Modeling the Effects of Transmission Gear Count, Ratio Progression, and Final Drive Ratio on Fuel Economy and Performance Using ALPHA

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Abstract

The Advanced Light-Duty Powertrain and Hybrid Analysis (ALPHA) tool was created by EPA to evaluate the Greenhouse Gas (GHG) emissions of Light-Duty (LD) vehicles [1]. ALPHA is a physics-based, forward-looking, full vehicle computer simulation capable of analyzing various vehicle types combined with different powertrain technologies. The software tool is a MATLAB/Simulink based desktop application. The ALPHA model has been updated from the previous version to include more realistic vehicle behavior and now includes internal auditing of all energy flows in the model [2]. As a result of the model refinements and in preparation for the mid-term evaluation (MTE) of the 2022-2025 LD GHG emissions standards, the model is being revalidated with newly acquired vehicle data.

This paper presents an analysis of the effects of varying the absolute and relative gear ratios of a given transmission on carbon emissions and performance. Energy-based methods of selecting absolute gear ratios are considered and the effects of alternative engine selections are also examined. An algorithm is presented for automatically determining ALPHAshift parameter sets based on the selected engine and transmission combination. It is observed that no single ratio progression optimizes fuel consumption for all applications, however, fuel consumption is also relatively insensitive to progression which implies a fixed set of ratios can still be used for a range of applications without necessarily compromising consumption. The energy-based ratio analysis may prove useful in determining the optimal overall top gear ratio for a given engine-vehicle combination and also helps to explain the relative insensitivity to ratio progression. Individual performance metrics can show high sensitivity to ratio progression, final drive ratio and shift calibration, in particular 30-50 and 50-70 MPH passing times.

Introduction

Background

During the development of the LD GHG and CAFE standards for the years 2017-2025, EPA utilized a 2011 light-duty vehicle simulation study from the global engineering consulting firm, Ricardo, Inc. The previous study provided a round of full-scale vehicle simulations to predict the effectiveness of future advanced technologies. Use of data from this study is documented in the August 2012 EPA and NHTSA "Joint Technical Support Document" [3].

The 2017-2025 LD GHG rule required that a comprehensive advanced technology review, known as the mid-term evaluation, be performed to assess any potential changes to the cost and the effectiveness of advanced technologies available to manufacturers. EPA has developed the ALPHA model to enable the simulation of current and future vehicles, and as a tool for understanding vehicle behavior, greenhouse gas emissions and the effectiveness of various powertrain technologies. For GHG, ALPHA calculates CO_2 emissions based on test fuel properties and vehicle fuel consumption. No other emissions are calculated at the present time but future work on other emissions is not precluded.

ALPHA will be used to confirm and update, where necessary, efficiency data from the previous study. It may also be used to understand effectiveness contributions from advanced technologies not considered during the original Federal rulemaking, such as continuously variable transmissions (CVTs) and naturally aspirated Atkinson engines for conventional vehicles.

This Paper's Focus

EPA engineers utilize ALPHA as an in-house research tool to explore in detail current and future advanced vehicle technologies. ALPHA is being refined and updated to more accurately model light-duty vehicle behavior and to include new technologies. To validate the performance of ALPHA, EPA is using newly acquired in-depth vehicle, engine, and transmission benchmarking data from several conventional and hybrid vehicles from 2013-2015 model years.

In broad terms, the transmission content of the current and future US light-duty fleet may be characterized in terms of transmission type, gear count and ratio spread. Within the given type, count and spread there are many possible configurations that may be used for a given application. In order to model possibilities for the future conventional light-duty fleet it is necessary to consider the interactions of various engines and transmissions.

Computer modeling of vehicle fuel economy, performance and sensitivity to design parameters is nothing new [4]. More recently, interesting and informative work has been done to inform manufacturers and stakeholders of the potential for various technologies to meet the 2025 standards [5, 6, 7, 8]. The findings here will not contradict any of the previous work but may shed some additional light on the topic by covering a few of the modeled factors in finer detail. For example, the papers referenced here cover only a few select final drive ratios and one or a handful of logarithmic ratio progressions. Also, the assumption of performance neutrality may be handled in a coarser manner as well, with engines scaled either not at all or perhaps only in certain cases. For the work presented here a selection of progressions and final drive ratios will be considered with relatively fine steps between configurations. For the combined progression and final drive ratio sweeps, 400 unique configurations are modeled, each adjusted for performance neutrality. In addition, an energy-based analysis of ideal gear ratios for a particular application will be presented and may help to explain some of the sensitivities observed.

This study will take advantage of the ALPHAshift algorithm which generates shift points for a given engine and transmission combination, subject to a set of tunable control parameters [9, 10].

This paper's focus is to explore the sensitivity of carbon emissions and acceleration performance to transmission ratio progression, final drive ratio and engine type in order to make informed decisions for current and future modeling work. In particular, the assumption used in previous work that a given set of ratios (e.g. from a production transmission) can be applied to a range of engine-vehicle applications by simply varying final drive ratio is explored and validated. Since the goal of this work is to explore model sensitivity, not transmission design, some simplifying assumptions are made which may not necessarily apply to production designs but which allow the effects of ratio progression and overall gearing to be isolated and studied for simulation purposes.

The modeling process and baseline transmissions will be described followed by a study of carbon emissions sensitivity to ratio progression. An energy-based analysis of ideal gear ratios will be presented followed by an analysis of the effects of varying final drive ratio. Performance metrics and performance neutrality will be discussed at the end of the paper along with some notes on the shift algorithm assumptions used for this study.

Modeling Process

The modeling process for this study is straightforward and proceeds as follows:

1. Determine estimated baseline vehicle performance (average power at the wheels over a reference vehicle speed range) as defined by a baseline engine/transmission/final drive package
2. Select an engine, determine new transmission and/or final drive ratios
3. Determine the estimated performance of the new package and iteratively adjust the engine displacement in 1% increments until equivalent or better performance is achieved
4. Perform the simulation and record the results

For this study, the baseline vehicle driveline consists of a 2008-era naturally aspirated 2.4L engine with a five-speed automatic transmission and a 3.23:1 final drive ratio. Nominally this vehicle represents a 2008 Toyota Camry. The 2025 LD GHG rule is based on technology packages that are meant to be performance neutral when compared with the 2008 fleet in order to determine technology effectiveness and carbon emissions improvements on a level playing field. Further details on performance neutrality are discussed in the Scaling Engines for Performance Neutrality section of this paper.

Step Gear Transmission Ratio Progressions

For a given gear count and spread, many possible progressions from maximum to minimum ratio are possible. A set of possible progressions is described by equation (1) [11] and has been used directly or with some modification in previous work of a similar nature [6,7,8].

$$i_n = \left[\frac{Spread}{\varphi_2^{\,0.5(z-1)(n-1)}}\right]^{\frac{z-n}{z-1}} \tag{1}$$

Where:

Spread = transmission spread (the ratio of maximum to minimum gear ratios)

z = total number of gears (> 1)

n = gear number of ratio being calculated (varies from 1..z)

φ_2 = progression factor (typically □1 to 1.2)

i_n = the nth gear ratio

The progression factor, φ_2, determines the curvature of the progression and indicates the change in steps between gears and will be further described below. Lower factors are more linear and higher factors reach lower ratios sooner. A φ_2 of 1.0 generates a progression where each gear ratio is a fixed percentage of the preceding ratio, also known as a geometric progression. Figure 1 and Figure 2 show a family of progressions generated by sweeping φ_2 over a range of values for six- and eight-speed transmissions respectively.

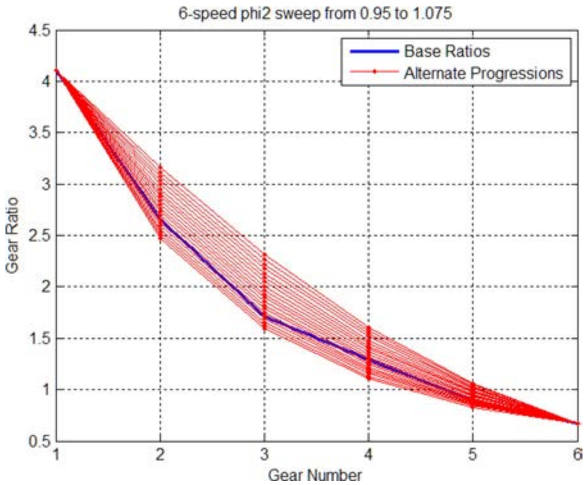

Figure 1. A sweep of φ_2 from 0.95 to 1.1 for a six-speed transmission

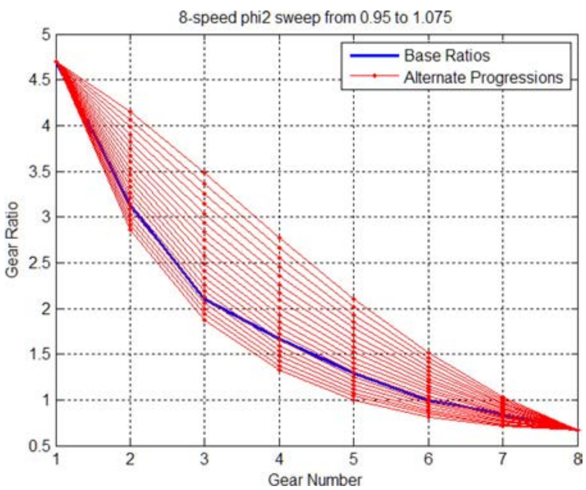

Figure 2. A sweep of φ_2 from 0.95 to 1.075 for an eight-speed transmission

As may be noticed from the figures, the range of φ_2 that generates reasonable values varies depending on the number of gears in the transmission. For the eight-speed transmission a φ_2 of 1.1 would generate a 7th gear with a ratio slightly below that of 8th gear. Values lower than 0.95 start to generate excessive convexity in the progression. For these reasons the range of φ_2 values for the eight-speed is limited to between 0.95 and 1.075 and reasonable values for higher speed transmissions would also require some investigation. There is no absolute value of φ_2 that generates optimal progressions for all transmissions. Base ratios for the six-speed transmission are based on the GM6T40 transmission from the Chevy Malibu, scaled down to compensate for the default final drive ratio used in this study (3.23 versus 2.89 for the Malibu, 5th gear ratio would normally be 1:1) [12]. Base ratios for the eight-speed transmission are inspired by the ZF8HP70 from ZF [13] but differ slightly.

