

Analytic Model of Powertrain Drive Cycle Efficiency, with Application to the US New Vehicle Fleet

Patrick Philips
Ford Motor Company

ABSTRACT

An analytic model of powertrain efficiency on a drive cycle was developed and evaluated using hundreds of cars and trucks from the US EPA 'Test Car Lists'. The efficiency properties of naturally aspirated and downsized turbocharged engines were compared for vehicles with automatic transmissions on the US cycles.

The resulting powertrain cycle efficiency model is proportional to the powertrain marginal energy conversion efficiency K , which is also its upper limit. It decreases as the powertrain matching parameters, the displacement-to-mass ratio (D/M) and the gearing ratio (n/V), increase.

The inputs are the powertrain fuel consumption, the vehicle road load, and the cycle work requirement. They could be modeled simply with only minor approximations through the use of absolute inputs and outputs, and systematic use of scaling.

On the Highway test, conventional automatic transmission vehicles of moderate performance achieve between 25% and 30% powertrain efficiency. On the City test, the efficiency is reduced to 15% to 20% because of the higher fraction of open torque converter launches and power-off operation.

Compared to naturally aspirated engines, turbocharged and downsized engines are a trade-off between reduced losses and reduced efficiency. When the losses are comparatively high, as in the City test or in high D/M vehicles, the benefit is substantial. But when the losses are comparatively low, as in the Highway test or in low D/M vehicles, the benefit is small.

CITATION: Philips, P., "Analytic Model of Powertrain Drive Cycle Efficiency, with Application to the US New Vehicle Fleet," *SAE Int. J. Fuels Lubr.* 9(1):2016, doi:10.4271/2016-01-0902.

INTRODUCTION

As vehicle fuel consumption is a topic of continuing interest, not only to vehicle users, but also to the authorities that regulate it out of environmental and economic concerns, there is a need to improve understanding of how vehicles use fuel. Simple metrics and models can be of great help, by clarifying the primary factors involved, and by illustrating the status as well as any limitations that might apply.

Ton-MPG is a simple metric of vehicle transport efficiency on the EPA Metro-Highway cycles. The progress of the US new vehicle fleet in Ton-MPG is updated every year by the EPA [1]. Figure 1 shows that, per unit of mass, efficiency has doubled over a forty year period. This metric incorporates both improvements made to the vehicle to reduce the required engine work, and improvements to the powertrain to raise its efficiency. The metric is useful because fuel consumption is roughly proportional to vehicle weight for a given technology level. Ton-MPG thus provides a metric for the efficiency of a fleet at a point in time, across the range of weights of the vehicles.

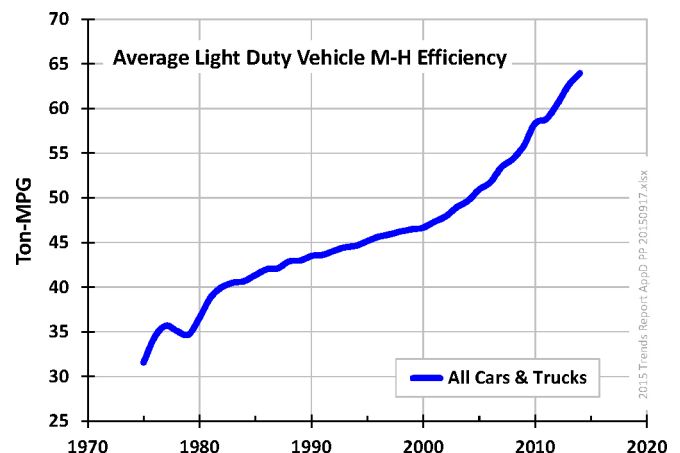


Figure 1. Evolution of the average efficiency of the US new vehicle fleet on the EPA Metro-Highway tests.

But Ton-MPG metric has some limitations. In these traditional units it is not possible to estimate the actual efficiency of the cycle efficiency of the vehicle, which cannot be directly compared to

known peak powertrain efficiencies, so that it is not possible to estimate whether the rapid recent progress seen in [figure 1](#) can be continued, or whether it is approaching fundamental limits. The Ton-MPG metric also does not distinguish between vehicle and powertrain aspects or provide any relation to other vehicle or powertrain properties.

The primary use of simple fuel consumption models is to predict future improvements in regulated fuel consumption. In daily industry practice, the combined effect of a few small changes to a vehicle can initially be estimated by simply adding their individual benefits. This has led to the ‘accounting’ approach to fuel economy, where for initial planning purposes benefits, costs, and weights are just added up. This practice is acceptable as long as both the individual and the combined changes are small, and provided the estimates are confirmed by detailed physical modeling and testing.

In the case of regulatory forecasts to justify large improvements over long periods of time, these assumptions are not met, and the accounting approach has been shown to be unsuitable due to multiple pitfalls [2]. As a result, attempts were made to improve the accounting approach with ‘lumped parameter’ models [3, 4]. Here the energy use of a base vehicle is broken down to a discrete set of causes, the lumped parameters, and a corresponding fraction of fuel is assigned to each ‘bin’. The benefit of each technology is similarly broken down to the discrete causes or bins. The claimed benefit of the lumped parameter approach is that it will prevent oversubscription of any one bin when applying multiple technologies attacking the same problem.

