Evaluation of the Automatic Transmission Model in HVE Version 7.1

Eric S. Deyerl, P.E., Michael J. Fitch
Dial Engineering

ABSTRACT

The Automatic Transmission Model in HVE Version 7.1 is evaluated via simulation of the motion of a full-scale test vehicle subjected to straight line acceleration runs. Fidelity of the simulated output data to measured vehicle parameters is discussed, as are limitations of the Automatic Transmission Model, which presently lacks the incorporation of drivetrain slip in the form of a clutch or torque converter model.

INTRODUCTION

With the introduction of HVE Version 7 in the summer of 2009, an Automatic Transmission Model (ATM) became available for users of the SIMON physics module. This model allowed SIMON users to designate the transmission of the simulated vehicle as being “automatic”, thereby allowing SIMON to assign the proper gear for the vehicle's transmission at the start of the simulation based on the assigned initial speed of the vehicle. The model also allows the transmission to shift “automatically” based on user-entered parameters regarding transmission shifts points.

A detailed discussion of the functioning of the HVE ATM is provided in [1]. In essence, the ATM seeks to mimic the functioning of an automotive automatic transmission by causing the simulated transmission to change gear ratios depending on the current throttle position and engine speed as designated in the user-entered Transmission Data dialog.

The functioning of the HVE ATM is presently evaluated by capturing powertrain data from a test vehicle subjected to straight-line acceleration runs.

TEST VEHICLE

The test vehicle was the 2009 Mercury Grand Marquis depicted in Figure 1. This vehicle is equipped with an electronic Powertrain Control Module (PCM), depicted in Figure 2, which contains microprocessors which monitor and control various aspects of the vehicle's powertrain, including the engine and transmission.

Figure 1 – 2009 Mercury Grand Marquis test vehicle

Figure 2 – Powertrain Control Module

The PCM in this Ford vehicle also stores data relating to powertrain operation which can be extracted via the Bosch Crash Data Retrieval (CDR) system. In the case of the 2009 Grand Marquis, data stored in the PCM includes 25.4 seconds' worth of vehicle speed, engine speed, accelerator pedal position, engine throttle position, brake switch status and driveline torque, among other values, recorded at intervals of 0.2 seconds. Thus, in this study, the PCM-CDR system was used as a means of data acquisition against which the simulation output could be compared.

The 2009 Mercury Grand Marquis is also contained within the HVE Vehicle Database, Figure 3, allowing a convenient starting location to model the full-scale vehicle.

Figure 3 – Simulated vehicle

The default data for the vehicle in the HVE database were evaluated and modified in the following areas:

WEIGHT AND LONGITUDINAL CG LOCATION - Were adjusted for the presence of two occupants and test equipment.

YAW MOMENTS OF INERTIA AND CG HEIGHT - Were calculated via the methods described in [2] and [3].

TIRES - The default 225/60R16 tires were exchanged for the 225/60R17 tires found on the test vehicle.

ENGINE - The default engine power curve at wide open throttle (WOT) for the 4.6-liter V-8 engine in the test car was found to have the proper peak horsepower (224 hp @ 4800 RPM) and torque (275 lb-ft at 4000 RPM) as compared to data found in [4] and [5].

Per reference [6], the shape of the default power and torque curves in the HVE Vehicle Database at other values are estimated by EDC to approximate the shapes depicted in reference [6].

Power and torque curves for a 2005—2009 Mercury Grand Marquis as tested at WOT on a dynamometer were procured by the authors from source [7]. While the numerical values of the data from these curves cannot be directly used for the HVE vehicle since they are values of power and torque as measured at the drive wheels, rather than at the engine, these measured curves were used to modify the shape of the default HVE power and torque curves, as shown in Figures 4 and 5.

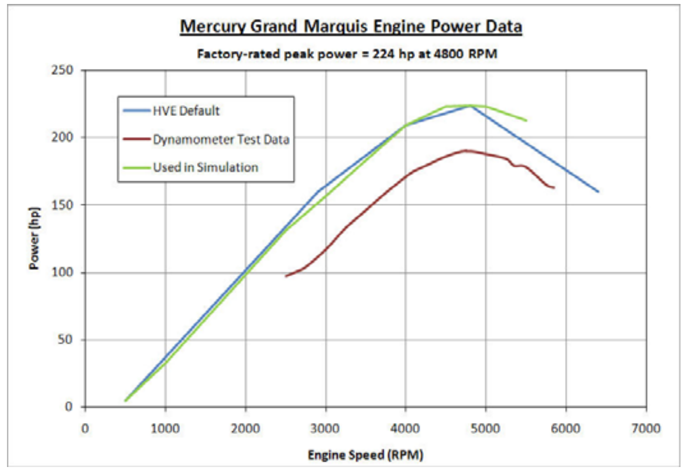

Figure 4 – Mercury engine power curves

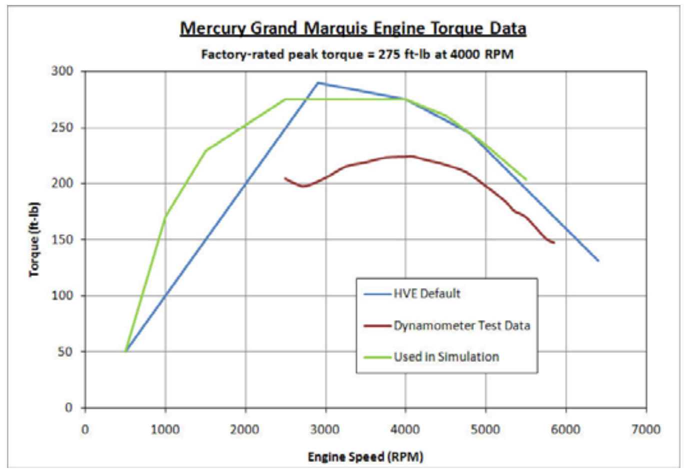

Figure 5 – Mercury engine torque curves

TRANSMISSION AND DIFFERENTIAL - The default transmission gear ratios for the 4-speed automatic transmission in the 2009 Grand Marquis as provided in the HVE database were found to be accurate as compared to data provided in reference [8]. The final drive (differential) gear ratio provided in HVE was also found to be accurate based on reference [8].

The default upshift curve provided in HVE for the Grand Marquis as depicted in Figure 6 was modified based on data collected from the test vehicle, as discussed below. The default downshift curve was not changed.

The Vehicle Data report for the vehicle used to simulate Test Run 03 is provided in the Appendix.

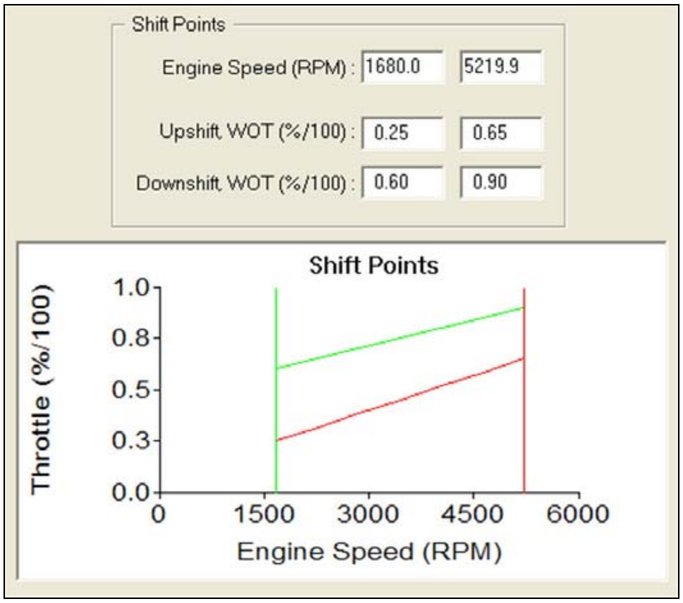

Figure 6 – Default shift points for Mercury Grand Marquis in HVE database

TEST METHODOLOGY

The test vehicle was taken to a straight and level road and subjected to a straight line acceleration maneuver, followed by braking to bring the vehicle to a stop. Because the amount of data stored in the vehicle's PCM is limited to just over 25 seconds and because this data is continuously overwritten while the vehicle's ignition key is in the "ON" position, the test runs were limited to the data which could be collected during a 25-second acceleration and stop cycle.

