FMTV TRANSMISSION FUEL ECONOMY STUDY: EVALUATION OF AMT PERFORMANCE USING EXPERIMENTAL AND ANALYTICAL METHODS

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ABSTRACT

Fuel economy improvements were investigated for the FMTV platform considering alternative transmissions and final drive ratio. An FMTV-M1078 with Caterpillar C7 engine and Allison 3700SP transmission was the target vehicle of this study. Experimental data were collected while vehicle was operated over the FTP72 test cycle. Base vehicle data (vehicle weight, coast-down times, etc.) were collected to provide comparison data for establishing the baseline analytical vehicle model.

Experimental data were processed to determine road load parameters, engine BSFC map, transmission shift schedule and similar for populating the analytical model. Modeling was performed using GT-Drive. The model was analyzed over the same defined drive cycle used to collect the experimental data. Once the model was correlated to the experimental data, updates were made for the variants in transmission and drive-line parameters to be used in the fuel economy study.

The difference between experimental average and the baseline analytical model was approximately 3.1% or 0.2 mi/gal. With a mature base model the transmission variants were introduced. Without changing the final drive ratio the AMT transmissions demonstrated an average 13.4% or ~ 1.0 mi/gal improvement in fuel consumption when compared to an optimized version of the Allison. Changing the final drive ratio provided a more realistic model that takes full advantage of the wider gear range of the AMT transmissions. With the revised final drive ratio the AMT transmissions produced similar results of approximately 22% or 1.75 mi/gal improvement in fuel consumption when compared to an optimized version of the Allison.

1. INTRODUCTION

Military vehicle fuel economy improvement has and continues to be a significant initiative for the US Army. Considerable investment has been made in the area of emerging technology; largely resulting in complex hybrid powertrain solutions. Although many of these hybrid powertrain solutions have demonstrated significant fuel economy improvement they do generally require extensive vehicle & powertrain architectural changes and would require significant development and validation programs. This project is based on the premise of applying existing COTS technologies that have the potential for near-term application and consequently the possibility for a more immediate impact to fuel economy.

The purpose of this study was to determine realistic approximations of the impact to fuel economy that can be expected through the application of an automated manual transmission (AMT) to the FMTV platform. The existing FMTV platform uses an Allison automatic transmission. The two AMT evaluated were the ZF AS Tronic 12 speed and the Eaton UltraShift® *PLUS* 10 speed. Both of these transmissions align with the COTS premise described above. Additionally, an AMT provides the same user interface as an automatic transmission, i.e. accelerator and brake pedals but no clutch.

This study utilized a combination of on-road experimental vehicle data and CAE simulation. This report presents the experimental and analytical fuel economy assessment methods, results, and conclusions. Included in this report are the subject hardware attributes, experimental test methods & results, analytical modeling strategy & model parameterization, experimental to analytical correlation, AMT transmission modeling strategy & results, analytical model variation investigation, and overall study conclusions. Considerations such as design, packaging, development, durability or manufacturing are outside the scope of this study.

2. APPROACH – MERGING THE REAL AND VIRTUAL WORLDS

An FMTV test vehicle (C7 engine, Allison transmission) was lightly instrumented to collect among other information: engine speed, vehicle speed, accelerator pedal position, manifold pressure, gear position, fuel flow rate, and wind speed & direction. Data was collected while vehicle was driven over a fuel economy test cycle. Base vehicle data (vehicle weight, coast-down times, etc.) were collected to help in modeling and simulation. The combined data set was used as the baseline for building, comparing, and maturing of an analytical model.

An analytical model of the vehicle and driveline system was created using GT-Drive. This model was analyzed over the defined drive cycle to correlate the model to the experimental vehicle data. Once correlated, the model was updated with the design parameters from the transmission variants to establish fuel economy impact estimates. The model was then evaluated with a revised final drive ratio to establish fuel economy sensitivity to this factor.

3. VEHICLE and TRANSMISSION VARIANTS

3.1. FMTV Vehicle Attributes

The vehicle that is the subject of this investigation is the FMTV A1 R M1078 A1 - 2.5 ton Standard Cargo produced by the Tactical Vehicle Systems Division of Armor Holdings. The manufacturers' specifications sheet [1] was used to collect base vehicle information. What follows is a brief summary of the vehicle specification:

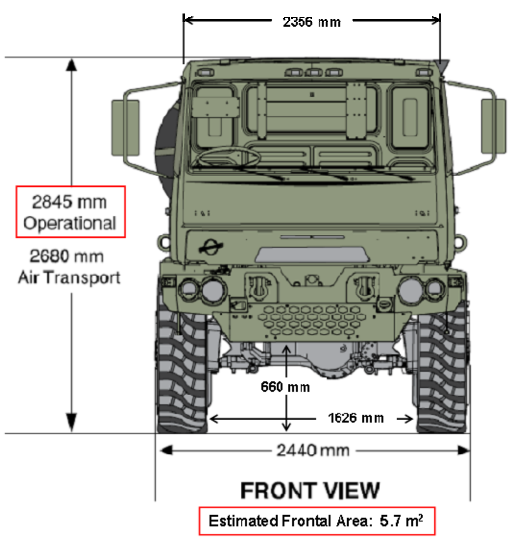
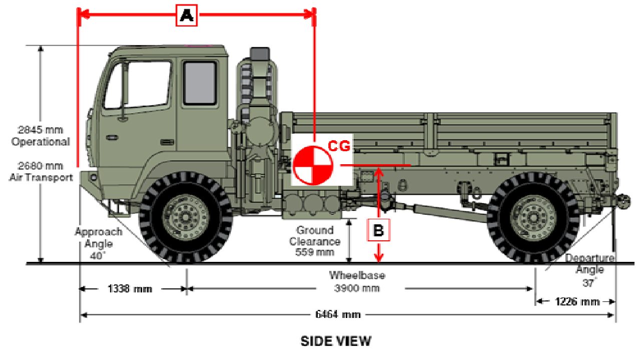


Figure 1 - Exterior dimensions of FMTV-M1078 (1)



MAX Payload: 2269 kg

WEIGHT RATING	DISTANCES (mm)		STATIC AXEL LOADS (kg)		
	A	B	Front	Rear	Total
CURB	2718	1186	5,302	2,903	8,206
GROSS	3099	1384	5,746	4,728	10,474

Figure 2 - Exterior dimensions of FMTV-M1078 (2)

FMTV M1078 engine specifications:

Manu.:	Caterpillar	
PART:	330 Hp C7-engine	
Part No.'s	SERIAL #: FML07676	
DESCRIPTION	**UofM**	
Displacement	L	7.24
Moment of Inertia	kg.m2	0.9725
Idle engine spd	RPM	700-800, programable
Max engine spd	RPM	2600

Figure 3 – C7 Engine specifications

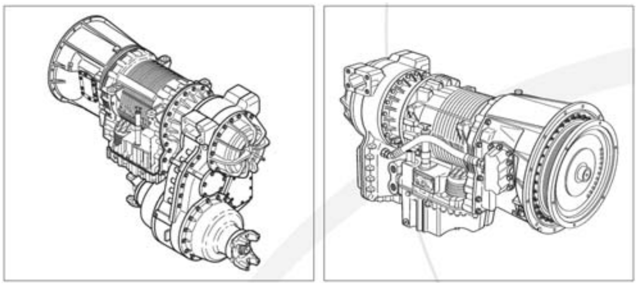
WOT performance data

ENGINE SPEED-RPM	ENGINE POWER-BHP	ENGINE TORQUE-LB.FT
800	76	499
900	99	578
1000	129	678
1100	162	773
1200	197	862
1300	213	861
1400	228	855
1440	236	859.99
1500	245	857.78
1600	259	851.14
1700	271	837.87
1800	282	821.64
1900	291	804.68
2000	301	789.19
2100	309	773.7
2200	317	756.74
2300	324	739.04
2400	330	722.07
2500	252	528.83
2600	166	335.59

Figure 4 – C7 Engine performance data

FMTV M1078 Transmission Specifications:

ALLISON 3700SP automatic transmission with integrated torque converter (TC-418, stall ratio of 1.98) and transfer case with 30/70 and 50/50 torque split modes.



From the manufacturers' specifications sheet [2] there are three shifting sequence calibrations available. Based upon experimental testing it was determined that the following shift sequence was used for this application:

Option 3: 1C–[1L]–2C–2L–3L–4L–5L–6L–7L
[C = Converter mode (lockup clutch disengaged); L = Lockup mode (lockup clutch engaged)]

Additionally, the transmission is calibrated to skip 1st gear unless specifically requested by the driver. This behavior was confirmed during experimental testing.

Gear #	(1)Ratio i	Gear Ratio Step fraction	(2)Inertia kg.m2	(3)Efficiency fraction
1	6.930	1.658	0.16	0.954
2	4.180	1.866	0.17	0.961
3	2.240	1.325	0.20	0.971
4	1.690	1.408	0.23	0.975
5	1.200	1.333	0.34	0.980
6	0.900	1.154	0.46	0.980
7	0.780		0.66	0.980

NOTES:
1 Gear ratios do not include TC multiplication
2 Inertias estimated based upon other transmissions
3 Efficiencies estimated based upon other

Figure 5 – Allison transmission gear ratio detail

3.2. Transmission Variant Attributes

Automated manual transmissions provide the same user interface as an automatic transmission, i.e. accelerator and brake pedal, with the efficiency benefits of a manual transmission. The physical shifting process of an automated manual transmission is the same as in a manual transmission: opening of the clutch, shifting, and closure of the clutch, with the difference, that the shifting and clutching operations are being initiated and operated by an Electronic Control Unit. Both the ZF AS Tronic 12 speed and the Eaton UltraShift *PLUS* 10 speed are similar in this regard.

Gear #	Ratio i	Gear Ratio Step fraction	Inertia kg.m2	(1)Efficiency fraction
1	10.369	1.230	0.084	0.945
2	8.428	1.299	0.108	0.948
3	6.487	1.230	0.091	0.952
4	5.273	1.261	0.118	0.956
5	4.182	1.230	0.107	0.96
6	3.399	1.371	0.143	0.963
7	2.480	1.231	0.116	0.968
8	2.015	1.299	0.156	0.971
9	1.551	1.230	0.172	0.975
10	1.261	1.261	0.241	0.979
11	1.000	1.230	0.302	0.99
12	0.813		0.437	0.98

NOTES:
1 Efficiency estimated based upon other transmissions

Figure 6 – ZF AS Tronic transmission gear ratio detail

Gear #	Ratio i	Gear Ratio Step fraction	[1]Inertia kg.m2	[2]Efficiency fraction
Lo-Lo	14.560	1.546	0.110	0.94
Low	9.420	1.510	0.150	0.95
1	6.240	1.348	0.160	0.96
2	4.630	1.362	0.165	0.96
3	3.400	1.344	0.180	0.96
4	2.530	1.383	0.190	0.96
5	1.830	1.346	0.220	0.98
6	1.360	1.360	0.290	0.98
7	1.000	1.351	0.400	0.99
8	0.740		0.700	0.98

NOTES
1 Inertias estimated based upon other transmissions
2 Generic values for Eaton AMT 10 speeds

Figure 7 – Eaton transmission gear ratio detail

4. EXPERIMENTAL TESTING

Experimental testing was performed to collect real world vehicle data to populate and validate the analytical vehicle model. Fuel economy drive cycle and coast down testing were performed. Only base vehicle testing, as delivered, with Allison transmission was performed. Instrumented vehicle test weight and daily weather data were also acquired. All testing was performed using pump Diesel DF2 fuel.

The experimental testing was performed between February 17 and February 19, 2009 at the Chrysler Arizona Proving Ground in Yucca Arizona. The oval test track was the only surface used for this project. The following sections describe the test procedures, vehicle instrumentation, experimental results and subsequent analysis.

4.1. Fuel economy test cycle

Despite early efforts a military specific fuel economy cycle was not available to support this project. This appears to be an open issue for the US Army as several sources [3] have been found that document this concern specifically as it applies to evaluation of hybrid powertrain technologies.

In absence of a military test cycle it was agreed to use the U.S. FTP-72 (Federal Test Procedure) cycle. This cycle is also called the Urban Dynamometer Driving Schedule (UDDS) or LA-4 cycle [CFR 40, 86, App.I]. The same engine driving cycle is known in Sweden as A10 or CVS (Constant Volume Sampler) cycle and in Australia as the ADR 27 (Australian Design Rules) cycle. The cycle simulates an urban route of 12.07 km (7.5 mi) with frequent stops. The maximum speed is 91.2 km/h (56.7 mi/h) and the average speed is 31.5 km/h (19.6 mi/h).

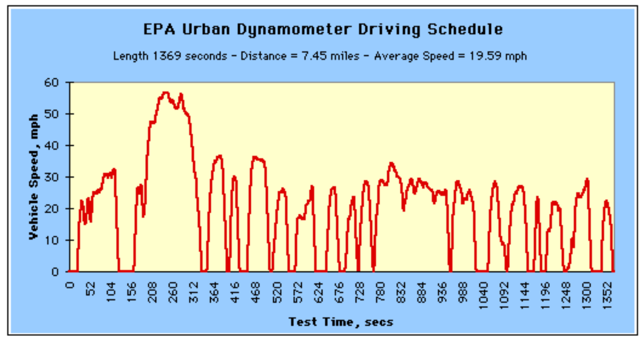


Figure 8: FTP-72 Drive Cycle

4.2. Coast down method

Coast down testing was performed to establish the drag forces that act on the vehicle. In this method the vehicle is accelerated to a desired speed, transmission set to neutral gear and then allowed to decelerate under the action of various drag forces. The general form of the equation of motion is:

$$M_e \cdot \frac{dV}{dt} = D_m + D_a + D_g$$

where:

V - vehicle speed, in Km/h

$M_e = M + \frac{I_f + I_r}{R^2}$ - effective vehicle mass including rotating components), in kg

where:

M - vehicle mass, in kg

I_f, I_r - mass moment of inertia for the front/rear axle, in $kg \cdot m^2$

R - tire rolling radius, in m

The second term in the expression of the effective vehicle mass is the effective mass of the rotating components, which represent the inertia of the rotating components expressed as additional linear mass.

The mechanical drag is modeled as a second degree function of vehicle speed. Its expression is:

$$D_m = A_m + B_m \cdot V + C_m \cdot V^2$$

where:

A_m, B_m, C_m - mechanical drag coefficients

The aerodynamic drag is modeled as a 4-th degree function of yaw angle and second degree relative wind speed. Its expression is as fellows:

$$D_a = \frac{1}{2} \cdot \rho_{air} \cdot A \cdot V_r^2 \cdot \left(a_0 + a_1 \cdot Y + a_2 \cdot Y^2 + a_3 \cdot Y^3 + a_4 \cdot Y^4\right)$$

where:

ρ_{air} - air density, in kg/m3

V_r - relative wind speed, in km/h

Y - wind yaw angle, in deg

a_0, a_1, a_2, a_3, a_4 - aerodynamic drag coeff.

The gravitational drag is a function of slope and vehicle mass. Its expression is:

$$D_g = \pm M \cdot g \cdot \frac{dh}{ds}$$

where:

$g = 9.81 \frac{m}{s^2}$ - gravitational constant,

$\frac{dh}{ds}$ - slope

The sign in the gravitational drag equation is a function of direction of movement (uphill = ”+” or downhill= ”-“).

The equation of motion is based on the following assumptions:

- The decrease in the mechanical drag due to lift forces is very small and is disregarded
- The tire slip angle due to aerodynamic side forces and yawing moments are negligible and do not influence the tire rolling resistance
- The variation of the aerodynamic drag coefficient over the speed range of the test is negligible
- The variation of the aerodynamic drag coefficient with yaw angle can be adequately modeled with a 4-th degree polynomial in yaw angle
- The variation of the mechanical drag with vehicle speed can be adequately modeled with a 2-nd degree polynomial in speed

4.3. Vehicle instrumentation

The test vehicle was lightly instrumented to collect among other information: engine speed, vehicle speed, accelerator pedal position, manifold pressure, gear position, fuel flow rate, and wind speed & direction. The analog and digital channels were acquired with the vehicle CAN data using a Roush Industries LapDAQ data acquisition system. The LapDAQ system also served as the driver’s aid for operating the vehicle over the specified drive cycle.

