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16. Abstract

This research project is to inform NHTSA and EPA's development of Phase 2 Greenhouse Gas Emissions and Fuel Efficiency Standards for Medium- and Heavy-Duty Engines and Vehicles (Class 2b – 8). A literature review is used to identify potential fuel saving technologies and review the state of the art. A large number of engine and vehicle technologies are selected for additional analysis, and their fuel saving performance is simulated to project the fuel savings potential of each technology over a wide range of duty cycles. Wherever possible, experimental data is used to inform and validate the simulation results. All baseline engine and vehicle models are validated against experimental data. Trade-offs between engine-out NOx levels and fuel consumption are reviewed. There is a review of fuel consumption, fuel economy, and GHG metrics, and of test and analysis procedures. The existing international truck fuel consumption and GHG regulations are compared and contrasted. Certification issues are explored, including the potential for including trailers in future regulations.

Note: This report was subjected to external peer review per OMB guidelines for a Highly Influential Scientific Assessment (HISA). Materials from the peer review process are publicly available in accompanying documents.

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EXECUTIVE SUMMARY

In 2011, the National Highway Traffic Safety Administration (NHTSA) and Environmental Protection Agency (EPA) jointly issued a first phase of fuel efficiency and greenhouse gas (GHG) standards that apply to medium- and heavy-duty on-highway engines and vehicles for model years (MY) 2014 to 2018 and beyond. These regulations are commonly referred to as "Phase 1" of the Heavy-Duty National Program. The standards cover all vehicles in weight classes 2b through 8, which encompasses most vehicles with gross vehicle weight ratings (GVWR) over 8,500 pounds except for a limited number of passenger vehicles covered under the light duty corporate average fuel economy (CAFE) standards, and recreational vehicles, which were included in EPA's GHG standards but not NHTSA's fuel efficiency standards. Phase 1 has two implementation stages. EPA's greenhouse gas emission standards are mandatory beginning with model year 2014. NHTSA's fuel consumption standards are voluntary in model years 2014 and 2015, becoming mandatory with model year 2016 for most regulatory categories. Commercial trailers were not regulated in Phase 1. The Phase 1 GHG and fuel consumption standards were developed using input from a number of studies that evaluated the fuel saving technologies that are available, such as the NESCCAF 2009 report [1] and the NHTSA and NAS 2010 reports [2, 3].

The research project described in this report has been completed for NHTSA to help to inform the next phase ("Phase 2") of the regulations, which would set standards in coordination with EPA for model years beyond 2018. In order to prepare for Phase 2, NHTSA directed SwRI to update prior research on fuel saving technologies to reflect the effects of the Phase 1 regulations, as well as to include technical progress that has been made over the last few years. In particular, SwRI was tasked with assessing the current commercial fleet technology baseline at the time of contract award (MY 2011/2012) and assessing the effectiveness and cost of potential fuel efficiency/GHG improving technologies for the Phase 2 timeframe (post MY 2018).

When considering potential fuel efficiency/GHG-reducing technologies, NHTSA directed SwRI to include a range of factors: design, functionality, duty cycle, use (type of work done by the vehicle), and factors that can influence the effectiveness, feasibility, and cost. Vehicle utility and performance are also to be considered. The content of report sections is summarized below.

After an introduction (Section 1), Section 2 provides a literature review covering the following topics:

- Fuel saving technologies for MD and HD engines and vehicles
- Market segmentation of fleets
- Current and planned fuel economy regulations in markets around the world

Four references were found regarding vehicle segmentation. This is a challenge in the medium- and heavy-duty vehicle world, where there are hundreds of applications, and where any given vehicle type may be exposed to a wide range of different drive cycles. SwRI determined that the most promising segmentation approach has been developed by CalHEAT in California. Calheat breaks the Class 2b through Class 8 market into six segments:

1. Heavy duty pickups and vans
2. Long haul tractors
3. Short and regional haul tractors
4. Work site vocational trucks such as dump trucks, concrete truck, utility service trucks, etc.
5. Urban vocational trucks such as refuse haulers, busses, delivery trucks, transit bus, school bus, etc.
6. Rural vocational trucks, including motor coach, forestry, petroleum, heavy haulers, etc.

Nineteen references were found regarding government regulations of truck fuel consumption and GHG emissions around the world. A review of the literature shows that the world is moving towards quite different fuel consumption / GHG regulations in different markets. The differences in national regulations reflect the unique characteristics and needs of each market, as well as variation in regulatory philosophy. Japan has the first regulation to go into effect. Its regulations use a simple vehicle simulation model where factors such as vehicle weight, aerodynamic drag, and rolling resistance are held constant. The Japanese regulation effectively targets engine brake specific fuel consumption (BSFC) and transmission match.

China is implementing a regulatory approach based on a combination of chassis dynamometer testing and simulation. Vehicles are assigned a target fuel consumption based on vehicle type and Gross Vehicle Weight (GVW) rating. A modified version of the World Harmonized Vehicle Cycle (WHVC) is specified for the drive cycle, with different weightings of the urban, rural, and motorway cycles as a function of vehicle application.

At the end of the literature review task in January 2013, NHTSA, EPA, and SwRI agreed on a list of vehicle and engine technologies that form the main subject of this project.

Section 3 addresses the heart of the project, which is a performance analysis of technologies that could be used to comply with a future Phase 2 fuel consumption / GHG regulation, for the time frame beyond 2018. The analysis included both engine technologies and vehicle technologies, using a technology list described in Section 3.1 that was developed during the literature review. Four basic engines and four vehicles were selected for simulation. All of these vehicles and engines had experimental data available from other projects, and full use of experimental data was made to calibrate the models before additional technologies were evaluated. The following baseline engine models were created and calibrated using available experimental data:

Engine Type	**Displacement, liters**	**Configuration**	**Rating**	**Applications**
Gasoline	3.5	V-6 Turbo	355 HP @ 5500 RPM	MD and Pickup Trucks
Gasoline	6.2	V-8	380 HP @ 6000 RPM	MD and Pickup Trucks
Diesel	6.7	Inline 6	300 HP @ 2500 RPM	Medium Duty Truck
Diesel	6.7	Inline 6	385 HP @ 3000 RPM	Pickup Truck
Diesel	15	Inline 6	485 HP @ 1800 RPM	Long Haul Truck

The following vehicle models were built and calibrated using experimental data:

Vehicle Type	Model	Empty Wt.	Payload	GVWR	Applications
Pickup Truck	Ram 2500	6,876 lb.	3,124 lb.	10,000 lb.	HD Pickup Truck
Straight Truck	Ford F-650	15,640 lb	6,360 lb	22,000 lb.	Tow Truck
Straight Truck	KW T270	17,140 lb.	8,860 lb.	26,000 lb.	Box Delivery Truck
Tractor-Trailer	KW T-700	33,960 lb.	46,040 lb.	80,000 lb.	Long Haul (OTR)

Each engine and vehicle technology was exercised over a range of drive cycles at zero payload, 50% payload, and 100% payload. For the pickup truck, the 100% payload point included pulling a trailer with a total combined vehicle weight of 25,000 pounds. The range of drive cycles is critical, in order to exercise the technologies over a range of speeds, loads, and grades. The table below summarizes which drives cycles were used for each truck model. The Greenhouse gas Emissions Model (GEM) cycles include an urban stop-and-go cycle developed by California Air Resources Board (CARB), a 55 MPH steady-state cruise, and a 65 MPH steady-state cruise. The cruise cycles do not include any grade, so they effectively represent a single operating point for the engine. The World Harmonized Vehicle Cycle (WHVC) cycle includes urban, rural, and European-style motorway segments. The Combined International Local and Commuter Cycle (CILCC) and Parcel delivery cycles are stop-and-go cycles meant to reflect delivery truck operation. The Northeast States Center for a Clean Air Future (NESCCAF) cycle includes a small portion of urban driving, but most operation is on the highway at target speeds of 65 to 68 MPH. This cycle does include grades of +/- 1% and +/- 3%. Section 3.2 describes the analysis approach used in this study.

Vehicle Type	Drive Cycles
Ram Pickup	FTP City, FTP Highway, US06, SC03, WHVC, 65 MPH Cruise
T270 Box Truck	GEM Cycles, CILCC, Parcel Delivery Cycle, WHVC
F-650 Tow Truck	GEM Cycles, CILCC, Parcel Delivery Cycle, WHVC
T-700 Tractor	GEM Cycles, WHVC, NESCCAF Long Haul Cycle

Section 3.3 describes the results from an evaluation of both engine and vehicle level technologies. The baseline 15-liter truck engine is the MY 2011 Detroit DD15. This engine uses a turbocompound for waste heat recovery, a feature that has been on the engine since its launch in 2008. One surprising result is that removing this feature actually provided a modest *reduction* in fuel consumption. This result was confirmed when Detroit started offering a non-turbocompound version of the DD15 in 2013, with claims of better fuel consumption than the 2011 turbocompound engine used as our baseline. When SwRI tried to model all the changes that Detroit implemented in 2013, it resulted in a benefit of 1 to 1.5%, similar to their claims. These results do not match previous published work on other engines, which showed a 2% to 3% benefit for turbocompound. Turbocompound systems work best when the engine runs at high load, so a smaller displacement engine or a more heavily loaded truck would produce a better result with turbocompound. There are also other forms of waste heat recovery such as a Rankine cycle (or bottoming cycle), and this will be discussed later.

Reduced engine friction provides a benefit of 2% to nearly 5% on the long haul T-700 truck, with the actual benefit depending on payload and drive cycle. As expected, more highly loaded cycles show a smaller benefit. Downspeeding and downsizing also both provide benefits in the 2% to 4% range. A water-based bottoming cycle shows a potential benefit of 4% to 5% for long haul drive cycles, while an R245-based cycle provides about a 3% benefit.

Vehicle related changes on the long haul tractor T-700 offer the potential for larger fuel savings. Ranges were evaluated for aerodynamic drag reduction, tire rolling resistance, and reductions in vehicle empty weight for the T700. For example, a 25% reduction in drag coefficient lowers fuel consumption by approximately 12% on a high-speed cycle, while a 30% reduction in tire rolling resistance provides benefits of 4% to 9%, depending on drive cycle and payload. However, this study did not examine potential trade-offs with tire traction, durability, and other performance characteristics. Evaluating empty vehicle weight reduction only (and not changes in payload), a vehicle mass reduction of 2,200 pounds only provides about a 1% fuel savings on high-speed cycles, but about a 2% benefit on stop-and-go cycles. A 6 X 2 drive axle configuration offers savings of around 1.5% on all operating cycles and payloads. Setting a lower speed limit on the vehicle speed controller is worth about 1.5% fuel savings for every 1 MPH reduction in cruise speed. The savings only accrue when driving conditions would normally allow a higher speed, and these fuel savings are offset by lower truck productivity, which would require a larger fleet and a higher vehicle miles traveled (VMT) to deliver the same freight volume. Automated manual transmissions appear to offer a significant fuel savings benefit in transient operation, although their impact at the steady-state maximum cruising speed is effectively zero. Increasing the number of transmission ratios from 10 to 18 provided no benefit.

It should be noted that the benefits of aerodynamic drag, tire rolling resistance, and weight reduction improvements are very linear. For example, if a 25% coefficient of drag (Cd) reduction provides a 12% fuel savings, then a 12.5% Cd reduction will be worth about 6% fuel savings. Not all technologies have such a linear behavior.

On the two baseline medium-duty trucks, the Ford F-650 and Kenworth T270, three different engines were evaluated: the medium-duty version of the 6.7-liter diesel, the 6.2-liter naturally aspirated gasoline V-8, and the 3.5-liter turbocharged, direct injection V-6. On all drive cycles and payloads but one (65 MPH at full GCW), the V-6 performed better than the V-8. The biggest benefits were at light loads on low speed drive cycles. The diesel provided up to 30% lower fuel consumption than the 6.2 V-8 on higher speed and load drive cycles, but on the Combined International Local and Commuter Cycle (CILCC) cycle, which has a high amount of time at idle and very gentle accelerations, the gasoline V-6 actually outperformed the diesel at light and moderate payloads. The cause of this result was tracked to the high idle fuel consumption of the diesel, which in turn was caused by the tighter torque converter match that is required by the low speed, high torque diesel. Future work will evaluate a feature to unload the torque converter at idle.

A collection of low friction engine features including fuel, water, and oil pumps, reduced piston cooling, as well as improved bearings, rings, and liners, offers significant potential for improving diesel engine efficiency in medium trucks. Benefits of 2.4% to 8.5% were found, with the larger benefits occurring on lightly loaded vehicles on low speed drive cycles. A high engine-out NOx approach (no exhaust gas recirculation (EGR) and a more efficient turbo) can also provide a significant benefit. This approach assumes future technical advances in aftertreatment that would mitigate the effects of higher engine-out NOx.

For the 3.5 liter V-6, adding variable valve actuation and lift (VVA/VVL) to reduce pumping work provides a benefit of about 3%. Cylinder deactivation can provide up to a 2% benefit on very lightly loaded drive cycles, but zero at high speed. Lean gasoline direct injection (GDI) offers significant benefits (2% to 13%), with bigger benefits at low speeds and light loads. Note that there are issues that need to be overcome to make production implementation of a lean burn gasoline engine feasible. The largest issue is the lack of a suitable NOx aftertreatment system that can work with the exhaust temperatures of a lean GDI engine. A stoichiometric EGR approach has benefits of 4 to 6% for most drive cycles and payloads. The combination of EGR and downspeeding provides benefits of 2% to 10%, again with the larger benefits at higher speeds and loads. Downspeeding hurts fuel consumption on cycles with significant idle time, such as the Parcel cycle, which is over 50% idle. The lower speed, higher torque engine requires a tighter torque converter match, which in turn hurts idle fuel consumption. Engine friction and turbocharger efficiency are minor factors on this relatively small displacement engine.

The 6.2-liter V-8 shows larger benefits from VVA/VVL and cylinder deactivation, compared to the V-6. This is because the larger displacement results in higher pumping losses that these technologies can address. VVA/VVL provides 3.5 to almost 7% benefits, while cylinder deactivation provides zero to 7% benefits. In both cases, the larger benefits are for light payloads and low vehicle speeds. Stoichiometric EGR is worth about a 4% fuel savings on most drive cycles, but it offers up to 10% on the large, heavy T270 at 65 MPH.

Vehicle technologies were all evaluated using the baseline 6.7-liter diesel engine. The medium-duty trucks have less potential for aerodynamic improvement than the long haul trucks, for two reasons. First, they are less amenable to aerodynamic treatments. Second, many medium-duty trucks spend a lot of their life in lower speed urban traffic, where aerodynamic drag is not a major factor. However, there is still some potential for improvement from aerodynamic improvements. Rolling resistance reduction can provide benefits from 3% to 10%, depending on the drive cycle and payload. However, this study did not address potential trade-offs with tire traction, durability, or other performance characteristics. The benefit is larger with increased payload. A substantial 1,100-pound weight reduction provides modest benefits ranging from 1% to just under 3%, depending on payload and drive cycle.

An improved efficiency 8-speed automatic (compared to the 5-speed baseline) provides benefits ranging from 1% to 6%. Only the CILCC cycle had benefits over 3%, however. Switching from the baseline automatic transmission to an automated manual transmission (AMT) offers benefits ranging from 3% to over 10%. The large benefits for AMTs occur in drive cycles with a significant amount of idle time. At idle, the parasitic drag of the torque converter is very high compared to a manual or AMT transmission in neutral, which has essentially zero torque drag from the transmission. Follow-up work is planned to see how an idle neutral feature affects the relative performance of AMT and automatic transmissions. It should be noted that AMTs come with two significant disadvantages in highly transient operation: the frequent power interrupts for shift events cause significantly slower acceleration times, and drivability suffers from the power interruptions. These penalties must be weighed against the performance advantages.

The baseline Ram pickup truck used the same two gasoline engines as the medium-duty trucks, along with a modified version of the 6.7 liter diesel with higher power, torque and speed range. In comparing the 3.5 V-6 to the 6.2 V-8, benefits of -1% to 25% in fuel consumption were found for the V-6, with the largest benefits coming on the most lightly loaded drive cycles. The 3.5 V-6 had lower fuel consumption than even the diesel in a few cases. On the lightest drive cycle (FTP City) at zero payload, the 3.5 V-6 was 7.5% better than the diesel. On the other hand, on highly loaded drive cycles at 25,000-pound GCW, the small V-6 was up to 36% worse than the diesel, despite offering lower performance. This result shows the diesel engine is better for applications where heavy loads are towed.

The diesel performs 15% to 28% better than the 6.2 V-8 at 50% payload, with the biggest advantages coming on the higher speed, more aggressive drive cycles. Lean GDI, VVA/VVL, reduced engine friction, and cylinder deactivation all provide more benefit in the pickup application than in the larger, heavier medium duty applications, except in the case where the pickup is towing a loaded trailer. The benefit of exhaust gas recirculation (EGR) on the gasoline engines is 3% to over 6% in the Ram, which is about the same as the benefits observed in the medium trucks.

The diesel engine shows a 4% to 8% benefit for reduced engine friction (only including components required to run the engine) at 50% payload, which is more than the benefit observed in medium trucks. Removing EGR and increasing turbocharger efficiency has an effect similar to that in medium trucks. A 4-cylinder diesel version, derived from the 6.7 liter 6-cylinder, was evaluated in the pickup. This engine is downsized by 33% both in size and in power, so performance is reduced (while still being significantly more powerful than pickup diesels of 20 years ago). The downsized engine provides results ranging from zero benefit on some cycles at 25,000 pounds GCW up to 12% better on the gentle FTP City cycle at zero payload. The 4-cylinder provides a 3.9% fuel savings on the aggressive US06 drive cycle at 50% payload.

The table below summarizes the fuel consumption reductions described above. Values in the table are for fuel savings of individual technologies, not for any combination of technologies. Technology combinations will be covered by a separate report.

Engine / Vehicle	Range of Fuel Savings	Notes
DD15 w/o WHR	0 – 5.7%	Downspeeding provided the largest benefits, mainly on low speed cycles
DD15 with WHR	2.7 – 6.0%	Largest benefit at high speed and payload. Includes asymmetric turbo + WHR
ISB Diesel	0 – 13%	Friction reduction and downsizing give largest benefits, both mainly at low speed, light load
6.2 L Gasoline V8	0 – 15%	Lean GDI gives the largest benefit, and works best at light load, but there are implementation issues
3.5 L Gasoline V6	0 – 15%	Lean GDI gives the largest benefit, and works best at light load, but there are implementation issues

T700 Tractor	0 – 15%	Reduced Cd gives the largest benefit at high speed, reduced Crr at 55 MPH
T270 Box Truck	0 – 11%	Reduced Crr and AMT transmission provide the largest benefit
F-650 Tow Truck	0 – 11%	Reduced Crr and AMT transmission provide the largest benefit
Ram Pickup	0 – 7%	Reduced Crr and high efficiency 8-speed automatic provide the largest benefits

Section 3.4 addresses the trade-offs between engine-out NOx and fuel consumption. Diesel engines have always had a strong relationship between these two key parameters. Over time, the trade-off has been improving, with a better NOx / BSFC trade-off possible today than in the past. The addition of selective catalytic reduction (SCR) by nearly all diesel engine manufacturers beginning in MY2010 has greatly reduced the trade off in fuel consumption caused by the requirement to reduce NOx emissions to 2010 levels, by allowing higher engine-out NOx. Further improvements in air handling and controls refinements should provide some additional improvement of the trade-off of fuel consumption against engine out NOx in the future, but the fact remains that low engine-out NOx (below 2 g/bhp-hr, and especially near 0.2 g/bhp-hr) extracts a significant fuel consumption and CO_2 penalty. Potential future lower NOx regulations are likely to require a reduction in engine-out NOx from current levels, and thus are likely to extract a fuel consumption penalty.

As stated above, SCR systems with high conversion efficiency can reduce the need for low engine-out NOx. However, in this case, fuel consumption is traded for urea (Diesel Exhaust Fluid - DEF) consumption. Typically, each gram of engine-out NOx requires the use of a volume of DEF equal to 1.4% of fuel. Thus, an engine running 2g/hp-hr engine-out NOx will consume DEF at a rate of about 2.8% of fuel burn. At early 2015 fuel prices, this DEF will cost about 2.4% of fuel cost. If minimum fuel consumption and GHG values are the goal, a high engine-out NOx level can be combined with a very efficient SCR system and high DEF consumption. This drives up the total fluid cost to the operator, however, so OEMs are likely to prefer lower engine-out NOx levels that offer the lowest total fluid consumption.

Section 4 evaluates testing and simulation approaches. The first subsection covers fuel efficiency metrics. Current regulations use fuel consumption in gallons per ton-mile. Other units that can be considered include fuel consumption per passenger-mile (for busses) and fuel consumption per unit volume-mile (for trucks that normally operate fully loaded, but at less than the legal weight limit (cubed-out). Longer, heavier truck combinations can provide large increases in the denominator (tons or cubic volume) for a relatively modest increase in the numerator (fuel consumption), with the potential for large increases in freight efficiency.

Section 4.2 assesses the ability of test and simulation procedures to quantify fuel consumption effects for individual technologies. Some technologies can be readily quantified by testing or simulating a complete vehicle or engine. Many smaller opportunities for increased efficiency, however, produce benefits too small to be reliably measured in a full engine or vehicle test. For these technologies, laboratory rig tests, combined with duty cycle data from actual vehicle use, can provide an accurate picture of benefits. Another issue that is discussed is technologies such as downspeeding, where the benefit may not be fully realized on an engine test cycle. A combination of engine test and vehicle drive cycle simulation may be needed to fully quantify the benefit of certain technologies. The section contains a list of technologies that may not be fully captured by the existing GEM model and engine certification protocol.

Section 4.3 looks at regulatory approaches in China, Japan, and Europe. Each of the countries and regions is adopting a different regulatory approach. To the extent that these different regulations work in ways that assign different benefits to a given technology, this can have the effect of driving different technical solutions in different markets, and increasing complexity for manufacturers. Some of the variation in regulatory approach is driven by differences in local market factors such as fuel price or length and weight regulations, but some differences are due to different regulatory philosophies.

Section 4.4 addresses certification of tractor-trailer vehicles. Trailers represent about 30% of the overall vehicle rolling resistance, so including trailer tire rolling resistance in the regulation will cover the 30% of rolling resistance not covered by current regulations. Requirement of SmartWay level aerodynamic features on a trailer provides about 5% fuel savings on highway operation. A portion of today's fleet already uses trailer aerodynamic features and low rolling resistance tires, and the share of these features in the fleet is growing. As a result, regulatory analysis should not claim all of the benefits from adding these features to the regulatory requirement.

Sections 4.4 and 4.5 provides data comparing the current regulatory cycles for certifying engines (the SET and FTP cycles) to actual truck fleet duty cycle data. There is a significant mismatch, which suggests that better regulatory cycles could be developed. There is a discussion of the potential for extending regulations for vocational trucks beyond the current parameters of engine efficiency and tire rolling resistance.

Section 4.6 describes how different technologies perform over a range of duty cycles. Some technologies perform best on drive cycles that emphasize low speed, light load engine operation, while others prefer high speeds and loads. A few technologies have performance that is almost independent of duty cycle.

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LIST OF ABBREVIATIONS AND ACRONYMS

6X2....................Tractor with a front axle, a drive axle, and a non-driven axle
6X4....................Tractor with a front axle and dual drive axles (tandem)
A/C....................Air Conditioning
AES...................Automatic Engine Shutdown
AFR ....Air/Fuel Ratio
ALVW………..Vehicle test weight for pickup trucks equal to the empty weight plus half of the payload that can go in the bed, with no trailer
AMT ....Automated Manual Transmission
APU ....Auxiliary Power Unit
BMEP..............Brake Mean Effective Pressure (A unit to compare the relative load on engines of different size)
BSFC ....Brake Specific Fuel Consumption
CAFE................Corporate Average Fuel Economy
CARB ....California Air Resources Board
Cd ....Coefficient of Drag (Aerodynamic drag)
CFD ....Computational Fluid Dynamics
CH_4 ....Methane
CILCC.............Combined International Local and Commuter Cycle
CNG..................Compressed Natural Gas
CO.....................Carbon Monoxide
CO_2 ....Carbon Dioxide
Crr ....Coefficient of Rolling Resistance (Tire rolling resistance)
DD15.................Detroit 15 liter heavy duty truck engine (formerly Detroit Diesel)
DEF...................Diesel Exhaust Fluid (Urea mixture used in SCR catalysts)
DPF...................Diesel Particulate Filter
E10....................Gasoline with 10% ethanol content
ECM ....Engine Control Module
EGR..................Exhaust Gas Recirculation
EPA...................United States Environmental Protection Agency
EVO ....Exhaust Valve Opening (Valve timing)
F-650.................Ford Class 5 and 6 truck model
FMEP ....Friction Mean Effective Pressure (Unit for comparison of friction between different engines)
GDI ....Gasoline Direct Injection
GEM ....Greenhouse gas Emissions Model (EPA tool for determining compliance with truck GHG regulations)
GCW.................Gross Combination Weight (Weight of the vehicle and trailer combined)
GHG ....Greenhouse Gas (CO_2, N_2O, CH_4, and others. In this report, CO_2 is the focus)
GT-POWER ....Commercial 1-dimensional engine simulation code. Part of GT-SUITE.
GVW.................Gross Vehicle Weight
GVWR.............Gross Vehicle Weight Rating (Vehicle mass with maximum allowed payload)

LIST OF ABBREVIATIONS AND ACRONYMS (CONT'D)

HCCI ....Homogeneous Charge Compression Ignition
HD.....................Heavy Duty (Typically refers to Class 8 trucks with engine of 10 liters or more displacement)
HPCR ....High Pressure Common Rail (Diesel fuel system)
ICCT.................International Council on Clean Transportation
IEA ....International Energy Agency
ISB ....Cummins 6.7 liter diesel engine (also available as a 4.5 liter 4-cylinder)
IVC ....Intake Valve Closing (Valve timing)
LD ....Light Duty (Typically refers to Class 2b and 3 trucks. Note that to passenger car manufacturers, Class 2b and 3 are called "Heavy Duty". This leads to considerable confusion between people with car and truck backgrounds.
LTC ....Low Temperature Combustion
MD....................Medium Duty (Typically refers to Class 4 through "Baby 8" trucks with engine displacements below 10 liters)
mm....................millimeter
MY....................Model Year
N_2 ....Nitrogen
N_2O ....Nitrous Oxide
NOx...................Nitrogen Oxides
NAS...................National Academy of Science
NESCCAF........Northeast States Center for a Clean Air Future
NH_3 ....Ammonia
NHTSA............National Highway Traffic Safety Administration (Responsible for fuel economy regulations)
NREL ....National Renewable Energy Laboratory
NMHC.............Non-Methane Hydrocarbons
NO.....................Nitric Oxide
NO_2 ....Nitrogen Dioxide
NOX..................Oxides of Nitrogen
O_2 ....Oxygen
DOC..................Diesel Oxidation Catalyst
ppm...................Parts per Million
PFI ....Port Fuel Injection
PM ....Particulate Matter
RCCI ....Reactivity Controlled Compression Ignition
rpm ....revolutions per minute
SCR...................Selective Catalytic Reduction
SwRI ....Southwest Research Institute
T270..................Kenworth Class 6 truck model
T-700.................Kenworth Class 8 long haul tractor model
TCPD................Turbocompound
VIUS ....Census Bureau Vehicle Inventory and Use Survey
VSL...................Vehicle Speed Limiter (also called road speed governor)
VVA/VVL ....Variable Valve Actuation/Lift (Variable lift and duration)

LIST OF ABBREVIATIONS AND ACRONYMS (CONT'D)

VVT ....Variable Valve Timing (Typically cam phasing, but constant lift & duration)
WHR.................Waste Heat Recovery
WHSC ....World Harmonized Steady-State Cycle (An engine dyno test cycle)
WHTC..............World Harmonized Transient Cycle (An engine dyno test cycle)
WHVC..............World Harmonized Vehicle Cycle (Truck test cycle with urban, rural, and motorway segments)

1.0 INTRODUCTION

In 2011, the National Highway Traffic Safety Administration (NHTSA) and Environmental Protection Agency (EPA) jointly issued a first phase of fuel efficiency and greenhouse gas (GHG) standards that apply to medium- and heavy-duty on-highway engines and vehicles for model years (MY) 2014 to 2018 and beyond. These regulations are commonly referred to as "Phase 1" of the Heavy-Duty National Program. The standards cover all vehicles in weight classes 2b through 8, which encompasses most vehicles with gross vehicle weight ratings (GVWR) over 8,500 pounds except for a limited number of passenger vehicles covered under the light duty corporate average fuel economy (CAFE) standards, and recreational vehicles, which were included in EPA's GHG standards but not NHTSA's fuel efficiency standards. Phase 1 has two implementation stages. EPA's greenhouse gas emission standards are mandatory beginning with model year 2014. NHTSA's fuel consumption standards are voluntary in model years 2014 and 2015, becoming mandatory with model year 2016 for most regulatory categories. Commercial trailers were not regulated in Phase 1. The Phase 1 GHG and fuel consumption standards were developed using input from a number of studies which evaluated the fuel saving technologies that are available, such as the NESCCAF 2009 report [1] and the NHTSA and NAS 2010 reports [2, 3].

The research project described in this report has been completed for NHTSA to help to inform the next phase ("Phase 2") of the regulations, which would set standards in coordination with EPA for sometime beyond model year 2018. In order to prepare for Phase 2, NHTSA directed SwRI to update prior research on fuel saving technologies to reflect the effects of the Phase 1 regulations, as well as to include technical progress that has been made over the last few years. In particular, SwRI was tasked with assessing the current commercial fleet technology baseline at the time of contract award (MY 2011/2012), projecting the post-Phase 1 technology baseline (MY 2018), and assessing the effectiveness and cost of potential fuel efficiency/GHG improving technologies for the Phase 2 timeframe (post MY 2018).

When considering potential fuel efficiency/GHG-reducing technologies, NHTSA directed SwRI to include a range of factors: design, functionality, duty cycle, use (type of work done by the vehicle), and factors that can influence the effectiveness, feasibility, and cost. Vehicle utility and performance are also to be considered.

NHTSA issued two requests for proposal (RFP) documents via the General Services Administration (GSA) in 2012. The first document, DTNH22-12-R-00599, was issued on August 9, 2012. A revised version, with the same name plus the suffix CAFE-9-1-2012, was released on September 1, 2012. In the RFP documents, NHTSA described two main purposes for doing a new study. First, the work should update the key findings of the prior NHTSA and NAS studies (which provided technology projections up to MY 2020) so that it could help the agency to develop standards beyond MY 2020. Secondly, the work should provide updated findings and analysis that could be used to inform the following regulatory considerations:

- Appropriate test procedures and methodologies for measuring fuel efficiency of MD and HD vehicles
- Appropriate metrics for measuring and expressing fuel efficiency
- The range of factors, including, but not limited to, design, functionality, use, duty cycle, infrastructure, and total overall energy consumption and operating costs, that affect MD and HD vehicle fuel efficiency
- Other factors and conditions that could impact a program to improve MD/HD fuel efficiency

Since the regulation of MD/HD engine and vehicle fuel efficiency/GHG emissions is relatively new, a "learning curve" to some extent is to be expected. One of the goals of this research project is to use the Phase 1 regulations to look for and evaluate areas for improvement for the Phase 2 program. Certain aspects of the Phase 1 regulations may provide learning opportunities for Phase 2. For instance, there is the possibly that the existing regulations do not fully capture significant fuel saving opportunities that exist and can be cost-effectively achieved. An example of this would be the potential fuel savings from aerodynamic and rolling resistance improvements to trailers, use of special low friction lubricants, or the use of automated manual transmissions.

The project has been divided into tasks. The first task was a literature survey, covering the following topics:

- Fuel saving technologies for MD and HD engines and vehicles
- Market segmentation of fleets
- Current and planned fuel economy regulations in markets around the world

At the end of the literature review task in January 2013, NHTSA, EPA and SwRI agreed on a list of vehicle and engine technologies that form the main subject of this project. The parties also agreed on the selection of engines and vehicles to be used in the project. The selection was constrained to include only engines and vehicles for which extensive experimental data was available. One vehicle type that was not included as a result of this criteria was a Class 8 straight truck. This is a tandem axle straight truck typical of dump trucks, concrete trucks, and waste haulers, among other applications.

The second task requires an analysis of the range of fuel efficiency and GHG reduction performance for technologies that were selected at the end of the literature review task. SwRI used the SwRI Vehicle Simulator tool, a vehicle simulation tool developed in-house, to model vehicle performance over a range of drive cycles. A range of vehicle models has been created to cover the range of Class 2b through Class 8b. There was an increased emphasis on vocational truck applications, since less information is available on the performance of these vehicles as compared to larger segments such as long haul. The commercial software GT-POWER (Gamma Technologies, Inc.) was used to model engine performance, fuel consumption, and CO_2 emissions over the full speed-load range. A range of two gasoline and two diesel engines have been simulated, with additional permutations to cover gaps in the engine size range. Note that GT-POWER is not an appropriate tool for evaluating other greenhouse gas emissions, such as N_2O or CH_4, so these are not addressed by this project.

The third task is a cost effectiveness analysis of the efficiency and emissions reduction technologies for the Phase 2 timeframe. A subcontractor, Tetra Tech, Inc. performed the cost analysis, and their results are to be provided in a separate report. The cost analysis also includes indirect costs and benefits that may occur with the various technologies.

A fourth task includes a review of fuel efficiency metrics that take vehicle work and use into account. It also includes a review of engine efficiency test procedures, vehicle efficiency test procedures (whole vehicle on-road, chassis dyno tests, etc.), and efficiency simulation approaches.

The final task calls for post-report support of NHTSA, including the provision of presentations, meeting support, and documents. This task will not be included in either final report. The remaining tasks will be covered by Final Report #2, which is due to NHTSA in early 2015.

2.0 LITERATURE REVIEW

Several researchers were involved in the literature review. There was no specific search methodology prescribed, so each section author used his own background and favorite search approach. The goal is to have a representative overview of literature related to the topics of concern, but not to have every possible paper included.

The scope of this review has included literature pertaining to light, medium, and heavy trucks. Passenger car studies have been included only in cases where the technologies and fuel economy benefits could be directly applied to light truck applications as well. Transit bus and motor coach applications were not included.

The majority of this work was done in calendar years 2012 and early 2013. As a result, the majority of references are through 2012. A few later references are included based on specific topics added during document review and revision. These have been added by exception, and do not consider the full scope of 2013 and 2014 publications.

2.1 Market Segmentation

Several approaches were considered in the development of market segmentation for commercial vehicles. The National Research Council, which is a part of the National Academy of Sciences, used an approach based primarily on weight class [VS-1].* The U.S. Census Bureau formerly provided detailed vehicle inventory data with populations by weight class and application. The information contained in the Census Bureau Vehicle Inventory and Use Survey (VIUS) study is excellent. It includes detailed information about the fleet size, types of operations, annual vehicle miles traveled, and more. Unfortunately, the final year for which such data was compiled was 2002 [VS-2]. After reviewing the various possible approaches, SwRI and NHTSA agreed to use the market segmentation approach developed by CalHEAT [VS-3, VS-4]. CalHEAT agreed to provide technical support, and the segmentation used in this project is as detailed in references VS-3 and VS-4. The segmentation was developed by CalHEAT in conjunction with industry representatives during a series of web meetings, the results of which are documented in VS-3 and VS-4.

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* Designations in brackets indicate the references used in the study, and summarized in this literature review. The references are organized by section, using the following prefixes:

VS	Vehicle Segmentation
R	Regulations
ET	Engine Technology
VT	Vehicle Technology

The CalHEAT approach to Class 2b – 8 vehicle segmentation involves 6 truck categories, which are listed below:

1. Class 2b/3 Pickups and Vans
2. Class 3-8 Urban Vocational Work Trucks
3. Class 3-8 Rural/Intracity Work Trucks
4. Class 3-8 Work Site Support Trucks
5. Class 7-8 Short and Regional Haul Tractors
6. Class 7-8 Over the Road Tractors

2.2 Regulations

The emission and CO_2 regulatory requirements provide a necessary framework from which the project was conducted. The regulatory standards studied, and the sources used, are summarized in this section. In the following subsections, the terms fuel *economy* and fuel *efficiency* may both be used. Fuel *economy* is measured in distance traveled per unit of fuel consumed, such as miles per gallon. Fuel *efficiency* is inversely proportional to fuel *economy*, and is expressed in terms of units of fuel consumed per distance traveled, such as gallons per 100 miles. If a change is introduced that increases efficiency by 25%, this means that a given transport task can be completed using 25% less fuel. Units of fuel *economy* overstate the benefit. A 25% increase in *efficiency* (25% decrease in fuel consumption) equates to a 33% increase in fuel *economy*. Figure 2.1 shows the relationship between fuel economy changes and fuel consumption / GHG emissions changes. For small changes, the values for change in fuel economy and fuel consumption are almost equal, but for larger changes, the values diverge rapidly. A 10% increase in fuel economy equals a 9.1% reduction in fuel consumption, but a 100% increase in fuel economy only represents a 50% reduction in fuel consumption. Some of the results described in sections below will be in terms of fuel economy, while others are in terms of fuel consumption.

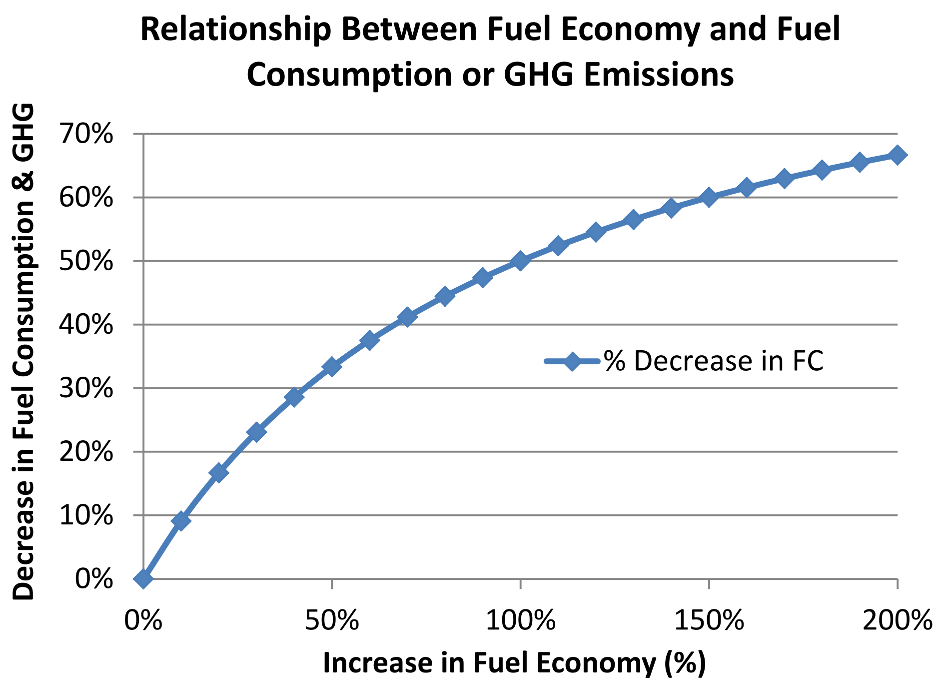

FIGURE 2.1 RELATIONSHIP BETWEEN CHANGES IN FUEL ECONOMY AND CHANGES IN FUEL CONSUMPTION AND GHG EMISSIONS

2.2.1 North American Fuel Economy Regulations

Central to the project are the current U.S. medium- and heavy-duty vehicle fuel efficiency and greenhouse gas (GHG) regulations, developed jointly by the NHTSA and the U.S. Environmental Protection Agency (EPA), and available at the web site, www.regulations.gov [R-1]. Supporting documentation, including the Regulatory Impact Analysis and further background studies, is also available on the NHTSA web site [R-2]. A NRC report provided independent technical recommendations for NHTSA to consider when developing the regulatory approach for Phase 1 [R-3], with supporting NRC data [R-4]. Canadian regulations are aligned with the U.S. 2014-2018 regulations, but are expressed only in terms of GHG limits, and not fuel consumption limits [R-18].

2.2.2 Worldwide Fuel Efficiency Regulations

The work being done in this study was also placed in the context of worldwide regulations and regulatory trends. A review was conducted of studies evaluating fuel efficiency regulations worldwide as part of this literature review. An overview of worldwide current and proposed regulations was presented in slides by Bandivadekar of the International Council on Clean Transportation (ICCT) [R-10]. A more recent slide summary was presented by Muncrief, from the same organization [R-19].

The challenges facing implementation of fuel efficiency standards, and the intended timeline were summarized in a slide presentation by Wang [R-9]. A more recent publication that provides the best available reference regarding development of the Chinese heavy-duty vehicle fuel efficiency test procedure was prepared by Zheng and co-authors from several agencies [R-14]. Fuel consumption regulations and test procedures in China are briefly summarized in a three-page document produced by the Chinese Ministry of Industry and Information Technology (MIIT) [R-5]. Base vehicle models are tested on a chassis dynamometer, using the Chinese low power variation of the World Harmonized Vehicle Cycle (C-WHVC) [R-15]. Variants of the base model have the option of being tested using a chassis dynamometer or by using simulation. Further test details are reported regarding parallel test standards administered by another Chinese government agency [R-16], and further details of current and proposed standards have been summarized [R-17].

Information regarding the 2015 Japanese regulations is summarized in the report of activities chaired by Ikegami [R-11]. The "Top Runner" program was instituted by Japan's Ministry of Economy, Trade and Industry, Agency for Natural Resources and Energy, with the stated objective of "developing the world's best energy-efficient appliances." It addresses a wide range of industries and appliances, with freight vehicles covered in Section 7.2 of the summary report [R-12]. Test procedures for fuel consumption calculation are summarized in a slide presentation from Japan's National Traffic Safety and Environment Laboratory [R-13]. The two driving cycles, JE05 and the Interurban driving mode are combined using weighting factors, and fuel consumption is calculated based on engine fuel maps. The overall impact of the Top Runner regulation is to drive improvement in engine efficiency, but not in vehicle efficiency or power demand.

It would be very useful to have a comparison of world-wide regulations in the same terms, so that the relative stringency of different regulations could be evaluated. Unfortunately, regulations are, in practice, very hard to compare. The US Phase 1 regulations, for example, cover engines and vehicles separately. There is a CO_2 and fuel consumption standard for engines, and a separate standard for vehicles. US engine standards are in units of emissions per unit of work. The vehicle standard assumes a generic engine fuel map provided by the regulators, and are in terms of emissions per unit of distance traveled. The Chinese standard, on the other hand, measures vehicle fuel consumption on a given cycle, at a payload determined by the regulation. The vehicle fuel consumption is influenced by the engine used, the transmission and driveline, and all other aspects of the vehicle. It is impossible to separate out engine and vehicle requirements in the Chinese standard, because there are no separate requirements.

The European Union is in the process of developing an approach to future regulation of truck fuel consumption and CO_2 emissions. The EU approach will use a sophisticated vehicle simulation model to project emissions over a range of drive cycles.

2.2.3 Fuel Efficiency Test and Analysis Methodology

The primary methodology for this study was to use engine and vehicle simulation approaches and drive cycles to estimate the fuel efficiency improvements of different technologies across applicable vehicles, loads, and drive cycles. This project is based on simulation, with experimental validation where existing test results are available. SwRI made an effort to include simulation methods and drive cycles that have been validated by previous researchers. For instance, SwRI is conducting a parallel project for EPA that includes developing a correlation between full vehicle on-road tests, chassis dynamometer tests of full vehicles, and powertrain tests conducted in a test cell. The work by Sharpe and Lowell [R-6] provides a good, high-level overview comparing the drive cycle fuel efficiency methodologies.

In setting the stage for a more detailed assessment of specific engine and vehicle technologies, the International Energy Agency prepared a technology roadmap [R-7]. Pages 23 through 28 of [R-7] discuss heavy-duty vehicles, and Table 8 lists the available technologies, estimated fuel efficiency improvements, technology costs, and provides an assessment of technology readiness. Table 9 estimates technology payback times in various applications. In a related publication, the IEA presents a proposed policy plan in support of vehicle fuel efficiency improvement [R-8].

2.3 Engine Technologies

This section summarizes the findings of the literature search on efficiency gains resulting from engine and aftertreatment development. It is divided into five sub-sections: Engine Systems, Air Handling, Combustion, Aftertreatment, and Friction/Parasitics. The first sub-section begins with a look at spark-ignition engines and the transition from spark-ignition to diesel engines. It is applicable only to light-duty applications (Class 2b/3). The remainder of the Engine Systems sub-section and the sub-sections that follow address diesel engines, in light, medium, and heavy-duty applications.

2.3.1 Engine System

2.3.1.1 Direct-Injection

The transition from port-injection to direct-injection in spark-ignition engines began several years ago, and is expected to continue, with most new engines following this trend. Attractions include a two to three percent volumetric efficiency improvement and the ability to increase the compression ratio by one to two numbers, or around 10 to 20 percent [ET-14]. These capabilities are utilized primarily to increase specific output, but the latter may provide a one to two percent fuel efficiency improvement.

2.3.1.2 Lean Direct-Injection

Another approach to direct injection spark-ignition engines that was taken in its early development is that of the lean burn, stratified charge combustion system. This approach held the promise of greater efficiency gains, but the lean NO_X aftertreatment requirements negated most of the gains, especially under light duty cycle conditions, when the exhaust temperature was insufficient for NO_X trap regeneration.

Several researchers have demonstrated the fuel economy improvement potential of the lean burn direct-injection engine. Stovel and his colleagues ran back-to-back fuel economy tests over five driving cycles, using a 1998 Toyota equipped first with its production port-injected engine, and then with lean direct-injection. Fuel economy improvements ranged from 3.7 percent on the U.S. highway cycle to 16.5 percent on the New York City Cycle and 17.2 percent on the Japan 10-15 cycle [ET-26]. Alkidas obtained similar results, and identified the primary source of the fuel economy improvement as reduced pumping at part load; this explains the far greater benefit seen in urban driving cycles as compared to highway driving. Further benefits were attributed to the higher compression ratio achievable with direct injection, and the reduced heat transfer of a stratified charge engine [ET-27].

Most of the work on stratified charge direct injection engines was done from the late 1990s, when several engines of this type were introduced to production in Europe and Japan, through the early 2000s. At that point the emphasis shifted to stoichiometric direct injection engines as it was concluded that the lean NOx aftertreatment systems could not be made sufficiently efficient for light duty cycle applications. Baumgarten and his colleagues demonstrated a 22 percent fuel consumption improvement over the European city driving cycle, but concluded that "New technologies and strategies in the field of exhaust gas aftertreatment are required" [ET-28]. Stovel and his colleagues continued their studies with evaluation of aftertreatment efficiencies versus duty cycle, and concluded that only by adding grade loads to the duty cycles could they gain sufficient aftertreatment efficiency [ET-29].

Another approach to lean, or dilute combustion is utilizing exhaust gas recirculation under not only part-load but also full-load operation. Researchers at Southwest Research Institute have developed this concept in the form of High Efficiency Dilute Gas Engines (HEDGE), and more recently as Dedicated Exhaust Gas Recirculation (D-EGR). An early modeling study projected diesel-like fuel efficiency in light and medium truck applications [ET-37]. More recent studies have reported fuel efficiency improvements of between five and 30 percent over the engine operating map [ET-38, ET-39]. Under most operating conditions the reported gains are on the order of five percent, while the larger gains are seen under full-load conditions, where air-fuel ratio enrichment would normally be utilized.

2.3.1.3 Cylinder Deactivation

This technology has been included only as a spark-ignition technology as it is difficult to implement in highly turbocharged engines due to turbocharger surge problems [ET-18]. The fuel consumption improvement in spark-ignition engines is estimated at 5 to 6.5 percent [ET-21]. This is achieved through the reduced pumping losses associated with firing only a portion of cylinders under part-load conditions, and thus working the remaining cylinders at a higher load, lower pumping-loss condition. The benefit of cylinder deactivation depends on factors including engine size, the maximum BMEP where a benefit can be obtained, vehicle power demand, and drive cycle. Results presented in Section 5 below will show that smaller engines in heavier vehicles achieve smaller benefits, and that for highly loaded duty cycles, the benefit can be zero.

2.3.1.4 Throttle-Free Operation

Another technology that has been used to reduce light-load throttling losses in spark-ignition engines is "throttle-free" operation. Variable valve event lift and timing are used to control engine load in lieu of intake throttling. This system has been developed by BMW, and they report a 20 percent specific fuel consumption reduction under very light load operating conditions. A ten percent tank mileage improvement is reported over passenger car driving cycles. The mileage improvement drops as vehicle duty cycle increases [ET-35].

2.3.1.5 Downsizing and Boosting

Simultaneously downsizing and turbocharging a direct-injected spark-ignition engine is widely seen as an important fuel efficiency improvement strategy. The Ford EcoBoost engine is a well developed example, and Ford reports a twelve percent fuel economy improvement as compared to a larger displacement port-fuel-injected engine in the same application [ET-23]. The result is reported over the FTP-75 light-vehicle test cycle. Note that actual results for this approach will vary from engine to engine, and are highly sensitive to drive cycle. Boggs reports a 17 percent fuel consumption improvement through downsizing, but this is a research trends presentation, and it is not clear how much of the reported gain is proven [ET-3].

2.3.1.6 PFI to Diesel

A 25 to 30 percent fuel economy improvement (20 to 23% fuel *efficiency* improvement) is generally accepted as the magnitude attributable to replacing a port-injected gasoline engine with a direct-injected diesel engine. Cummins reports a 30 percent fuel economy improvement versus gasoline in a light truck/SUV chassis [ET-5]. This study was done using a direct comparison between the two drivetrains over the North American driving cycle.

The remaining items listed in this section pertain to further efficiency improvements that are believed achievable through diesel engine optimization. Overall gains are quite consistently reported as between twenty and twenty-five percent, with the latter number including idle-off and parasitic reductions. It is important to note that the individual improvements listed below are not completely additive as the result may include combined effects (VVA and/or Miller Cycle and turbocharging, for example), or technologies may be ones that cannot be applied together without reducing the efficiency gain of each (turbocompounding and a bottoming cycle, for example). The DOE newsletter presents an overall summary of diesel efficiency objectives, highlighting the twenty percent improvement goal (from 42 to 50 percent brake thermal efficiency (BTE)) of the SuperTruck program [ET-6]. Note that the SuperTruck program engine goal focuses on achieving 50% brake thermal efficiency (BTE) at a single operating point, and that many of the technologies used in this program may not be commercially viable in the 2015 timeframe.

2.3.1.7 Idle Reduction

Idle shutdown technology is covered in the Vehicle Technologies section, but one study is listed here. A line-haul fleet field study, with idle shut-down and an APU, was conducted on a fleet comprised of 16 trucks. An override system was used that would not allow the APU to stay on when the engine was running. The result was an almost 16 percent fleet fuel economy improvement, from 6.04 to 6.99 MPG. This result was reported for the actual use of the sixteen trucks (not for a single, defined duty cycle). Percent idle time for each vehicle, before and after the idle shut-down feature was installed is reported. Prior to installation, the percent idle time ranged from 25 percent to nearly 70 percent. After installation the percentage ranged from less than one to 20 percent, with the majority of vehicles idling less than five percent of the time [ET-25].

2.3.1.8 Bottoming Cycle

Organic Rankine Cycle waste heat recovery (WHR) systems have been studied for many years. Their basic approach is to use engine exhaust waste heat to evaporate a working fluid in a boiler unit. The gas is passed through a turbine to create mechanical or electrical power, whereupon it is re-condensed prior to pumping it again into the boiler unit. Cost and complexity remain high, and along with package size and transient response challenges that limit the application to line haul vehicles only. In EGR engines they offer the benefit of eliminating the EGR cooler. To the extent that WHR systems use exhaust heat, they increase the overall cooling system heat rejection requirement, which can have negative impacts on cooling fan power needs, as well as on vehicle aerodynamics. Eckerle and Koberlein report potential efficiency gains from WHR on the order of six percent [ET-10, ET-16]. de Ojeda reports three to five percent gains [ET-7]. Greszler reports four to five percent [ET-12]. Sisken reports 4.5 percent [ET-18].

2.3.2 Air Handling

2.3.2.1 Exhaust Gas Recirculation (EGR)

At low levels EGR is close to fuel efficiency neutral in a diesel engine, but as EGR level is increased it has a negative impact on fuel efficiency through two mechanisms – increased back pressure, adversely impacting pumping work, and slowing down the combustion heat release rate. Taking a typical EGR system map as the baseline, Eckerle reports the potential for approximately three percent BSFC improvement through further system optimization [ET-10]. This is primarily through improved system effective flow area, reducing the adverse impact on back pressure.

It should be noted that even as aftertreatment becomes more effective, the most efficient solution in practice may retain some EGR, as noted by Koberlein [ET-16]. He also reports less opportunity for improvement through reduced EGR restriction (about 0.5 percent).

2.3.2.2 Turbocharger Efficiency Improvement

Both Czarnowski and de Ojeda estimate that further advances and optimization in turbocharger efficiency provide on the order of two percent fuel efficiency improvement potential [ET-4, ET-8]. Koeberlein reports about 0.5 percent over a driving cycle [ET-16].

2.3.2.3 Engine Breathing Improvements

The combined result of various breathing improvements may be estimated as the sum total of intake and exhaust pressure drop reductions divided by brake mean effective pressure (BMEP). Koeberlein reports 1.4 percent through optimization [ET-16]. Sisken projects a two percent fuel efficiency improvement through air handling system development [ET-18]. Jadin predicts almost four percent through a combination of variable intake valve closing timing (IVC), turbocharger efficiency and match improvements [ET-15]. A few plots in this reference show another four percent, but these are not explained.

2.3.2.4 Variable Valve Event Timing

The primary gain in diesel engines is achieved by varying the exhaust valve opening (EVO) event versus engine speed and load, in conjunction with turbocharger optimization to minimize blowdown losses. de Ojeda reports a 1.25 percent fuel consumption improvement [ET-8].

The Miller Cycle is sometimes considered to improve specific engine output, but there are no definitive studies that quantify its fuel consumption impacts. Attractions include its potential role in engine downsizing, but the technology requires a greater portion of the compression process to be conducted in the turbocharger compressor, at lower isentropic efficiency. Similarly, the Atkinson Cycle may be considered in its integration with hybrid drivetrains, and fuel efficiency benefits are reported as part of the impact of hybridization.

2.3.2.5 Turbocompound – Mechanical

On-highway demonstrations of this technology began in the early 1980s. Results are duty cycle dependent, and require significant time at high load to see a fuel efficiency improvement. Light load factor vehicles can expect little or no benefit. Greszler reports two to four percent fuel consumption improvement in line haul applications [ET-12].

2.3.2.6 Turbocompound – Electric

Further gains over the mechanical turbocompound system might be available by using a turbo-generator instead of conventional power turbine. These are attained through better vehicle integration and lower backpressure impacts. The power turbine speed is no longer linked to crankshaft speed, which allows more efficient operation of the turbine. De Ojeda reports on the order of a 1 to 1.5 percent efficiency improvement over mechanical turbocompound systems at 0.5 to 0.7 gm/hp-hr engine-out NOx levels, but dropping at lower engine-out NOx [ET-8]. Zero benefit is reported at 0.3 to 0.4 gm/hp-hr engine-out NOx, due to lower available temperature. Jadin reports a 1.6 percent fuel efficiency improvement, again as compared to a mechanical turbocompound system [ET-15].

2.3.2.7 Asymmetric Turbocharging

Detroit Diesel Corporation has recently replaced a mechanical turbocompounding system with an asymmetric turbocharger. This approach uses a twin entry turbocharger in which one portion is sized smaller than the other. The more restrictive side provides the back pressure required for EGR operation, while reducing the overall exhaust restriction. Fuel economy improvements have been reported relative to their earlier turbocompound engine. The fuel economy is reported as equivalent throughout the load range from full load to approximately 25 percent load. At lighter loads the fuel consumption is reduced, with a maximum reported reduction of five percent at ten percent load and below [ET-36].

2.3.3 Combustion

2.3.3.1 Low Temperature Combustion

While combustion efficiency drops slightly with lower combustion temperature, this is more than offset by a more favorable specific heat ratio during expansion, and reduced heat transfer, allowing greater work extraction during the expansion process. A significant increase in closed-cycle (gross indicated) efficiency has been demonstrated by several researchers. The remaining question is how much these gains might be negated by increased pumping work. The paper by Teetz provides a detailed evaluation of an HCCI engine in a multi-cylinder configuration; interestingly, fuel consumption changes go unreported [ET-20]. Zhang reports a 7.4 percent fuel consumption improvement at best operating conditions when using a dual fuel (E85/diesel) RCCI engine as compared to a baseline diesel engine [ET-24]. Jadin reports an improvement of greater than three percent at 5.8 bar [ET-15]. Jadin's presentation predicts overall 50 percent brake thermal efficiency with diesel fuel and 55 percent as a "bold goal" with dual-fuel engines. Wagner reports a 7.1 percent brake thermal efficiency improvement in a light duty (1.9 liter) engine at best operating point [ET-22]. Each of these researchers point out that low temperature combustion also comes with downsides, such as difficult control of combustion during transient operation, and increased combustion noise caused by high rates of heat release at the beginning of combustion and under high load conditions.

2.3.3.2 In-Cylinder Optimization

It's difficult to quantify efficiency improvements through combustion system optimization, since this is closely coupled with aftertreatment technology. For example, allowing higher in-cylinder NOx as SCR efficiency improves is an important avenue. The biggest lever for further optimization is rate shape control and injection split optimization as common rail technology advances. In an attempt to separate SCR from in-cylinder optimization one can look at specific fuel consumption changes at constant in-cylinder NOx. Eckerle reports between 3 and 3.5 percent specific fuel consumption improvement at constant NOx through fuel system rate shape optimization [ET-10]. Jadin reports on the order of four percent improvement through a combination of in-cylinder optimization approaches (CFD optimization, injection pressure and fuel system losses) [ET-15].

Another important aspect of in-cylinder optimization is the impact of combustion temperature on reaction kinetics. Two fundamental studies are included in the references that demonstrate limiting parameters regarding in-cylinder NOx control. The work of Hu explains a significant slowing of energy release as combustion temperature drops [ET-30]. Flynn's paper demonstrates the impact of the slowed reaction rates on in-cylinder efficiency [ET-31].

2.3.3.3 Increased Peak Cylinder Pressure

Continued development of cylinder head, piston, and head gasket technology allows increased peak cylinder pressure. This can be taken advantage of for efficiency improvements through increased compression ratio and increased rate of heat release. de Ojeda reports a one percent improvement of fuel efficiency resulting from compression ratio increase as reasonable in the projected timeframe [ET-7]. Koeberlein reports just under one percent in a driving cycle [ET-16]. de Ojeda reports a compression ratio impact of two percent at higher engine speeds [ET-8]. Sisken projects a fuel consumption improvement of just under five percent in HD applications through "downsizing" [ET-18]. This is later defined as running at higher BMEP. Sisken also projects a 1.5 percent improvement through compression ratio increase [ET-18]. It should be pointed out that increasing cylinder pressure often requires an extensive or complete redesign of the engine, and reliability/durability of the engine also becomes an issue.

2.3.3.4 Model-Based Control

A topic that is receiving attention as an aid to diesel combustion system calibration is model-based control. This approach replaces look-up tables in collecting transient engine test data and using it in "desk-top" calibration optimization. Several recent papers have demonstrated its efficacy in resource reduction and system optimization when developing diesel combustion system calibrations [ET-32 through ET-34]. It is important to recognize that there is no inherent fuel consumption improvement resulting from model-based control. Its ability to improve the optimization process may result in improved calibrations in some cases.

2.3.4 Aftertreatment

Two approaches involving aftertreatment systems can be applied to improve fuel efficiency: better combustion system optimization through increased aftertreatment efficiency, and reduced backpressure through further development of the devices themselves. de Ojeda reports a seven to eight percent improvement projected through a combination of higher cylinder pressure, injection optimization, and engine/aftertreatment optimization [ET-8]. Koeberlein reports a 0.5 percent improvement through improved aftertreatment flow (catalyst size optimization and improved NOx surface utilization) [ET-16]. Sisken projects a two percent fuel efficiency improvement through reduced EGR. Thinner wall DPF, improved SCR cell density, and catalyst material optimization allow greater NOx conversion efficiency in the aftertreatment, reducing the amount of EGR required. [ET-18].

2.3.5 Friction and Parasitic Losses

Piston, ring, and bearing friction are not separated in most studies. de Ojeda identifies a combined improvement of up to two percent through reduced bearing friction, reduced piston and ring friction, and unspecified lube pump improvements [ET-7]. In his 2012 follow-up paper he reports 5.5 percent through a combination of friction reduction and both lube and cooling system improvements [ET-8]. Later in the same presentation he specifies 0.45 percent demonstrated through water pump improvements and 0.3 percent through lube pump improvements. The total number of 5.5 percent seems optimistic, but is a projection of further improvements. Koeberlein reports a combined number of 3 percent [ET-16]. Sisken reports a combined number of two percent, with 0.5 percent coming from improved water pump efficiency [ET-18]. de Ojeda reported a 1.3 percent improvement in HD diesel fuel efficiency over the NA HD on-highway cycle [ET-17]. Jadin shows a 0.9% benefit for a variable speed water pump and variable displacement oil pump; piston/ring/liner friction reduction as 0.5 percent; bearing friction reduction as 0.6 percent [ET-15]. It should be noted that water pump improvements include both pump efficiency improvement, and variable speed or on/off controls. Lube pump improvements are primarily achieved using variable displacement pumps and may also include efficiency improvement.

Engine downspeeding is mentioned by Sisken but without description or quantification [ET-18]. The challenges to downspeeding include: higher driveshaft and axle torques, maintaining drivability and grade capability at lower cruise RPM, and engine durability when running at higher BMEP. Downspeeding is identified by both Cummins and Volvo as holding fuel economy advantages on their current web sites:

http://cumminsengines.com/smartadvantage?%20-%20overview#overview
http://www.volvotrucks.com/trucks/na/en-us/products/powertrain/xe/Pages/xe.aspx

2.3.5.1 Variable Displacement Lube Pump / Variable Speed Water Pump

Sliding-vane, variable displacement lube pumps are just starting to see production in new European diesel passenger cars. Analysis shows lube pump driving power to be cut in half at highest engine speeds. This in itself is reported by Excell as reducing pump power demand by two to three kilowatts at rated speed in a heavy-duty diesel engine (approximately one percent) [ET-11]. Larger improvements are seen if the engine also uses piston cooling nozzle cut-outs for light load, high speed operation. Large gains are reported under cold start conditions (six to eight percent) [ET-1, ET-17, ET-19].

One reference reported a 2.2 percent fuel efficiency improvement through piston, ring pack, and bearing optimization, and the addition of a variable displacement oil pump. The results were over the European passenger car driving cycle [ET-2]. The DD15 water pump uses a variable speed viscous clutch controlled by the engine management system based on engine load and coolant temperature [ET-9].

2.3.5.2 Flow Circuit and Thermostat Advances

An interesting approach to flow circuit and thermostats is taken in the new Volkswagen TSI gasoline engine. Two thermostats were used, with separate flow circuits around the cylinder walls, and through the oil cooler and cylinder head. Flow around the cylinder walls is separately regulated to keep wall temperatures at a more uniform (and generally higher) temperature, and to reduce cylinder wall lubricant viscosity. While this is a small gasoline engine, the same concept could be applied in small, lower duty cycle diesel engines [ET-13].

2.4 Vehicle Technologies

An extensive study of reference material pertaining to vehicle technologies was conducted on this project. Vehicle technologies include those affecting aerodynamics, rolling resistance, transmission and driveline after the engine, and vehicle auxiliary loads are included in this section. Because many of the potential improvements are highly dependent on both the vehicle type and its application or duty cycle, specific fuel efficiency projections are for the most part not summarized in this section. Examples from specific applications can be found in many of the references, and information from these references was applied to the analyses reported elsewhere in this project.

2.4.1 Identification of Vehicle Technologies

The National Research Council [VT-1, VT-2] and Department of Energy [VT-3] summaries provide recent evaluations of the technologies available for vehicle efficiency improvement. These reports provide a starting point to gain perspective on the work being done through such programs as the 21st Century Truck Partnership. The final rule for medium and heavy duty fuel efficiency includes a further listing of fuel efficiency improvement technologies [VT-5]. Also addressed [VT-5] are questions pertaining to alternative standards that may better include vehicle technologies not accounted for in the current rules.

A CalHEAT study [VT-4] identifies further vehicle technology pathways including electrification, low carbon and alternative fuels, powertrain efficiency, hybridization, mass and drag reduction. Also considered in this study is improving the efficiency of truck utilization. Annual vehicle mileage and ton-mpg values for medium and heavy duty applications are addressed in several of the studies [VT-11, VT-14].

A 2009 NESCCAF study focused specifically on engine and vehicle technologies for long haul applications [VT-6]. It was further summarized, including technologies anticipated through the 2017 requirements in a study led by Cooper [VT-7]. Projections toward further, future technology implementation are provided by Kobayashi [VT-9]. The timeframe to 2050 is addressed in the Kobayashi study. An IEA study provided projections to the year 2050 that included especially useful graphical depictions of the impact of fuel efficiency improvement technologies by vehicle type and expected implementation date on fuel usage and GHG mitigation [VT-12].

Emphasizing cost/benefit analysis, Harrington identified problems capturing and quantifying the vast array of vehicle fuel efficiency improvement technologies with the current fuel consumption standards. These problems are driven by the wide range of applications and duty cycles encountered in the field. The work covered the vehicle range from Class 2B through Class 8 [VT-8]. Alternative approaches are presented with an emphasis on capturing improvements resulting from further vehicle technologies.

In a study by Saricks and colleagues, potential market penetration of various technologies is considered [VT-10]. A base case, in which innovation proceeds at its current pace, and an accelerated implementation pace, are considered and compared through the year 2020. Both engine and vehicle technologies are considered.

A particular medium duty vehicle was evaluated in an Argonne study [VT-13]. Technologies including aerodynamic drag reduction, rolling resistance reduction, transmission improvements, and vehicle weight reduction were applied to a baseline vehicle. Each technology was considered individually, and then various technology groupings were studied.

An Oregon state government study was previously cited for its consideration of vehicle utilization efficiency [VT-14]. This extensive study also covers a wide range of vehicle technologies including auxiliary power units, automatic engine shut-down, automated transmissions and speed governors, low rolling resistance and single-wide tires, automated tire inflation and nitrogen tire inflation, as well as an array of aerodynamic devices. The impacts of management policies including driver education, idle time reduction, and service practices are addressed. The study also includes information system technologies such as wireless transport management, on-board fuel system monitoring, and advanced routing and scheduling systems. Each of these technologies, and several others, are identified with an initial discussion of their potential, but this is not a detailed study.

Several light duty studies were also considered, especially if the findings could be applied to medium or heavy duty vehicles. An ICCT study focused on Class 2B and 3 pickup trucks and vans, and included cost information [VT-15]. An EPA study that included cost analysis for light duty vehicles provided a costing approach that was identified as applicable to medium and heavy duty vehicles [VT-26]. A roadmap on fuel efficiency prepared by the International Energy Agency (IEA) included vehicles from two-wheeled through cars and light trucks [VT-16]. This study also outlined the roles of various stakeholders, and ways they can work together in achieving fuel consumption reduction goals.

The interaction between emission factors and fuel efficiency was addressed in a university study [VT-17]. Vehicle classification was addressed in this study, and various engine and vehicle technologies were included.

2.4.2 Evaluations of Specific Vehicle Technologies

Each of the technologies described in this section were identified in some of the references cited in the previous section on Vehicle Technology Identification [VT-1 through VT-17]. The publications summarized in this section are detailed studies of the specific technologies listed under each heading, often including measured or calculated fuel economy improvements. Included as an important reference is the users' guide to EPA's GEM model [VT-25]. It is a tool for evaluation of the GHG reduction potential of specific technologies.

2.4.2.1 Low Resistance and Wide-Based Tires

Several studies have documented the role of rolling resistance reduction in improving vehicle fuel economy. These include both modeling [VT-34, VT-36] and experimental [VT-35], and both passenger car [VT-35, VT-36] and heavy-duty [VT-34]. While this plays out in various efforts to improve tire technology, the most visible changes being seen are the wide-based tires now seeing increased application in Class 8 trucks.

As a well-developed technology, now seeing increased implementation, there are numerous studies quantifying the potential benefits of wide-based single tires in heavy duty applications [VT-18 through VT-21]. In an oral presentation from Michelin, wide-based singles were reported to reduce rolling resistance by ten percent compared to conventional dual tire heavy-truck installations and to reduce vehicle weight by about 800 pounds (363 kg) including the effect of the spare tire on the trailer [VT-30].

A study by the International Energy Agency [VT-18] emphasizes European truck applications, and provides specific recommendations regarding the use of low resistance tires. The study led by Lascurain at Oak Ridge National Lab included measured fuel economy benefits when comparing new generation single wide-based tires to standard dual tires on a Class 8 long haul truck [VT-19]. A nine to ten percent fuel economy improvement was recorded, over actual duty cycles. The improvement increased as vehicle weight was increased.

Hausberger and his colleagues developed specific coast down and constant speed tests, and included low resistance tires, trailer aerodynamics, and trailer weight in their measurements [VT-20]. Over this defined test the reduced rolling resistance tires alone resulted in a 4.5 percent fuel consumption reduction. In combination with improved aerodynamics the improvement was 6.5 percent. Reducing the trailer weight by 800 kg, combined with the rolling resistance and aerodynamic improvements, brought the total fuel consumption improvement to eight percent.

In a study led by the American Truck Association (ATA), a two to three percent fuel economy improvement is reported, with a maximum improvement of eight percent [VT-21]. This report also includes driver feedback comments on the pros and cons of wide-based tires.

2.4.2.2 Aerodynamic Improvements

As identified in the previous sub-section, trailer aerodynamics was among the factors tested by Hausberger and his colleagues [VT-20]. An ICF International study provides a comprehensive look at a variety of aerodynamic devices [VT-22]. Fuel economy gains attributed to each are reported over various duty cycles. Also reported are current and projected market penetration for the various devices.

As part of the DOE-sponsored SuperTruck program, one of the technology groups being studied is Class 8 vehicle aerodynamics. Tractor improvements, trailer improvements, and integrated, tractor/trailer system improvements have been studied, with results reported in terms of drag coefficient reduction. The most recent presentations at the time of this writing were presented at the May 2014 National Research Council Phase 3 Review of the SuperTruck partnership. The three presentations are summarized here. Koberlein reported a 46 percent drag coefficient reduction when comparing the SuperTruck tractor/trailer combination to today's baseline. This was improved to 49 percent when camera cab mirrors replaced the conventional mirrors. A further improvement (49.6 percent drag coefficient reduction from the baseline) was reported for an advanced concept tractor [VT-31]. Greszler presented a breakdown, beginning from a 2009 "best in class" baseline. The drag coefficient was reduced to 80 percent of the baseline through trailer add-on devices, and to 76 percent of the baseline through tractor modifications. Co-optimizing the tractor and trailer combination reduced the drag coefficient to 70 percent of the baseline, with the longer-term integrated SuperTruck design reducing the drag coefficient to 58 percent that of the 2009 baseline [VT-32]. Finally, Kayes reported on the order of a 15 percent drag coefficient reduction with several trailer modifications. Two tractor design concepts were evaluated, each resulting in a further 15 percent drag coefficient reduction. Scale model wind tunnel tests of the full vehicle (integrated tractor and trailer concepts) resulted in a 39 percent lower drag coefficient than that of their baseline truck [VT-33].

2.4.2.3 Hybrid Drivetrains

A projection of hybrid and plug-in hybrid cost effectiveness over a range of duty cycles is included in an especially useful reference on hybrid drivetrains [VT-23]. The study focused on medium duty parcel delivery vehicles, and explored various cost scenarios pertaining to fuel, batteries, and drivetrain components.

2.4.2.4 Weight Reduction

Regarding vehicle weight reduction, a study by the Environmental Defense Fund looked at the effect of vehicle tare weight in Class 3 through 6 medium duty trucks [VT-24]. It included considerations regarding truck class downsizing on fleet operating costs. The Hausberger study cited previously [VT-20] included the impact of reducing trailer weight in heavy duty vehicles.

2.4.2.5 *Improved Drivetrain Lubricants*

A study conducted by researchers at Shell Global Solutions on a Mercedes Benz OM 460LA heavy-duty diesel engine run under the World Harmonized Transient Cycle (WHTC) and World Harmonized Stationary Cycle (WHSC), used a combination of a SAE 5W-30 engine oil, SAE 75W-80 gearbox oil and SAE 75W-90 axle oil. The combination yielded average fuel economy improvements of 1.8 percent over the WHTC and 1.1 percent over the WHSC, relative to a SAE 15W-40 engine oil, SAE 80W gearbox and SAE 90 axle oil [VT-27]. The baseline lubricants represent current mainstream products, and the new lubricants were top-tier formulations focusing on modified viscometric effects. Using the WHSC cycle, significant variations in the individual lubricant contribution under different speed and load conditions within the cycle were identified. Additionally, an average fuel economy improvement of 1.8 percent was observed using medium-duty trucks under a range of typical European driving conditions in a controlled field trial.

2.4.2.6 *Improved Transmission Shifting*

A report by Stanton describes the Cummins High Efficiency, Clean Combustion program (HECC) for the 15L ISX engine and the 6.7L ISB engine [VT-28]. Among the technologies assessed, an additional two to three percent fuel economy improvement was achieved through engine down speeding, transmission shift pattern, and proper selection of axle ratio.

2.4.2.7 *Electrified Accessories*

In a Hyundai heavy duty vehicle study, various cooling system improvements were assessed. One of the improvements was that of replacing the engine-driven fan with a variable speed electric fan. The authors reported a nearly ten percent fuel economy improvement through reduced fan speed and the ability to shut the fan off over a driving cycle [VT-29].

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Note that where available, web links to the referenced documents have been included. These links were active at the time the document was written, but web addresses may change or be eliminated over time.

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3.0 PERFORMANCE ANALYSIS OF TECHNOLOGIES FOR BEYOND MODEL YEAR 2018

This section contains an evaluation of fuel saving technologies that are expected to be available for implementation beyond model year 2018. Engine, vehicle, and trailer technologies are included in the evaluation. SwRI used Polk data to determine popular engine and vehicle models in the various segments when benchmarking/testing engines and vehicles for prior programs. As a result, data from these engines and vehicles was available to inform the current simulation effort. In one case, the Ford F-650 tow truck, the vehicle was selected despite being a relatively low volume vehicle for two reasons: because it extended the range of vocational trucks towards the smaller, lighter end of the market, and because extensive test data was available to input to the simulation model.

With information from NRC 2010, the literature review, and meetings with industry stakeholders, the staffs at SwRI, NHTSA, EPA, and CARB used their engineering judgment to select feasible engine and vehicle technologies for performance analysis. In a few cases, potentially promising technologies such as hybrids or RCCI technologies could not be simulated as they were beyond the technical capabilities of the software and/or the program lacked access to physical test data (prior engine/vehicle testing) from which to develop models. Other technologies such as continuously variable transmissions and variable compression ratio gasoline engines were left out because they have not yet been demonstrated for the vehicle classes under consideration, and are not expected to be commercially viable in the time frame for this study.

Engine technologies have been evaluated for both diesel engines and gasoline engines. Due to their low market penetration and the need for extensive experimental data to calibrate the simulation models, natural gas engines, other alternative fuels, and dual fuel engines are beyond the scope of this study. The engine technology evaluations were conducted using GT-Power, which is a commercially available one dimensional (1-D) engine modeling tool. GT-Power models of the engines were built and calibrated using actual engine test data. The GT models were then used to explore a range of potential engine technologies.

Two basic diesel engine models were selected for the project. The 6.7 liter Cummins ISB engine is used at high volume in ¾ ton Class 2b and one ton Class 3 Ram pickup trucks, and is also the most popular engine in medium-duty trucks through Class 7 and into the low end of Class 8. The ratings of this engine are quite different between medium-duty trucks and pickup trucks, so we modeled two different calibrations. The medium-duty rating is 300 HP @ 2500 RPM, while the pickup rating is 385 HP @ 3000 RPM. The pickup rating is chassis certified, so it does not use EGR at full load.

The 14.8 liter Detroit DD15 is a popular heavy-duty truck engine for Class 8 long haul operation, and the 2011 version that serves as our baseline met the 2014 GHG requirements in SwRI's benchmarking tests. To cover the range of displacements used in Class 2b through Class 8, we developed some derivatives of these base engines. We created a 4.5 liter 4-cylinder version of the ISB. We also generated a 12.3 liter 5-cylinder version of the DD15. Because of the level of effort involved in creating and calibrating GT-Power models, we used derivatives of our two basic diesel engines rather than creating entirely new GT models of different engines to cover the desired displacement range. There is a 4-cylinder version of the ISB in production, so our model of that engine matches an actual production engine. The 12.3 liter version of the DD15 does not exist, however. Instead of this version, we could have modeled the 12.8 liter DD13, but we lacked the design data required to build GT models of this engine. We also lacked the test data required to calibrate GT models of this engine. We believe our approach of doing a smaller version of the DD15 provide realistic approximations of the true performance of engines in their size range, which can achieve the same objective of evaluating downsizing impact.

Gasoline engines are used today by all OEMs in Class 2b and 3 vehicles. Ford currently offers a gasoline engine in Class 4 through 7 trucks up to 30,000 pounds GVWR. Gasoline engines have one huge advantage in the Class 2b to Class 7 market: initial cost. In a heavy-duty pickup, the diesel engine option typically costs the customer about $8,000 above the base gasoline engine. In a model year 2014 Ford F-650 Class 6 truck, the diesel engine is a $9,216 option. Other advantages of gasoline engines include a wider operating speed range, lower price fuel, lighter weight, and, in many cases, higher horsepower ratings.

The disadvantages of gasoline engines in Class 2b – 7 trucks include higher fuel consumption (measured on a fuel volume basis, e.g. MPG). Gasoline engines also have lower maximum torque values and reduced durability compared to diesel engines. This section will explore technologies that could reduce the fuel consumption disadvantage of gasoline engines. Turbocharging can reduce the torque disadvantage of gasoline engines. The durability disadvantage stems from two primary sources: gasoline engines operate at higher temperatures, and existing medium-duty gasoline engines were originally designed for Class 2b and 3 applications, not for heavier Class 4 - 8 applications.

The fuel consumption penalty is inherent in the operating cycle of gasoline engines, and in the lower energy content per unit volume of gasoline compared to diesel fuel. A gallon of diesel fuel contains about 16% more chemical energy than a gallon of E10 gasoline [Energy.gov, 2014]. The same reference states that the CO_2 emissions from a gallon of gasoline are about 14% lower than for a gallon of diesel, so energy content and CO_2 emissions are very tightly linked for these two fuels. The combination of lower peak torque, higher peak power, and a wider operating speed range are typical features of gasoline engines. High peak power and a wide operating speed range can be advantages, but many heavy-duty applications require a higher peak torque than currently available gasoline engines can deliver.

The durability issue faced by gasoline engines is largely based on two factors. The first is the higher temperatures experienced in a stoichiometric combustion engine. The second factor is that the engines used in these trucks are derived from engines originally developed for light duty applications. It is possible to design a more durable gasoline engine, although that would erode the price advantage to some degree. High levels of cooled EGR can be used to reduce the temperatures experienced in stoichiometric combustion, although it is generally not possible to get diesel-like temperatures without experiencing misfire issues. Up to this time, OEMs have evidently found that there is not enough volume potential in a heavy-duty gasoline engine intended specifically for Class 4 – 7 applications to justify the development of one. Existing GM, Ford, and Chrysler heavy-duty gasoline engines were developed for (and with the exception of Ford) are only applied in Classes 2b and 3. Ford has expanded the application range of their

6.8 liter V-10 up into Class 7. Using information from the separate cost report that is included with this project, it is clear that the cost increment for an upgraded gasoline engine with EGR and a 3-way catalyst is far lower than that of a diesel engine with SCR and DPF. SwRI believes that the large price difference between gasoline engines with simple 3-way catalyst aftertreatment, compared to diesel engines with expensive HPCR fuel systems and SCR + DPF aftertreatment, may in the future lead to more gasoline engines being developed for and used in medium-duty vehicles (Classes 4 – 7). This is particularly likely if the fuel consumption and durability issues of heavy-duty gasoline engines can be improved.

Two gasoline engine GT-Power models were developed for this project. One model is a 6.2 liter V-8 engine with port injection. This naturally aspirated engine is now used in Class 2b and 3 applications. This engine could in the future be developed for heavier duty applications. The other GT model is of a 3.5 liter V-6 with turbocharging and direct injection. This engine is now applied to Class 2a trucks, but future versions could be developed for heavier duty applications. We did not develop any other versions of these two models to represent different displacements.

The Kenworth T700 Class 8 tractor trailer truck, the Kenworth T270 Class 6 box delivery truck, and the Ford F-650 Class 5 tow truck were selected for the program because extensive vehicle test data was available from an EPA program. This data was used to calibrate the vehicle simulation models. These trucks are also good representatives of medium- and heavy-duty trucks. The T270 and F-650 were modeled using the Cummins ISB engine that they come standard with. The T700 offers both PACCAR and Cummins engines, but this vehicle was simulated with the Detroit Diesel DD15. This engine could be used in the T700, but for business reasons it is not. The DD15 was used for this study because of the extensive engine benchmarking data that is available.

While engine technologies have been simulated in substantial detail, this study takes a different approach to vehicle technologies. In this study, the fuel consumption sensitivity of the vehicle to various forms of power demand reduction is explored, without attempting to simulate specific features. Therefore, reductions in Cd and Crr are assumed, in line with values that the literature suggests are feasible. In other cases, available but proprietary experimental data was employed, such as for transmission and axle mechanical efficiency.

Note that all results presented in this report are in terms of percent change in fuel consumption, not fuel economy. Fuel consumption units align well with the units used in EPA and NHTSA GHG and fuel consumption regulations.

Table 3.1 below summarizes how the selected vehicles and engines fit into the US vehicle classification system, and Table 3.2 summarizes the fundamental characteristics of the engine in their baseline form.

TABLE 3.1 VEHICLE AND ENGINE CLASSIFICATION

Class	Vehicle	Diesel	Gasoline	Base Transmission
2b	Ram Pickup	Cummins 6.7 Liter 385 HP (base), 4.5 Liter 256 HP	3.5 L V-6, 6.2 L V-8	6-Speed Automatic
3				
4				
5	F-650 Tow Truck	Cummins 6.7 Liter 300 HP (base), 4.5 Liter 256 HP	3.5 L V-6, 6.2 L V-8	5-Speed Automatic
6	T270 Box Truck			
7				
8	T700 Tractor-Trailer	Detroit 14.8 L DD15 (base), 12.3 L Derivative	None	10-Speed AMT

TABLE 3.2 ENGINE CHARACTERISTICS

Engine	Displacement Liters	Rated HP @ RPM	Torque Peak lb-ft @ RPM	Best BSFC g/kW-hr	Other
ISB Pickup	6.7	385 @ 3000	850 @ 1600	198.6	Part load EGR
ISB MD	6.7	300 @ 2500	750 @ 1300	207.8	Full time EGR
V-6	3.5	370 @ 5500	420 @ 3500	238.0	Turbo, DI
V-8	6.2	316 @ 5500	400 @ 4200	236.5	NA, PFI
DD15	14.6	485 @ 1800	1650 @ 1240	185.7	Turbocompound

3.1 Technology Lists

A wide range of both engine and vehicle technologies were explored in this project. These technologies are listed in tabular form below. Additional details are provided in Appendix A (gasoline engine technologies), Appendix B (diesel engine technologies), Appendix C (vehicle technologies), and Appendix D (waste heat recovery systems).

3.1.1 Engine Technologies

The tables below list technologies that were applied to the engines in the program. There are limited combustion related technologies in this list, since GT-POWER is not capable of modeling these technologies with any degree of confidence, unless extensive experimental data is available to calibrate the model. Examples of combustion technologies left out of the study include low temperature diesel combustion and dual fuel engines. In the literature, combustion technologies have generally been shown to offer benefits in the 1 to 2% range (see Section 2).

