

Simulations of the Fuel Economy and Emissions of Hybrid Transit Buses over Planned Local Routes

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ABSTRACT

We present simulated fuel economy and emissions of city transit buses powered by conventional diesel engines and diesel-hybrid electric powertrains of varying size. Six representative city drive cycles were included in the study. In addition, we included previously published aftertreatment device models for control of CO, HC, NO_x, and particulate matter (PM) emissions. Our results reveal that bus hybridization can significantly enhance fuel economy by reducing engine idling time, reducing demands for accessory loads, exploiting regenerative braking, and shifting engine operation to speeds and loads with higher fuel efficiency. Increased hybridization also tends to monotonically reduce engine-out emissions, but tailpipe (post-aftertreatment) emissions are affected by complex interactions between engine load and the transient catalyst temperatures, and the emissions results were found to depend significantly on motor size and details of each drive cycle.

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INTRODUCTION AND BACKGROUND

Included among the seven million medium-duty (MD) vehicles currently registered in the U.S., there are a significant number of city transit buses that receive \$26 billion/year in federal, state and local subsidies [1,2]. Most city transit buses travel for more than ten hours and cover a hundred miles daily along local stop-and-go routes. The high frequency of stop-and-go and idling operation would appear to make transit buses good candidates for electric hybridization. However, despite the expected potential benefits, MD bus hybridization is still at a relatively early stage of development [3, 4].

Hybrid vehicle powertrains are generally grouped into three main configuration types: series, parallel, and series-parallel [5, 6]. Series-parallel hybrids are more complex than are series or parallel systems, and most commercial hybrid bus powertrains tend to be series or parallel [7, 8, 9, 10, 11]. In series configurations, propulsion is provided entirely by an electric motor, and the engine drives a generator that converts the mechanical power into electricity. The electricity either directly powers an electric motor (which drives the wheels) or it is stored in a battery for future use. While series hybrids have some advantages in heavy stop-and-go conditions and allow the engine to operate at a peak efficiency condition most of the time it runs, there is an efficiency penalty associated with the dual-step conversion of energy (i.e., mechanical to electric to mechanical) [9]. In addition, series hybrids require larger

motors and batteries, which increase the cost significantly. Recent examples of the high costs of series hybrid buses are the diesel-electric Aero Star Eco buses sold by Mitsubishi Fuso Truck and Bus Corp. [7] and those sold by Orion International [8].

Parallel hybrid configurations appear to be the most attractive choice for transit buses. In parallel configurations the bus can be propelled separately by either the engine or electric motor, or by both at the same time. The presence of two power sources makes it possible to keep the electric motor and battery smaller, thus reducing cost [6]. Many manufacturers, including GM, Eaton and Allison [9, 10, 11], are commercially producing diesel-electric parallel hybrid buses. Transit agencies have reported that these parallel hybrid buses yielded an average fuel economy improvement of 20-40% based on their daily revenue service [9]. However, the availability of detailed performance data, including fuel consumption and emissions, from these hybrid buses is rather limited. This is at least partly due to the limited availability of qualified test facilities as well as the significant cost associated with making the necessary measurements. Some previous research has been done to model the performance of parallel diesel hybrid buses [12, 13], but these results are rather limited, especially in regard to emissions. Thus we were motivated to undertake more detailed simulations that accounted for the rapidly evolving diesel aftertreatment technologies and the increasing constraints imposed by new emissions regulations [14].

As is the case for other diesel vehicles, U.S. EPA regulations require hybrid buses to limit emissions of carbon monoxide (CO), hydrocarbons (HCs), nitrogen oxides (NOx), and particulate matter (PM) in their exhaust [15]. In some cases, it is possible to constrain engine operation (e.g., speed and load) to conditions where engine-out emissions are low, and thus, at least partially, achieve the required control. This approach is often used in engine control strategies, but it typically results in engine operation that is less fuel efficient. This efficiency penalty can be reduced by relying on exhaust aftertreatment technologies such as diesel oxidation catalysts (DOCs), urea-selective catalytic NOx reduction (urea-SCR), and catalyzed diesel particulate filters (DPFs) to remove the engine-out emissions before the exhaust is released. However, these aftertreatment devices can also require additional fuel to function. In general, the direct fuel penalty for operating DOC and SCR catalysts is quite small, but DPFs create back pressure on the engine and require occasional heat addition to oxidize trapped PM, both of which consume significant amounts of fuel. However, if the fuel penalty and cost of aftertreatment can be kept sufficiently low, it is possible to match the engine and aftertreatment operation in ways that boost overall fuel efficiency while still meeting the required constraints for tailpipe emissions. The options for achieving such matching are enhanced in hybrid vehicles, where the battery and electric motor provide additional degrees of freedom. Therefore, it is important to include the combined behavior of the coupled engine, aftertreatment, and electric systems when simulating hybrid bus performance under realistic drive cycle conditions.