The lumped parameter models do not appear to have been extensively documented, so it is not clear what relations were used between the terms. Nevertheless, what can be said is that the binning of effects only represents an upper bound on the potential benefit of combined technologies. As the lumped parameter models do not describe the physical effects, they cannot estimate the actual physical interactions.

What is needed is the simplest possible physically correct representation of the physical phenomena involved [5,6,7,8], and their relation to each other. Even then, with physically sound models, long term predictions will remain fraught with uncertainties, as future technologies are by definition not fully understood, nor are future external changes such as shifts in the market preferences, fuel prices or emissions requirements.

An analytic model of powertrain fuel consumption was developed previously [8], and was used to estimate statistically a high level vehicle model of fuel consumption by technology type. Here the same approach will be used to estimate vehicle cycle work based on known vehicle and cycle properties. By taking the ratio of the cycle work to the fuel usage a simple model of the powertrain cycle efficiency is obtained. The model is applied to the recent US new vehicle fleet as documented in the EPA ‘Test Car Lists’, and the cycle powertrain efficiency of vehicles with downsized turbocharged (TC) engines is compared to that of vehicles with naturally aspirated (NA) engines.

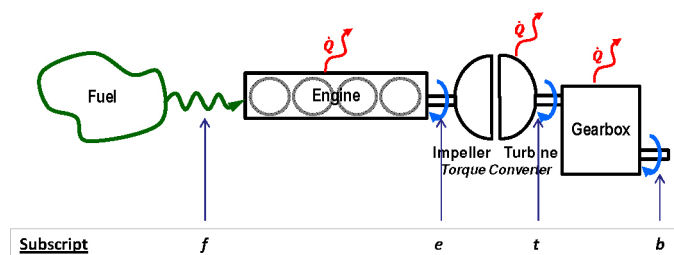


Figure 2. Fuel is provided to a system consisting of an engine, a torque converter and a gearbox. The output of the gearbox is the power required by the vehicle. Subscripts are used to differentiate different stages of the process.

POWERTRAIN MODEL

The discussion of the powertrain model is largely a summary of [8]. The system being considered is shown in [figure 2](#). Fuel energy is provided to the engine at a finite rate, mechanical power leaves the engine, to go through a torque converter and then a gearbox before driving the car at the wheels.

Engine Fuel Consumption Model

The fuel consumption of a conventional automotive naturally aspirated spark ignition engine is shown in [figure 3](#) in load sweeps at different engine speeds. The traditional metric of brake specific fuel consumption (bsfc), the mass of fuel used per unit of work delivered, is shown as a function of engine load in the most used speed range from 1,500 to 2,500 rpm.

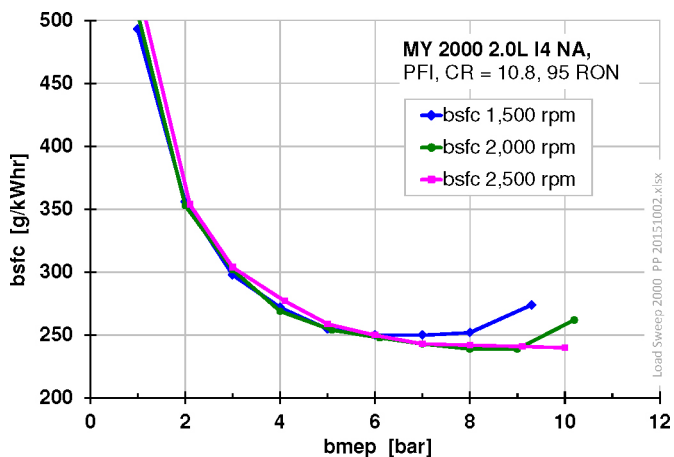


Figure 3. Brake specific fuel consumption (bsfc) of an automotive 2L PFI NA SI engine as a function of load (bmep) over a range of typical engine speeds.

The engine is a model year 2000, 2 liter four cylinder inline, with 4 valves per cylinder, dual overhead cams, port fuel injection, and a central spark plug, with a rated output of 107kW at 6,000 rpm [9].

The bsfc curves show the typical ‘fish hook’ shape as a function of brake mean effective pressure (bmep), which is the torque of the engine normalized for its displacement. The fuel consumption behavior does not change appreciably with engine speed at the lighter loads. At the highest loads the bsfc is degraded at low rpm due to knock sensitivity, but this is nearly eliminated as the engine speed increases to 2,500 rpm.

