



## Testing and Benchmarking a 2014 GM Silverado 6L80 Six Speed Automatic Transmission

2017-01-5020

Published 11/17/2017

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**CITATION:** Stuhldreher, M., Kim, Y., Kargul, J., Moskalik, A. et al., "Testing and Benchmarking a 2014 GM Silverado 6L80 Six Speed Automatic Transmission," SAE Technical Paper 2017-01-5020, 2017, doi:10.4271/2017-01-5020.

### Abstract

As part of its midterm evaluation of the 2022-2025 light-duty greenhouse gas (GHG) standards, the Environmental Protection Agency (EPA) has been acquiring fuel efficiency data from testing of recent engines and vehicles. The benchmarking data are used as inputs to EPA's Advanced Light Duty Powertrain and Hybrid Analysis (ALPHA) vehicle simulation model created to estimate GHG emissions from light-duty vehicles.

For complete powertrain modeling, ALPHA needs both detailed engine fuel consumption maps and transmission efficiency maps. EPA's National Vehicle and Fuels Emissions Laboratory has previously relied on contractors to provide full characterization of transmission efficiency maps. To add to its benchmarking resources, EPA developed a streamlined more cost-effective in-house method of transmission testing, capable of gathering a dataset sufficient to broadly characterize transmissions within ALPHA. This technique targets the range of transmission operation observed during vehicle testing over the EPA city and highway drive cycles.

This paper describes the method and test results of the benchmarking process used to gather transmission data. With this method, the transmission is tested as a complete system, as opposed to disassembling the transmission components and testing each separately. To develop this test method, a 6-cylinder EcoTec 4.3L engine with a 6L80 six speed automatic transmission from a 2014 Chevrolet Silverado was installed in an engine dyno test cell. The transmission dataset includes gear efficiencies, torque converter slippage and K factors, oil temperature and pressure, and OBD/epid CAN bus data.

The transmission data collected using this benchmarking method were supplied as inputs to the EPA ALPHA model including transmission gear efficiency, torque converter K factors, and spin losses. The ALPHA results were then validated to vehicle chassis dynamometer test data.

### Introduction

During the development of the light-duty GHG standards for the years 2017-2025 [1], EPA utilized a 2011 light-duty vehicle simulation study from the global engineering consulting firm, Ricardo, Inc. This study provided a round of full-scale vehicle simulations to predict the effectiveness of future advanced technologies.

The LD GHG regulation required that a comprehensive advanced technology review, known as the midterm evaluation, be performed to assess any potential changes to the cost and the effectiveness of advanced technologies available to manufacturers for MYs 2022-2025. For the midterm evaluation, EPA is using its full vehicle simulation model, called the Advanced Light-duty Powertrain and Hybrid Analysis Tool (ALPHA) [2], to supplement and expand upon the previous study used during the Federal rulemaking. ALPHA will be used to confirm and, if necessary, update efficiency data from the previous study, to include the latest efficiencies of advanced downsized turbo and naturally aspirated engines. ALPHA will also be used to simulate and investigate effectiveness contributions from advanced technologies not considered during the original Federal rulemaking, such as continuously variable transmissions (CVTs) and Atkinson-cycle naturally aspirated engines.

In the past, EPA has relied on contractors to provide full benchmark testing and characterization of transmissions to build comprehensive efficiency maps which were used in its midterm evaluation of the 2022-2025 light-duty GHG standards. Although these transmission efficiency and operational maps used for the midterm evaluation were detailed and highly accurate, the mapping process is relatively costly and can be time consuming depending on the availability of others' test capacity.

For these reasons, EPA decided to explore a more streamlined alternative transmission benchmarking process for use in-house at EPA's laboratory that could provide comparable quality transmission data suitable for use within ALPHA. To simulate drive cycle performance, the ALPHA model requires various vehicle parameters as inputs, including vehicle inertia and road loads, component efficiencies, and vehicle operation data. The benchmarking study used an engine dynamometer test to measure both the efficiency of the vehicle's engine and its automatic transmission for input to the ALPHA model.

This paper provides an overview of EPA's complete benchmarking work on the 2014 Chevrolet Silverado including both engine dynamometer and vehicle chassis testing to characterize the engine and transmission operation over EPA city and highway test cycles.

A "tethered" engine and streamlined transmission benchmarking method was used to test a Chevrolet 4.3L LV3® engine and 6L80 six speed transmission from a 2014 Chevrolet Silverado. The engine and transmission were mounted in an engine dyno test cell, tethered with a lengthened engine wiring harness to a complete Silverado vehicle outside the test cell. This benchmarking process allowed both the engine and transmission mapping to be conducted at EPA's laboratory using the stock ECU and TCU, and their calibrations. Additional information on the benchmarking for this vehicle can be found in an associated SAE paper covering the benchmarking of the Chevrolet 4.3L LV3® engine [4]. Data from the benchmarking was subsequently configured as engine and transmission inputs for ALPHA.

To evaluate the suitability of EPA's streamlined transmission benchmarking process, results from the vehicle chassis tests were used together with the transmission data from the new streamlined benchmarking method to process ALPHA validation simulations. As part of this evaluation, a validation of the Silverado was performed in the ALPHA model. While complete results of the vehicle validation are outside the scope of this paper, the validation results are provided to inform the discussion. Helpful additional details on ALPHA simulations can also be found in previously published SAE papers and technical documents on the topic [5, 6, 8, 10].

## Description of Test Article and Setup

The 2014 Chevrolet Silverado engine and transmission used in this project were a 4.3L EcoTec3 V6 and a 6L80 six speed automatic transmission. [Table 1](#) summarizes information that describes the vehicle and engine used in this test program.

Table 1. Summary of Vehicle and Engine Identification Information

<b>Vehicle (MY, Make, Model)</b>	2014 Chevrolet Silverado
<b>Vehicle Identification Number</b>	1GCNCPEH2EZ171727
<b>Engine (displacement, name)</b>	4.3L EcoTec3 V6
<b>Rated Power</b>	285 HP (213 kW) @ 5300 RPM
<b>Rated Torque</b>	305 lb.-ft (413 Nm) @ 3900 RPM
<b>Recommended Fuel</b>	Regular unleaded or E85
<b>Transmission</b>	6L80 six speed automatic transmission
<b>Transmission Gear Ratios</b>	1 <sup>st</sup> : 4.027 2 <sup>nd</sup> : 2.364 3 <sup>rd</sup> : 1.532 4 <sup>th</sup> : 1.152 5 <sup>th</sup> : .852 6 <sup>th</sup> : .667

## Test Site

Testing was performed in a light duty engine dynamometer test cell located at the National Vehicle Fuels and Emissions Laboratory (NVFEL) in Ann Arbor, Michigan. The test cell equipment and instrumentation is listed in [Table 2](#).

