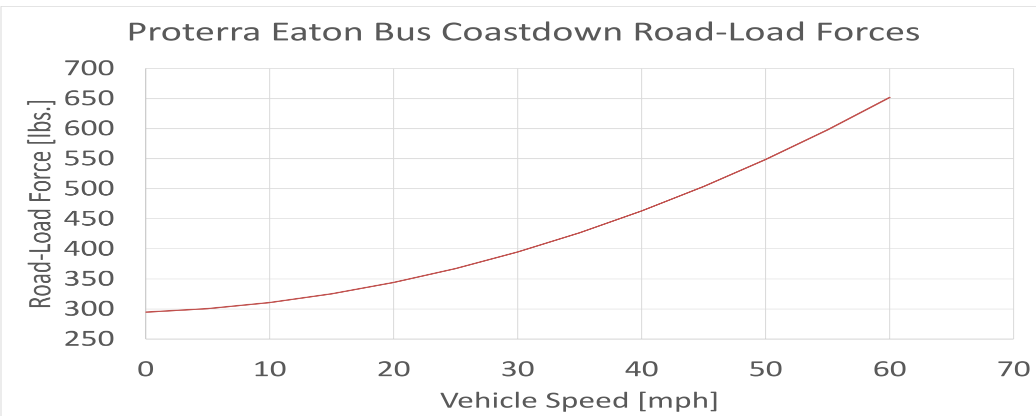


Figure 141. Resulting road load force curve

Subtask 14.3 - Vehicle Transient Drive Cycles Testing on Chassis Dyno

Five chassis dynamometer drive cycles were chosen because they represent typical transit bus operation: Manhattan Bus, Orange County Transit Authority (OCTA), Urban Driving Dynamometer Schedule (UDDS), World Harmonized Vehicle Cycle (WHVC), and a modified Altoona cycle (which is a modified business-arterial-commuter bus cycle). The original Altoona cycle has multiple repeated sub-cycles and consumes a substantial amount of energy. The time for each recharge for a complete Altoona cycle extended well beyond the standard soak times and was not sustainable for driving multiple repeats in one day. Therefore, NREL modified the Altoona cycle to run each sub-cycle only once. The test cycle profiles and statistics are shown in Figure 142 and Table 79.