TABLE 3.3 ENGINE TECHNOLOGIES EVALUATED ON THE DD15

Technology	Hardware Content	Comments
1. Baseline DD15	Production 2011 DD15	Complies with 2014 GHG requirement, but with zero margin
2. Optimized Mechanical Turbocompound	Downsized power turbine	Attempt to improve BSFC at cruise, at the expense of high RPM
3. Optimized Electrical Turbocompound	Delete power turbine gear train, add electrical generator and electric motor	Decouples power turbine speed from crankshaft speed. Power fed back to crank via electric motor
4. No EGR	Remove EGR cooler, valve, and plumbing. Turbocharger resized to match.	Would require a very high conversion efficiency SCR to meet NOx requirement. OBD could be very challenging
5. Turbocompound Removed	Remove power turbine, turbine gear train, and plumbing. Turbocharger resized to match.	Still a single stage, fixed geometry turbo. Unable to flow adequate EGR below 1400 RPM, so would require very high SCR efficiency
6. EGR and Turbocompound Removed	Delete EGR and turbocompound hardware. Turbocharger resized to match	Still a single stage, fixed geometry turbo. Would require a very high conversion efficiency SCR to meet NOx requirement
7. Asymmetric Turbo	Delete turbocompound, add fixed geometry turbo with asymmetric volute, drive EGR from only cylinders 1-3	Represents 2014 DD15. Only half the engine suffers from negative Δp required to flow EGR. This technology is covered by a Daimler patent, so other OEMs would need to license it or design around it.
8. Reduced Exhaust Backpressure	Higher flow capacity aftertreatment, engine unchanged	Determine BSFC sensitivity to exhaust restriction
9. Reduced Inlet Restriction	Higher flow capacity intake system, engine unchanged	Determine BSFC sensitivity to intake restriction
10. Reduced CAC Restriction	Higher flow capacity charge air cooler, engine unchanged	Determine BSFC sensitivity to charge air cooler restriction
11. Reduced Engine Friction	Variable speed water pump, variable displacement oil pump, on/off piston cooling, reduced piston/ring/liner friction, low viscosity lube	Assume a FMEP reduction of 10% at high speed and load, increasing to 35% at low speed, light load, where cooling needs are reduced
12. High Efficiency Turbo	Upgrade fixed geometry turbo efficiency by 10% (half compressor, half turbine).	Determine BSFC sensitivity to turbo efficiency. Note that additional technology would be required to maintain EGR flow if desired, such as turbocompound or an intake throttle
13. No EGR, No Turbocompound, High Efficiency Turbo	Delete EGR and turbocompound systems, upgrade turbo efficiency by 10% (half compressor, half turbine)	Determine effect of turbo efficiency on a non-EGR engine. Would require a very high conversion efficiency SCR to meet NOx requirement. OBD could be a major issue.
14. Downspeed A	No hardware change	Increase low speed torque, reduce rated speed, maintain PCP limit and maximum engine power. Operate engine at lower speed during cruise
15. Downspeed B	No hardware change	A more radical version of Downspeed A. Both versions of downspeeding involve changes to vehicle gearing, but are classified as an engine technology for this report
16. Downsize, Constant Torque	higher BMEP (See Appendix B for details)	Vehicle performance not affected. Engine loads increased.
17. Downsize, Constant BMEP	same BMEP as baseline (See Appendix B for details)	Power and torque reduced 16.7%
18. Variable Valve Train	Add VVT hardware	Variable valve lift and duration
19. Water-Based Bottoming Cycle	EGR and exhaust heat exchangers, expander, condenser, pump	See Appendix B for details
20. R-245 Based Bottoming Cycle	EGR and exhaust heat exchangers, expander, condenser, pump	See Appendix B for details

TABLE 3.4 ENGINE TECHNOLOGIES EVALUATED ON THE ISB 6.7 MEDIUM-DUTY ENGINE

Technology	Hardware Content	Comments
21. Baseline ISB	Production 2012 ISB	Most popular MD truck engine
22. Reduced Exhaust Restriction	Higher flow capacity aftertreatment, engine unchanged	Determine BSFC sensitivity to exhaust restriction
23. Reduced Engine Friction	Variable speed water pump, variable displacement oil pump, on/off piston cooling, reduced piston/ring/liner friction, low viscosity lube	Assume a FMEP reduction of 10% at high speed and load, increasing to 35% at low speed, light load, where cooling needs are reduced
24. EGR Removed	Remove EGR cooler, valve, and plumbing. Turbocharger resized to match.	Would require a very high conversion efficiency SCR to meet NOx requirement. OBD could be a major issue
25. High Efficiency Turbo	Upgrade turbo efficiency by 10% (half compressor, half turbine)	Determine BSFC sensitivity to turbo efficiency.
26. 8-Cylinder Version	Log-style exhaust manifold, rescaled turbo size, unchanged combustion parameters, EGR rate, and AFR.	Lower peak torque and rated speeds to reflect typical 9 liter engine ratings
27. Variable Valve Train	Add VVT hardware	Variable valve lift and duration. This technology was dropped due to disappointing results. The technology is described in Section 1.2 of Appendix A

TABLE 3.5 ENGINE TECHNOLOGIES EVALUATED ON THE ISB 6.7 ENGINE FOR CLASS 2B/3 PICKUPS

Technology	Hardware Content	Comments
28. Baseline ISB	Production 2012 ISB for Ram	Higher power and torque than MD version, no EGR at high loads, chassis certified
29. Reduced Exhaust Restriction	Higher flow capacity aftertreatment, engine unchanged	Determine BSFC sensitivity to exhaust restriction
30. High Efficiency Turbo	Upgrade turbo efficiency by 10% (half compressor, half turbine)	Determine BSFC sensitivity to turbo efficiency.
31. Reduced Engine Friction	Variable speed water pump, variable displacement oil pump, on/off piston cooling, reduced piston/ring/liner friction, low viscosity lube	Assume a FMEP reduction of 10% at high speed and load, increasing to 35% at low speed, light load, where cooling needs are reduced
32. 4-Cylinder Version	Log-style exhaust manifold, resized single-entry turbo, unchanged combustion parameters, EGR rate, and AFR.	Constant BMEP, so power and torque are reduced by 1/3 from baseline.

TABLE 3.6 ENGINE TECHNOLOGIES EVALUATED ON THE 3.5 LITER V-6 TURBO GDI

Technology	Hardware Content	Comments
33. Baseline 3.5 V-6	2012 Ford EcoBoost 3.5 used in F-150 (Class 2a)	Not used in heavier applications yet, but the potential is there; used as a representative of a downsized, boosted engine for heavier duty applications
34. Variable Valve Train with Cam Phaser	Add VVT and cam phaser hardware	Variable valve lift and duration, plus valve event phasing. This technology is much more promising on gasoline engines
35. Cylinder Deactivation	Cylinder deactivation hardware for OHC engine	Operate engine on 3, 4, or 6 cylinders as required. Gives light load benefit
36. Lean Burn GDI	SCR or other NOx aftertreatment	Requires NOx aftertreatment. Exhaust temperature is a problem
37. Stoich EGR	Low pressure loop EGR system, EGR valve and cooler, high energy ignition	High energy ignition needed to ignite dilute air/fuel mixture
38. EGR + Downspeed	Requires higher cylinder pressure capability	Increase low speed torque, reduce rated speed, increase cylinder pressure. Operate engine at lower speed during cruise. Higher BMEP to retain vehicle performance and rated power
39. Reduced Engine Friction	Variable speed water pump, variable displacement oil pump, on/off piston cooling, reduced piston/ring/liner friction, low viscosity lube	Assume a FMEP reduction of 10% across the range, given the higher level of past friction reduction work on gasoline engines compared to diesel
40. High Efficiency Turbo	Upgrade turbo efficiency by 10% (half compressor, half turbine)	Determine BSFC sensitivity to turbo efficiency.

TABLE 3.7 ENGINE TECHNOLOGIES EVALUATED ON 6.2 LITER PORT-INJECTED V-8

Technology	Hardware Content	Comments
41. Baseline 6.2 V-8	2012 6.2 V-8 used in Class 2b / 3 pickup trucks	Not used in heavier applications yet, but the potential is there
42. Convert to GDI	GDI fuel system	No change to power and torque
43. Lean Burn GDI	SCR or other NOx aftertreatment	Requires NOx aftertreatment. Exhaust temperature is a problem. Rated power maintained by going rich at full load
44. Variable Valve Train with Cam Phaser	Add VVT and cam phaser hardware	Variable valve lift and duration, plus valve event phasing. Value of cam phasers also evaluated
45. Cylinder Deactivation	Cylinder deactivation hardware for OHC engine	Operate engine on 4 or 8 cylinders as required. Gives light load benefit
46. Stoich GDI EGR	Low pressure loop EGR system, EGR valve and cooler, high energy ignition	High energy ignition needed to ignite dilute air/fuel mixture
47. Reduced Engine Friction	Variable speed water pump, variable displacement oil pump, on/off piston cooling, reduced piston/ring/liner friction, low viscosity lube	Assume a FMEP reduction of 10% across the range, given the higher level of past friction reduction work on gasoline engines compared to diesel

3.1.2 Vehicle Technologies

The tables below list technologies that were applied to the vehicles in the program.

TABLE 3.8 VEHICLE TECHNOLOGIES EVALUATED ON RAM PICKUP (CLASS 2B / 3)

Technology	Hardware Content	Comments
A. Baseline Ram	ISB diesel, 6-speed automatic transmission	Coastdown results (Cd and Crr values) from the 2007 Ram 2500, powertrain data from 2011
B. 6.2 V-8 Port Injected	Replace ISB diesel with baseline gasoline V-8	
C. 3.5 V-6 Turbo GDI	Replace ISB diesel with baseline gasoline V-6	
D. Reduced A/C Power Demand	Combination of increased cab insulation, reduced reheat, more efficient compressor	Assumed a 40% power demand reduction from a baseline A/C power demand of 1.5 kW
E. Improved Cd	Radiator shutters, belly pan, cab and bed tweaks	Assumed 25% Cd reduction
F. Improved Crr	Low rolling resistance tires	Assumed 30% Crr reduction
G. Transmission Upgrade	Replace baseline 6-speed automatic with 8-speed	Modified shift schedule and gearing with the 8-speed to improve fuel economy and at least maintain vehicle performance
H. Weight Reduction	Material substitution – 500 pounds	Weight difference between engines also included
I. Chassis and Driveline Friction	Synthetic lube, improved axle efficiency, improved bearings	Assume 30% chassis friction reduction
J. Hybrid Systems	TBD	To be in Final Report #2

TABLE 3.9 VEHICLE TECHNOLOGIES EVALUATED ON KENWORTH T270 BOX TRUCK (CLASS 6)

Technology	Hardware Content	Comments
K. Baseline T270	ISB diesel engine, Allison 5-speed automatic	13’ 2” X 102” delivery box height X width. Coastdown data available from EPA project.
L. 6.2 V-8 Port Injected	Replace ISB diesel with baseline gasoline V-8	
M. 3.5 V-6 Turbo GDI	Replace ISB diesel with baseline gasoline V-6	
N. Reduced A/C Power Demand	Combination of increased cab insulation, reduced reheat, more efficient compressor	Assumed a 40% power demand reduction from a baseline A/C power demand of 1.5 kW
O. Improved Cd	Roof fairing, cab-to-box fairing, side skirts	Assumed 15% Cd reduction
P. Improved Crr	Low rolling resistance tires	Assumed 30% Crr reduction
Q. Automatic Transmission Upgrade	Replace baseline 5-speed automatic with 8-speed	Modified shift schedule and gearing with the 8-speed to improve fuel economy and at least maintain vehicle performance
R. AMT Alternatives	6-speed and 10-speed AMT	Compare to automatics
S. Weight Reduction	Material substitution – 1000 pounds	Weight difference between engines also included
T. Chassis and Driveline Friction	Synthetic lube, improved axle efficiency, improved bearings	Assume 30% chassis friction reduction
U. Hybrid Systems	TBD	To be in Final Report #2

TABLE 3.10 VEHICLE TECHNOLOGIES EVALUATED ON FORD F-650 TOW TRUCK (CLASS 6)

Technology	Hardware Content	Comments
V. Baseline F-650	ISB diesel engine, Allison 5-speed automatic	Tow truck equipment mounted on chassis. Coastdown data available from EPA project.
W. 6.2 V-8 Port Injected	Replace ISB diesel with baseline gasoline V-8	
X. 3.5 V-6 Turbo GDI	Replace ISB diesel with baseline gasoline V-6	
Y. Reduced A/C Power Demand	Combination of increased cab insulation, reduced reheat, more efficient compressor	Assumed a 40% power demand reduction from a baseline A/C power demand of 1.5 kW
Z. Improved Cd	Front air dam, cab tweaks, side skirts	Assumed a 10% Cd reduction
AA. Improved Crr	Low rolling resistance tires	Assumed a 30% Crr reduction
BB. Automatic Transmission Upgrade	Replace baseline 5-speed automatic with 8-speed	Modified shift schedule and gearing with the 8-speed to improve fuel economy and at least maintain vehicle performance
CC. AMT Alternatives	6-speed and 10-speed AMT	Compare to automatics
DD. Weight Reduction	Material substitution – 1000 pounds	Weight difference between engines also included
EE. Chassis and Driveline Friction	Synthetic lube, improved axle efficiency, improved bearings	Assume 30% chassis friction reduction
FF. Hybrid Systems	TBD	To be in Final Report #2

TABLE 3.11 EVALUATION OF TRACTOR AND TRAILER TECHNOLOGIES ON KENWORTH T-700 TRACTOR (CLASS 8)

Technology	Hardware Content	Comments
GG. Baseline T-700	DD15 diesel engine, 10-Speed Eaton AMT, 53’ box van trailer	SmartWay tractor, standard (NOT SmartWay) trailer. Coastdown data available from EPA project.
HH. Reduced A/C Power Demand	Combination of increased cab insulation, reduced reheat, more efficient compressor	Assumed a 40% power demand reduction from a baseline A/C power demand of 1.5 kW
II. Improved Cd	Trailer: full skirt, boat tail, gap reducer. Tractor: cameras replace mirrors, radiator shutters, gap reducing fairings, skirt over drive axles, under - chassis air flow mgmt. devices	Assumed a maximum of 25% Cd reduction, and also calculated benefits for 5%, 10%, 15%, and 20% Cd reduction
JJ. Improved Crr	Low rolling resistance tires	Assumed a 30% Crr reduction
KK. Weight Reduction	Material substitution and redesign of cab and chassis components – 6.5% and 13% weight reductions	1000 and 2000 kg (2200 and 4400 pound) reductions
LL. Chassis and Driveline Friction	Synthetic lube, improved axle efficiency, improved bearings	Assume 20% chassis friction reduction in the tractor
MM. 6X2 Axles	Replace 6X4 with 6X2	Remove friction of one drive axle. Single drive axle is more efficient than front axle of a tandem
NN. Road Speed Governor	Use existing feature	Evaluate unlimited vs. 65, 60, and 55 MPH limits
OO. 18-Speed AMT	Replace 10-speed with 18-speed	Smaller splits allow the engine to stay in higher efficiency zones
PP. Manual Transmission	Replace AMT with manual	Shift points 200 RPM higher than AMT

3.2 Modeling Methodology

The engines and engine technologies were modeled in GT-Power, which is a commercially available simulation tool. Each baseline model was calibrated using experimental engine data. In many cases, detailed combustion heat release data was available from engine testing. This allowed heat release to be input directly into the model, rather than estimated by GT. We also used actual turbocharger efficiency maps as an input, although these maps were not necessarily from the engine being simulated. The turbo maps were scaled up or down to achieve the required air flow for each specific engine technology we simulated.

One dimensional CFD tools such as GT-Power have certain advantages and limitations. Some advantages relative to 3-D CFD tools include:

- Rapid solution time
- Accurate calculation of engine air flows, pressures, and temperatures (provided the input geometry data is correct)
- Very useful for predicting the effects of basic parameters such as compression ratio, combustion timing, air/fuel ratio, intake and exhaust restriction, etc.
- Very useful for determining required turbocharger match
- Fairly accurate representation of overall fuel consumption and CO_2 emissions (typically within +/- 3%)
- More accurate representation of small changes in fuel consumption and CO_2 as a result of a technology change (differences of less than 1% can often be reliably predicted)

Areas of weakness in tools like GT-Power include:

- Unreliable predictions of NOx, PM, and other criteria emissions
- Predictions of combustion parameters such as rate of heat release are simplified unless experimental data is available to use as an input. Because of that, the prediction on combustion related technologies, such as LTC and RCCI would not be as reliable as prediction of air handling system technologies.
- Predictions of turbocharger performance are based on the maps that are provided to the program, which sometimes do not reflect real-life performance on the engine. Can be improved if measured engine data is available to baseline and tweak the turbo maps to match actual, on-engine turbo performance.

One issue with simulating turbocharger performance is the fact that turbo efficiency maps are measured on a gas stand. The gas stand has steady flow, unlike the pulsating flow of an actual engine. As a result, the gas stand will miss performance that is a function of fluctuating flow, such as the benefit of a dual entry turbine housing, which utilizes the pulsation energy from blow-down pluses in the exhaust manifold. If engine test data is available, the turbo performance maps determined on a gas stand can be modified to reflect actual on-engine performance.

To overcome the weaknesses of GT-Power, SwRI used measured combustion heat release data whenever it was available. All of the engine models used measured heat release for the baseline technology, and in many cases, experimental data was available for specific technologies that had an effect on combustion. Except in specific cases that are noted in the discussion, EGR rates and air/fuel ratios were controlled to match the baseline engine performance. Technologies that would affect heat release rates and combustion duration will have effects on efficiency that are not captured in the GT-POWER models used in this project. Technologies where assumptions had to be made regarding combustion are noted in the results section. SwRI also used actual turbocharger performance maps as a basis, and scaled them to match given engines and technologies. This approach provides turbocharger performance that matches at given points and has the right characteristics across the engine speed/load range. Having data on a full family of turbochargers that would be applied to the engine being simulated would be even better. Unfortunately, full turbo map data was not available for all of the engine permutations in our study.

For each engine and engine technology, the GT-Power model was run over a range of speed and load conditions. The resulting fuel consumption and CO_2 data were used to create a fuel consumption map. The map provides projected fuel consumption over a range of 20 speeds and 20 loads, for a total of 400 data points. Not all of these data points were actually simulated – many were generated by interpolation between simulated speed/load points. Appendix A and B provide details on the number of speed/load points that were simulated for each engine. The 20 X 20 point fuel maps were then provided to the vehicle simulation tool to represent the engine performance. In addition to the fuel maps, the full load torque curve and motoring torque curve (the torque required to spin the engine with zero fuel) were provided to the vehicle simulation tool.

Note that all engine simulations are done for steady state operating conditions. During transient operating conditions such as a shift event or accelerator pedal position change, the engine may not have the same torque capability and efficiency as it does at steady state. On highly transient drive cycles, this can lead to an over-estimation of vehicle performance and efficiency. Modeling of complete engine and vehicle performance during transients is possible, but very time consuming. For this large study that evaluates many technologies, transient simulation was judged to be too time consuming to perform.

Appendix A includes details of each gasoline engine model, including sources of input data and comparisons to experimental results. The assumptions made for each technology are also described. Appendix A also includes the fuel map results for each gasoline engine and technology. Appendix B includes the same information for the two diesel engines.

Vehicles and vehicle technologies were modeled using the SwRI Vehicle Simulator tool. This MATLAB-based tool is similar to the NREL tool called Advisor. The Vehicle Simulator tool is a deterministic model (has no randomness) and has the ability to handle a wide range of vehicle technologies including automatic transmissions, automated manual transmissions, hybrid systems, etc. One advantage of the SwRI tool is that features can easily be ported to the GEM tool. Any desired drive cycle can be put into the Vehicle Simulator tool. The following drive cycles were used for this program:

TABLE 3.12 VEHICLES AND DRIVE CYCLES USED IN STUDY

Vehicle	Drive Cycles
Ram Pickup	FTP City, FTP Highway, US06, SC03, WHVC, 65 MPH
T270 Box Truck	GEM Cycles, CILCC, Parcel Delivery Cycle, WHVC
F-650 Tow Truck	GEM Cycles, CILCC, Parcel Delivery Cycle, WHVC
T-700 Tractor	GEM Cycles, WHVC, NESCCAF Long Haul Cycle

The cycles listed above are described in detail in Appendix C. The current version of GEM includes 3 cycles. One is a low speed urban cycle developed by CARB. The second is a constant 55 MPH with no grade or wind. The final cycle is a 65 MPH constant speed with no grade or wind. For the 55 and 65 MPH cycles, only data from the steady-state portion of the cycle is reported. The US06 and SC03 cycles are carried over from light-duty applications. Since heavy-duty pickup trucks are often used as a car, it is appropriate to evaluate their performance on car-like drive cycles. The acronym CILCC stands for Combined International Local and Commuter Cycle. The parcel delivery cycle was derived from the operations of a Class 6 parcel delivery truck in the US market. The CILCC and Parcel Delivery cycles were used to represent the local operations that are typical for many vocational trucks.

WHVC stands for World Harmonized Vehicle Cycle. This cycle is intended for medium- and heavy-duty vehicles, and includes a low speed urban segment, a moderate speed suburban segment, and a highway cruise segment. The highway segment of the WHVC does not include grade or wind, and the speeds on this segment reflect the European practice of installing road speed governors on trucks to limit maximum vehicle speed to 90 km/h (56 MPH). The WHVC is the only cycle other than the GEM cycles that were applied across the complete range of Class 2b through Class 8 vehicles. The NESCCAF long haul cycle includes brief urban/suburban segments and 4 extended highway cruise segments at 65 to 70 MPH. One of highway cruise segments includes a cyclic grade of +/- 1%, and another highway cruise segment includes a cyclic grade of +/- 3%. Like all the other cycles, the NESCCAF cycle does not include the effect of any wind.

The reason for exploring several duty cycles is to develop an understanding of how different engine and vehicle technologies perform across a range of drive cycles. Certain technologies may be insensitive to payload or drive cycle, while others can be extremely sensitive. One of the questions to be explored by this study is whether the existing GEM cycles do an adequate job of covering the range of drive cycles experienced in the field.

Appendix C describes the input data required by the SwRI Vehicle Simulator tool. In most cases, SwRI used measured data from test vehicles and components, or information provided by OEMs and component suppliers as inputs. In certain cases, such as axle and transmission efficiency, SwRI used test data from other SwRI projects as inputs to the simulation. This existing data is proprietary to SwRI and its specific clients, and was not created or derived from federally funded work. In these limited cases, SwRI cannot provide the actual input data used in the simulation runs.

One of the many inputs required by the SwRI Vehicle Simulator tool is the transmission shift schedule. For automatic and automated manual transmissions, the shift schedule can be relatively straightforward. It can be measured directly in vehicle or powertrain testing, or it can be provided by the transmission manufacturer. Determining shift schedules for manual transmissions is more difficult. Different drivers may have dramatically different shifting habits, which will have a substantial effect on fuel consumption. Extensive recording of driver performance in the field would be required to arrive at either an "average driver" shift schedule, or a range of shift schedules that covers most of the driver population. Since this data was not available, we have simulated manual transmissions in this report by adding 200 RPM to the upshift points of the automated manual transmission. This offset is an attempt to reflect typical driver behavior. In order to provide consistency when evaluating engine and vehicle technologies, the AMT transmission with programmed shift schedules was used as the baseline for all runs.

One factor that is neglected in the simulation is the weight increases that result from adding efficiency technologies to the vehicle. Some technologies, such as low rolling resistance tires, can be weight neutral or even a weight reduction. Other technologies, such as aerodynamic features and waste heat recovery systems, definitely add weight. Unfortunately, good weight estimates for many technologies are not available in the literature, so the simulations performed in this project do not adjust weight for most technologies. If a reader would like to take into account the impact of a known or assumed weight increase, the vehicle fuel consumption sensitivity to changes in weight is provided for all vehicles.

The situation is simple for any vehicle not operating at the legal weight limit. For example, if vehicle fuel consumption increases 0.7% for every 1,000 pound weight increase, there would be a fuel penalty of 0.35% for a 500 pound weight increase. However, if the vehicle is operating at the weight limit, then payload must be reduced to stay within the weight limit. Consider the example of a tractor-trailer with an empty weight of 34,000 pounds, running at the legal limit of 80,000. In this case, an increase in empty weight of 500 pounds has no effect on vehicle fuel consumption, because the loaded vehicle weight is unchanged. However, the empty weight increase will raise load-specific fuel consumption by about 1.1%, because payload is reduced by about 1.1%. This example shows how the impact of a weight increase is greater if the vehicle is operating at maximum weight, when payload must be reduced.

Another factor that is not considered in the simulation is the fuel required to maintain the performance of the emissions aftertreatment system. This includes fuel used to achieve rapid light-off of the aftertreatment during a cold start, fuel used for thermal management to keep the aftertreatment at the desired temperature during light load operation, and fuel used to regenerate the aftertreatment to prevent plugging or excessive sulfur loading. Getting an accurate prediction of the fuel used for these purposes requires an extensive understanding of the complex control strategies employed. These strategies vary greatly from one manufacturer to another, and reverse engineering of these strategies is a difficult challenge. As a result, this study did not explore fuel consumption related to aftertreatment operation.

3.3 Results

This section will cover the results of the engines and vehicle types listed in Sections 3.1.1 and 3.1.2. Results presented in this report are in terms of percent reduction in fuel consumption (also called percent fuel savings). These results translate directly into percent reduction in grams of CO_2 or gallons of fuel per hp-hr, as well as grams of CO_2 or gallons per 1000 ton-miles, which are the units used in the Phase 1 regulations.

3.3.1 Class 8 Tractor-Trailer Truck Engine Technology Results

Class 8 tractors are typically powered by 11 to 16 liter engines. For standard 80,000 pound long haul applications, 15 liter engines such as the Cummins ISX and Detroit DD15 tend to predominate in terms of sales volume, but there are also 13 liter engines used in this market that are supplied by Volvo, Navistar, Detroit Diesel and PACCAR. The Kenworth T-700 vehicle simulated for this project was chosen because it was used in an EPA program looking at Phase 2 GHG regulation test methodologies. As a result, a substantial amount of test data was available to calibrate the simulation model. This data is summarized in Appendix E. The T-700 is a SmartWay certified tractor, with extensive aerodynamic treatment. The trailer used for the simulation, however, was a basic 53 foot box van trailer with no aerodynamic treatment and with standard (not low rolling resistance) tires. This combination reflects the type of vehicle that will comply with the 2018 GHG and fuel consumption requirements.

The Kenworth T-700 is sold with many ratings of two basic engines: the PACCAR MX-13 (a 12.9 liter engine) and the Cummins ISX (a 15 liter engine). The Detroit DD15 engine used in this study is not offered in the T-700, but it has performance comparable to the ISX, and the DD15 is certainly an engine that *could* be applied in the T-700. The DD15 was chosen because of data available from a SwRI benchmarking program that was performed on a 2011 DD15. The Kenworth T-700 tractor and standard 53 foot box van trailer used to acquire the experimental coast-down data that was used to calibrate the simulation model are shown in Figure 3.1 below.

FIGURE 3.1 KENWORTH T-700 TRACTOR AND STANDARD 53 FOOT BOX VAN TRAILER

3.3.1.1 Results for Baseline Kenworth T-700 / DD15 Tractor-Trailer Vehicle

The tractor-trailer combination was run at the following payloads and combined vehicle weights: 0% payload and a vehicle weight of 33,960 pounds, 50% payload (23,020 pounds of payload) and a vehicle weight of 56,980 pounds, and 100% payload (46,040 pounds of payload) and a vehicle weight of 80,000 pounds. Figure 3.2 shows the *fuel economy* performance of the baseline vehicle, across all 5 drive cycles and 3 payloads. Figure 3.3 shows the same results, but in terms of *fuel consumption*. All of the remaining results in Section 5.3.1 will be given in terms of percent difference in *fuel consumption* compared to the baseline performance. The same data is presented in tabular form in Tables 3.11 and 3.12 below.

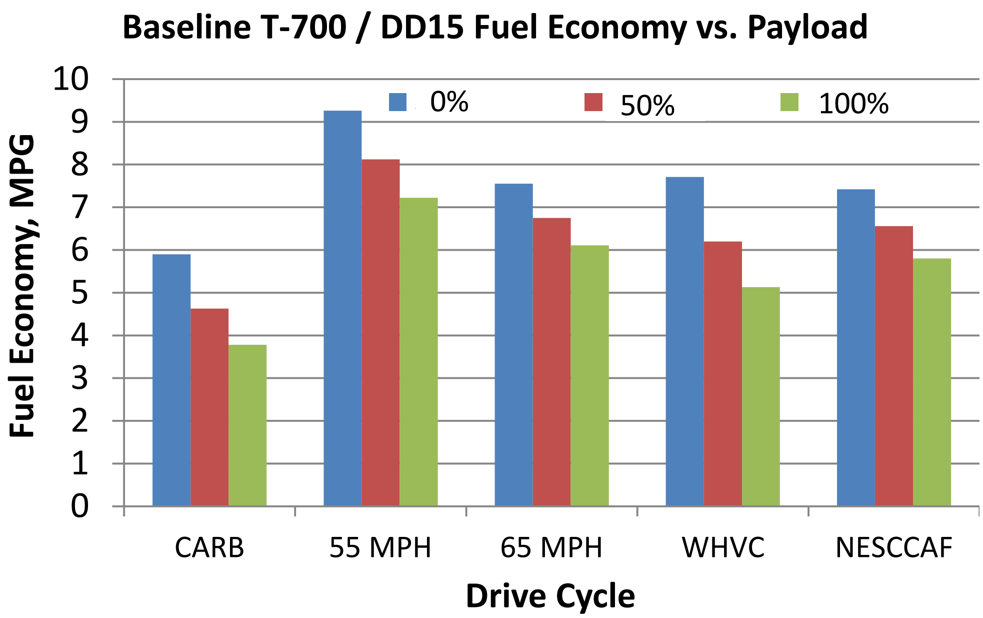


FIGURE 3.2 BASELINE T-700 / DD15 FUEL ECONOMY IN MPG AS A FUNCTION OF PAYLOAD

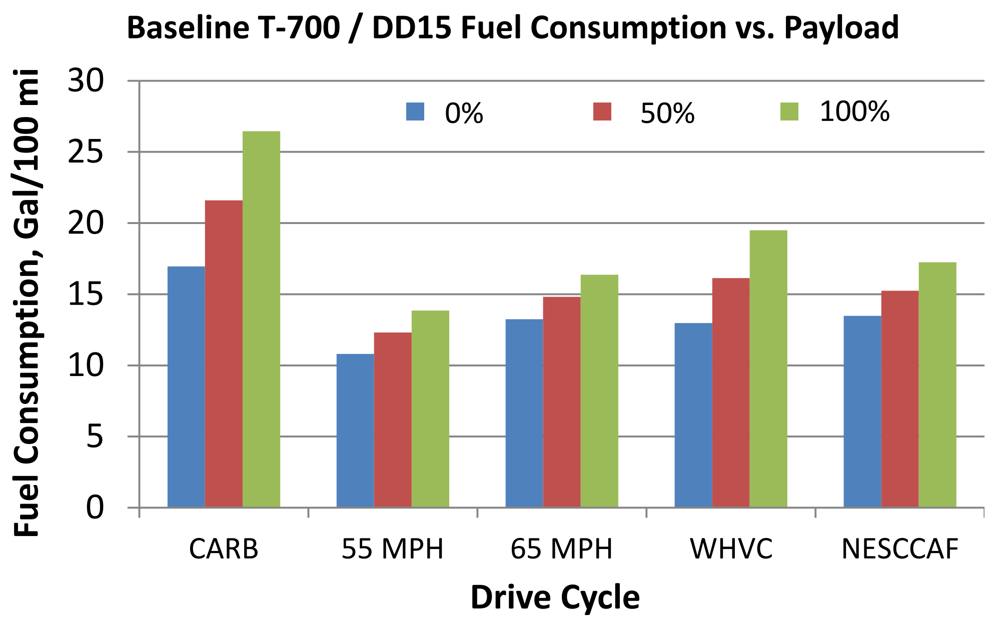


FIGURE 3.3 BASELINE T-700 / DD15 FUEL CONSUMPTION IN GALLONS PER 100 MILES AS A FUNCTION OF PAYLOAD

TABLE 3.13 FUEL ECONOMY OF THE BASELINE T-700 / DD15 VEHICLE IN MPG

	MPG @Percent Payload		
Drive Cycle	0%	50%	100%
CARB	5.90	4.63	3.78
55 MPH	9.26	8.12	7.22
65 MPH	7.55	6.75	6.11
WHVC	7.71	6.2	5.13
NESCCAF	7.42	6.56	5.80

TABLE 3.14 FUEL CONSUMPTION OF THE BASELINE T-700 / DD15 VEHICLE IN GALLONS/100 MILE

	Gallons/100 mi. @Percent Payload		
Drive Cycle	0%	50%	100%
CARB	16.9	21.6	26.5
55 MPH	10.8	12.3	13.9
65 MPH	13.2	14.8	16.4
WHVC	13.0	16.1	19.5
NESCCAF	13.5	15.2	17.2

3.3.1.2 Summary of DD15 engine technology results in T-700 truck

The engine technologies listed in Tables 3.13 and 3.14 above were all evaluated using the baseline tractor-trailer configuration. Appendix B describes the details of the DD15 model, its calibration, as well as the assumptions and parameters involved in simulating each of the considered technologies.

Table 3.15 below summarizes the results of engine technology simulations. Results are provided for each technology on 5 drive cycles. Each drive cycle is run at 3 payloads, to provide information on how sensitive a given technology is to payload. As a result, the table provides 15 data points for each of the 17 technologies that were evaluated. The results shown are in terms of percent reduction in fuel consumption compared to the baseline production 2011 DD15 engine GT model.

TABLE 3.15 FUEL SAVINGS RESULTS OF DD15 ENGINE, USING THE BASELINE T700 TRACTOR-TRAILER

Technology	Fuel Consumption Reduction (Percent) On Drive Cycle and at Percent of Maximum Payload														
	CARB			55 MPH			65 MPH			WHVC			NESCCAF		
	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%
2. Opt. TCPD	0.7	0.6	0.4	0.7	0.7	0.8	0.6	0.6	0.6	0.7	0.6	0.5	0.4	0.4	0.3
3. Elec. TCPD	0.9	0.9	0.8	0.8	0.9	1.0	0.8	0.9	1.0	0.9	0.9	0.9	0.8	0.9	0.8
4. No EGR	0.1	0.3	0.5	0.6	0.7	0.7	0.5	0.6	0.7	0.2	0.4	0.5	0.5	0.6	0.7
5. No TCPD	1.5	1.2	0.9	0.9	0.7	0.7	1.3	1.2	1.1	1.3	1.0	0.8	1.3	1.0	1.0
6. No EGR or TCPD	1.7	1.6	1.5	1.6	1.6	1.6	2.0	2.0	2.0	1.6	1.5	1.4	1.9	1.7	1.7
7. Asym. Turbo	1.6	1.3	1.0	1.0	0.8	0.8	1.4	1.3	1.2	1.4	1.1	0.9	1.3	1.1	0.9
8. Low Back Pres.	0.3	0.4	0.4	0.3	0.3	0.4	0.4	0.4	0.5	0.3	0.4	0.4	0.4	0.5	0.5
9. Low Intake Rest.	0.0	0.1	0.1	0.0	0.0	0.1	0.1	0.1	0.1	0.0	0.1	0.1	0.1	0.1	0.1
10. 0 CAC Rest.	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
11. Reduced FMEP	6.1	4.6	3.7	3.8	3.1	2.7	2.4	2.1	1.9	5.3	3.9	3.1	2.8	2.3	2.0
12. + 5% Turbo Eff	2.4	2.3	2.2	2.3	2.4	2.5	2.8	2.8	2.9	2.3	2.3	2.2	2.6	2.5	2.5
13. No EGR, no TCPD, +5% T	2.4	2.5	2.5	2.5	2.5	2.7	3.0	3.1	3.2	2.4	2.5	2.5	2.8	2.8	2.8
14. Downspeed A	5.0	3.9	3.0	3.2	2.7	2.3	2.3	2.0	1.5	4.4	3.4	2.7	3.1	2.4	2.1
15. Downspeed B	5.7	4.2	3.3	0.1	0.8	1.1	4.5	4.0	3.6	4.0	3.1	2.5	5.0	3.8	3.4
16. Downsized Torque	5.0	3.7	2.9	2.3	1.8	1.6	2.3	2.0	1.8	4.1	2.9	2.3	2.5	2.0	1.7
17. Downsized BMEP	5.6	3.9	2.5	2.1	1.5	0.8	1.7	1.6	0.3	4.2	2.7	1.7	2.5	1.9	2.0
18. VVA	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
19. Water BC	N/A	N/A	N/A	3.7	4.1	4.4	4.3	4.6	4.8	N/A	N/A	N/A	4.2	4.7	5.0
20. R245 BC	N/A	N/A	N/A	2.6	2.8	2.9	2.5	2.7	2.8	N/A	N/A	N/A	2.4	2.6	2.6

One technology listed in Table 3.3 does not have results shown in Table 3.15. The benefits from a study of variable valve lift and duration at a few engine operating points were disappointing, with fuel savings of under 1%. This result may come as a surprise to those with spark ignited engine experience, where VVA is much more effective, but this result is typical for diesel engines [NESCCAF, 2009]. Given the substantial effort involved in developing a new fuel map with VVA (the valve events must be optimized separately at every operating condition), the decision was made to not progress this technology to a full engine map and vehicle simulation evaluation. See Appendix B for more details.

The results shown in Table 3.15 can be presented in graphical form. To simplify the graph, only results for the 50% payload are shown in Figure 3.4 below. The maximum Y-Axis value of 16% is used to maintain consistency with later plots of vehicle technology results.

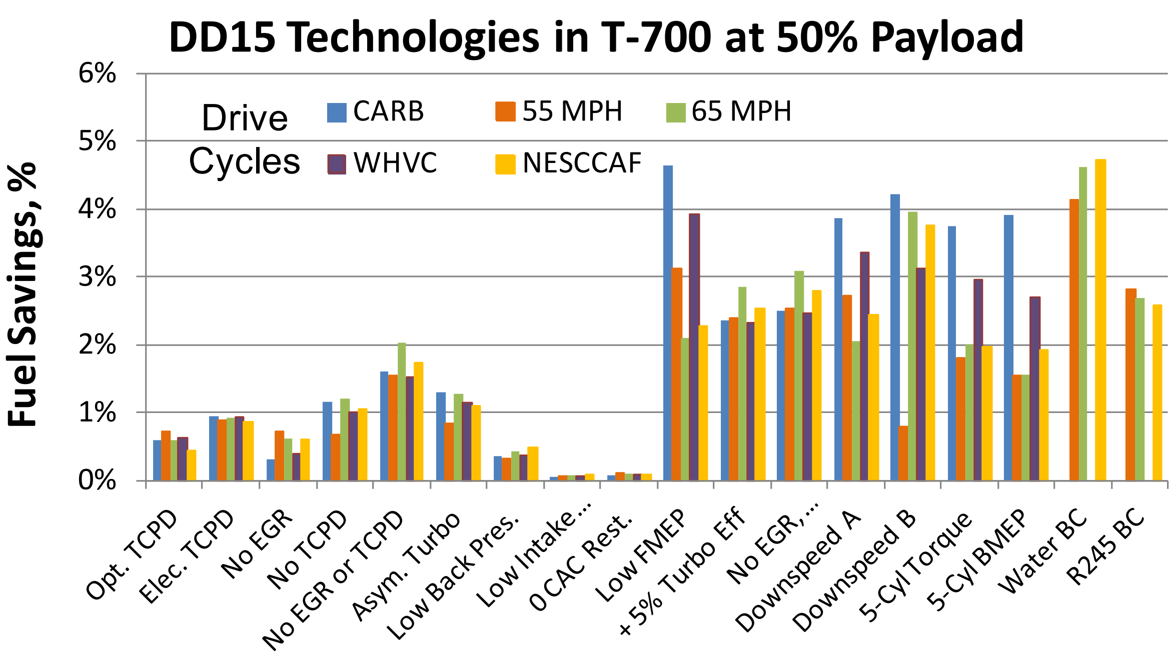


FIGURE 3.4 FUEL SAVINGS RESULTS OF DD15 ENGINE, USING BASELINE T-700 TRACTOR-TRAILER

In the following subsections, each engine technology is given its full name in the section heading, along with the abbreviated name used in Table 3.15 and Figure 3.4. The abbreviations are provided in parentheses.

3.3.1.3 Technology #2, Optimized Turbocompound (Opt. TCPD)

The Optimized Turbocompound technology represents an effort to modify the base engine by going to a power turbine that is downsized, in an effort to extract more exhaust energy at cruise conditions. The penalty for this is higher backpressure (and thus pumping loss) on the engine at high speed and load. In the simulated drive cycles, the engine rarely operates near rated speed, so this penalty is insignificant. Nevertheless, as the results in Table 3.15 and Figure

3.4 show, the benefit from downsizing the turbocompound is small (under 1%), regardless of the drive cycle.

3.3.1.4 Technology #3, Electric Turbocompound (Elec. TCPD)

An electric turbocompound uses the output of the power turbine to drive a generator. The electricity is then used to drive an electric motor which is connected to the crankshaft. This allows the power turbine speed to be independent of engine speed, which helps achieve better efficiency across a wider portion of the engine operating range. However, as the results presented above show, the fuel consumption and GHG benefit is limited to 0.8% to 1% on all of the drive cycles.

3.3.1.5 Technology #4, No EGR

Driving EGR flow in an engine requires pumping work. The exhaust manifold pressure must be maintained higher than the intake manifold pressure in order for EGR to flow. Removing EGR has two benefits: pumping work can be reduced, and combustion heat release rates will increase. In the case of the DD15 engine, however, removing EGR does not eliminate much pumping work, because the power turbine keeps exhaust manifold pressure high. As a result, removing EGR has only a marginal benefit on this engine – less than 1%. Note that this no-EGR technology involves very high engine-out NOx, which in turn requires a very high SCR conversion efficiency and high urea consumption, which could present significant challenges in OBD.

3.3.1.6 Technology #5, Turbocompound Removed (No TCPD)

For 2013, Detroit removed the turbocompound system from one version of the DD15. SwRI simulated this approach in two steps. In the first step, we removed the turbocompound system, but retained a standard dual entry fixed geometry turbocharger and the EGR system. This technology is labeled “No TCPD” in the table and figure above. The turbo was rematched to accommodate the removal of the turbocompound power turbine. This configuration is not able to flow EGR at lower engine speeds. One option is to add an intake throttle to drive EGR flow. This approach would increase fuel consumption at lower engine speeds, due to pumping losses. The other alternative is a higher conversion efficiency SCR and higher urea consumption, which is the option modeled here. It is something of a surprise that removing the turbocompound system provides a slight fuel consumption benefit of 0.7% to 1.5% on all drive cycles and at all payloads. Note, however, that none of the evaluated drive cycles and payloads require the engine to operate at full load for a significant period of time. In the past, several researchers have found benefits from adding turbocompound (See Section 2), but a recent paper from Daimler describing the new version of the DD15 [MTZ 2013] provides experimental data similar to SwRI’s simulation results.

3.3.1.7 Technology #6, EGR and Turbocompound Removed (No EGR or TCPD)

Once the turbocompound system is removed, the engine can benefit from not requiring backpressure to drive EGR flow. This technology simulation determined that the benefit, assuming no changes in turbocharger efficiency, approaches 2%. Note again that this no-EGR technology involves very high engine-out NOx, which in turn requires a very high SCR conversion efficiency and high urea consumption.

3.3.1.8 Technology #7, Asymmetric Turbo (Asym. Turbo)

This technology is a simulation of the 2013 model DD15 recently introduced by Detroit. In this version, there is no turbocompound. The conventional dual entry turbocharger is replaced by an asymmetric dual entry, fixed geometry turbocharger. One turbine volute is substantially smaller than the other. The small volute imposes a backpressure on the front 3 cylinders, enabling EGR flow from these three cylinders. The back 3 cylinders are able to run with a positive intake to exhaust pressure ratio, also known as a positive Δp. There is a wastegate on the larger turbine volute, which can be modulated to retain the desired back pressure on the front 3 cylinders, so that they achieve the required EGR flow. This technology provided results 0.8% to 1.6% better than the baseline across the range of drive cycles and payloads. Note that none of the simulated drive cycles and payloads require full engine power for a significant period of time. It should be noted that this technology approach is currently unique to Detroit (Daimler) engines in the heavy-duty industry. All other manufacturers use a variable geometry turbocharger to regulate the manifold pressure ratio and thus EGR flow.

3.3.1.9 Technology #8, Reduced Exhaust Backpressure (Low Back Pres.)

Engine efficiency can be improved by reducing restrictions on air flow. Modern aftertreatment systems impose higher back pressure on the engine than the muffler-only systems used before 2007. For this technology simulation, exhaust back pressure was reduced by about 60%. This amount of back pressure reduction would be very difficult to achieve in production feasible hardware, but even with the very optimistic reduction in back pressure, the effect was 0.5% or less for all drive cycles and payloads.

3.3.1.10 Technology #9, Reduced Inlet Restriction (Low Intake Rest.)

This simulation evaluated the effect of lower restriction through the intake air system, air filter, and plumbing connecting to the turbocharger compressor inlet. A 50 – 60% reduction in restriction was modeled. Very small benefits (under 0.1%) were found for all drive cycles and payloads.

3.3.1.11 *Technology #10, Reduced Charge Air Cooler Restriction (0 CAC Rest.)*

The charge air cooler system includes the plumbing from the turbocharger compressor outlet to the charge air cooler, and then back to the intake manifold. A 50 – 60% reduction in restriction was modeled, and extremely small benefits (around 0.1%) were found for all drive cycles and payloads.

3.3.1.12 *Technology #11, Reduced Engine Friction (Low FMEP)*

Assuming a reduction in FMEP ranging from 10% at high speed and load, to 35% at low speed and light load (see Appendix B, page 33 for details), significant benefits were found. The benefits are duty cycle and payload dependent. Low speed, low average power demand cycles such as the CARB cycle show the biggest benefit, exceeding 6% at zero payload. High average power demand cycles such as 65 MPH cruise and the NESCCAF cycle show a 2% to 2.5% benefit, less than half the benefit found on the CARB cycle. For all drive cycles, the largest benefit occurs with zero payload, and the smallest benefit at 100% payload. This result is to be expected, since at light load the FMEP makes up a greater portion of overall engine output.

3.3.1.13 *Technology #12, High Efficiency Turbo (+5% Turbo Eff.)*

SwRI applied an assumption of a 5% increase in both compressor and turbine efficiency, under all operating conditions, resulting in an overall turbocharger efficiency increase of 10%, not 10 points (for example, from 50% to 55% overall efficiency). This technology provides a fuel savings of 2.2 – 2.9% across all drive cycles and payload levels. Turbocharger efficiency can be improved with different bearing technology, tighter wheel to housing clearances, and other features. SwRI is aware of some experimental turbochargers that around 10% more efficient than current products, but none for which published results are available. The benefit of increased turbo efficiency is linear. As a result, if a 2% overall improvement is obtained, the fuel savings will be about 20% of the values provided in this report. More discussion can be found in Appendix B.

3.3.1.14 *Technology #13, No EGR, No Turbocompound, High Efficiency Turbo (No EGR, TCPD, +5%T)*

This simulation represents a combination of three technologies. The EGR system and turbocompound systems are removed, and a high efficiency turbocharger replaces the original turbocharger. The turbo efficiency increase is the same as in Technology #12, but in this case the turbo is rematched to work without a turbocompound system. The benefit of this technology is 2.4 to 3.2% fuel savings. The performance of the technology is slightly higher on the high power demand cycles, such as the 65 MPH cruise. Note that this no-EGR technology involves very high engine-out NOx, which in turn requires a very high SCR conversion efficiency and high urea consumption. Achieving very high conversion efficiency across the entire operating range is a significant technical challenge. Providing OBD functionality that can detect any problems such a system with sufficient resolution is an even larger challenge.

3.3.1.15 Technology #14, Downspeed A

The engine torque curve was modified to generate higher torque at low speed, and the maximum engine speed was reduced. Details of the engine BSFC map are provided in Appendix B. The current cylinder pressure limit of just over 200 bar was retained. The vehicle's axle ratio was modified from 3.36 to 2.97 to reduce cruise RPM from 1368 RPM to 1209 RPM at 65 MPH, while retaining approximately the same power margin at the 65 MPH cruise point. Details of the changes are shown in Appendix C. The downspeeding strategy had the largest benefits on the low speed drive cycles and at zero payload, where benefits over 4% can be found. The value at high speed and load is closer to 2%.

3.3.1.16 Technology #15, Downspeed B

Downspeed strategy B is more radical than Downspeed A. The peak torque RPM was pushed down to 1000 RPM, and torque was increased again. The rated engine speed was reduced to 1600 RPM. Details of the engine BSFC map are provided in Appendix B. The vehicle's axle ratio was modified from the baseline of 3.36 to 2.58 to reduce cruise RPM to 1051 RPM at 65 MPH, while retaining approximately the same power margin at the 65 MPH cruise point. Downspeed B provided larger fuel saving benefits under all drive cycles except for the 55 MPH cruise and the WHVC. At 55 MPH, the vehicle needed to run a gear down compared to Downspeed A, which resulted in a smaller fuel savings than was achieved by Downspeed A. The WHVC high speed cruise portion is also at low road speed (about 50 MPH), so Downspeed B runs that a gear down. The savings at 55 MPH were still in the 1 to 2% range.

3.3.1.17 Technology #16, Downsize at Constant Torque

The DD15 model was downsized to 12.5 liter. The goal is to reduce displacement, and thus engine friction. At light loads, the smaller size engine operates at higher BMEP than the 15 liter engine, which provides an efficiency benefit. For this configuration, the smaller engine retains the same power and torque as the original, which means that full load BMEP levels are increased. Vehicle performance in this case is unchanged. There is often a fuel consumption penalty at high BMEP, because timing must be retarded to limit cylinder pressure. The extent of the timing retard, and the BMEP level at which a retard is required, are functions of the cylinder pressure limit of a given engine. As expected, this technology provides the largest benefits (up to 5%) in the low speed CARB drive cycle at zero payload. At high speeds and loads, the benefit drops under 2%.

3.3.1.18 Technology #17, Downsize at Constant BMEP

This technology is similar to #16, except that the torque curve is reduced to keep maximum BMEP equal to that of the original 15-liter 6-cylinder engine. As a result, vehicle performance will suffer due to a 17% reduction in power and torque. On the 5 drive cycles and 3 payloads evaluated, this configuration provided fuel savings similar to or slightly lower than those of Technology #16.

As a simple metric for the penalty of a lower power rating, consider the minimum speed reached on the 3% grade segment of the NESCCAF cycle. For a vehicle at 100% payload, the minimum speeds on a 3% grade are:

Baseline DD15	46.3 MPH
Downsized DD15	36.9 MPH

The vehicle was able to follow the remaining drive cycles without an issue, despite the reduction in available power and torque. In real-world driving, there would be penalties on acceleration times and speeds on a grade. For the NESCCAF cycle, the cycle time is lengthened for the lower power engine, so that the vehicle covers the same distance, but at a slightly lower average speed.

3.3.1.19 Technology #18, Variable Valve Actuation (VVT)

VVT was explored in the GT-POWER model of the engine, but the fuel consumption benefits were very small (well under 1%). Given the high cost of the system and the intensive analysis to optimize VVT over the whole operating range, VVT was dropped from the project. It should be noted, however, that VVT can be of value in providing thermal management (increased heat) for the aftertreatment system. There are several alternatives for thermal management, including an intake throttle, retarded ignition timing, and changes in exhaust valve opening timing (from a VVT system). The VVT approach may be the lowest fuel consumption approach to thermal management, but exploring this approach was outside of the project scope.

3.3.1.20 Technology #19, Water-Based Bottoming Cycle (Water BC)

A bottoming cycle is a Rankine cycle heat engine that in this case extracts energy from the EGR stream and the post-aftertreatment exhaust stream. Water is boiled with EGR and exhaust heat. The water drives an expander (turbine or piston expander) to extract power. A condenser located at the front of the truck condenses the post-expansion steam back into water. A pump then drives the water under pressure back to the boiler. Details of the bottoming cycle, including assumptions about heat exchangers, temperatures, and pressures, are in Appendix D. Because bottoming cycles have very poor transient response, and the simulation model does not have transient capability, the simulation results are only reported for cycles with extensive steady state components. On these cycles (55 MPH, 65 MPH, and NESCCAF), the water based bottoming cycle provides a benefit of 4.1 to 4.7%.

Note that a bottoming cycle system adds significant hardware, and thus weight, to the truck. This analysis does not include the effect of increased weight. On the test cycles run, this effect will be very small. However, for operators who run at the maximum legal weight, the addition of a bottoming cycle will reduce their payload, and thus the freight efficiency, compared to the performance shown here. Another important note is that while all other engine technologies are compared to the baseline DD15 engine, the bottoming cycle results are based on the asymmetric turbo technology (5.3.1.8). This is because the turbocompound system is a method of extracting energy from the exhaust, as is a bottoming cycle. The performance of the bottoming cycle is compromised if there is another energy extraction feature in front of it.

3.3.1.21 Technology #20, R245 Refrigerant-Based Bottoming Cycle (R245BC)

R245 is an alternative material that avoids the freezing issue of a water-based system. R245 has a high GHG factor, so there are other materials with similar thermodynamic properties that are under development. From a thermodynamic perspective, however, a model using R245 should represent the performance of these alternative materials. The main thermodynamic difference between water and R245 is that the refrigerant has a low temperature limit. To stay under that limit, a much higher mass flow is used in the bottoming cycle system. Because bottoming cycles have very poor transient response, and the simulation model does not have transient capability, the simulation results are only reported for cycles with extensive steady state components. The efficiency of the cycle is lower with R245, because of the lower operating temperature. See Appendix D for details. On the drive cycles of interest (55 MPH, 65 MPH, and NESCCAF), the R245 bottoming cycle provides fuel savings of 2.6% to 2.8%. These results are lower than results reported in the SuperTruck program. At least one of those bottoming cycles includes a recuperator, which should increase the performance of the cycle. This feature will be evaluated in the second project report.

3.3.2 T-700 Class 8 Tractor-Trailer Truck Vehicle Technology Results

The vehicle technologies listed in Table 3.9 were all evaluated using the baseline version of the DD15 engine. Appendix G describes the details of each vehicle technology, including parameters and assumptions. Performance of the baseline vehicle is described above in Figures 3.1 and 3.2, as well as Tables 3.11 and 3.12.

Table 3.14 below summarizes the simulation results for all of the vehicle technologies. Results are provided for each technology on 5 drive cycles. Each drive cycle is run at 3 payloads, to provide information on how sensitive a given technology is to payload. As a result, the table provides 15 data points for each of the 10 technologies that were evaluated. The results shown are in terms of percent reduction in fuel consumption compared to the baseline production T-700 tractor-trailer vehicle model.

Note that the road speed governor, Technology NN, was only simulated on the NESCCAF cycle. The remaining drive cycles have N/A shown for the road speed governor simulation.

TABLE 3.16 FUEL SAVINGS RESULTS OF T-700 VEHICLE TECHNOLOGIES, USING THE BASELINE DD15 ENGINE

Technology	Fuel Consumption Reduction (Percent) On Drive Cycle and Percent of Maximum Payload														
	CARB			55 MPH			65 MPH			WHVC			NESCCAF		
	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%
HH. A/C -40%	1.0	0.8	0.7	1	0.5	0.4	0.4	0.4	0.3	0.9	0.7	0.5	0.4	0.4	0.3
II. Cd - 25%	2.0	1.6	1.3	14	12	11	16	15	13	5.7	4.5	3.7	14	12	10
JJ. Crr - 30%	3.0	4.0	4.6	6	9.1	11	5.2	7.8	10	4.4	5.7	6.5	4.8	6.9	8.4
KK. Weight – 2200 lb	2.5	2.1	1.8	1.3	1.2	1.1	1.1	1.0	0.9	2.2	1.9	1.7	1.2	1.2	1.1
KK. Weight – 4400 lb	5.0	4.2	3.6	3	2.4	2.1	2.2	2.0	1.8	4.3	3.9	3.3	2.3	2.4	2.1
LL. Chassis Fr - 20%	0.5	0.6	0.6	0.9	0.9	0.9	0.9	0.9	0.9	0.7	0.7	0.7	0.8	0.8	0.8
MM. 6X2 Axles	1.3	1.4	1.4	1.7	1.7	1.6	1.7	1.6	1.6	1.5	1.5	1.5	1.6	1.6	1.5
NN. Vmax = 65	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	2.2	1.6	1.4
NN. Vmax = 60	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	10	8.0	6.8
NN. Vmax = 55	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A	18	14	12
OO. 18-spd AMT	0.5	0.4	0.2	0.5	0.7	0.8	0.7	0.8	0.8	-0.8	-0.7	-0.7	0.6	0.6	0.8
PP. 10-spd Manual	-8.3	-6.3	-5.2	-16	-13	-11	0.0	0.0	0.0	-9.3	-6.9	-5.4	-0.8	-1.1	-0.8

The results shown in Table 3.16 can be presented in graphical form. To simplify the graph, only results for the 50% payload are shown in Figure 3.5 and 3.6 below. The maximum Y-Axis value of 16% is consistent with the engine results in Figure 3.4. Again, note that the road speed governor was only simulated on the NESCCAF cycle, so the alternative road speed governor settings have only a single bar of data. A review of Figure 3.5 shows that certain vehicle technologies have a much larger effect than the engine technologies shown in Figure 3.4. It is also apparent that the fuel savings offered by most vehicle technologies is very duty cycle and payload dependent. Figure 3.6 expands the Y-axis to cover negative values. The 18-speed AMT has a slight penalty compared to the 10-speed AMT on the WHVC. This is because the 10-speed AMT runs the high speed portion of the WHVC in top gear, at approximately 1133 RPM, while the 18-speed AMT runs this portion of the cycle in 17th gear, at approximately 1335 RPM. The 10-speed manual transmission has a penalty on all cycles except the 65 MPH cruise. The largest penalty is at 55 MPH, where the manual transmission runs a gear down. Recall that there is no physical difference in gearing between the 10-speed manual and AMT transmissions. The manual has upshift speeds set 200 RPM higher than the AMT, to simulate “typical” driver behavior.

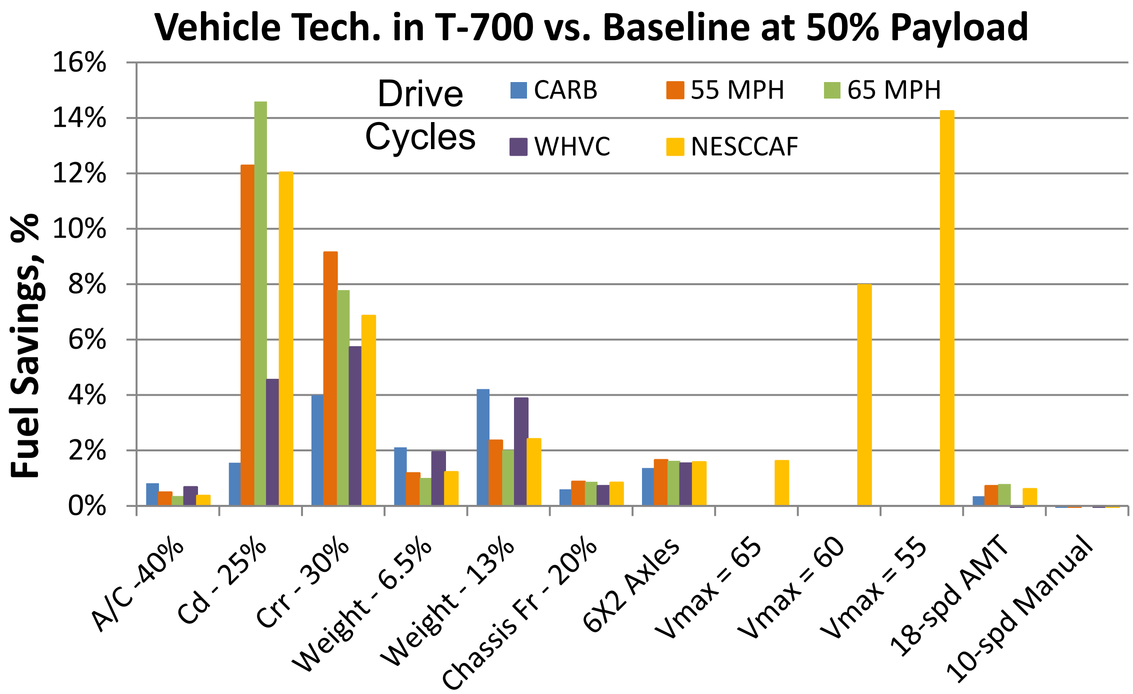


FIGURE 3.5 FUEL SAVINGS RESULTS OF VEHICLE TECHNOLOGIES IN THE T-700 TRACTOR-TRAILER, USING THE BASELINE DD15 ENGINE

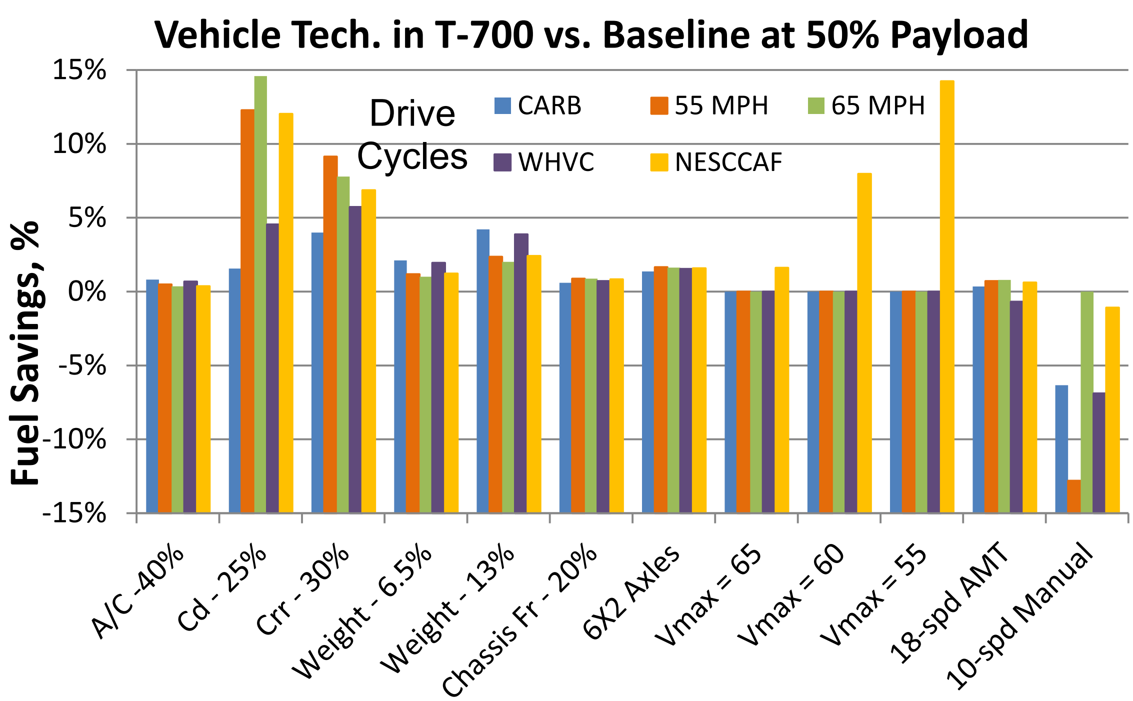


FIGURE 3.6 SAME RESULTS AS FIGURE 3.5, BUT SHOWING NEGATIVE FUEL SAVINGS VALUES FOR THE 18-SPD AMT AND THE 10-SPD MANUAL TRANSMISSIONS ON CERTAIN CYCLES

In the following subsections, each vehicle technology is given its full name in the section heading, along with the abbreviated names used in Table 3.16 and Figure 3.5. The abbreviations are provided in parentheses.

3.3.2.1 Technology HH, Reduced A/C Power Demand (A/C -40%)

For all duty cycles evaluated in this project, the air conditioner is operating. Average A/C power demand is assumed to be 1.5 Kw, and was applied as a steady load. The reduced A/C power demand technology set is assumed to reduce power demand by 40% to 900 watts. This provides a 0.3% to 1% benefit, with the largest benefit occurring on the CARB cycle at zero payload, where the average engine operating loads are the lowest. The smallest benefit is observed on the 65 MPH cruise cycle, where vehicle power demand is high. Note that in actual service, the air conditioner is not used all the time, so the real-world benefits will be less than those shown in Table 3.16. On the other hand, this technology can represent any feature that reduces auxiliary load by 600 Watts.

3.3.2.2 Technology II, Improved Cd (Cd – 25%)

This technology involves a substantial reduction in aerodynamic drag for the tractor-trailer combination. Both tractor and trailer features will be required to achieve a 25% reduction in Cd. Detailed assumptions about the features required to achieve a given level of Cd improvement are provided in Appendix C. No specific set of features is simulated here. A reduction in Cd based on the literature review was assumed, and the manufacturer is free to put together a set of features that can meet the target. The results from a Cd reduction vary widely, from a 1.3% benefit on the CARB cycle at 100% payload up to an impressive 16% at 65 MPH with zero payload. The results are also strongly dependent on payload. Light payloads reduce the power demand from tire rolling resistance and the requirement to accelerate the vehicle inertia, and this makes aerodynamic drag a larger portion of the total power demand. Aerodynamic features will provide a significant benefit on any vehicle that runs a significant portion of its mileage at high road speeds.

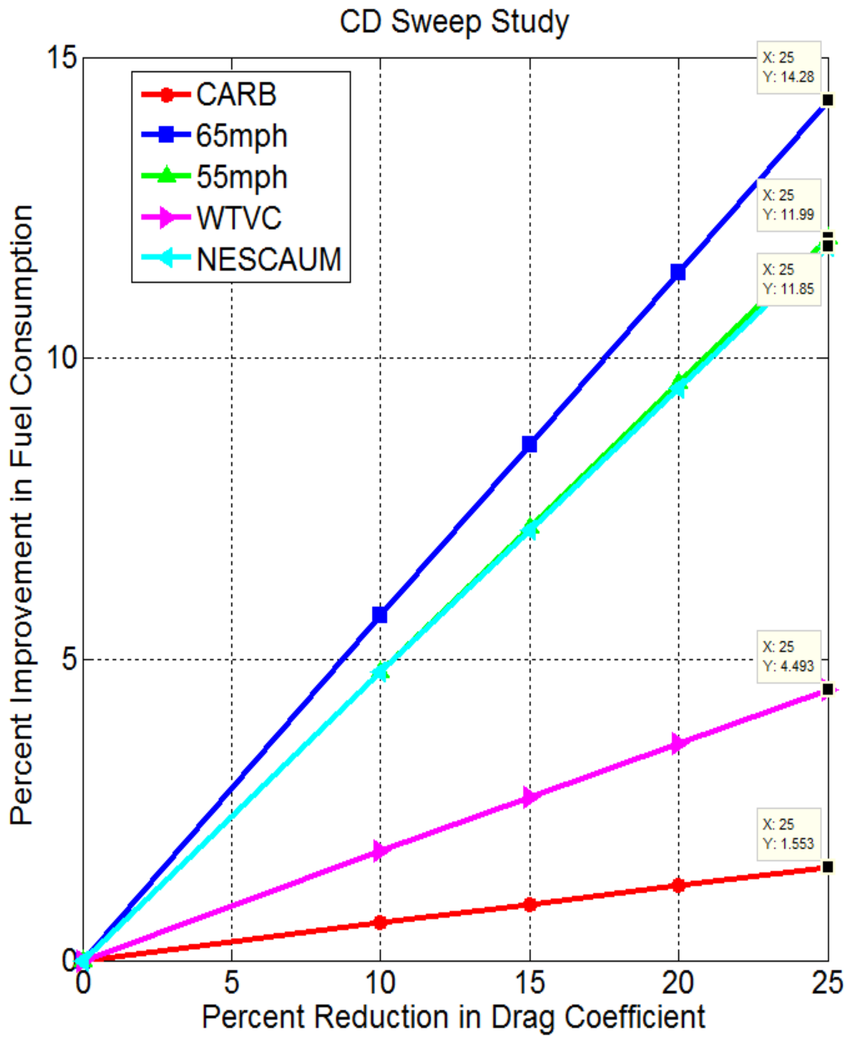


FIGURE 3.7 FUEL SAVINGS OVER A RANGE OF CD REDUCTIONS AT 50% PAYLOAD

Because of the large benefits obtained with Cd reductions, they were evaluated over a range of Cd values. These results are shown in Figure 3.7 below. A linear relationship between Cd reduction and fuel savings is shown. The first increment of Cd improvement, 10%, is based on the application of a full length trailer skirt. The 15% improvement increment assumes both a trailer skirt and boat tail. The 20% improvement increment assumes a full length trailer skirt that extends over the trailer bogey. This feature would require regulatory changes or design changes to stay within the width limit. The 20% improvement package also includes a gap reduction feature. The 25% package includes extending the tractor side skirts over the drive axles, and a feature to completely close the trailer gap. It is worth noting that most of the aerodynamic improvement features are added to the trailer, not the tractor. See Section 2 for a discussion of the potential for tractor aerodynamics separate from the trailer.

3.3.2.3 Technology JJ, Improved Crr (Crr – 30%)

This technology assumes an improvement in tire rolling resistance by 30%, based on the literature review. The specific features that would be required to achieve this improvement were not simulated. Tire rolling resistance is most important at moderate speeds in steady state operation, such as the 55 MPH cruise cycle. At higher speeds, aerodynamic drag grows, which

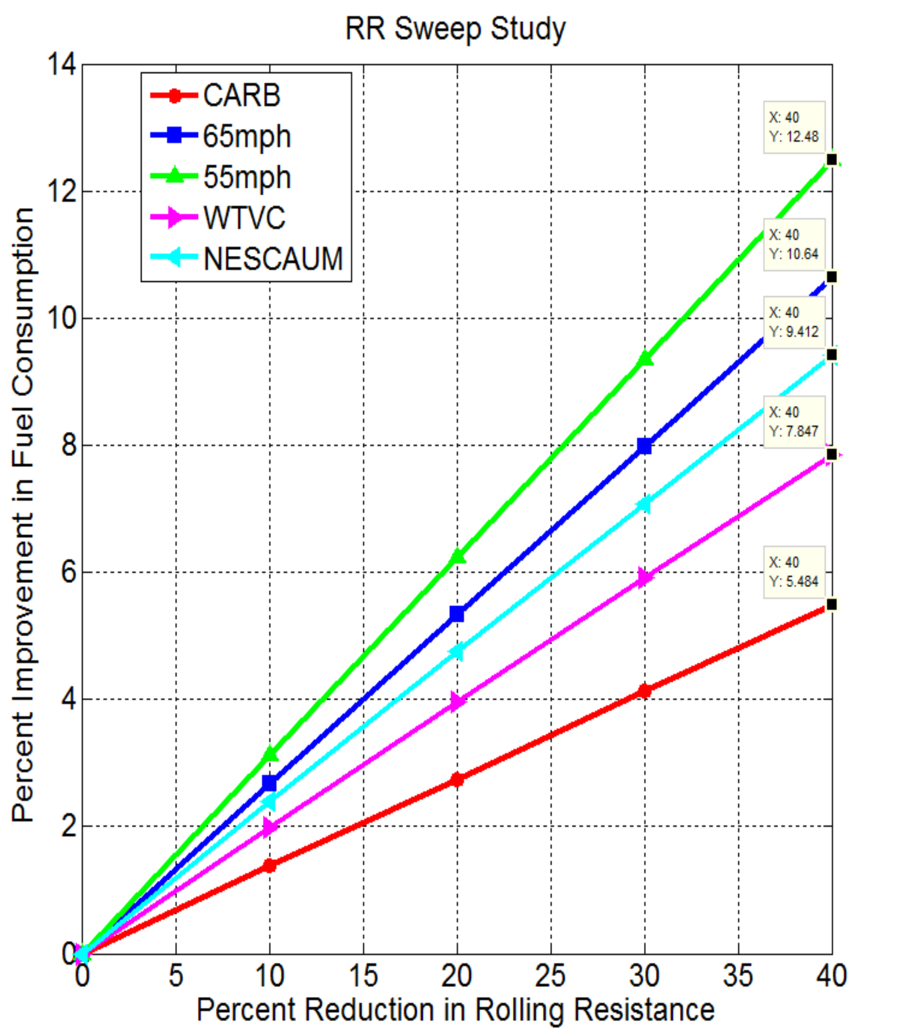


FIGURE 3.8 FUEL SAVINGS OVER A RANGE OF CRR REDUCTIONS AT 50% PAYLOAD

reduces the rolling resistance share of total vehicle power demand. In highly transient cycles such as CARB, the power required to accelerate the vehicle inertia overshadows the rolling resistance power demand. The benefit of lower Crr is very payload dependent. At 100% payload, rolling resistance plays a larger share in overall vehicle power demand than at 0% payload.

Because of the large benefits obtained with Crr reductions, they were evaluated over a range of Crr values from a 10% improvement to a 40% improvement, which is larger than the project target of 30%. These results are shown in Figure 3.8 above. A linear relationship between Crr reduction and fuel savings is shown. At 100% payload, the trailer tires carry 42.5% of the vehicle weight, so trailer tires make a substantial contribution to overall vehicle rolling resistance. With the Crr values assigned to the steer, drive, and trailer axles, 36.2% of the total vehicle rolling resistance can be attributed to the trailer tires at full payload. At zero payload, the trailer tires carry about 22% of the total vehicle weight, and account for about 17% of rolling resistance.

Some background on the math in the previous paragraph: at 100% payload, the truck weighs 80,000 pounds. Each tandem axle set (tractor tandem and trailer tandem) is allowed to carry 34,000 pounds, for a total of 68,000 pounds on the tandems. This leaves 12,000 pounds on the steer axle. The trailer tires thus carry 34,000 out of 80,000 pounds, or 42.5% of the total vehicle weight. At zero payload, the vehicle weighs 34,000 pounds total, including an empty trailer that weighs 15,000 pounds. The trailer tandem carries half the trailer weight, or 7,500 pounds. This means that the trailer tires carry 7,500 pounds out of the total of 34,000, or 22%. The rolling resistance coefficient of trailer tires is normally less than that of the tractor tires, especially the traction tires used on the drive axles, so the trailer accounts for a slightly lower percentage of total vehicle rolling resistance than the portion of vehicle weight carried by the trailer tires.

3.3.2.4 Technology KK, Weight Reduction (Weight – 2,200 lb. and Weight – 4,400 lb.)

The baseline T-700 tractor-trailer vehicle has an empty weight of 34,000 pounds. This technology simulation looked at the effect of a 2,200 and 4,400 pound weight reduction (6.5% and 13% of the empty vehicle weight). The payload weight was left unchanged, so the weight of the loaded vehicle decreased by the same amount as the empty weight. The fuel savings found were linearly proportional to the weight reduction. A 6.5% weight reduction was worth 0.9% to 2.5%, while the 13% weight reduction provided benefits from 1.8% to 5%. The benefits of weight reduction are greatest in transient operation, where lower weight translates into reduced power demand for acceleration. At steady speed, the weight reduction only affects rolling resistance. Therefore, the largest benefits are seen on the CARB cycle, and the smallest benefit is on the 65 MPH cycle. The benefit of weight reduction is linear, so an 1100 pound weight reduction will have half of the fuel savings of the 2200 pound weight reduction.

Operators who run below the maximum legal weight will see benefits from weight reduction as described in the paragraph above. Operators who run at the maximum legal load will increase payload to take advantage of a weight reduction. This means that their vehicle fuel consumption will be unchanged, but their load specific fuel consumption (gallons per 1000 ton-miles) will improve significantly. In practice, OEMs will struggle to maintain constant weight as new aerodynamic and safety features are added to trucks. Penalties for weight increases will essentially match the benefits for a weight reduction of the same size. More discussion of weight reduction technologies can be found in Appendix C.

3.3.2.5 Technology LL, Chassis and Driveline Friction Reduction (Chassis Fr -20%)

This simulation evaluates the potential for features such as improved axle efficiency, improved wheel bearings, and synthetic lubricants used in the driveline. The benefits range from 0.5% to 0.9%, with the high speed cycles showing the largest benefits.

3.3.2.6 Technology MM, 6X2 Axles

6X2 drive configurations are popular in Europe, but are just coming onto the market in North America. With the conventional 6X4 arrangement, the front axle in a tandem is less efficient than the rear axle of the tandem. This is because the front axle needs to split torque and pass along a portion of the torque to the rear axle in the tandem. There are more gear sets and oil seals in the front axle of a tandem. With the 6X2 setup, the single drive axle is basically a beefed-up version of the rear axle in a tandem pair. One significant penalty of the 6X2 setup is limited traction. Only a small portion of the total vehicle weight rests upon the drive axle. This makes 6X2 arrangements unsuitable for vehicles that need to go off road, or that frequently operate in low friction conditions. Benefits for the 6X2 configuration range from 1.3 to 1.7%. The larger benefits are on high speed cycles.

3.3.2.7 Technology NN, Road Speed Governor (Vmax = 65, Vmax = 60, Vmax = 55)

Road speed governors only have an effect on a duty cycle where the maximum vehicle speed is higher than the road speed governor setting. As a result, the road speed governor was evaluated on the NESCCAF cycle, which has cruise speeds ranging from 65 to 70 MPH. The drive cycle automatically compensated for the lower speeds by lengthening the cycle time, so that the same distance is covered regardless of cruise speed. On this cycle, a 65 MPH governor setting has only a modest benefit (1.4% to 2.2%), because the speed reduction is small. At 60 MPH, the benefits grow to a range of 6.8% to 10%, and at 55 MPH, the fuel savings range from 12% to 18%. The larger benefits are at zero payload, while the smallest benefits are at 100% payload. These fuel savings must be offset against the costs of longer trip times. If trip time does not increase, that means that speed is being limited by some factor other than the governor setting (traffic and road conditions, posted speed limits, vehicle capability, etc.). Longer trip time drives higher labor cost, and a larger number of vehicles will be required to deliver the same freight volume (ton-miles). Additional vehicles add both cost and congestion, offsetting at least some of the fuel savings from a lower governor setting.

3.3.2.8 Technology OO, 18-Speed AMT

The baseline 10-speed AMT is replaced by an 18-speed AMT to evaluate the potential benefit of closer gear splits. The narrow splits offer the potential to keep the engine closer to its best BSFC operating point. One downside of a higher gear count is more shift events. Every shift event with a manual or AMT causes a power interruption. Power interruptions reduce vehicle performance and annoy the driver. Power interruptions also have the effect of making the engine operate briefly in a less efficient transient condition. Unfortunately, since the modeling analysis is based on steady state engine fuel maps, the effect of adding additional transients cannot be evaluated.

The 18-speed AMT provided slight benefits (<1%) on all the drive cycles except the WHVC, where it caused a slight penalty (<1%). An examination of the engine fuel map shows that there is very little benefit to the slightly narrower operating range that an 18-speed AMT provides. In field service, the frequent shifting of the 18-speed would more than offset the slight advantage on fuel efficiency in transient operating conditions. In practice, as one of many examples, the 18-speed is used in heavy haul applications, where keeping the engine at maximum power is important.

3.3.2.9 Technology PP, 10-Speed Manual

Ten speed manual transmissions dominate the long haul market today, but the market share of AMT transmissions is growing. For this study, the manual and AMT have identical gear ratios. The only difference is the addition of hardware to enable computer-controlled shifting in the AMT. AMT transmissions have the advantage of taking away shift point decisions from the driver. The very best driver using a manual transmission may be able to beat the fuel efficiency of an AMT by a small margin, but the average driver cannot match AMT performance. The worst drivers use significantly more fuel with a manual than with an AMT. In this study, the "average" driver's shift strategy was modeled by simply pushing the upshift points higher by 200 RPM compared to the AMT shift schedule. Full throttle upshifts occur at 1400 – 1500 RPM for the AMT, and 1600 – 1700 RPM for the "manual". It should be noted that the AMT shift schedule used for this study (based on the Eaton Ultrashift production shift schedule) has the effect of limiting vehicle performance, since engine power drops significantly after each upshift at 1400 to 1500 RPM. A driver interested in good performance (as opposed to good fuel economy) would use higher engine speeds to maintain power after upshifts.

At 65 MPH cruise, there is no difference between the 10-speed manual and AMT, because in both cases, the truck is running in top gear. At 55 MPH, the manual is running a gear down. The higher engine speed hurts the fuel efficiency of the manual running in 9^{th} gear, but pushing engine BMEP down to a lower level (higher speed = lower torque) hurts efficiency even more. The efficiency penalty at 55 MPH for the manual is worst at 0% payload (16%). On the CARB and WHVC cycles, efficiency penalties of 5% to 9% are seen with the manual. On the NESCCAF cycle, the penalty is around 1%, because most of this cycle is run at higher road speed, with the transmission in top gear regardless of whether it is a manual or an AMT.

Another way of looking at the 10-speed manual results is as the inverse of the effectiveness of an AMT. To enhance the accuracy and repeatability of the vehicle simulations, the baseline vehicle was modeled with an AMT with programmed shift patterns rather than an operator-dependent manual transmission. Therefore, the inverse results for the CARB and WHVC cycles for 10-speed AMT indicate an efficiency increase of 5% to 9% over a 10-speed manual.

3.3.3 *Kenworth T270 Class 6 Delivery Truck Engine Technology Results*

Class 6 trucks are typically powered by 5 – 8 liter diesel engines. Ford, however, offers a 6.8 liter V-10 gasoline engine for Classes 4 – 7. Given the cost advantage of gasoline engines in medium-duty vehicles, SwRI expects that gasoline will become more popular in MD provided their efficiency and durability disadvantages can be mitigated.

Three engines are included in the evaluation of Class 2b – 6 vehicles for this study: the Cummins ISB 6.7 liter diesel, a 6.2 liter V-8 gasoline engine that is currently offered in Class 2b and 3 vehicles, and which may in time replace the older V-10 in heavier vehicles, and a Ford 3.5 liter turbocharged V-6 which is currently offered in Class 2a vehicles. The ISB diesel is considered the baseline engine for these vehicles. The Cummins ISB, branded under the PACCAR PX-7 name, is the standard engine sold in the T270 truck. Data for modeling the Kenworth T270 truck came from the example shown in the photographs below. A listing of vehicle parameters is provided in Appendix C.

FIGURE 3.9 KENWORTH T270 CLASS 6 BOX DELIVERY TRUCK

3.3.3.1 Baseline engine and vehicle results for T270 truck

The T270 truck was evaluated at the following payloads and vehicle weights: 0% payload and a vehicle weight of 17,141 pounds, 50% payload (4430 pounds of cargo) and a vehicle weight of 21,570 pounds, and 100% payload (8860 pounds of cargo) and a vehicle weight of 26,001 pounds. Figure 3.10 below shows the *fuel economy* performance of the T270 truck with the three engines in their baseline form, all evaluated at 50% payload. Figure 3.11 shows the *fuel consumption* results for the same engines and payload. The same data is provided in tabular form in Tables 3.17 and 3.18 below the figures.

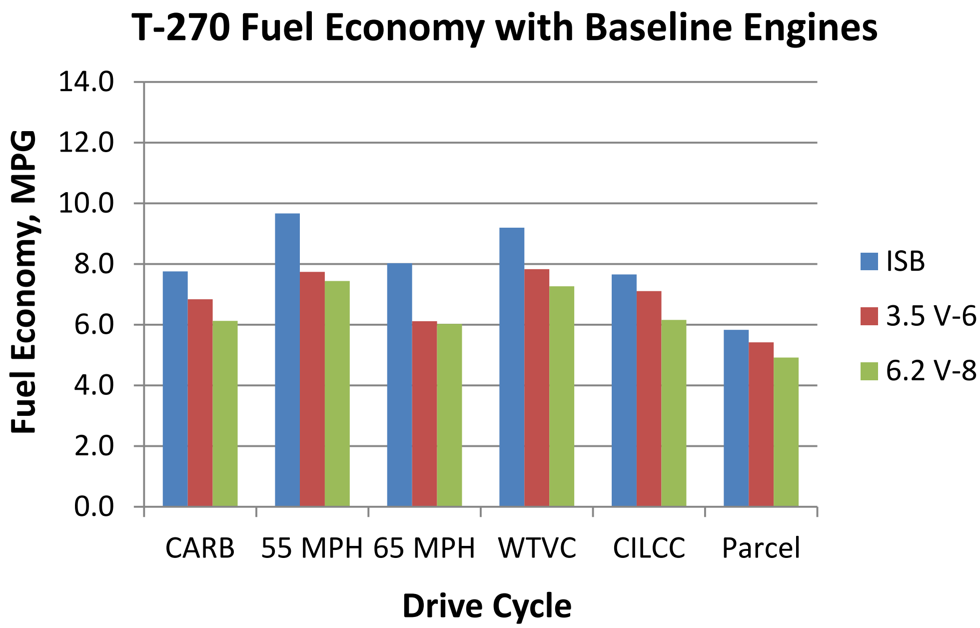


FIGURE 3.10 FUEL *ECONOMY* OF THE KENWORTH T270 BOX DELIVERY TRUCK AT 50% PAYLOAD, COMPARING THREE ENGINES

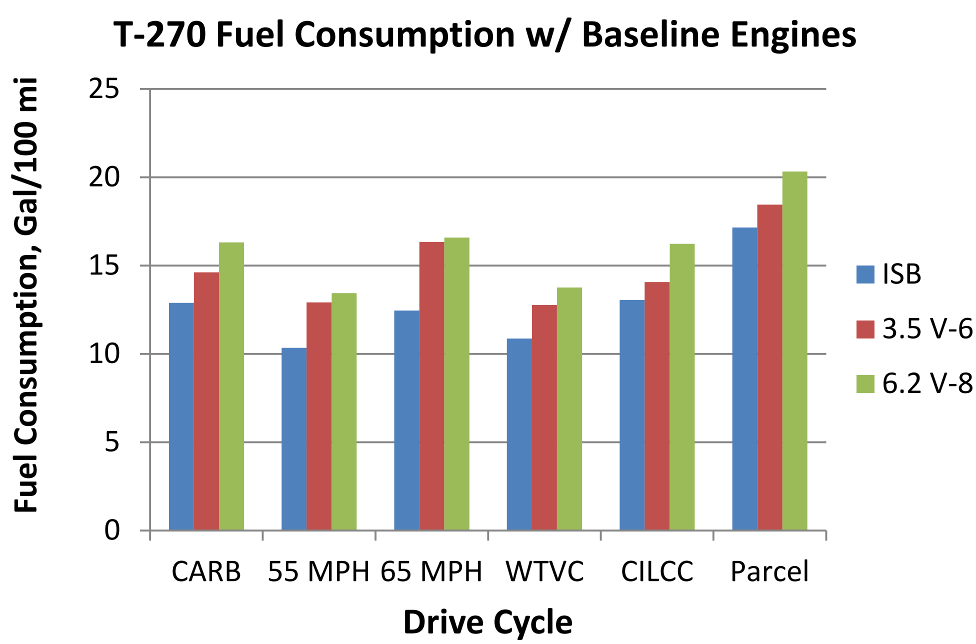


FIGURE 3.11 FUEL *CONSUMPTION* OF THE KENWORTH T270 BOX DELIVERY TRUCK AT 50% PAYLOAD, COMPARING THREE ENGINES

TABLE 3.17 FUEL *ECONOMY* OF THE KENWORTH T270 BOX DELIVERY TRUCK AT 50% PAYLOAD, COMPARING THREE ENGINES

Drive Cycle	ISB	3.5 V-6	6.2 V-8
CARB	7.7	6.8	6.1
55 MPH	9.7	7.7	7.4
65 MPH	8.0	6.1	6.0
WTVC	9.2	7.8	7.3
CILCC	7.6	7.1	6.2
Parcel	5.8	5.4	4.9

TABLE 3.18 FUEL *CONSUMPTION* OF THE KENWORTH T270 BOX DELIVERY TRUCK AT 50% PAYLOAD, COMPARING THREE ENGINES

Drive Cycle	ISB	3.5 V-6	6.2 V-8
CARB	12.9	14.6	16.3
55 MPH	10.3	12.9	13.4
65 MPH	12.5	16.3	16.6
WTVC	10.9	12.8	13.8
CILCC	13.2	14.1	16.2
Parcel	17.1	18.5	20.3

An alternative way of comparing the three engine options in the T270 is provided in Figure 3.12. For this figure, the 6.2 liter V-8 gasoline engine is considered to be the baseline, and the other two engines are compared to it. The 3.5 V-6 results provide a look at the benefits of downsizing and boosting, compared to a larger naturally aspirated engine. On lightly loaded duty cycles, such as the CARB cycle and the CILCC cycle, the smaller gasoline engine has a large advantage over the bigger V-8 (10 – 13%). However, on the most highly loaded cycle, 65 MPH cruise, there is no advantage at all for the smaller engine. The diesel comparison shows the benefit of converting a medium-duty truck from gasoline V-8 to diesel. On lightly loaded cycles, the diesel has about a 20% fuel consumption advantage. At high loads, the diesel's advantage approaches 30%. Remember that gasoline has about 16% less energy per gallon, so the thermal efficiency of the gasoline engines is not as poor as this comparison makes it appear. Also, keep in mind that these comparisons are all for the baseline engines, without any potential improvement technology.