Table 2. Test Cell Equipment and Instrumentation

<b>Instrument Name</b>	<b>Purpose/Measurement Capabilities</b>	<b>Manufacturer</b>
Dynamometer (AC)	Engine speed, torque, power	Meidensha
Torque Sensors	In line shaft torque	HBM

## Tethered Wire Harness

In modern vehicles the engine control unit (ECU) is no longer the main computer. The ECU now requires communication with the body control module (BCM), the transmission control unit (TCU) and other various modules to monitor the entire vehicle operation (security, entry, key on, dash board signals, etc.). Because the ECU needs signals from these other modules to operate as calibrated by the manufacturer, the signals need to be extended to the test cell. The wiring harnesses connecting the ECU to the rest of the vehicle were lengthened to allow the engine and transmission in the dynamometer cell to be tethered to the vehicle chassis located outside the cell. [Figure 1](#) illustrates the tethered wiring harness. Signal wires from the ECU to the engine and transmission were tapped to allow the signals to be either monitored or replaced as needed. This ensured testing could be performed without setting ECU/TCU fault codes and in a manner consistent with expected transmission operation in the vehicle.

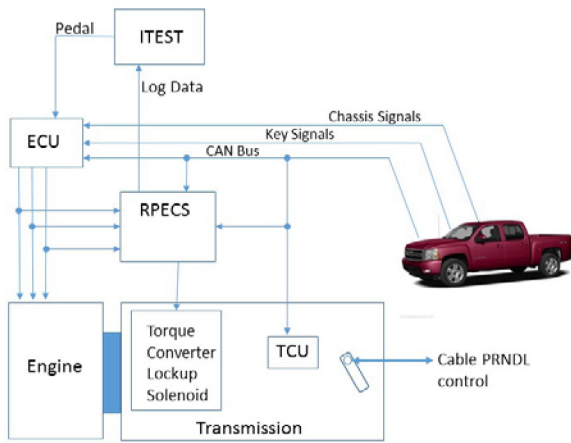


Figure 1. Vehicle and Engine Tethered Wire Harness

### Data Collection and Controls Systems

Test cell data acquisition and dynamometer control were performed by iTest, a software package developed by A&D Technology, Inc. Test cell data including temperatures, pressures, speed and torque are logged by iTest. Engine and transmission ECU inputs and outputs were measured using RPECS, a hardware/software package for engine control and supplemental data acquisition developed by Southwest Research Institute (SwRI). RPECS data is logged by iTest via an Ethernet connection and combined into a single output file. The transmission control and analysis software packages are summarized below in Table 3.

Table 3. Engine Control and Analysis Software

Software	Developer	Description	Data Rate
iTest	A&D Technology, Inc.	<ul style="list-style-type: none"> <li>Controls dyno</li> <li>Collects test cell data</li> <li>Master data logger</li> <li>Commands pedal</li> </ul>	10 Hz
RPECS	Southwest Research Institute	<ul style="list-style-type: none"> <li>Collects TCU CAN and analog transmission data</li> <li>Controls torque converter lock up solenoid</li> </ul>	1/engine cycle

### Engine Systems

A production Chevrolet Silverado 4.3L engine was used to support this transmission testing. Specific details for the engine setup and testing are described in the associated SAE Technical Paper. [4] To control engine torque for this testing, the chassis throttle pedal inputs were utilized. The production vehicle throttle pedal signals were duplicated and controlled by the iTest dyno controls.

### Transmission Setup

To incorporate the transmission into the test cell, a system was designed to separate and instrument both engine and transmission output speed and torque. Figure 2 shows a schematic for the engine and transmission setup in the test cell. A SOLIDWORKS® model of the complete assembly, including the inline torque sensor (Figure 3) was developed to assist in determining mounting mechanisms and locations.

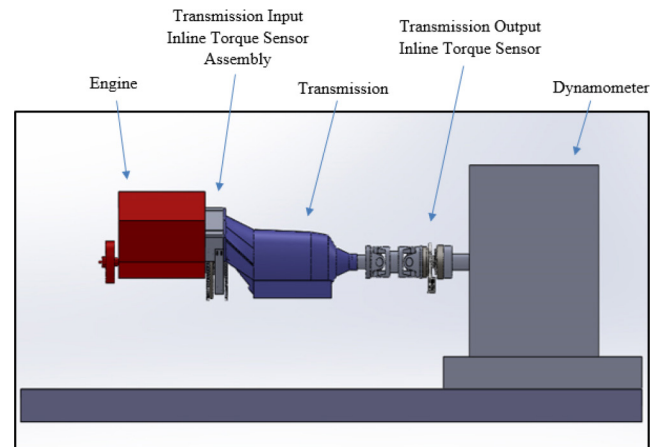


Figure 2. Test Cell Model with Engine and Transmission

The Transmission Input Inline Torque Sensor is illustrated with an exploded view in Figure 3. This assembly was designed to maintain the concentricity and axial spacing of the transmission torque converter and engine flywheel. Figure 4 shows the final engine and transmission assembly after having been installed into the test cell.

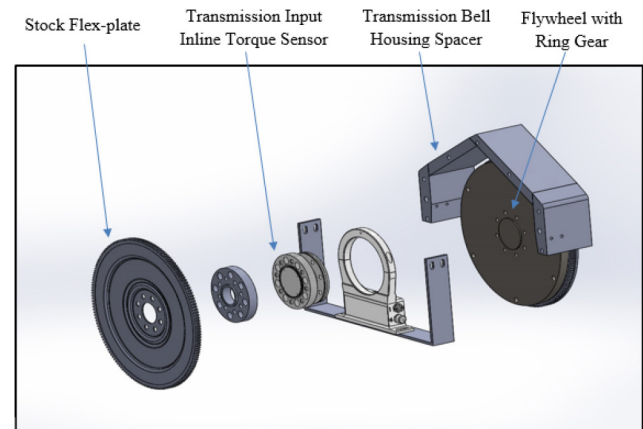


Figure 3. Transmission Inline Torque Sensor and Spacers Assembly



Figure 4. Final Engine and Transmission Test Cell Installation

### Transmission System and Control

Some key aspects of this streamlined transmission testing process are shifting/gear selection, fluid temperature control, engine start, and integration of an inline torque sensor. To properly control the transmission and record the appropriate data, a series of modifications and procedural steps were required.

## 1. PRNDL Shift Controls

The transmission shifting is controlled by the PRNDL shift lever, normally mounted on the vehicle steering column. The shift lever moves a shift cable which connects to the transmission PRNDL selector mounted internally in the transmission. The steering column shift lever also has a button to control the electronic gear selection when the PRNDL is in manual mode. For the test cell setup, a PRNDL lever was mounted in the test cell and tethered to the iTest console. The PRNDL manual mode was used to hold the transmission in a specific gear and the shift lever +/- button was used to control the actual gear selection.

## 2. Transmission Fluid Cooling

The transmission fluid was cooled by circulating the fluid out of the transmission to a radiator cooler and then back into the transmission. For the test cell setup, and to best emulate a production cooling system, the transmission fluid was cooled by a plate-style heat exchanger which replaced the production vehicle radiator cooler. The external cooling loop of the heat exchanger was connected to the engine coolant heater core circulation loop using production transmission cooler lines and routing. The transmission thermostat was not altered or modified for the test cell setup.

## 3. Input Inline Torque Sensor, Bell Housing, and Flywheel

An inline torque sensor was used to measure the transmission input torque. Adapters were fabricated to connect the inline torque sensor to the engine flywheel and transmission flex-plate as shown in [Figure 3](#). To accommodate the torque cell, a spacer was fabricated to separate the engine and transmission, and an aftermarket flywheel mounted to the engine. The flywheel ring gear was mounted in a position similar to the production transmission configuration to allow for starting with the stock starter motor.