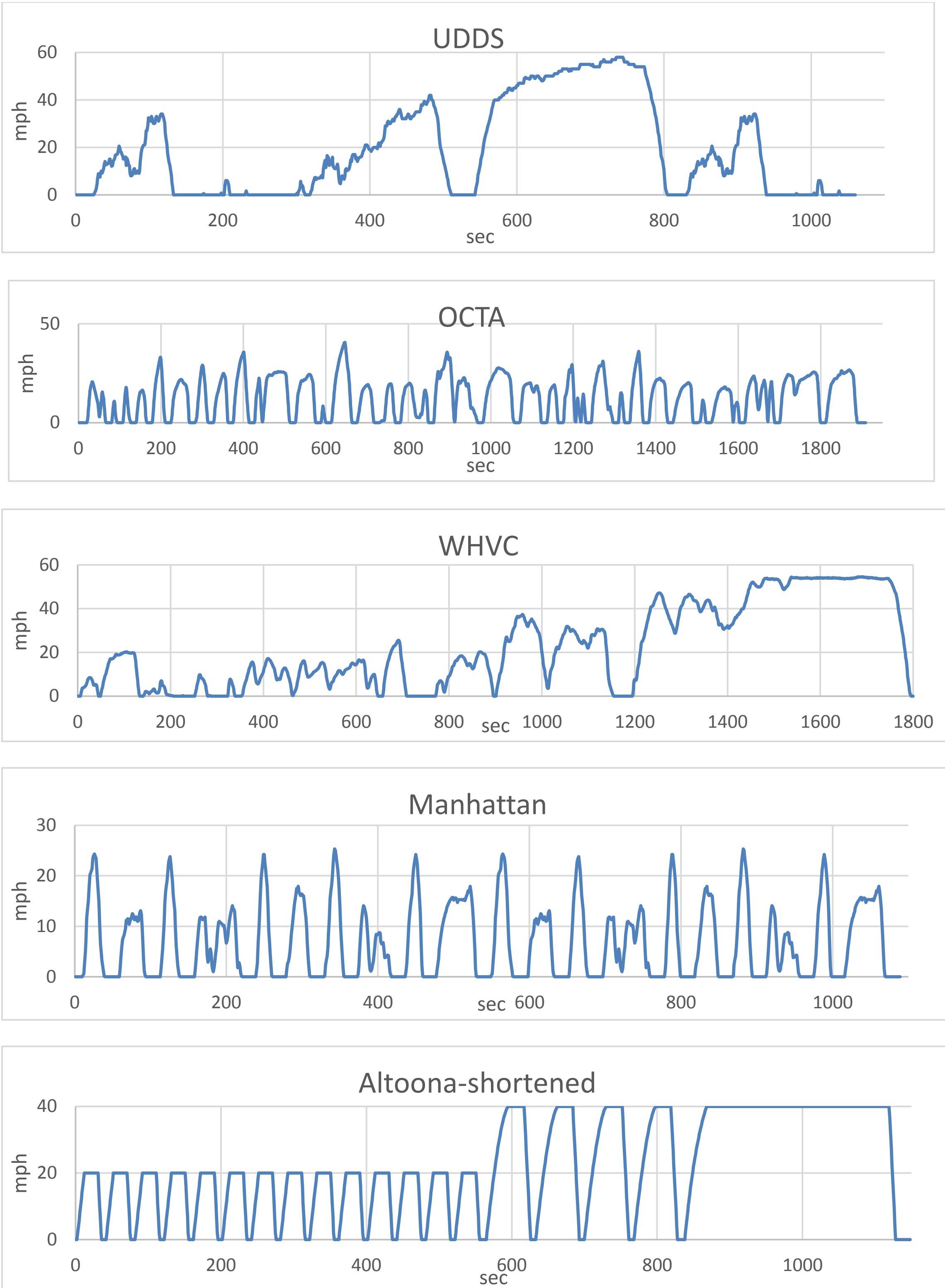


Figure 142. Chassis dynamometer drive cycles

Table 79. Chassis dynamometer drive cycle information

	OCTA	MAN	WHVC	ALTOONA_S	UDDS
Duration (sec)	1910	1089	1800	1148	1061
Distance (miles)	6.54	2.07	11.22	7.05	5.55
Total Avg. Speed (mph)	12.33	6.83	22.44	22.08	18.84
Avg. Driving Speed (mph)	15.67	10.67	25.98	25.65	28.23
Max Speed (mph)	40.63	25.4	54.56	40	58
Kinetic Intensity (1/mile)	3.59	9.14	0.39	0.86	0.61
Total # of Stops	31	20	12	19	14
Stops per Mile	4.74	9.66	1.07	2.7	2.52

Table 80. Energy consumption for all test cycles using different transmission modes

		Cycle Dist.	Dyno Dist.	Battery Energy Net		Dies. Eq.
Transmission Config	Cycle	[Miles]	[Miles]	[kWh]	[kWh/Mile]	mpg_de
4 speed - 1st gear launch	Manhattan	2.07	2.08	-4.07	-1.96	17.87
2 speed - 1st & 3rd gear	Manhattan	2.07	2.11	-4.60	-2.14	16.65
1 speed - 3rd gear	Manhattan	2.07	1.97	-4.47	-2.28	16.26
4 spd improve over 2sp	Manhattan			13%	9%	9%
4 spd improve over 1sp-3	Manhattan			10%	17%	17%