A total of 13 runs during which data were collected were made, ranging in peak speeds of 28 to 84 miles per hour. Attempts were made to undertake runs with varying degrees of accelerator application ranging from mild to full throttle. No periods of extended level-throttle "cruising" were made due to the limited data acquisition time available.

TEST DATA

The data from three test runs were evaluated via simulation: Run 03, a "firm" throttle application run (55% accelerator pedal application, 58 mph peak speed); Run 10, a "heavy" throttle application run (100% accelerator pedal application, 77 mph peak speed); and Run 12, another "firm" throttle application run (61% accelerator pedal application, 66 mph peak speed).

Relevant data from Run 03 is depicted in Figure 7. It is interesting to note that the recorded position of the engine throttle is not always directly proportional to the recorded position of the accelerator pedal. This is physically possible given that the vehicle is equipped with Electronic Throttle Control (ETC) in which the position of the throttle valve is controlled via an electronic servo under computer control by the PCM rather than by a mechanical cable from the accelerator pedal. The pedal cable linkage is in turn replaced by an electronic position sensor at the foot pedal with electrical leads providing position data to the PCM. Thus, the relation between pedal position and engine throttle is determined by algorithm(s) in the PCM, and not by a mechanical linkage between the accelerator pedal and the engine throttle valve.

Also of note is the shape of the trace relating to the speed of the engine. As can be seen, this trace possesses a "sawtooth" pattern in which the peaks of the trace coincide with the moments at which the transmission changes gear ratios. By identifying the location of each peak, one can identify which gear the transmission is in during any particular phase of the test run. The estimated transmission gear positions for Run 03 are identified in Figure 7.

Another item of interest in the data traces is the relationship between engine speed and vehicle speed during the first second or so after throttle application begins. As can be seen, the engine speed increases rapidly, but the vehicle speed does not change significantly during this time. This is related to the drivetrain slip allowed by the vehicle's torque converter, the fluid coupling which allows the engine to turn while the vehicle is stopped and the transmission is in gear. The lack of a model of a clutch or torque converter in the HVE ATM proved making a precise simulation of the motion of the test vehicle from a stop impossible to achieve, as discussed below.

SIMULATION INPUTS

Initially, attempts were made to simulate the test vehicle's motion by using the accelerator pedal position data as throttle input in HVE, but it was found that the simulated vehicle's motion far underpredicted vehicle speed as compared to using engine throttle position data. Thus, the "Throttle %" parameter recorded by the PCM was used as the basis for throttle input values in the Driver Controls dialog for each test run evaluated.

The Driver Control inputs for reach run consisted of a throttle time history which duplicated the data captured in the test vehicle's PCM followed by a constant-brake-pedal-force input initiated at a time which resulted in a vehicle speed history which best matched the speed data captured by the PCM. As can be seen by the relatively constant slope of the declining speed curves after the peak speed, it was felt that this constant-force approach for the brake input was appropriate.

As mentioned previously, the transmission "upshift map" (the relation of engine speed and throttle position which results in an upshift) in the HVE Transmission Data dialog was adjusted based on test data. Each of the 13 total test runs were analyzed to identify readily-determined points at which transmission upshifts