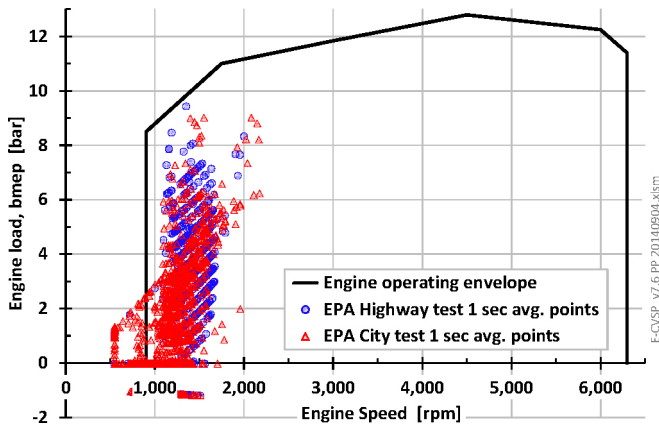


Figure 4. Second by second operating points for a naturally aspirated engine with an eight speed planetary automatic transmission on the EPA City and Highway tests.

Figure 4 shows the distribution of engine speed/load points over the EPA drive cycles for a vehicle with an eight speed automatic transmission. The operating points are seen to be concentrated in the low speed and low to mid load ranges. Advanced automatic transmissions minimize fuel consumption by keeping engine speed low to raise the engine load, which reduces fuel consumption as shown in figure 3. The high loads (>8 bar bmeq) are only rarely used on the EPA City and Highway test cycles.

Looking at the same operating points as a trace through their time sequence, figure 5, reveals that the primary variation in engine operating points while installed in a vehicle with an automatic transmission is in the load direction (vertical in the plot), and not in engine speed.

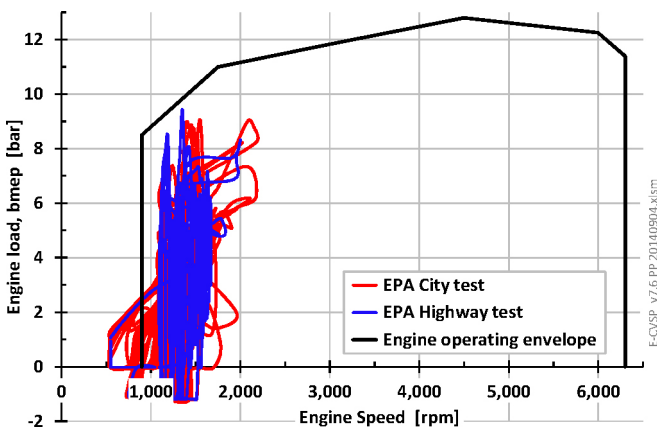


Figure 5. Transient trace of operating points for a naturally aspirated engine with an eight speed planetary automatic transmission on the EPA City and Highway tests.

The load sweeps are then not only a convenient description of the data, they show the typical ‘movement’ of the operating points in vehicle operation. However, load changes at approximately constant speed necessarily imply output work changes, and therefore the representation of fuel consumption *per unit work*, as in figure 3, may not be the most direct or useful. In particular, there is no obvious way to model figure 3 simply, nor does it indicate that an engine works in an approximately linear manner.

A practical metric expresses the fuel use in terms of energy use, and normalizes it in the same way the output of the engine is normalized. The fuel *energy* flow rate, P_f , is defined as the fuel mass flow rate multiplied by the lower heating value of the fuel, H_f ,

$$P_f = H_f \dot{m}_f = \frac{1}{2} D n \text{ fumeq} \quad (1)$$

This fuel ‘power’ is normalized for displacement D and engine speed n , to define a ‘fuel mean effective pressure’, *fumeq*, the factor $\frac{1}{2}$ resulting from four stroke operation. One can convert between the different ways of expressing engine efficiency, bsfc, efficiency and fumeq using

$$\text{fumeq} = H_f \text{ bmeq bsfc} / 3600 = \text{bmeq} / \eta \quad (2)$$

The same engine data is re-plotted in figure 6 as absolute inputs and outputs normalized per unit displacement, and normalized per engine cycle instead of per unit time, i.e. as *fumeq* (bmeq).

Now the measurements at the three engine speeds collapse onto a single straight line at part load, there is no singularity or zero at zero load, and the results are independent of the heating value of the fuel. The data in figure 6 can be summarized by a simple linear regression, covering the low speed range and the load range from 0 to 7 bar. This means that nearly the whole engine operating range used in the EPA test cycles (fig.4 & 5) is very well described by a linear model, a very powerful enabler for an analytic model. The continuous linear behavior always extends down to zero load, although the zero load data are less commonly found. Then as the load becomes negative, fuel is cut off, firing is interrupted, and the engine is spun as a resistance.

