United States
Environmental Protection
Agency

Office of Transportation
and Air Quality

EPA420-P-05-001
February 2005

Fuel Consumption Modeling of Conventional and Advanced Technology Vehicles in the Physical Emission Rate Estimator (PERE)

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Executive Summary

This study proposes a modeling methodology for conventional and advanced technology vehicles including:

- Conventional motorcycles
- Conventional gasoline and diesel (light and heavy duty)
- Advanced gasoline internal combustion engine
- Advanced diesel internal combustion engines
- Hybrid electric and gasoline/diesel
- Hydrogen fuel cell and hybrid
- Hydrogen internal combustion engine

The model called PERE (Physical Emission Rate Estimator) is designed to support the new EPA energy and emissions inventory model, MOVES (MOtor Vehicle Emissions Simulator). PERE is a series of stand-alone spreadsheets, which can be run and modified by an informed user. It outputs Pump-to-Wheel (PTW) fuel consumption rates. The purpose of PERE is to fill data gaps in MOVES and to provide a tool to extrapolate to future projections of energy and emissions. The current version of PERE will model the fuel consumption from many of the advanced technologies in MOVES in addition to conventional vehicles. Criteria pollutant emissions will be modeled in a later version. If the user is knowledgeable of the vehicle parameters required in PERE, other technologies can be modeled that are not described in this report (e.g. alternative fuels, minimum/mild hybrids, etc.).

MOVES is currently being integrated with GREET (developed at Argonne Laboratory), which produces upstream Well-to-Pump (WTP) energy and emissions rates. The combination of PERE, MOVES, and GREET will provide a powerful comprehensive modeling suite for anyone requiring emissions inventory projections or life cycle (energy/emissions) studies for mobile sources into the future.

As the name implies, PERE uses physical principles to model propulsion systems in the vehicle. It is therefore built on a relatively strong foundation. It is based on a sound, yet elegantly simple model for the internal combustion engine. Modeling of hybrids is achieved by inserting a secondary power source (usually a battery/motor combination). Fuel cell vehicles are simulated by replacing the engine with an aggregate fuel cell system. PERE takes vehicle input parameters, then runs the vehicle through driving cycles defined by the user and outputs second-by-second fuel consumption rates. All of the processes are transparent to the user, which allows for easy updates.

The model is validated to four production conventional gasoline (it has been validated to more in previous publications), a number of motorcycles, buses and heavy-duty trucks. Also validations were performed on five advanced (and hybrid) vehicles by comparing the PERE output to the rated fuel economy figures. The model performs well, in most cases the predictions are within 10%, as the figures below show. This demonstrates the relative robustness of PERE to model a wide range of technologies.

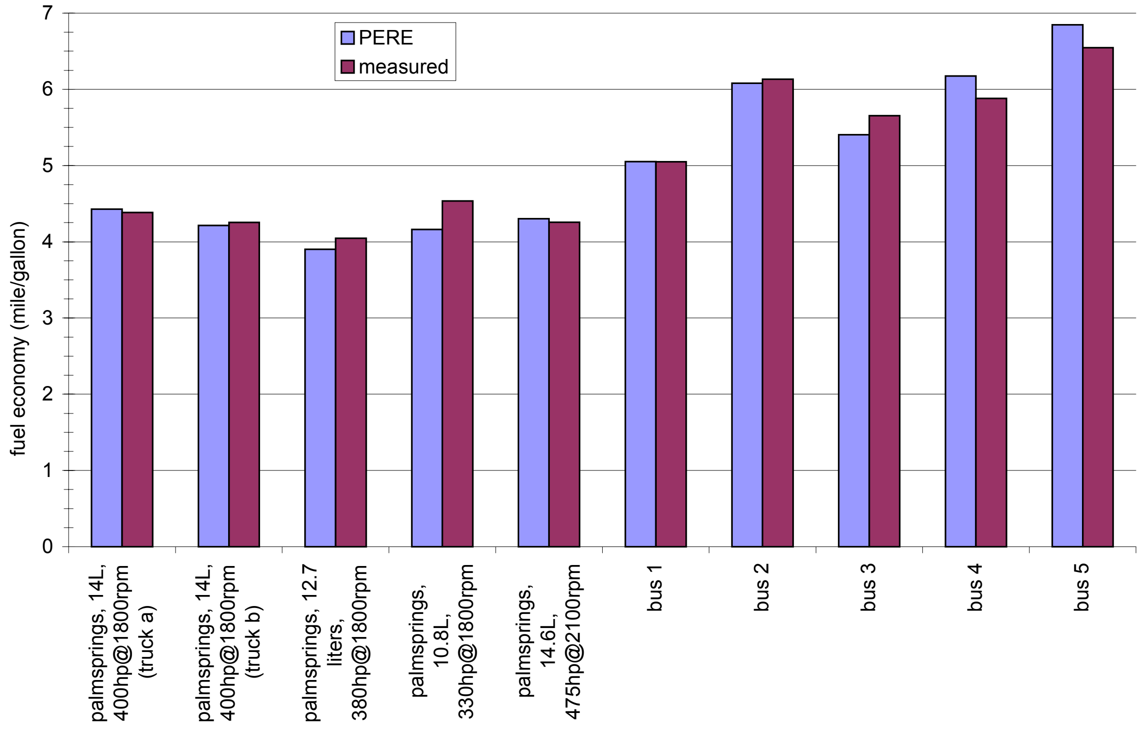


Figure ES1. Validation of PERE to heavy-duty trucks and buses as measured on-road.

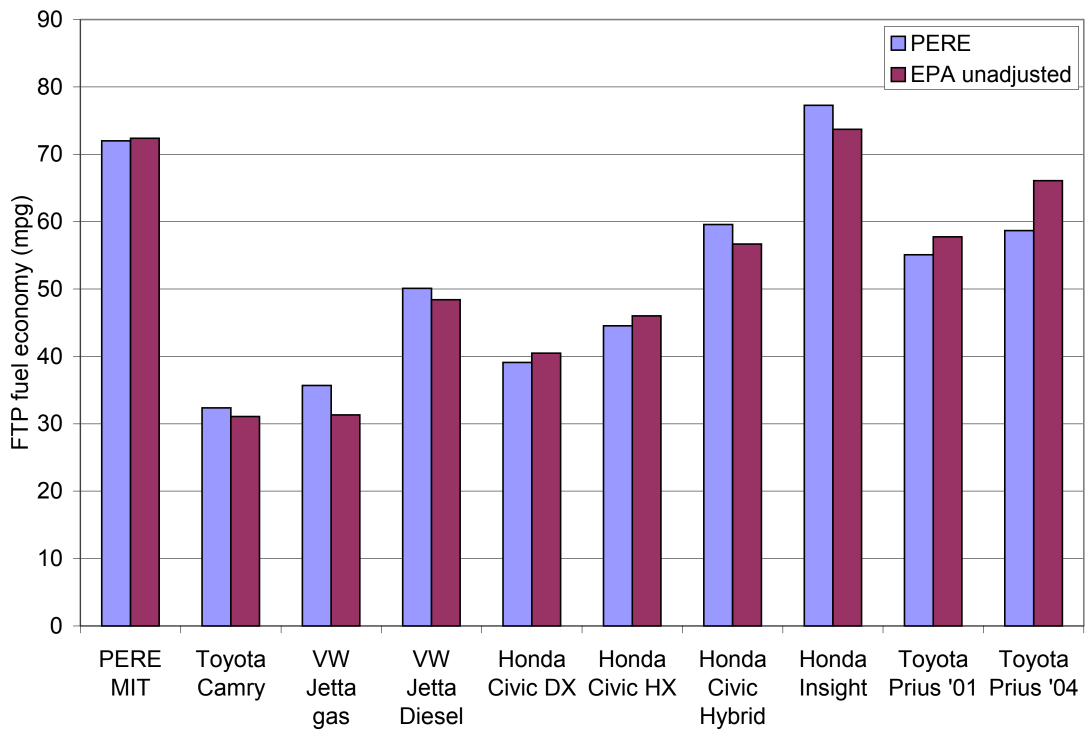


Figure ES2. Combined fuel economy (city/highway) estimates from PERE compared to vehicle rated values.

The fuel cell model describes a direct hydrogen PEM fuel cell and hybrid. It is validated to the fuel economy results from the Honda FCX vehicle. The model captures the fuel economy to within 5% of the measured value.

The report also includes a sensitivity analysis, which will help guide users to determine which parameters require more accuracy.

Finally, the report describes how PERE might be used to support MOVES. It provides guidance for how parameters may be estimated in order to perform future projections. It also describes a methodology by which the PERE output is integrated into MOVES.

Introduction

The EPA is tasked to develop models, which can calculate the inventory of emissions from mobile sources as well as estimate future projections. The model presently used for this purpose is MOBILE6. MOVES (MOtor Vehicle Emission Simulator) is the next generation model and will eventually replace MOBILE. MOVES is designed to be a data-driven model, so that the emissions rates are derived from direct measurements. However, there are substantial portions of the fleet where data is sparse or non-existent. A separate model is needed to fill these "data holes" in MOVES. Moreover, a model is required that can capture the behavior of advanced technology vehicles, which behave very differently from conventional vehicles.

The goal of this project is to develop a model, which can capture the behavior of a variety of conventional as well as future or advanced technology vehicles for the purpose of estimating emissions inventories. The Physical Emissions Rate Estimator (or PERE) is expected to generate fuel (or energy) consumption rates for MOVES. So, in its present form, the "E" in PERE should stand for "energy" more than "emissions". The model will eventually characterize criteria pollutants [Nam, 2003], but fuel consumption is seen as the first step toward validation of the model. Modeling this "pump-to-wheel" fuel consumption also allows for a life cycle, "well-to-wheel", model if PERE output is combined with GREET, which models upstream ("well-to-pump") emissions [Wang, 2001].

The primary users of PERE are the MOVES developers and users. However, PERE will be made available to the public, when others may find it useful for modeling and comparing fuel consumption from conventional or advanced technology vehicles.

The current report describes the modeling of fuel (and energy) consumption of conventional and advanced technology vehicles using physical principles. PERE is based on a number of models that exist in the literature (the references are peppered throughout this document). The model is developed for the following conventional technology types (light and heavy duty): gasoline, diesel, and motorcycle. Previous publications have already presented results for many light duty gasoline vehicles. PERE also models advanced technologies such as: advanced gasoline/diesel, "moderate" parallel hybrid, "full" parallel hybrid, hydrogen internal combustion engine, electric only, and fuel cell vehicles. Validations are presented where available. These are the technologies, which seem to have near or long term possibilities for market penetration. The number of other advanced technology combinations that could exist are enormous. Such examples are hydraulic hybrids, series hybrids, plug-in hybrids, etc. This report does not model all of the possible combinations, however the hope is to demonstrate that PERE is sufficiently robust and/or "generic" such that it can accommodate many of these technologies if the need arises.

Though the model is based on mathematical and physical principles, it is intended to be aggregate, and is not appropriate for engineering or product design. Thus it is designed to model a "typical" vehicle of technology type, rather than a specific vehicle. However, the validations are conducted on specific vehicles for a general comparison. Due to the approximations made in some of the parameters of the model, it is not expected to accurately capture fuel consumption of specific vehicles better than within 10% of measured values.

The basic mechanism of PERE involves calculating the road load energy required to move the vehicle mass along a driving trace, then distributing that energy demand to the various vehicle components (engine, electric motor, fuel cell, etc). The energy components are modeled using overall systems efficiencies. The validations in this report are conducted mainly with certification fuel economy data (though there are two modal test results presented). Where appropriate, simplifications and approximations are made using physical constraints, or based on publications in the literature. The model currently only models hot running operation. Cold start factors are handled in MOVES separately, and will only briefly be described in this report.

For the purposes of modeling the future fleet, our goal is to allow as many of the significant assumptions as possible to be under the control of the user. However, default values will be presented in this paper. An attempt will be made to justify the assumptions in each case.

The report begins by describing conventional vehicles, both gasoline and diesel (light and heavy duty). It then goes on to briefly examine advanced engines. Hybrid vehicles are modeled and validated, followed by fuel cell vehicles. The report caps off with a sensitivity analysis and describes how PERE rates might feed into MOVES. Each section is broken up into subsections representing the primary parts of the model: Vehicle, engine (or fuel cell), transmission, and motor.

For simplicity, PERE is currently in a spreadsheet format and should be usable for most users, who have a nominal amount of background information on hybrids and fuel cells (as well as spreadsheet skills). The final form that the model takes for the integration may be different from what is presented in this report.

Section I: Road Load Basics

Vehicle Model

As Figure 1 shows, the model is basically a "backwards-looking" model in that it takes a driving cycle (second-by-second speed vs. time) input and outputs the energy consumption required to follow the drive trace. The power demand is based on overcoming inertia, road grade, tire friction, and aerodynamic loss. This road-load methodology was mainly introduced by Sovran and Bohn [1981]. This is essentially the VSP equation shown in numerous publications.

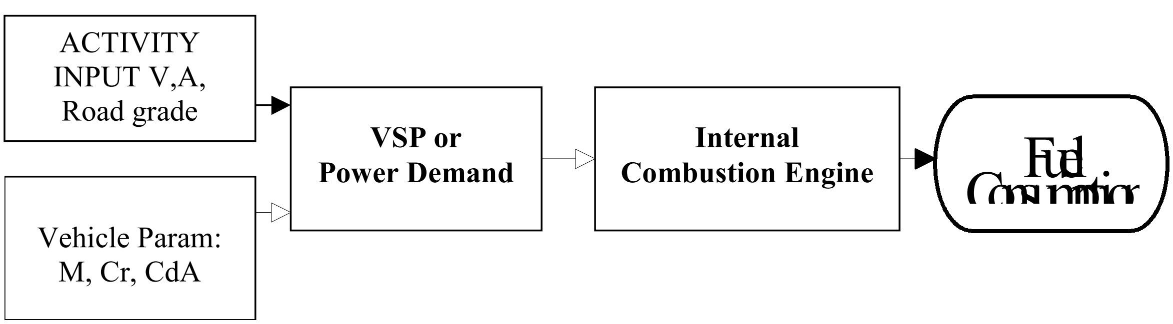


Figure 1. Conventional internal combustion engine vehicle model flow.

The power demand (in Watts) is the brake (or tractive) power or, VSP*m:

Equation 1 $$P_b = VSP*m = mv[a(1+\varepsilon) + g*grade + gC_R] + 0.5\rho C_D A_F v^3$$

where:

v: is vehicle speed (assuming no headwind) in m/s (or mps)
a: is vehicle acceleration in m/s^2
ε: is mass factor accounting for the rotational masses (~0.1)
g: is acceleration due to gravity (9.8 m/s^2)
grade: is road grade
C_R: is rolling resistance (~0.009)
ρ: is air density (~1.2 kg/m^3)
C_D: is aerodynamic drag coefficient (~0.3)
A_F: is the frontal area in $meters^2$ (~2 m^2)
m: is vehicle mass in metric tonnes.

The rotating mass term, ε, is assumed to be 0.1 [Jimenez, 1999]. However it increases at lower gears [Gillespie, 1992]. Other publications have a smaller effect [Thomas and Ross, 1997], however the results are not very sensitive to this term. The rolling resistance for radial tires, C_R, can range from 0.008 – 0.013 for a majority of the on- road passenger car tires. This value can be larger depending on state of inflation, temperature, ground surface type, and speed (though this effect is minor) [Bosch, 2000; Gillespie, 1992]. Heavy-duty trucks tend to have lower resistance. The aerodynamic drag, C_D, varies according to the body shape, and is sometimes provided by the manufacturers. The frontal area, A_F, of the vehicle is also sometimes supplied. Where it is unknown, PERE uses 93% of the box frontal area based on a number of vehicles examined by the author. This is calculated using the dimensions available for the vehicles:

Equation 2 $$A_F = (H-GC)*W*0.93$$

where: H is the vehicle height
GC is the ground clearance
W is the width.

Alternatively to **Equation 1**, is the dynamometer load equation:

Equation 3 $$P_b = Av + Bv^2 + Cv^3 + mv[a + g*grade]$$

where the coastdown coefficients: A, B, C, are rolling, rotating and aerodynamic resistive coefficients, respectively. These are determined from track coast-down tests and are available from the EPA certification database for certified light duty vehicles. The “A” coefficient roughly corresponds to the tire rolling resistance terms in **Equation 1**. “B” tends to be small, and describes higher order rolling resistance factors in addition to mechanical rotating friction losses. The “C” term represents the air drag coefficient terms. These two equations are not necessarily identical. When available, the coastdown (track) coefficients should be used in Equation 3 since this more accurately represents the test conditions. This will be discussed in greater detail in the sensitivity section, where PERE is run using both equations for comparison.

Older vehicles (pre-2000) only have a single Tractive Road Load HorsePower (TRLHP) rating provided by manufacturers [Sierra, 2000]. For the purposes of Inspection and Maintenance (IM) testing, A, B, and C coefficients have been estimated from TRLHP using the following equations:

Equation 4

$$\begin{aligned} A &= (0.35/50)*TRLHP(hp/mph) \\ B &= (0.10/50)*TRLHP(hp/mph^2) \\ C &= (0.55/50)*TRLHP(hp/mph^3) \end{aligned}$$

The power is distributed 35%, 10% and 55% into each of the 3 load categories [USEPA, 1994].

The power (or energy) determined from the road load can then be distributed, on a second-by-second basis, throughout the powertrain. After engine/transmission losses are included, this energy consumption forms the basis for PERE. For conventional vehicles, the primary energy conversion device is the internal combustion engine, which is the subject of the next section.

Section II: Conventional Gasoline Internal Combustion Engines

Gasoline Spark Ignited Internal Combustion Engine

The internal combustion engines (ICE) converts chemical into mechanical energy. This combustion process, as well as the losses inherent to it, is a critical element to the modeling of vehicles. Modern vehicles powered with gasoline have been the subject of several models in the literature. This paper remains consistent with the approach developed by Ross and An [1993]; Barth et al. [1999]; and Weiss et al. [2000] as well as other authors. The formalism is described well in Ross [1997], which will be briefly reviewed here.

The basis for the engine model lies in the linear relationship between brake power and fuel consumption, both represented in mean effective pressure (mep in bar) units. The power equivalent of fuel is:

Equation 5

$$P_f = FR*LHV$$

where FR is the fuel consumption rate in grams per second
LHV is the lower heating value of the fuel (approximately 43.7 kJ/g for gasoline)

In mep [kPa], this becomes:

Equation 6

$$\text{fuel mep} = 2000*P_f/(VN)$$

where V is the engine displacement in liters
N is the engine speed (rps)

One can model it as follows:

Equation 7 **Fuel mep = k + bmep/η**

or

Equation 8 **fuel mep = (fmep + bmep)/η**

where k is the engine friction related term (in friction mep terms, k = fmep/η) and η is a measure of the indicated or thermal efficiency of the engine, i.e., it is the fraction of the energy in the fuel which is converted to useful work. bmep is the brake mean effective pressure. This efficiency describes engine properties such as compression ratio, fuel mixing, fuel type, cylinder temperature, valve timing, spark timing, combustion chamber geometry, etc. (though some of these affect friction as well) [Muranaka, 1987; Ross, 1997]. Ideally, this is limited by engine heat cycle and thermodynamic 2nd law losses at the upper limit.

The indicated thermal efficiency is not to be confused with "mechanical" efficiency, or even with "overall" engine efficiency (discussed below). k and fmep reflect the mechanical losses in the engine mainly from rubbing, pumping, and auxiliary load losses. Specifically, the rubbing losses can originate from valves, gears, piston rings, lubricant, piston, crankshaft, etc [Milington & Hartles, 1968]. The pumping losses are derived from the difference between the intake cylinder and ambient pressures. The throttle plate, and intake, exhaust and valve geometry on gasoline engines cause this loss in "volumetric efficiency" [Patton, et al., 1989]. As an engine "breathes" better, the pumping losses decrease and its volumetric efficiency increases. Thus an engine with 4 valves per cylinder is typically more fuel efficient than one with 2 valves. Also diesel engines lack throttle plates, so they have improved volumetric efficiency over gasoline engines. Throttling results in about 25% of the friction losses [Ross, 1997]. The product of mechanical efficiency (which we have not quantified yet) and thermal efficiency is the overall engine efficiency, which will be discussed in more detail below.

Given the fuel mep, to obtain a mass fuel rate, one only needs engine displacement and speed. The bmep, is equivalent to brake power and can be determined from road load using the ubiquitous road load Equation 3. Equation 8 is equivalent to the one used by Weiss , et al. [2000]:

Equation 9 **imep = fmep + bmep**

where imep is the indicated mep equivalent to fuel mep*η.

This relationship has been plotted on Figure 2 below for a series of 10 different engines at many different operating conditions, where the wide-open throttle points have been omitted [Nam & Sorab, 2004]. According to the figure the indicated or thermal efficiency (inverse slope) is $\eta \sim$ 0.40, while k ~ 4.24 bar and fmep ~ 1.72 bar for this diverse family of engines. This procedure for estimating the engine friction analytically is referred to as the "Willans Line" method [Millington & Hartles, 1968]. It is similar to experimentally motoring the engine down. The largest uncertainty associated with this methodology lies in the slight curvature of the lines especially at high load. This is why the high load points were omitted, though there is still some curvature [Pachernegg, 1969]. Despite the limitations, the method is quite robust as the lack of scatter in Figure 2 demonstrates. For the scale of modeling in this report, this estimate is quite sufficient.

It has been found that the indicated efficiency has not changed significantly over the preceding 3 decades [Nam & Sorab, 2004; Weiss, et al. 2000]. It should be noted that this holds for engines operating at stoichiometry, where there is exactly the right quantity of air to combust the fuel completely. Older vehicles can run rich under various high power driving conditions, and those need to be modeled separately [Goodwin, 1996]. It is assumed that present and future vehicles (which have to meet SFTP standards) will not undergo significant periods of enrichment. However future vehicles may have improved efficiency if lean burn engines (or other advanced technologies) become more prevalent.

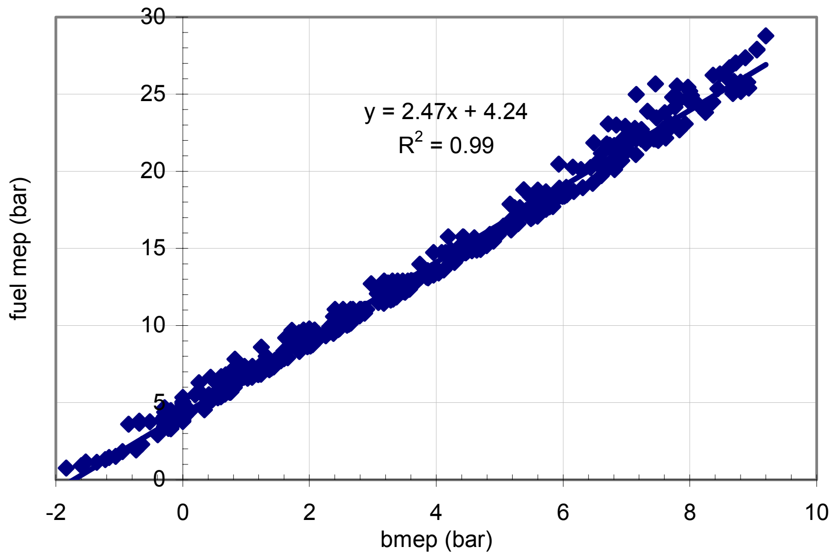


Figure 2. Fuel mean effective pressure as a function of brake mep for 10 engines from 4 different manufacturers, omitting wide-open throttle points [Nam & Sorab 2004].

It was mentioned earlier that “indicated thermal” efficiency is not the same as “overall” engine efficiency. The latter includes the effect of mechanical or frictional losses. This can be demonstrated most dramatically in Figure 2. The overall (or “brake”) efficiency of the engine at a given operating (load) point, is the x-axis value (output energy) divided by the y-axis value (input energy). For example, at bmep = 2 bar, the fuel mep = 10 bar, meaning the overall engine efficiency is 20%. Also at bmep = 8 bar, fuel mep = 24 bar, meaning the engine efficiency is 33%. It is clear that the overall efficiency increases with the load (omitting wide-open throttle and fuel enrichment) while the indicated thermal efficiency (inverse slope) remains constant at roughly 40%. Thus the indicated efficiency is the thermodynamic efficiency limit of the engine – here the peak total efficiency is 36%. The product of the indicated and mechanical efficiency is the overall engine efficiency. Thus the mechanical efficiency is dependent on load and engine speed, and the most efficient operating points are those with low engine speed and high load. This highlights the advantage that hybrids have over conventional vehicles: the engine operates more at higher loads and thus higher efficiencies, while the battery (or other energy storage medium) operates during low engine efficiency modes (allowing it to shut down). To calculate a final overall vehicle (or "pump-to-wheel" efficiency, it is also necessary to factor in transmissions and final drive losses. Accessories can also play a minor role. In general the overall efficiency of the vehicle is defined as the amount of fuel energy converted to useful work:

Equation 10

$$Efficiency = \frac{EnergyOut}{EnergyIn} = \frac{Pb(tractivework)}{fuelconsumed * LHV}$$

Though the scatter in Figure 2 is small, there is some systematic relationship, which can explain some of the variation. It is well known that the friction (or offset) is dependent on the engine speed. This is demonstrated in Figure 3. The base friction term can thus be modeled

Equation 11

$$\mathbf{k = k_0 + k_1 N}$$

where: $k_0 = 3.283$ bar and $k_1 = 0.000515$ bar/rpm for the present gasoline engines (multiply by 100 to get kPa, then divide by 2000 to get the proper units for the fuel rate equation). Trends for other years are reported in a later section. The friction at idle is significantly higher (about 50%). Alternatively, one can interpret the efficiency being lower at idle - the end result for fuel consumption in the model is identical, though this may not be the case if studying and isolating the effect of new engine technologies at low engine speeds. There is a slight curvature evident in the figure and some references include higher order terms, but these are probably more necessary for higher revving engines [Yagi, et al., 1990].

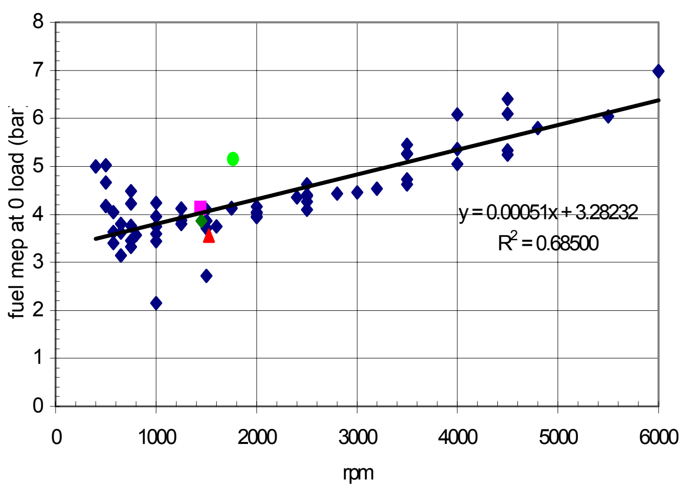


Figure 3. Engine friction as a function of engine speed for 10 engines. [Nam & Sorab 2004].

Though the indicated efficiency has changed little over the years, the friction has decreased over the past 30 years at a rate of roughly 10% per decade [Nam & Sorab, 2004; Sandoval & Heywood, 2003]. Various technologies have helped accomplish this: overhead cam, multiple valves per cylinder, improved lubricants, variable valve timing, etc. Additionally, engine efficiency can be improved with "advanced" technologies such as lean burn gasoline (e.g., Honda HX), Atkinson cycle (as in Toyota Prius), Direct Injection Spark Ignition (DISI – no current model in US), Homogeneous Charge Compression Ignition (HCCI), or with variable compression ratio. Fuel economy can also be improved with variable displacement, cylinder deactivation, or adding a turbocharger with engine downsizing in the future. There may come a day when engines will be "variable everything".

Despite the many different models and manufacturers of gasoline engines, there is remarkable homogeneity in their performance characteristics (neglecting advanced engines). As a result, it is possible to develop "rules of thumb", which give rough estimates, of engine efficiency, friction, peak torque and peak power, given only its displacement (and model year). Figure 4 shows generic peak torque and power curves for a 2.0 liter gasoline engine based on the bmep curves (**Error! Reference source not found.**) in Weiss et al. [2000]. The curves can be scaled to different engine sizes using standard torque/bmep relationships in Sandoval and Heywood [2000]. The equation for four stroke engines is:

Equation 12. $$\mathbf{mep(kPa) = 4\pi\tau/V_d}$$

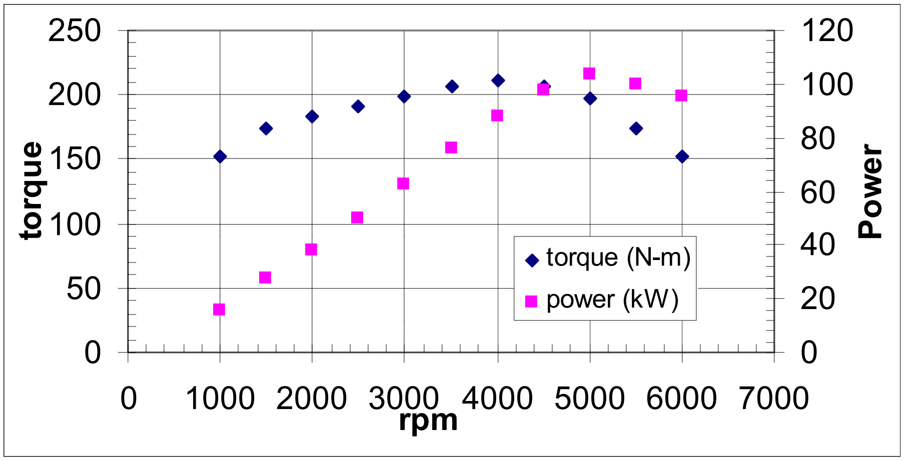
Where: τ is torque in Newton meters
V_d is engine displacement in Liters

The relation may also shift over time (described in the section: Advanced Engines and Hybrid Vehicles). While not all 2.0 liter engines will have the same shape and peak values, the general trends appear in most engines and this relationship is sufficient for our modeling needs. These peak curves are mainly used to determine transmission downshift points and for cut-point determination for hybrids (where engines may be significantly downsized), therefore the PERE output is typically not highly sensitive to the shape of the curves. In a future version of PERE, these peak torque curves may also define enrichment strategy, which is modeled in a different fashion for older vehicles [Nam, 2003].

In PERE, the bmep curve is modeled with a simple 7th order polynomial:

Equation 13. $$\mathbf{bmep = a_0 + a_1N + a_2N^2 + a_3N^3 + a_4N^4 + a_5N^5 + a_6N^6 + a_7N^7}$$

Where
N is engine speed in rps
a_0 = -1200.51
a_1 = 298.934
a_2 = -17.5860
a_3 = 0.563420
a_4 = -.0104629
a_5 = 0.000113228
a_6 = -6.64513e-7
a_7 = 1.63097e-9



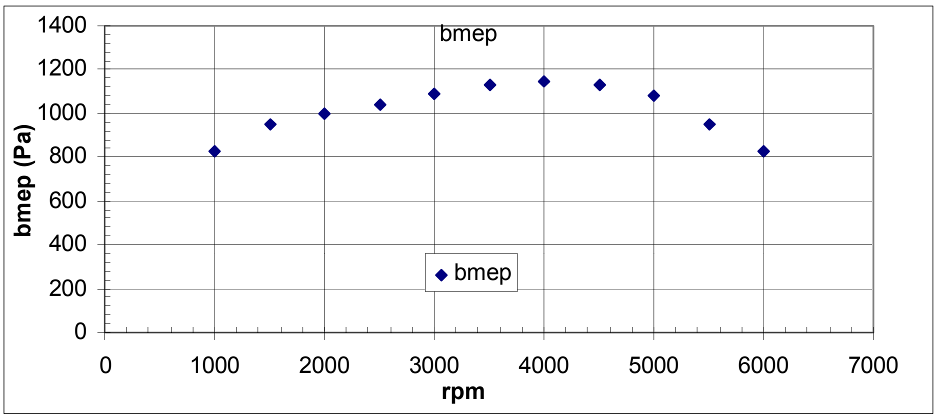


Figure 4. Engine peak torque and power for a "generic" 2.0L gasoline engine (top) based on bmep curve (bottom).

Section III – Transmissions and Fuel Rate

Transmissions

Another element required to calculate total vehicle efficiency is transmission and final drive efficiency. All vehicle powerplants must connect to the wheels via a transmission. Sometimes this is as simple as a single gear reduction (as in some fuel cell and electric vehicles). More often though, there are multiple gears to take advantage of the engine's limited operating range. The multi-gear transmission model used for PERE is based on Thomas and Ross [1997]. While transmissions and their shift strategies vary tremendously between vehicles, the overall fuel consumption rate is not highly sensitive to this sub-model as long as 'sensible' parameters are employed.

The engine speed is:

Equation 14 $$N = (N/v)_{top} * (60rps/rpm) * (g/g_{top}) * v$$

where N is engine speed in rpm
$(N/v)_{top}$ is the rpm to speed ratio in top gear
v is vehicle speed in mph
(g/g_{top}) is the gear ratio at the various gears

$(N/v)_{top}$ is assumed to be a constant for light duty vehicles, but a future version of PERE may make it dependent on engine size. Smaller engines tend to rev higher than larger engines. One could also calculate engine speed from final drive ratio and tire radius.

In order to meet the power demand of some driving cycles, it is necessary to downshift. The algorithm for downshifting is as follows:

If TorqueDemand > MaxTorque
Then Downshift

The algorithm is repeated until the Torque demand is met, or gear 1, whichever comes first. There is no consideration for shift or ride quality in this simplified model.

Efficiency

Transmission efficiency directly affects fuel consumption. In PERE, the efficiency simply depends on what gear the vehicle is in. The efficiencies used in MOVES are shown on Table 1 based on the published work on Bishop and Kluger [1996]. These values were based on an average over the combined EPA city/highway driving cycles (excluding idle). The efficiencies are lower in 1st gear due to “slippage”, and better in the higher gears due to the gears “locking-up” in the torque converter.

Table 1. Transmission efficiencies of a composite automatic transmission (based on Bishop & Kluger, 1996)

Gear	Efficiency
1	72.2
2	80.9
3-5	87

More realistically, the efficiencies also vary with engine speed as well as torque. Due to the current spreadsheet nature of PERE (where it is difficult to do loops), these models were deemed impractical. If PERE is coded in the future, they could be easily applied. Efficiency can be a modeled as a linear function of engine speed [Park et al, 1996], or as a linear function of both speed and torque [Greenbaum et al., 1994].

Manual transmissions range in efficiency from 87-99%, and fixed value of 95% has been assumed in this report (independent of gear). An overall 1.5% improvement in transmission efficiency could correspond to a 0.1 km/L increase in fuel economy [Greenbaum, et al., 1994; Kluger, et al., 1995; Bishop, et al., 1996]. Continuously variable transmissions (CVT), have been estimated to have cycle efficiencies of approximately 88% [Weiss et al, 2000].

Vehicles whose drivetrains run off of motors do not necessarily require complex transmissions employing gear shifts. These motor driven vehicles only require a single gear due to the large operating range of motors. Single gear transmissions naturally tend to be very efficient and are assumed to be 95% efficient in this report. Though multi-gear transmissions for motor-driven vehicles may be advantageous for some applications, the hybrids and fuel cell vehicles in this version of PERE use only single gear transmissions.

Light Duty Transmissions

It has already been shown that due to the nature of the engine model, PERE (for light duty vehicles) is less sensitive to the specifics of a transmission model, than other models can be. Many other engine models are based on maps of torque, rpm and efficiency (or brake specific fuel consumption). The fuel consumption points are selected off of the map by the transmission shift strategy, which can make those models much more sensitive to transmission parameters. In PERE, the fuel consumption is defined by road load power and engine friction (which are more or less independent). Therefore the losses due to the transmission are primarily captured with the transmission efficiencies. However, an estimate of the engine speed is still required for the friction calculation.

While PERE users can adjust powertrain parameters as they see fit, when PERE is used to calculate fuel consumption rates for the fleet, it is necessary to make some generalizations. For MOVES, PERE assumes all light duty vehicles to have 4 speed automatic transmission with overdrive (or a 5 speed automatic transmission). The vast majority of light duty vehicles on the road (in the United States) have automatic transmissions, and many modern vehicles are equipped with an overdrive for higher speed driving. The shift points and gear ratios are shown in Table 2. Older vehicles are assumed to have 4 speed automatic transmissions.

Table 2. Shift point and gear ratios for light duty vehicles (5 speed) [Thomas & Ross, 1997**].**

Speed (mph)	Gear	g/gtop
0-18	1	4.04
18-25	2	2.22
25-40	3	1.44
40-50	4	1.0
50+	5	0.9

Due to the downshifting model (in the previous section), vehicle speed is not the only variable defining gear.

For gasoline vehicles in the fleet some other generalizations were made:
- the engine idle speed is fixed at 700 rpm
- the (N/v)top (engine speed to vehicle speed ratio at gear ratio 1) is fixed at 35.6
- the cruising top gear ratio is usually 1 by our definition of N/v

Fuel Rate

If transmission speed and efficiency is added to Equation 5 through Equation 8, we would obtain the fuel rate equation used in PERE [Nam, 2003]:

Equation 15 $$\mathbf{FR = \varphi\ [kNV + (P_b/\eta_t + P_{acc})/\eta]\ /\ LHV}$$

or

Equation 16 $$\mathbf{FR = \varphi\ [fmepNV + P_b/\eta_t + P_{acc}]\ /\ \eta LHV}$$

where

- φ: is the fuel air equivalence ratio (mostly =1 for gasoline)
- k: is the engine friction (can depend on engine speed).
- fmep: is the friction mean effective pressure.
- N: is the engine speed, depending mostly on vehicle speed
- V: is the engine displacement volume
- η_t: is a transmission and final drive efficiency
- η: is a measure of the engine indicated efficiency
- P_{acc}: is the power draw of accessories such as air conditioning, pumps electrical loads etc. Without AC, it is some nominal value ~ 0.75kW.
- LHV: is the factor lower heating value of the fuel (~43.7 kJ/g for gasoline, 41.7 for diesel).

The only times (during hot running operation) when the fuel air equivalence ratio does not equal 1, is when the engine is in command enrichment (for more power and for catalyst cooling). This modal regime is modeled separately for older vehicles, which tend to go more into enrichment than newer ones [Nam, 2003]. Lean burn is modeled separately (as described below).

Section IV – Motorcycles

In principle, Motorcycles are modeled the same as passenger cars in PERE. However different parameters are required for the road-load, engine, and transmission.

Motorcycle Road Load

Motorcycle road load forces are typically parameterized [USEPA, 2000 and UN, 2003] with mass dependent A and C terms which take into account rolling resistance and aerodynamic drag,

Equation 17. $$F = A + Cv^2$$

where F is the total road load force in Newtons, A is the rolling resistance in Newtons and C is related to the aerodynamic drag, C_dA_F, and has units of Newtons/(meters/second)^2. A and C are further parameterized in terms of the inertial mass, M, of the vehicle and driver, i.e.,

$$A = a_0 + a_1 M$$

Equation 18.

$$C\left(= \frac{1}{2}\rho_{air} C_d A_F \right) = c_0 + c_1 M$$

where a_0, a_1, c_0, and c_1 are constants and ρ_{air} is the density of air (~1.2 kilograms/meters^3), C_d is the aerodynamic drag coefficient, and A_F is the frontal area of the motorcycle in meters^2. The EPA uses a driver mass of 80kg, whereas the UN uses a driver mass of 75kg.

