Heavy-duty truck test cycles: combining driveability with realistic engine exercise

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Abstract: Heavy-duty engine certification testing uses a cycle that is scaled to the capabilities of each engine. As such, every engine should be equally challenged by the cycle's power demands. It would seem that a chassis cycle, similarly scaled to the capabilities of each vehicle, could successfully bridge the gap between 'universal driveability' and 'realistic engine exercise.' The purpose of this paper is to present just such a cycle.

Keywords: chassis cycle, driveability, heavy duty truck test cycles.

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Metric equivalents

Readers more familiar with the metric system may use the following equivalents to convert any quantities that appear in non-metric units:

1 mile = 1.609 km

1 brakehorsepower/hour = 0.7457 kWh

1 foot = 0.3048 m

1 pound (lb) = 0.4536 kg

1 HP = 745.7 W

Introduction

Dynamometer testing is a critical component of the US Environmental Protection Agency's (EPA's) strategy for assessing and regulating emissions from mobile sources. By US federal law, a vehicle cannot be sold in the USA unless it complies with the applicable emissions standards for that model year. Compliance is demonstrated by dynamometer testing, using equipment and procedures detailed in Title 40 Code of Federal Regulations Part 86 (Federal Register, 1991). Passenger cars and light duty trucks are certified, on a chassis dynamometer, to standards that are specified in grams/mile. The results of the certification test are reported to EPA, which uses the certification data for compliance assessment and future standards planning. These data are also used for modelling purposes, to estimate the contribution of mobile sources to emissions inventories.

Heavy-duty vehicles are not certified on a chassis dynamometer, and emissions are not reported in grams/mile. Because of the enormous variety of engine, drive train, and vehicle combinations that are sold in the heavy-duty vehicle marketplace, EPA's heavy-duty standards are applied to the engines rather than completed vehicles. Heavy-duty engines are certified, on an engine dynamometer, to standards that are specified in grams/brakehorsepower/hour. From a regulatory standpoint, a work-specific emissions unit is acceptable to compare engines and to eliminate those that over-emit. However, for many of the same reasons that chassis testing was impractical for certification (i.e., the variety of vehicles and drivetrains that can incorporate a given engine), results of engine testing are proving impractical for the calculation of emissions inventories.

Among the mobile source emissions measurement and modelling communities, there is somewhat of a consensus that chassis dynamometer testing could do a better job of characterizing heavy-duty vehicle emissions (Kitchen and Damico 1992; Ferguson *et al.*, 1992; Heirigs, and Caretto, 1997; Browning, 1998). Unfortunately, such a consensus does not extend to the dynamometer test procedures. In the real world, heavy-duty vehicles operate under a wide range of conditions, including just about any combination of payload weight, roadway topography, traffic flow, and weather. All of these parameters affect vehicle emissions; as such, an effective dynamometer test must simulate a representative set of conditions for each test vehicle. Payload weight dictates the total vehicle inertia, and it substantially impacts road load power demand (i.e., the power that is required to overcome all of the frictional forces that oppose the vehicle at a given speed). Roadway topography and traffic flow contribute to the 'duty cycle' aspects of the operating condition (i.e., the varying sequence of speeds, grades, accelerations, decelerations, and idle time that is required of the vehicle). Weather affects the road load power demand, but it also has a direct impact on vehicle emissions through its effect on the engine's combustion air (Krause *et al.*, 1973).

Chassis dynamometer facilities have a number of adjustable parameters that allow them to simulate the real world. Vehicle inertia is simulated by rotating flywheels, motor-generators, or some combination thereof. Road load power demand is simulated by motor-generators or some other form of power absorber. All of these inertia and power components work together, under a precise feedback control system, to simulate real-world reactions to driver inputs. A key difference between engine testing and chassis testing is the extent of driver input control. An engine dynamometer facility controls all inputs, including the engine throttle; a chassis dynamometer facility relies on an experienced driver to operate the test vehicle through a test cycle.

A test cycle is meant to be a representative duty cycle for the type of vehicle that is being tested. Usually consisting of a timed sequence of vehicle speeds, literally dozens of chassis test cycles have been developed for heavy duty vehicles. Many were developed for urban buses, but some claim to be equally applicable to trucks. All of the cycles fit into two general categories: A 'stylised' cycle, often taking on a geometric appearance, is a regular sequence of accelerations, constant speeds, and decelerations; and a 'realistic' cycle, appearing much more random in its sequencing, is often based on real-world data. Figure 1 shows examples of each type of cycle.