## 4. Torque Converter Clutch Lockup

The torque converter clutch is normally controlled by the transmission control unit (TCU). This testing required the torque converter clutch to be controlled directly. This was done by tapping into the wires connecting the clutch solenoid and the TCU. The signal coming out of the TCU was read by RPECS and a new signal was passed to the clutch solenoid that would allow either a locked or an unlocked clutch position as desired. This enabled the transmission gear efficiency to be measured with the torque converter locked and the torque converter K Factors to be measured with the torque converter unlocked.

## Transmission Testing

The in-house EPA transmission benchmarking procedure described in this paper included measuring the transmission's total efficiency in each gear and torque converter K-factors. The intent was to run all component and chassis transmission tests with transmission oil temperature at or above a target of 90°C. Performing all the testing at high temperature simplifies the analysis and eliminates the need to map operation of the transmission with multiple temperature sweeps.

However, due to practical limitations in this initial pilot program, the operating target of 90°C or above was not always achieved for either the chassis testing or the transmission benchmarking. The actual measured transmission temperature data were observed to be from approximately 70°C to 105°C. Therefore, during the validation process, a torque loss adjustment method was developed to normalize the measured transmission loss data to represent values as though they were measured at a consistent 100°C. The paper's section entitled "*Building Temperature Compensation into ALPHA's Spin Loss Look-up Table*" describes this temperature related adjustment.

Previously, ALPHA assumed that all transmission losses were measured from a fully warmed up transmission and did not need a temperature sensitive torque loss adjustment method for vehicle simulations. After the temperature profile data was added to the transmission's efficiency map, validation simulations in ALPHA can now adjust transmission loss data to account for actual oil temperatures encountered during the vehicle chassis tests. This adjustment process is described in the section entitled "*Validation of Transmission Data using ALPHA*."

*Note: It should be noted that the following charts in [Figures 5, 6, 7, 8, 9, 10](#) show measured data as collected, and represent transmission efficiency before any temperature dependent spin loss adjustment was applied.*

## Transmission Efficiency Testing

The transmission gearbox efficiency test was done by holding the transmission in a selected gear and locking up the torque converter. The transmission input speed and load were controlled to a fixed value and held until stable. Each gear was tested in steady state mode over a range of transmission output speeds and loads. For each speed and load combination, the data were logged at a 10 Hz sampling frequency for 10 seconds, then averaged to create a single average data point. Transmission efficiency, reported in percentage, was calculated according to [Equation \(1\)](#) using the values obtained from iTest.

$$\text{Transmission efficiency} = \frac{\text{Torque Out} \cdot \text{Speed Out}}{\text{Torque In} \cdot \text{Speed In}} * 100 \quad (1)$$

The measured test points for the transmission efficiency test covered the torque and speed range shown in [Figure 5](#). Data were measured in each gear for specific speed values over increasing input torque and have no adjustments for transmission temperature. The dataset logged during all modes of transmission testing included parameters such as transmission speed in and out, torque in and out, oil temperature and pressure, gear selection, and additional epid CAN data.

The transmission input torque in each gear was limited by the maximum transmission output torque that could be absorbed by the dynamometer (500 Nm). As a result, the number of data points able to be measured for first and second gear was limited. The input torque at or near the engine WOT line was captured for gears 3-6, which provided adequate coverage for the transmission operation.



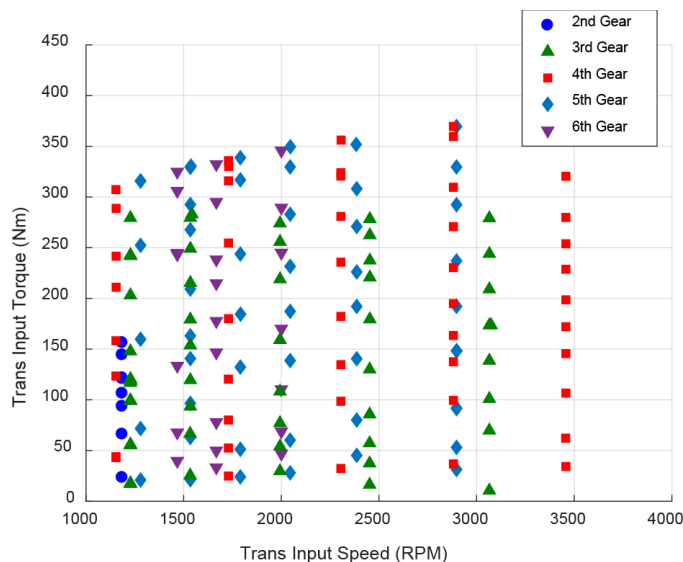


Figure 5. Measured Test Data Points

The transmission efficiency, reported in percentage, for each individual gear as measured at various speeds is shown in [Figures 6, 7, 8, 9, 10](#).

[Figure 6](#) shows the transmission efficiency with the gear selection held in second gear. The legend box (noting “locked up” or “unlocked”) refers to the torque converter clutch state. The data point labels are the transmission efficiency values.

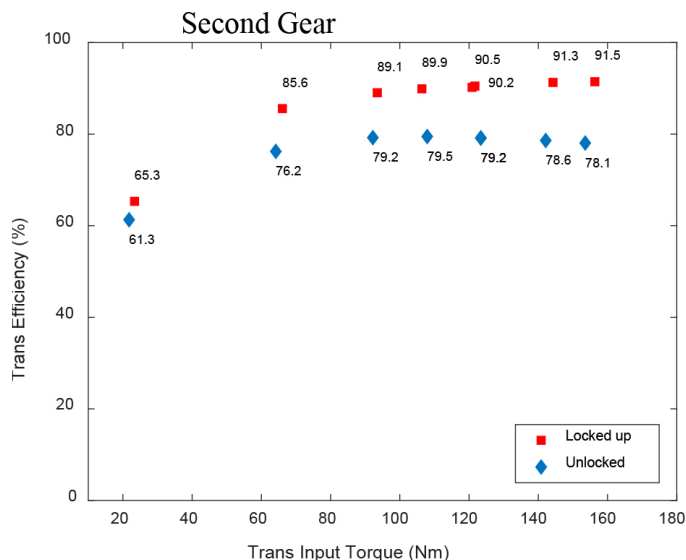


Figure 6. 2014 Chevrolet 6L80 Transmission Second Gear Measured Efficiency

[Figures 7, 8, 9, 10](#) show the transmission efficiency with the gear selection held in the listed gear. The legend box (rpm) refers to the transmission input speed. The data point labels are the transmission efficiency values for the specified data points.