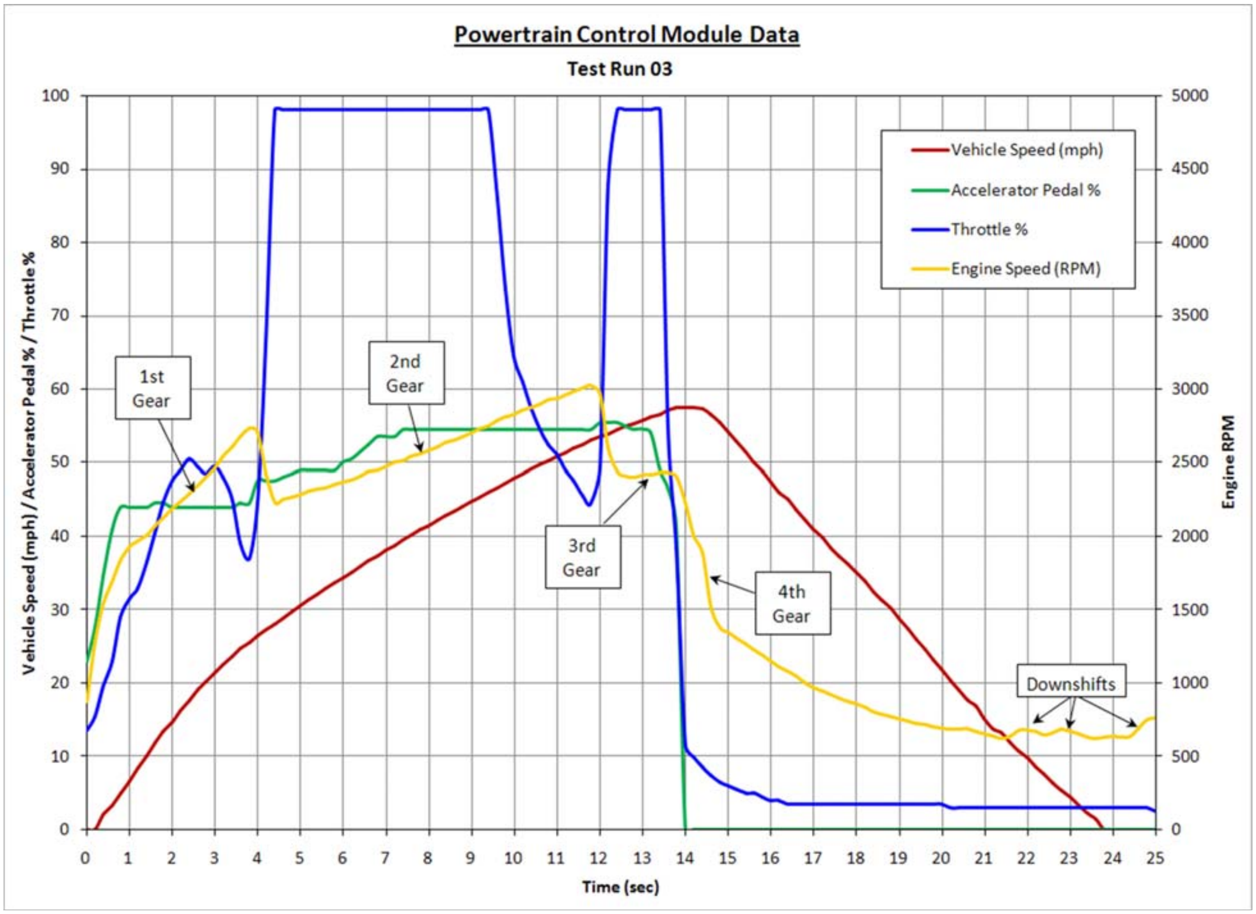

Figure 7 – Data collected from PCM during Test Run 03

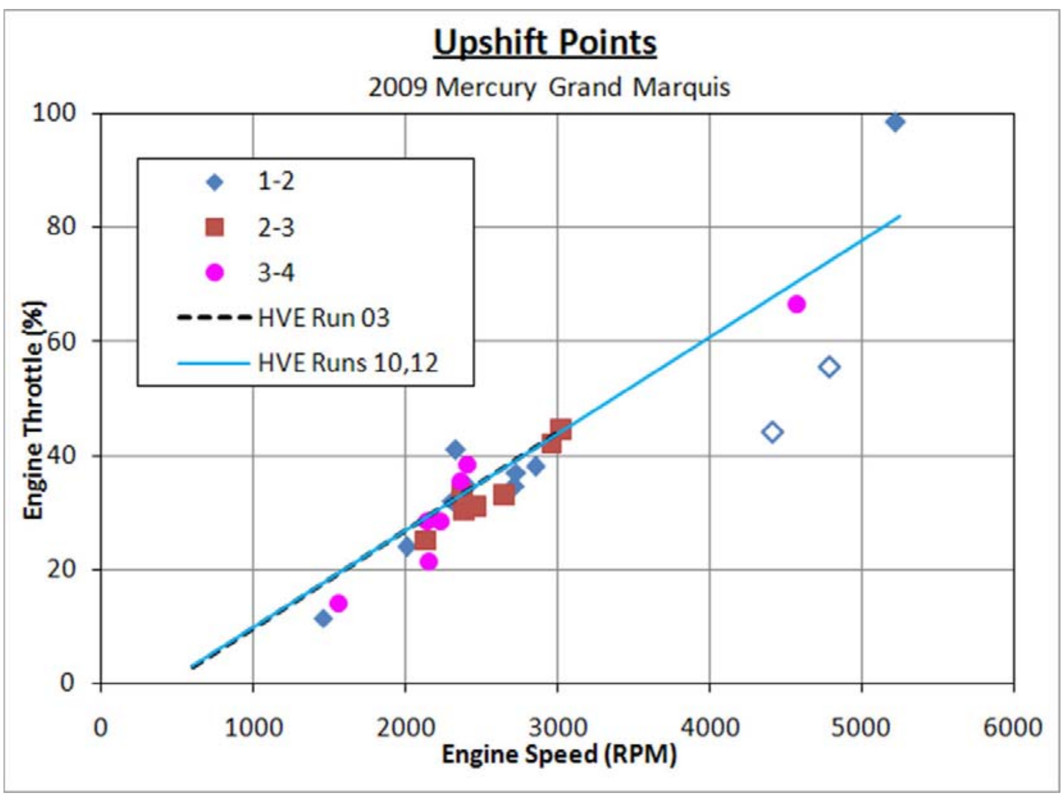

Figure 8 – Upshift data for test vehicle (points)
and relation used in simulations (lines)

occurred, and the engine speed and engine throttle positions for each shift were noted.

Based on the collected data, an upshift curve for the 2009 Mercury Grand Marquis test vehicle was developed, and is shown in Figure 8. As noted, with the exception of a few points recorded during rapid change in throttle position (in which the timing of the data recorded in the PCM may hold an effect on the values recorded), the data seem to follow a relatively linear relationship. A line estimating the relationship of throttle to engine speed was passed among the data points, without attempt at a regression analysis. This relationship was used as the basis for the upshift map in the HVE Transmission Data dialog, as shown in Figure 9. The downshift curve was left at the default values.

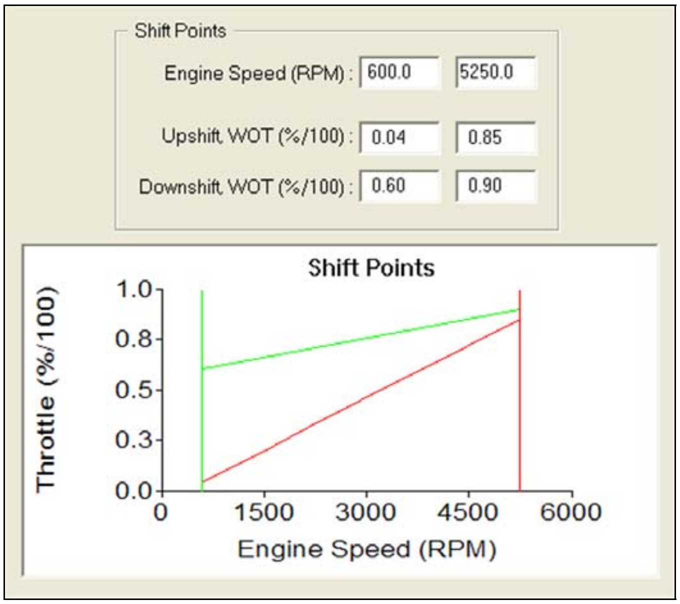

Figure 9 – Shift points used in simulations of Test Runs 10 and 12

COMPARISON OF SIMULATION RESULTS TO TEST DATA

As identified above, the HVE ATM as of Version 7.1 does not possess a clutch or torque converter model. Because of this, the simulated engine RPM's must follow in lock step with the speed of the simulated vehicle. Thus, unlike the full-scale vehicle, there is no slip allowed in the driveline, the amount of which proved to be more than insignificant in the test vehicle.

TEST RUN 03 - Plotted in Figure 11 along with other recorded vehicle parameters is torque converter slip as calculated throughout test Run 03. This parameter is calculated via the relation:

$$TC\_Slip = \frac{RPM_{engine} - RPM_{trans\_input}}{RPM_{engine}} \qquad [1]$$

Thus, torque converter slip is here defined as the difference in speed between the engine and the input shaft of the transmission divided by engine speed, where transmission input speed is calculated from measured vehicle speed via the relation:

$$RPM_{trans\_input} = \frac{V_{vehicle} \times r_{trans} \times r_{diff}}{2 \times \pi \times r_{rolling}} \qquad [2]$$

where

$V_{vehicle}$ = vehicle speed from PCM
$r_{rolling}$ = rolling radius of drive wheels
r_{trans} = gear ratio of transmission
r_{diff} = gear ratio of differential

with appropriate unit conversion factors.

In this calculation, the changes in gear ratios were presumed to have occurred instantaneously rather than over a finite time period as actually occurred in the test vehicle.

As can be seen in Figures 10, 12, and 14, torque converter slip is maximum at the start of vehicle motion (by definition = 100%), reduces in value as the vehicle gains speed, and peaks again after each gear change. Also noticeable in Figure 10 is that torque converter slip for Run 03 reached a minimum of about 3 to 5% prior to upshift. This is meaningfully greater than the 0% driveline slip modeled in the HVE ATM. Also, in all three of these test runs, torque converter slip reaches its minimum value after a period of 4 to 6 seconds.

The lack of slip in the simulated driveline prevents one from precisely simulating the motion of the vehicle from a dead stop, as unlike the real vehicle, the engine of which is able to develop significant power by increasing RPM while the vehicle is almost standing still, the simulated vehicle is limited to the power developed at the RPM of the engine as rigidly coupled to the drive wheels.

Thus, initial attempts at simulating vehicle motion from a stop led to results which were not fidelic to the test data, and so further attempts to do so were not undertaken. Instead, the simulations were started at a point where the torque converter slip was approaching a minimum value with the transmission still in first gear.

Presented in Figure 11 is the HVE simulation data as plotted against the collected test data from Run 03. The simulation of this particular run happened to provide the most fidelic output data out of the three runs analyzed. As seen in this figure, the ATM modeled the timing of upshifts fairly reasonably – within 1 second or less of the actual gear change. Further, HVE seems to have identified a shift into 4th gear at a point in the test run