Figure 9: FMTV Drivers aid & LapDAQ PC

The NovaLynx model 200-WS-02F Wind Sensor was used to measure the relative speed and direction of the wind. The sensor assembly consists of a wind vane mounted on top of a 3-cup anemometer. Because this style of wind measurement devise is not sensitive to vertical movement it’s a good fit for automotive testing where vehicles experience pitch and suspension travel during on-road testing.

The wind vane is coupled to a 20Kohms potentiometer with a linearity of ±1%. There is a gap of approx. 5 degrees (“dead band”) at the end of the resistor. The anemometer uses 3 magnets that activate a normally-open magnetic Reed switch. The approximate speed constant is 1.25mph/Hz.

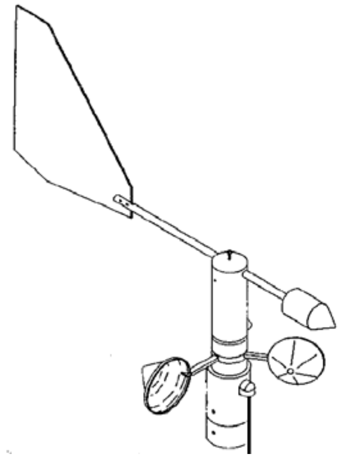


Figure 10: NovaLynx Wind Sensor

Figure 11: FMTV wind sensor installation

Specifications

Wind Vane	
Dead band	approx. 5°
Vane threshold	1.2mph
Measurement range:	0-360°
Azimuth accuracy:	± 3°
Anemometer:	
Maximum wind speed	125mph
Speed Threshold:	1.2mph
Measurement range:	0-99mph
Accuracy:	1mph (± 3%)

The wind sensor assembly was mounted approximately 3m in front of the vehicle and at roughly half its height (1.4 m) – see figure 11. This was done in order to place the wind speed/direction measuring device in an area where it is not influenced by the boundary layer created around the moving vehicle. The tripod-like frame that sustains the wind sensor assembly was made in such way that its contribution to the overall aerodynamic drag is minimized.

4.4. Drive cycle experimental testing and analysis

The drive cycle tests were performed on the oval track at the Chrysler Arizona Proving Ground in Yucca Arizona. The cycle was performed using a driver's aid system, LapDAQ as described in section 4.3, similar to that found in a vehicle emissions laboratory. The driver's aid system provides the vehicle operator a graphical scrolling vehicle speed target, with upper and lower limits, as well as current actual vehicle speed. This enables the driver to operate the vehicle according to the defined drive cycle.

Prior to each test the vehicle was preconditioned for at least thirty minutes. The preconditioning was done by running the vehicle at constant speed (50-60 km/h), in order to ensure that the tires and all the fluid temperatures were warm and stable.

After preconditioning the vehicle was brought to rest for approximately one minute before initiating the drive cycle driver's aid trace. The vehicle was driven according to the drive cycle until completion, approximately 7.5 miles. Three cycles were performed on February 17th and four cycles were performed on February 19th. February 18th was determined to be too windy for data collection.

The average fuel economy for the seven tests was 6.29 mi/gal with a minimum of 5.80 mi/gal and maximum of 6.62 mi/gal. The resulting standard deviation is 0.27 mi/gal & coefficient of variation (COV) of 4.31%. One of the data sets, 021709_0005, was somewhat of an outlier from the other six tests so was temporarily removed for a quick analysis. The average fuel economy for just the six tests was

6.38 mi/gal with a standard deviation of 0.176 mi/gal and coefficient of variation (COV) of 2.76%. Despite the improved STDEV and COV it was decided to use all seven tests for the study.

In addition to the fuel economy analysis the experimental data was mined for other vehicle component performance information to populate the analytical model. In many cases the required data was not available from the manufacturer. The following are two examples:

Caterpillar C7 BSFC map:
The part load BSFC map is one of the foundation requirements for the analytical GT-Drive model. Ideally, the part throttle and wide open throttle BSFC maps would be obtained directly from dynamometer data; however, these data were viewed as being proprietary in nature and were unavailable from Caterpillar, the engine manufacturer. To produce the BSFC map for this study, an alternate approach was employed with sufficient prediction accuracy utilizing the experimental data captured during the drive cycles over the various speeds and loads as described by the procedure below.

The key enabler for using the experimental data for creating the BSFC model was the filtering methods employed to identify and extract data near steady-state operating conditions. The filtering is considered important because preliminary data analysis revealed that during transient operating conditions the fuel flow rate measurement would lag actual engine consumption, based on observed torque, and that the acquired vehicle CAN data was also being broadcast at a rate that in certain situations could entirely miss a transient event (aliasing). Preliminary BSFC models that included transient operations helped identify this issue with the experimental data. Removing the transient data significantly improved the BSFC regression and produced a high quality model; results are shown below. The following describes the filtering and regression methods employed.

1. All seven of the experimental fuel economy data sets were combined into one large database containing 127,353 lines of data. This data set contained both steady state operating conditions, i.e. where torque, accelerator pedal position and engine speed remained relatively constant, and transient operating conditions where torque, accelerator pedal position and engine speed varied; typically during vehicle acceleration or deceleration.
2. Derivatives of accelerator pedal position (APP) and % torque actual (%TQ) were calculated. Rounding functions were applied, 3rd order was final choice, to minimize noise in the calculation, and for later data filtering use.
3. Data was filtered using various levels of cut-off for derivative of APP vs. time, and derivative of %TQ vs. time. It was determined that %TQ was the most appropriate filtering term with a cut-off derivative of 0.5. This reduced the data set size to 71,331 lines.
4. The data set was further reduced by removing deceleration events. Torque values less than 10% were removed to eliminate erroneous fuel flow measurements, many times zero, during deceleration events. Base engine idle in drive requires approximately 11% torque. This further reduced the data set size to 51,950 lines.
5. The data set was additionally filtered based on vehicle speed. The experimental data revealed some type of vehicle speed limit strategy being applied at vehicle speeds greater than 90 kph. When the speed limiter was applied it upset the requested torque and fuel flow calculations. For this reason all speeds of 90 kph and above were removed from the data set. This finally reduced the data set size to 47,774 lines ~ 35 % of original combined data set. This was still a significant data set for regression.
6. Regression analysis was performed resulting in a nine term polynomial with an Rsq of 92.3%. The regression results, analysis, and residual plots can be found in Appendix 1. Below is a graphical representation of the regression derived steady state BSFC map that was used for the simulation.

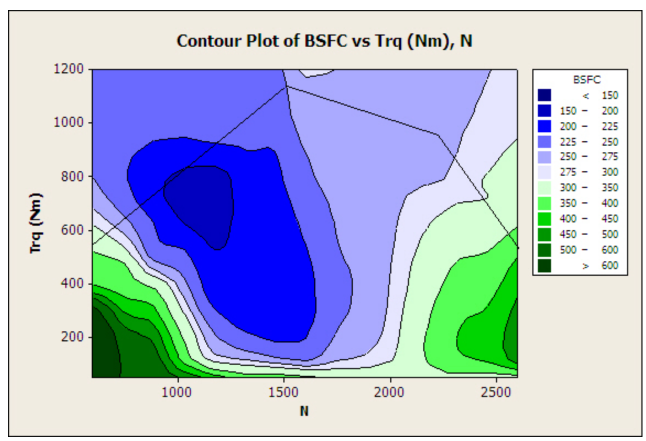


Figure 12: Caterpillar C7 BSFC map

Allison shift schedule:
Shift schedule data was not available from Allison to support this study. For the purpose of aligning and validating the base FMTV analytical model it was important for the

analytical shift schedule to match the experimental schedule. Additionally, prior to optimization of the shift schedule it was important to have a starting point which to work from. For these reason the experimental data was mined to determine the significant terms that affect the shift schedule and to determine a starting point for analytical simulation. The following describes the method used for determining the shift schedule.

1. All seven of the experimental fuel economy data sets were reviewed for transmission behavior. It was determined that one set would be adequate for this analysis as the shift points from test to test had very small variations. Data from experimental test 021709_0009 was selected.
2. This experimental data set was filtered to extract only the events when a change in gear position was being requested, not when the shift actually occurred which is delayed from request. This was done by comparing the current gear parameter to the selected gear parameter.
3. This shift event only data set was then split into two separate sets; one with just up-shifts and one with just down-shifts.
4. These data sets were further divided into specific gear-set and direction changes, e.g. all the 3rd to 4th gear up-shifts were put into one data set. This resulted in the following ten data sets: US_2-3, US_3-4, US_4-5, US_5-6, US_6-7, DS_7-6, DS_6-5, DS_5-4, DS_4-3, DS_3-2.
5. These individual US and DS data sets were evaluated to identify the significant terms that impact shift schedule. It was suspected that some form of engine load, percent torque, or driver demand would have a significant influence on the vehicle speed at which the shifts occurred or at a minimum show some trends. Evaluation of these data sets did not confirm this suspicion and in fact show very small influences due to engine load and in some cases showed conflicting load trends in some cases. For this reason it was determined that the shift schedule was largely based on vehicle speed alone.
6. The average vehicle speed at which the shifts occurred was determined by taking the average mean values from histogram plots of the individual shift sets.

Below is the resulting vehicle speed based shift schedule. This shift schedule was used for the base FMTV GT-Drive simulation. Example data is shown below and further supporting data can be found in Appendix 2.

Allison 3700SP Shift Scheduling

vehicle speed based shifting
shift values averaged from experimental data

Shift Event	up 1-2	up 2-3	up 3-4	up 4-5	up 5-6	up 6-7
Veh_Spd [kph]	skip	11.05	28.23	22.46	33.67	53.68
Shift Event	down 2-1	down 3-2	down 4-3	down 5-4	down 6-5	down 7-6
Veh_Spd [kph]	skip	8.72	16.95	24.05	31.99	47.30

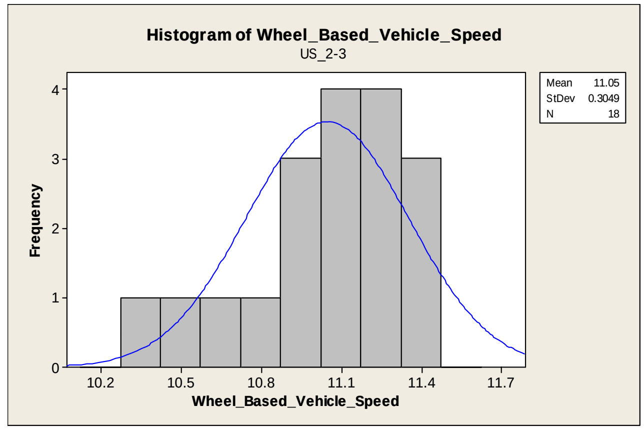


Figure 13: Example 2nd to 3rd gear up-shift

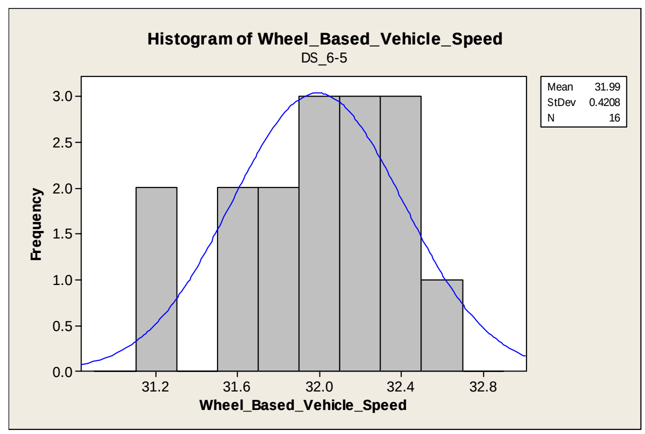


Figure 14: Example 6th to 5th gear down-shift

4.5. Coast down experimental testing and analysis

The coast down tests were performed on the oval track at the Chrysler Arizona Proving Ground in Yucca Arizona. On this track, the straight portion (section AB and CD in the picture below) is approximately 1.6km long and the radius of the curved portion is approximately 0.8km. There is no slope on either straight portion of the track. The curved portion instead shows a mild, constant slope of approximately 1:42; down-hill between D and A and up-hill between B and C.

According to SAE J2263 / Oct.1996 [5], the vehicle was preconditioned for at least thirty minutes before each coast down test. The preconditioning was done by running the vehicle at constant speed (50-60 km/h), in order to ensure that the tires and all the fluid temperatures were warm and stable.

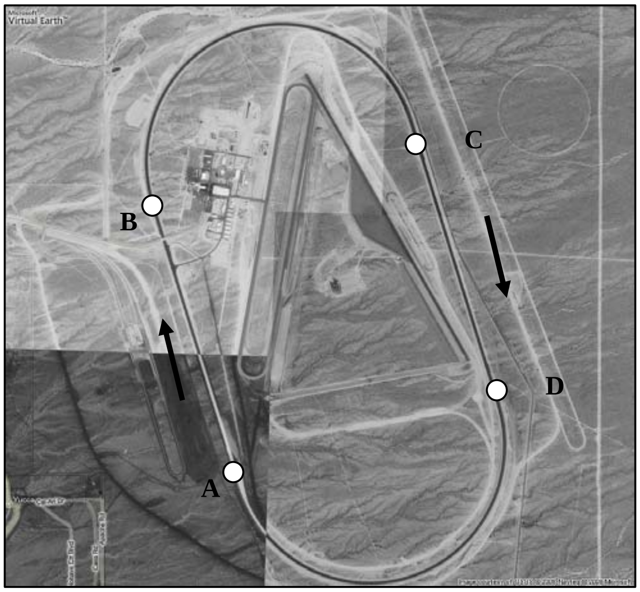


Figure 15: Chrysler APG aerial picture

Immediately after the preconditioning, the coast down test started. This consists in minimum 10 consecutive runs, each one starting at the beginning of the straight portion (points A and point C in the picture below). The vehicle was accelerated on the curved portion of the track to the maximum speed of 90kph and allowed to stabilize. Then, as soon as the beginning of the straight portion was reached, the transmission was set to neutral and the accelerator pedal was released. The vehicle was allowed to coast down until its speed reached approx. 10km/h. During this time the following parameters were recorded:

- Elapsed time
- Ambient air temp
- Wind direction in relation to the direction of travel
- Relative wind speed
- Vehicle speed (wheel based)

The recording was done continuously, and started at the beginning of preconditioning and ended at the end of the last coast down run. The sampling rate was 10 Hz. In addition to the parameters mentioned above, fuel economy related data and engine/vehicle key parameters were recorded as well (fuel flow, accelerator pedal position, selected gear, engine oil pressure, etc).

Approximately fifty coast down runs were performed in all with a minimum of 10 consecutive runs performed each day.

Weather - related data (ambient temperature, pressure, track temperature, relative humidity, wind speed and direction, etc.) was provided by the Yucca - Arizona Proving Ground meteorological station. The sampling rate was 0.1 Hz. For each day these values were averaged for the duration of the test and used to establish the correction factors.

Batch data post-processing was performed using a Matlab program specifically created for this project that follows the SAE J2263 procedures.

The first step in post-processing the raw data collected by the LapDAQ system was to eliminate the outliers. These are considered to be more than 3*StdDev away from the mean value of the dataset. The second step was to filter the data using a moving average filter. The post-processing program also compensated for the shift created by this filter. The third step was to eliminate the extreme data points. In this step, the following data points are eliminated:

- Points for which the absolute value of the yaw angle is greater than 20 deg (to eliminate side winds)
- Points for which the relative wind speed is smaller than 5 km/h (to avoid backwind conditions)
- Points for which the vehicle speed is less than 15 km/h and greater than 90 km/h.