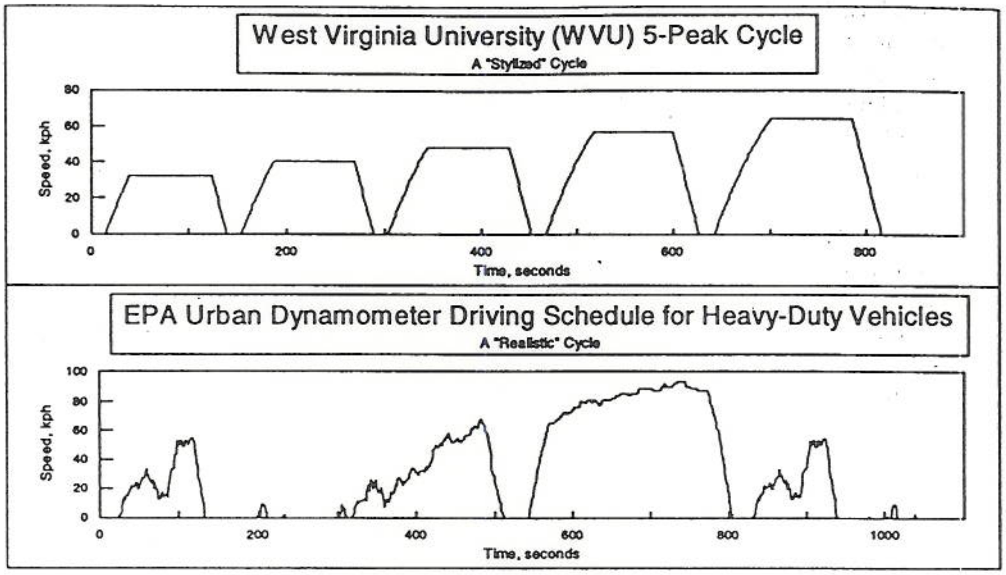

Figure 1 *Driving cycle examples*

Figure 1 also illustrates the two design philosophies that govern cycle development. The West Virginia University (WVU) 5-peak cycle (Clark *et al.*, 1994) is designed for 'universal driveability.' Its mild accelerations and low maximum speed are within the capabilities of buses, garbage trucks, and highway trucks alike. The EPA cycle (Federal Register, 1991) is designed for 'realistic engine exercise.' Its combination of idle, full throttle acceleration, rapid transitions, and low and high speed cruising are meant to represent the full range of vehicle/engine operating modes. Due to the wide variability of power-to-weight ratios in heavy-duty vehicles, any cycle that challenges the strongest and fastest vehicles will become too much of a challenge for slower/weaker ones. Many vehicles, particularly trucks with unsynchronized transmissions, cannot keep pace with the accelerations and decelerations that the EPA cycle specifies. Conversely, the WVU 5-peak cycle would challenge only the weakest or slowest vehicles, leaving the majority of vehicles to perform an unrealistically 'easy' cycle.

Heavy-duty engine certification testing uses a cycle that is scaled to the capabilities of each engine. As such, every engine should be equally challenged by the cycle's power demands. It would seem that a chassis cycle, similarly scaled to the capabilities of each vehicle, could successfully bridge the gap between 'universal driveability' and 'realistic engine exercise.' The purpose of this paper is to present just such a cycle.

Background

One of the first cycle development efforts for heavy-duty vehicles was carried out by US EPA in the mid-1970s. The CAPE-21 programme, a heavy-duty vehicle driving pattern and use survey jointly sponsored by EPA and the Coordinating Research Council (CRC), provided operating data from 95 vehicles performing their normal duties in New York City and the Los Angeles Basin. EPA used the CAPE-21 database as the reference population to produce thousands of candidate cycle segments by Monte Carlo simulation (Smith, 1978). Because these candidate cycle segments were actually derived from a sequence of computer-generated random numbers, it is highly unlikely that any of them corresponded to the actual real time data from any of the test vehicles. Given the fact that each cycle segment consists of several hundred records, each of which is the function of a random number, it is likely that some cycle segments represent operating sequences that have never actually taken place in the real world. Nonetheless, all of the candidate cycle segments were filtered by comparing a number of overall statistics (e.g., average vehicle speed, percent time idling) between the CAPE-21 database and the candidate cycle segments. From among the cycle segments that passed the statistical filter, the final composite cycles were selected and sequenced based on engineering judgment and predetermined goals regarding 'balance' (NY versus LA, freeway versus non-freeway) (Wysor and France 1978; France 1978).

Because the CAPE-21 database contained both vehicle and engine data, it was used to generate both chassis and engine test cycles. The chassis cycle, often referred to as Schedule-d because of its heading in the CFR, is a 1060 second sequence of vehicle speeds (Federal Register, 1991). If a vehicle were able to closely follow the sequence, it would travel a distance of just under 9 kilometres. This cycle is used for evaporative emissions testing, where failure to 'keep up' is not considered a fatal error; there is no federally mandated use of this cycle for exhaust emissions testing. The engine cycles, collectively referred to as the FTP (federal test procedure), are sequences (1167 seconds for gasoline, 1199 for diesel) of 'normalised' torque and engine speed readings (Federal Register, 1991). These normalised readings serve as dimensionless index pointers into the operating ranges of the subject engine. Thus, the actual test sequence is a function of the maximum and minimum engine speeds, and the maximum torque at each engine speed. Assuming that the normalized values are properly converted to engineering units (N m of torque, revolutions per minute engine speed), the entire cycle should be within the operational capabilities of the engine.

As a consequence of EPA's reliance on engine testing, there is no EPA-endorsed cycle for exhaust emissions testing of heavy-duty vehicles on a chassis dynamometer. It is largely because of this void that there are so many cycles, many of which are used only by the researchers that developed them. The utility of a chassis cycle can be assessed by a number of traits, including similarity to real-world driving patterns, driveability/repeatability, and suitability to the physical limitations of the vehicle or test facility. Another issue, one which comes up frequently in discussions of heavy-duty chassis dynamometer cycles, is comparability to the engine FTP. Since engine certification results constitute the most extensive database of heavy-duty emissions measurements available, there would be considerable value to being able to relate that database to real-world emissions measurements (e.g. in grams/kilometre).

Unfortunately, 'FTP comparability' is a function of many different variables. First, the FTP cycle itself is different for each engine, dictated by the engine's idle speed, governed speed, and Maximum Applied Power (MAP) curve. Second, many engines can be coupled to any number of drive trains, from 3- and 4-speed automatic transmissions to the multiple-gear, dual-range unsynchronized transmissions that are used in most tractor-trailers. Finally, different vehicles have different inertia and road load power characteristics, thus requiring different dynamometer settings. The FTP is an engine-specific cycle; any chassis cycle that claims to be comparable to the FTP must not only be engine-specific, but vehicle-specific, as well.

Researchers from WVU have investigated the possibility of 'adapting' the engine FTP to a chassis cycle (Clark and McKain 1995). The first option they explored was creating an engine/vehicle-specific chassis cycle that would create an FTP-like engine operating sequence. It was not difficult to prove that it is virtually impossible to duplicate the FTP sequence of engine speeds and torques using a chassis test. However, the second approach explored by the WVU researchers, the conservation of energy approach, may be the closest that any chassis cycle can come to being 'comparable' to the FTP. Using 'typical' truck and engine specifications, the researchers derived a chassis cycle, then used a computerised 'driving model' to demonstrate its driveability. Although the results were encouraging, there has since been no indication of this type of cycle's being used for widespread vehicle testing.

Engine FTP - energy conservation adaptation

Figure 2 shows how the engine FTP is converted to a vehicle speed sequence. For clarity, only a small segment of the 20-minute sequence is shown. Traces (a) and (b) are the normalised cycle values, as printed in the certification test procedures. The torque sequence, trace (b), includes records that are designated as 'Closed Rack Motoring' in the CFR listing. These records indicate when the throttle should be shut off, and the engine rotated externally by its flywheel. From an energy perspective, motoring is the application of 'negative' torque by the engine, and will translate to a proportional 'negative' power input.