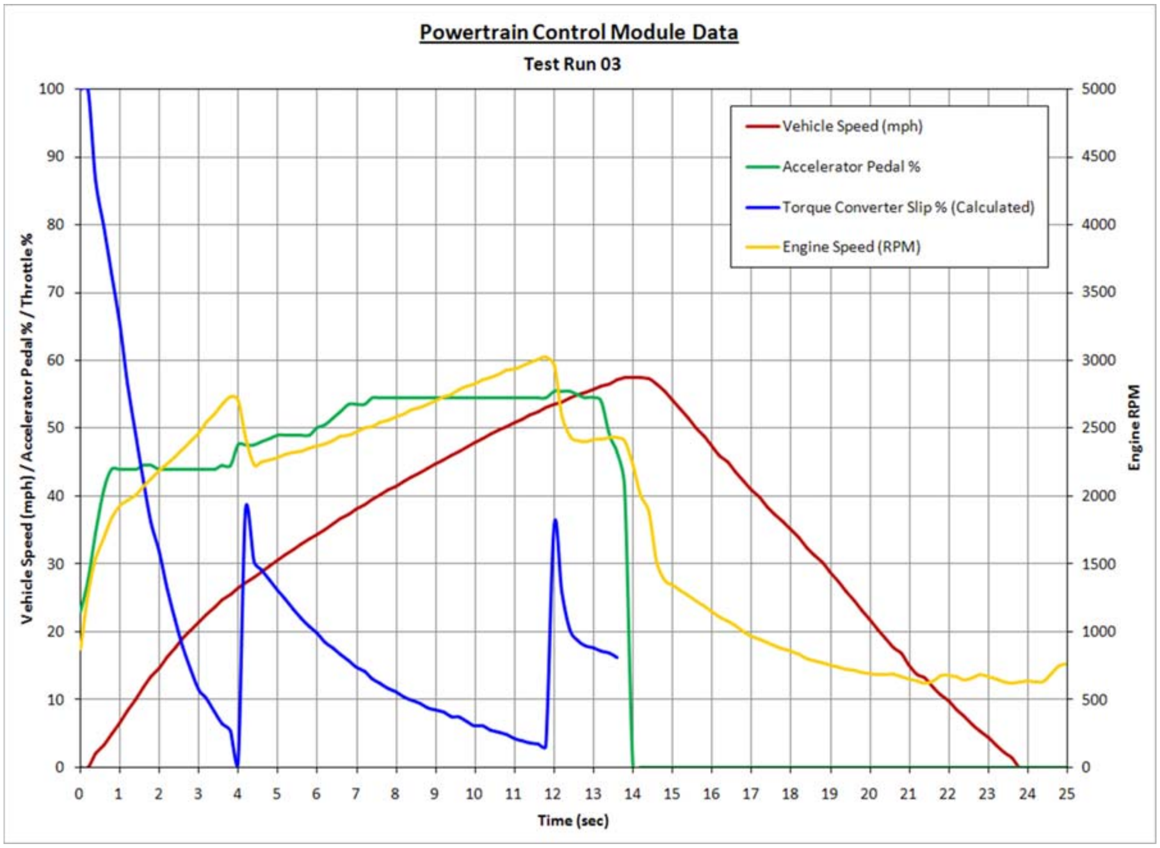

Figure 10 – Calculated torque converter slip plotted against PCM data for Test Run 03

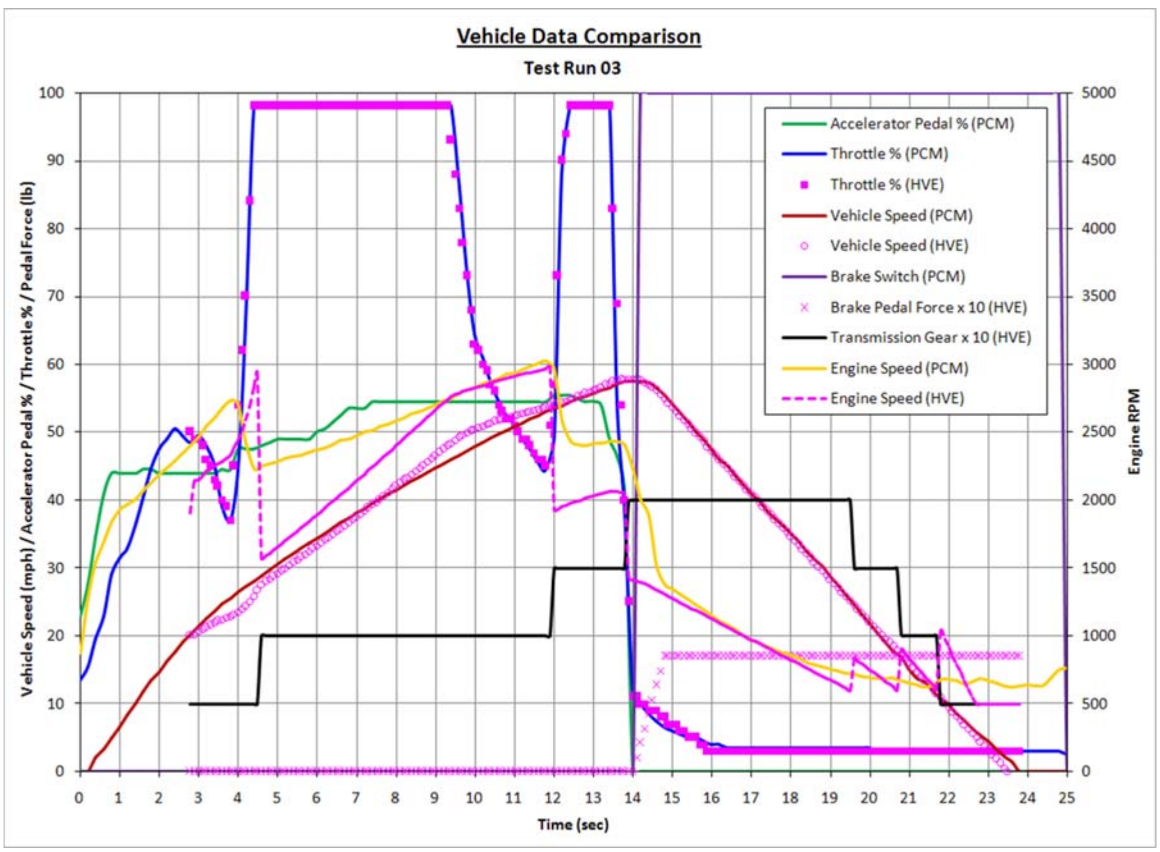

Figure 11 – Simulation output plotted against PCM data for Test Run 03

coinciding with an episode of decreasing engine RPM which would have been difficult to discern by examining the test data alone. And, importantly, the calculated top speed of the vehicle (57.8 mph) was within 0.5% of the recorded top speed of the test vehicle, 57.5 mph.

However, there are discrepancies between the calculated engine RPM and the test vehicle's RPM, particularly near the start of the run. Here, the lack of a torque converter model in the ATM manifests itself in the mismatch between the engine speed of the test vehicle versus that of the simulated vehicle. As can be seen in Figure 11, although the initial speed of the simulated vehicle is set to be the same as that of the test vehicle, the test vehicle's engine speed was several hundred RPM higher than that of the simulated vehicle at the start of the simulation.

Further, at the moment when the test vehicle shifted from 1st to 2nd gear, the RPM trace of the simulated vehicle overshot the peak RPM of the test vehicle by approximately 200 RPM and then undershot the low point of the test vehicle RPM by approximately 700 RPM. The latter is again attributed to a lack of HVE to model slip in the simulated vehicle's drivetrain.

Also of note is the slope of the simulated engine speed curve between the first and second gear shifts. As observed in Figure 11, the simulated engine speed increases at a greater rate than that of the test vehicle.

In order to prevent the simulated vehicle's engine speed from overshooting that of the test vehicle near the shift points, the simulated vehicle's Transmission Data table in Run 03 was limited to a top engine speed of 3000 RPM. This may be an effect of the manner in which the particular PCM in the test vehicle is programmed to command engine throttle based and pedal input. It is theorized that shift points may occur at varying engine speeds in the actual vehicle, as compared to the linear and unchanging relationship presumed in the HVE Transmission Data dialog.

In spite of the above discrepancies, the ATM within SIMON provided a reasonable prediction of vehicle speed and shift timing for this particular test.

TEST RUN 10 - This run was a wide-open throttle run in which both the accelerator pedal position and the engine throttle position remained at the maximum range of travel during the acceleration portion of the test run. The PCM data and calculated torque convertible slip for this run are shown in Figure 12. The test data and simulation results for this run are depicted in Figure 13.

As the data depicts, here again the simulated vehicle begins the run at an engine speed which differs from the test vehicle. In this case, the simulated vehicle's engine speed is approximately 300 RPM higher than the test vehicle's was at the same vehicle speed. Further, the simulated vehicle's first shift occurs approximately 1.3 seconds prior to that of the test vehicle.