Both the EPA and the UN have similar parameterization of the mass dependence of motorcycle tire rolling resistance, but differing parameters. The UN's parameterization is

Equation 19. $\mathbf{A_{UN} = (0.088 m/s^2) M}$

which corresponds to a rolling resistance coefficient similar to that of passenger cars:

Equation 20. $\mathbf{C_r\ (=A/Mg) = 0.009.}$

The mass, M, corresponds to the curb weight of the vehicle plus a 75 kg driver. The EPA's parameterization uses

Equation 21. $\mathbf{A_{EPA} = (0.0874 m/s^2) M - (8.79 Newtons)}$

which yields a static rolling resistance coefficient, C_r, of

Equation 22. $\mathbf{C_r\ (= A_{EPA}/Mg) = (0.0874 m/s^2)/(9.8 m/s^2) - (8.79 Newtons)/(M9.8 m/s^2)}$

$$\cong 0.009 - (0.9kg/M)$$

The mass, M, in the USEPA's parameterization assumes the vehicle mass plus an 80kg driver. Table 3and Figure 5 summarizes the parameterized approaches as a function of mass.

Table 3. USEPA, UN, and Gillespie's passenger car mass parameterization of the rolling loss road load coefficient, A(=MgC$_r$).

Rolling loss force parameterizations, A (= MgC$_r$) = a$_0$ + a$_1$M			
Organization / Emissions Certification Procedure	a_0 Newtons	a_1 meters/second^2	C_r (or A/Mg) -
EPA Certification Procedure, CFR 40 part 86.529	-8.79	0.0874	0.0089– (0.90kg/M)
UN Report, "Worldwide Harmonised Motorcycle Emissions Certification Procedure"	0	0.088	0.0090
T.D.Gillespie, 1992	-	-	0.01(1 + v/100mph)

Motorcycle aerodynamic drag is governed by both its relatively small frontal area and its relatively large aerodynamic drag coefficient, which is due to its non-continuous shape / non-aerodynamic features of the motorcycle and the driver. Drag coefficients, C_d, for motorcycles can vary between 0.4 to 0.6. Frontal areas, A_F, of the motorcycle and driver range between 0.4 to 1 m^2 [Arakai et al., 2001]. Hence, the majority of motorcycles have aerodynamic loss coefficients, C_dA_F, which lie between about 0.2 m^2 and 0.6 m^2.

Both the EPA and the UN have established aerodynamic losses for different motorcycles based on the mass of the vehicle and the driver. The UN parameterization is

Equation 23. $$C_{UN} = 0.26\ N/(m/s)^2 + 1.94 \times 10^{-4}\ M\ [N/kg(m/s)^2]$$

which corresponds to an aerodynamic drag coefficient of $C_dA_F \cong 0.50m^2$ when M=227kg(≅500lbs). The EPA's parameterization is similar except it has two mass ranges, i.e.,

Equation 24. $$C_{EPA} = 0.254\,N/(m/s)^2 + 0.35 \times 10^{-4} \times M\,[N/kg \cdot (m/s)^2]\ \ for\ M \le 450kg$$
$$and$$
$$C_{EPA} = 0.376\,N/(m/s)^2 + 0.804 \times 10^{-4} \times M\,[N/kg \cdot (m/s)^2]\ \ for\ M > 450kg$$

In Figure 5, the graphs of the aerodynamic loss coefficient as a function of mass for both the EPA and UN parameters are displayed.

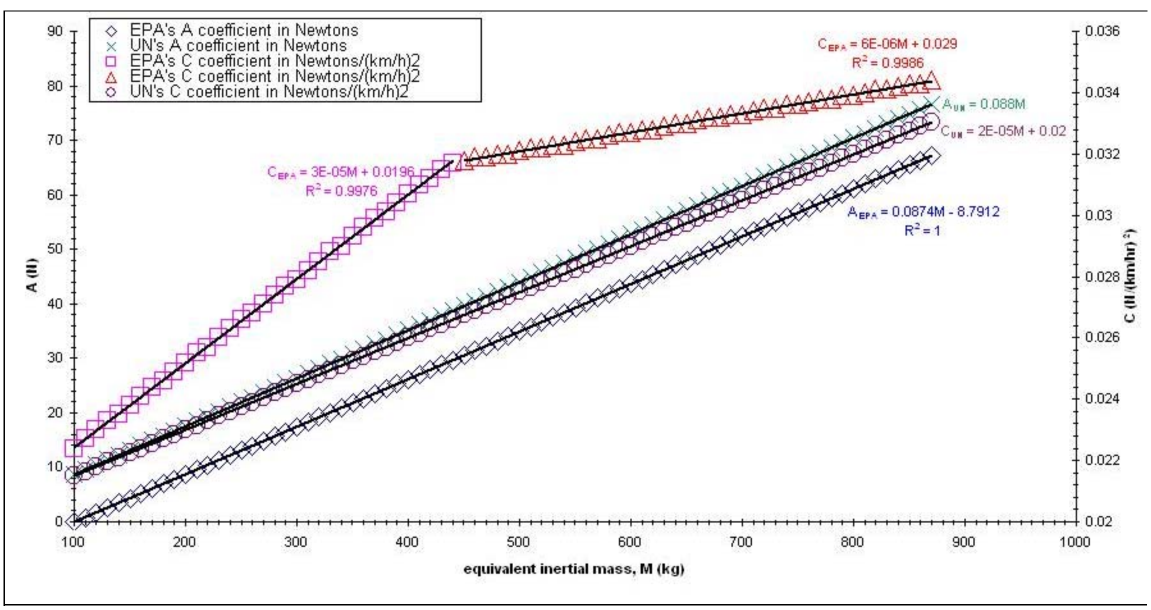


Figure 5. USEPA and UN A and C road load coefficients as a function of the mass of the motorcycle and driver.

Figure 6 shows the total road load force as a function of speed comparing the UN and EPA methodologies. The vehicle plus driver mass for these plots is 700lbs or about 318kg. The largest differences (absolute differences greater than 10%) between the two road load parameterizations occurs at speeds below 30km/hr (18 mph) and above 60km/hr (36mph). At about 38km/hr (24mph) the UN road load force becomes less than the EPA road load values. Below speeds of about 38km/hr the UN road load force is greater than the EPA values.

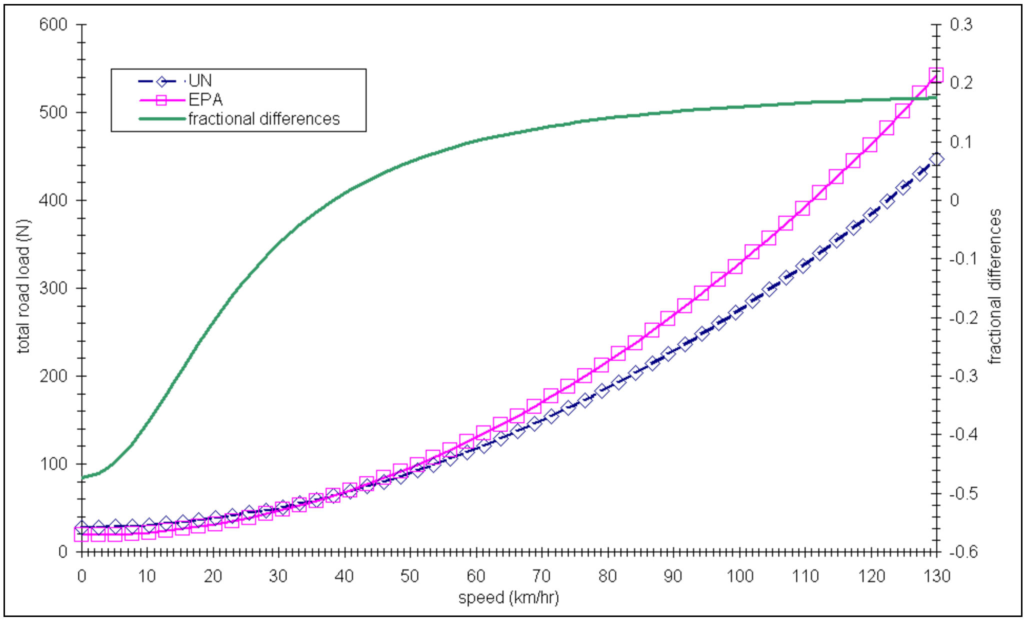


Figure 6. USEPA and UN total road load force as a function of speed along with the fractional differences between the two road load forces at each value of motorcycle speed.

In PERE (and MOVES), the UN methodology for motorcycle road load is employed. However, the US standard driver weight of 80kg is assumed.

Motorcycle Engines

Current literature states that the typical (4-stroke) motorcycle engine thermal efficiency ranges between 32% to 40% [Yamamoto, 1999]. PERE will use the same efficiency that has been used for other SI gasoline engines, i.e., 40%.

Motorcycle engine friction for engine sizes ranging between 0.125 to 1.0 L has been studied extensively over the past 15 years [Yagi, et al., 1990; Yagi, et al., 1991; Tsuchida & Tsuzuku, 1991, 1995, 1999; Boretti, 1996]. In these studies, motorcycle engine friction has typically been parameterized as a second order function of engine speed dependent on the engine bore, stroke, and mean equivalent crank shaft diameter. Explicitly, the functional form in terms of the frictional mean effective pressure or fmep, is

Equation 25.

$$fmep = \frac{(S \cdot D)^{1/2}}{B} \cdot \left(C_0^{'} + C_2^{'} \cdot N^2 \right)$$

where S is the stroke, B is the bore and D is the mean equivalent crank shaft diameter (average diameter of the crankshaft journals and crankshaft pins), C_0' is a constant proportional to the lubricating oil viscosity, and C_2' is a constant due to engine pumping losses.

Equation 26

$$C_0 = \frac{(S \cdot D)^{1/2}}{B} \cdot C_0' \quad and \quad C_2 = \frac{(S \cdot D)^{1/2}}{B} \cdot C_2'$$

From the literature cited above average coefficients, for over 20 motorcycle engines are listed in Table 4. Also listed are the standard deviations representing vehicle-to-vehicle variability (not measurement errors). Figure 7 illustrates the motorcycle engine friction mep dependence on engine speed compared to light duty spark ignition engines determined earlier (which do not usually operate at such high speeds).

Table 4. Parameters found for motorcycle engine friction from the literature. The engines ranged in size from 0.125 liter to 1 liter.

C_0 (MPa)	C_0 standard deviation (MPa / rpm^2)	C_2 (MPa/rpm^2)	C_2 standard deviation (MPa / rpm^2)
0.152	0.031	1.52E-09	6.23E-10

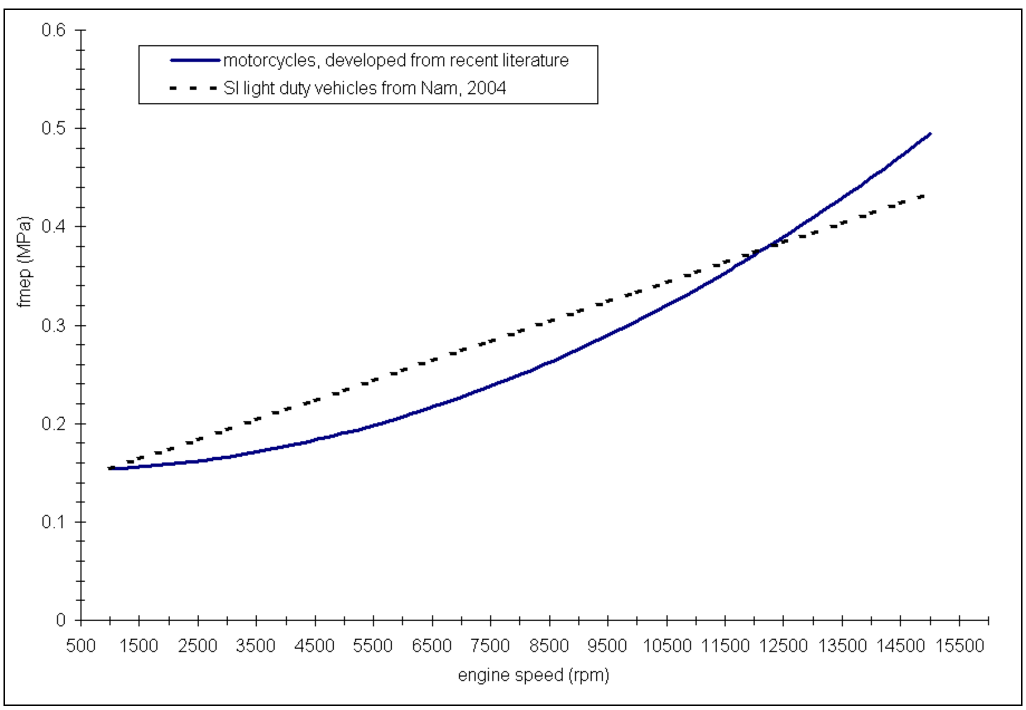


Figure 7. Comparison of engine speed dependence of motorcycle and recent light duty passenger vehicle engine friction.

Finally, the accessory loading for motorcycles in PERE was dropped down to 0.25 kW (compared to 0.75 kW for passenger cars).

Motorcycle Transmissions
Due to the high RPM range of motorcycle engines, the transmission model is somewhat different than that of light duty manual transmissions. Transmission modeling parameters for motorcycles were derived from a number of sources. The N/v is determined from the UN Document: "Worldwide Harmonised Motorcycle Emissions Certification Procedure," [2003] (Figure 2).

The N/v at top gear is 115 rpm/mph. The 6 gear ratios are 3.1, 2.34, 1.75, 1.4, 1.17, and 1.0. In PERE, these are spaced out among 5 gears: 3.1, 2.3, 1.7, 1.4, and 1.0. N/v is normalized to be 1 in top gear, so the actual gear ratios may differ on vehicles.

The shift point is based on engine speed, rather than vehicle speed. This is because the shifting occurred at excessively high RPM when fixed speeds were used. The shift point in PERE is set at ~5,000 rpm, though it is surely lower for the heavier bikes. The transmission efficiency was assumed to be the same as light duty manual transmissions (~0.95).

Motorcycle Validation
The validation data is the EPA certification database. The EPA maintains a database (The Certification and Fuel Economy Information System or CFEIS, e.g., http://www.epa.gov/otaq/crttst.htm) which contains (among other vehicle types) motorcycle emissions certification information. It covers, as of this writing, motorcycles with model years ranging from 1995 to 2004. Included in the database are the vehicle model name, model year, weight, engine displacement, maximum torque and horsepower, the ratio of engine speed, N, to vehicle speed, v, at the highest gear (N/v), and emission certification levels for HC and CO. This includes 55 motorcycles of model year 2004-2005, ranging in weight from 240-580kg and engine displacement from 0.5–2.3L. The fuel economy is calculated from the CO_2 and emissions reported by the manufacturers, however these figures were not confirmed by the EPA.

Figure 8 shows the scatter plot of the fuel economy comparison, with the 1:1 line included. On average, the lighter motorcycles seem to be captured well. However, it is evident from the plot, that there is a systematic underprediction of fuel economy for the heavier (lower fuel economy) motorcycles, i.e. PERE seems to be less efficient than expected. This is due to the fact that a limited transmission model was used. Since motorcycles have significantly lower road loads than passenger cars, their fuel consumption is more sensitive to transmission (and engine friction losses). For example, the Harley Davidson motorcycles tend to have lower N/v values, lower peak RPM, and lower shift points, thus changing their fuel consumption significantly. Using more appropriate transmission values, PERE captured the 39 mpg rating of a heavier Harley Davidson (2004 FLHR Road King, 723lb, 1.44L, port fuel injected, N/v = 29, top gear ratio = 3.76, assuming a shift point ~3500RPM) [Harley-Davidson, 2004]. The heavier bikes have transmission parameters that are more similar to passenger cars than the lighter motorcycles. Using more specific values for the transmissions of these motorcycles increases the likelihood that the results from PERE will be more accurate. This can be changed in a future version of the model.

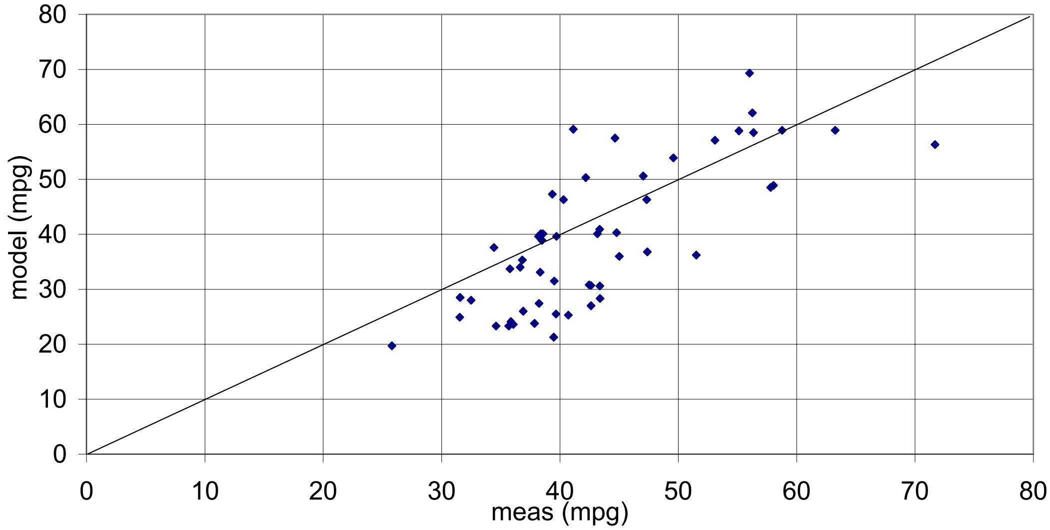


Figure 8. Fuel Economy model compared to measured for 55 motorcycles (2004-2005) in the EPA certification database. The 1:1 line is included.

Section V – Light Duty Diesel Vehicles

This section describes the model parameters for light-duty diesel vehicles. The model validation is lumped together with the advanced technology section.

Light Duty Diesel Vehicle Road Loads

Apart for the powertrain, light duty diesel vehicles are very similar to their gasoline counterparts in PERE. The main exception to this is in the vehicle weight. Diesel vehicles tend to be somewhat heavier, mainly due to their heavier engines.

It is useful to conduct a comparison of gasoline and diesel light duty vehicles currently sold. Comparing the current technologies can help us to extrapolate how they will compare in the future. Table 5 shows the power and weight of various vehicles in the American and European markets.

Based on the table, we note that diesel engines tend to be heavier than their gasoline counterparts (to accommodate higher pressures), thus they often need to be downsized in order to fit into the same frame. According to the ratios of these same vehicles, the turbocharged diesel engines should be approximately 5% smaller. Due to the many differences between gasoline and diesel engines, it is impossible to do a perfect apples-to-apples comparison. However, based on these vehicles, the equivalent vehicle weight of a diesel is assumed to be 4% higher than the gasoline vehicle when PERE fills rates for MOVES.

Table 5. Diesel vehicles used for comparison

model	max torque	max power	displacement (l)	curb weight (kg)
Ford Focus Diesel	200N-m@ 2000rpm	66kW@ 4000rpm	1.753	1258
Ford Focus Gasoline	160Nm@ 4400rpm	85kW@ 5500rpm	1.8	1200
Ford Mondeo Diesel	245N-m@ 1900rpm	85kW@ 4000rpm	1.998	1498
Ford Mondeo Gasoline	190Nm@ 4500rpm	107kW@ 6000rpm	1.999	1376
peugot 607 diesel	250N-m@ 1750rpm	79kW@ 4000rpm	1.997	1500
peugot 607 gasoline	217Nm@ 3900rpm	116kW@ 5650rpm	2.231	1455
VW Beetle Diesel	210N-m@ 1900rpm	66kW@ 3750rpm	1.896	1270
VW Beetle Gasoline	172Nm@ 3200rpm	85kW@ 5400rpm	1.984	1222
VW Jetta Diesel	240Nm@ 1800rpm	75kW@ 4000rpm	1.9	1347
VW Jetta Gasoline	165Nm@ 2600rpm	86kW@ 5200rpm	2.0	1331

* All sources are either from manufacturer website, Car&Driver, or Road&Track

Light Duty Diesel Engines

Diesel engines are different from gasoline engines in several important ways. They run on diesel fuel, which has higher density, energy density, and carbon content. The fuel is self-ignited by compression rather than with a spark (hence it is also known as compression ignition engine). The increase in compression ratio required for compression ignition significantly improves the thermal efficiency compared to a gasoline spark ignition engine. The engines can also combust fuel in an environment that is lean of stoichiometry. A stoichiometric fuel air mixture has just enough air to completely combust (or oxidize) all of the fuel, so a lean mixture has more air, and rich mixture has more fuel. Naturally, lean mixtures result in higher fuel efficiency. Modern diesel engines inject the fuel directly in the chamber with electronically controlled bursts, which increases the pumping efficiency (decreases friction) by doing away with the losses inherent to a throttle. Unfortunately, diesel engines tend to have higher NOx and particulate matter (PM) emissions. The excess NOx emissions is due to the lack of aftertreatment system, and the PM emissions is due to the combustion mechanism with fuel droplets. Advanced diesel engines complying with federal Tier 2 emission standards will require aftertreatment technologies, which may require the fuel combustion to run at stoichiometry on occasion so that NOx and PM can be stored, then treated. This will in turn, have a small negative effect on fuel economy.

In PERE, Diesel indicated efficiency is assumed to be 0.48. This is 20% more efficient than current gasoline engines [Wu & Ross 1997]. Weiss et al. [2000] employ an indicated efficiency of 0.51 for advanced diesel engines.

The friction in diesel engines is typically lower than their gasoline counterparts [Heywood, 1988, Weiss, et al., 2000, Wu&Ross 1997] – though this is mostly in the speed independent term (k_0 in Eq. 8). However, Ball et al. [1986] measured a diesel engine to have similar (total) friction compared to an equivalent gasoline engine at higher operating conditions. In a diesel engine, the pumping losses are significantly lower, but the piston/crank assembly as well as auxiliary load (mainly from high pressure fuel injector) losses are higher. This hints that the speed dependent term (k_1) should be higher for diesels, or that higher order terms in Equation 11 may be necessary. For the efficiency and friction parameters in PERE, we use the heavy-duty friction results presented in the next section. These results are consistent with Wu and Ross' average measurements from 4 light duty diesel engines [1997].

The peak torque and power curves are generically determined from analysis from engine map data from the EPA database. This is discussed further in the heavy-duty transmission section of this report. The results are roughly consistent with bmep curves from Weiss et al. [2000]. Figure 9 shows the relationship for bmep (from which torque and power can be calculated). The torque curves end up flatter than that of gasoline engines. This high low-end torque gives diesel engines the advantage in towing as well as decent 0-30 mph acceleration compared to their naturally aspirated gasoline equivalents (of same displacement). However the peak power tends to be lower, thus the overall 0-60 mph acceleration may be lower depending on the vehicle.

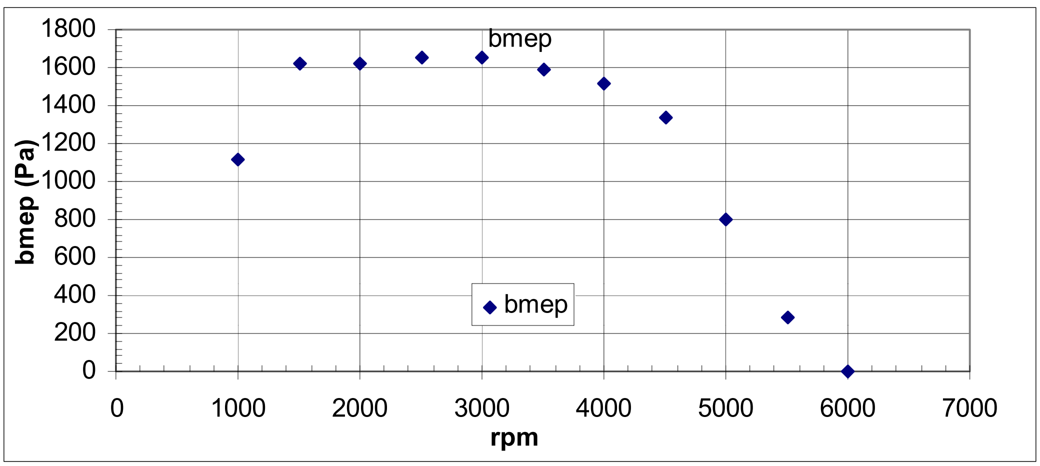


Figure 9. The peak bemp curve for a "generic" turbo-charged diesel engine.

In PERE, this generic bmep curve is modeled by a simple 7[th] order polynomial fit:

Equation 27. $$\mathbf{bmep = a_0 + a_1N + a_2N^2 + a_3N^3 + a_4N^4 + a_5N^5 + a_6N^6 + a_7N^7}$$

Where

N is engine speed in rps
$a_0 = -19950.8$
$a_1 = 3479.90$
$a_2 = -231.809$
$a_3 = 8.25775$
$a_4 = -0.169919$
$a_5 = 0.00202259$
$a_6 = -1.2921e-5$
$a_7 = 3.42208e-8$

Diesel engines are ideally suited for turbo charging largely due to their unthrottled operation. A turbo charger employs the exhaust heat energy to spin a turbine, which pumps more air charge into the engine cylinders. This efficient use of energy generates more power with a smaller engine. As a result virtually all light and heavy duty diesels employ turbo chargers. Unfortunately, the variety of turbocharger sizes makes generating peak torque and power curves, given only engine displacement, a difficult exercise. This would matter more if we were constructing the model using a performance basis (e.g. 0-60 mph time), rather than a fuel economy basis.

Diesel Light Duty Transmissions

The transmission model for light duty diesel vehicles is identical to that of gasoline, with the exception that diesel engines tend to rev lower. The $(N/v)_{top}$ is estimated to be 75% that of gasoline. This figure is obtained by comparing peak rpm power values from an assortment of vehicles, which have a diesel option (US and European). E.g. VW Jetta, VW Beetle, Ford Focus, Ford Mondeo, & Peugeot 607. See Table 5.

Section VI – Heavy Duty Diesel Vehicles

This section describes the method for determining a complete set of model parameters for heavy duty diesel vehicles, including in-use city buses, heavy duty diesel trucks, as well as some non-road diesel engines. The model validation is also presented.

Data

Emissions, global positioning (GPS), fuel, and engine control module data were taken from 15 in-use Ann Arbor, MI city transit buses [Ensfield, 2002] measured by SENSORS and EPA, as well as from 12 heavy duty diesel tractor-trailers measured on-road by the University of California Riverside (CE-CERT) [Barth et al., 2004]. The 17 non-road diesel engine data were taken from two EPA studies [Fritz and Starr, 1998; and Starr, 2003] contracted through Southwest Research Institute. The non-road diesel engines varied from a small 0.2 liter electric generator diesel engine to a 34.5 liter diesel locomotive engine. The data for these non-road engines were collected from engine dynamometers. Table 6 below summarizes the vehicle and engine specifics from which emissions were measured and parameters for road load and engine characteristics were determined. More details about the vehicles and engines can be found in the references listed in the text.

Table 6. Abridgement of the vehicle and engine specifications which were used in this study.

source	# of vehicles or engines	model year range	vehicle weight range	odometer range	rated torque@RPM	engine displacement	# of gears
-	-	-	kilogram	kilometer	Newton-meter@RPM	liters	-
CE-CERT (in-use/ on road)	12	1997 to 2001	26,535 to 28,145	12,875 to 83,864,682	1830@1200 to 2373@1200	10.8 to 14.6	9 to 13
EPA	15	1995 to	12,020	320,260 to	1207@1200	8.5	6

AATA bus (in-use/ on road)		1996		457,055			
EPA nonroad engines (dynamometer)	17	1988 to 1999	-	-	230@2200 to 3586@1400	0.2 to 34.5	-

Both the CE-CERT trucks and the Ann Arbor bus emissions testing measured emissions, vehicle speed, position, fuel use and engine parameters during on-road driving. The CE-CERT trucks were driven on approximately 20 highway and engineered trips including vehicle coast downs. This study used four of these trips, the UDDS, the Riverside to Palm Springs, the creep, and the ARB cruise trips. Examples of the coast down and Riverside, CA to Palm Springs, CA speed-time traces for one of the trucks are illustrated in Figure 10.

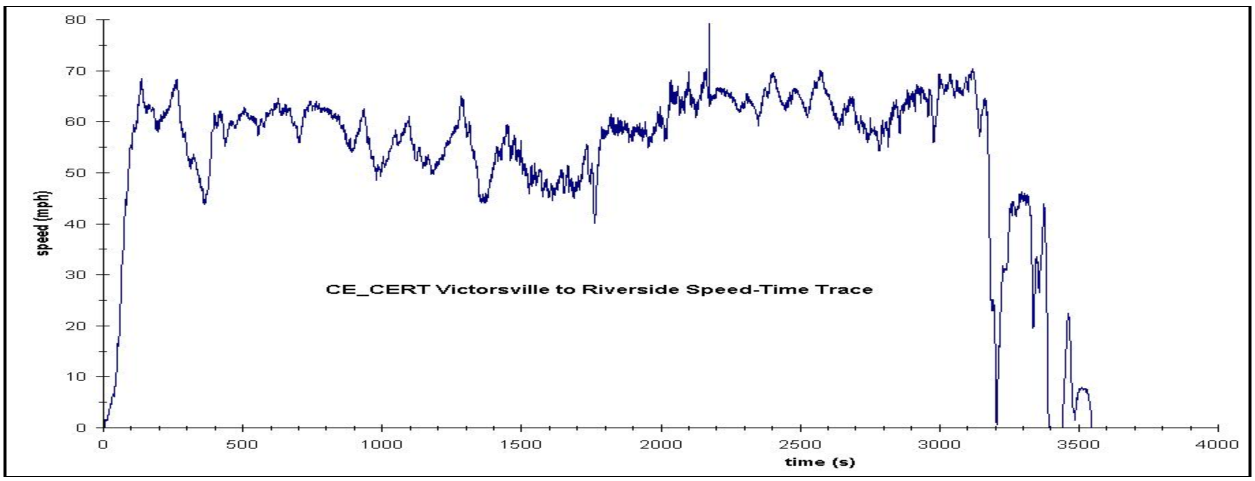


Figure 10. Speed-time trace of a complete CE-CERT heavy duty diesel tractor-trailer trip between Riverside, CA and Palm Springs, CA.

Measurements from the AATA buses were collected on bus routes used by the Ann Arbor Transit Authority. An example of a speed-time trace for one of the bus trips is shown in Figure 11.

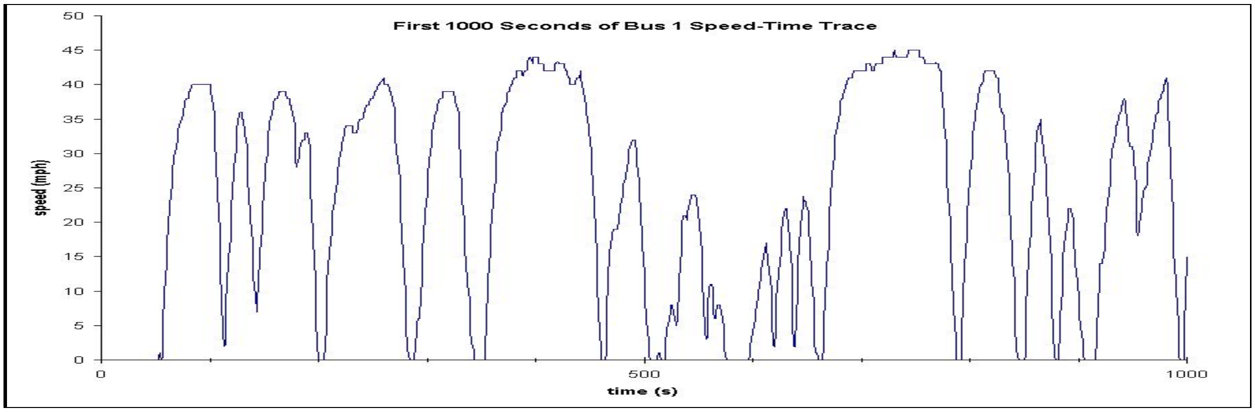


Figure 11. Speed time trace for the first 1000 seconds of AATA bus 1 in-use emissions testing. The trips in the on-board measurements were from a typical AATA bus route whereby the bus did not carry any passengers.

EPA non-road engines were tested on engine dynamometers. The engines were run at different loads and RPM points throughout the operating range of the test engine. Table 7 lists the test schedule used for the 2003 on-road engines. In particular, each engine was operated at 39 different RPM, engine load pairs, which are listed in the table as percent of the highest engine RPM and percent of maximum load. More details of these test cycles and the measurements taken can be found from the respective references.

Table 7. The non-road engine tests. The percentages are the percent of the highest rated engine speed and the maximum rated engine load. These tests included an additional operating mode at the engine's low idle speed, with no load applied, which is not listed in this table.

Percent of Highest Engine Speed	Percent of the Maximum Rated Engine Load								
	100%	85%	82%	75%	63%	50%	40%	25%	10%
100%	1	2		3		4		5	6
91%			37						
80%					38				
75%	7	8		9		10		11	12
63%							39		
60%	13	14		15		16		17	18
50%	19	20		21		22		23	24
25%	25	26		27		28		29	30
10%	31	32		33		34		35	36

Heavy Duty Vehicles Road Load

Tractive road-load power coefficients for heavy-duty vehicles include tire rolling and aerodynamic losses. Coefficients for both of these effects are developed from a relatively comprehensive study in terms of the range of vehicles that are considered [Petrushov, 1997]. In particular, the vehicles range from passenger cars to heavy-duty tractor-trailers pulling two cargo trailers. Although the vehicles in this study are all from the Euro-Russian fleet, comparisons of the parameters developed are mostly consistent with general studies done in the United States on vehicles and tires used in the U.S. fleet [e.g., Yong, 2001; Gillespie, 1992; USEPA, 2002; Andrei, 2001; McCallen, et al.,1999; USDOT, 1977; Younglove, 2003]. However, the aerodynamic loss coefficients (C_dA_F) are approximately 35% lower and will be updated in a future version of MOVES.

Dynamometer based emissions testing use empirical A, B, and C tractive road-load parameters which can be related to the rolling resistance and aerodynamic drag coefficients through the speed dependence of the road load equation. Because they are empirically determined they may include additional factors such as axle rotational inertia which can account for approximately 6% of an 18 wheel tractor-trailer's kinetic energy. (This can be approximated by a simple kinematic analysis of the rotational energy of the truck's wheels, tires, and axles). Also, additional aerodynamic factors from transverse, or cross, winds are not accounted for with the aerodynamic parameters presented here.

Aerodynamic drag coefficients from Petrushov are developed for heavy-duty delivery trucks and for buses. The truck aerodynamic drag coefficients are divided into three classes of delivery trucks. Single unit delivery trucks are divided into two classes according to length and the third group included all long-haul tractor-trailers. The two classes of single unit trucks have lengths between 5 to 6 meters or lengths between 7 to 8 meters. Buses vary in length between 9 to 12 meters. Table 8 below lists the values of these drag coefficients along with a standard deviation, which includes the inter-vehicle differences and the reported measurement errors. Also included in the table for comparison with the U.S. fleet, are values of aerodynamic drag coefficients from recent literature [USEPA, 2002; Andrei, 2001] and values determined in this work from the CE-CERT truck data. Figure 12 below illustrates the speed dependence of the aerodynamic drag coefficients listed in the table. These comparisons reveal that the tractor-trailer numbers are about 35% smaller than those typical values reported in the U.S. fleet (Table 8). The frontal areas of the tractor-trailers in the Petrushov work are on average about 9 m^2 whereas in the U.S. the frontal areas are 10 to 12m^2. Also, values of C_D from the U.S. fleet can vary between 0.5 and 0.8.

Table 8. Comparison of aerodynamic drag coefficients.

source	vehicle description	C_DA (m^2)	standard deviation (m^2)
CE-CERT from coast downs	tractor-trailer combo	7.2	4.5 (63%)
Petrushov, SAE 970408	tractor-trailer combo	4.8	1.6 (33%)
Paul Andrei, WVU thesis, 2001	buses	4.2	-
Petrushov, SAE 970408	buses	5.4	0.8 (14%)
Petrushov, SAE 970408	box trucks and vans, length ~5 to 6 m	2.4	0.6 (24%)
Petrushov, SAE 970408	box trucks and vans, length ~7 to 8 m	3.2	0.7 (20%)
Petrushov, SAE 970408	cars	1.2	0.6 (48%)

source	Aerodynamic loss reduction feature	C_DA (m^2) (frontal area $A=10m^2$)
"Industry options for Improving Ground Freight Fuel Efficiency", ICF Consulting, prepared for USEPA, 2002/ Detroit Diesel Spec Manager Software	None, 40" trailer gap	7.5
	Roof deflector	6.4
	Full aerodynamic package	5.8
	Reduced trailer gap, 18"	5.6
	Cab over engine	5.1

Heavy duty truck tire rolling loss coefficients are developed from the Petrushov work. In contrast to the light duty tire rolling loss, Petrushov includes the second order (in speed) rolling resistance force term (3rd order in power) which is accounted for in the "C" term of the road load Equation 3. Thus, the following relationship can be determined between the physical and the A, B, C terms from Equation 1 and Equation 3:

Equation 28.

$$A = C_{R0}mg$$
$$B = 0.0$$
$$C = \frac{C_d A_r \rho_{air}}{2} + C_{R2}mg$$

Where the C_{R0} and the C_{R2} are the zero and second order (in speed) rolling resistance force terms. Figure 12 displays the speed dependence of the aerodynamic force term.

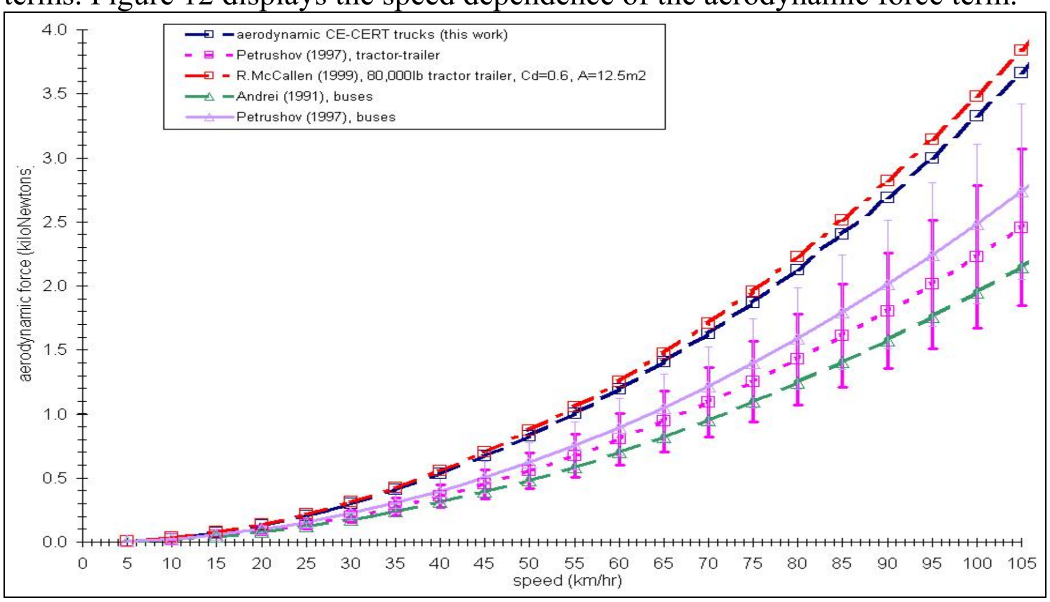


Figure 12. Comparison of aerodynamic forces from existing literature and from values determined using speed-time traces of CE-CERT truck coast downs.