Part of the normal test procedure is the 'unnormalisation' of the tabulated values to actual engine speeds and torques, which are specific to the engine that is being tested. For this adaptation, each speed/torque combination is then converted to power; trace (c) shows the power trace segment for a 1989 Cummins NTC-315 diesel engine. It should be noted that several of the lower power 'spikes' in the power trace result from conditions where the torque goes positive and the engine speed remains at idle. These conditions were considered unrealistic and unreproducible, and were therefore removed in subsequent processing; the net result is a ~0.5% decrease in total cycle energy.

At this point, the characteristics of the vehicle itself come into play, determining how the engine power is converted into kinetic, potential, and thermal energy. Potential energy is held constant by testing on level ground. Thermal energy is the continuous energy drain produced by rolling resistance and aerodynamic drag; these quantities are often estimated (but can be determined experimentally) as a function of vehicle speed. Subtracting road load power from engine power leaves the net power that accumulates as kinetic energy. For a known vehicle mass, this kinetic energy can be readily converted to velocity. The velocity segment shown in trace (d) represents a 1989 Ford CL-9000 coupled to a 45-foot cargo van trailer; the gross combined weight (GCW) is 49,900 pounds.

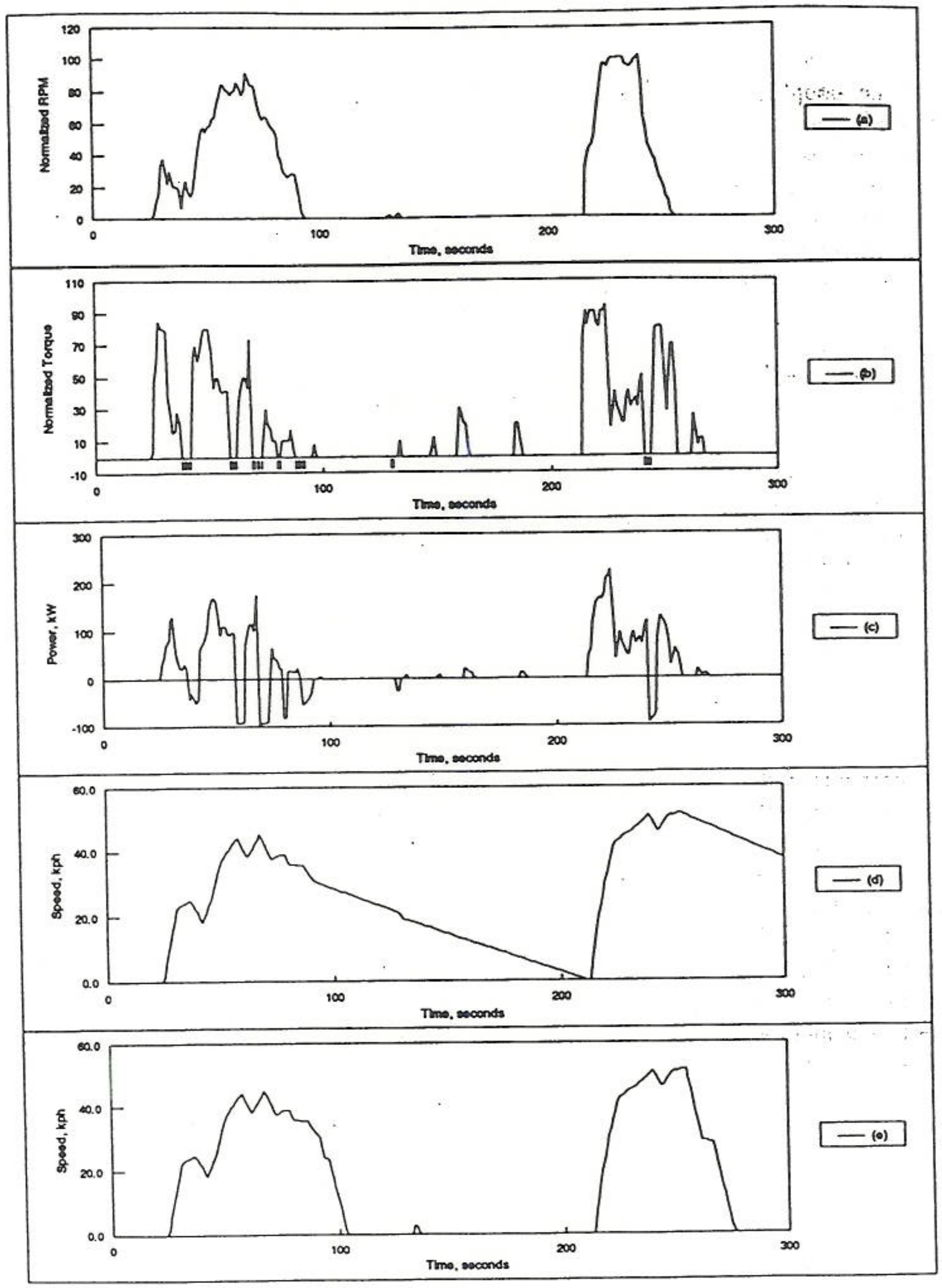

Figure 2 *Engine FTP cycle adaptation*

Up until this point in the adaptation, the energy balance has assumed that road load power demand (i.e., thermal energy loss) is the only mechanism by which energy is removed from the system, hence the lengthy deceleration in trace (d) that ultimately brings the vehicle speed to zero. Since coasting to a full stop is not a typical vehicle behaviour, it will be replaced with braking. Trace (e) shows the brakes being applied at each point where the engine speed and torque are both zero. Braking is one of several modifications to the adaptation that make it more driveable and more representative of typical operation.