It is believed that the two effects above combine to result in a simulated maximum vehicle speed (87.8 mph) which is 14% higher than that of the test vehicle (77.3 mph). And, while the simulated gear change from 2nd to 3rd gear occurred essentially simultaneously with that of the test vehicle, the simulated vehicle's engine speed at the time of that shift was almost 900 RPM higher than that of the test vehicle, exacerbating the higher simulated speed.

It is theorized that the discrepancy between simulated and actual engine speed values and gear change timing may be the result of the PCM's programming as compared to the simulated vehicle's presumed shift map logic.

TEST RUN 12 - This was a run made under constant firm accelerator pedal application and is one in which the engine throttle position was caused to open to close to a maximum value for much of the run, other than at times of gear changes. PCM data and calculated torque convertible sip for this run are shown in Figure 14. PCM data and simulation results for this run are depicted in Figure 15.

In this particular run, the simulated engine speed and vehicle speed at the start of the run are both very close to those of the test vehicle. And, the simulated engine peak speed and low speed during the first gear change are each within approximately 200 RPM of the test vehicle's. However, the gear change in the simulated vehicle again occurs too early – about 1.8 seconds sooner than that of the test vehicle, which contributes to the simulated vehicle's speed diverging upward from the test data for a period of time before returning briefly to that of the test vehicle.

The slope of the simulated engine speed curve is again noted to be steeper than that of the test vehicle, which likely contributes to the elevated vehicle speed predicted by the simulation.

Also, even though the second gear change of the simulated vehicle occurs at about the same time as that of the test vehicle, by the time that shift occurs, the simulated vehicle's engine speed has again overshot the test vehicle data by some 400 RPM. This is likely what is resulting in the calculated maximum speed of 73.5 mph exceeding the recorded maximum speed of 65.7 mph by 12%.

In this run, it appears that the discrepancy in vehicle speeds between the simulated and test vehicles is a result of the steep ramp-up of engine RPM in the simulated vehicle. It is theorized this may again be due to the functioning of the torque converter in the test vehicle which is not modeled by the simulation.

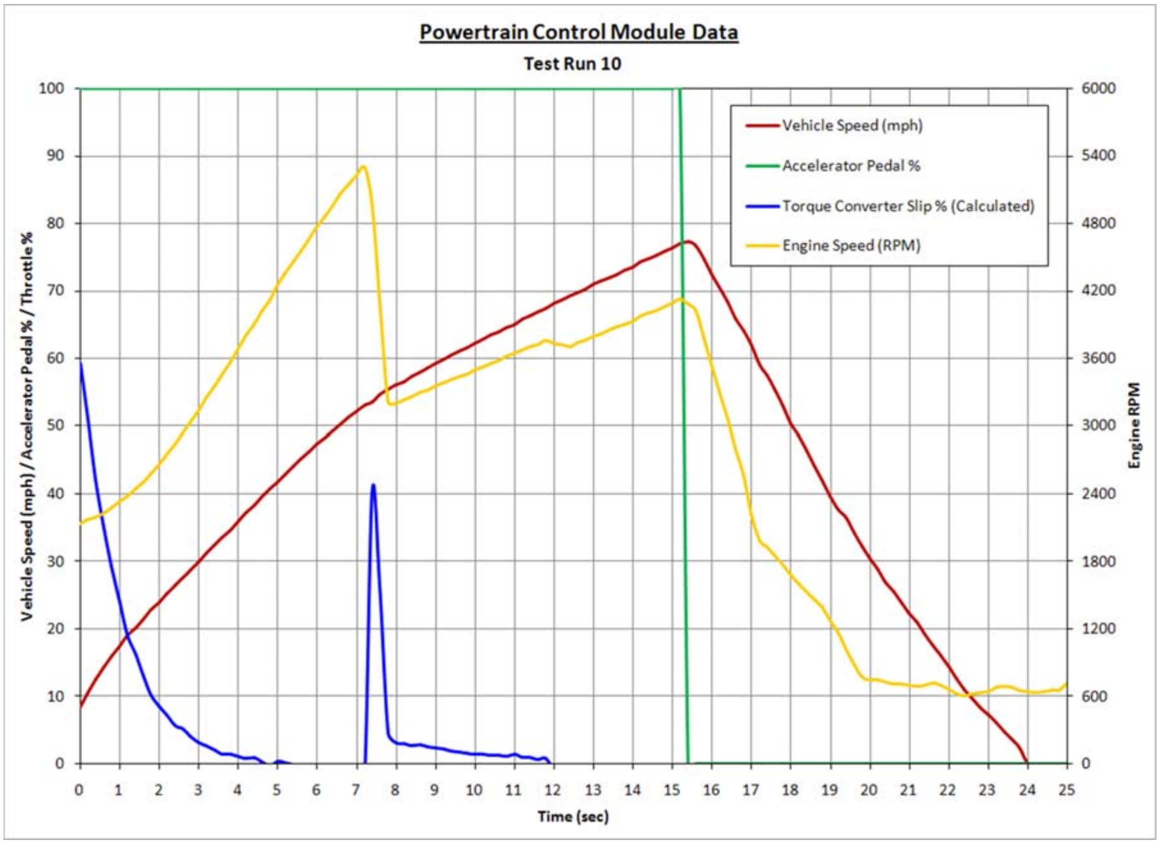

Figure 12 – Calculated torque converter slip plotted against PCM data for Test Run 10