		Cycle Dist.	Dyno Dist.	Battery Energy Net		Dies. Eq.
Transmission Config	Cycle	[Miles]	[Miles]	[kWh]	[kWh/Mile]	mpg_de
4 speed - 1st gear launch	UDDS	5.55	5.54	-7.93	-1.43	25.94
2 speed - 1st & 3rd gear	UDDS	5.55	5.57	-8.19	-1.47	25.18
1 speed - 3rd gear	UDDS	5.55	5.51	-8.04	-1.46	25.62
4 speed - 2nd gear launch	UDDS	5.55	5.55	-8.11	-1.46	25.47
4 spd improve over 2sp	UDDS			3%	3%	3%
4 spd improve over 1sp-3	UDDS			1%	2%	2%
4 spd improve over 4sp2	UDDS			2%	2%	2%

		Cycle Dist.	Dyno Dist.	Battery Energy Net		Dies. Eq.
Transmission Config	Cycle	[Miles]	[Miles]	[kWh]	[kWh/Mile]	mpg_de
4 speed - 1st gear launch	OCTA	6.54	6.61	-10.59	-1.60	23.34
2 speed - 1st & 3rd gear	OCTA	6.54	6.63	-12.05	-1.82	20.71
1 speed - 3rd gear	OCTA	6.54	6.44	-11.66	-1.81	20.54
4 spd improve over 2sp	OCTA			14%	13%	13%
4 spd improve over 1sp-3	OCTA			10%	13%	13%

		Cycle Dist.	Dyno Dist.	Battery Energy Net		Dies. Eq.
Transmission Config	Cycle	[Miles]	[Miles]	[kWh]	[kWh/Mile]	mpg_de
4 speed - 1st gear launch	WHVC	11.22	11.20	-14.31	-1.28	29.40
2 speed - 1st & 3rd gear	WHVC	11.22	11.22	-14.74	-1.31	28.49
1 speed - 3rd gear	WHVC	11.22	11.19	-14.51	-1.30	29.01
4 spd improve over 2sp	WHVC			3%	3%	3%
4 spd improve over 1sp-3	WHVC			1%	2%	2%

		Cycle Dist.	Dyno Dist.	Battery Energy Net		Dies. Eq.
Transmission Config	Cycle	[Miles]	[Miles]	[kWh]	[kWh/Mile]	mpg_de
4 speed - 1st gear launch	ALT-comp	13.10	13.12	-23.10	-1.76	21.32
2 speed - 1st & 3rd gear	ALT-comp	13.10	13.19	-24.62	-1.87	20.11
1 speed - 3rd gear	ALT-comp	13.10	12.94	-24.38	-1.89	19.93
4 speed - 2nd gear launch	ALT-comp	13.10	13.08	-22.82	-1.75	21.53
4 spd improve over 2 spd	ALT-comp			7%	6%	6%
4 spd improve over 1sp-3	ALT-comp			6%	7%	7%
4 spd improve over 4sp2	ALT-comp			-1%	-1%	-1%

Table 80 lists the efficiency gain results of the drive cycle experiments. The UDDS and Altoona cycles were also tested in four speed transmission mode, starting in first gear and in one-speed, two-speed, and four-speed modes starting in second gear. Because the Altoona test cycle was tested in a shortened version, the composite values were calculated by assembling modal data from the CBD, Commuter, and Arterial sections.

Table 81 compares energy consumption simulation and test results. Simulations underestimated the energy consumed per mile compared to the test results. The HIL test results were in between the simulations and the chassis dyno test results.