There is one oddity in the results shown in Figure 3.12. This is the relatively poor performance of the diesel engine on the Parcel Delivery cycle. This cycle includes a lot of accelerations that are more aggressive than those of the CILCC cycle, which should tend to favor the diesel. The key to the poor performance of the diesel on this cycle, however, is the high percentage of time spent at idle on the Parcel cycle, where the baseline truck does not include the feature "auto neutral at stop". Typically, a diesel engine will have lower fuel consumption at idle than a gasoline engine. However, once the parasitic power demand of the transmission torque converter at idle is taken into account, the diesel has a higher idle fuel consumption than the gasoline engines. The diesel has a higher peak torque and a lower peak torque speed, so the converter match is tighter. As a result, the stall torque of the converter at idle speed is much higher on the diesel engine than on the gasoline engine. The diesel engine idles at 750 RPM at a stall torque of 59 Nm, while the two gasoline engines idle at 650 RPM with a stall torque of 35 Nm.

Some MD and HD automatic transmissions now have, or will soon have, a feature such as "auto neutral at stop" that unloads the torque converter at idle speed. This would give the diesel engine an advantage on all the drive cycles with idle time, and especially on the Parcel cycle. These features will be evaluated for the second report.

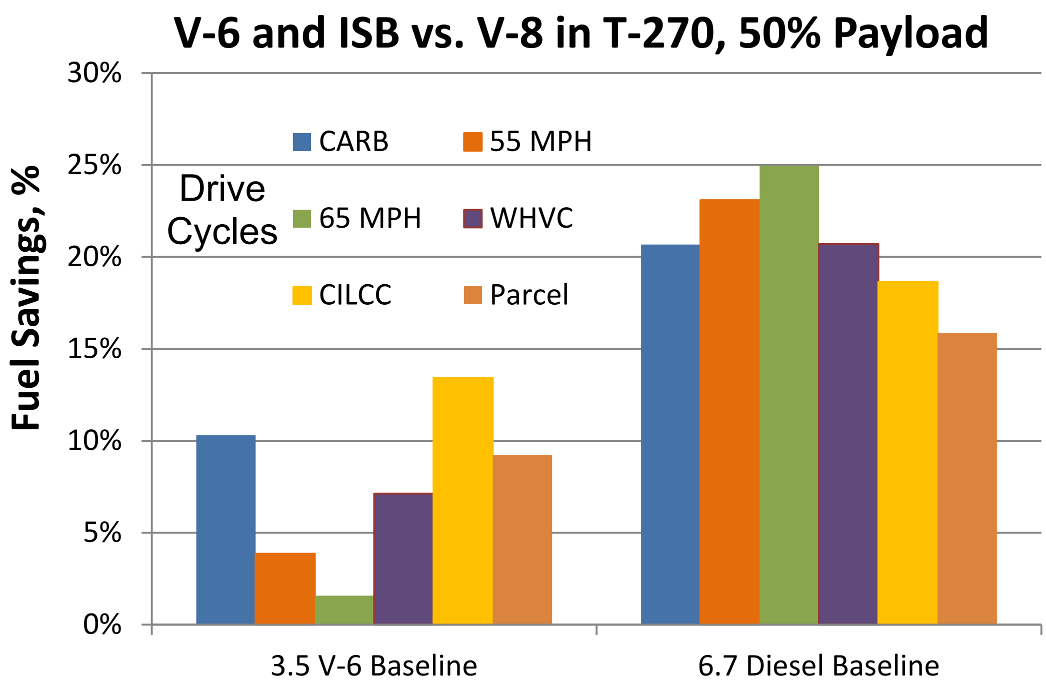


FIGURE 3.12 PERFORMANCE OF 3.5 V-6 GASOLINE AND 6.7 DIESEL BASELINE ENGINES AGAINST THE 6.2 V-8 GASOLINE ENGINE IN THE T270 TRUCK

3.3.3.2 Summary of engine technology results in T270 baseline truck

The engine technologies listed in Tables 3.17 and 3.18 above were all evaluated using the baseline T270 truck configuration. Appendix B describes the details of the ISB diesel engine model, its calibration, as well as the assumptions and parameters involved in simulating each of the considered technologies. Appendix A covers the 6.2 liter V-8 gasoline engine and the 3.5 liter V-6 gasoline engine model. Table 3.19 summarizes the fuel savings performance of all the engine technologies that have been evaluated. Each technology's performance is compared to the baseline for that engine, and all are evaluated using the baseline T270 truck. The comparison of the baseline 3.5 V-6 to the baseline diesel is highlighted in bold type, as is the comparison of the baseline 6.2 V-6 to the baseline diesel.

Note that there are some technologies available that cannot be simulated in GT-POWER. Any technology that changes combustion characteristics needs experimental data or CFD predictions to provide GT-POWER with input data. It is also difficult or impossible to handle transient operation with complex model-based controls in GT-POWER. Section 2 provides results from the literature that suggest the level of benefits that could be achieved with some of these technologies.

TABLE 3.19 FUEL SAVINGS RESULTS OF ENGINE TECHNOLOGIES, USING BASELINE T270 TRUCK

Technology	Fuel Consumption Reduction (Percent) On Drive Cycle and at Percent of Maximum Payload																	
	CARB			55 MPH			65 MPH			WHVC			CILCC			Parcel		
	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%
20. ISB Low Back Pres.	0.1	0.2	0.2	0.3	0.4	0.4	0.5	0.5	0.5	0.1	0.2	0.2	0.0	0.0	0.1	0.1	0.2	0.2
21. ISB Low FMEP	7.6	6.8	6.2	2.8	2.6	2.5	2.1	2.0	1.9	5.8	5.1	4.6	8.8	8.0	7.5	6.8	6.1	5.4
22. ISB No EGR	3.5	3.4	3.3	3.0	2.9	2.8	2.2	2.0	2.1	3.5	3.3	3.2	3.6	3.5	3.4	2.9	2.8	2.8
23. ISB + 5% Turbo Eff	1.6	1.6	1.6	1.7	1.7	1.7	1.4	1.4	1.4	1.7	1.7	1.7	1.5	1.5	1.5	1.4	1.4	1.5
32. 3.5 V-6 VVA / VVL	2.8	2.9	3.1	2.6	2.7	2.8	3.6	3.4	3.2	2.5	2.6	2.7	3.3	3.3	3.2	2.2	2.3	2.4
33. 3.5 V-6 Cyl. Deact.	1.7	1.4	1.2	0.0	0.0	0.0	0.0	0.0	0.0	0.7	0.5	0.4	2.8	2.2	1.8	0.9	0.7	0.5
34. 3.5 V-6 Lean GDI	9.1	7.8	7.8	3.0	2.5	2.1	2.5	2.3	1.2	7.2	6.2	5.4	11	9.8	9.3	7.2	6.2	5.7
35. 3.5 V-6 Stoich EGR	4.1	4.6	5.5	4.9	5.3	5.9	9	10	11	4.6	5.0	5.8	3.9	4.2	4.8	3.8	4.6	5.8
36. 3.5 V-6 EGR + Dwnspd	6.0	6.2	7.2	5.2	5.5	5.6	10	10	11	5.2	5.5	6.1	6.1	5.9	5.9	1.1	1.9	3.1
37. 3.5 V-6 Low FMEP	0.7	0.7	0.7	0.5	0.5	0.5	0.5	0.4	0.2	0.6	0.6	0.6	0.8	0.8	0.7	0.6	0.5	0.5
38. 3.5 V-6 + 5% Turbo Eff	0.1	0.1	0.1	0.0	0.0	0.2	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
3.5 V-6 to Base ISB	**9.7**	**12**	**14**	**19**	**20**	**21**	**23**	**24**	**25**	**13**	**15**	**16**	**4.0**	**6.0**	**7.9**	**5**	**7.3**	**8.7**
40. 6.2 V-8 GDI	-0.1	-0.1	-0.1	0.0	-0.3	-0.3	0.3	0.4	0.5	0.0	-0.1	-0.1	-0.1	0.0	0.0	0.0	0.1	0.1
41. 6.2 V-8 Lean GDI	11	9.8	9.7	8.0	8.2	8.6	11	12	11	9.7	9.3	9.1	10	10	11	8.5	8.2	7.8
42. 6.2 V-8 VVA / VVL	5.2	4.4	4.1	3.0	2.9	2.8	3.3	3.0	2.7	3.9	3.5	3.3	5.8	5.2	4.8	3.5	3.2	2.9
43. 6.2 V-8 Cyl. Deact.	5.7	4.2	3.5	-0.1	-0.1	-0.1	0.0	0.0	0.0	2.7	2.0	1.5	7.4	6.1	5.2	3.1	2.4	1.8
44. 6.2 V-8 EGR	4.4	4.3	4.9	5.0	5.3	5.8	9.0	10	11	4.4	4.8	5.4	3.5	3.8	4.2	3.3	3.9	4.8
45. 6.2 V-8 Low FMEP	1.3	1.2	1.2	1.0	0.9	0.9	1.1	1.0	0.9	1.2	1.1	1.1	1.4	1.3	1.3	1.0	1.0	0.9
6.2 V-8 to Base ISB	**21**	**21**	**20**	**23**	**23**	**23**	**25**	**25**	**25**	**21**	**21**	**21**	**19**	**19**	**18**	**15**	**16**	**16**

The data provided in Table 3.19 can also be presented in graphical form. Figures 3.13, 3.14, and 3.15 below show the performance of the various engine technologies on the 6.7 liter diesel, the 6.2 liter gasoline engine, and the 3.5 liter gasoline engine. In each figure, the fuel saving benefit of a given technology is listed relative to that specific engine's baseline performance.

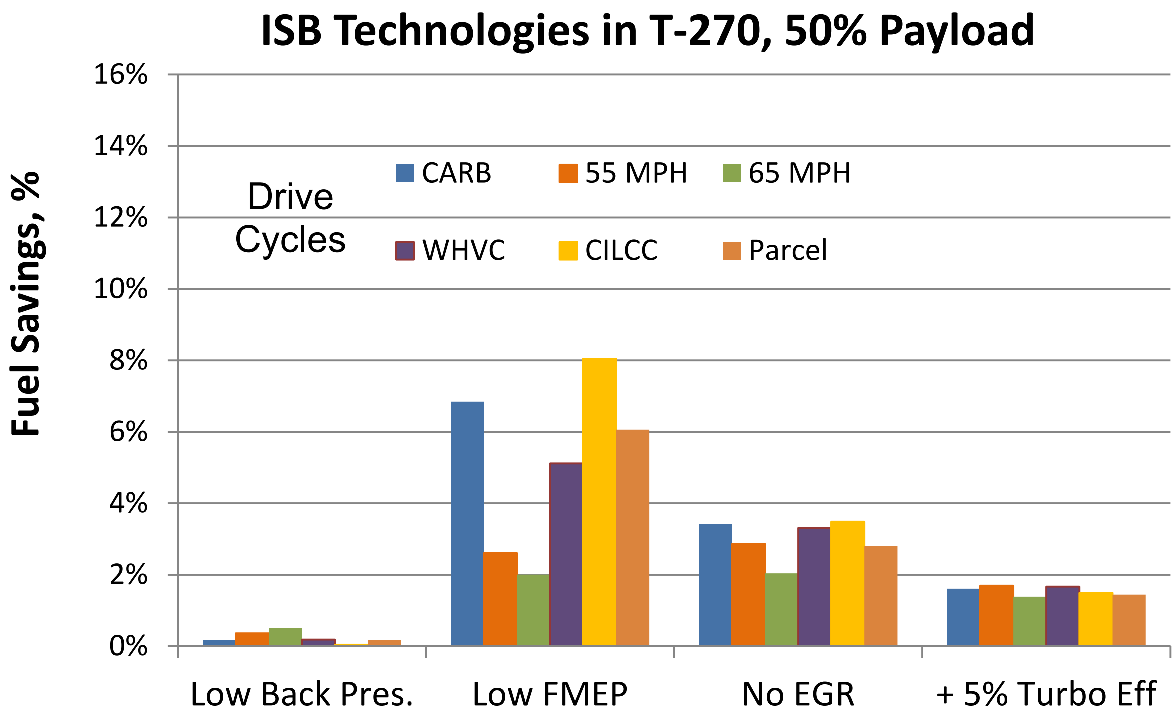


FIGURE 3.13 FUEL SAVINGS RESULTS OF 6.7 LITER DIESEL TECHNOLOGIES IN THE BASELINE T270 TRUCK

3.3.3.3 Technology #20, ISB Reduced Exhaust Restriction (Low Back Press)

Modern aftertreatment systems impose higher back pressure on the engine than the muffler-only systems used before 2007. For this technology simulation, exhaust back pressure was reduced by about 60%. Similar to the results found on the DD15 engine in the T-700 tractor-trailer, reduced exhaust restriction has only a slight impact on efficiency (see section 5.3.1.9). The simulated fuel savings were 0.1% to 0.5%, with the largest benefits coming at high speeds and high payloads.

3.3.3.4 Technology #21, ISB Reduced Engine Friction (ISB Low FMEP)

Assuming a reduction in FMEP ranging from 10% at high speed and load, to 35% at low speed and light load (see Appendix B for a complete listing of the technologies included in this friction reduction assessment), significant benefits were found. The benefits are duty cycle and payload dependent. Low speed, low average power demand cycles such as the CARB and CILCC cycles show the biggest benefit, with a range of 6% to 8% fuel savings. High average power demand cycles such as 65 MPH cruise show just a 2% benefit for low FMEP, which is less than a third of the benefit found on the CARB cycle. For all drive cycles, the largest benefit occurs with zero payload, and the smallest benefit at 100% payload. This result is to be expected, since at light load the FMEP makes up a greater portion of overall engine output.

3.3.3.5 Technology #22, ISB EGR Removed (ISB No EGR)

Driving EGR flow in an engine requires pumping work. The exhaust manifold pressure must be maintained higher than the intake manifold pressure in order for EGR to flow. Removing EGR has two benefits: pumping work can be reduced, and combustion heat release rates will increase. As a result, removing EGR provides a fuel savings of 2.7 – 3.7%, with the largest benefit coming on the most lightly loaded cycles. Note that this no-EGR technology involves very high engine-out NOx, which in turn requires a very high SCR conversion efficiency and high urea consumption, which make OBD much more challenging.

3.3.3.6 Technology #23, ISB High Efficiency Turbo (+5% Turbo Eff)

SwRI applied an assumption of a 5% increase in both compressor and turbine efficiency, under all operating conditions, resulting in an overall turbocharger efficiency increase of 10%, not 10 points (for example, from 50% to 55% overall efficiency). This technology provides a fuel savings of 1.4 – 2% across all drive cycles and payload levels. In this case, it was possible to maintain EGR flow with the higher efficiency turbo, so no impact on NOx emissions is expected. The exhaust manifold pressure remained higher than the intake manifold pressure across the operating range. It should be noted that on many engines, an increase in turbocharger efficiency will reduce or eliminate EGR flow in certain operating conditions. This issue was also experienced on the DD15 engine without turbocompound (see Section 5.3.1.6).

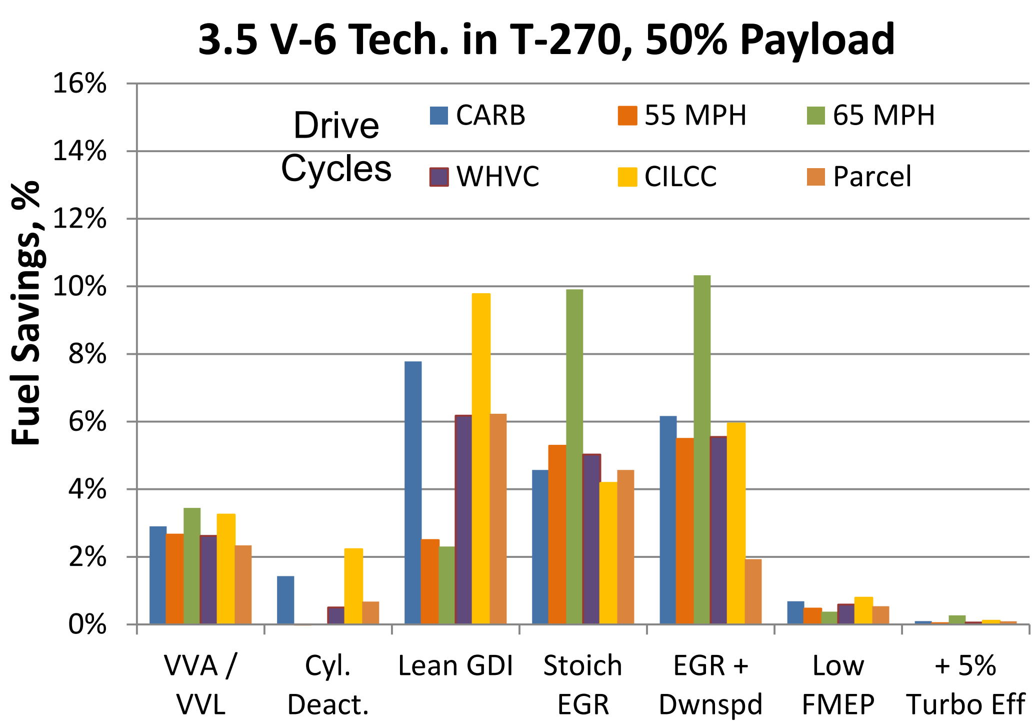


FIGURE 3.14 FUEL SAVINGS RESULTS OF 3.5 LITER V-6 GASOLINE ENGINE TECHNOLOGIES IN THE BASELINE T270 TRUCK

3.3.3.7 Technology #32, 3.5 V-6 Variable Valve Train with Cam Phaser (VVA/VVL)

The use of variable valve timing and lift is a well-known method for improving the efficiency of stoichiometric, spark-ignited engines. The application of VVA/VVL eliminates the need for a throttle, and for its associated pumping losses. In addition, engine breathing can be optimized over the wide speed range typical of spark-ignited engines (in this case, 600 to 5,500 RPM). See Section 1.2 of Appendix A for more details. In the case of this relatively small engine, however, throttling losses are small. As a result, the benefits of a VVA/VVL system are fairly modest but still significant, with a range of 2.2% to 3.6% over all the drive cycles and payloads.

3.3.3.8 Technology #33, 3.5 V-6 Cylinder Deactivation (Cyl. Deact.)

Cylinder deactivation is used to reduce pumping losses. By running the remaining cylinders at a higher BMEP, less throttling is required. This technology works best on light duty cycles and with low payloads. Given the small displacement of the 3.5 liter V-6, there isn't much opportunity to actually use cylinder deactivation in a medium-duty truck. Fuel savings benefits vary with drive cycle and payload. Cylinder deactivation does not occur on the cruise cycles at all, but benefits up to 2.8% were found on the very lightly loaded CILCC cycle.

3.3.3.9 Technology #34, 3.5 V-6 Lean Burn GDI (Lean GDI)

Lean burn GDI provides two efficiency advantages. First, it reduces pumping losses from throttling at light loads. The second advantage is that lean GDI does not require timing retard at low speed and higher loads to prevent knock. The disadvantage of lean GDI is a big one: it results in high engine-out NOx (which was not predicted in this project). Because of lean operation, a 3-way catalyst cannot be used to reduce NOx. Therefore, lean GDI requires a very effective NOx aftertreatment, such as SCR, lean NOx trap (LNT), or a combination of the two. We limited the lean GDI exhaust temperature in order to allow for aftertreatment durability and conversion efficiency. This requires going rich at loads above 14 bar BMEP, so the full load efficiency of this engine is comparable to the base engine. The largest benefits for lean burn are found on lightly loaded cycles. Up to 11% savings was recorded on the CILCC cycle, but only 1.2% was obtained at 65 MPH cruise with 100% payload.

The improvements shown here from the GT-POWER simulation results do not include the fuel penalty for regenerating an LNT, or account for the urea required by an SCR aftertreatment. These penalties can increase fuel or DEF consumption on the drive cycle and potentially cancel part or all of the benefits of lean operation. See Appendix A for more detail.

3.3.3.10 Technology #35, 3.5 V-6 Stoichiometric EGR (Stoich EGR)

Stoichiometric EGR in a spark-ignited engine has benefits similar to lean burn (which is described in the subsection above). EGR flow reduces throttling losses, eliminates the need for rich operation, and greatly reduces the need to retard spark to avoid knock. Stoichiometric EGR avoids the need for NOx aftertreatment, because a conventional 3-way catalyst can be used. Challenges that come with EGR include the need for higher ignition system energy to avoid misfire, and the difficulty of controlling EGR rates adequately in transients to avoid misfire. EGR also has the effect of slowing combustion rates. This effect is accounted for in the simulation by using combustion data from a 3.5 liter EGR engine. SwRI has been developing EGR technology for spark-ignited engines for about 10 years now under the HEDGE program. This acronym stands for High Efficiency Dilute Gasoline Engine, although the technology is also applicable to other fuels, such as natural gas.

Unlike the lean technology, EGR provides the largest benefit at high load, where enrichment is avoided entirely. At 65 MPH, fuel savings of 9% to 11% are obtained. On the lightly loaded drive cycles, the benefit is in the 4 – 6% range.

3.3.3.11 Technology #36, 3.5 V-6 EGR + Downspeed (EGR + Dwnspd)

One advantage of turbocharged engines is that (within reason) the power level becomes independent of both displacement and speed. Engines tend to have poor efficiency at high speed, so modifying the engine to allow higher BMEP (higher torque) enables the same power to be developed at lower engine speed. The primary limitation on BMEP is cylinder pressure, which increases with BMEP. The technology evaluated here involves reducing rated speed from 6,000 RPM to 4,500 RPM, and increasing BMEP to retain power and vehicle performance. The axle ratio is modified by a factor of 4500/6000 to keep vehicle performance identical.

Under light load conditions, the engine operates at higher BMEP, reducing pumping losses and thus increasing efficiency. At high load, the engine operates at lower RPM, which generally helps efficiency. On the 65 MPH cruise cycle, benefits of 10% to 11% are obtained, with the largest benefit coming at 100% payload. The remaining cycles except the Parcel cycle show benefits that are mostly in the 4% to 7% range. The Parcel cycle results are disappointing – less than the benefits of the EGR-only technology. The reason for this disappointing performance is that the downspeed engine comes with a tighter torque converter match. The tight converter causes a higher torque converter stall torque at idle, and thus a higher load on the engine when the vehicle is stationary. The Parcel cycle includes a significant portion of time spent with the vehicle stationary.

3.3.3.12 Technology #37, 3.5 V-6 Reduced Engine Friction (Low FMEP)

Gasoline engines have seen much more effort on friction reduction than diesel engines over the past 20 to 30 years. As a result, the scope for improvement in gasoline engines is less than for diesel engines, so a target of 10% reduction in FMEP was selected for this study. Reducing friction is of greatest benefit in light load operation, where friction makes up a larger portion of the power produced. Given the small displacement of the 3.5 liter V-6, the average BMEP on a drive cycle is relatively high, which limits the potential benefit of reduced friction. Fuel savings benefits vary with drive cycle and payload, ranging from 0.2% at 65 MPH and 100% payload up to 0.8% on the CILCC cycle at zero payload.

3.3.3.13 Technology #38, 3.5 V-6 High Efficiency Turbo (+5% Turbo Eff.)

SwRI applied an assumption of a 5% increase in both compressor and turbine efficiency, under all operating conditions, resulting in an overall turbocharger efficiency increase of 10%, not 10 points (for example, from 50% to 55% overall efficiency). Because the gasoline engine is throttled, the potential fuel savings benefit is relatively small compared to the diesel engine case. The high efficiency turbo provides a fuel savings of only 0.1 – 0.3% across all drive cycles and payload levels. This benefit does not match the benefit found on the two diesel engines. In the case of gasoline engines, however, the turbocharger works against an engine throttle, in order to maintain transient response. The throttle cancels most of the benefit obtained by the more efficient turbo.

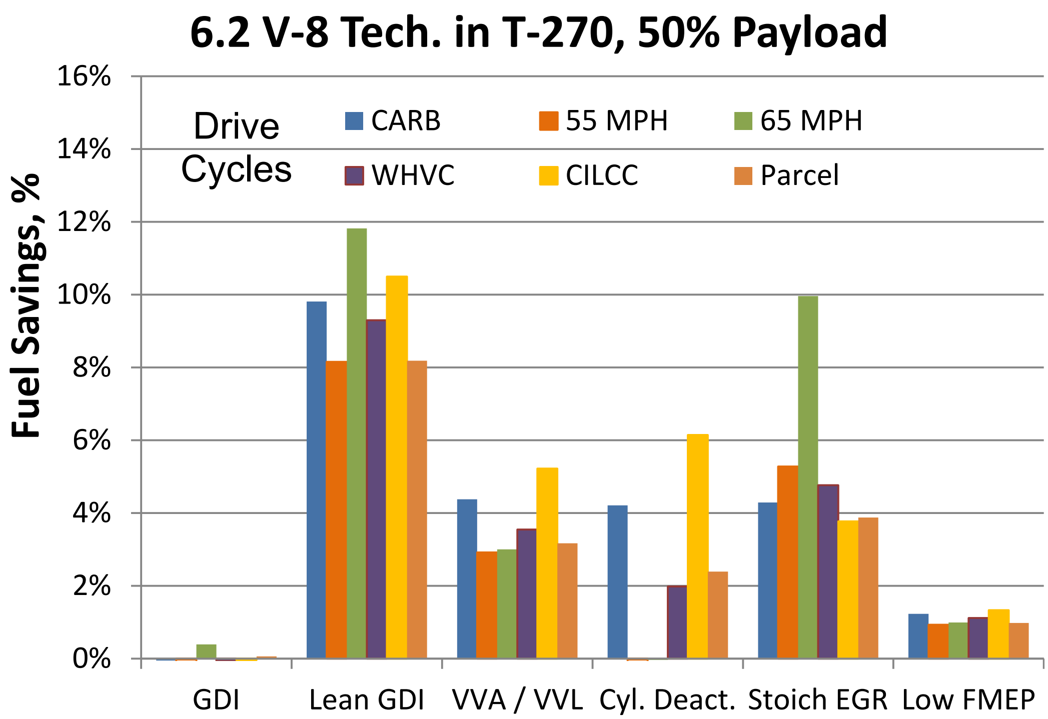


FIGURE 3.15 FUEL SAVINGS RESULTS OF 6.2 LITER V-8 GASOLINE ENGINE TECHNOLOGIES IN THE BASELINE T270 TRUCK

3.3.3.14 Technology # 40, 6.2 V-8 Convert to GDI (GDI)

Unlike the 3.5 V-6, which uses gasoline direct injection (GDI) in its baseline form, the 6.2 liter V-8 is a port injected engine in its baseline form. The addition of GDI is normally used to extract slight power and torque increases, but that was not done for this project. The original torque curve was retained to maintain constant vehicle performance. The other impact of GDI is that the cylinder charge cooling effect of direct injection reduces the tendency to knock. This typically allows an increase in compression ratio, which leads to a slight efficiency improvement. In this study, the original compression ratio was increased by 1.5 points. The benefit from the compression ratio increase is partly offset by a reduction in combustion efficiency and the parasitic power demand of the high pressure GDI fuel pump. Over most of the range of drive cycles and payloads evaluated in the T270 truck, the change in fuel consumption was negligible, and in many cases slightly negative. At 65 MPH, a benefit of 0.3 to 0.5% was found.

3.3.3.15 Technology #41, 6.2 V-8 Lean Burn GDI (Lean GDI)

Lean burn GDI provides two efficiency advantages. First, it reduces pumping losses from throttling at light loads. The second advantage is that lean GDI does not require timing retard at low speed and higher loads to prevent knock. The disadvantage of lean GDI is a big one: it requires a very effective NOx aftertreatment. We limited the exhaust temperature in order to allow for aftertreatment durability and conversion efficiency. This requires going rich at loads above 9 bar BMEP below 4,000 RPM and 8 bar BMEP above 4,000 RPM, so the full load efficiency of this engine is comparable to the base engine. Lean GDI on a large naturally aspirated engine is fairly insensitive to drive cycle and payload, with fuel savings of 7.8% to 12%. The largest benefits come on the 65 MPH cruise cycle. This cycle imposes a relatively high load on the engine, but not full load.

The improvements shown here from the GT-POWER simulation results do not include the fuel penalty for regenerating an LNT, or account for the urea required by an SCR aftertreatment. These penalties can increase fuel or DEF consumption on the drive cycle and potentially cancel part or all of the benefits of lean operation. See Appendix A for more detail.

3.3.3.16 Technology #42, 6.2 V-8 Variable Valve Train with Cam Phaser (VVA/VVL)

The use of variable valve timing and lift is a well-known method for improving the efficiency of stoichiometric, spark-ignited engines. The application of VVA/VVL eliminates the need for a throttle, and for its associated pumping losses. In addition, engine breathing can be optimized over the wide speed range typical of spark-ignited engines (in this case, 600 to 6,000 RPM). Section 1.2 of Appendix A describes the approach used to evaluate the potential of VVA/VVL. Compared to the results for the 3.5 liter engine (see Section 5.3.3.7), the 6.2 V-8 has a larger displacement and thus larger pumping losses. As a result, the benefits of a VVA/VVL system are in the 2.7% to 5.8% range, with the largest benefits coming at zero payload on a light drive cycle (the CILCC), and the smallest benefit appearing at a highly loaded condition: 65 MPH and 100% payload.

3.3.3.17 Technology #43, 6.2 V-8 Cylinder Deactivation (Cyl. Deact.)

Cylinder deactivation is used to reduce pumping losses. By running the remaining cylinders at a higher BMEP, less throttling is required. This technology works best on light duty cycles and with low payloads. The larger displacement of the 6.2 liter V-8 offers more potential for cylinder deactivation than is available for the smaller V-6. Fuel savings benefits vary with drive cycle and payload. Cylinder deactivation does not occur on the cruise cycles at all, but benefits up to 7.4% were found on the very lightly loaded CILCC cycle. On the CARB, Parcel Delivery, and WHVC cycles, fuel savings from 1.5% to 5.7% were obtained.

3.3.3.18 Technology #44, 6.2 V-8 Stoichiometric GDI EGR (EGR)

Stoichiometric EGR in a spark-ignited engine has benefits similar to lean burn (which is described in Subsection 5.3.3.15). EGR flow reduces throttling losses, eliminates the need for rich operation, and greatly reduces the need to retard spark to avoid knock. Stoichiometric EGR avoids the need for NOx aftertreatment, because a conventional 3-way catalyst can be used. Challenges that come with EGR include the need for higher ignition system energy to avoid misfire, and the difficulty of controlling EGR rates adequately in transients to avoid misfire. SwRI has been developing EGR technology for spark-ignited engines for about 10 years now under the HEDGE program. This acronym stands for High Efficiency Dilute Gasoline Engine, although the technology is also applicable to other fuels, such as natural gas.

Unlike the lean technology, EGR provides the largest benefit at high load, where enrichment is avoided entirely. At 65 MPH, fuel savings of 9% to 11% are obtained. On the lightly loaded drive cycles, the benefit is in the 3.3% – 4.9% range.

3.3.3.19 Technology #45, 6.2 V-8 Reduced Engine Friction (Low FMEP)

Gasoline engines have seen much more effort on friction reduction than diesel engines over the past 20 to 30 years. As a result, the scope for improvement in gasoline engines is less than for diesel engines, so a target of 10% reduction in FMEP was selected for this study. Reducing friction is of greatest benefit in light load operation, where friction makes up a larger portion of the power produced. For the 6.2 liter V-8, the average BMEP on a drive cycle is lower than for the 3.5 liter V-6, and so the potential benefit of reduced friction is larger. Fuel savings benefits vary with drive cycle and payload, ranging from 0.9% at 65 MPH and 100% payload up to 1.4% on the CILCC cycle at zero payload.

3.3.4 T270 Delivery Truck Vehicle Technology Results

The vehicle technologies listed in Table 3.7 were all evaluated using the baseline version of the ISB diesel engine. Appendix G describes the details of each vehicle technology, including parameters and assumptions. Performance of the baseline vehicle is described above in Figures 3.10, 3.11, and 3.12, as well as Table 3.20 and Figure 3.16.

Table 3.20 above summarizes the simulation results for all of the vehicle technologies. Results are provided for each technology on 6 drive cycles. Each drive cycle is run at 3 payloads, to provide information on how sensitive a given technology is to payload. As a result, the table provides 15 data points for each of the 8 technologies that were evaluated. The results shown are in terms of percent reduction in fuel consumption compared to the baseline production T270 box delivery truck. Figure 3.16 provides the same information in graphical form.

TABLE 3.20 RESULTS OF T270 VEHICLE TECHNOLOGY SIMULATIONS, USING THE BASELINE ISB DIESEL ENGINE

Technology	Fuel Consumption Reduction (Percent) On Drive Cycle and at Percent of Maximum Payload																	
	CARB			55 MPH			65 MPH			WHVC			CILCC			Parcel		
	0%	50%	100 %	0%	50%	100 %	0%	50%	100 %	0%	50 %	100 %	0%	50 %	100 %	0%	50%	100 %
N. A/C -40%	1.9	1.7	1.5	0.6	0.6	0.6	0.4	0.5	0.4	1.4	1.3	1.1	2.2	2.0	1.8	2.4	2.2	2.5
O. Cd - 15%	1.6	1.4	1.3	7.4	7.1	6.8	8.7	8.5	8.4	3.9	3.6	3.3	1.6	1.4	1.3	1.3	1.2	1.2
P. Crr - 30%	5.5	6.2	6.9	6.6	8.0	9.2	5.7	6.9	8.1	6.2	7.2	7.8	6.4	6.6	6.8	3.8	4.4	5.7
Q. 8-Spd Auto	1.8	2.1	2.3	0.4	0.2	0.1	0.1	0.1	0.0	2.1	2.0	2.0	7.1	7.2	7.0	2.6	2.7	3.9
R. 6-Spd AMT	4.3	3.9	3.8	3.7	3.5	3.3	3.3	3.2	3.1	5.1	4.5	4.0	11	10	9.2	11	9.5	9.7
R. 10-Spd AMT	4.5	4.4	4.4	3.7	3.5	3.3	5.4	4.1	2.8	5.2	4.8	6.4	6.9	6.4	5.9	9.6	8.9	9.2
S. Wght - 1000 lb.	2.6	2.5	2.3	1.5	1.5	1.4	1.3	1.3	1.2	2.2	2.1	2.0	2.2	1.9	1.8	2.2	2.1	3.0
T. Chassis Fr - 30%	0.7	0.8	0.9	0.9	1.0	1.1	0.9	1.0	1.1	0.8	0.9	1.0	0.7	0.7	0.8	0.6	0.6	1.4

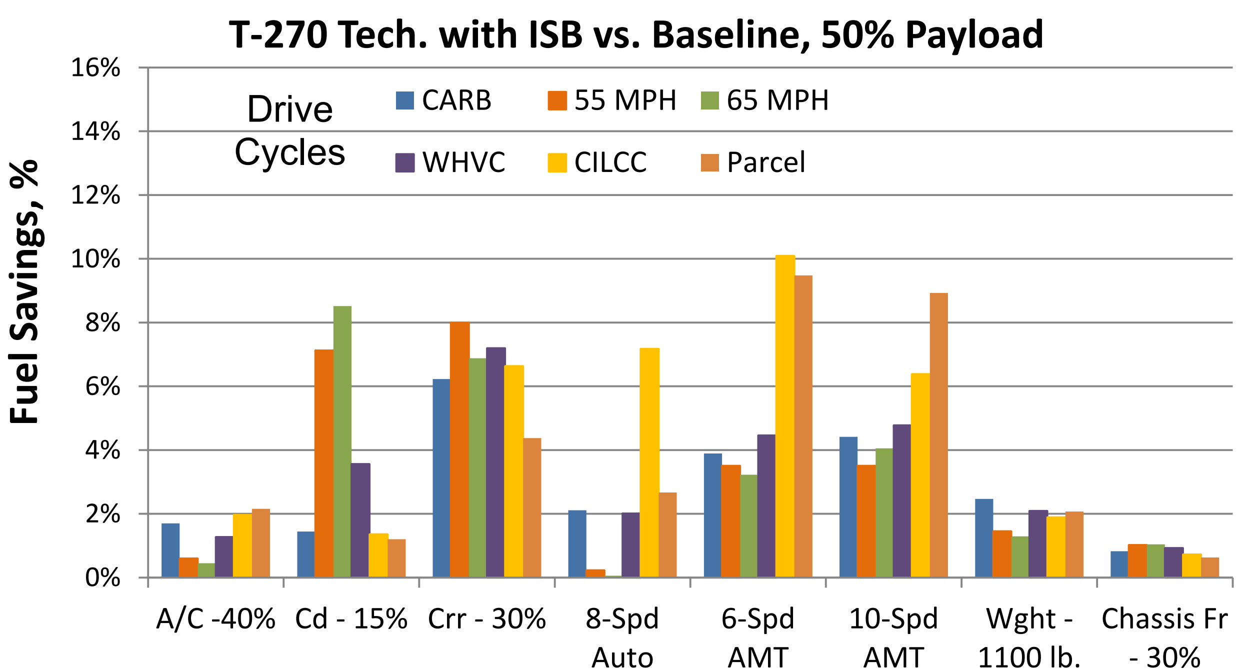


FIGURE 3.16 PERFORMANCE OF T270 TRUCK TECHNOLOGIES WITH BASELINE DIESEL ENGINE

3.3.4.1 Technology N. T270 Reduced A/C Power Demand (A/C -40%)

For all duty cycles evaluated in this project, the air conditioner is operating. Average A/C power demand is assumed to be 1.5 kW. The reduced A/C power demand technology set is assumed to reduce power demand by 40% to 900 watts. This provides a 0.4% to 2.4% benefit, with the largest benefit occurring on the CILCC and Parcel Delivery cycles at zero payload, where the average engine operating loads are the lowest. The smallest benefit is observed on the 65 MPH cruise cycle, where vehicle power demand is high.

3.3.4.2 Technology O. Improved Cd (Cd – 15%)

This technology involves a reduction in aerodynamic drag for straight truck with a large cargo box. Several features will be required to achieve a 15% reduction in Cd. Detailed assumptions about the features required to achieve a given level of Cd improvement are provided in Appendix C. The simulation results reported here do not evaluate any specific aerodynamic feature. This study evaluates the effect of reduced vehicle power demand on a range of duty cycles. Note that different features may be required for a range of vocational segments such as refuse, utility, cement mixers, dump trucks, etc., and the level of potential improvement will vary with application type. As indicated by the results in Figure 3.7, fuel efficiency benefits over a sweep of Cd values appear to be relatively linear and can be interpolated for various Cd improvements. For the 15% reduction in Cd evaluated, the results vary widely, from a 1% benefit on the Parcel Delivery cycle up to a more impressive 8.7% at 65 MPH with zero payload. The results are also dependent on payload. Light payloads reduce the power demand from tire rolling resistance and the requirement to accelerate the vehicle inertia, and this makes aerodynamic drag a larger portion of the total power demand. Aerodynamic features will provide a significant benefit on any vehicle that runs a significant portion of its mileage at high road speeds.

3.3.4.3 Technology P. Improved Crr (Crr – 30%)

This technology assumes an improvement in tire rolling resistance by 30%. As indicated by the results in Figure 3.8, fuel efficiency benefits over a sweep of Crr values appear to be relatively linear and can be interpolated for various Crr improvements. Tire rolling resistance is most important at moderate speeds in steady state operation, such as the 55 MPH cruise cycle, where a benefit of up to 9.2% fuel savings was found. At higher speeds, aerodynamic drag grows, which reduces the rolling resistance share of total vehicle power demand. In highly transient cycles such as the Parcel Delivery cycle, the power required to accelerate the vehicle inertia overshadows the rolling resistance power demand. As a result, on the Parcel cycle the fuel savings ranges from 3.9% to 6.1%. The benefit of lower Crr is payload dependent. At 100% payload, rolling resistance plays a larger share in overall vehicle power demand than at 0% payload.

3.3.4.4 Technology Q. Automatic Transmission Upgrade (8-Spd Auto)

The baseline 5-speed automatic transmission was replaced with a more efficient 8-speed automatic. The efficiency data for the 8-speed transmission was based on that of the most efficient pickup truck automatic measured by SwRI to date. The benefits are about 2% on most duty cycles. On the very lightly loaded (gradual accelerations) CILCC cycle, benefits up to 6.4% were achieved. The ~1% benefit on most cycles came from improved transmission efficiency when the vehicle is moving. The 5- and 8-speeds share a torque converter, so there is no difference at idle. On the CILCC, the engine appears to gain some benefit from the closer gear ratio spacing, but this does not show up in the other cycles. The 8-speed was geared to have the same cruise RPM as the 5-speed. If the 8-speed would be geared taller (i.e., use a lower numerical axle ratio) to take advantage of its wider ratio range, a larger benefit can be expected.

3.3.4.5 Technology R. AMT Alternatives (6-Spd AMT and 10-Spd AMT)

Automated manual transmissions offer higher mechanical efficiency than torque converter automatic transmissions. On the other hand, AMTs require a power interrupt for every shift event. Also, drivability with AMT is an issue in this application. Given two otherwise identical vehicles, the vehicle with an automatic transmission will accelerate faster and operate more smoothly than the vehicle with an AMT.

The 6-speed AMT was geared for the same cruising RPM as the automatic, so the benefits of 3.1 to 3.7% that are observed at 55 and 65 MPH are the result of higher mechanical efficiency of the AMT compared to the automatic. There are larger benefits provided by the AMT on the transient cycles, and especially on the Parcel delivery cycle and CILCC cycle. Additional evaluation of the results revealed that the primary culprit for the high fuel consumption of the automatic transmission on these cycles is due to the high input torque of the torque converter when the vehicle is stationary and the engine is at its idle speed of 750 RPM. The AMT imposes no drag torque on the engine at idle. Thus, at idle, the fuel consumption with an AMT is significantly lower than that of the automatic. This issue is particularly potent with the diesel engines and downspeed gasoline engines, which require a tighter converter match, and thus high engine torque at idle with the vehicle stationary. Technologies for unloading the torque converter at idle will be explored later in this project, and are expected to diminish, but not eliminate, the advantage enjoyed by AMT transmissions.

The 10-speed AMT was geared taller (lower engine speed in top gear) to take advantage of its wider ratio range. Thus, the 10-speed AMT provided a benefit of up to 5.4% at 65 MPH. On most cycles, the 10-speed provided similar efficiency benefits to the 6-speed AMT.

3.3.4.6 Technology S. Weight Reduction (Wght – 1100 lb.)

The baseline T270 box truck has an empty weight of 17,140 pounds. This technology simulation looked at the effect of a 1,100 pound weight reduction (6.4%). The payload weight was left unchanged, so the weight of the loaded vehicle decreased by the same amount as the empty weight. Benefits range from 1.2% on the 65 MPH cruise cycle to 2.7% on the highly transient CARB cycle at zero payload.

3.3.4.7 Technology T. Chassis and Driveline Friction (Chassis Fr – 30%)

This simulation evaluates the potential for features such as improved axle efficiency, improved wheel bearings, and synthetic lubricants used in the driveline. Benefits of 0.6% to 1.1% are achieved, with the larger benefits on the high speed cruise cycles (55 and 65 MPH).

3.3.5 Ford F-650 Truck Engine Technology Results

Three engines are included in the evaluation of The Ford F-650 for this study: the Cummins ISB 6.7 liter diesel, a 6.2 liter V-8 gasoline engine that is currently offered in Class 2b and 3 vehicles, and which may in time replace the older V-10 in heavier vehicles, and a Ford 3.5 liter turbocharged V-6 which is currently offered in Class 2a vehicles. The ISB diesel is considered the baseline engine for these vehicles. The Cummins ISB is the standard engine sold in the F-650 truck, although there is also a 6.8 liter V-10 gasoline engine option. Data for modeling the Ford F-650 truck came from the example shown in the photographs below.

The biggest difference between the F-650 and the T270 truck covered above is that the cargo box is replaced by a flatbed for towing purposes. Vehicle parameters for the F-650 can be found in Appendix C. The flatbed greatly reduces the frontal area of the truck, which will reduce aerodynamic drag. At high road speed, the F-650 can be expected to require significantly less power than the T270. For example, at 65 MPH, the baseline F-650 has a power requirement of 88 kW, while the T270 demands 129 kW, a 47% increase.

FIGURE 3.17 FORD F-650 ROLL-OFF TOW TRUCK

3.3.5.1 Baseline Engine and Vehicle Results for F-650 Truck and Engines

Figure 3.18 below shows the *fuel economy* performance of the F-650 truck with the three engines in their baseline form, all evaluated at 50% payload. Figure 3.19 shows the *fuel consumption* results for the same engines and payload. The same data is provided in tabular form in Tables 3.21 and 3.22 below the figures.

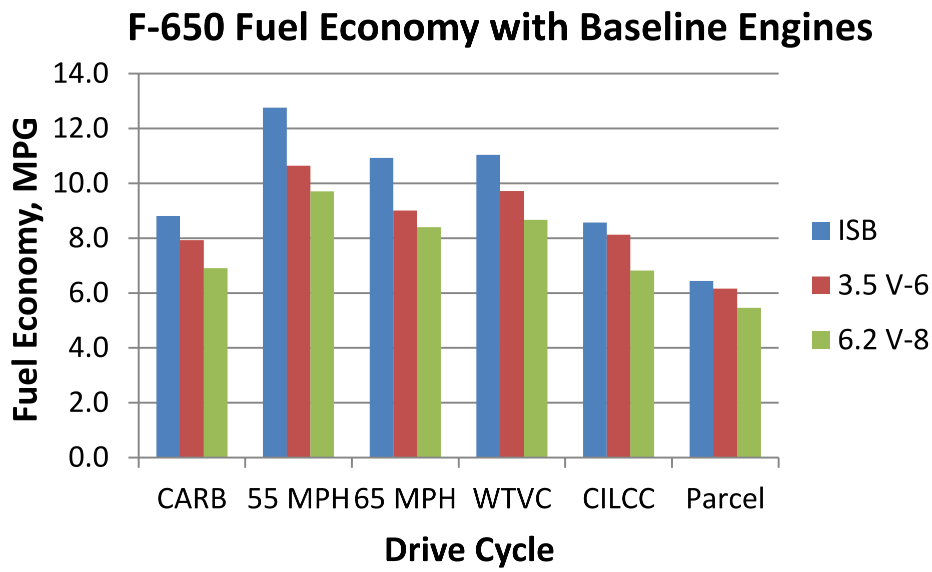


FIGURE 3.18 FUEL *ECONOMY* OF THE BASELINE F-650 AT 50% PAYLOAD, WITH THREE ENGINE OPTIONS

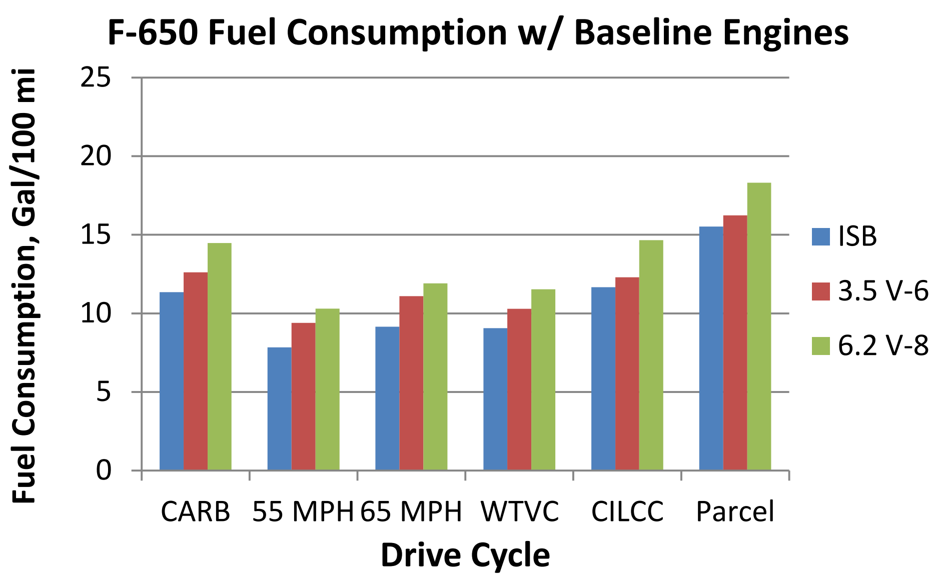


FIGURE 3.19 FUEL *CONSUMPTION* OF THE BASELINE F-650 AT 50% PAYLOAD, WITH THREE ENGINE OPTIONS

There is one oddity in the results shown in Figures 3.18 and 3.19. This is the relatively modest performance of the diesel engine on the CILCC and Parcel Delivery cycles, particularly when compared to the gasoline V-6. These cycles include a lot of accelerations, which should tend to favor the diesel. The key to the relatively high fuel consumption on the diesel on these cycles, however, is the percentage of time spent at idle in the two cycles, especially on the Parcel cycle, where the baseline truck does not include the feature "auto neutral at stop". Typically, a diesel engine will have lower fuel consumption at idle than a gasoline engine. However, once the parasitic load from the transmission torque converter at idle is taken into account, the diesel

TABLE 3.21 FUEL *ECONOMY* (MPG) OF THE BASELINE F-650 AT 50% PAYLOAD, WITH THREE ENGINE OPTIONS

Drive Cycle	ISB	3.5 V-6	6.2 V-8
CARB	8.8	7.9	6.9
55 MPH	12.8	10.6	9.7
65 MPH	10.9	9.0	8.4
WTVC	11.0	9.7	8.7
CILCC	8.6	8.1	6.8
Parcel	6.4	6.2	5.5

TABLE 3.22 FUEL *CONSUMPTION* (GAL/100 MI) OF THE BASELINE F-650 AT 50% PAYLOAD, WITH THREE ENGINE OPTIONS

Drive Cycle	ISB	3.5 V-6	6.2 V-8
CARB	11.4	12.6	14.5
55 MPH	7.8	9.4	10.3
65 MPH	9.1	11.1	11.9
WTVC	9.1	10.3	11.5
CILCC	11.7	12.3	14.7
Parcel	15.5	16.2	18.3

has more idle fuel consumption than the gasoline engines. The diesel has a higher peak torque and a lower peak torque speed, so the converter match is tighter. As a result, the stall torque of the converter at idle speed is much higher on the diesel engine than on the gasoline engine. The diesel engine idles at 750 RPM at a stall torque of 59 Nm, while the two gasoline engines idle at 650 RPM with a stall torque of 35 Nm.

Some MD and HD automatic transmissions now have, or will soon have, a feature such as "auto neutral at stop" that unloads the torque converter at idle speed. This would give the diesel engine an advantage on all the drive cycles with idle time, and especially on the Parcel cycle. These features will be evaluated for the second report.

An alternative way of comparing the three engine options in the F-650 is provided in Figure 3.20. For this figure, the 6.2 liter V-8 gasoline engine is considered to be the baseline, and the other two engines are compared to it. The 3.5 V-6 results provide a look at the benefits of downsizing and boosting, compared to a larger naturally aspirated engine. On lightly loaded duty cycles, such as the CARB cycle and the CILCC cycle, the smaller gasoline engine has a large advantage over the bigger V-8 (13 – 17%). However, on the most highly loaded cycle, 65 MPH cruise, the advantage of the smaller engine shrinks to 9%. The diesel comparison shows the benefit of converting a medium-duty truck from gasoline V-8 to diesel. On lightly loaded cycles, the diesel has about a 22% fuel consumption advantage. On the 55 and 65 MPH cruise cycles, the diesel's advantage is 22 - 25%. Remember that E10 gasoline has about 16% less energy per gallon, so the thermal efficiency of the gasoline engines is not as poor as this comparison makes it appear. Also, keep in mind that these comparisons are all for the baseline engines, without any potential improvement technology.

A comparison of Figure 3.20 with Figure 3.12, which compares the same three engines in the larger, heavier T270, is instructive. In the F-650, the 3.5 V-6 enjoys a larger advantage on every drive cycle than it has in the T270. The difference is particularly large on the 65 MPH cruise cycle, where the 3.5 V-6 has a 7.2% advantage over the V-8 in the F-650, while it has only a 2% advantage in the larger T270. The ISB diesel, on the other hand, retains nearly the same advantage over the V-8 in the F-650.

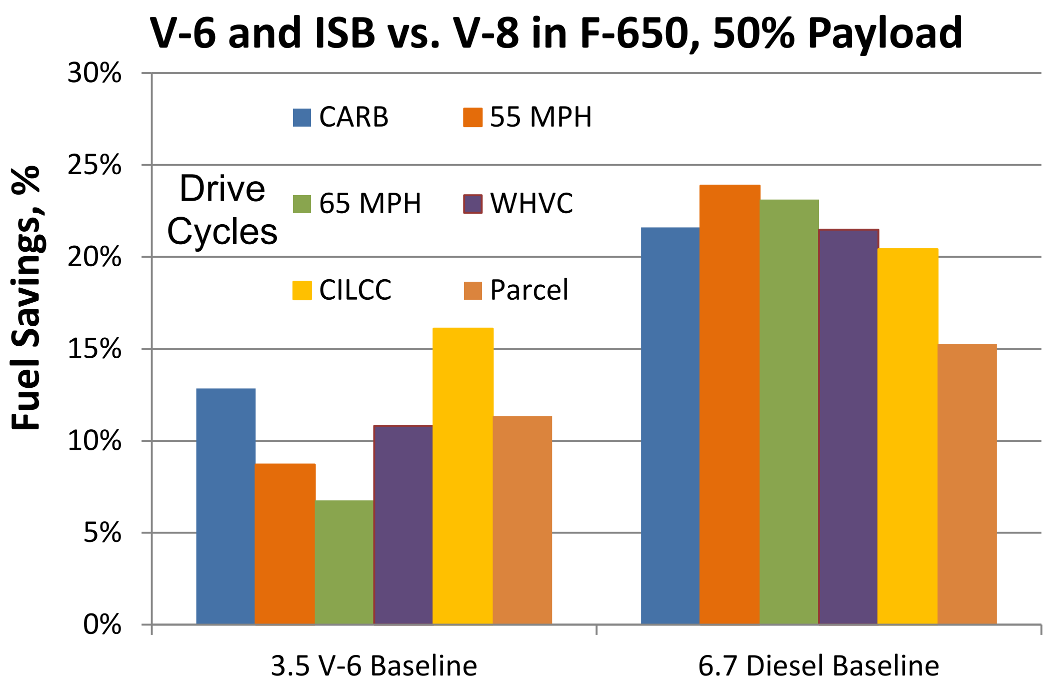


FIGURE 3.20 FUEL SAVINGS OF 3.5 V-6 GASOLINE AND 6.7 DIESEL BASELINE ENGINES AGAINST THE 6.2 V-8 GASOLINE ENGINE IN THE F-650 TRUCK

The engine technologies listed in Tables 3.2, 3.4, and 3.5 above were all evaluated using the baseline F-650 truck configuration. Appendix B describes the details of the ISB diesel engine model, its calibration, as well as the assumptions and parameters involved in simulating each of the considered technologies. Appendix A covers the 6.2 liter V-8 gasoline engine and the 3.5 liter V-6 gasoline engine model. Table 3.23 summarizes the fuel savings performance of all the engine technologies that have been evaluated. Each technology's performance is compared to the baseline for that engine, and all are evaluated using the baseline T270 truck.

TABLE 3.23 FUEL SAVINGS RESULTS OF ENGINE TECHNOLOGIES, USING BASELINE F-650 TRUCK

Technology	Fuel Consumption Reduction (Percent) on Drive Cycle @ Percent of Maximum Payload																	
	CARB			55 MPH			65 MPH			WHVC			CILCC			Parcel		
	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%	0%	50%	100%
20. ISB Low Back Pres.	0.1	0.1	0.1	0.2	0.2	0.2	0.6	0.6	0.6	0.1	0.1	0.1	0.0	0.0	0.1	0.1	0.1	0.2
21. ISB Low FMEP	8.5	7.7	7.1	4.3	4.0	3.6	3.2	3.0	2.8	7.1	6.4	5.7	10	9.4	8.8	7.3	6.8	6.2
22. ISB No EGR	3.7	3.6	3.4	3.4	3.3	3.2	4.0	3.8	3.5	3.7	3.6	3.4	3.8	3.7	3.6	2.9	2.8	2.7
23. ISB + 5% Turbo Eff	1.6	1.6	1.6	1.9	1.8	1.8	2.3	2.2	2.0	1.7	1.7	1.7	1.5	1.5	1.5	1.4	1.4	1.4
32. 3.5 V-6 VVA / VVL	2.9	2.9	2.9	2.4	2.4	2.4	3.5	3.5	3.5	2.6	2.5	2.6	3.5	3.4	3.3	2.2	2.3	2.4
33. 3.5 V-6 Cyl. Deact.	2.1	1.8	1.6	0.3	0.2	0.1	0.0	0.0	0.0	1.2	0.9	0.7	3.4	3.0	2.5	1.3	1.0	0.8
34. 3.5 V-6 Lean GDI	11	9.4	8.4	12	9.3	6.8	8.3	5.9	3.6	11	9.5	8.0	12	11	10	8.0	7.2	6.6
35. 3.5 V-6 Stoich EGR	3.8	4.0	4.3	4.1	4.3	4.4	4.2	4.3	4.3	4.0	4.3	4.5	3.6	3.8	3.9	3.4	3.8	4.3
36. 3.5 V-6 EGR + Dwnspd	8.4	7.7	7.2	8.0	7.8	7.6	9.4	9.0	8.6	7.5	7.0	6.8	8.9	8.3	7.7	1.5	1.6	2.1
37. 3.5 V-6 Low FMEP	0.8	0.8	0.7	0.9	0.9	0.8	1.0	0.9	0.8	0.8	0.8	0.7	0.9	0.9	0.8	0.6	0.6	0.6
38. 3.5 V-6 + 5% Turbo Eff	0.1	0.1	0.1	0.2	0.2	0.2	0.4	0.3	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
3.5 V-6 to Base ISB	8.7	10	11	16	17	17	17	18	18	11	12	13	3.8	5.2	6.2	3.0	4.4	6.2
40. 6.2 V-8 GDI	0.0	0.0	-0.1	0.9	0.7	0.6	1.0	0.9	0.7	0.3	0.2	0.1	0.0	0.0	0.0	0.0	0.0	0.1
41. 6.2 V-8 Lean GDI	11	10	10	13	11	10	8.7	8.6	8.4	12	11	10	11	11	11	8.6	8.5	8.3
42. 6.2 V-8 VVA / VVL	5.2	4.8	4.6	4.1	3.7	3.4	4.6	4.4	4.2	4.7	4.3	3.9	6.4	6.0	5.6	3.9	3.6	3.3
43. 6.2 V-8 Cyl. Deact.	6.3	5.4	4.6	3.4	2.4	1.4	0.6	0.2	0.0	4.7	3.7	2.9	8.8	7.8	6.9	3.9	3.3	2.7
44. 6.2 V-8 EGR	3.7	3.9	4.1	4.2	4.3	4.4	4.7	5.0	5.2	3.9	4.1	4.4	3.2	3.3	3.6	2.9	3.3	3.7
45. 6.2 V-8 Low FMEP	1.4	1.3	1.3	1.6	1.5	1.3	1.5	1.4	1.3	1.4	1.3	1.3	1.5	1.4	1.4	1.0	1.0	1.0
6.2 V-8 to Base ISB	22	22	21	24	24	24	23	23	23	22	21	21	21	20	20	16	15	15

The data provided in Table 3.23 can also be presented in graphical form. Figures 3.21, 3.22, and 3.23 below show the performance of the various engine technologies on the 6.7 liter diesel, the 6.2 liter gasoline engine, and the 3.5 liter gasoline engine. In each figure, the fuel saving benefit of a given technology is listed relative to that specific engine's baseline performance.

3.3.5.2 Engine Technology Performance in the F-650 Truck

Because these technologies have already been reviewed in the T270 truck section, this section will only describe situations where there are significant differences in the performance of engine technologies between the two trucks.

For the ISB diesel, the friction reduction technology provides somewhat better benefits on the smaller F-650. This is because when vehicle power demand is lower, friction makes up a greater percentage of the vehicle power demand.

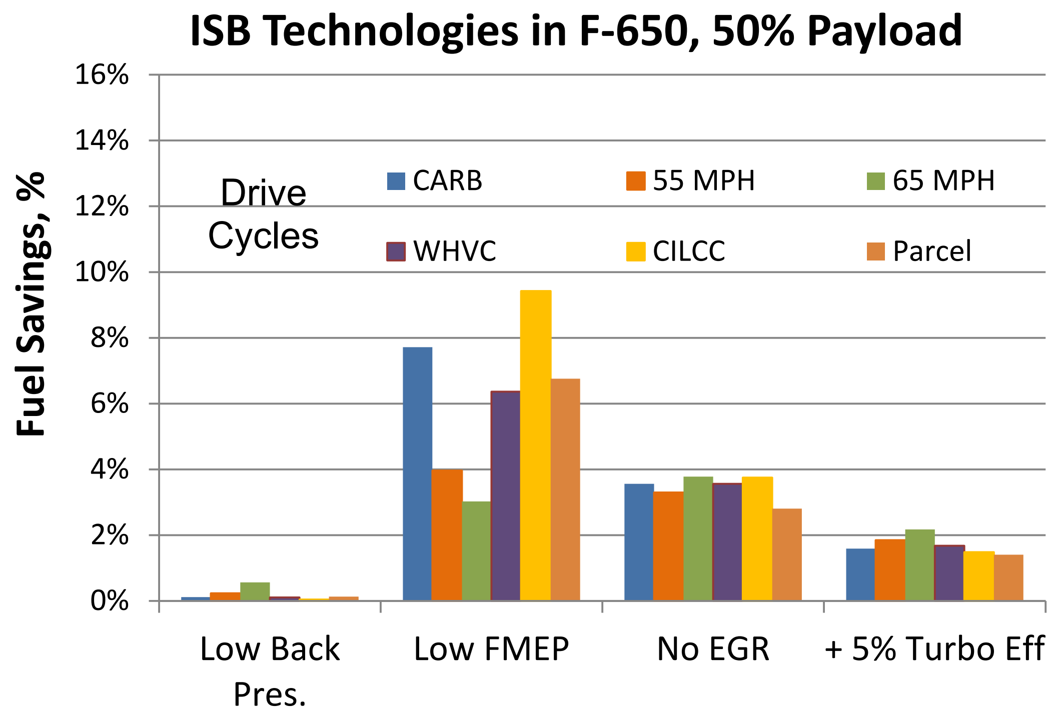


FIGURE 3.21 ISB DIESEL TECHNOLOGIES IN THE F-650 AT 50% PAYLOAD

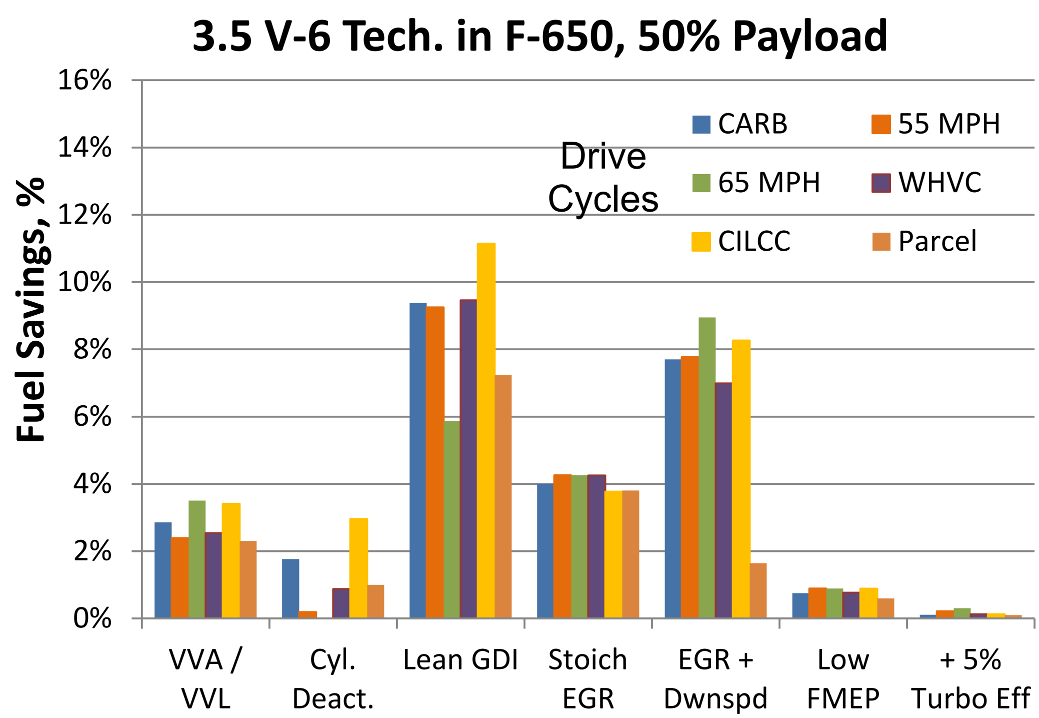


FIGURE 3.22 3.5 V-6 GASOLINE ENGINE TECHNOLOGIES IN THE F-650 AT 50% PAYLOAD

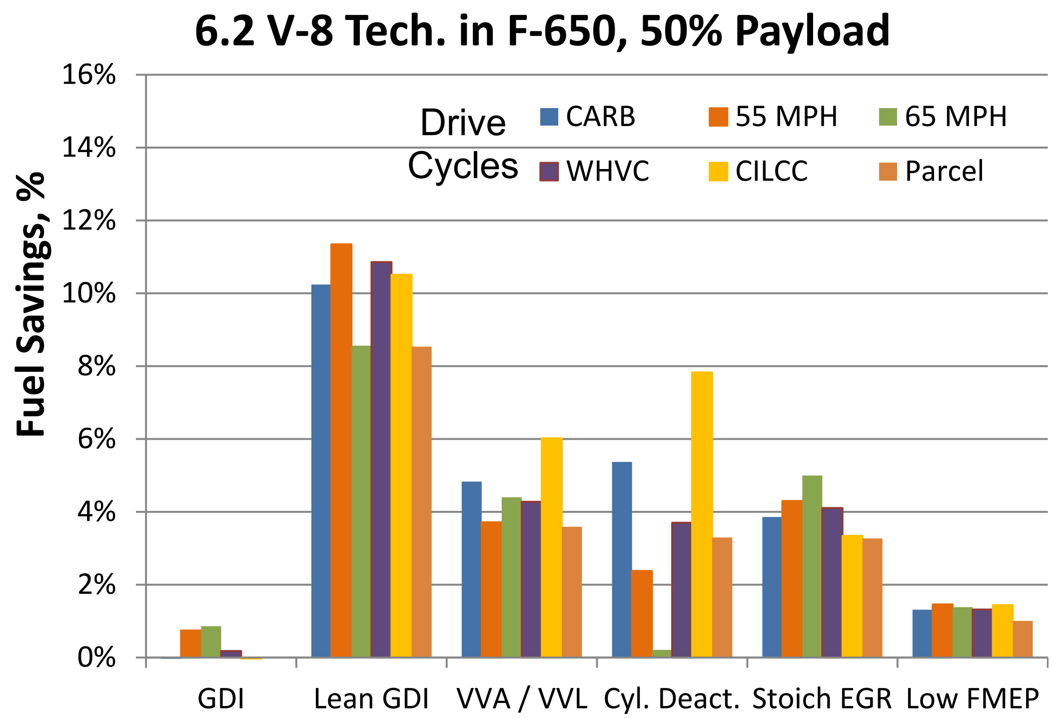


FIGURE 3.23 6.2 V-8 GASOLINE ENGINE TECHNOLOGIES IN THE F-650 AT 50% PAYLOAD

3.3.5.3 Vehicle Technology Performance in the F-650 Truck

Because these technologies have already been reviewed in the T270 truck section, this section will only describe situations where there are significant differences in the performance of engine technologies between the two trucks.

Drag reduction technology has less benefit on the F-650 than on the T270. There are two reasons for this. First, because there is less opportunity to improve the Cd of the flatbed F-650, only a 10% Cd reduction was simulated, vs. 15% for the T270 box truck. Second, since the frontal area of the F-650 is much less than that of the T270, a given change in Cd provides less benefit.

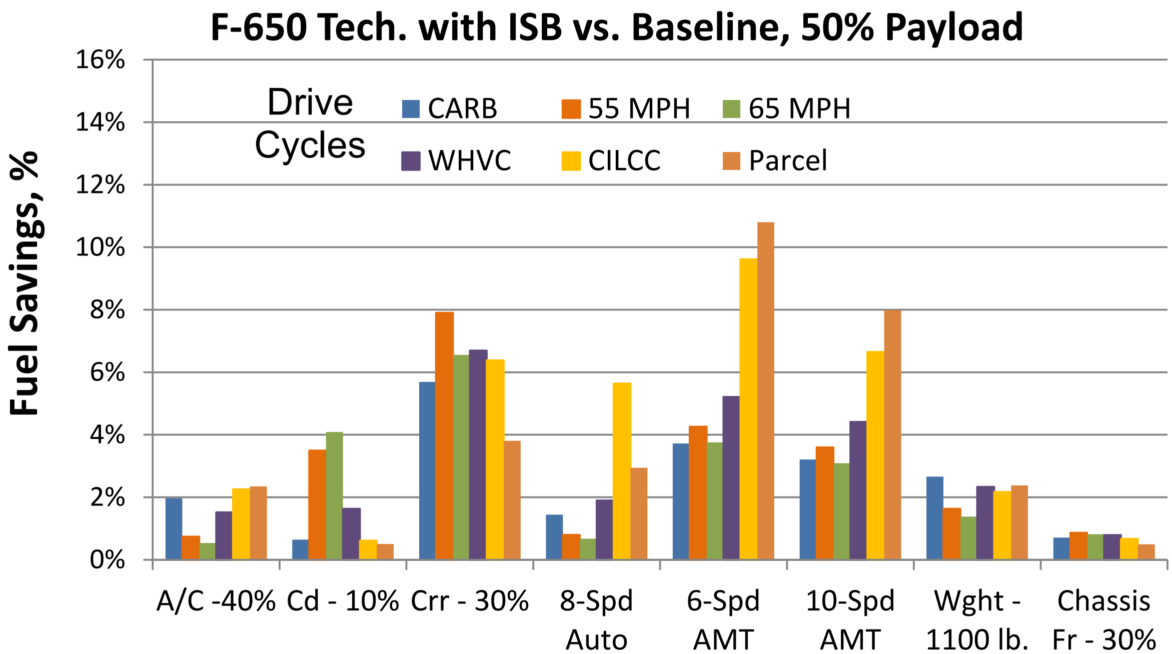


FIGURE 3.24 VEHICLE TECHNOLOGIES WITH THE DIESEL ENGINE IN THE F-650 AT 50% PAYLOAD

3.3.6 Ram Pickup Engine and Vehicle Technology Performance

Class 2b and 3 trucks and vans offer a mixture of gasoline and diesel engines. Gasoline V-8 engines of 5.7 to 6.4 liters are offered along with diesel engines of 4.5 to 6.7 liters. Diesel engines hold the largest market share, but there is a significant market for gasoline engines in this segment.

Three engines are included in the evaluation of Class 2b – 3 vehicles for this study: the Cummins ISB 6.7 liter diesel, a 6.2 liter V-8 gasoline engine that is currently offered in Class 2b and 3 vehicles, and a Ford 3.5 liter turbocharged V-6 which is currently offered in Class 2a vehicles. The ISB diesel is considered the baseline engine for the Ram. The Cummins ISB is also the standard diesel engine sold in the Ram, and it is offered in three different ratings for 2014. The lowest rating of 350 HP and 660 lb-ft is offered with a manual transmission. This is the only manual transmission identified by SwRI among the three high volume pickup and van makers. There is a 370 HP / 800 lb-ft rating offered with Chrysler-built 6 and 8-speed automatic transmissions, and a 385 HP / 850 lb-ft rating offered with an Aisin 6-speed automatic. All of these ratings are chassis certified. SwRI modeled the top diesel rating with the 6-speed automatic. The Ram also offers 5.7 and 6.4 liter V-8 gasoline engines with 6-speed automatics. For 2014, the 6.4 liter gasoline engine offers EGR, which is advertised as a fuel saving technology. Chassis cab models offered by Ram use engine dynamometer certified gasoline and diesel ratings.

For this study, the 385 HP / 850 lb-ft top rating of the ISB was modeled. See Appendix B for details of this engine, and for the differences between this version and the medium truck version of the ISB that was used in the T270 and F-650. The gasoline engine models for the pickup are described in Appendix A. They are identical to the models used in the medium duty trucks.

From the engine’s point of view, the biggest difference between the pickup truck and the medium duty trucks is the lower vehicle power demand of the pickup truck. The drag coefficient is lower, frontal area is lower, and vehicle mass is lower. The pickup has a wider range of vehicle masses across the payload options, however. The zero payload mass of the pickup is 6,876 pounds, while the mass at GCW with a loaded trailer is 25,000 pounds, almost as much as the T270 at full load.

FIGURE 3.25 RAM PICKUP TRUCK

3.3.6.1 Baseline Engine and Vehicle Results for Ram and Engines

Figure 3.26 below shows the *fuel economy* performance of the Ram truck with the three engines in their baseline form, all evaluated at 50% payload. Figure 3.27 shows the *fuel consumption* results for the same engines and payload. The same data is provided in tabular form in Tables 3.24 and 3.25 below the figures.

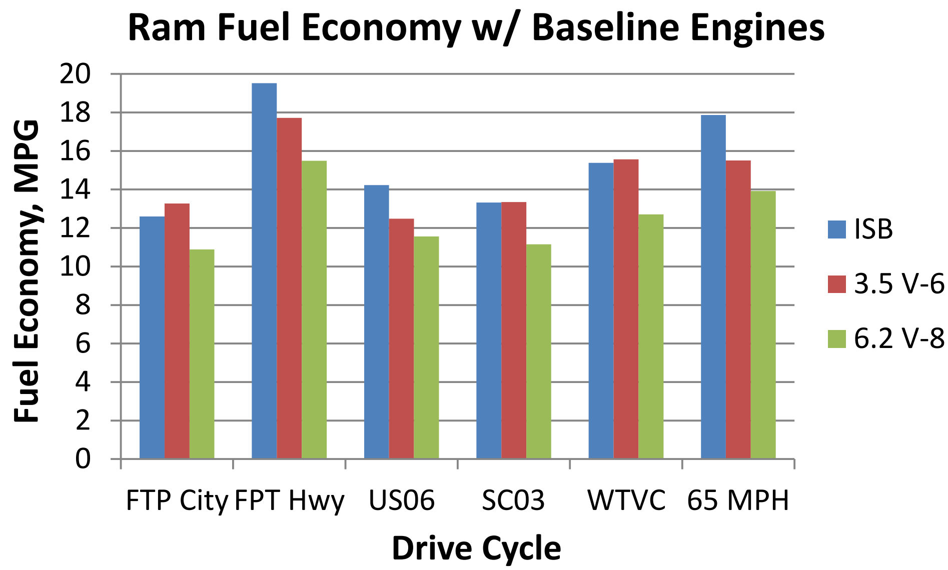


FIGURE 3.26 FUEL ECONOMY OF THE RAM WITH 3 ENGINES, ON 6 DRIVE CYCLES, AT ALVW

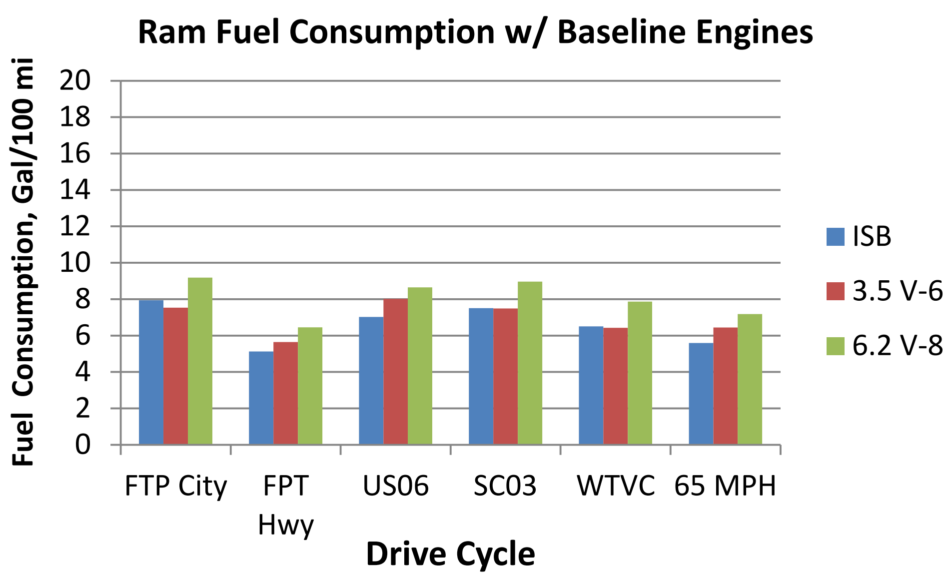


FIGURE 3.27 FUEL CONSUMPTION OF THE RAM WITH 3 ENGINES, ON 6 DRIVE CYCLES, AT ALVW

TABLE 3.24 FUEL ECONOMY OF THE RAM WITH 3 ENGINES, ON 6 DRIVE CYCLES, AT ALVW

Drive Cycle	ISB	3.5 V-6	6.2 V-8
FTP City	12.6	13.3	10.9
FTP Hwy	19.5	17.7	15.5
US06	14.2	12.5	11.6
SC03	13.3	13.4	11.2
WTVC	15.4	15.6	12.7
65 MPH	17.9	15.5	13.9

TABLE 3.25 FUEL CONSUMPTION OF THE RAM WITH 3 ENGINES, ON 6 DRIVE CYCLES, AT ALVW

Drive Cycle	ISB	3.5 V-6	6.2 V-8
FTP City	7.9	7.5	9.2
FTP Hwy	5.1	5.6	6.5
US06	7.0	8.0	8.7
SC03	7.5	7.5	9.0
WTVC	6.5	6.4	7.9
65 MPH	5.6	6.4	7.2

One surprise of the results shown above is that the 3.5 V-6 actually outperforms the diesel on some of the lower speed drive cycles. These cycles include time at idle, and as noted in the medium truck sections above, the diesel engine suffers from higher torque converter power absorption at idle. In the second phase of the project, the benefits of an idle neutral feature for the automatic transmission will be evaluated. Another factor is that the large diesel engine is not very efficient when running at very low loads.

An alternative way of comparing the three engines in the Ram pickup is provided in Figure 3.28 below. This compares the 3.5 liter V-6 and the diesel engine against a baseline of the 6.2 liter V-8. All engines are compared in their baseline form. The V-6 performs 8 to 18% better than the V-8. This is primarily due to the fact that for any given vehicle power demand, the V-6 engine is running at almost twice the BMEP as the larger displacement V-8. This means that the V-6 is often near the sweet spot of the fuel map, while the V-8 is often running at a relatively inefficient low BMEP point. It should be noted that in real life drive cycles where loads are heavy or the driver is aggressive, this advantage for the smaller V-6 will disappear. At high speeds and loads, the baseline V-6 runs in the enrichment portion of the map, which greatly reduces efficiency.

The ISB diesel performs 11 to 24% better than the baseline V-8, depending on the drive cycle. The ISB has a slightly larger displacement than the V-8, which is a small BMEP disadvantage. On the other hand, the lower speed range of the diesel makes it operate at lower RPM for a given vehicle power demand, which increases BMEP and (at lighter loads) improves efficiency. The diesel also enjoys a lower minimum BSFC than either of the gasoline engines (see Appendix A and B).

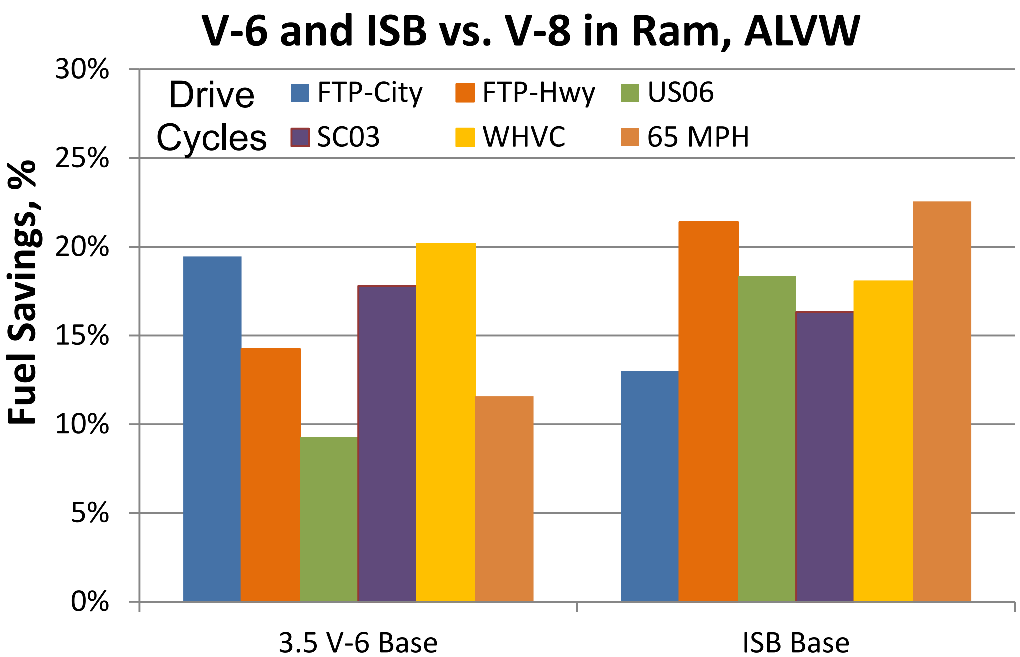


FIGURE 3.28 PERFORMANCE OF 3.5 V-6 GASOLINE AND 6.7 DIESEL BASELINE ENGINES AGAINST THE 6.2 V-8 GASOLINE ENGINE IN THE RAM TRUCK AT ALVW

3.3.6.2 Engine Technology Performance in the Ram Truck

The engine technologies listed in Tables 3.24, and 3.25 above were all evaluated using the baseline Ram truck configuration. Appendix B describes the details of the ISB diesel engine model, its calibration, as well as the assumptions and parameters involved in simulating each of the considered technologies. Appendix A covers the 6.2 liter V-8 gasoline engine and the 3.5 liter V-6 gasoline engine model. Table 3.26 summarizes the fuel savings performance of all the engine technologies that have been evaluated. Each technology's performance is compared to the baseline for that engine, and all are evaluated using the baseline Ram truck.

TABLE 3.26 FUEL SAVINGS RESULTS OF ENGINE TECHNOLOGIES IN THE RAM TRUCK. THE COLUMN OF RED NUMBERS UNDER THE US06 DRIVE CYCLE AT GCW INDICATES SIMULATIONS WHERE THE VEHICLE WAS UNABLE TO FOLLOW THE DRIVE CYCLE

Technology	Fuel Consumption Reduction (Percent) on Drive Cycle @ Payload																	
	FTP-City			FTP-Highway			US06			SC03			WHVC			65 MPH		
	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW
ISB + 5% Turbo Eff	2.5	2.4	1.6	2.5	2.4	1.5	1.7	1.6	0.9	2.2	2.1	1.3	2.4	2.4	1.7	1.8	1.8	1.2
ISB Low FMEP	8.0	7.6	4.1	7.2	6.7	3.1	4.2	3.8	1.7	7.4	6.8	3.4	8.5	8.0	4.6	4.6	4.5	2.4
ISB Low Back Pres.	0.9	0.8	0.4	0.4	0.4	0.3	0.3	0.3	0.4	0.5	0.4	0.3	0.6	0.5	0.2	0.0	0.0	0.4
No EGR	3.6	3.6	2.2	4.7	4.5	2.0	2.9	2.5	1.1	3.4	3.2	1.8	3.8	3.9	2.5	4.6	4.2	1.2
ISB 4-Cylinder	13	12	6.8	8.5	8.3	3.7	4.9	4.0	8.1	11	11	5.0	13	13	7.8	6.1	5.8	0.3
3.5 V-6 VVA / VVL	3.4	3.1	2.8	2.7	2.5	2.8	2.6	2.7	2.7	3.2	3.0	2.8	3.3	3.1	2.6	2.1	2.1	3.0
3.5 V-6 Cyl. Deact.	3.4	2.8	0.6	1.5	1.1	0.1	0.6	0.5	0.1	2.8	2.2	0.4	3.2	2.6	0.5	0.3	0.1	0.0
3.5 V-6 Lean GDI	13	12	5.5	15	13	3.9	8.7	7.3	1.5	13	11	4.6	15	14	6.1	15	13	0.8
3.5 V-6 Stoich EGR	3.5	3.7	7.9	4.0	4.2	7.7	4.8	6.0	17	3.6	3.9	11	3.5	3.6	5.2	3.4	3.6	6.4
3.5EGR + Downspd	7.2	7.0	7.0	10	9.5	7.9	7.5	8.0	20	8.1	7.5	10	7.4	7.4	5.7	8.2	8.0	5.6
3.5 V-6 Low FMEP	0.9	0.9	0.6	0.9	0.9	0.5	0.7	0.7	0.7	0.9	0.8	0.6	1.0	0.9	0.6	1.0	0.9	0.5
3.5 + 5% Turbo Eff	0.1	0.1	0.1	0.1	0.1	0.0	0.1	0.1	0.2	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
3.5 V-6 to Base ISB	-8.0	-5.3	11	8.3	9.2	20	10	12	25	-1.8	-0.2	16	-2.7	-1.2	11	12	13	22
6.2 V-8 GDI	0.0	0.0	0.2	0.3	0.3	0.0	0.3	0.3	1.3	0.0	0.0	0.4	0.1	0.1	-0.3	0.8	0.7	-0.1
6.2 V-8 Lean GDI	11	11	8.4	14	13	7.8	11	10	4.3	11	11	7.4	12	12	9.3	15	14	8.5
6.2 V-8 VVA / VVL	7.0	6.5	3.4	6.5	5.7	2.6	4.3	3.8	1.6	6.8	6.3	3.0	7.3	6.6	3.3	4.7	3.8	2.2
6.2 V-8 Cyl. Deact.	9.4	8.2	2.2	6.6	5.4	0.7	3.2	2.4	0.3	8.5	7.1	1.7	10	8.4	1.9	4.7	3.5	0.0
6.2 V-8EGR	3.0	3.2	6.4	4.2	4.4	7.0	4.6	5.6	11	3.3	3.5	7.5	3.2	3.5	4.9	4.4	4.6	6.5
6.2 V-8 Low FMEP	1.6	1.5	1.1	1.7	1.6	1.0	1.3	1.2	1.0	1.6	1.5	1.1	1.7	1.6	1.0	1.7	1.6	1.1
6.2 V-8 to Base ISB	13	14	16	21	21	22	18	19	30	16	16	20	18	17	16	23	22	21

The data provided in Table 3.26 can also be presented in graphical form. Figures 3.29, 3.30, and 3.31 below show the performance of the various engine technologies on the 6.7 liter diesel, the 6.2 liter gasoline engine, and the 3.5 liter gasoline engine. In each figure, the fuel saving benefit of a given technology is listed relative to that specific engine's baseline performance.