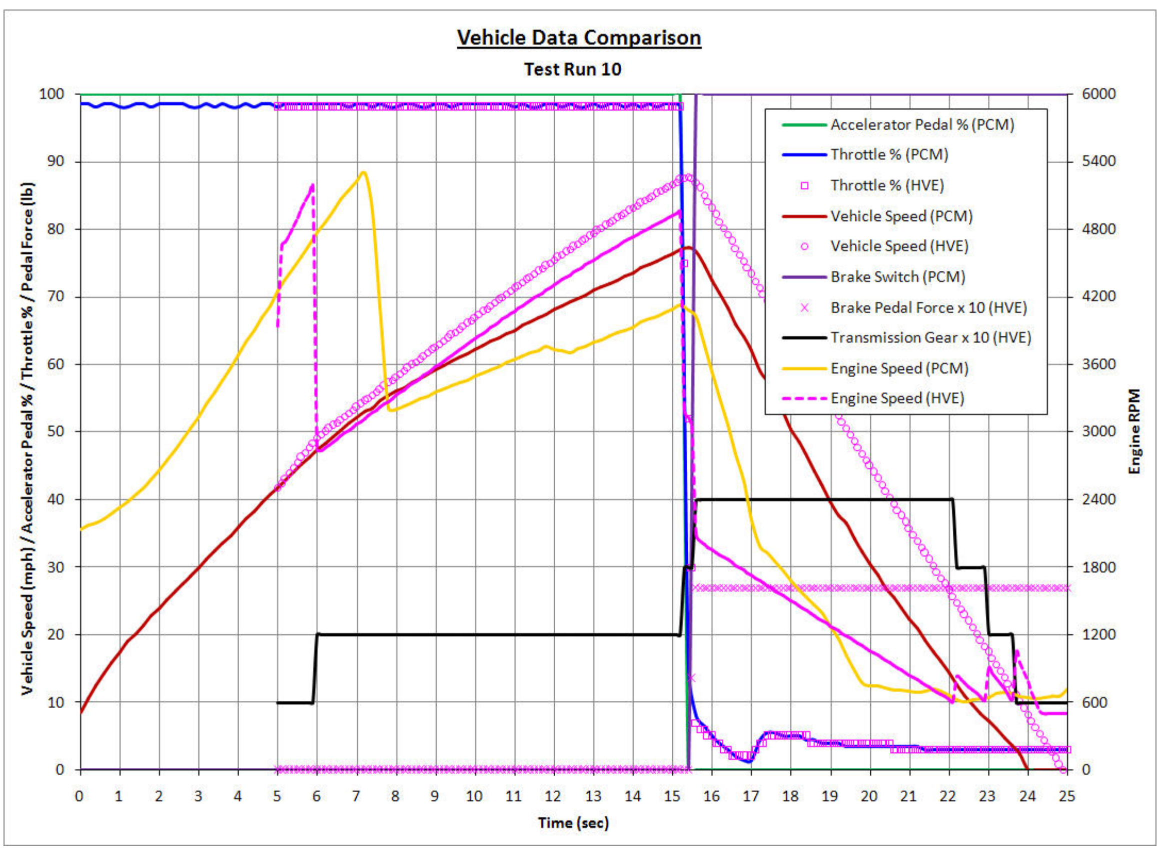

Figure 13 – Simulation output plotted against PCM data for Test Run 10

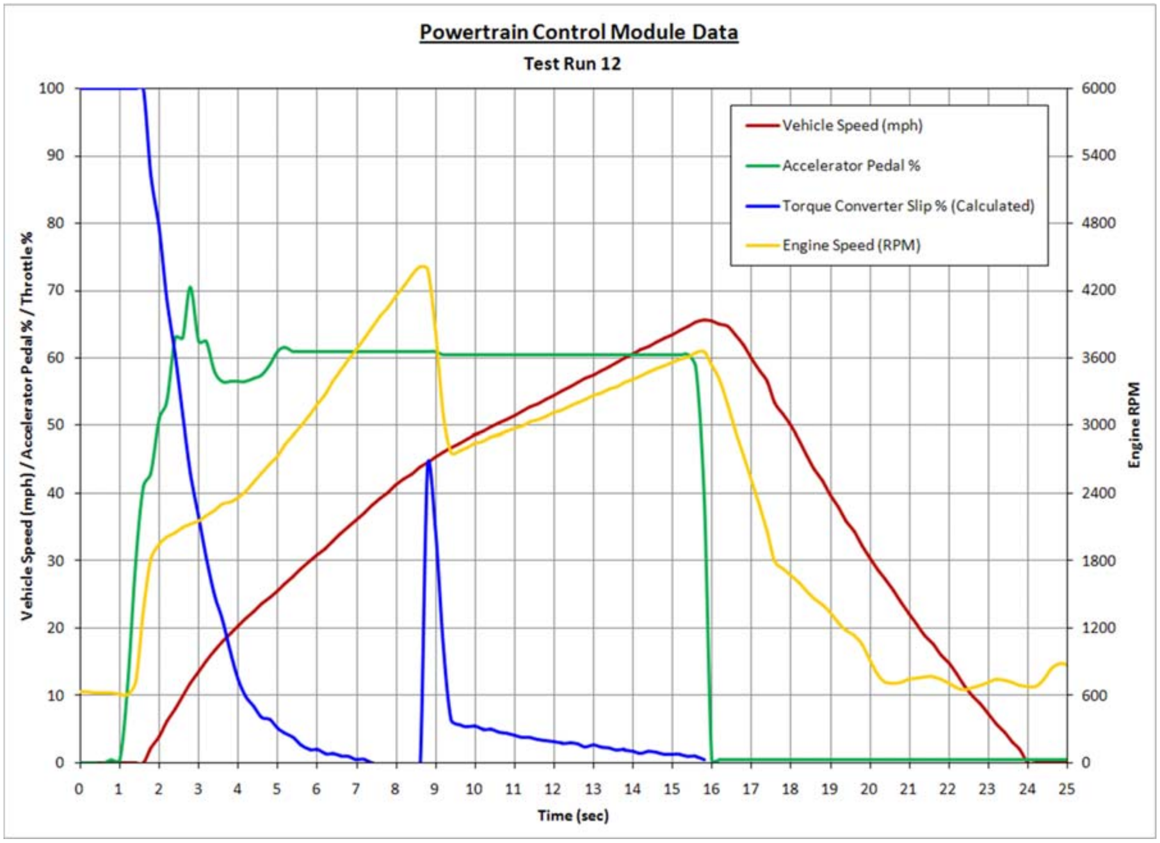

Figure 14 – Calculated torque converter slip plotted against PCM data for Test Run 12

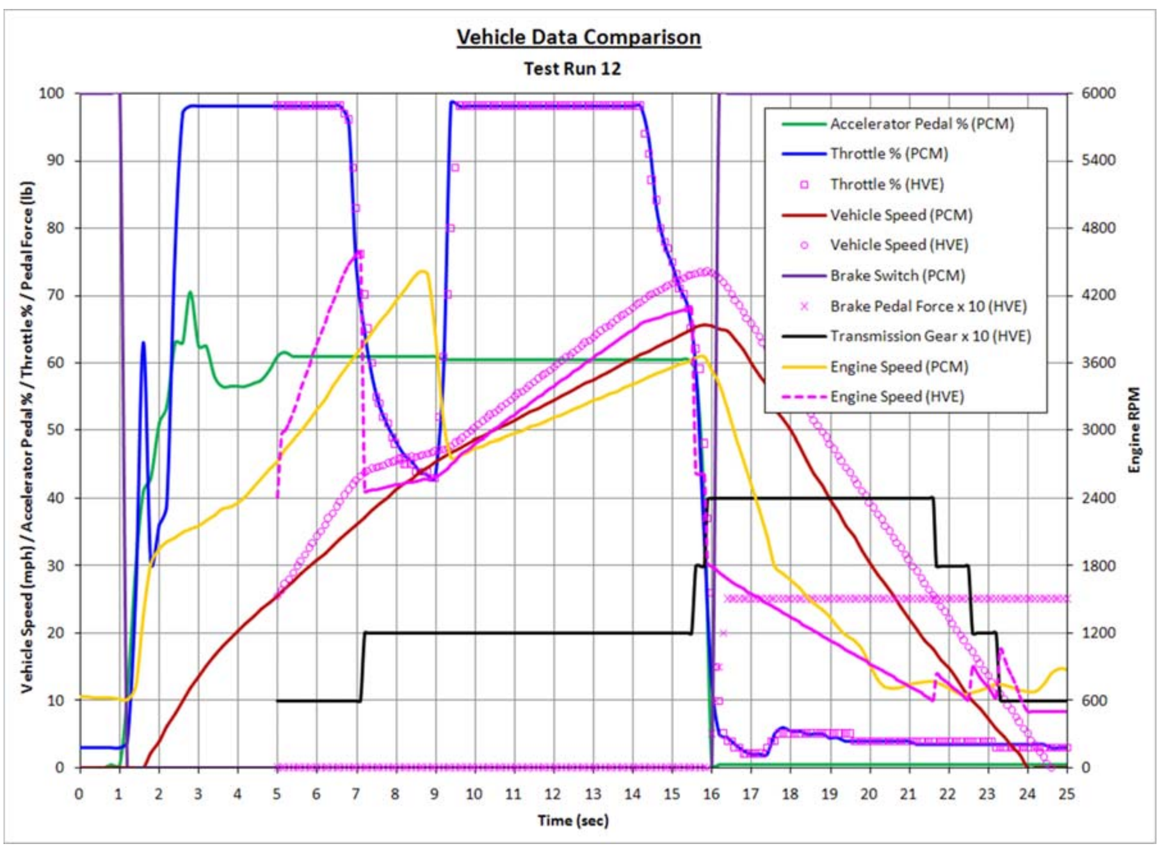

Figure 15 – Simulation output plotted against PCM data for Test Run 12

OBSERVATIONS AND CONCLUSIONS

Based on the testing and analysis undertaken in the current research, it is observed that:

- The Automatic Transmission Model (ATM) within HVE-SIMON has the capability to properly model the shift timing and vehicle speed history of a full-scale vehicle, given proper initial vehicle conditions and appropriate parameters in the Transmission Data tables.

- Default data in the HVE ATM for the particular vehicle test were found to be accurate in aspects such as gear ratios and linearity of the throttle position versus engine speed relation.

- The lack of a clutch or torque converter model in the HVE Automatic Transmission Model limits the fidelity of the model when attempting to match the vehicle speed, engine speed and shift timing profiles of a particular test vehicle at low initial vehicle speeds.