Our goal in this study has been to use computational simulation to compare the estimated fuel economy and emissions of city transit buses powered by conventional diesel engines with those incorporating a parallel hybrid electric powertrain. By doing this, we hope to identify potential advantages and technical barriers for hybrid bus technology. To keep the problem tractable, we restricted our consideration to a single hybrid powertrain configuration and engine-battery control strategy to reveal general trends rather than identify optimal combinations of all possible parameters.

The main software platform for our simulations was Autonomie, which is an open-architecture powertrain and vehicle systems simulation tool developed at Argonne National Laboratory [16, 17]. In addition to the components included in Autonomie's default bus configuration, we added engine and aftertreatment models developed at the Oak Ridge National Laboratory (ORNL) that account for transient fuel consumption, engine-out emissions, and aftertreatment component performance for comparable conventional and hybrid buses operating over representative city drive cycles. Since engine and battery size are key factors in the cost, we considered multiple sizes for the hybrid powertrain. The results reported here should be considered only as an early first step in understanding the potential benefits of city bus hybridization, since it is very likely

that the simulated hybrid configuration, control strategy and aftertreatment system technology will almost certainly be further optimized in the future.

SIMULATION TOOLS AND ASSUMPTIONS

In the following sub-sections, we summarize the drivetrain and aftertreatment component models, the vehicle configuration, and the drive cycles that were implemented in Autonomie for this study.

Transient Engine Model

In order to accurately estimate fuel economy and tailpipe emissions, it is important to accurately model the transient fuel consumption and engine-out exhaust properties, since these can change dramatically from their steady-state values, especially during cold start and stop-and-go engine operation. We addressed this issue here by utilizing previously developed dynamic correction factors for steady-state maps that account for recent engine history to give more accurate estimates of engine fuel consumption and engine-out exhaust temperature and composition. Briefly, these correction factors model the dynamic engine response as first-order lags associated with the global engine temperature response. The lag parameters are calibrated with transient engine data and/or known engine characteristics. Details of this approach are explained more fully in a previous publication [17]. For the present study, transient engine-out and fuel consumption calibration data were not available for the specific Cummins ISB engine simulated here. Thus it was necessary to estimate the transient parameter values based on overall engine characteristics and measurements from other diesel engines. While this is admittedly a rough approximation, we believe it still provides the correct general trends. Clearly, this is an area where additional studies should be made when experimental transient calibration data for this specific engine becomes available.

DOC Model

Diesel oxidation catalysts oxidize unburned HC and CO in the exhaust. Details of the DOC model used in the paper have been reported in a previous publication [18]. Briefly, three global one-step reactions are considered, including CO, HC, and NO oxidation. We assume that the global performance of the DOC device can be approximated by a single, representative 1-D channel with transient plug flow. All reactions are assumed to occur in the washcoat. All hydrocarbons are represented by their propylene (C_3H_6) equivalent. The model includes a differential energy balance to track axial and temporal temperature variations. The initial conditions, engine-out temperature and species variations, and ambient heat loss are all included. The kinetic parameters used in this study for estimating the oxidation rates of CO, HC, and NO were based on typical literature values and adjusted slightly to match experimental dynamometer measurements as reported in [18]. In the appendix, we summarized the DOC

reaction mechanism, and also listed main parameters used for the DOC simulation, including geometry, material properties, and catalyst loading parameters used in the DOC model.