The last step is to apply the correction factors to the mechanical and aerodynamic drag terms and to calculate the coefficients of the equation of motion. The coefficients are as follows:

$$K_m = 1 + 0.0081 \cdot (T - 20)$$ - mechanical drag

$$K_a = \frac{(273 + T)}{293} \cdot \frac{98.21}{p}$$ - aerodynamic drag

Where:

T - average ambient air temperature for the duration of the test, in degC

p - average ambient barometric pressure for the duration of the test, in kPa

The output of this Matlab program is a text file which contains the following:

Column 1: Name of the data file corresponding to one run
Column 2: Run's duration
Column 3: Number of points analyzed (post processing)
Column 4: Coefficient Am from the equation of motion
Column 5: Coefficient Bm from the equation of motion
Column 6: Coefficient Cm from the equation of motion
Column 7: Coefficient a0 from the equation of motion
Column 8: Coefficient a1 from the equation of motion
Column 9: Coefficient a2 from the equation of motion
Column 10: Coefficient a3 from the equation of motion
Column 11: Coefficient a4 from the equation of motion

In order to determine the coefficients of the equation of motion mentioned above, the program uses the Least Square method. As shown in Appendix 3, the program also creates a graphical representation of the pre and post-processed data. In these particular graphs the total drag is calculated based on the equation of motion (referred as "Polynomial Model") and also the mechanical, aerodynamic and gravitational drag for both raw and processed data.

The mechanical and aerodynamic drag as a function of vehicle speed for the experimental tests performed on February 17 and February 18, 2009 is illustrated in Appendix 4.

5. VEHICLE SYSTEM MODELING APPROACH

The software package used for the analytical simulation portion of this project was GT-Drive. GT-Drive is capable of very complex simulations that use thermo-fluid engine models that consider temperature and environmental factors. For this study a relatively simple approach was taken that uses steady state engine performance maps to determine fuel rate for a desired engine torque at a given engine speed. It has been shown [7] that fuel economy can be accurately predicted using simple steady state engine maps. GT-Drive can be configured to solve a vehicle model using two primary methods:

- Kinematic method – is a solve backward approach where the desired vehicle speed is forced on the model, the driveline condition is known, and therefore the engine is forced to provide the appropriate output. The modeled vehicle speed will exactly match the desired vehicle speed provided the engine is capable of meeting the demand at a specific engine speed.
- Dynamic method – is a solve forward approach where the desired vehicle speed is an input to a driver / PID controller which actuates the accelerator position therefore requesting torque from the engine which in turn produces a response through the driveline. This method is more realistic in that it controls the engine similar to the way an actual driver does and does not impose or directly control the driveline components. The tuning of the driver / PID determines how accurately the analytical vehicle follows the desired vehicle speed trace.

Although both kinematic and dynamic methods were evaluated it was determined that the dynamic method produced the most realistic results. Driver / PID tuning will be presented later in this report.

5.1. High level vehicle model

Along with steady state engine data and the vehicle speed controller the GT-Drive model requires significant driveline information for accurate simulation. Following is a list of the key information areas: clutch or torque converter object performance tables; transmission object with gear ratios, inertias, efficiencies and shift schedule; transfer case object with gear ratio, inertias and torque split percentage between front and rear; drive shaft and axle objects with inertias; final drive objects with gear ratio and inertias; tires with radius, rolling resistances and any traction-limit data; brake objects with braking torque maps and any other data related to vehicle resistances such as aerodynamic loading.

Other control components were added to the model to simulate an idle speed controller and a simple fuel shutoff control during deceleration events; producing more accurate simulation of actual operating conditions. The model does not account for any hardware limitations beyond those described by the performance tables. The following two pages provide graphical representations of the FMTV powertrain layout and the applicable GT-Drive model objects and connections used to simulate this powertrain.

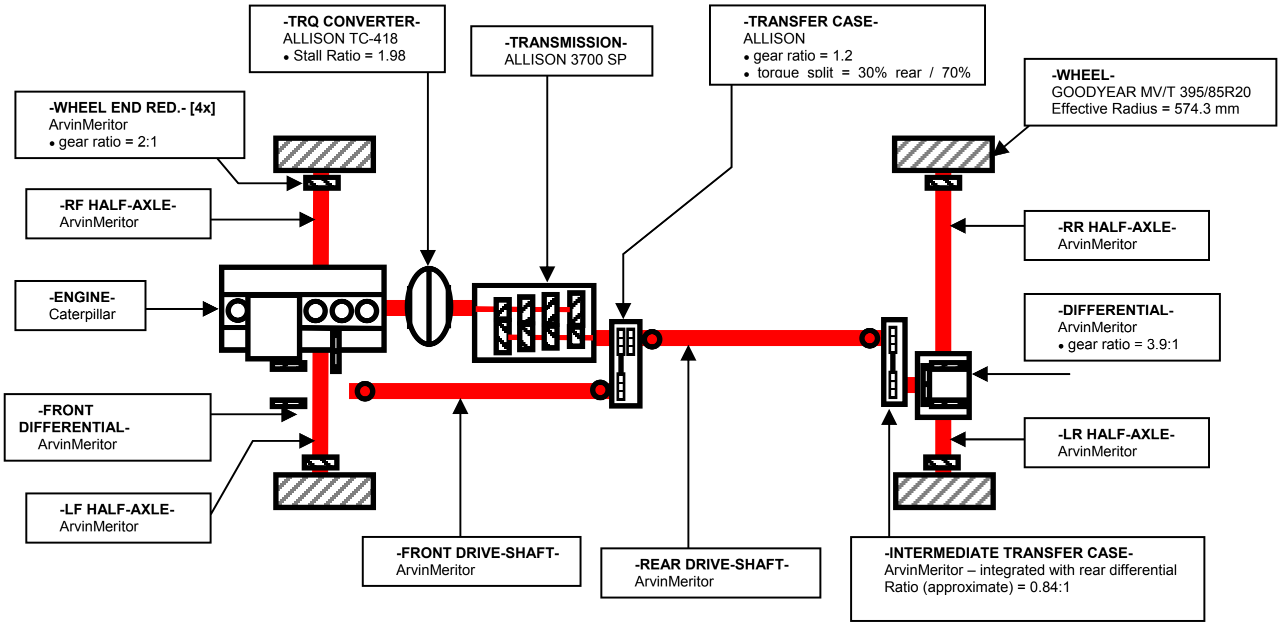


Figure 16 – FMTV-M1078 Powertrain Layout

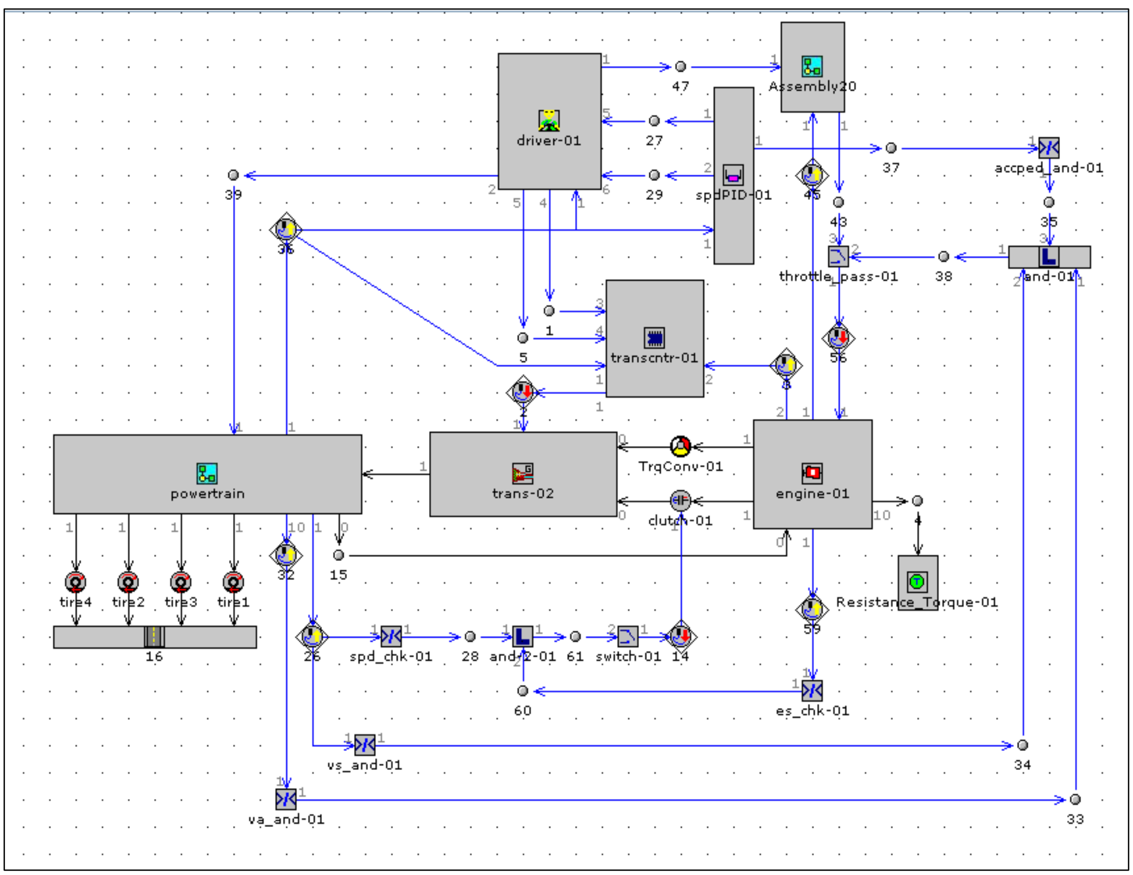


Figure 17 - GT-DRIVE model layout

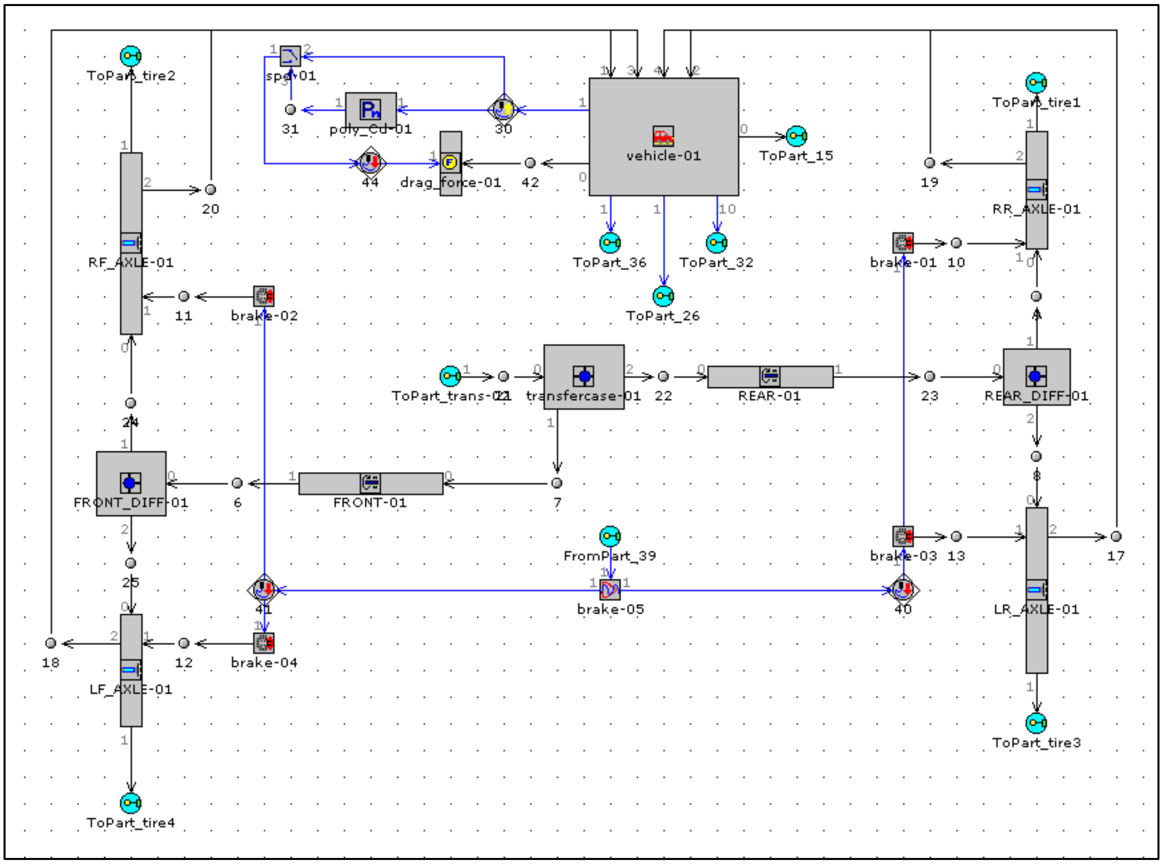


Figure 18 - GT-DRIVE model, powertrain layout

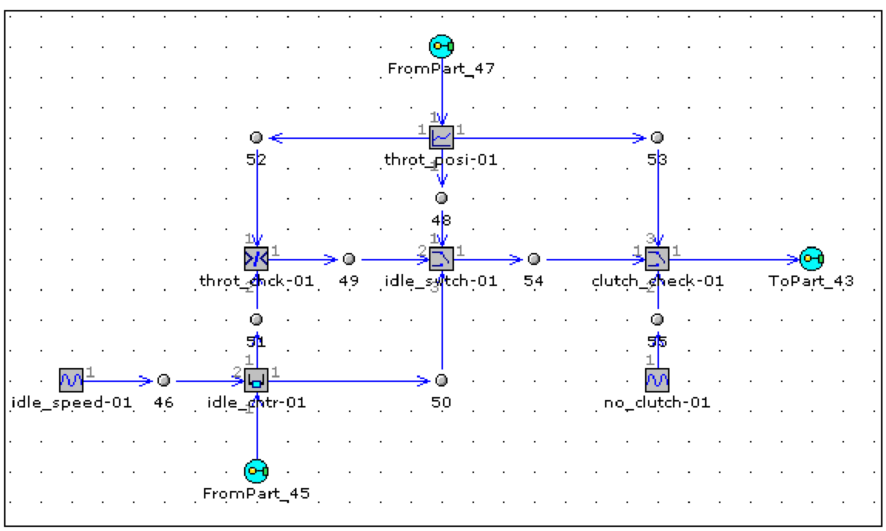


Figure 19: GT-Drive engine idle controller

5.2. Model parameterization

The GT-Drive model was parameterized based on data from both published and experimental sources. AMT data was obtained directly from the manufacturer. In some cases object data was not available and therefore needed to be based on similar objects, inference, or engineering judgment. The following two sections describe the source, method and reasoning behind the information used to parameterize the GT-Drive model objects.

5.2.1. Published and experimental data sources

Base FMTV dimensions
- Armor Holdings cut sheet for FMTV-M1078 [1]
- Physical measurements taken to support frontal area and MOI calculations

Engine – Caterpillar 330 hp C7, serial#: FML07676
- Full load torque data provided by CAT dealer
- Part load torque and BSFC/fuel rate maps derived from experimental data

Torque converter TC-418
- Stall ratio from Allison cut sheet [2,4]
- Torque / speed ratio was inferred based on combination of known and published data [2, 9]

Allison transmission – 3700SP
- Gear ratios from cut sheet and confirmed with experimental data [2]
- Efficiency data interpolated / extrapolated from Eaton provided data [11]
- Inertia data interpolated / extrapolated from ZF AS Tronic manual [12]

Transfer Case
- Select pages from Allison 3700SP manual [4]
- Physical measurements to support MOI calculations and to verify gear ratios

Axles, drive shafts, final drive, and tire/wheel
- Armor Holdings cut sheet for FMTV-M1078 [1]
- Physical measurements
- Axle efficiencies estimated based upon engineering judgment and published data [9]
- Tire / wheel data obtained from manufacturers' website and physical measurements

ZF AS Tronic transmission
- Gear ratios from AS Tronic manual [12]
- Efficiency data interpolated / extrapolated from Eaton data [11]
- Inertia data from AS Tronic manual [12]
- Clutch defined using surrogate GT-Drive values [10]

Eaton UltraShift *PLUS* Transmission
- Gear ratios provided by manufacturer [11]
- Generic 10 speed AMT efficiency data provided by manufacturer [11]
- Inertia data interpolated / extrapolated from ZF AS Tronic manual [12]
- Clutch defined using surrogate GT-Drive values [10]

5.2.2. Data source details

Caterpillar C7 BSFC map:
See section 4.4 and Appendix 1.