The ratios used for the baseline transmission cases are shown in Table 1. In the table, "Step" refers to the ratio of the gear ratios between successive gears. For example, for the six-speed gears 1 and 2, 4.101/2.653 = 1.546. The "Step Ratio" is the ratio of the Steps, calculated in similar manner. As will be seen below, the Step Ratio corresponds to φ_2 in equation (1).

Table 1. Baseline transmission ratios for 6 and 8 speed AT transmissions

Gear Number	6-speed Ratios	Step	Step Ratio	8-speed Ratios	Step	Step Ratio
1	4.101			4.700		
2	2.653	1.546		3.130	1.502	
3	1.711	1.551	0.9969	2.100	1.490	1.007
4	1.294	1.322	1.173	1.670	1.257	1.185
5	0.895	1.446	0.9142	1.290	1.295	0.9714
6	0.668	1.340	1.079	1.000	1.290	1.004
7				0.840	1.190	1.084
8				0.670	1.254	0.9495
Final Drive Ratio	3.23			3.23		
Average Step Ratio			1.041			1.033
Spread	6.139			7.015		

As a concrete example of equation (1) in use, see the results shown in Table 2 which shows ratio progressions for the six- and eight-speed transmissions using the average Step Ratios from Table 1 as the values for φ_2. For each transmission the spread and minimum and maximum ratios are held constant and the intermediate ratios are determined by equation (1). For the six-speed, the ratios are quite a good match with 4th gear showing the largest difference. For the eight-speed, the baseline ratios drop more quickly for gears 2 and 3 but 4 through 8 are quite similar.

Table 2. Sample six- and eight- speed transmission ratios using φ_2 values of 1.041 and 1.033 respectively

Gear Number	6-speed Ratios	Step	Step Ratio	8-speed Ratios	Step	Step Ratio
1	4.101			4.700		
2	2.632	1.558		3.228	1.456	
3	1.759	1.496	1.041	2.290	1.409	1.033
4	1.224	1.438	1.041	1.678	1.364	1.033
5	0.8861	1.381	1.041	1.271	1.321	1.033
6	0.668	1.327	1.041	0.9938	1.279	1.033
7				0.8028	1.238	1.033
8				0.670	1.198	1.033

In order for the progression study to provide meaningful results, some simplifying assumptions were made based on initial modeling runs:

- All gears have the same efficiency, regardless of gear ratio
- All torque converter clutches lock up in 2nd gear and above

Without the first assumption there occurred results where the best "eight-speed" was a "six-speed" - the results favored using the eight-speed spread but only using the first 6 gears since they had higher efficiency than the last two gears, although admittedly these were extreme cases. In itself this was an interesting result in that it suggested that transmission spread may be as important as or more important than the number of gears, but further exploration would be required in order to draw any conclusions on this observation. Running the model with gear efficiencies from actual transmissions changes the absolute carbon emissions results but the overall trends are comparable and using default fixed efficiencies eliminates some spurious cases. To further confirm this, gear efficiencies from [7] were used for the transmission and final drive and results were nearly indistinguishable from the results presented here so the simplification does not negatively affect the results or the conclusions of this paper.

The second assumption prevents step-changes in the carbon emissions based on the original transmission lockup tables and their sensitivity to new and unexpected ratio progressions. For this reason, the lockup strategy was reduced to the simplest form possible.

Carbon Emissions and Gear Ratios

Engine Selections

The first analysis will examine carbon emissions sensitivity to ratio progression for six- and eight-speed automatic transmissions mated with four different engines:

- A 2010-era engine with dual cam phasing (DCP) [14, 15]
- The 2013 Chevy Malibu engine as used in previous modeling efforts [12]
- The 2014 Mazda Skyactiv 2L, 13:1 compression ratio (CR) [16]
- A hypothetical “2020” downsized boosted engine, 24 bar, with cooled EGR [14, 15]

The engines represent a range of past, current, and future technologies. Three of the engines are naturally aspirated and the fourth represents a hypothetical future downsized boosted engine. It should be noted that for this study the engines are not compared to each other and the results for each engine are normalized to that particular engine. The purpose of including multiple engine packages is to see if any distinct trends emerge for a given engine technology package rather than to compare and contrast the engine technologies themselves, as has already been done [3, 5, 14].

Progression Sensitivity

Figure 3 and Figure 4 show the normalized combined EPA city-highway (UDDS-HFET) carbon emissions for six- and eight-speed automatic transmissions as a function of φ_2. The vertical lines represent the best carbon emissions point, color-coded by engine (the same color-coding is used for all plots for these engines). In this case, unique progressions were optimal for each engine.

For a wide range of φ_2, the results are comparable for a given engine. Many of the results are within 0.5% of peak and almost all are within 1%. Generally speaking, there is not a strong trend in carbon emissions as a function of φ_2 for different engine types and number of gears in the transmission, although the 24 bar engine shows slightly less sensitivity than the other engines, as might be expected with its large high efficiency plateau.

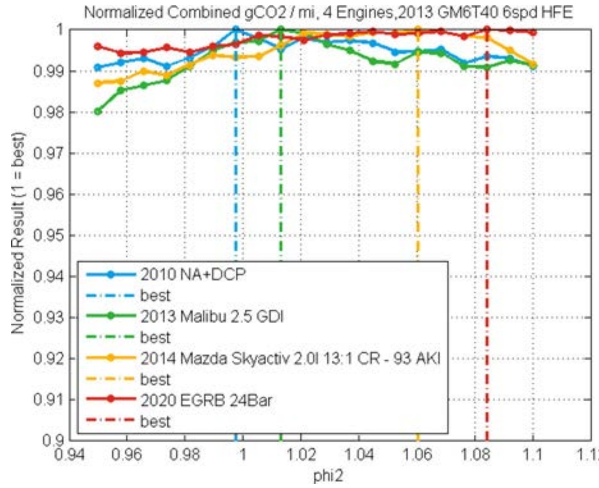

Figure 3. Six-speed AT normalized carbon emissions progression sensitivity

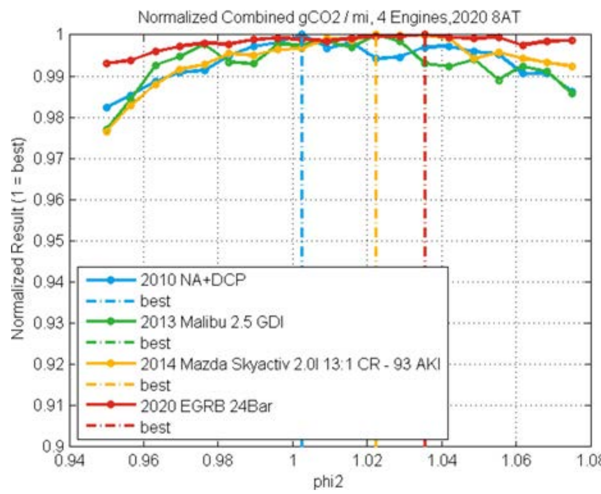

Figure 4. Eight-speed AT normalized carbon emissions progression sensitivity, the Malibu and the Mazda share the same optimal progression

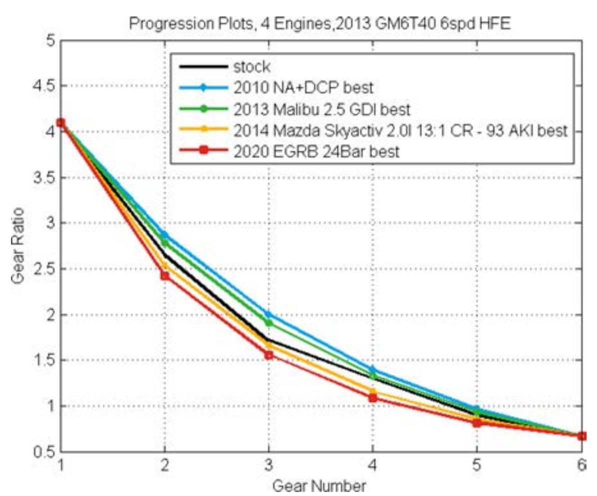

Figure 5. Six-speed AT optimal progressions

Figure 5 and Figure 6 show the optimal progressions as a function of gear number. In general the progressions are similar to the base progressions, especially given the range of possibilities as seen in Figure 1 and Figure 2. This, combined with the relative insensitivity to φ_2, gives some confidence that as long as reasonable ratio progressions are used (possibly from a variety of production transmissions) engines and transmissions can be mixed and matched without a detrimental effect on carbon emissions from using a generic progression.