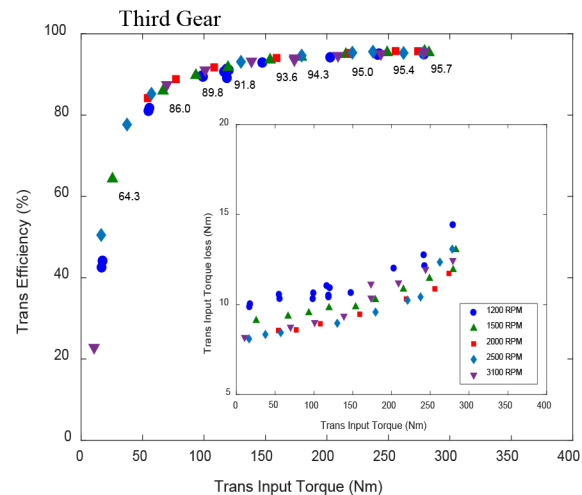


Figure 7. 2014 Chevrolet 6L80 Transmission Third Gear Measured Efficiency and Torque Loss (insert). Labeled efficiencies are for 1500 rpm points.

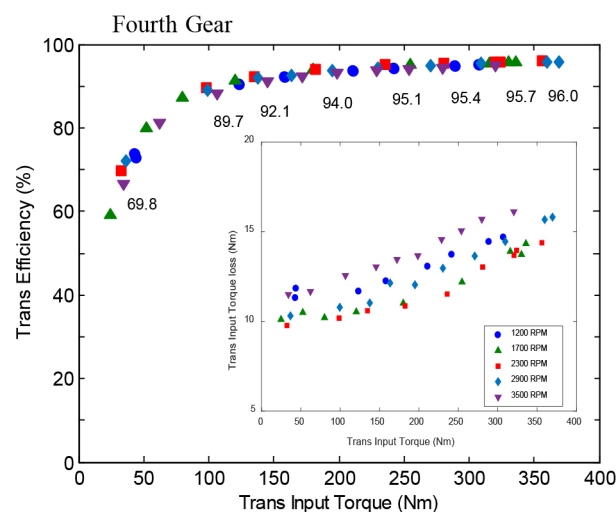


Figure 8. 2014 Chevrolet 6L80 Transmission Fourth Gear Measured Efficiency and Torque Loss (insert). Labeled efficiencies are for 2300 rpm points.

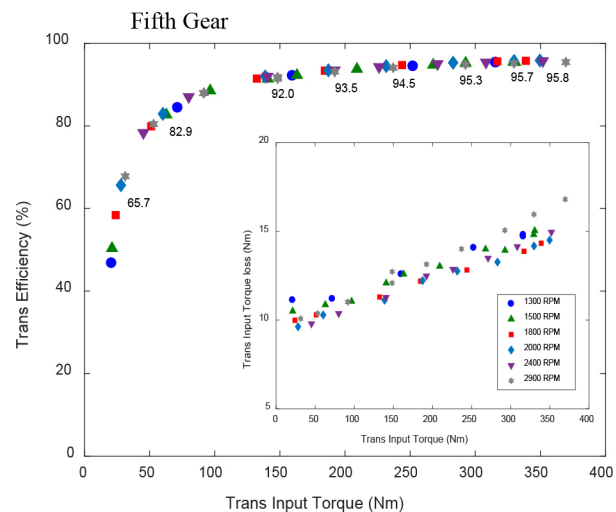


Figure 9. 2014 Chevrolet 6L80 Transmission Fifth Gear Measured Efficiency and Torque Loss (insert). Labeled efficiencies are for 2000 rpm points.

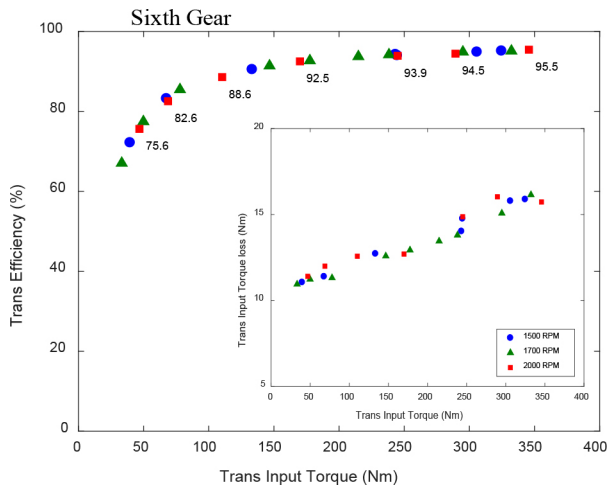


Figure 10. 2014 Chevrolet 6L80 Transmission Sixth Gear Measured Efficiency and Torque Loss (insert). Labeled efficiencies are for 2000 rpm points.

### K-Factor Testing

The torque converter K-Factor test was conducted by holding the transmission in a selected gear (fifth), with the torque converter unlocked and the dyno speed held constant, and sweeping input speed by ramping engine pedal request. The data were logged at a 10 Hz sampling rate until the sweep was complete. The transmission output speed was varied between 1500 and 3000 rpm in selected increments. The transmission input torque (engine torque) was swept from 0 to 250 Nm over 50 seconds. The K-Factor for the torque converter was calculated using Equation (2) using the test results obtained from iTest. The results of the K Factor calculation are shown in Figure 11.

$$K \text{ Factor} = \frac{\text{Input Speed}}{\sqrt{\text{Torque In}}}$$

(2)

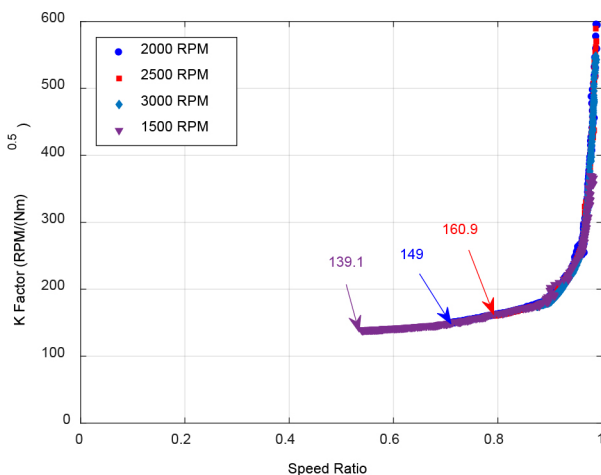


Figure 11. 2014 Chevrolet 6L80 Transmission Torque Converter K-Factor Results

### Drive Cycle Coverage of Streamlined Test Data

The streamlined transmission efficiency testing was undertaken to develop transmission maps that were adequate for simulating fuel consumption over standard EPA drive cycles. While the streamlined transmission test method and its associated efficiency results in this

paper do not cover the entire possible operational range of the transmission, the data does cover the core operating range used during standard EPA cycles.

Measured efficiency test points for each gear (previously shown in Figure 5) reveal that the test process covered different speed and load ranges in each gear, depending on the dynamometer and engine limitations. Figure 12 shows the amount of fuel consumed in each gear during the UDDS, HWFET, and US06 cycles, as recorded during Silverado vehicle chassis testing. Over the HWFET and US06, the majority of fuel was consumed in sixth gear. Over the UDDS, the majority of fuel was consumed primarily in third and fourth gears.