Torque converter TC-418:
Torque / speed performance data was not available from Allison. Allison cut sheet included the stall ratio for the TC-418 converter. This value was compared with typical torque converter characteristics [9] data to create an approximate curve as shown below. This was deemed acceptable since the experimental data showed relatively quick torque converter lockup once the vehicle speed reached approximately 5 kph. Considering the residency time in this low speed region is minimal it was deemed adequate to characterize the torque converter using this method.

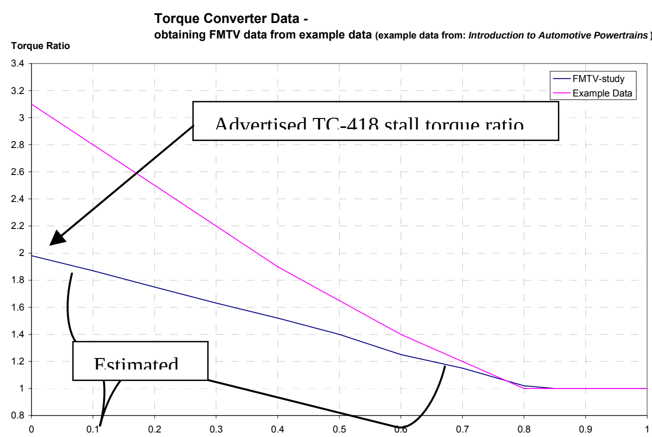


Figure 20: Torque converter performance curve

Transmission inertia and efficiency:

Gear inertia data was provided for the ZF transmission [12]. Gear efficiency data was provided [11] for the Eaton transmission. Based upon gear ratio (x-axis), these values were interpolated and extrapolated for the other transmissions. The inertia numbers are so small relative to the rest of the powertrain that even significant errors in this area have insignificant influence on the analytical fuel economy values.

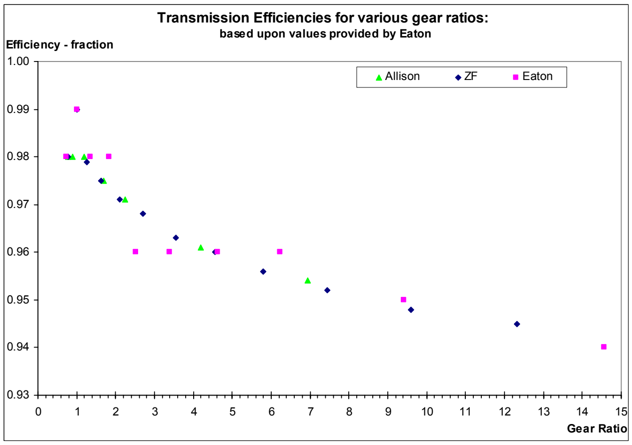


Figure 21: Transmission efficiency characteristics

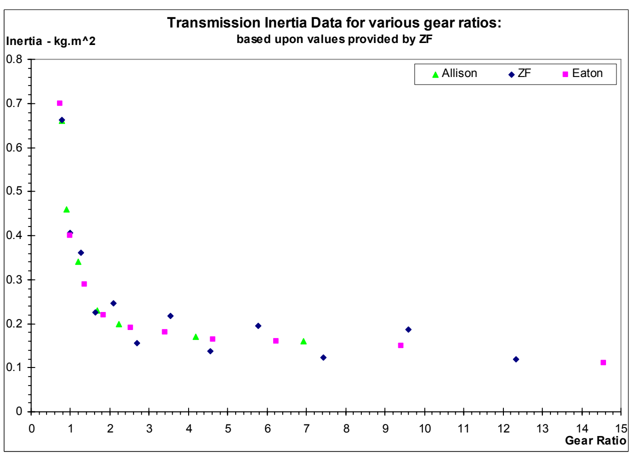


Figure 22: Transmission inertia characteristics

Allison transmission shift schedule:

See section 4.4 and Appendix 2.

Transfer case / drive shafts / axles and final drive:

All components were physically measured to determine reasonable MOI approximations. Material properties and hidden component dimensions, eg. drive shaft wall thickness or gear dimensions, were assumed based on engineering judgment and housing dimensions.

Final drive ratio, intermediate (rear) transfer case:

The M1078 specifications sheet [1] described the final drive ratio as 3.9:1 plus a 2:1 reduction at the wheels, making for an overall ratio of 7.8:1. After an initial investigation of the actual powertrain layout, physical measurements of under-vehicle housing sizes, and on-road data collection of engine speed versus vehicle speed, the overall final drive ratio was determined to be further reduced by the Rear Transfer Case by a 0.84:1 ratio.

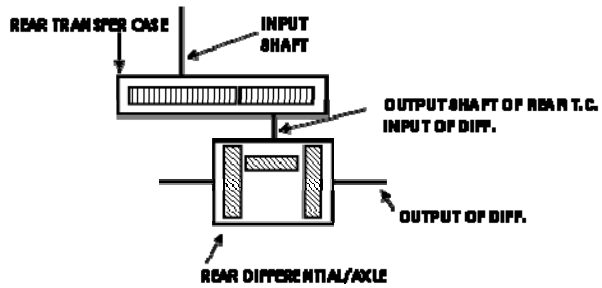


Figure 23: Rear transfer case and differential

Wheels and end reduction:

Wheel end reduction, 2:1, is part of the Arvin Meritor axle assembly. No direct data was obtained from the manufacturer. Reduction gear sizes for the inertia calculation were based upon the measured hub housing diameter, estimated thickness and estimated clearance.

Wheel and tire information was obtained from the manufacturer; Goodyear MV/T 395/85R20. Measurements of the test vehicle wheel were also taken.

Moment of inertia calculations:

Based upon the powertrain component measurements and gear ratios an MOI calculation was done to estimate effective mass (linear mass plus rotational mass).

Based upon a linear mass of 8402 kg, a combined rotational mass of 229.6 [kg-m^2], and an effective wheel rolling radius of 574 mm; the total calculated mass of the vehicle is 9099 kg. This value does not include the rotational mass of the transmission or the engine. The value of effective mass calculated above was used for calculating road-load coefficients during coast-down testing per SAE J2263.

The relative importance of the individual inertia values was evaluated to gain an appreciation for the analytical fuel economy sensitivity to potential errors. As an example the model was run with an error in the transmission inertia; 1 kg-m^2 was changed to 200 kg-m^2. This change resulted in less than 0.3% difference in overall fuel use. A study was also conducted to determine if there was any difference between using a lump-at-wheels, also called reduced-to-wheels, approach versus a distributed-among-components approach to modeling inertia. No difference between the methods existed.

This result may have been due to the fact that this vehicle has a large percentage of the actual powertrain inertia existing in the wheels. From this study we were able to conclude that inaccuracies in the small object inertia values would not have a significant impact on fuel consumption.

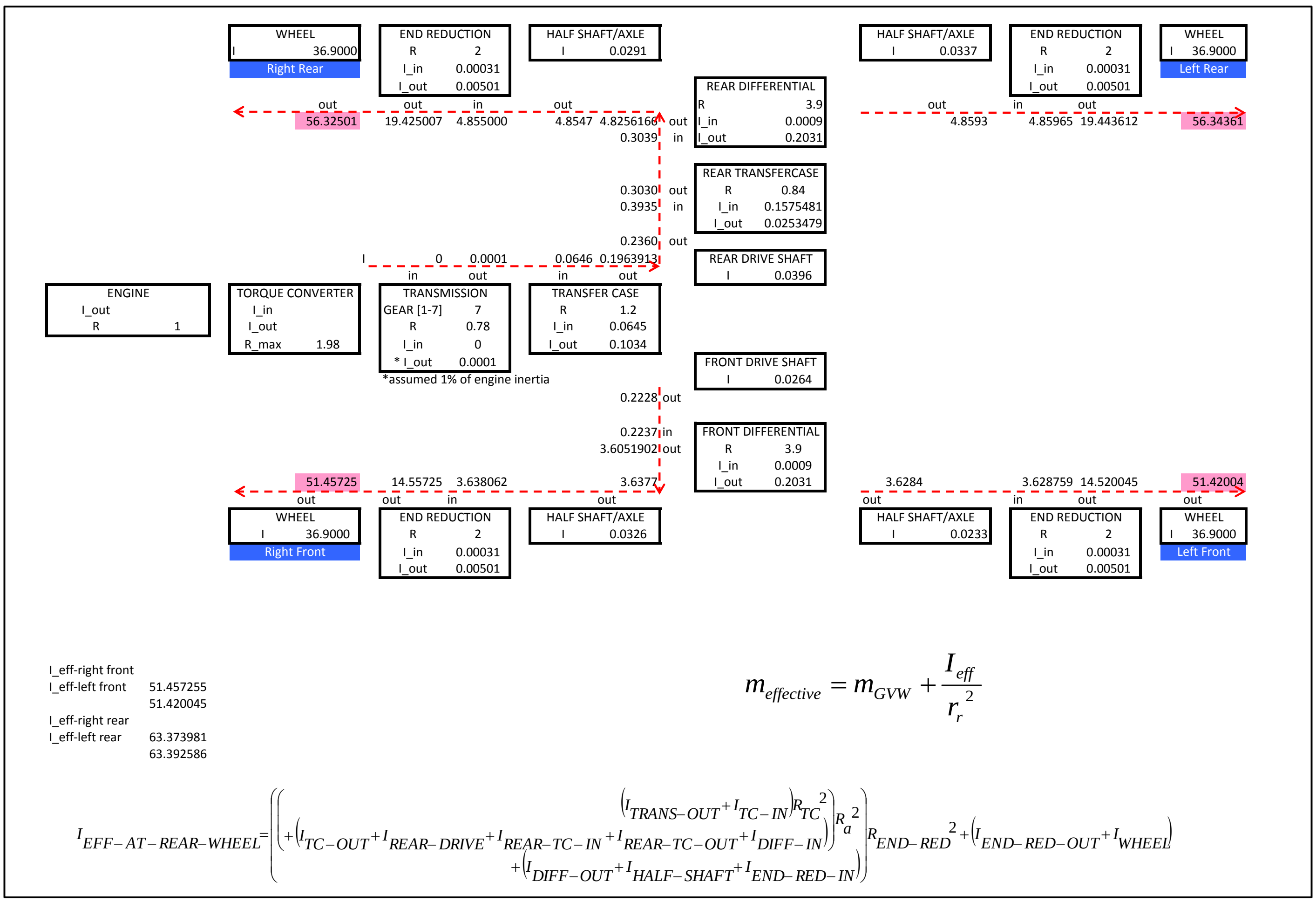


Figure 24: MOI Calculation

6. ANALYTICAL TO EXPERIMENTAL BASELINE MODEL CORRELATION

Creation of the base FMTV analytical model and correlation to the experimental data was the first step in the modeling process. After vehicle model parameterization it was necessary to ensure the analytical model closely matched the experimental results. Three primary areas were investigated to tune, correlate and validate the analytical model: road load, shift schedule over cycle, and engine speed & torque and vehicle speed over cycle. The following sections address these three areas.

6.1. Road load correlation

If the analytical road load was different from the experimental road load it would lead to significant inaccuracies in the fuel economy calculations as the required engine torque to move the vehicle through the virtual cycle would be higher or lower than actual required. To validate the road load parameterization analytical coast-down tests were performed. The results of the simulated (analytical) coast-down runs are shown below. Included are the coast-down time comparisons from 90 kph to 20 kph and a two graphs showing time vs. vehicle speed for runs with the least and most amount of error in coast down time.

Runs, 2-17-2009			ERROR	
	Experimental [sec]	Analytical [sec]	Δ TIME: (ANLYT - EXP) [sec]	Δ TIME / ANLYT %
run1	75.8	76.672	0.872	1.1%
run2	84.3	79.104	-5.196	-6.6%
run3	74.1	72.704	-1.396	-1.9%
run4	78.8	77.056	-1.744	-2.3%
run5	76.6	76.928	0.328	0.4%
run6	77.1	75.648	-1.452	-1.9%
run7	73	75.264	2.264	3.0%
run8	75.8	73.856	-1.944	-2.6%
run9	72	67.328	-4.672	-6.9%
avg_1-9		75.392		

Figure 24: 02-17-09 coast down comparison

Runs, 2-18-2009			ERROR	
	Experimental [sec]	Analytical [sec]	Δ TIME: (ANLYT - EXP) [sec]	Δ TIME / ANLYT %
run2	80.2	77.568	-2.632	-3.4%
run3	72.9	74.88	1.98	2.6%
run6	76.5	74.24	-2.26	-3.0%
run7	71	74.624	3.624	4.9%
run8	80.2	78.464	-1.736	-2.2%
run9	69.7	73.984	4.284	5.8%
run10	79.5	76.672	-2.828	-3.7%
run11	68.3	73.344	5.044	6.9%
run12	81.4	77.696	-3.704	-4.8%
run13	68.4	74.624	6.224	8.3%
Avg_3,6-13		75.136		

Figure 25: 02-18-09 coast down comparison

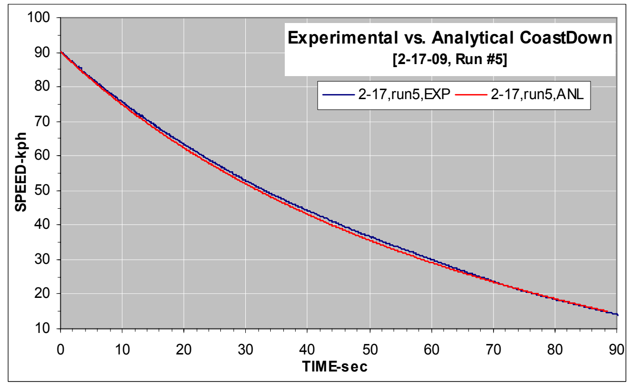


Figure 26: 02-17-09 Run#5 experimental vs. analytical coast-down

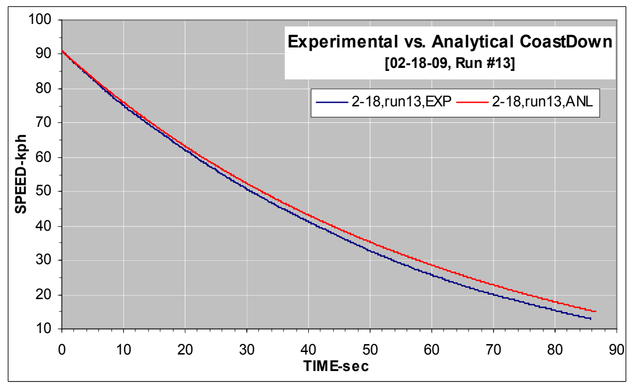


Figure 27: 02-18-09 Run#13 experimental vs. analytical coast-down

As illustrated above, the error between the analytical and experimental data can vary based on the comparator experimental data set chosen. The experimental variation was due primarily to the difference in wind, side and head or tail, present during each experimental run. Keep in mind that the analytical coast-down runs assume ideal environmental conditions. No experimental test was performed with ideal environmental conditions so any comparison of experimental to analytical results need to bear in mind the environmental conditions present during the experimental run.