Figure 3 shows the complete cycle corresponding to Figure 2, trace (e). This compares to 'modified energy conservation method driving speed cycle' shown in the Clark and McKain paper [13]. The following assumptions (some of which will be re-visited in a later section) are incorporated into the Figure 3 adaptation:

- Torque curves do not vary significantly among engines of the same model (i.e., the Cummins/Ford example cycle used a torque curve from another Cummins NTC-315 engine).
- During 'closed rack motoring' (identified as '(^{1})' in the cycle listing), negative torque varies linearly with engine speed; for the Cummins example, T N m = $-0.210 \times$ (RPM - idle).
- Effective inertial mass of a heavy-duty truck (i.e., mass of vehicle plus effective inertia of rotating components) equals GCW + 150 lb/wheel (49,900 + 2700 pounds for the Ford CL-9000 example).
- There is no change in elevation (i.e., potential energy) during the cycle.
- Road load power demand, including drivetrain losses, can be accurately represented by a two-parameter equation (e.g., for the 49,900 lb GCW Ford, kW = $0.0314 \times \text{kph}^3 + 166.8 \times \text{kph}$) derived from coast-down data.
- Brakes are applied anytime both normalized-RPM and normalized-torque are zero.
- Anytime brakes are applied, vehicle will decelerate at a rate of 2 MPH/second (3.2 kph/second) until speed reaches zero, or until the next non-zero horsepower record.
- If a '(^{1})' record is reached when the adapted vehicle speed is zero, the vehicle speed will remain zero, and the accumulated energy input parameter will remain unchanged.

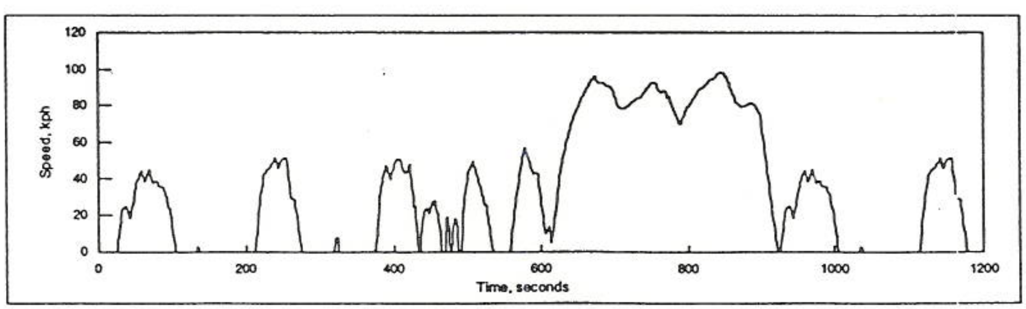

Figure 3 *Chassis cycle from FTP energy conservation adaptation*

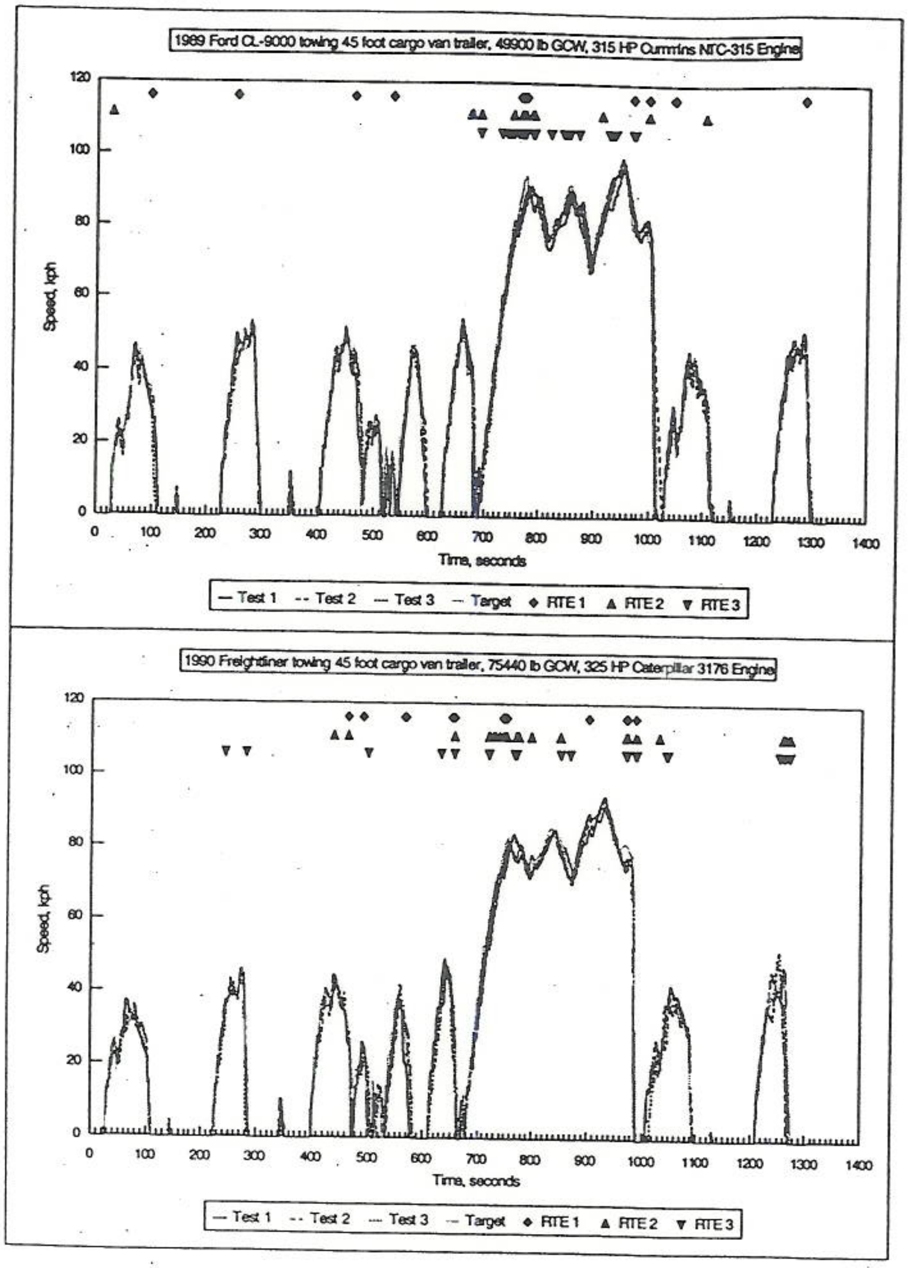

Figure 4 *First generation modified energy conservation cycle (MEC/FTP1)*

One other assumption that is reflected in the Figure 3 adaptation is that power flows continuously from the engine to the vehicle during positive torque segments. For a vehicle with an automatic transmission, the errors of this assumption may be small. However, for many heavy-duty trucks, particularly those with unsynchronized transmissions, power delivery is far from continuous during accelerations. To account for these discontinuities, shift delays are added to the acceleration segments. The result is a cycle that has a similar sequence of speeds, but that takes longer to complete. Nonetheless, the total cycle energy is unchanged, and the delays are the only way to create a driveable cycle that follows the energy profile of the engine FTP.

