

Simulated Real-World Energy Impacts of a Thermally Sensitive Powertrain Considering Viscous Losses and Enrichment

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ABSTRACT

It is widely understood that cold ambient temperatures increase vehicle fuel consumption due to heat transfer losses, increased friction (increased viscosity lubricants), and enrichment strategies (accelerated catalyst heating). However, relatively little effort has been dedicated to thoroughly quantifying these impacts across a large set of real world drive cycle data and ambient conditions. This work leverages experimental dynamometer vehicle data collected under various drive cycles and ambient conditions to develop a simplified modeling framework for quantifying thermal effects on vehicle energy consumption. These models are applied over a wide array of real-world usage profiles and typical meteorological data to develop estimates of in-use fuel economy. The paper concludes with a discussion of how this integrated testing/modeling approach may be applied to quantify real-world, off-cycle fuel economy benefits of various technologies.

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INTRODUCTION

A number of studies have demonstrated large negative impacts on fuel consumption and emissions due to colder ambient temperatures. Fuel enrichment and spark timing adjustments for catalyst light-off strategies, high rates of heat transfer, and non-linear viscosity of engine lubricants combine to negatively affect powertrain and drive cycle efficiency in cooler ambient conditions [1, 2, 3, 4, 5, 6]. These effects are also present in hybrid powertrains and may be magnified due to the powertrain operating at lower than optimal temperatures [7, 8, 9]. Additionally, regional drive cycle variability plays a large role in overall vehicle efficiency [10]. Together, these factors represent important real-world considerations for powertrain design and efficiency.

In 2012 new light-duty fuel economy standards were set for the North American market. The U.S. Environmental Protection Agency (EPA), National Highway Traffic Safety Administration (NHTSA), California Air Resources Board (CARB), original equipment manufacturers (OEMs), non-government entities, and other stakeholders collaborated to define 2012-2025 national fuel economy and greenhouse gas standards. The program called for a 4%-5% annual improvement in fuel economy with the final car and light-duty

truck standard set to 54.5 miles per gallon (mpg) [11, 12]. Vehicle manufacturer fleet fuel economy certification for this Corporate Average Fuel Economy (CAFE) regulation is determined via a combination of on- and off-cycle methods. On-cycle certification is evaluated from weighted measured testing results of EPA's test cycles [13]. On-cycle certification can be supplemented with off-cycle credits representing estimated real-world vehicle efficiency not captured by on-cycle testing. The EPA and NHTSA currently recognize three pathways in which technologies can qualify for off-cycle credit:

1. **On-Table** - An OEM gets a predefined credit value for technologies that are included in the credit table.
2. **5-Cycle** - An OEM uses a predefined 5-cycle test methodology to determine credit value.
3. **Alternative Method** - An OEM may develop and justify a test methodology and credit value using real-world data.

The first two pathways are set and standardized. The third pathway, the alternative method, requires the use of real-world data in which OEMs and suppliers can demonstrate the viability of the technology

and receive appropriate credit for its implementation. The work presented in this paper may serve as a framework to better understand and assess real-world fuel economy impacts for certain technologies.

The approach outlined in this paper includes experimental vehicle chassis dynamometer tests conducted over a wide range of temperatures and loads, coupled with simplified models developed from the experimental data, that are used to demonstrate real-world economy over a broad range of drive cycles and ambient temperatures. Experimental dynamometer tests were conducted at Argonne National Laboratory (ANL) on a 2011 Ford Fusion (2.5L, 6-speed automatic) over a broad range of temperatures. From this data, simplified response surface methodology [14] and lumped capacitance thermal models were developed to simulate vehicle efficiency. These simplified models were integrated into the National Renewable Energy Laboratory's (NREL) Transportation Secure Data Center (TSDC) [15] and coupled with U.S. typical meteorological data [16]. Goodness of fit statistics for the simplified modeling approach are presented with fleet fuel economy impacts quantified for multiple large combinations of real-world drive cycles and historical ambient conditions.