The test with the best environmental conditions was run#5 from 02-17-09. This run had the smallest amount of overall error with only 3% of the points outside the ±5 deg yaw-angle range and 17% outside the relative wind speed range of ±5 kph. Sections 4.2 and 4.5 describe the calculations and filtering methods used to account for varying environmental conditions when determining the coefficients of the equation of motion used for the simulation.

After reviewing the data and the relative wind differences during the runs, it was determined that coefficients derived from experimental runs made on 02-17-09 would have the least amount of correction due to environmental conditions. The average of the filtered coefficients from runs on 02-17-09 resulted in the following road load terms: A=819.135 [N], B=18.3576 [N/kph], C=0.30751 [N/kph^2]. These values were used in the model to run all FTP72 fuel economy cycles.

6.2. Shift schedule correlation

Baseline shift scheduling needed to match the experimental data for correlation. Transmission calibration allowed for starting in 2nd gear, thus 1st gear was removed in the analytical model. Experimental data was statistically evaluated in each gear to find the mean speeds at which up shifting and down shifting occurred, see section 4.4 of this report. Below are the tabular and graphical versions of this shift schedule.

Allison 3700SP Shift Scheduling

vehicle speed based shifting
shift values averaged from experimental data

Shift Event	up 1-2	up 2-3	up 3-4	up 4-5	up 5-6	up 6-7
Veh_Spd [kph]	skip	11.05	28.23	22.46	33.67	53.68
Shift Event	down 2-1	down 3-2	down 4-3	down 5-4	down 6-5	down 7-6
Veh_Spd [kph]	skip	8.72	16.95	24.05	31.99	47.30

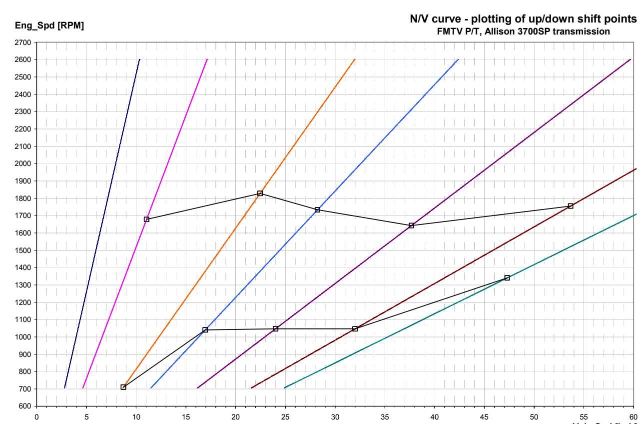


Figure 28: base Allison shift schedule

For the initial correlation the above shift schedule was implemented into the GT-DRIVE model and run over the FTP72 drive cycle. After correcting a few simple model errors it was determined that the analytical shift schedule aligned reasonably well with the experimental data. Experimental data set 021909_0003 was used for this comparison as it was determined to be a reasonable experimental average test. The lower part of the figure below compares the experimental and analytical gear positions.

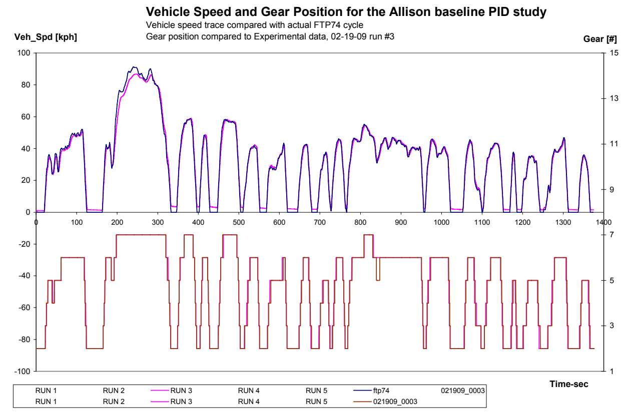


Figure 29: analytical vs. experimental gear position

6.3. Driver PID tuning & cycle correlation

With confidence in the road load simulation and in the shift schedule it was time to begin tuning the vehicle driver to provide similar engine torque, engine speed and vehicle speed behavior as the experimental data. Initial results revealed fluctuations in torque and fuel consumption rate that did not appear to match the experimental data and generally looked noisy or too busy. These results are shown on the next page. These fluctuations were being introduced by the model controller as confirmed by reviewing the analytical accelerator position over the cycle. Driver tuning was the obvious next step.

DoE (design of experiments) methods were used to evaluate the model response to various PID configurations. The primary goals were to reduce torque fluctuations, i.e. smooth the analytical torque, and to provide the best vehicle speed trace matching. Preliminary analytical vs. experimental results supported the PID values of 10, 0.1, 0. Upon further investigation, the PID values of 40, 0.4, 0 were chosen for the remainder of the study. This larger gain (aggressiveness factor) produced better vehicle speed matching. Torque smoothing was not as good as with the 10, 0, 0 PID but still aligned well with the experimental data. The results of applying these PID values are shown below.

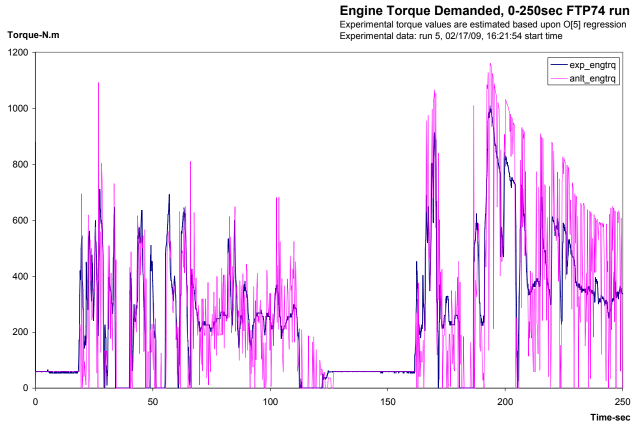


Figure 30: Analytical torque fluctuations vs. experimental torque

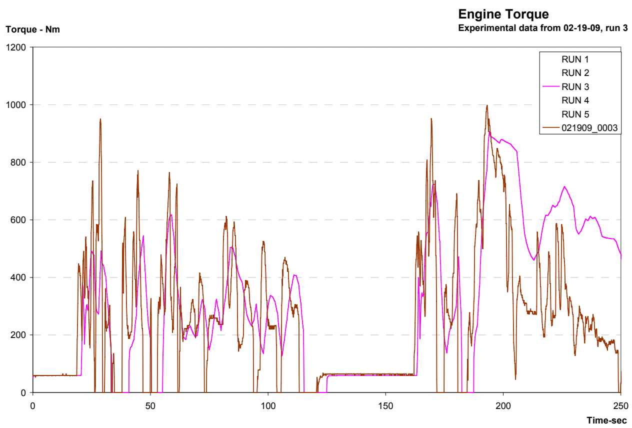


Figure 31: Engine Torque, comparing experimental and analytical w/ PID (10-0.1-0) control

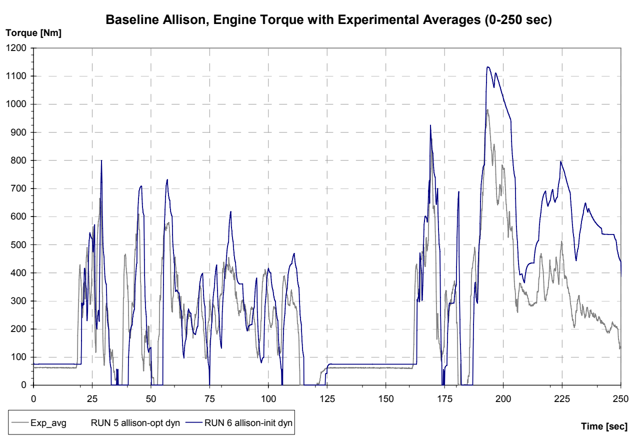


Figure 32: Engine Torque, comparing experimental and analytical w/ PID (40-0.4-0) control

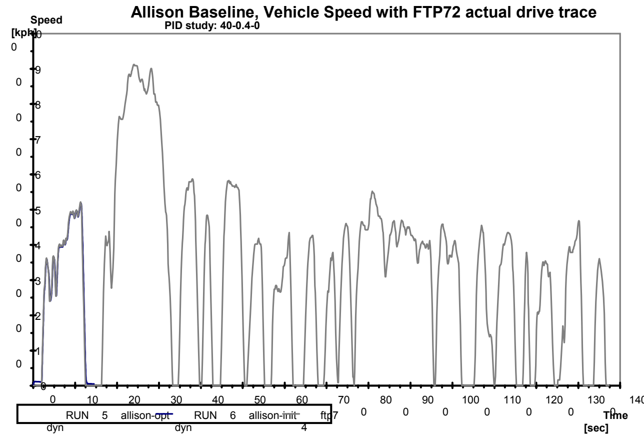


Figure 33: Engine speed trace, comparing experimental and analytical w/PID (40-0.4-0) control

6.4. Baseline model results and comparison

With the mature road load characterization, matching shift schedule, and tuned driver PID the next step was to compare the analytical to experimental fuel consumption. An experimental mean comparator data set was created for this comparison that was simply the average of all experimental tests. The results of this comparison are detailed below.

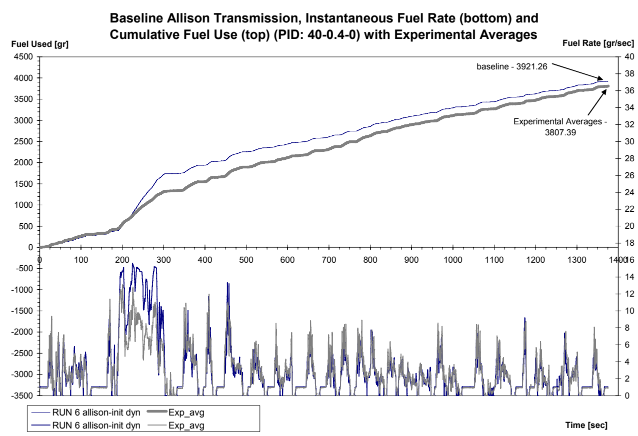


Figure 34: Cumulative fuel consumption & torque variation, comparing experimental and analytical

The end of cycle difference between experimental and baseline analytical model is 114 grams. This translates into a fuel economy difference of ~ 0.2 mi/gal as the analytical result of 3921.26 grams translates to 6.10 mi/gal and the experimental average of 3807.39 grams translates to 6.29 mi/gal. The analytical fuel economy result falls within the variation of the experimental data; recalling from section 4.4

that the minimum experimental result was 5.80 mi/gal and maximum was 6.62 mi/gal.

The above figure illustrates that the cumulative fuel consumption of the experimental average and analytical results have significantly different slopes in the 200 to 300 second range. The root of this problem is illustrated in figures 31 and 32 where it's clear to see that in the 200 to 300 second range the experimental torque is considerably less than the analytical torque. Despite significant efforts this torque difference was not resolved. The one distinction that separates this 200 to 300 second range from the rest of the drive trace is vehicle speed. This is the highest vehicle speed portion of the trace where speeds approach 90 kph. The natural thought would be a road load error at higher vehicle speeds but even with drastic reductions in higher speed road loads the required analytical torque was still appreciably higher than the experimental average. Although not confirmed the two areas suspected for introducing this error are inaccuracies in the ECU data that was collected during experimental testing or some unknown engine or powertrain characteristics that significantly improve efficiency under these operating conditions. Regardless, the higher required torque in this region is also present in the simulations of the transmission variants, as will be shown in the next section, so this torque difference does not impact the comparison results.

7. APPLICATION OF TRANSMISSION VARIANTS

With the baseline FMTV-Allison model created and correlated to the experimental results the next step is the application of the ZF AS Tronic 12 speed and the Eaton UltraShift *PLUS* 10 speed to the base model. After the transmission parameterization described in section 5.2 the next most influential element of the transmission objects are the shift schedules. Described below is the method used to define the shift schedules and the subsequent results. Additionally presented below are the fuel economy results for each transmission variant.

7.1. Shift schedule optimization

With the help of experts at GTisoft [10] an optimizer method was created for targeting minimum fuel consumption. With this method we assume shifting is solely a function of vehicle speed. This is a limiting assumption which may provide somewhat unrealistic results or contain shifting points that conflict with drivability and calibration limitations. The results of this vehicle speed based method should be interpreted as the ideal shifting schedule only from the point-of-view of best fuel economy. To help understand the magnitude of potential shift schedule and fuel economy differences an analysis was performed using the Allison transmission to compare the differences between an optimizer derived shift schedule and the experimentally derived shift schedule. The results of this comparison will be shown in this section and also used for explaining the optimizer.

To start the optimizer initial values must be defined for all up and down shift points. For the Allison transmission this shift schedule was already determined by statistical averaging of the experimental data. For the other transmissions, without defined shift schedules, starting schedules were created largely based on the Allison shift schedule.

Setting the target parameter to minimum fuel consumption over the total cycle, the program systematically varies the shift schedule one gear at a time over the possible range of values. This range is constrained by the neighboring up and down shift points and min and max vehicle speeds in the specific gear. The program finds the vehicle-speed-based shift point within the specified range that results in the lowest total fuel consumed. The optimizer routine then moves onto the next gear. It does not replace the initial value with the, newly found, optimized value.

During the optimization of some gears the optimizing function is relatively flat. In other words, over a good portion of the possible vehicle speed range the total fuel consumed varies within 0.5-1.0 gram. The nature of certain gears having very insignificant influence on fuel economy means that in the real world these shift points could be modified for drivability or similar without having a measurable impact to fuel economy. The figure on the next page is an example of a flat optimization function.

After a single optimization iteration the whole set of up shift and down shift points have been evaluated with the assumption that changing one gear ratio will not affect the fuel consumption in other gears – assuming gear independence. This is a limiting assumption, but that is why the set is then re-iterated; changing all initial values to the new optimized values. The iterations continue until points converge; this being more a subjective measurement of convergence as opposed to an exact number. Convergence typically appeared after the third or fourth iteration of the optimizer. As an example the Allison transmission optimization points converged with three iterations. Results are shown in the figure on the next page.

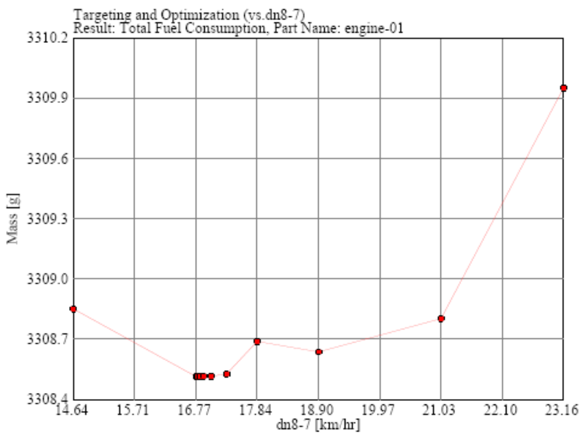


Figure 35: Shift schedule optimization, flat optimizing function

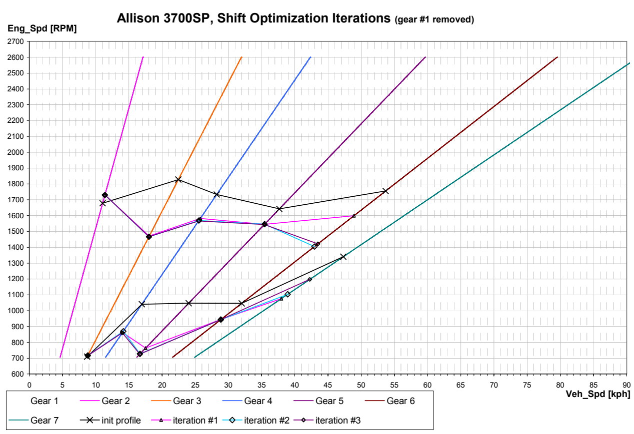


Figure 36: Allison shift schedule optimization - graphical

Iteration #3 Results			
shift event	kph	shift event	kph
2-3 up	11.392	3-2 down	8.806
3-4 up	18.027	4-3 down	14.012
4-5 up	25.514	5-4 down	16.652
5-6 up	35.411	6-5 down	28.860
6-7 up	43.481	7-6 down	42.256

Figure 37: Allison shift schedule optimization - tabular

Because the transmission variants are twelve and ten speeds they have lower gears ratios than the Allison. These transmissions are designed to be used with lower final drive ratios. For simplicity sake the initial optimizations were performed using the base FMTV final drive ratio. Later they will be re-optimized with a revised final drive ratio. Considering the high final drive ratio of the base FMTV it was necessary to remove several of the lower gears as they would effectively be super creeper gears. For the ZF transmission gears 1-4 were skipped, for the Eaton transmission gears 1-3 were skipped.