- It is suggested that the HVE ATM be provided with a model of a torque converter or a clutch which includes the incorporation of driveline slip and finite shift intervals throughout a range of vehicle speed and acceleration rates as demonstrated in the testing performed in this study.

- The engine throttle position of a vehicle equipped with Electronic Throttle Control (ETC) may not always be in direct proportion to the position of the accelerator pedal, particularly when gear changes are occurring.

- When simulating a full-scale vehicle acceleration run in which data has been collected from a Ford Powertrain Control Module, it is most appropriate to use the engine throttle position data and not the accelerator pedal position data as throttle input for the simulated vehicle.

- The automatic transmission shift map of a modern vehicle may vary based on the extent and speed at which accelerator pedal inputs are made by the driver. The linear relationship between throttle and engine speed in the Transmission Data table in HVE may not fully describe the shift map in vehicles with non-linear shift programs.

REFERENCES

1. Technical Newsletter, Summer 2009, Engineering Dynamics Corporation
2. MacInnis D.D., et. al., “A Comparison of Moment of Inertia Estimation Techniques for Vehicle Dynamics Simulation”, SAE 970951, 1997.
3. Allen R.W., et. al., “Estimation of Passenger Vehicle Inertial Properties and Their Effect on Stability and Handling”, SAE 2003-01-0966, 2003.
4. ’09 Mercury Grand Marquis product brochure, www.mercuryvehicles.com
5. VinLinNK Full Report for VIN 2MEHM75V59X632218, ESP Data Solutions, Inc., www.vinlink.com
6. “Building Vehicles for HVE”, Garvey, J.T., HVE White Paper #2000-6, Engineering Dynamics Corporation, 2000
7. “Hypertech Dynamometer Test Results, 2005-2009 Ford Crown Victoria/Grand Marquis/Town Car – 4.6L 2V V8 Engine”, Hypertech, 2008
8. 2009 Crown Victoria/Grand Marquis Workshop Manual, Ford Motor Company, 2008

CONTACT

Eric Deyerl, P.E. and Michael Fitch are engineers with Dial Engineering in Culver City, California. They provide consulting services in the areas of mechanical engineering, mechanical design, vehicular accident reconstruction, computer animation, and photographic analysis.

APPENDIX

The Vehicle Data report for the vehicle used to simulate Test Run 03 follows.

VEHICLE DATA

General Information ---

Vehicle Name:	Run 03
Vehicle Type:	Passenger Car
Vehicle Make:	Mercury
Vehicle Model:	Grand Marquis
Vehicle Year:	2003-2009
Vehicle Body Style:	4-Door
Version No:	V 5.20 (RCS $Revision: 1.12
Number of Axles:	2
Driver Location:	Left Side
Engine Location:	Front Engine
Drive Axle(s):	Axle 2

Sprung Mass Dimensional Data ---

Overall Length (in):	211.50
Overall Width (in):	73.80
Overall Height (in):	60.48
Ground Clearance (in):	12.48
Wheelbase (in):	114.60
CG to Front Axle (in):	51.78
CG to Back Axle (in):	-62.82
CG Height (in):	22.20
Front Overhang (in):	42.32
Rear Overhang (in):	54.58

Sprung Mass Inertial Data ---

Total Weight (lb):	4299.00
Sprung Weight (lb):	3912.29
Sprung Mass (lb-sec^2/in):	10.12
Sprg Mass Rot Inertia (lb-sec^2-in) - Roll:	6329.69
Pitch:	33367.66
Yaw:	34568.84
XZ Product:	0.00

Sprung Mass Aerodynamic Parameters ---

Surface Name:	Front
Drag Coefficient:	0.3500
Proj. Surface Area (in^2):	3378.24
Center of Pressure (in) - x:	89.00
y:	0.00
z:	0.00

Brake System Data ---

Brake Pedal Ratio (psi/lb):	17.00
ABS System:	Tire Slip Algorithm
ABS Controller Location:	This Vehicle
Sample Method:	Wheel-Based
Delay Method:	Wheel-Based
Threshold Pressure (psi):	10.00
Threshold Velocity (mph):	4.00

Steering System Parameters ---

First Axle:	Steerable
Steering Gear Ratio (deg/deg):	16.40

	Right Side	Left Side
Caster (deg):	6.20	6.20

Inclination Angle (deg):	10.08	10.08
Steering Offset (in):	0.00	0.00
Stub Axle Length (in):	2.46	2.46
Initial Steer Axis Coord (in) - x:	51.78	51.78
y:	28.94	-28.94
z:	8.36	8.36

Second Axle: Not Steerable

Drivetrain Parameters ---

Engine Description:	4.6L V-8 4-Spd Auto
Maximum Power (HP):	224
Maximum Torque (ft-lb):	275
Transmission Forward Speeds:	4
Differential Speeds:	1

Wide-open Throttle, Speed (RPM):	500	1000	1500	2500	3000	3500	4000	4500	4800	5000
Power (HP):	5	32	65	131	157	183	209	223	224	223
Torque (ft-lb):	50	171	229	275	275	275	275	260	245	234

Closed Throttle, Speed (RPM):	500	2900	4000	4800	6400
Power (HP):	-1	-28	-54	-77	-137
Torque (ft-lb):	-9	-51	-71	-85	-113

Transmission Type: Automatic

Shift Points -	Min	Max
Engine Speed (RPM):	600	3000
Shift Up, WOT (%/100):	0.04	0.44
Shift Down, WOT (%/100):	0.04	0.19

Transmission Gear:	Rev	1st	2nd	3rd	4th
Numerical Ratio:	-2.32	2.84	1.55	1.00	0.70

Differential Gear Ratio: 2.730

Wheel Location Information, First Axle ---

	Right Side	Left Side
Initial Wheel Coordinates (in) - x:	51.78	51.78
y:	31.40	-31.40
z:	8.38	8.38

Suspension Information, First Axle ---

Suspension Type:	Independent
Auxiliary Roll Stiffness (in-lb/deg):	9221.32

	Right Side	Left Side
Spring Rate (lb/in):	219.80	219.80
Viscous Damping (lb-sec/in):	10.17	10.17
Coulomb Friction (lb):	50.00	50.00
Friction Null Band (in/sec):	5.00	5.00
Deflection to Jounce Stop (in):	-5.00	-5.00
Stop Linear Rate (lb/in):	300.00	300.00
Stop Cubic Rate (lb/in^3):	600.00	600.00
Stop Energy Ratio (%/100):	0.50	0.50
Deflection to Jounce Stop (in):	5.00	5.00
Stop Linear Rate (lb/in):	300.00	300.00
Stop Cubic Rate (lb/in^3):	600.00	600.00
Stop Energy Ratio (%/100):	0.50	0.50
Roll Steer Const. Coef (deg):	0.00	0.00
Roll Steer Linear Coef (deg/in):	0.00	0.00

Roll Steer Quadratic Coef (deg/in):	0.00	0.00
Roll Steer Cubic Coef (deg/in):	0.00	0.00

Camber and Half-track Tables

-------- Right Side -------			-------- Left Side --------		
Susp Defl (in)	Camber (deg)	1/2-track Change (in)	Susp Defl (in)	Camber (deg)	1/2-track Change (in)
-5.00	-0.40	0.00	-5.00	-0.40	0.00
0.00	-0.40	0.00	0.00	-0.40	0.00
5.00	-0.40	0.00	5.00	-0.40	0.00