Figure 13 displays the speed dependence of the rolling force term. For comparison, earlier work done at the University of Michigan (e.g., Yong and Gillespie) are also displayed. The aerodynamic force coefficient determined from the CE-CERT truck coast downs is also included. Note that at a certain speed, the aerodynamic effects overcome the rolling resistance effects. The agreement with University of Michigan [1984] work is within the 20% measurement and vehicle-to-vehicle variability. The uncertainty bars on the Petrushov tractor-trailer points illustrate these uncertainties. One exception out of the range of the uncertainties is the 1974 radial ply tires which are an additional 15% (a total of about 35%) lower than the rolling resistance losses developed from the Petrushov work.

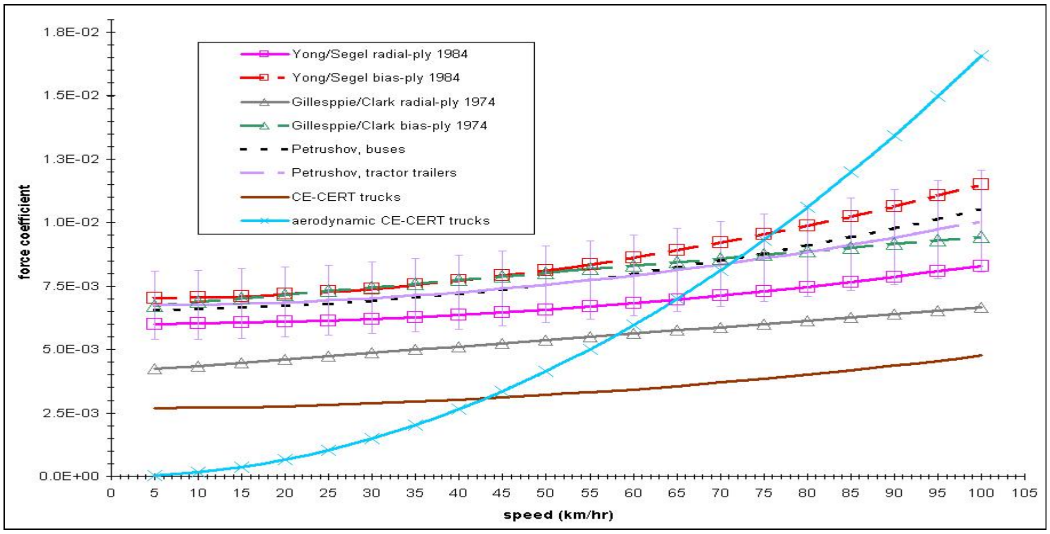


Figure 13. Comparison of rolling resistance force coefficients from existing literature and from values determined using speed-time traces of CE-CERT truck coast downs.

Finally, Table 9 lists the form of the coefficients that are developed from Petrushov as a function of vehicle mass and according to the three heavy duty truck and the single bus categories described above. The three parameters are listed as the typical road load A, B, and C coefficients. The four different categories are two single unit delivery categories, the tractor-trailer category, and the bus category. The two single unit delivery categories correspond to the two smallest weight categories.

Table 9. Road load parameters developed from V.A.Petrushov. (Note: Although the frontal area of the trucks in the study were relatively smaller than the CE-CERT trucks, the aerodynamic drag terms were similar and within statistical and measurement errors.)

Road Load Coefficients for Heavy Duty Trucks and Buses				
	8500 to 14000 lbs (3.855 to 6.350 tonne)	14000 to 33000 lbs (6.350 to 14.968 tonne)	>33000 lbs (>14.968 tonne)	Buses
A (kW*s/m)	$\frac{0.0996M}{2204.6}$	$\frac{0.0875M}{2204.6}$	$\frac{0.0661M}{2204.6}$	$\frac{0.0643M}{2204.6}$
B (kW*s^2/m^2)	0	0	0	0
C (kW*s^3/m^3)	$1.47+\frac{5.22\times10^{-5}M}{2204.6}$	$1.93+\frac{5.90\times10^{-5}M}{2204.6}$	$2.89+\frac{4.21\times10^{-5}M}{2204.6}$	$3.22+\frac{5.06\times10^{-5}M}{2204.6}$

In summary, these tire rolling loss road load parameters are in reasonable agreement with values from recent literature, but the aerodynamic loss coefficients are lower than what might be expected in the U.S. fleet. The tire rolling losses lie within the variability range of tire rolling losses determined within the last 30 years. The agreement is better for the latter U.S. fleet measurements. The uncertainty in the rolling losses is within 20%. Petrushov aerodynamic loss coefficients were found to be lower than the reported values from the U.S. fleet. For example, the values for an 80,000lb tractor-trailers are about 35% lower, with a C_D of 0.6 and a frontal area of 12.5 m^2 [McCallen, et al., 1999]. The Petrushov aerodynamic losses have an uncertainty of about 25% which includes a 5% measurement error along with about a 17% vehicle–to-vehicle variability. MOVES aerodynamic heavy-duty road load coefficients will be updated to reflect the higher values of aerodynamic losses more typical to the U.S. fleet.

Heavy Duty Diesel Engines

As with light duty vehicles, the Willans line methodology is used to determine both the engine efficiency and the engine friction of the heavy-duty diesel engines studied in this report. To construct the Willans line for these engines, instantaneous values of fuel mep and bmep were determined from measurements of fuel flow, engine speed, and engine load. The non-road engine measurements were made on an engine dynamometer [Fritz & Starr, 1998], the in-use / on-road measurements for the CE-CERT trucks, and the AATA buses were taken from the engine control module (ECM) with portable emissions measurement systems described elsewhere [Ensfield, 2002; Miller, et al., 2002; Barth, et al.,2004]. This section describes a new method of estimating engine efficiency and friction from on-road data.

Data from the non-road engine study, AATA buses and the CE-CERT trucks had fuel flow measurements from which the fuel mean effective pressure could be determined:

Equation 29. $$\textbf{fuel mep} = n_R * (1000/100)[\text{bar/MPa}] * FR * LHV / V_d N$$

where the fuel rate, FR, has units of kg/s, the diesel fuel lower heating value (≅43MJ/kg), engine speed, N, is in units of rev/s, the engine displacement, V_d, is in units of m^3, and n_R is 2 for four stroke engines. The factor of 1000/100 is a conversion factor from MPa to bar as indicated within the boxed parentheses.

Brake mean effective pressure or bmep is determined from engine torque maps and engine speed for the non-road engines. For both the buses and the CE-CERT trucks, the bmep is determined from engine maximum load curves, engine speed (N), and engine percent load measurements taken from the engine control module (ECM). From above quantities and the engine displacement, the bmep is calculated as follows,

Equation 30. $$bmep = \left(\frac{\%load}{100}\right) \times \frac{P_{max}(N)}{V_d N}$$

where %load is the measured percent of the maximum engine load, V_d is the engine displacement, N is the measured engine speed, and $P_{max}(N)$ is the maximum engine load [and includes volumetric efficiency (Ensfield, 2002)] at the current engine speed, N, determined from manufacturer specified engine maximum load curves.

Figure 14 and Figure 15 illustrate this methodology for all of the non-road engine dynamometer based measurements and for the CE-CERT truck 1 on-road measurements. The CE-CERT truck plot includes only those points where the vehicle accleration is greater than zero. Clearly, the on-road measurements illustrate the scatter associated with operation of the engine at many of the possible points of the engine map and indicate that a further refinement or data reduction will be needed to determine an accurate engine speed-engine friction relationship. The non-road engines, operated on a dynamometer, are restricted to the constant load operating points without transmissions (Table 7).

Table 10 lists the individual non-road engine results for indicated engine efficiency and the fit statistics. The non-road engine efficiencies range between 39% and 46% with an average value of 43% and a standard deviation of about 2%. These engines range in size from 0.2 liters to 34.5 liters (and were from 7 different engine manufacturers). The efficiencies show no indication of a dependence on engine size. Beyond 5L, there also appears to be little dependence of engine friction on engine size.

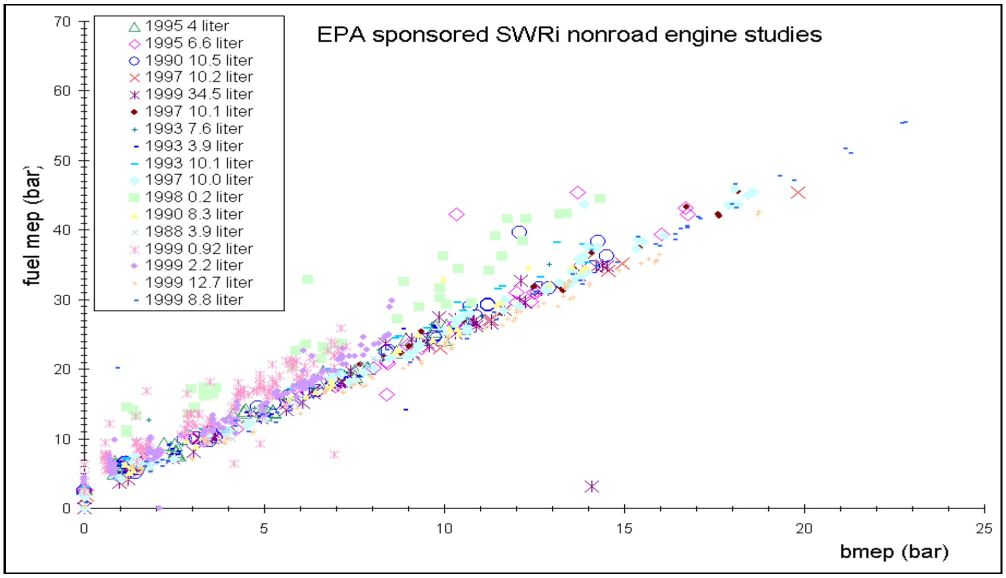


Figure 14. Willans line for nonroad dynamometer based measurements of fuel consumption and brake mean effective pressure.

Table 10. Engine efficiency results along with the linear fit statistics for the diesel non-road engines.

engine displacement (liters)	k (bar)	k standard error (bar)	$1/\eta$	$1/\eta$ standard error	R^2	η
0.2	8.7	0.9	2.43	0.1	0.93	0.421
0.92	5.2	0.4	2.37	0.09	0.83	0.406
2.2	3.4	0.2	2.46	0.05	0.95	0.444
3.9	2.5	0.2	2.25	0.04	0.99	0.462
3.9	3.1	0.5	2.16	0.09	0.95	0.451
4	3.3	0.3	2.22	0.04	0.99	0.413
6	2.3	1.0	2.42	0.1	0.92	0.392
7.4	2.0	0.4	2.55	0.05	0.98	0.446
7.6	3.6	0.3	2.24	0.05	0.99	0.424
8.3	2.4	0.5	2.36	0.06	0.98	0.437
8.8	2.4	0.6	2.29	0.05	0.98	0.430
10	2.3	0.2	2.33	0.02	0.99	0.410
10.1	3.1	0.3	2.44	0.04	0.99	0.458
10.2	2.2	0.4	2.18	0.03	0.998	0.419
10.5	2.5	0.5	2.39	0.06	0.98	0.461
12.7	1.8	0.2	2.17	0.02	0.99	0.441
34.5	2.3	0.2	2.27	0.03	0.99	0.412

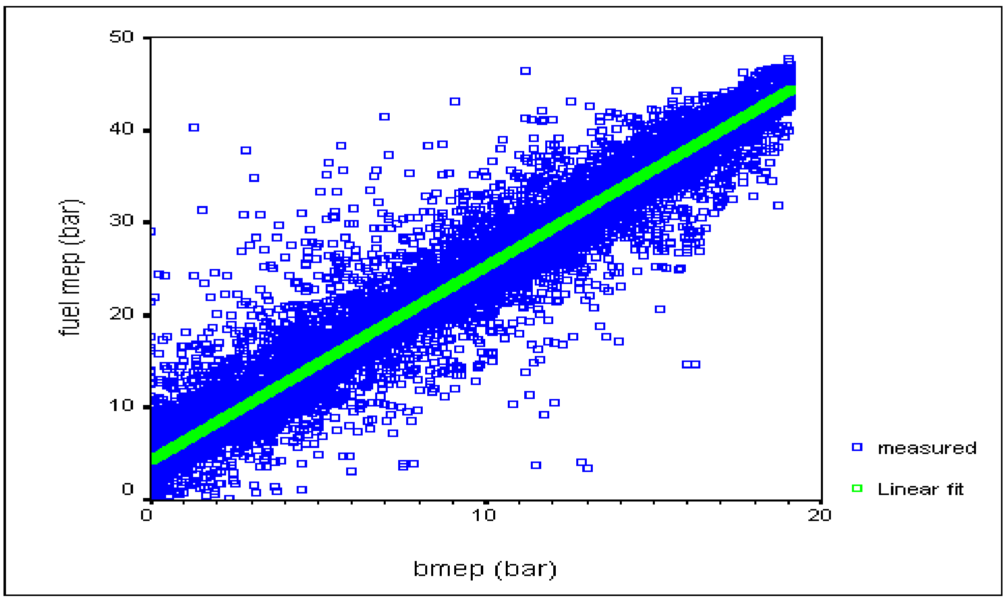


Figure 15. Willans line for CE-CERT truck 1 on road based measurements of fuel consumption and brake mean effective pressure determined from engine maps and engine % load measurements. The green line is a fit to the data. The points were limited to those with accelerations greater than zero.

Average engine efficiencies over all the on-road tractor trailer diesel powered vehicles is approximately 0.48 (with standard deviation of about 0.02) and the AATA buses have a slightly lower average of 0.46. Based on this, PERE uses 0.48 for all heavy-duty on-road diesel applications. Table 11 below lists the values for each of the individual trucks and a single number for all of the buses. Some of the linear fit statistics are included in the table.

A summary of the results are shown in Table 12. The on-road diesel trucks are in agreement with the smaller passenger car results of Wu and Ross [1997], who looked at four light duty cars with engine sizes ranging from 1 to 2.5 liters. R^2 of the fits are typically at or better than 0.9 and the standard errors in the slope parameter of the linear fits are around 20% of the fit value. The non-road diesel engine efficiencies are typically about 5% lower than the on-road vehicles and the buses are approximatel 2% less efficient than the diesel tractor-trailers which may be due to how the vehicles were driven (including significantly more transients).

Table 11. Engine efficiency results along with the linear fit statistics for the on-road CE-CERT tractor trailer and the AATA bus engines.

engine displacement (liters)	vehicle	k (bar)	$1/\eta$	standard error in $1/\eta$	R^2	η
8.5	buses	0.926	2.17	0.01	0.95	0.461
10.8	truck 2	1.81	2.14	0.01	0.92	0.468
11.9	truck 9	1.39	2.21	0.02	0.85	0.452
12.7	truck 6	1.90	1.97	0.03	0.63	0.507
12.7	truck 7	-0.043	2.15	0.02	0.86	0.466
12.7	truck 8	0.611	2.10	0.02	0.83	0.475
12.7	truck 3	-1.10	2.13	0.02	0.83	0.469
14	truck 4	3.91	2.05	0.01	0.95	0.488
14.6	truck 1	4.27	2.10	0.01	0.95	0.476

Table 12. Diesel engine indicated efficiencies for the engines in this study.

source	average efficiency	standard deviation (vehicle-to-vehicle differences)	engine displacement range	fuel delivery	# of vehicles or engines
-	-	-	(liters)	-	-
CE-CERT trucks	48%	2%*	10.8 to 14.6	Turbo- EUI	8
AATA buses	46%	-	8.5	TDI	15
Nonroad engines	43%	2%*	0.2 to 34.5	varied	17
Wu & Ross, 1997	47% to 49%		1.08 to 2.46	TDI	4

*R^2 of the fits are typically at or better than 0.9 and the standard errors in the slope parameter of the linear fits are about 20% of the fit value. So the total percent error in these engine efficiencies is ~ 25%.

In order to determine engine friction coefficients, it is necessary to further filter out the effect of transients - especially if our goal is to determine the relationship between engine friction and engine speed (Equation 11). A novel method of doing this is to isolate only low acceleration operating modes. Dynamometer based measurements of fuel rate and brake mean effective pressure are done at a single value of engine load and engine speed. On-road/in-use based measurements don't yield such ideal conditions. Therefore, different ranges of accelerations are used to limit the range of engine load points for a set of different engine speeds.

The methodology to find the frictional losses from in-use data is:

(1) construct a family of fuel mep – bmep graphs for different values (average values of intervals) of engine speed; the range of bmep's were limited by using positive vehicle accelerations near zero, e.g., large non-zero positive accelerations may be where transmission shifts typically occur and may add step-wise changes in engine speed; also negative values of accelerations where braking occurs will also not represent true engine map values of bmep and engine speed.

(2) curve fit each of the different graphs, yielding a set of k's and 1/η's for the range of engine speed.

(3) from these sets of coefficients, the slopes or engine efficiencies should be relatively constant. However, the fuel mep-intercepts should yield an engine speed dependence, which defines the engine friction.

(4) finally, fit the fuel mep intercepts (k) to a line dependent on engine speed

Figure 16 and Table 13 illustrate this procedure and show the relatively constant values of engine efficiency as well as the engine speed dependence of engine friction. When these values are fit to the (average value of the interval) engine speed, it yields the complete fuel mep – bmep dependence for the engine under consideration.

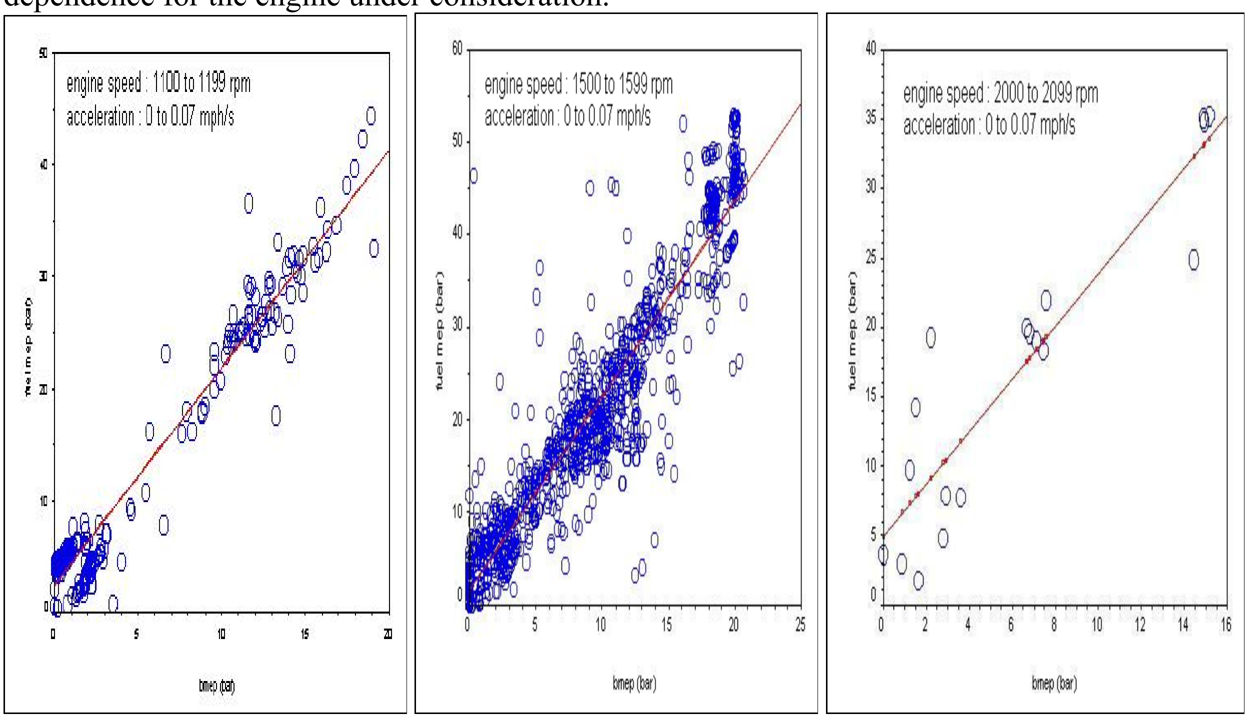


Figure 16. A series of figures, showing fuel mep vs bmep for engine speed values between 1100 and 2099 rpm, and accelerations between 0 and 0.007 mph/s. The linear fit to these data is the solid line.

Table 13. Example of the series of fuel mep vs bmep curves at different engine speeds, N, required to determine the speed dependence of engine friction. The usable data are from (9 of 12 usable) trucks on four drive cycles. (Note that the units of k(N) in this table are in bar and the units of engine speed are in rpm whereas in Table 14 the units are in kPa and rps.)

0 mph/s < acceleration < 0.07 mph/s					
k(N) (bar)	k(N) standard error (bar)	Mean N +/- Std. Deviation (RPM)	R^2	linear fit results for $k(N) = k_1 N + k_0$	
3.54	0.13	1045 +/- 26	0.21		
2.53	0.19	1154 +/- 27	0.95		
2.94	0.27	1260 +/- 28	0.88		
2.05	0.20	1352 +/- 28	0.89		
3.58	0.19	1448 +/- 28	0.89		
1.69	0.27	1549 +/- 30	0.89		
0.54	0.32	1647 +/- 28	0.89	k_0 (bar)	1.09
3.08	0.33	1747 +/- 29	0.85	k_0 standard error (bar)	1.6
2.55	0.46	1842 +/- 28	0.86		
3.7	1.1	1936 +/- 30	0.63	k_1 (bar/rpm)	0.00116
4.94	1.7	2050 +/- 30	0.83	k_1 standard error (bar/rpm)	0.0009
5.27	1.4	2135 +/- 21	0.84		
2.68	1.1	2228 +/- 14	0.84	R^2	0.04

The results of this study are shown on Table 14 and in Figure 17. A comparison is provided to both Millington and Hartles [1968] and a more recent work by Wu and Ross [1997] (averaging the four of their light duty engines). The comparison of this study's heavy-duty engine friction to the light duty in Wu and Ross is good, however, the comparison is between the former study's light duty to the present study's heavy duty engines. The fmep formula in Wu and Ross may not extrapolate accurately to large engine sizes (due to their limited data set). However, neither this work nor the Wu and Ross work are able to discriminate the second order effects of mean piston speed as was done by Millington and Hartles. These results also demonstrate the improvements in engine friction, which have occurred in the last thirty years.

Table 14. CE-CERT truck engine friction determination from on-road measurements [of 9 trucks]. *Values are standard deviations in the averages. The actual uncertainties are larger.

Acceleration range (mph/s)	k_0 (kPa)	k_1 (kPa/rps)	k_2 (kPa / (m/s))
(1) 0.001<accel<0.25	-9.32	12.6	-
(2) 0.001<accel<0.2	13.7	11.4	-
(3) 0.001<accel<0.15	96.1	7.85	-
(4) 0.001<accel<0.1	157	5.17	-
(5) 0.001<accel<0.07	109	6.98	-
Ave. of ranges 3, 4, and 5	121 (+/-32*)	6.66 (+/-1.37*)	-
Wu and Ross, 1997	135	5.4	-
Millington and Hartles	179	6.1	0.83

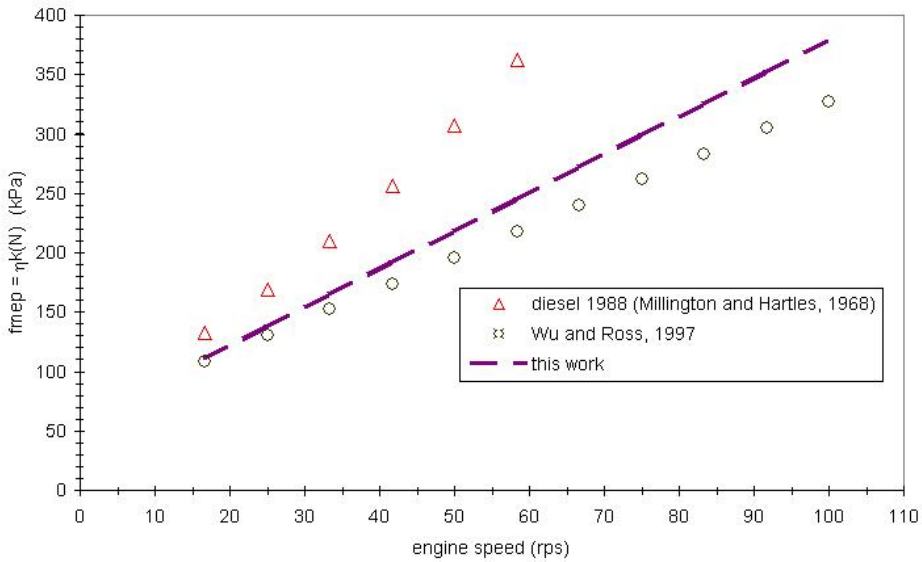


Figure 17. Engine friction results of this work compared with previous compression ignition studies from the comprehensive work of Millington and Hartles (and Heywood, 1988) and from a more recent work by Wu and Ross (1997).

Heavy Duty Transmissions

Heavy duty transmissions in PERE are assumed to work similarly to the light duty transmissions described earlier. The number of gears and shift points would certainly differ, but the algorithms and efficiencies are assumed to be identical.

The on-road measurements of engine and vehicle speed are measured simultaneously, thus allowing for the empirical determination of gear ratios, shift points, and N/v based on the methodology presented earlier. Because these speed-gear distributions will be inherently different for different vehicles classes, a relationship between transmission shift points and vehicle speeds are deduced for different vehicle transmissions classes in this section. Two transmissions are modeled; one for medium duty (based on the AATA buses, which were automatic) and the second for the heavy duty-diesel tractor-trailers (manual). These are supplemented with vehicle downshifting (during accelerations).

Figure 18 shows an example of the speed distributions for one of the CE-CERT heavy duty diesel tractor-trailers. Each of the 12 gears has a distinct distribution of speeds which tails off toward the lower speeds in the distribution. Although the distributions for the lower gears seem to be smeared, if that portion of the graph is expanded, the lower gear distributions appear distinct.

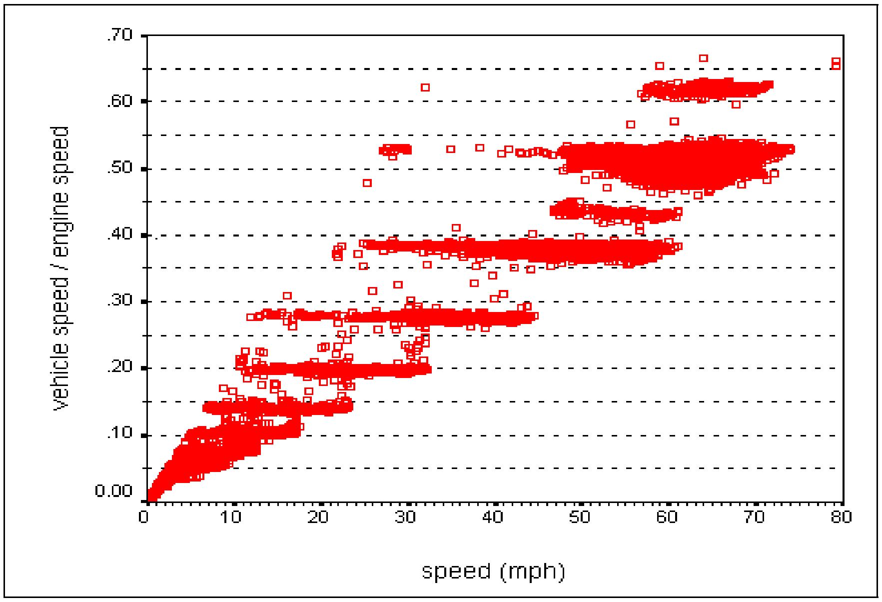


Figure 18. CE-CERT truck 1 vehicle speed to engine speed ratios vs vehicle speed to determine average speeds for a given gear.

The CE-CERT trucks have between 10 and 13 total forward gears. An average speed from each of the distributions is determined along with a standard deviation in that average speed. This was also done for the AATA buses. Figure 19 displays the results of those averages along with a polynomial fit to the two different vehicle speed-(engine speed/vehicle speed) ratio relations. From these, the current gear and engine speed are determined.

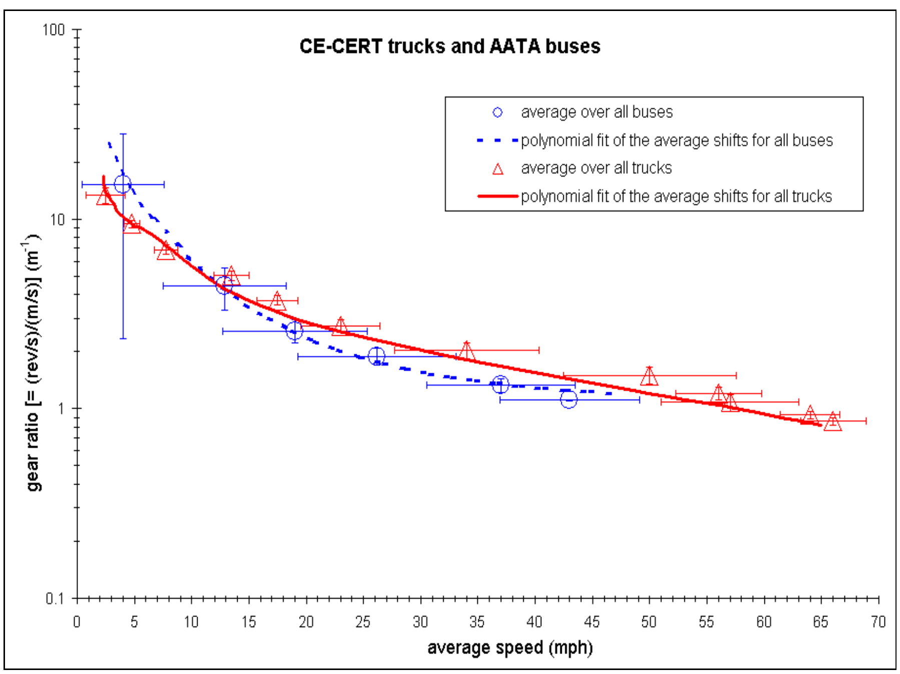


Figure 19. Final engine speed - vehicle speed relationships for buses (red) and the heavy duty trucks (blue). The error bars are standard deviations determined from the distributions depicted in vehicle speed - engine speed ratio graph.

The transmission parameters chosen for the medium duty 6-speed automatic, and the heavy-duty 12 speed manual transmissions modeled in PERE are displayed in the following table.

Table 15. Transmission parameters for medium duty and heavy duty vehicles in PERE.

Vehicle Type	HD	MD
Transmission Type	Manual 12	Auto 6
N/v (rpm/mph)	26.7	26.7
Nidle (rpm)	700	700
trans eff	0.95	0.85
Shift point 1-2 (mph)	2.48	6.2
Shift point 2-3	4.75	15.6
Shift point 3-4	7.75	22.9
Shift point 4-5	13.5	30.3
Shift point 5-6	17.5	37.7
Shift point 6-7	23	
Shift point 7-8	34	
Shift point 8-9	50	
Shift point 9-10	56	
Shift point 10-11	57	
Shift point 11-12	64	
g/gtop 1	13	14.1
g/gtop 2	9.4	4.46
g/gtop 3	6.9	2.66
g/gtop 4	5	1.91
g/gtop 5	3.7	1.34
g/gtop 6	2.7	1.11
g/gtop 7	2	
g/gtop 8	1.5	
g/gtop 9	1.2	
g/gtop 10	1.1	
g/gtop 11	0.93	
g/gtop 12	0.86	

The last component of the transmission model is the (power) downshifting. Vehicle power must be checked against the actual maximum manufacturer specified engine power for a given engine speed. A downshift occurs whenever the vehicle tractive power is greater than the maximum engine power at a given engine speed. Because PERE's primary purpose is to model a class of vehicles, rather than individual engine/vehicle combinations, an average engine maximum torque/power curve is used. A scalable torque (or bmep) curve is determined from a number of different diesel engines ranging in size from 8 to 15 liters and is shown in Figure 20. The map is scaled depending on the maximum power or engine displacement. The level of turbo-charging cause some variability in these scaling factors. (Note: Actual validation of these heavy duty algorithms used the manufacturer listed maximum engine power). This figure is consistent with Figure 9.

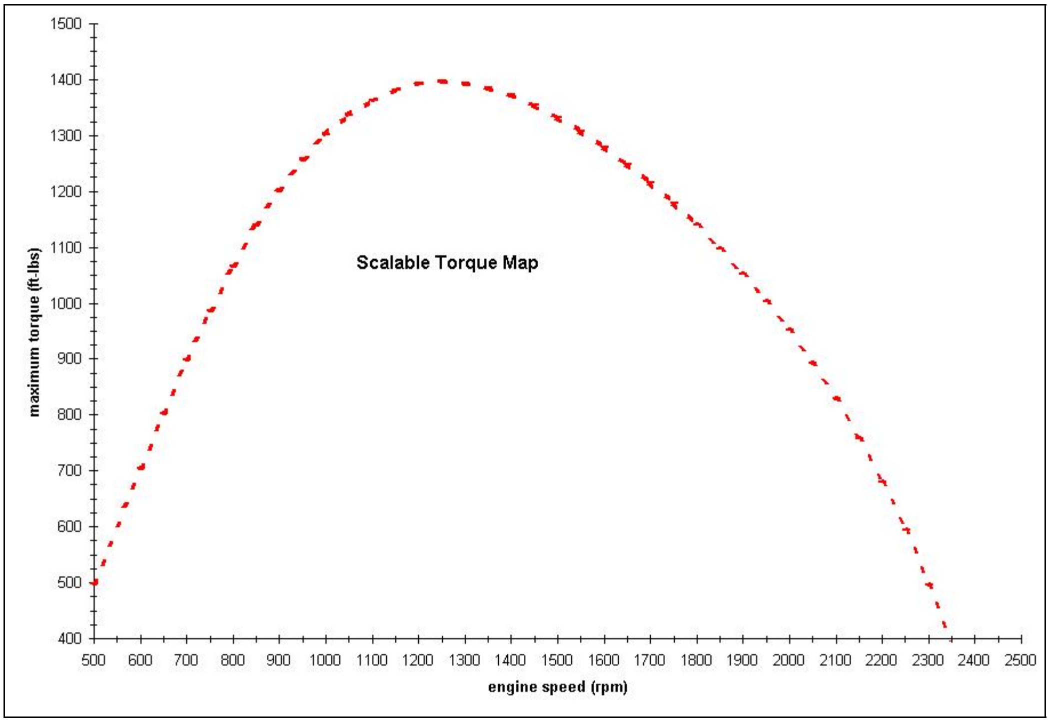


Figure 20. Scalable torque determined from comparison of 13 individual diesel engine maps from 4 different engine manufacturers. This scalable map is based on a 380 hp, 12 liter diesel engine extrapolated to 500 rpm and 2350 rpm.

Another method employed is to check the vehicle maximum bmep against a single highest value of bmep based on an average maximum engine torque. From the USEPA's CFEIS database an average maximum engine torque from over 638 diesel engines ranging in size from 1.8 to 27 liters is calculated. It is used to find an average maximum bmep value and a standard deviation,

$$bmep_{\max} = \frac{2\pi n_R \tau_{\max}}{V}$$

$$\overline{bmep_{\max}} = 1648 \pm 440^{*} \; N \cdot m$$

This method is not used for MOVES rates, but was used for an earlier version of model development.

Validation

Using the fuel rate equation along with the other elements of the PERE model described above, calculated values of fuel consumption are compared with the measured fuel rate for an independent vehicle trip. Both bus and truck calculated and measured emissions and fuel consumption numbers are compared on Table 16. All calculated values are within 10% of the measurements. For all cases considered here the values are neither systematically high nor low. However, the heavy-duty trucks had relatively large discrepancies for driving cycles with average speeds less than 40mph. At the time of this writing, driving cycles with average speeds less than 40 mph had calculated values of fuel consumption more than 20% lower than actual measured values. These discrepancies are currently being addressed and will be adjusted in the next version of PERE.

Table 16. Percent differences between calculated and measured value of fuel consumption for 5 buses and two CE-CERT truck 1 trips. All differences are within 10%.

Vehicle/ trip	fuel % difference between measured and calculated mass values
AATA bus 1	9%
AATA bus 2	-2%
AATA bus 3	4%
AATA bus 4	-5%
AATA bus 5	-6%
CE-CERT truck 1, Victorsville	9%
CE-CERT truck 1, Palm Springs	-1%
CE-CERT truck 2, Victorsville	3%
CE-CERT truck 2, Palm Springs	9%
CE-CERT truck 4, Palm Springs	1%
CE-CERT truck 4, Palm Springs	5%
CE-CERT truck 5, Palm Springs trip 1	-1%
CE-CERT truck 5, Palm Springs trip 2	8%
Calculated fuel consumption is typically 20% or more under predicted for drive cycles with average speeds less than 40mph (see text for explanation)	

Section VII: Advanced Engines and Hybrid Vehicles

From reading manufacturer press releases, and seeing the new vehicles being offered, it is becoming evident that advanced technology vehicles, such as hybrids, will become more commonplace in the near future. A hybrid vehicle combines two forms of propulsion in order to optimize efficiency and fuel economy. The incremental cost of hybrid vehicles is expected to decrease as volumes increase, and as public acceptance increases. There are also other types of advanced vehicles, such as clean diesel, and fuel cell vehicles, which may claim a portion of the market in the future. It is important to understand the possible effects that these vehicles will have on the fleet and the environment.

The following sections describe the assumptions that went into modeling advanced internal combustion engines (gas and diesel), hybrid and fuel cell vehicles. Validation results are also be presented.

Advanced Internal Combustion Engine (ICE)

In this report, "advanced" technology is defined as a vehicle (or component) that is improved over those in most vehicles currently. Advanced ICE vehicles might employ one or more non-standard engine or driveline technologies with superior thermal, and/or driveline efficiencies, which lead to better fuel economy and performance. Examples include: lean-burn gasoline engines, variable displacement, direct injection gasoline, and continuously variable transmission (CVT).

In order to model future vehicles, it is educational to study current technologies comparing them with their predecessors. Two areas that have seen marked improvement in the past, are engine friction and power. The former likely leading to the latter. Typically, improvements in (overall) engine efficiency can either improve fuel economy, or increase engine size in order to increase power performance (thus maintaining the same fuel economy). The latter has been the main trend over the past few decades, though there is strong evidence to show that specific power (power per unit displacement) has also improved [Chon and Heywood, 2000].

Based on the research of Chon and Heywood, the relationship between engine power and model year has been established for the past 15 years. This trend has been projected forward by Weiss et al. [2000] and is shown in Figure 21. The friction reduction projections have also been added to the figure, based on the research of Sandoval and Heywood [2003] and Nam and Sorab [2004] at a rate of approximately 10% per decade. Advanced diesel trends are assumed to be the same, though their starting values may be different.