The initial ZF and Eaton shift schedules were based on interpolated & extrapolated versions of the Allison experimentally derived shift schedule. Following are the ZF and Eaton optimization runs and resulting shift schedules.

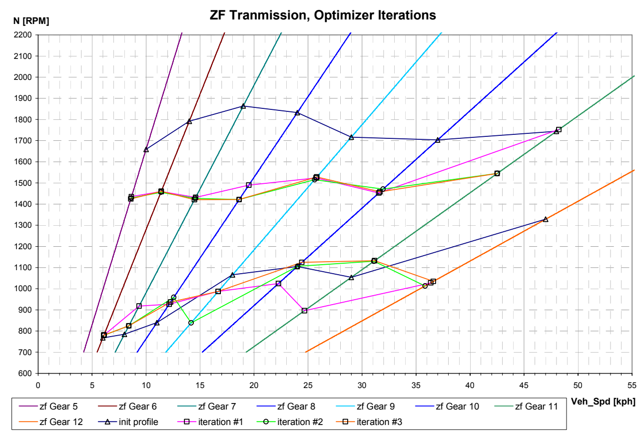


Figure 38: ZF shift schedule optimization - graphical

iteration #3	shift event	kph	RPM
	u5-6	8.5895	1424.691
	u6-7	11.4122	1460.391
	u7-8	14.485	1420.999
	u8-9	18.6153	1421.049
	u9-10	25.8159	1527.989
	u10-11	31.6605	1457.493
	u11-12	42.5195	1544.900
	d6-5	6.09726	780.254
	d7-6	8.40668	824.708
	d8-7	12.2721	936.818
	d9-8	16.6811	987.318
	d10-9	24.4242	1124.368
	d11-10	31.1626	1132.260
	d12-11	36.6012	1034.634

Figure 39: ZF shift schedule optimization - tabular

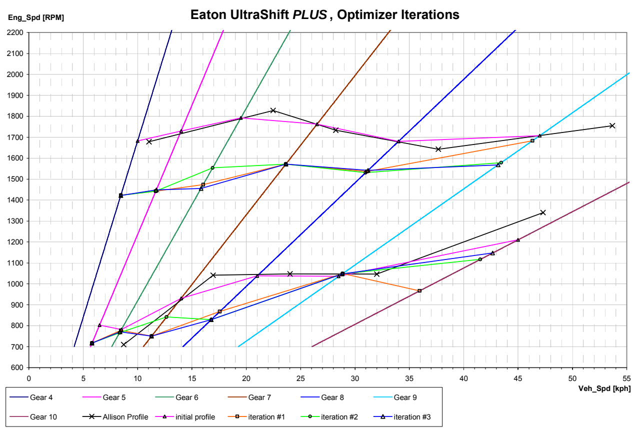


Figure 40: Eaton shift schedule optimization - graphical

iteration #3	shift event	kph	RPM
	u4-5	8.451	1421.711
	u5-6	11.728	1448.791
	u6-7	15.833	1455.458
	u7-8	23.639	1571.796
	u8-9	31.201	1541.770
	u9-10	43.160	1568.189
	d5-4	5.800	716.505
	d6-5	8.381	770.435
	d7-6	11.278	749.918
	d8-7	16.771	828.712
	d9-8	28.822	1047.205
	d10-9	42.659	1146.968

Figure 41: Eaton shift schedule optimization - tabular

7.2. Fuel economy results comparison

The optimized shift schedules were run over the FTP72 drive cycle. Presented below are comparisons of the Allison experimental average data set, experimentally derived shift schedule and optimized shift schedule. Additionally there is a separate comparison of the optimized Allison, optimized ZF and optimized Eaton.

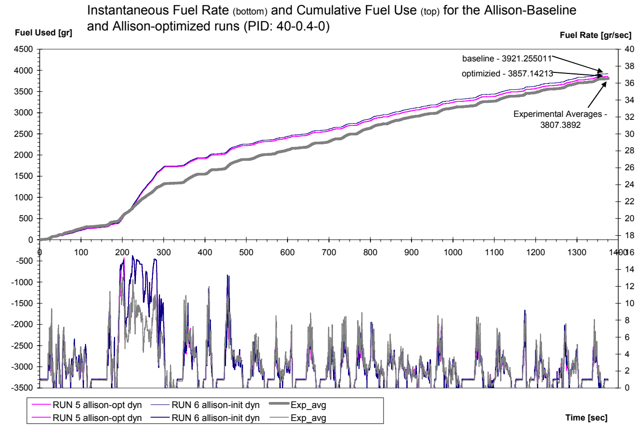


Figure 42: Allison fuel consumption comparison – exp. vs. shift schedule variation

Results indicate a 1.64% (64 gram) end-of-cycle improvement between the Allison experimentally derived shift schedule and the Allison-optimized shift schedule. To ensure a fair comparison of the transmissions the optimized version of the Allison was used for comparison with the ZF and Eaton variants.

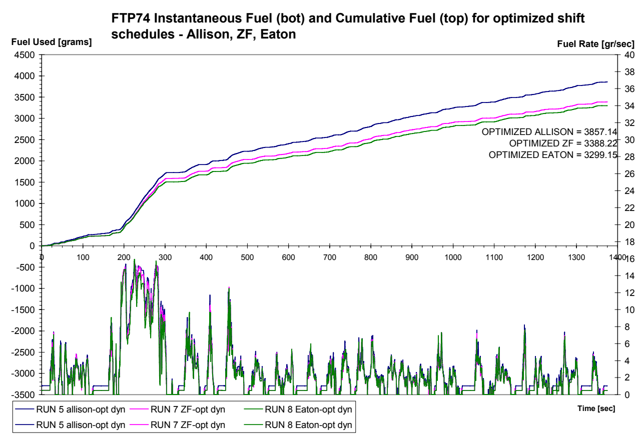


Figure 43: Fuel consumption comparison – Allison opt. vs. ZF & Eaton

Optimized Shft Schd	Total Fuel Used [g]	Δ FUEL [g]	% difference
Allison	3857.14	-	-
ZF	3388.22	468.92	12.16%
Eaton	3299.15	557.99	14.47%

Figure 44: Fuel consumption comparison – Allison opt. vs. ZF & Eaton - tabular

Compared with the optimized Allison the ZF shows a 12.2% improvement and the Eaton shows a 14.5% improvement in fuel consumption. This translates to 0.86 mi/gal improvement with the ZF and 1.05 mi/gal improvement with the Eaton over the FTP72 drive cycle. Keep in mind that these numbers were generated with the base FMTV final drive ratio and by skipping the first several gears in both the ZF AS-Tronic 12 speed and the Eaton UltraShift *PLUS* 10 speed transmissions. Changing the final drive ratio will have a significant influence on these numbers.

7.3. Final drive ratio change

To take advantage of the wider operating range offered by the ZF and Eaton transmissions it was necessary to change the vehicle final drive ratio. The new final drive was selected by maintaining the current max vehicle speed of 90 kph, selecting an efficient portion of the engine map (1200 rpm) and using an average of both final transmission gears (average of .778 and .74 = 0.76). With these assumptions we can calculate the desired final drive ratio.

$$\frac{N}{V} = \frac{(2652.5824) * R_{TOTAL}}{R_R} = 13.333\left[{}^{RPM}\!/_{KPH}\right]$$

[2652.5824 comes from unit conversion constants]

$$R_{TOTAL} = R_{FINAL\_DRIVE} * R_{TRANSMISSION} * R_{TRANSFER\_CASE} * R_{REAR\_T.C.}$$

$$R_{TRANSMISSION}(last\_gear\_average) = 0.76$$

$$R_{TRANSFER\_CASE} = 1.2$$

$$R_{REAR\_T.C.} = 0.84$$

$$R_R = 574[mm]$$

$$\frac{\frac{N}{V} * \frac{R_R}{(2652.5824)}}{R_{TRANSMISSION} * R_{TRANSFER\_CASE} * R_{REAR\_T.C.}} = R_{FINAL\_DRIVE} = 3.766$$

For the GT-DRIVE model, rear transfer case (REAR_TC) and final drive were combined into one value/parameter (the final drive ratio in the model). This value is then equal to:

$$R_{GT-DRIVE\_FINALDRIVE} = R_{FINAL\_DRIVE} * R_{REAR\_T.C.} = 3.766 * 0.84 = 3.163$$

Upon investigation of the final drive ratio it was determined that the tractive power available in the highest gear (~0.76) with the 3.766 final drive was too close to the road-load curve, see figure on next page. Although this level of power would be sufficient for moving the vehicle down the road it was determined that its proximity to the road-load curve would cause frequent down shifts with increased grade, cargo weight, frontal wind or similar loads. For this reason the final drive was adjusted up to a value of 4.50 (effective ratio ~ 3.78) to provide a more realistic analysis. This final drive adjustment moves the tractive power available in the highest gear (~0.76) to a more reasonable location in reference to the road load curve providing a greater torque reserve for additional loads.

Regarding first gear / creeper gear operation the current baseline FMTV with Allison has a first gear N/V of 251. With the 4.50 FDR the Eaton would have a first gear N/V of 305 and the ZF would have a first gear N/V of 258; both surpass the baseline Allison. It could be argued that the Eaton should be evaluated with a numerically smaller FDR to better match the first gear N/V of the baseline FMTV Allison and to take advantage of potential further fuel economy improvement. For transmission comparison sake and simplicity this FDR study was performed using the single 4.50 FDR.

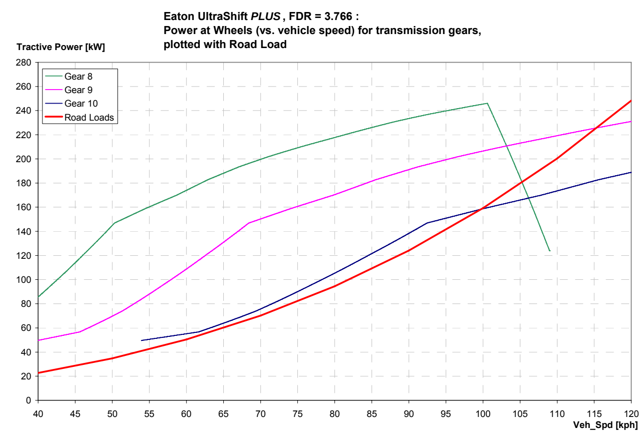


Figure 45: Road load vs power at wheels for Eaton with FDR = 3.766

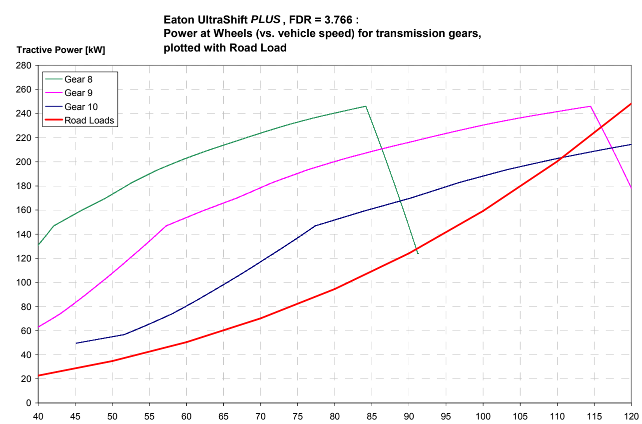


Figure 46: Road load vs power at wheels for Eaton with FDR = 4.5

With the new final drive ratio in place the ZF and Eaton transmission models were updated to activate all gears. It was then necessary to re-run the shift schedule optimizer to take advantage of the new final drive ratio and the full set of gears.

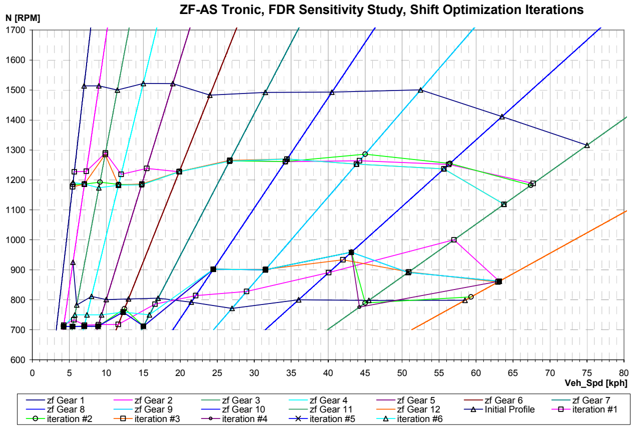


Figure 47: ZF full shift schedule optimization - graphical

iteration #6	shift event	kph	RPM
	u1-2	5.511	1191.680
downshifts limited to 750-rpm	u2-3	7.051	1186.236
only upshifts 1-2 through 7-8	u3-4	9.000	1173.874
were re-run	u4-5	11.653	1182.447
	u5-6	14.786	1184.101
	u6-7	19.865	1227.355
	u7-8	26.662	1262.834
	u8-9	34.446	1269.584
	u9-10	43.831	1252.557
	u10-11	55.649	1236.889
	u11-12	63.767	1118.649
	d2-1	4.225	710.709
	d3-2	5.750	750.000
	d4-3	7.392	750.000
	d5-4	9.365	750.000
	d6-5	12.288	759.206
	d7-6	15.834	750.000
	d8-7	24.481	902.317
	d9-8	31.490	899.904
	d10-9	43.127	958.564
	d11-10	50.797	891.126
	d12-11	63.144	861.799

Figure 48: ZF full shift schedule optimization - tabular

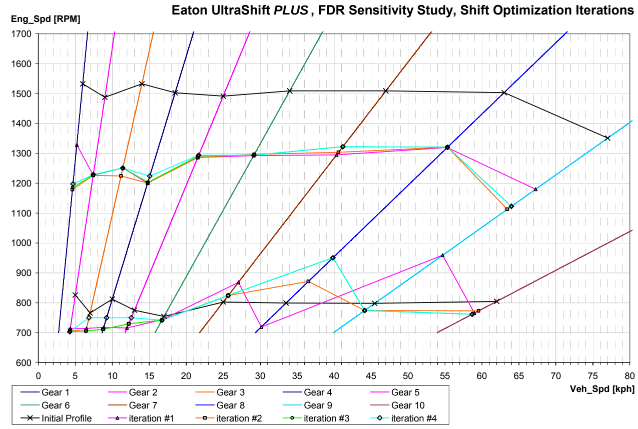


Figure 49: Eaton full shift schedule optimization – graphical

iteration #4	shift event	kph	RPM
	u1-2	4.689	1197.569
downshifts limited to 750-rpm	u2-3	7.438	1229.164
upshifts were re-run,	u3-4	11.426	1250.816
but not the downshifts	u4-5	15.070	1224.050
	u5-6	21.690	1293.709
	u6-7	29.152	1293.844
	u7-8	41.178	1321.956
	u8-9	55.345	1320.433
	u9-10	63.988	1122.526
	d2-1	4.256	703.343
	d3-2	6.851	750.000
	d4-3	9.234	750.000
	d5-4	12.574	750.000
	d6-5	16.718	741.991
	d7-6	25.679	824.391
	d8-7	39.841	950.533
	d9-8	44.149	774.495
	d10-9	58.677	761.724

Figure 50: Eaton full shift schedule optimization - tabular

With the optimized shift schedules defined for full gear range and revised final drive ratio the models were updated and run over the FTP72 drive cycle. Presented below are comparisons of the original FDR Allison with optimized shift schedule, original FDR ZF, original FDR Eaton, new FDR full range ZF, and new FDR full range Eaton.