Tire Information, First Axle ---

	Right Side	Left Side
Tire Name:	Generic	Generic
Tire Manufacturer:	Generic	Generic
Tire Model:	Generic	Generic
Tire Size:	P225/60R17	P225/60R17
Version No:	es\db\g	es\db\g
Unloaded Radius (in):	13.82	13.82
Nominal Width (in):	8.86	8.86
Tread Width (in):	7.09	7.09
Init. Radial Stiffness (lb/in/tire):	1500.00	1500.00
2nd Radial Stiffness (lb/in/tire):	15000.00	15000.00
Defl. @ 2nd Stiffness (in):	4.26	4.26
Max Deflection (in):	5.32	5.32
Rebound Energy Ratio (%/100):	1.00	1.00
Spin Inertia (Tire+Whl+Brk, lb-sec^2-in/	17.08	17.08
Steer Inertia (Tire+Whl+Brk, lb-sec^2-in	8.54	8.54
Weight (Tire+Whl+Brk, lb/tire):	59.00	59.00
Roll Resistance Const:	0.01	0.01
Roll Resististance Linear Coef (sec/in):	0.00	0.00
Min Fz For Skidmark (lb):	413.25	413.25
Pneumatic Trail (in):	-0.97	-0.97
Lateral Stiffness (lb/in):	1500.00	1500.00

Cornering Stiffness (lb/deg/tire):	Right Side			Left Side		
Loads (lb):	826.5	1653.0	2479.5	826.5	1653.0	2479.5
Speeds (in/sec):	528.0			528.0		
Load No.:	1	2	3	1	2	3
Speed No. 1:	186.3	349.5	406.7	186.3	349.5	406.7

Camber Stiffness (lb/deg/tire):	Right Side			Left Side		
Loads (lb):	826.5	1653.0	2479.5	826.5	1653.0	2479.5
Speeds (in/sec):	528.0			528.0		
Load No.:	1	2	3	1	2	3
Speed No. 1:	18.6	35.0	40.7	18.6	35.0	40.7

Tire Friction Table:	Right Side			Left Side		
Loads (lb):	826.5	1653.0	2479.5	826.5	1653.0	2479.5
Speeds (in/sec):	528.0			528.0		
Speed No. 1, Load No.:	1	2	3	1	2	3
Peak Mu:	1.1500	1.1000	1.0500	1.1500	1.1000	1.0500
Slide Mu:	0.9000	0.8500	0.8000	0.9000	0.8500	0.8000
Slip @ Peak Mu (%/100):	0.1600	0.1600	0.1600	0.1600	0.1600	0.1600
Long. Stiffness (lb/slip):	7000.0	13000.0	18000.0	7000.0	13000.0	18000.0

Brake Information, First Axle ---

	Right Side	Left Side
Brake Assembly Type:	Generic Brake	Generic Brake
Brake Time Lag (sec):	0.0000	0.0000
Brake Time Rise (sec):	0.0000	0.0000
Pushout Pressure (psi):	0.00	0.00
Nominal Brake Torque Ratio (in-lb/psi):	19.58	19.58
ABS Parameters ---		
Min Wheel Slip (%/100):	0.0500	0.0500
Max Wheel Slip (%/100):	0.1500	0.1500
Apply Delay (sec):	0.0500	0.0500
Pri Apply Rate (psi/sec):	5000.00	5000.00
Sec Apply Rate (psi/sec):	500.00	500.00
Release Delay (sec):	0.0500	0.0500
Release Rate (psi/sec):	10000.00	10000.00

Wheel Location Information, Second Axle ---

	Right Side	Left Side
Initial Wheel Coordinates (in) - x:	-62.82	-62.82
y:	32.80	-32.80
z:	8.38	8.38

Suspension Information, Second Axle ---

Suspension Type:	Solid Axle
Axle Roll/Yaw Inertia (lb-sec^2-in):	74.20
Axle Roll Ctr Ht Below CG (in):	5.38
Axle Roll Steer (deg/deg):	0.00
Lateral Spring Spacing (in):	36.00
Nominal Track Width (in):	65.60
Total Unsprung Weight (Axle+Wheels, lb):	268.70
Auxiliary Roll Stiffness (in-lb/deg):	0.00

	Right Side	Left Side
Spring Rate (lb/in):	130.00	130.00
Viscous Damping (lb-sec/in):	6.49	6.49
Coulomb Friction (lb):	50.00	50.00
Friction Null Band (in/sec):	5.00	5.00
Deflection to Jounce Stop (in):	-5.00	-5.00
Stop Linear Rate (lb/in):	300.00	300.00
Stop Cubic Rate (lb/in^3):	600.00	600.00
Stop Energy Ratio (%/100):	0.50	0.50
Deflection to Jounce Stop (in):	5.00	5.00
Stop Linear Rate (lb/in):	300.00	300.00
Stop Cubic Rate (lb/in^3):	600.00	600.00
Stop Energy Ratio (%/100):	0.50	0.50
Camber Constant (deg):	0.00	0.00

Tire Information, Second Axle ---

	Right Side	Left Side
Tire Name:	Generic	Generic
Tire Manufacturer:	Generic	Generic
Tire Model:	Generic	Generic
Tire Size:	P225/60R17	P225/60R17
Version No:	es\db\g	es\db\g
Unloaded Radius (in):	13.82	13.82
Nominal Width (in):	8.86	8.86
Tread Width (in):	7.09	7.09
Init. Radial Stiffness (lb/in/tire):	1500.00	1500.00

2nd Radial Stiffness (lb/in/tire):	15000.00	15000.00
Defl. @ 2nd Stiffness (in):	4.26	4.26
Max Deflection (in):	5.32	5.32
Rebound Energy Ratio (%/100):	1.00	1.00
Spin Inertia (Tire+Whl+Brk, lb-sec^2-in/	17.08	17.08
Steer Inertia (Tire+Whl+Brk, lb-sec^2-in	8.54	8.54
Weight (Tire+Whl+Brk, lb/tire):	59.00	59.00
Roll Resistance Const:	0.01	0.01
Roll Resististance Linear Coef (sec/in):	0.00	0.00
Min Fz For Skidmark (lb):	413.25	413.25
Pneumatic Trail (in):	-0.97	-0.97
Lateral Stiffness (lb/in):	1500.00	1500.00

Cornering Stiffness (lb/deg/tire):	Right Side			Left Side		
Loads (lb):	826.5	1653.0	2479.5	826.5	1653.0	2479.5
Speeds (in/sec):	528.0			528.0		
Load No.:	1	2	3	1	2	3
Speed No. 1:	186.3	349.5	406.7	186.3	349.5	406.7

Camber Stiffness (lb/deg/tire):	Right Side			Left Side		
Loads (lb):	826.5	1653.0	2479.5	826.5	1653.0	2479.5
Speeds (in/sec):	528.0			528.0		
Load No.:	1	2	3	1	2	3
Speed No. 1:	18.6	35.0	40.7	18.6	35.0	40.7

Tire Friction Table:	Right Side			Left Side		
Loads (lb):	826.5	1653.0	2479.5	826.5	1653.0	2479.5
Speeds (in/sec):	528.0			528.0		
Speed No. 1, Load No.:	1	2	3	1	2	3
Peak Mu:	1.1500	1.1000	1.0500	1.1500	1.1000	1.0500
Slide Mu:	0.9000	0.8500	0.8000	0.9000	0.8500	0.8000
Slip @ Peak Mu (%/100):	0.1600	0.1600	0.1600	0.1600	0.1600	0.1600
Long. Stiffness (lb/slip):	7000.0	13000.0	18000.0	7000.0	13000.0	18000.0

Brake Information, Second Axle ---

	Right Side	Left Side
Brake Assembly Type:	Generic Brake	Generic Brake
Brake Time Lag (sec):	0.0000	0.0000
Brake Time Rise (sec):	0.0000	0.0000
Pushout Pressure (psi):	0.00	0.00
Nominal Brake Torque Ratio (in-lb/psi):	9.99	9.99
Brake Proportioning Pressure (psi):	200.00	200.00
Brake Proportioning Ratio:	0.33	0.33

ABS Parameters ---

Min Wheel Slip (%/100):	0.0500	0.0500
Max Wheel Slip (%/100):	0.1500	0.1500
Apply Delay (sec):	0.0500	0.0500
Pri Apply Rate (psi/sec):	5000.00	5000.00
Sec Apply Rate (psi/sec):	500.00	500.00
Release Delay (sec):	0.0500	0.0500
Release Rate (psi/sec):	10000.00	10000.00