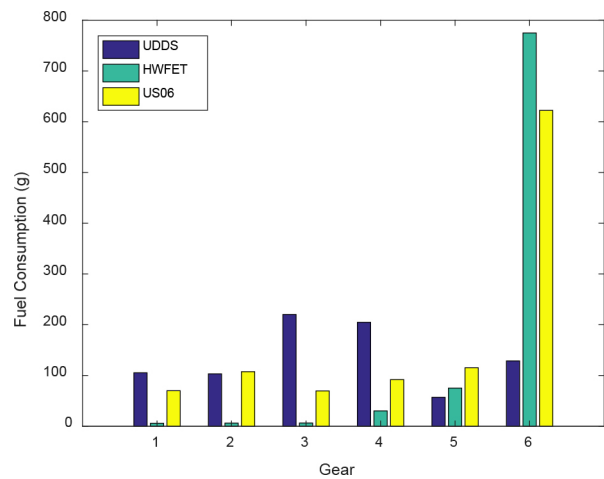


Figure 12. Fuel consumed in each gear for the Silverado over three standard cycles, recorded as grams at 10 Hz from chassis dynamometer testing.

Figures 13 and 14 show the engine operational area for third and sixth gears as recorded during the chassis dynamometer testing of the Silverado. The shaded areas on the engine BSFC maps in the two figures represent the energy weighted engine operation. The areas in red and yellow represent areas of primary engine operation, the areas in blue indicate areas of lower engine operation, and the areas in white represent no engine operation. The test points recorded during the streamlined transmission efficiency testing are represented by the pink circles (previously shown in Figure 5). In both figures, the upper heat map shows engine operation over the UDDS and HWFET cycles, and the lower heat map covers the US06 test cycles.

Figure 13 indicates the operational range of the transmission over the UDDS, HWFET and US06 test cycles for third gear is sufficiently represented by the data points measured in the test cell. Although some of the energy weighted heat map points of the US06 are slightly higher in torque, the operational time in third gear over the US06 is relatively short. Fourth gear results are substantially similar, except the transmission efficiency map reaches higher torque.

Likewise, Figure 14 shows the operational range of the transmission over the UDDS, HWFET and US06 test cycles for sixth gear. In this example, the operational torque range of the transmission is entirely covered by the efficiency test data points and the map reaches the WOT line. However, the operational speed range, particularly for the UDDS/HWFET cycles, extends to slightly lower speeds than the efficiency testing covered, and thus requires some moderate extrapolation.

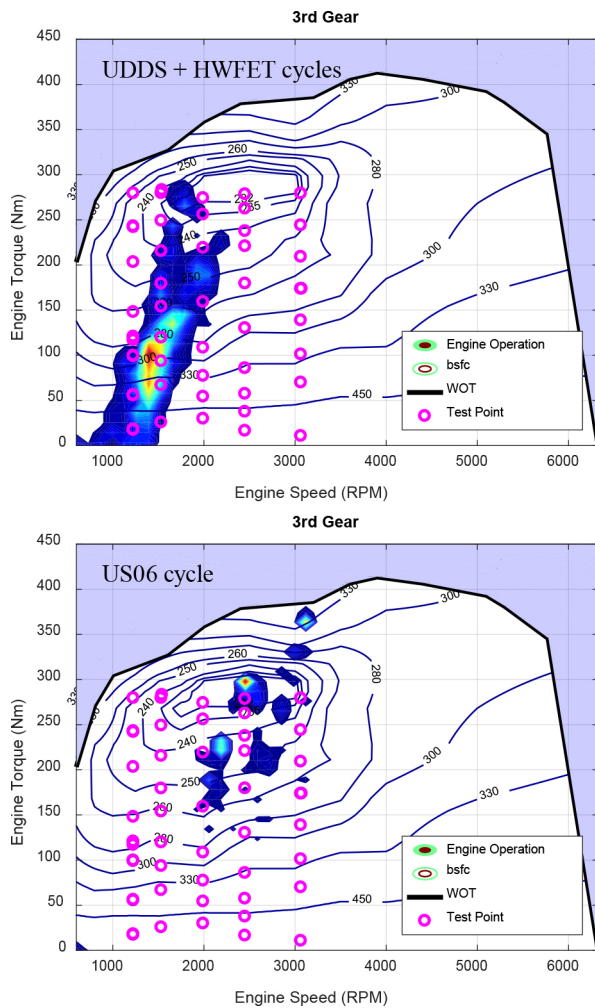


Figure 13. Energy weighted heat maps showing time spent in third gear, taken from data recorded during chassis dyno testing. (a) The top map covers the UDDS + HWFET cycles, and (b) The bottom map covers the US06 cycle. The pink circles represent actual data points used to construct the transmission efficiency map.

Figures 13 and 14 show that for third and sixth gears the data samples recorded during the streamlined transmission testing are reasonably representative of the transmission's operational areas during the cycles of interest. All of the available measured data from gears 2 to 6 were used to construct a full ALPHA transmission efficiency map as discussed in the next section.

However, as the top chart in Figure 14 illustrates, there are some cases where not all of engine/transmission operations are inside the region covered by test points. Such coverage gaps are relatively limited and actual operation is very close to the area covered, so minimal extrapolation is required to construct the full transmission efficiency map.

Table 4 provides a sense of how well the data available from the streamlined transmission testing cover a full transmission efficiency map required for modeling for the specified gears over different driving cycles. The coverage percentage is computed by counting the number of operating points inside a convex hull constructed from test points and integrating the fuel usage over these points.

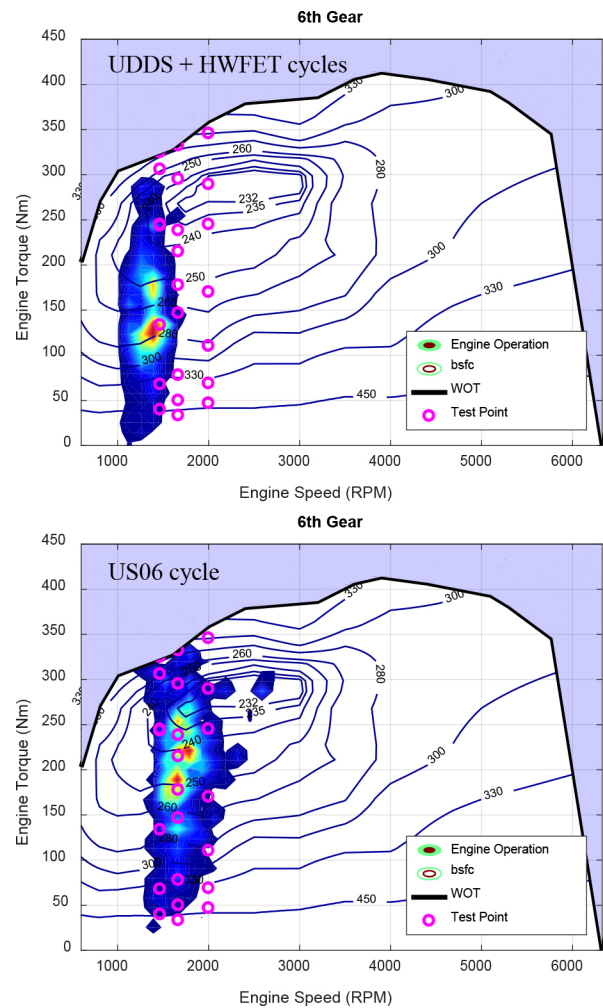


Figure 14. Energy weighted heat maps showing time spent in sixth gear, taken from data recorded during chassis dyno testing. (a) The top map covers the UDDS + HWFET cycles, and (b) The bottom map covers the US06 cycle. The pink circles represent actual data points used to construct the transmission efficiency map.