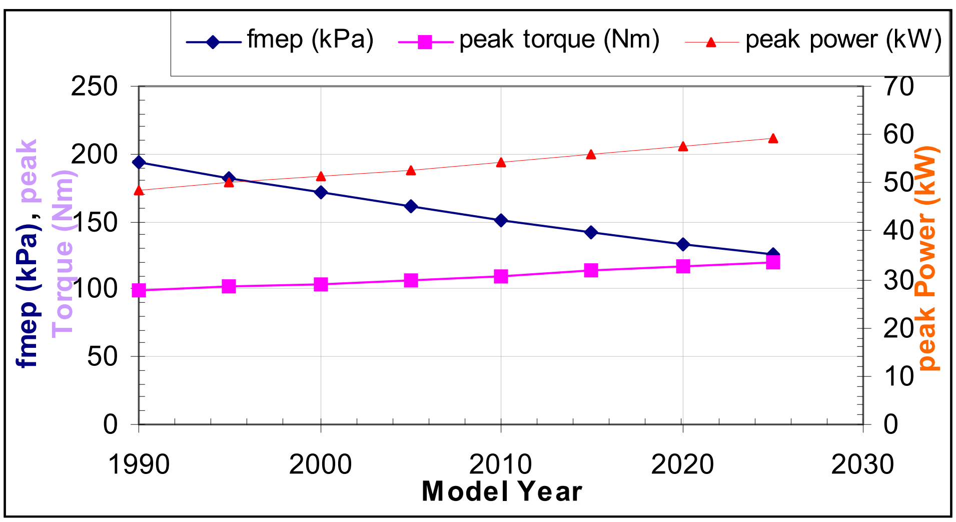


Figure 21. Projected gasoline engine trends over time [Chon & Heywood, 2000; Nam & Sorab, 2004].

In PERE, this trend is modeled with a quadratic fit on the friction term k_0 (Equation 11) in the following equation. k_1 is assumed to remain constant over time. The trends for diesel are less well known, but it is assumed to follow the same progression as gasoline.

Equation 31. $$k_0\ (kJ/L) = a_0 + a_1Y + a_2Y^2$$

Where Y is year and

	Gas	Diesel
a_0	7.1505E+01	3.5205E+01
a_1	-6.8667E-02	-3.3833E-02
a_2	1.6498E-05	8.1286E-06

The current version of PERE does not currently change peak torque (or bmep) curves with model year, due to the relative lack of sensitivity of the model to this term. Future versions of the model may include this option.

Beyond incremental improvements in engine friction, it is difficult to predict how engines will change in the next 30 years. It is conceivable that production gasoline engines can have improved indicated efficiencies. This could be accomplished with the trend of "variable everything". The Atkinson cycle engine (as in the current Toyota Prius) is a good example of variable valve timing being used to increase efficiency at the cost of power. The lean burn engines of Honda (currently installed in the Civic HX, Hybrid, and Insight models) improve efficiency by burning gasoline under lean conditions at low loads.

The Honda Lean burn engine parameters are determined from Ogawa et al. [2003]. The engine improves the best bsfc (best brake specific fuel consumption, or "sweet spot") over conventional engines by approximately 20%, and decreases friction by 8%. PERE models this with an indicated efficiency of ~0.48, with an 8% friction improvement. During higher engine speeds, the engine reverts to stoichiometric operation, where it is still more efficient than conventional engines (0.44). The engine speed breakpoint is not provided in the reference, so in PERE it is assumed to be 50% of the peak power rpm (6100 rpm*0.5). These are probably optimistic estimates, since the vehicle must still meet stringent NOx emissions standards. Also, improvement in peak efficiency already includes improvements in friction.

There is less information on the Toyota Atkinson engine. PERE assumes that the efficiency is 15% greater than that of conventional engines, until the rpm cut point is reached [Heywood, 1988]. The peak torque values are decreased correspondingly. Beyond that point, it behaves as a conventional engine.

Gasoline Direct Injection (GDI) or Homogeneously Charged Compression Ignition (HCCI) engines are also a promising alternative, though these are still mainly laboratory engines. In the near future, there may be engines with variable displacement, variable compression ratio, cylinder deactivation, etc. coming on to the scene. Moreover the current move toward 42 Volt power systems on the vehicle will allow for Integrated Starter Generator systems to start and stop the car during idle. This has been referred to as "minimal" or "mild" hybrid.

The specific technologies described above are for PERE validation purposes only. For the "generic" advanced internal combustion (AIC) engine vehicle in MOVES, PERE uses target coefficients, rather than choosing a suite of specific technologies. These target values are assumed to be a 10% improvement in indicated efficiency (0.44), and engine friction equivalent of 2015 (Figure 21). The diesel AIC engine assumes slightly less ambitious targets (η = 0.50), and friction improvement of 15%.

In addition to technical advances in engine design, it is possible that tire rolling resistance and vehicle aerodynamics can improve in the future. Currently passenger cars have tire rolling resistance of approximately 0.009. It is unlikely that these will improve significantly without sacrificing other performance measures. The aerodynamic resistance and vehicle frontal area of vehicles have room for improvement, but here, form tends to follow function, or design. Current cars have coefficients of drag approximately 0.30 - 0.35 (Toyota Camry C_d = 0.30). Vans and SUVs have Cd ~ 0.35 – 0.40 and pickups range from 0.40 – 0.45 [Gillespie, 1992]. The exceptional 2004 Toyota Prius has a C_d of 0.26, and the Honda Insight has the best in class coefficient of 0.25.

Hybrid Vehicles

Currently the most successful method implemented commercially to improve the overall efficiency of vehicles is to hybridize their drivetrains, by adding an electric propulsion path to the wheels.

There are a number of electric hybrid vehicles on the road today: The Honda Insight was the first to be introduced in the United States in 1999. It was followed by the Toyota Prius and the Honda Civic (the Prius was first sold in Japan in 1997). Other manufacturers have also announced that they will release hybrid models of various kinds within the next few years.

There are several different kinds of hybrid vehicles under development. All share the common trait that there are two power sources that drive the vehicle forward during different operating modes. Most methods hybridize an internal combustion engine with an energy storage device such as a battery, ultracapacitor, or even hydraulic pump (fuel cell hybrids will be discussed in the next section). The hybrid configuration is usually either series, parallel, or some combination of the two. Series hybrids run off electric power only, and the batteries are recharged by the engine, which can run (continuously) in an efficient mode. Series hybrids suffer from the disadvantage that both a large battery and motor are required, while losses are incurred in both charging and discharging the battery. Parallel hybrids run the drivetrain alternately with the engine, battery or both. This allows for a smaller battery/motor than the series hybrid and for the engine to run efficiently. The disadvantage compared to the series configuration is that the system requires more sophisticated strategy, packaging and transmission. Some models require a separate generator. The Honda hybrids are parallel, while the Toyota Prius is a combination series/parallel.

The spectrum of hybrids soon to be on the road is broad. General Motors will soon sell a “minimal” or “mild” hybrid version of its Sierra/Silverado truck, which is expected to improve fuel economy by 12%. By 2007, they announced a displacement on demand (cylinder deactivation) hybrid system for the Tahoe and Yukon line [GM, 2004]. Ford released a full hybrid version of the Escape compact SUV in 2004. Honda is selling a hybrid version of the

Accord in 2004, which will have Variable Cylinder Management (VCM) in its V6 engine [Honda, 2004]. Toyota announced the Highlander hybrid in 2005, while the Lexus 400 hybrid is expected in 2004 [Toyota, 2004]. All of the hybrids mentioned above are in (mostly) parallel configuration and store secondary energy in a battery. They also use "advanced" gasoline engines. All of these hybrids recharge the batteries by employing the motor to recapture energy from the brakes, however, they do not all use the motor for traction.

With the promise of hybrids hitting many of the markets segments in the near future, it is imperative that any modeling of future vehicles include this class of vehicle. There have been a number of models developed, which captures the behavior of hybrid vehicles. ADVISOR (developed at NREL) is widely used in the research community [Kelly, 2001]. PSAT is another model developed at Argonne National Laboratories [Rousseau et al., 2004]. They both model conventional as well as hybrid and fuel cell vehicles. They are also relatively sophisticated models that run on the MATLAB® platform. Unfortunately, the models could not easily be integrated into MOVES for several reasons. Namely, the intensive data and effort required to calibrate the model is prohibitive for trying to model broad sections of the fleet required. It was thus necessary to adjust the approach so that a model could be run on the PERE spreadsheet. From there, it may either be run in spreadsheet mode, or programmed directly to link with MOVES in the future. The basic framework for PERE is similar to ADVISOR though: starting with a driving cycle input, calculate the road loads, then distribute the power and losses to the various energy conversion and storage media. Many other models exist as well, but the hybrid architecture in PERE is roughly based on the simple control logic described in Weiss et al., [2000] at MIT.

Strategy

Figure 22 shows the PERE flow chart for the parallel hybrid design. The battery can be any energy storage device (ultracapacitor or hydraulic). The logic control is relatively simple [Weiss et al., 2000 (MIT)]. When power demand is less than the hybrid threshold (2kW in the MIT case), the car runs on battery alone. Therefore, the engine shuts off during decelerations in the model. Beyond the threshold, the car only runs on the gasoline engine. When power demand is greater than the peak engine power, then the battery assists or provides the needed boost. When braking, part of the energy is recaptured into the batteries. Each of the power paths described has its unique losses. This strategy is modified slightly in PERE to require that the hybrid threshold be set such that the state of charge on the battery is sustained over the course of the driving cycle input, i.e. the battery ends the test with the same charge with which it started. In the validation, charge is conserved over the Federal Test Procedure driving cycle (city and highway). Other differences to the MIT model will be described below. The parallel power and state of charge algorithms are shown in Appendix A.

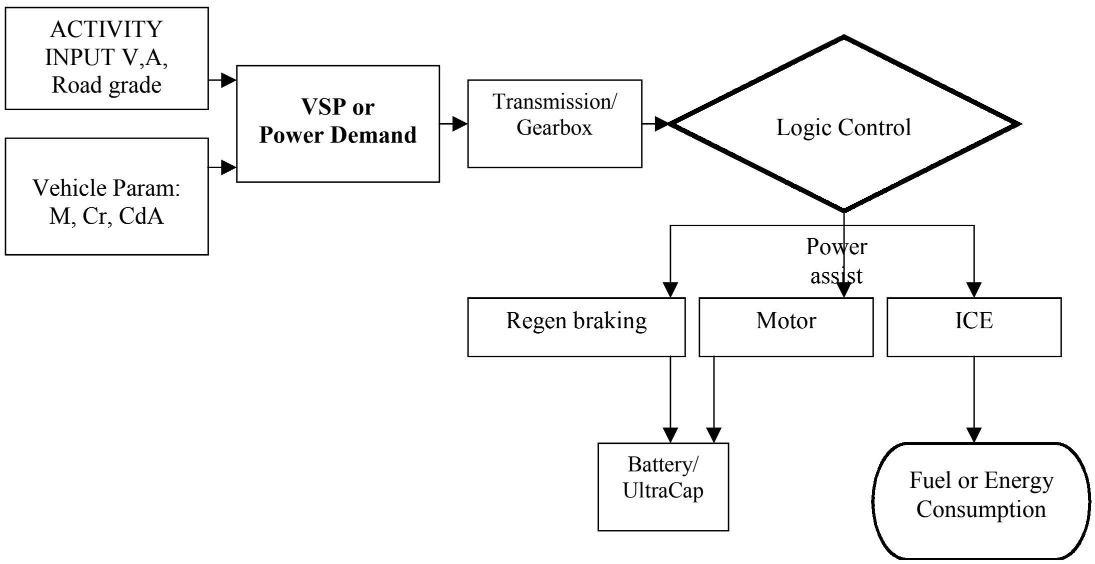


Figure 22. Parallel hybrid (gasoline/battery) flow chart.

Actual hybrid strategies can get much more sophisticated, where the battery can get recharged by the engine directly, during operation (as is the case for the Toyota Prius THS - Toyota, 2003). This system is a series/parallel architecture and is especially appropriate for stop and go driving. However, the PERE methodology takes the basic approach of distributing energy in the system defined by the strategy above, and taking into account losses. Thus, whether the energy is stored in the battery, or converted by the engine, the same overall energy is required, therefore the overall predicted fuel economy should be similar to what is measured (within 10%). The main omissions of PERE at this time is that it does not include a cold-start module. Cold start factors will be inserted by MOVES, not PERE. Hybrid vehicle fuel consumption can be significantly larger during cold start to heat the catalyst to light-off temperatures and also to charge up the batteries or capacitors (if needed). Cold start factors are discussed in greater detail in the Sensitivity section.

Hybrid threshold was chosen in order to give close to net zero state of charge over the driving cycle input (FTP city and HWY). This loosely simulates a “charge sustaining” strategy (‘pseudo-charge sustaining strategy’). For the hybrids modeled in this report, this threshold fell between 2 and 4 kW. Values are in Appendix B. This strategy dictates that the batteries are only being recharged through regenerative braking, and not the engine. This can still result in a charge sustaining strategy if the batteries are not used excessively for traction.

Unfortunately, MOVES cannot at this time accommodate hybrid vehicles with a series element. This is due to the design of VSP (vehicle specific power) based emissions in the model, which relates road load causally to fuel consumption. Series hybrids are not strictly “load-following” in this manner, therefore, the fuel consumption can follow some time after the road-load transient. This “history-effect” is impossible to model in the current MOVES framework.

Motor/Generator/Inverter

The conversion between electrical and mechanical energy is done with a motor. In reverse this is called a generator. For hybrids, we will use the term “motor” to indicate both devices. In PERE the motor operates both forwards and backwards at different times depending on the driving, with different losses. Weiss et al. [2000] estimates the motor system (including converters and inverters) average efficiency to be 76% with an additional 15% loss during regenerative braking. This estimate is likely low given the recent improvements in motor system efficiencies. The peak motor efficiency (alone) is higher, greater than 90% for a permanent magnet brushless DC motor, though it depends on the size of the motor [Laramie & Dicks, p. 279 2000, Ogawa, et al., 2003]. However, the MIT is a cycle average and value includes other system losses (such as from an inverter, which can have efficiencies of ~94%). Motor efficiency depends somewhat on the operating mode (speed and torque), but PERE also assumes a fixed value for simplicity.

The motor power and torque curve is shown in Figure 23. It is scalable with peak power [Weiss et al., 2000]. In the present version of PERE, the motor in the model is connected directly to the wheels (with a single gear step), bypassing the multi-step transmission (and thus the losses). However, it is more likely that real-world hybrid motors will require a multi-step transmission to maintain efficiency and torque performance. A future version of PERE may shift the motor to connect to the engine output shaft. At the level of modeling in PERE, however, the overall system losses would likely be similar.

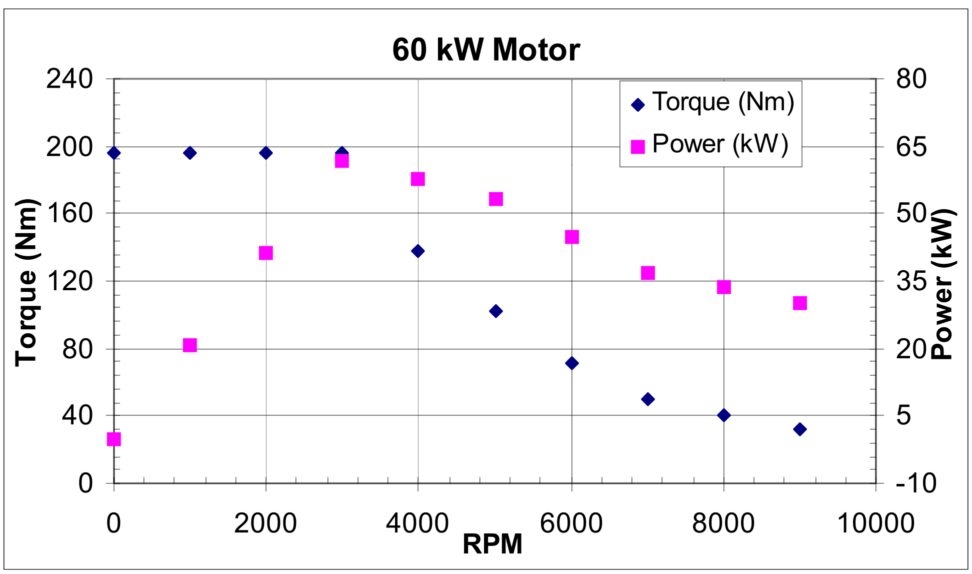


Figure 23. Motor peak torque and power from Weiss et al., [2000].

In PERE, the peak motor torque is scaled by power rating (calibrated to 60kW) according to the following polynomial:

Equation 32. $T\ (\mathrm{Nm}) = P_m(a_0 + a_1N + a_2N^2 + a_3N^3 + a_4N^4 + a_5N^5 + a_6N^6 + a_7N^7)/60$

Where

P_m is motor power rating (kW)

$a_0 = 196.26$

$a_1 = -1.2542$

$a_2 = 0.085004$

$a_3 = -5.1976e-4$

$a_4 = -4.4345e-5$

$a_5 = 8.1616e\text{-}7$
$a_6 = -5.2169e\text{-}9$
$a_7 = 1.1566e\text{-}11$

For the sake of nomenclature, we define the terms "moderate" and "full" hybrid. There is no standard definition in the literature since there are no clear demarcations, only a continuous spectrum of possibilities. In one reference, the separating value is defined by the ratio of peak motor power to peak motor + engine power [An & Santini, 2004]. So that a conventional vehicle powered with an engine has a ratio of 0, and a "pure" battery electric vehicle has a ratio of 1. This report remains consistent with this approach and defines moderate hybrid as having a ratio < 0.25. An example is the Honda Civic Hybrid (~0.14). The full hybrid has a ratio =>0.25. The 2004 Toyota Prius has a rating of ~ 0.47. This definition does not necessarily imply that moderate hybrids are worse, or less efficient than full hybrids. In reality, however, the size of the motor is meaningless without a battery (or ultracapacitor) that can manage the current flows in and out of it. The effect of hybridization naturally depends on the hardware, and what functions are built in, more so than the ratio of any two numbers.

Energy Storage Devices – Batteries, Ultracapacitors and Hydraulics
Batteries on hybrid vehicles tend to be of the nickel metal hydride (NiMH) variety. They have relatively high energy densities and can withstand many charge/discharge cycles, while being relatively affordable. Compared to a conventional vehicle, however, batteries add significant cost and weight. PERE does not model battery weight independently, but wraps the electrical system weight to an aggregate added weight to the vehicle (described below). In the future (when the prices may drop), lithium ion batteries may significantly reduce the weight of electrified vehicles.

In PERE, each second the power draw (discharge) from the battery is adjusted by the discharge efficiency (95%) as well as the motor efficiency. To be more accurate, this efficiency really depends many factors, including the state of charge of the battery, temperature, history etc. but is approximated as constant. During brake recharge, the rate includes a regenerative brake efficiency of 85% [Weiss et al., 2000]. There is added loss due to fact that front wheel drive hybrids only can recapture braking power from the front wheels (not the rear). Thus up to approximately 75% of the braking energy is available for recapture [An & Santini, 2004]. However, the new Prius brake by wire system can recapture significantly more [Toyota, 2003]. After the losses are included, the power is integrated over each second of the driving cycle, and then the discharge (positive) and recharge (negative) totals are added to give a final state of charge (SOC), as a fraction of the kWhr rating. Currently, PERE is designed so that the user adjusts the hybrid power threshold (manually) until the state of charge is close to zero for the given driving cycle. A more sophisticated charge sustaining strategy would require coding of the strategy and is considered for a future version of PERE. A more accurate battery model would include voltage/SOC/temperature (or similar) plots, and the spreadsheet would need to ensure that the battery can handle transients. However, due to the relative complexity of battery systems, there are many possible levels of modeling.

The battery is assumed to have a rating of 0.936 kWhr on the moderate hybrid, the nominal rating for the Honda Insight (the Civic is 0.864). For the full hybrid, the battery is assumed to be similar to the ’04 Prius, or 1.31kWhr. However, because we have a ‘pseudo-charge sustaining strategy’ and the vehicle is not assumed to drive on any extremely aggressive driving cycles, this number does not affect the model. In reality it is more important to define the limitations of the hybrid based on the total electrical power generated from the battery/motor system. PERE simplifies this, by placing emphasis on the motor system only. This assumes that manufacturers will integrate a “properly” sized battery for the motor. If PERE is required for high performance (0-60mph) runs, this, and other, portions of the model may require revision.

Replacing a battery with an ultracapacitor changes the model only slightly. The discharge and recharge efficiencies may be different, as would its ability to recapture energy from higher power braking events.

Hydraulic hybrids combine internal combustion engines with an hydraulic energy storage device. This storage device stores the energy in pressurized gases. The hydraulics would respond similar to an ultracapacitor except that the link would be mechanical, rather than electrical. This would increase recapture efficiency. The EPA is developing this technology with industry partners through cooperative research and development agreements, including implementation of the technology on a number of prototype vehicles [Alson et al., 2004]. Since hydraulic hybrids do not require large batteries (which are expensive), they lend themselves well to a series configuration.

Vehicle Weight and other Specifications

Hybrid vehicles are usually able to downsize the engine in order to save weight and fuel, however the added components of the motor, inverter, battery, and other connecting components more than compensate for the engine weight loss. Previous studies have summed the weights of the components based on power density in order to estimate a vehicle weight. This method has limitations in that other structural and component weights may be omitted. This report uses a different approach. The weights of existing hybrids are compared with their conventional vehicle counterparts. Table 17 shows the weights of various hybrid vehicles. In particular the Honda Civic Hybrid is compared with the Civic DX, and the Dodge Durango hybrid is compared with the Durango 4WD. While the Durango is no longer planned for production, it is still useful to compare specifications. The Prius and Insight obviously have no direct partner, so no direct weight comparison could be done. The weight ratio of hybrid to non-hybrid is found to be 1.07, or a 7% increase. This is based on the ratio of test weights, which is curb weight + 300 lbs. Though not shown, the Silverado “mild” hybrid (4WD) is 5% heavier than its conventional truck counterpart. The diesel hybrid is assumed to have a 4% increase over its gasoline hybrid counterpart. As more hybrid versions of vehicles are released, this mass increase ratio estimate can be improved.

Table 17: Hybrid Vehicle Specification in Comparison to Conventional (if exists). All sources are from EPA CFEIS database, Manufacturer, Car&Driver or Road&Track

websites.

	model	max torque	max power	displacement (l)	curb weight (kg)	Motor Power	Battery Capacity (kWhr)
Hybrid	**Honda Insight**	89.5N-m@ 2000rpm	50kW@ 5700rpm	0.995	853	10.4kW@ 3000rpm	0.936
	Honda Civic Hybrid	118Nm@ 43300rpm	63.4kW@ 5700rpm	1.339	1213	10.kW@ 3000rpm	0.864
	2003 Toyota Prius	111N-m@ 4200rpm	52kW@ 4500rpm	1.497	1254	33kW@ 1540rpm	1.31
	2004 Toyota Prius	111Nm@ 4200rpm	57kW@ 5000rpm	1.497	1311	50kW@ 1540rpm	1.31
	Dodge Durango	305N-m	130.6kW	3.9	2389	66.4kW	?
Conventional	**Honda Civic DX**	149Nm@ 4500rpm	86kW@ 6100rpm	1.668	1111		
	Honda Civic HX	151Nm@ 4500rpm	87.3kW@ 6100rpm	1.668	1111		
	Toyota Camry SE	220Nm@ 4000rpm	117kW@ 5600rpm	2.4	1425		
	Dodge Durango 4wd	475Nm@ 3000rpm	186kW@ 4200rpm	5.9	2251		

Many of the other vehicle attributes that aren't published had to be approximated. Often coefficients of drag and weights are available, but rolling resistance is not. Frontal area is approximated from the published dimensions (Equation 2) if a direct measurement is not provided.

The table of hybrid coefficients are also in Appendix B. Many of the variables may have a model year relationship into the future. E.g. component weights, Cr, Cd, Pacc, motor efficiency, regeneration braking efficiency, discharge/recharge efficiency, etc. However, most of these are expected to drop somewhat and level off in the near future. The accessory term (Pacc) is expected to rise in the next few years with the addition of more electrical loads (and luxuries).

Accessories

Currently, PERE does not model air conditioning or supplemental cabin heat in hybrid (or fuel cell vehicles). Therefore the accessory loads are a relatively small portion of the power required (~0.75 kW). However it is not insignificant, if the accessories were run completely on the batteries, there would be less power remaining for accelerations. In addition it is expected that accessory power will increase in the future.

In PERE, the accessories are run off the battery when the battery is on a discharge cycle. Otherwise it runs off the generator (engine). The algorithms are listed in the Appendix A.

Air conditioning and heating are potentially large loads for hybrid vehicles. From a fuel economy standpoint, the accessory loading can be large. This is due in large part to the fact that the engine might have to continue running under conditions when it would normally shut off (idle and deceleration). In MOVES, the effect of air conditioning will be applied to the base AC-off energy consumption rates developed by PERE. A direct air conditioning model for PERE is considered for the future.

Model Calibration
Since the PERE hybrid strategy model is based on Weiss, et al. [2000], the model is 'calibrated' to the model "MIT hybrid" vehicle. The MIT report gives most of the vehicle parameters required for the model. Figure 24 compares the fuel economy results of PERE with the MIT hybrid. Figure 25 compares the state of charge of the batteries following the FTP test. Since the models are not identical, differences are expected. However, the fuel economies are an excellent match. This indicates that the modeling methodologies are quite similar.

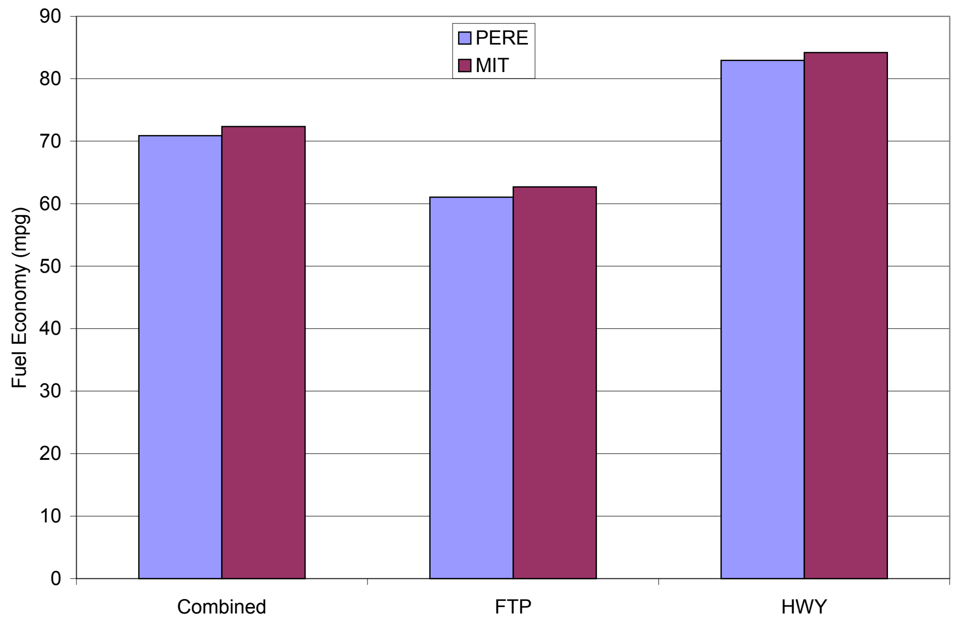


Figure 24. Fuel economy of a model year 2020 hybrid car compared to MIT study [Weiss et al., 2000].

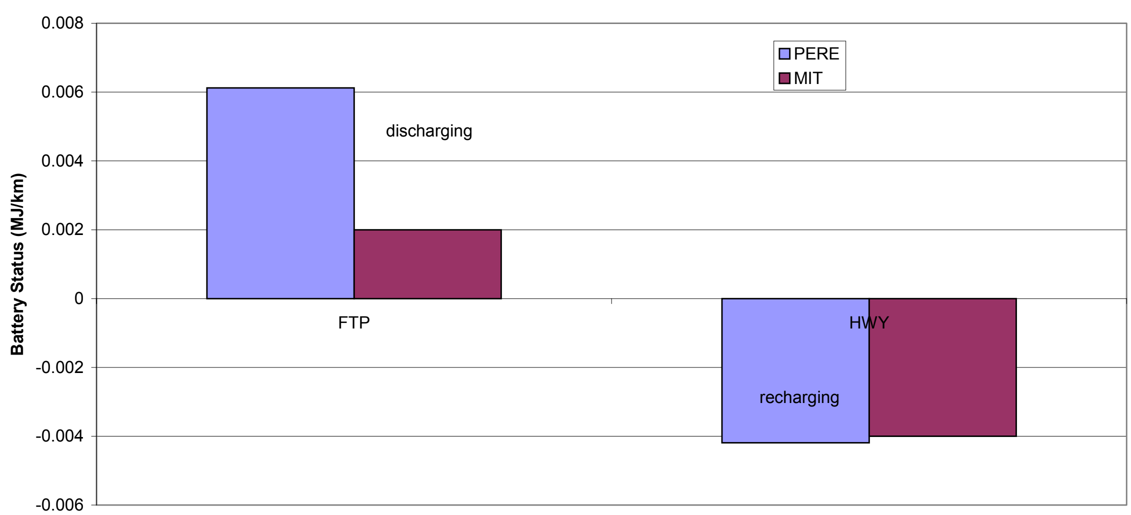


Figure 25. State of Battery compared to MIT study [2000].

The differences between the MIT and PERE models are listed below:
1- PERE does not include bag 1 cold start. The MIT report does not mention if cold start effects are included in their model.
2- MIT transmission model is unknown.
3- Hybrid threshold is determined differently
4- Battery recharge model probably is not identical.
5- PERE adds additional loss from front wheel drive regenerative braking only.
6- Engine friction model is slightly different.

The actual parameters that MOVES and PERE will employ will differ significantly from the MIT model vehicles.

Fuel Economy Validation Results
The initial validations are conducted in comparison to unadjusted EPA certification fuel economy numbers. The second set of validations are done on two hybrid vehicles tested at the EPA laboratories. It is difficult to conduct validation results with real hybrid vehicles, when the model is only an approximation of a hybrid vehicle. Actual hybrid vehicles have much more sophisticated control strategies, as well as efficiencies that depend on various operating conditions. However, the energy flows still follow a similar physical and logical progression. Given a power demand the fuel consumption rate should be similar if the loss terms are characterized sufficiently well.

The fuel economy validation is conducted on a number of vehicles (hybrid and non-hybrid), whose specifications could be found on either the manufacturer, or a trade magazine website (such as Car and Driver and Road and Track). All weights are test weights (curb weight plus 300lbs). The list of vehicles is on Table 18. The Honda Civic HX is the lean burn engine version of the Civic series. This report does not attempt to validate PERE to a larger variety of conventional vehicles (light duty vehicles and trucks), since that has already have been conducted successfully in the literature.

Table 18. Validation vehicles – all model years are 2004 unless otherwise specified.

Mfr	Model
Toyota	Camry
VW	Jetta gas
VW	Jetta Diesel
Honda	Civic DX
Honda	Civic HX
Honda	Civic Hybrid
Honda	Insight
Toyota	Prius '01
Toyota	Prius '04

The rated fuel economy numbers for production vehicles are adjusted based on real world driving correction factors: adjusted FTP = unadjusted FTP*0.9 and adjusted HWY = unadjusted HWY*0.78. PERE output should be compared with the dynamometer measured fuel economy, which are the unadjusted numbers, since PERE is modeling the physical loads on the vehicle as imposed by the dynamometer directly. The combined fuel economy (miles per gallon) is a weighted average 0.45 (FTP) and 0.55 (HWY), calculated in the following standard fashion [Schaefer, 1994]:

Equation 33 **Combined F.E. = 1/(0.55/FTP + 0.45/HWY).**

Figure 26, Figure 27, and Figure 28 show the validations for the vehicles (city, highway and combined respectively). The model performs especially well for conventional vehicles. There were several other conventional vehicles (light cars and trucks) validated, the Camry, Civic, and Jetta are the only ones shown. All of the conventional vehicles matched rated fuel economy to within 5% (on city and highway) with the exception of the Volkswagen Jetta (gasoline). There were also several conventional vehicles validated to dynamometer tests in previous studies, using a similar, but older version of PERE [Nam, 2003]. Although the Jetta (1.9L TDI) results were within 5%, the reason for the discrepancy with the Jetta (gas) is unknown. For a small fraction of vehicles, the fuel economy is not accurately captured by PERE. This is most likely due to many different factors unique to each make and model. It is also interesting to note that PERE slightly overestimates fuel economy from the city for many of the vehicles. This is not too surprising since cold start factors are not included.

The “advanced vehicle” validations begin with the Honda Civic HX, with the lean burn engine. The predictions for this vehicle is very good (within 5%). This suggests that the powertrain model is correctly modeled, based on the description of the lean burn engine by Ogawa et al. [2003]. All of the advanced vehicles are accurately capture by PERE for highway driving (within 5%). With the exception of the Toyota Prius, the city estimates are also all within 10%. PERE overpredicts the fuel economy on the Honda Civic Hybrid on the city cycle by around 9%. This is likely due to two reasons: PERE shuts the engine down during deceleration events and idle (running off only battery) but the Honda does not shut down all the time. This is more evident in the next section of this paper. Also, there is no cold start module in the present version of PERE. Also, the Honda engines may not shut off during short idles as PERE models it (see modal results below). Hybrids are especially expected to have significantly higher fuel consumption during the cold start first phase (bag 1) of the FTP. This is due to the fact that the engine has to stay on in order to ensure that the catalyst lights off in a timely fashion (after which it can start breaking down the criteria pollutants). Many conventional engines also run slightly rich during the start up period. Presently PERE shuts the engine down during decelerations and idle, which would tend to significantly under predict fuel consumption (over predict fuel economy) during start-up for hybrids, and to a lesser extent conventional engines. Modifying PERE so that the engine remains on for the first two minutes would be relatively simple. When this is done, PERE still underpredicts city fuel economy for the Prius by 12% and 18% for the 2001 and 2004 models respectively. This demonstrates how different the strategy of the Prius is from the rudimentary strategy employed in PERE. The Toyota Hybrid System (THS) is a series/parallel hybrid design. A powersplit device routes some of the power from the engine to recharge the batteries, thus the motor is used much more frequently than in parallel hybrid [Toyota, 2003] to supplement power. The hybrid system can be fine-tuned to give optimal fuel economy. The brake-by-wire system on the 2004 Prius also helps to regain maximal energy from regenerative braking. In this way, the Prius can and does get better fuel economy in the city than on the highway (a difficult condition for PERE to duplicate). Moreover, the specifications (efficiency improvements) for the Prius Atkinson cycle engine are unknown.

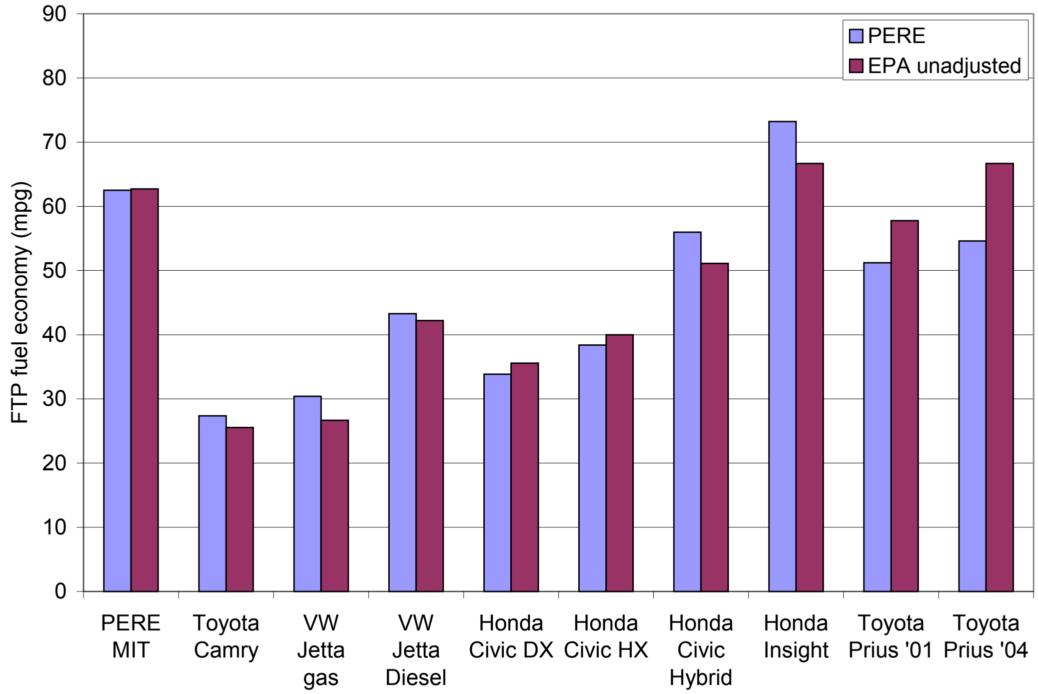


Figure 26. Fuel Economy validation for the FTP city (UDDS) compared to unadjusted EPA values.

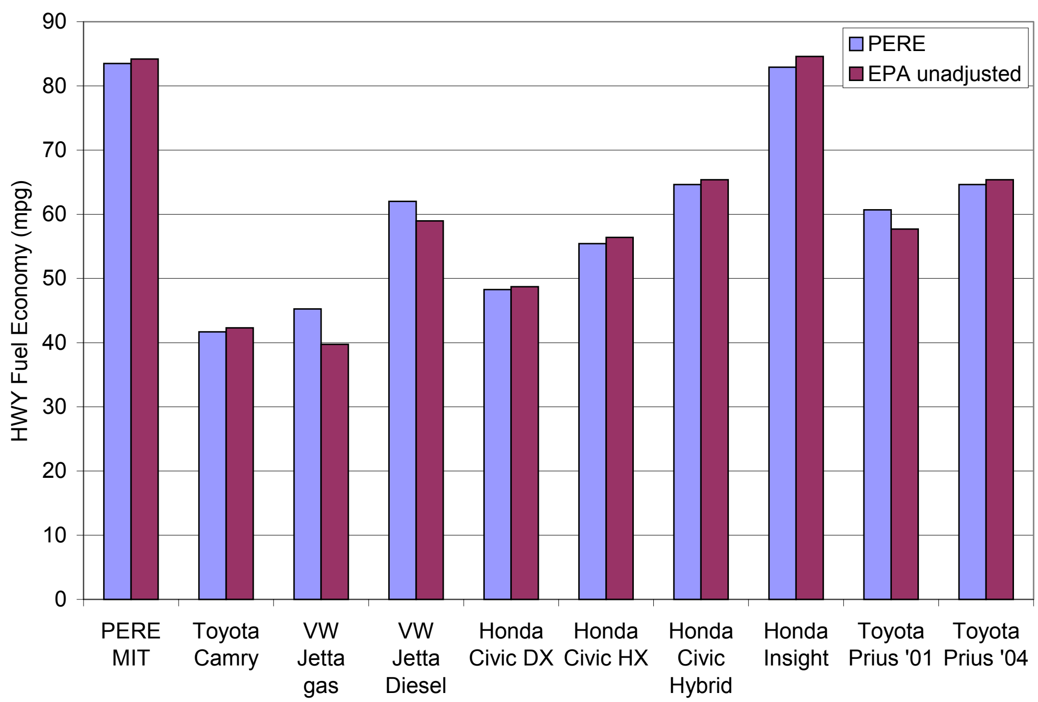


Figure 27. Fuel Economy validation for the FTP (HWY) compared to unadjusted EPA values.