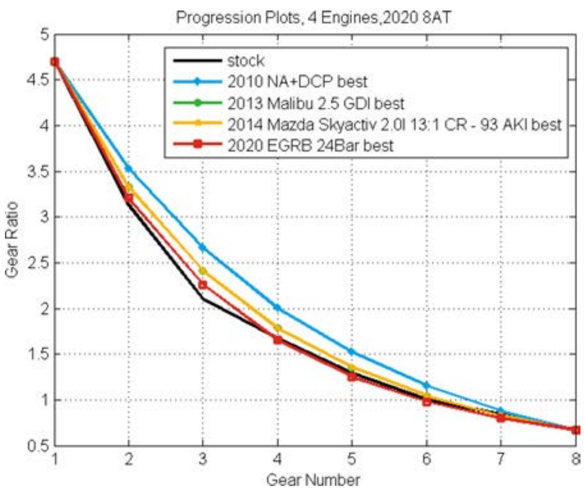

Figure 6. Eight-speed AT optimal progressions, the Malibu and Mazda curves are superimposed and follow the same progression

Ratio Preference Based on Energy-Weighted Power Demand and Ideal Operating Points

One method of understanding the sensitivity of carbon emissions to gear ratios is to try to answer the question "what ratios would this engine prefer for this given application?" It is possible to calculate, for each engine, a minimum BSFC operating speed and load as a function of desired engine power. In addition, for a given desired power there may be a range of suitable operating points within some tolerance of optimal, e.g. almost anywhere on an engine's peak efficiency plateau. If the engine's operating power as a function of vehicle speed can be determined then an optimal gear ratio can be determined for each vehicle speed since vehicle speed is proportional to transmission output speed and gear ratio is defined as the ratio of the transmission input (engine) and output speeds.

Three energy-based analysis methods are presented. The first uses average engine power as a function of vehicle speed and the engine's optimal BSFC line to determine desired gear ratios. The second uses average engine power and additionally considers operating points within one BSFC percent of the optimal BSFC line. The final method considers the instantaneous power demand and optimal and the same near optimal operating points.

For simplicity, only one engine will be considered for the following analysis, the 2013 Mazda Skyactiv 2L as shown in Figure 7. The magenta curve represents the optimal BSFC line and the light green curves above and below it represent the boundaries of operation within +- 1% of optimal. The highlighted colored area shows the energy weighted operation over the drive cycle depicted in Figure 8 with the baseline 6AT and 3.23:1 FDR.

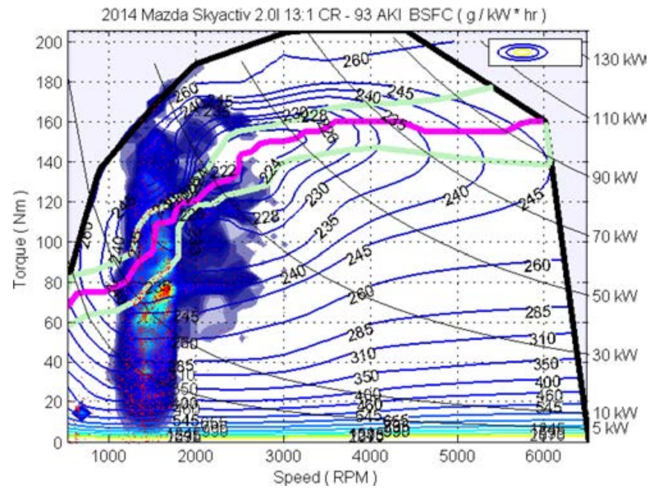

Figure 7. Mazda Skyactiv engine BSFC map and energy weighted operating points over the drive cycle depicted in Figure 8 with the baseline 6AT and 3.23:1 FDR

For this energy analysis, the FTP and HFET cycles were concatenated to produce the cycle shown in Figure 8. For simplicity the drive cycle power demand in the following figures has not been weighted by the factors used to create a "combined" city/highway carbon emissions result, although this is certainly possible.

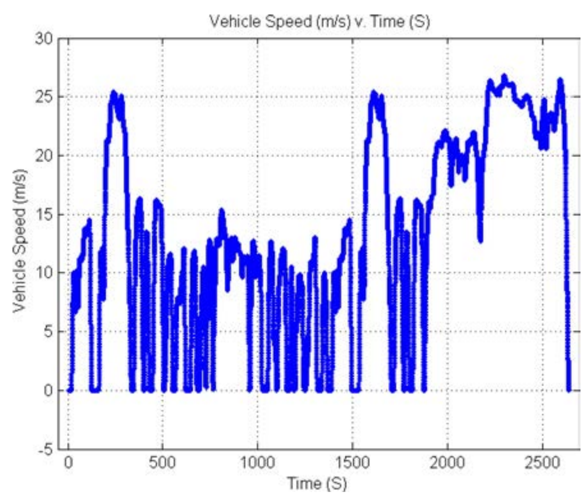

Figure 8. FTP and HFET concatenated drive cycle

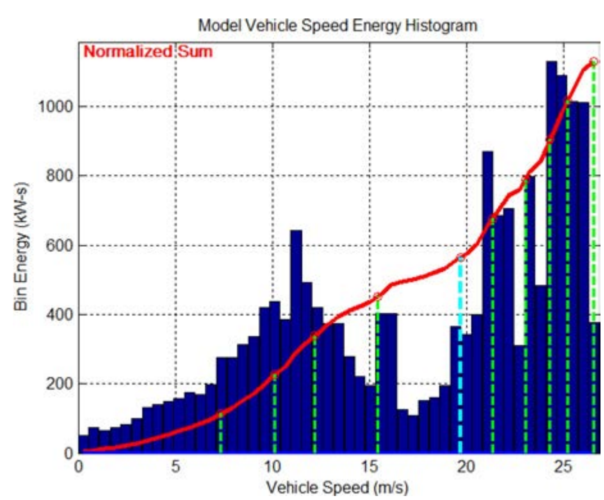

Figure 9. Energy signature for the concatenated drive cycle, positive engine powers and speeds above 0.2 m/s.

For this drive cycle, the engine's positive (driving) energy consumption histogram was determined as a function of vehicle speeds above 0.2 m/s (0.5 MPH) as seen in Figure 9. Every drive cycle has a unique energy signature. The vertical cyan line represents the point at which half the energy was consumed at higher speed and half consumed at lower speed - the 50% point of total consumed energy. The vertical green lines are in increments of 10% of total energy usage. The red cumulative energy line is normalized to the height of the tallest blue bar.

This drive cycle has a peak in energy consumption around 11 m/s (25 MPH) and other peaks at about 21 and 25 m/s (47 and 56 MPH). For this drive cycle the halfway mark for energy consumption is around 20 m/s (44 MPH) and the overall signature is essentially bimodal - a low speed cruise area and a high speed cruise area and not much inbetween.

Given the energy consumed at each vehicle speed the average power can be calculated if the time spent at each speed is known. Therefore, the total time at each speed was also determined (Figure 10).

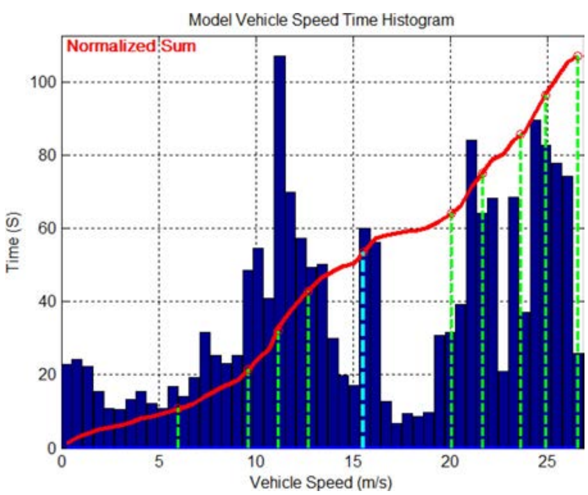

Figure 10. Time signature for the concatenated drive cycle, for positive engine powers and speeds above 0.2 m/s.