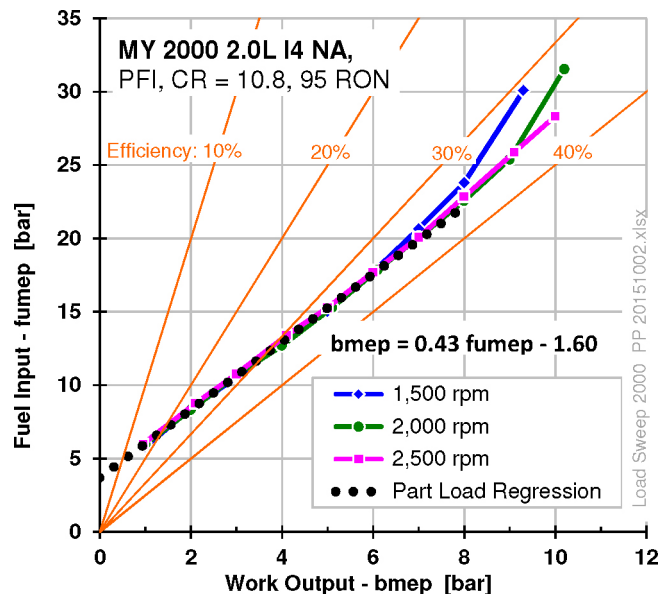


Figure 6. Normalized fuel input per cycle (fumeq) of an automotive 2L PFI NA SI engine as a function of normalized load (bmeq) over a range of typical engine speeds. Same engine data as in figure 3.

As the figure shows absolute work output versus absolute fuel input, lines of constant efficiency are radial straight lines from the origin, marked with xx%. The efficiency at any one load can therefore also be qualitatively read from these figures.

The linear relation observed between fuel input and work output was shown in [8] to be fundamental to the way an engine works. The relation may be written

$$bmep = \mu fume_p - lme_p$$

or

$$fume_p = (bmep + lme_p) / \mu \quad (3)$$

where μ is called the marginal efficiency because it is the slope of the relation between input and output, and lme_p is the offset term, or loss mean effective pressure.

The two forms of Eq. (3) are mathematically equivalent, but reflect two different perspectives. The upper version shows the perspective of the process flow, of converting fuel to shaft power. It can be put into words as the fuel energy $fume_p$ is transformed into work on the piston with an efficiency μ , and then mechanical losses lme_p are subtracted to come to a net output work $bmep$. The lower version takes the engineering perspective and expresses how much fuel $fume_p$ it takes to generate an output $bmep$. It says that to the output $bmep$ one needs to add the mechanical losses lme_p , and then divide the sum by the effective combustion process efficiency μ to find the required fuel energy $fume_p$.

At a high level this is the same approach as taken in [2, 5]. The engine data in figure 6 show $\mu = 0.43$, whereas the indicated efficiency of such an engine would be expected to lie in the range 37-39% [2]. More detailed analysis [8] shows that μ is higher than indicated combustion efficiency because the pumping losses in SI engines fall off gradually from light loads to near zero at high loads. The observed marginal efficiency μ is the gross indicated efficiency plus the rate of pumping loss decrease as engine load increases. The loss mean effective pressure lme_p is primarily related to the mechanical losses in the engine, the friction plus the pumping at zero indicated load, with a smaller contribution from the fuel that burns late or not at all.

For a boosted engine, the evolution of fuel mep as a function of bmep is shown in figure 7 for a turbocharged PFI SI engine of the same period and technology level as the naturally aspirated engine of figures 3 and 6. In the load range of 0 to 10 bar, the behavior of the two engines is very similar qualitatively, but the marginal efficiency of the turbocharged engine is substantially lower because of the lower compression ratio.

Koffler and Rohde-Brandenburger [7, 10] found that over a range of typical current engines, the marginal efficiency drops from an average of 0.42 for NA engines to 0.40 for TC engines, a 5% penalty. The loss mep of the turbocharged engine is also higher, because the higher loads on the components lead to stronger, heavier components and

larger bearings. Note that the turbocharged engine in figure 7 was run on higher octane fuel (98 RON) than the naturally aspirated engine of figure 6 (95 RON). If the turbo engine had been run on 95 RON fuel, knock would have set in at a lower load, reducing peak efficiency, and further deteriorating fuel consumption at high loads.

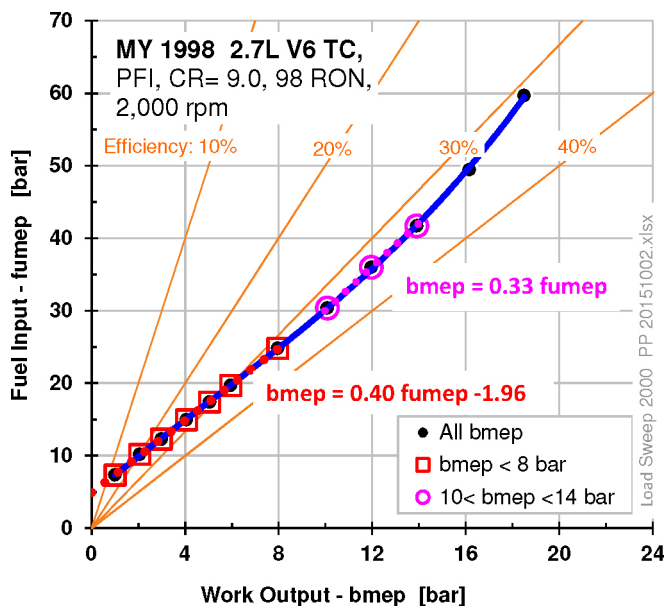


Figure 7. Normalized fuel input ($fume_p$) as a function of output ($bmep$) at 2,000 rpm for a 1998 PFI turbocharged V6 engine running on 98 RON fuel. Regressions at part load and at peak efficiency are shown as dotted lines.