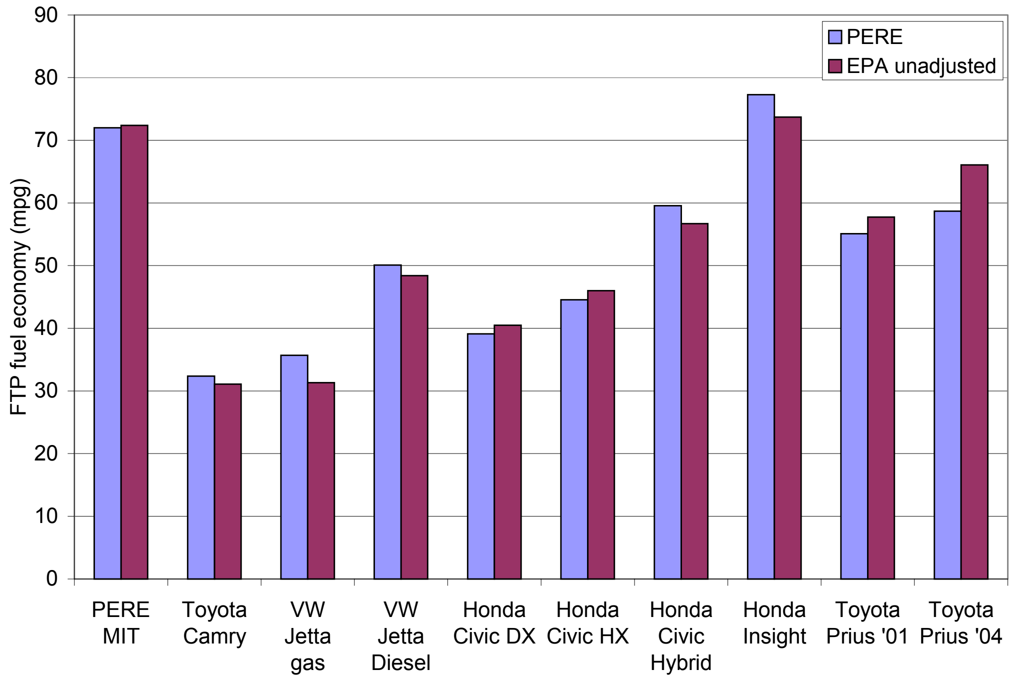


Figure 28. Fuel Economy validation for the FTP/HWY Combined compared to unadjusted EPA values.

It has been demonstrated that PERE can capture the fuel consumption behavior of gasoline, diesel, and "generic" parallel hybrid vehicles. The vehicle parameters required from the user are model year, weight, body type (resistive coefficients and size, which can be approximated), engine displacement, motor power, and fuel type. With the exception of body type, these are the primary "source" inputs for MOVES.

This simplification of hybrid control strategy modeling could result in under-predicting fuel economy. A more optimized control strategy would be to run the battery at low load and allow the engine to recharge the battery at higher loads, where the engine can operate more efficiently (as the Prius does). This was not done because it was difficult to put in spreadsheet format. If PERE is coded, then a more sophisticated control algorithm could be added.

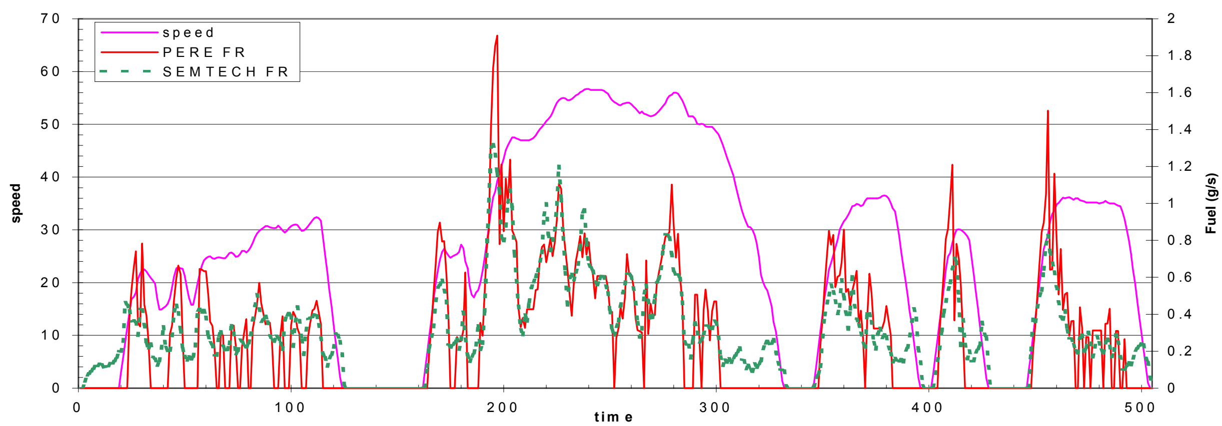
Modal Validation Results

PERE is based on a quasi-steady state powertrain model, designed to model "generic" vehicles (conventional and advanced). The hybrid vehicle modeling is especially simplified since it distributed energy loads using relatively straightforward algorithms. It can be somewhat misleading to compare to actual second-by-second data taken on a dynamometer. We should not be surprised if the results vary significantly. However, exploring these discrepancies are of interest.

Two in-use hybrid vehicles were tested at the National Vehicle and Fuel Emissions Laboratory in 2003 as part of an in-use compliance survey. The 2001 Honda Insight (9,400 miles) and 2001 Toyota Prius (58,100 miles) were tested on a chassis dynamometer on a 4 bag FTP test (no highway). Both vehicles were certified to federal ULEV standards. Some of the tests included bench modal data (second-by-second) and some included measurements from an portable on-board emissions analyzer. The SEMTECH-G manufactured by SENSORS Inc. was incorporated on the instrumented vehicle tests. The SEMTECH-G measures real-time concentrations of CO_2, CO, HC and NOx. It also measures the exhaust volume flow independently using a hot wire anemometer. Combining concentration and flow (after time alignment) gives second-by-second mass emissions measurement. The experiments and results are described in greater detail in [Nam et al., 2005]

Of the 12 tests performed on these two vehicles, 1 representative sample from each is compared to the model. The Prius data is dynamometer modal data, whereas the Insight only has on-board emissions data. The fuel consumption is calculated from mass CO_2 (and emissions) measured.

Figure 29 shows the bag 3 results of the Honda Insight. The hot wire anemometer was incapable of accurately measuring flow below 10cfm. Due to the small engine on the Insight, the flow can approach this value during low load operation. Therefore, the low flow bag 2 measurements were not used. Both the instantaneous and cumulative emissions are displayed in the figure for comparison. One can see from the figure that the measured fuel rate is higher in the peaks than PERE. However PERE shuts off the engine during deceleration and idle, thus giving lower fuel rate than measured during these modes of driving. The cumulative fuel rate plot shows that the overall fuel rate mode for PERE is quite accurate for this type of vehicle. Though not shown, the total fuel consumption between model and measurement compare within the bounds of the test to test variability.



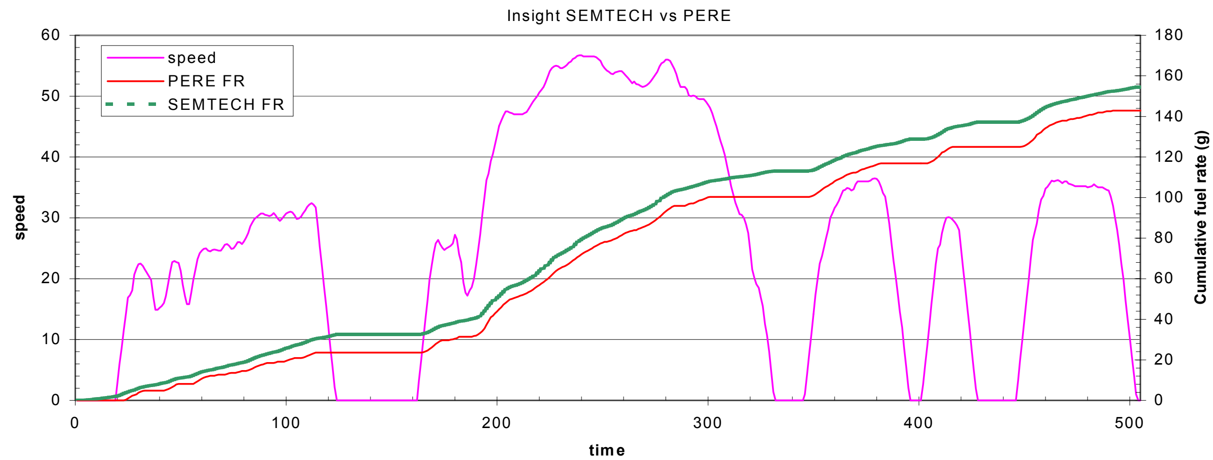
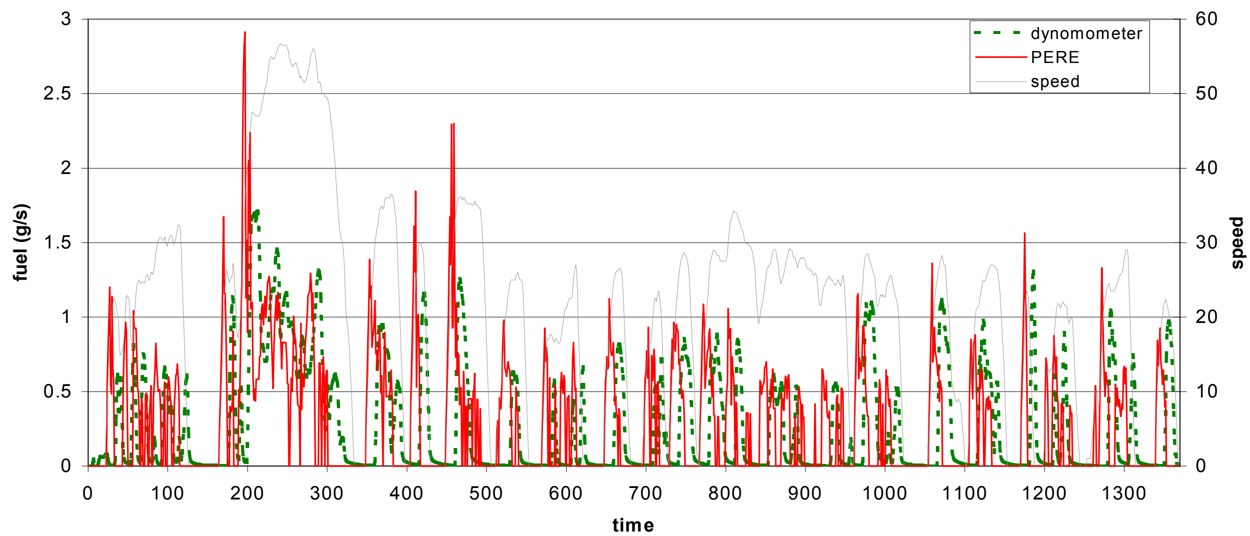


Figure 29. Bag 3 second by second fuel rate and cumulative fuel rate comparing PERE with SEMTECH on a 2001 Honda Insight.

Figure 30 shows the bag 3 and 4 results for the Toyota Prius. It appears at first glance that there may be a timing shift error. The effect is real. Because of the series-parallel nature of the Prius hybrid system, the engine does not deliver power until a few seconds after it is demanded. Thus the battery is "launching" this vehicle during a moderate load acceleration. The engine turns on later in order to recharge the recently depleted battery.



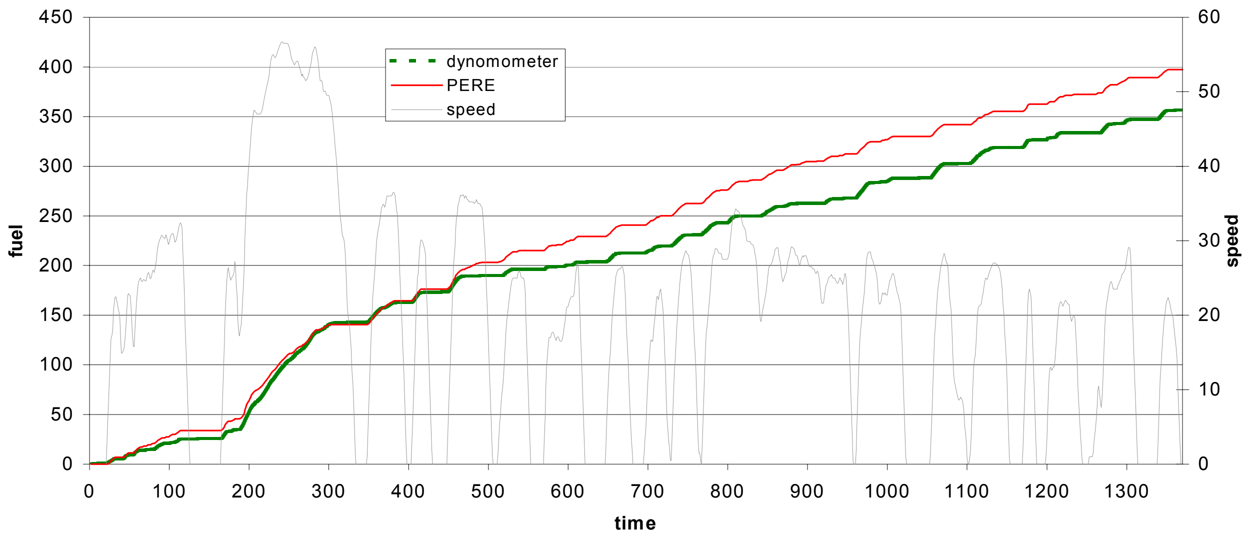


Figure 30. Bag 3 and 4 fuel rate and cumulative fuel consumption modeled and measured (on a chassis roll) for the 2001 Toyota Prius.

The difficulty with using the Prius data in MOVES is that it is not directly "load-following", i.e. the emissions event does not immediately follow the driving, which caused it. This "problem" may apply for all hybrid vehicles that have a series element (where the engine recharges the battery rather than drive the wheels). Since MOVES and PERE are modal or load-based models, it may be necessary to adjust the data so that they are more aligned with the models. If the timing on the Prius data were shifted 12 seconds to align with the speeds and accelerations, we would observe more of a load-following behavior. This was conducted on the FTP measurements, however, other driving traces may have different or no engine lag. The Prius data was shifted and then was binned by VSP (this binning procedure is described in greater detail in a later section and in other MOVES documents) in Figure 31. The straight binning (without time alignment for load correction – indicated by squares) gives poor results. This is because the VSP event coincident to the fuel consumption is non-causal. Instantaneous emissions rates clearly cannot be based on this method. However, after time shifting the data, the comparison between the model and the data is good. Any more rigorous comparison is beyond the abilities of PERE at this time. The difficulty of using this load correcting time alignment procedure is that the time shift will vary depending on a number of conditions: magnitude of acceleration, road grade, state of charge of battery, air conditioning, and others. Unfortunately, this poses potential problems when including the data from any hybrid vehicle with a series (power storage) element into the MOVES database.

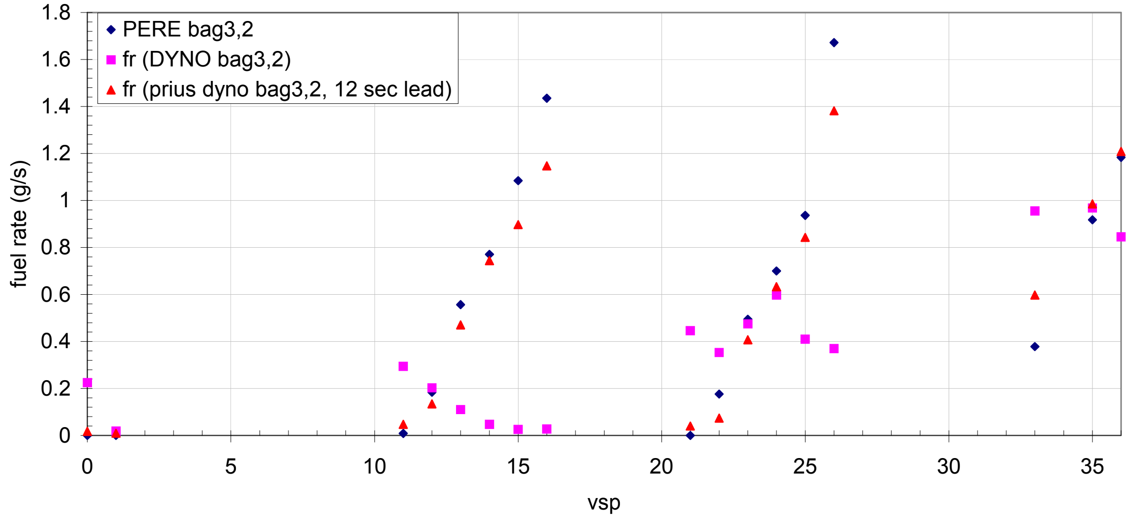


Figure 31. Prius fuel rate as a function of Vehicle Specific Power Bin with and without time alignment of the data.

Electric Vehicles

At this time, MOVES does not put strong emphasis on electric (only) vehicles (EV). EVs are modeled in PERE simply as hybrid vehicles with the internal combustion engine removed. Most of the major components are already included (battery, traction motor, regenerative braking, etc). In a future version, there may be further attempts to develop and validate this model.

Section VIII – Hydrogen Internal Combustion Engine Vehicles

As of the date of the publication of this report, hydrogen ICE vehicles are not yet in production. However, there are a number of prototypes constructed by various manufacturers, notably Ford and BMW. Due to the relative advanced stage of development of Ford's vehicle, we obtain PERE parameters primarily from a series of Ford SAE papers [Tang et al., 2002, Natkin et al., 2003].

The hydrogen vehicle is approximately 25% heavier than its conventional ICE counterpart [Natkin, 2004]. Much of this is due to the added weight of the hydrogen tanks and the added structures required for holding the tank securely. Fuel Cell vehicles have the same weight additions (see Table 19). The Ford vehicle also has hybrid electric elements. Moreover, this is a prototype vehicle, so weight reductions are inevitable if this vehicle were taken to full production.

The indicated efficiency of the (boosted) H_2 ICE is taken to be 45% (though it may well be somewhat higher). However, the specific power of the naturally aspirated engines are significantly lower than their gasoline counterparts. In order to achieve similar power performance it is necessary to boost the intake air. PERE assumes that the H_2 ICE is supercharged engine (like the Ford engine). Because of this, the friction is also increased. The friction is determined from the motoring torque curves in Natkin et al. [2003] and is shown in Figure 32. The friction losses from supercharging are significant.

In PERE, the transmission for the H2 ICE vehicle is assumed to the identical to conventional transmissions.

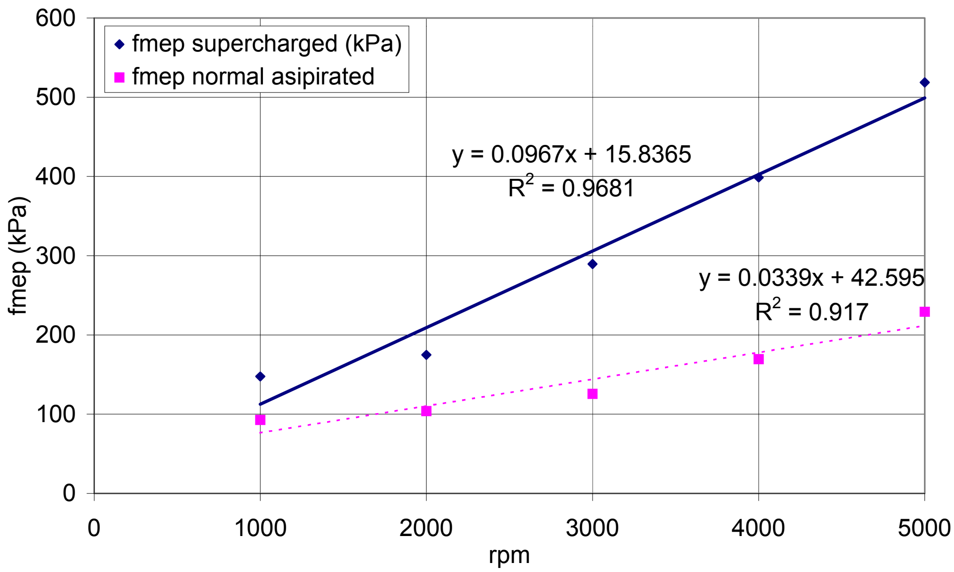


Figure 32. Friction as a function of engine speed for the Ford 2.3L supercharged and naturally aspirated H_2 ICE [Natkin et al., 2003].

Unfortunately, a validation could not be performed on the PERE H_2 ICE vehicle at this time since there is no comparison vehicle. This will be attempted in the future.

Section IX – Hydrogen Fuel Cell Vehicles

Fuel Cell and Hybrid Fuel Cell Vehicles

While, there are several types of fuel cells, the hydrogen Proton Exchange Membrane (PEM) fuel cell is the fuel cell of choice for vehicle traction applications. A fuel cell is a power source that runs on hydrogen to make current across a Proton Exchange Membrane. Each cell produces a small amount of current and (like a battery) these are stacked together to produce a relatively efficient power pack. There are other types of fuel cells, but PEM cells are the most commonly used for automotive applications today. Most prototype vehicles are equipped with Hydrogen PEM stack fuel cells. Hydrogen is the primary energy source, which produces energy when combined with oxygen. This reaction produces water, which is the only tailpipe emissions. Some fuel cell systems use a reformer to break hydrocarbons from liquid fuels into gaseous hydrogen, but this technology appears to be falling out of favor for light-duty vehicle use due to its size and relative inefficiency.

In PERE, fuel cell vehicles are modeled as "series hybrid" vehicles. This means that there is only one mechanical path. However, there is still a parallel electrical power path, in that either the battery or fuel cell can drive the motor. This model is similar to the one proposed by Weiss et al. [2000 & revised in 2003] (there is still a hybrid threshold required). The power flow chart is shown on Figure 33. It is essentially the same as the hybrid model above, but the engine is replaced with a fuel cell, and there is no transmission (only a single gear driven by the high speed motor). The fuel cell system efficiency (from the same reference) is shown on Figure 34. The fuel cell efficiency curve is a projection and reflects an optimistic estimate. The figure shows the efficiency points (scaled to a 72 kW cell) including a 7th order polynomial fit for PERE. The equation is shown below (calibrated to a 80kW stack):

Equation 34. $$\mathbf{FCeff = a_0 + a_1P_{frac} + a_2P_{frac}^2 + a_3P_{frac}^3 + a_4P_{frac}^4 + a_5P_{frac}^5 + a_6P_{frac}^6 + a_7P_{frac}^7}$$

Where

$P_{frac} = 80*P_d/P_{fc}$ (scaled)

P_{fc} is fuel cell power rating (kW)

P_d is power demand (kW)

$a_0 = 0.067284$

$a_1 = 0.166336$

$a_2 = -0.018361$

$a_3 = 9.6832e-4$

$a_4 = -2.7509e-5$

$a_5 = 4.2972e-7$

$a_6 = -3.4719e-9$

$a_7 = 1.1328e-11$

Again in this version of PERE, there is no cold start modeled, the effects of which may be significant. We also use the battery, similar to a parallel hybrid, to ‘launch’ the vehicle below the hybrid threshold (initial acceleration). This avoids the very low fuel cell system efficiencies at low load (see Figure 34). The model is similar in structure to the hybrid presented above. The “calibration” comparison to the MIT results is shown in Figure 35. The fuel economy figures are calculated in miles per gallon gasoline equivalent by converting the work equivalent hydrogen into gasoline mass (with their respective lower heating values). This vehicle is found to have a tank to wheel efficiency of 50% (this is also what Toyota claims for their fuel cell vehicle efficiency). The tank to wheel efficiency is simply defined here as the useful energy out/fuel energy in. The useful energy out is only the (positive) energy required for the car to follow the driving cycle. It would be slightly lower if the accessories were included.

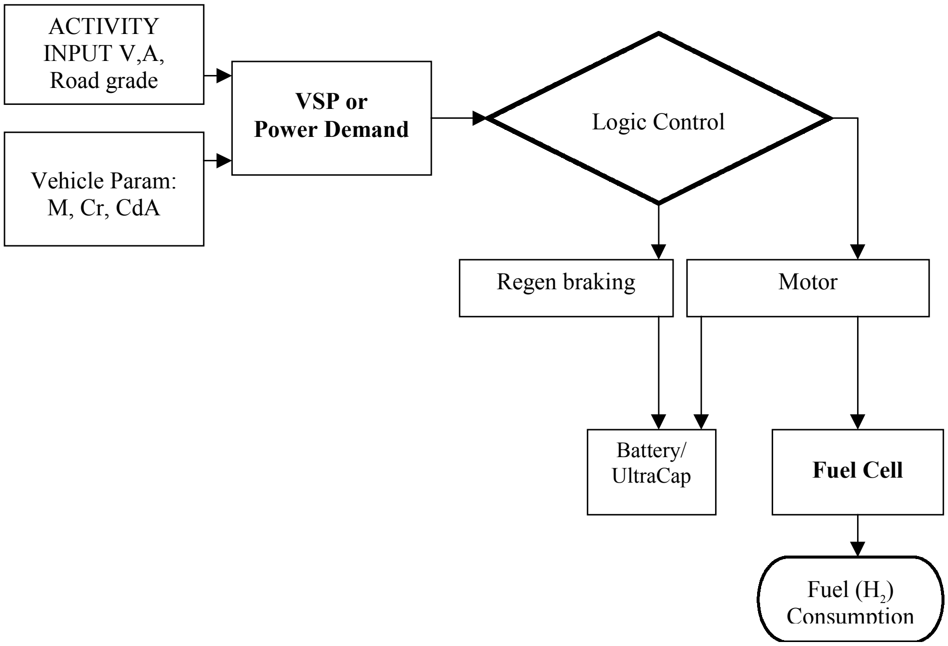


Figure 33. Power flow chart for the hydrogen hybrid fuel cell vehicle.

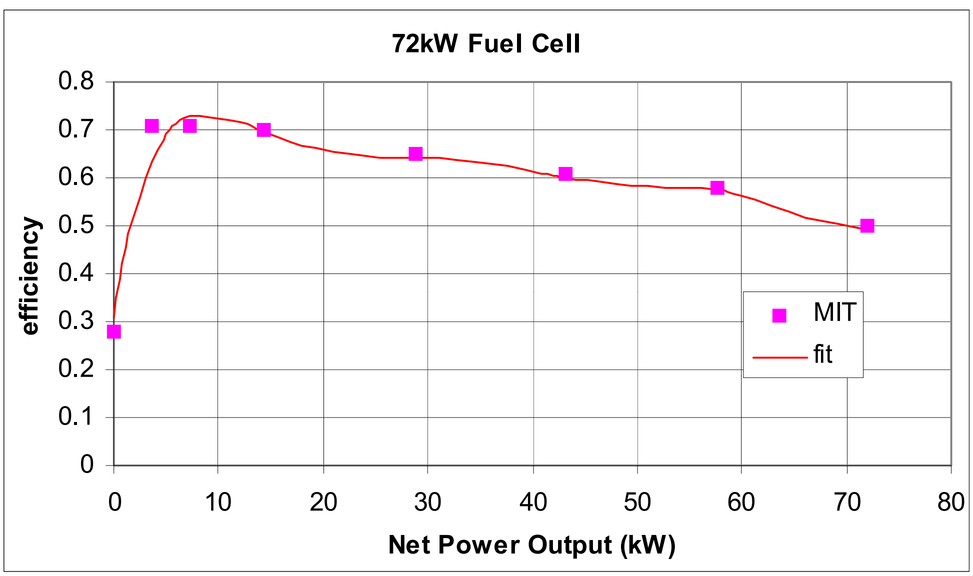


Figure 34. Fuel cell integrated system efficiency [Weiss et al, 2000, 2003] including a 7th order polynomial fit.

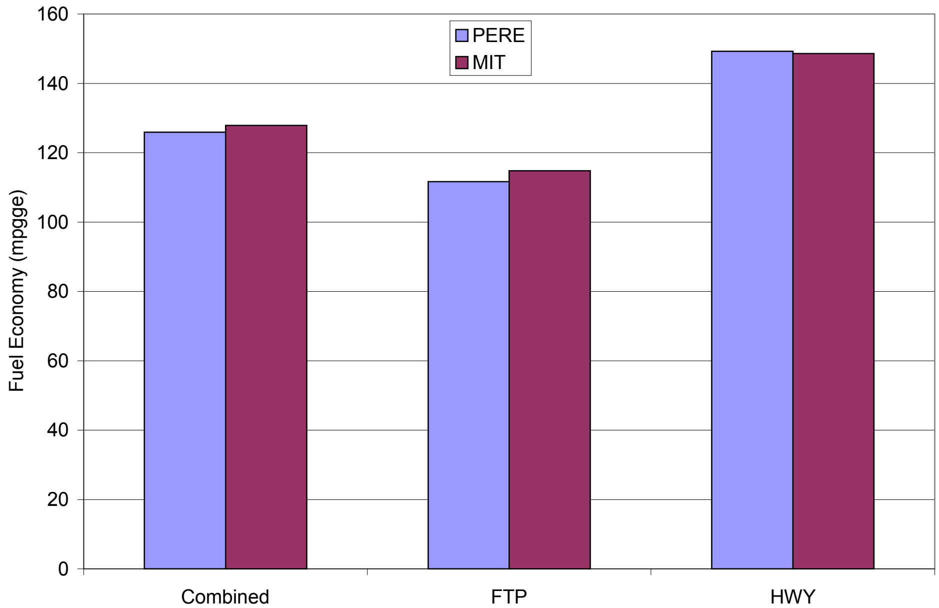


Figure 35. Fuel economy comparison between PERE and Weiss et al [2003]. The units are in miles per gallon gasoline equivalent.

Validation
It is difficult to conduct a validation of the fuel cell model since there isn't a tremendous amount of data available for the prototype vehicles that exist. Some specifications have been collected for a number of manufacturer prototypes. They are listed on Table 19 along with the conventional vehicle body on which the vehicle is built (the Daimler Chrysler F-Cell vehicle weight is unknown). The Honda FCX has a unique body, so it does not have a conventional vehicle equivalent (though it does have a similar electric vehicle). The average increase in weight of the first 3 vehicles compared to its equivalent conventional vehicle is 23% (interestingly this is consistent with the increase in the weight of the Ford H_2ICE). This increase may be less for larger (medium and heavy-duty) vehicles.

The Ford, DaimlerChrysler, and Honda vehicles all use the Ballard® Mark 902 fuel cell stack. The GM Hydrogen 3 and Toyota FCHV use different stacks. The former is not a hybrid. As the fuel cell energy densities increase in the future, the weight increase is expected to be reduced from the current estimate of 23%. The estimated system efficiency for the 80kW fuel cell is different from the one on Figure 34 (which can be used for future projections). The more conservative estimates are shown on Figure 36. This is used for the validation below.

Table 19. List of some fuel cell vehicles. (References are all from information sheets distributed during ride & drives by manufacturer representatives, or from website)

	model	Eng power (kW)	curb weight (kg)	Motor Power (kW)	FC power (kW)
Fuel Cell	Ford Focus		1600	65	80
	Toyota FCHV		1860	80	90
	GM Hydro-Gen3		1700	60	94
	DC FCELL		N/A	65	80
	Honda FCX		1684	60	80
Conventional	Ford Focus	97	1213		
	Toyota Highlander	119	1597		
	GM Opel Zafira	74	1393		
	DC Mercedes A-Class	76	1115		

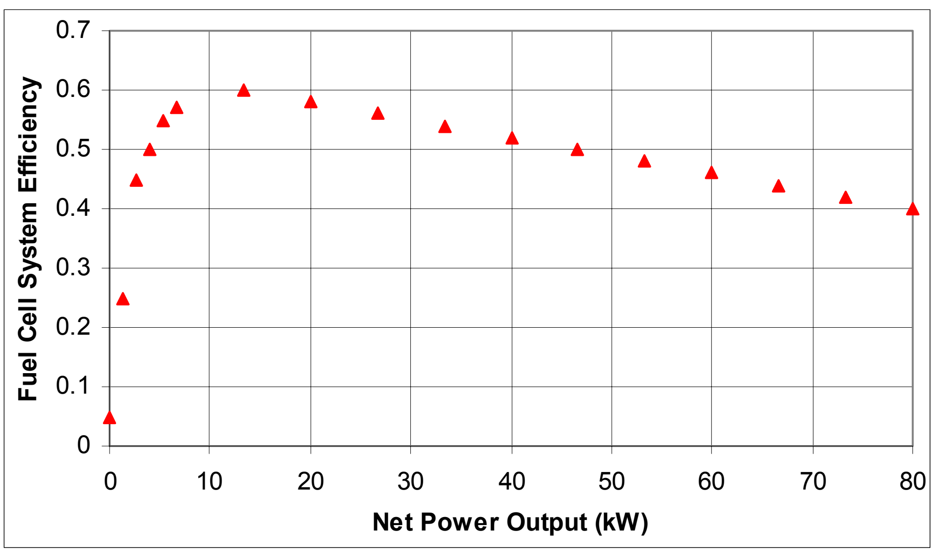


Figure 36. Fuel Cell System efficiency for an 80kW stack [Nelson, 2003].

Validation can only be conducted on the Honda FCX, since this is the only vehicle whose fuel economy equivalent numbers are known.

In order to model the Honda FCX, some of the energy storage parameters had to be adjusted since the vehicle is equipped with an ultracapacitor, rather than a battery and does not have a DC-DC converter. The motor system, and recharge efficiencies are higher. The Honda FCX has an adjusted EPA fuel economy rating of 51/46 miles per kilogram hydrogen (city/highway). A kg of hydrogen is roughly equal to a gallon of gasoline in energy. In comparison, PERE predictions are shown on Figure 37. The discrepancy is within 5%.

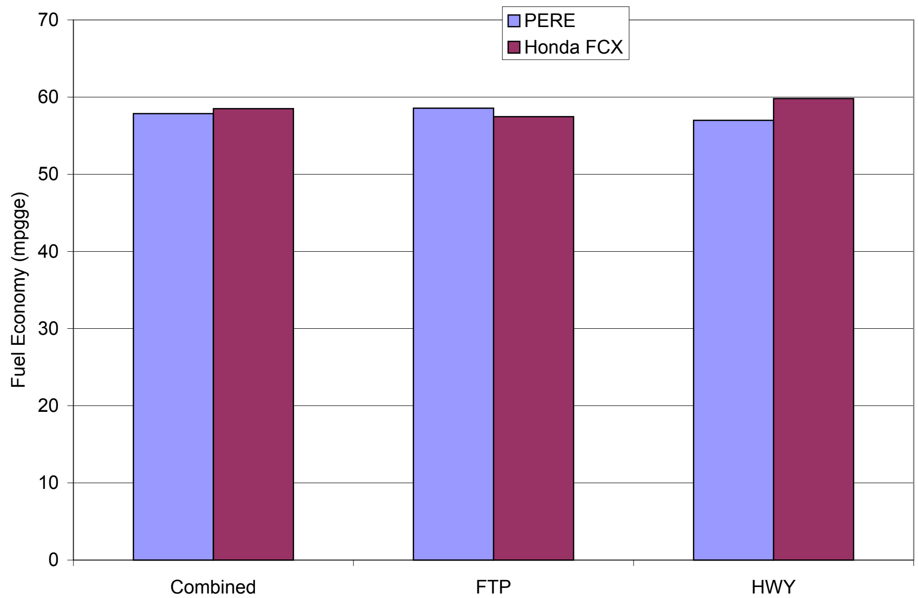


Figure 37. Fuel economy (in miles per gallon gasoline equivalent) for the Honda FCX.

Section X - Sensitivity

This section is devoted to a sensitivity analysis of PERE. The analysis determines which parameters are most important to characterize accurately. A static sensitivity is performed in relation to a change in city vs highway fuel consumption. It is necessary to do a separate analysis since the parameters affect the results from two different driving types in a disparate fashion. The simple static sensitivity test assumes a 10% variation in each parameter (though some of the parameters are correlated). The change in the fuel consumption result is ranked and compared. In order to perform a more complete uncertainty analysis, it is necessary to quantify the variation in the parameters. This is a potentially difficult undertaking, since some of the parameters are estimates (and not measurements). Also, it is quite probable that different alternate driving cycles will give different sensitivity results. Futhermore, some of the parameters may vary by significantly more than 10% (some by an order of magnitude). A more complete analysis may be performed in a future report.

Table 20 shows the sensitivity results, in % difference, for a typical modern passenger car modeled using PERE. To get the relative sensitivity, divide by 10. TRLHP is the Track Road Load Horsepower. It is a single quantity on which the (track) road load coefficients (A,B,C) are based in MOVES. PERE does not typically use TRLHP, but MOVES does. As expected, the parameters most affecting fuel consumption tend to be different for city vs highway driving. It is not surprising though, that the efficiency, weight, and engine displacement all dominate the top ranking. Fortunately, these quantities are typically known inputs for the model, or in the case of indicated efficiency, should not vary much. For those advanced engines where the efficiency is unknown, it clearly poses a problem for the model. The high ranking of the transmission efficiency might support the need for a speed or load based transmissions model. Aside from efficiency and N/v, however, PERE seems to be relatively insensitive to the other transmission parameters. However, it is important to make sure that the gear ratios are not systematically off from the actual gear ratios. It is interesting also to note that the model is quite insensitive the peak torque curves. It would be more crucial if high acceleration (full throttle) performance modeling was being performed. Since the activity in MOVES describes representative "real-world" driving, these driving modes are not stressed in the model, nor is it in PERE. Though it is not included the table, the model was not sensitive (<1%) to the rotational term in **Equation 1** (ε).

Table 20. Static sensitivity test for the PERE parameters used to model a typical conventional passenger car. Parameters are adjusted 10%

RANK	Parameter	Error City
1	P/T indicated eff (eta)	4.93
2	Engine Displ (L)	4.57
3	trans eff	4.46
4	Vehicle wgt (kg)	3.98
5	k0 (N indep friction kJ/l	3.72
6	N/v (rpm/mph)	3.31
7	TRLHP	1.95
8	Shift point 3-4	1.31
9	g/gtop 4	1.26
10	Shift point 2-3	1.24
11	Shift point 4-5	1.11
12	Cr0 (rolling resistance)	1.05
13	g/gtop 5	0.96
14	Cd (drag coeff)	0.93
15	A (frontal area m^2)	0.93
16	Nidle (rpm)	0.90
17	k1 (N dependent fric)	0.85
18	Pacc (accessory - kW)	0.52
19	g/gtop 1	0.38
20	g/gtop 2	0.22
21	g/gtop 3	0.16
22	Shift point 1-2 (mph)	0.04
23	torque curve up 10%	0.03

RANK	Parameter	Error Hwy
1	P/T indicated eff (eta)	5.92
2	trans eff	5.55
3	TRLHP	5.14
4	Engine Displ (L)	3.49
5	Cd (drag coeff)	3.11
6	A (frontal area m^2)	3.11
7	Vehicle wgt (kg)	3.01
8	k0 (N indep friction kJ/l	2.75
9	Nidle (rpm)	2.27
10	g/gtop 2	2.25
11	g/gtop 1	2.25
12	Shift point 2-3	2.22
13	Shift point 1-2 (mph)	2.21
14	N/v (rpm/mph)	1.89
15	g/gtop 3	1.88
16	Cr0 (rolling resistance)	1.88
17	Shift point 4-5	1.46
18	Shift point 3-4	0.91
19	k1 (N dependent fric)	0.74
20	g/gtop 4	0.67
21	Pacc (accessory - kW)	0.40
22	g/gtop 5	0.23
23	torque curve up 10%	0.00

The sensitivity results for a "typical" full hybrid are presented in Table 21. Here the motor is power split such that it can bypass the transmission. This improves the efficiency of the vehicle. This is a fictional hybrid vehicle designed using PERE only, but assuming the same body and overall power as the previous vehicle. With the exception of efficiency, the transmission parameters have been omitted from this analysis. The hybrid threshold was adjusted to maintain the pseudo-charge sustaining strategy with each run. It is clear from this table that the engine parameters are still more important for fuel economy modeling compared to the electric parameters. Since the small engine is operating at much higher loads, the model is much more sensitive to changes than the conventional vehicle. Again, since the vehicle is able to meet the drive cycle (sometimes with the help of the motor), the battery limit parameters are not important for the model.