APPROACH AND TEST SETUP

Tests were conducted at ANL's Advanced Powertrain Research Facility (APRF) four-wheel-drive dynamometer test cell [17]. This dynamometer test facility is designed to handle light- to medium-duty sized (maximum 6,350 kg) vehicles capable of producing up to 373 kW of wheel power. The test cell is EPA 5-cycle capable with ambient temperature capability from -7°C to $+36^{\circ}\text{C}$. Additionally, the test cell can go to colder temperatures (for this work -17°C). A vehicle fan located at the front of the test cell provides cooling airflow to the vehicle and its powertrain during testing. The simulation fan is a standard vehicle speed-matching fan that fulfills the test regulations for the SC03 air-conditioning (A/C) test. The cell also contains solar lamps simulating a multitude of solar loading conditions experienced in the real-world environment, with a typical target solar loading of 850 W/m^2 at the base of the windshield and/or rear window. The vehicle is restrained to the dynamometer by a tie-down system. Two posts are bolted to T-slot rails on the floor, each post containing a height-adjustable system to restrain the vehicle and to remove vertical load from the wheels on the dynamometer roll.

The test cell contains emission benches capable of bag measuring the criteria emissions total hydrocarbons, oxides of nitrogen, carbon monoxide, soot, as well as carbon dioxide for cycle fuel economy. Additionally a turbine wheel fuel cart is used to accurately measure fuel. A data acquisition system is integrated that allows for a multitude of controller area network, analog, and digital signals to be collected. All data is collected and time aligned at 10 Hz frequency. The test vehicle and APRF test facility is shown in [Figure 1](#).



Figure 1. 2011 Ford Fusion test vehicle on APRF dynamometer for testing.

The vehicle was extensively instrumented to capture pertinent thermal and energy/power nodes. The engine was instrumented with K-type thermocouples to determine coolant, oil, inlet/exhaust temperature at various locations (exhaust port to pre-, mid-bed, and post-catalyst). Accuracy lies within 5°C below 1250°C of measurement, decreasing to 1.1°C at 100°C . Additional K-type thermocouples were included in the transmission and vehicle interior. A strain based torque measurement system was installed on the engine flex plate and half shafts enabling measurement of engine out, transmission input, and transmission output torques and power levels. For the half shafts, full scale torque measurements were set at 3400 Nm with a maximum static measured error of 0.2%. The flex plate full scale torque was set to 500 Nm with a maximum static measured error of 0.2%. Flow measurements for fuel are calibrated within 1.8% of measured error. Engine speed and transmission gear were recorded via CAN signals. [Figure 2](#) represents the pertinent instrumentation and nodes completed for the work.

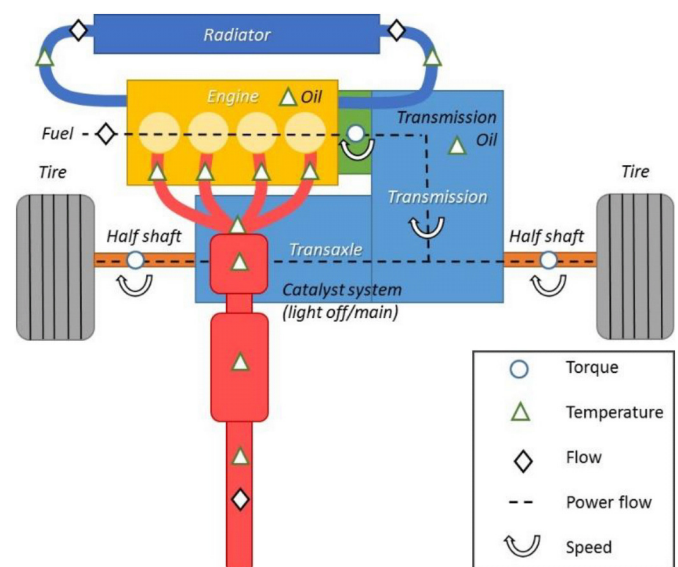


Figure 2. Test vehicle instrumentation layout

To minimize the number of tests completed while maximizing data fidelity, speed/load data from the vehicle was analyzed over the Urban Dynamometer Driving Schedule (UDDS), the Highway Fuel Economy Test (HWFET), and the aggressive US06 cycle. These tests were conducted at +20°C. Both the drive cycle velocities and engine speeds/loads are shown in Figure 4. By comparing the engine loading over the cycles it was determined that testing the vehicle using only the UDDS and US06 cycles would be sufficient in gathering a broad speed/load range, with the HWFET cycle speed/load ranges overlapping the other two cycles, therefore deemed unnecessary.