Table 4. Coverage of operation for tested points in each transmission gear over the test driven cycles, as a percentage of total fuel used in the designated gear.

Gear	UDDS+ HWFET	US06	Total
1	NA	NA	NA
2	0	0	0
3	77.9	49.1	71.9
4	71.9	86.8	76.0
5	53.6	83.1	67.2
6	13.1	85.7	42.6
All	37.1	82.5	53.7

Below is a detailed discussion of some of the coverage percentages from this pilot study:

- **First Gear:** Coverage data shows "NA" because gear efficiency tests were not conducted for first gear.
- **Second Gear:** Data are zero percent because these efficiencies were measured at only one speed condition (1185 rpm), which was just outside the range of second gear operation on the UDDS, HWFET and US06 cycles.

- **Third Gear:** Coverage data during the US06 cycle is relatively low, as engine operates at higher torques. Torque data for these lower gears was limited by the torque capacity of the engine dynamometer used during this pilot program.
- **Fifth Gear:** Data during the UDDS/HWFET cycles could have been mapped at 1100 RPM, instead of at 1200 RPM, to improve its coverage.
- **Sixth Gear:** Coverage data during the UDDS/HWFET cycles were actually measured at 1200 RPM, but there were some measurement issues with the data that prevented it from being used during the mapping process.

Simply measuring efficiency data at 1100 RPM during future uses of this streamlined transmission benchmarking process for UDDS and HWFET cycles would greatly improve the coverage area for fifth and sixth gears.

## Creating Transmission Data Inputs for ALPHA

To evaluate the suitability of EPA's streamlined transmission benchmarking process, results from the vehicle chassis tests were used together with the transmission data from the new streamlined benchmarking method to process ALPHA validation simulations. As part of this evaluation, a validation of the Silverado was performed in the ALPHA model. While complete ALPHA model calculations are outside the scope of this paper, some calculations and validation results are provided to inform the discussion. Helpful additional details on ALPHA simulations can also be found in previously published SAE papers and technical documents on the topic [5, 6, 8, 10].

To simulate drive cycle performance, the ALPHA model requires various vehicle parameters as inputs, including transmission benchmarking data. The benchmarking study used an engine dynamometer test to measure both the efficiency of the vehicle's engine and its automatic transmission for input to the ALPHA model. See Transmission loss data within ALPHA are typically contained in multi-dimensional look-up tables, which can reflect the influence of speed, torque, gear number and oil temperature, when available. The collected data were processed and analyzed according to [equation \(3\)](#) to develop transmission torque loss maps for the transmission model implemented in ALPHA.

$$\begin{aligned} \tau_{\text{loss\_gbx}}(\tau_{\text{gbx\_in}}, \omega_{\text{gbx\_in}}) = \\ \tau_{\text{gear\_loss}}(\tau_{\text{gbx\_in}}, \omega_{\text{gbx\_in}}) + \tau_{\text{spin\_loss}}(\omega_{\text{gbx\_in}}) \end{aligned} \quad (3)$$

Where:

- $\tau_{\text{loss\_gbx}}$  = total transmission torque loss
- $\tau_{\text{gbx\_in}}$  = transmission gearbox input torque
- $\omega_{\text{gbx\_in}}$  = transmission gearbox input speed
- $\tau_{\text{gear\_loss}}$  = torque consumed by specific gear mesh
- $= (1 - \eta_{\text{gear}}) * \tau_{\text{gbx\_in}}$
- $\tau_{\text{spin\_loss}}$  = torque required to spin the unit with no output load applied (including pump losses)
- $\eta_{\text{gear}}$  = gear efficiency

For compatibility with the structure of the model, the full transmission efficiency maps for each gear were represented as torque losses (third gear is as shown as an example in [Figure 15](#)).

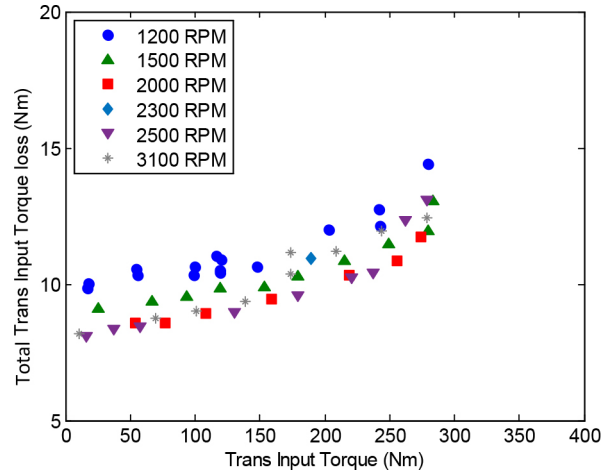


Figure 15. Total Transmission Torque Losses in Third Gear as a Function of Input Speed and Torque for the 2014 Chevrolet 6L80.

The **total transmission losses** in each gear were further decomposed into spin losses and mechanical losses (or gear meshing losses) as described in a technical paper by Seetharaman [7], so that temperature effects could be applied to the spin losses as needed.

**Spin losses** primarily include both churning and oil pump losses; these losses in each gear were estimated by using a regression model with zero torque input as shown in [equation \(7\)](#).

$$\tau_{\text{spin\_loss}} = \tau_{\text{loss\_gbx}}(0, \omega_{\text{gbx\_in}}) - \tau_{\text{gear\_loss}}(0, \omega_{\text{gbx\_in}}) \quad (4)$$

Substituting the formula for  $\tau_{\text{gear\_loss}}$  yields:

$$\tau_{\text{spin\_loss}} = \tau_{\text{loss\_gbx}}(0, \omega_{\text{gbx\_in}}) - [(1 - \eta_{\text{gear}}) * \tau_{\text{gbx\_in}}] \quad (5)$$

Substituting zero for input torque,  $\tau_{\text{gbx\_in}}$ , yields:

$$\tau_{\text{spin\_loss}} = \tau_{\text{loss\_gbx}}(0, \omega_{\text{gbx\_in}}) - [(1 - \eta_{\text{gear}}) * 0] \quad (6)$$

$$\tau_{\text{spin\_loss}} = \tau_{\text{loss\_gbx}}(0, \omega_{\text{gbx\_in}}) \quad (7)$$

**Mechanical losses** for each gear were computed as the total torque loss without spin losses. As an example, [Figure 16](#) shows computed mechanical losses as a function of input speed and torque in third gear. The mechanical losses induced by gear meshing and rolling element bearings were assumed to be independent of operating temperature [7].

To generate a mechanical loss map for the transmission's second gear (where data were available at only one speed, 1185 rpm), the mechanical loss data points were calculated by assuming that mechanical losses as a function of speed would have a similar relationship as the average speed relationships in other gears.



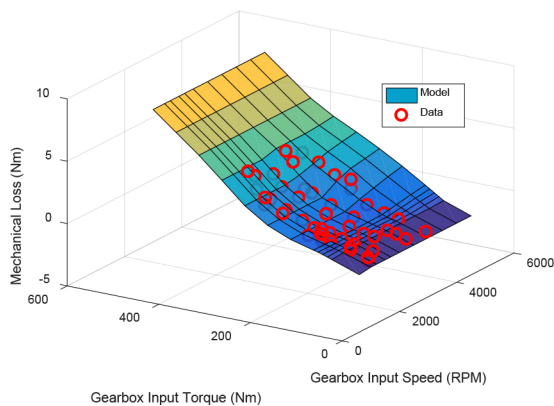


Figure 16. Extrapolated mechanical loss map generated for third gear, using the measured streamlined test data.