A sensitivity analysis is not conducted on the fuel cell hybrid, however, it is obvious from the above two studies that the fuel cell system efficiency curve is the most important parameter. The fuel cell peak power also being of moderate importance.

Table 21. Static sensitivity test for the PERE parameters used to model a fictional full electric hybrid passenger car.

RANK	Parameter	Error City
1	P/T indicated eff (eta)	8.11
2	Vehicle wgt (kg)	7.18
3	Engine Displ (L)	5.42
4	overall motor efficiency	3.38
5	Cr0 (rolling resistance)	3.32
6	Cd (drag coeff)	3.12
7	A (frontal area m^2)	3.12
8	torq curve up 10%	2.96
9	Regen Brake Eff	1.99
10	FWD power frac	1.99
11	Motor peak power (kW)	1.08
12	fmep0 (N indep friction kJ	1.01
13	Pacc (accessory - kW)	0.73
14	fmep1 (N dependent fric)	0.39

RANK	Parameter	Error Hwy
1	P/T indicated eff (eta)	8.52
2	Cd (drag coeff)	3.47
3	A (frontal area m^2)	3.47
4	Vehicle wgt (kg)	2.35
5	Cr0 (rolling resistance)	2.21
6	Engine Displ (L)	2.08
7	fmep0 (N indep friction kJ	1.81
8	overall motor efficiency	0.61
9	fmep1 (N dependent fric)	0.49
10	Pacc (accessory - kW)	0.44
11	Regen Brake Eff	0.30
12	FWD power frac	0.30
13	torq curve up 10%	0.24
14	Motor peak power (kW)	0.05

Road Load and Track Coefficients

Finally a brief comparison was run between using the road load coefficients Cr, Cd, A_F (frontal area), etc, vs using the track (or target) coast-down coefficients (A, B, C). The fuel consumption was compared using both techniques for the conventional and hybrid vehicles. This is essentially a sensitivity analysis of the B term. It was found that the choice of methods made little difference for city fuel consumption results, but were significant for highway. For the city, the fuel consumption was underpredicted by about 2% (or less). For the highway cycle, the fuel consumption was underpredicted by 5% for conventional vehicles and 8% for hybrids on average. This implies that at high speed, the load curves from the estimated coefficients tend to be lower than those measured on the track. This could be due to the lack of a first order speed dependent term (B in force units), which tends to be quite small (or sometimes negative) on track tests. Additionally, there could be higher order rolling resistance terms [Gillespie, 1992]. It may be further aggravated in hybrids because of motor drag and their lack of a true neutral gear.

With production vehicles, the issue of choosing the method of calculating road loads is simple, since most of the track A, B, C coefficients are publicly available. Unfortunately, this cannot be done with vehicles that have not yet been produced. It is impractical to combine **Equation 1** with Equation 3 in order to back-derive physical quantities from track

A, B, and C coefficients, since the values are determined from empirical fits to coastdown data. Thus, it is necessary to have a methodology for estimating the effect of the B coefficients for future vehicles. Two methods of estimating its magnitude are compared here. The first method involves fitting to the known parameters.

The following coast-down force equations are based on Equation 3:

Equation 35 $$F_{RL} = mgC_R + 0.5\rho C_D A_F v^2$$

Equation 36 $$F_T = A + Bv + Cv^2$$

Where the subscripts "RL" and "T" stand for "Road load" and "Track" respectively. The equations are compared for the validation vehicles in the study. Then a B "correction" term is added into Equation 35 and empirically fitted to Equation 36.

Equation 37 $$F_{RL} = mgC_R + Bv + 0.5\rho C_D A_F v^2$$

The second method is to use the higher order rolling resistance term from Gillespie [1992] modified slightly here:

Equation 38 $$C_R = C_{R0}*(1+v/44.7)$$

Where C_{R0} is the base rolling resistance (typically ~.009). An estimate of B is

Equation 39 $$B' = MgC_{R0}/44.7 \quad \text{[N/mps]}$$

The speed dependent parameter in the denominator is not precise. It is estimated to double the rolling resistance factor at 100 mph. In other references, the speed dependent rolling resistance is embedded in the C parameter [Petrushov, 1997]. Unfortunately, this approach does not take other vehicle (non-tire) frictional factors into account.

Table 22 shows the fitted B coefficients along with their 1 standard deviation uncertainties. The third column displays the percentage underprediction in fuel consumption from the use of the road load **Equation 1** compared to using the track Equation 3. The last column (B') shows the estimated value of B using Gillespie's method. First it is clear that the B term has a significant effect on fuel consumption. In all but the Camry, the difference is over 5% on the highway cycle. Comparing the fitted B with the estimated B', there is some agreement, but the trends break down for certain vehicles (e.g. Camry). Unfortunately, the tires (and rolling resistance) used for testing these vehicles were unknown. Moreover, the lack of correlation with some vehicles only supports the notion that other factors are inherent to the B term, such as rotating friction in bearings, transmission, and final drive, tire windage, motor drag etc. Because of the motor drag (and other hybrid differences), the official fuel economy numbers for certain hybrids may not necessarily match what one would achieve in the real world [Rechtin, 2003].

Table 22. Fitted B coefficients and 1 standard deviation. CV = Conventional Vehicle, HEV = Hybrid Electric Vehicle.

	B fit (N/mps)	sigma	HWY Fuel Cons Effect %	B'
Civic DX	2.01	0.19	8.60	2.44
Camry	0.67	0.07	1.48	3.09
Jetta	2.85	0.11	8.87	2.89
Civic hybrid	2.32	0.11	8.18	2.37
Insight	1.37	0.15	6.27	1.73
Prius 01	3.11	0.35	10.46	2.13
avg CV	**1.84**	**0.13**	**6.31**	**2.81**
avgHEV	**2.27**	**0.20**	**8.30**	**2.08**
avg ttl	**2.06**	**0.16**	**7.31**	**2.44**

This subject clearly requires further research. However, if a vehicle's A, B, C track coefficients are unknown, PERE will assume Gillespie's (or Petrushov's) methodology, until an improved methodology is presented.

The subjects of dynamometer, track, and road load coefficients are complex. This brief study, only begins to scratch at the surface. A more full discussion of this topic is beyond the scope of this report.

Cold Start

Modern vehicles are capable of eliminating nearly all of the criteria pollutants during hot running periods. However, during cold starts, the engine and the catalytic converter are cold, thus preventing the breakdown of emissions. For the vehicles meeting the cleanest standards, the cold start emissions account for a large fraction of the vehicle's operating emissions. Though it is significant for the sensitivity of PERE, the impact on fuel consumption is not as dramatic as it would be for criteria pollutants.

In order for criteria pollutants to be treated in the catalytic converter, the catalyst must first obtain a sufficiently high temperature to break down the pollutants. Before this can happen, the engine must warm up to a point where its exhaust is hot enough to heat the catalyst. When this temperature is reached, this condition is referred to as "light-off". In order for the catalyst to light-off, the heat from the engine exhaust must heat the catalyst, therefore the engine must be running until this point. At this time, PERE cannot predict at what time light-off will occur, this must be done empirically. Typical gasoline vehicles light-off within 2 minutes into the first portion (bag 1) of the FTP. There are advanced technologies, which help minimize this light-off time, but they usually come at a cost. For example, the engine may run rich briefly, or electrical power may be needed to pre-heat the catalyst. In either of these cases, energy is being consumed. Additionally, the engine runs less efficiently while it is cold.

Kim and Lee [2004] measured the cold vs hot fuel economy for the (pre-2004) Toyota Prius. Their report indicates that the city cold start fuel economy was 12.4% worse than a hot start test. However, their test vehicle may have had a different calibration than one in the United States. In the EPA testing, the cold start fuel consumption (defined here as bag 1 – bag 3) was 53g for the 1.5L Prius. This is an average of 5 tests and corresponds to an increase of 27% compared to bag 3. For the 1.0L Insight, the cold start fuel consumption was 20g. This is an average of 7 tests and corresponds to an increase of 12% over bag 3. The fuel consumption at the beginning of bag 3, did not appear unusually high for either vehicle. It is interesting to compare these figures to those obtained from the displacement dependent cold start factors described in Koupal et al., [2004]. The MOVES equation describing gasoline cold start factors is included below and the table compares the results.

Equation 40. Cold-Start Fuel Consumption (grams) = 31.867 + 10.863 * Displacement (L)

Table 23. Cold start rates compared for the Insight and Prius.

	Measured	MOVES model
1.0L Insight	20g	43g
1.5L Prius	53g	48g

The model is adequate for the Prius but overpredicts for the Insight. Due to the large uncertainties involved in cold start modeling, it is decided to use the conventional vehicle cold start factors in MOVES for hybrids until the model can be improved in the future.

An attempt is also made to model cold start modally in PERE. The fuel strategy on a 1.1L full hybrid vehicle (used in the sensitivity study) is adjusted so that the engine remains on for the first 2 minutes of the test. This time was chosen based on estimating the cold duration on numerous conventional vehicles. The battery power is set to zero for the duration, so that vehicle is acting much like a conventional vehicle, with the exception that the regenerative brakes are still active. The fuel never goes to zero in this scenario, but is minimal at engine idle. The additional energy (fuel) consumed in this period is assumed to balance any alternative technologies such as an electrically heated catalyst.

The fuel consumption from this modified hybrid vehicle increases by 11% on bag 1 vs bag 3 (10% lower fuel economy). By subtracting the hot running bag 3 from the bag 1, this corresponds to a cold start factor of 22.3 grams of fuel per start, which is curiously consistent with the measured Honda Insight results (which has a similar engine displacement). The model would naturally underpredict the Prius cold start however. This start factor will depend on the type of hybrid, size of engine, and other vehicle parameters. This sub-model is not yet used due to its developmental nature.

There is as yet, no cold start model for fuel cell vehicles in PERE.

Section XI - Application to MOVES

PERE has been developed fundamentally to produce energy consumption and emission rates for MOVES for vehicle categories (known as "source bins" in MOVES terminology) where in-use data is insufficient to populate these rates directly. Thus PERE will produce these rates for most of the advanced technology vehicles (the focus of this report) and some of the conventional vehicles where data is lacking ("holes"). The potential for PERE to compliment MOVES in this way is discussed in the USEPA report [**2002 -MOVES GHG emission analysis report**].

Presently PERE is a standalone spreadsheet model capable of modeling on a finer scale than MOVES requires. PERE also has the potential to model a wider assortment of technologies than MOVES currently has. This flexibility allows for more source bins to be filled in the future as the need arises. However, the development time for each technology type using PERE is significant, so MOVES will model some advanced technologies directly using energy and emissions rates measured from other studies such as GREET (e.g. alternative fueled vehicles).

The process of how PERE results could be integrated into MOVES is the subject of this section.

Hole Filling Using PERE

The original intent of MOVES is to derive all energy and emissions rates from existing data. However, it is not possible to obtain data from all mobile sources in the vast fleet. The MSOD (the database upon which MOVES obtains its emissions rates – [Koupal et al., 2004]) currently accounts for approximately 95% of the on-road fleet. Therefore, it is necessary to come up with a method of "filling these holes". It was demonstrated in a series of reports, that PERE provides a fairly robust fuel consumption model for conventional vehicles [Nam, 2003; Nam & Giannelli, 2004]. The model also serves the purpose of allowing for the study of physical effects on vehicle fuel consumption and emissions by isolating certain factors. As it currently stands, PERE will be employed to fill some of the more "common" source bin holes in MOVES. "Common" is defined as the source bins that are most frequently represented in the fleet. The following table describes a parsed source bin ID along with the assumed values used by PERE for conventional vehicles. This bin happens to be the highest ranked hole, i.e. of all the holes, this bin represents the largest fraction of the fleet (~0.78%).

Table 24: Source bin nomenclature using the highest ranked hole in MOVES: Heavy duty diesel (70,000lb).

Source Bin: 10201000450990800000

1:	leading digit (no physical meaning)
02:	Fuel type: diesel
01:	Engine Technology: Conventional Internal Combustion
00:	Regulatory Class: N/A
04:	Model Year: 1991-1999 (assumed 1995)

5099: Engine Size: >5.0L (assumed 6.9L – see Nam, Giannelli, 2004)
0800: Weight: 60,000-80,000 lbs (Heavy Duty – assumed 70,000)
00: Trailing zeroes (no physical meaning)

Some of the common source bin holes (ranked top 45 or so) are listed in the table below (grouped by model year). Also included are the central values used in PERE for model year, weight and engine size. With these filled by PERE, > 97% of the fleet will be included.

Table 25. List of the top source bins sorted by fuel type and model year.

rank	Source bin ID	fuel ID	Fuel	MY ID	MY	engsz	Wt ID
15	1010100032025006000	1	Gas	3	86-90	2025	60
32	1010100044050002500	1	Gas	4	91-99	4050	25
37	1010100044050007000	1	Gas	4	91-99	4050	70
36	1010100044050009000	1	Gas	4	91-99	4050	90
4	1010100045099016000	1	Gas	4	91-99	5099	160
7	1010100045099019500	1	Gas	4	91-99	5099	195
18	1010100045099026000	1	Gas	4	91-99	5099	260
38	1010100045099033000	1	Gas	4	91-99	5099	330
35	1010100053035014000	1	Gas	5	01-10	3035	140
43	1010100055099002500	1	Gas	5	01-10	5099	25
44	1010100055099003000	1	Gas	5	01-10	5099	30
42	1010100055099003500	1	Gas	5	01-10	5099	35
40	1010100055099004000	1	Gas	5	01-10	5099	40
5	1010100055099016000	1	Gas	5	01-10	5099	160
20	1010100055099019500	1	Gas	5	01-10	5099	195
14	1010100055099026000	1	Gas	5	01-10	5099	260
25	1010100055099033000	1	Gas	5	01-10	5099	330
65	1020100043540007000	2	Diesel	4	91-99	3540	70
16	1020100043540014000	2	Diesel	4	91-99	3540	140
62	1020100043540019500	2	Diesel	4	91-99	3540	195
19	1020100045099003500	2	Diesel	4	91-99	5099	35
21	1020100045099004000	2	Diesel	4	91-99	5099	40
11	1020100045099004500	2	Diesel	4	91-99	5099	45
12	1020100045099005000	2	Diesel	4	91-99	5099	50
13	1020100045099006000	2	Diesel	4	91-99	5099	60
10	1020100045099016000	2	Diesel	4	91-99	5099	160
1	1020100045099080000	2	Diesel	4	91-99	5099	800
2	1020100045099100000	2	Diesel	4	91-99	5099	1000
27	1020100055099007000	2	Diesel	5	<90	5099	70
26	1020100055099008000	2	Diesel	5	<90	5099	80
17	1020100055099009000	2	Diesel	5	<90	5099	90
29	1020100055099010000	2	Diesel	5	<90	5099	100
8	1020100055099014000	2	Diesel	5	<90	5099	140
9	1020100055099016000	2	Diesel	5	<90	5099	160
3	1020100055099080000	2	Diesel	5	<90	5099	800
6	1020100055099100000	2	Diesel	5	<90	5099	1000
28	1020100045099130000	2	Diesel	4	91-99	5099	1300

1010100012025003500	**1**	**Gas**	**1**	**<80**	**2025**	**35**
1010100013035003500	**1**	**Gas**	**1**	**<80**	**3035**	**35**
1010100013540003500	**1**	**Gas**	**1**	**<80**	**3540**	**35**
1010100013540004000	**1**	**Gas**	**1**	**<80**	**3540**	**40**
1010100013540004500	**1**	**Gas**	**1**	**<80**	**3540**	**45**
1010100014050004500	**1**	**Gas**	**1**	**<80**	**4050**	**45**
1010100015099006000	**1**	**Gas**	**1**	**<80**	**5099**	**60**
1010100020020002000	**1**	**Gas**	**2**	**81-85**	**20**	**20**
1010100022025004500	**1**	**Gas**	**2**	**81-85**	**2025**	**45**
1010100023035004500	**1**	**Gas**	**2**	**81-85**	**3035**	**45**
1010100024050003000	**1**	**Gas**	**2**	**81-85**	**4050**	**30**
1010100024050005000	**1**	**Gas**	**2**	**81-85**	**4050**	**50**
1010100024050006000	**1**	**Gas**	**2**	**81-85**	**4050**	**60**
1010100025099002500	**1**	**Gas**	**2**	**81-85**	**5099**	**25**
1010100030020002000	**1**	**Gas**	**3**	**86-90**	**20**	**20**
1010100032530002500	**1**	**Gas**	**3**	**86-90**	**2530**	**25**
1010100033540003000	**1**	**Gas**	**3**	**86-90**	**3540**	**30**
1010100034050006000	**1**	**Gas**	**3**	**86-90**	**4050**	**60**
1010100035099007000	**1**	**Gas**	**3**	**86-90**	**5099**	**70**
1010100043035002500	**1**	**Gas**	**4**	**91-00**	**3035**	**25**
1010100043540002500	**1**	**Gas**	**4**	**91-00**	**3540**	**25**
1010100043540003500	**1**	**Gas**	**4**	**91-00**	**3540**	**35**
1010100044050002000	**1**	**Gas**	**4**	**91-00**	**4050**	**20**
1010100045099003000	**1**	**Gas**	**4**	**91-00**	**5099**	**30**
1010100995099004500	**1**	**Gas**	**4**	**91-00**	**5099**	**45**
1010100045099009000	**1**	**Gas**	**4**	**91-00**	**5099**	**90**
1010100052530004000	**1**	**Gas**	**5**	**01-10**	**2530**	**40**
1010100053540004000	**1**	**Gas**	**5**	**01-10**	**3540**	**40**
1010100984050005000	**1**	**Gas**	**5**	**01-10**	**4050**	**50**
1010100985099004500	**1**	**Gas**	**5**	**01-10**	**5099**	**45**

The parameter selection for PERE is a somewhat involved process. The decision tree is outlined below. The parent variables (underlined) are above the dependent variables. The parent variables define the decision source (tree) of the parameters. The dependent variables are the parameters that are determined from the parent variable. Some dependent parameters vary with multiple parents (e.g. engine friction). A few other parameters are fixed.

SOURCE BIN

Weight
- LD/MD/HD
- Road load: C_r, C_d, A_F, or A, B, C
- Enrichment Threshold: FRth

Model Year
- Engine Friction: k0, k1
- peak bmep fit
- Enrichment Threshold: FRth

Engine Size
- Peak torq fit
- Enrichment Threshold: FRth

Gas/Diesel
- Ind Efficiency: η
- Engine Friction: k0, k1
- peak bmep fit
- fuel: LHV, ρ_f

LD/MD/HD/Motorcycle
- Driving cycle, Transmission: N/v, shift points, gear ratio, manual/auto, accessory: Pacc

FIXED PARAMETERS Auto transmission efficiency (by gear), manual transmissions efficiency, density of air, fuel enrichment slope (gasoline only), inertial rotational mass term.

The determination of general vehicle weight classification is the very first step in the parsing process. Since MOVES does not have explicit splits for light, medium and heavy duty, PERE assumes that the light to medium split occurs at weighclassID=80 (7,000-8,000 lbs), where above this weight bin is medium duty (single unit delivery trucks, buses, etc). The medium to heavy duty split occurs at weightclassID=400 (33,000-40,000 lbs). This in turn defines whether the transmission will be a 5 speed automatic (LD), 6 speed automatic (MD), 12 speed manual (HD), or 5 spd manual (motorcycle). Motorcycles have their own weight and engine size categories as well as a different accessory loading (0.25 vs 0.75kW).

The weight determines the road-load coefficients for the vehicles directly. For model years 2000 to present, A, B, and C track coefficients are provided by manufacturers. Older vehicles only have the single Tractive Road Load Horsepower (TRLHP) figure. The TRLHP for a typical light-duty vehicle (and truck) in the source bin can be approximated from weight using the relation in the MOVES Documentation [2004]. For heavy duty vehicles, the coefficients are determined from the equations presented earlier in this report.

The weight class also defines the driving cycles that PERE will run. Before model year 1999, weight also is a term in the enrichment equation. After 1999, the vehicles are assumed to go into enrichment so rarely that it is ignored by PERE.

The model year also affects the performance of the engine, both in terms of friction and peak bmep (which mainly influences power downshifting). Finally, the engine/fuel type determines, the efficiency and friction characteristics of the engine. All of these relationships can be found in this report. Finally, the model year also defines the fuel parameters since gasoline and diesel have different physical properties.

The following table shows the result parameters from the truck source bin described above.

Table 26. Parameters for the PERE 70,000lb heavy duty truck.

Vehicle	
Model Year	1995
Vehicle wgt (kg)	31752
Cr0 (rolling resistance)	N/A
Cd (drag coeff)	N/A
A (frontal area m^2)	N/A
A (N)	2098.81
B (N/mps)	0.000
C (N/mps^2)	4.2268
Pacc (accessory - kW)	0.75
Engine	
Engine Displ (L)	6.9
k0 (N indep friction kJ/Lrev)	0.0605
k1 (N dependent fric)	0.00333
P/T indicated eff (eta)	0.48
Transmission	
N/v (rpm/mph)	26.7
Nidle (rpm)	700
trans eff	0.95
Shift point 1-2 (mph)	2.48
Shift point 2-3	4.75
Shift point 3-4	7.75
Shift point 4-5	13.5
Shift point 5-6	17.5
Shift point 6-7	23
Shift point 7-8	34
Shift point 8-9	50
Shift point 9-10	56
Shift point 10-11	57
Shift point 11-12	64
Shift point 12-13	64
g/gtop 1	13
g/gtop 2	9.4
g/gtop 3	6.9
g/gtop 4	5
g/gtop 5	3.7
g/gtop 6	2.7
g/gtop 7	2
g/gtop 8	1.5
g/gtop 9	1.2
g/gtop 10	1.1
g/gtop 11	0.93
g/gtop 12	0.86
g/gtop 13	0.86
Fuel	
LHV (kJ/g)	43.2
density gas (kg/L)	0.8114

The motorcycle engine size and weight classifications fall under discrete bins as well. The divisions are marked in the following figure and are based on the fleet mix.

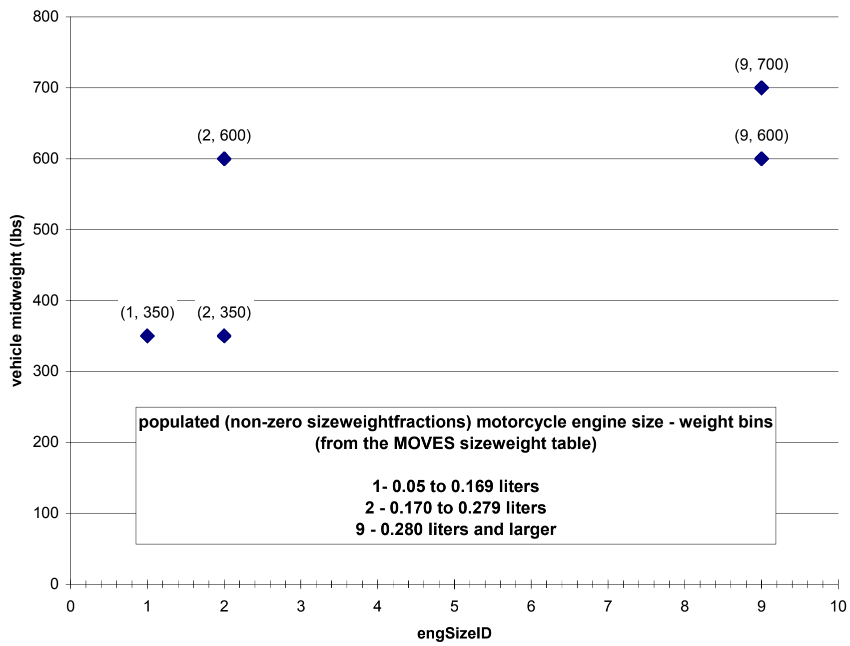


Figure 38. Engine size-weight categorization of motorcycles in MOVES.

Driving Cycles and Running PERE

The output to PERE is second-by-second energy consumption, therefore the next step is to input the driving cycles. The driving cycles input into PERE help determine the VSP "binned" energy consumption rates for MOVES.

It is important to capture a representative sampling of real-world driving. For PERE, the driving cycle input is dependent on the weight of the vehicle. 14 driving cycles for light duty applications have been selected for this purpose from MOVES. These cycles represent a broad spectrum of driving, from very low to very high speeds. The cycles and their average speeds are shown on the following table. When merged, the cycles run 6,981 seconds. It is also important to artificially set the accelerations to zero in between the cycles since they do not all start (and end) at rest. It is likely that similar binned rates could be obtained from a smaller sample of driving cycles, thus making the model easier to manage.

Table 27. The 14 MOVES light-duty driving cycles used as input to PERE.

CYCLE	avg spd
FWYHI1	63.2
FWYAC	59.7
FWYD	52.9
FWYE	30.5
FWYF	18.6
FWYG	13.1
RAMP	n/a
ARTAB	24.8
ARTCD	19.2
ARTEF	11.6
NYCC	7.1
FWYHI2	68.2
FWYHI3	76.0
LOWSPEED1	2.5

For medium and heavy-duty applications, the driving cycles where drawn from the medium and heavy-duty schedules used to populate the MOVES default database [Beardsley et al., 2004]. The medium duty drive cycles are listed in the table below and totaled 6,050 seconds.

Table 28. MOVES medium-duty driving cycles used as input to PERE.

cycle ID	Name	avg spd
201	MD 5mph Non-Freeway	1.8
202	MD 10mph Non-Freeway	10.5
203	MD 15mph Non-Freeway	15.6
204	MD 20mph Non-Freeway	20.4
205	MD 25mph Non-Freeway	24.4
206	MD 30mph Non-Freeway	30.8
251	MD 30mph Freeway	37.4
252	MD 40mph Freeway	45.3
253	MD 50mph Freeway	55.5
254	MD 60mph Freeway	60.1

The heavy-duty cycles (in the table below) are also a subset from Beardsley et al. [2004]. Together it totals 28,313 seconds of driving weighted mainly with freeway driving. The length of this file makes PERE cumbersome, and in the future, the number of cycles should be reduced.

Table 29. MOVES heavy-duty driving cycles used as input to PERE.

ID	Name	avg spd
301	HD 5mph Non-Freeway	1.2
302	HD 10mph Non-Freeway	10.8
303	HD 15mph Non-Freeway	15.2
304	HD 20mph Non-Freeway	19.8
305	HD 25mph Non-Freeway	24.9
306	HD 30mph Non-Freeway	30.8
351	HD 30mph Freeway	34.9
352	HD 40mph Freeway	46.9
353	HD 50mph Freeway	54.3
354	HD 60mph Freeway	59.5

The motorcycles are run only on the FTP and HWY driving cycles. The small engine scooters, are run only on the reduced speed FTP scooter cycle.

With the driving cycles entered, the binner macro (on the spreadsheet) is run in order to get the VSP binned fuel consumption rates, which are in turn converted to MOVES energy rates.

The holes in MOVES not filled by PERE are then filled using interpolation and extrapolation techniques described in the next section and in Koupal et al. [2004]. Finally after the process of interpolation and extrapolation is complete, there were a few bins remaining, which PERE filled. These are listed in the following table:

Table 30. "Leftover" holes after interpolation/extrapolation.

1020100010020002500
1020100012025003500
1020100020020002500
1020100020020003000
1020100022025003000
1020100030020002500
1020100030020003000
1020100032025003000
1020100054050002000
1020100054050002500
1020100054050003000

Finally, the motorcycle source bins are in the table below and includes model years into the future (the rates are unchanged).

Table 31. Motorcycle source bins filled by PERE.

1010110010001000500'
1010110020001000500'
1010110030001000500'
1010110040001000500'
1010110050001000500'
1010110060001000500'

1010110070001000500'
1010110010002000500'
1010110020002000500'
1010110030002000500'
1010110040002000500'
1010110050002000500'
1010110060002000500'
1010110070002000500'
1010110010002000700'
1010110020002000700'
1010110030002000700'
1010110040002000700'
1010110050002000700'
1010110060002000700'
1010110070002000700'
1010110010009000700'
1010110020009000700'
1010110030009000700'
1010110040009000700'
1010110050009000700'
1010110060009000700'
1010110070009000700'
1010110010009000900'
1010110020009000900'
1010110030009000900'
1010110040009000900'
1010110050009000900'
1010110060009000900'
1010110070009000900'

Filling Advanced Technology Using PERE

Step 1: Choose the Source Bin

The first step is to determine what kind of vehicle PERE will model. In MOVES, these are "source bins" defined by vehicle characteristics which differentiate energy consumption and emissions. For energy consumption, source bins have 5 dimensions: fuel type, engine technology, model year group, engine size, and loaded weight [**Koupal et al., 2004**]. The list of the combined fuel types and engine technologies to be included in MOVES are in Table 32. PERE is used to fill a majority of these technologies. The categories not filled by PERE are marked with an asterix (*). In these cases, energy rates are derived from already-published analyses.

Table 32. Examples of advanced technologies, as defined by fuel types and engine technologies, to be included in MOVES. "C" is conventional, "A" is advanced, and "IC" is internal combustion. This list is not final.

- **Gasoline conventional (IC)**
- **Gasoline Advanced IC**

- **Gasoline Hybrid -CIC Moderate**
- **Gasoline Hybrid -CIC Full**
- **Gasoline Hybrid -AIC Moderate**
- **Gasoline Hybrid -AIC Full**
- **Diesel Fuel conventional (IC)**
- **Diesel Fuel Advanced IC**
- **Diesel Fuel Hybrid -CIC Moderate**
- **Diesel Fuel Hybrid -CIC Full**
- **Diesel Fuel Hybrid -AIC Moderate**
- **Diesel Fuel Hybrid -AIC Full**
- **Compressed Natural Gas (CNG) conventional (IC) ***
- **Liquid Propane Gas (LPG) conventional (IC) ***
- **Ethanol (E85 or E95) conventional (IC) ***
- **Methanol (M85 or M95) conventional (IC) ***
- **Gaseous Hydrogen Advanced IC**
- **Hydrogen -Fuel Cell**
- **Hydrogen Hybrid -Fuel Cell**
- **Electricity electric only**

Not all of the engine technologies are represented fully across the other 5 dimensions, for example, it is highly unlikely that long haul trucks will be hybridized in the future, due to the relatively minor effect it has on highway fuel economy compared to the already efficient diesels. However, hybridization is quite promising for urban buses. Due to the sheer number of tests required, simplifications are necessary. Examples of these are described in greater detail below.

Step 2: Define the Vehicle Specifications
Based on the engine technology source bin, the vehicle parameters such as weight, engine size, vehicle shape etc, can be defined. The weight and engine size values are simply the central value of the bin. Thus PERE effectively models an "average" vehicle in the source bin. For the source bins on the "edge" of the matrix (<2000 lbs; >130,000lbs; <2.0L; and >5.0L), their results will be extrapolated - see Step 6, or average values are determined from the fleet mix data.

Since engine size, has limited meaning for hybrids and no meaning for fuel cells and electric vehicles, it is necessary to define a power surrogate for engine size. This has already been mentioned in this report [Chon & Heywood, 2000]. Though it is not perfectly equivalent, we will take the convention that motor power + engine peak power = total peak power. The fuel cell definition of power is simpler since there is only one mechanical power device (motor) and the battery and fuel cell should be sized optimally within the motor limitations.

Because the source bins do not have a dimension for body type, the road load coefficients are estimated based on the weight. Lighter light duty weights tend to be (compact) passenger cars, then midsize cars, luxury, compact pickups, SUVs, minivans on up to medium-duty trucks. Advanced technology heavy-duty trucks are not included in MOVES. The variation in body types will certainly lead to variation (or uncertainty) in the emission rates.

Step 3: Run PERE on the FTP if Light duty fuel economy required.
For the light-duty vehicles, once the vehicle specifications have been finalized, the next step is to run the model over the FTP driving cycle (city and highway). This is to determine the fuel economy of the vehicle. If fuel economy is not required, this step can be skipped.

Step 4: Define the Driving Cycles
The output to PERE is second-by-second energy consumption, therefore the next step is to input the driving cycles. The driving cycles input into PERE help determine the "binned" energy consumption rates for MOVES (see Figure 1). Henceforth "binned" refers to operating mode bin and not source bin, which has already been defined.

The binned energy consumption and emissions rates are dependent on the driving cycle input. It is important to capture a representative sampling of real-world driving. Three driving cycles were chosen for this purpose: FTP (urban), FTP (highway), and LA92. The latter is a self-weighted cycle based on more recent real-world driving, and includes harder accelerations and higher speeds. For hybrids, the state-of-charge was maintained over the total of the three cycles. For conventional vehicles, more driving cycles are used to get a better representation within a VSP bin.

For medium-duty applications, 3 urban bus plus the UDDS cycles were employed for a total of 3,260 seconds. The following table lists them with their average speeds.

Table 33. Medium duty drive cycles for advanced technologies.

Name	avg spd
CBD	12.7
NYBUS	3.7
NYCCT	8.8
UDDS	18.8

Step 5: Run PERE and Bin Output
After PERE is run, the rates must be binned into the respective operating modes. These are the 17 VSP and speed operating mode bins as defined in **Koupal** [2003]. The binning procedure is straightforward and can be programmed in any script.

Figure 39 shows a sample of predicted fuel consumption as a function of VSP for a fictional hybrid passenger car. This is an indication of how an output from PERE would be translated into an emissions rate in MOVES. The uncertainty bars are from PERE generated variations within a bin, and do not adequately reflect the true uncertainty of the emission rates, which will be added in future versions of PERE.

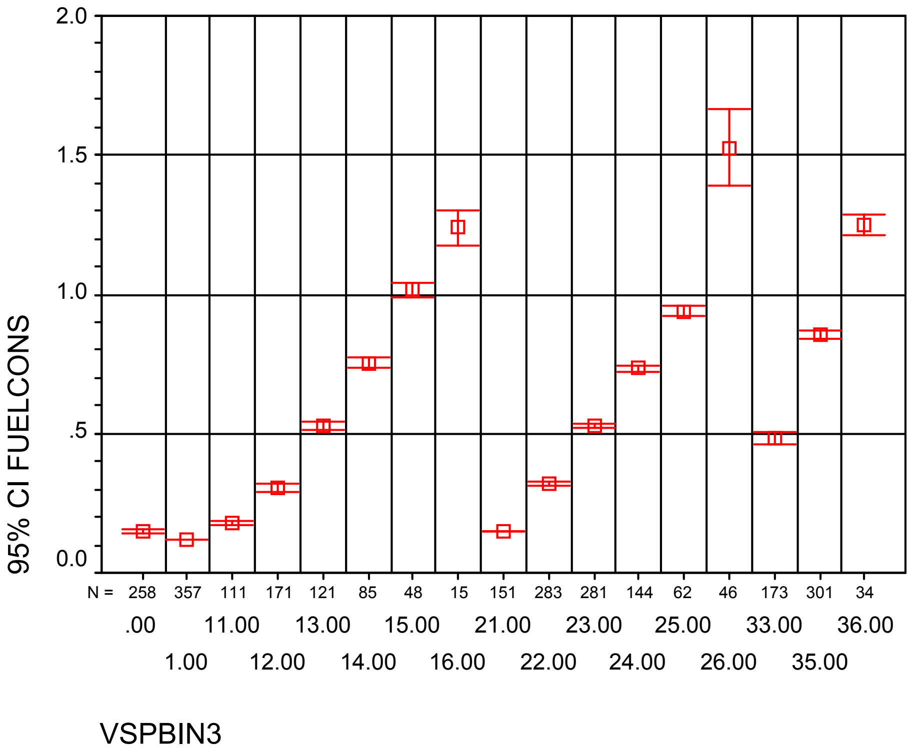


Figure 39. Hybrid fuel consumption as a function of VSP bin.

Step 6: Repeat for the Rest of the Source Bins
Rather than run PERE for every operating mode bin within every advanced technology source bin (there are hundreds), it is more efficient to extrapolate from one bin to another along a dimension. A dimension could be weight or power (engine displacement), so the rates could be interpolated between the endpoints (e.g. lightest and heaviest weight class. To demonstrate the validity of the linear extrapolation technique, PERE was run on a conventional gasoline powered passenger car for a variety of weights and engine displacements. The driving cycles employed were the city and highway portions of the FTP. The base car is a 3500lb, 2.5L passenger car. For the different engine sizes, the VSP values are identical, however, this is not true along the weight axis. Here the second-by-second fuel consumption as well as the VSP values are binned.

Figure 40 shows the fuel consumption rates as a function of VSP bin. Each color (shade) represents a separate engine displacement, as labeled along the right. It is evident that given the minimum and maximum points, that the other could be linearly interpolated.

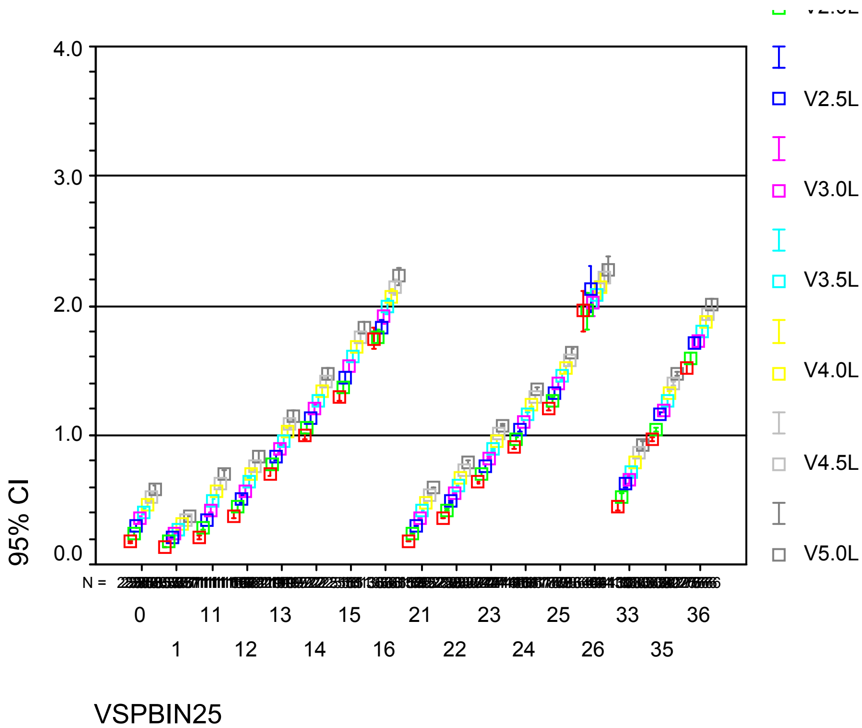


Figure 40. Fuel Rate (g/s) as a function of VSP bin for engine sizes 1.5 – 5.0L.

Figure 41 shows the same relationship but with vehicle weight varied instead of engine displacement. Despite the curvature between the VSP bins, within a bin, an interpolation is quite reasonable.