SCR Model

Urea-SCR systems utilize injected aqueous urea slurry to reduce the NO_x in engine-out exhaust to nitrogen gas (N₂). The SCR model we employed here utilizes 1-D differential transient balances similar to the above DOC model. It is assumed that the rate limiting steps in the SCR device occur after urea is converted to NH₃ [14, 18], thus the model does not account for vaporization and thermolysis of the urea slurry. The present model accounts for three main global catalytic reactions: 1) the standard reaction between NH₃ and NO; 2) the fast reaction between NH₃ and equimolar levels of NO and NO₂; and 3) the reaction between NH₃ and NO₂. The effects of NH₃ adsorption and desorption and gas phase reactions involving NH₃, O₂, and nitrogen oxides were also included, but N₂O formation was not considered here. Details of the SCR model and associated kinetic parameter estimates based on laboratory measurements of a commercial Cu-chabazite zeolite catalyst are summarized in [14], where details of the SCR model validation are also included. In the [appendix](#), we briefly provided the current NH₃-SCR reaction mechanism, and also listed main parameters used for the SCR simulation, including geometry, material properties, and ammonia storage capacity. In this study, we assumed a simple urea injection strategy such that the instantaneous urea rate was always adjusted to produce the optimal stoichiometric ratio of NH₃ to NO at the SCR catalyst inlet when the inlet temperature was above 150°C. For lower temperatures, urea injection was stopped.

Since SCR technology is still rapidly evolving, it is likely that urea injection strategies and catalyst parameter values will continue to improve as more advanced catalytic materials are developed and more understanding is gained about their functionality. However, we expect that the general trends resulting from the present SCR model are unlikely to change.

Catalytic DPF Model

Catalyzed diesel particulate filters (CDPFs) remove particulates from the engine exhaust by mechanical filtration and subsequently oxidize the solid carbon and residual HCs. Adequate oxidation of the particulate is necessary to prevent buildup of back pressure, which penalizes engine output. Some degree of PM oxidation is always occurring, but the rate is an exponential function of local temperature in the DPF. So in cases where PM oxidation rates are low, it is necessary to occasionally stimulate PM oxidation by burning extra fuel to raise the engine-out exhaust temperature. This is referred to as active DPF regeneration. Thus accurate simulation of the accumulated DPF soot levels is important for determining fuel economy.

In this study, we used a simplified 3-zone catalytic DPF model to approximate the DPF as a combination of an inlet channel zone, an outlet channel zone, and a filter wall. Temperature can vary from zone to zone but is assumed to be uniform within each zone. Deposited PM is considered to effectively include two layers: a deeper layer in close proximity to the catalyst and an outer bulk soot cake. PM oxidation by O₂ and NO₂ occurs in the cake layer without any catalytic acceleration, while oxidation by O₂ is catalytically accelerated in the deeper layer. Details of the global oxidation kinetics parameters have been published elsewhere [19]. We also included one-step reactions in the deep layer for oxidation of CO, HCs, and NO. Except for catalyst loading, the kinetic parameters for these oxidation reactions are assumed to be identical to those in the DOC model described above. We briefly introduce the current DPF reaction mechanism and list major parameters used for the current simulation in the [appendix](#).

Consistent with our previous work, we assumed here that active DPF regeneration would only be required when the engine-out back pressure reached a specified threshold. Under these circumstances, active regeneration is accomplished by adding extra fuel to raise the exhaust temperature and stimulate accumulated PM oxidation. Passive regeneration (where no extra fuel is needed) can occur when engine out temperatures are sufficiently high to promote spontaneous PM oxidation. Passive regeneration rates were very low in the cases studied here because of the extremely low exhaust temperatures. But as noted below, the rates of PM accumulation in the DPF were still sufficiently low to keep below the back pressure threshold during each of the drive cycles. Thus our results do not include the effects of active regeneration. Future studies will need to consider multi-cycle periods in order to address the impacts of active regeneration.

Bus Configurations

We started with a conventional transit bus configuration based on the 2005 Optima LF-34 bus, which has been extensively characterized in the ORNL MD conventional bus database. The 2005 Optima is powered by a Cummins ISB-02 5.9-L diesel engine with a 5-speed automatic transmission. Since the Cummins ISB-02 engine map was not available, we estimated a fuel consumption map by downsizing a map derived from measurements of a Caterpillar 3126E 7.2-L diesel engine. [Table 1](#) lists the basic specifications of the modeled conventional transient bus, where the vehicle weight, frontal area, rolling resistance, and aerodynamic drag coefficient were estimated using a tractive energy analysis [20] of measurements from the ORNL MD conventional bus database. Constant average values for the power consumed by accessories were used in the model.