Table 81. Energy consumption simulations and test results in kWh/mile for four-speed

Electric Bus with FDR 6.2 and at SLW	Energy consumed with 4-spd (kWh/mile)			
	UDDS	OCC	NYC	ADB
Simulations at Eaton	1.28	1.17	1.26	1.59
HIL tests at ORNL	1.44	1.37	1.52	1.49
Chassis Dyno at NREL	1.43	1.60	2.06	1.76

The efficiency improvements in mpgde between four- and two-speed modes are 9%, 3%, 13%, and 6% in Manhattan, UDDS, OCTA and Altoona (ADB) cycles in chassis dyno tests. These numbers agree very well with the HIL test results presented in Figure 123 and summarized again in Table 82. On the other hand, the simulation results presented earlier in Table 48 and summarized in Table 82 grossly underestimated the benefits of four-speed transmission compared to the real test results. Inaccuracies in simulations occur for several reasons. First, the motor efficiency map in low speed does not have accurate and detailed information needed for accurate calculations. Secondly, the simulations overestimate the regenerative braking energy captured by two-speed mode because information on the motor response rate and the contributions from the foundation braking is lacking. Hence, there is room to improve the vehicle and powertrain models used in the simulations. The HIL and the chassis dyno tests provide important test data to develop high-fidelity vehicle models.

Table 82. Comparison of simulation, HIL and Chasis Dyno test results:
% efficiency gains (% mpgde improvement) with four-speed mode over two-speed mode for BE35 electric bus with FDR 6.2.and at Seated Load Weight (SLW).

Electric Bus with FDR 6.2 and at SLW	% Efficiency Gain with 4-spd over 2-spd (% of mpgde)			
	UDDS	OCC	NYC	ADB
Simulations at Eaton	-0.9	7.2	9.4	3.3
HIL tests at ORNL	3.5	13.5	15.4	5.6
Chassis Dyno at NREL	3	13	9	6

Subtask 14.4 - Vehicle Gradeability Testing on Chassis Dyno

Gradeability experiments assessed powertrain performance on a simulated grade. Prior to grade testing, the vehicle and dynamometer were conditioned by driving over a UDDS cycle. The vehicle was tested for maximum speed at 10% and 20% grade and for maximum grade at 15 mph and 20 mph. The original plan was to test max grade at 10 mph instead of 15 mph, but the 10-mph point was not possible due to a torque limitation of the chasses dynamometer.

Additional gradeability tests determined the time required for the EV motor stator to overheat, which results in a power de-rate at 10 mph and 20% grade and at 20 mph and 10% grade.

All gradeability experiments were performed at the GVW of 37,530 lb. Since the vehicle coast downs were not actually performed in the field at this test weight, the road load had to be corrected for the higher weight. The theoretical road load A coefficient is a constant that is a product of the vehicle weight and a tire rolling resistance coefficient. To make the test weight correction for the road load force, the A coefficient was increased proportionally to the increase of the test weight (A2 = A1 * TW2 / TW1) The resulting corrected road load force coefficients for the GVW TW were A=330.78lb, B=0.7176lb/mph, C=0.0872lb/mph^2

Due to extreme grades, the transmission could be operated only in first gear below 17 mph and in second gear above that. Table 83 lists the results of testing maximum attainable speed at 10% and 20 % grade. Table 84 lists the results of testing maximum grade capacity of the bus at 15 and 20 mph. Note that original plan to test for maximum speed at 10 mph was not possible due to dynamometer limitations. Table 85 lists the results of testing how long it takes for the traction motor windings to overheat and cause a de-rate condition at which the vehicle is no longer able to sustain the given grade and speed.

Table 83. Results of testing maximum speed at 10% and 20% grade

Max speed at grade tests			
Grade %	Grade force lb	Max speed mph	Gear #
10	3734	21.9	2
20	7360	11.7	1

Table 84. Results of testing maximum grade at 15 mph and 20 mph

Max grade at speed tests					
	Speed mph	gear #	max grade Force lb	max grade degree	max grade %
test1	20	2	4140	0.11	11.10
test2	20	2	4299	0.11	11.53
test1	15	1	6293	0.17	17.01
test2	15	1	6360	0.17	17.20

Table 85. Time to de-rate at 10% and 20% grade

Time to de-rate due to winding overheat		
Grade %	Speed mph	Time from start sec
10	20	153
20	10	128