Taking the total energy at each speed and dividing by time spent at each speed results in an average power for each speed, as shown by the red line in Figure 11. For example, the energy consumed in the 15.5 m/s histogram bin was 400 kW-s and the time spent in the same speed bin was 60 seconds so the average power for that speed range would be 6.7 kW as indicated by the red line at the same speed in Figure 11. The blue dots represent the instantaneous power demand versus vehicle speed during the drive trace shown in Figure 8. It can be seen that the energy-weighted average looks reasonable but there is quite a range of operating points when looking at the instantaneous powers. The bump in average desired power over the 17 to 20 m/s range is due to the fact that the operation in that speed range, for this drive cycle, is based on accelerating the vehicle up to higher speeds and does not include any low-power cruising.

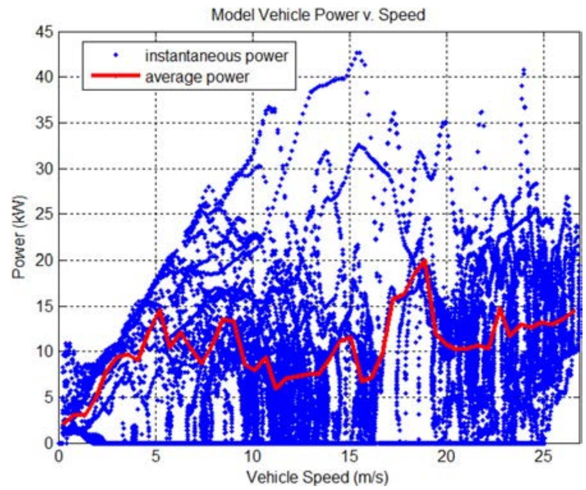

Figure 11. Instantaneous and energy-weighted average engine power versus vehicle speed

From transmission output speed as a function of vehicle speed, the optimal and boundary BSFC lines from Figure 7 and average desired engine power, a desired gear ratio can be calculated for each vehicle speed as shown in Figure 12. The dark blue curve represents the desired ratio based on the optimal BSFC line and the red and magenta curves represent the ratios for the upper and lower BSFC boundary curves (where BSFC is within +/- 1% of optimal) respectively. For reference, the cyan lines represent the ratios chosen by the ALPHAshift algorithm for this drive cycle and naturally bound the desired gear ratios where possible.

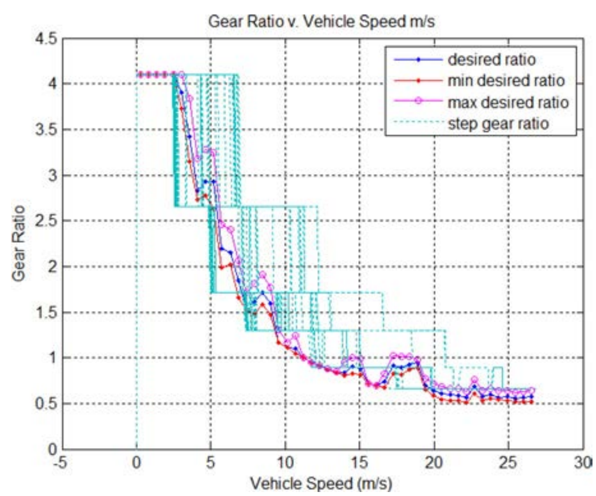

Figure 12. Desired and actual gear ratios versus vehicle speed

An energy weighted desired ratio histogram can be created by taking each point along the blue curve in Figure 12 and putting *n* points into gear ratio bins where *n* is the amount of energy consumed at each point from Figure 9. This process can be referred to as "voting" for preferred gear ratios and then tallying the vote as seen in Figure 13. For example, considering operation at about 6 m/s in Figure 12, the ideal ratio is about 2.2:1 and looking at the histogram in Figure 9 we can see the energy consumption was around 175 kW-s at the same vehicle speed, which results in the vote tally in Figure 13 for the bin between 2 and 2.5. The red lines represent the baseline six-speed ratios for the Mazda engine from Table 1.

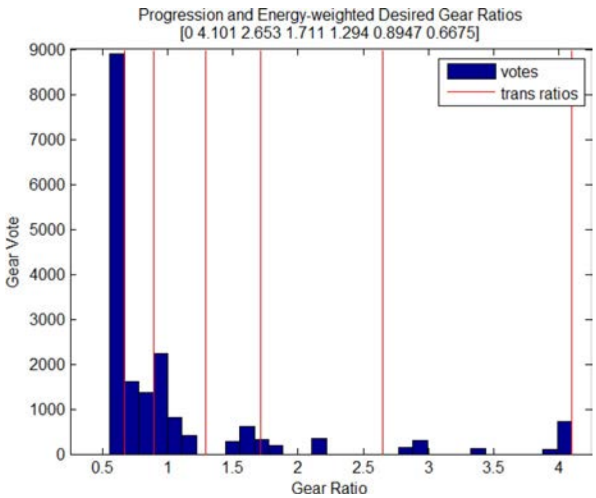

Figure 13. Energy weighted desired gear ratio histogram

One may also consider the minimum and maximum desired ratios and apply a weighting factor based on distance from optimal. This approach would recognize that the optimal ratio might be best but close ratios might also be worth considering. Several weighting factors are possible. Figure 14 shows an example weighting function for an optimal ratio of 2.758:1 and boundary ratios of 2.553:1 and 3.161:1 which represent operating points with 1% BSFC of optimal. For the histograms in Figure 16 and Figure 17 a linear weighting function as shown here was used. Points on the boundary received zero weight, as shown in Figure 14 and Figure 15.

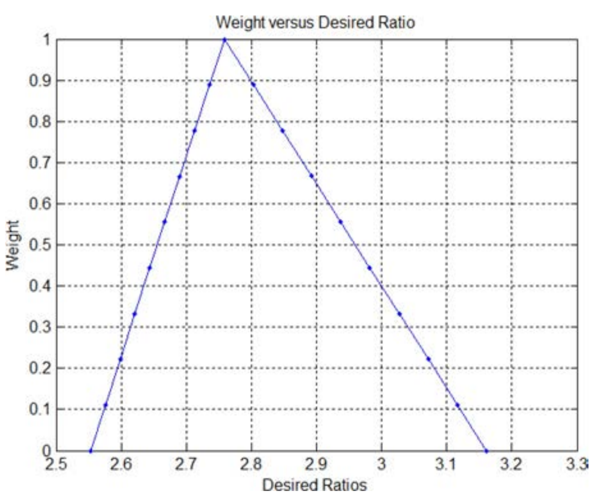

Figure 14. Desired gear ratio and weighting factor based on distance from optimal BSFC

In general, differing weighting factors proved to have little effect on the overall result other than slightly increasing or decreasing the overall number of votes.

An example chart showing the application of the weights is shown in Figure 15. The vertical red line represents the optimal ratio and the green lines represent the boundary ratios for the case shown in Figure 14. The length of the blue bars is determined by the weight factor as a function of distance from the optimal BSFC multiplied by the drive energy from Figure 9 for that particular operating point's vehicle speed.

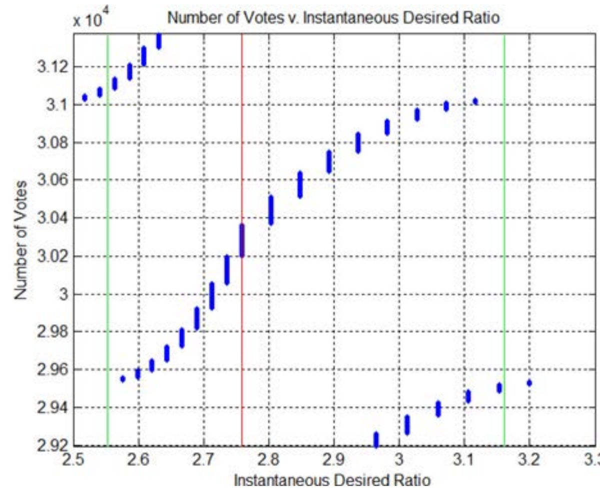

Figure 15. Example of weighting factor applied to a range of desired ratios

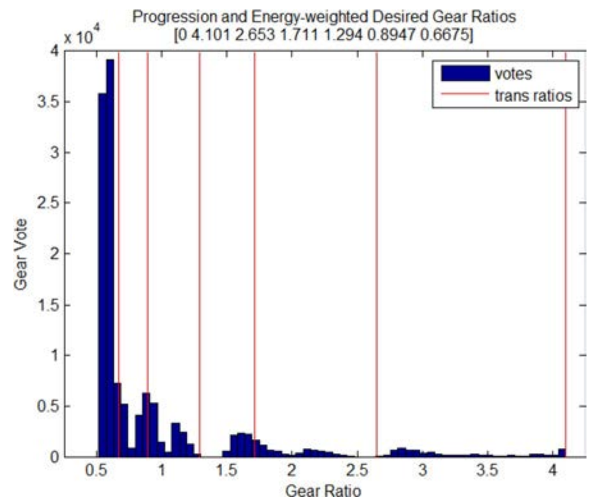

Figure 16. Energy weighted desired gear ratio histogram considering optimal and near optimal ratios

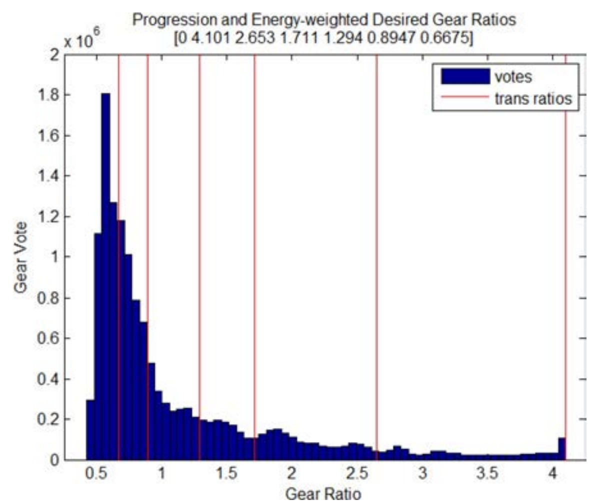

Figure 17. Energy weighted desired gear ratio histogram considering optimal, near optimal ratios and instantaneous power demand

On the basis of these charts alone it may not be possible to determine optimal ratios, although after experimenting with tweaking the ratios a little bit one way or the other, it was found the improvements were modest at best and sometimes the results were slightly worse. Considering instantaneous power (as shown in Figure 11) instead of average power as well as minimum and maximum desired ratios the histogram in Figure 17 is obtained which ultimately helps to explain the relative insensitivity to variations in gear ratio.