Table 1. Specifications of the simulated conventional transit bus.

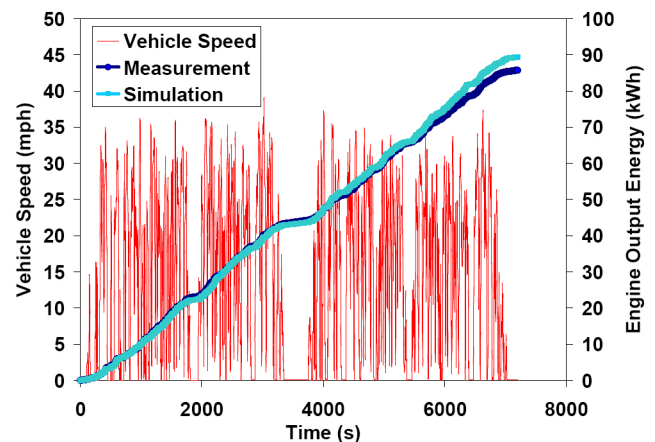
Diesel engine	Max torque	1024Nm@1450 rpm
	Max power	191kW@2200 rpm
	Idle speed	845rpm
Transmission Gear ratios	1st gear	3.49
	2nd gear	2.86
	3rd gear	1.41
	4th gear	1.00
	5th gear	0.75
Final Drive	Final ratio	3.73
Wheel	Wheel radius	0.45m
Chassis	Vehicle mass	11,636kg
	Frontal area	8.5m ²
	Rolling resistance coefficient	0.0098
	Aerodynamic drag coefficient	0.79
Accessories	Ave Electrical power	360W
	Ave Mechanical power	6,790W

To confirm that the above assumptions gave reasonable simulation results for a conventional bus, we compared our 2005 Optima simulations with on-road measurements made by ORNL and the Knoxville Area Transit (KAT) Authority of the same bus [21]. For a travel distance and time of 28.6 miles and 7200s, respectively, the predicted and measured trip fuel consumptions were 6.68 gallon and 6.56 gallon, respectively. This difference is probably within measurement accuracy. Figure 1 illustrates the good agreement between the simulated and measured cumulative fuel consumption and engine work between the simulations and measurements, implying that our basic simulation assumptions concerning the engine and powertrain configuration were indeed reasonable.

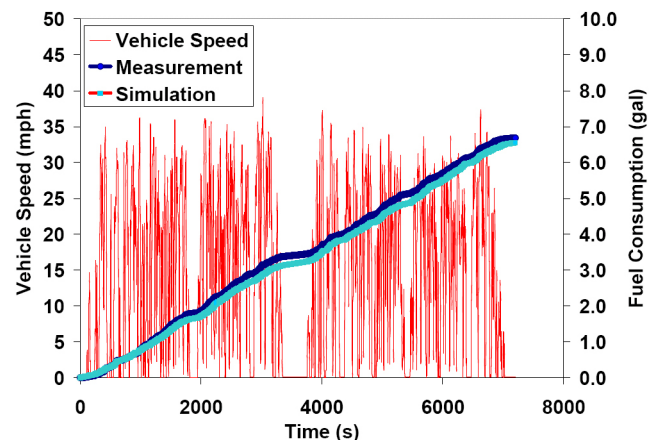
For MD hybrid bus simulations, we specified a pre-transmission parallel configuration with a single motor between the clutch and gearbox. The electric motor was rated at 120 kW, and the battery was rated at 28Ah with 140 kW peak power. We included an additional 200kg in the mass of the vehicle to account for the weight penalty of the motor, battery, and accessory components. Table 2 summarizes the main electrical system features. Other powertrain components and parameters are the same as those used in the conventional bus configuration.