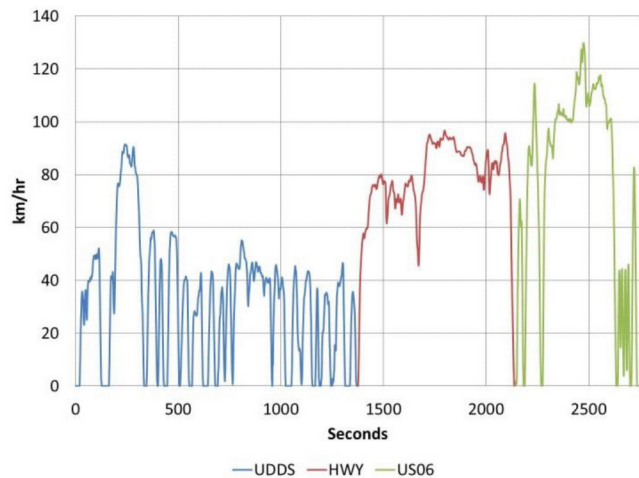


Figure 3. 2011 Ford Fusion velocity and speed/load points over the UDDS, HWFET, and US06 cycles.

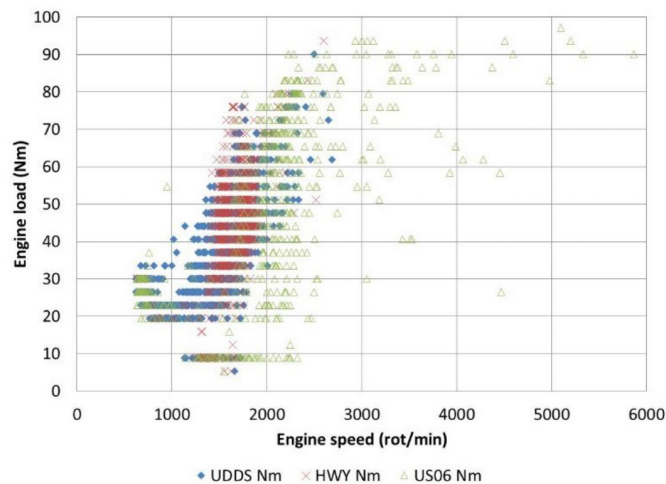


Figure 4. (Continued).

In addition to a wide range of driving behavior, the scope of the work required a broad sweep of thermal conditions for assessment. Both the UDDS and US06 cycles were tested at ambient conditions ranging from -17°C to +35°C. Each of these cycles was tested from cold and hot start conditions. Here, a cold start is defined as the vehicle soaking overnight at the test cell temperature. This allowed for data at cooler powertrain operating states to be collected for both high and low power levels. For hotter tests, the vehicle air conditioner was not used so that the powertrain thermal effects would not be masked by the increased A/C compressor load. For cold tests, the vehicle heater was used and set on the median level. After the

dynamometer test, cool-down data were recorded to determine the rate at which the coolant and lubricants cooled following peak operational temperature. Note that additional tests were conducted with the heater in the “on” and “off” positions, and no significant measurable load difference was recorded. Solar loading was not included in the testing as this has no impact on the component efficiency at a components given operational temperature (note: solar loading does, however, increases the rate of component warming thus impacts cycle efficiency). Table 1 lists the final testing matrix.

Table 1. Matrix of test conditions run at APRF (16 tests in total).