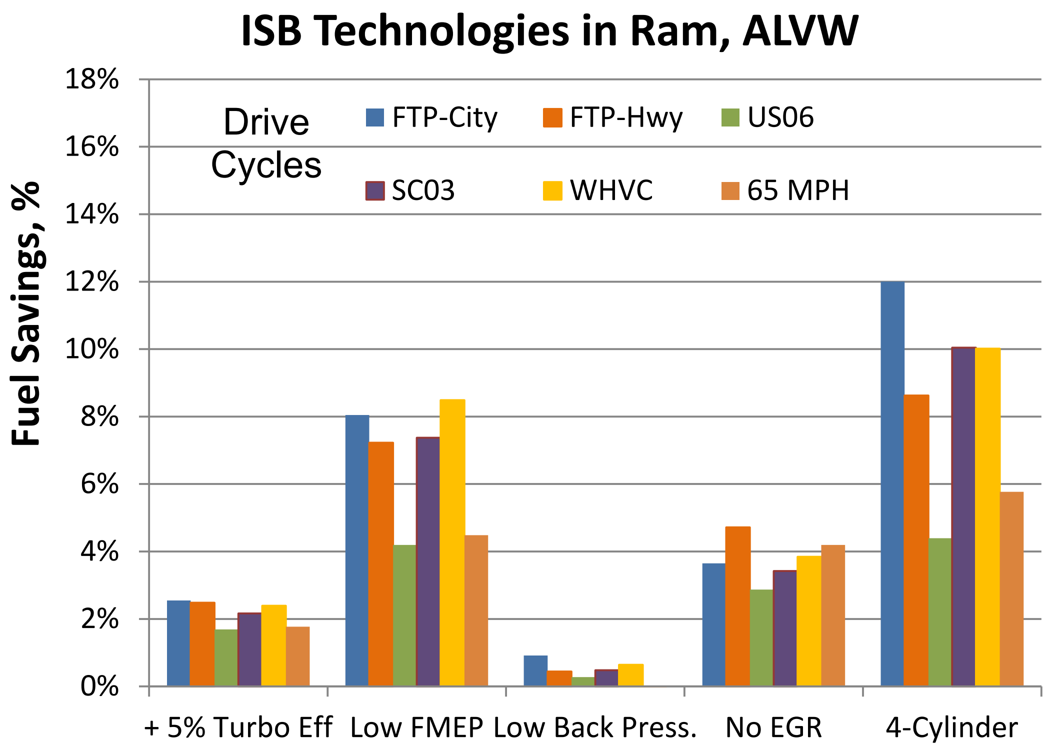


FIGURE 3.29 PERFORMANCE OF DIESEL TECHNOLOGIES IN THE RAM TRUCK

An increase in turbocharger efficiency provides about a 2% fuel consumption reduction for the diesel, at the expense of losing control of EGR rate (and thus NOx) at low RPM. The benefit declines with more aggressive drive cycles and at higher payloads. Reduced engine friction provides a 4% to 8% benefit, depending on drive cycle. The benefit of reduced friction is somewhat greater on the Ram than for the larger medium duty trucks. This makes sense: if vehicle power demand is reduced, friction becomes a larger portion of total power demand. As with the medium duty applications, low back pressure provides only marginal benefits. The benefit of removing EGR in the diesel is generally in the 3% to 4% range, which is a bit higher than was observed in the medium duty trucks. The BSFC benefit of eliminating EGR tends to be larger at light loads. The 4-cylinder diesel option provides substantial fuel savings, particularly on the more gentle drive cycles. The smaller engine operates at a higher BMEP, which makes it more efficient under light load conditions. Note that the 4-cylinder engine has only 2/3 of the power and torque of the 6-cylinder engine, so this option will not meet the needs of customers who want high power.

It is worth mentioning, however, that the 4-cylinder has more power and torque than any pickup diesel sold in the 1990s. Twenty years ago, the Cummins engine sold in Ram pickups had 160 HP and 400 lb-ft of torque, compared to the current top rating of 385 HP an 850 lb-ft. There has been quite the HP race among the pickup diesel engines. The performance of the 4-cylinder alternative shows that trading some of this increased performance for fuel consumption may be attractive to at least some customers.

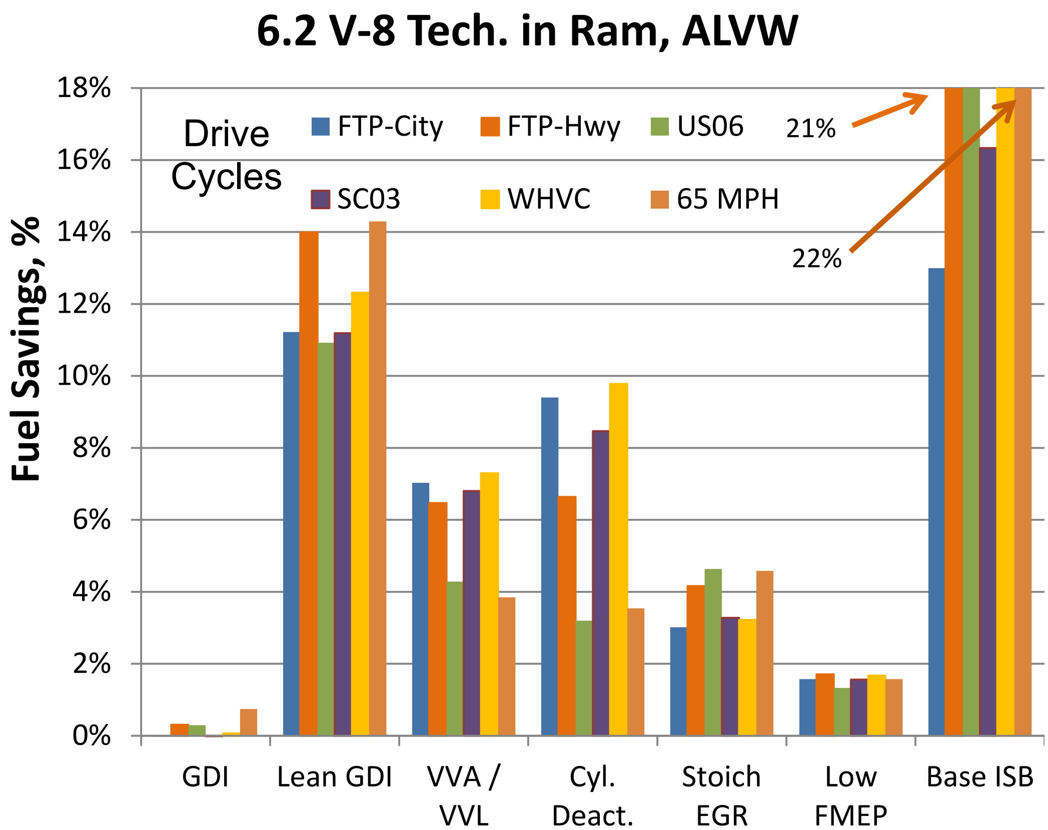


FIGURE 3.30 PERFORMANCE OF 6.2 V-8 GASOLINE TECHNOLOGIES IN THE RAM TRUCK, AND COMPARISON TO THE BASELINE DIESEL

As with the medium duty vehicles, GDI on its own provides little or no fuel consumption benefit for the V-8. The GT-POWER simulation showed that the higher efficiency provided by increased compression ratio is mostly canceled by the higher power demand of the high pressure fuel pump. (The fuel pump power demand was experimentally determined.) Lean GDI provides substantial benefits in the V-8, but there are extensive issues that would need to be resolved in order to come up with NOx aftertreatment that is reliable, durable, and which does not eliminate much of the fuel savings. The benefit of lean GDI is reduced when the vehicle runs at full GCW. At moderately high loads, a lean GDI engine runs stoichiometric, and at high speed and load, the engine runs rich, which eliminates the advantage compared to the baseline engine.

On this relatively large V-8 engine, reductions in pumping work are more important than on a smaller engine like the 3.5 V-6. As a result, VVA/VVL and cylinder deactivation provide significant benefits, especially on the more gentle drive cycles where the engine spends most of its time at light load. Lower friction also is a larger factor for the big V-8, but the benefits are still under 2%. EGR provides a 3% to 4% benefit with zero payload and at ALVW, but more when the vehicle is run at full GCW. At high loads, EGR allows the engine to avoid enrichment. The differences between the V-8 and the diesel show one of the reasons that the diesel is a popular option. Especially on highly loaded drive cycles, the diesel shines in comparison to the big V-8.

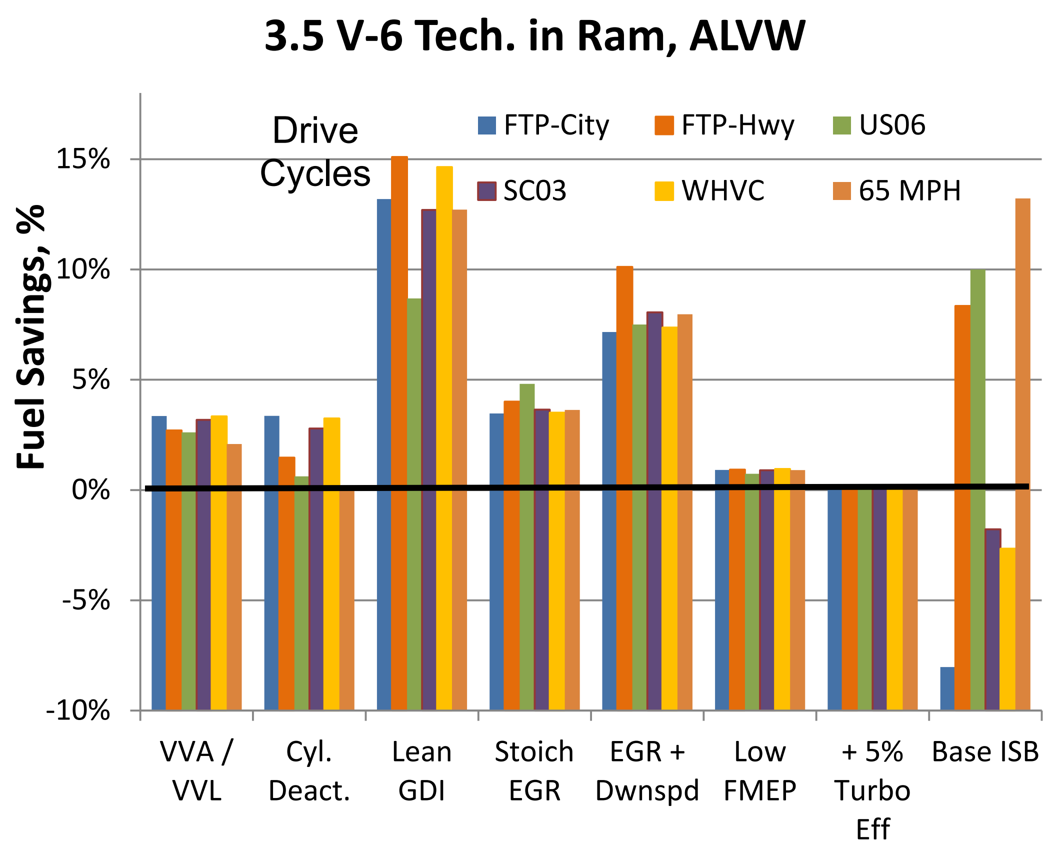


FIGURE 3.31 PERFORMANCE OF 3.5 V-6 GASOLINE TECHNOLOGIES IN THE RAM TRUCK, AND COMPARISON TO THE BASELINE DIESEL

Note that zero percent fuel savings is near the center of the plot in Figure 3.31. VVA/VVL and cylinder deactivation provide a smaller benefit on the 3.5 V-6 than on the larger V-8, because the smaller engine needs to run at higher BMEP to deliver a given power requirement. Lean GDI provides impressive benefits in the V-6, but has the same implementation issues as are described for the V-8. The benefit of lean GDI is greatly reduced when the vehicle runs at full GCW with the 3.5 V-6. At moderately high loads, a lean GDI engine runs stoichiometric, and at high speed and load, the engine runs rich, which eliminates the advantage compared to the baseline engine. EGR provides around a 4% benefit in the V-6 at zero payload and at ALVW, but substantially more at full GCW. For the downspeeding option, the engine maximum speed was reduced from 5,500 RPM to 4500 RPM, and the torque curve was increased to provide identical vehicle performance at the lower engine speed. The higher BMEP would require upgrade to the engine so it can tolerate higher cylinder pressure. In practice, this is likely to be an all-new engine. The benefits of EGR + downspeeding are substantial, however, at 7% to 10%.

Low friction contributes less than a 1% benefit on this small V-6 engine. The comparison to the diesel engine is a mixed bag. The V-6 has a fuel consumption advantage on drive cycles that involve low speed and idle operation. Two factors work in favor of the V-6 on these drive cycles: higher BMEP due to having little over half the displacement of the diesel, and the fuel sapping torque converter power demand on the diesel at idle. On the higher speed drive cycles, the diesel retains a significant advantage.

3.3.6.3 Vehicle Technology Performance in the Ram Truck

The performance of fuel saving vehicle technologies is summarized in Table 3.27 and Figure 3.32 below.

TABLE 3.27 PERFORMANCE OF VEHICLE TECHNOLOGIES ON THE RAM TRUCK USING THE BASELINE 6.7 DIESEL ENGINE. RESULTS ON THE US06 CYCLE AT GCW ARE MARKED IN RED, BECAUSE THE VEHICLE WAS UNABLE TO FOLLOW THE CYCLE

Technology	Fuel Consumption Reduction (Percent) On Drive Cycle @ Percent of Maximum Payload																	
	FTP-City			FTP-Highway			US06			SC03			WHVC			65 MPH		
	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW	0%	ALVW	GCW
A/C -40%	2.2	2.2	1.2	1.5	1.4	0.7	1.1	1.0	0.5	2.1	2.0	1.1	2.4	2.2	1.4	0.9	0.9	0.5
Cd - 10%	0.7	0.6	0.4	2.6	2.5	1.3	2.7	2.4	1.2	0.7	0.6	0.4	3.2	3.1	1.6	3.8	3.5	2.1
Crr - 30%	2.6	2.7	4.6	4.0	4.4	7.0	2.6	2.9	4.3	2.2	2.8	4.3	2.8	3.3	6.7	3.4	4.0	7.0
8-Spd Auto	4.8	4.1	3.7	3.7	3.3	1.4	0.8	1.0	1.9	2.4	2.7	2.7	3.5	3.3	1.6	1.9	1.7	0.2
Wght - 500 lb.	2.0	1.7	1.3	2.2	2.1	1.2	2.5	2.6	1.3	2.2	2.3	1.6	1.5	1.5	1.1	1.8	1.7	1.0
Chassis Fr - 30%	0.5	0.3	0.7	0.5	0.6	0.8	0.6	0.6	0.7	0.4	0.4	0.8	0.4	0.4	0.8	0.6	0.6	0.8

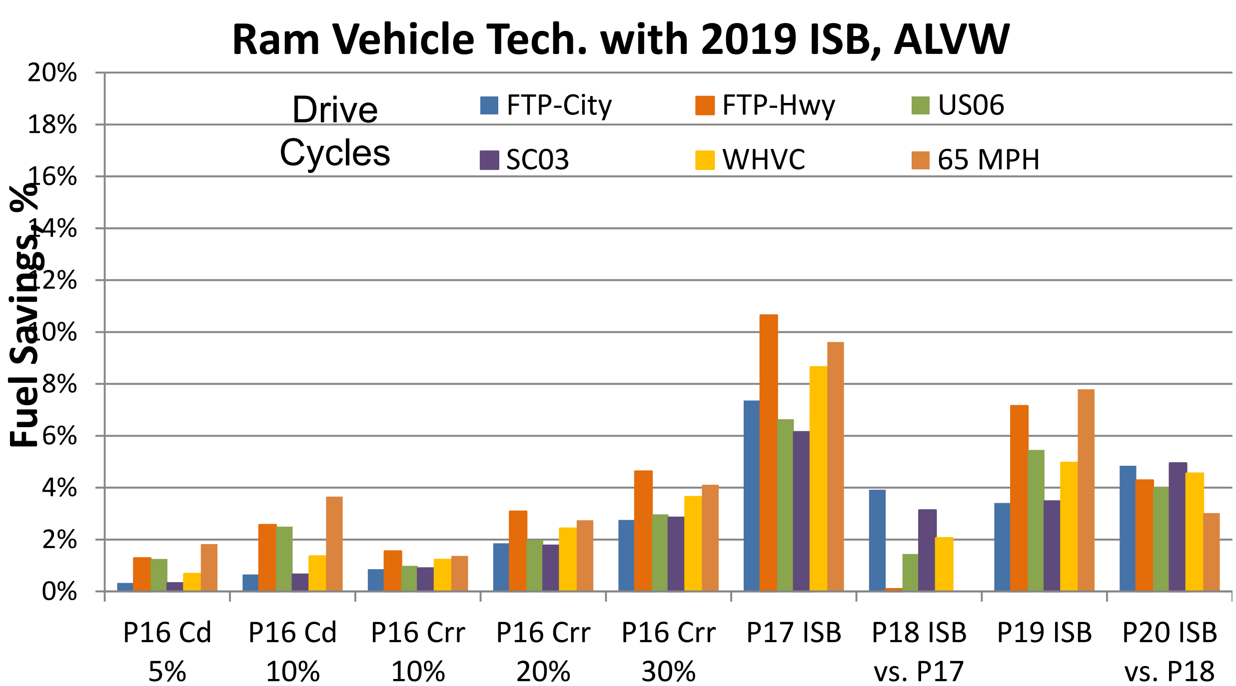


FIGURE 3.32 PERFORMANCE OF VEHICLE TECHNOLOGIES IN THE RAM TRUCK WITH THE BASELINE 6.7 LITER DIESEL ENGINE

The value of a 600 Watt reduction in air conditioning power demand is largest on lightly loaded drive cycles, and at zero payload. Aerodynamic drag reduction plays a relatively small role, for two reasons. First, an improvement of only 10% was projected. Second, the frontal area of a pickup truck is much lower than that of a larger truck, so aerodynamic drag is a smaller portion of overall power demand. A 30% reduction in rolling resistance provides a 2.2% to 4% benefit at zero payload and at ALVW, but larger improvements at GCW. Note that the trailer tire rolling resistance was also improved in this simulation.

An 8-speed automatic transmission provides a 1% to 4% improvement over the baseline transmission. Most of this improvement is due to higher mechanical efficiency of the 8-speed unit, rather than any benefit from closer matching of gear ratios to the engine fuel map. In this study, the 8-speed transmission was geared to cruise at the same engine RPM as the baseline transmission. In some cases, the OEM might gear the 8-speed taller, to take advantage of the wider ratio range from first gear to the top gear. The second report will look at this opportunity.

A 500 pound weight reduction provides a benefit of about 2% at zero payload and at ALVW. At the full GCW, the benefit of a 500 pound weight reduction is about 1%. Reduced chassis friction provides a benefit of less than 1%.

3.4 Trade-Offs Between Fuel Consumption / CO2 and Future Emissions Standards

For many years, developers of diesel engines have struggled with trade-offs between NOx emissions and two other parameters: PM emissions and fuel consumption. Historically, steps taken to lower engine-out NOx have tended to increase both PM and fuel consumption. The introduction of diesel particulate filters (DPF) in 2007 greatly reduced the trade-off with NOx and tailpipe PM. The DPF is so effective that engine-out PM can run at very high levels without exceeding the tailpipe requirement. The primary downside of high engine-out PM is the need for more frequent DPF regeneration, which costs fuel.

All of the figures in this section show data for 12 to 15 liter long-haul truck engines. This section will focus on heavy-duty engines, for which more data is available. Figure 3.33 below shows the trade-off between NOx and fuel consumption, using data acquired by SwRI over a number of years on several different projects. These engines were equipped with unit injector fuel systems, which are capable of only single-shot injection. Unit injectors also have high injection pressure at rated speed, but injection pressure falls off as engine speed goes down. These engines represent a variety of combustion chamber designs. Best point fuel consumption is shown in the figure, not the SET fuel consumption result, which will be higher. At 0.2 g/bhp-hr engine-out NOx, SwRI has determined the typical best point BSFC to be about 225 g/kW-hr. Fuel consumption drops in an exponential fashion as NOx is increased, with a value of 188 g/kW-hr at about 10 g/bhp-hr NOx (14 g/kW-hr). The total fuel consumption penalty for reducing engine-out NOx across this range is almost 20%.

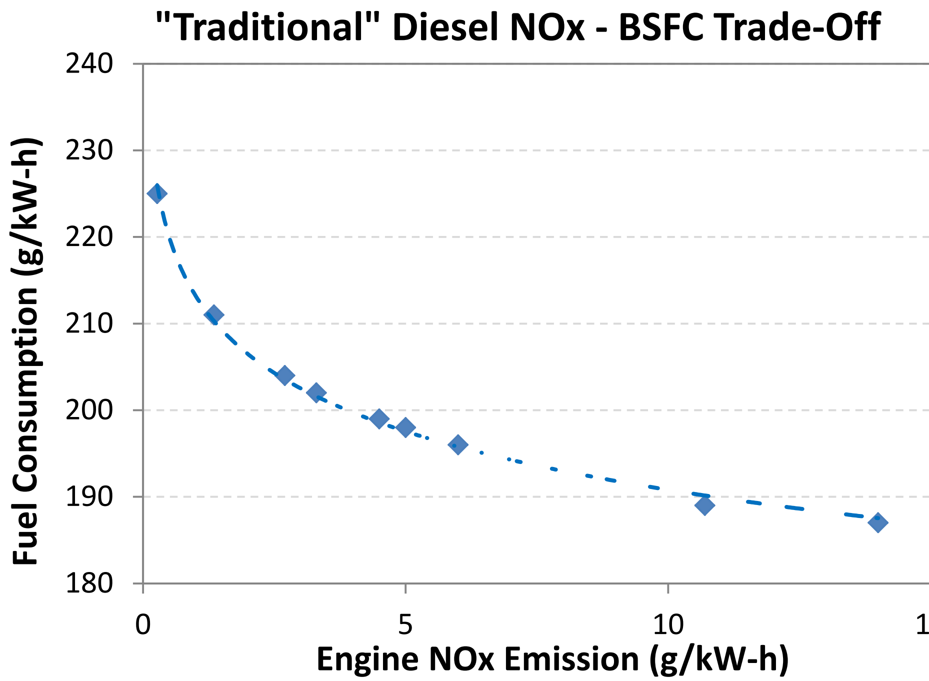


FIGURE 3.33 NOX / BEST POINT BSFC TRADE-OFF WITH TRADITIONAL ENGINE TECHNOLOGY

Over time, researchers and engine manufacturers have learned to drive the NOx / BSFC trade-off curve down a bit, and make the "elbow" of the curve much steeper. Figure 3.34 shows 13-mode SET fuel consumption data taken by SwRI on production engines complying with 2004, 2007, and 2010 emissions requirements. The thick red curve represents the trade-off for 2004 compliant engines. The thick blue curve shows the NOx / BSFC trade-off for 2007 and 2010 compliant engines. Finally, there is also a thin blue curve representing the possible future direction of development, and how that will improve the NOx / BSFC trade-off.

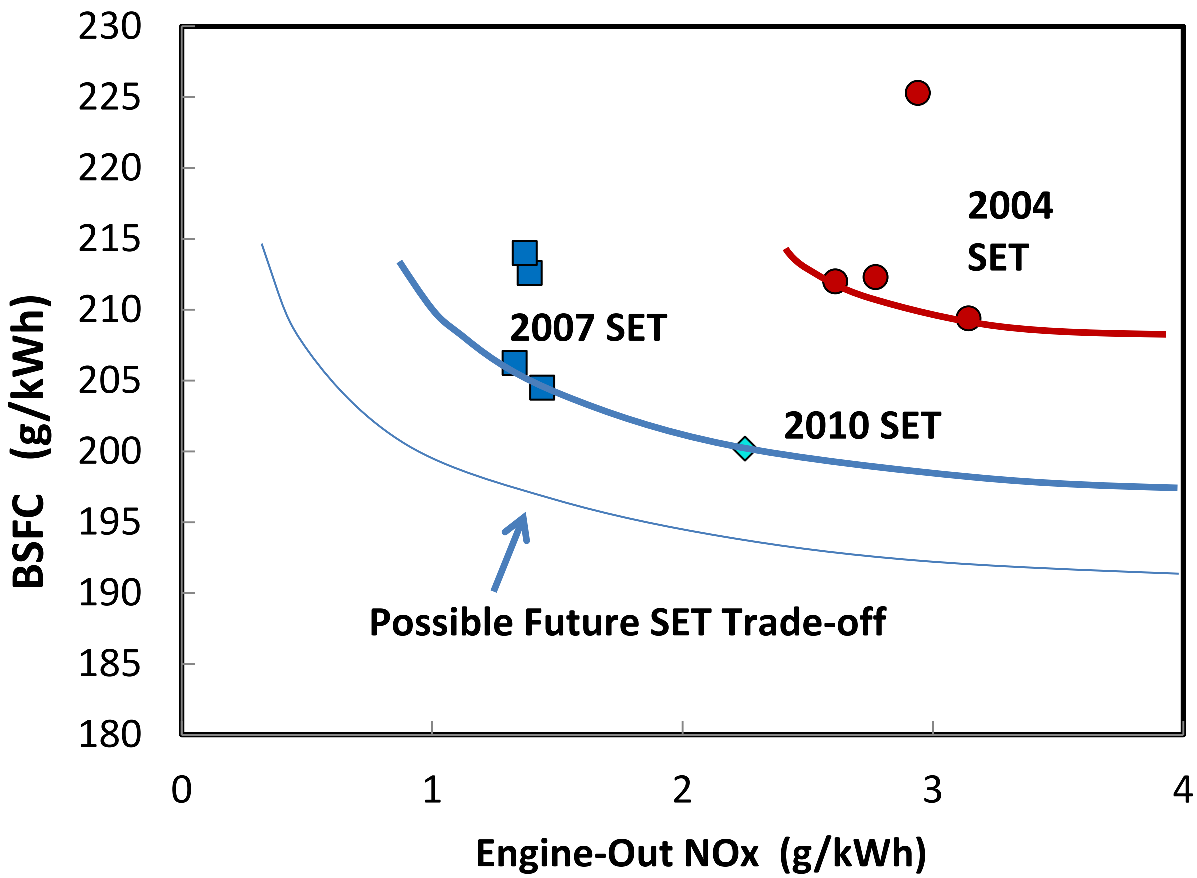


FIGURE 3.34 EVOLUTION OF NOX / BSFC TRADE-OFF OVER TIME, USING SET RESULTS

In order to enable the production implementation of new fuel saving technologies, some key limiting-issues must be addressed, including:

1.) "Best-point" fuel efficiency may not scale to full-operating range fuel efficiency. A significant part of production-development work is devoted to unavoidable tradeoffs in best local-optimum operation versus best drive-cycle operation and best regulatory cycle operation. These three targets are often mutually-exclusive to some degree. It is important that regulatory cycles be as similar as possible to real-world use, in order to limit this discrepancy. Sections 4.4.3 and 4.5.2 expand on this issue.
2.) An engine designed to provide the highest fuel efficiency may not meet other production requirements, such as NVH, durability, cost, etc.
3.) OBD requirements have not been considered and will play a key role in selection of producible, saleable engines. For example, if a very high efficiency SCR system is applied, diagnosing failures at the required level becomes very difficult.

As will be explained below, there isn't much that can be done to improve fuel consumption at very low engine-out NOx levels approaching the current tailpipe regulatory limit of 0.2 g/bhp-hr. There does appear to be scope for improving upon the trade-off curve shown in

Figure 3.34 at higher levels of engine-out NOx, such as in the 1 to 3 g/bhp-hr range. Research is under way in the engine industry to determine how much improvement is possible in the 1 to 3 gram NOx range.

The cause of high fuel consumption at low engine-out NOx levels is primarily due to the combination of pumping losses (driven by high dilution/EGR for low NOx) and extended combustion duration. High EGR rates (high dilution) lead to slower combustion and thus longer combustion duration. As combustion duration gets longer, less of the expansion stroke is available to extract energy from the combustion event. This leads to a decline in fuel efficiency.

As diesel combustion begins, the dominant combustion reaction path depends on formation of an alkyl peroxyl radical that becomes less stable as the temperature increases. These reactions start slowly, and because of reduced stability actually slow down further as the temperature increases. But with a further increase in temperature another reaction mechanism becomes predominant. In this second reaction path, hydrogen peroxide is formed, each molecule of which easily breaks into two hydroxyl radicals (OH). This chain branching reaction path rapidly increases the combustion rate, and continues to speed up with increased temperature. It is this chain branching reaction path that is critical to the rapid energy release required for efficient operation. These two reaction paths, and the change in importance of each with temperature, were reported by Professor James Keck at MIT, and his graduate student [ET-30].

Based on these reaction paths, Flynn and his colleagues at Cummins demonstrated the impact of flame temperature in the diesel engine combustion chamber [ET-31]. In practical terms, at lower in-cylinder temperatures the slower initiation reaction scheme predominates, and it is only at higher temperatures that the much faster, branching reaction path takes over. The transition in dominance between these reaction paths occurs at flame temperatures that correspond to the knee shown in the NOx/BSFC curves of Figures 3.33. As the flame temperature is reduced further, the fuel consumption penalty gets continually worse, with combustion becoming prohibitively slow as the engine approaches 0.2 grams in-cylinder NOx production.

Since 2010, most truck engines sold in the USA use SCR to reduce tailpipe NOx levels, and this approach will soon be universal. SwRI has measured the performance of several 2010 to 2013 engines and SCR systems. On the composite FTP transient cycle, NOx conversion efficiencies of 90 to 92% are common. This has allowed compliance with the current NOx standard using an engine-out NOx level of 1.5 to 2 g/bhp-hr, allowing some margin for production compliance. Under the steady-state SET test cycle, conversion efficiencies reach 98% or more. This would in theory allow very high engine-out NOx levels, but manufacturers have tended to limit engine-out NOx to around 2 – 3 g/bhp-hr in order to limit urea consumption. Urea consumption increases linearly with engine-out NOx. Even though urea consumption has a small contribution to CO_2 emissions compared to using diesel fuel, the cost of urea to operators is significant.

Figure 3.35 below shows the engine-out NOx vs. BSFC data from Figure 3.33, along with urea consumption (DEF or Diesel Exhaust Fluid) and total fluid consumption. This figure suggests a minimum total fluid consumption at around 3 to 4 grams engine-out NOx per bhp-hr.

In Figure 3.35, if the cost of diesel fuel and DEF is the same ($/gallon), then the green curve representing total fluid consumption would also represent the operating cost for fuel and DEF. In practice, DEF prices vary widely, with lower prices for bulk delivery by tanker truck, and much higher prices when purchased in 1 gallon jugs. On March 4, 2014, Flying J Truck Stops web site quoted a price of $2.79 per gallon for DEF from the pump at their stations, with diesel fuel costing $3.79 to $3.99 per gallon. Large fleets that buy DEF in bulk will pay less than a driver at a truck stop. At the other extreme, automotive web sites quote $6/gallon for DEF in 2.5 gallon jugs. If the lower per gallon price of DEF for many operators is factored in, and fluid consumption is replaced by fluid cost, then the optimum engine-out NOx level is somewhat higher – around 5 to 6 grams at early 2014 diesel fuel prices.

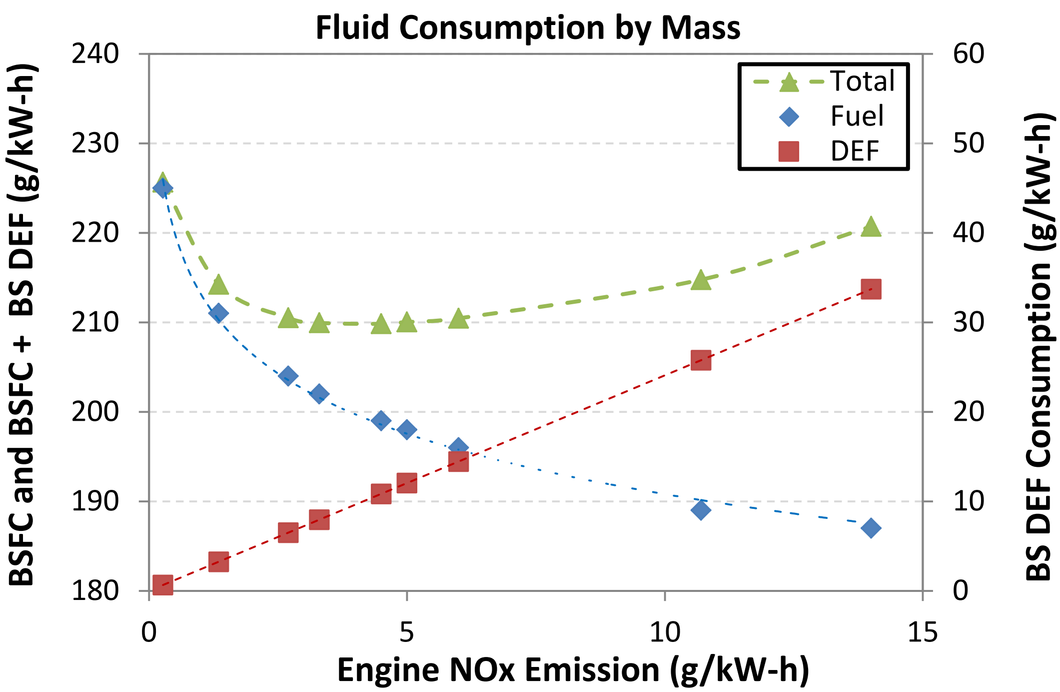


FIGURE 3.35 FUEL CONSUMPTION AND TOTAL FLUID CONSUMPTION (LEFT AXIS) AND DEF CONSUMPTION (RIGHT AXIS) AS A FUNCTION OF ENGINE-OUT NOX

Figure 3.36 below shows the same data, but assuming a future, improved NOx vs. BSFC trade-off. Because these results are speculative, the actual values have been removed. In this situation, the minimum fluid consumption occurs at an engine-out NOx level of about 1.5 to 2 g/bhp-hr. The minimum fluid cost point at current diesel fuel and DEF prices would be around 2 to 2.5 grams NOx, assuming $2.80 for DEF and $3.90 for diesel fuel.

If the tailpipe NOx standard is reduced using current SCR systems, engine-out NOx may have to be reduced from current levels in order to maintain tailpipe NOx compliance. Other alternatives include improved SCR system efficiency and/or thermal management of the SCR.

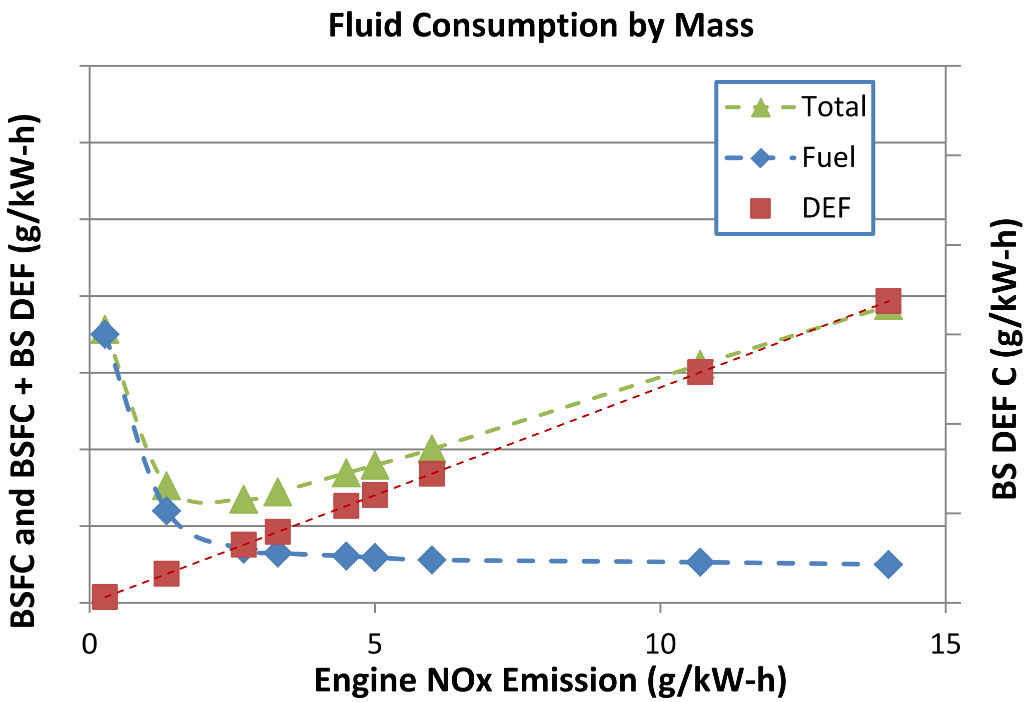


FIGURE 3.36 FUEL CONSUMPTION AND TOTAL FLUID CONSUMPTION (LEFT AXIS) AND DEF CONSUMPTION (RIGHT AXIS) AS A FUNCTION OF A POTENTIAL FUTURE NOX / BSFC TRADE-OFF

Note that thermal management usually entails a fuel consumption penalty. As the figures above show, this could cause an increase in fuel consumption and CO2 emissions. In order to bring the fuel consumption penalty of lower NOx standards down to a more acceptable level, more efficient SCR systems will be required, and OBD issues will need to be overcome.

There has been rapid progress in SCR conversion efficiency over the last 10 years, since these systems first went into production for European trucks around 2004. In fact, one of the major sources of reduced fuel consumption in 2014 model year GHG certified engines compared to their 2010 predecessors is higher engine-out NOx, enabled by more efficient SCR systems. SwRI's expectation is that for 2017 Phase 1 GHG compliance, engine manufacturers are likely to take advantage of improving SCR systems to push up engine-out NOx to some degree. This will enable a modest reduction in fuel consumption and CO_2 emissions.

It can be anticipated that further increases in SCR system efficiency will occur. However, cycle-weighted conversion efficiencies may have to increase by an order of magnitude (from 90 – 92% to over 99% on the composite FTP) in order to allow a factor of 10 decrease in the NOx standard, if there is to be no fuel consumption penalty compared to today's engines. Similarly, if the NOx standard is reduced by a factor of 4 (to 0.05 g/bhp-hr), SCR efficiencies may need to increase by a factor of 4 to avoid an increase in fuel consumption. Even if these more efficient SCR systems become available, OBD issues may prevent OEMs from taking full advantage of them.

4.0 EVALUATION OF TESTING AND SIMULATION APPROACHES

4.1 Fuel Efficiency Metrics

As part of this project, SwRI evaluated the fuel efficiency metrics used in the current GHG / fuel consumption regulations, and considered potential alternatives or additions. The units used in the current regulations are based on emissions or fuel consumption per unit work, and are summarized in Table 4.1 below:

TABLE 4.1 FUEL EFFICIENCY AND GHG METRICS IN CURRENT REGULATIONS

Regulated System	Type of Regulation	Metric Used
2b/3 Trucks	GHG	Grams of CO_2 /mile
2b/3 Trucks	Fuel Consumption	Gallons/100 mile
Vocational and Tractor Vehicles	GHG	Grams of CO_2 per ton-mile
Vocational and Tractor Vehicles	Fuel Consumption	Gallons per 1000 ton-miles
Engines	GHG	Grams of CO_2 per bhp-hr
Engines	Fuel Consumption	Gallons per 100 bhp-hr

EPA and NHTSA set standards for HD pickups and vans based on the proposed "work factor" attribute that combines a portion of the vehicle payload capacity and vehicle towing capacity, in pounds, with an additional fixed adjustment for four-wheel drive (4wd) vehicles. This adjustment accounts for the fact that 4wd, critical to enabling the many off-road heavy-duty work applications, adds roughly 500 lb to the vehicle weight. **Citation: 76 FR 57162**

For heavy-duty trucks, both combination and vocational, the agencies adopted standards expressed in terms of the key measure of freight movement, payload ton-miles or, more simply, ton-miles. Hence, for EPA the final standards are in the form of the mass of emissions from carrying a ton of cargo over a distance of one mile (g/ton-mi). Similarly, the final NHTSA standards are in terms of gallons of fuel consumed over a set distance (one thousand miles), or gal/ 1,000 ton-mile. Finally, for engines, EPA is adopting standards in the form of grams of emissions per unit of work (g/ bhp-hr), the same metric used for the heavy-duty highway engine standards for criteria pollutants today. Similarly, NHTSA is finalizing standards for heavy-duty engines in the form of gallons of fuel consumption per 100 units of work (gal/100 bhp-hr). **Citation: 76 FR 57115**.

[NRC 2010, pages 20 - 28] devoted significant effort to a discussion of metrics for fuel efficiency. This report pointed out that the most popular metric for fuel efficiency in North

America, miles per gallon, has a number of disadvantages. First, MPG is a non-linear metric. A 2 MPG improvement from a baseline of 6 MPG represents a huge benefit, while a 2 MPG improvement from a baseline of 50 MPG is almost insignificant. Another issue is that a 100% increase in MPG only represents a 50% reduction in fuel consumption, fuel cost, and GHG emissions. These factors make it easy for people to make incorrect decisions when dealing with a complex data set.

As discussed in Section 2.2, fuel consumption is inversely proportional to fuel economy. Fuel consumption can be measured in units of gallons per 100 miles traveled or, in metric units, liters per 100 km. Fuel consumption is a linear metric. If the fuel consumption of a vehicle is reduced by 2 gallons per 100 miles, the cost of driving 100 miles is reduced by a fixed amount (2 gallons times the fuel price), regardless of whether the vehicle started with a fuel consumption of 3 gallons or 30 gallons per 100 miles. Also, a 50% change in fuel consumption will translate directly into a 50% reduction in fuel cost and GHG emissions. For vehicles that perform work, such as the delivery of goods or passengers, units of work-specific fuel consumption are desirable [NRC 2010]. The units used in the current GHG regulations conform with the NRC's recommendations.

Metrics for units of work include ton-miles, but there are other types of work that vehicles can perform. A bus could be evaluated in terms of passenger-miles. Depending on cargo density, many trucks can be completely full but below the legal weight limit (i.e., cube-out). This fact suggests the possibility of a work unit in terms of gallons per cubic volume-mile, resulting in a work specific metric such as gallons consumed per million cubic foot - miles.

A passenger and luggage load can be readily converted into a mass-based metric such as tons, so a metric in terms of gallons per 1000 passenger-miles is probably redundant. However, it is more risky to completely avoid a volume-based metric. Some truck configurations lend themselves to efficiently moving low-density freight, while others favor high-density freight. For example, when moving high-density freight, it would be easy to reduce frontal area (and thus aerodynamic drag), because the cargo space can be relatively small. Low-density freight, on the other hand, is moved most efficiently in a vehicle with a high cargo volume. The use of both mass and volume-based metrics would seem particularly useful in situations where potential changes to vehicle length and weight regulations are being considered. If there is no intention to consider changes in vehicle cargo capacity (either mass or volume), then the existing mass-based metrics should be adequate. In the cases of vehicle mass reduction or increased payload with existing payload volume, the existing mass-based metric will capture the benefit.

Consider the example of a vehicle configuration being used to a limited extent in Canada today: turnpike doubles. A turnpike double consists of a standard tractor pulling two 53-foot box van trailers. This configuration has a total of nine axles, four of which support the second trailer. If the current US axle weight limits are retained, this vehicle could operate at a GCW of 148,000 pounds. However, to limit loading of bridges and for safety reasons, such a vehicle might be limited to 120,000 pounds (in Canada, the limit is 140,000 pounds, or 63,500 kg). Assume an empty weight of the turnpike double vehicle of 48,000 pounds, compared to 34,000 pounds for a standard 53-foot single trailer configuration. The change in payload possible at a 120,000 or 140,000-pound limit would be:

TABLE 4.2   COMPARISON OF PAYLOADS FOR LONG COMBINATION VEHICLES

Vehicle Configuration	Weight Limit	Empty Weight	Payload	Payload Increase, %
53 foot single	80,000	34,000	46,000	N/A
53 foot double	120,000	48,000	72,000	56.5%
53 foot double	140,000	48,000	92,000	100.0%

While the available payload increases by 56.5% for a 120,000-pound turnpike double, the volume available for cargo increases by exactly 100%. For operators carrying low-density freight who cube-out rather than gross-out, the appropriate efficiency metric would be fuel consumption per volume-mile, because these operators would double the amount of freight they can deliver with a single vehicle. Weight limited operators would see a smaller but still substantial benefit. At a 140,000-pound weight limit, there would be no difference between the increase in cargo mass and volume. Note that in the case of these large doubles, the fuel consumption of a given vehicle does increase compared to a traditional single trailer configuration. However, fuel consumption increases by much less than payload mass or payload volume increases, so there is a substantial increase in fuel efficiency [NESCCAF, 2009].

Another issue is that not all medium- and heavy-duty vehicles transport goods or passengers. Some trucks simply deliver a capability to a job site, usually in the form of some sort of equipment. These capabilities include examples such as towing, well drilling, or bringing maintenance tools and parts to a job site. A utility bucket truck is another example of a vehicle used to bring a capability to a job site, rather than to deliver freight. Despite the lack of a "cargo", the best efficiency metric for these vehicles may still be units of gallons per 1000 ton-miles. In this case, cargo mass is replaced with the mass of the equipment installed on the vehicle, including, if appropriate, a typical load of replacement parts and other removable equipment or tools that are needed to perform the job.

It is possible to quibble with the units of fuel consumption now used for engine fuel consumption certification. Gallons per 100 bhp-hr is not a unit familiar to any engine development engineer (or to any member of the public). Mass-based fuel consumption units are typically used in engine development: brake specific fuel consumption is measured in terms of grams of fuel per kW-hr in metric units, or pounds per bhp-hr for English units. One reason for using mass based units is that volume based units are very sensitive to fuel density. For example, a gallon of gasoline has about 13% less energy content than a gallon of diesel. On the other hand, gasoline and diesel fuel have similar energy content per unit mass, because of the higher density of diesel fuel. If a switch from the current volume-based units to more familiar mass-based units is considered, the regulators must keep in mind the different densities (and energy content and carbon content) of fuels such as gasoline and diesel. Note that ethanol (used in E10 gasoline) has a lower energy content both on a per unit volume and per unit mass basis, compared to gasoline and diesel fuel.

The units used for GHG certification of engines are an odd mix of metric and English units (grams per bhp-hr), but at least these units are familiar from their use in criteria pollutant regulations.

4.2 Matching Test and Simulation Procedures to Technologies

Many engine and vehicle technologies that can have an effect on vehicle fuel consumption. It is important to match the test or simulation procedure to the technology being evaluated in order to accurately account for the performance of the technology under consideration.

As a first step, the technologies can be separated into two categories:

1. Technologies that affect the efficiency of generating power
2. Technologies that affect the vehicle's demand for power

Or, the issue can be expressed using very simple terms:

1. How efficiently can I make power?
2. How much power do I need to do the job?

Generally, the first category includes engine technologies, but things like waste heat recovery and hybrid systems can fall into this category as well. The second category encompasses any feature that has an influence on vehicle power demand. This basic split fits well with the current regulatory approach of having an engine efficiency standard and a separate vehicle efficiency standard. Unfortunately, not every technology can be effectively evaluated using the same testing or simulation approach. In this section, we will consider the range of fuel saving technologies, and look at the available approaches for evaluating technology performance.

It is also worth noting that some technologies perform best at full payload, while others perform best at zero payload, and some technologies are insensitive to load. As a result, it is best to evaluate technologies over a range of payload, to ensure that real-world performance is captured.

4.2.1 Engine (Powertrain) Efficiency

The current regulatory approach for evaluating engine efficiency is an engine dynamometer test. This test provides results in terms of fuel consumption or CO_2 emissions per unit of work. Engine (or powertrain) efficiency can also be predicted using simulation, as we have done in this study. Simulation is very helpful for evaluating technology options and for specifying hardware to be used in a development project. However, a lot can happen between a simulation and a finished product, so we do not recommend using engine simulations to certify engine efficiency. A dynamometer test on an appropriate duty cycle is a more reliable way to determine efficiency.

On which technologies does the current engine test certification procedure perform well? Generally, it does a good job on the following technologies:

- Engine friction reduction, including lubricants, bearings, etc.

- Combustion systems
  - Combustion chamber shape
  - Fuel injection characteristics, including injection timing
  - Swirl, tumble
  - Ignition system characteristics, including ignition timing
  - Control of heat release rate
  - Engine-out emissions strategy
- Turbochargers
- EGR systems (or the lack of one)
- Turbocompound systems
- Downsizing (but not downspeeding, see the discussion below)
- Reduced restrictions (intake, charge air cooler, exhaust)
- Variable valve actuation
- Cylinder cutouts

One area where the current engine dynamometer test approach falls short is when we would like to accurately determine the performance of a feature that only makes a very small difference in overall fuel consumption. In some cases, these technologies would be better evaluated using an alternative approach. For example, there are a number of power consuming accessories on an engine that could be improved, but where the magnitude of improvement is small enough to result in a large measurement uncertainty if an engine level test is used. Examples of these technologies include:

- Engine water pump
- Engine oil pump
- Engine high pressure fuel pump
- Friction reduction features, where individual features typically provide less than a 1% benefit
- Vacuum pump
- Air compressor
- Power steering pump
- Air conditioner compressor
- Active control of accessory power demand, such as on/off piston cooling nozzles

For technologies like these, we recommend two different approaches. For components that are part of the base engine, and which operate over the current regulatory dynamometer test cycle (for example, the water, oil, and fuel pumps, plus friction reduction features), any improvement can simply be considered part of the overall engine recipe. In these cases, the existing regulatory cycle is sufficient to capture the benefit, even if the engine dynamometer test is not sensitive enough to capture the benefit of a specific technology change. Other technologies (such as the vacuum pump, air compressor, power steering pump, and A/C compressor) will not be part of the normal certification test, so a different approach is required. The power consumption of the accessory can be very accurately measured in a test rig ("bench test") across a range of simulated engine operating conditions. Next, the change in accessory power demand for each portion of the engine test cycle can be determined, and that data is used to determine the change in engine efficiency on its test cycle.

As an example, let us consider the idea of adding a clutch to an engine driven air compressor. The purpose of the clutch is to disengage the compressor from the drive at any time compressed air is not needed, similar to the way an air conditioner compressor clutch is used. According to data published by Cummins [Secrets of Better Fuel Economy, 2007], the compressor absorbs about 0.4 HP at cruising RPM when it is not active. It is very difficult to measure the change in fuel consumption or CO_2 from a 0.4 HP improvement with any certainty, if we use an engine dynamometer test. We are looking for approximately a 0.2% benefit, which is a smaller change than can be reliably measured using an engine dynamometer test.

However, measuring the power demand of the air compressor on a test rig can be accomplished to within 1% or so. Combined with vehicle duty cycle data, (what percent of the time is the compressor idle?) a very accurate estimate of the fuel savings from a compressor clutch can be generated. The technology provides a small benefit (0.2% in our example), but that small benefit can be very accurately determined with a combination of rig testing, duty cycle evaluation, and data analysis. The same basic approach can be applied to all the engine accessory technologies listed above. To cover the full list of technologies, a range of test setups and data evaluation approaches will be required. Since it would be very difficult for the regulator to cover all of the possibilities, we recommend that it be left to industry groups such as SAE to develop approaches for validating the performance of fuel saving technologies that fall into this realm. The regulators could develop a checklist for manufacturers. The checklist would require the manufacturer to show that their method of evaluating a technology properly takes duty cycle into account, and that their evaluation achieves an acceptable accuracy or error band size when estimating the technology's benefit.

There is one technology listed above where the effect would only be partially described by a rig test. On/off piston cooling nozzles would reduce oil pump power demand during times when piston cooling is not required. This benefit can be measured on a rig. However, turning piston cooling off has other effects that a rig test would not capture. The piston would run hotter under light loads, when the cooling is turned off. This will have two benefits: exhaust temperatures will go up slightly, which is a benefit to the turbo and to the aftertreatment, and there will be a (tiny) improvement in engine efficiency from increased piston power. These benefits may be too small to be measured directly in an engine dynamometer test, but they can be captured on the existing dynamometer test cycle as part of the overall engine efficiency result. As an alternative, since on/off piston cooling nozzles would be active on the current regulatory cycle, this technology could just be considered part of the base engine.

There is another group of technologies that are not amenable to either an engine test or a component level test. For example, the vehicle-level benefits of a downspeeding strategy may not show up, at least in full, on an engine-only certification cycle. The engine certification cycle is run at fixed speed and load points. However, with downspeeding, the engine operates at lower speed, and thus at a higher BMEP, for a given vehicle power demand. When power demand is below roughly 60%, an increase in BMEP will lead to higher efficiency. A powertrain efficiency test cycle or vehicle simulation could determine the true vehicle level performance of a downspeeding strategy.

Also, as transmissions become increasingly computer controlled, there is an opportunity to improve the efficiency of power delivery by integrating the control of the engine and transmission. If you add vehicle electrification or other forms of hybrid such as hydraulic hybrids into consideration, this is another opportunity to provide power more efficiently. For these technologies, the best approach is a powertrain efficiency test, which is also referred to as a powerpack or powertrain test that includes both the engine and transmission. The powertrain is exercised over a defined power demand and road speed cycle (including, for hybrids, defined periods of negative power demand where regenerative braking can be performed). The powertrain test cycle would include specification of the powertrain output shaft speed and torque as a function of time, to simulate a given vehicle drive cycle chosen by the regulators. This test would be run on a dynamometer, and the powertrain controller would be left to determine how best to meet the power and speed demand. Just as with the existing engine test cycles, the result would be in terms of fuel consumption and GHG emissions per unit of work performed. Selection or development of a realistic vehicle power demand cycle is the key to getting appropriate results with this approach.

Note that there are issues that need to be resolved in order to create a powertrain test cycle. Existing engine cycles do not represent any specific vehicle drive cycle. A number of assumptions need to go into a powertrain cycle, such as vehicle configuration and mass, gear ratios, etc. Choices for these parameters will dictate the output shaft speed and torque as a function of time.

There are also issues that need to be addressed to keep a powertrain test cell test as realistic as possible. These issues are shared with chassis dynamometer testing. All the aftertreatment must be present in a realistic configuration, and heat losses from the exhaust system components need to reflect real-world conditions. The engine cooling system and charge air cooling must accurately represent the conditions found in real-world driving. A fair amount of test development is required to make the engine and aftertreatment perform in a test cell environment the same way that it performs in the vehicle.

4.2.2 Vehicle Power Demand

The current Phase 1 regulatory approach for evaluating vehicle power demand is the GEM model. For tractor-trailer trucks, the inputs to the GEM model are:

- Aerodynamic drag coefficient, determined by
    - Coastdown testing,
    - Wind tunnel testing,
    - CFD analysis, or
- Constant speed testing
- Steer and drive tire rolling resistance coefficients, determined by an ISO test procedure

- Vehicle empty weight reduction from a baseline value
- Road speed governor (vehicle speed limiter, or VSL) setting (if used, and if factory set for the life of the vehicle or for a defined time or mileage)
- Extended idle reduction feature, also called automatic engine shutdown, or AES (if used, and if factory set for the life of the vehicle or for a defined time or mileage)

In discussions with OEMs, SwRI was told that some customers resist taking advantage of the vehicle speed limiter (VSL) and automatic engine shutdown technology (AES) features that are factory set to qualify for GHG / fuel consumption credits, even though most tractors are used by fleets with these features active. Many fleets use VSL and AES features, but they want the flexibility to adjust vehicle speed limits or disable AES, depending on the application. To retain resale value, original purchasers want to allow the second owner to be able to decide whether and how to use these features. In the case of VSLs, to qualify for GHG/FC credit, the maximum speed must be factory set for the useful life of the truck or for expiration after a predetermined number of miles. If the maximum vehicle speed can be adjusted by the owner, rather than being preset by the factory, then the VSL does not qualify for credit under the regulation. The VSL's value is prorated based on the speed setting, with no credit given for a setting at 65 or above. Temporary increases in maximum speed (soft top) are allowed for safety purposes related to maneuvering and passing on-road. To get credit for AES, a feature that automatically shuts off the main engine after 300 seconds or less must be installed for the useful life of the vehicle or for expiration after a predetermined number of miles. If the AES feature can be turned on and off by the owner, rather than being preset by the factory, then the AES feature does not qualify for credit.

The simplified approach to estimating vehicle power demand that is implemented in GEM in Phase 1 means some vehicle technologies are not directly captured in the GEM model (though manufacturers may request innovative technology credits for technologies not in common use with heavy-duty vehicles before model year 2010 that are not reflected in the GEM simulation tool). Examples of potential fuel saving technologies not reflected by GEM in Phase 1 include:

- Matching of axle ratio to the powertrain and vehicle requirements
- Power transmission efficiency of the transmission, driveline, and axles
- Number of transmission ratios, control of shift events, efficiency features
- Power demand of vehicle accessories, such as
    - Air compressors (which can be clutched to reduce power demand)
    - Hydraulic pumps
    - Air conditioner systems
    - Electrical systems
    - Cooling fans
    - APUs and other anti-idle "hotel load" systems
- "Driver management" features, such as
    - Gear-down protection (not allowing a vehicle speed above X unless the transmission is in top gear)
    - Progressive shift algorithms (reducing engine governed speed at lower road speed)

- Load based speed control (estimating vehicle mass and using it to set the engine speed governor)
- "Smart" cruise control (operating the engine at its best BSFC load as much as possible, allowing some variation from the set speed)
- GPS based cruise control
- Driver reward systems (tracking acceleration, deceleration, cruise speed, and shift patterns, estimating the driver's achieved efficiency compared to the best possible efficiency, and determining a driver bonus)
- Downspeeding (setting the vehicle up so that it cruises at a given road speed using a lower engine speed)
- Shift optimization (using grade and load sensors)
- Electronic E-coast feature, which disengages the transmission

Many of the technologies listed above can use the same approach described in Section 4.2.1 for engine accessories. The power demand and duty cycle of vehicle accessories can be measured and used to calculate fuel savings on a given regulatory drive cycle. The power transmission efficiency of transmissions, axles and driveline components can be measured on dynamometers and used with duty cycle data to determine fuel savings on a regulatory drive cycle. For example, it is very hard to accurately determine the contribution of an alternative axle lube from vehicle testing, because the potential benefit is within the range of test-to-test variability. However, the benefit of an alternative axle lube can be measured with great accuracy in an axle dynamometer test. It should be noted that there remain some technologies on the list above which do not fit easily into a component level test scheme. This is particularly true of the driver management features.

The match of engine rating, transmission type, and axle ratio can be optimized for any given payload and drive cycle. Unfortunately, if the vehicle duty cycle changes, the vehicle is no longer optimized for its new duty cycle. For example, if a tractor is purchased for hauling potato chips on flat terrain, the buyer could specify it in a way that minimizes fuel consumption for this type of application. However, if that tractor is later used to haul heavy loads in the mountains, the performance will be very poor. For this reason, users at the extremes (especially at the light end) of the duty cycle range tend not to fully optimize for their drive cycle, in order to preserve some flexibility for future applications. It is also not clear what regulatory approach could be used to improve the market's performance regarding vehicle powertrain and driveline specifications. OEMs have very sophisticated tools that are routinely used to optimize specifications for customer applications. These tools provide guidance with high confidence even at very small fuel consumption differences, based on specific customer payloads and driving routes. One market failure here is that smaller operators who buy only one or a few trucks at a time do not often get the benefit of an OEM analysis of their application.

Driver management features also represent a difficult area for regulators to deal with. Consider the example of gear-down protection. For example, assume that the truck has a vehicle speed limiter set to 65 MPH, and a gear-down setting of 60 MPH. This means that the truck will not be able to exceed 60 MPH unless it is in top gear (or going downhill). It is easy to quantify the fuel economy benefit of running at 65 MPH in top gear vs. one gear down. The hard part is to determine with any degree of confidence how often this would happen in the field, and therefore the actual fuel saving performance of the feature. What percentage of the time would the average driver run over 60 MPH without being in top gear, if the gear-down feature is not in use? A lot of real-world data logging with and without the feature would be required to get a reliable estimate of the benefit.

Progressive shift is another example of this difficulty. Twenty years ago, almost every driver did upshifts only as the engine approached the high-speed governor. Given that driving style, the use of progressive shift, which lowers governed speed under many conditions, will make a big improvement for applications with a lot of transient operation. Today, however, most drivers no longer automatically rev the engine to the governor, because driver training has a strong focus on keeping the revs down to improve fuel economy. As a result, a lot of data logging would be required to determine the actual fuel saving benefit of progressive shift. A lot more data would be required to determine how often the benefits of progressive shift come into play across the vehicle population. Other features on the list above have different issues regarding quantification of benefit. For example, the performance of smart and GPS-based cruise control is extremely sensitive to the specific route being driven and to traffic conditions. On the current GEM cycles (and on some real-world routes), the value of these features is not captured. However, there are other real-world routes where smart or GPS-based cruise control will provide significant benefits. Characterizing the driving conditions of the vehicle fleet, and then determining the benefits of a specific feature, is a large challenge.

4.2.3 Alternative Approaches

The discussion in Sections 4.2.1 and 4.2.2, and the current regulatory framework, are both focused on a certification test approach. This means that the engine or vehicle is put into a specific class, and its performance is measured against a duty cycle that is intended to represent the typical use for that class of vehicle. If the actual use of the vehicle in the field diverges significantly from the certification cycle, there will be situations where the real-world benefit or penalty for a given technology is significantly different from what is measured on the regulatory cycle. This could limit regulatory incentives for certain technologies that might have a useful benefit in certain actual duty cycles, but not on the regulatory cycles. Driver management features are particularly likely to fall into this situation.

Other performance measurement approaches may be better suited for capturing some benefits. For example, the performance of a technology could be measured while the vehicle is operating in normal use, rather than on a regulatory cycle. For example, consider an OEM who would like to get credit for a GPS-based cruise control and automated transmission control. To determine the performance of the feature, the ECM could contain a model of the unimproved vehicle system (i.e., with the standard cruise control and AMT control). The ECM can then track the commanded fuel consumption for the baseline case, and compare it to the actual commanded fuel consumption with the GPS-based technology. The difference in performance between the (simulated) baseline technology fuel consumption and the actual in-use fuel consumption would form the basis for the fuel consumption and GHG credit awarded to this technology. However, in order to implement this approach, both the fuel consumption prediction model for the baseline case, and the "measured" fuel consumption for the technology case would need validation. Manufacturers could monitor and quantify the in-use benefits of difficult to measure technologies using the method described here, and then request innovative technology credit for new vehicles that incorporate these features. The OEM would need to show the regulator that both models accurately reflect the real-world fuel consumption of the vehicle.

4.3 Assessment of Current Regulatory Testing and Simulation Approaches

4.3.1 China's Regulatory Approach

The Chinese government has developed fuel consumption regulations for medium and heavy-duty trucks. The regulations are based on fuel consumption of fully loaded vehicles, measured in liters per 100 km. [SAE Paper 2011-01-2292] describes the methods used. The test and simulation procedures used in the Chinese regulations are based on the World Transient Vehicle Cycle (WTVC). This cycle consists of three segments: an urban low-speed segment, a rural secondary road medium-speed segment, and a motorway high-speed segment that is based on the EU truck speed limit of 80 km/h (50 MPH) and the EU maximum vehicle speed limiter setting of 90 km/h (56 MPH). There are no grades in the WTVC. Because many Chinese trucks operate with a relatively low power/weight ratio, the Chinese regulators have modified the WTVC to limit acceleration and deceleration rates. The resulting modified cycle that is used in the Chinese regulation is called the C-WTVC.

The Chinese standards provide a table of fuel consumption targets by GVW/GCW for each vehicle market segment. The market segments are described in the next paragraph below. There are two steps in the current regulation. Stage 1 targets apply to existing production vehicles in July 2014. Stage 2 standards apply to new vehicles launched after July 2014, and existing vehicles after July 2015. The targets were set based on tests of a sample of over 300 different existing production vehicles. The Stage 1 targets were set at a level that would require improvement in about the worst 10% of current vehicles. The Stage 2 targets are 10 to 15% more stringent, and require improvement to almost 50% of current production vehicles [ICCT Update on China HDV, 2013, page 4]. The regulators did not describe the current product baseline vehicles in detail, but they appear to be mostly 2010 and 2011 vehicles that meet China Stage 3 and 4 (roughly Euro 3 and 4) emissions requirements.

Under the new Chinese efficiency regulation, base models of each vehicle type are required to undergo chassis dynamometer testing. Coastdown tests conducted with a fully loaded vehicle are used to determine the aerodynamic drag and rolling resistance values for the chassis dynamometer testing. Depending on the vehicle type, different weightings are applied to the three C-WTVC cycle segments. For example, large semi-tractors skip the urban cycle, and use 10% rural and 90% motorway, while city busses use 100% urban cycle. Table 1 in [SAE Paper 2011-01-2292] shows the market segmentation of MD and HD vehicles, and what proportion of urban, rural, and motorway segments apply to each market segment. The basic market segments are: semi-tractor, dump truck, straight truck, city bus, and motor coach. Some of these vehicle segments are further subdivided by weight class. Some important details of the chassis dynamometer test procedure are not spelled out in the available literature. For example, how is cooling air handled? Is there a supply provided by a wind tunnel? Does the air supply velocity track simulated road speed? The way cooling air is handled will have a large effect on cooling fan power demand, assuming some sort of fan clutch is used in the vehicle.

The parent, or standard, configuration of a vehicle type must be certified on the chassis dynamometer test. For variants of a basic vehicle type, the manufacturer has a choice of chassis dynamometer or vehicle simulation. The simulation procedure uses the same C-WTVC test cycle. The model calculates the transient driving power demand required to follow the speed-time profile. This is translated into an engine torque and speed requirement. The regulation spells out the shifting strategy to be used with manual transmissions. Using a measured engine fuel map, fuel consumption is calculated by the vehicle simulation model. The results on the three test cycle segments are weighted in the same way as they are for the chassis dynamometer test standard.

The Chinese regulation puts an overall efficiency target on the vehicle. It is up to the OEM to set system and subsystem targets that allow the overall vehicle to meet the requirement. One OEM may decide to focus more on engine technologies, while another focuses more on aerodynamic features or weight reduction. The Chinese regulations do not address some of the issues raised in Section 4.2. Another area that the Chinese regulation does not address is that of fleet averaging. Every vehicle must comply, so there is no opportunity to use good performance on some portion of an OEM's fleet to subsidize other less efficient vehicles. This may have an effect on the vehicle market, especially in smaller niches where the engineering cost to meet the standard may prove excessive.

4.3.2 Japan's Regulatory Approach

Japanese regulators introduced MD and HD vehicle efficiency standards as part of a broader effort to improve the efficiency of all power consuming devices from trucks to refrigerators. The program is called "Top Runner," and the overall approach is to find the most efficient vehicle in a given class, and eventually require that all vehicles in the class meet the same efficiency. Top Runner went into effect in 2005, based on test results of MY 2002 vehicles [ICCT 2013]. Hybrid vehicles were not eligible for Top Runner status, since this would have the effect of forcing hybrid technology to be applied across the board. The initial regulatory requirement, which began in 2006, is that vehicle literature must prominently discuss efficiency technologies that have been implemented. All vehicles must meet the Top Runner requirement in 2015, which means that they must have efficiency equal to the best vehicle in class from MY2002. For tractors, the regulation requires an improvement of 9.7% from the 2002 average. Straight trucks and busses have targets ranging from 11 – 13% compared to the 2002 average.

The Top Runner program has goals in terms of fuel economy, measured in km/liter. As a result, the actual reduction in fuel consumption will be somewhat less than the headline increase in fuel economy. For example, a 10% increase in fuel economy is equivalent to a 9.1% reduction in fuel consumption or GHG emissions. Regulatory compliance is determined by vehicle simulation. An engine dynamometer test provides the fuel map for the vehicle simulation. Other vehicle information, such as the number of gears and gear ratios, is entered into the model. Values for Cd, Crr, and vehicle weight are pre-determined by the regulator for each vehicle class, based on average values from the 2002 model year. The simulation model has an algorithm to determine shift points, based on the engine torque curve and vehicle gearing. The model then predicts vehicle fuel economy on two drive cycles. The first is the JE05 drive cycle, which includes a mix of mostly urban driving with a small highway segment [ICCT World Regs, 2013]. The second drive cycle is a constant 80 km/h "Interurban Mode" that includes varying grades of up to +/- 5%. The overall regulatory result is a weighted average of the two drive cycles.

Since vehicle weight, Cd, frontal area, and tire rolling resistance are all input values specified by the regulation (the values vary as a function of vehicle class), the Top Runner program basically translates into a requirement for engine efficiency (fuel map values) and the match between transmission ratios and engine torque curve (keeping the engine in a favorable part of the fuel map as often as possible). Factors that influence the vehicle power demand are deliberately left out of the regulation. Many of the technologies discussed in Section 4.2 would not play a role in determining fuel economy according to the Japanese regulations.

4.3.3 The EU Regulatory Approach

Very high fuel prices in Europe provide a strong market motivation to achieve good vehicle efficiency. Average European diesel fuel prices at the pump as of January 2, 2014 were:

- France $6.88 / gallon
- Germany $7.14 / gallon
- UK $8.60 / gallon
- Italy $8.75 / gallon
- USA $3.91 / gallon (for comparison)

These prices are driven by high taxes on fuel. Given the strong market incentives and the complexity of the truck market, with its huge range of applications and duty cycles, European regulators are taking a slow, conservative approach to regulation of MD and HD vehicle fuel consumption and GHG emissions. As a first step, the EU is developing a vehicle efficiency simulation tool. This tool will be used to meet a vehicle efficiency labeling requirement, but the introduction data for the labeling requirement has not been finalized yet. Once experience has been gained from the labeling program, the EU will consider whether regulatory requirements are needed, and whether the simulation tool provides results useful enough to drive a regulatory requirement. In other words, the EU approach is to develop a system for determining vehicle efficiency, test that system out in a labeling program, fix any deficiencies that are identified, and then determine if an efficiency regulation is justified.

As a safety measure, the EU has required trucks to have a vehicle speed limiter set to 90 km/h (56 MPH) since 1992 [EU web site]. Even though this regulation is driven by safety rather than fuel economy, it has significant fuel economy and GHG implications for vehicles that operate on long distance routes.

The EU is currently developing a program to require vehicle labeling to show CO_2 and fuel consumption performance. There has been an active research program for several years to explore ways of obtaining CO_2 and fuel consumption values that represent real-world experience. Research organizations, universities, and the industry have been participating in this effort. The EU is developing a vehicle simulation tool called VECTO. This tool will use a range of vehicle drive cycles. The drive cycles used for a specific vehicle will be selected based on the intended application of that vehicle. The VECTO model will project fuel consumption and CO_2 emissions on the selected drive cycle(s).

The EU's VECTO model will use the following inputs [ICCT 2013]:

- Engine fuel maps from engine dynamometer testing
- Aerodynamic drag measured using a constant speed on-road test
- Tire rolling resistance, based on the ISO 18164 and 28580 test procedures
- Transmission ratios and efficiency map data from manufacturers
    - Default transmission efficiency values can also be used
- Auxiliary systems power demand from efficiency maps

According to [http://ec.europa.eu/clima/policies/transport/vehicles/heavy/index_en.htm], accessed on January 31, 2015, it is expected that regulations for the EU fuel economy and GHG labeling requirement will be proposed in 2015.

Unlike the US regulation, the proposed EU system does not regulate engines and vehicles separately. Engine efficiency maps are an input to the EU's VECTO vehicle efficiency model. One OEM may decide to focus more on engine technologies, while another focuses more on aerodynamic features or weight reduction. It can be argued that this is a cost effective way to achieve a given efficiency target, although independent engine manufacturers may face a range of efficiency expectations from different OEM customers. The EU approach uses a constant speed on-road test to determine aerodynamic drag, while the US approach is to use coast-down testing. Both test approaches can be extremely sensitive to test variables such as slight changes in grade, wind, and ambient temperature. The EU approach includes transmission ratios and transmission efficiency maps, which are not part of the current US regulation. The EU approach also includes power demand from vehicle accessories, which are not part of the current US regulation.

4.3.4 Canada's Regulatory Approach

So far, the Canadian government has followed the US EPA and NHTSA regulations for truck fuel economy and GHG, with some tweaks to accommodate differences between the US and Canadian truck market. It is too early to know if this approach will continue for the next phase of regulations.

4.4 Recommendations for Certification of Tractor-Trailer Vehicles

4.4.1 Realism of Current GEM Cycle Weightings for Sleeper Tractors

The current GEM regulatory drive cycles could be criticized for not exercising the vehicle power demand over the whole range. Addition of a cycle including grade would address this concern. While it is true that the engine certification cycle already exercises the engine over a wide speed and load range, the SET test cycle is increasingly removed from typical engine operation in tractors, as will be shown below.

It can be argued that the NESCCAF cycle is more representative of sleeper-equipped tractor-trailer operations than the blend of cycles used in the Phase 1 GEM model. Tractors with sleepers use a summation of drive cycles consisting of 86% at 65 MPH cruise, 9% at 55 MPH cruise, and 5% urban transient to determine the GEM result (95% steady-state cycles). Discussions with long haul operators have indicated that it is unusual for a vehicle to average much over 50 MPH in long haul service, and very difficult to average over 55 MPH. The lower average speed obtained in real-world operation is driven by time spent driving off the interstate network, time in congestion, and local speed limits or road conditions including grades. The NESCCAF cycle averages 54 MPH for a truck at 50% payload, with a slight reduction for 100% payload, and a slight increase in average speed when unloaded. The variations in cycle average speed are caused by the 3% grade segments, which cause the truck to lose speed. When compared to the NESCCAF cycle, the Phase 1 GEM approach tends to predict higher benefits for aerodynamic and rolling resistance technologies, as shown in Table 4.3 below. There is little difference between the NESCCAF results and GEM results for the other vehicle technologies that were evaluated in Section 3.

TABLE 4.3 COMPARISON OF TECHNOLOGY PERFORMANCE ON GEM AND NESCCAF CYCLES AT 50% PAYLOAD.

Technology	Fuel Savings Performance		GEM – NESCCAF	GEM Benefits Higher By
	GEM Cycle	NESCCAF Cycle		
25% Cd Reduction	13.74%	12.00%	1.70%	14.1%
30% Crr Reduction	7.72%	6.90%	0.85%	12.5%

If the GEM cycle was modified to the following weightings, the Cd and Crr effects come close to matching the NESCCAF cycle: 79% @ 65 MPH, 0% @ 55 MPH, and 21% on CARB cycle, which has a 15.3 MPH average speed. The reason for eliminating the 55 MPH segment from the GEM summation would be to bring the rolling resistance benefits in line with the NESCCAF results. Note that at 55 MPH cruise, the effect of a change in rolling resistance is very high, because there is no power demand related to inertia (no acceleration) and aerodynamic forces are relatively low compared to 65 MPH. Overall, the suggested GEM weighting of 79% at 65 MPH and 21% on the CARB cycle gives an average speed of 54.6 MPH, very close to the average speed of the NESCCAF cycle, and near the top of the range of typical long haul drive cycles. Results for the suggested reweighting of GEM cycles are shown in Table 4.4.

TABLE 4.4 COMPARISON OF TECHNOLOGY PERFORMANCE ON MODIFIED GEM AND NESCCAF CYCLES AT 50% PAYLOAD

Technology	Fuel Savings Performance		GEM – NESCCAF	GEM Benefits Higher By
	Modified GEM Cycle	NESCCAF Cycle		
25% Cd Reduction	11.86%	12.00%	-0.18%	-1.47%
30% Crr Reduction	6.99%	6.90%	0.13%	1.83%

Rather than simply implementing the change in GEM weighting shown above, it would make sense to take extensive field data from long-haul trucks and use that data to modify the GEM weighting parameters.

4.4.2 Value of Including the Trailer in a Regulation

In a standard 80,000 pound tractor-trailer combination with an empty weight of 34,000 pounds, the weight on the trailer axles represents 22% of the total for an empty vehicle, and 42.5% of the total for a fully loaded vehicle. (See Section 3.3.2.3 for details.) Since trailer tires have somewhat lower rolling resistance than traction tires used on the drive axles, the trailer share of rolling resistance varies from just under 20% when empty to about 40% when fully loaded. Assuming that trucks carry an average of 50% payload, leaving the trailer out of the regulation sacrifices about 30% of the fuel saving benefit that is available from the use of low rolling resistance tires. Note that some operators already take advantage of low rolling resistance trailer tires today on their own, under the SmartWay program, or under California regulations, so a regulation would only affect those who currently use higher rolling resistance tires.

Trailers have an even larger effect on vehicle aerodynamics. Compared to today's most aerodynamic tractors, there remains a potential of up to a 5% fuel consumption reduction due to improvements in tractor aerodynamics, as demonstrated in the SuperTruck program [ET-15 and ET-18]. The potential for improvement on the trailer side is much greater. A SmartWay trailer skirt or boat tail provides about a 5% savings, and the two combined are worth about 8%.

Another factor that should be considered is the question of what sort of trailer the tractor designer should work with. If tractor performance is measured using a "standard" trailer with no aerodynamic features, the tractor designers will optimize for that. If the tractor designer works with a more aerodynamic trailer, the tractor design will include different features with different dimensions, in order to work better with the more aerodynamic trailer. Including trailer aerodynamic performance in the regulation would allow tractor designers to optimize around a more aerodynamic trailer, resulting in greater overall fuel savings. Defining the "standard" trailer against which tractor performance will be judged is a topic that deserves consideration.

Since the trailer owner is often not the same as the tractor owner, the trailer owner has little or no economic incentive to invest in fuel saving technology, since the fuel savings are retained by the tractor owner. This makes a strong argument in favor of regulation of trailer rolling resistance and aerodynamic characteristics. There are typically 3 to 4 trailers per tractor [NRC 2010], so the average trailer has a much lower annual miles traveled than a tractor. This makes achievement of a good cost/benefit ratio for trailer technologies more difficult. Consideration of trailer technologies needs to take this issue into account.

4.4.3 Comparison of SET Test Points with Long Haul Truck Duty Cycles

The SET test cycle is currently used for engine fuel consumption and CO_2 emissions certification. This was done to maintain commonality between emissions and fuel economy test cycles. Unfortunately, the SET test cycle exercises the engine in a way quite different from the way the engine is actually used in the field. This difference in duty cycle can potentially lead to efforts to optimize engine performance in parts of the speed/load range that are irrelevant to actual in-use fuel consumption. Such an approach could reduce the potential benefits of optimizing for actual applications, and misstate the benefits of many technologies. There are also issues with having the emissions test cycle operate in a way that is not representative of real-world operation.

There is another potential drawback of having an emissions and fuel efficiency test cycle that is not representative of real-world operation. Manufacturers could put a high priority on criteria emissions reductions at operating points that are rarely used in actual service, and focus on fuel efficiency at operating points extensively used in service. This sort of approach would optimize real-world fuel consumption and GHG emissions, but at the expense of higher real-world criteria emissions.

Figure 4.1 below shows the percentage of operating time spent in each speed range on the SET ramped modal cycle, compared to field data logged on current production long-haul sleeper tractors in revenue service. This data was provided to SwRI by Volvo. The tractors are equipped with a 13-liter engine and downspeeding technology. The engine cruise RPM on these trucks is set up to be 1160 RPM at 65 MPH road speed. These trucks are also equipped with AMT transmissions, which helps limit the in-use engine speed range. Note that results for other engine and transmission types will vary from the results presented here. Comparing the two data sets, it is clear that the SET test cycle emphasizes higher RPM operation, where the engine rarely operates in the field. Less than 1% of the engine's field duty cycle is at the B speed or higher, while on the SET the B and C speeds represent over half of the total duty cycle. The SET test under-represents idle speed time and time around the A speed, which for this particular vehicle configuration closely matches the highway cruise speed.

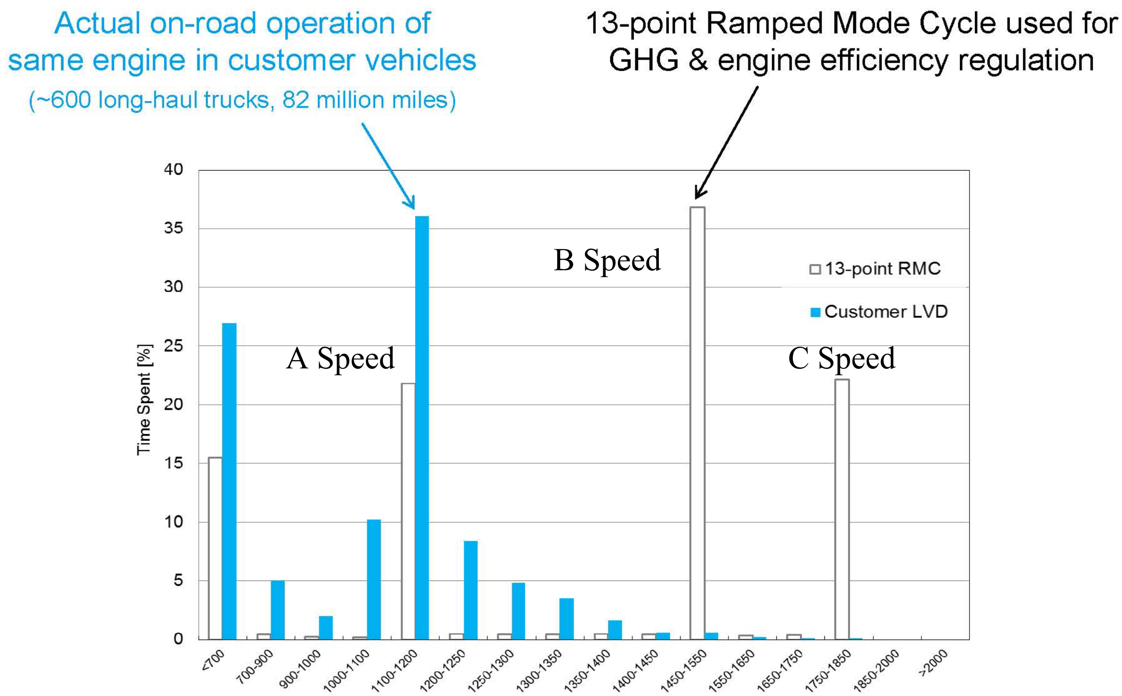


FIGURE 4.1 COMPARISON OF SET TEST CYCLE SPEEDS AND ON-HIGHWAY DUTY CYCLE SPEEDS

It must be noted that the data in Figures 4.1 and 4.2 is for a downspeeded engine with an AMT. These configurations are becoming popular on the market, and are likely to increase in market share in the future. Today, however, manual transmission applications geared to cruise at around 1300 RPM @ 65 MPH are most common. If the comparison of field data and SET data shown in Figure 4.1 was repeated for "standard" engine and transmission configurations, the results would show a large peak around 1300 PM, which is about midway between the A and B speeds. There would also be a little more high RPM operation. Even this "standard" configuration, however, would show very little real-world operation above the B speed.

Figure 4.2 below compares engine torque on the SET test cycle with actual on-road duty cycle data for the same fleet of trucks with downspeeding and AMT. The SET cycle roughly matches the actual duty cycle for the percentage of time spent at the highest torques, but the SET cycle does not accurately represent the relatively even distribution of real-world torque values across the wide range between 300 and 2200 Nm. The SET also missed the very high portion of time spent in the field at very light load (0 to 300 Nm). Combined with the wide discrepancy in engine speeds shown in Figure 4.1, the results suggest that the SET cycle is not a very good representation of an actual long-haul truck duty cycle. There is a risk that optimization of the engine to perform better on the SET cycle will need to focus on high speed, medium and high load operation that is rarely used in actual long-haul duty cycles. It is worth noting that the increasing use of aerodynamic tractors and trailers, low rolling resistance tires, and other vehicle efficiencies, driven both by regulation and customer demand, will significantly reduce the torque profile of future vehicles even further.

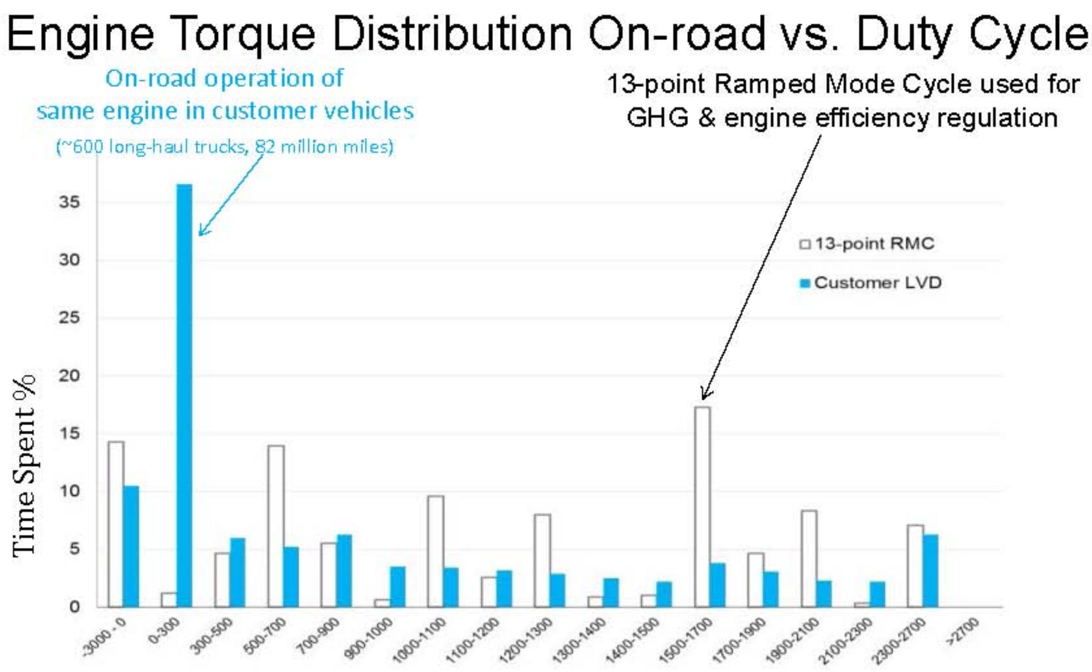


FIGURE 4.2 COMPARISON OF SET TEST CYCLE TORQUES AND ON-HIGHWAY DUTY CYCLE TORQUES

Based on a comparison of in-use duty cycle data and the regulatory certification cycle, SwRI recommends that EPA and NHTSA consider modifying the duty cycle used for certification of fuel consumption and GHG emissions.