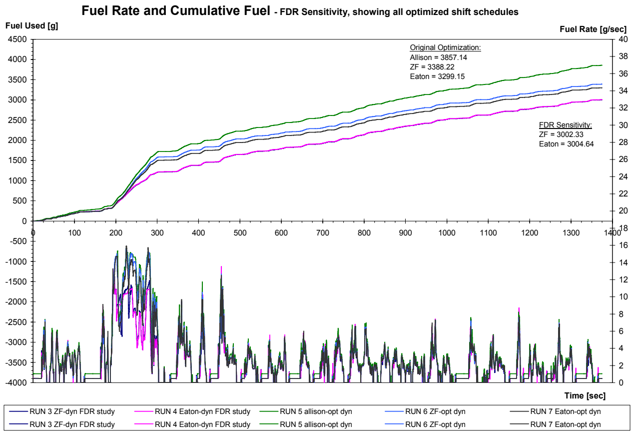


Figure 51: Fuel consumption comparison – revised FDR - graphical

Shft Schd	Fuel Used [g]	Δ FUEL [g]	% difference
Allison_opt	3857.14	-	-
ZF_opt	3388.22	468.92	12.16%
Eaton_opt	3299.15	557.99	14.47%
ZF_FDR study	3002.33	854.81	22.16%
Eaton_FDR study	3004.64	852.50	22.10%

Figure 52: Fuel consumption comparison – revised FDR – tabular

Compared with the optimized Allison the ZF and Eaton both produced a 22% improvement in fuel consumption. This translates to roughly 1.75 mi/gal improvement over the FTP72 drive cycle.

8. CONCLUSIONS & FUTURE WORK

Both the ZF AS Tronic 12 speed and the Eaton UltraShift *PLUS* 10 speed automated manual transmissions demonstrated significant (22%) fuel economy improvement over the FTP72 drive cycle. These fuel economy gains can be attributed to the greater efficiency of manual transmissions and the increased number of gears available in these transmissions to keep the engine operating in its most efficient region. The key enabler to the 22% improvement is the final drive ratio. The final drive ratio change is currently being discussed with Arvin Meritor. It's not clear at this point if Arvin Meritor offers a similar ratio gear set as a standard product.

Both of these transmissions can be considered COTS solutions. As mentioned in the introduction, this study did not consider areas such as design, packaging, development, durability, manufacturing or other commercial issues related to these transmissions. Further investigation in these areas would be appropriate to determine if any hard points exist with either transmission variant. In addition to identifying the primary technical and commercial considerations this investigation would be used to distinguish one AMT variant to be the subject of a vehicle demonstration program.

Aside from assessing the impact to fuel consumption over various drive cycles a demonstration program would also make possible the evaluation of driveability and soft terrain performance. Soft terrain performance was raised as an area of potential concern at the beginning of this study because of the AMT characteristic torque interruption during shifts which could result in vehicle stopping in soft terrain.

AMT transmissions are for all practical purposes a manual transmission with an ECU controlling the clutch and gear position. This type of shifting can be characterized by a so called shifting with load interruption. This simply means that for the time the clutch is open there is no traction on the wheels. When on road or hard terrain this is not an issue. However, at low speed in rough or soft terrain the vehicle may come to a standstill during shifting. In standstill the ECU has to shift down to the first or second gear and the driver has to try to accelerate again. The principle of an automatic powershift transmission, however, which is usually used for high mobility vehicles, lies in the feature that during the shifting operations there no load interruption as the hydraulic torque converter provides for hydraulic coupling during shifting. So during shifting there is still traction on the wheels and therefore less risk of being stuck in soft terrain.

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[11] Rich A Nellums; Eaton Truck Technology, Manager – Technology Planning & Innovation
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CONTACT DETAILS

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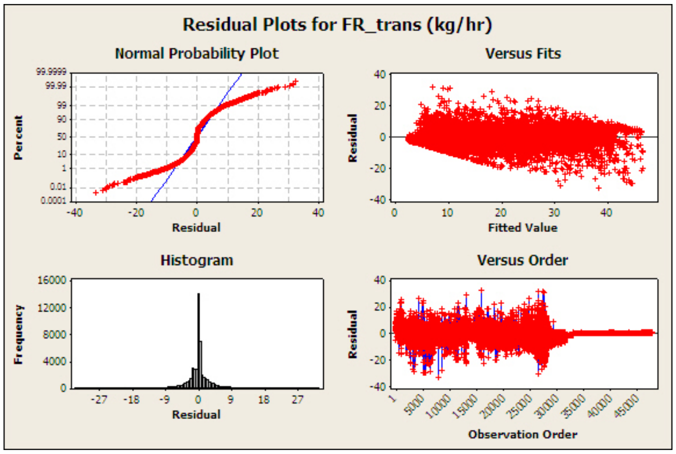
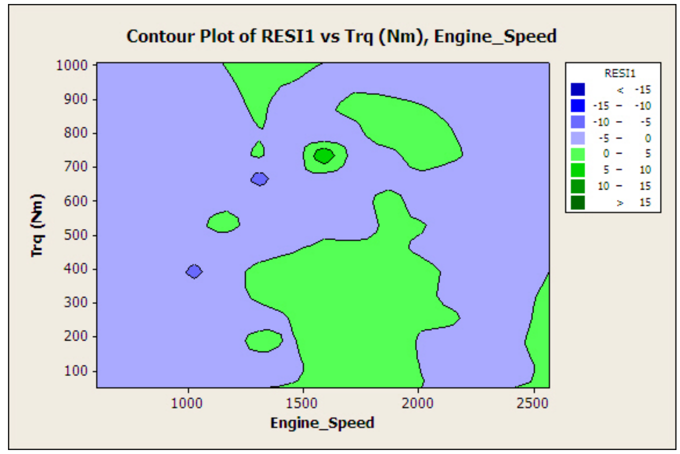
APPENDIX 1

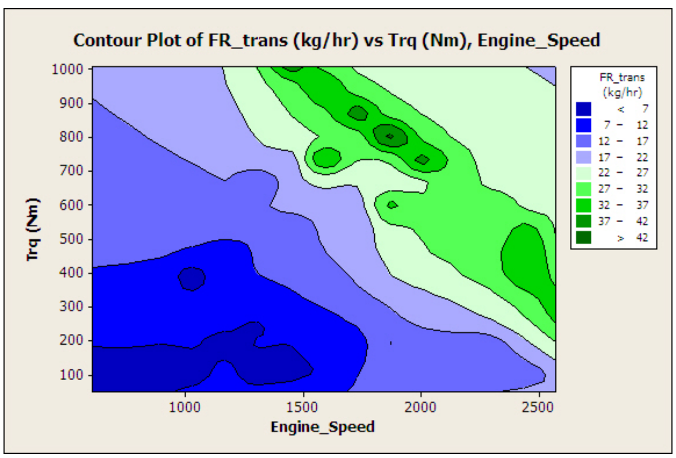
The regression equation is:

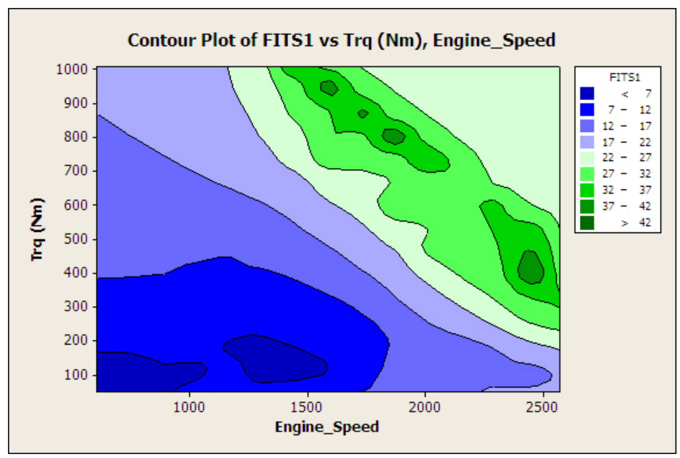
$$\begin{aligned} FR\_trans\ (kg/hr) = & -8.58 + 0.0141\,N - 0.000005\,N^2 + 0.236\,Trq - 0.000343\,Trq^2 \\ & - 0.000308\,N{*}Trq + 0.000000\,N{*}Trq^2 + 0.000000\,N^2{*}Trq \\ & - 0.000000\,N^2{*}Trq^2 \end{aligned}$$

S = 3.11923   R-Sq = 92.3%   R-Sq(adj) = 92.3%

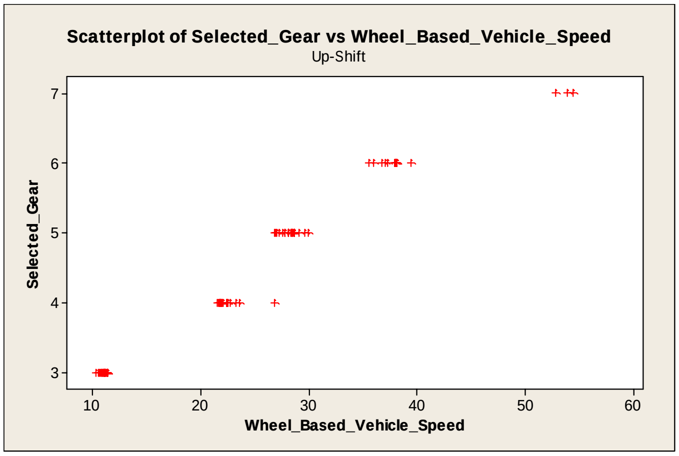








APPENDIX 2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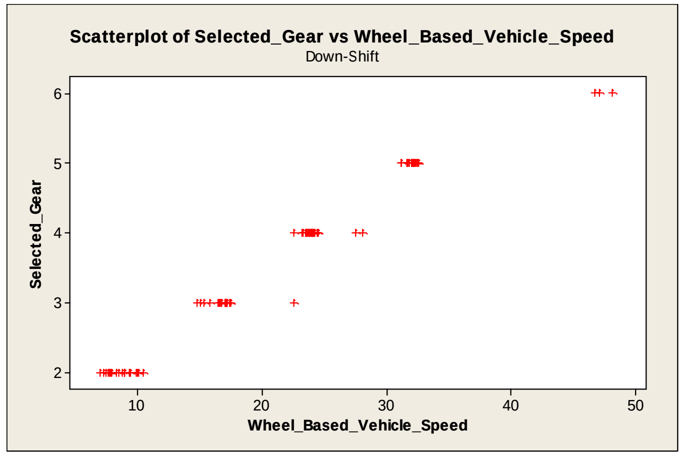
Up-Shift
Selected_Gear
Wheel_Based_Vehicle_Speed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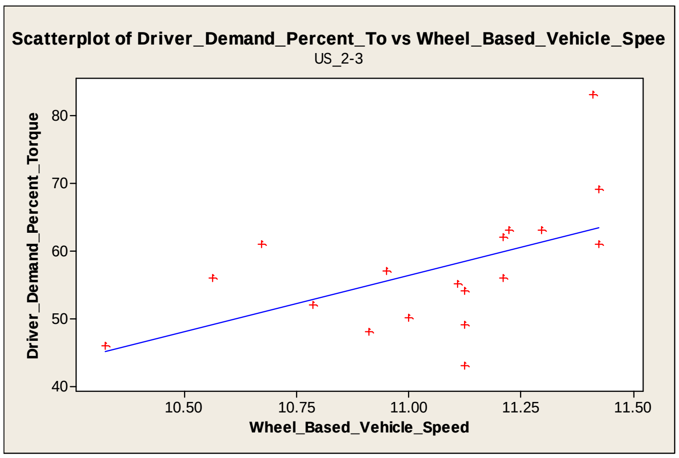
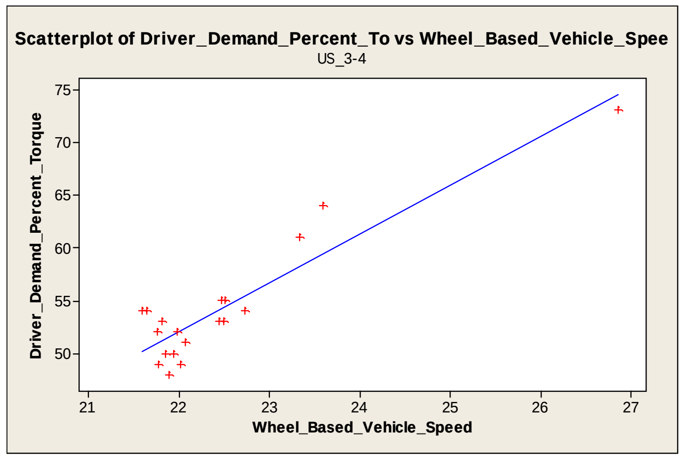
Down-Shift
Selected_Gear
Wheel_Based_Vehicle_Speed


Scatterplot of Driver_Demand_Percent_To vs Wheel_Based_Vehicle_Spee
US_2-3
Driver_Demand_Percent_Torque
Wheel_Based_Vehicle_Speed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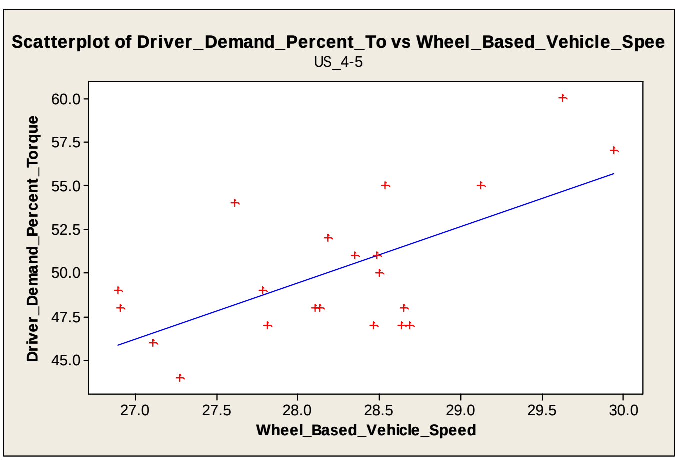
US_3-4
Driver_Demand_Percent_Torque
Wheel_Based_Vehicle_Speed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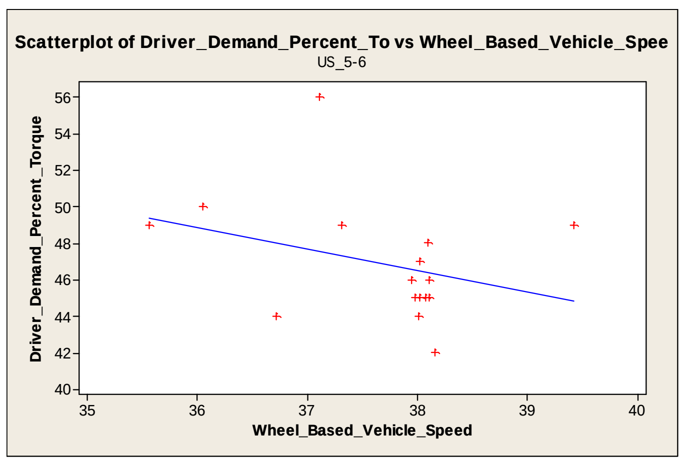
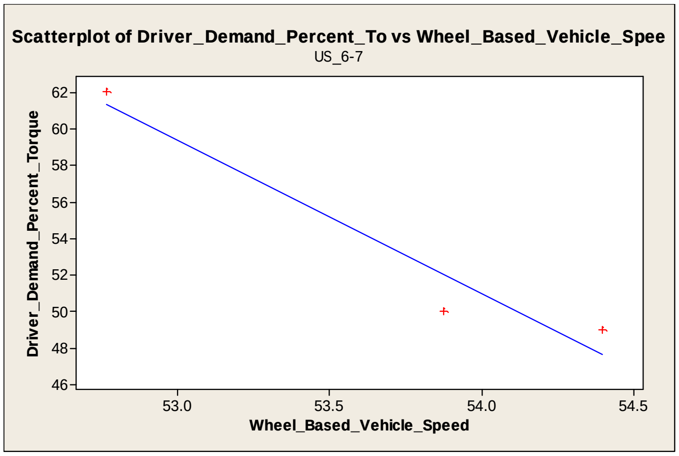
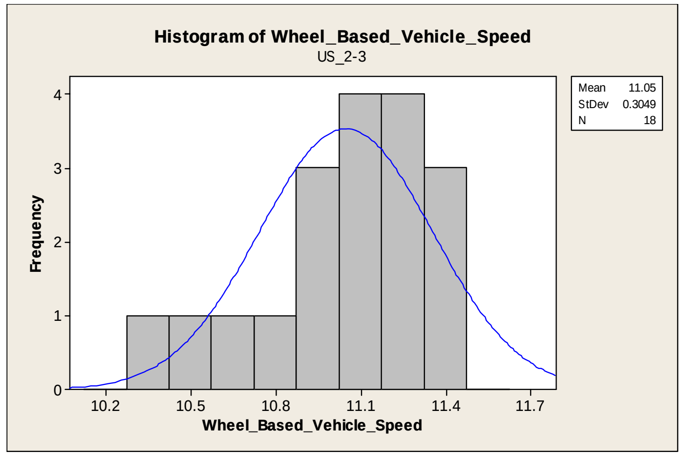
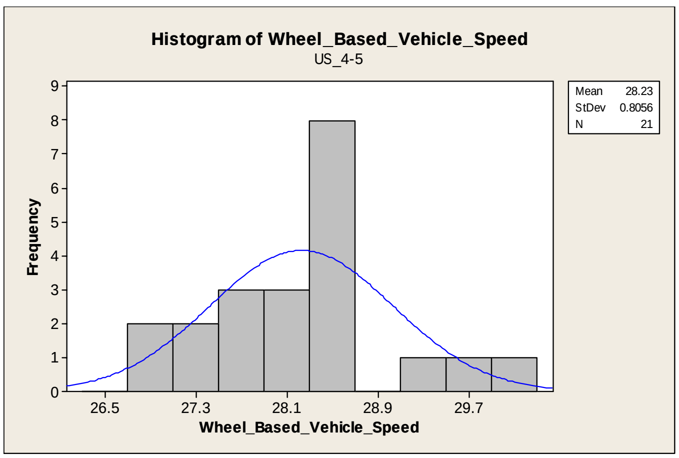
US_4-5
Driver_Demand_Percent_Torque
Wheel_Based_Vehicle_Speed