Drive Cycle	UDDSt2, US06x2, non-operational cool down
Start Condition	Hot Start, Cold Start
Test Cell Temperature	-17°C, -7°C, +22°C, +35°C

Results

Figure 5 and 6 contrast measured engine oil temperature and integrated cycle fuel consumption over two repetitions of the UDDS drive cycle with the test cell ambient temperature ranging from -17°C to +35°C. As shown, the oil temperature rises from the initial ambient cold-start temperature during the cycle to the final steady state powertrain operational temperature. At -17°C, the engine oil steady state temperature is approximately 18°C cooler than the steady state temperature achieved at +35°C ambient. This is due to the increased convective heat transfer from the powertrain at cooler temperatures. These data are indicative of the results seen for the higher-load US06 cycle (not shown for sake of brevity).

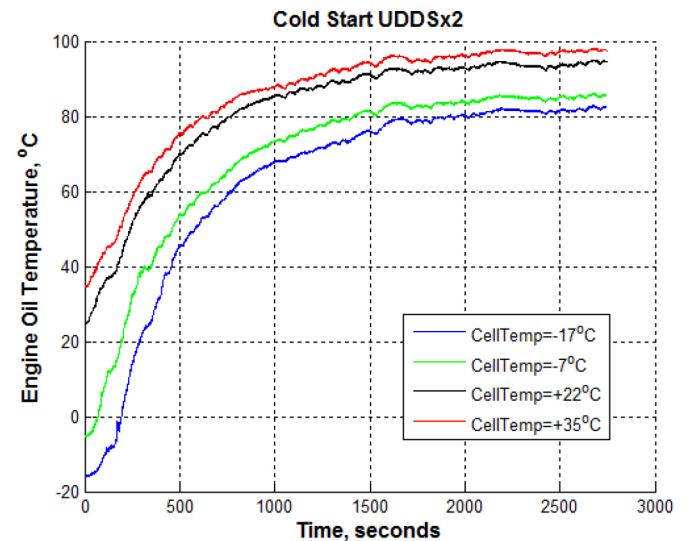


Figure 5. Engine oil warm-up temperatures (measured at dipstick) over two consecutive UDDS cycles following soak to four distinct ambient conditions.

Comparing these results in Figure 6, a significant variation in fuel consumption relative to the temperatures is evident. The increase in fuel consumption for the cold start UDDS cycle at -17°C relative to the +35°C cold start is approximately 22%. Although considered relatively warm, a 3% fuel consumption decrease at +20°C relative to the +35°C UDDS test exists. These results further demonstrate the importance of understanding the real-world thermal effects on

vehicle efficiency. This would assist in quantifying the effect that engineering solutions to reduce thermal losses could have on increasing vehicle efficiency.

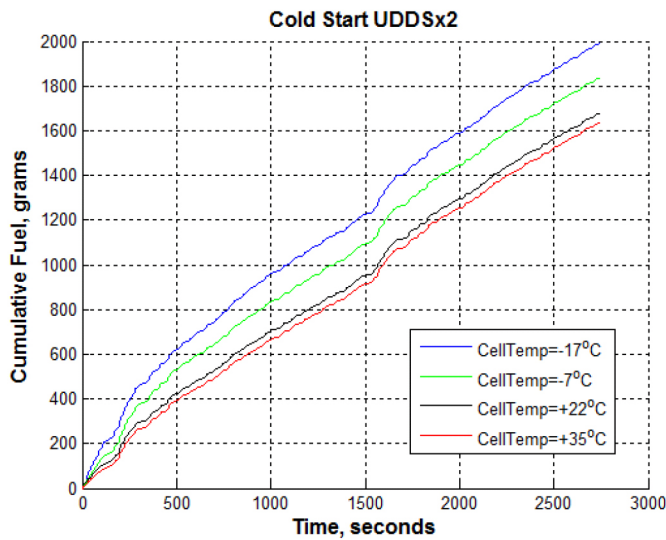


Figure 6. Cumulative fuel use over two consecutive UDDS cycles following soak to four distinct ambient conditions.