4.5 Recommendations for Certification of Vocational Vehicles

4.5.1 Parameters Considered for Vocational Vehicle Certification

The current regulation for vocational vehicles includes only tire rolling resistance. As Figure 4.3 below shows, tire rolling resistance is a significant factor in vocational truck fuel consumption. Note that the advantages shown for the AMT transmissions on the CILCC and Parcel cycles will be reduced when a neutral idle feature is applied to the automatic transmission. Aerodynamic drag is also an important factor, but only for vehicles that drive a significant portion of their miles at high speed.

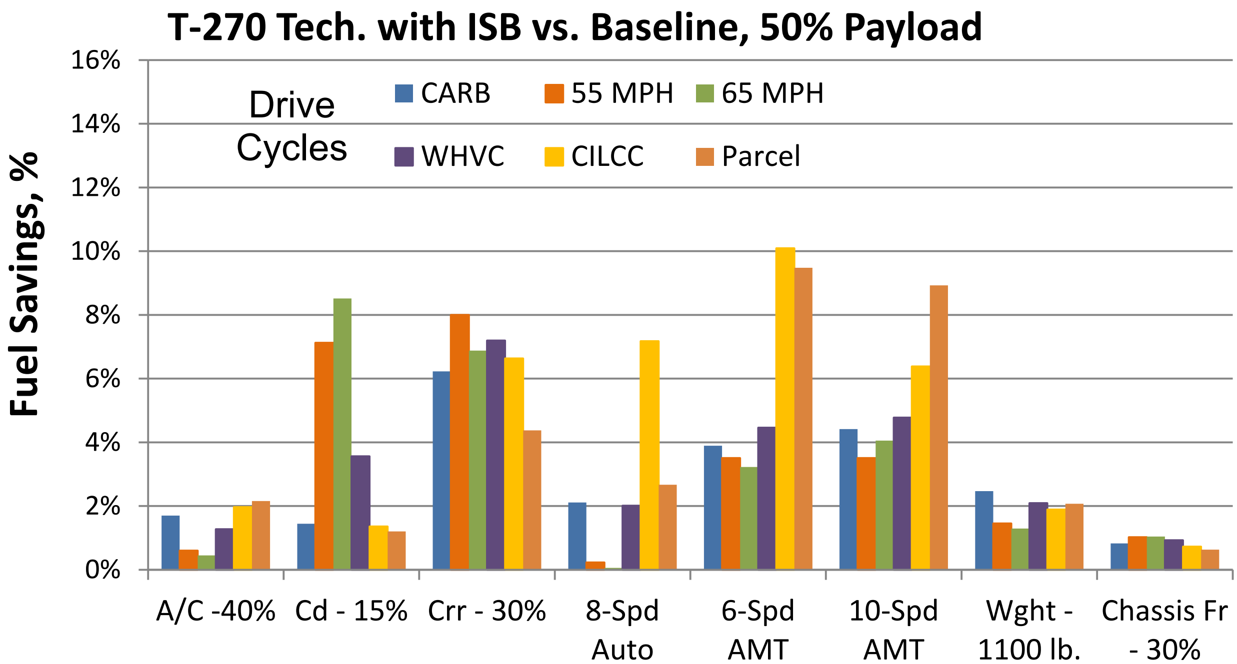


FIGURE 4.3 FUEL SAVINGS OF VEHICLE TECHNOLOGIES APPLIED TO THE T270 VOCATIONAL TRUCK

Many vocational vehicles are delivered from the OEM as a running chassis, with the bodywork and equipment added by a separate company. There are hundreds of companies in the bodybuilding business, which would make regulation very difficult. The body and equipment drive both the Cd and the weight of the completed vehicle.

It might be possible for the regulator to drive small incremental improvements through additional requirements on the chassis manufacturer. For example, some reduction in parasitic power from air conditioning, power steering, and other accessories is possible. It might be feasible to require a chassis aerodynamic skirt on most vocational trucks, although exemptions would be needed for cases where the skirts would interfere with the bodywork or with vehicle function. Modifications to the GEM model would be needed to allow these factors to be accounted for.

4.5.2 Comparison of FTP Test Points with Vocational Truck Duty Cycles

As was the case with sleeper tractor trucks and the SET certification cycle, a comparison of the FTP test cycle and actual vocational truck duty cycles can provide insight into the usefulness of the regulatory cycle for predicting real-world fuel consumption and GHG performance. As is the case with the SET cycle, there are significant discrepancies between the FTP test cycle and in-use duty cycles.

Figure 4.4 below shows the speed distribution of the FTP cycle compared to recorded in-use data for a variety of vocational trucks and transmissions using an 11-liter engine. This data comes from trucks that do not use downspeeding technology, and which use mostly manual transmissions. These trucks are set up to cruise at engine speeds ranging from 1488 RPM to 1774 RPM at 65 MPH road speed. This data was also provided to SwRI by Volvo. Results from other manufacturers may vary. For the Volvo data, the FTP cycle clearly over-represents the time the engine spends at speeds over 1650 RPM by a factor of 2 to 5. The FTP cycle under-represents the time the engine spends in the 700 to 900 RPM range, as well as the more important 1000 to 1550 RPM range.

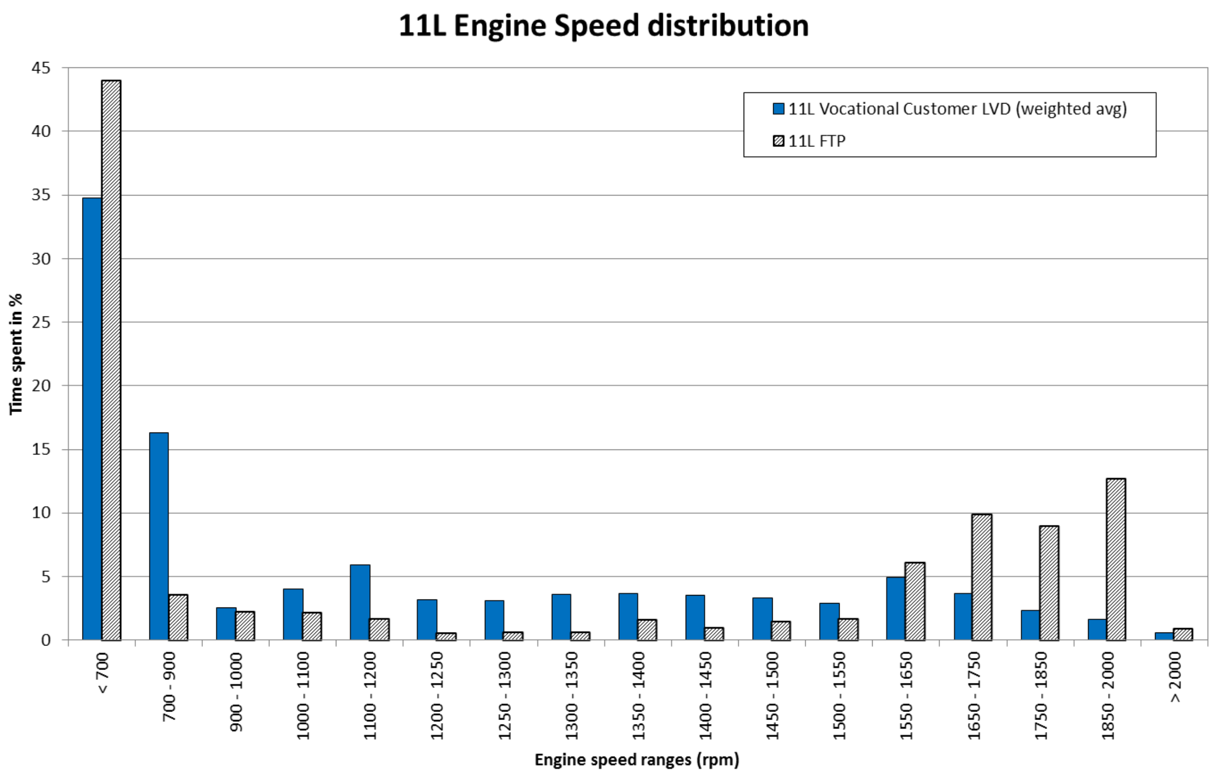


FIGURE 4.4 ENGINE SPEED DISTRIBUTION ON THE FTP CYCLE AND IN-USE OPERATIONAL DATA

Figure 4.5 shows the torques operated by the 11-liter engine on the FTP cycle and in field service. The FTP cycle over-represents the higher torques in the 1500 to 1900 Nm range, but in general, the FTP cycle gets the torque distribution approximately correct. One thing that does not show up in these figures of speed and torque is the combination of speed AND torque. The FTP includes a fair amount of high speed, low and medium torque operation that is not found in real-world duty cycles. In general, during the relatively few occasions that drivers use high engine speed, it is because they are looking for maximum power from the engine. Driver training emphasizes that high RPM, light load operation is bad for fuel economy.

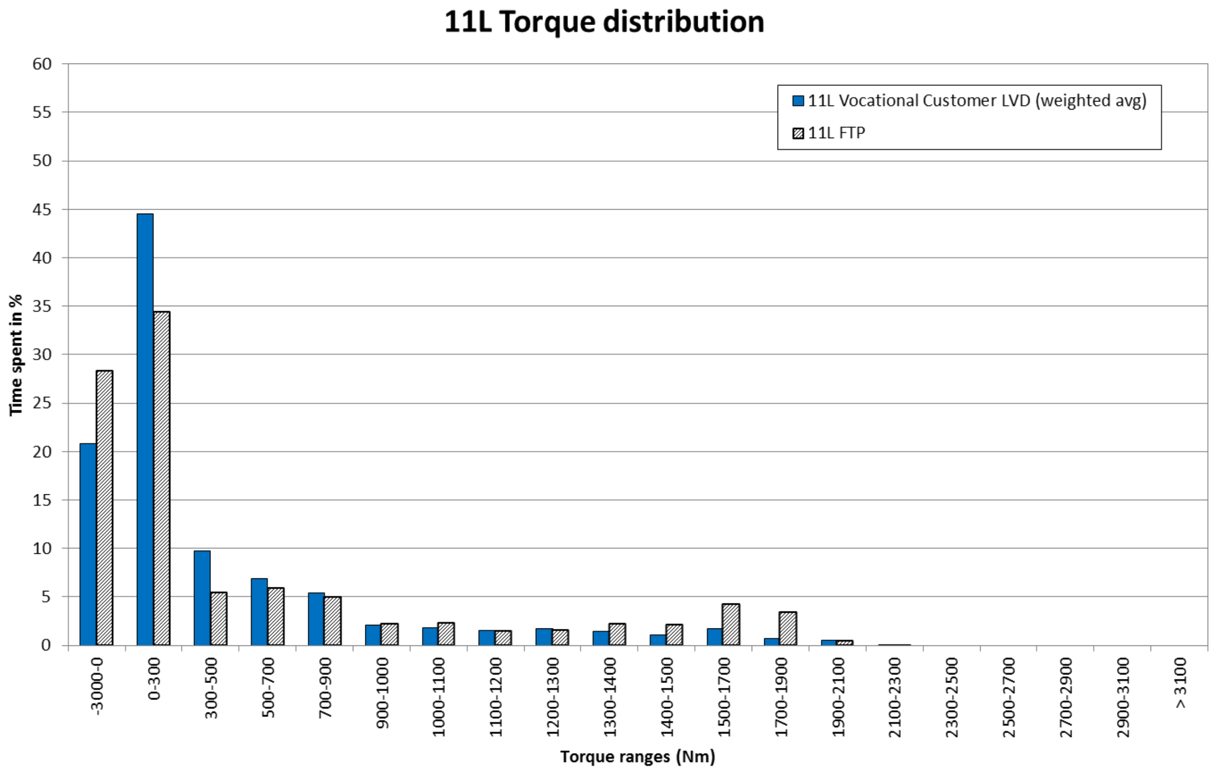


FIGURE 4.5 COMPARISON OF FTP TEST CYCLE TORQUES AND ON-HIGHWAY DUTY CYCLE TORQUES

SwRI recommends that EPA and NHTSA evaluate the usefulness of the FTP cycle for predicting the fuel consumption and GHG emissions of engines in field service.

4.6 Effect of Drive Cycle on Technology Performance

Drive cycles and payload can significantly affect the performance of vehicle and engine technologies. In this section, technologies will be evaluated over a range of payload and drive cycle.

The T-700 tractor and DD15 engine are generally intended for long haul operation, but vehicles like this can sometimes find use in low speed drayage operations. Figure 4.6 shows the performance of five technologies on the 65 MPH steady state cycle. Aerodynamic drag has a huge impact on this drive cycle, but the benefit of an aerodynamic drag improvement declines with increasing payload. Tire rolling resistance has the second largest fuel savings on the 65 MPH cycle, but its value *increases* with higher payload. There is a trade-off between the portion of vehicle power demand going to aerodynamics, which is independent of payload, and the portion of power demand going to rolling resistance, which increases with payload. The third largest benefit comes from Downspeeding, which provides around a 4% benefit. This benefit declines slightly as payload increases. The values of reduced engine friction and reduced vehicle weight are in the 2% and 1% range respectively, and these values also decline slightly with increasing payload.

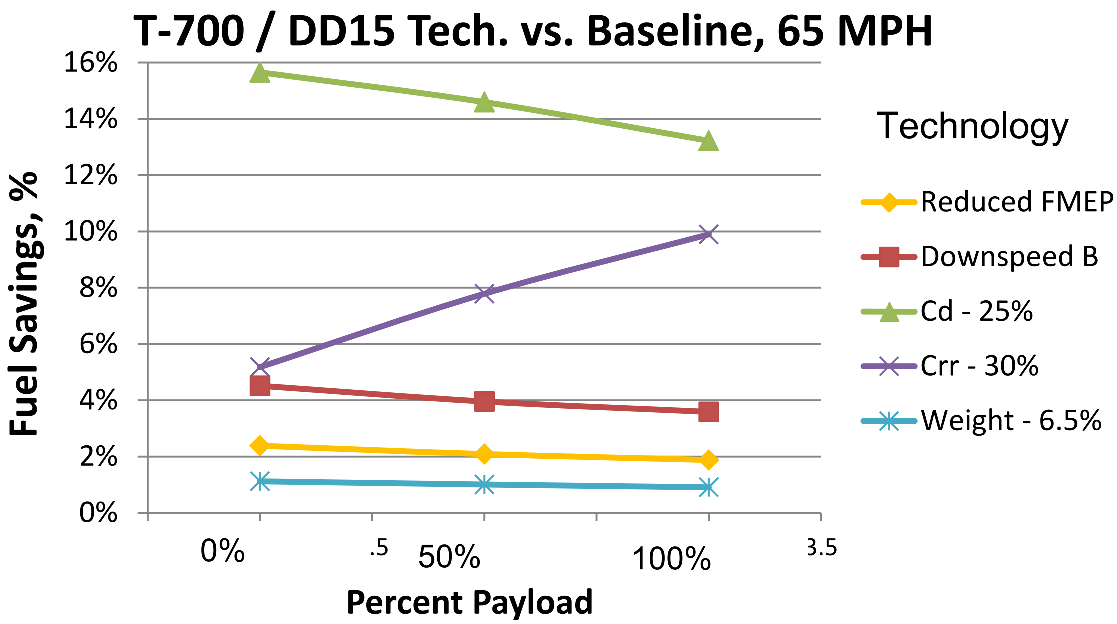


FIGURE 4.6 SENSITIVITY OF T-700 AND DD15 TECHNOLOGIES TO PAYLOAD AT 65 MPH

At the other extreme, the CARB urban cycle has an average speed of about 15 MPH, and it represents stop and go urban driving. Most technologies perform very differently on this cycle. Aerodynamic improvement, which gives the largest benefit at 65 MPH, has the smallest benefit on the CARB cycle (Figure 4.7). The load sensitivity is still the same, with aerodynamics becoming less important as payload increases. Downspeeding moves up to 2nd place in the benefit ranking, from 3rd place at 65 MPH. The benefit of downspeeding is more sensitive to payload on the low speed CARB cycle than at 65 MPH. It should be noted, however, that downspeeding is the only technology with nearly identical benefit on both drive cycles. Reduced engine friction moves up to become the #1 benefit on the CARB cycle, up from 4th place at 65 MPH. The benefit of friction is also strongly payload dependent. Weight reduction provides about twice the benefit on the CARB cycle as is found on the 65 MPH cycle. Overall, note that no technology exceeds a 6% benefit on the CARB cycle, while two technologies exceed 6% (and in one case, 15%) on the 65 MPH cycle. Results on the other drive cycles fall between the extremes defined by the 65 MPH cycle and the CARB cycle.

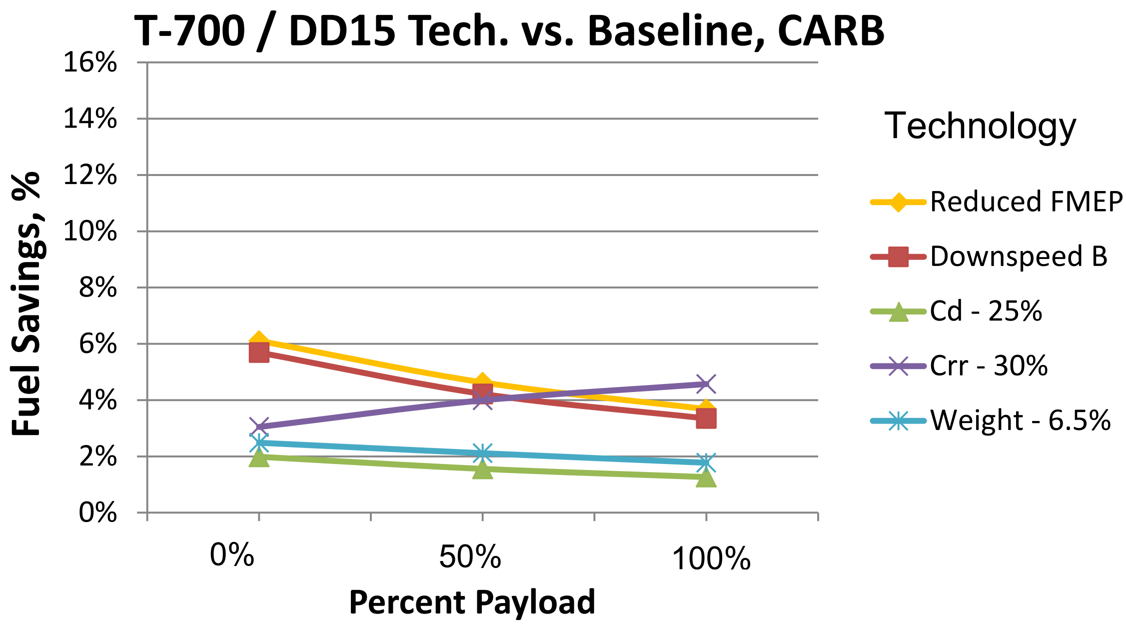


FIGURE 4.7 SENSITIVITY OF T-700 AND DD15 TECHNOLOGIES TO PAYLOAD ON THE CARB CYCLE

Vocational vehicles also see changes in technology performance depending on drive cycle and payload. Figure 4.8 shows the performance at 65 MPH of six engine and vehicle technologies on the Ford F-650 tow truck. The largest benefit is provided by the addition of EGR and downspeeding to the baseline 3.5 liter V-6 gasoline engine. The benefit of this engine technology declines slightly with higher payload. Tire rolling resistance provides the second

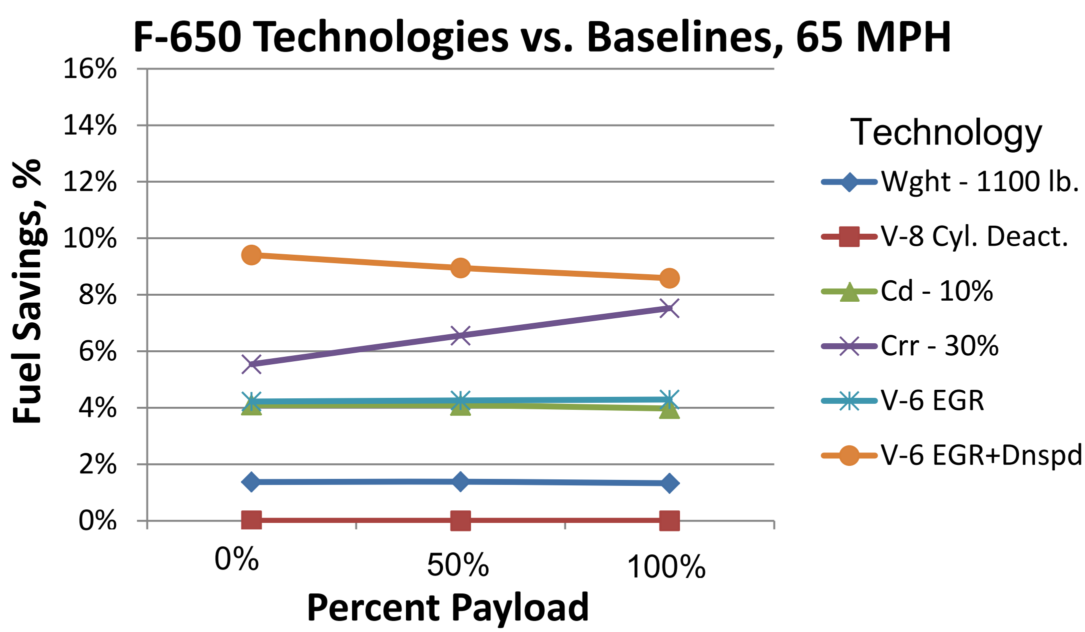


FIGURE 4.8 SENSITIVITY OF F-650 AND ENGINE TECHNOLOGIES TO PAYLOAD AT 65 MPH

largest benefit, and this benefit strongly increases with payload. Applying EGR only to the 3.5 liter V-6 provides a benefit just over 4%, regardless of payload. Reducing aerodynamic drag by 10% provides about a 4% improvement, independent of payload. Reducing vehicle mass by 1100 pounds provides a 1.3% fuel savings, independent of payload. Finally, cylinder deactivation on the 6.2 liter V-8 provides no benefit at all, because at the 65 MPH operating point, the engine needs all 8 cylinders.

The comparison between 65 MPH and the CARB cycle is not quite as dramatic with the F-650 as it was on the sleeper tractor T-700 above, but there are still a number of important differences in performance between the two cycles. On the CARB cycle, the EGR + Downspeeding technology on the V-6 engine provides the largest benefit, as it did at 65 MPH. The benefit is a bit lower on the CARB cycle, however. Cylinder deactivation on the 6.2 liter V-8, which provides no benefit at 65 MPH, comes in 2nd place on the CARB cycle, with benefits of 4 to 6 percent. The lower benefit comes at full payload. Tire rolling resistance roughly ties for 2nd on the CARB cycle. The benefit of lower rolling resistance consistently increases with higher payload. EGR on the 3.5 V-6 engine provides almost the same benefit on both drive cycles (around 4%), and is relatively insensitive to payload. Weight reduction is more important on the CARB cycle that at 65 MPH, a result that matches the T-700 results described above. Reduced Cd is the least effective technology on the CARB cycle, down substantially from its performance at 65 MPH.

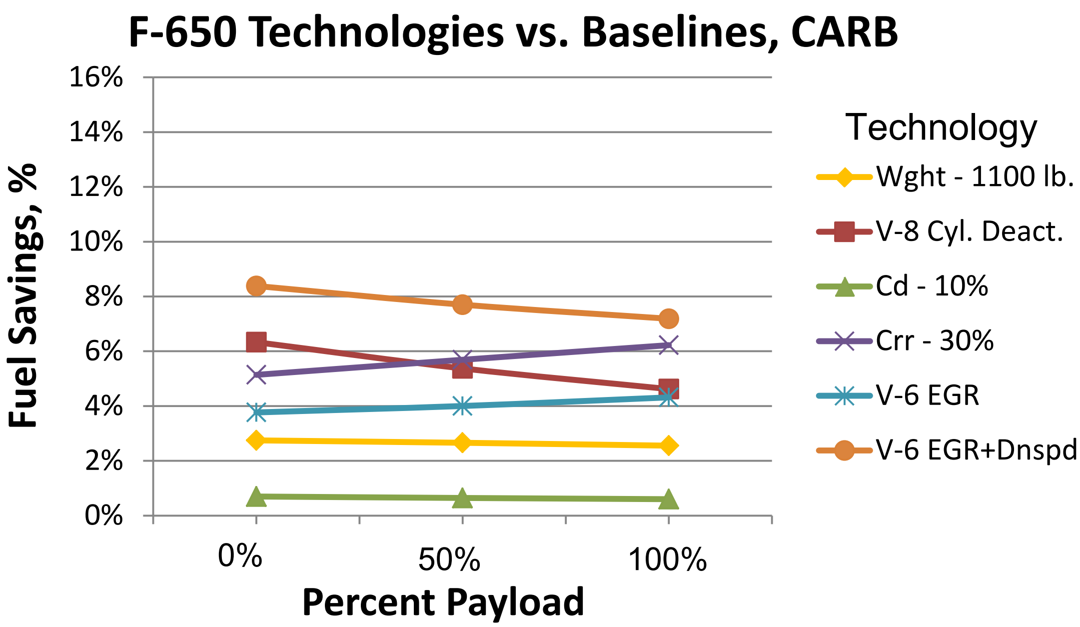


FIGURE 4.9 SENSITIVITY OF F-650 AND ENGINE TECHNOLOGIES TO PAYLOAD ON THE CARB CYCLE

Figure 4.10 below shows the performance of several technologies on the Ram 2b/3 pickup truck as a function of payload on the US06 test cycle. The US06 is an aggressive drive cycle that includes rapid accelerations and speeds up to 80 MPH. The three payload levels are not evenly distributed. The baseline vehicle weighs 6876 pounds. At ALVW, there is a 1562-pound payload, for a total vehicle weight of 8,438 pounds. At GCW, a trailer is added to provide a total vehicle weight of 25,000 pounds, along with a 50% increase in frontal area to account for the additional aerodynamic drag of the trailer. As a very important caveat, it should be noted that the fuel savings benefits predicted for the GCW case are ***not reliable***. This is because the vehicle was unable to follow the aggressive drive cycle very well, with discrepancies in speed for 10% or more of the total drive cycle. For the 4-cylinder diesel, there was a speed discrepancy about 20% of the time. Changes that affect either engine power or vehicle power demand resulted in changes in the speed/time history on the US06. This makes the fuel consumption results unreliable.

On the US06 cycle, the most effective technology shown in Figure 4.10 is the combination of EGR and downspeeding on the 3.5 V-6 gasoline engine. This technology performs best at full GCW, where the baseline engine often has to use extensive enrichment to avoid excessive engine and catalyst temperatures. EGR alone on the 3.5 V-6 is the second most effective technology on the US06, with benefits from 5% at zero payload to over 17% at GCW. Reduced rolling resistance provides benefits from 6 to 9%. These benefits also increase with payload, but not nearly as strongly as the two EGR technologies. The 4-cylinder version of the diesel provides a 4% to 6% benefit. A 500-pound weight reduction is worth 3% at zero payload, falling to 1.3% at GCW. Cylinder deactivation on the 6.2 V-8 gasoline engine is worth 3.1% at zero payload, falling to 0.3% at GCW. A 10% improvement in Cd is worth just over 2% at zero payload, falling to 1% at GCW.

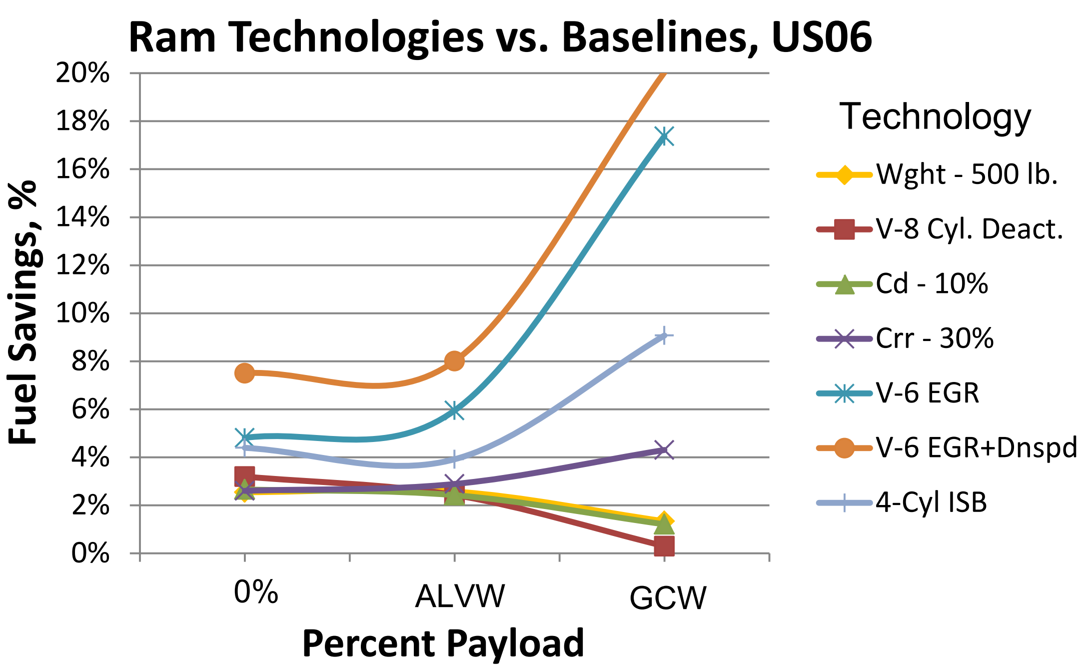


FIGURE 4.10 SENSITIVITY OF PICKUP VEHICLE AND ENGINE TECHNOLOGIES TO PAYLOAD ON THE US06 CYCLE

Figure 4.11 below shows the performance of the same set of technologies on the low speed FTP-City cycle. On this cycle, the 4-cylinder diesel is the best performing technology. The downsized diesel benefits from an increase in average BMEP compared to the baseline. This technology ranked #4 on the US06 cycle. Note that the benefit of the smaller diesel drops off considerably at full GCW, where the engine runs near full load much of the time. The number 2 performer on the city cycle is EGR and downspeeding, applied to the V-6 gasoline engine. This technology provides a benefit of around 7% that is independent of payload. Cylinder deactivation on the V-8 gasoline engine provides over 9% benefit at zero payload, falling to 2% at GCW. At full GCW, there is little opportunity to use cylinder deactivation. The benefits of reduced rolling resistance and reduced weight are similar on the city cycle and on the US06. Finally, a reduction in Cd has almost no effect on the low speed city cycle.

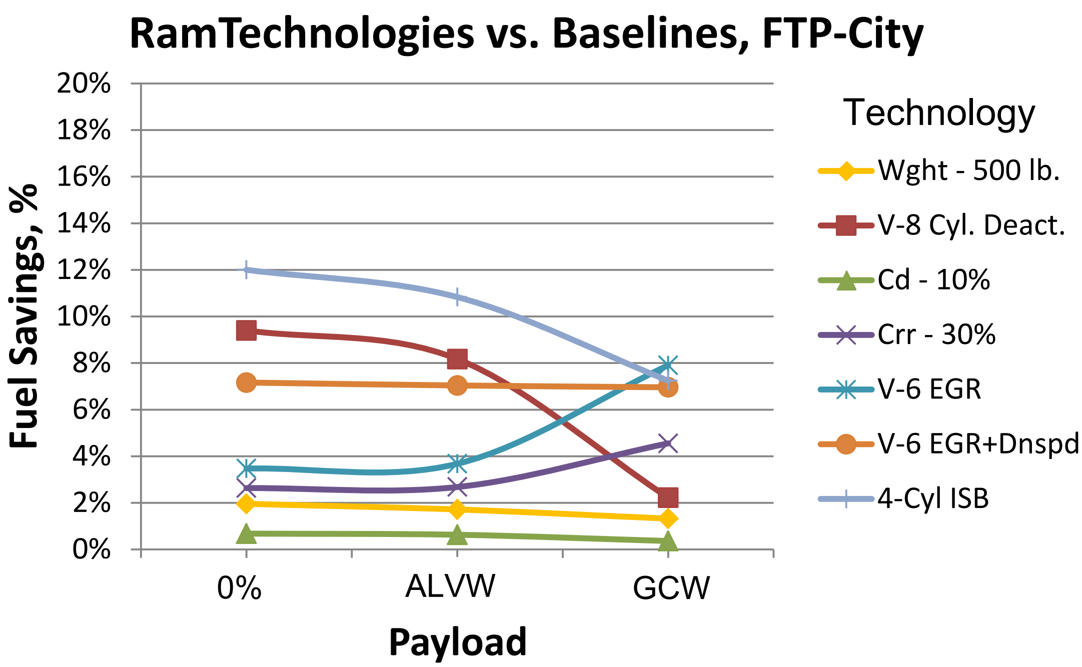


FIGURE 4.11 SENSITIVITY OF PICKUP VEHICLE AND ENGINE TECHNOLOGIES TO PAYLOAD ON THE FTP-CITY CYCLE

5.0 SUMMARY

The tractor-trailer truck shows the largest potential percentage fuel savings of any of the vehicles evaluated. This is fortunate, since tractor-trailers account for a significant portion of overall medium- and heavy-duty vehicle fuel consumption. Another factor in favor of fuel saving technologies for tractor-trailer vehicles is the high average miles traveled per year, at least for tractors. The high VMT gives an opportunity to pay back the cost of what are often expensive technologies (see the separate cost report for cost information). Since there are three to four trailers per tractor, benefits for trailer technologies accrue based on their respective VMT.

Given the technologies evaluated in this report, the most promising long haul truck engine technologies (in terms of fuel savings – cost will be considered in a separate report) are:

- Friction reduction, downspeeding, and downsizing: 2% to 4% each
- Elimination of EGR, turbo efficiency improvement: up to 2% each (note that EGR elimination requires very high conversion efficiency from the aftertreatment)
- Waste heat recovery systems: 3% to 5% on highway cycles, but little benefit in urban driving

Note that downsizing and downspeeding both have the same effect, which is to drive up the average load on the engine (BMEP), so these two technologies can only be combined to a limited extent. Also note that friction reduction is not always compatible with downsizing or downspeeding. Since both downsizing and downspeeding tend to push up cylinder pressure, the engine design must accommodate higher cylinder pressure, and this tends to increase friction.

Tractor-trailer trucks show a significant potential for vehicle power demand reduction, and thus for fuel consumption reduction. The most promising technologies here include:

- 25% Cd reduction: < 2% fuel savings in urban driving, 12% to 14% on high speed cycles
- 30% Crr reduction: ~ 4% fuel savings in urban driving, 6% to 8% on high speed cycles
- 2,200 pound empty weight reduction: 1% to 2%
- 6 X 2 axles: ~1.5% fuel savings
- Road speed governor: 1.2% per 1 MPH speed reduction (only effective when the speed would otherwise be higher than the governed limit)

The potential of medium duty truck diesel engine technologies is more limited than for tractor-trailer engines, primarily because waste heat recovery systems are not practical for use in transient operating conditions. This is due to the extremely slow transient response of waste heat systems. The high cost of WHR systems will also be an issue in medium trucks. The most promising medium duty diesel technologies are:

- Friction reduction: 2% to 8%, with larger benefits on low speed, lightly loaded cycles

- EGR elimination: 2% to 3.5% (note that this option requires very high aftertreatment conversion efficiency)
- Improved turbocharger efficiency: about 1.5%

Gasoline engines show a high potential for fuel consumption improvement in medium truck applications. The limited gasoline engine offerings in medium trucks today are large displacement, port injected, naturally aspirated engines. These engines suffer from very high fuel consumption at high loads, where they use enrichment to limit engine and exhaust temperatures. Downsizing and boosting is one option. Using today's technology, this option has a fuel savings potential of up to 16% compared to the large naturally aspirated engine, on the most lightly loaded vehicles and gentle drive cycles. Unfortunately, that benefit shrinks to zero on the most heavily loaded and aggressive cycles in the study (and none of the medium truck cycles in the study is very aggressive, since there are no grades). Gasoline engines with today's technology have a substantial fuel consumption penalty compared to a medium-duty diesel. The diesel's advantage is largest on the more heavily loaded and aggressive duty cycles, where gasoline engines run in the enrichment portion of their maps.

Compared to the naturally aspirated V-8, the diesel uses 15% to 25% less fuel than the gasoline alternative. Note that about 13% of this difference is due to the lower energy content of a gallon of gasoline compared to a gallon of diesel, while the remainder is a difference in efficiency. The CO_2 emissions from a gallon of diesel are about 14% higher than from a gallon of gasoline, so the CO_2 advantage of the diesel ranges from 1% to 11%. The following technologies can improve the fuel consumption of the naturally aspirated V-8:

- VVA/VVL: 3% to 5% fuel savings, with the largest benefits on lightly loaded cycles
- Cylinder deactivation: 0% at high speed and load, up to 8% on the most lightly loaded vehicle and cycle
- Stoichiometric EGR: 3% to 10% fuel savings on the drive cycles used in this study, with the largest savings under high load conditions. Up to 30% savings at 100% load.
- Lean GDI: 8% to 11%, with the highest benefits on lightly loaded cycles (note the lack of available, durable lean aftertreatment for spark ignited engines)

The downsized and boosted 3.5 liter V-6 evaluated in this study often has a fuel consumption advantage over the 6.2 liter V-8. For a given power demand, the V-6 runs at higher BMEP. This gives the V-6 an efficiency benefit at light load, but a penalty at high load, where enrichment causes high fuel consumption. The light load advantage of the V-6 comes from reduced pumping work (throttling loss) and lower friction (from the smaller size). Beyond the benefits of downsizing and boosting, the fuel consumption of the V-6 can be reduced significantly with the following technologies:

- VVA/VVL: 2% to 3% fuel savings (less than for the V-8, because the smaller displacement means that there is less pumping work to reduce)
- Cylinder Deactivation: 0% to 3% (much less than the V-8, because the V-6 rarely runs at a light enough load to make use of cylinder deactivation)
- Stoichiometric EGR: 4% to 10%, with the largest fuel savings under high load conditions. Up to 30% savings at 100% load.

- EGR + Downspeed: 2% to 11% fuel consumption reduction, with the smallest savings coming on the Parcel cycle, which has 50% idle time. The penalty on the Parcel cycle is due to higher idle power demand on the engine from a tighter torque converter. On the remaining drive cycles, downspeeding provides a 1% to 2% fuel savings on the large, heavy T270, and about 4% on the smaller, lighter F-650.
- Lean GDI: 2% to 11%, with the highest benefits on lightly loaded cycles (note there is a lack of available, durable lean aftertreatment for spark ignited engines)

With EGR and downspeeding, the 3.5 liter V-6 comes within 3% of the fuel consumption of the diesel in the F-650 on the CARB cycle. This means that under low speed, light load conditions, the small V-6 gasoline engine can exceed the thermal efficiency of the much larger diesel, as well as offering up to 11% lower CO_2 emissions. Note that the diesel retains an 18% fuel consumption and a 4% CO_2 advantage under higher load operation, such as at 65 MPH cruise in the larger T270. The harder the engine needs to work, the larger the diesel's advantage will be. These results do indicate that for more urban applications, an advanced technology gasoline engine can compete with a diesel on thermal efficiency. Given the huge cost advantage of gasoline engines with their simple 3-way catalyst aftertreatment, and the lower price of gasoline compared to diesel fuel, this is an important finding.

Medium trucks have less potential for vehicle power demand reduction than tractor-trailer trucks. There is less opportunity for aerodynamic improvements, so this study evaluated a potential improvement in Cd of 15% on the T270 box truck and 10% on the F-650 tow truck. However, as for the tractor trailer trucks, these trucks were evaluated with a potential 30% tire rolling resistance reduction. The largest medium truck fuel consumption reductions came from:

- 15% / 10% Cd reduction: 1% to 8% fuel savings on the T270, 0.5% to 4% on the F-650, which has less frontal area as well as a smaller Cd improvement potential. The largest benefit is at 65 MPH cruise in both trucks.
- 30% Crr reduction: about 6% fuel savings on most cycles. Higher savings at 55 MPH cruise, lower on the Parcel cycle, which has 50% idle time.
- High efficiency 8-speed automatic: 0% to 2% on most cycles, more on the very gentle CILCC cycle.
- Automated Manual Transmission (AMT): 3% to 4% fuel savings on most cycles, up to 10% on the CILCC and Parcel cycles. The advantage on the parcel cycle is mostly due to elimination of torque converter load at idle. Note that the AMT will have significantly less acceleration capability and driving smoothness in transient conditions.
- 1100 pound weight reduction: 1% to 2%, with the larger benefits on transient cycles

The pickup truck shares the same gasoline engines with the medium trucks. The diesel has the same displacement, but for the pickup truck the rated engine speed is increased to 3,000 RPM from 2,500. The pickup diesel also has more power and torque than the medium duty version: 385 HP and 850 lb-ft, compared to 300 HP and 750 lb-ft for medium trucks. In the pickup truck, the fuel savings potential of eliminating EGR and for a more efficient turbocharger are very similar to the results reported above for the medium duty version. Friction reduction is more important in the more lightly loaded pickup, with fuel savings of 4% to 8% when the truck operates at ALVW. The benefit of friction reduction is much less when the truck operates at full GCW, because of the higher average engine load.

One additional diesel variant was considered for the pickup: a 4.5 liter 4-cylinder version of the original 6.7 liter inline-6. The 4-cylinder retained the BMEP and speed range of the larger engine, so there is a 33% reduction in available power and torque. The 4-cylinder engine still has substantially more power and torque than pickup diesels of 15 years ago, however. The 4-cylinder provides fuel savings of 0% to 13%. The 0% figure comes at 65 MPH and full GCW, while many cycles show a benefit of over 10%. The primary benefit is operation at higher BMEP for a given road load, which at light load pushes the 4-cylinder into a more efficient portion of its operating map. The 4-cylinder also benefits from lower friction. Many commercial pickup operators may be willing to trade the performance of the 6-cylinder for the substantial fuel consumption improvement of the smaller diesel, provided that they do not routinely pull large, heavy trailers.

For the 6.2 V-8 in the pickup, VVA/VVL, cylinder deactivation, and lean GDI all perform somewhat better than in medium trucks, because of the lower average vehicle power demand:

- VVA/VVL: 4% to 7%, with the largest benefits on lightly loaded cycles
- Cylinder deactivation: 0% at high speed and load, up to 10% on the most lightly loaded vehicle and cycle
- Stoichiometric EGR: 3% to 11%, with the largest savings under high load conditions (full GCW). 3% to 6% savings on most cycles, but up to 30% savings at 100% load.
- Lean GDI: 8% to 14%, with the highest benefits on lightly loaded cycles (note the lack of available, durable lean aftertreatment for spark ignited engines)

As with the V-8, the 3.5 liter V-6 engine sees slightly larger fuel savings from several of the technologies in a pickup compared to the medium truck results:

- VVA/VVL: 2% to 3% (less than for the V-8, because of the smaller displacement)
- Cylinder Deactivation: 0% to 3% (much less than the V-8, because the V-6 rarely runs at a light enough load to make use of cylinder deactivation)
- Stoichiometric EGR: 4% to 10%, with the largest savings under high load conditions. 17.4% savings on the US06 cycle at full GCW. Up to 30% savings at 100% load, due to elimination of enrichment.
- EGR + Downspeed: 6% to 10% on most cycles, with the smallest savings coming on lightly loaded cycles. 20% savings on the US06 cycle at full GCW. Downspeeding provides a 3% to 5% fuel savings over EGR only at zero payload and at ALVW, but much less at full GCW.
- Lean GDI: 8% to 15% fuel savings at zero payload and at ALVW, but only 1% to 6% at full GCW, where high power demand often pushes the engine out of the lean operating region (note the lack of available, durable lean aftertreatment for spark ignited engines).

In general, the pickup truck proved less sensitive to Cd and Crr than the larger trucks. With its smaller frontal area, the pickup will have a lower CdA value. Also, tire rolling resistance represents a smaller portion of total vehicle power demand. On the other hand, the pickup is more sensitive to empty weight reductions and auxiliary power demand. Vehicle technologies applied to the pickup truck provided the following benefits:

- 10% Cd reduction: 0.4% to 3.8% fuel savings, with the largest benefit at 65 MPH cruise.
- 30% Crr reduction: 2.2% to 4.4% at zero payload and at ALVW, and up to 7% at full GCW (assuming a 30% reduction in trailer tire Crr).
- High efficiency 8-speed automatic: 0.2% to 4.8% fuel savings, with larger savings at light load and on gentle cycles.
- 500 pound weight reduction: 1% to 2.6%, with the larger benefits with light payloads and on transient cycles
- 600 watt auxiliary power demand reduction (A/C): 0.9% to 2.2% at zero payload and at ALVW, with the largest benefit on low speed cycles. 0.5% to 1.2% at full GCW.

Auxiliary power demand reduction was also evaluated on the larger trucks, but it has the largest benefits on the pickup truck, where the average vehicle power demand is lowest.

The results presented in this report show that known technologies can provide significant fuel savings across a wide range of Class 2b through Class 8 trucks. These results can be combined with cost information (to be provided in a companion report) and VMT data to estimate cost effectiveness. A second report will evaluate technology combinations. This is an important topic, because while some technologies may be synergistic, in many cases, adding a second technology will reduce the benefit obtained from the first technology.

APPENDIX A

GASOLINE ENGINE TECHNOLOGIES

Gasoline Engine Technologies

Objective: Simulate Future Gasoline Engine Technologies to demonstrate Fuel Economy Improvement Potential in Class 2b through 7 vehicles utilizing:

3.5L V6 turbocharged, direct injected gasoline engine

6.2L naturally aspirated, port injected V8 engine

1. **3.5L V6 turbo technology evaluation plan:**
    - 1.1. **Baseline V6 Validation (Turbocharged, Direct Injection, Stoichiometric Operation)**
    - 1.2. **Explore Variable Valve Lift, Duration, and Timing**
    - 1.3. **Explore Cylinder Deactivation**
    - 1.4. **Lean Burn GDI**
    - 1.5. **Explore GDI with cooled EGR (HEDGE)**
    - 1.6. **EGR Engine Down speeding with increased BMEP and PCP**
    - 1.7. **Explore FMEP Improvements**
    - 1.8. **Explore turbo efficiency improvement**

2. **6.2L naturally aspirated V8 engine evaluation plan**
    - 2.1. **Baseline V8 Validation (Naturally Aspirated, Port Injected)**
    - 2.2. **Stoichiometric Gasoline Direct Injection**
    - 2.3. **Lean Burn GDI w/SCR**
    - 2.4. **Variable Valve Lift**
    - 2.5. **Cylinder Deactivation**
    - 2.6. **GDI with EGR**
    - 2.7. **FMEP Improvements**

1.1 Baseline V6 Validation

Figure A 1. Ford EcoBoost 3.5 liter engine

- **Ford EcoBoost**
    - 3.5L V6
    - Twin Turbo
    - Single Cam Phaser (Intake)
    - 4 valves per cylinder
    - Gasoline
    - 265/272 kW (car/truck)
    - 475/569 Nm (car/truck)

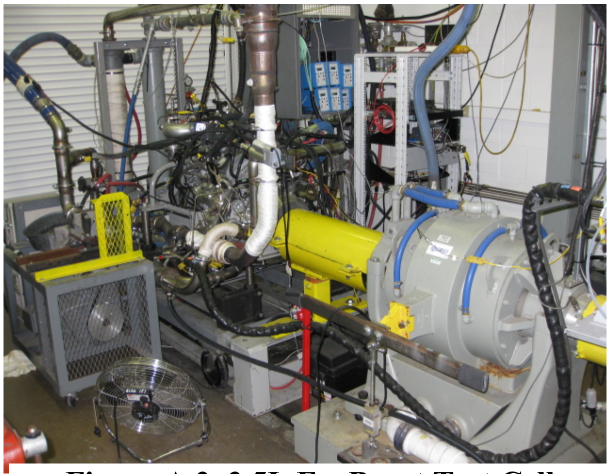


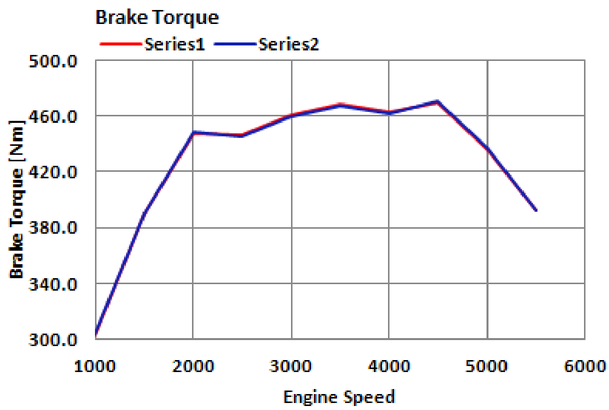
Figure A 2. 3.5L EcoBoost Test Cell

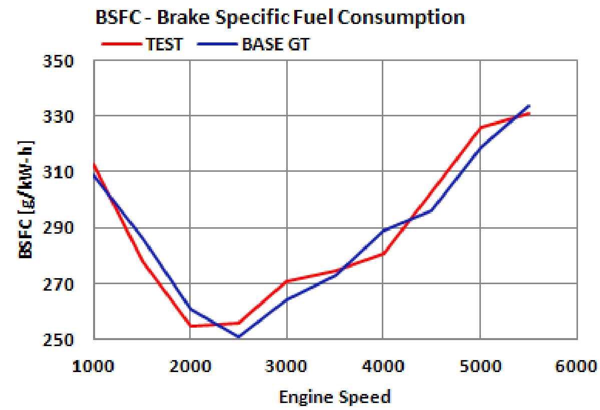
A GT Power model was built using the geometry of a 3.5L Ford EcoBoost engine currently installed in a SwRI test cell. The engine was fully instrumented, including temperatures, pressures, speed, torque and high speed cylinder pressure measurements. A test matrix that covered the entire engine operating range, consisting of 78 part and full load conditions, was run in the test cell. Engine data from the test cell, shown below, was used as model input data.

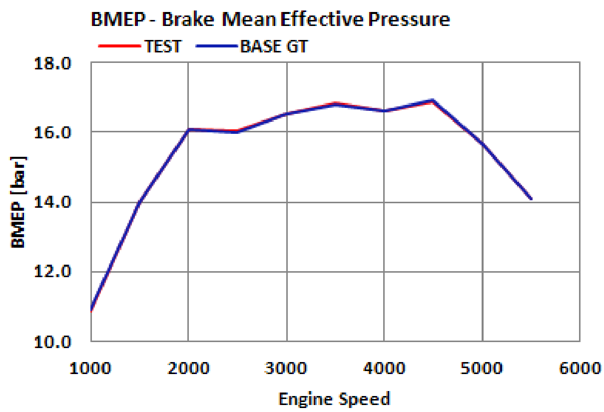
50% Mass Fraction Burned (CA50)
10-90 Burn Duration
Valve Flow Coefficients
FMEP
Cam Phaser Position
Air/Fuel Ratio

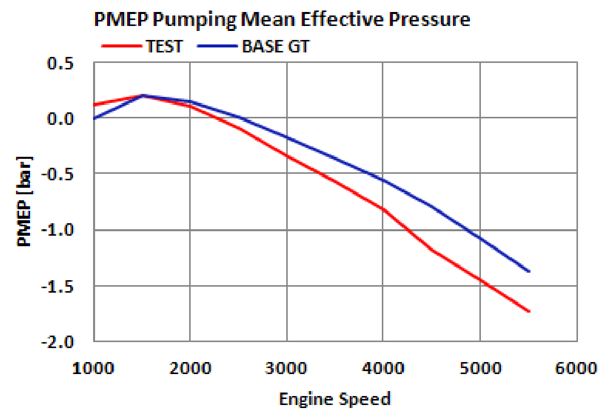
The GT Power model was then operated over the same 78-point test matrix to compare the following variables to the actual engine data. Figure A 3 shows the comparison between actual engine testing and GT Power model predictions. The following operating parameters were considered in the GT Power model validation:

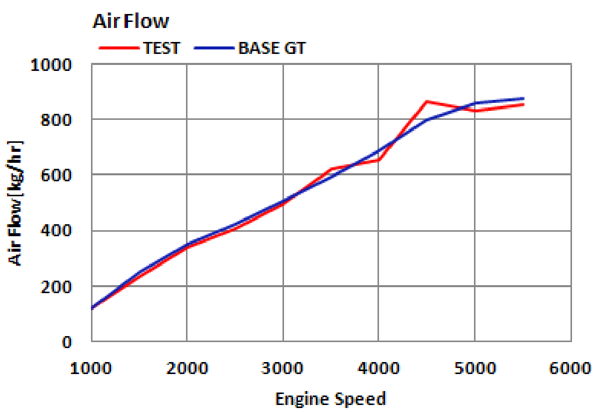
Brake Torque
BSFC
BMEP, IMEP, PMEP
Air and Fuel Flow
Pre/Post Throttle pressures and temperatures
Pre/Post Turbine pressures and temperatures
Combustion Phasing
Combustion Duration

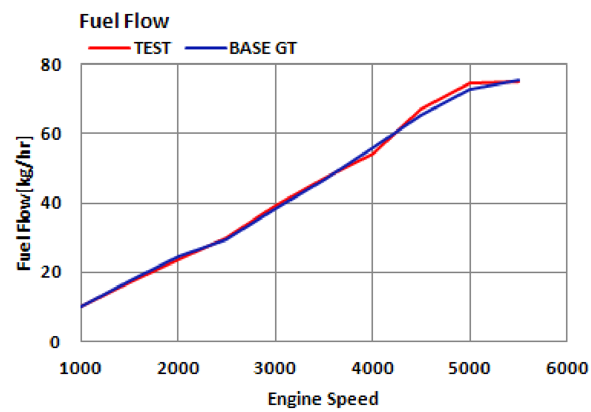

Brake Torque
Series1
Series2
Brake Torque [Nm]
500.0
460.0
420.0
380.0
340.0
300.0
1000
2000
3000
4000
5000
6000
Engine Speed

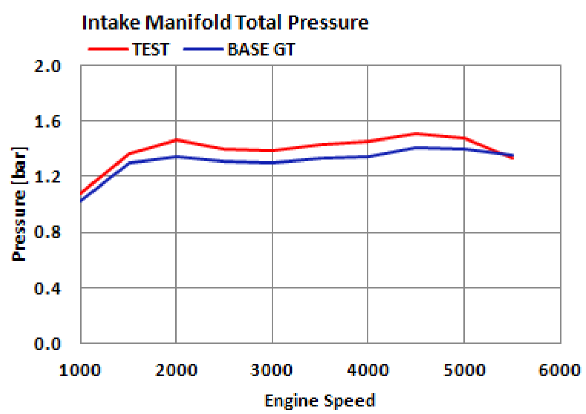

BSFC - Brake Specific Fuel Consumption
TEST
BASE GT
BSFC [g/kW-h]
350
330
310
290
270
250
1000
2000
3000
4000
5000
6000
Engine Speed

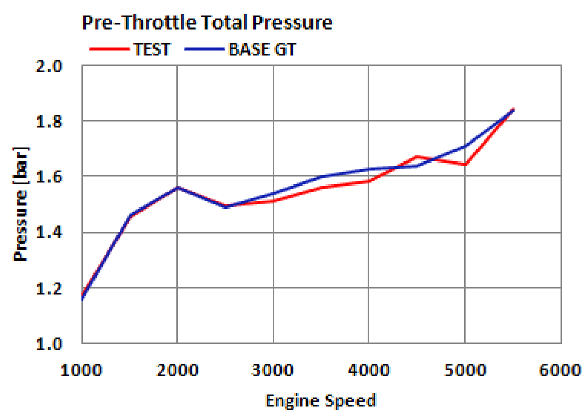

BMEP - Brake Mean Effective Pressure
TEST
BASE GT
BMEP [bar]
18.0
16.0
14.0
12.0
10.0
1000
2000
3000
4000
5000
6000
Engine Speed


PMEP Pumping Mean Effective Pressure
TEST
BASE GT
PMEP [bar]
0.5
0.0
-0.5
-1.0
-1.5
-2.0
1000
2000
3000
4000
5000
6000
Engine Speed


Air Flow
TEST
BASE GT
Air Flow [kg/hr]
1000
800
600
400
200
0
1000
2000
3000
4000
5000
6000
Engine Speed


Fuel Flow
TEST
BASE GT
Fuel Flow [kg/hr]
80
60
40
20
0
1000
2000
3000
4000
5000
6000
Engine Speed


Intake Manifold Total Pressure
TEST
BASE GT
Pressure [bar]
2.0
1.6
1.2
0.8
0.4
0.0
1000
2000
3000
4000
5000
6000
Engine Speed


Pre-Throttle Total Pressure
TEST
BASE GT
Pressure [bar]
2.0
1.8
1.6
1.4
1.2
1.0
1000
2000
3000
4000
5000
6000
Engine Speed

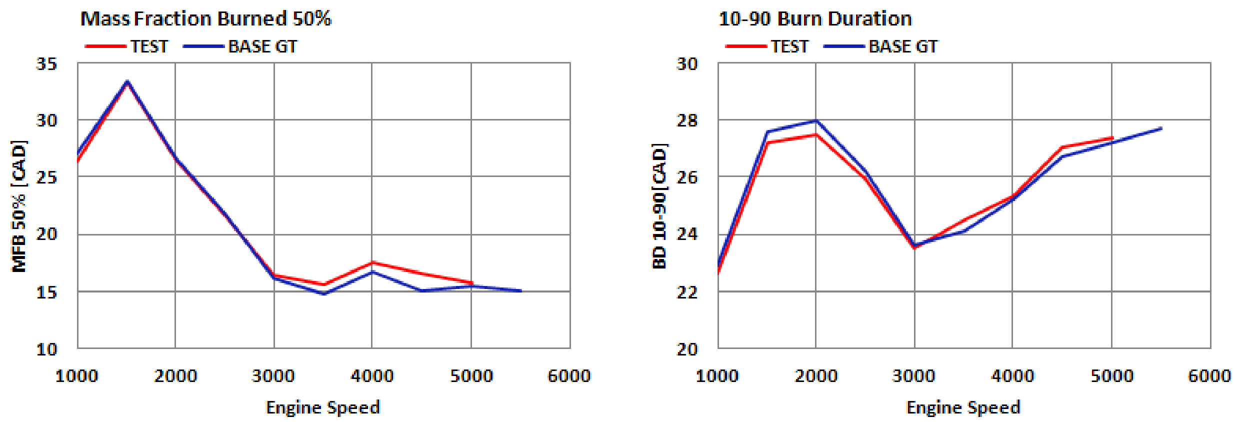


Figure A 3. Baseline Validation

Once the GT-POWER simulated conditions outside of the cylinder were close to the experimental data, the in-cylinder pressure data was compared to model data and the heat rejection was fine tuned until simulated and measured cylinder pressure shape and amplitude were similar for multiple speeds and loads.

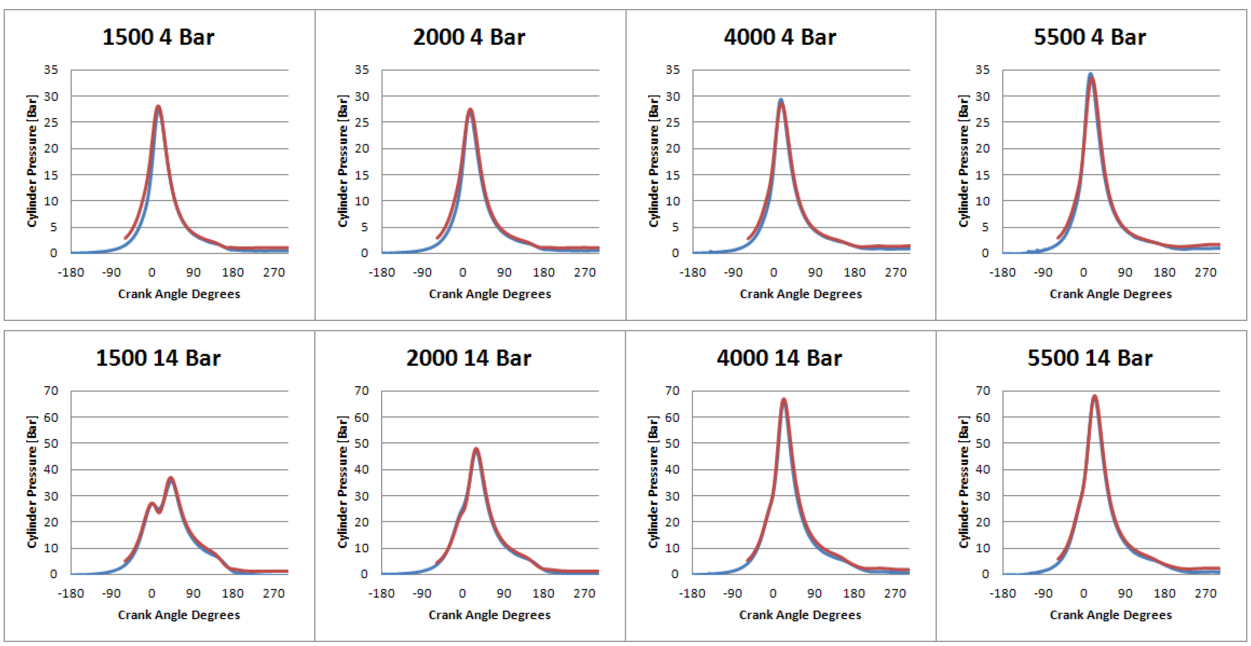


Figure A 4. In-Cylinder Pressure Data Comparison Between Simulation and Test

With the "matched" operating conditions, the 78 operating points were run again to confirm that the simulated BSFC was within +/- 3% of the experimental results. Shown below is the comparison between the engine data and model reported BSFC. Also shown in Figures A6-A7 are the full BSFC and equivalence ratio contours.

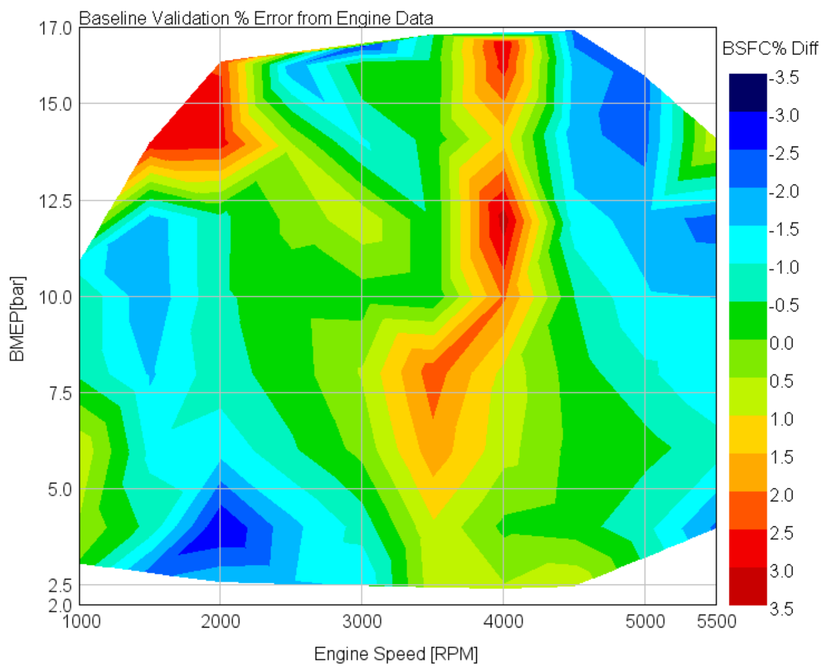


Figure A 5. BSFC Model Error

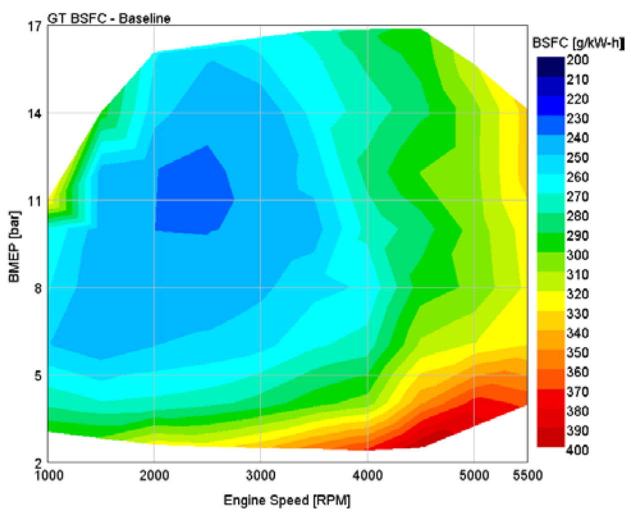


Figure A 6. Baseline Model BSFC

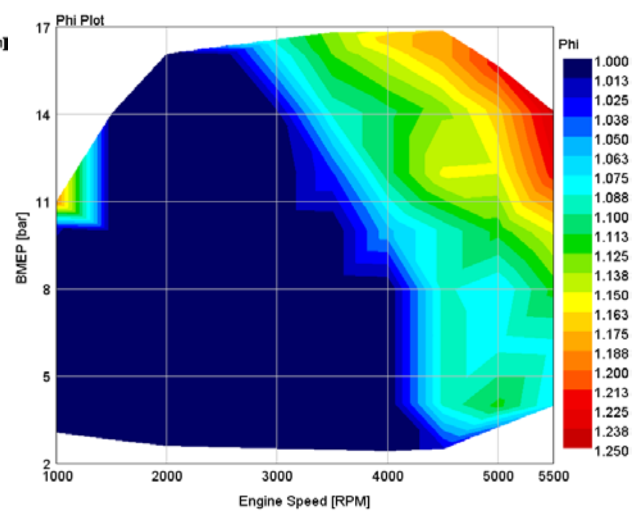


Figure A 7. Baseline Equivalence Ratio

1.2 Explore Variable Valve Lift (VVA/VVL)

- **Variable Valve Lift and Phasing**
  - Reduced pumping losses (reduced or eliminated throttling)
- **Additional Components include:**
  - Special cams and actuators to allow for variable valve lift
- **Variable Valve lift model modifications**
  - The engine throttle is removed and the load is controlled by changing the lift and duration of the intake valve
  - Valve lift for the 78 point test matrix shown in Figure A10 below
  - Low valve lift affects flow fields and might not be a realistic operating condition

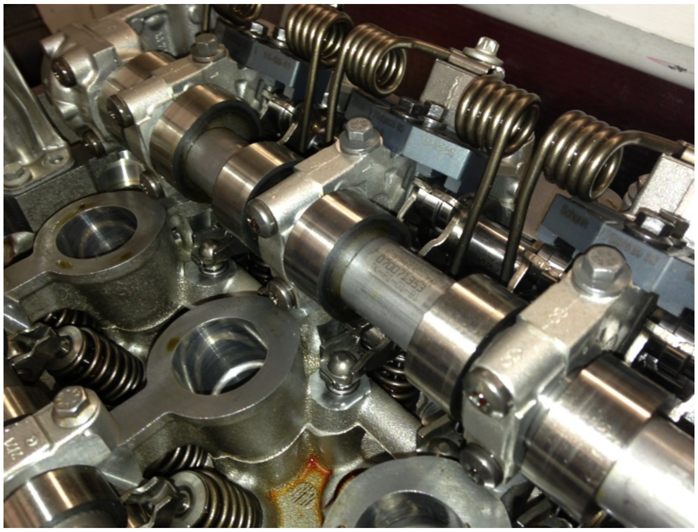


Figure A 8. VVL Actuators

- **Variable Valve lift Example**
  - Baseline Pumping loop, shown below in Figure A 9, shows the pumping work that is wasted under a "typical" throttled condition
  - Improved Pumping work from VVL, shown below in Figure A10

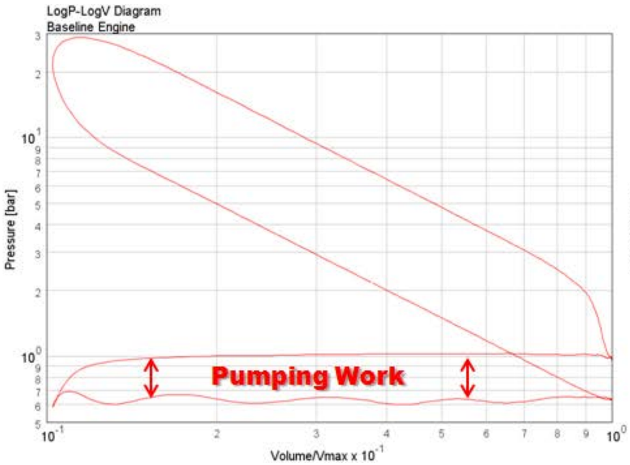


Figure A 9. Pumping Work

Figure A 10. Energy Recovery

Variable valve lift allows for improvement in efficiency during part load or "normally" throttled conditions. By closing the intake valve early and allowing the cylinders to pull a vacuum on the downward stroke, some of the energy is recovered as the piston comes back up.

This is more evident at higher engine speeds, and the benefit phases out as load increases. Special cam actuators are required to allow for variable lift and phasing as shown in the figures above. As the valve lift and duration are shortened, the in-cylinder charge motion may be affected negatively and cause unstable combustion at the low lift, or low load conditions. If the engine has two intake valves per cylinder, a potential solution might be to vary the lift on one valve much more than the other to keep the velocity high through at least one of the intake ports during low load operation. For the purposes of this model, charge motion was assumed not to be affected.

The variable valve lift model is a modified version of the baseline 3.5L V6. The throttle was removed and the load was controlled by the intake valve lift. Intake valve duration was a mathematical function of the lift that attempted to emulate a realistic VVL system. In addition to the variable valve lift, the intake and exhaust cams have independent phasing capability. To optimize fuel efficiency, more than 1400 intake and exhaust timing combinations were run for the 78 point test matrix. Shown to the right in Figure A 11, is the final intake and exhaust valve timing for the 78 speed/load conditions. Variable valve lift allows for improvement in BSFC at throttled conditions. The benefit is more evident at higher engine speeds, and it phases out as load increases. Shown below are the BSFC and BSFC improvement over the baseline engine contour maps.

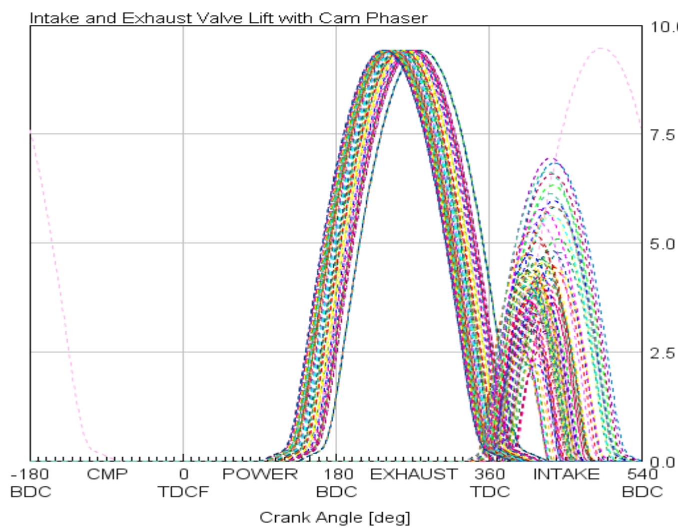


Figure A 11. 3.5L V6 VVL Cam Phasing

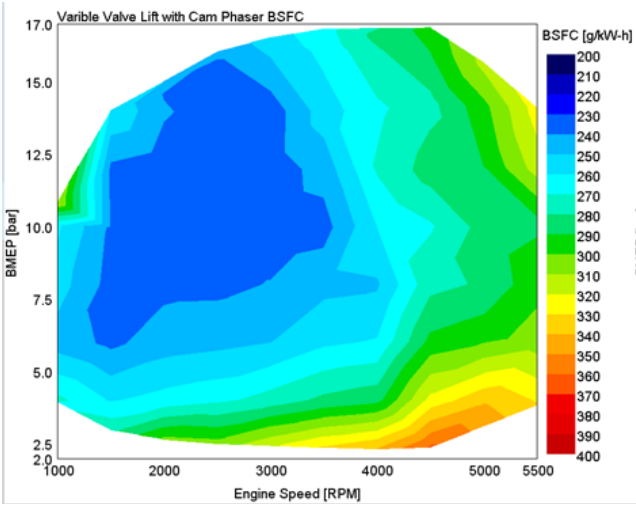


Figure A 12. Variable Valve Lift BSFC

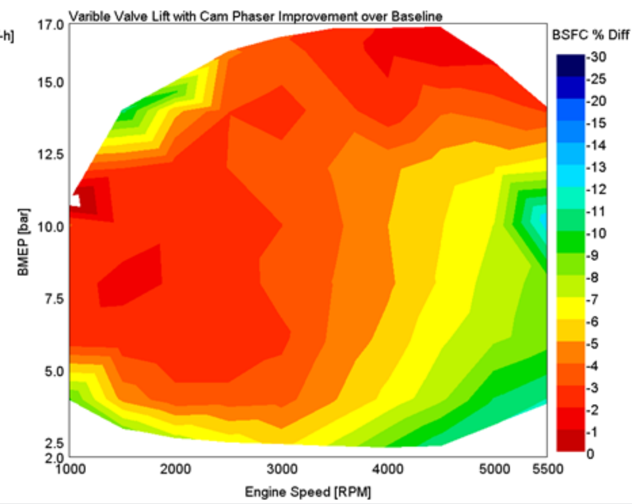


Figure A 13. BSFC Improvement over Baseline

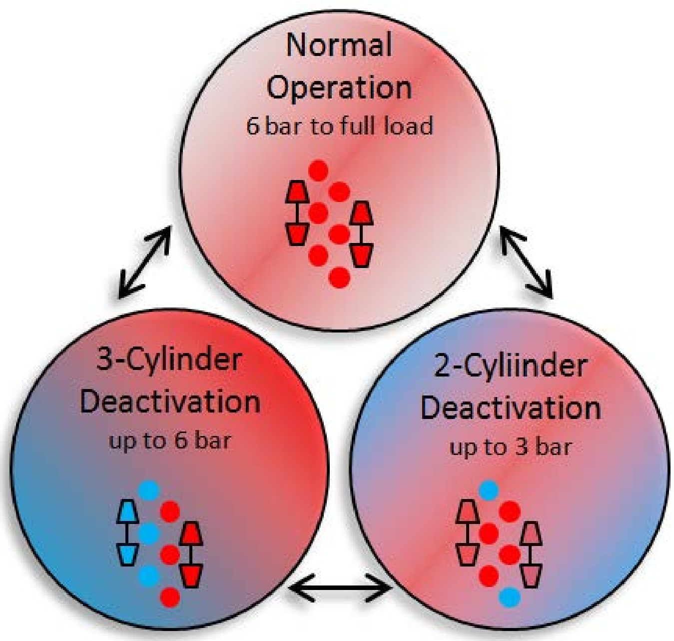


Figure A 14. Cylinder Deactivation Method

1.3 Explore Cylinder Deactivation

- **Two and Three Cylinder Deactivation**
  - Reduced pumping losses (less throttling)
  - Reduced energy loss from deactivated valves
  - Applicable for loads up to 6 bar BMEP
- **Additional Components include:**
  - Special cams and actuators to allow for valve deactivation
  - Active Engine mounts/dampers

Engines that have six cylinders have demonstrated operation with two and three cylinder deactivation, such as Honda's Variable Cylinder Management (VCM). The engine at idle conditions will operate on all six cylinders to minimize vibration, but between idle and three bar BMEP, three cylinders are deactivated by cutting fuel injection and valve lift. To ensure adequate throttle response, compressor outlet pressures were maintained. The model was run up to the maximum load that three-cylinder combustion would allow, nine bar BMEP. (see Figure A 15) Then the operating areas that were less efficient than the baseline six-cylinder were removed. (see Figure A 16)

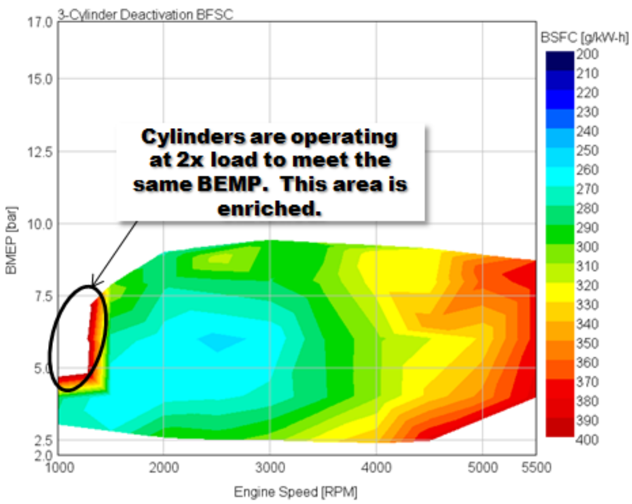


Figure A 15. Full load Three-Cylinder

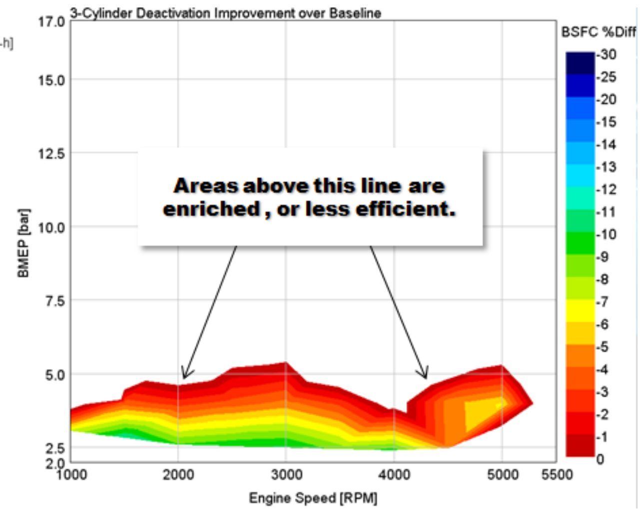


Figure A 16. Potential Operating Range

Because this engine has one turbocharger per bank of cylinders, when 3 cylinders are deactivated, one full bank is shut down. This approach allows normal turbocharger operation on the functioning cylinder bank as shown below in Figure A 17.

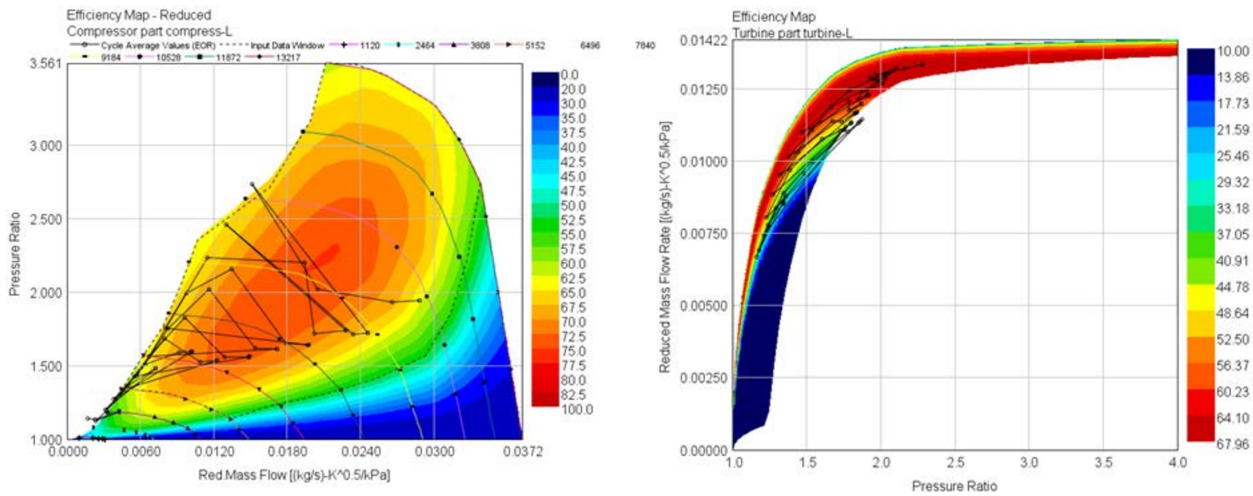


Figure A 17. Turbo Operation in Three-Cylinder Mode (Compressor on left, turbine on right.)

As load increases above three bar BMEP, an additional cylinder will be reactivated allowing four active cylinders to reach engine loads of up to six bar BMEP. In addition, to ensure original throttle response, compressor outlet pressures were maintained at the baseline levels. The model was run up to the maximum load the four-cylinder combustion would allow which was 13 bar BMEP (see Figure A 18). The areas of 4-cylinder operation that were less efficient than the baseline six-cylinder operation were removed. (see Figure A 19)

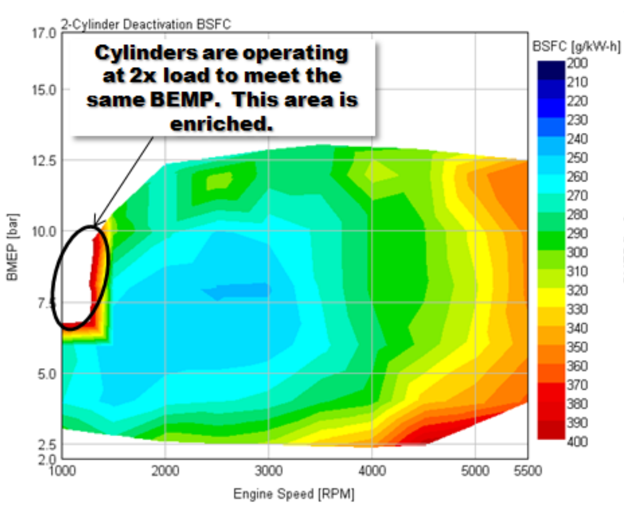


Figure A 18. Full load 4-Cylinder

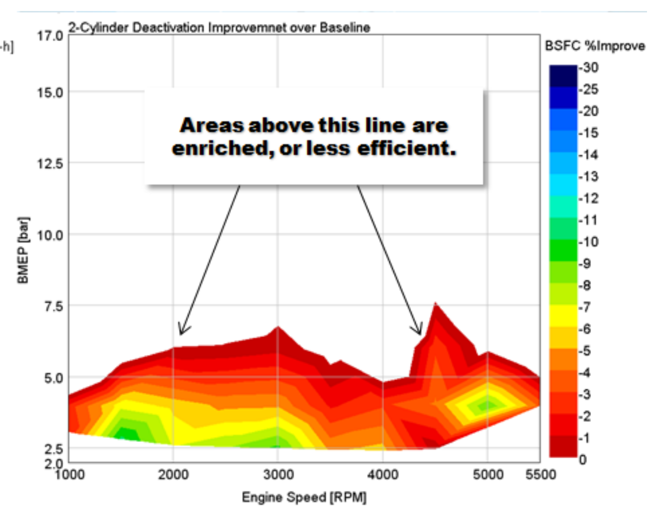


Figure A 19. Potential 4-Cyl. Operating Range

The four-cylinder combustion mode deactivates one cylinder per bank and cuts the mass flow to each turbo by one third. The overall operating points on the turbo maps are shifted to the left, which pushes some of the high engine load points into compressor surge. This is not a problem in four cylinder operation, since the engine will not operate at these high load points in four-cylinder combustion mode due to lower efficiency.

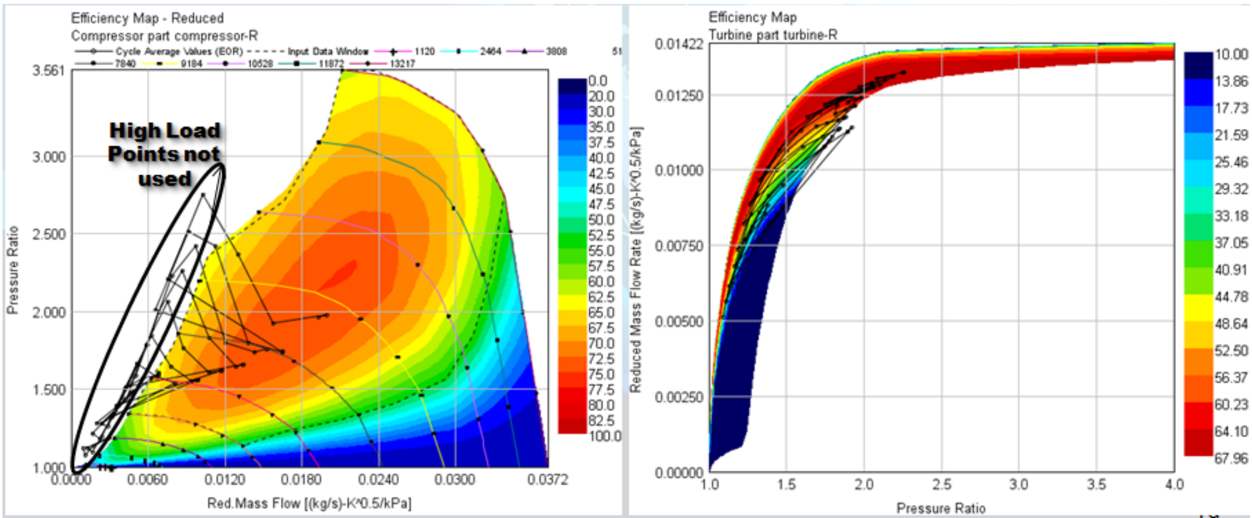


Figure A 20. Turbo Operation in 4-Cylinder Mode. Compressor map (left), Turbine map (right)

The engine efficiencies were compared for the three-cylinder, four-cylinder, and six-cylinder combustion modes and a composite operating range of best efficiencies was compiled. Figures A 21 - A 23 below, shows the engine operating ranges, BSFC, and percent BSFC improvement over the baseline engine.