Considering that each gear must cover a range of operation and that the change in ratio preference is generally gradual (for this drive cycle) it can be seen that once a reasonable set of ratios is applied there is not much change in preference for small variations in ratio. Different drive cycles have different results, for example, for the NEDC drive cycle a fairly strong preference is seen that could lead to ideal ratios for the top two or three gears of the transmission, as may be seen in the Appendix. The gear ratios as depicted in this case cover the high preference region of the histogram quite nicely from around 2.6:1 down. The overall trend in this chart is a higher preference for lower gear ratios and especially something on the order of about 0.5:1 in this case. Since the transmission's top gear is limited to around 0.67:1 one might naturally consider an alternative final drive ratio in an attempt to satisfy the overall gearing required for optimal carbon emissions. Such a study is performed in the following section.

Carbon Emissions and Final Drive Ratio

The effect of varying the final drive ratio over a range on combined city-highway carbon emissions is shown in Figure 18 and Figure 19 for six- and eight-speed transmissions respectively.

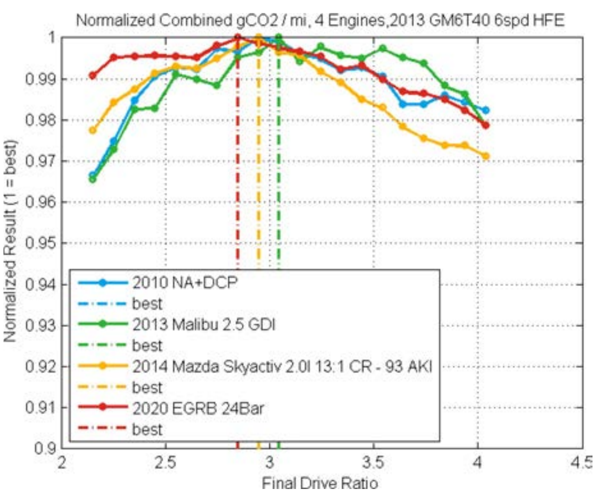

Figure 18. Six-speed AT carbon emissions sensitivity to final drive ratio, the baseline engine and the Mazda engine were optimal at the same final drive ratio

Figure 18 and Figure 19 show a larger potential impact of varying the final drive ratio on carbon emissions compared with varying the ratio progressions (Figure 3 and Figure 4). For each engine there appears to be a range of final drive ratios that provide carbon emissions within 1% of peak or less. It can be seen from Figure 18 that a lower ratio than the baseline was preferred regardless of engine but that improvement was modest - less than 1%. Extreme values of final drive ratio cause a pronounced increase in carbon emissions as might be expected. Favored final drive ratio varied little by engine, compared with the range of possible values, and the overall trends are very similar between the six- and eight-speed transmissions but with the six-speed showing a higher sensitivity to the highest and lowest final drive ratios.

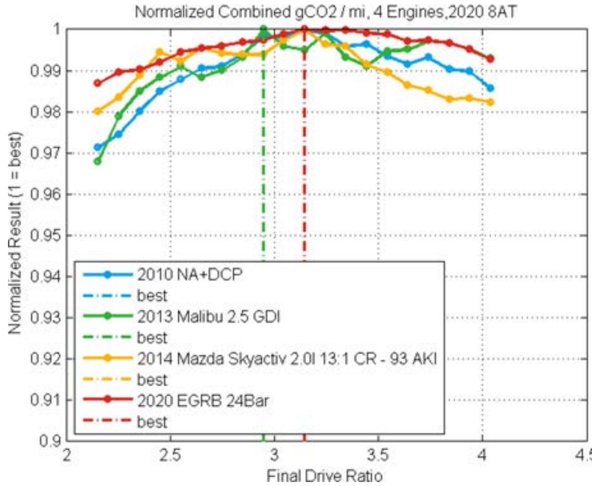

Figure 19. Eight-speed AT carbon emissions sensitivity to final drive ratio, the Malibu had the lowest preferred ratio, the other engines all preferred the ratio shown by the red vertical line.

Given the sensitivity of carbon emissions to final drive ratio (versus progression) and the relative ease of altering it for a particular application (compared with designing an entirely new gear set), it makes sense that a common set of transmission ratios may be applied across a range of vehicles while tuning the final drive ratio to the application. For example, the GM6T40 used in the Chevy Malibu has a 2.89:1 final drive ratio while the sister Ford 6F35 transmission in the FWD Escape uses a 3.21:1 final drive.

Combined Effects of Ratio Progression and Final Drive Ratio

The combined effects of ratio progression and final drive ratio can be considered to see if there are any positive or negative synergies between them. For brevity, combined effects will only be considered for the Mazda engine and a six-speed transmission, although plots for the eight-speed transmission are available in the Appendix. In the normalized gCO_2 / mi plots presented here the results are normalized to the operating point which has the lowest gCO_2 / mi such that higher numerical values represent lower carbon emissions.

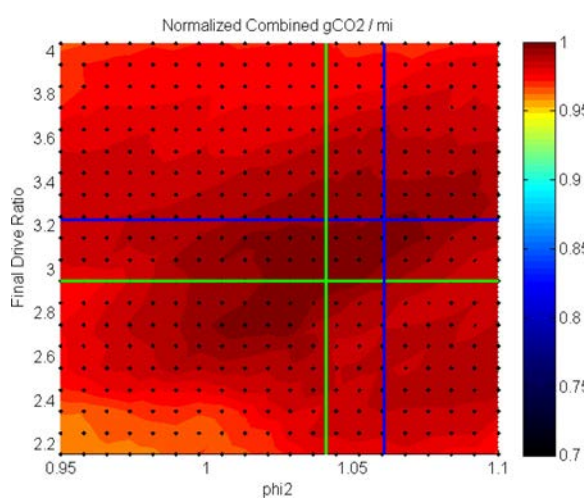

Figure 20. Normalized combined UDDS/HFET carbon emissions as a function of φ_2 and final drive ratio

Figure 20 shows the normalized combined city/highway carbon emissions as a function of φ_2 and final drive ratio. The intersection of the blue lines represents the optimal point from Figure 3 with a φ_2 of approximately 1.06 and final drive ratio of 3.23:1. The intersection of the green lines represents the optimal point from Figure 18 with a φ_2 of approximately 1.04 and final drive ratio of 2.95:1. For reference, the baseline six-speed transmission and final drive would be represented by the intersection of the vertical green line and the horizontal blue line, and is not a poor operating point. There are certainly a range of combined φ_2 / final drive ratio values that are capable of reaching equivalent carbon emissions. Combined carbon emissions increase at the highest and lowest final drive ratios, as seen previously. To understand the impact on combined carbon emissions, each phase of the drive cycle can be considered separately.