The default vehicle level hybrid controller from Autonomie was utilized to manage powertrain components (such as engine, electric motor and transmission) and maximize fuel consumption [5, 16]. The controller commands the engine to be on if the accelerator pedal or power request exceeds specified thresholds, if the motor is saturating, or if the battery state-of-charge (SOC) is below a threshold value. Otherwise, the

engine is off and all power is supplied by the motor and battery. Motor torque assist is utilized when the requested torque exceeds the engine's peak torque at a transient speed. The clutch is engaged when the engine is on and disengaged any time the engine is off but the electric motor is on. During braking, the engine is turned off and the motor acts as a generator, providing power back to the battery. If the braking power during regeneration exceeds the battery power charging limit, it is then assumed that the recovered braking power is equal to the maximum battery power charging ability. An additional control strategy was added so that when the vehicle experiences long periods of idling conditions, the engine is periodically switched on to maintain a reasonable battery SOC. This simple hybrid control strategy does not adjust according to the bus engine load, nor is it a synergistic strategy that considers tailpipe emissions. This approach is admittedly rather simplistic, so there are several opportunities to improve the control strategy to achieve a more robust design.



(a). Engine cumulative energy output



(b). Fuel consumption

Figure 1. Comparison of simulation results and actual on-road measurements for conventional transit bus engine energy output and fuel consumption. The simulated vehicle mass was 11,636kg.

Table 2. Specifications of hybrid electrical components.

Motor	Max power	120kW
	Continuous power	60kW
	Max torque	920Nm
	Max power	6200rpm
	Idle speed	1000rpm
Battery	Capacity	28Ah
	Peak chg power	120kW
	Peak dis power	140kW
	Normal Voltage	324V
Accessory	Electrical power	2,490W
	Mechanical power	1,580W
Mass penalty	Additional hybrid components weight	200kg

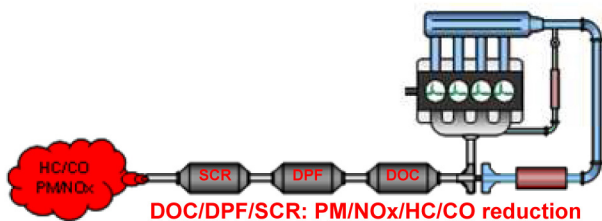


Figure 2. Configuration of the simulated DOC/DPF/SCR after treatment train system.

To account for emissions controls, we assumed that both the conventional and hybrid buses were equipped with the 2.3-L DOC, 9.7-L catalyzed DPF, and 7.7-L SCR catalyst, positioned as shown in Figure 2. By locating the DPF upstream of the SCR catalyst, chances for passive regeneration of the DPF were increased, thereby potentially reducing the need to expend extra fuel for DPF regeneration. In the present study, we did not consider the heat loss from exhaust pipes connected between these components. Additionally, the power consumed for urea dosing is not considered, due to the small low-pressure urea injection rate, which is not likely to significantly affect the fuel economy.

City Drive Cycles

Many different bus drive cycles have been proposed for metropolitan buses [21, 22, 23, 24]. For the present study, we selected five previously proposed driving cycles and one new cycle to include in our simulations. The five previously proposed cycles were the Central Business District (CBD) cycle, New York bus (NYBC) cycle, Manhattan bus cycle, Orange County Transit Authority (OCTA) bus cycle, and Washington Metropolitan Area Transit Authority (WMATA) cycle.

The CBD cycle originated with the US Department of Transportation and the Urban Mass Transit Association. It includes a “sawtooth” driving pattern, with multiple repetitions of an idle, acceleration, cruise, and deceleration sequence [25].

The OCTA cycle is a chassis dynamometer test for heavy-duty vehicles developed by the West Virginia University (WVU), based on driving patterns of urban transit buses in the Los Angeles area [25]. The Manhattan Bus Cycle was developed for chassis dynamometer bus testing based on observed driving patterns of buses in Manhattan. This cycle is characterized by frequent stops and very low speeds [25]. The WMATA cycle was derived from vehicle speed data logged from transit buses in Washington, D.C., and surrounding areas [23]. The NYBC cycle, on the other hand, was developed by EPA to reflect the driving patterns of transit buses in New York City. It simulates rapid stop-and-go traffic with long passenger transfer times by combining rapid accelerations, rapid decelerations, and long idling periods [25].