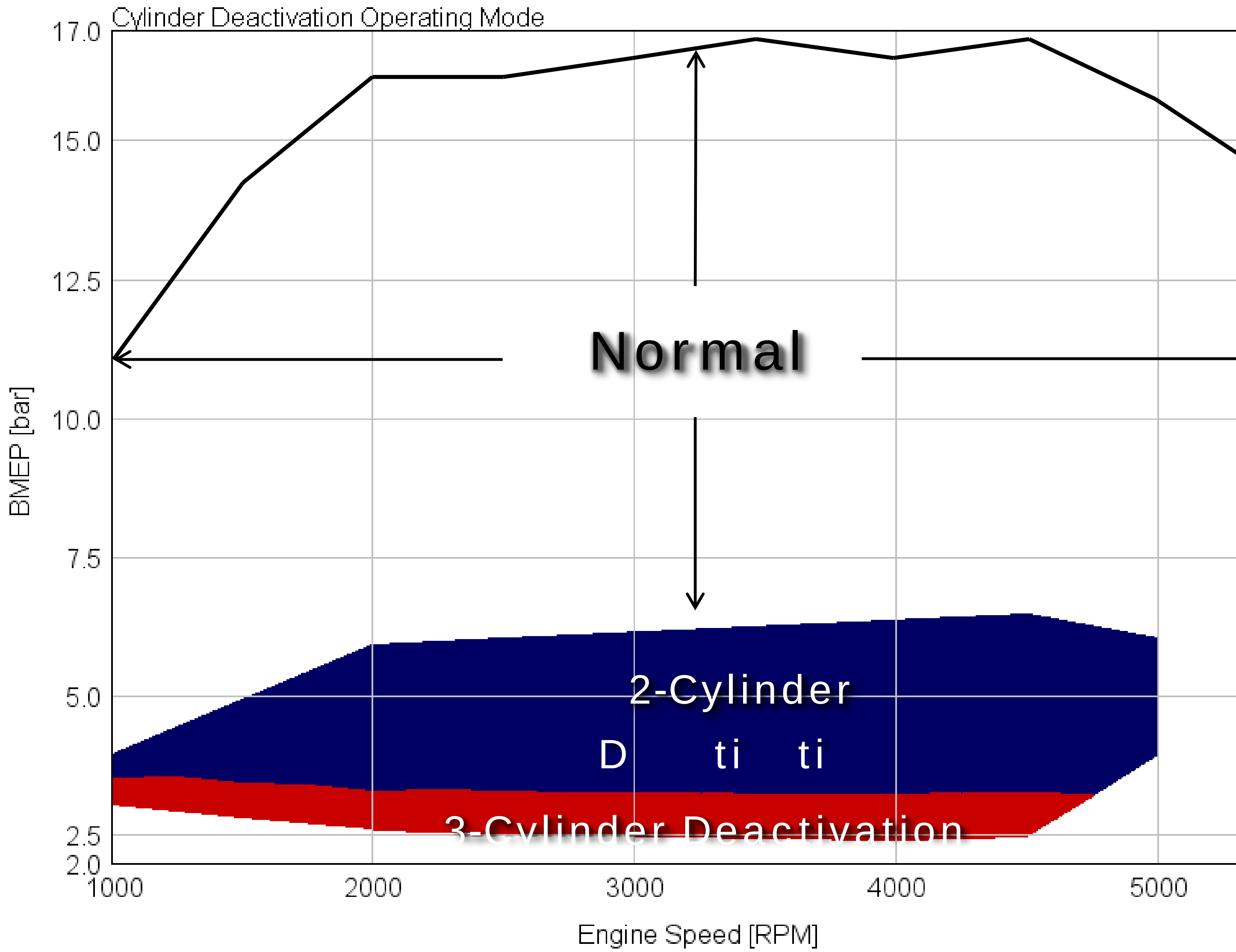
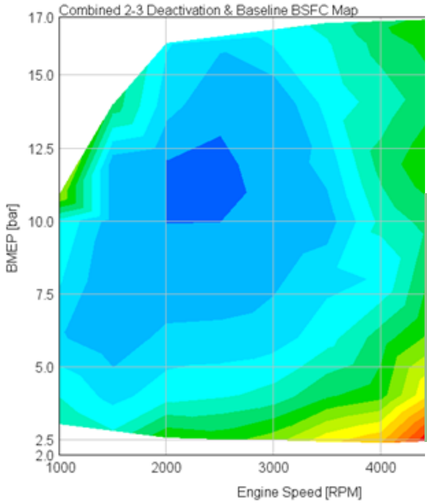


Figure A 21. Composite Engine Operating Range



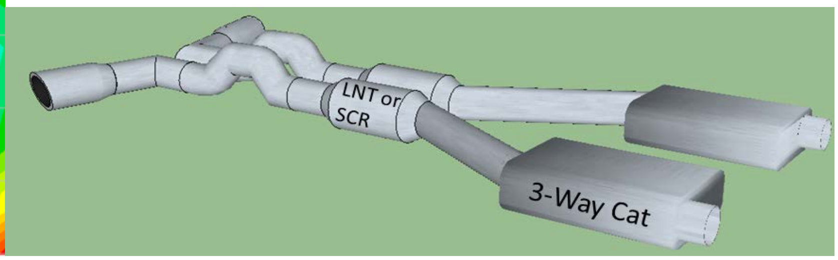


Figure A 22. Combined BSFC Map

Figure A23. BSFC Improvement over Baseline

1.4 Lean Burn GDI

- Lean Burn Benefits
  - Reduced pumping losses
  - Improved combustion efficiency
  - Improved working fluid
  - Reduced heat transfer
- **Additional Components include:**
  - Piezo Fuel Injectors
  - NOx Sensor
  - NOx Trap
  - Exhaust Gas Temperature Sensor

Lean burn operation can improve engine efficiency by reducing throttling losses, increasing combustion efficiency due to excess oxygen and by improving gamma of the working fluid. In the stratified operating area, the majority of improvement comes from a pumping work improvement and reduced heat transfer due to cooler in-cylinder temperatures. In addition, the homogeneous lean area benefits some due to pumping work and greatly in the areas where the baseline engine utilized enrichment.

The improvement shown from the GT Power simulation show favorable improvements in its steady state operation, however, it does not show the fuel penalty for the regeneration of the LNT or account for the Urea needed in an SCR. In addition to these penalties, the location of the NOx aftertreatment would have to be located sufficiently far downstream to minimize the inlet temperatures. This distance will help the durability of the aftertreatment, but another fuel penalty will be added at startup until the NOx aftertreatment is up to operating temperature. During this time, the engine will operate in stoichiometric mode to allow the three-way catalyst to operate. These fuel penalties can increase fuel comsumption on the drive cycle and potentially remove any of the benefits of lean operation.

To operate lean, additional components, such as a lean NOx traps or SCR systems, NOx sensors and Piezo fuel injectors for precise multiple injections will be needed. In order for this vehicle to be successful with current and future emission standards, a high NOx conversion efficiency will be required. For high LNT conversion efficiencies, relatively low space velocities are required; therefore a very large volume LNT will be required for conversion of NOx at high load, resulting in a costly LNT. For high SCR conversion efficiencies, high urea dosing rates will be required.

Some of the potential issues with operating in lean mode are degradation of the NOx aftertreatment at high load/temperature conditions and using US fuels with high sulfur content.

The pre-turbine temperature limit is 1223 K for the baseline engine and the maximum obtained during homogeneous operation was 1200 K. Ideally, the maximum temperature at NOx aftertreatment inlet would be less than 1050 K to avoid premature aging and less than 850 K for good conversion efficiency. A certain amount of heat loss will occur across the turbine and piping, but heat addition can take place at the catalyst. Since modeling of heat loss in the aftertreatment system was outside the scope of the project, the pre-turbine temperature limit was allowed to reach 1200 K and the lean operation limits were chosen based upon typical operating ranges from literature. [1, 2] It was assumed that a suitable location for the NOx aftertreatment that met inlet temperature requirements would be used for all vehicles.

High sulfur content will lead to sulfur poisoning and frequent desulfation of a LNT, which will lead to a shortened life of the aftertreatment system. There is the potential that future US gasoline fuel quality specifications will reduce sulfur content, and thus minimize the damage caused by sulfur in the fuel, but that is not presently the case.

This engine model configuration has three operating modes, stratified lean mixture, homogeneous lean mixture, and homogeneous stoichiometric to rich mixture. Shown below in Figure A 25, the engine operates at low speed/load, medium speed/load, and high load in stratified lean, homogeneous lean, and homogeneous stoichiometric modes respectively. Near full load, the engine operates rich, similar to a conventional gasoline engine.

Due to limited published engine data and inability to run stratified charge on our test engine, combustion phasing was assumed to be the same as the baseline operation. Stratified charge burn durations were chosen to be the same as the homogenous lean engine data collected on the test engine. Homogeneous lean and stoich combustion phasing/duration data was collected from the test engine. It should be noted that efficiency is a strong function of combustion phasing and a weak function of burn duration.

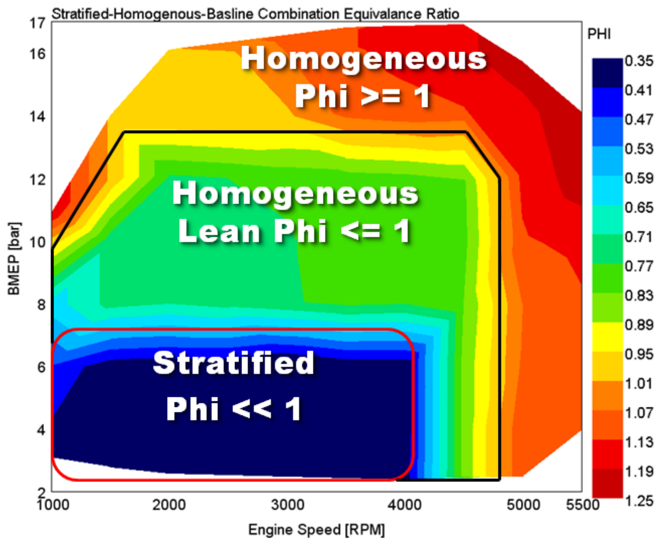


Figure A25. Engine Operating Modes

The engine efficiencies for the three combustion modes were combined and a composite engine operating map was generated. Figure A - A 27, show the composite engine BSFC, and percent BSFC improvement over the baseline engine.

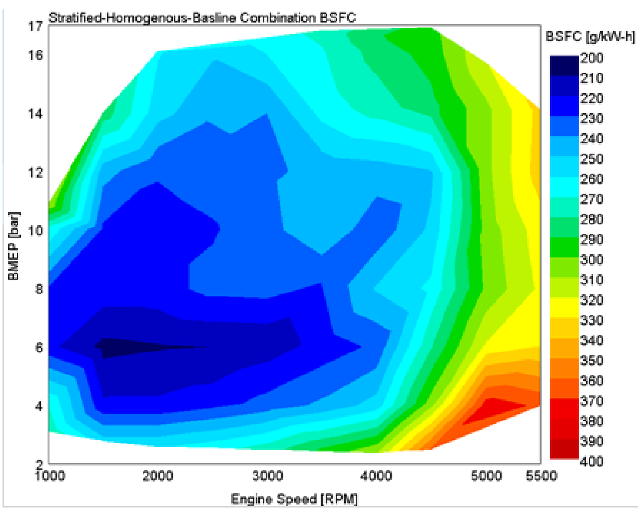


Figure A 26 Combined BSFC Map

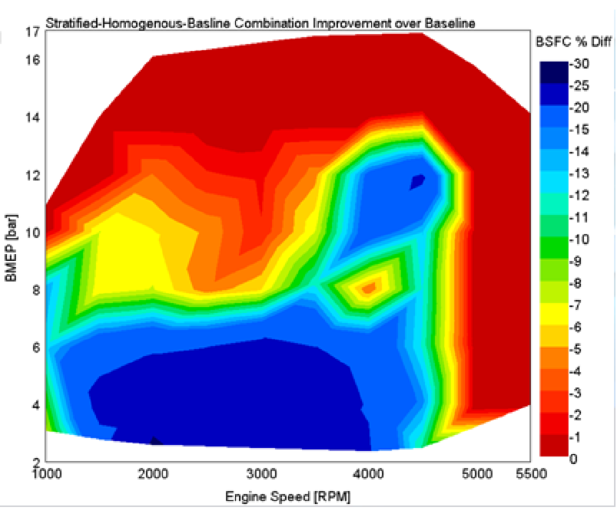


Figure A 27. BSFC Improvement over Baseline

1.5 Explore GDI with cooled EGR (HEDGE)

- **EGR Benefits**
  - Reduced pumping losses
  - Improved working fluid
  - Improved knock tolerance
    - Improved combustion phasing
    - Increased compression ratio
  - Reduce heat transfer
  - Eliminate enrichment for catalyst protection
- **Additional Components include:**
  - EGR Valves
  - EGR Cooler
  - High Energy Ignition

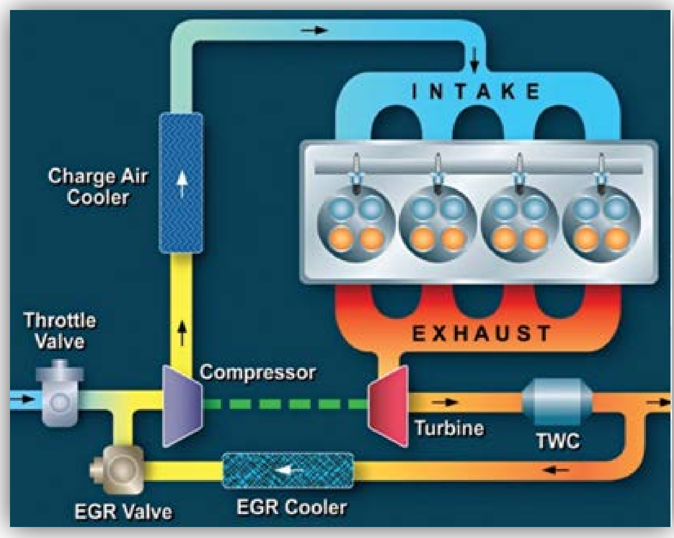


Figure A 28. Typical LPL-EGR Configuration

SwRI's High Efficiency Dilute Gasoline Engine (HEDGE) consortium has proven that engines with exhaust gas recirculation (EGR) show large benefits in efficiency and emissions. EGR engines require an EGR cooler and valve to recirculate the exhaust gasses, and a high energy ignition system to ignite the dilute mixture. [3]

A model based on the baseline engine model was developed to represent a low pressure loop exhaust gas recirculation (LPL-EGR) engine. EGR can improve efficiency in several ways. It enables full-map stoichiometric operation, improves the ratio of specific heats of the working fluid, lowers heat loss in-cylinder due to reduced combustion temperatures and reduces knock tendency which allows more favorable combustion phasing and/or an increase in compression ratio. All of these effects can be observed in the test cell and in GT Power.

Engine parameters, such as CA50, MFB 10-90, compression ratio, EGR rates, and cam timing were selected for this engine model based upon SwRI's experience in converting and calibrating EGR engines. [4,5,6,7] It is understood that while the values will not be exact, the values chosen are based upon the operating characteristics of the baseline engine and should be a close representation of an actual EGR conversion. The compression ratio was increased to 11.5:1, and trapped residuals were kept below 38 percent to ensure robust combustion. Appropriate EGR valves, coolers and controls were added to the model. Combustion phasing (CA50 and 10-90 burn duration) and EGR rates were adjusted to match SwRI experience with LPL-EGR engines and some limited data generated on the 3.5L EcoBoost test cell engine. The CA50 map was modified from the baseline to change any value that was less than or equal to 13 degrees after top dead center on the firing stroke (aTDCf) to a Maximum Brake Torque timing (MBT) value of 8 degrees aTDCf. The remaining area of the map advanced the timing to 50% of the angle between MBT and the baseline CA50 value. For example, if the base CA50 is 16 deg aTDCf and MBT is 8 deg, then (Base CA50 - MBT) x 50% + MBT = (16-8)x0.5+8 = 12 deg aTDCf. The 10-90 burn duration was uniformly lengthened by 10 percent due to the slower burning EGR mixture.

EGR flow rates were defined as follows: at low speed and light load, 12% EGR is used. At high speed light load, 15 % EGR is used. Under full load, EGR is 15% at low speed, increasing to 18% at high speed. Linear interpolation is used to determine the EGR rate for any point in the map.

Intake cam phasing was altered from baseline cam timing to an EGR optimized cam timing based upon SwRI's EGR experience. At low loads more cam overlap is used with normal to late intake valve closing. Mid loads used low overlap and later intake valve closing. High loads operated at best volumetric efficiency positions with normal to high overlap and normal intake valve closing.

Engine BSFC, and percent BSFC improvement over the baseline engine are shown below in Figures A 29- A 30.

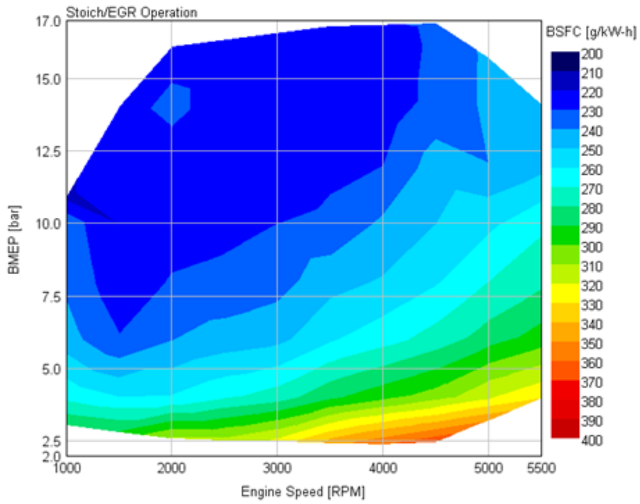


Figure A 8. Stoichiometric EGR BSFC Map

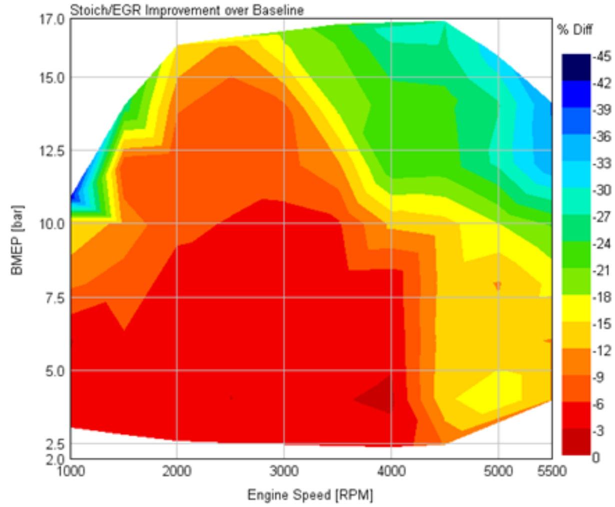


Figure A 30. BSFC Improvement over Baseline

1.6 EGR EGR Engine Down speeding with increased BMEP and PCP

- **Down speed Benefits**
  - Reduced pumping losses
- **Additional Components include:**
  - EGR Valve
  - EGR Cooler
  - High Energy Ignition
  - Improved cylinder pressure capability

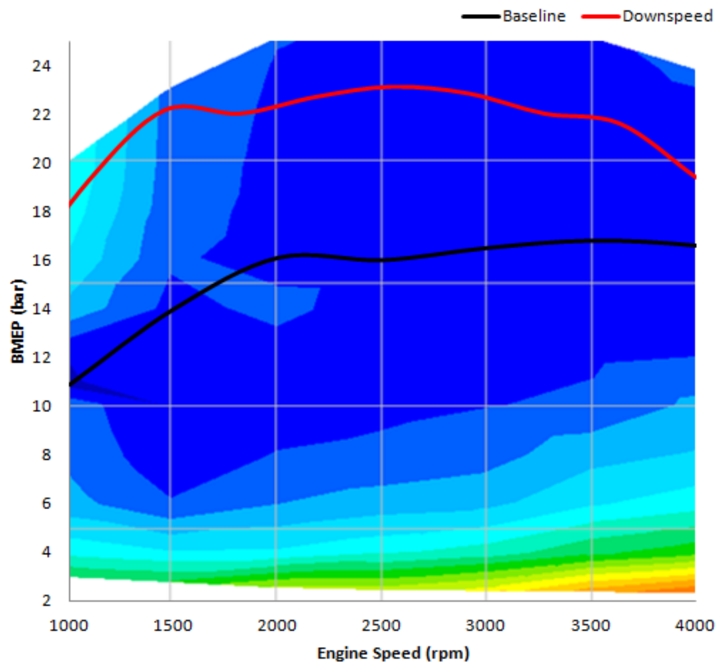


Figure A 31. BMEP Comparison of Standard and Down Speed Torque Curves

A model based on the LPL-EGR engine model was developed to down speed the engine from 5500 rpm to 4000 rpm while maintaining the same peak power. To accomplish this, the BMEP requirement of the engine becomes greater as demonstrated by the red line in Figure A 31 above. The model was allowed to run to even higher loads, (areas above red line) but the final torque curve and fuel map was limited to keep the power levels the same as the baseline and to keep the maximum cylinder pressures below 105 bar.

1.7 Explore FMEP Improvements

Reducing engine friction can be achieved in several ways, such as low viscosity oils, smaller or dynamic bearings, as well as smart controlled oil and water pumps. The modeling effort for this portion lumped together the friction items and reduced their value by 10 percent from the baseline model.

The baseline engine model with the reduced friction was run over the same 78 point test matrix and the BSFC and BSFC improvement over baseline can be seen in the plots below. As expected, the largest benefits occur at light load and higher engine speeds.

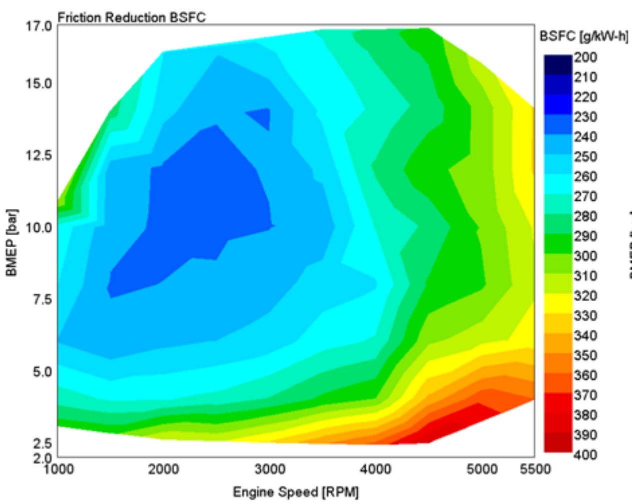


Figure A 32. Friction Reduction BSFC

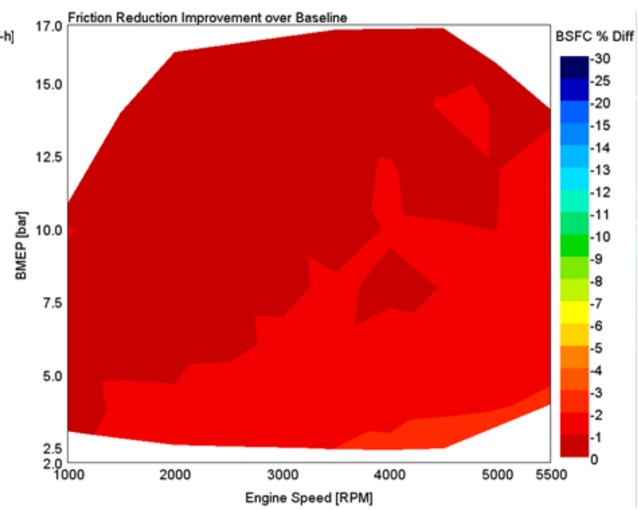


Figure A 33. BSFC Improvement over Baseline

1.8 Explore turbo efficiency improvement

Turbo Efficiency Improvement Benefits

- Reduced pumping losses
- Improved waste heat recovery

- **Additional Components include:**
  - Potentially more expensive turbo

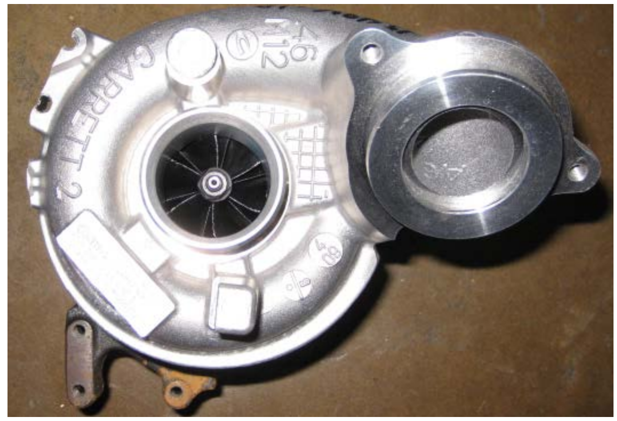


Figure A 34. Example turbocharger hardware

Improvements in turbo designs, bearings, and materials are leading to more efficient and robust turbochargers. An engine model based upon the baseline engine model was built to examine a five percent improvement in turbocharger compressor efficiency. In reality, a waste gated turbocharger, like the one on the 3.5L EcoBoost, operating with improved efficiency, would require additional waste gating to maintain the same engine power. To compensate for the five percent compressor efficiency increase, the turbine size was increased by six percent in the model such that the same amount of waste gating occurred. The baseline engine model with improved turbocharger efficiency was run over the same 78 point test matrix and the BSFC and BSFC improvement over baseline can be seen in the plots below in Figures A 35- A 36.

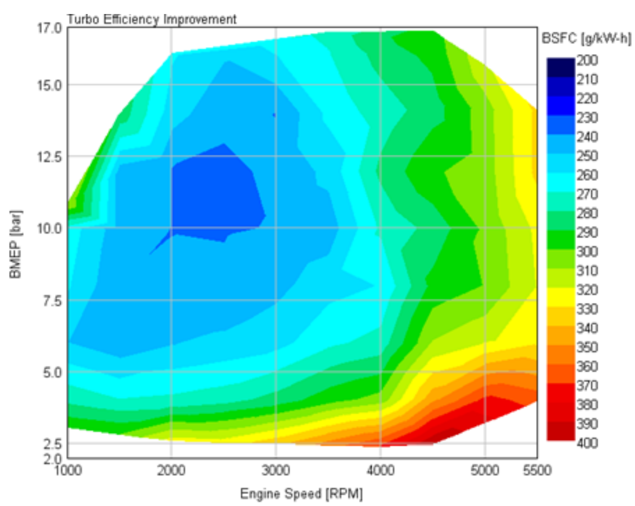


Figure A 35. Turbo Efficiency BSFC Map

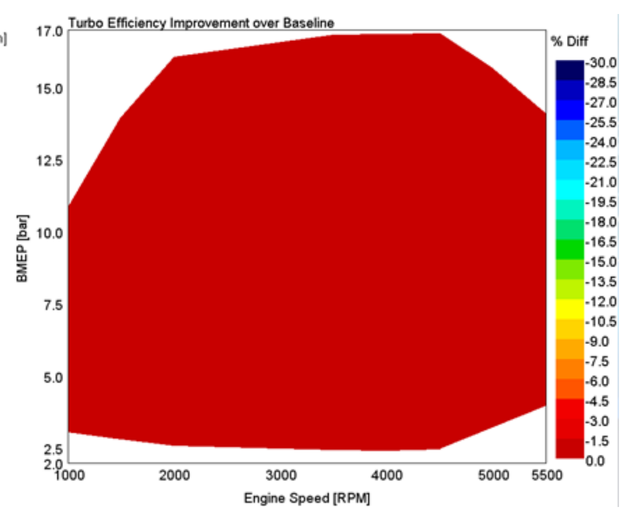


Figure A 36. BSFC Improvement over Baseline

2.1 Baseline V8 Validation

- **OEM V8**
  - 6.2L V8
  - 9.8:1 compression ratio
  - Naturally Aspirated
  - Single Overhead Cam w/Phaser
  - 2 valves per cylinder
  - Gasoline
  - 287/236 kW (<10K/>10K GVWR)
  - 550/538 Nm (<10K/>10K GVWR)

A model of a 6.2L V8 was built with the help and input parameters from an engine OEM. To validate the engine model, 82 part and full load test points were run on an instrumented engine at the OEM facilities and supplied to Southwest Research Institute. Similar to the 3.5L, the model inputs are shown below:

1. **50% Mass Fraction Burned**
2. **10-90 Burn Duration**
3. **Valve Flow Coefficients**
4. **FMEP**
5. **Cam Phaser Position**
6. **Air/Fuel Ratio**

The GT Power model was then run over the same 82-point test matrix and validated by the following operating parameters:

1. **Pre/Post Throttle pressures and temperatures**
2. **Air and Fuel Flow**
3. **BSFC**
4. **BMEP, IMEP, PMEP**

With the "matched" operating conditions, the 82 operating points were run again to confirm that the BSFC calculated by GT-POWER was within +/- 3% of the experimental data. Shown in Figure A 37 below is the comparison between the engine data and model reported BSFC. Also shown in Figures A 38- A 39 are the full BSFC and equivalence ratio contour.

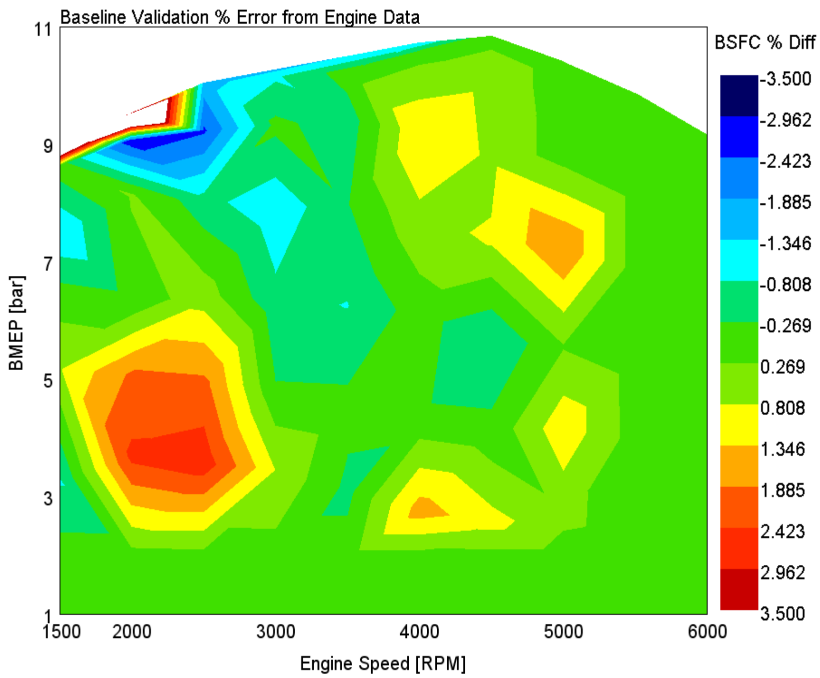


Figure A 37. BSFC Model Error

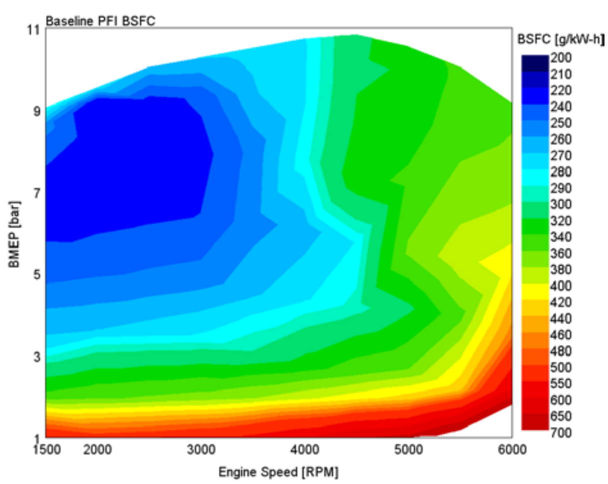


Figure A 38. Baseline Model BSFC Map

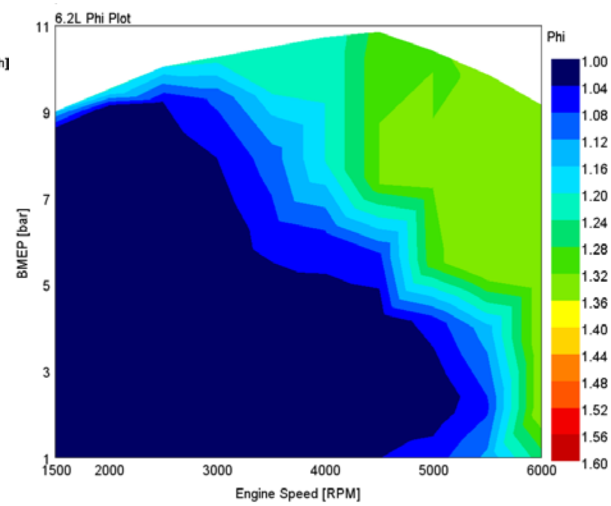


Figure A 39. Baseline Equivalence Ratio

2.2 Stoichiometric

Figure A 40. Gasoline Direct Injector

- **GDI conversion**
  - Allows for in-cylinder charge cooling
  - Better Fuel control
  - Higher power output is available (but not used in this project)
  - Allows higher compression ratio
- **Additional Components include:**
  - High pressure pump
  - High pressure injectors
  - High pressure fuel lines
  - High voltage electronics

Gasoline direct injection is typically found on boosted engines with increased power density or lean burn engines that need precise fuel control. In the case of this naturally aspirated V8 engine, a higher compression ratio and improved combustion phasing might be possible, offset with a penalty of degraded combustion efficiency and increased parasitic power losses for the high pressure fuel pump. The baseline GT power model was modified to represent a GDI engine. Port fuel injectors (PFI) were replaced with GDI injectors, compression ratio was increased by 1.5 points, combustion efficiency was reduced by two percent. There has been continual improvement in GDI fuel systems, (i.e. injector relocation from side to central) to improve combustion efficiency. Direct comparisons of PFI and GDI show a 10-90 % increase in CO emissions throughout the engine operating range, indicating a reduction in combustion efficiency with GDI. [8] For this study, a conservative average of combustion efficiency reduction was taken across the operating range. GDI fuel pump loads were calculated based upon engine conditions and the required fuel flow and typical pressures. This power was directly removed from the crankshaft in the GT Power model.

The baseline engine model with gasoline direct injection was run over the same 82 point test matrix and the BSFC and BSFC improvement over baseline can be seen below in Figures A 41 - A 42.

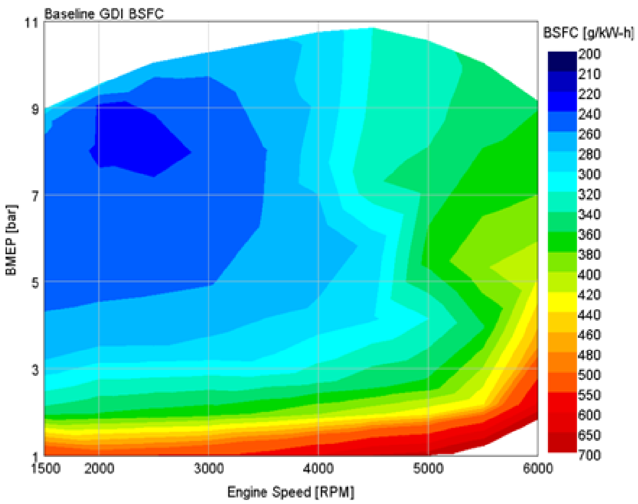


Figure A 41. GDI BSFC Map

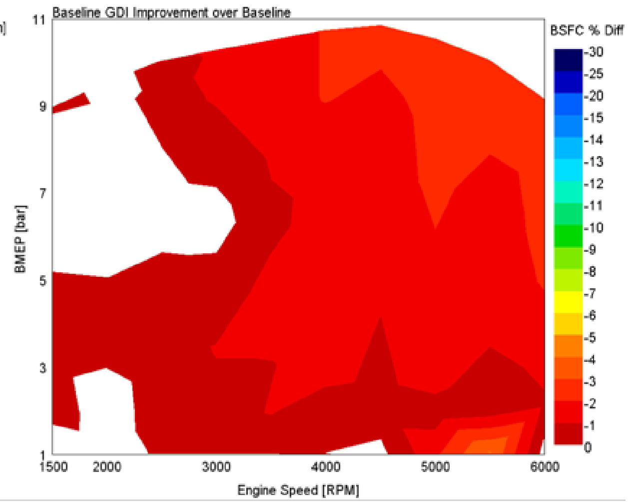


Figure A 42. BSFC Improvement over Baseline

2.3 Lean Burn GDI w/SCR

- **GDI conversion**
  - Allows for in-cylinder charge cooling
  - Better Fuel control
  - Higher power output
  - Reduced pumping losses
  - Improved combustion efficiency
- **Additional Components include:**
  - High pressure pump
  - High pressure injectors
  - High pressure fuel lines
  - Higher voltage electronics
  - NOx Trap or SCR Catalyst
  - NOx Sensor
  - Urea/Injector/mixer
  - Exhaust Gas Temperature Sensor

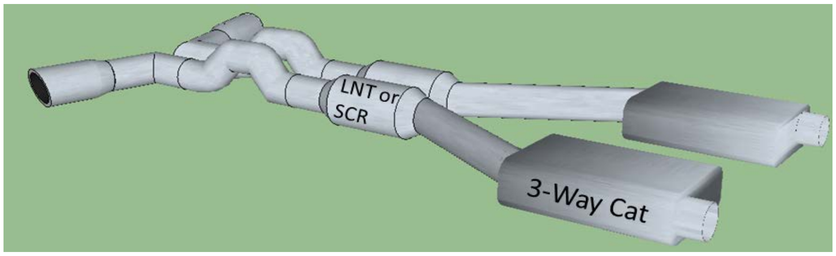


Figure A 43. Lean Aftertreatment

Lean burn operation can improve engine efficiency by reducing throttling losses, increasing combustion efficiency due to excess oxygen and by improving gamma of the working fluid. In the stratified operating area, the majority of improvement comes from a pumping work improvement and reduced heat transfer due to cooler in-cylinder temperatures. In addition, the homogeneous lean area benefits some due to pumping work and greatly in the areas where the baseline engine utilized enrichment.

The improvement shown from the GT Power simulation show favorable improvements in its steady state operation, however, it does not show the fuel penalty for the regeneration of the LNT or account for the Urea needed in an SCR. In addition to these penalties, the location of the NOx aftertreatment would have to be located sufficiently far downstream to minimize the inlet temperatures. This distance will help the durability of the aftertreatment, but another fuel penalty will be added at startup until the NOx aftertreatment is up to operating temperature. During this time, the engine will operate in stoichiometric mode to allow the three-way catalyst to operate. These fuel penalties can increase fuel comsumption on the drive cycle and potentially remove any of the benefits of lean operation.

To operate lean, additional components, such as a lean NOx traps or SCR systems, NOx sensors and Piezo fuel injectors for precise multiple injections will be needed. In order for this vehicle to be successful with current and future emission standards, a high NOx conversion efficiency will be required. For high LNT conversion efficiencies, relatively low space velocities are required, therefore a very large volume LNT will be required for conversion of NOx at high load, resulting in a costly LNT. For high SCR conversion efficiencies, high urea dosing rates will be required.

Some of the potential issues with operating in lean mode are degradation of the NOx aftertreatment at high load/temperature conditions and using US fuels with high sulfur content. The exhaust temperature limit is 1175 K for the baseline engine and the maximum obtained during homogeneous operation was 1175 K. Ideally, the maximum temperature at NOx aftertreatment inlet would be less than 1050 K to avoid premature aging and less than 850 K for good conversion efficiency. A certain amount of heat loss will occur across the turbine and piping, but heat addition can take place at the catalyst. Since modeling of heat loss in the aftertreatment system was outside the scope of the project, the pre-turbine temperature limit was allowed to reach 1200 K and the lean operation limits were chosen based upon typical operating ranges from literature. [1, 2] It was assumed that a suitable location for the NOx aftertreatment that met inlet temperature requirements would be used for all vehicles. High sulfur content will lead to sulfur poisoning and frequent desulfation of a LNT, which will lead to a shortened life of the aftertreatment system. There is the potential that future US gasoline fuel quality specifications will reduce sulfur content, and thus minimize the damage caused by sulfur in the fuel, but that is not presently the case.

This engine model configuration has three operating modes, stratified lean mixture, homogeneous lean mixture, and homogeneous stoichiometric to rich mixture. Shown below in Figure A 43, the engine operates at low speed/load, medium speed/load, and high load in stratified lean, homogeneous lean, and homogeneous stoichiometric modes respectively. Near full load, the engine operates rich, similar to a conventional gasoline engine.

Due to limited published engine data and inability to run stratified charge on our test engine, combustion phasing was assumed to be the same as the baseline operation. Stratified charge burn durations were chosen to be the same as the homogenous lean engine data collected on the test engine. Homogeneous lean and stoich combustion phasing/duration data was collected from the test engine. It should be noted that efficiency is a strong function of combustion phasing and a weak function of burn duration.

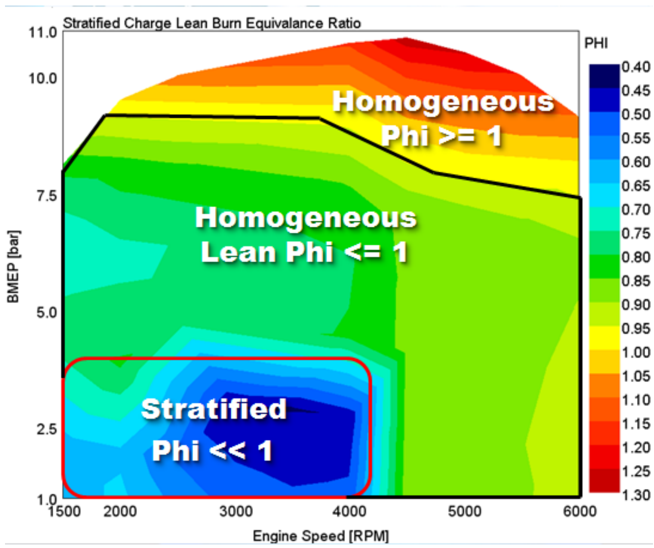


Figure A 44. Engine Operating Modes for lean GDI

The engine efficiencies for the three combustion modes were combined, and a composite engine operating map was generated for the same 82 operating points. Figures A 45 and A 46 below show the composite engine BSFC, and percent BSFC improvement over the baseline engine.

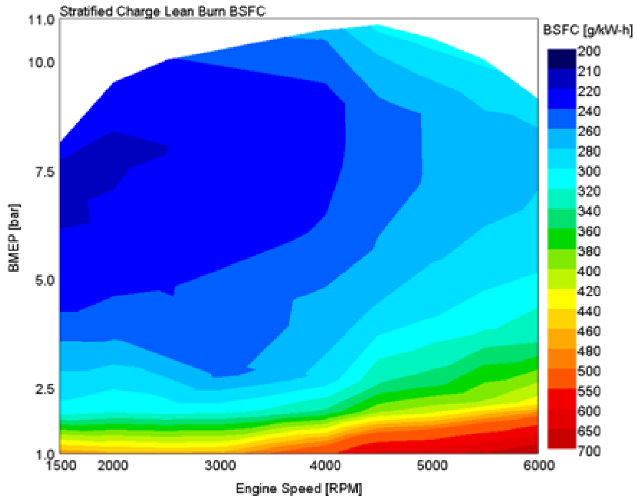


Figure A 45. Combined Lean Burn BSFC

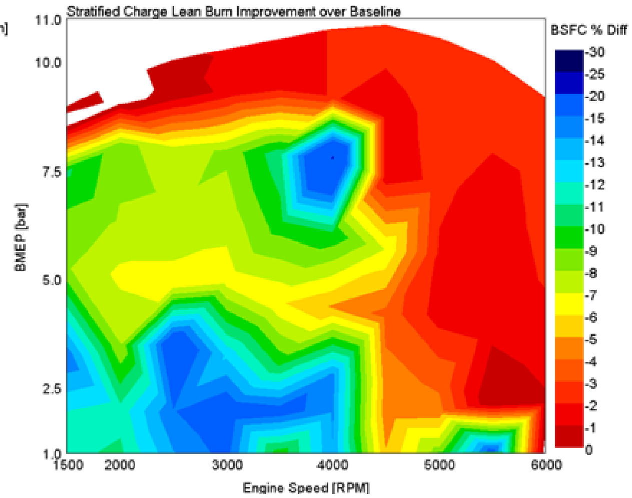


Figure A 46. BSFC Improvement over Baseline

2.4 Variable Valve Lift

Figure A 47. VVL arrangement

- **Variable Valve Lift (VVL) and Phasing**
  - Reduced pumping losses (reduced or eliminated throttling)
- **Additional Components include:**
  - Special cams and actuators to allow for valve deactivation

Variable valve lift (VVL) allows for improvement in efficiency during part load or "normally" throttled conditions. By closing the intake valve early and allowing the cylinders to pull a vacuum on the downward stroke, some of the energy is recovered as the piston comes back up. The benefit is more evident at higher engine speeds, and it phases out as load increases. Special cam actuators are required to allow for variable lift and phasing as shown in Figure 50 above. As the valve lift and duration are shortened, the in-cylinder charge motion may be affected negatively and cause unstable combustion at the low lift, or low load conditions. If the engine has two intake valves per cylinder, a potential solution for might be to vary the lift on one valve much more than the other to keep the velocity high through at least one of the intake ports during low load operation. For the purposes of this model, given the fact that this engine only has one intake valve, charge motion was assumed not to be affected.

The variable valve lift model is a modified version of the baseline 6.2L V8. The throttle was removed and the load was controlled by the intake valve lift. Intake valve duration was a mathematical function of the lift that attempted to emulate a realistic VVL system. In addition to the variable valve lift, the intake and exhaust cams are now have independent phasing capability. To optimize fuel efficiency, more than 1300 intake and exhaust timing combinations were run for the 82 point test matrix. Shown below in Figure A 48, is the final intake and exhaust valve timing for the 82 speed/load conditions.

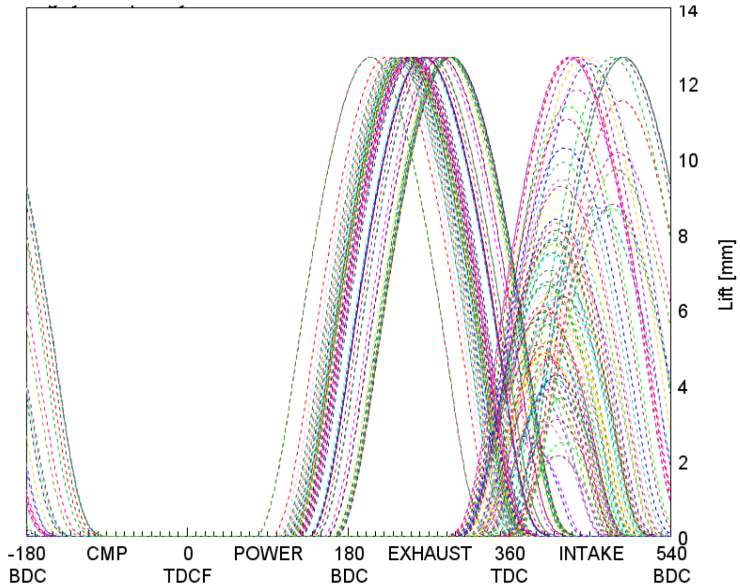


Figure A 48. 6.2L V8 Cam Phasing

Variable valve lift allows for improvement in BSFC at throttled conditions which is more evident at higher engine speeds, but the benefit phases out as load increases. Shown below in Figures A 49- A 50, are the BSFC and BSFC improvement over the baseline engine contour maps.

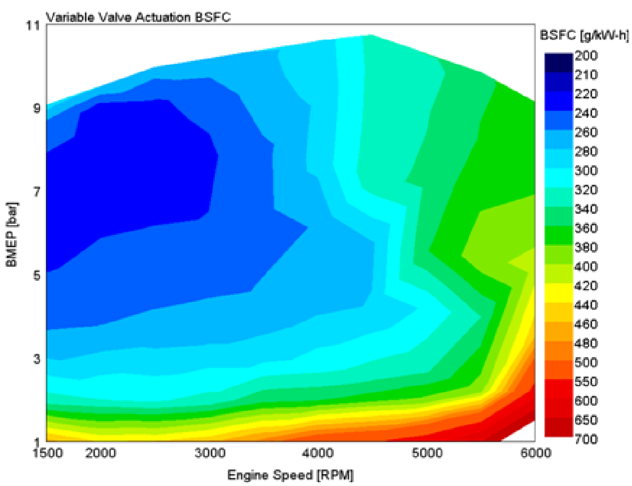


Figure A 49. Variable Valve Lift BSFC Map

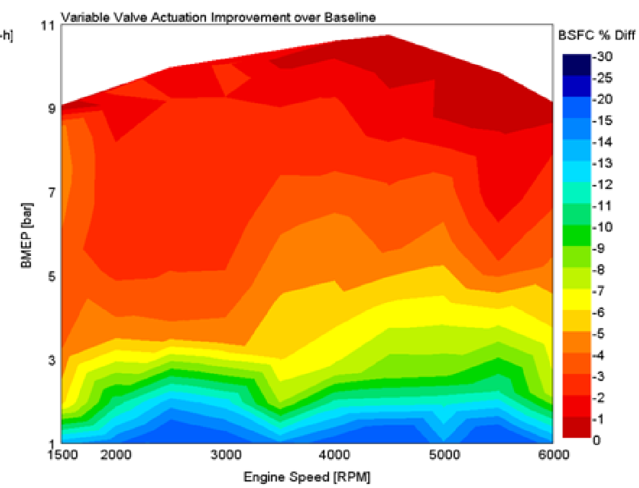


Figure A 50. BSFC Improvement over Baseline

2.5 Cylinder Deactivation

- **Four Cylinder Deactivation. GM Active Fuel Management (pushrod engines) and Audi cylinder on demand (OHC engines)**
    - Eliminates pumping losses in deactivated cylinders
    - Deactivation of every second cylinder by firing order (1 5 4 8 6 3 7 2)
    - Reduced pumping losses in active cylinders (less throttling)
    - Reduced energy loss from deactivated valves
    - Possible at loads up to 6 bar BMEP

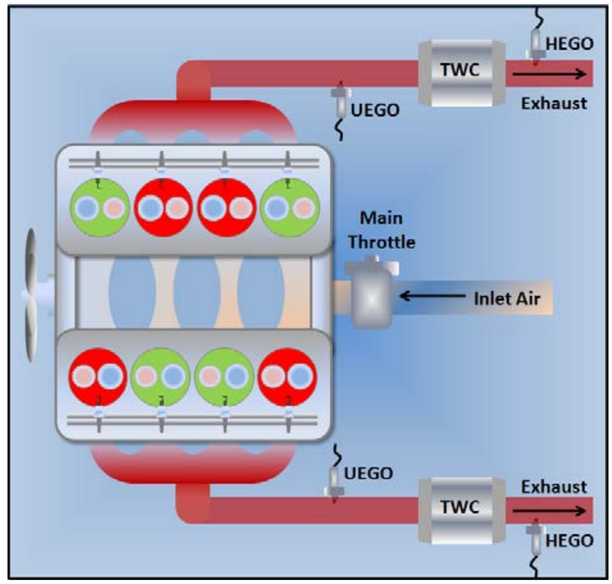


Figure A 51. Cylinder Deactivation

- **Additional Components include:**
    - Special cams and actuators to allow for valve deactivation
    - Active Engine mounts/dampers
    - Special exhaust to handle 2nd and 4th order

Engines that have eight cylinders have demonstrated operation with four cylinder deactivation such as General Motors Active Fuel Management and Audi Cylinder on Demand. The engine at idle conditions will operate on all eight cylinders to minimize vibration, but between idle and approximately five bar BMEP, four cylinders will cut fuel injection and valve actuation. The model was run up to the maximum load the four-cylinder would allow, five bar BMEP (see Figure A 52) and the areas that were less efficient than the baseline eight-cylinder were removed. (see Figure A 53) Loads above five bar BMEP will transition back to eight-cylinder operation.

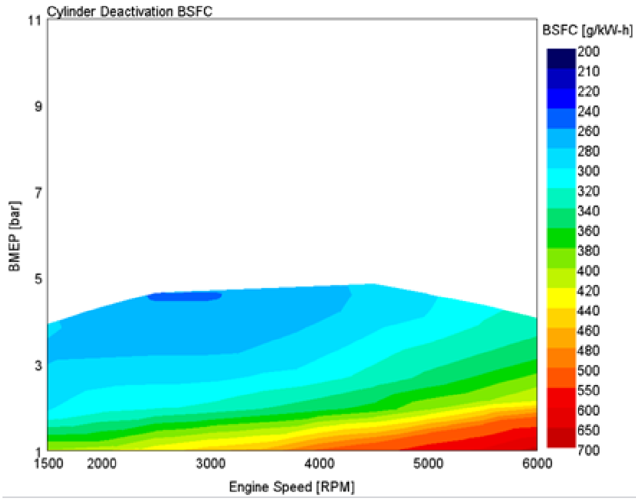


Figure A 52. Cylinder Deactivation BSFC Map

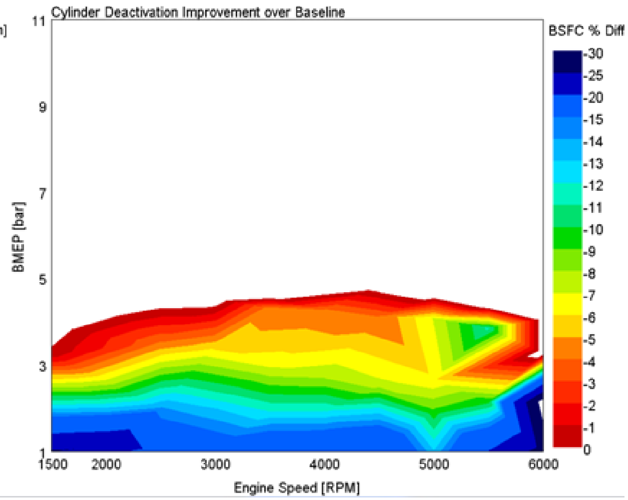


Figure A 53. BSFC Improvement over Baseline

2.6 GDI with EGR

- **EGR Benefits**
  - Reduced pumping losses
  - Improved working fluid
  - Improved knock tolerance
    - Improved combustion phasing
    - Increased compression ratio
  - Reduce heat transfer
  - Eliminate enrichment for catalyst protection
- **Additional Components include:**
  - EGR Valves
  - EGR Cooler
  - High Energy Ignition

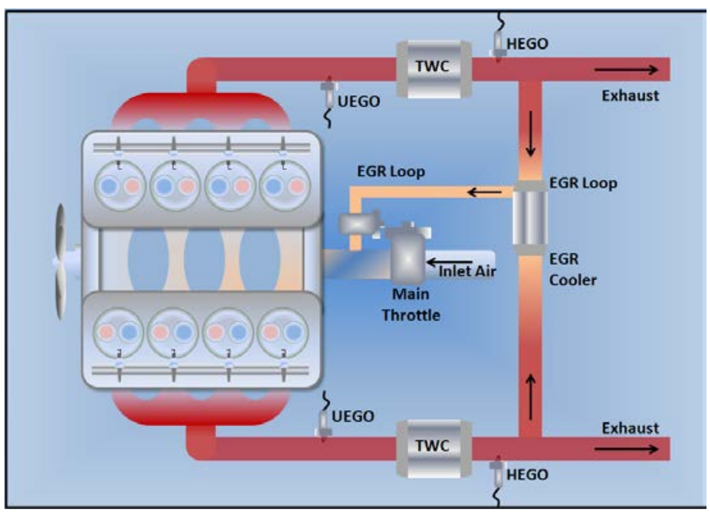


Figure A 54. EGR System for naturally aspirated V-8

SwRI's High Efficiency Dilute Gasoline Engine (HEDGE) consortium has proven that engines with exhaust gas recirculation (EGR) show large benefits in efficiency and emissions.

EGR engines require an EGR cooler and valve to recirculate the exhaust gasses, and a high energy ignition system to ignite the dilute mixture.

A model based on the baseline engine model was developed to represent a low pressure loop exhaust gas recirculation (LPL-EGR) engine. EGR can improve efficiency in several ways. It enables full-map stoichiometric operation, improves the ratio of specific heats of the working fluid, lowers heat loss in-cylinder due to reduced combustion temperatures and reduces knock tendency which allows more favorable combustion phasing and/or an increase in compression ratio. All of these effects can be observed in the test cell and in GT Power.

Engine parameters, such as CA50, MFB 10-90, compression ratio, EGR rates, and cam timing were selected for this engine model based upon SwRI's experience in converting and calibrating EGR engines. [4,5,6,7] It is understood that the values will not be exact, however, the values chosen are based upon the operating characteristics of the baseline engine and should be a close representation of an actual EGR conversion. The compression ratio was increased to 11.5:1, and trapped residuals were kept below 38 percent to ensure robust combustion. Appropriate EGR valves, coolers and controls were added to the model. Combustion phasing (CA50 and 10-90 burn duration) and EGR rates were adjusted to match SwRI experience with LPL-EGR. The CA50 map was modified from the baseline to change any value that was less than or equal to 13 degrees aTDCf to a MBT value of 8 degrees aTDCf. The remaining area of the map advanced the timing to 50% of the angle between MBT and the baseline CA50 value. For example, if the base CA50 is 16 deg aTDCf and MBT is 8 deg, then (Base CA50 - MBT) x 50% + MBT = (16-8)x0.5+8 = 12 deg aTDCf. The 10-90 burn duration was uniformly lengthened by 10 percent due to the slower burning EGR mixture.

EGR flow rates were defined as follows: at low speed and light load, 12% EGR is used. At high speed light load, 15 % EGR is used. Under full load, EGR is 15% at low speed, increasing to 18% at high speed. Linear interpolation is used to determine the EGR rate for any point in the map.

Intake cam phasing was altered from baseline cam timing to an EGR optimized cam timing based upon SwRI's EGR experience. At low loads more cam overlap is used with normal to late intake valve closing. Mid loads used low overlap and later intake valve closing. High loads operated at best volumetric efficiency positions with normal to high overlap and normal intake valve closing.

Figures A 55- A 56 below, show the engine BSFC, and percent BSFC improvement over the baseline engine.

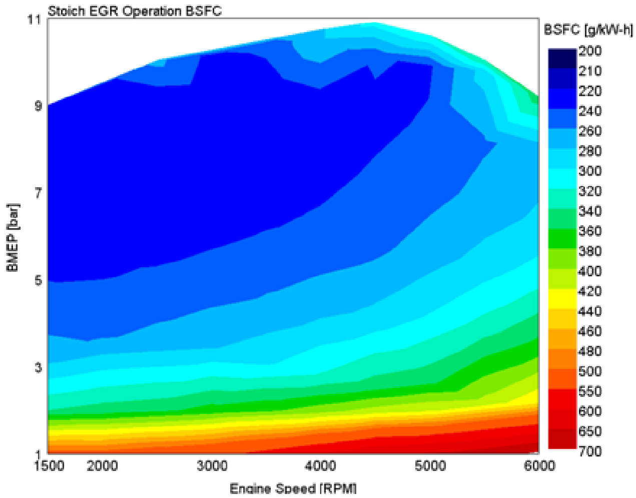


Figure A 55. Stoichiometric EGR BSFC Map

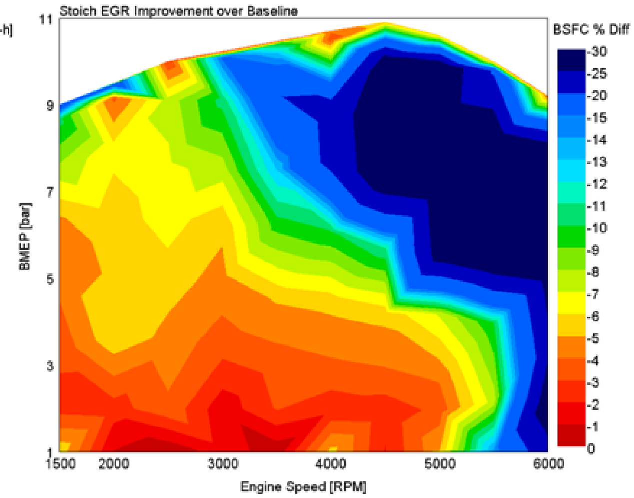


Figure A 56. BSFC Improvement over Baseline

2.7 FMEP Improvements

Reducing engine friction can be achieved in several ways, including low viscosity oils, smaller or dynamic bearings, as well as smart controlled oil and water pumps. The modeling effort for this portion lumped together the friction items and reduced their value by 10 percent from the baseline model.

The baseline engine model with the reduced friction was run over the same 82 point test matrix and the BSFC and BSFC improvement over baseline can be seen in the plots below.

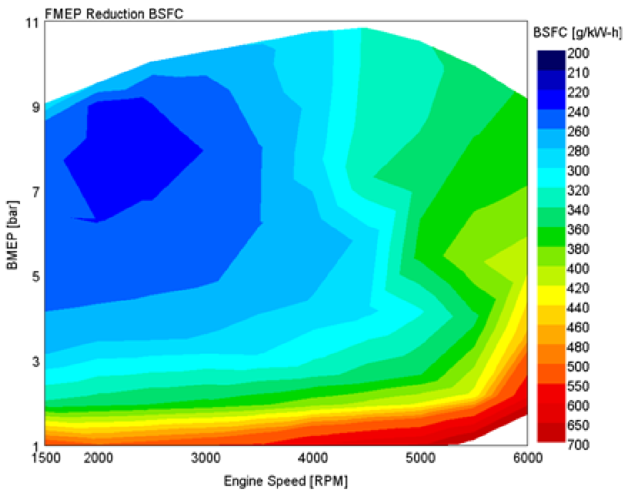


Figure A 57. Friction Reduction BSFC Map

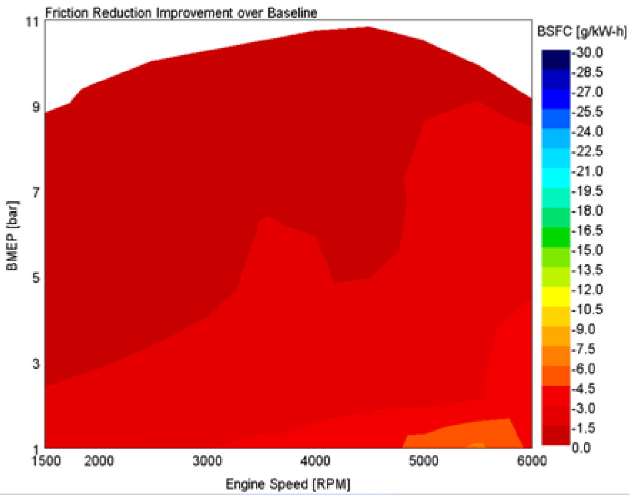


Figure A 58. BSFC Improvement over Baseline

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APPENDIX B

DIESEL ENGINE TECHNOLOGIES

Part 2: Diesel Engine Technologies

Objective: Simulate Future Diesel Engine Technologies to demonstrate Fuel Economy Improvement Potential in Class 2b through 7 vehicles utilizing:

- **14.6L turbocharged diesel engine**
- **6.7L VG turbocharged Medium Duty 300 bhp diesel engine**
- **6.7L VG turbocharged Pick Up 385 bhp diesel engine**

1. 15L turbocharged diesel engine evaluation plan:

1.1. **Baseline Validation**
1.2. **Reduced Exhaust Backpressure**
1.3. **Reduced Inlet System Restriction**
1.4. **Optimized Mechanical Turbocompound (APT)**
1.5. **Optimized Electrical Turbocompound (APT)**
1.6. **No EGR**
1.7. **Turbocompound Removed (No APT)**
1.8. **No EGR or APT**
1.9. **Asymmetric Turbocharger and EGR circuit**
1.10. **Down Speeding with Increased BMEP**
1.11. **Downsizing with Constant BMEP**
1.12. **Downsizing with Constant Torque**
1.13. **Explore FMEP Improvements**
1.14. **Explore Turbo Efficiency Improvement**
1.15. **Variable Valvetrain Actuation**

2. 6.7L VG turbocharged Medium Duty 300 bhp diesel engine evaluation plan

2.1. **Baseline Validation**
2.2. **Reduced Exhaust Backpressure**
2.3. **No EGR**
2.4. **Explore Turbo Efficiency Improvement**
2.5. **FMEP Improvements**

3. 6.7L VG turbocharged Pick Up 385 bhp diesel engine evaluation plan

3.1. **Baseline Validation**

3.2. **Reduced Exhaust Backpressure**

3.3. **No EGR**

3.4. **Explore Turbo Efficiency Improvement**

3.5. **4.5L Version (4 cyl), Constant BMEP**

3.6. **FMEP Improvements**

1.1 DD15Baseline Validation

- **Basic engine specification**
  - 14.6L inline 6 cylinder
  - Single Fixed Geometry Turbo
  - Auxiliary Power Turbine (Turbocompound) with geared link to engine
  - 4 valves per cylinder
  - Diesel
  - 376 kW @1800 rpm
  - 2200 Nm @ 1240-1400 rpm

GT Power model was built using the geometry of a DD15 engine previously benchmarked at SwRI and run in a test cell. The engine had been fully instrumented, including temperatures, pressures, speed, torque and high-speed cylinder pressure measurements and had been operated over a full test matrix that covered the entire engine operating range. Engine data from the test cell, shown below, was used as model input data.

- **Combustion data – from analysis of high-speed cylinder pressure data**
  - **50% Mass Fraction Burned**
  - **10-90 Burn Duration**
- **Valve Flow Coefficients**
- **FMEP**
- **Air/Fuel Ratio**
- **EGR Rates**

The GT Power model was then run over a speed range of 1000 – 2100 rpm at 10, 20, 40, 60, 80 & 100% load (see figure B1a), and validated by the following operating parameters:

- **BMEP, IMEP, PMEP**
- **BSFC**
- **Air and Fuel Flow**
- **Pre/Post compressor pressures and temperatures**
- **Pre/Post Turbine pressures and temperatures**
- **EGR Rates**

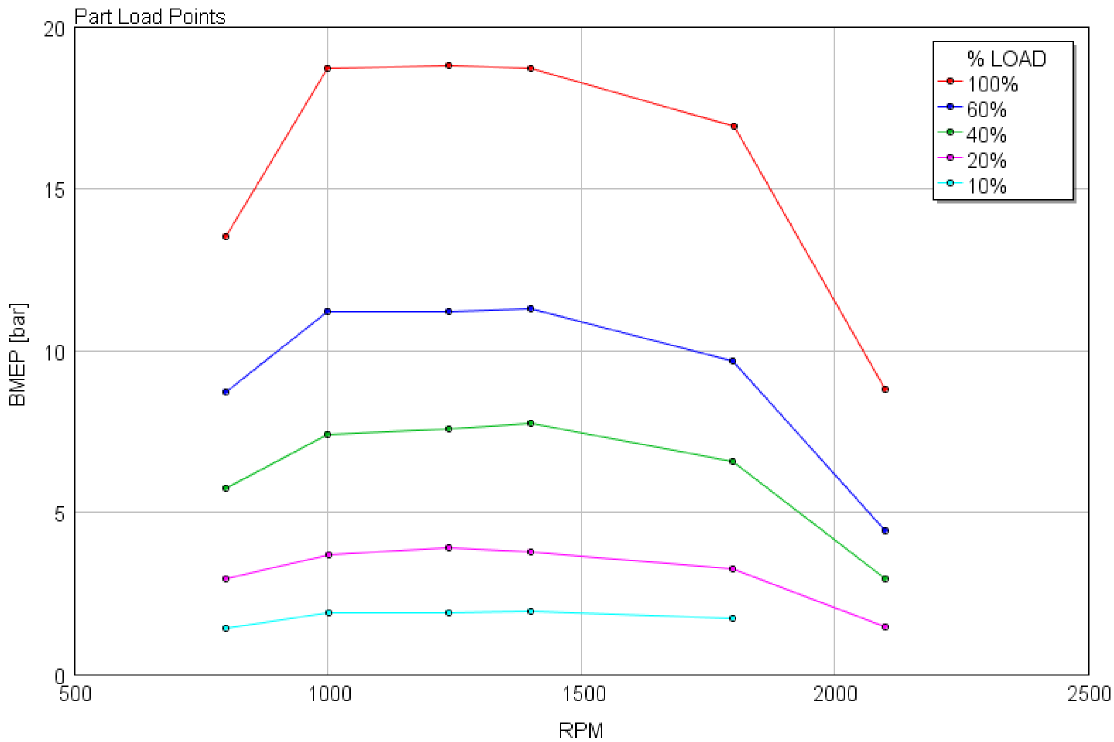


Figure B1a. DD15 GT baseline operating points

The GT model was setup using combustion data derived from the measured high-speed cylinder pressure data. A typical correlation between measured (Test) data and the GT simulated data are shown in the figure below.

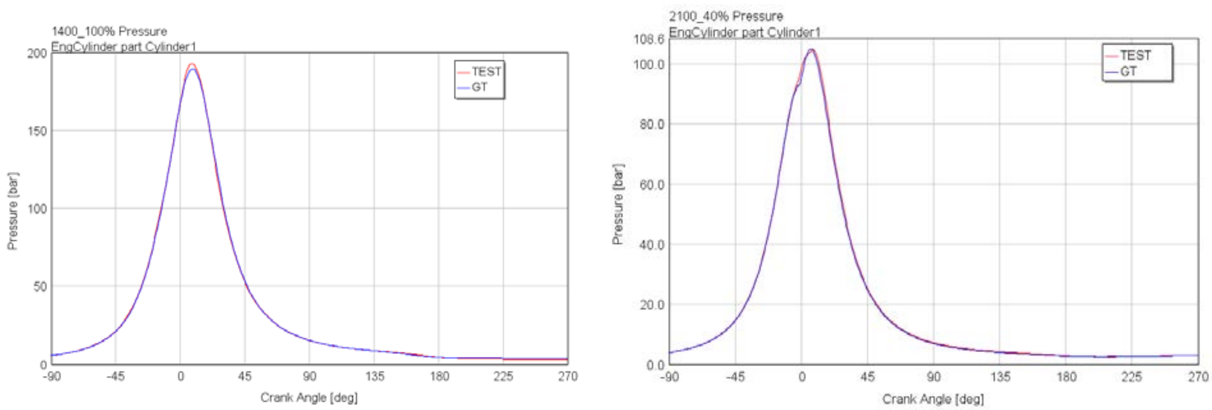


Figure B1b. DD15 GT Cylinder pressure correlation

The geometry was constructed from measurements of the real hardware and port flow data and valve lift was from flow-rig tests. The actual turbo maps for both the main turbo (twin entry) & the APT were not available, so existing maps from a similarly sized engine were used and scaled so that that the measured inlet & outlet conditions were matched to test data.

A review of published data indicated that a match of +/- 3-5% should be achievable with a correctly setup GT model. Two examples of papers relating to models for similar medium duty engines are:

- “Creation and Validation of a High Accuracy, Real-Time-Capable Mean-Value GT-Power Model” by Tim Prochnau, International Truck & Engine Corporation. Presented at the 11th GT-Suite Users’ Conference, November 13, 2007
- “Transient engine modeling at John Deere using GT-Power” by John Deere. Presented at GT-Suite North American Conference 2009

Further papers showing the modelling, accuracy and application of GT Power models can be found at https://www.gtisoft.com/knowledgebase .

With the "matched" operating conditions, the operating points were run again to confirm that the simulated BSFC was within +/- 3% of the experimental results. Shown below is the comparison between the engine data and model reported BSFC. Also shown is the full range of matched data. The accuracy of the model was also evaluated by comparing simulated and experimental pumping mean effective pressure (PMEP), air flow, EGR rate, peak cylinder pressure, as well as turbocharger pressures and temperatures. See Figure B1d for comparison plots of these parameters.

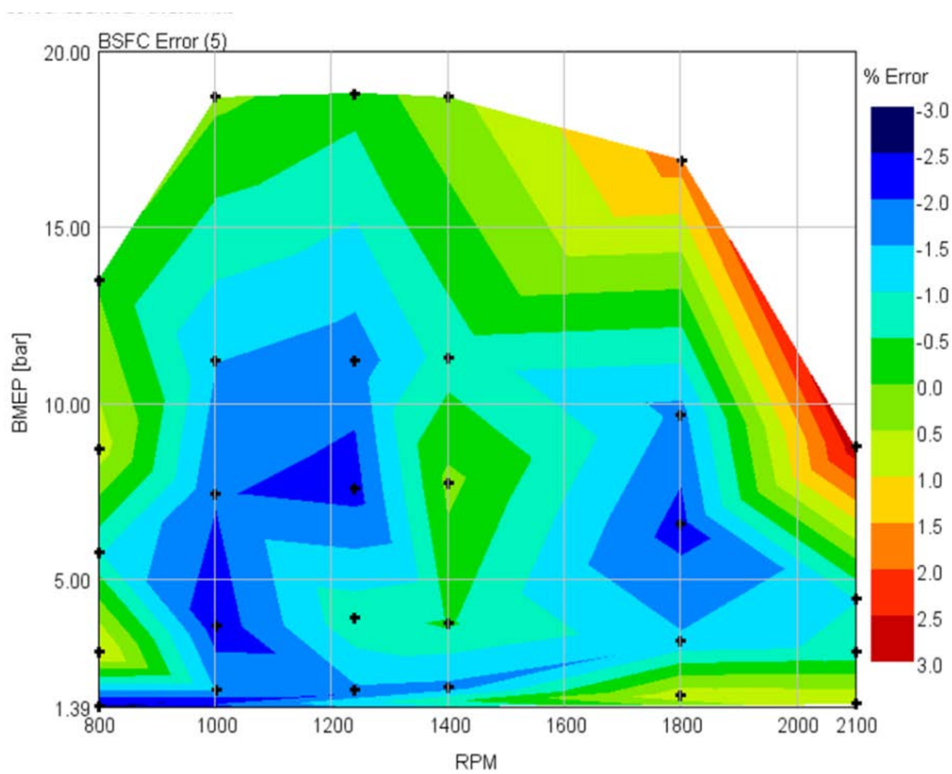
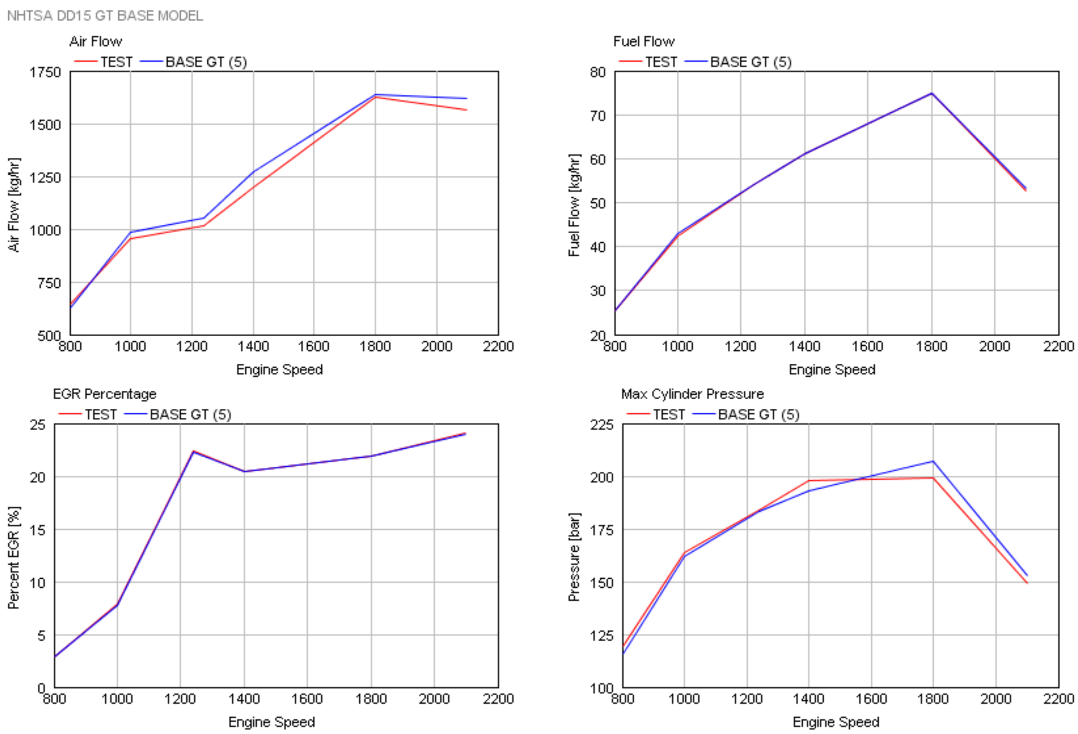


Figure B1c. BSFC Model Error



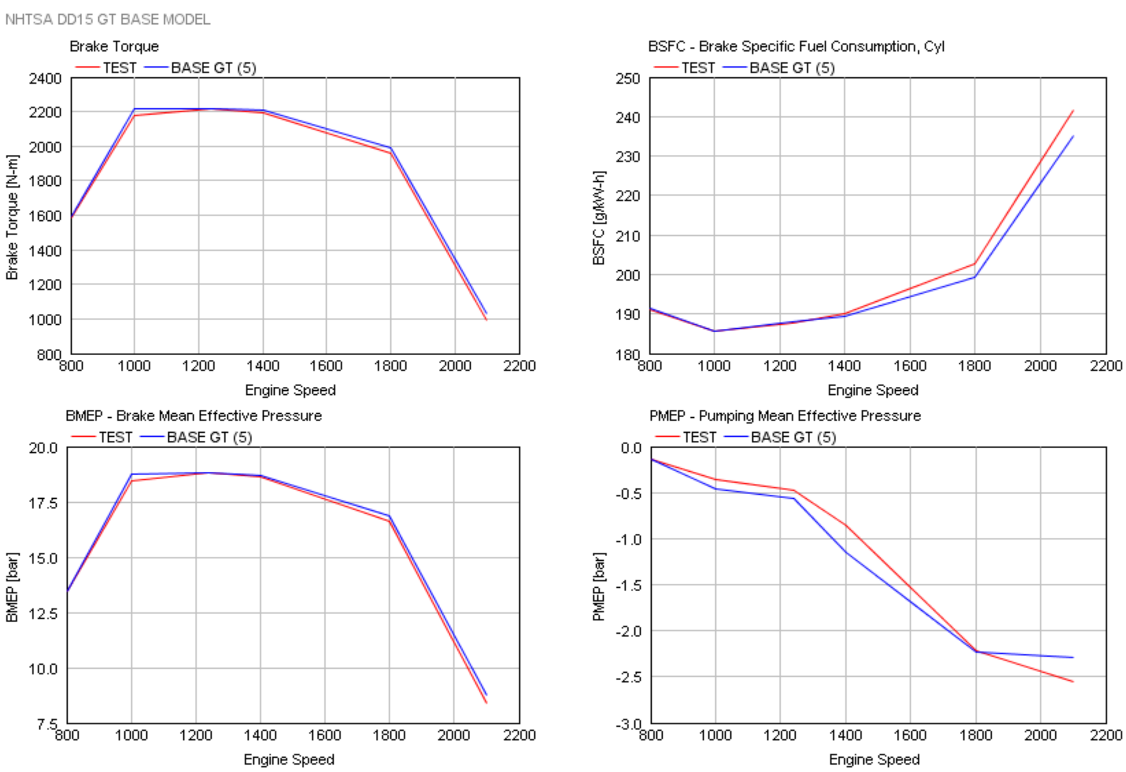


Figure B1d. (Part 1) Baseline Model Data Correlation at Full Load

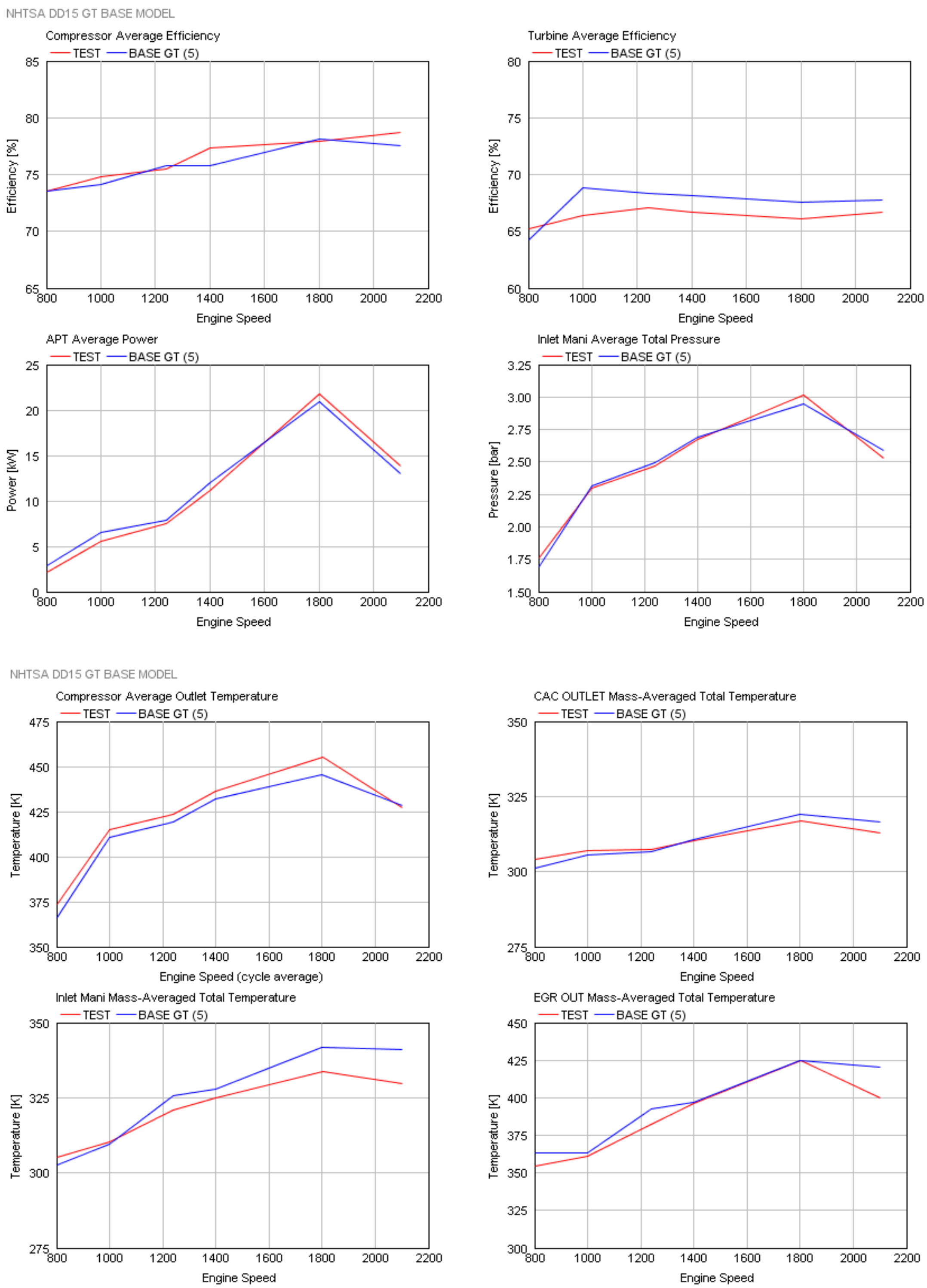


Figure B1d. (Part 2) Baseline Model Data Correlation at Full Load

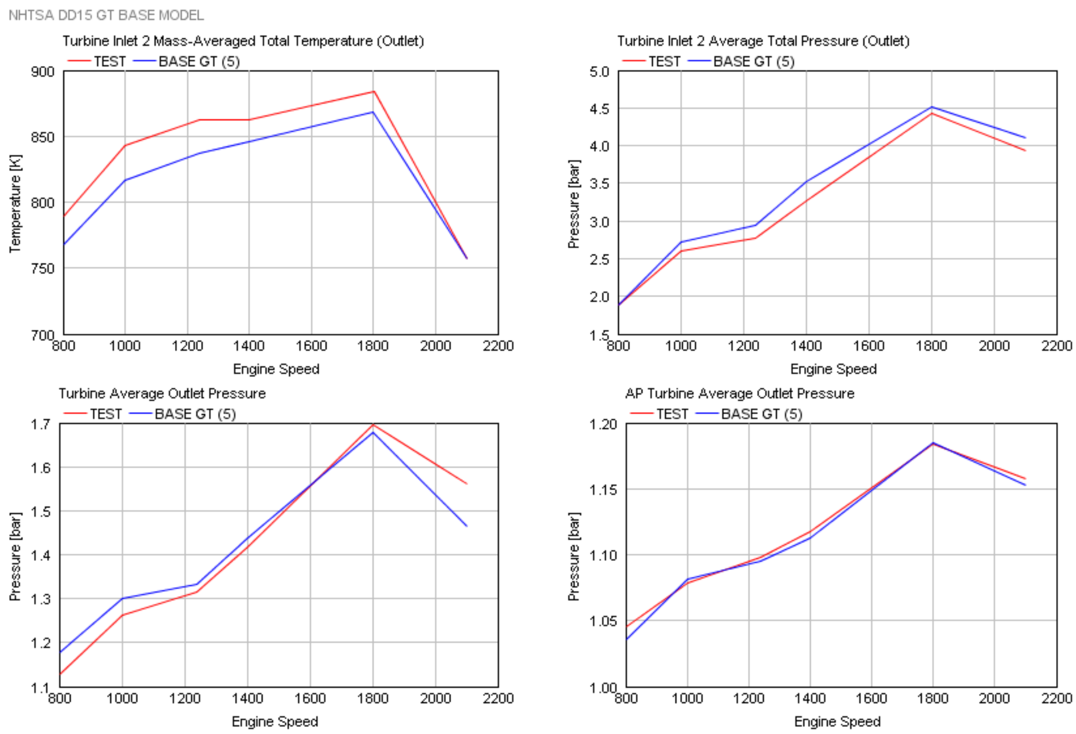


Figure B1d. (Part 3) Baseline Model Data Correlation at Full Load

1.2 Reduced Exhaust Restriction

The aim of running the model with reduced exhaust restriction is to simulate the effect of improved flow after-treatment systems for emissions control and the resulting reduction in pumping losses. The baseline GT model was originally setup without modeling the emissions control hardware, but the effective backpressure seen in the exhaust at the APT exit is simulated by the use of a simple orifice. To model the reduced exhaust backpressure the orifice is resized to achieve the desired target backpressure at the rated condition. In this case, the backpressure has been reduced by 45%, from 18.5 to 10 kPa. The effect on the engine BSFC is shown in figure B2a. Assuming that the after-treatment effectiveness is retained, there will be no effect on either engine-out or tailpipe emissions of criteria pollutants.