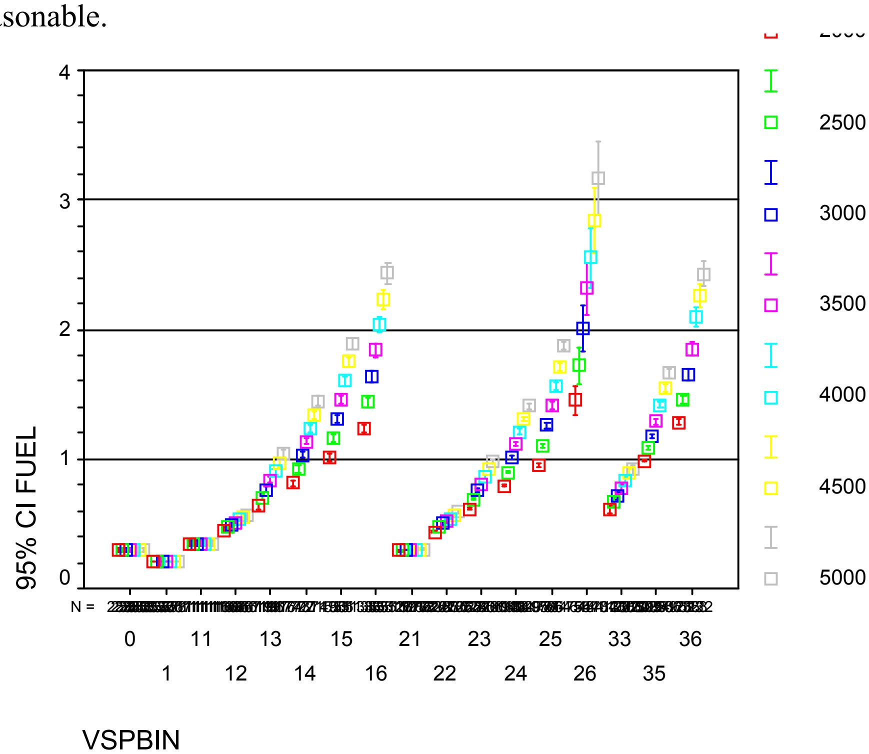


Figure 41. Fuel Rate (g/s) as a function of VSP bin for weights ranging from 2000 – 5000 lbs.

Given the rates in any two cells near the edge of the matrix, the rest of the emission rates can be interpolated or extrapolated. Figure 42 shows a matrix and defines an example bin filling strategy. The red bins are possible engine size (power) and weight combinations. The green bins are filled using PERE and the black arrows represent the direction of interpolation or extrapolation. The rest of the bins can be filled interpolating from the edge bins. In this case, PERE needs to be run a minimal of four times. Note that each of the bins below has 17 operating mode bins to fill.

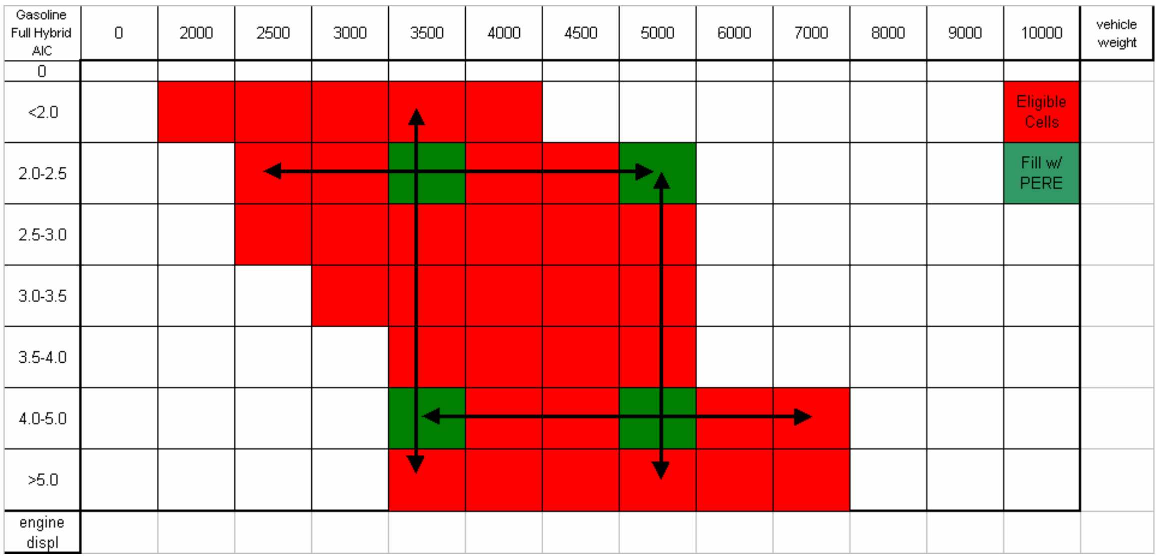


Figure 42. A sample source bin matrix hole filling strategy.

When choosing bins to fill, the absolute edge bins (<2000 lbs; >130,000lbs; <2.0L; and >5.0L) should not be run since the masses and or engine sizes are not well defined. This is the reason for choosing the "outer" bins in Figure 42, but not the "edge" bins.

If it is easier to fill each bin with PERE, rather than interpolating, this can also be done.

Step 7: Insert Estimated Cold Start Energy/Emission Factors
As discussed before, cold start factors will be approximated using the MOVES approach. The rates are dependent on engine displacement. A more detailed model is a subject of future study.

Advanced Technology Hole Filling Results
While the method of filling holes (step 6 above) is still being considered, due to time constraints, it was decided to take an even simpler approach for this first version of MOVES. For each of the advanced technology categories in Table 10, a representative vehicle was modeled using PERE. The rates from these vehicles were then ratioed to their corresponding conventional vehicle rates (gasoline equivalent). These ratios were then used to populate all of the advanced technology source bins using a simple computer program.

Due to its relative frequency in the fleet, the "representative" vehicle was chosen to lie in a MOVES source bin with weight ranging from 3,500 – 4,000 lbs (test weight), and engine size ranging 3.0 – 3.5 Liters (automatic transmission). This corresponds to a PERE conventional vehicle of 3,750 lbs, 3.25L, and 155kW (peak power). It also corresponds to a 1.7L engine full hybrid with a 72kW motor, or a 2.8L engine moderate hybrid with a 22kW motor (gasoline and diesel).

The ratio of energy (or fuel consumption) for some of the advanced technologies to conventional is shown in the series of figures. Note that it is possible for the ratios to exceed 1 in some operating mode bins. This is due to the fact that the hybrids are heavier vehicles, driven with smaller engines, yet still follow the same driving trace. Thus, it is quite possible for the fuel consumption to be higher than the conventional vehicle during certain modes of driving. However, over a given driving cycle, the fuel consumption is significantly lower. In most driving cycles, the time spent in modes where the ratio is > 1 is usually quite small (high acceleration events). These results support the well known fact that the advantage of hybrids is best seen in stop-and-go driving.

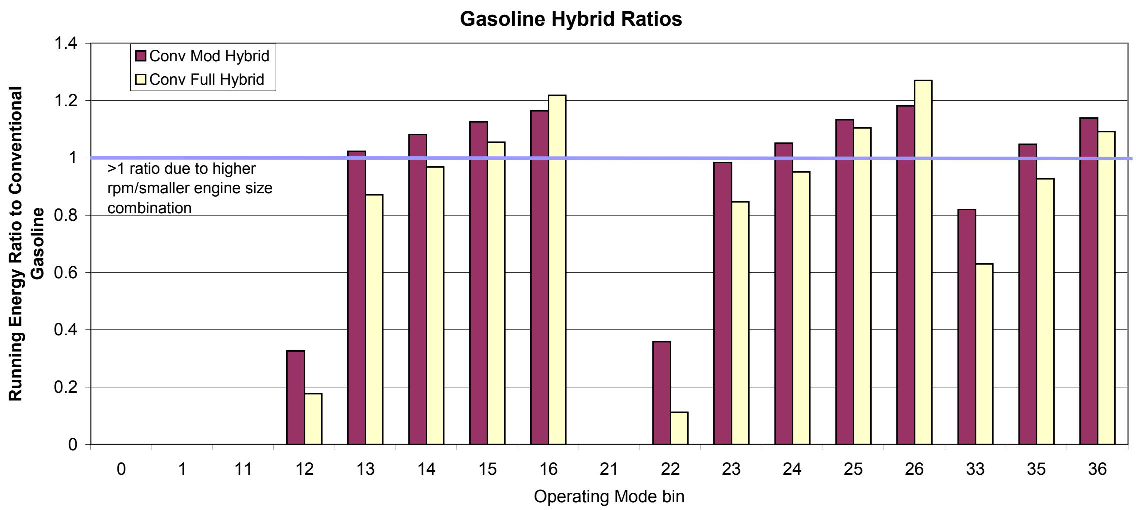


Figure 43. The ratio of energy (or fuel consumption) for gasoline moderate and full hybrid vehicles to conventional by operating mode.

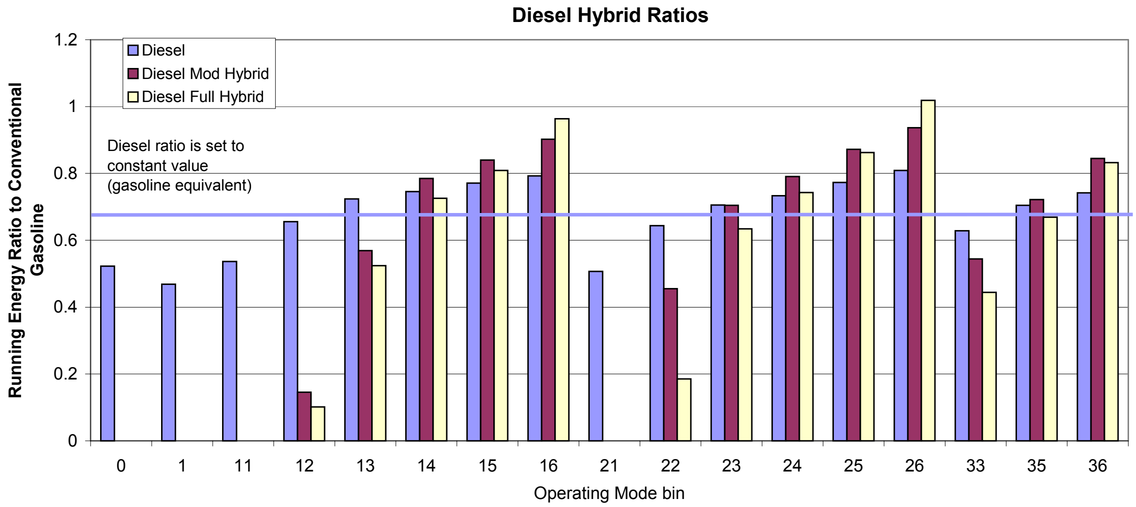


Figure 44. The ratio of fuel consumption for diesel, moderate, and full hybrid vehicles with respect to gasoline conventional by operating mode (using gasoline equivalent numbers).

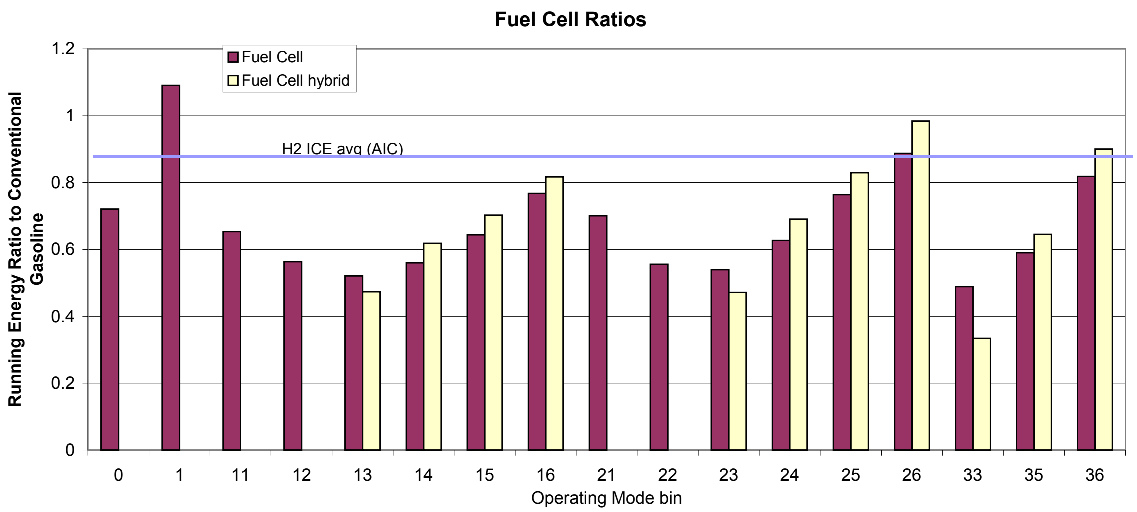


Figure 45. The ratio of energy for H_2 fuel cell, H_2 hybrid fuel cell, and H_2 ICE vehicles with respect to gasoline conventional (using gasoline equivalent numbers).

The ratios are simplified to a single constant value (for all operating modes) when the ratio of energy consumption is less dependent on the type of driving (mainly for non-hybrids). These ratios are shown as bars on the above figures. The ratio of H_2 ICE is nearly identical to that of Advanced ICE (~0.9).

This method for determining MOVES rates for hybrids is clearly superior to having a single fixed fuel economy ratio (over an FTP cycle for example) as other studies have proposed. This is due to the fact that hybrid fuel economy depends in large part on the type of driving, idle time, decelerations etc., which are washed out in a single cycle average number. However, aside from being more complex, the methodology also has limitations, which are presently explored.

The ratios are based on a single (though common) representative vehicle class. We might expect vehicles of different weight but same power-to-weight ratio to give similar rates. However, vehicles with lower power-to-weight ratios would (for example) require more assistance from the engine, both during launch and assist (power boost). This would necessarily change the shape of the curves. An example of this is shown in Figure 46. The varying power-to-weight ratios having different shapes. The effect is mainly pronounced in the "2" bins, i.e. bins 12, and 22. These correspond to moderate acceleration from low to medium speed (launch).

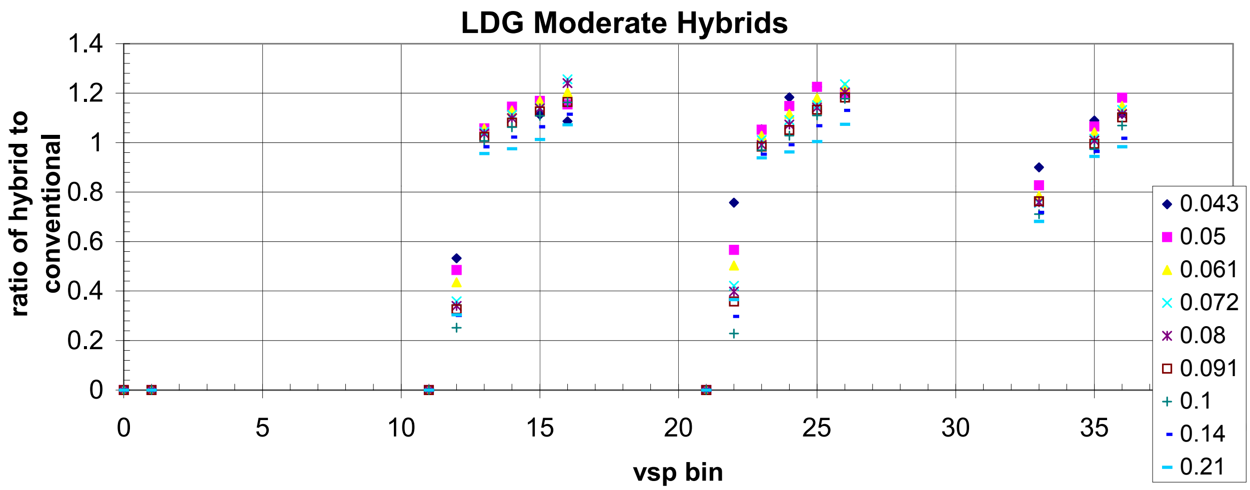


Figure 46. Fuel consumption ratios for a series of power-to-weight ratio (kW/kg) light-duty gasoline hybrid vehicles.

The shape difference is even more pronounced in medium-duty vehicles where the power-to-weight ratio can fall below 0.01. This is shown on Figure 47 for a series of medium-duty diesel hybrid vehicles. Note that the fuel consumption drops at higher loads for the underpowered vehicles. This is due to the fact that the engine cannot keep up with the cycle by itself and more of the total power is supplied by the battery/motor. It is also likely that these vehicles are unable to follow the driving cycles. These hybrids were run on the following medium-duty driving cycles: CBD (Central Business District), NYBUS, NYCCT, and UDDS-D.

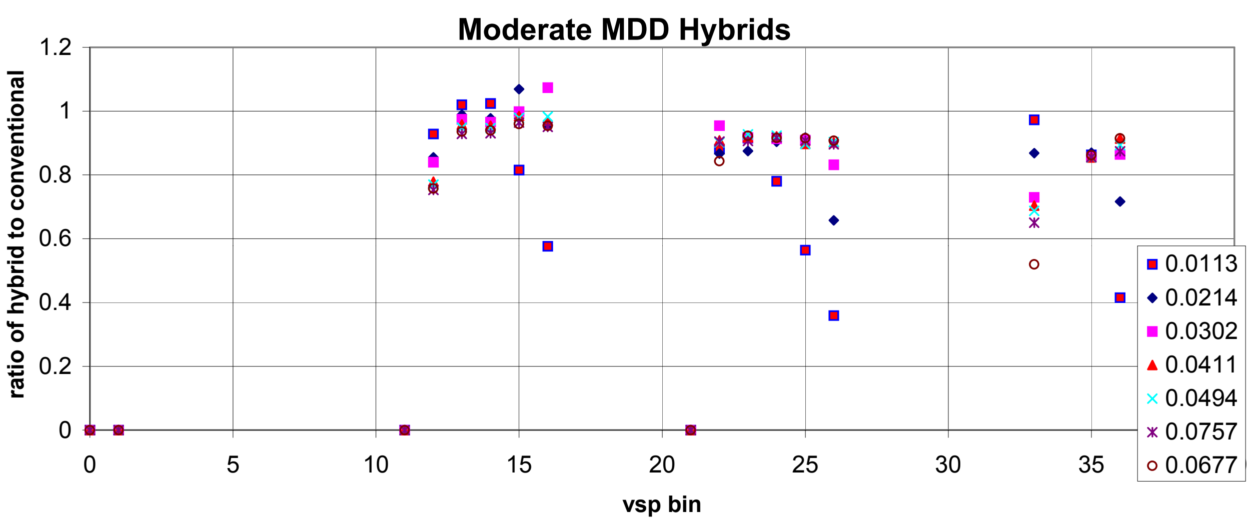


Figure 47. Fuel consumption ratios for a series of power-to-weight ratio (kW/kg) medium-duty diesel hybrid vehicles.

For future versions of MOVES, the rates may likely be separated by power-to-weight (P/Wt) ratio bins, rather than engine displacement or weight separately in order to differentiate the energy ratios for the different vehicle classes. An example of such a split is shown in Table 34. The bands of color represent possible P/Wt ratios to group together into bins. The boxed cells are modeled explicitly using PERE to obtain the results in the previous figure.

Table 34. Possible Power-to-weight ratio bins (kW/kg) for turbo diesel hybrids.

		Engine displacement or **avg pwr**					
lo wt bin	avg wt in bin	2	2.5	3	3.5	4	5
lbs	**kg**	**95.2**	**116.3**	**137.5**	**158.6**	**190.3**	**291.9**
2000	**1021**	0.0933	0.1140	0.1347	0.1554	0.1865	0.2860
2500	**1247**	0.0763	0.0932	0.1102	0.1271	0.1526	0.2340
3000	**1474**	0.0646	0.0789	0.0933	0.1076	0.1291	0.1980
3500	**1701**	0.0560	0.0684	0.0808	0.0932	0.1119	0.1716
4000	**1928**	0.0494	0.0603	0.0713	0.0823	0.0987	0.1514
4500	**2155**	0.0442	0.0540	0.0638	0.0736	0.0883	0.1355
5000	**2495**	0.0382	0.0466	0.0551	0.0636	0.0763	0.1170
6000	**2948**	0.0323	0.0394	0.0466	0.0538	0.0645	0.0990
7000	**3402**	0.0280	0.0342	0.0404	0.0466	0.0559	0.0858
8000	3856	0.0247	0.0302	0.0357	0.0411	0.0494	0.0757
9000	4309	0.0221	0.0270	0.0319	0.0368	0.0442	0.0677
10000	5443	0.0175	0.0214	0.0253	0.0291	0.0350	0.0536
14000	6804	0.0140	0.0171	0.0202	0.0233	0.0280	0.0429
16000	8051	0.0118	0.0144	0.0171	0.0197	0.0236	0.0363
19500	10319	0.0092	0.0113	0.0133	0.0154	0.0184	0.0283
26000	13381	0.0071	0.0087	0.0103	0.0119	0.0142	0.0218

Section XII - Acknowledgments

The authors would like to thank the following staff at EPA for their help and support: Megan Beardsley, Emily Chen, Kent Helmer, John Koupal, Larry Landman, Joe McDonald, Arvon Mitcham, Gene Tierney, James Warila, and Tad Wysor. Feng An and Dan Santini from Argonne National Laboratory provided valuable feedback. We would also like to acknowledge Robert Natkin from Ford Motor Company for his technical advice on H2 ICE. Matthew Barth, George Scora, and Ted Younglove offered significant assistance in the heavy-duty diesel modeling section.

Finally, we acknowledge the contributions of the reviewers who provided valuable comments about the report. Some of these comments are in the following appendices. Comments from three SAE (Society of Automotive Engineers) reviewers, however, are not included. These comments were from a paper formed from portions of the hybrid sections (both ICE and fuel cell) of this report, and published in 2005.

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Appendix A: PERE Algorithms

Key:

FR	Fuel rate (g/s)
KNV	Friction term
LHV	Lowere heating value of fuel
P_{acc}	Accessory power
P_{batt}	Battery power
P_d	Power demand
P_{eng}	Engine power
P_{FC}	Fuel Cell power
P_{engmax}	Maximum power of engine
P_{fcmax}	Maximum power of fuel cell
P_{motmax}	Maximum power of motor
P_{min}	Minimum regeneration power
P_{mot}	Motor power
P_{th}	Hybrid threshold power
η	Engine indicated efficiency
η_{disch}	Battery discharge efficiency
η_{FC}	Fuel cell efficiency
η_{FD}	Final drive efficiency (for motor only)
η_{FWD}	Front wheel drive power fraction
η_{mot}	Motor efficiency
η_{tran}	Transmission efficiency
η_{regen}	Regenerative braking recharge efficiency

Parallel Hybrid Algorithm

Engine Power (nested if statements)
IF OR($v < 2$, $P_d < P_{th}$) THEN $P_{eng} = 0$ 'idle & decel engine shut-off
IF $P_d > P_{engmax}$ THEN $P_{eng} = P_{engmax}$ 'can't exceed max
    ELSE $P_{eng} = P_d$
Fuel Rate
    IF $P_d = 0$ THEN FR = 0 'fuel shut-off
        IF $P_d > P_{engmax}$ THEN FR = $[KNV + P_{eng} / \eta\eta_{tran}]/LHV$ 'no P_{acc} – slightly overpower
ELSE FR = $[KNV + (P_{eng} / \eta_{tran} + P_{acc})/\eta]/LHV$
Battery discharge ($P_{batt} > 0$)
    IF AND($P_d \le P_{th}$, $P_d > 0$) THEN $P_{batt} = (P_d / \eta_{mot}\eta_{FD} + P_{acc})/\eta_{disch}$ 'launch
IF $P_d > P_{engmax} + P_{motmax}$ THEN $P_{batt} = P_{motmax}$ 'can't exceed max
IF $P_d > P_{engmax}$ THEN $P_{batt} = [(P_d - P_{engmax})/\eta_{mot}\eta_{FD} + P_{acc}]/\eta_{disch}$ 'assist
    ELSE $P_{batt} = 0$
Battery recharge ($P_{batt} < 0$)
    IF $P_d \le -P_{motmax}/\eta_{FWD}$ THEN $P_{batt} = -P_{motmax}\eta_{mot}\eta_{regen}\eta_{FD}$ 'can't exceed max
        If $P_d < -P_{min}/\eta_{FWD}$ THEN $P_{batt} = P_d\eta_{FWD}\eta_{mot}\eta_{regen}\eta_{FD}$ 'can't fall below min
            ELSE $P_{batt} = 0$

Fuel Cell Hybrid Algorithm

Fuel Cell Powerplant
IF OR($v < 2$, $P_d < P_{th}$) THEN $P_{FC} = 0$ 'idle & decel fc shut-off
IF $P_d > P_{motmax}$ THEN $P_{FC} = P_{fcmax}$ 'can't exceed max
ELSE P $P_{FC} = [P_d/(\eta_{mot}\eta_{FD}) + P_{acc}]/\ \eta_{FC}$
Battery (ultracapacitor) discharge
IF AND($P_d <= P_{th}$, $P_d > 0$) THEN $P_{batt} = (P_d/\eta_{mot}\eta_{FD} + P_{acc})/\eta_{disch}$ 'launch
IF AND ($P_d > P_{fcmax}$, $P_d < P_{motmax}$) THEN $P_{batt} = [(P_d - P_{fcmax})/\eta_{mot}\eta_{FD} + P_{acc}]/\eta_{disch}$
IF $P_d <= 0$ THEN $P_{batt} = P_{acc}/\eta_{disch}$
ELSE $P_{batt} = 0$
Battery recharge
Same as for hybrid (above)

Appendix B: Tables of parameters for validation vehicles.

Vehicle	MIT	Camry	Jetta	Jetta TDI	Civic DX	Civic HX	Civic Hybrid	Insight	Prius '01	Prius '04
Model Year	2020	2004	2004	2004	2004	2004	2004	2004	2001	2004
Vehicle wgt (kg)	1154	1565	1467	1483	1239	1224.2664	1354	989	1390	1447
Cr0 (rolling resistance)	0.006	0.009	0.009	0.009	0.009	0.009	0.008	0.008	0.009	0.007
Cd (drag coeff)	0.22	0.3	0.3	0.3	0.3	0.3	0.28	0.26	0.26	0.26
A (frontal area m^2)	1.8	2.4	2.11	2.11	2.14	2.14	2.14	1.92	2.11	2.33
Pacc (accessory - kW)	1	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75
A (N)		127.36	111.25	111.25	105.47	105.47	125.58	53.76	86.33	88.63
B (N/mps)		0.9578	3.6834	3.6834	5.4276	5.4276	-0.9000	2.2837	2.2355	1.3849
C (N/mps^2)		0.4374	0.3764	0.3786	0.2670	0.2670	0.4474	0.3013	0.4144	0.3645
Engine										
Engine Displ (L)	1.11	2.4	2	1.9	1.7	1.7	1.35	1	1.5	1.5
k0 (N indep friction kJ/Lrev)	0.153	0.164	0.164	0.123	0.164	0.15088	0.15088	0.15088	0.164	0.164
k1 (N dependent fric)	0.00155	0.00155	0.00155	0.00215	0.00155	0.00155	0.00155	0.00155	0.00155	0.00155
P/T indicated eff (eta)	0.405	0.405	0.405	0.45	0.405	0.48	0.48	0.48	0.46575	0.46575
Transmission										
N/v (rpm/mph)	35.6	35.6	35.6	26.7	35.6	35.6	35.6	35.6	35.6	35.6
Nidle (rpm)	700	700	700	700	700	700	700	700	700	700
trans eff	0.88	0.88	0.88	0.88	0.88	0.88	0.88	0.88	0.88	0.88
Shift point 1-2 (mph)	18	18	18	18	18	15	18	18	18	18
Shift point 2-3	25	25	25	25	25	25	25	25	25	25
Shift point 3-4	40	40	40	40	40	40	40	40	40	40
Shift point 4-5	50	50	50	50	50	50	50	50	50	50
g/gtop 1	4.04	4.04	4.04	4.04	4.04	4.04	4.04	3.461	4.04	4.04
g/gtop 2	2.22	2.22	2.22	2.22	2.22	2.22	2.22	1.75	2.22	2.22
g/gtop 3	1.44	1.44	1.44	1.44	1.44	1.44	1.44	1.096	1.44	1.44
g/gtop 4	1.00	1.00	1.00	1.00	1.00	1	1.00	0.86	1.00	1.00
g/gtop 5	0.90	0.90	0.90	0.90	0.90	0.9	0.90	0.71	0.90	0.90
Fuel										
LHV (kJ/g)	43.7	43.7	43.7	41.7	43.7	43.7	43.7	43.7	43.7	43.7
density gas (kg/L)	0.737	0.737	0.737	0.856	0.737	0.737	0.737	0.737	0.737	0.737
Motor										
overall efficiency	0.76						0.76	0.76	0.76	0.76
Regen Brake Eff	0.85						0.85	0.85	0.85	0.85
FWD power frac	0.7						0.75	0.75	0.7	0.95
Motor peak power (kW)	30						10	10	33	50
min regen (kW)	2.8						2.8	2.8	2.8	2.8
Motor Energy (kWhr)	1.8						1.8	1.8	1.8	1.8
Battery										
Initial SOC	0.56						0.56	0.56	0.56	0.56
Batt Energy (kWh)	1.8						0.936	0.936	1.8	1.3104
min SOC	0.2						0.2	0.2	0.4	0.4
max SOC	0.8						0.8	0.8	0.8	0.8
discharge eff	0.95						0.95	0.95	0.95	0.95
Hybrid										
hybrid threshold (kW)	1.75						2.2	1.5	2.18	2.9

Appendix C- Comments from Alliance of Automobile Manufacturers, including responses. [1]

These comments were made regarding the previous version of this paper "Advanced Technology Vehicle Modeling in PERE" EPA # EPA420-D-04-002, March, 2004.
The response to comments will be separated into general and specific. The specific responses are below the comments, which are copied over from the original text.

General Response:
We appreciate the time and effort that AAM members put into commenting on the report: "Advanced Technology Vehicle Modeling in PERE." The depth and breadth of the comments indicate that careful thought and consideration went into many of the specific comments.

However, from the nature of the more technical comments, it is possible that the report failed to communicate the limitations and use of the model. While, it is stated more clearly in the revised report, it will be clarified here. The PERE model will NOT be used in MOVES to capture the behavior of a specific vehicle, but rather a fleet of vehicles. Therefore, the accurate representation of its components (transmission, motor, etc) was not the goal. If users are aware of improved coefficients or algorithms, the model will be available for changes so that fuel consumption from individual vehicles can be more closely estimated. The validation to individual vehicle (make and model) is an extreme validation when the model is pushed to its limits. This will be clarified in some of the response to comments below.

The original report (dated March, 2004) was released as a draft version. As a result, this new report supercedes the previous version, and addresses some of the comments. Two sections were added to the report (Heavy-duty and motorcycle modeling). However, the fundamental structure of the model (PERE), remains unchanged. The present report also benefits from the support and cross referencing of other MOVES documentation, which help describe how PERE will supply rates to MOVES.

Comments on EPA "Advanced Technology Modeling in PERE"
Alliance of Automobile Manufacturers
April 19, 2004

Executive Summary
On March 11, 2004, EPA released a report entitled Advanced Technology Vehicle Modeling in PERE (Physical Emission Rate Estimator). The report describes a model created by EPA to estimate

[1] Comments were also submitted from several SAE reviewers when the following paper was submitted: E.K. Nam, "Fuel Consumption Modeling of Hybrid Vehicles in PERE.," SAE2005-01-0627, 2005. This SAE paper describes the hybrid modeling portions of PERE. While these comments are not included here, the reviews were generally positive. The relevant corrections were also applied to this draft.

fuel consumption over any input driving cycle from different types of vehicles, including advanced gasoline internal combustion engines, advanced diesel internal combustion engines, hybrid electric and gasoline/diesel powertrains, and hydrogen fuel cell hybrids. Basically, PERE takes vehicle input parameters, then runs the vehicle through driving cycles defined by the user and outputs second-by-second fuel consumption rates. The purpose of the model is to "fill data gaps in MOVES and to help it extrapolate to future projections of energy and emissions."

The report states that the model is validated to four production conventional gasoline and five advanced (and hybrid) vehicles by comparing the PERE fuel consumption for these vehicles to the rated fuel economy figures. The report further states that the predictions are within 10% of the rated values.

Alliance members reviewed the PERE report. However, in order to do a thorough review, we would need to evaluate the actual PERE model. The model was not provided with the report, and we hope that in time EPA will provide the actual PERE model to enable a more thorough review. Since we were only able to evaluate the report, our comments should be viewed as preliminary. Further, given the technical issues raised below, we recommend EPA redraft the report with additional description of how it will be implemented into the MOVES model (as well as other projected uses) and re-release the report for external peer review.

A summary of our major comments is outlined below. Together, these comments raise concerns with the technical content of the report as well as over-arching issues related to the use and purpose of the model.

Model Design and Validation

Comment:
• **The design of the model does not incorporate vehicle performance requirements,** which can have a significant effect on predicted fuel economy.

Response:

There are a number of reasons that power was the only performance requirement applied to PERE.

1- Power is a surrogate variable (for engine displacement) that is capable of being transferred directly into the MOVES framework. MOVES is incapable of distinguishing vehicles with varying 0-60 acceleration, gradeability, or trailer tow capability.
2- The report mentions that the PERE model is meant to fill holes in MOVES. This also implies that the activity patterns (driving cycles) employed to run the models are typical driving patterns. The extreme driving patterns are captured by PERE (e.g., full out acceleration, or steep grades), but its accuracy has not yet been determined.
3- As a spreadsheet model, PERE could not easily capture 0-60 (acceleration) performance criteria. PERE may need to be coded (due to multiple feedback loops required). This may be done in the future.

Comment:

• **The model sensitivity test results raise technical concerns.** The static sensitivity tests performed by EPA for conventional vehicles demonstrate weaknesses of PERE. For example, PERE's sensitivity to idle speed is greater on the federal Highway cycle than in the lower speed FTP cycle. The term 'powertrain indicated efficiency' has no physical meaning; indicated efficiency only has meaning in the context of the heat engine alone. That said, the sensitivity to powertrain indicated efficiency reported in Table 5 is about
half of the expected 10% for engine indicated efficiency. The sensitivities to aero and rolling resistance differ between conventional and hybrid vehicles.

Response:

-The highway fuel consumption is sensitive to idle speed because in PERE, the engine defaults to the idle state when there is no power demanded of the engine, such as during decelerations. By this definition, there is actually significant engine "idle" operation, though it is not "modally" idle.

-"powertrain indicated efficiency" has been changed in the paper to "engine indicated efficiency".

-Due to the approach that PERE takes (equation 11), the indicated efficiency affects fuel consumption in the brake term, and not the friction term (directly). This may be slightly counter-intuitive, and would probably be more correct if an "fmep" value of friction were used instead of "K", which already has the indicated efficiency inherent to it. "K" was used to remain consistent with past modeling efforts. However, as the model stands, this correction would make little difference (except in the sensitivity study or when indicated efficiency changes, but K does not). This will probably be changed in a future version of PERE.

-The sensitivities to road coefficients for hybrids do indeed vary between hybrids and conventional vehicles. The degree to which the sensitivity differs is difficult to predict due to the intermix of power demand, driving cycle, and regenerative braking.

Comment

• **The report does not demonstrate the rationale for a 10% validation criterion.** The report indicates that the model predicts fuel economy for several validation vehicles within 10%. However, it is not clear how EPA has determined that 10% is an acceptable level. Depending on the use of the model, predictability within 10% over the FTP and Highway Test may not be acceptable. EPA should explain in more detail in the report the various uses it intends for the model so that the reviewers can decide whether 10% is acceptable. Further, EPA should determine whether PERE makes random errors or systematic errors, such as consistently over-predicting or under-predicting FE for all or particular powertrains.

Response:

The 10% criterion was determined by combination of the coarseness of following factors: the fleet and vehicle source bins in MOVES, the (VSP) operating mode definitions (including cycle sensitivities), and PERE. It would not be practical for PERE to predict fuel economy to within 2% given the limitations of MOVES. Other models are more suited to such accuracy levels. It is expected that most of the uncertainties are "random", reflecting the uncontrolled variability in the parameters due to operating region or fleet characteristics, rather than "systematic" effects.

Comment:
• **The report does not account for cold starts:** The report compares FTP and Highway tests with cold start effects to the PERE model without cold start effects. Cold start CO_2 should be 3% to 5% higher than fully warmed up CO_2.

Response:

Cold start is modeled separately and described in a separate document. However, a supplemental discussion of hybrid cold start rates is described in the revised document. This is preliminary and only based on 2 hybrid vehicles tested.

Comment:
• **The report does not demonstrate overall robust validation.** In spite of claims in the PERE report and the apparent agreement between PERE predictions and unadjusted vehicle FE, the PERE model has not been validated. There are a number of reasons for this, as follows:
- In validating the model, only a narrow range of vehicles was considered. The model should be validated on a wider range of vehicle weights and vehicle classes.
- Nearly all of the vehicles used for validation were assumed to have the same transmission and shift point parameters, instead of those of the actual vehicles in the study. Meaningful validation can only be conducted when the input parameters to the model are identical to the characteristics of the vehicles being studied.
- Only the FTP and Highway test cycles were considered during validation. However, the MOVES model is based on 14 other test cycles that have higher speeds and accelerations than the FTP and Highway tests, and no validation is performed on these cycles.
- Benchmarking PERE against the MIT studies is not sufficient to demonstrate the quality of the predictions. There must be a comparison to actual vehicle data.

Response:

-The PERE model has been validated against 41 "conventional" passenger vehicles driven on the FTP as well as US06 cycles in a previously published report: (Nam, EPA document number:420-R-03-005, Table 2). In the same document, analyses were conducted against another 17 vehicles driven on a series of 8 cycles developed by CARB (UCC cycles). Moreover, the same modeling methodology has been proven robust by a number of researchers in several different (peer-reviewed) publications. These authors include (Feng An, Matt Barth, Marc Ross, among others…). Some of these publications include truck, diesel, and even advanced technology vehicle modeling over several different driving cycles.

-PERE does not actually incorporate a detailed transmission model. Since the engine model is based on power (and not torque), transmission takes on secondary importance. [This also reduces the importance of proper transmission/engine matching – a comment below]. The transmission parameter that is of primary importance is the efficiency, which will discussed in more detail below (and in the new version of the paper). The shift points, gear ratios, etc., only affect the friction term of the fuel rate equation 11. If PERE was based on Torque/RPM/bsfc engine maps (as are many other more detailed models), transmission would play a more significant role. Due to the "coarseness" of the PERE powertrain model (and the coarseness of the MOVES vehicle source bin definitions) precise transmission models were not deemed necessary. To further demonstrate this point, the validation in this report (and in

other references mentioned above) is proven to be quite robust, despite the use of the same "average transmission" for many of the vehicles.
-PERE is compared to the MIT report merely to demonstrate that when the same vehicle is modeled, PERE gives the same result. This does not imply that all of the same assumptions made in the MIT report (especially about future technologies) were made in PERE or MOVES. The hybrid results from PERE are quite different from the results obtained in the MIT report due to these difference in assumptions. Naturally, comparisons to data will be conducted as data is received.

Comment:
Model Uses and Purpose
• **The report does not describe the data gaps to be filled by PERE.** Complete information regarding the application of the PERE model for use in MOVES is central to determining the sufficiency of modeling assumptions, accuracy and/or validation. Without this key information the reviewer does not have adequate information to fully evaluate the adequacy of the model as described in the report.

Response:
The "hole-filling" report is described in a separate document to be released concurrently with this one. Also, the section has been expanded in the revised report.

Comment:
• **The report does not explain need for new modeling tool.** EPA has not fully explained why it had to create a new model for the purpose of examining fuel consumption from advanced technologies, rather than using other available models such as the Argonne vehicle model utilized in GREET.

Response:
Models such as ADVISOR, PSAT, etc. were considered, but deemed impractical for integration with MOVES. ADVISOR (for example) is more detailed and complex (than PERE) and is designed to model individual vehicles, rather than fleets of vehicles. Additionally, ADVISOR is coded in the MATLAB/SIMULINK environment, which the users of MOVES are not assumed to possess. ADVISOR is also soon to be commercialized and not freely available to users. It is possible that in the near future, PERE might be bechmarked with PSAT (developed at Argonne National Laboratory), however, it will be important to remember that there are many differences between the models.
It was necessary to provide a model that was more accessible to MOVES users, who may want to adjust the PERE rates. The specific user and purposes of PERE adjustments are not yet known. Finally, the management of EPA considers it important to keep the expertise in this type of modeling in-house, rather than relying on contractor support.