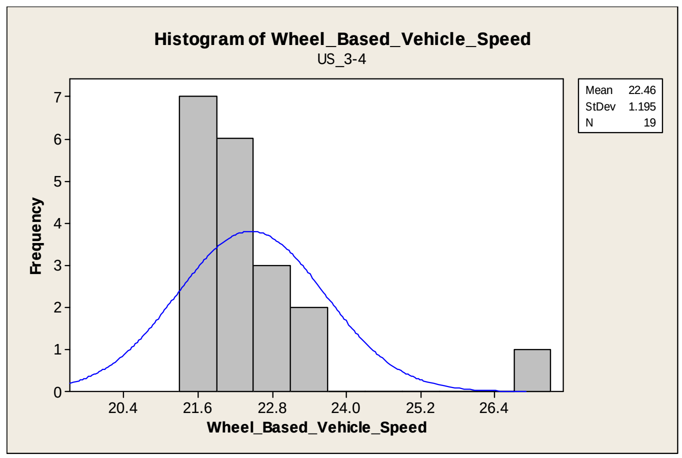

Scatterplot of Driver_Demand_Percent_To vs Wheel_Based_Vehicle_Spee
US_5-6
Driver_Demand_Percent_Torque
Wheel_Based_Vehicle_Speed

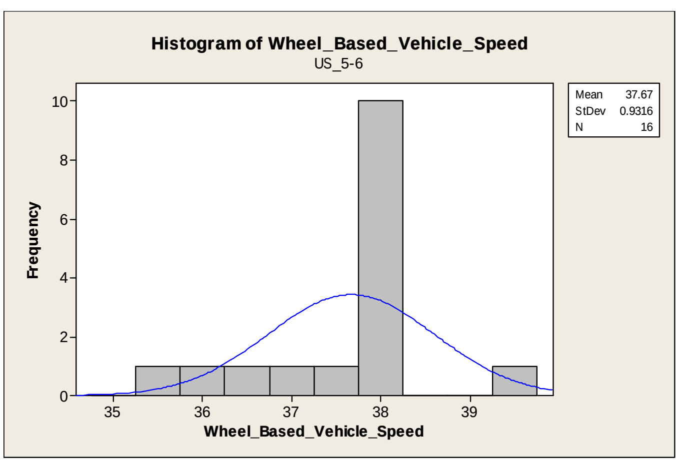
APPENDIX 2 (continued)

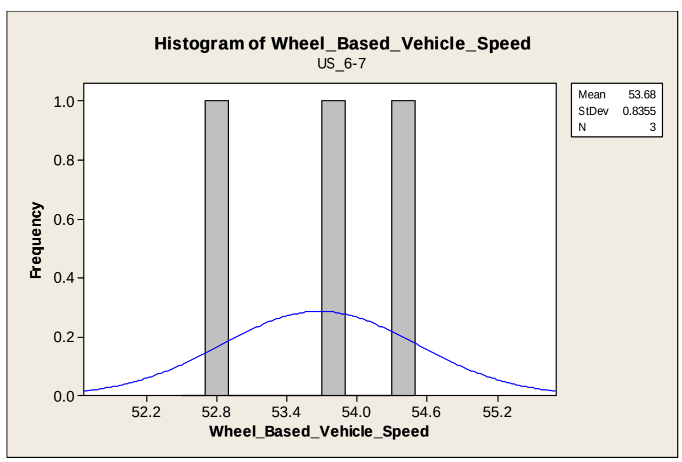




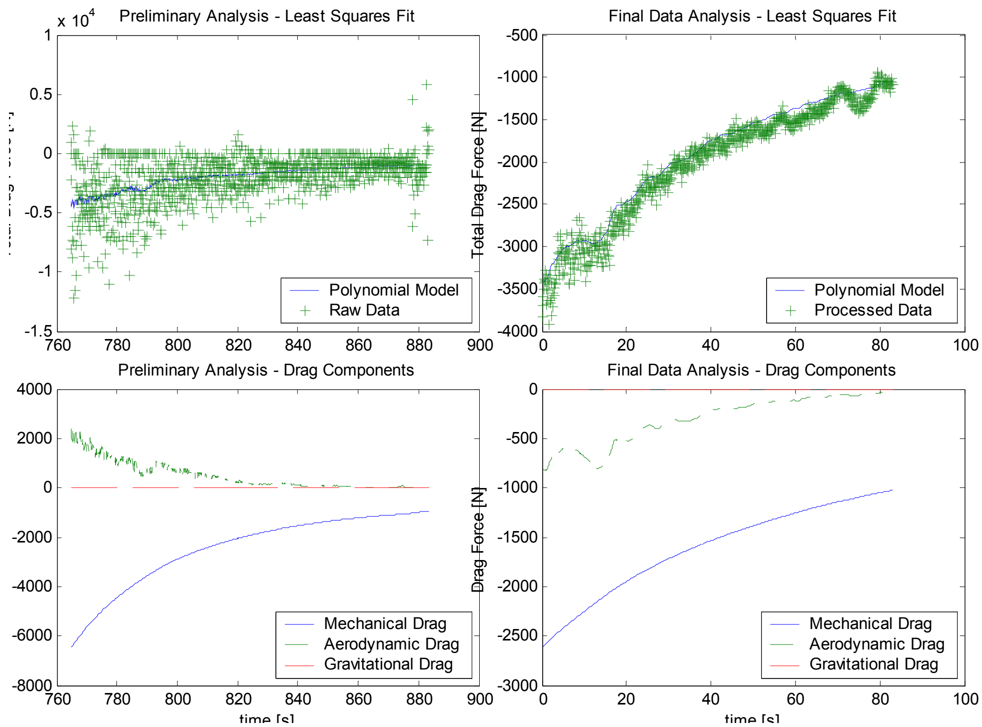




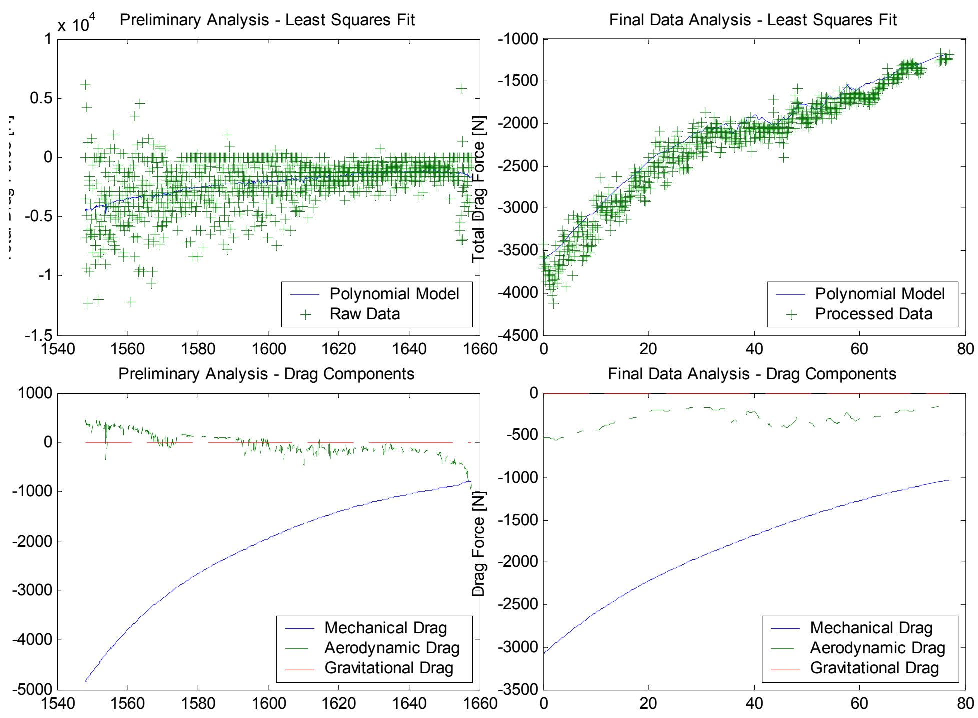




APPENDIX 3

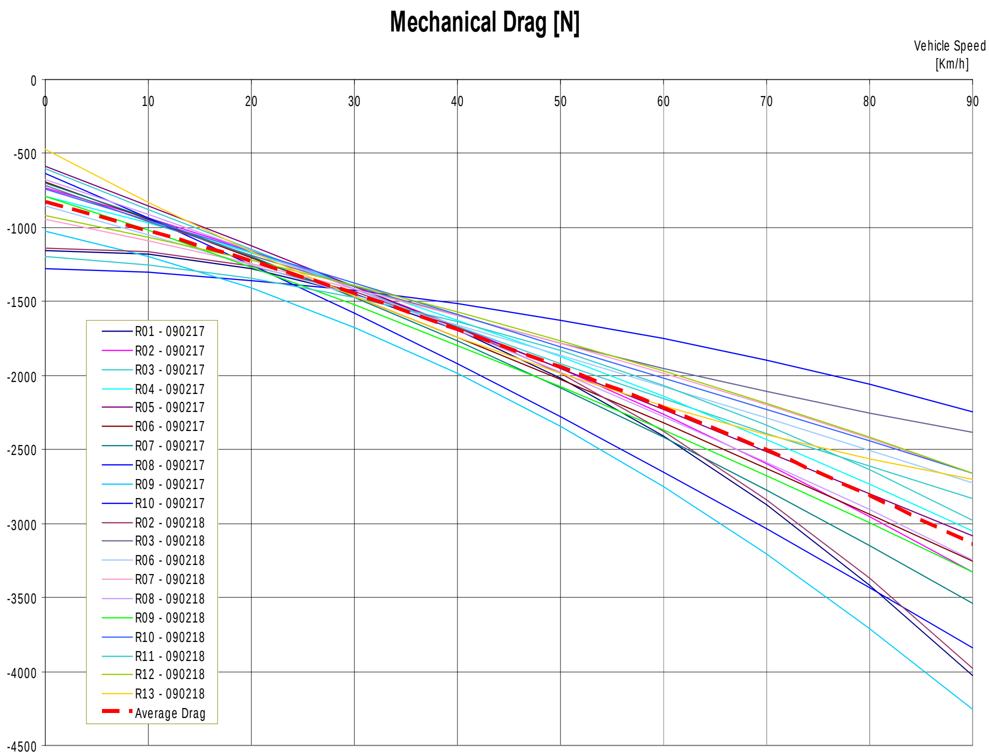


Coast down data for run 4 from February 17, 2009



Coast down data for run 7 from February 18, 2009

APPENDIX 4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Vehicle Speed [Km/h]
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10
20
30
40
50
60
70
80
90
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-1000
-1500
-2000
-2500
-3000
-3500
-4000
-4500
R01 - 090217
R02 - 090217
R03 - 090217
R04 - 090217
R05 - 090217
R06 - 090217
R07 - 090217
R08 - 090217
R09 - 090217
R10 - 090217
R02 - 090218
R03 - 090218
R06 - 090218
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R09 - 090218
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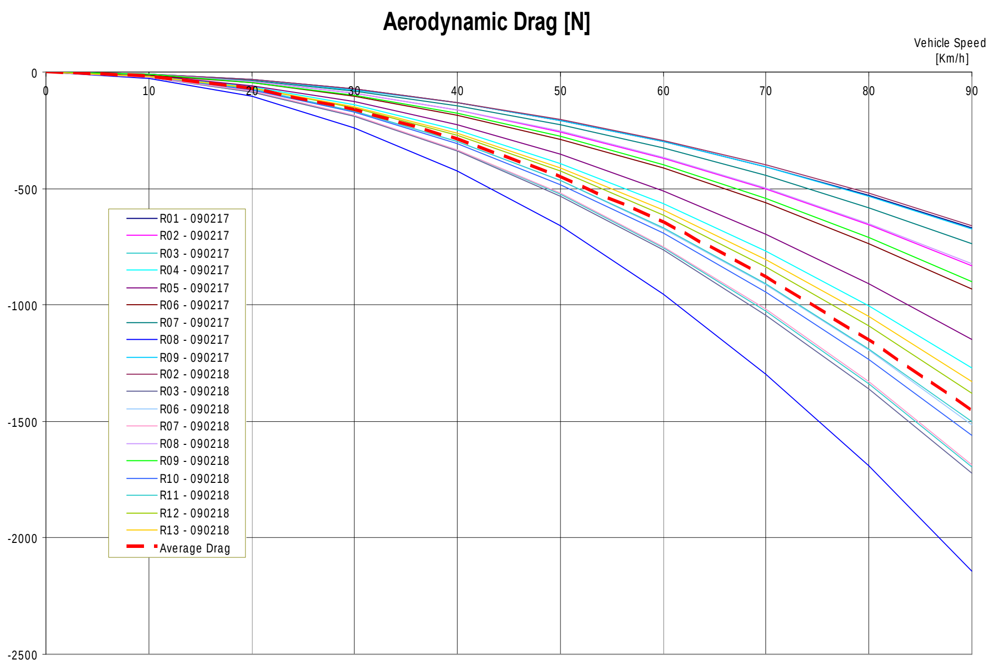
R11 - 090218
R12 - 090218
R13 - 090218
Average Drag


Aerodynamic Drag [N]
Vehicle Speed [Km/h]
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20
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50
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70
80
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Average Drag