The drop in fuel conversion efficiency at the high loads can be clearly seen, with the consequence that the model (5) would be too optimistic at the high loads. Fortunately, an even simpler model provides a good approximation, namely that of constant, maximum brake efficiency,

$$bmep = \eta_M fume_p \quad (4)$$

where η_M is the engine maximum brake efficiency. This model provides a very good approximation of the data in figure 7 for the load range from 10 to 14 bar, and transitions seamlessly to Eq. (3) at lighter loads. For this older PFI turbocharged engine, the peak brake efficiency is 33% over a wide load range. In this range, the indicated efficiency is falling off at an increasing rate with increasing load; but the friction losses become a smaller fraction of the total fuel used, so that the overall efficiency is approximately constant.

At higher loads still, the engine brake thermal efficiency starts falling off. The engine is becoming increasingly knock limited, forcing ignition to be retarded to avoid damage and leading to deteriorating efficiency and elevated exhaust temperatures. The high load behavior of turbocharged engines is a three way balance between desired maximum bmep, fuel anti-knock quality (RON) and compression ratio. In this example, with Port Fuel Injection good efficiency could be preserved to the full load of 18 bar through the reduction of compression ratio to 9:1 as well as the use of premium 98 RON fuel. Running this engine on US 91 RON regular fuel would have caused the efficiency to start dropping off at lower loads, leading to lower peak efficiency and unacceptable performance at full load.

Gearbox Efficiency Model

The transmission model is also formulated to bring out scaling factors, and to enable comparisons between transmissions of different sizes. The obvious transmission 'size' parameter is the torque capacity. An ideal gearbox converts speed to torque while conserving power,

$$P_b \approx P_t \quad (5)$$

where the subscripts refer to the stage of the overall process, 'b' for gearbox output and 't' for torque converter output, as defined in figure 2. The analysis is therefore done in terms of power, normalized by a torque times a speed. The normalizing torque is the engine torque capacity of the transmission, T_{cap} , and the normalizing speed is the gearbox input speed, ω_i .

The real gearbox cannot convert speed to torque perfectly. The power flowing through the gearbox does so with an efficiency gamma, γ close to but smaller than one. In addition, a real gearbox incurs 'overhead' losses through friction in bearings, seals, pumps, etc. The overhead loss torque is a small fraction of the torque capacity of the gearbox, but nearly independent of the power flowing through the gearbox. It is therefore modeled as a separate additive term,

$$P_b / T_{cap} \omega_i = \gamma (P_t / T_{cap} \omega_t) - \phi$$

with $\gamma \approx 1 < 1$, $0 < \phi \ll 1$ (6)

where ϕ is a small, non-dimensional loss coefficient, the normalized 'spin loss' of the transmission, as discussed in more detail in [8]. As in the case of the engine, we have formulated a linear model of gearbox output power in terms of gearbox input power.

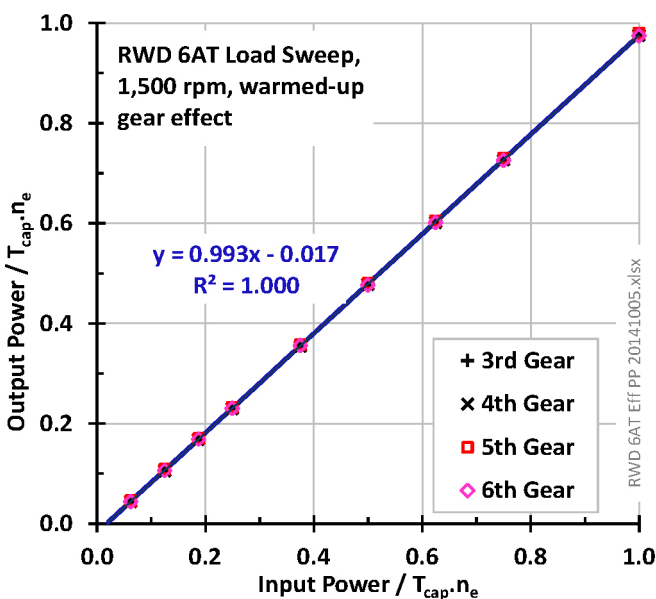


Figure 8. Effect of gear on normalized output versus input power for a six speed rear wheel drive automatic transmission.