Clark and McKain (1995) tested the driveability of their 'modified energy conservation' cycle (with inserted shift times, of course) using a computerized 'driving model' and a 'hypothetical vehicle' that was chosen to represent a 'typical' vehicle/drivetrain/engine configuration. Figure 4 shows our comparable first generation Modified Energy Conservation FTP cycle, MEC/FTP1, as derived for (and tested on) two trucks. Also shown on each graph are the 'as driven' traces, which give an indication of real-world driveability. The symbols indicate recommended tolerance exceedances (RTEs, using tolerances defined in the EPA recommended procedure) (France *et al.*, 1979). The tolerances are applied only when the engine is actually powering the vehicle (as opposed to motoring or idling, which make up just over 50% of the FTP) because those are the only segments where the driver's inputs affect emissions. A majority of the tolerance exceedances are random and occur at different times among the triplicate tests; these are related to the driver's attention's being divided between following the trace and operating the vehicle safely on a public roadway. It is expected that, under more controlled conditions with a more focused effort (i.e., a dynamometer simulation), these random exceedances can be avoided.

There is one systematic exceedance (i.e., showing up in all three tests) in each cycle, occurring near the 800 second mark in the Ford cycle and near the 1000 second mark in the Freightliner cycle. It appears that, in spite of each cycle's being tailored to the truck, there are places where the trucks simply cannot keep up within 2 MPH (3.2 kph). This would indicate that the cycle adaptation algorithm is slightly overpredicting the trucks' capabilities. Our analysis has shown that several of the assumptions that went into the first-generation cycle may need to be revisited; the goal is to develop a second-generation cycle that better represents the FTP energy profile, and has improved driveability,

Adaptation refinements: the second-generation cycle

In conducting our follow-up analysis of the MEC/FTP1 cycle development and testing, several shortcomings were observed: (1) the cycle development process was quite interactive and subject to individual interpretation, (2) there was no systematic discrimination between the various deceleration modes, (3) inertia selection was quite arbitrary and showed little regard for inter-vehicle comparability, (4) transmission losses were not properly accounted for, and (5) given the dissociation of 'engine cycle' RPM and 'chassis cycle' RPM, there were no built-in assurances that the vehicle could actually deliver the necessary power to keep up with the cycle. The primary goal of the second-generation cycle development effort is to create an automatic, repeatable algorithm that will generate a chassis cycle with little more trouble than the engine FTP cycle is now generated. The secondary goal is to make the cycle more driveable and more representative of the engine FTP.

Table 1 Zero torque events

Record Range			*Event*
1	-	24	Braking
93	-	94	Coasting
97	-	128	Coasting
131	-	132	Coasting
134	-	146	Braking
149	-	157	Braking
164	-	183	Braking
187	-	213	Braking
256	-	262	Braking
268	-	320	Braking
329	-	376	Braking
435	-	436	Braking
483	-	484	Braking
523	-	523	Coasting
526	-	543	Coasting
545	-	547	Coasting
550	-	551	Braking
556	-	557	Braking
580	-	580	Braking
596	-	605	Braking
707	-	707	Coasting
900	-	926	Braking
995	-	996	Coasting
999	-	1030	Coasting
1033	-	1034	Coasting
1036	-	1048	Braking
1051	-	1059	Braking
1066	-	1085	Braking
1089	-	1115	Braking
1158	-	1164	Braking
1170	-	1199	Braking

Table 2 Motoring events

Record Range			*Brakes*
38	-	42	No
59	-	62	No
69	-	73	Yes
80	-	81	No
88	-	92	Yes
129	-	130	No
241	-	244	Yes
326	-	328	No
392	-	396	Yes
410	-	414	No
423	-	427	No
430	-	434	Yes
447	-	456	No
456	-	458	No
460	-	469	No
473	-	477	No
485	-	497	No
508	-	513	Yes
514	-	522	No
544	-	544	No
548	-	549	Yes
578	-	579	Yes
581	-	589	Yes
595	-	595	Yes
612	-	613	No
675	-	676	No
697	-	705	No
757	-	759	No
770	-	772	No
775	-	781	No
784	-	788	No
855	-	860	No
862	-	865	No
893	-	893	No
897	-	899	Yes
940	-	944	No
961	-	964	No
971	-	975	Yes
982	-	983	No
990	-	994	Yes
1031	-	1032	No
1143	-	1199	Yes

Unpowered segments of the MEC/FTP1 cycle were handled quite arbitrarily. By inspecting the unmodified cycle (e.g., Figure 2, trace (d)), engineering judgement was used to select the unrealistic coasting periods for conversion to braking events. This approach is valid, and is still used for MEC/FTP2, but the modification is made a part of the input cycle (i.e., the engine FTP) for the sake of consistency between vehicles. Of the cycle records where normalized torque is zero, some are designated as 'braking' events, while some are left as 'coasting' events. Table 1 shows how the zero torque segments are designated.

Vehicle deceleration occurs under two other conditions during the cycle. One condition occurs when the scheduled power input is not sufficient to maintain the current speed, and is fully accounted for in the energy balance. The remaining deceleration conditions correspond to the 'Closed Rack Motoring' records of the engine FTP. Designated as '(^{1})' in the CFR listing, these records indicate when the throttle should be shut off, and the engine rotated externally by its flywheel. This corresponds to 'engine braking' that is used quite often by heavy-duty vehicles. In the MEC/FTP2 adaptation, these events are represented as negative torques proportional to engine speed. What is not so straightforward, however, is the status of the brakes during these motoring events. In normal operation, it is common for vehicles to experience motoring decelerations both with and without brakes. Therefore, the cycle must once again be analyzed to determine when the brakes should be applied. Braking status was assigned based on (1) the engine speed profile (rapidly slowing engine would indicate braking), (2) brakes status of adjacent records, and (3) whether or not the vehicle would be coming to a complete stop during the records that immediately follow. Table 2 shows how braking was assigned to the motoring records of the engine FTP.