Table 3 summarizes the details of the above driving cycles, while the vehicle speed profiles are plotted in Figure 3. We feel it is important to note that the above drive cycles were developed without consideration of real road grades that are frequently encountered in many areas of the country and considerably change engine operations [22]. Thus we proposed a new characteristic drive cycle including grades, as depicted in Figure 3(f). The new transit bus cycle was constructed using the ORNL MD conventional bus database, which includes one year of measurements from three Class-7 buses operated by KAT in Knoxville, TN. We refer to this cycle as the KAT drive cycle. The KAT cycle consists of two distinct types of operating conditions: frequent stop-and-go during the first 4000s, and an extended idling period for the remainder of the drive cycle. These operations were found to be very typical of the Knoxville data. The stop-and-go periods were observed to be dominated by relatively low accelerations and decelerations. Extended engine idling occurred roughly 50% of the time (much of it at single stops). Figures 4(a) and 4(b) compare the statistical speed and acceleration distributions derived in the selected cycle with those measured from three KAT buses over the complete measurement period. The three city buses compared were all 2005 Optima LF-34 models, but they were operated over different routes. From Figure 4, the selected cycle exhibits similar speed and acceleration patterns as those experienced by all three buses, but also captures idle operation. Our analysis indicates that the extended idling exhibited by the buses resulted in approximately 20% more fuel consumption than would have occurred without the idling, which by itself represents a very significant energy and emissions reduction potential for the hybrid bus technology.

Figure 5 compares the distributions of simulated engine efficiencies for a conventional bus operating over each of the above six drive cycles. Because of the rather artificial characteristics of the CBD cycle, it does not resemble the other cycles that are based more directly on measurements of actual transient data. While probably more realistic, the Manhattan and NYBC cycles are also dominated by low engine loads. The OCTA, WMATA, and KAT cycles result in the broadest spectrum of engine operating conditions, and thus these cycles may offer the biggest opportunities for understanding and exploiting the benefits of hybridization.

Table 3. Details of representative drive cycles and characteristic drive cycles constructed from the ORNL MD bus database.

Cycle	CBD	OCTA	Manhattan	WMATA	NYBC	KAT
Duration (s)	567	1909	2178	1839	620	7200
Distance (mile)	2.01	6.54	4.13	4.26	0.61	17.15
Avg. Speed (mph)	12.73	12.33	6.83	8.34	3.57	8.57
Max Speed (mph)	20.00	40.63	25.30	47.50	30.70	39.06
% Idle Time	20.6%	26.9%	39.2%	43.8%	68.9%	54.5%
Max Accel (m/s^2)	2.30	4.06	4.60	3.00	6.10	2.12
Max Decel (m/s^2)	-4.70	-5.13	-5.60	-4.50	-4.30	-2.28