Finally, mechanical losses for first gear (where no test data were taken) were assumed to be the same as the losses for second gear. This assumption was made based on analyzing data from previously benchmarked transmissions [8].

### Building Temperature Compensation into ALPHA's Spin Loss Look-up Table

As discussed earlier, transmission oil temperature during this pilot project was not always able to be maintained at or above a target of 90°C, resulting in some efficiency data being measured at lower than the desired fully warmed up operating temperature. The bottom graph in Figure 17 shows the oil temperatures measured during the transmission benchmarking of third gear. The top chart in Figure 17 indicates the transmission's actual measured torque loss values, which are also shown in Figure 15.

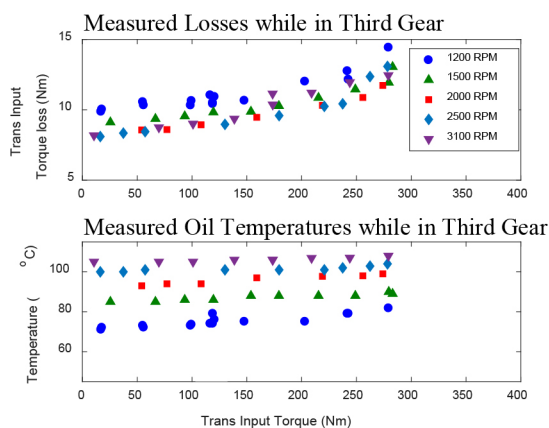


Figure 17. Measured transmission input torque loss (top graph) at measured oil temperature (bottom graph).

For previous work supporting the EPA's midterm evaluation of light-duty greenhouse gas standards [5, 6, 8, 9], ALPHA assumed that basic transmission loss data were obtained from a fully warmed up transmission. For this project, a temperature sensitive torque loss adjustment method was derived to adjust torque loss values to what they would have been if the transmission had been benchmark tested at higher better controlled temperatures, such as 100°C<sup>1</sup>. Using temperature profile data from previously benchmarked

transmissions<sup>2</sup>, the model's resulting torque loss look-up table for this pilot program was designed to adjust torque loss values. The thermal relationship that was derived from these transmissions is shown in Figure 18.

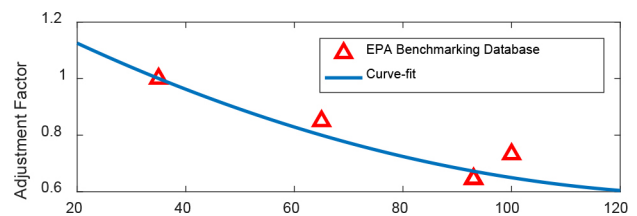


Figure 18. Temperature relationship used to adjust for transmission temperature in computing spin losses.

The data points shown in this chart represent averaged adjustment factors (AF) over all gears and speeds. The adjustment factor curve was approximated by a quadratic function. The curve represents the ratio of actual torque loss proportional to the torque loss at 35°C. Below are two examples that use adjustment factors from this chart to illustrate adjusting torque loss values.

Assuming that a torque loss at 65°C is measured at 14.0 Nm, and that AF@35=1.0, AF@65=.85, and AF@90=.65, then mathematically the torque loss would be adjusted to:

$$\begin{aligned}
 &16.5 \text{ Nm at } 35^\circ\text{C} \\
 &= 14.0 * (\text{AF}@35 / \text{AF}@65) \\
 &= 14.0 * (1.0 / 0.85) \\
 &10.7 \text{ Nm at } 90^\circ\text{C} \\
 &= 14.0 * (\text{AF}@35 / \text{AF}@65) * (\text{AF}@90 / \text{AF}@35) \\
 &= 14.0 * (1.0 / 0.85) * (0.65 / 1.0)
 \end{aligned}$$

Figure 19 shows an example of the results of adjusting the measured transmission torque losses of this benchmarked transmission for third gear as a function of input speed and load at 100°C. The adjustment factor function chart in Figure 18 was applied to the actual measured torque losses (top chart in Figure 17) obtained during benchmarking at the measured transmission oil temperatures indicated in the bottom chart in Figure 17.

The temperature adjusted transmission torque loss data points were then extrapolated to cover the full torque spread required for use in the ALPHA model. Losses for torques above about 300 Nm are linearly extrapolated. It should be noted that, in general, the extrapolation is relatively straightforward and the extrapolated portions of the transmission loss map represent areas where there is relatively low operation over the FTP and HWFET. Since these cycles are the primary focus of EPA's work to estimate GHG from emerging vehicle technologies, the extrapolated areas have little effect on modeling results when the final data are used.

For each of the measured data points, the temperature adjusted torque losses were recombined with mechanical losses to create ALPHA's look-up table for transmission loss as a function of each gear, input speed, input torque, and temperature.

1. As has been done for previous work, it is possible to more tightly control transmission temperature to better mimic a fully warmed up vehicle and transmission, which simplifies the modeling effort.

2. This work assumed that the thermal characteristics of this tested transmission were similar to those of the previously benchmarked GM 6T40 [6] and ZF 845RE [8] transmissions.

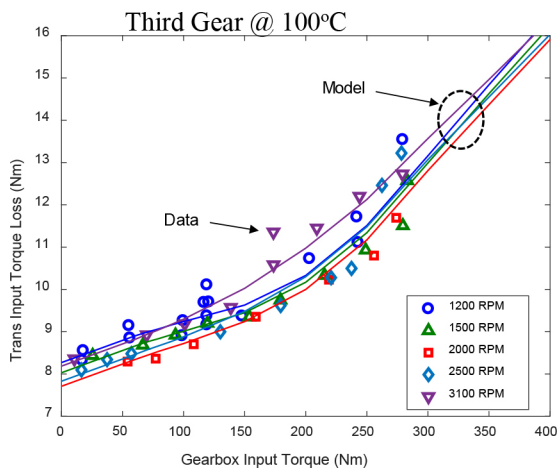


Figure 19. Predicted total transmission input torque loss at 100°C for third gear, along with temperature-adjusted data for 100°C.

As a verification of the temperature adjustment algorithm, 100°C data in the look-up table were used to back calculate the torque loss values at the measured oil temperature. The back-adjusted results were in good agreement with measured losses as seen in Figure 20. The solid points represent the actual data and the hollow points are the model simulation data. Of particular interest is the circled data point, which was an additional warmup data point recorded at 52°C which was added to this verification exercise to demonstrate that the algorithm can even handle adjustments for more extreme temperature measurements. The measured torque loss was reasonably well predicted by the loss maps, which makes it possible to investigate the influence of actual transmission temperature conditions on fuel economy during ALPHA simulations as discussed in the next section.