Incorporating the information from Tables 1 and 2 transforms the cycle from a simple sequence of speeds to a sequence of vehicle operating modes. As such, it is expected that the MEC/FTP2 cycle will require an enhanced computer/driver interface as well as an experienced driver who has practiced the cycle several times. In deriving the cycle for three trucks of various configurations, our experience has led us to believe that, in spite of its truck-specific origins, the cycle will take on a similar shape for most trucks. So, once a driver becomes experienced with the cycle, that experience should carry forward to performing the cycle on other trucks with few 'practice runs.'

Figure 5 shows the MEC/FTP2 cycle as derived for three trucks. The most noticeable difference between this cycle and the first-generation MEC/FTP1 cycle is the three coasting segments that show up in the first, second, and fourth quarter-cycles. These segments are necessary to properly simulate the motoring conditions that follow them. Braking the vehicle during those segments would leave no momentum to motor the engine. On a dynamometer with motoring capability, the coasting could be eliminated in favour of a simulated downhill roll just prior to the motoring segments. Nonetheless, for the cycle to be as universally useful as possible, the coasting segments will have to suffice.

An important characteristic of this cycle that will not show on the figure is that it incorporates a driving model that compares the power demands at each point to the engine's capabilities. Because the 'as-tested' RPM profile will bear little resemblance to the engine FTP, driveability is not assured by the modified energy conservation technique. Clark and McKain (1995) used a driving model to verify cycle driveability, which was quite good (only a handful of deviations, the largest being ~2.5 kph). What we have done is taken the procedure a step further by incorporating a driving model into the cycle itself, for *assured* driveability.

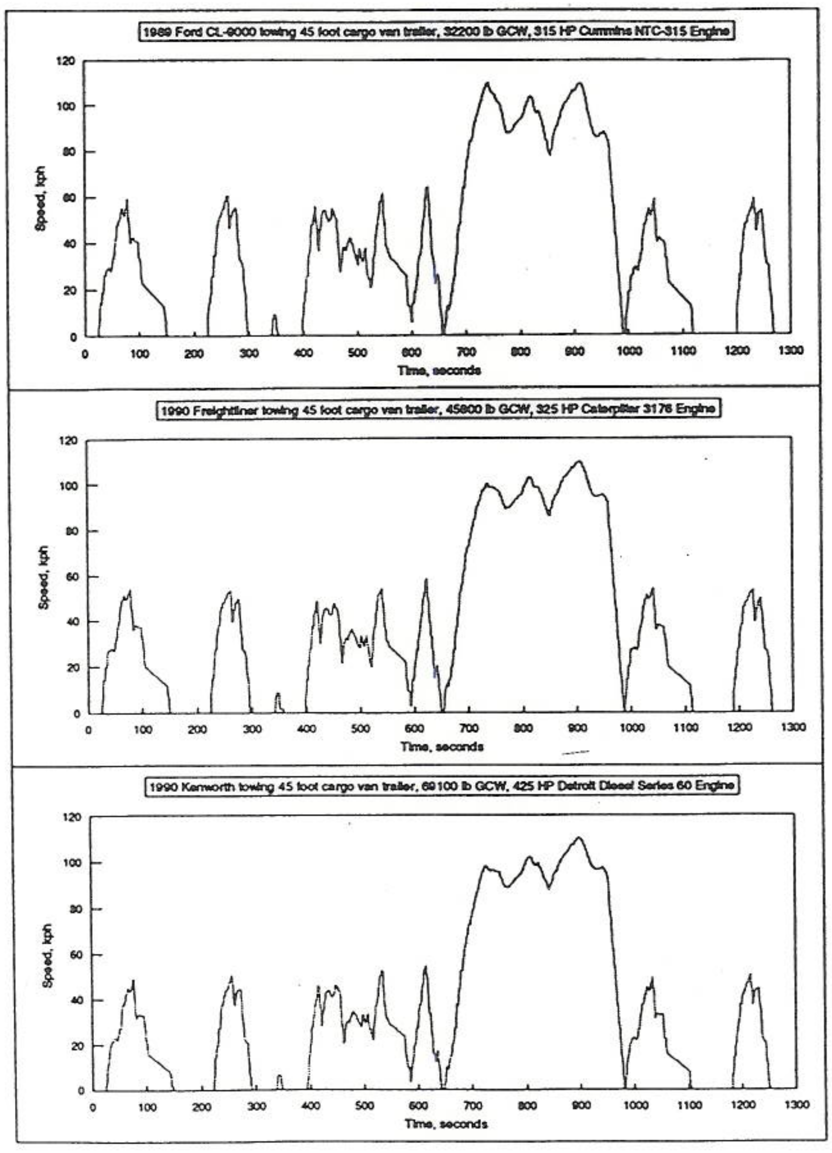

Figure 5 *Second generation modified energy conservation cycle (MEC/FTP2)*

MEC/FTP2: Cycle generation algorithm

The first step in generating the MEC/FTP2 cycle is to generate the engine FTP that it is meant to simulate. As the procedure is described in the CFR (Federal Register, 1991) the first step is mapping the engine: determining the maximum torque values for the full range of engine speeds between idle and the 'maximum mapping speed' (which is defined, as a function of rated speed, for each of four engine types). Since engine removal and testing is burdensome, it is expected that vehicle testing facilities will either use a 'representative' torque curve for the given engine, or develop a procedure to accurately determine the engine torque curve on a chassis dynamometer. It will also be necessary to generate or estimate a 'negative torque' curve to properly simulate the motoring conditions. Once the engine torque curves are available, converting the normalised values from the CFR table is straightforward. The result is a 20-minute sequence of torque and RPM values which can be readily converted to engine power.

Converting the engine power sequence to speeds requires knowledge of the vehicle: its weight, its gear ratios, and the frictional forces that it must overcome as it moves. Vehicle test weight is a topic that is as often discussed (and never resolved) as test cycles. Like the duty cycle, weights vary widely among in-use vehicles, and weight fundamentally affects on-road power demand and emissions. Fortunately, the MEC/FTP2 cycle can simulate the FTP power demand profile at any number of weights. One of the outcomes of our efforts to automate the cycle development process is that the cycle can be generated repeatedly, even iteratively, to observe or quantify the sensitivity of the cycle to changes in the input parameters. The test weights for the cycles in Figure 5 were varied in an iterative fashion to make each cycle have a maximum speed of ~110 kph.