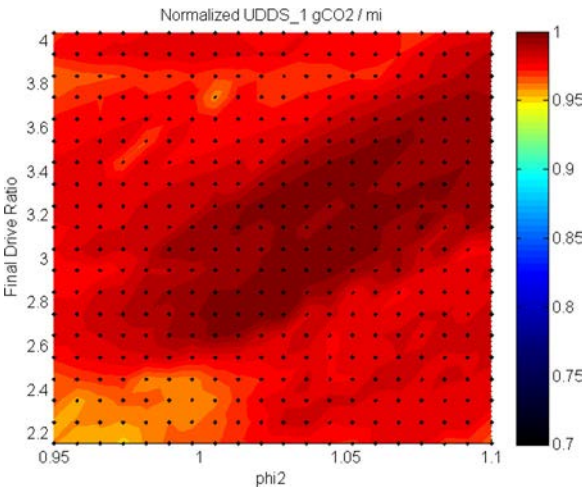

Figure 21. Normalized UDDS phase 1 carbon emissions as a function of φ_2 and final drive ratio for the Mazda engine and a six-speed transmission

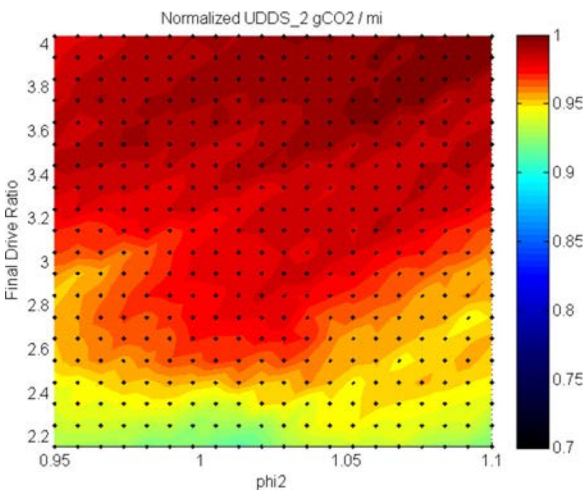

Figure 22. Normalized UDDS phase 2 carbon emissions as a function of φ_2 and final drive ratio for the Mazda engine and a six-speed transmission

Figure 21 and Figure 22 show normalized carbon emissions for the UDDS phase 1 and phase 2 respectively. Here the effect of low final drive ratios on UDDS phase 2 can be seen, which is the major contributor to the combined carbon emissions increase at low ratios seen in Figure 20.

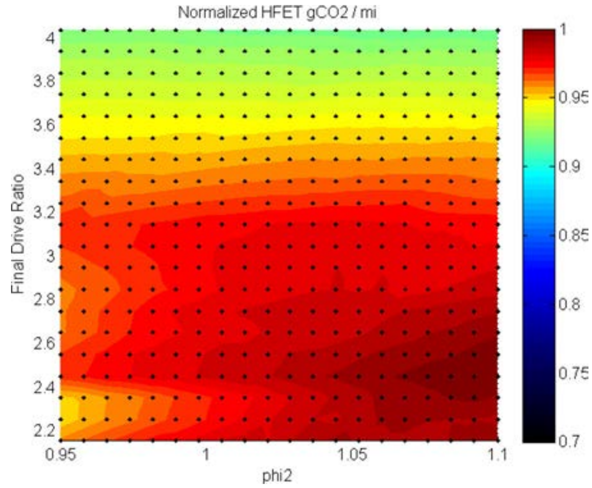

Figure 23. Normalized HFET carbon emissions as a function of φ_2 and final drive ratio for the Mazda engine and a six-speed transmission

Figure 23 shows the results for the HFET drive cycle. Here the effect of high final drive ratios can be seen, which is the major contributor to the combined carbon emissions increase at high ratios seen in Figure 20. Carbon emissions improves as final drive ratio reduces with a slight preference for higher φ_2 values where the top gears are closer in ratio.

There are a wide range of reasonable φ_2 and final drive ratio values to choose from when considering overall combined carbon emissions, in this case the optimal range appears to be about 1.0 to 1.07 for φ_2 and 2.75 to 3.30 for final drive ratio.

Performance Metrics

Carbon emissions are not the only consideration in vehicle design and drivability and performance must also be evaluated. Given the wide range of progressions and final drive ratios that provide reasonable carbon emissions there should be options available to also provide reasonable acceleration performance and drivability without large carbon emissions tradeoffs.

The 2017-2025 LD GHG rule assumes carbon emissions standards based on vehicles that perform no worse than 2008-era vehicles. In order to build “performance neutral” vehicle packages, performance metrics must be defined. A performance drive cycle is used to measure:

- 0-60 time
- ¼ mile time
- ¼ mile speed
- 30-50 passing time (full acceleration from 30 MPH cruise)
- 50-70 passing time (full acceleration from 50 MPH cruise)

From these metrics, for this study, overall performance is calculated from the sum of the 0-60 time, ¼ mile time and passing times. Using the combined times reduces the sensitivity to any individual metric since some of them can be extremely sensitive to small changes in driveline configuration as will be seen below. Our target tolerance for packages to be considered performance neutral is baseline +- 5%. Of these metrics, ¼ mile time and speed are the least sensitive to variations in φ_2 and final drive ratio, being primarily determined by overall power to weight ratio.

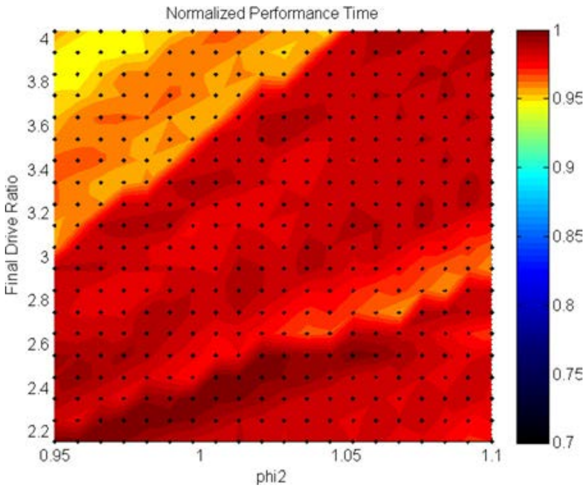

Figure 24. Normalized overall performance time for the Mazda engine and a six-speed transmission

Figure 24 shows the normalized combined performance time for the Mazda engine and a six-speed transmission. Performance was held fairly constant over the set of driveline configurations, as desired, although there is a slight bias towards higher performance at the lower final drive ratios and higher phi2 values due a slight over-scaling of the engines towards the lower right hand side of the plot (this trend is more easily observed in Figure 26). Overall, the results are relatively insensitive to φ_2 although there is some relationship between φ_2 and final drive ratio, as may be seen in the upper left quadrant of the figure. As φ_2 increases, gear ratios decrease and this can be offset by raising the final drive ratio, hence the southwest to northeast trend in this and following charts.

Individual metrics can be examined to see their trends as a function of progression and final drive ratio.

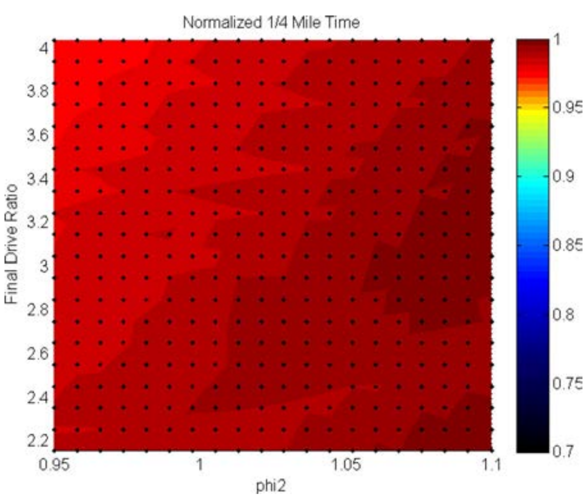

Figure 25. Normalized ¼ mile time for the Mazda engine and a six-speed transmission

Figure 25 shows that ¼ mile time shows little sensitivity to final drive ratio or φ_2, all the results are within a 5% range.

Of all the performance metrics, ¼ mile speed is (generally) the least sensitive to powertrain configuration, and is depicted in Figure 26. This is one reason why it's not part of the combined performance metric - it's not very informative as far determining transmission and final drive ratios although it can still be compared with the baseline vehicle to make sure high speed performance hasn't degraded in some way. In this case there's a slight trend towards improvement from the upper left corner to the lower right corner of the plot, this trend follows the engine scaling, which is approximate, as described later, and which increases along the same axis. The lowest performing ¼ mile speed case was still within 4% of the reference powertrain and the best cases showed a 3% improvement.

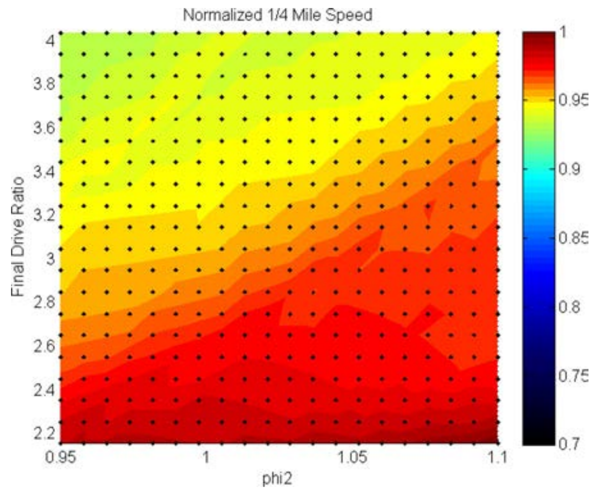

Figure 26. Normalized overall ¼ mile speed for the Mazda engine and a six-speed transmission

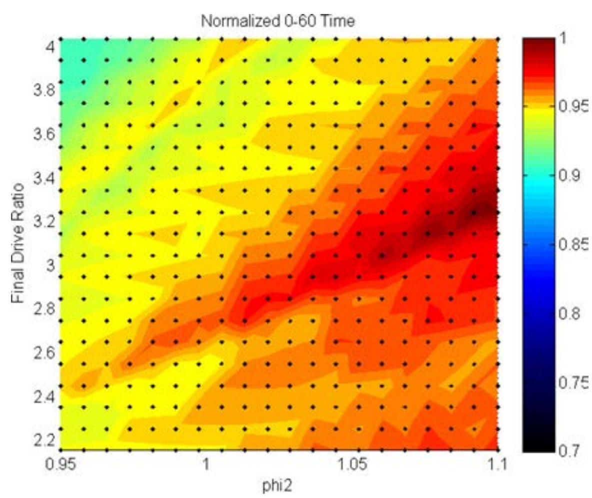

Figure 27. Normalized 0-60 time for the Mazda engine and six-speed transmission

Figure 27 shows the 0-60 results for the Mazda engine and a six-speed transmission. In the case of the six-speed transmissions considered here, 60 MPH is achieved in second gear so only one shift point is included in the 0-60 run and therefore there is little variation since first gear ratio is identical for all points on the chart. The worst performing case was about 0.5 seconds worse than the baseline and the best case was about 0.25 seconds faster and the average case was within 2.5% of the baseline. Results from the eight-speed transmissions showed similar trends.