The model Eq. (6) is demonstrated using efficiency mapping data from a rear wheel drive six speed automatic transmission, for different gears in figure 8, and for different engine speeds in figure 9. The model is seen to provide an excellent description of the data across the most used gears, and across the most used engine speeds. By enabling a description in a single linear relation Eq. (6), a drastic compression of the data is achieved. In first and second gear the behavior is similar, but the slope is lower, reflecting a lower gearing efficiency as a result of the high gearing ratio.

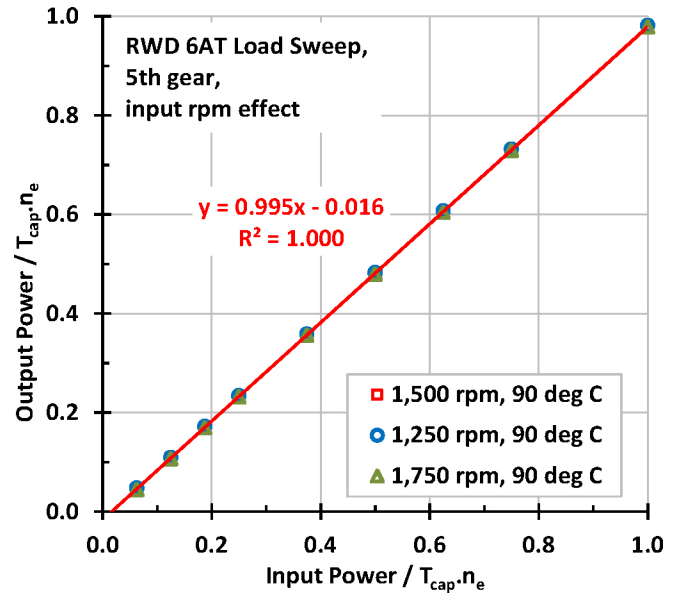


Figure 9. Effect of engine speed on normalized output versus input power for the same transmission as in figure 8.

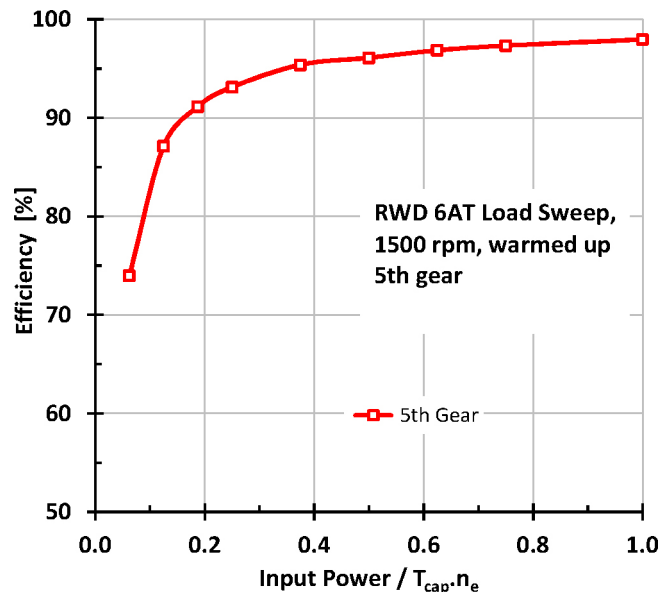


Figure 10. Efficiency of a rear wheel drive six speed automatic transmission in a load sweep at 1,500 rpm engine speed in 5th gear, torque converter locked.

Moreover, compared to the traditional description in terms of efficiency curves, figure 10, it is much easier to describe efficiency properties quantitatively. In [8] it was shown that Eq. (6) also accurately describes manual and CVT transmissions.

Complete Powertrain Model

As we have linear models for both the engine, Eq. (3), and the gearbox, Eq. (6), an expression for powertrain fuel consumption at part load can easily be found by combining them,

$$P_f = [P_b + \frac{1}{2} Lmep Dn_t] / \kappa \quad (7)$$

where κ is the combined energy conversion marginal efficiency of engine, torque converter and gearbox, and $Lmep$ is the mechanical loss mep for the combined powertrain. The model Eq. (7) is to first order in small coefficients only, meaning that products of small parameters (second order terms) are neglected. If the engine is operating in the best efficiency range, the required fuel power for the powertrain from Eq. (4) and Eq. (6) is simply

$$P_f = [P_b] / \gamma \eta_M \quad (8)$$

again to first order in small coefficients. Once the engine is operated in its best efficiency region, comparing Eq. (7) and Eq. (8) shows that reductions in speed or displacement will not directly reduce fuel consumption, but that improvements in peak efficiency are needed.

A hypothetical but realistic example of a complete powertrain consisting of a naturally aspirated SI engine with a six speed automatic transmission with a locked torque converter at part load is discussed in figure 11. The naturally aspirated engine examples discussed earlier have marginal efficiencies in the low forty percent, and a low speed BMEP of a little more than 10 bar. The automatic transmissions were found to have marginal efficiencies in the low to high nineties. The combined marginal efficiency will therefore be in the neighborhood of 40%. The loss term for engines was found to be typically $l_{mep} \sim 1.6$ bar, to which need to be added a few tenths for accessory loads and another few tenths for transmission friction, for a total of about 2 bar. If the powertrain uses a turbocharged engine with a lower compression ratio, the marginal efficiency would typically be lower by about two percent points [7, 8], and the mechanical loss term increased by several kPa. If instead of dealing with a locked torque converter there is an unlocked one, the marginal efficiency will drop by ten or more percent relative to the locked case.