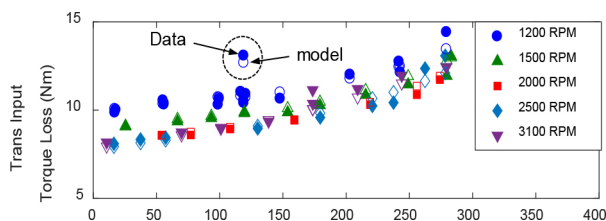


Figure 20. Verification of the temperature adjustment algorithm.

## Validation of Transmission Data Using ALPHA

The transmission loss map developed for ALPHA was validated in the ALPHA model by simulating fuel economy for a 2014 Chevrolet Silverado with a Chevrolet 4.3L LV3® engine and 6L80 six speed transmission over the three driving cycles and comparing to measured chassis dynamometer results for the vehicle.

Figure 21 shows the measured engine coolant and transmission oil temperatures during UDDS, HWFET and US06 tests for the purpose of comparison. The engine was operating fully warmed up during tests, but it was observed that the transmission was not fully warmed up for the UDDS validation.

The UDDS cycle data was constructed from FTP test cycle data. Warm UDDS operation was simulated by using a segment from a warm start FTP bag 3 along with the FTP bag 2 segment from the same test. This piecewise construction of the UDDS results in the discontinuity in measured transmission temperature seen in Figure 21.

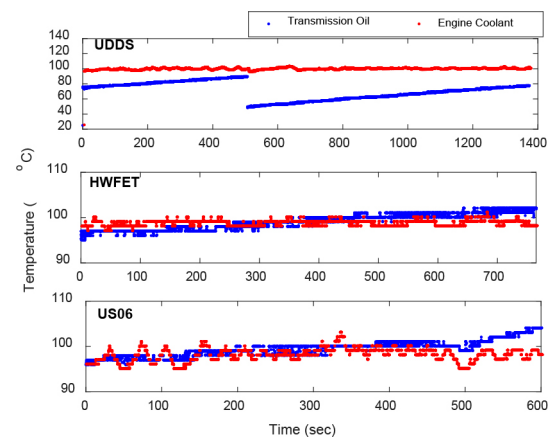


Figure 21. Transmission oil and engine coolant temperatures over three test cycles.

The measured transmission oil temperatures shown in Figure 21 were used as the inputs in ALPHA simulation to determine the temperature adjusted spin loss values matching the cooler actual transmission temperatures observed during the dyno tests. The comparison of the average fuel economy results obtained through these simulations and chassis dynamometer tests are presented in Table 5. The results were within  $\pm 1.5$  percent for all phases of the UDDS, HWFET, and US06.

Table 5. Comparison of fuel economy results in MPG obtained from dynamometer tests and ALPHA simulations

Vehicle	Drive Cycle	Chassis Test (avg mpg)	ALPHA Simulation (avg mpg)	FE Diff (%)
Chevrolet Silverado	UDDS Phase 1	22.30	21.99	-1.4
	UDDS Phase 2	19.66	19.47	-1.0
	UDDS Total	20.85	20.61	-1.1
	HWFET	30.47	30.57	+0.3
	US06 Phase 1	13.24	13.41	+1.2
	US06 Phase 2	21.83	21.93	+0.5
	US06 Total	19.08	19.21	+0.7

Over the UDDS and HWFET cycles (which are core elements of the cycles used for the light-duty vehicle GHG standards), the maximum absolute difference was 1.4 percent. Therefore, it is reasonable to conclude that fuel economy results from ALPHA simulation with the inclusion of data from the streamlined transmission benchmarking are sufficiently accurate for the intended purpose of determining combined CO<sub>2</sub> grams per mile over the EPA city and highway cycles.

An additional sensitivity comparison was calculated by not applying the adjustment for temperatures observed during the dyno tests during the ALPHA simulation runs. The unadjusted percent difference for the “UDDS Total” was only 1.2 percent greater than the corresponding difference in Table 5 which was temperature adjusted according to the data in Figure 20. Both sets of results support our expectation that ALPHA validation results should agree within 3 percent of the chassis test results.

EPA also ran ALPHA simulations using the US06 cycle, a more aggressive cycle with high fuel consumption per mile. The maximum absolute difference between ALPHA simulations and average test

data for the US06 Total cycle was 1.2 percent for Phase 1. Phase 1 of the US06 cycle is a very short cycle with high transient rates, and this level of agreement is remarkably good.

## Summary/Conclusions

EPA's method of benchmarking a transmission coupled to an engine has been demonstrated. This method of adding the transmission to the engine dynamometer setup to include inline torque measurements is straightforward and does not add significantly to the test cell complexity.

This streamlined benchmarking method for a transmission is lower cost and less complex than an independent transmission component benchmarking test. In addition, the tethered methodology ensures that engine and transmission are controlled in tandem according to the manufacturer's calibrations. However, the data set for this method is limited by the dynamometer's ability to absorb torque, especially in the lower gear ranges.

The transmission data collected using this benchmarking were supplied as inputs to the ALPHA model including transmission gear efficiency, torque converter K factors, and temperature adjusted spin losses. Considering the limitations on the available test data from this pilot project, the ALPHA chassis simulation results showed a very good match to chassis test data confirming our confidence in this streamlined transmission benchmarking method.

Even though this test program showed good results, there are three opportunities for improvement of this streamlined transmission benchmarking method that will be considered by EPA for future transmission testing programs.

- a. Additional testing at a wider range of speeds could extend the transmission operation coverage over standard cycles currently addressed in this paper with modest extrapolation.
- b. Additional testing with a broader range, and better controlled, transmission oil temperatures could result in a higher fidelity for the temperature sensitive torque loss adjustment method.
- c. Separating the temperature dependent losses by directly measuring oil pump losses would allow a more accurate accounting of individual contributions of transmission losses, and a better prediction of fuel consumption during engine idling.

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## Acknowledgments

The authors would like to thank Karla Butters, Brian Olson, Raymond Kondel, Mike Murphy and Greg Davis, in the National Center of Advanced Technology at the National Vehicle and Fuel Emissions Laboratory for their assistance and contributions and SwRI for technical support. Youngki Kim was previously with SwRI and currently with the University of Michigan - Dearborn.

## Definitions/Abbreviations

**6L80** - GM 4.6 six-speed longitudinal transmission; part of the 2014 Silverado powertrain

**ALPHA** - Advanced Light Duty Powertrain and Hybrid Analysis

**BCM** - body control module

**CAN** - controlled area network

**CVT** - continuously variable transmission

**ECU** - engine control unit

**EPA** - Environmental Protection Agency

**epid** - CAN message channel

**GHG** - greenhouse gas

**HIL** - hardware in the loop

**HWFET** - Highway fuel economy test, the "highway cycle"

**iTest** - Test cell data acquisition and dynamometer control software

**LD** - Light duty

**LV3®** - GM 4.3L engine; part of the 2014 Silverado powertrain

**OBD** - onboard diagnostics

**PRNDL** - park, reverse, neutral, drive, low gear selector

**RPECS** - rapid prototyping engine control unit

**SwRI** - Southwest Research Institute

**TCU** - transmission control unit

**UDDS** - Urban dynamometer drive cycle, the “city cycle”

**US06** - A high-speed and high-acceleration dynamometer cycle.

**WOT** - Wide open throttle; i.e., maximum torque.

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ISSN 0148-7191

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