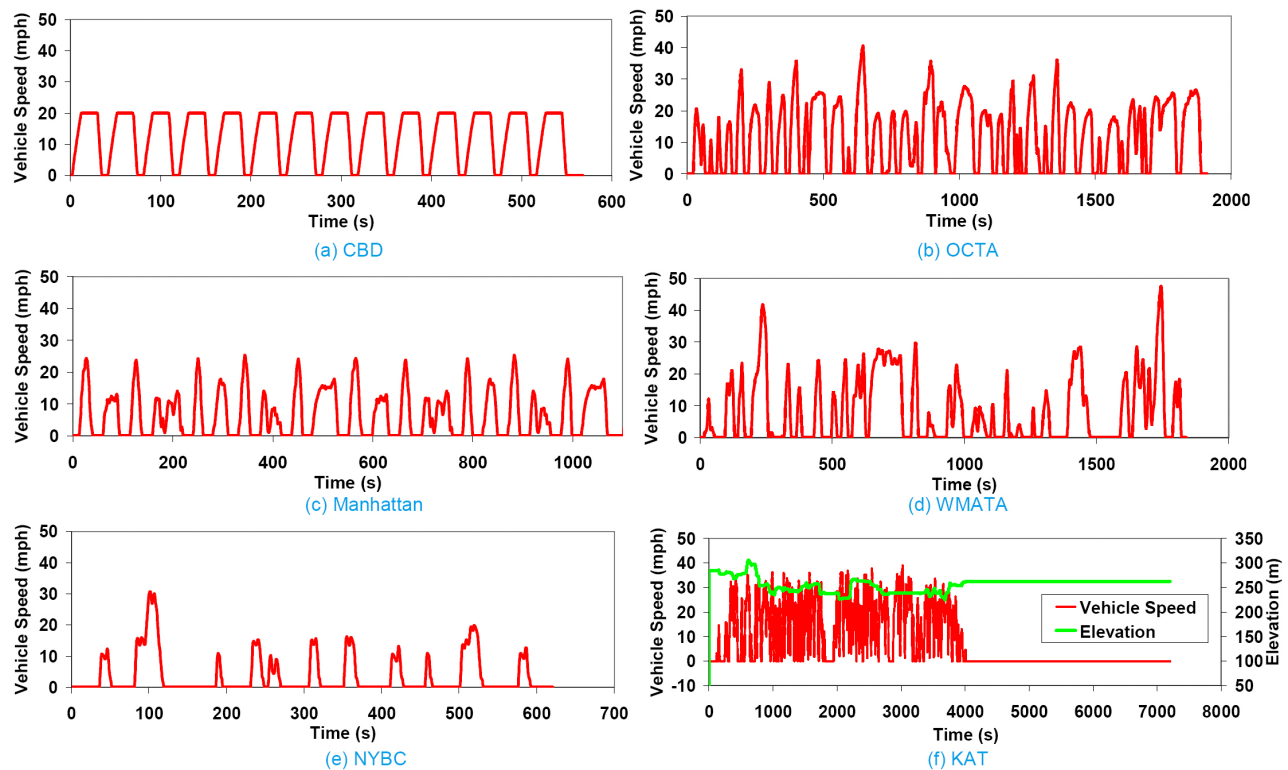


Figure 3. Speed profile of representative drive cycles and characteristic drive cycles constructed from the ORNL MD bus database.

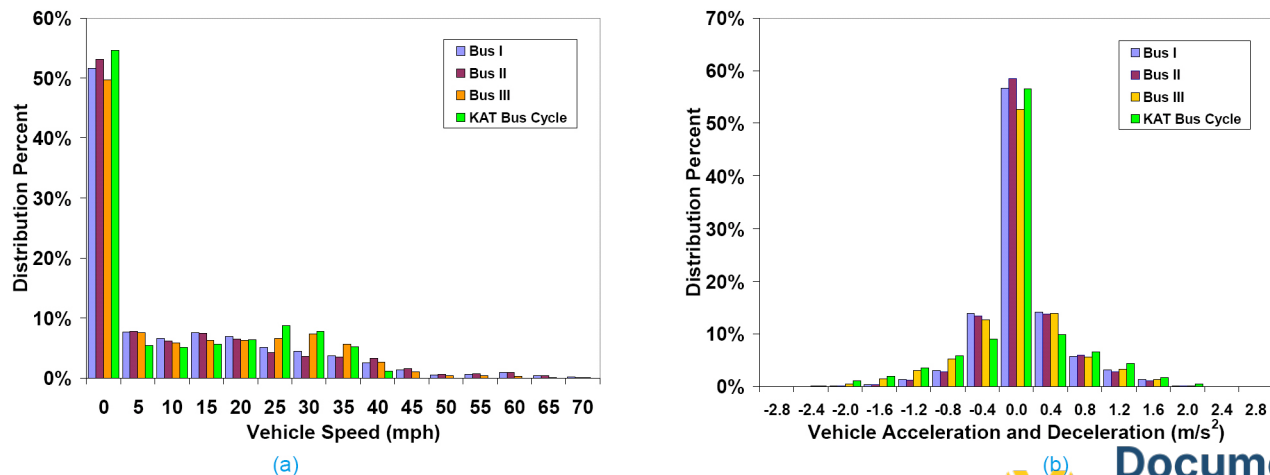


Figure 4. Measured speed and acceleration distributions of the Knoxville city buses compared to the selected KAT drive cycle.

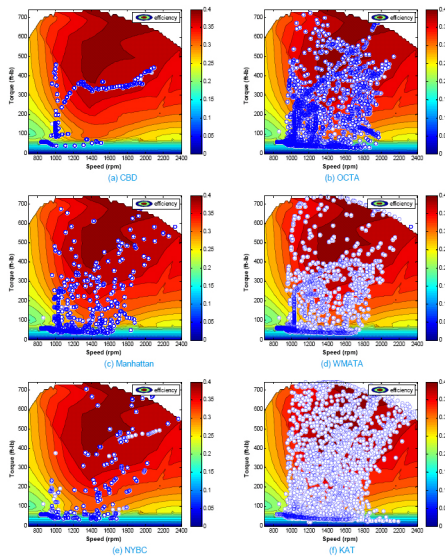


Figure 5. Simulated engine efficiency distribution for conventional city bus over the representative six city bus drive cycles; the color contour in the map is engine efficiency.