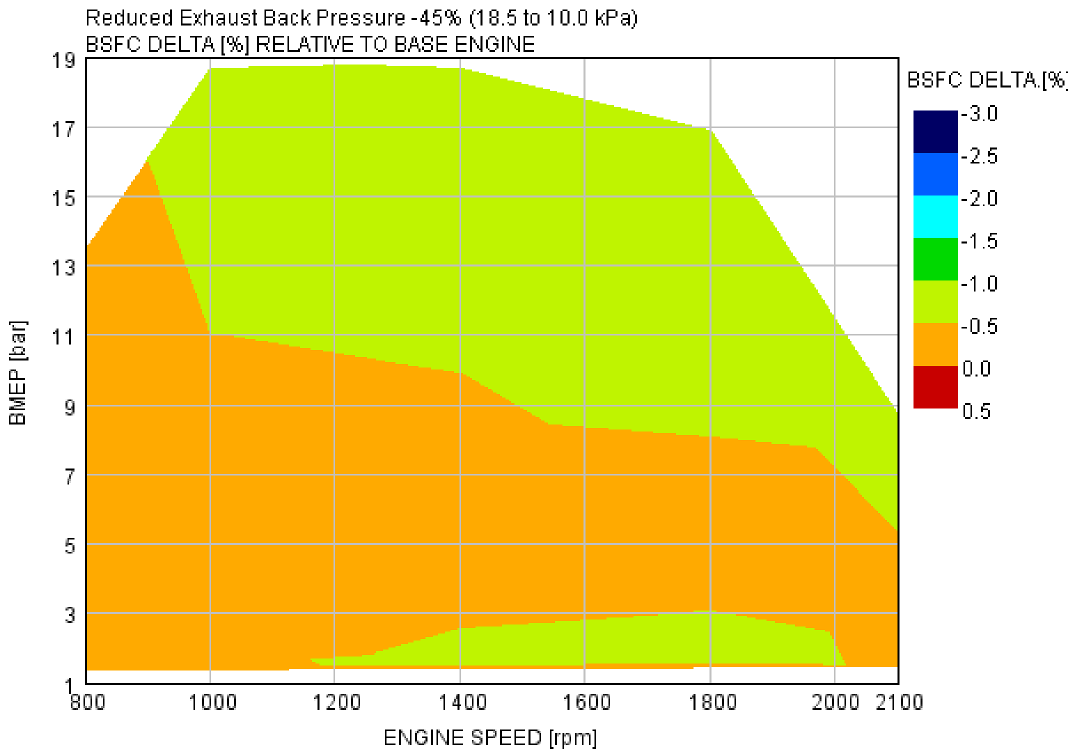


Figure B2a. Reduced Exhaust backpressure BSFC Improvement

1.3 Reduced Inlet System Restriction

The two areas of the inlet system considered in this section where the inlet filter and the charge air cooler (CAC). The filters effective restriction was reduced by 50% and the CAC restriction was virtually eliminated by opening up the GT model orifice that controlled the restrictions. Despite the dramatic sounding changes, the overall effect was minimal with < 0.5% BSFC improvement across the range. This technology as modeled has no effect on EGR rates, and is therefore not predicted to have measureable effects on engine-out or tailpipe criteria emissions.

1.4 Optimized Mechanical Turbocompound (APT)

The base engine model incorporates an axial power turbine (APT) that recovers exhaust energy after it has passed through the main engine turbo, and is connected to the engine crankshaft via a gearbox & fluid clutch. The setup is shown in figure B4a. The system is setup to deliver maximum power at engine rated power, so for this study the turbine was re-sized to improve the power delivery at part load drive cycle speeds, even at the expense of maximum power. Additionally the gear ratio between the APT and the crankshaft was modified to improve the overall performance. The remainder of the engine system, including the main turbo was left unmodified. By changing the APT setup the effective backpressure seen by the main turbo changed, leading to it potentially operating in a different point on its map. The backpressure seen by the APT was maintained as per the baseline model by controlling the exhaust orifice.

As shown in figure B4b, the revised APT improved BSFC across the majority of the operating map by 1-2%, but at the expense of the higher speed section. In reality the engine rarely operates in this zone, so it has little effect on the drive cycle fuel consumption. This technology as modeled has no effect on EGR rates, and is therefore not predicted to have measureable effects on engine-out or tailpipe criteria emissions.

Figure B4a. DD15 Axial Power Turbine portion of the turbocompound system

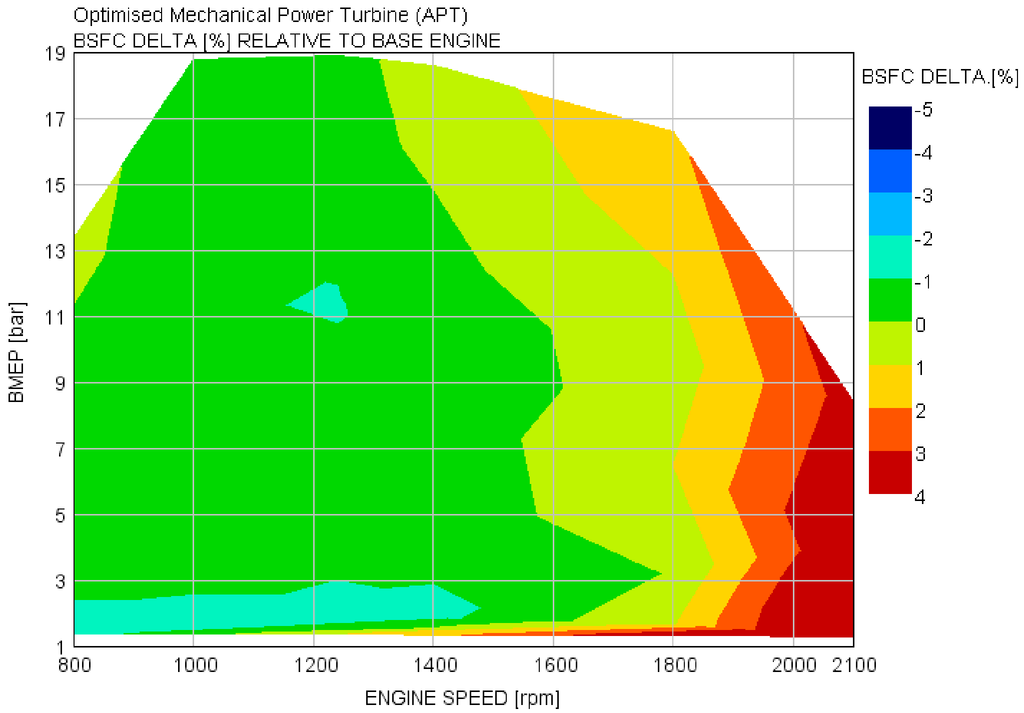


Figure B4b. BSFC improvement from Optimized Mechanical Axial Power Turbine

1.5 Optimized Electrical Turbocompound (APT)

For the electrical APT model the energy recovered is stored in an electrical form. On a vehicle this energy can be used to power electrical accessories – fans, pumps, lighting and entertainment system. In this analysis, the stored energy was used to drive a motor attached to the crankshaft, through a gearbox. An electrical generator efficiency map was used for the storage process, shown in figure B5a, and a motor efficiency of 90% is assumed for the power motor. The resulting effect on BSFC is shown in figure B5b. In reality, the energy could be stored and used only at critical or pre-determined conditions, but the results show the effect of the energy produced at the given speed/load point. As for the mechanical system, the APT is re-sized to give improvements at cruise speeds & loads, but in this case the penalty at higher speed & loads, where it does not operate so frequently on a typical drive cycle, is less significant. There will be no effect on engine-out or tailpipe criteria emissions from this technology.

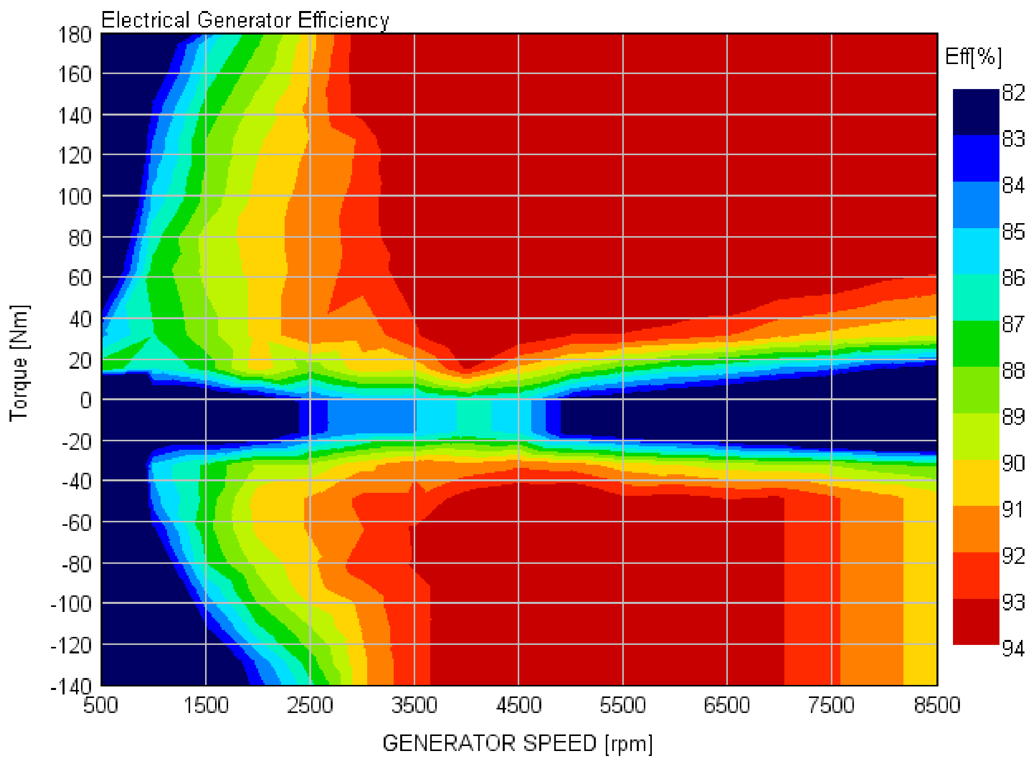


Figure B5a. DD15 Electrical generator efficiency map

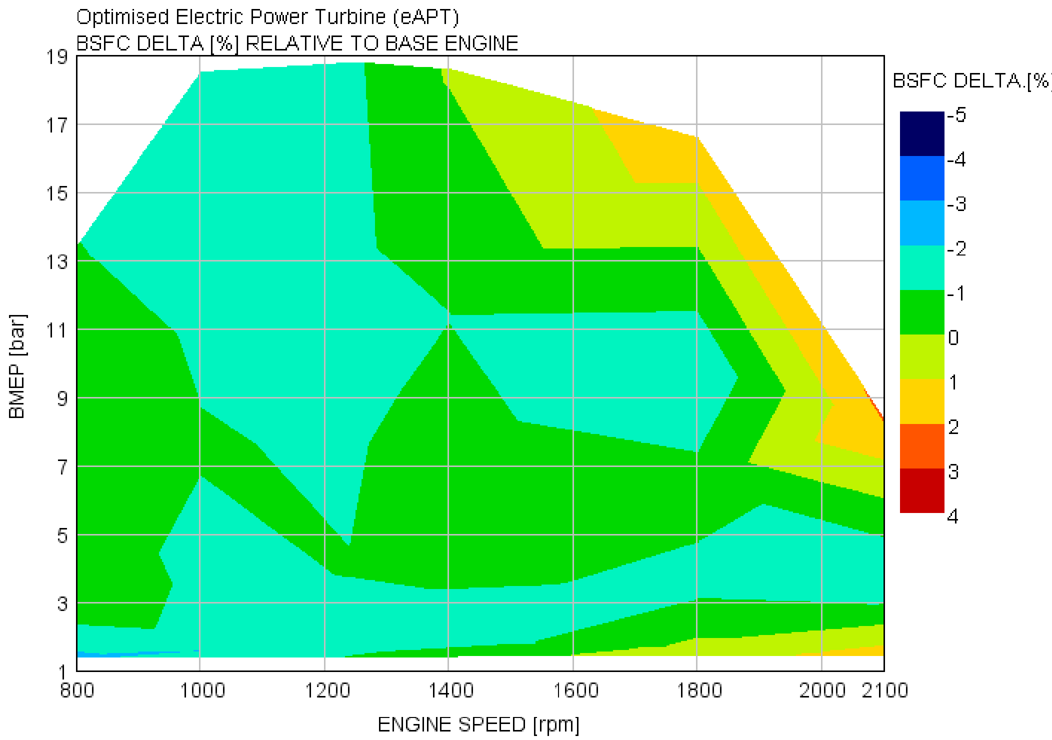


Figure B5b. BSFC improvement from Optimized Electrical Axial Power Turbine

1.6 No EGR

The base DD15 engine features a high-pressure EGR circuit, with 2 pre-turbine exhaust manifold feeds, 2 valves, an EGR cooler and venturi system, which dumps the EGR into the inlet system pre manifold but post heater & throttle valve. To assess the engine performance with no EGR demand, the entire EGR system set of system components was removed from the GT model, with the heater & throttle maintained. The engine was run with the same performance targets but an increased AFR limit (+2 ratios), and the combustion & injection parameters left as standard. In practice, eliminating EGR will shorten combustion duration, which should reduce fuel consumption slightly, so the results here are likely to slightly understate the benefit of EGR removal. This technology would involve very high engine-out NOx, and require a very efficient SCR system with a high urea consumption rate.

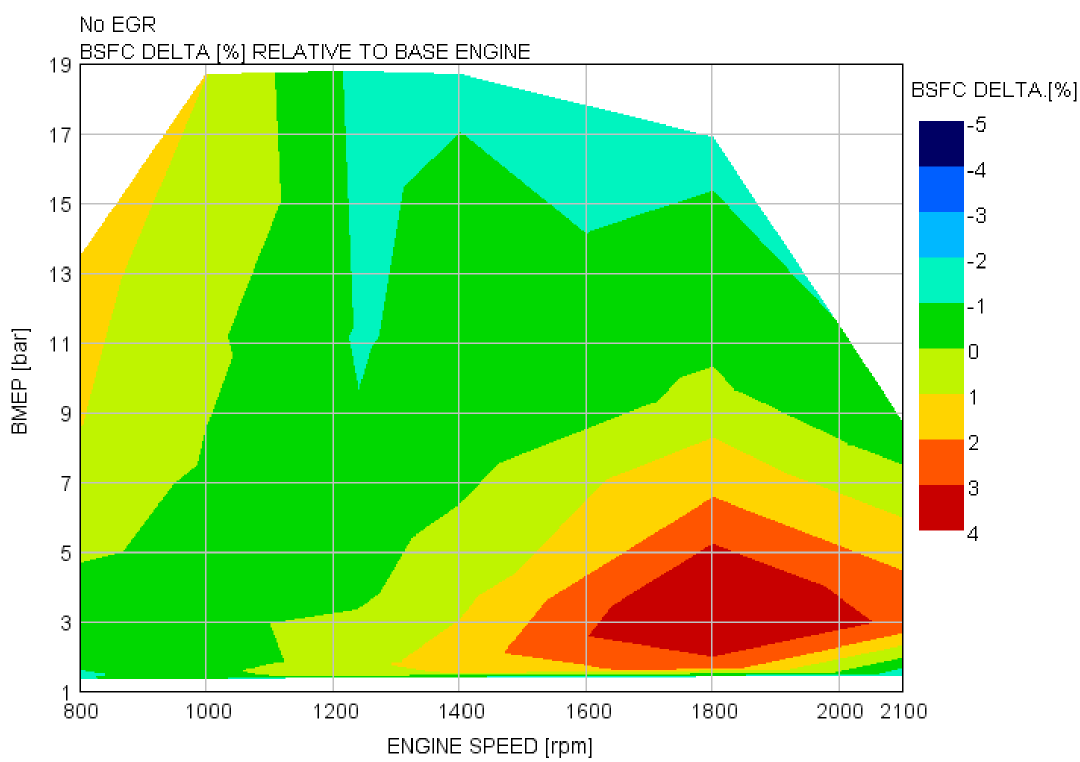


Figure B6a. BSFC improvement from EGR removal

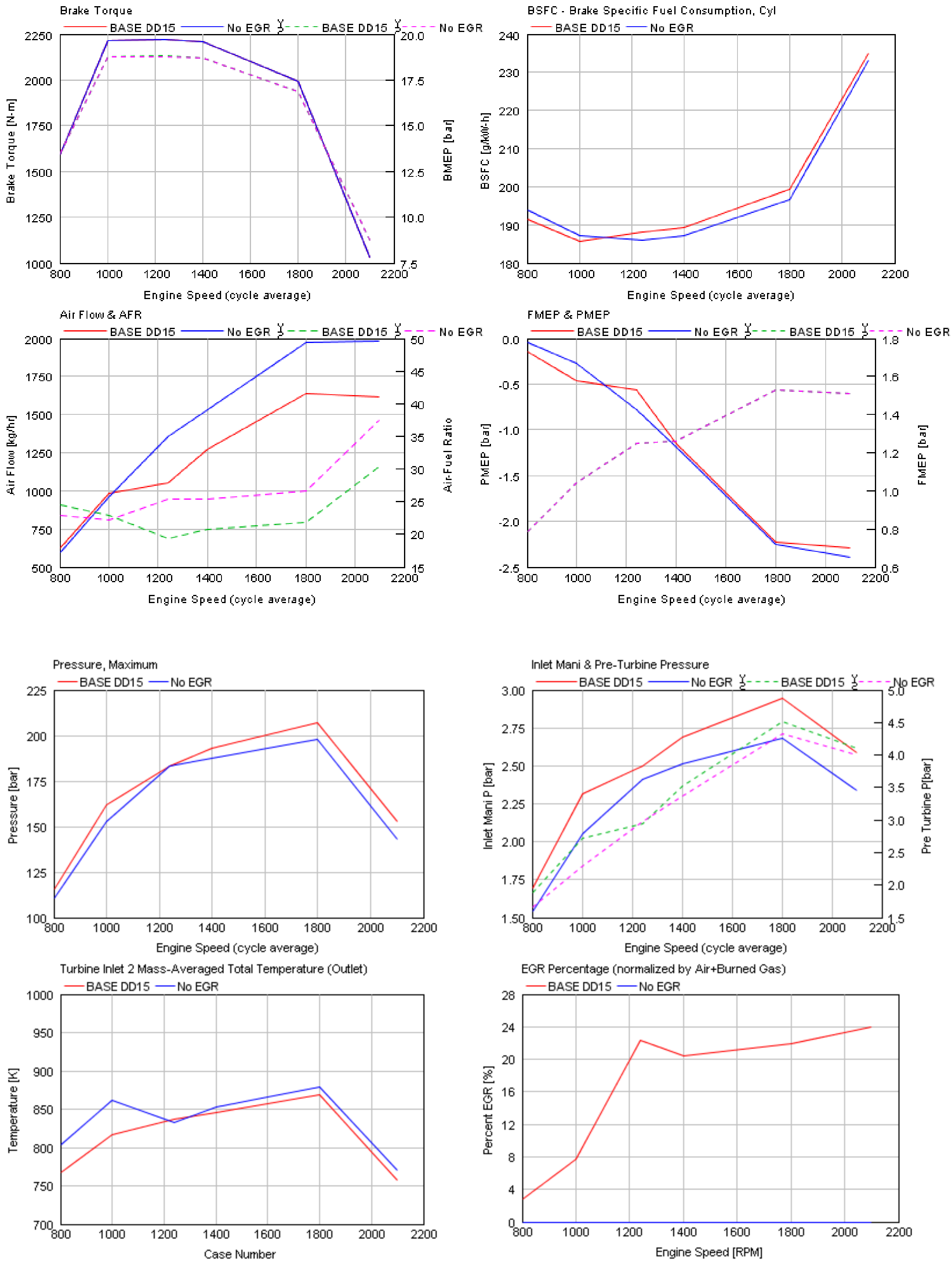


Figure B6b. Effect of EGR Removal on Engine Performance at Full Load

1.7 No Turbocompound (APT)

The base engine model incorporates an APT which recovers exhaust energy after it has passed through the main engine turbo, and is connected to the engine crankshaft via a gearbox & fluid coupling. In this model, the APT is removed from the engine and the back pressure seen by the main turbo was reset to match the baseline engine backpressure. The main turbo was not rescaled in this case as the contribution by the APT to the overall engine performance is relatively limited, particularly at part load. The latest generation of this engine has removed the APT (along with several other changes described in section 1.9) and claims improved BSFC. At lower engine speed, this configuration does not provide sufficient back pressure to drive EGR flow. As a result, this technology would involve very high engine-out NOx, and require a very efficient SCR system with a high urea consumption rate. Alternatives include the use of a variable geometry turbo (VGT). This would enable EGR flow, but would impose a BSFC penalty due to reduced turbo efficiency at speeds where this is required to drive EGR flow. VGT is by far the most common technology approach in the HD engine market today.

Note that in Figure B7b, the pumping work (PMEP) is reduced by about a half bar on the torque curve, which would normally result in lower fuel consumption. The fuel consumption on the torque curve is almost unchanged. The reason for this effect is that the power contribution from the APT has been removed. At full load, the lower PMEP cancels the power turbine contribution. At lighter loads, the benefit of reduced PMEP is greater than the penalty of reduced ATP contribution, so there is a net fuel consumption benefit at part loads.

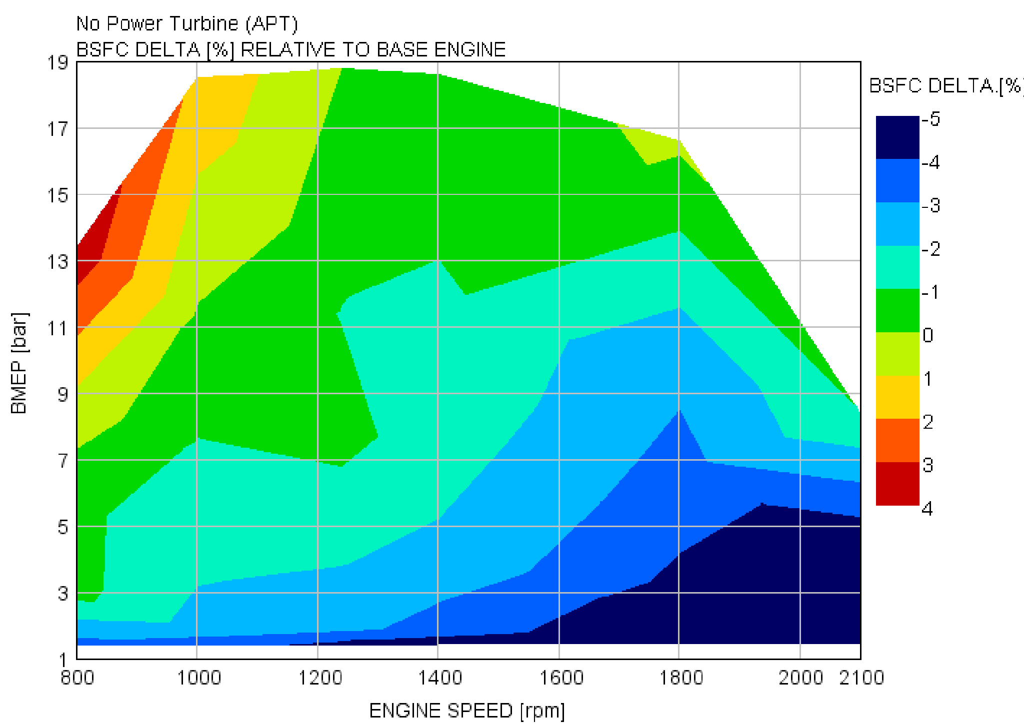


Figure B7a. BSFC improvement from removing turbocompound

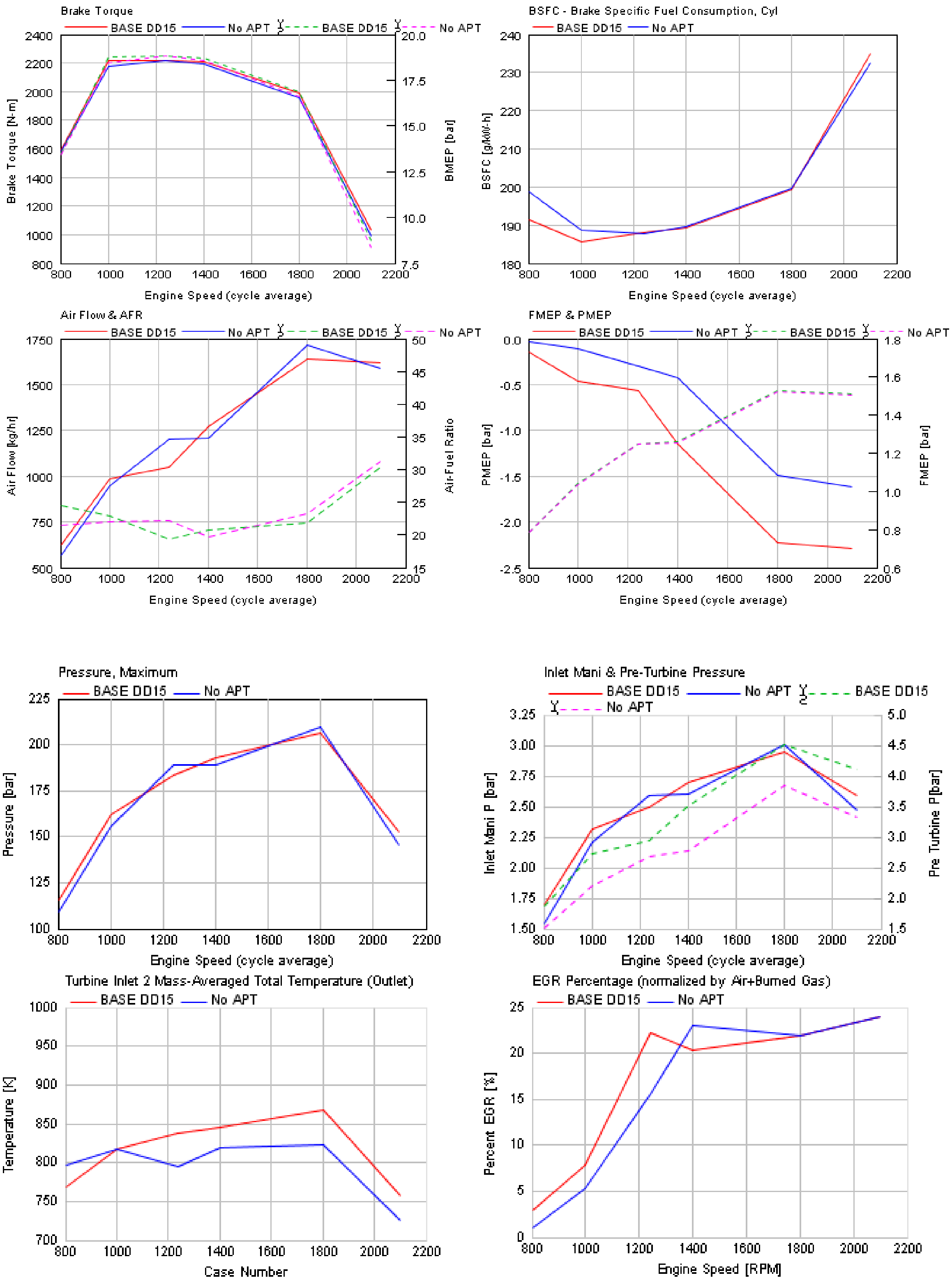


Figure B7b. Effect of Turbocompound Removal on Engine Performance at Full Load

1.8 No EGR or APT

This technology setup involved the removal of both the EGR system & APT as described in sections 1.6 & 1.7 above. This required the main turbo to be re-sized and the back pressure to be reset to achieve comparable performance with the base engine. Combustion & injection characteristics were left unchanged. This technology would involve very high engine-out NOx, and require a very efficient SCR system with a high urea consumption rate. In reality, eliminating EGR flow will shorten combustion duration, which has a positive effect on BSFC. We did not have sufficient data to model this effect accurately, but the net effect is that the fuel consumption reduction is probably slightly understated.

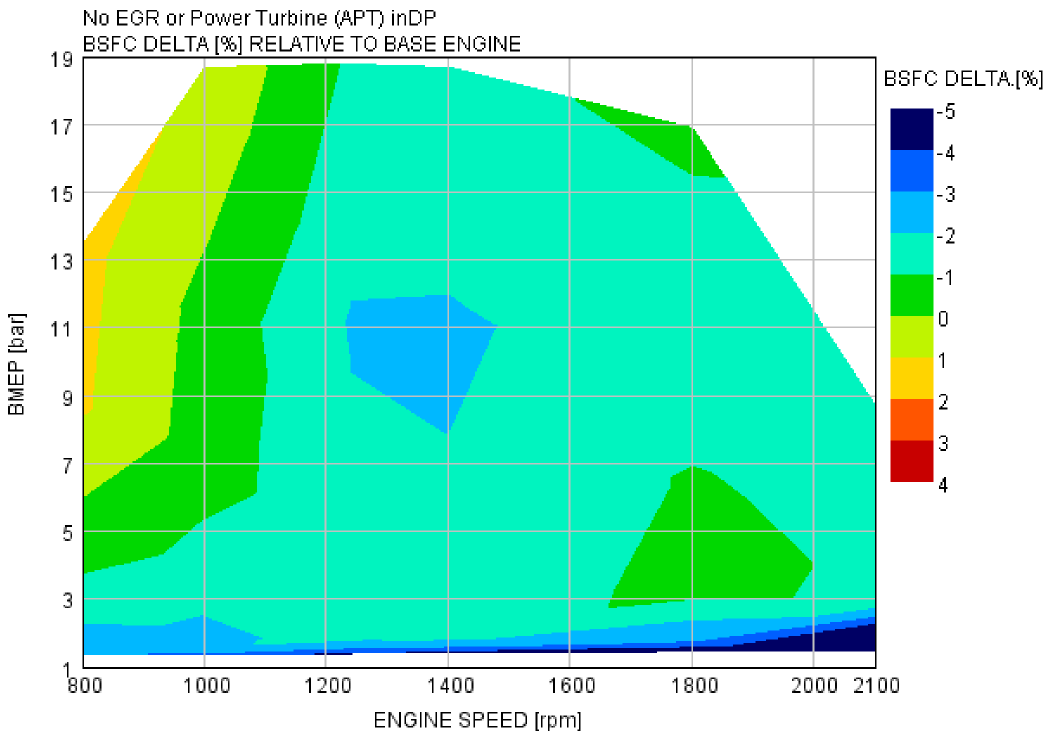


Figure B8a. BSFC improvement from EGR & APT removal

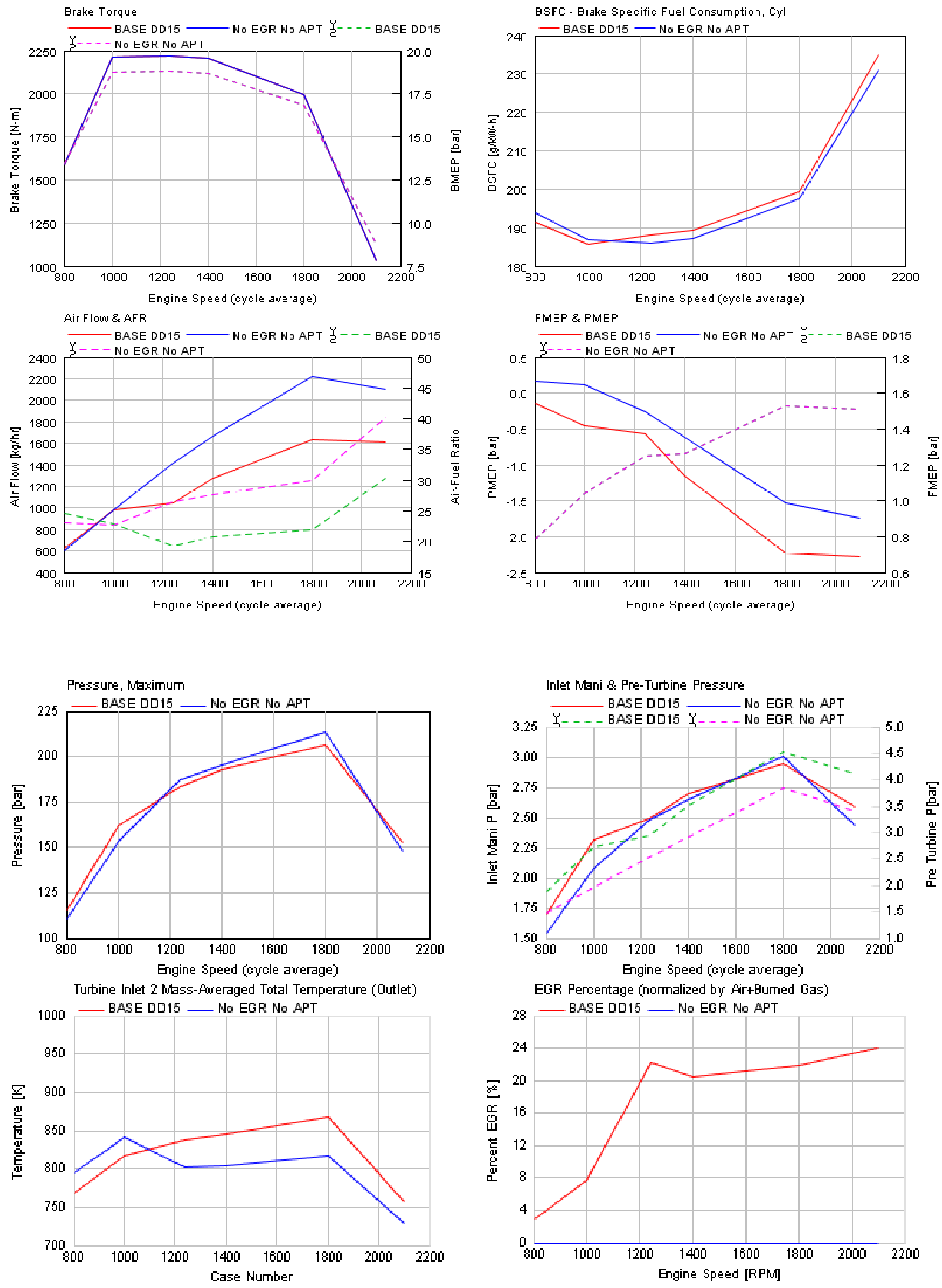


Figure B8b. Effect of EGR & APT Removal on Engine Performance at Full Load

1.9 Asymmetric Turbocharger and EGR Circuit

As described above in section 1.6, the base DD15 engine features 2 pre-turbine exhaust manifold feeds & 2 EGR valves, so effectively both the front & rear halves of the engine supply equal amounts of EGR. The latest generation of this engine has a revised EGR system that only takes EGR from one half of the engine. As the turbine is still twin entry, this leads to the requirement for an asymmetric turbine entry setup due to the differing flow rates from either side of the engine. The eventual ratio between the 2 entries was 37:63, and the EGR system was sized to achieve the same EGR rates are the baseline engine. Additionally, the APT has been removed. All other engine setups were left unchanged. The asymmetric turbo approach causes a fuel consumption penalty over most of the full load curve, especially near rated speed. The benefit of the power turbine is largest at maximum speed and load, so this result is to be expected. There is a modest improvement in BSFC in the cruise RPM range, especially at light loads, where the power turbine does not contribute. At high speed, light load, the benefit is large. At this condition, the power turbine hurts (from pumping work due to exhaust restriction) more than it helps.

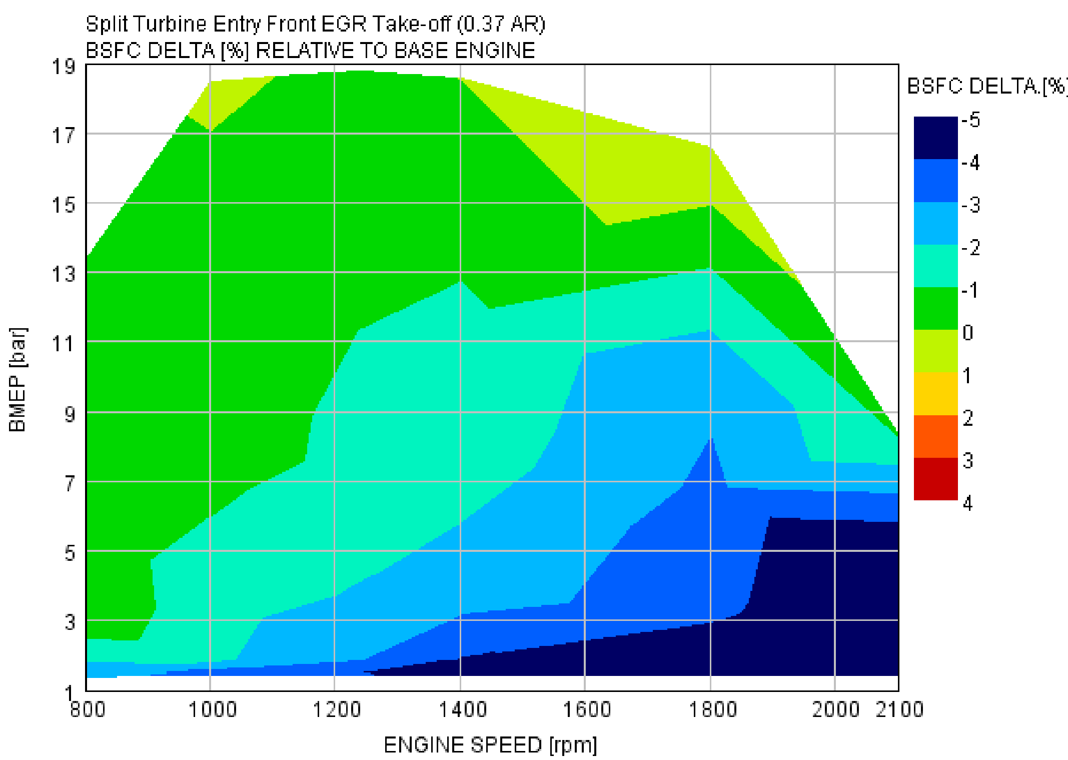


Figure B9a. BSFC improvement from Asymmetric Turbo and EGR Supply

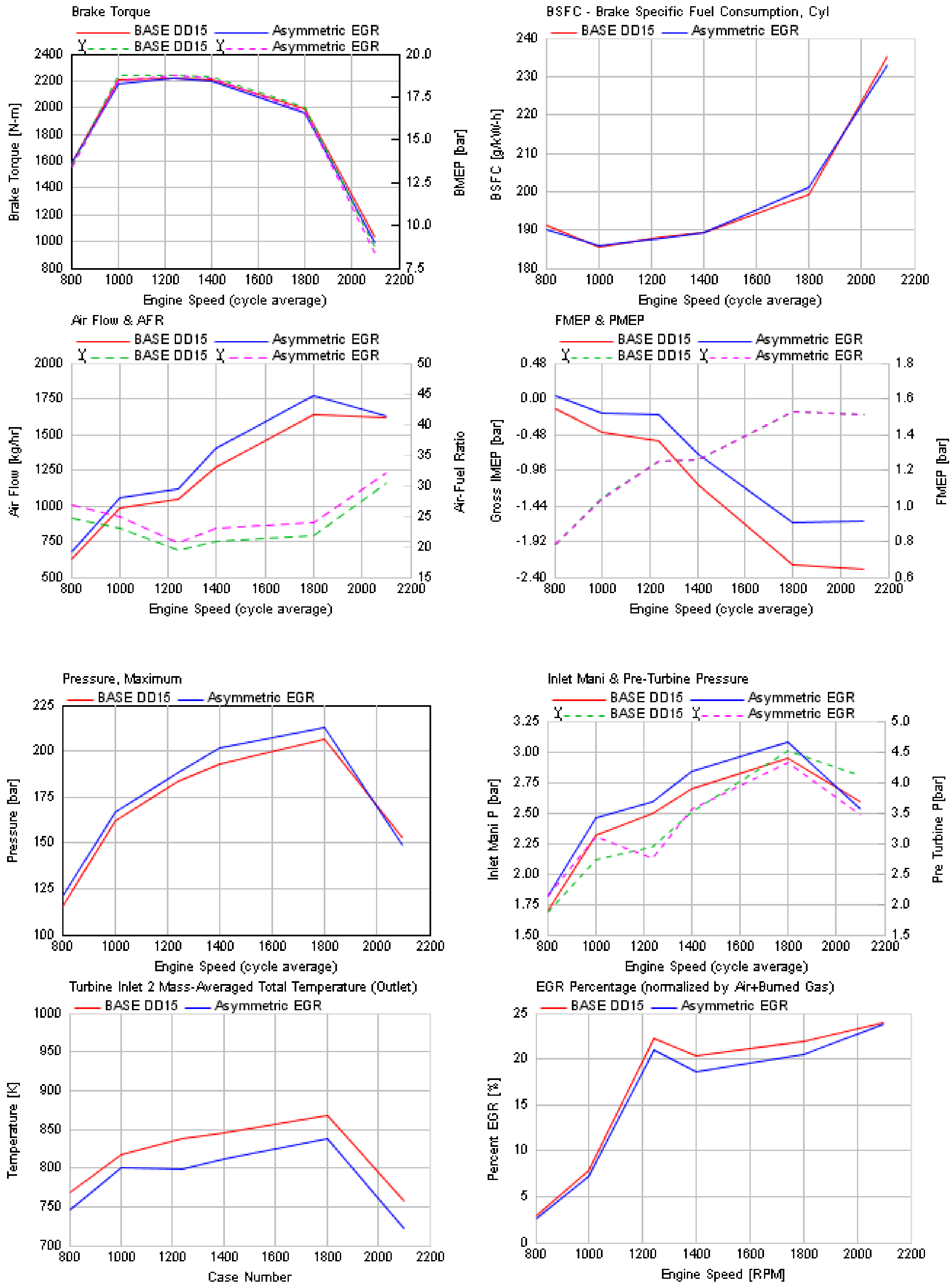


Figure B9b. Effect of Asymmetric Turbo and EGR Supply on Engine Performance at Full Load

1.10 Downspeeding with Increased BMEP

A common approach to improving the drive cycle fuel consumption is engine downspeeding. Typically, the peak torque engine speed is reduced and max torque available is increased. This allows the engine to run at a lower speed for a given vehicle speed. For this study, 2 downspeed cases were assessed, both reducing the peak torque speed to 1000-1200rpm (from 1200-1400rpm) with torque increases of 17 & 33% respectively. The higher torque levels were achieved by increasing injected fuel quantities and resizing the turbocharger and power turbine to match the new, higher power levels. Injection timing was retarded where required to keep cylinder pressure within the 207 bar limit. Compression ratio was left unchanged. In both downspeeding cases, the turbo systems where re-sized to achieve comparable minimum AFR (~19:1) and the same or slightly increased EGR (increased at 1000 rpm to retain NOx control at the lower peak torque speed). Vehicle cruising speeds were reduced from 1368 RPM @ 65 MPH for the baseline case, to 1209 RPM for Downspeed A, and 1051 RPM for Downspeed B.

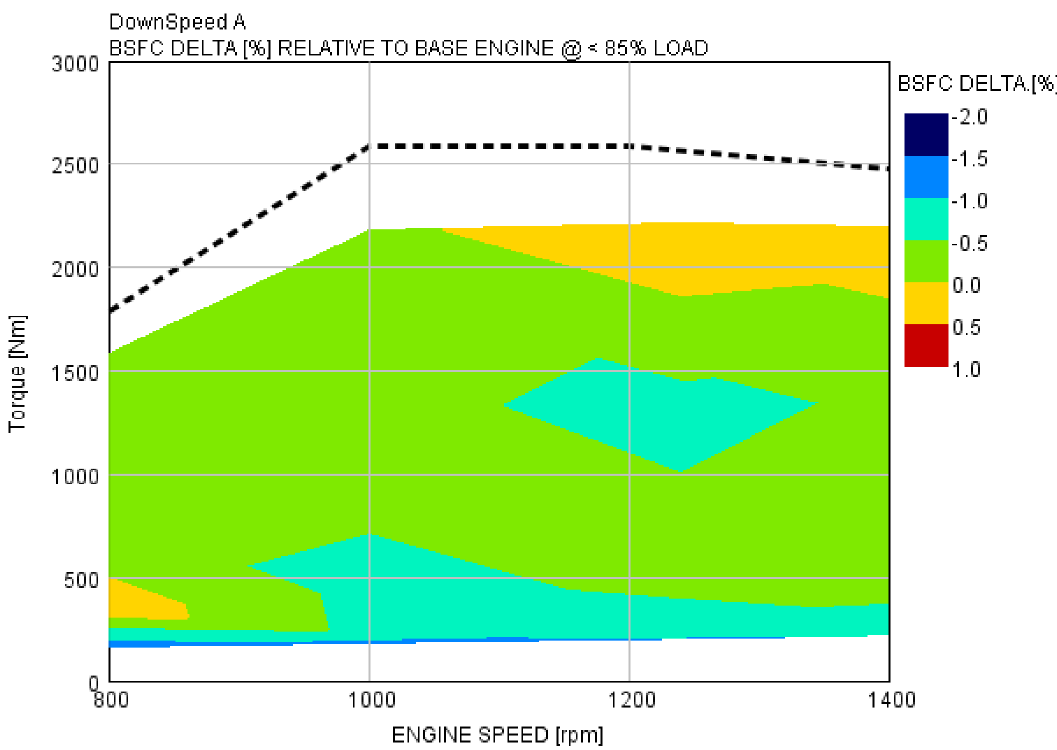


Figure B10a. DownSpeed 'A' BSFC improvement

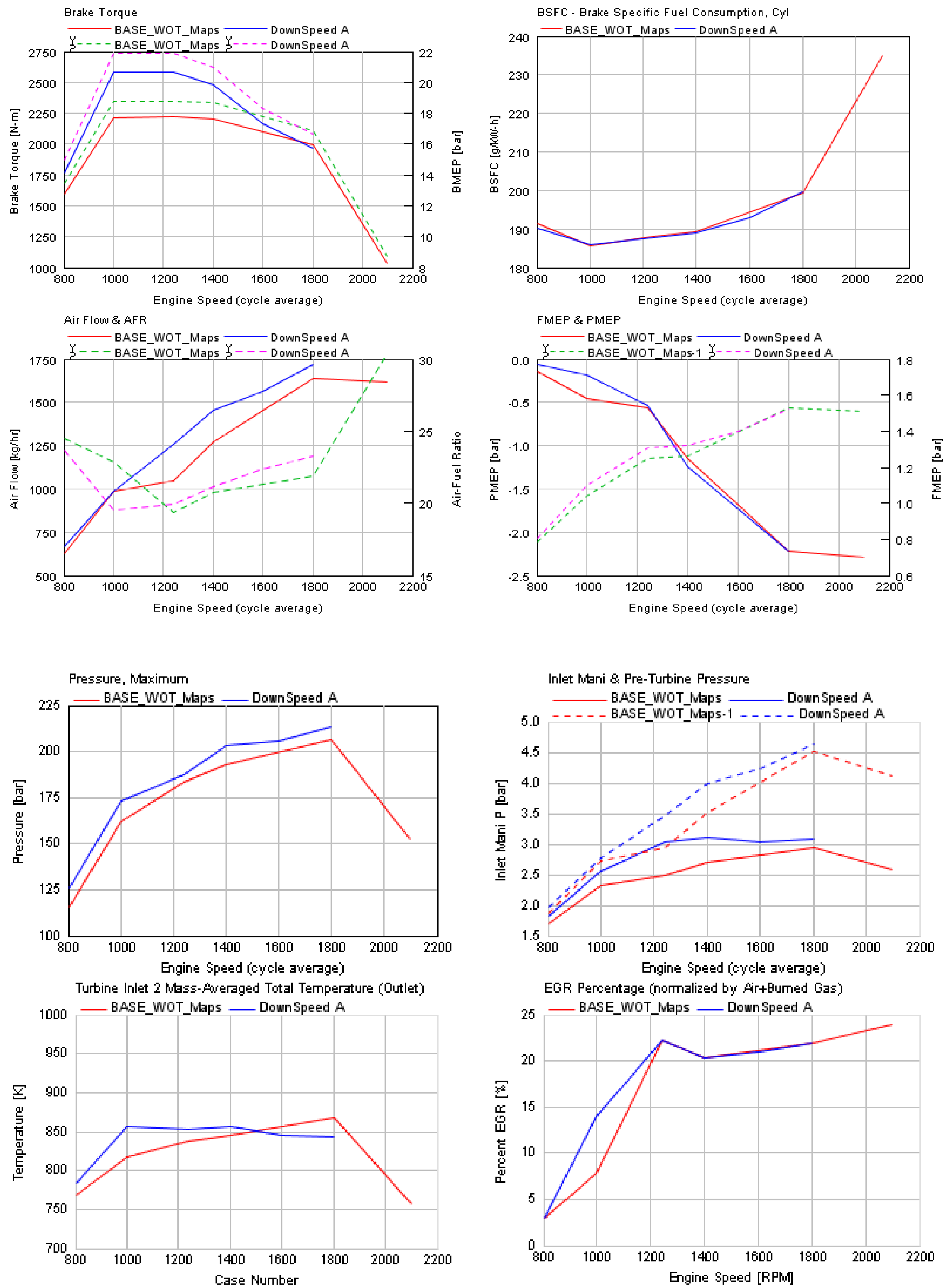


Figure B10b. Effect of DownSpeed 'A' on Engine Performance at Full Load

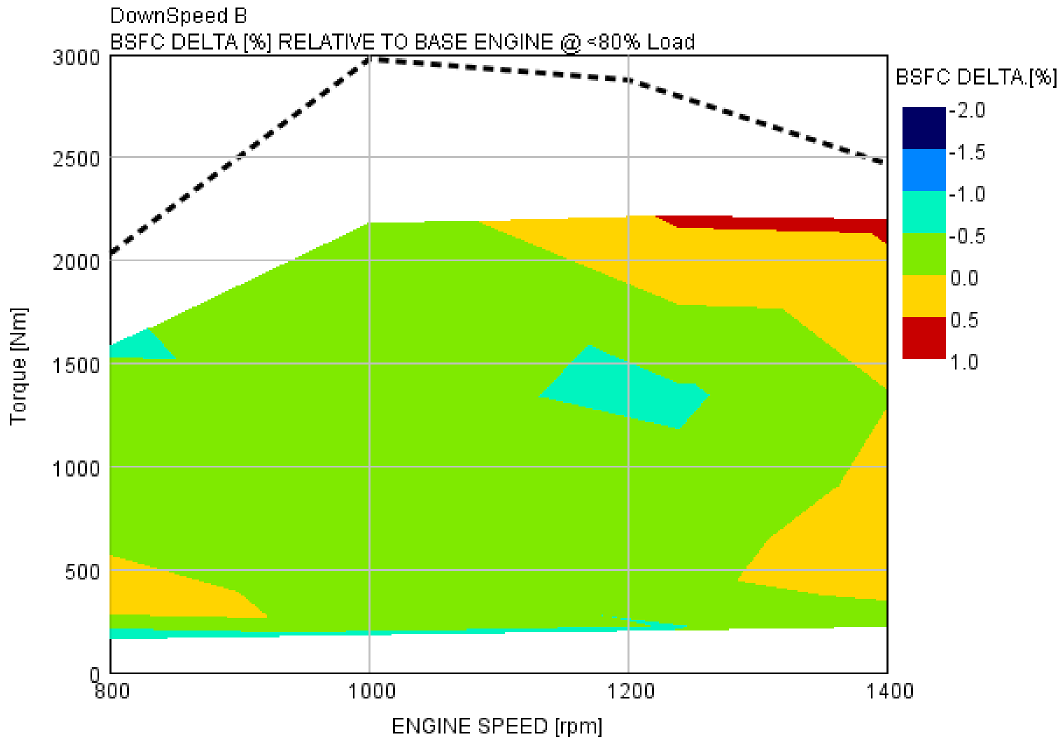


Figure B10c. DownSpeed 'B' BSFC improvement

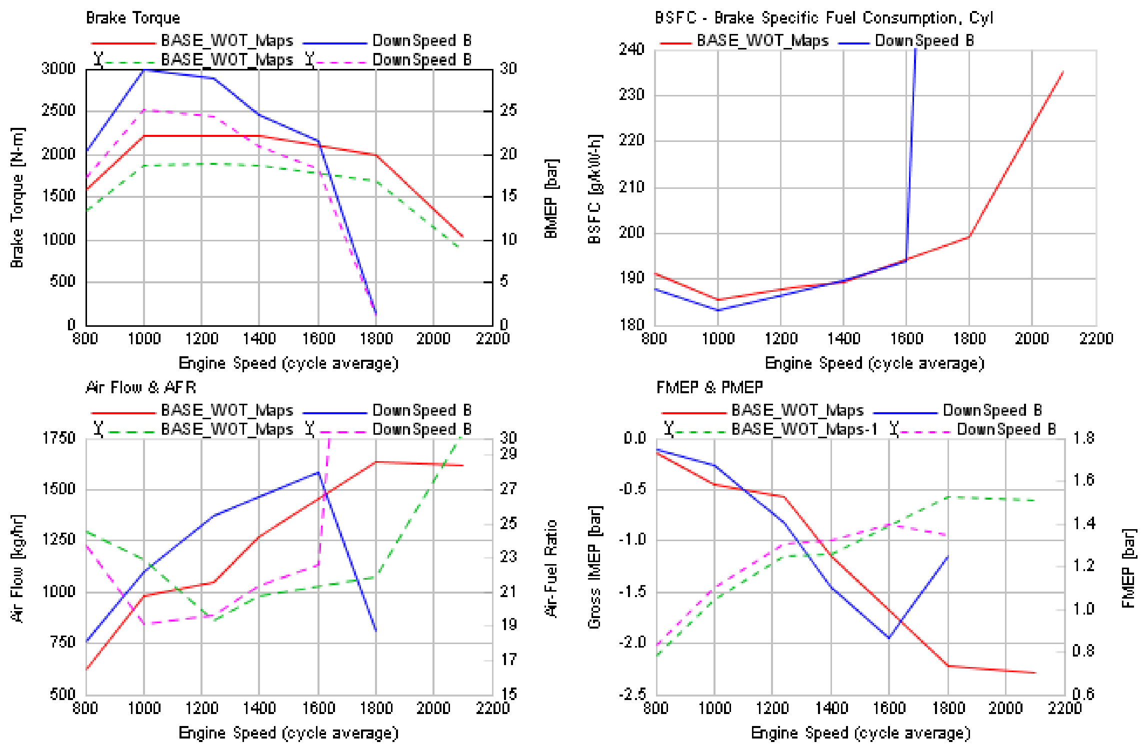


Figure B10d. (Part 1) Effect of DownSpeed 'B' on Engine Performance at Full Load

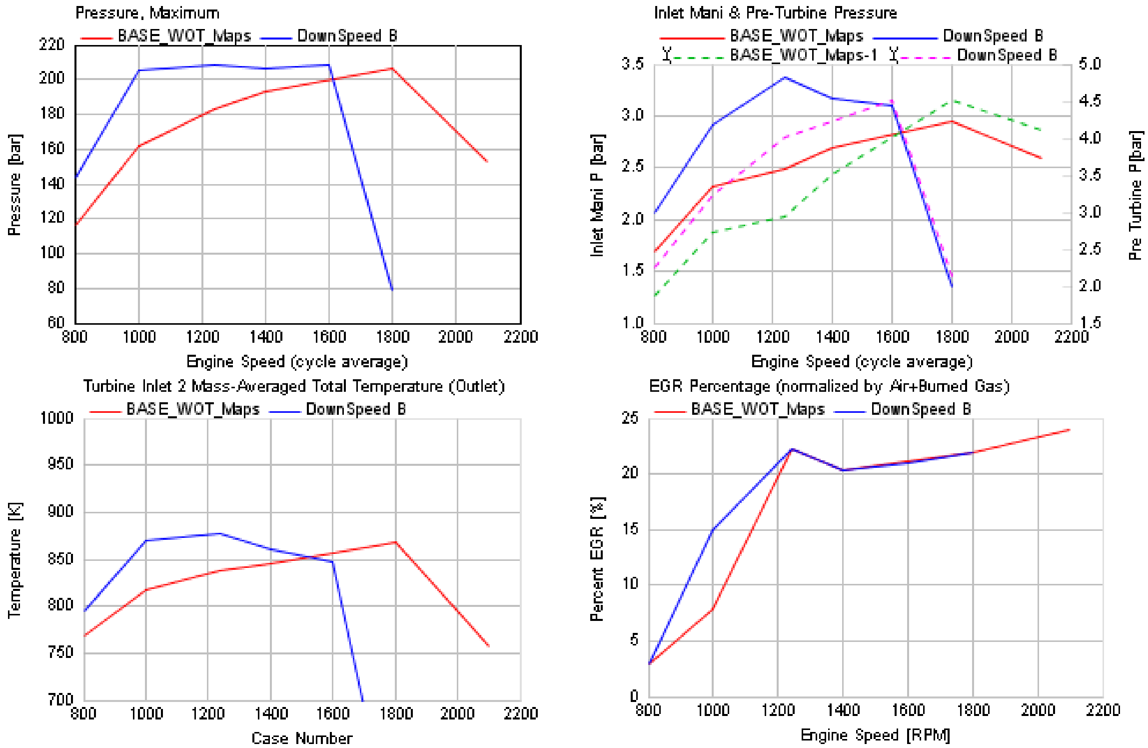


Figure B10d. (Part 2) Effect of DownSpeed 'B' on Engine Performance at Full Load

1.11 Downsizing with Constant BMEP

A 12.5L displacement version of the DD15 was constructed to assess the effect of downsizing. The approach taken was to remove 1 cylinder, so making it a 5-cylinder unit. This method was chosen because it was simpler to reconfigure the manifold layouts rather than re-size all the ports, valves & associated pipework. An ideal approach would be to model a smaller diesel engine, such as the Volvo D13 or the Detroit DD13. Unfortunately, this approach would require purchase and testing of an engine to provide input data for the model, building a new GT-POWER model, and calibrating it. This effort was beyond the scope of the project. In our experience, the performance differences between a 5 and 6-cylinder of comparable displacement will be small. With a 5-cylinder layout, the firing order has a number of options, which were assessed in conjunction with 2 exhaust manifold layout options of split or log type. The best setup, in terms of BSFC, was found to be a log style manifold with a firing order of 1-4-3-2-5.

The FMEP values are the same as for the base engine, so the friction torque or power at a given operating point is reduced by 5/6, due to the smaller displacement and cylinder count.

The engine was run at the same BMEP as the base engine, which results in a lower torque. The turbo was re-sized to achieve comparable Air Fuel Ratios as the base engine, and this has a slightly detrimental effect on PMEP. The start of injection (SOI) was adjusted to achieve the same max cylinder pressure (207 bar) as the base engine. The APT was also maintained and re-sized. Given the requirement for same BMEP, the boost requirements should be broadly the same, which they are. EGR levels from the base engine where maintained to achieve NOx control.

The results show that the down sized engine is marginally better at part load conditions, but worse at higher loads and speeds due to the increased PMEP. This high load BSFC penalty can be assigned to the reduced performance of a turbocharger set up for downsized operation on a log manifold, compared to the more efficient dual entry turbo setup that is used on the baseline 6-cylinder engine. Plot B11a shows the BSFC delta on a like-for-like BMEP basis, while plot B11b show the fuel rate for both the baseline 6-cylinder engine & the downsized engine on a torque scale, to highlight the difference in torque.

In the vehicle simulation, the downsized engine performs better than Figure B11a would lead one to expect. This can be explained using the following logic. At any given point on the drive cycle, the vehicle imposes a certain power demand on the engine. Given the transmission gearing, that translates into a given torque and speed demand on the engine. If the engine is smaller, but still operates at the same speed and torque, the BMEP of the smaller engine will be higher. At lighter loads, BSFC improves rapidly as BMEP increases. Thus, for situations where the baseline engine would operate at a BMEP under approximately 10 bar, there will be a significant improvement in BSFC for the downsized engine, even if it is slightly less efficient at a given BMEP level.

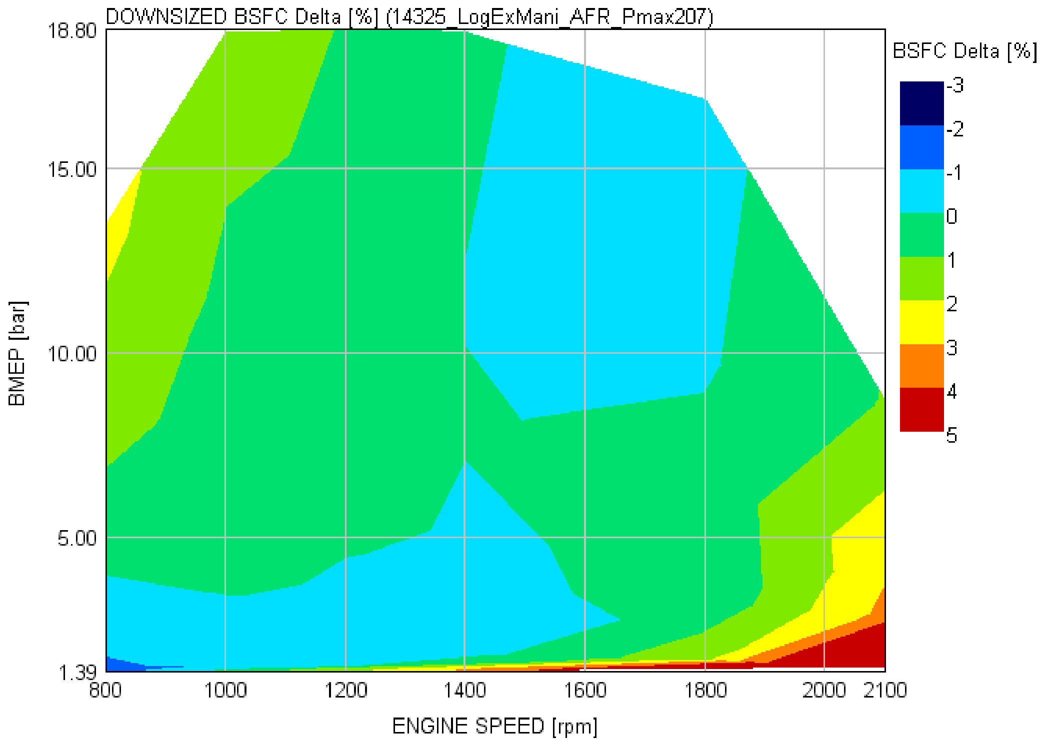


Figure B11a. BSFC improvement of Downsize Engine with Constant BMEP

Figure B11a indicates that over most of the operating range, the efficiency of the downsized engine is within +/- 1% of the 6-cylinder engine, when the two are compared at the same BMEP level. However, the vehicle does not care about BMEP, it cares only about speed and torque, so Figure B11b provides an alternative way of comparing the downsized and 6-cylinder engines. In this figure, BSFC is compared on an equal torque basis. Since the downsized engine has a lower torque curve, the comparison is only shown up to the maximum torque of the downsized variant. This figure shows that there is a BSFC penalty at low RPM and high torque, but the downsized version enjoys a significant BSFC advantage across most of the operating range, especially at light load.

Another way to compare the two engines is to look at the BSFC maps, which are shown in Figure B11c. These look similar at first glance (other than the lower maximum torque of the downsized engine), but a look at the light load portion of the maps (Figure B11d) shows significant advantages for the 5-cylinder.

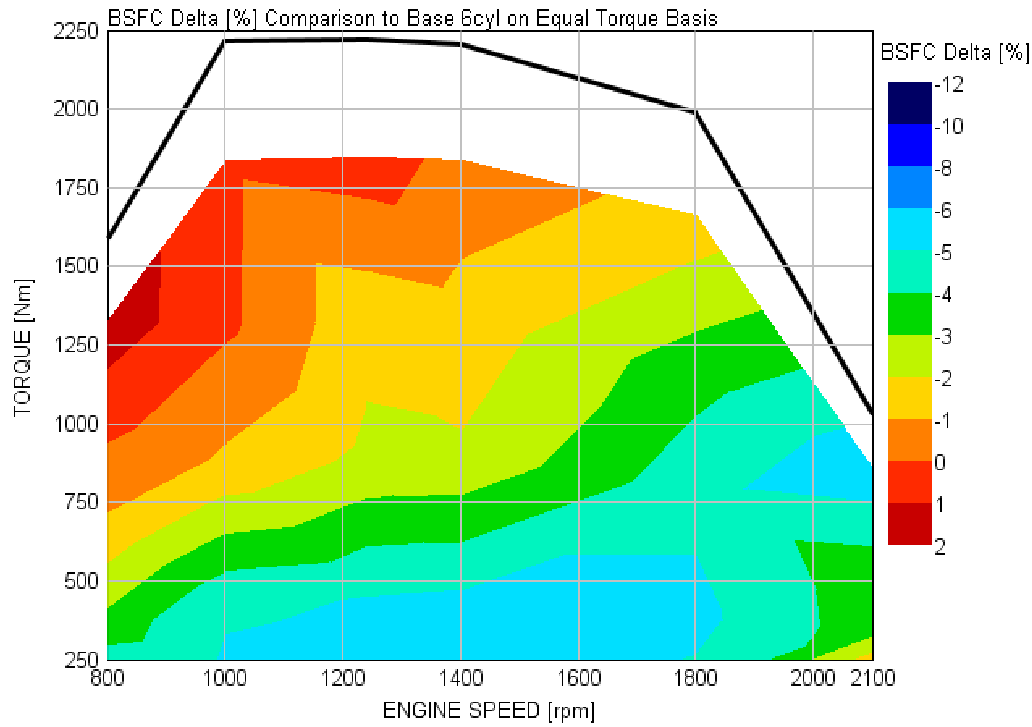


Figure B11b. BSFC improvement of Downsize Engine with Constant BMEP – Downsized Engine, with BSFC interpolated & compared at equivalent Base engine torques

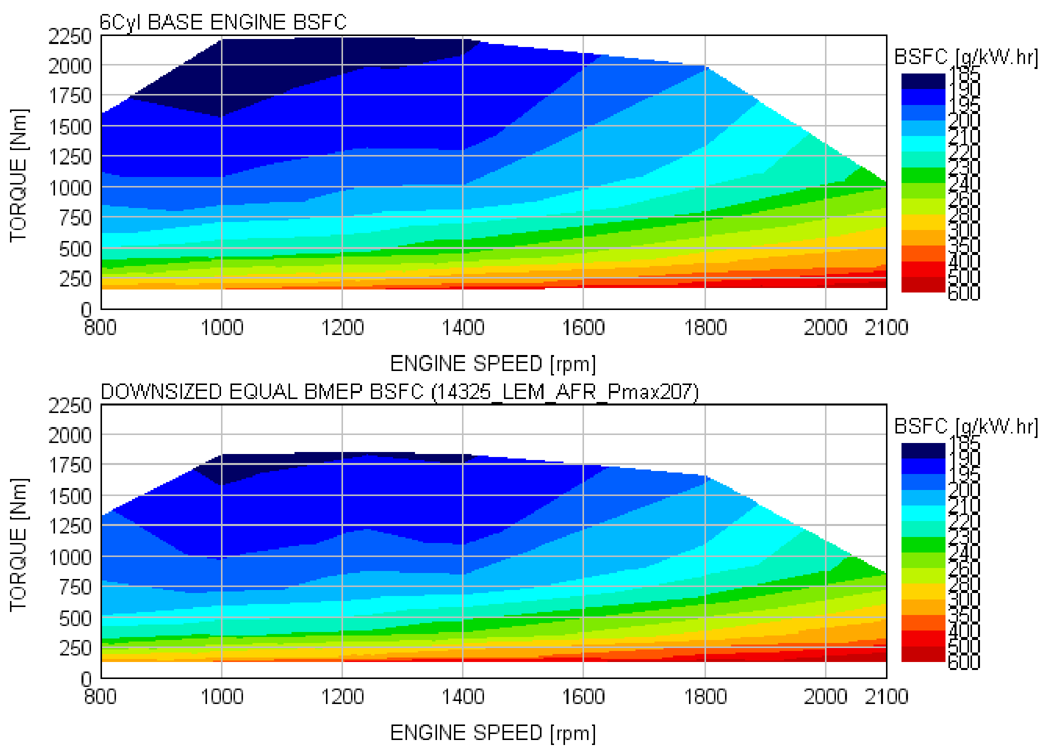


Figure B11c. Comparison of 6 Cyl vs. Downsized BSFC on a Torque basis

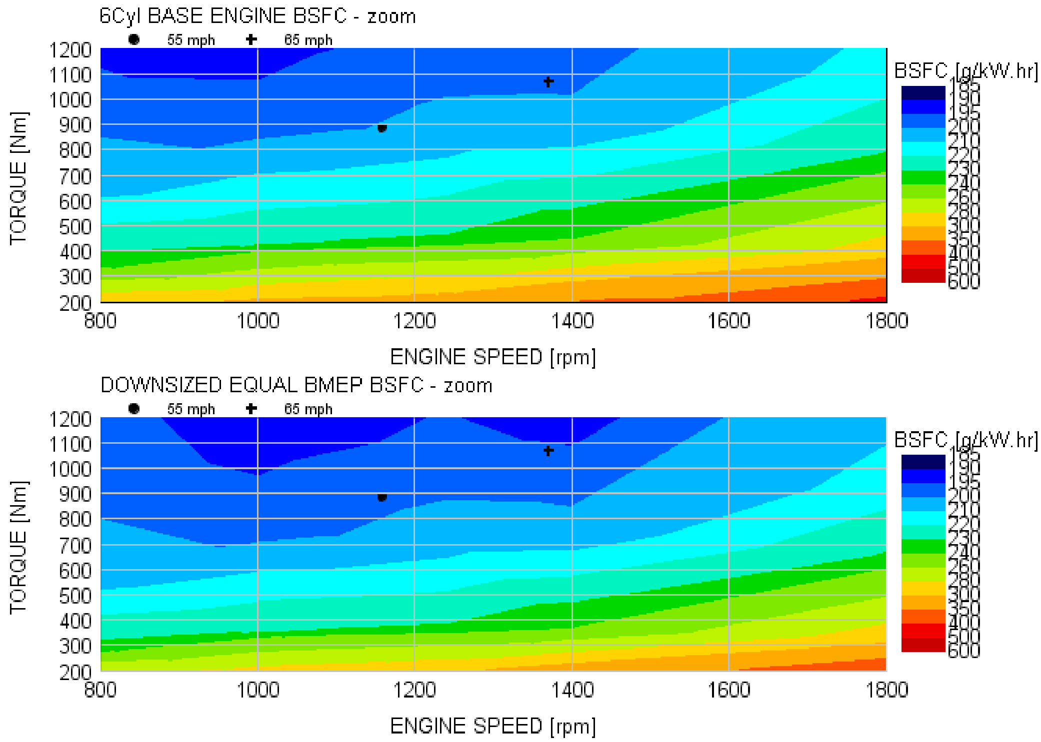


Figure B11d. Comparison of 6 Cyl vs. Downsized BSFC on a Torque basis – zoomed. Note the 55 and 65 MPH operating points marked on both plots. The downsized engine enjoys a BSFC advantage at both points.

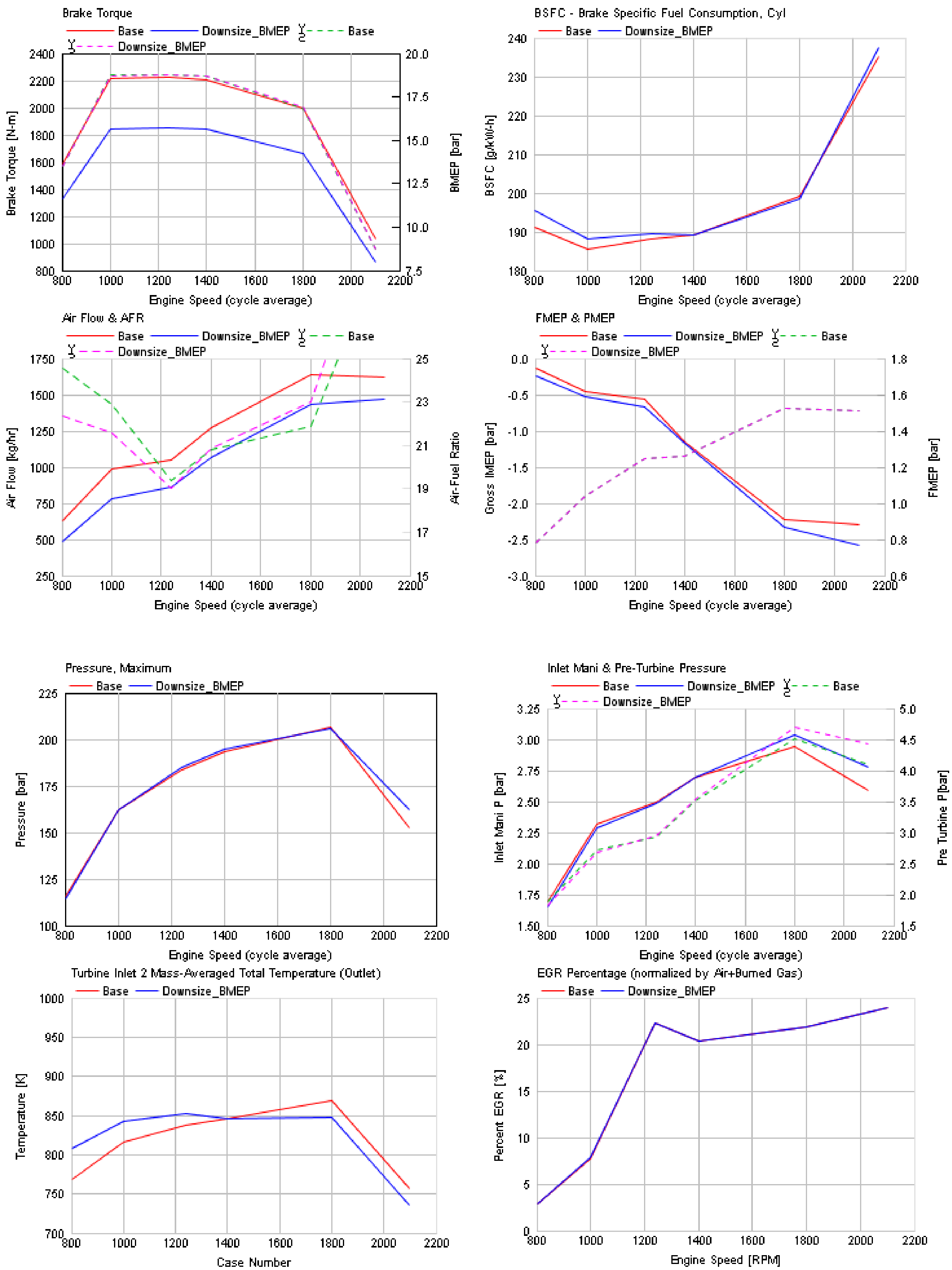


Figure B11e. Effect of Downsized Engine with Constant BMEP on Engine Performance at Full Load

1.12 Downsizing with Constant Torque

The engine model developed in section 1.11 was then run at the same torque levels as the base engine, which results in higher BMEP levels. Again, AFR & max cylinder pressure levels were maintained. Subsequently higher boost levels where required, PMEP is higher and FMEP is also higher as the engine is running at a higher BMEP on the speed-load lookup map.

The results show that the down sized engine is significantly better at part load conditions, but worse at higher loads and speeds due to the increased FMEP & PMEP. Plot B12a shows the BSFC delta on a like-for-like torque basis, while plot B12b show the fuel rate for both the baseline 6 cylinder engine & the downsized engine on a BMEP scale, to highlight the difference in BMEP.

It should be noted that the combustion model is setup so that a fixed set of Wiebe functions are used for a given speed & %load per cylinder condition: for example the 1400 rpm 10% load combustion data is the same for both models even though the value of 10% load per cylinder is 20% higher for the downsized model. At low load conditions the combustion duration can change by several degrees with a small change in load, so this approach introduces some uncertainty in light load fuel consumption values. Additionally, at full load the SOI is adjusted to maintain the 207 bar max cylinder pressure limit. This has the effect of making full load operation slightly less efficient for the downsized engine, compared to the baseline.

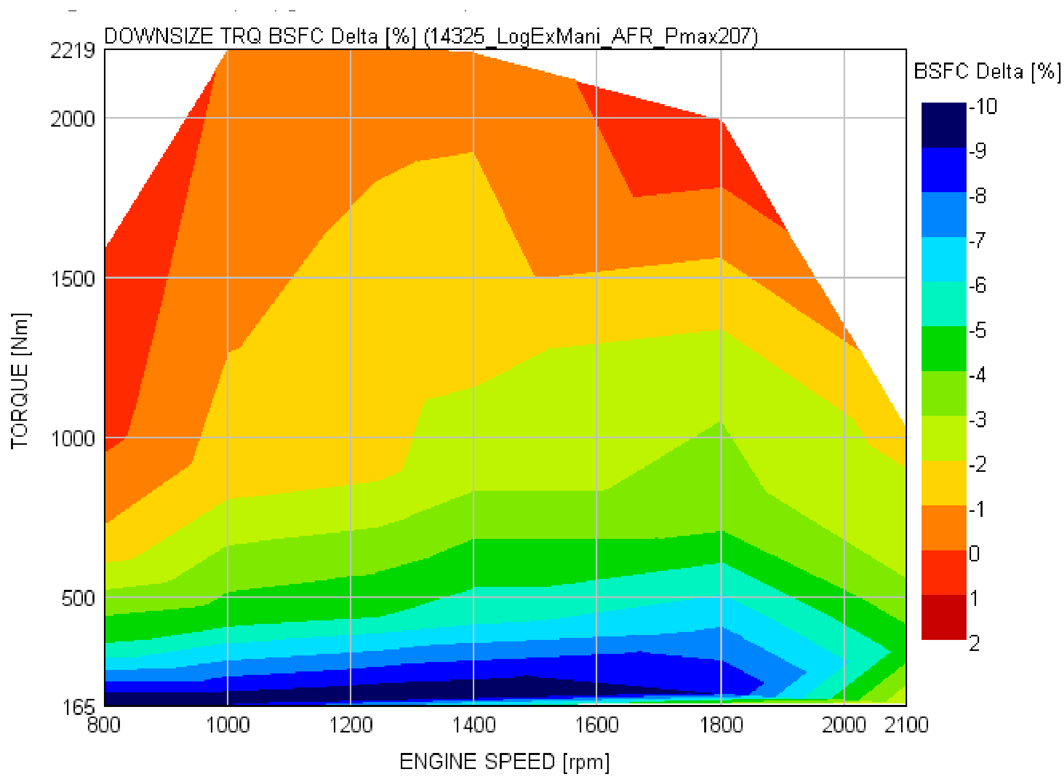


Figure B12a. BSFC improvement of Downsize Engine with Constant Torque

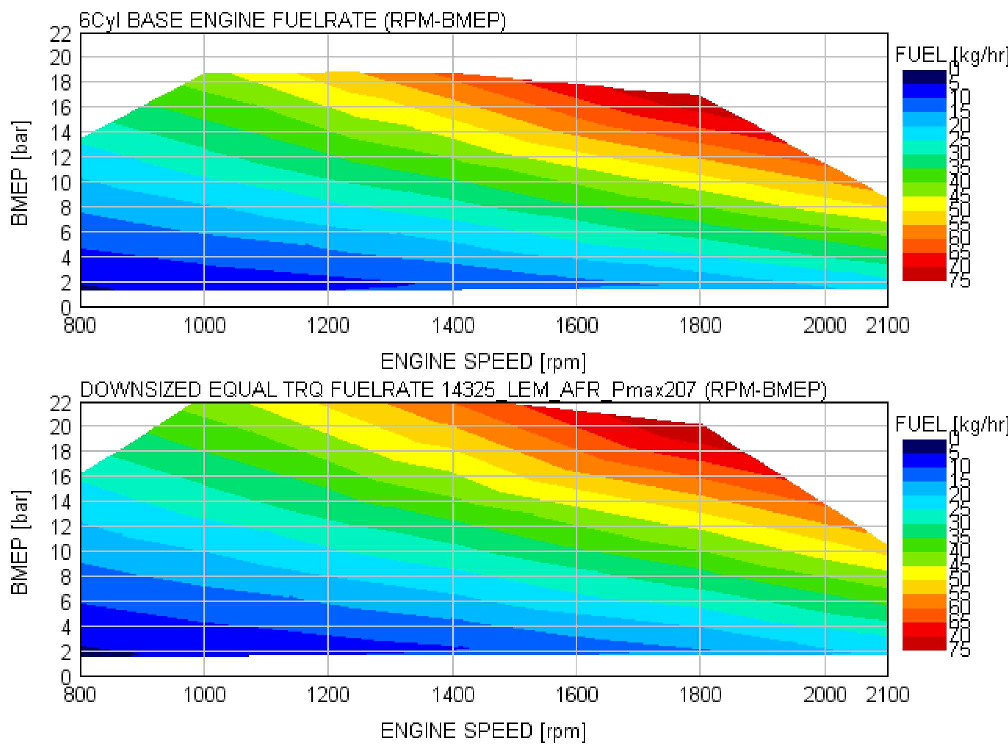


Figure B12b. Comparison of 6 Cyl vs. Downsized Fuel Rate on a BMEP basis

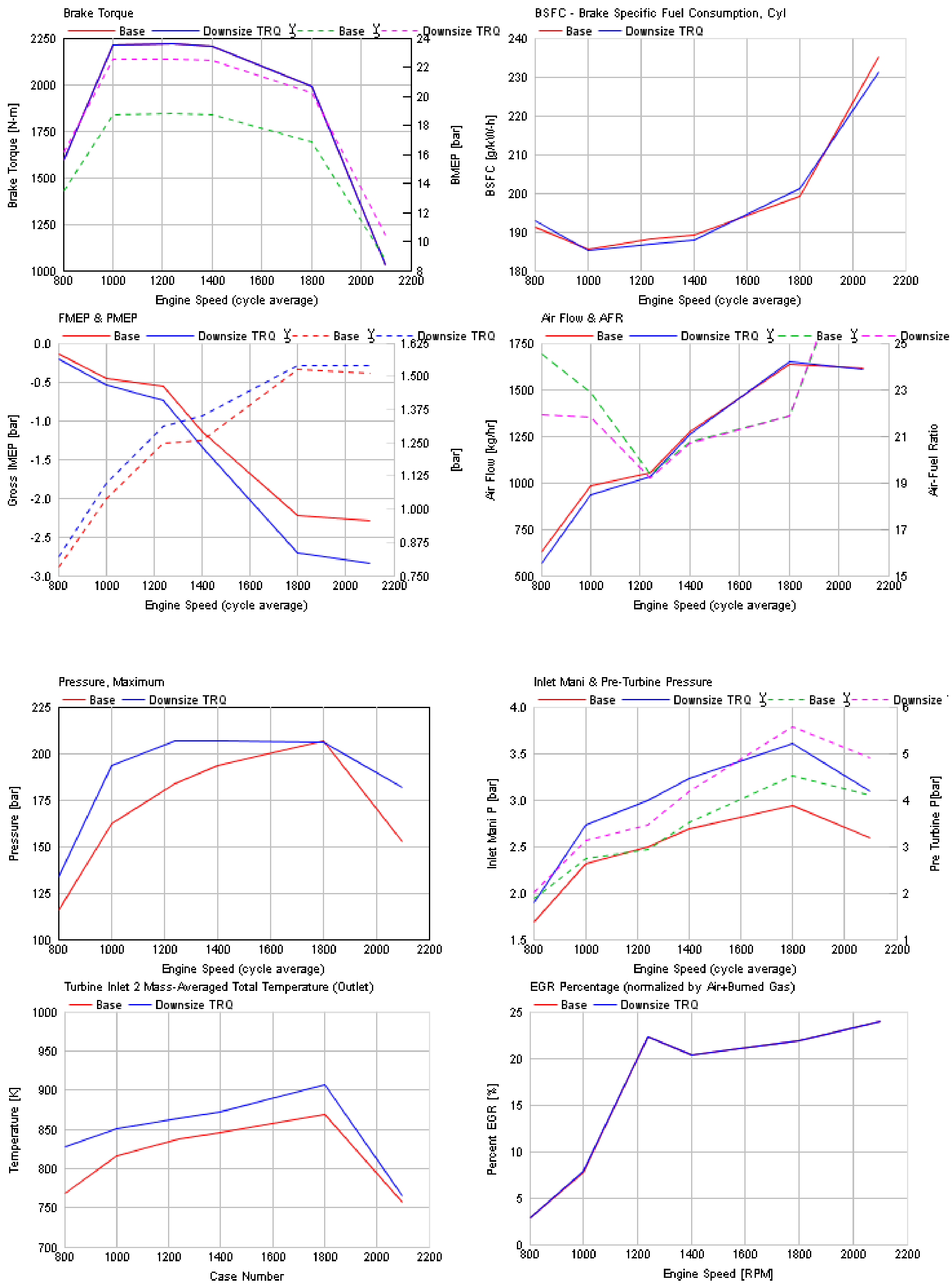


Figure B12c. Downsized Engine Full Load Performance with Constant Torque

1.13 Explore FMEP Improvements

The baseline engine model with a reduced friction map was run at the same conditions of as the baseline model. The friction (FMEP) was reduced on a speed-load relationship, with full load reductions of 10% and part load reduction of up to 35%. As expected, the largest benefits occur at light load and lower engine speeds.

Reducing engine friction can be achieved in several ways, such as low viscosity oils, smaller or dynamic bearings, ring/piston/liner interface improvements, a more efficient fuel pump, as well as smart controlled oil and water pumps. In this study, all the friction improvements where lumped together. Note that these would be a lot of development required to achieve the sort of friction reductions modeled here, and it is not certain that this size of benefit could be achieved without reliability/durability issues.