Subtask 14.5 - Vehicle Acceleration Testing on Chassis Dyno

Acceleration experiments assessed how transmission shifting affects the vehicle's rate of acceleration. Prior to performing acceleration tests, the vehicle and the dynamometer were conditioned by running a UDDS drive cycle. Acceleration experiments were carried out at the vehicle test weights of 33,500 lb (SLW) and 37,530 lb. (GVWR). The vehicle was operated at 100% accelerator pedal position to accelerate the vehicle from 0 to 50 mph as quickly as possible. After each acceleration run, the vehicle was coasted back down to 0 mph to allow for battery power regeneration. The driver waited for two minutes before running the next acceleration event to ensure all the runs started at the same temperature conditions. The transmission configurations alternated for every run in a random fashion. This process was repeated multiple times to ensure statistical confidence in the data results. The results of the acceleration test are summarized in Table 86 for the 33500 lb test weight and in Table 87 for the 37530 lb test weight.

Table 86. Acceleration times (seconds) at 33500 lb test weight (SLW)

	0-50 mph	0-30 mph	30-50 mph
4 spd - 1st gear launch	32.5s	13.7s	18.9s
1 spd - 3rd gear	35.9s	18.6s	17.4s
2 spd - 1st & 3rd gear	33.3s	15.9s	17.4s
4spd1 improve over 2spd	10%	36%	-8%
4spd1 improve over 1spd3	2%	16%	-8%

Table 87. Acceleration times (seconds) at 37530 lb test weight (GVW)

	0-50 mph	0-30 mph	30-50 mph
4 spd - 1st gear launch	35.7s	14.8s	20.9s
1 spd - 3rd gear	39.9s	20.7s	19.2s
2 spd - 1st & 3rd gear	36.4s	17.2s	19.2s
4 spd - 2nd gear launch	35.6s	15.0s	20.6s
4spd1 improve over 2spd	12%	40%	-8%
4spd1 improve over 1spd3	2%	16%	-8%
4spd1 improve over 4spd2	0%	1%	-1%

Chassis dyno tests of four-speed mode showed 36% and 40% faster acceleration in 0-30-mile range with GVW of 37530 lb. compared to the two-speed mode. The gradeability and acceleration performance results of simulations and tests are compared in Table 88. The simulations overestimated the acceleration and gradeability compared to the HIL tests and chassis dyno tests. HIL test results and chassis dyno test results were close enough for all practical purposes.

Table 88. Comparison of simulation, HIL, and Chasis Dyno acceleration and gradeability. Test results of Proterra BE35 electric bus with four-speed transmission, FDR 6.2, and SLW.

Electric Bus with four-speed, FDR 6.2 and at SLW	Acceleration Time (s) @ SLW		Gradeability (%) @ GVW	
	0-30 mph	30-50 mph	10 mph	20 mph
Simulations at Eaton	9.7	16.6	19.5	9
HIL tests at ORNL	12.2	17.9	22.7	11.2
Chassis Dyno at NREL	13.7	18.9	N/A	11.1

3.0 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., Deng, Y., Naghshtabrizi, P, Genise, T., "Modular Multi-Speed Transmission for MD EV," CTI Symposium China, Automotive Transmissions, HEV and EV Drives, 5th International Congress and Expo, 21-23 September 2016, Shanghai, China.
5. One display unit of the new four-speed EV Transmission was displayed at the IAA Commercial Vehicles Trade Fair in Hannover on September 22-29, 2016.
6. Chavdar, B., FY 2016 Vehicle Systems Annual Progress Report (FY2016 VS APR), October 2016.
7. Chavdar, B., Genise, T., Naghshtabrizi, P., Papp, G., "Development of Robust and Modular Drive System for MD-EV," 11th International CTI Symposium, Automotive Transmission, HEV and EV Drives, 15-18 May 2017, Novi, MI, USA.
8. Chavdar, B., FY 2016 Vehicle Systems Annual Progress Report (FY2017 VS APR), October 2017

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