COMPONENT MODEL DEVELOPMENT

While many modeling approaches exist to estimate vehicle efficiency relative to road loads and powertrain component efficiency, less effort has been expended characterizing the efficiency of internal combustion engines with respect to thermal state. The focus of this work centered on developing a simplified methodology to understand thermal effects in real-world driving conditions without using overly complex models dependent upon accurate measurement of comprehensive component data. Not only are such data not generally available, but simulation of hundreds of thousands of real-world drive cycles from around the country using an unnecessarily complex model would currently be computationally intensive.

The following simplified component models provide a means for predicting component warm-up times and steady-state operating points sensitive to ambient conditions. The generalized approach involves response surface methodology models coupled with simplified lumped capacitance parameter-based models. Application of the Nelder-Mead non-linear optimization method solves for the unknown coefficients to minimize model error relative to vehicle measured results of fuel consumption over the drive cycles and thermal conditions listed in Table 1.

Engine Efficiency

A simplified model of the engine fueling rate as a function of engine power, engine oil temperature, and catalyst light-off was developed. The engine fueling rate is described by a third-order function response surface model with engine output power and engine oil temperature as model inputs [2]. The oil temperature in this case is a differential between the nominal operating temperature and the

current oil temperature. The catalyst model is a simplified exponential decay equation that takes into consideration the catalyst temperature until light-off. This equation adds a decaying amount of fuel until a certain catalyst temperature is reached. The engine fueling equations are shown here for reference:

$$fuel = f_1(P_{out}, T_{oil}) + f_2(T_{cat}) \quad (1)$$

$$f_1(P_{out}, T_{oil}) = a_0 + a_{1,1}P_{out}^3 + a_{1,2}P_{out}^2 + a_{1,3}P_{out} + a_{2,1}dT_{oil}^3 + a_{2,2}dT_{oil}^2 + a_{2,3}dT_{oil} + a_{3,1}P_{out}^3 dT_{oil}^3 + a_{3,2}P_{out}^2 dT_{oil}^2 + a_{3,3}P_{out}dT_{oil} \quad (2)$$

$$f_2(T_{cat}) = \max(0, a_1 * (e^{a_2 * (T_{cat} - a_3)} - 1)) \quad (3)$$

$$dT_{oil} = T_0 - T_{oil} \quad (4)$$

Following the Nelder-Mead solution of the coefficients, plots of the engine efficiency as a function of the engine oil temperature and power output were generated, as well as the time-based comparison between the model's predicted fuel flow and the actual measured data. As an example of the time-based comparison, Figure 7 shows results for a 250-second UDDS cycle segment conducted at -17°C . As these results show, the model matches relatively well with the actual fuel flow measurements recorded during testing.

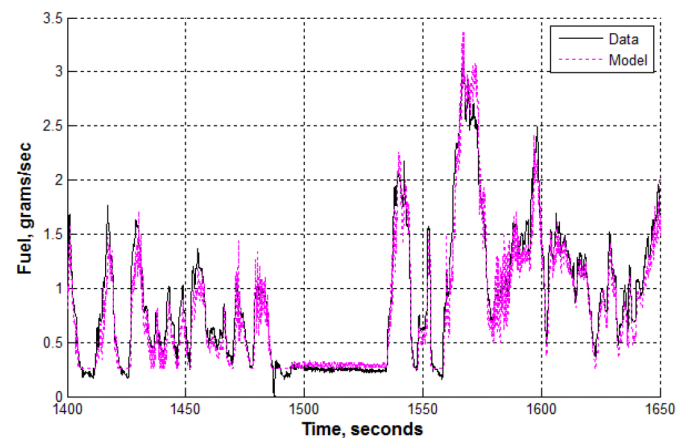


Figure 7. Example time series data of engine fueling rate (250-second section of UDDS). Measured test data from chassis dynamometer (solid black) overlaid with model estimate (dashed magenta).

Figure 8 contrasts the significant impact engine lubricant temperature has on efficiency. An island of optimal efficiency (approximately 36%) exists at power levels between 80 kW and 110 kW and an oil temperature of 100°C . Yet at identical power levels, as engine oil temperature decreases to -17°C , engine efficiencies fall to between 23%-26%, an approximate 33% decrease in overall efficiency for identical power outputs. This underscores the tremendous effect decreased ambient temperatures play on heat engine efficiency.