We note here that all of the simulated cycles above assumed an initial cold start at 20°C. No consideration was given to lower starting temperatures. Since lower temperatures could be relevant to the fuel consumption required to initially warm the passenger cabin and other vehicle systems during periods of extreme cold, this would be an important area to include in future studies of hybrid buses.

RESULTS AND DISCUSSIONS

To study the impact of hybrid technology on fuel savings and emissions reductions, we simulated both conventional and hybrid city buses by repeating each drive cycle until reaching a total duration of 7200s (2 hours). Each case was simulated using an initial cold start of 20°C. In this section, we summarize the simulation results.

Fuel Economy and Energy Analysis

Figure 6 illustrates the estimated hybrid bus fuel savings over each of the six city drive cycles. The hybrid configuration summarized in Table 2 improved fuel economy in all cases by anywhere from 26% to 80%. The main factors behind these savings were: (1) a boost in the engine efficiency by operating it at more favorable speeds and loads, (2) reduction of idle and accessory loads as a result of engine shutdown during stops, and (3) brake energy recovery. Explicitly quantifying the contributions from each of these items is complex because these factors frequently have strong interactions.

Figure 7 illustrates the differences in fuel energy partitioning between the conventional and hybrid cases in the KAT cycle. The total fuel consumption for the conventional bus is used as the basis for both Figures 7(a) and 7(b), so that the fuel savings due to hybridization is shown as a separate sector (green) in 7(b). These energy 'pie' charts make it clear that the hybrid bus is considerably more fuel efficient. The loss of engine exhaust heat from the hybrid bus drops to 42.7% of the total fuel energy consumed by the conventional bus, which expelled 67.5% of its energy to the environment. This is due to less idle time and more time spent at more optimal speeds and loads. Reduced engine idling is so important because there is so much time spent there. For example, in the KAT cycle, the conventional bus spends 54.5% of the time idling, during which 21.6% of the fuel energy is consumed. Significant Idle operation time is typical of the other bus cycles as well, as illustrated in Table 3.

Wheel energy loss (rolling resistance) in the hybrid bus accounts for 9.8% of the reference energy consumption, compared to 18.6% for wheel loss in the conventional bus. This is mainly due to brake energy recovery in the hybrid vehicle. On the other hand, the hybrid also has higher drivetrain losses and losses in the motor and battery. The higher hybrid drivetrain losses result from power transfers for braking energy recovery and battery charging. The energy losses of the motor and battery are limited, and their contribution to the total energy loss is less than 3%.

The impact of using the engine selectively in the hybrid bus to reduce fuel consumption is illustrated in Figure 8, which reveals that the hybrid bus engine spends considerably more time close to the most efficient region of the engine map. Thus the hybrid bus is able to achieve approximately 33% reduction in fuel consumption compared to the conventional bus in the KAT cycle.

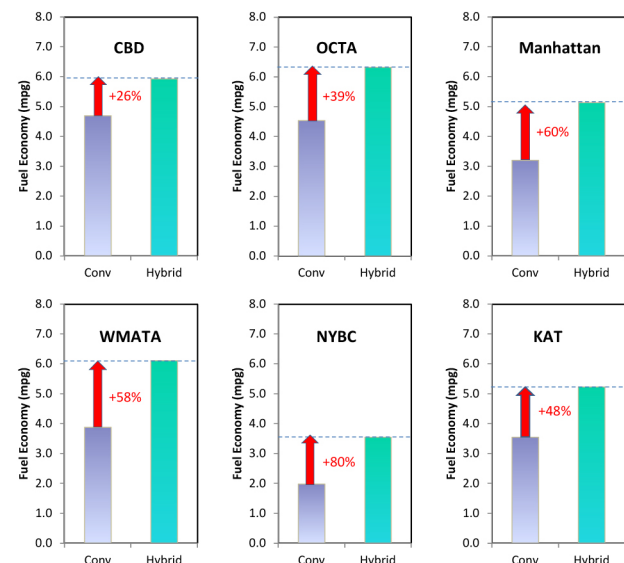


Figure 6. Comparison of simulated fuel economy for conventional versus hybrid city buses operating over six different repeating driving cycles till 7200s; all cases begin with a 20°C cold start.