The last two metrics, 30-50 and 50-70 passing time show much more interesting trends, as can be seen in Figure 28 and Figure 29.

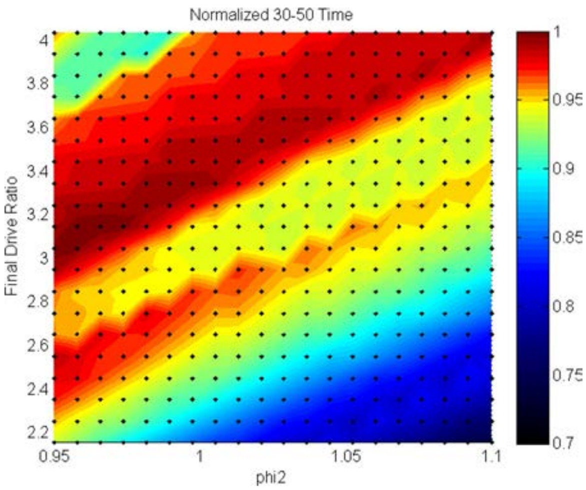

Figure 28. Normalized 30-50 passing time for the Mazda engine and six-speed transmission

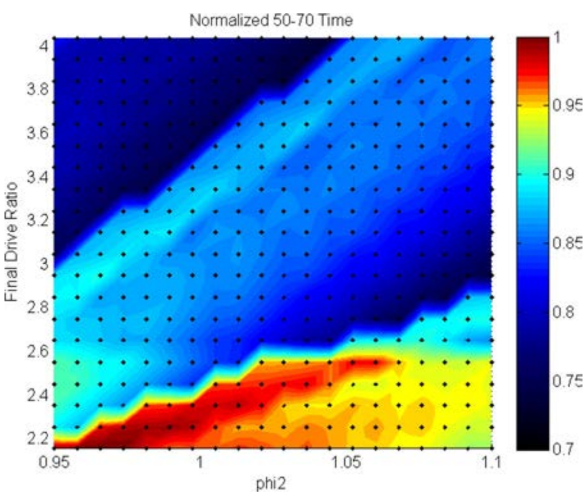

Figure 29. Normalized 50-70 passing time for the Mazda engine and six-speed transmission

The passing maneuvers are sensitive, on a normalized basis, in part because the times measured are fairly small - 30-50 time is in the range of 5 seconds and 50-70 time is under 8 seconds - and thus can be influenced by small changes in absolute time due to downshift delays or other factors. In addition, the passing times are highly sensitive to “cusp” events - downshifts that may or may not occur based on being right on the edge of meeting or not meeting the maximum allowed downshift speed requirement. Within the range of results one can see the trend is higher final drive ratios provide better performance until the cusp is crossed and performance degrades again. For this reason, these metrics are only used as part of the combined total performance time. In Figure 29, while it may appear as though performance for the majority of the cases may be inferior to the baseline, a closer inspection reveals that the best points are in fact almost 30 % faster than the baseline and the average point was about 10% faster. The worst performing case was only about 4% slower.

Scaling Engines for Performance Neutrality

When considering a range of engines and transmissions for a particular vehicle class and designing for performance neutrality, a method must be devised to scale engine performance for the particular application. For this study, engine displacement was scaled by estimating power at the wheels over a range of vehicle speeds and adjusting engine displacement to match the average power of the baseline configuration (as described in the Modeling Process section) over the reference speed range.

The approach taken for this paper is to approximate the power at the wheels using iterative calculation rather than using iterative simulation which would provide a more precise result while also increasing the simulation time, perhaps significantly. For this reason the performance is not quite held perfectly neutral, however the results are still quite reasonable. For example, for the Mazda with a 6AT, the sum of the performance metrics was within 3.5% of the baseline for all 400 runs shown in Figure 24. Future work will develop an iterative simulation approach although it is expected that the net effect on modeling results should be small.

For each vehicle speed over the reference speed range and each gear, transmission input and output speeds are calculated. Engine torque is determined for each vehicle speed based on the transmission input speed and the engine’s max torque curve. Torque and power at the wheels for each gear are calculated based on the engine torque and overall gear ratio. Engine redline speed (as determined by the minimum of the highest engine speed with at least 98% of full engine power available or a speed 650 RPM less than the highest speed point on the maximum torque curve) is taken as the transition point from one gear to the next and the average power over the speed range is calculated.

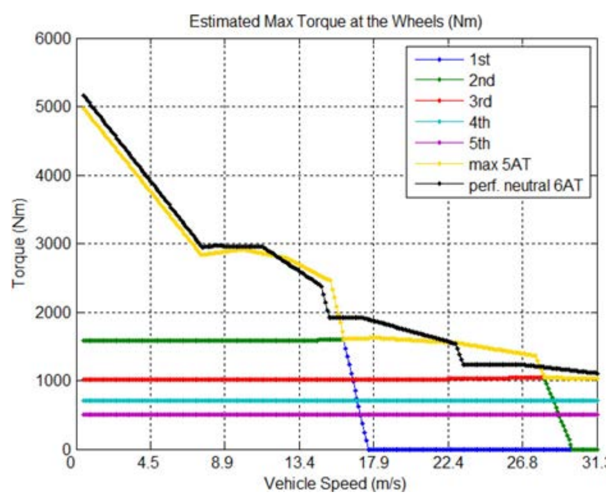

Figure 30. Estimated torque at the wheels for a 6AT and baseline 5AT configurations

The estimated maximum wheel torques for the baseline with 5AT and Mazda with 6AT are shown in Figure 30. The scale factor for the Mazda engine and six-speed transmission was 1.09, so the engine was upsized 9% to achieve comparable performance. The black curve represents the peak torque of the new powertrain configuration versus the yellow torque curve of the baseline configuration.

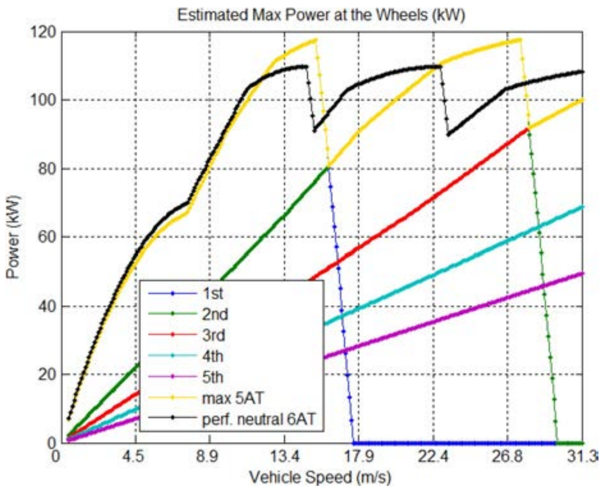

Figure 31. Estimated power at the wheels for a 6AT and baseline 5AT configurations

The estimated maximum wheel power for the baseline with 5AT and Mazda with 6AT are shown in Figure 31. In this case the baseline powertrain has a higher peak power, as indicated by the yellow curve, but the six-speed powertrain has a more consistent power with lower peaks and higher valleys as shown by the black curve in the figure. For both configurations the average power over the speed range was approximately 89 kW.

When scaling the engine displacement, the engine's max torque curve is scaled linearly by the scale factor and the engine's BSFC map is modified by a polynomial adjustment curve that has the effect of adjusting efficiency based on changes in engine displacement [5]. Lowering displacement reduces peak efficiency relative to the original engine while potentially providing overall carbon emissions benefits by operating at higher relative load. Increasing displacement increases peak efficiency slightly but generally increases fuel consumption by operating at lighter relative loads.