Comment:
PERE does not address powertrain matching considerations (that is, selection of engine displacement, transmission gear ratios and final drive) explicitly. Performance metrics are not specified or

calculated. This is especially critical in extending PERE to the analysis of other vehicle classes; particularly trucks where the trailer tow requirements can be significant. How is the user assured that the powertrain match is 'ballpark' correct? For example, if only 0-to-60 time is matched, SI engines are undersized, fuel economy is overstated and off-cycle vehicle performance is generally not acceptable. In industry vehicle simulations, vehicle performance requirements for passenger cars and trucks are specified a priori and include real world requirements such as performance times, passing time in top gear, vehicle launch acceleration, top vehicle speed, gradeability and trailer tow.

Response:

This point is addressed above.

Comment:

• In spite of claims in the PERE report and the apparent agreement between PERE predictions and unadjusted vehicle FE, the PERE model has not been validated. There are a number of reasons for this, as follows:

1. In the PERE validation effort, only a narrow range of vehicles was considered.
2. All of the vehicles used for validation were assumed to have the same transmission and shift point parameters, instead of those of the actual vehicles in the study.
3. Only the FTP and Highway test cycles were considered.
4. The report compares FTP and Highway tests with cold start effects to the PERE model without cold start effects.
5. Benchmarking PERE against the MIT studies is not sufficient to demonstrate the quality of the predictions.

These reasons are discussed further below.

1. In validating the model, only a narrow range of vehicles was considered. The report compared unadjusted fuel economy to the fuel economy as predicted by the model for the following vehicles: Toyota Camry, VW Jetta (gas and diesel), Honda Civic DX, Honda Civic HX, Honda Civic hybrid, Honda Insight, Toyota Prius (01), and Toyota Prius (04). The conventional gasoline vehicles are the Camry, Jetta, and Civic DX. None of these vehicles is larger than EPA mid-size car class. The comparison included no heavier light duty passenger cars, no SUVs, no minivans, and no light duty trucks. The report indicates the model is capable of estimating fuel economy for all conventional gasoline vehicles. It is not clear why the validation on conventional gasoline vehicles was so limited. A sample of representative vehicles from each of the EPA vehicle categories should be used to demonstrate that the model is valid for the range of vehicles that will be studied using PERE.

2. All of the vehicles used for validation were assumed to have the same transmission characteristics, instead of their own individual characteristics.

The vehicle descriptions in Appendix B show that the same transmission, gear ratios, N/V and shift points were used in making the PERE estimates for the MIT study, Toyota Camry, VW Jetta, Honda Civic DX, Honda Civic HX, Honda Civic Hybrid, Honda Insight, Toyota 2001 Prius and Toyota 2004 Prius. Meaningful validation can only be conducted when the input parameters to the model are identical to the characteristics of the vehicles being studied.

3. Only the FTP and Highway test cycles were considered.

EPA has only shown a demonstration of PERE for regulatory drive cycles where peak vehicle acceleration is below 0.165 G and peak vehicle speed is below 60 mph. Nothing was shown for customer driving where the vehicle speeds and accelerations are significantly higher. Extrapolating speeds and loads from regulatory cycles to simulate customer driving is not appropriate. The FE benefits of advanced technologies depend on the drive cycle (such as, the hybrid FE benefit on the federal city cycle being greater than that on federal highway cycle). The ability to use PERE to estimate corresponding FE benefits on unspecified customer cycles has not been demonstrated. Figure ES1 and Figures 11-13 in the report compare the PERE model predictions to the reported fuel economy values for both the FTP and the highway test. However, Table 9 of the report indicates that neither the FTP nor the highway test will be used to establish vehicle activity and emissions in MOVES; this vehicle activity and the emissions will be based on 14 other light duty driving cycles. These 14 driving cycles generally have higher accelerations and speeds than either the FTP or the highway test, and these cycles were also used in MOBILE6. The report has no comparisons of fuel economy for the PERE model and these cycles. In addition, the report states that while the model is calibrated so that the state of charge for hybrids is close to zero after the FTP, this may not be true for other driving cycles, like all of the driving cycles that will be used in MOVES. EPA has determined that these 14 cycles are more representative of overall in-use driving than either the FTP or the highway test. EPA should therefore check the accuracy of the PERE model on some of the cycles intended to be used in MOVES.

4. The report compares FTP with cold start effects to the PERE model without cold start effects. The PERE model currently does not model cold start effects on fuel consumption, nor is there any plan to add cold start effects to PERE. EPA plans to add cold start effects in MOVES. However, the FTP tests do have cold start effects included. Therefore, EPA compares PERE predictions of fuel economy without cold start effects to test data that includes cold start effects. This is an “apples to oranges” comparison. EPA should either add cold start effects to PERE and compare the model predictions to test data with cold start effects, or EPA should obtain test data on the FTP without cold start, and compare that to the PERE model predictions.

5. Benchmarking PERE against the MIT studies is not sufficient to demonstrate the quality of the predictions.

o Figures ES1, 11, 12 and 13 (that show agreement between PERE and the MIT study for a near-70 mpg vehicle) should clarify that the model results shown are for an advanced SI engine hybrid for which there is no experimental data. This does not mean either model is right or wrong; merely that the models seem to make similar predictions.

o Benchmarking PERE against the MIT "on the Road in 2020" and "Competitive Assessment of Fuel Cell Cars" is not sufficient to demonstrate the quality of the predictions. PERE has used several assumptions from the MIT study that are questionable, specifically: The energy density of gasoline per liter is 1.5% too large. For similar vehicles, the MIT estimate of FE would be 1.5% higher, everything else being equal.

The 91 RON BMEP of the naturally aspirated advanced SI engine in the MIT report is too optimistic. Specifically, peak BMEP exceeds 13.3 bar and that at 1500 rpm is 11.0 bar, compared to observed levels of ~12.2 bar and ~9.8 bar, respectively.

The MIT report equates equal power-to-weight to equal performance (*On the Road in 2020*, p. 3-5) which is the same as setting only full throttle acceleration, e.g., 0-60 time, equal. Other characteristics of vehicle performance, such as continuous gradeability, are not strongly related to the peak power-to-weight ratio. The equal power-to-weight assumption for SI engines results in undersized engines and overstated fuel economy. Conversely, this assumption is likely to result in oversized diesel engines.

The MIT reports assume very aggressive engine downsizing in hybrid vehicles that would produce compromised off-cycle driveability and overly optimistic projected FE benefits; for example, >40% projected highway cycle FE benefit.

The 2003 MIT report is based on very optimistic fuel cell vehicle assumptions: high fuel cell system performance, significantly reduced ancillary losses and high H_2 utilization.

Finally, there are several other studies published based on more realistic technology assumptions than the MIT study which could better serve for the purpose of calibrating the PERE model. Among these studies are the following:

"Well to Wheels Analysis of Future Automotive Fuels and Powertrains in the European Context", EUCAR, CONCAWE, and European Commission, Joint Research Center, Version 1b, January 2004.

"Potential of IC-Engines as Minimum Emission Propulsion System", Eichlseder and Wimmer, Atmospheric Environment, 37 (2003) 5227-5236

"A Comparison of Hydrogen, Methanol, and Gasoline as Fuels for Fuel Cell Vehicles: Implications for Vehicle design and Infrastructure Development", Ogden, Steinburgler, Kreutz", Journal of Power Sources 79 (1999) 143-168.

Response:

Most of these comments are addressed above. There are some specific additional responses:

3- As mentioned earlier, other studies have validated PERE modeling conventional technologies on other (more aggressive driving cycles. However, the reviewer is correct that PERE for hybrids has not been validated to other driving cycles. This is difficult due to the lack of data. The model will be improved as more data on hybrid vehicles becomes available. Some discussion was added based to published data from NREL, which shows some results from the US06 driving cycle for the Honda Insight. The hope is that at the level MOVES models energy rates, the fuel rates by operating mode bin (VSP) will be relatively independent of (most) drive cycles effects. Any added complexities would take the model details beyond the level of detail that MOVES is capable of capturing. However, to address some of the concerns from this comment, the modeled advanced technology vehicles were run on an additional cycle: the LA92.

4- Cold start will indeed have an affect on measured fuel economy. This was not quantified in the original report. Some measurements taken at the EPA now supplement the data. However, cold start in MOVES is modeled by other means (mainly as a function of engine displacement), and can be found among the MOVES documentation.

5- We appreciate the comments on the MIT report. Again, PERE only uses the same parameters as the MIT model does when comparing the same vehicle (the MIT vehicles). When modeling MOVES or existing vehicles, a different set of parameters are employed. For example, note that the fuel cell system efficiency curves are significantly lower than the ones used by MIT. Also, PERE is not identical to the MIT model since some of the details of the MIT model are not published. PERE is more like the models in other publications (An, Barth, Ross, et al).

Specific Technical Comments:
The following comments are organized in order of page number.

Page 6 (second paragraph):
The text states that the PERE Vehicle Specific Power estimate is lower than that obtained from the coastdown equation and that FE would be overestimated. No estimate of this error was provided.

Response:

This is discussed in a little more detail pp 33-35 (original report).

Page 7:
At part load, a four-valve SI engine is more efficient than a two-valve engine because improvements in indicated efficiency from the higher compression ratio and lower combustion chamber surface area more than offset the increase in valve train and cam drive friction. Differences in pumping are minimal.

Response:

The friction term (fmep) includes the improvement in "breathing" that accompanies the increase in area of the intake (and exhaust) openings. PERE does not assume a significant increase in indicated efficiency due to additional valves.

Page 9 and Page 15:
The opportunities for FE gains from lean operation are not likely to materialize because of the high levels of NOx conversion efficiency required at Tier 2 Bin 5 and PZEV tailpipe NOx standards. Major breakthroughs in lean after treatment systems are required to realize significant FE gains from lean cruise and stratified charge DI operation. GDI is already in production in Europe where the NOx emissions standards are more lenient. Any estimate of benefits from lean operation must also include an assessment of compliance with tailpipe NOx and HC standards and estimates of other factors such as vehicle weight and performance.

Response:

The modeling of the lean burn engines was of current vehicles (even though their lean operation may be very limited), which are not subject to Tier 2 standards. It is certainly possible that lean burn vehicles meeting Tier 2 will have fuel economy disbenefits to meet emissions standards. This will likely be incorporated into the model.

Page 11:

The efficiencies of the transmissions seem to be too simplistic and need to be checked. If average efficiencies are used then these should be obtained from experts and should reflect average efficiencies on the duty cycles. We believe it is poor argument that the efficiency of a manual is the similar to a continuously variable transmission (CVT). CVTs improve engine operating efficiency but are not more efficient on their own than discrete step transmissions (they have higher pumping losses, especially when designed to deliver ratio change responses to meet discrete step transmission performance requirements).

The shift logic implemented as described on p.11 is too simplistic and does not agree with data shown in the table on p.48. The loss in automatic transmission efficiency during gearshifts due to slippage in the torque converter is not a significant loss to worry about relative to other losses represented in the transmission models (this is usually combined into the average efficiency on the driving cycle already).

Drive quality or other real world performance requirements (launch acceleration, 0-60 time, and top vehicle speed) have a significant impact on fuel economy and are ignored by PERE. Tradeoffs between fuel economy and drive quality vary significantly between conventional and hybrid vehicles and some basic drive quality criteria can be easily incorporated into the overall analysis methodology.

Response:

> The transmissions efficiency discussion has been fleshed out after some discussion with transmission experts. Transmission models (especially automatic) can be extremely complex. As mentioned earlier, the transmission model was never meant to be detailed since PERE models a “typical” vehicle, rather than a specific one. As with other components, when run for MOVES, PERE may not specify the actual physical transmission, it will merely give a “target value” for efficiency. This is what was meant by advanced transmissions approaching manual transmission efficiency. It is agreed that CVT efficiency is not presently that efficient yet.

Page 12:
It is not clear if the efficiencies used in the PERE model are average duty cycle or maximum brake thermal efficiencies. This should be clearly stated.

The assumed 45% diesel engine indicated efficiency appears to represent the peak indicated efficiency and is too high for current engines. Applying this efficiency uniformly over the entire range of speeds and loads of a drive cycle is not appropriate. What size engine is assumed and what engine or aftertreatment technologies have been considered? It appears that PERE assumes for diesel engines that U.S. emission standards can be met with aftertreatment that has no fuel economy penalty. Diesel engine compliance with U.S. tailpipe NOx standards is a significant challenge; estimates of the fuel economy penalty for emissions compliance associated with lean NOx trap and catalyzed diesel particulate filter regeneration should be incorporated into PERE. Although diesels are used widely in Europe, the NOx standard is about 8 times more lenient there than in the U.S.

EPA estimates that vehicles with diesel engines weigh approximately 4% more than their gasoline counterparts. However, this assessment is based on current diesel vehicles that do not include aftertreatment and do not meet more stringent Tier 2 standards. The Tier 2 standards will require extensive aftertreatment, which will add to the weight of diesel vehicles relative to gasoline vehicles. EPA should include expected Tier 2 aftertreatment impacts in its assessment of the relative weight of gasoline and diesel vehicles.

Response:

> The efficiencies used in PERE are average indicated thermal efficiencies according to engine map data (excluding idle and wide open throttle operation). The paper (and one reference) gives a detailed description of what this indicated efficiency is. The use of this method has been proven to be quite robust in estimating fuel consumption in various papers (see above responses and references). The fuel economy penalty of diesels (and other lean burn technologies) was also discussed above. EPA will likely include a penalty for future model years as this becomes better understood.

Page 13:
The argument for engine sizing not being critical because this model is to be used for fuel economy and not for performance analysis disregards the most critical aspect of assessing and comparing the fuel economy potential of advanced powertrain technologies. This intentional oversight ignores the tradeoff between fuel economy and vehicle performance requirements; fuel economy comparisons must be made on an equal performance basis.

Response:

> As mentioned earlier, the performance metric used in PERE (and MOVES) are power and weight, (or perhaps power to weight ratio). This is really the only practical performance metric MOVES can accommodate.

Page 15:
EPA lists many reasons why diesel engines are more efficient than gasoline engines. EPA concludes that diesel engine indicated efficiency is assumed to be 45% and gasoline engine indicated efficiency is 40%. Therefore, the 48% indicated efficiency of the Honda lean burn engine is better than a diesel and should be revised.

There is no justification for assuming that the Atkinson cycle engine is 15% more efficient than current engines. Peak torque of Atkinson cycle engines are typically close to those of conventional engines, peak torque speed is higher because of the late intake valve closing. Also, Atkinson low speed torque is degraded. Consequently, Atkinson cycle engines are best used in hybrid powertrains and not with conventional transmissions.

Engine models for advanced technologies such as HCCI, SIDI must be described in more detail and demonstrated to be valid. The model should be reviewed by the engine experts for current and advanced technologies.

Response:

> The efficiency of the Honda lean burn is probably too high for PERE. However, it was obtained from their SAE paper [SAE 2003-01-0083]. Even if it is appropriate, the operating

regime for lean burn operation might be quite small, especially in the future (due to NOx requirements). Since PERE used generic coefficients for transmissions (see above), it should also have used generic coefficients to describe these advanced engines, rather than attempting to model them explicitly. This would be more in line with inputs to MOVES. However, since MOVES models "generic" hybrid vehicles, the parameters of this specific engine are not used.

Pages 17, 23-29, Appendix A.
The controls models are too simplistic. Control strategies are important for propulsion system modeling because hybrid control strategy has a strong impact on the fuel economy estimate. In the PERE model, incorporation of a variety of control architectures and optimization of the control strategy are difficult, if not impossible. There is no assurance that the assumed control strategy would result in appropriate vehicle performance off-cycle.

Response:

This is certainly true. However (as mentioned earlier), PERE is not meant to be a detailed model. Energy (or power) demand is distributed to engine, motor, etc. accounting for losses. The target for PERE is to be within 10% fuel consumption (more or less random effects), so a detailed model is not deemed necessary at this scale. The model will be continually validated as more data is collected.

Page 17:
PERE is a spreadsheet tool and should not be referred to as a 'simulation' tool

Response:

This is a matter of semantics, however, all references to "simulation" have been replaced with "model".

Page 18:
The charge-sustaining strategy in PERE's HEVs assumptions must be more thoroughly reviewed, as there appears to be a contradiction in description; i.e., there is a statement that batteries are only being charged through regenerative braking and not the engine, whereas in a prior section it was stated that the batteries are being charged during highway driving.

Response:

The batteries are regenerated also during some coasting operation. During the highway cycle, there are coasting as well as braking events. Thus, the assumptions are sound.

Page 20:
The purpose of mentioning hydraulic hybrids and ultracapacitors is not clear. The report discusses methods of modeling hybrids that utilize hydraulics and ultracapacitors, but does not discuss how so called "minimal" hybrids would be modeled with PERE.

Response:

Thus far, PERE has no plans to model “minimal” hybrids (ISG with idle shut-off and minimal regenerative braking). This may change in the future. The energy storage device is treated rather generically in PERE, regardless of whether that storage device is a battery, ultracap, or hydraulic in nature. The user can adjust the coefficients appropriately.

Page 21:
Vehicle mass estimation is very critical in the comparison of advanced engine and hybrid vehicle technologies impacting vehicle performance and fuel economy and thus the model input data should be carefully reviewed.

Page 22:
Description of handling the accessory loads for conventional and hybrid vehicles are not clear. Also, there is no description of the electric motor performance. An AC induction motor is used in the Ford Focus and DCX F-Cell while permanent magnet motors are used in the Toyota FCHV and Honda FCX. These motors have different performance and efficiency characteristics.

Response:

Once again, the components are generic in nature (since the model is for fleet modeling). Accessory loads (air conditioning) are modeled separately as a temperature correction in MOVES. If it is necessary to model it explicitly in PERE, this may be done in the future.

Page 23:
Figure 10 does not demonstrate that there is no change in battery state of charge for hybrid vehicles across the city and highway drive cycles. How does PERE assure that the battery state across customer driving and other off-regulatory cycles does not change? This is a critical input to the estimate of fuel use in the model.

Response:

Figure 10 merely demonstrates that the state of charge algorithm is different between PERE and the MIT model. The algorithm that MIT uses was not published. For the vehicles modeled in PERE, charge is conserved over the driving cycle. When PERE supplies rates to MOVES, charge will be conserved over the drive cycles (when it can). We realize that an optimized hybrid model is beyond the scope of this project.

Page 27-29:
Fuel cell vehicle modeling where the propulsion system characteristics are obtained from a spreadsheet produces rough estimates of fuel cell and fuel cell hybrid fuel economy (within ~10%). More accurate fuel economy estimates require a dynamic model that simulates the detailed performance of individual components interacting with the vehicle and complex control strategies.

Page 29:
There is insufficient data to validate the model predictions for fuel cell hybrid vehicles, as the author of this report pointed out. Only comparisons could be made with the Honda FCX with its unique hybrid system architecture and control strategy.

Response:

As more data is obtained, the model will be updated.

Page 30 Figure 17.
Fuel cell system performance seems to represent static performance. Efficiency during dynamic behaviors and at real world operating conditions (operating temperature is not stable in the real world usage) will be worse. The issue is how to incorporate these adjustments into the model. A dynamic model would include these characteristics.

Page 30:
The maximum efficiency at 60% for fuel cell vehicles seems high. One member company has modeled this at 55%.

Page 32-33:
The static sensitivity tests performed by EPA for conventional vehicles demonstrate weaknesses of PERE. For example, PERE's sensitivity to idle speed is greater on the federal Highway cycle than in the lower speed FTP cycle. The term 'powertrain indicated efficiency' has no physical meaning; indicated efficiency only has meaning in the context of the heat engine alone. That said, the sensitivity to powertrain indicated efficiency reported in Table 5 is about half of the expected 10% for engine indicated efficiency. Why is the sensitivity to aero and rolling resistance differ between conventional and hybrid vehicles?

Response:

See above.

Page 33:
For the road load and track coefficients – it is not very significant what method is used as long as this method is used consistently for all vehicle concepts modeled; however, using vehicle aerodynamic and tire rolling resistance coefficients may add less uncertainty to the predictions than assuming vehicle coast down data which combines too many unknown losses.

Response:

The point of this analysis was to demonstrate that a higher order rolling resistance term is necessary in modeling most vehicles, or else the loads (and hence fuel consumption) will be underestimated significantly. Most models in the literature either use a higher order rolling resistance term, or a “B” term explicitly. The MIT model (for example) does not.

Page 37:
The report indicates that the source bins do not have a dimension for body type, and that the estimates will be based on vehicle weight. Within a given weight range, however, there can be considerable variation in body type that can result in wide variation in aerodynamic drag coefficient. The uncertainty in each of the vehicle specific power bins could be very high because of this factor. To reduce this uncertainty, EPA may want to consider dividing each weight range into different levels by general body type or aerodynamic drag coefficient range.

Response:

> This will be taken under consideration. However, this consideration must be evaluated against the need to avoid excessive numbers of vehicle "bins" in MOVES. It is likely that this will remain one of the uncertainties (hopefully quantified).

Page 41.
The observation that emissions compliance is determined entirely on performance during the cold start period is based on experience with engine systems that operate at stoichiometry with a TWC. This is not true for advanced gasoline and diesel engine technologies where the engines operate lean throughout the drive cycles because the lean NOx conversion efficiencies are substantially lower than those of a TWC. For compliance with PZEV emission standards, NOx throughout the entire drive cycle is critical. Additionally, a statement about the tailpipe criteria emission standard must be made for all fuel economy estimates.

Response:

> This subject will have to be revisited and will likely be corrected by the time criteria pollutants are modeled in MOVES.

Page 48
The table of parameters for vehicle validation needs to be reviewed and explained. For example, the shift points presented for Prius are suspect.

Comments Regarding Modeling Purpose and Use
EPA states that the purpose of PERE is to "fill data gaps in MOVES and to help it extrapolate to future projections of energy and emissions." The following are our comments on the purpose.

• EPA is not specific about the "data gaps" EPA anticipates PERE will fill. There is also little or no information out these data gaps in the report, or how the model will be used for such a purpose.
The MOVES model is being developed from second-by-second data to estimate fuel consumption and emissions from user-inputted driving cycles. Similarly, PERE has been developed to estimate fuel consumption from user-inputted driving cycles, on both current and advanced technology vehicles. There appears to be an overlap between the two models, since both

models estimate fuel consumption for current conventional vehicles. EPA plans to integrate PERE into MOVES. What are the data gaps that EPA hopes to fill with PERE? Where will PERE be used instead of MOVES? Will this be a specific model year for each vehicle class, and if so, which one? Since the model has been developed to fill certain "data gaps", the report should be clearer about which data gaps the model will fill. Complete information regarding the application of the PERE model for use in MOVES is central to determining the sufficiency of modeling assumptions, accuracy and/or validation. Without this key information the reviewer does not have adequate information to fully evaluate the adequacy of the model as described in the report.

Response:

The filling of data gaps is a complicated subject to be handled in more detail in a separate document. There has also been some added tests to the new version of this report.

• **EPA has also not completely explained why it had to create a new model for the purpose of examining fuel consumption from advanced technologies.**

The PERE report indicates that EPA considered using the ADVISOR model, but decided against this approach because ADVISOR could not be readily integrated into MOVES. However, there are other models that estimate fuel consumption from advanced technology vehicles. For example, there is the Argonne National Lab (ANL) vehicle model used in GREET that also perform these functions. Was this model considered, and if not, why not?

Further, assuming that a creation of PERE is justified, how do the results of PERE compare with those of the other models (that is, ADVISOR, PSAT, GREET)? What is being done to reconcile differences, if they exist?

Response:

These comments have been addressed above. PERE has been validated to the some of the same vehicles that ADVISOR has been validated with.

• **It is not clear from the report who the "users" of PERE will be, or what level of information EPA will supply to the users.**

The executive summary of the EPA PERE report specifies that an "informed user" can run the model. An informed user is not defined. We are concerned about how much and what type of experience simulating the behaviors of advanced powertrains is required to use the model correctly. States and local governments doing emission inventory analyses will run the MOVES model. Does EPA anticipate that states and local governments will also be running PERE, and if so, what information will EPA provide to these users?

Response:

EPA does not expect state or local governments to run PERE. Nevertheless, the model will be readily available to all users to evaluate and use as they see fit. Support will be provided at some level.

• **Finally, the Alliance reiterates previous comments to the EPA to defer CO_2 modeling and fuel economy to the Department of Energy, DOT, and NHSTA.**
We are concerned with EPA's efforts to model fuel and energy consumption. We continue to recommend EPA defer CO_2 modeling and fuel economy to the Department of Energy, DOT, and NHSTA. (see previous comments to the EPA in March 4, 2003 and December 20, 2002 letters from Mr. Casimer Andary and Ms. Ellen Shapiro to John Koupal). We note that EPA has stated in its September 8, 2003 decision that the Clean Air Act does not authorize the regulation of emissions of CO_2 and other greenhouse gases from motor vehicles.

Response:

As stated in previous oral and written communications to the Alliance, MOVES, like MOBILE6, is designed to complement rather than compete with the national estimates prepared by the Department of Energy (DOE) of fuel consumption. Bottom up estimates of fuel consumption are essential not only for validation of MOVES2004 fleet and activity estimates, but also for the proper integration of GREET, for generating regional and local inventories, and for assessing the impacts on all pollutants in various changes in future technology mix. These are important functions that cannot be met with top-down modeling.

Appendix D- Comments from independent reviewer: Professor Marc Ross, including responses.

PERE Review 10/13/04 Marc Ross. These comments were made regarding the previous version of this paper "Advanced Technology Vehicle Modeling in PERE" EPA # EPA420-D-04-002, March, 2004.

OVERVIEW

In my opinion, PERE is an excellent motor-vehicle fuel consumption model, which should prove practical and powerful in use. (I may be somewhat biased in this because in the 1990s I was involved – not for EPA - in several aspects of this general approach to modeling of motor vehicle fuel consumption and emissions.) The model is relatively simple, requiring relatively few inputs, and yet rather accurate over a wide range of vehicles and driving patterns. The major strength of the model is its basis in physical principles, which are responsible for its generality and simpledescription. These principles are adequately approximated, over many vehicles and driving patterns, by simple algebraic expressions. Most of the parameters in these expressions are known for any type of light-duty vehicle, or are essentially constant for a given technology. This report clearly presents the model, and discusses the assumptions made, and relates them to many other referenced studies and measurements.

Of course the model requires appropriate validation in terms of measurements. But this is not primarily a statistical model. While the approximations and their parameterization need justification, *it is essentially a matter of physics rather than of complete coverage. Wherever an application involves different physics, then appropriate measurements are needed*. That means that since it and similar models have been validated for lighter light-duty vehicles, extensive data on the heavier light-duty vehicles is almost certainly not needed. However, more measurements may be needed on very heavy vehicles. (I have not carefully reviewed the heavy-duty, section VI - pp24 to 40.) Although the success of this type of modeling is encouraging, it does not mean PERE will necessarily be satisfactory in all applications in support of MOVES, unless there are additional measurements and analysis, as I briefly discuss further below.

In addition, there are some particular conditions and new technologies which are important, and which one knows are not adequately described by the model in its present form, i.e. in the absence of further measurements and analysis. Cold start, already addressed in part in the report under review, is, as stated, one such area. A serious beginning has been made on hybrid vehicles, but hybrids are likely to be a diverse class, and too few vehicles have been available for measurement to adequately implement PERE in that respect. Fuel cell vehicles represent another new technology where even fewer relevant specifications and measurements are available. But it is likely, if not certain, that this general modeling approach in its simple form will prove adequate in that case as well. In addition, criteria pollutants have been successfully addressed elsewhere using this kind of modeling, so I would be hopeful that PERE can contribute effectively to the MOVES program in that area as well. (It is not clear, however, whether particulate emissions, especially distributions in size and composition - which might in the future become an important regulatory concern, can be modeled in this simple a fashion.)

I applaud the position taken here that 5% variability, or even 10%, can be a very useful level of accuracy. Elaborate models with many more parameters, and elaborate measurement programs, often prove to be no more reliable *in practice*.

SIGNIFICANT SPECIFIC COMMENTS/QUESTIONS

1) Fill data holes (p5, p71): MOVES empirical basis, on-board measurements in random driving, may prove inappropriate in particular cases because: driving or weather conditions different from those sampled are (or become) of interest in applying MOVES. Comparison of predictions by PERE with the statistical results in MOVES in MOVES may also prove useful in unanticipated ways. In addition, PERE could offer alternative ways, as time goes on, to up-date information for MOVES.

RESPONSE: Good point. PERE could be used to isolate certain physical effects to study their impact on emissions and fuel consumption separately. Such a statements is added in this section.

2) I agree that application of PERE to advanced technologies (p5, 77ff) may not, in most cases, require a large accompanying measurement program.

3) Engine friction dependence on N (p11, motorcycle comparison p19, and Fig. 32 p59): In the figure, p11, N appears to go to 6000 rpm. Ignoring the details of low-N dependence, is there a significant difference between linear and quadratic to 6000 rpm? Indeed, a casual look suggests the data might fit a quadratic (a + bN^2) better than the linear dependence used, and have the further advantage of being similar to the motorcycle dependence.

Also the statement: "end result in the model (of assuming increasing friction vs assuming poorer indicated efficiency as N decreases) is identical" needs a brief discussion. The two assumptions are "identical", as stated, unless one considers applying the model to analysis of technologies focused on saving fuel at low N.

RESPONSE: Yes, a quadratic fit may also work. If this is advantages, it may be changed in the future. Regarding the 2nd comment, the sentence has been clarified. It is true that if a more detailed study is required for new engine technologies in this operating regime, the user would have to understand this fine distinction.

4) N/v)top is assumed to be a constant (p13). In my modeling work, I found N/v)top to be an important variable for predicting fuel rate. See Fig. 3 of my 1993 paper with Feng An (A Model of Fuel Economy and Driving Patterns, SAE 930328) where we used mass, displacement and N/v)top to predict the fuel intensity of many cars. On the other hand, since N/v is related to displacement, if somewhat roughly, the fuel rate may be insensitive to N/v, as such, in an empirical approach. [I fit N/v)top roughly to SQRT(N).]

RESPONSE: The sensitivity study in the paper confirms that N/v is an important variable for fuel consumption modeling. Bob Giannelli came up with some relation of N/v with displacement, which we used more for heavy duty applications.

5) "PERE is less sensitive to the specifics of the transmission model" aside from the transmission's efficiency. (See also bottom p99.) Good. This important insight had escaped me when I was working in this area. Of course the fuel rate still depends on N via the engine friction term.

RESPONSE: Of course, it is also sensitive to N/v as mentioned above. I added the parenthetical, that this generalization is for light duty. We found motorcycles (for example) to be quite sensitive to transmissions parameters (shift points, gear ratios, etc).

6) Top two paragraphs p 42: Is it really possible, as claimed, to improve efficiency substantially at low load by going lean without producing much NOx? I once estimated the improvement in fuel intensity by going lean (Eqn. 4b of my Contemporary Physics paper). That estimate suggests a smaller benefit; e.g. a reduction of 14% in second-by-second fuel rate if the fuel-air ratio is reduced by a highly lean one-half. But the dependence on fuel-air ratio in my expression may be greatly over-simplified.
See Thomas & Ross for an analysis of an effective threshold for engine-out NOx emissions. We found that threshold to be a fuel rate of 0.5 g/s for an early 1990s composite car. That implies, in the moderate driving of bag 2, most of the time very little NOx is emitted engine-out. In turn that suggests going lean in the bag 2 driving regime would not result in much engine-out NOx. However in more vigorous driving there is much engine-out NOx. So going lean will save fuel and not cause high NOx, but amount of low-power driving may be small, so the efficiency improvement may be small in practice.
RESPONSE: NOx control is definitely more of an issue with Tier2 regs. There is some concern over the lean engine model using parameters straight from the Honda paper. For example, improvement in peak efficiency already includes improvements in friction. This comment has been added to the paper. However, even with these liberal assumptions, the Honda Insight tests show that the system is even more efficient than the model is predicting during engine transients! This is likely due to hybrid strategy differences.

7) Hybrid vehicles (pp45, 46): If the "hybrid threshold" is taken to be higher (i.e. the control calls for somewhat more all-electric driving and some charging of the battery by the engine beyond that associated with regeneration), I believe the cycle fuel use would be further reduced. By how much would depend on regeneration efficiency and other characteristics, in line with the comment at the top of p47. One is likely to want a more detailed model for hybrids than we have here, as more-diverse examples of the technology become available. See my comment (10).
RESPONSE: True, but as the model becomes more and more complex, it evolves closer to a "design" model, and loses the relative simplicity that gives PERE its advantages.

8) Battery loss (long paragraph p47) and battery ratings paragraph p 47: Does this mean PERE tracks SOC over the cycle? On p 47, it says "this number does not affect the model" (for the cycles considered?). How great a variation in SOC occurs for the two vehicles and cycles considered here? Would the lowest SOC reached significantly affect battery efficiency? Would adding a moderate hill kill the SOC? Or are such issues not addressed in the current version of PERE?
RESPONSE: Yes, one can track SOC on the spreadsheet, but since the battery model is so limited, the results have room for improvement. A simple battery model could be added in the future if needed.

9) Accessories, p49: As stated, big accessory loads, especially A/C, would make a lot of difference. Does EPA have a dynamometer test for the mpg of hybrid cars which are expected to have A/C? If so, how is it done?

RESPONSE: This is beyond the scope of this report, though it is an important topic.

10) Big paragraph p52: There's a significant difference with PERE over-predicting Civic Hybrid city mpg by 9% and under-predicting Prius city mpg by 12 or 18%. Can you speculate as to a possible *specific* explanation? See my comment (7).

RESPONSE: As mentioned in the paper, and earlier, the Prius strategy is VERY different from the PERE hybrid strategy. The Prius is a series parallel hybrid and regenerates the engine during operation, PERE cannot do this currently. So the comparisons, should be taken with a grain of salt since these are essentially two different vehicles. The only things they have in common, are vehicle characteristics (weight, etc) and engine, motor sizing. The fact that the results differ should be of no great surprise. The Honda hybrids, however, are parallel hybrids. Even still, there are differences in the strategies, since the PERE strategy is extremely simplified. Also, note in the results that the Insight does not shut off the engine completely during most short idles. The Civic is likely to have a similar strategy. This would account for some of the overprediction of fuel economy. This comment has been added to the paper.

11) bottom p64, top p65, sensitivity analysis, also p 98: This sensitivity analysis is valuable for anyone who studies the model with at least some care; but it is likely to be misleading for others; because a few variables commonly differ from the assumed values by much more than 10%, while most others will differ by much less. Examples of highly variable variables are (1 – epsilon) where epsilon is transmission efficiency, N/v, and perhaps k0 .

RESPONSE: Good point, another statement of more variability has been added to the section.

12) pp 75-76 Has the accuracy of PERE been checked against measurement for both high and low speed/power driving, or only primarily for the standard city and highway cycle *totals*? I would expect PERE to be adequately accurate for moderate and high speed/power, but of uncertain accuracy at low speed/power, where the modeled fuel rate is lower relative to that in the standard cycles and the physical basis for the model, as is, is poorer. As shown here (and is known), the model, as is, is not realistic in coasting/braking modes; so one must be concerned if there are MOVES applications where coasting/braking is an important part of the whole of fuel consumption.

RESPONSE: Aside from the two modal tests presented in the paper, there is little data for a systematic study during different modes of driving. As more data becomes available, the model can be improved.

MINOR ISSUES

- Rotating mass term (p7). Jimenez' number seems high for high gears. Again the issue is not important for general driving cycles, but may be significant for special cycles

RESPONSE: Jimenez provides two references for this in his thesis: Bauer, 1996 (Bosch Handbook) and Emmelman and Hucho, "Performance of Cars and Light Trucks. Aerodynamics of Road Vehicles," SAE,1998). However this value is not entirely consistent with other works I've referenced in the paper. Even when I changed this value significantly (more than 10%) the effect on overall fuel consumption was minimal.

- Frontal area approximation (p7). I have used 0.83*height*width for cars.

RESPONSE: I also subtract the clearance height, could that explain the discrepancy?

- Units for figures 4 and 9 (p12 & 23). I assume the units are the same as on p46.
RESPONSE: The units are in the key box.

- It's important to be clear with terminology on diesel efficiency (pp21-23). We are talking about fuel energy, not fuel volume.
RESPONSE: I only deal with volumes when calculating fuel economy (at the end of all the calculations).

- Table 12, p33, * and ** are not defined.
RESPONSE: This has been clarified.

- Fig. 16 caption says "red line". What does that refer to?
RESPONSE: This has been clarified.

- 1st paragraph p 44: I think it can be claimed that all the hybrids listed use some regenerative braking. I think they don't all use electric motors for traction.
RESPONSE: This has been clarified.

- p54: Does "body type" simply refer to determination of CDA, or are there other uses of this info?
RESPONSE: This has been clarified.

- 11) Fig. 29, p56 (Insight). I must have missed where it says fuel is turned off during decels in PERE. The measured fuel rate in decels in the example here is substantial.
RESPONSE: This has been clarified in the section describing hybrid strategy (fig 22). Indeed for the Insight, the vehicle is even more efficient during accelerations than PERE would predict (even using a relatively generous lean-burn engine model).

- p57: What do you think is the reason for the diversion between calculation and measurement beyond 400s for Prius?
RESPONSE: As the paper mentions, the model is not entirely appropriate to use to model the Prius. This is because the Prius architecture is significantly different from the PERE parallel hybrid architecture. For example, the Prius recharges during driving, which is a strategy that PERE is unable to handle at the present. However, given these vast differences, the model seems to perform reasonably well on an aggregate level. Of course, further validations should be performed on more driving schedules (which will be done as more data is available).

- page 58: Ford gaseous H2 and probably soon also BMW liquid H2.
RESPONSE: This has been added.

- p59 bottom: Say: vehicle traction applications.
RESPONSE: This has been clarified.

- p50: Fuel cell efficiencies are relatively high at low load (although not at zero load), like a battery. What is the shape of the system efficiency between 0 and 4 kW in Fig. 34? Nelson's estimate (p63) indeed shows "very low" efficiency up to 4 kW, but I wonder if that is general, or the result of some very particular assumptions Nelson made.
RESPONSE: These low values are approximate.

Additional Marked up Comments (on paper)
- mid p16 "curb" weight
- mid p 42 1.20x0.40 = 0.48?
- top p 44 You say Many of the hybrids.. Doesn't that mean all those mentioned?
- p42 & 43 figure is repeated
- mid p 52 Honda "Civic" Hybrid
- fig 29 bot (p 56) not clear where SEMTECH line is in my copy
- bot p 59 veh "traction" app
- mid p 60 Fuel cell inefficiencies at low load are not obvious in Fig 34,
but are in Fig 36
- 2nd line from bottom p64: What about transmision efficiency?
- mid p 67 vehicles which have not yet been produced omit comma
- p 75 I may not have commented clearly enough in my review of PERE that the
model has not been adequately evaluated in the low speed cycles.
- pp87&88 I appreciate the citations, which I did not kmow, for the Bishop
and Kluger, and also for a couple of other papers.
- p89 I dont think the Heywood book has this date. My copy is 1988.
- p89 Nitkin is spelled wrong once.
- p96 3rd line above the solid line Omit the redundant "However".

RESPONSE: All of these comments have been addressed in the appropriate section.