Two load cases are considered, one at 2 bar and one at 6 bar mep. The first case might correspond to a vehicle on the lightly loaded European test cycle, with a manual transmission that keeps the engine speed higher, and therefore the bmep lower. The second case corresponds to a more highly loaded situation, with an advanced automatic transmission keeping the engine speed lower, and the bmep higher.

In the conventional view, once the load is known, the efficiency or bsfc can be looked up or interpolated from tables of data, as shown in figure 11(a). In this example, the total powertrain efficiency at 2 bar is 20%, and at 6 bar it is 30%.

Because the model presented here makes use of all the data over the part load range, some additional conclusions can be drawn in figure 11(b). As discussed above, this powertrain has internal losses of 2 bar l_{mep} , and a marginal efficiency of 0.40. From this one can infer that $2/0.4 = 5$ bar fumeq is required to run the powertrain at zero output. Every additional unit of net output then requires $1/0.4 = 2.5$ bar of fumeq. At 2 bar output, additional 5 bar of fumeq are needed, and at 6 bar, additional 15 bar of fumeq. Figure 11(b) and equation (7) show that it is straightforward to split the fraction of the fuel used for running the powertrain itself and the fuel needed to generate the output.

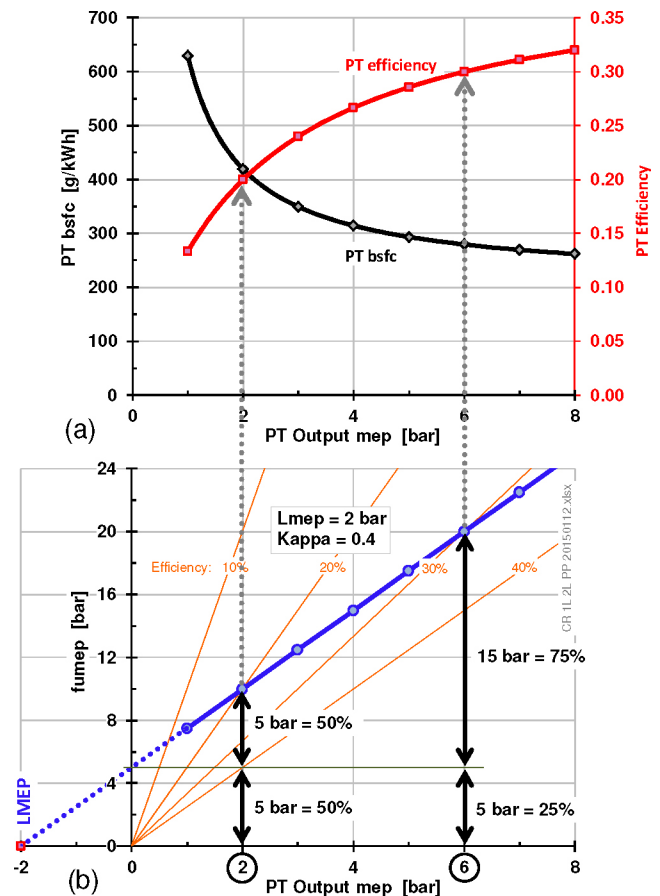


Figure 11. Comparison of powertrain level fuel use between (a) conventional bsfc and efficiency metrics, and (b) fuel mep for two operating points, at 2 and 6 bar.

At 2 bar output, and with internal losses of 2 bar l_{mep} , half the fuel is used to cover the internal losses and half goes toward generating the output. But as the energy conversion marginal efficiency is forty percent, the overall powertrain efficiency is $0.5 \times 40\% = 20\%$. Similarly at 6 bar, three quarters of the fuel is used to generate the output, so the overall efficiency is $0.75 \times 40\% = 30\%$.

The linear relation between work and fuel usage has been used by Rohde-Brandenburger [10] to discuss and quantify the different loads and losses present in the vehicle.

VEHICLE ROAD LOAD

The traction power P_t required to drive the vehicle is the sum of components for accelerating mass, overcoming road load resistance, and raising mass up a grade,

$$P_t = \alpha MV + RV + v g MV \quad (9)$$

where α is the acceleration rate, R is the road load resistance force, v is the grade, and g is the acceleration due to gravity. Government drive cycles specify a time-dependent velocity profile $V(t)$ and therefore acceleration profile $a(t)$ on a flat road, $v = 0$. When comparing different vehicles on a given drive cycle, this is a fixed and known profile.