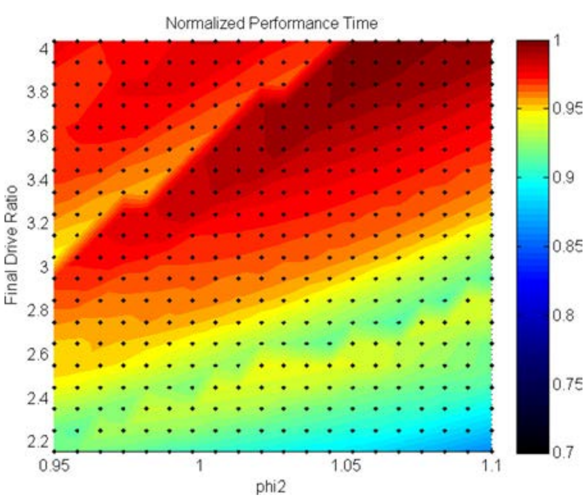

Figure 32. Normalized performance time without performance neutrality

Without adjusting the engine displacement for performance neutrality, a significantly different total performance trend emerges, as shown in Figure 32 compared with Figure 24 - performance increases with final drive ratio as would normally be expected. Interestingly, the overall carbon emissions trends and sensitivities were quite similar between the performance and non-performance equalized cases.

Adjusting Shift Strategy

For this and other modeling work, the ALPHAshift algorithm is used to generate shifts points during simulation based on driver demand and a cost map based primarily on an engine's fuel consumption map.

The ALPHAshift strategy, while adapting to each engine's BSFC performance, still benefits from some adjustment for each engine in order to provide reasonable operating speed ranges. Many approaches are possible. For this study, with reference to our previous description of ALPHAshift [9], the following parameters were adjusted based on the engine used for each configuration:

- min_speed_radps: set to the lowest speed which provides 12% of maximum available engine power for 1st gear up to the speed which provides 18% of maximum available engine power for top gear
- upshift_min_speed_radps: set to the greater of the lowest speed which provides 15% of maximum engine power or the min_speed_radps + 10 radians/sec
- max_speed_radps: set to the highest speed which provides 98% of full engine power, or to a speed 52.4 radians/sec slower than the zero torque maximum speed of the engine (last point of the full throttle torque curve), whichever is lower
- kickdown_trigger_ratio: scaled from 1.5 for the first half of available gears down to 1.2 for the highest gear

These parameter settings were based on observed shift points from vehicles used in previous validation and benchmarking work and should provide reasonable shift points. All other ALPHAshift parameters were set to shared values for the various engines.

Conclusions

For this paper, carbon emissions and performance effects of transmission gear count, ratio progression, final drive ratio and engine type were modeled. An energy-based analysis of preferred gear ratios was presented based on average and instantaneous power and methods for adjusting powertrain packages for performance neutrality were discussed. An adaptable set of ALPHAshift parameters was presented in order to adjust shift points automatically for different engines.

As a result of this study, one may reach the following conclusions:

- There is no single value of φ_2 that optimizes carbon emissions across multiple engine and transmission types
- All other things being equal, carbon emissions are fairly insensitive to φ_2 for a range of engine and transmission combinations
- An energy-based analysis of the engine's preferred operating points may be helpful in understanding the response of carbon emissions to varying gear ratios
- Final drive ratio can have a larger impact on carbon emissions than progression but there are a range of ratios that provide reasonable carbon emissions
- UDDS carbon emissions are relatively insensitive to φ_2 and variations in final drive ratio across a wide range of values
- HWFET carbon emissions are sensitive to high final drive ratios but relatively insensitive to φ_2
- Individual performance metrics can show high sensitivity to ratio progression, final drive ratio and shift calibration, in particular the 30-50 and 50-70 MPH passing times

Many interesting comparisons were beyond the scope of this paper and the data presented here are only a sample of the study results, but future work may include a study of the effects of transmission spread for CVTs and gear count and spread for step gear transmissions. Comparisons between step gear transmissions based on gear count or transmission architecture (DCT versus planetary AT, for example) might also be investigated with an eye towards performance tradeoffs such as gearing versus torque multiplication, etc.

Work on ALPHA is ongoing as data on current and emerging vehicles and technologies continues to be collected and analyzed.

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Definitions/Abbreviations

ALPHA - Advanced Light-Duty Powertrain and Hybrid Analysis modeling tool

AT - Automatic Transmission

DCP - Dual Cam Phasing

FTP - Federal Test Procedure, also refers to the drive cycle created by driving the UDDS then repeating its first 505 seconds.

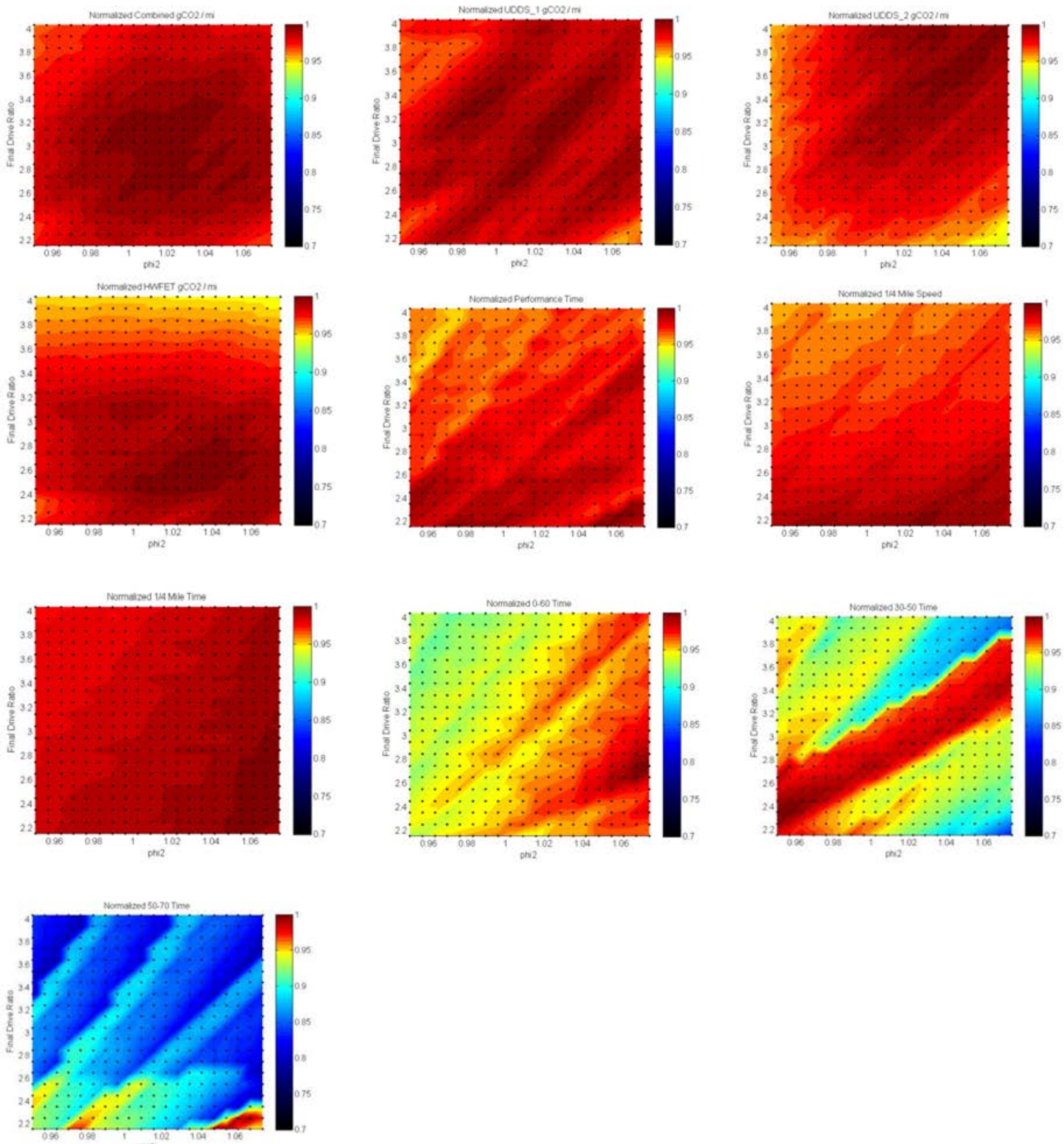
HWFET - EPA's Highway Fuel Economy Test, the "highway" cycle

NVFEL - EPA's National Vehicle and Fuel Emissions Laboratory located in Ann Arbor, MI

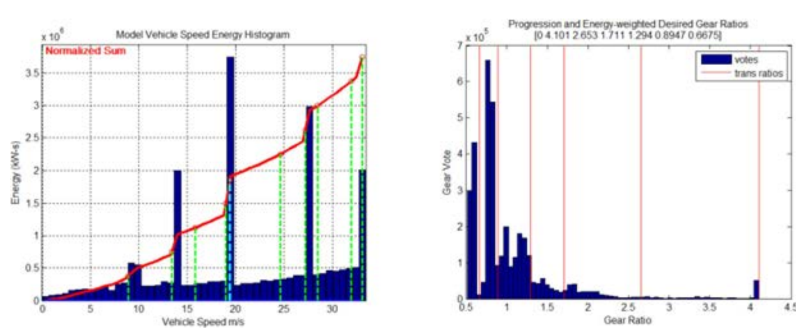
UDDS - EPA's Urban Dynamometer Drive Schedule, the "city" cycle

APPENDIX

Eight-speed sensitivity plots for the Mazda SkyActiv engine:



NEDC Energy and desired gear ratio plots:



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