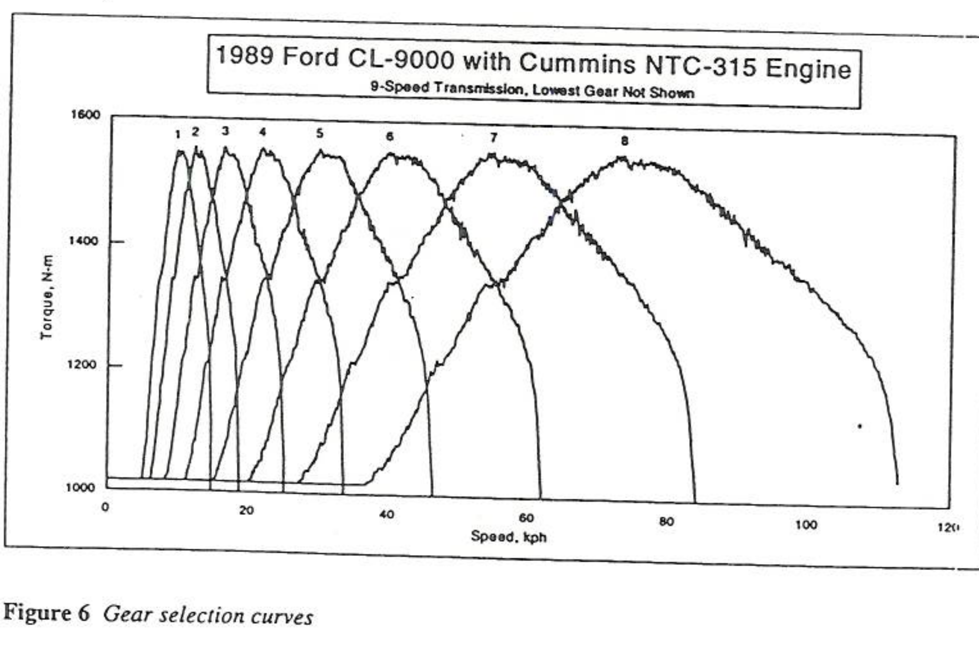

Figure 6 *Gear selection curves*

Table 3 Gearing for Ford CL-9000

Gear	*Minimum, kph*	*Maximum, kph*	*Ratio, kph/rpm*
1		14.15	0.00780
2	10.93	19.02	0.00984
3	14.29	25.96	0.01329
4	19.30	34.97	0.01777
5	26.10	48.09	0.02437
6	34.99	64.52	0.03261
7	47.67	84.72	0.04426
8	63.99	112.65	0.05955

Gear ratios are used by the driving model to schedule shifts, and to decide what gear the truck should shift into. Table 3 shows the ratio table for the Cummins/Ford example that has appeared throughout this paper. The maximum speed values and the ratios were determined by experiment, and may vary from what might be calculated from manufacturer's specifications and tyre diameters. The minimum speed column is defined as the minimum speed for which that gear will generate more engine torque than any other gear. These values correspond to the intersections of the curves in Figure 6. The idea is that a skilled driver, when choosing to shift gears, will always shift into the gear that gives the most engine torque. During accelerations, the driver may choose to shift when the minimum speed for the next gear is reached, or may choose to stay in gear until the engine reaches governed speed (the driver may even choose to skip gears, if the truck is loaded lightly). The driving model upshifts at governed speed, and it only skips gears during downshifts.

The kph/RPM column allows the driving model to calculate, based on the truck's gearing, the actual engine speed for all points in the cycle. For each of these engine speed values, the model returns to the torque curve and verifies that the engine can actually deliver the energy that is required at that point in the cycle. If it cannot, the engine is operated at maximum torque until the cumulative delivered energy has caught up to the scheduled energy. This is where *assured* driveability differs from *verified* driveability; without the built-in driving model, any energy that could not be delivered is lost, even if it could have been made up the very next second (as is often the case).

Table 4 Estimated transmission drag

Measured parameter	*Drag force, N*
Drivetrain With Transmission	177.5 + 3.673 * kph
Drivetrain Without Transmission	175.7 + 2.731 * kph
Transmission Drag by Subtraction	1.8 + 0.942 * kph

The remaining input parameter is the frictional force function. Vehicle testing facilities have their own procedures for determining and simulating road load power demand. Many of them involve combining published data, engineering assumptions, and dynamometer measurements to arrive at the power absorber settings that should make a dynamometer mimic the real world. Since the trucks corresponding to the Figure 5 cycles were tested on-road, our road load force is determined by the coast-down technique of White and Korst (White and Korst, 1972). The result of this technique is a two-parameter model of road load force as a function of vehicle speed. This force includes rolling resistance, aerodynamic drag, and drivetrain losses up to the transmission (the tests are done with the transmission in neutral). For all efforts up through the MEC/FTP1 development, it has been assumed that transmission losses were negligible. We've found only one technical paper that quantifies drive train friction as a function of speed (Walston *et al.* 1976), and have developed the relationship in Table 4 for inclusion in the MEC/FTP2 algorithm.