As with the engine and transmission, opportunities to bring out scaling factors should be sought. In this case, the acceleration and grade power is proportional to vehicle mass, and the question is therefore whether the road load is also related to mass. This approach aligns with the way vehicle road load force is determined experimentally. The vehicle is allowed to coast down, the deceleration rate is measured and multiplied by the mass. The road load force R is split into the coast down deceleration rate $\delta_c(V)$ multiplied by the mass M ,

$$R = \delta_c(V) M \quad (10)$$

Conventionally the road load force is broken up into three terms in powers of vehicle speed. As the mass of any one vehicle is constant, we can do the same with the deceleration rate,

$$\delta_c(V) = \delta_0 + \delta_1 V + \delta_2 V^2 \quad (11)$$

An example for a 2013 compact car is shown in figure 12. The quadratic model is seen to describe the data very well. The constant and linear terms are caused by tire, axle and brake friction, the quadratic term by aerodynamic losses.

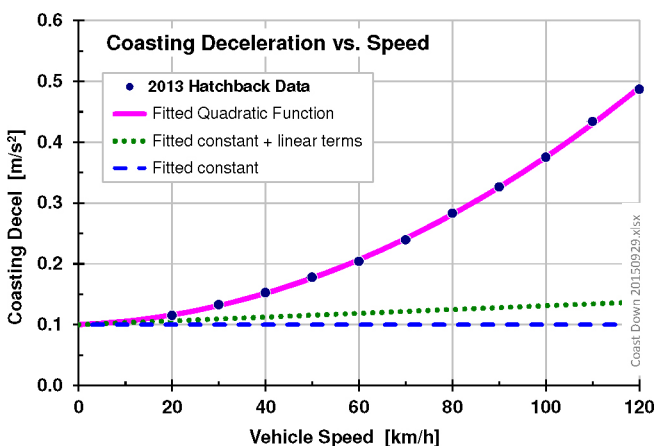


Figure 12. Fit of road load deceleration versus speed data for a compact car to a quadratic model. The constant and linear terms correspond to tire, axle and brake friction. The quadratic term results from aerodynamic drag.

The advantage of working in terms of coast down deceleration rate instead of deceleration force is that it is more normalized, so that all vehicles have very similar properties. This is shown in figure 13, where the deceleration rates observed in benchmarking tests of six very different vehicles are compared.

The vehicles range from a sports car and a luxury limousine with good aerodynamic properties and low cross sectional area in relation to the weight, to a compact hatchback and a panel van with larger cross section and less favorable aerodynamic shapes. The coast down deceleration curves of all these vehicles fall into a fairly narrow range.

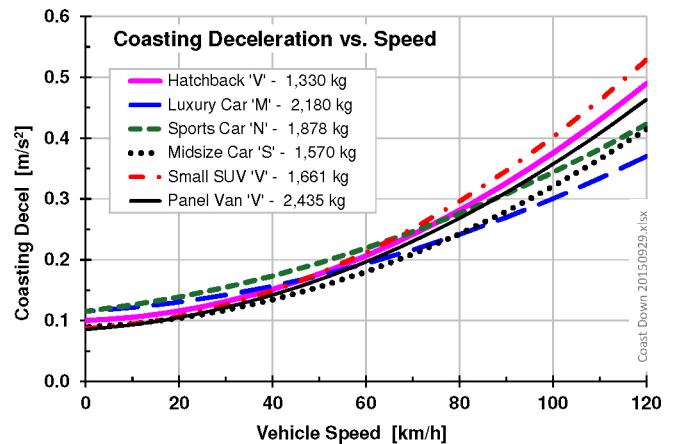


Figure 13. Coast down deceleration rates for a wide range of recent vehicles, from small car to large van.

The absolute road load power under steady driving on a flat road for these same vehicles is shown in figure 14.

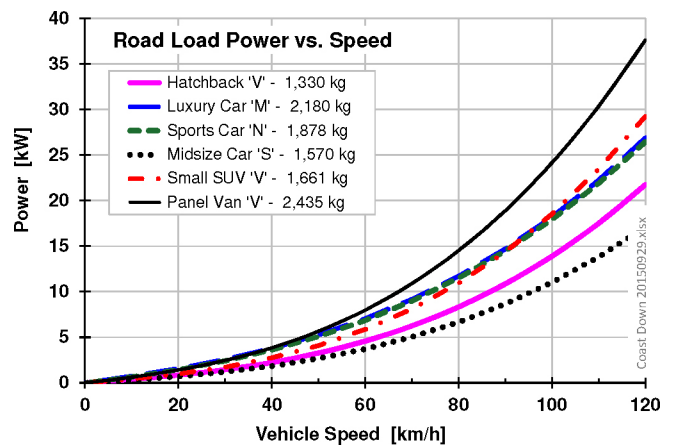


Figure 14. Absolute road load power for the same vehicles as in figure 13.

The wide range of power levels primarily reflects the range of vehicle weights, as can be seen by comparing with figure 15, which shows the road load power normalized for weight. The narrow range of power/weight ratios reflects the narrow range of coast down deceleration rates observed in figure 13.