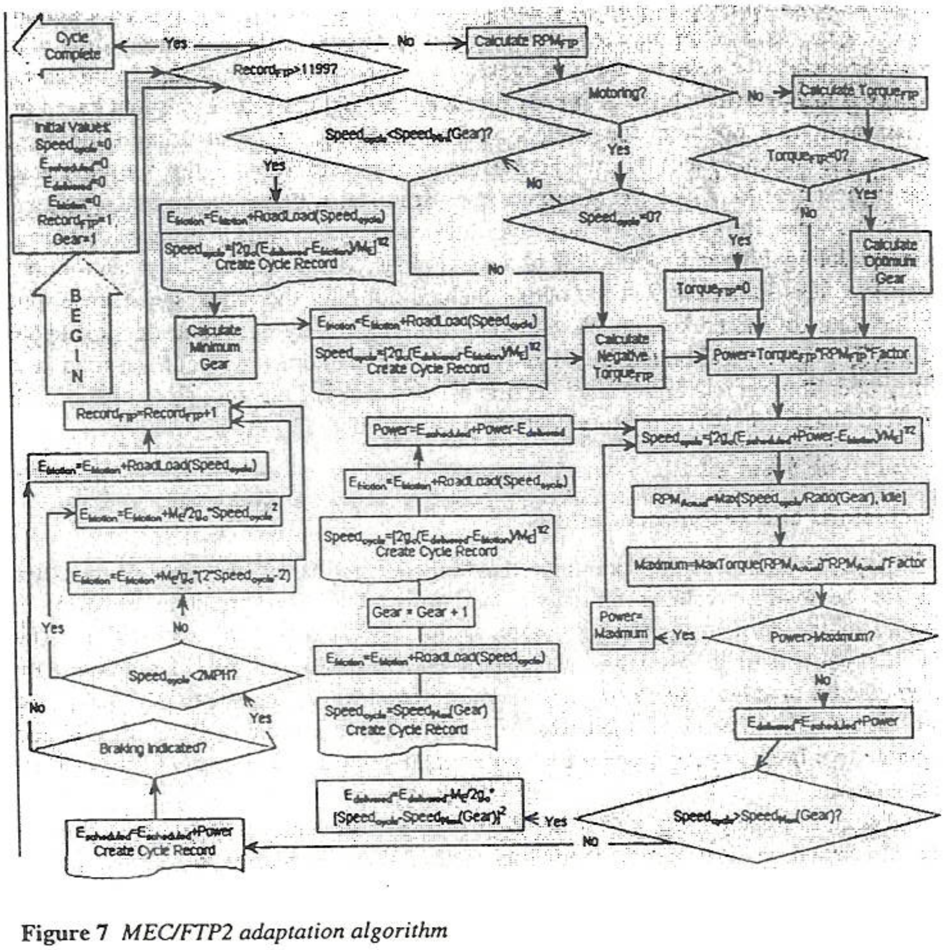

Figure 7 *MEC/FTP2 adaptation algorithm*

Figure 7 is a flow chart of the cycle adaptation algorithm (as calculated in MPH, for use in the United States). The processing takes place on a record-by-record basis; each record in the input cycle generates a corresponding record in the MEC/FTP2 cycle, and an additional two records are generated at each shift point. The conservation of energy methodology uses the energy terms $E_{scheduled}$, $E_{delivered}$, and $E_{friction}$ as accumulators. $E_{scheduled}$ is the energy from the input cycle, which only differs from $E_{delivered}$ during shifting, and on the rare occasions when the engine power falls short of the scheduled power input. $E_{friction}$ is the accumulator for where the power leaves the system, either through road load friction (rolling resistance, drivetrain friction, and aerodynamic drag) or through the brakes. Speed is calculated from the kinetic energy, which is the difference between $E_{delivered}$ and $E_{friction}$.

The 'Gear' parameter is incremented and decremented continuously during the cycle. When the current gear's 'Maximum kph' parameter (see Table 3) is reached, and the scheduled torque is still positive, a 2-second delay is inserted, and the Gear parameter is incremented. During extended motoring periods, the speed is compared to the 'Minimum kph' parameter. If the minimum is reached, and the scheduled torque is still negative, a 2-second delay is inserted, and the Gear is set to the lowest value for which the Maximum kph parameter is not exceeded. These downshifts often span more than one gear, as is often the case in real-world driving situations. Downshifting also takes place when the vehicle coasts or brakes in a non-motoring situation. In this case, no delay is inserted (the clutch is already disengaged), and the gear is selected based on the assumption of 'optimum' gear selection (i.e., using the gear that delivers the most torque at the current speed). For the Ford example shown in Table 3, the optimum gear is the highest gear for which the Minimum kph parameter is below the current speed.

For this work, the cycle adaptation algorithm was implemented in a spreadsheet, but the methodology is equally applicable to a stand-alone computer program or subroutine. Regardless, it is important that the output include not only the target speed trace, but also the status of the brakes, clutch, and gearshift. Accurately executing this test cycle requires a properly calibrated heavy-duty chassis dynamometer, or an on-road test facility operating on a straight, level section of road that is long enough to contain the full cycle distance (~14½ km for cycles with a top speed of 110 kph).

Conclusions and recommendations

This paper addresses two problems that impede the use of chassis dynamometer data for heavy-duty vehicle emissions modelling. The first problem, affecting a majority of the currently available cycles, is the 'one size fits all' assumption. Any cycle that consists of an inflexible sequence of vehicle speeds will be (1) undrivable by underpowered or otherwise slowly accelerating vehicles, (2) unrealistically 'easy' for powerful or agile vehicles, or (3) both. What is needed is a cycle that is scaleable and adaptable to a large variety of vehicles, while maintaining some comparability between the test results.

The second impediment to the use of chassis dynamometer data for modelling is fleet representation. The current modelling methodology represents the entire fleet of heavy-duty vehicles, because the engine certification database (i.e., the complete collection of engine FTP results) includes all engines sold in the USA. Achieving a similar level of representation using vehicle-based testing would be prohibitively costly. What is needed is a way to test a small sample of vehicles, and to relate those test results to the full population. Since the current heavy-duty vehicle population is represented in the engine certification database, the best way to expand the applicability of vehicle test results is to somehow relate them to engine test results.

The best solution to both problems is to develop an in-vehicle equivalent of the engine FTP. A properly adapted cycle would already be scaled to the vehicle's engine power. Adding other vehicle-specific elements would create a cycle that successfully balances drivability with realistic engine exercise. FTP comparability, in addition to facilitating statistical inferences about engine/vehicle samples and populations, would also ensure a certain degree of 'modal balance' (i.e., highway versus non-highway driving modes) which is somewhat lacking in some of the special-purpose cycles currently in use.

This paper has demonstrated that, while no chassis cycle will duplicate the FTP engine duty cycle, it is possible to duplicate its energy demands. So, to the extent that power demand is a controlling factor in engine emissions (and much of the current heavy-duty vehicle emissions modelling assumes that it is), then the MEC/FTP2 cycle should generate emissions that are comparable to the engine FTP. In many ways, the differences between this chassis cycle and the FTP (primarily engine speed) actually make it a more realistic cycle than the engine cycle from which it is derived. At the very least, the MEC/FTP2 cycle is more comparable to the FTP than the EPA cycle shown in Figure 1, which has been (somewhat erroneously) referred to as 'essentially the chassis version of the engine transient cycle' (Dietzmann and Warner-Selph, 1985). This cycle is now available for use for on-road and chassis dynamometer testing when FTP-comparability is desired.

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