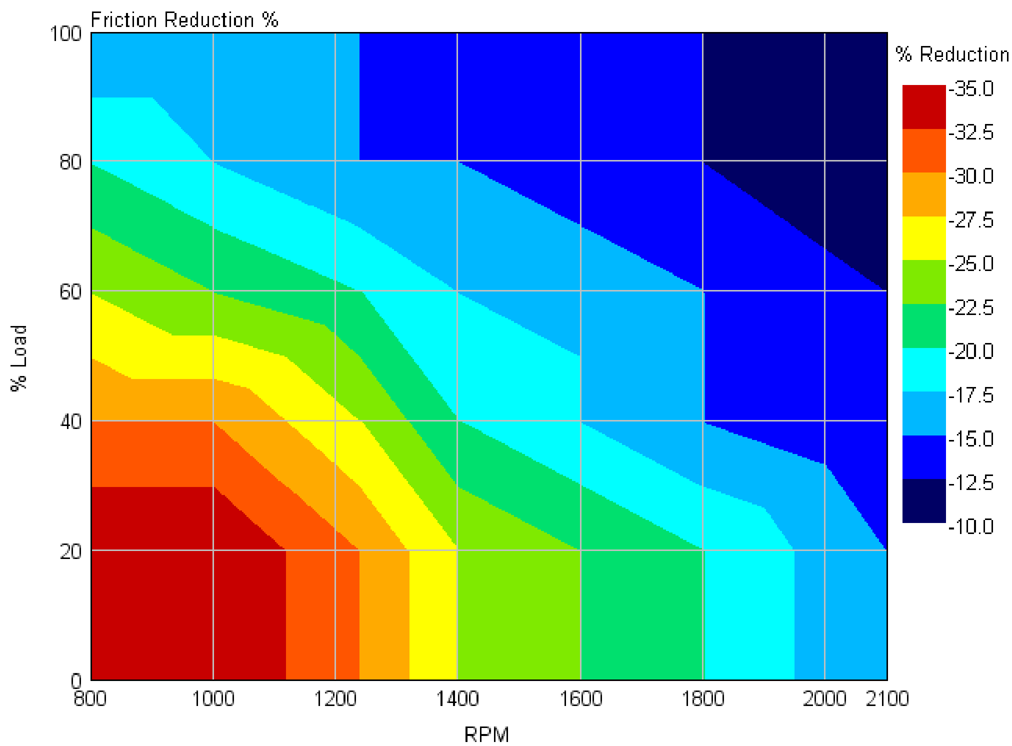


Figure B13a. Friction Reduction Assumptions

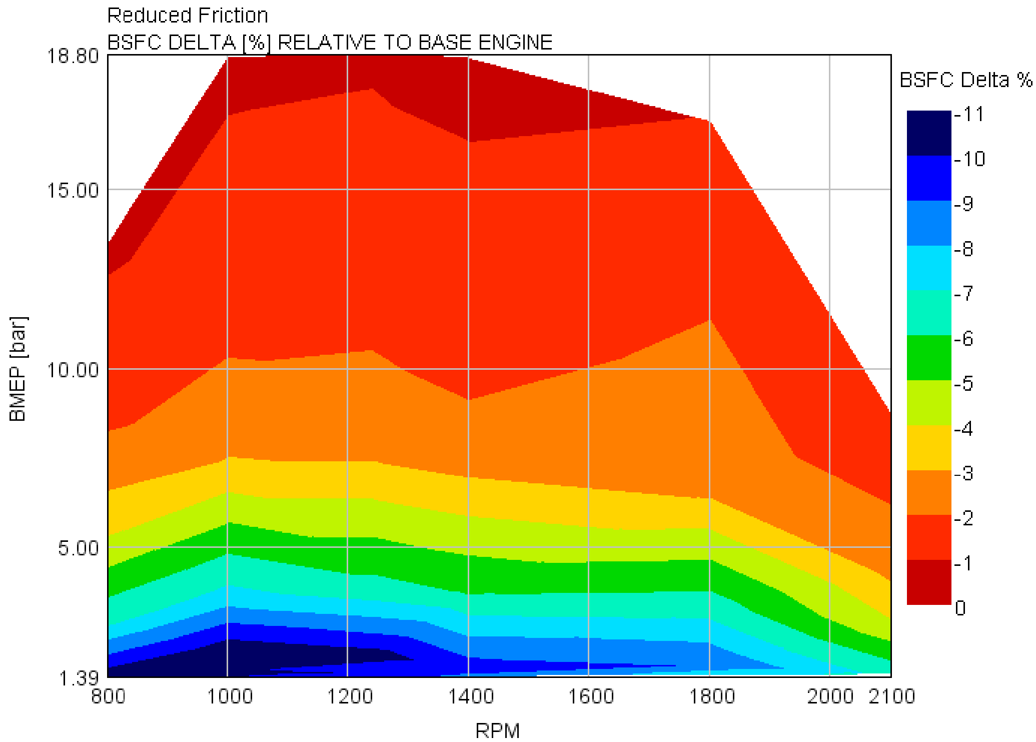


Figure B13b. Friction reduction BSFC Improvement over Baseline

1.14 Explore Turbo Efficiency Improvement

To simulate an improved efficiency turbo, the base engine model was run with a 5% efficiency increase on both the turbine & the compressor maps. If the overall turbocharger efficiency at a given operating point is 50%, this change would increase it by 5% to 52.5%.

Turbo efficiency improvement can be realized by improvements in turbo designs, bearings, and materials. Reduced pumping losses & improved waste heat energy recovery are typical benefits of improved turbo efficiency. For engines that use EGR and do not have turbocompound, there is limited scope for improving turbo efficiency. In these engines, which represent almost the entire market, improved turbocharger efficiency will result in a loss of EGR flow and thus a loss of NOx control. The results presented are run with no EGR demand on the engine and compared to a similar setup of the base engine.

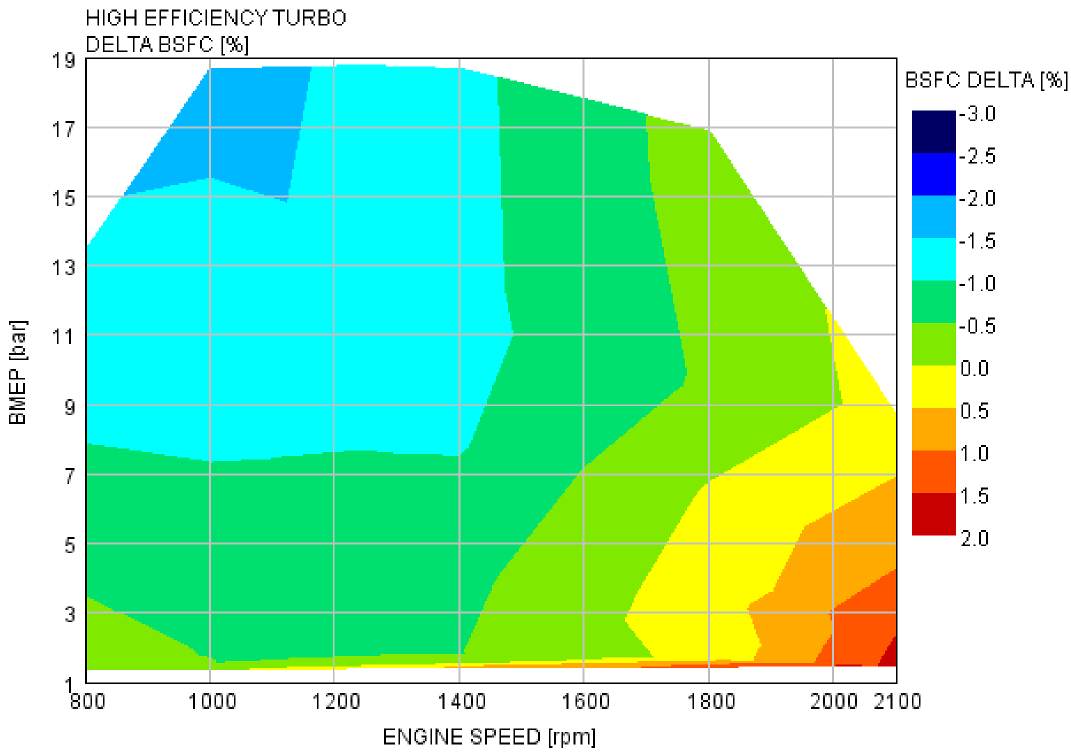


Figure B14a. BSFC improvement from High Efficiency Turbo

Note that the results above are compared to the base engine turbo setup with no EGR which has a 3-4% BSFC penalty at high speed & low load conditions, so the penalty seen in the graph above is actually an improvement compared the base engine (with EGR)

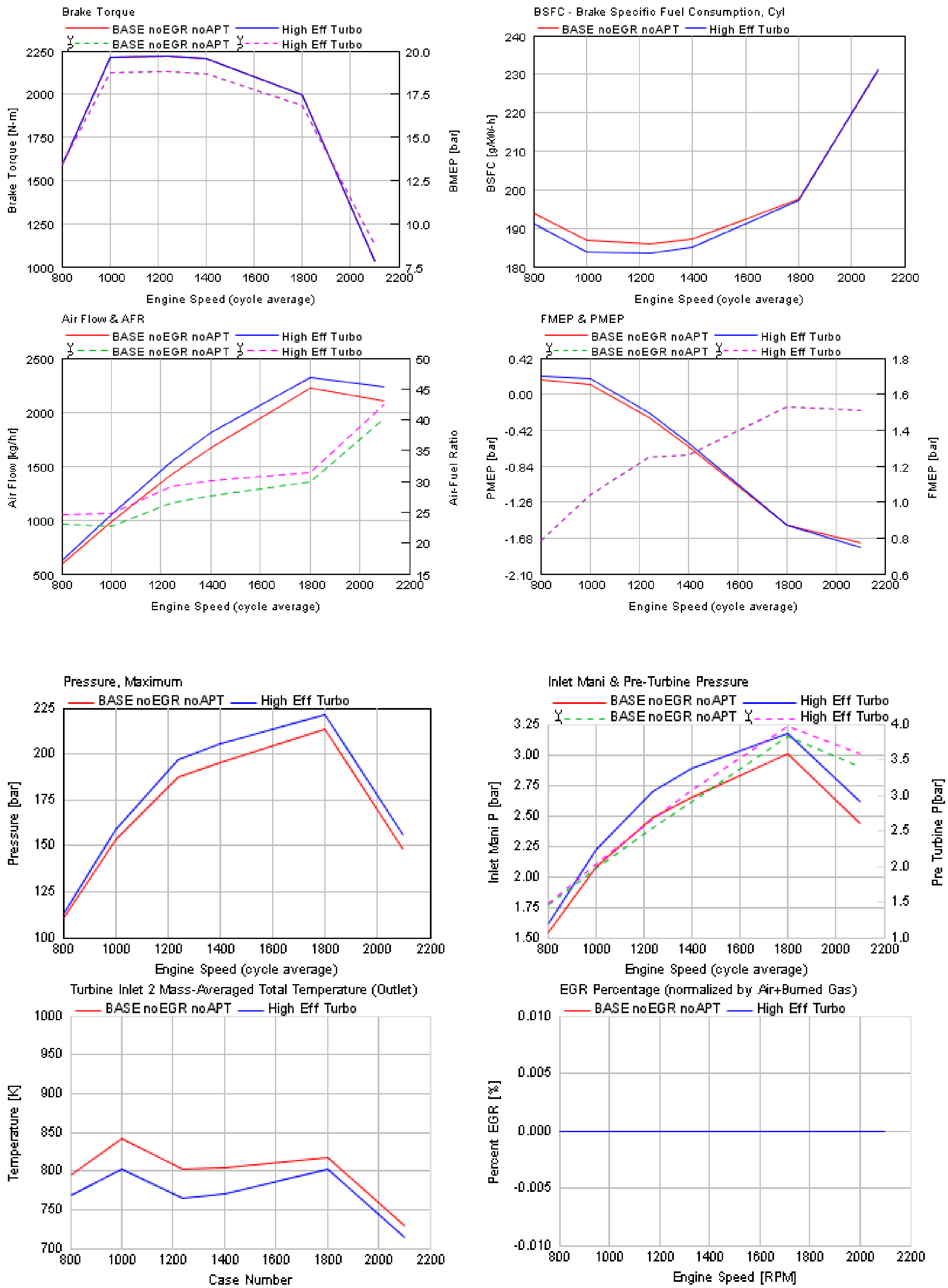


Figure B14b. Effect of High Efficiency Turbo on Full Load Engine Performance

1.15 Explore Variable Valvetrain Actuation Improvements

The application of variable valvetrain actuation (VVA) was investigated, with a full control system assumed, where both lift & duration and timing were adjustable. Previous experience and analysis of this type of technology on large diesel engines had shown that minimal performance gains were achievable. On the DD15 model, the mid speed & load range operating conditions were targeted for best gains on the drive cycles. The results indicated that a maximum of 1% BSFC improvement was achievable, at the conditions simulated. Given the complexity and cost of such a system it was decided not to perform a full speed & load mapping, which would have been very time consuming, and concentrate on other technologies. However, it is believed that VVA could be applied to the exhaust valvetrain to improve emissions equipment performance through exhaust gas temperature management.

2.1 ISB Medium Duty 300bhp Baseline Validation

- **Basic engine specification**
    - 6.7L inline 6 cylinder
    - Single Variable Geometry Turbo
    - 4 valves per cylinder
    - Diesel
    - 225 kW @ 2500 rpm
    - 900+ Nm @ 1300-2200 rpm

GT Power model was built using the geometry of an earlier generation (2007) ISB engine previously benchmarked at SwRI and run in a test cell. The main engine geometry – ports, manifolds, valve data etc. – was the same as the newer target engine (2011). Test high-speed data was used to give a guideline for the combustion but due to the different EGR system of the older engine, was not directly used. Further test data, but not high-speed, was taken from a vehicle running the 2011 specification engine at SwRI. A full speed/load range data set was taken, sufficient to calibrate the GT model.

The GT Power model was then run over a speed range of 1000 – 2500 rpm at 10, 20, 40, 60, 80 & 100% load and validated by the following operating parameters:

- **Torque, BMEP**
- **BSFC**
- **Air and Fuel Flow**
- **Pre/Post compressor pressures and temperatures**
- **Pre/Post Turbine pressures and temperatures**

Unfortunately, the VGT turbo & compressor maps where not available, so a VGT data set from a different engine was used and scaled to achieve the required performance, matching inlet & outlet conditions and airflow. The main issue with this approach was that the maps used did not have the correct performance range required, so the VGT turbine & compressor maps had to be scaled differently depending on speed & load conditions. This method is not ideal but did produce acceptable overall model correlation. This approach is acceptable, since all technology comparisons are done in a relative basis as opposed to providing absolute values of fuel consumption. When the different technologies were applied and a turbo re-size was necessary, the baseline scaling factors where globally multiplied (as opposed to on a speed-load basis).

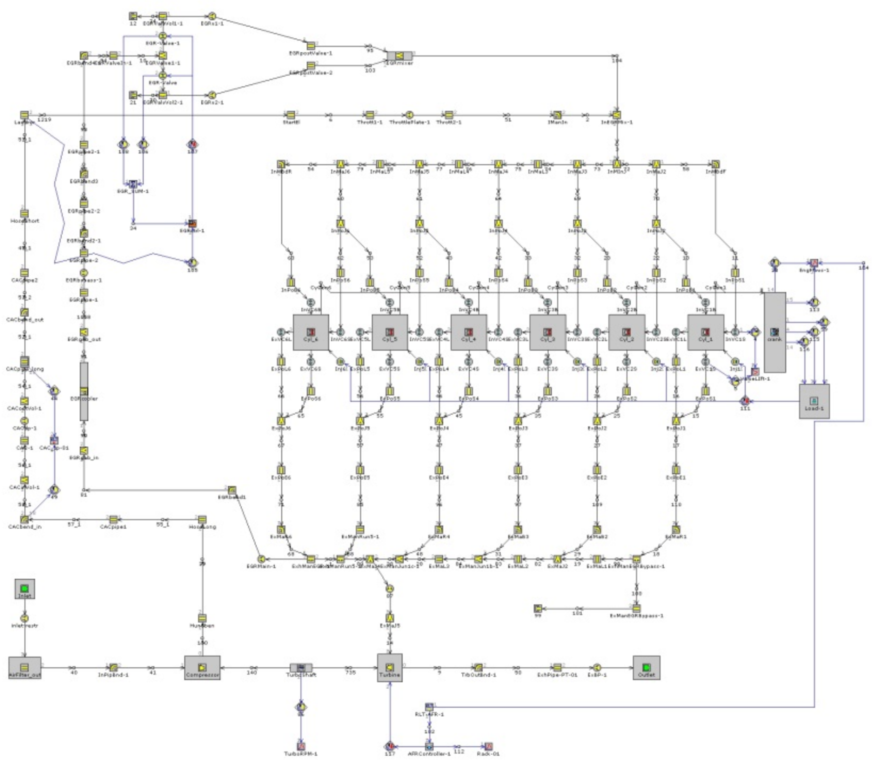


Figure B2.1a GT Model

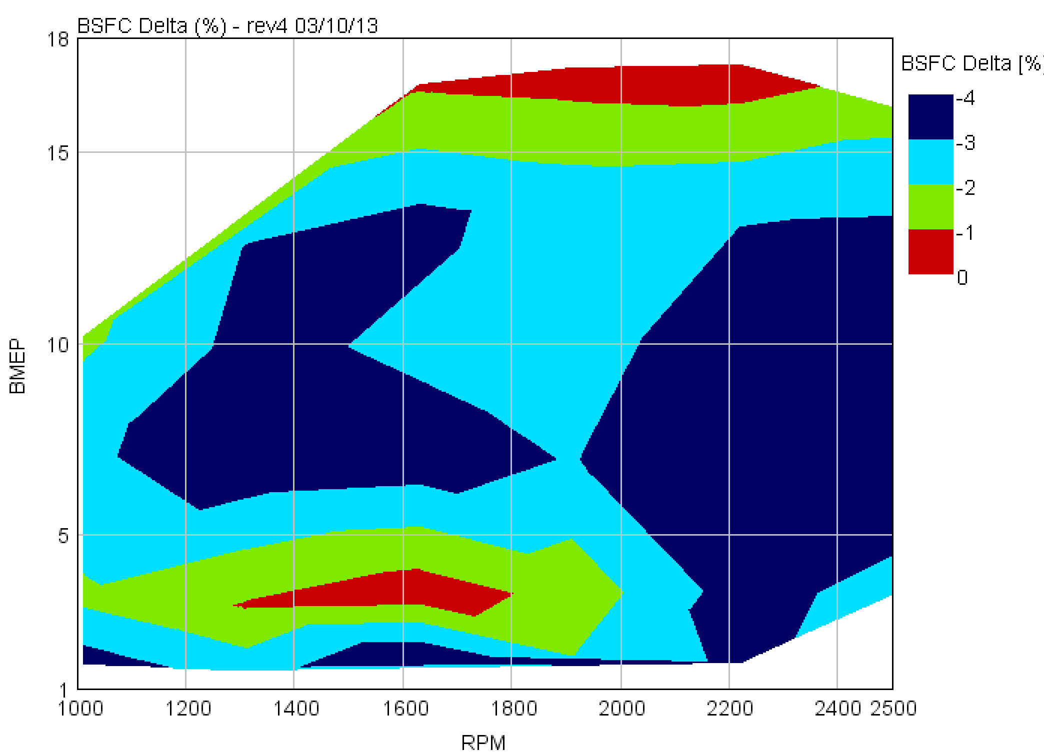


Figure B2.1b Baseline Model BSFC error

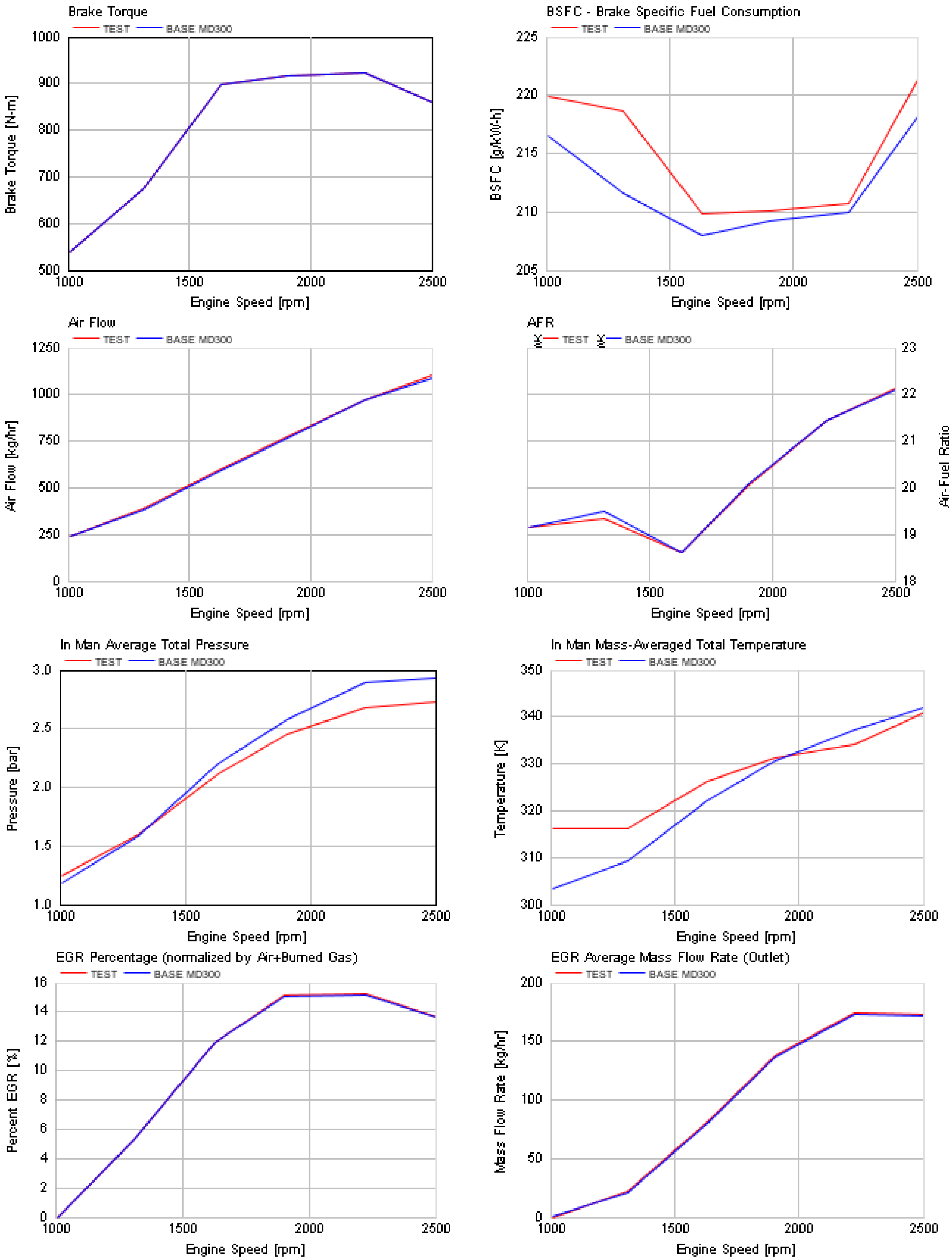


Figure B2.1c (Part 1) Baseline Model Data Correlation at Full Load

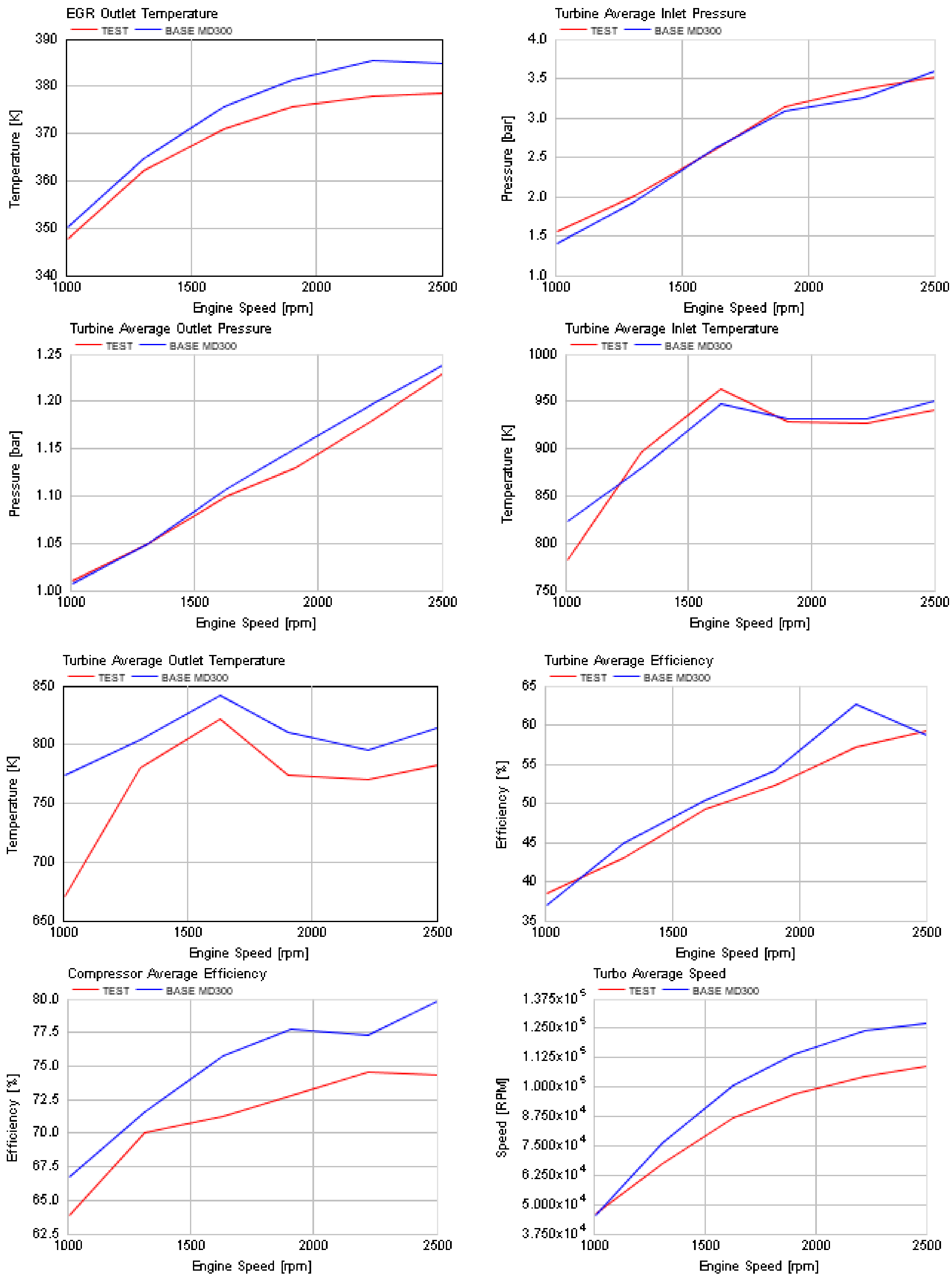


Figure B2.1c (Part 2) Baseline Model Data Correlation at Full Load

2.2 Reduced Exhaust Backpressure

The aim of running the model with reduced exhaust restriction is to simulate the effect of improved flow after-treatment systems for emissions control and the resulting reduction in pumping losses. The baseline GT model was originally setup without modeling the emissions control hardware, but the effective backpressure seen in the exhaust at the VGT exit is simulated by the use of a simple orifice. To model the reduced exhaust backpressure the orifice is resized to achieve the desired target backpressure at the rated condition. In this case, the backpressure has been reduced by 50%, from 24 to 12 kPa. The effect on the engine BSFC is shown in Figure B2.2a. Assuming that the after-treatment effectiveness is retained, there will be no effect on either engine-out or tailpipe emissions of criteria pollutants.

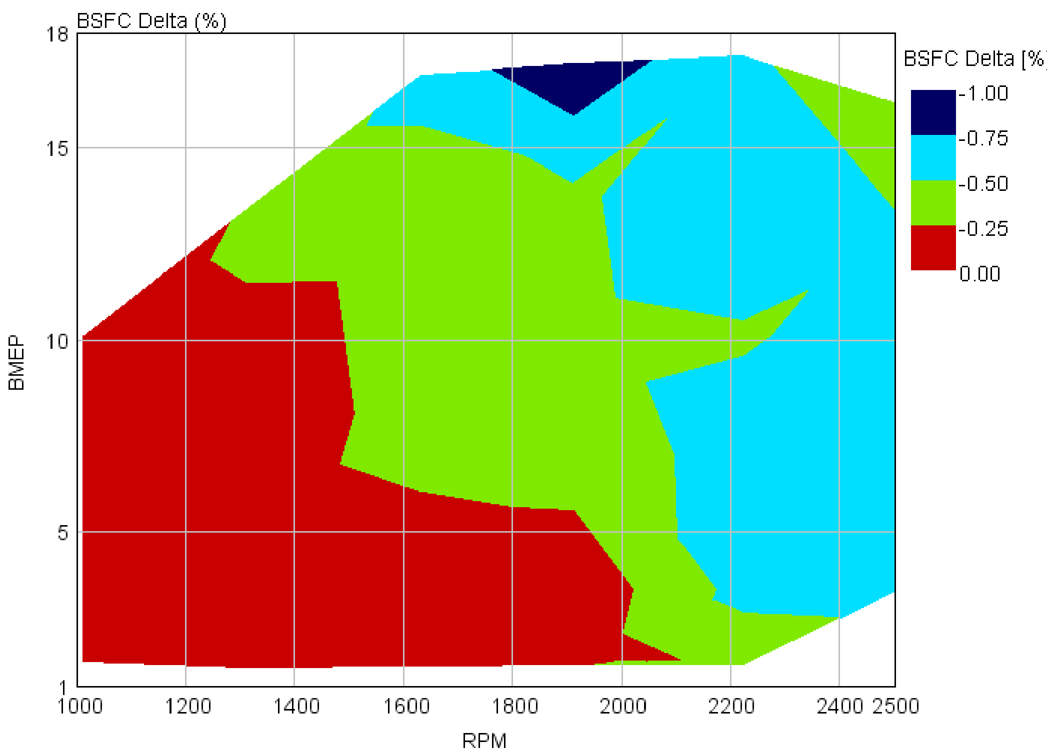


Figure B2.2a Reduced Exhaust backpressure BSFC Improvement

2.3 No EGR

The base ISB engine features a high pressure EGR circuit, with a pre-turbine exhaust manifold supply, an EGR cooler with a bypass valve for when cooling is not required. The valve unit is integrated into the inlet supply system that features an annular ring, which feeds into the inlet pipe pre manifold. To assess the engine performance with no EGR demand, the entire EGR system set of system components was removed from the GT model. The engine was run with the same performance targets but an increased AFR limit (+2 ratios), and the combustion effectively reduced in duration. This technology would involve very high engine-out NOx, and require a very efficient SCR system with a high urea consumption rate.

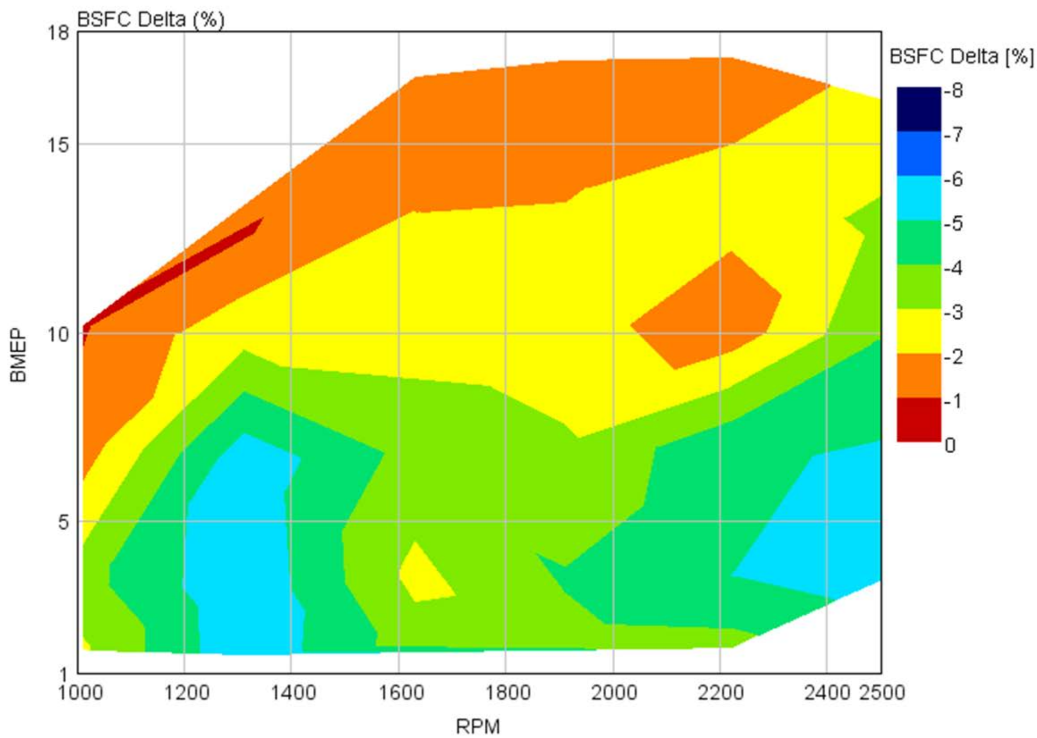


Figure B2.3a. BSFC improvement from EGR removal

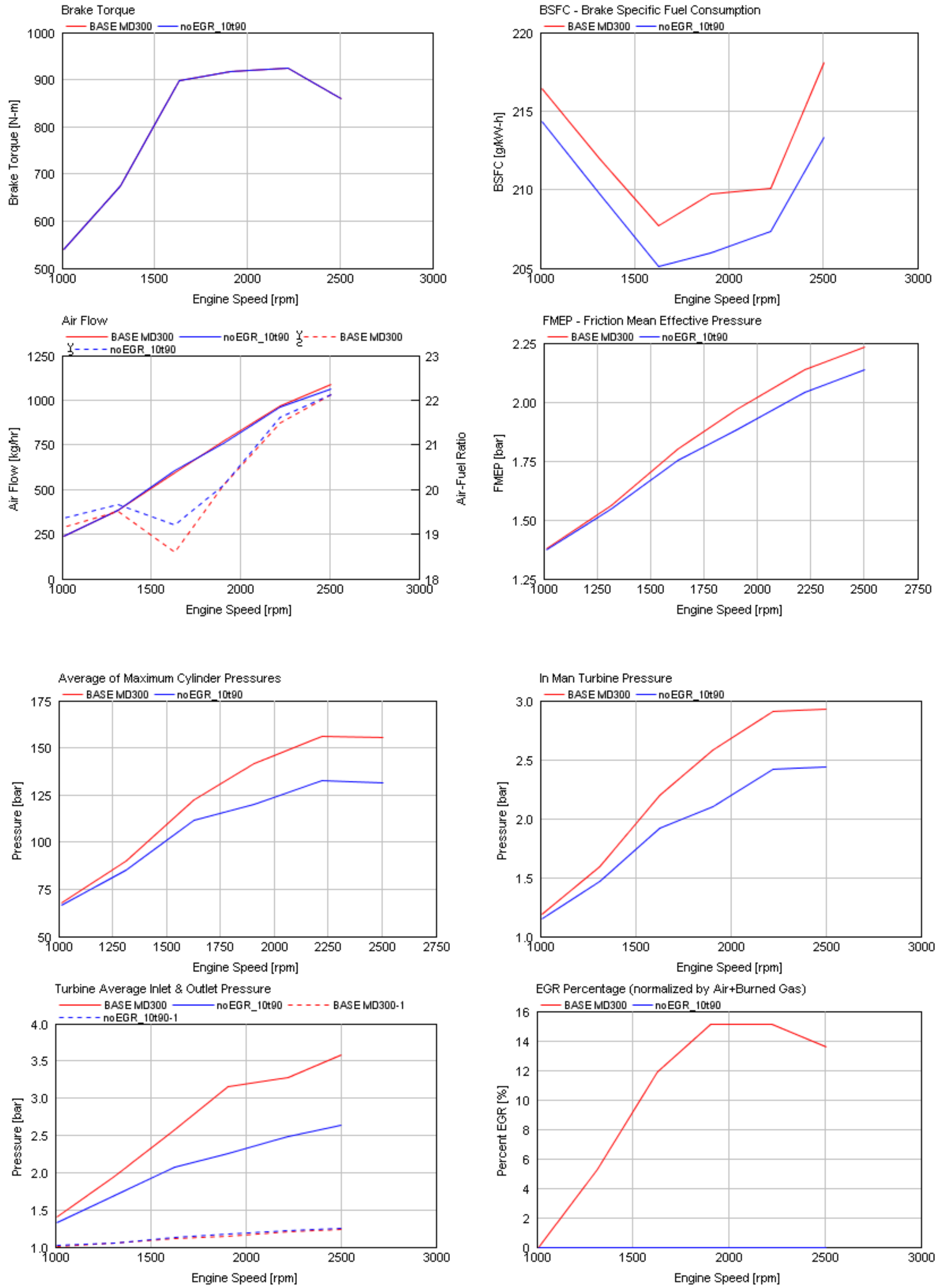


Figure B2.3b. Effect of EGR Removal on Full Load Engine Performance

2.4 Explore Turbo Efficiency Improvement

To simulate an improved efficiency turbo, the base engine model was run with a 5% efficiency increase on both the turbine & the compressor maps. In this case, where the actual engine turbo maps were unavailable and other maps where used and scaled to suit, these maps have been additionally (and globally) scaled to achieve the 5% improvement in efficiency. In a situation where the overall turbocharger efficiency is 50%, a 5% improvement results in a 52.5% overall efficiency. Turbo efficiency improvement can be realized by improvements in turbo designs, bearings, and materials. Reduced pumping losses & improved waste heat energy recovery are typical benefits of improved turbo efficiency. For engines that use EGR there is limited scope for improving turbo efficiency. In these engines, which represent almost the entire market, improved turbocharger efficiency will result in a loss of EGR flow and thus a loss of NOx control. In the case of this engine, there is a reduction in EGR flow with a 5% improvement in turbo efficiency. This will result in higher engine-out NOx, and a greater reliance on aftertreatment conversion efficiency.

The comparison shown is against the baseline engine operating with EGR, unlike the DD15, where the more efficient turbo result was compared against the version with no EGR.

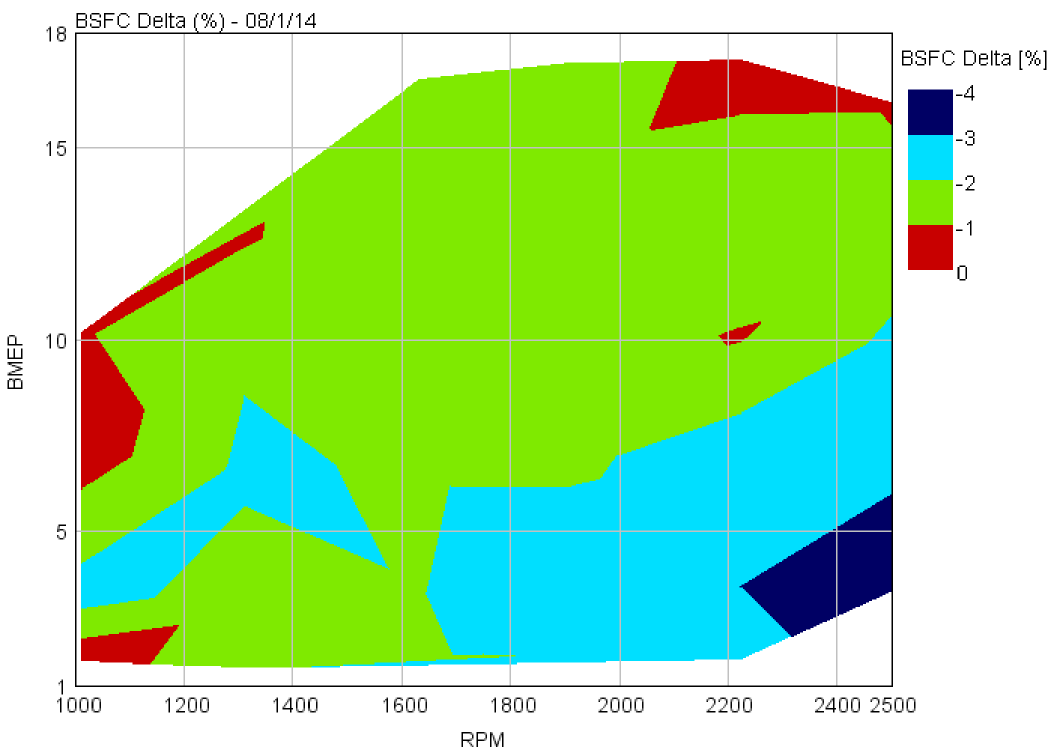


Figure B2.4a. High Efficiency Turbo BSFC Improvement

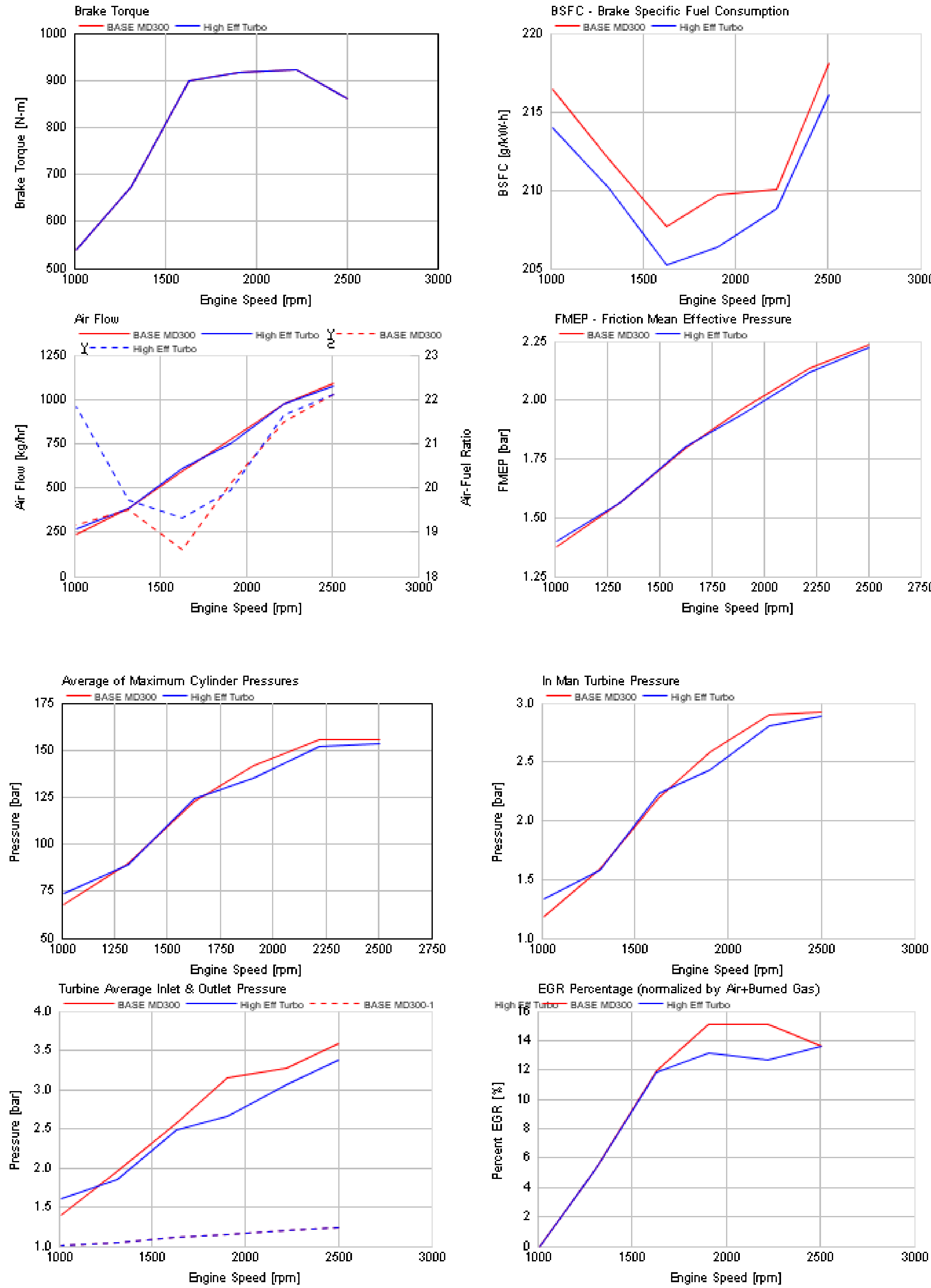


Figure B2.4b. Effect of High Efficiency Turbo on Full Load Engine Performance

2.5 Explore FMEP Improvements

The baseline engine model with a reduced friction map was run at the same conditions of as the baseline model. The friction (FMEP) was reduced on a speed-load relationship, with full load reductions of 10% and part load reduction of up to 35%. As expected, the largest benefits occur at light load and lower engine speeds.

Reducing engine friction can be achieved in several ways, such as low viscosity oils, smaller or dynamic bearings, ring/piston/liner interface improvements, a more efficient fuel pump, as well as smart controlled oil and water pumps. In this study, all the friction improvements where lumped together.

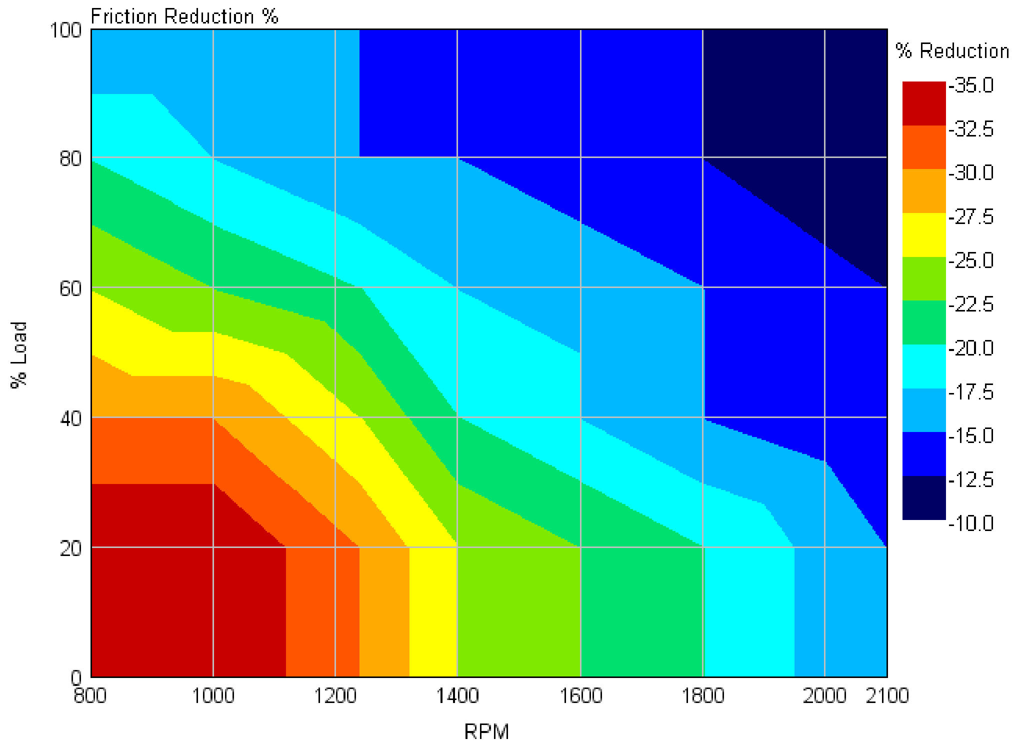


Figure B2.6a. Friction Reduction Assumptions

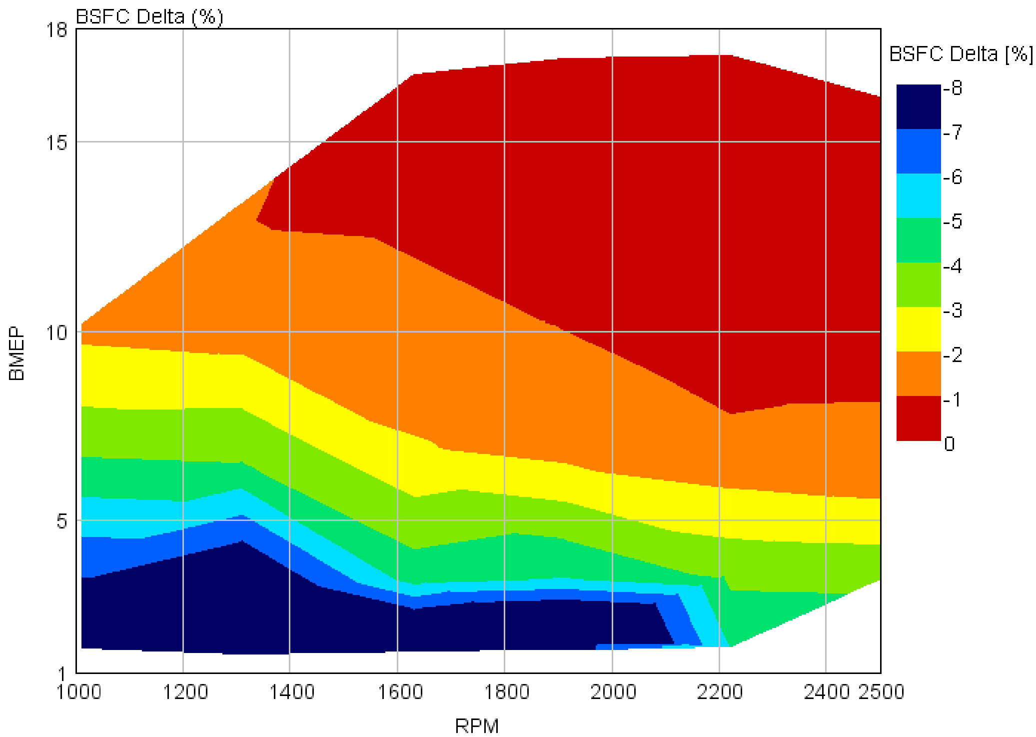


Figure B2.6b. Friction reduction BSFC Improvement over Baseline

3.1 ISB Pickup 385bhp Baseline Validation

- **Basic engine specification**
    - 6.7L inline 6 cylinder
    - Single Variable Geometry Turbo
    - 4 valves per cylinder
    - Diesel
    - 285 kW @ 2350 - 3000 rpm
    - 1150 Nm @ 1600 - 2350 rpm

The GT Power model for this version of the ISB engine was closely based on the validated model from the medium duty 300 bhp analysis. Low speed performance was maintained as per the 300bhp model (below 1300rpm), then power/torque was increased by ~25% across the remainder of the speed range, which was extended to 3000 rpm, as opposed to 2500rpm for the 300bhp engine. For this engine, a number of parameters where either maintained or revised, compared to the 300 bhp engine.

- Same AFR (min ~19:1)
- Reduced EGR flow (see Fig 3.1c)
- Increased maximum cylinder pressure limit (175 bar)
- Re-sized turbo

Although no test data was available to correlate the engine model, published data & benchmark data for similar engines where used to confirm the model predicted a reasonable level of performance.

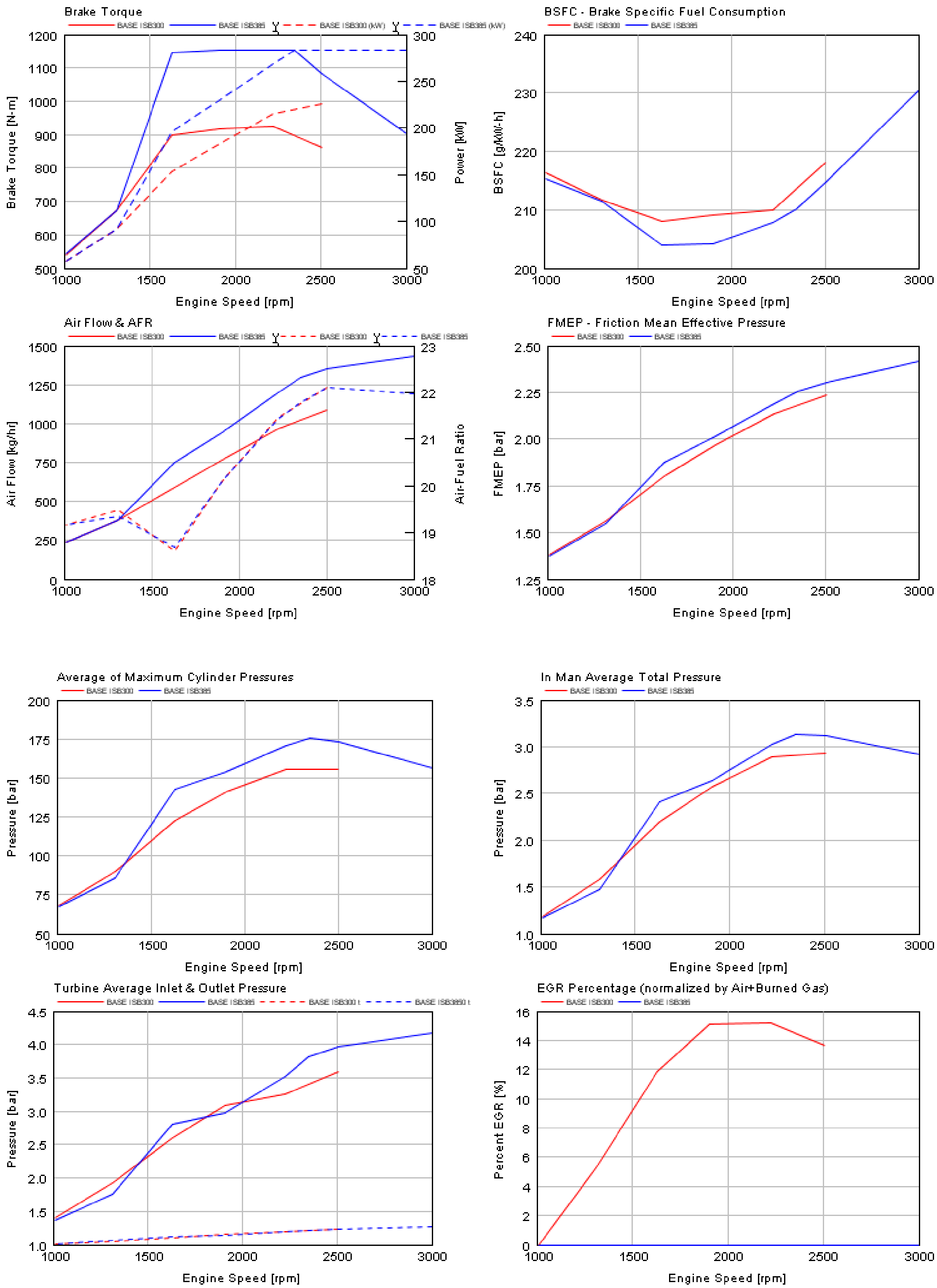


Figure B3.1a (Part 1) ISB MD300 vs PU385 Performance Comparison at Full Load

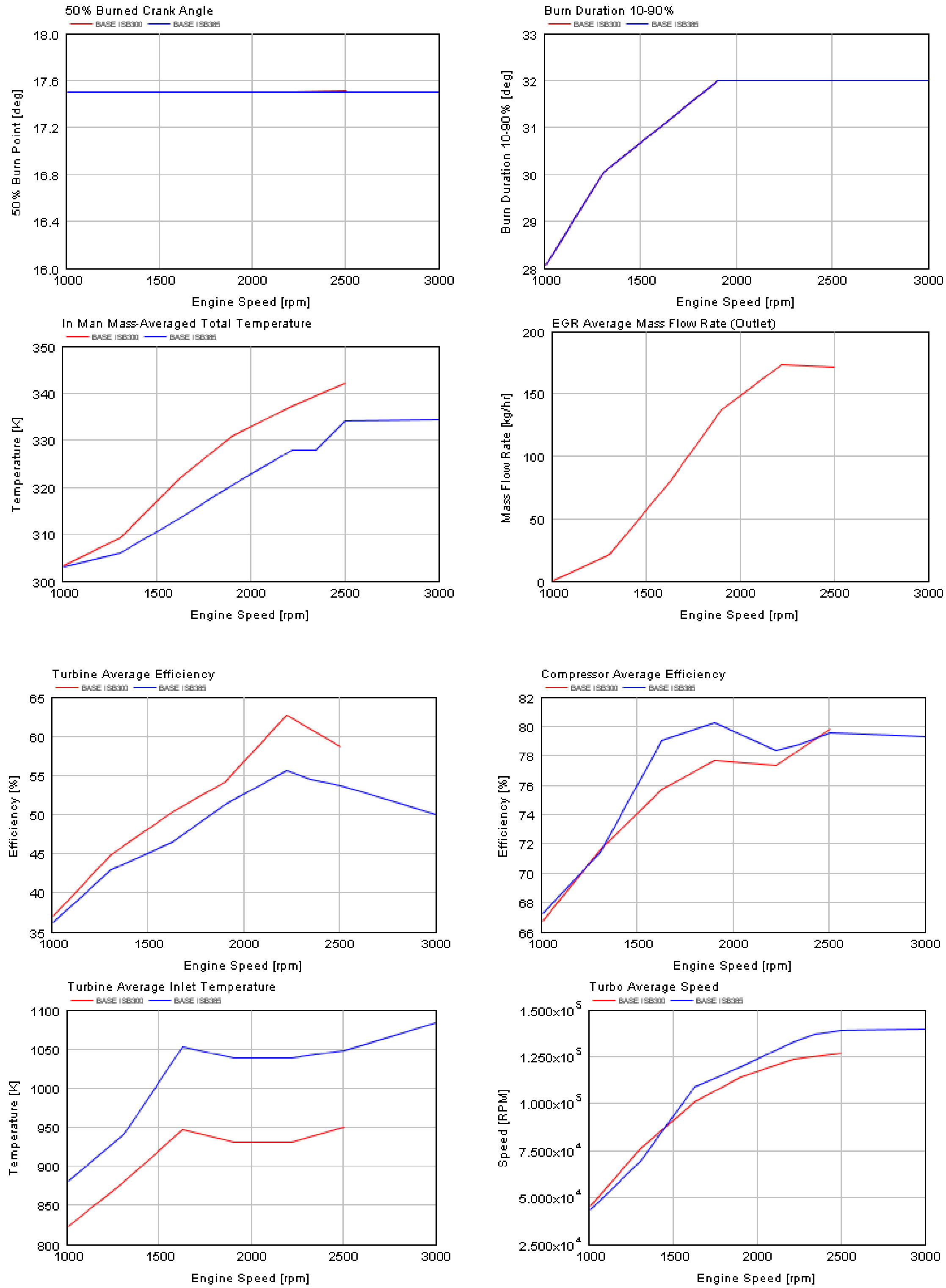


Figure B3.1a (Part 2) ISB MD300 vs PU385 Performance Comparison at Full Load

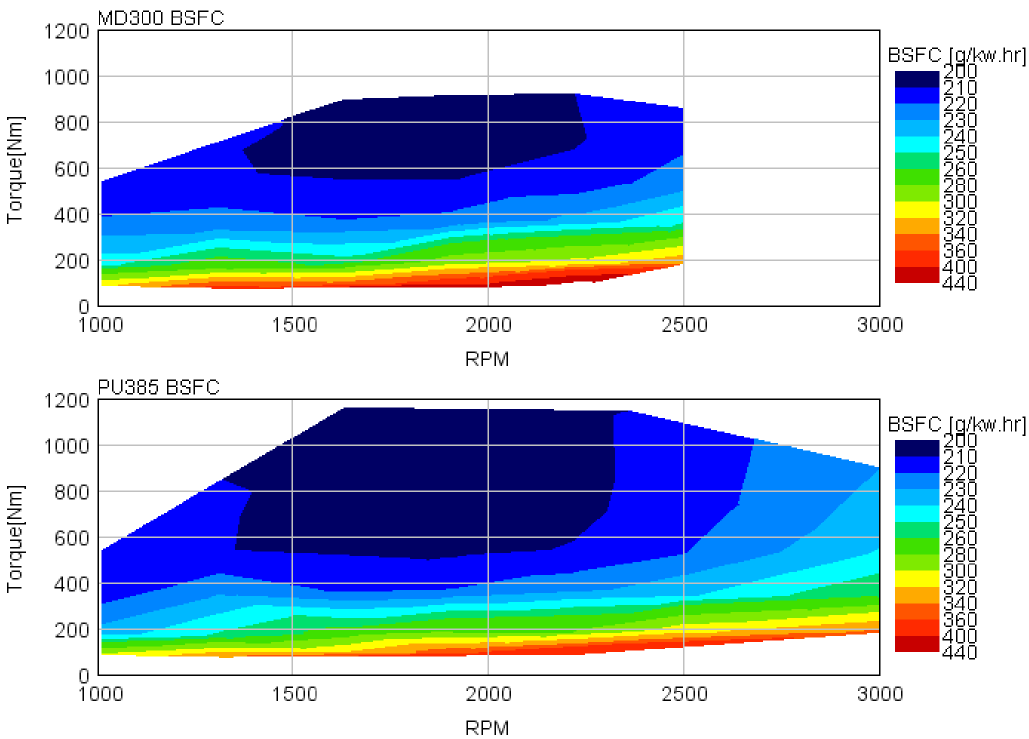


Figure B3.1b ISB MD300 vs PU385 BSFC Comparison

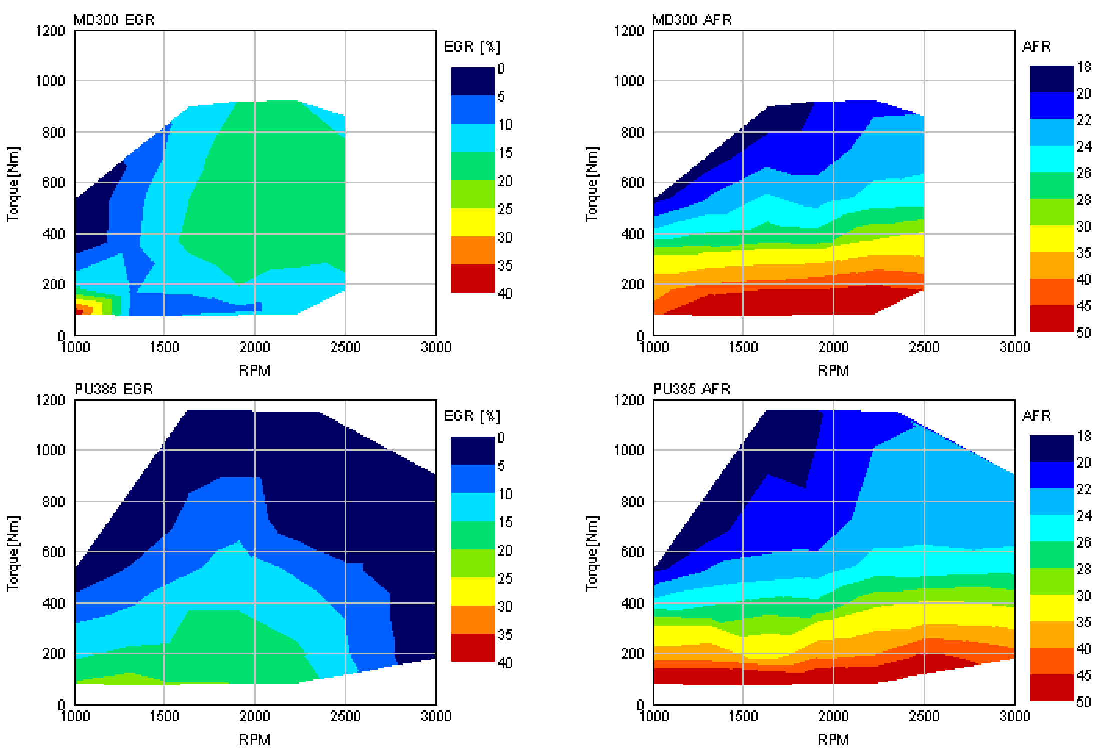


Figure B3.1c ISB MD300 vs PU385 EGR & AFR Comparison

The reason for the lack of EGR flow at higher speeds and loads on the 385 HP pickup version of the engine is the fact that the engine is chassis dynamometer certified for emissions. Since the vehicle drive cycle does not require high RPM or high torque operation, EGR is not required at these conditions to control NOx. The lack of EGR flow at high load is a key factor enabling the very high BMEP of this engine. If EGR was required at full load, the cylinder pressure would increase substantially. Either the power rating would need to be reduced, or the engine would need to be redesigned to tolerate higher cylinder pressure.

3.2 Reduced Exhaust Backpressure

The aim of running the model with reduced exhaust restriction is to simulate the effect of improved flow after-treatment systems for emissions control and the resulting reduction in pumping losses. The baseline GT model was originally setup without modelling the emissions control hardware, but the effective backpressure seen in the exhaust at the VGT exit is simulated by the use of a simple orifice. To model the reduced exhaust backpressure the orifice is resized to achieve the desired target backpressure at the rated condition. In this case, the backpressure has been reduced by slightly over 50%, from 26 to 12 kPa. The effect on the engine BSFC is shown in figure B3.2a. The effect is very small, so the figure only uses two colors despite a scale of 0.25% per color. Assuming that the after-treatment effectiveness is retained, there will be no effect on either engine-out or tailpipe emissions of criteria pollutants.

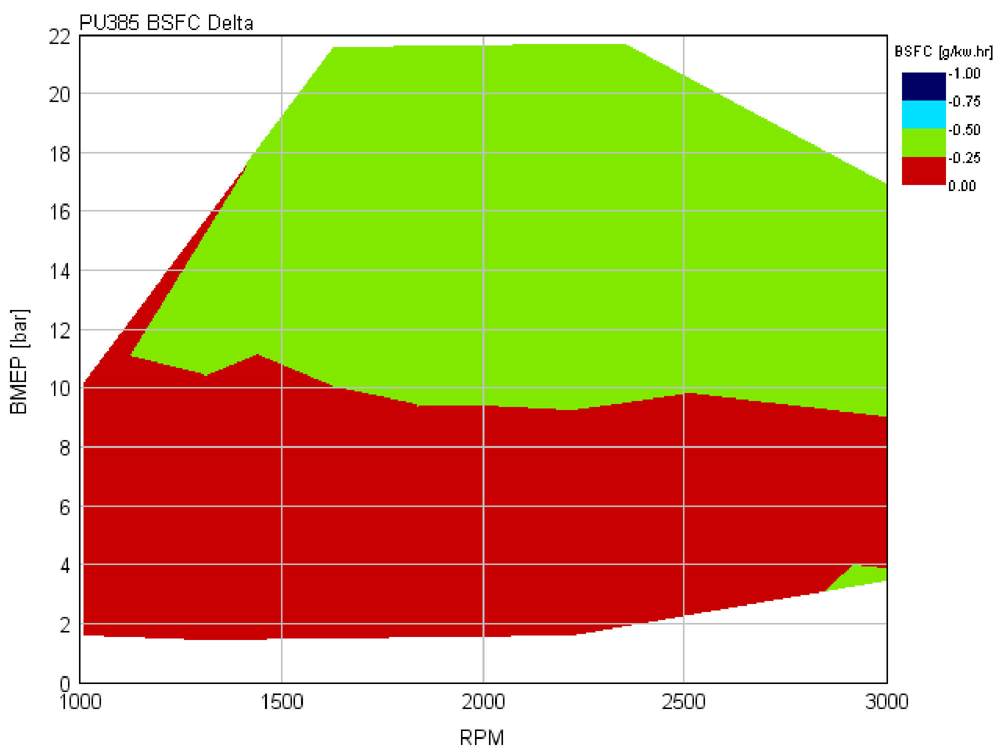


Figure B3.2a Reduced Exhaust backpressure BSFC Improvement

3.3 No EGR

The base ISB engine features a high pressure EGR circuit, with a pre-turbine exhaust manifold supply, an EGR cooler with a bypass valve for when cooling is not required. The valve unit is integrated into the inlet supply system that features an annular ring that feeds into the inlet pipe pre-manifold. To assess the engine performance with no EGR demand, the entire EGR system set of system components was removed from the GT model. The engine was run with the same performance targets but an increased AFR limit (+2 ratios), and the combustion effectively reduced in duration. This technology would involve very high engine-out NOx, and require a very efficient SCR system with a high urea consumption rate.

Although the base 385bhp engine has a relatively low EGR demand across the speed & load range compared to the 300bhp ISB engine, it was still necessary to re-scale the turbo setup to achieve suitable performance at the low load conditions, where the EGR rates are up to ~20% (see Fig 3.1c). This leads to change in performance on the full load curve, even though the EGR demand is unchanged (from zero).

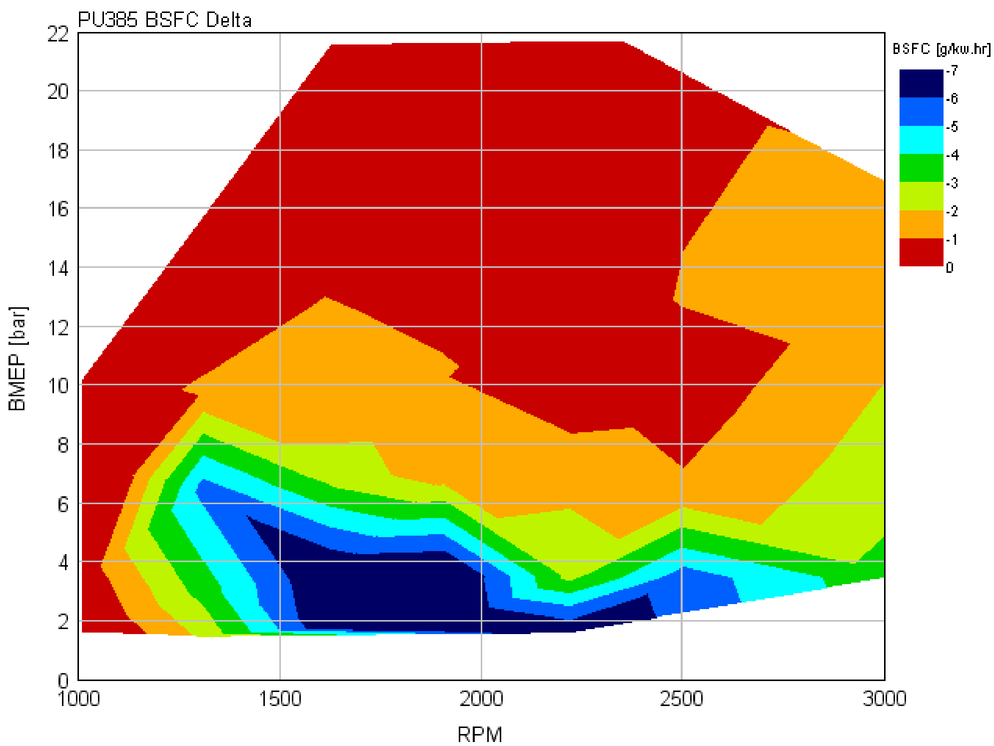


Figure B3.3a. BSFC improvement from EGR removal

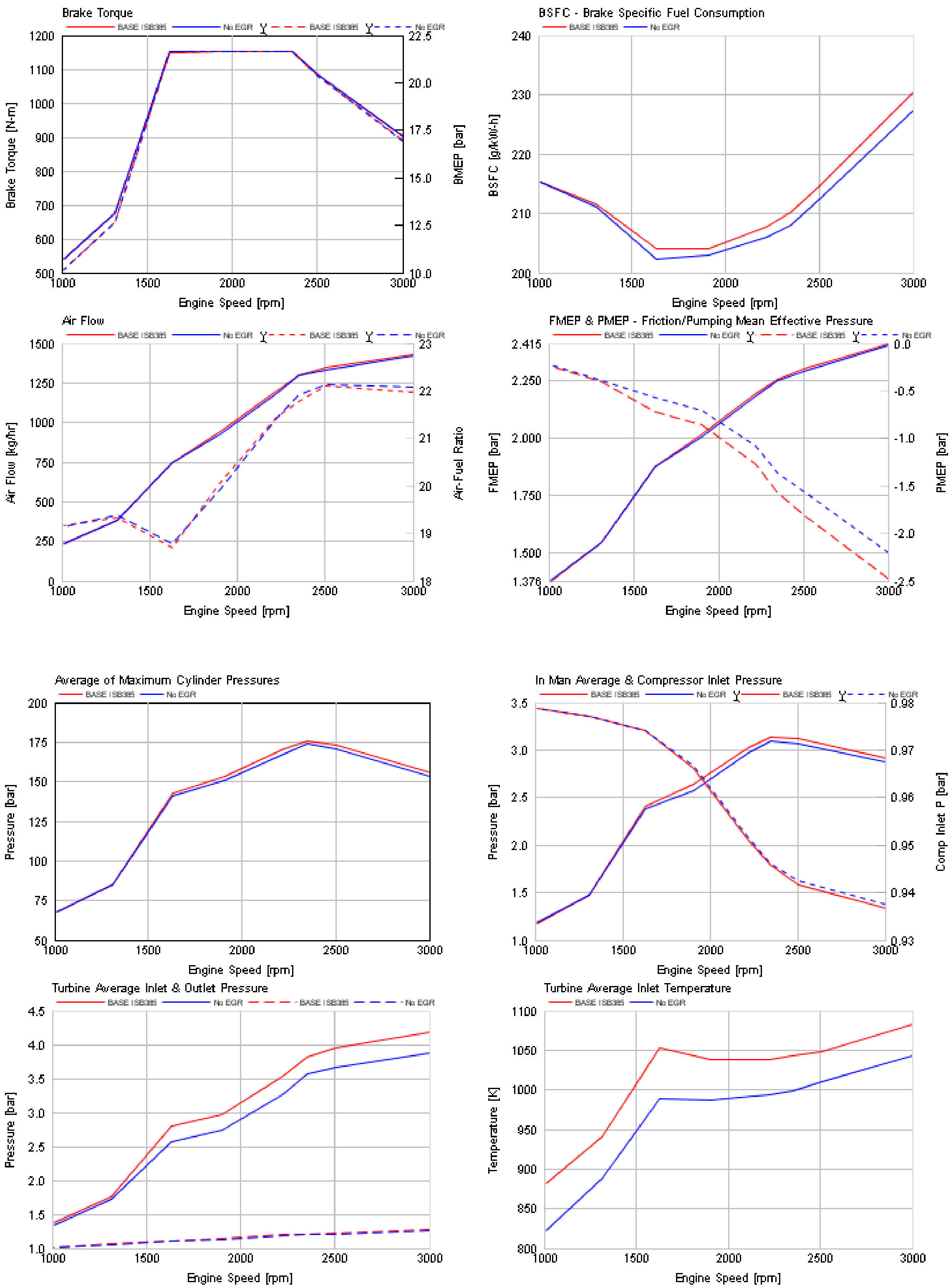


Figure B3.3b. Effect of EGR Removal on Full Load Engine Performance

3.4 Explore Turbo Efficiency Improvement

To simulate an improved efficiency turbo, the base engine model was run with a 5% efficiency increase on both the turbine & the compressor maps. In this case, where the actual engine turbo maps were unavailable and other maps where used and scaled to suit, these maps have been additionally (and globally) scaled to achieve the 5% improvement in efficiency. Turbo efficiency improvement can be realized by improvements in turbo designs, bearings, and materials. Reduced pumping losses & improved waste heat energy recovery are typical benefits of improved turbo efficiency. For engines that use EGR there is limited scope for improving turbo efficiency. In these engines, which represent almost the entire market, improved turbocharger efficiency will result in a loss of EGR flow and thus a loss of NOx control.

The comparison shown is against the baseline engine operating with EGR, unlike the DD15, where the more efficient turbo result was compared against the version with no EGR.

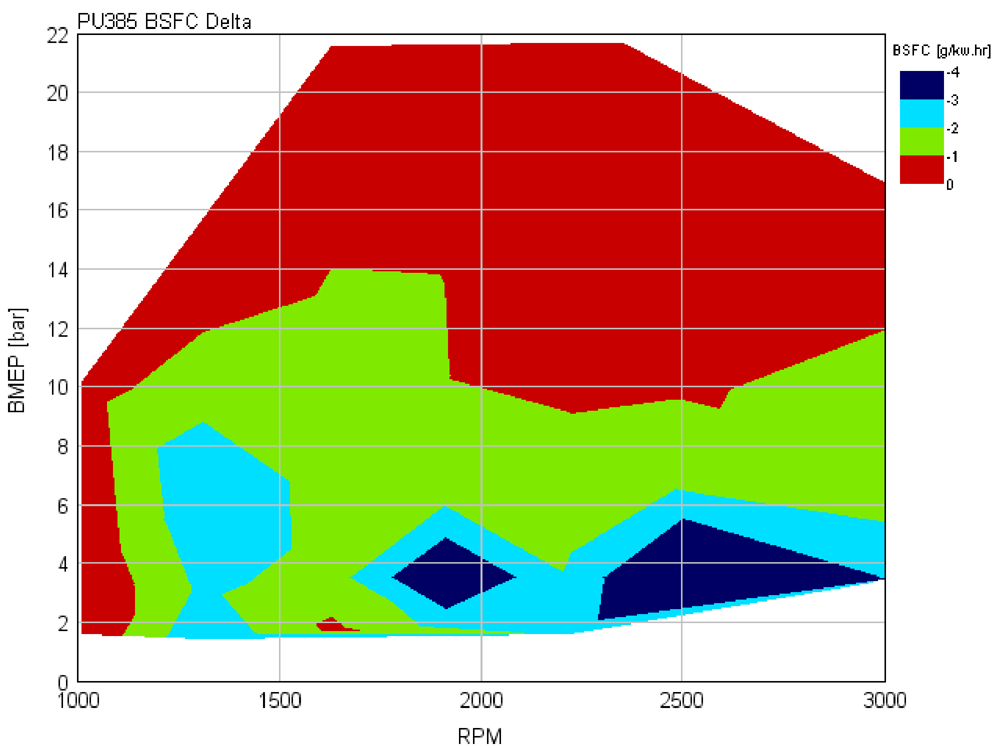


Figure B3.4a. High Efficiency Turbo BSFC Improvement

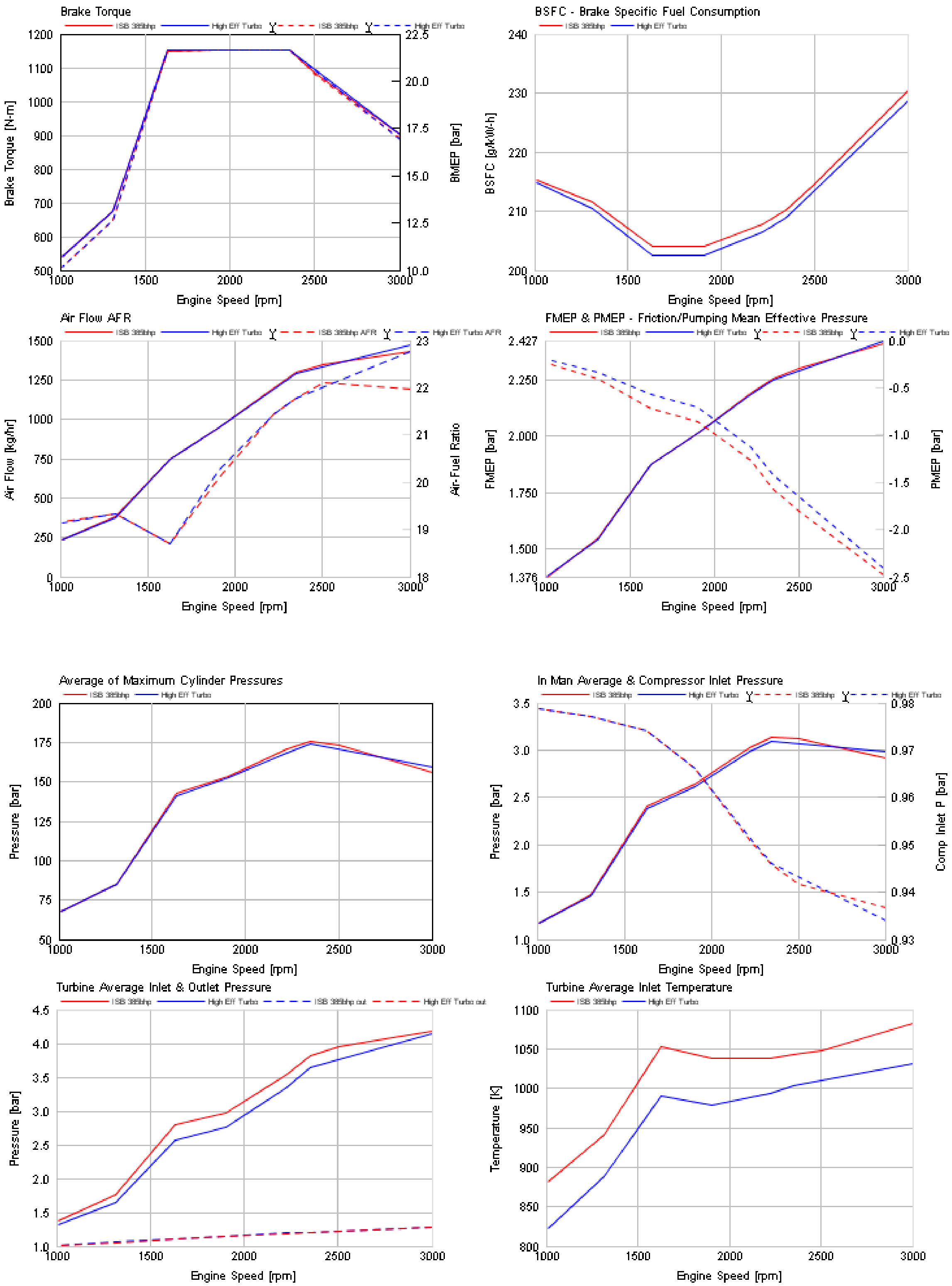


Figure B3.4b. Effect of High Efficiency Turbo on Full Load Engine Performance

3.5 In-line 4 Cylinder 4.5L Engine

An in-line 4-cylinder version of the ISB 385 was constructed, resulting in a displacement of 4.5L. The same BMEP levels as the 6-cylinder version were maintained, so a lower torque level was achieved. The peak torque speed was maintained from 1600 rpm to 2350 rpm. This technology is meant to reflect the performance of a smaller medium duty engine for Class 2b/3 applications. This size of engine is often used in urban area delivery trucks, and could be used in pickup trucks if customers are willing to accept less power in exchange for better fuel efficiency.

The existing single turbine entry exhaust manifold layout was carried over, as was the EGR system. The following model setup parameters & features where maintained as the 6 cylinder model

- Combustion settings
- EGR levels
- AFR
- Friction
- Cylinder pressure limits where maintained,

The FMEP was assumed to be the same as seen on the 6-cylinder, so for a given operating point (speed and BMEP level), the 4-cylinder will have 2/3 of the friction torque.

The combustion characteristics are also the same as for the 6-cylinder engine, and are related to operating BMEP levels (not torque).

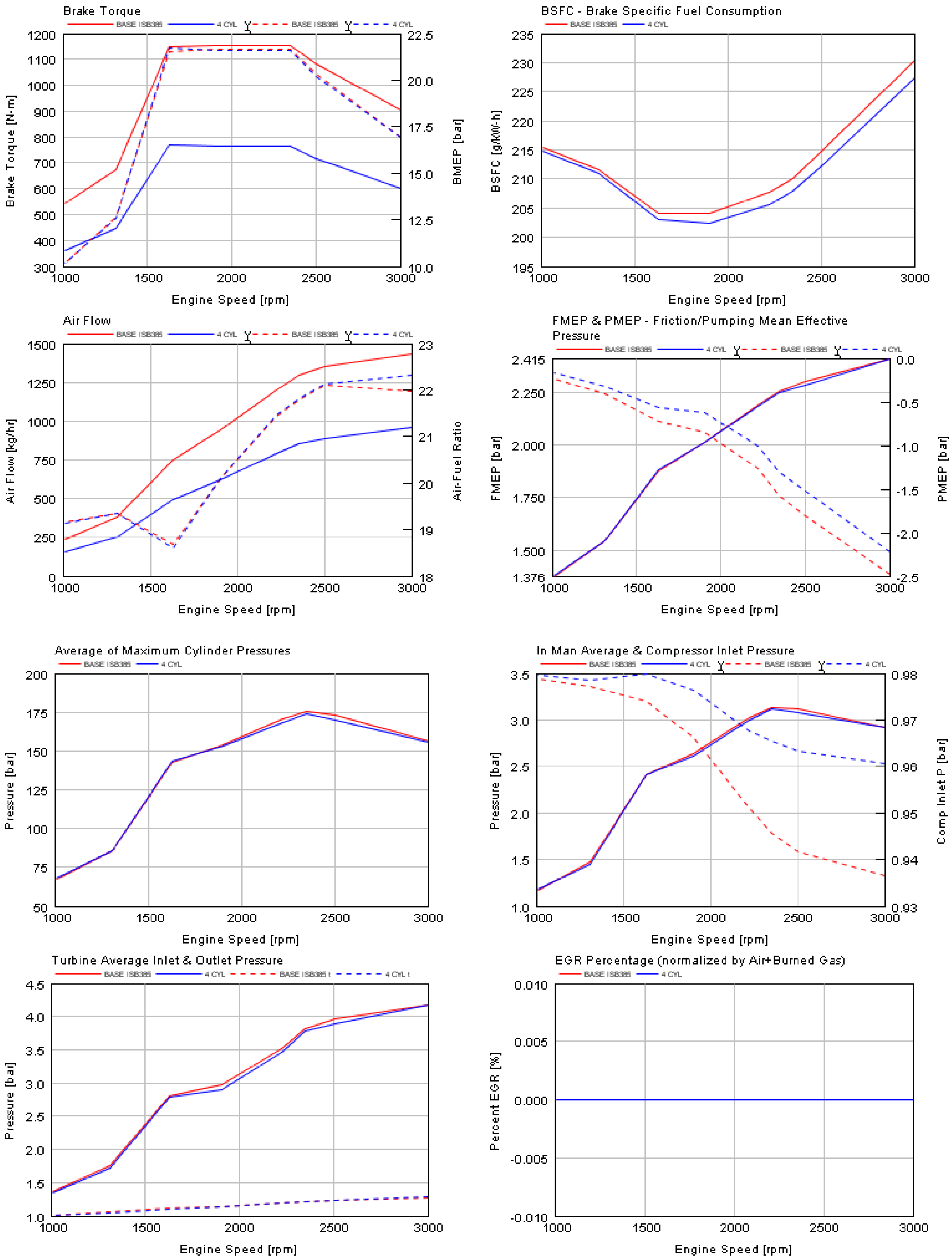


Figure B3.5a. Comparison of 6 & 4 Cylinder ISB versions at Full Load

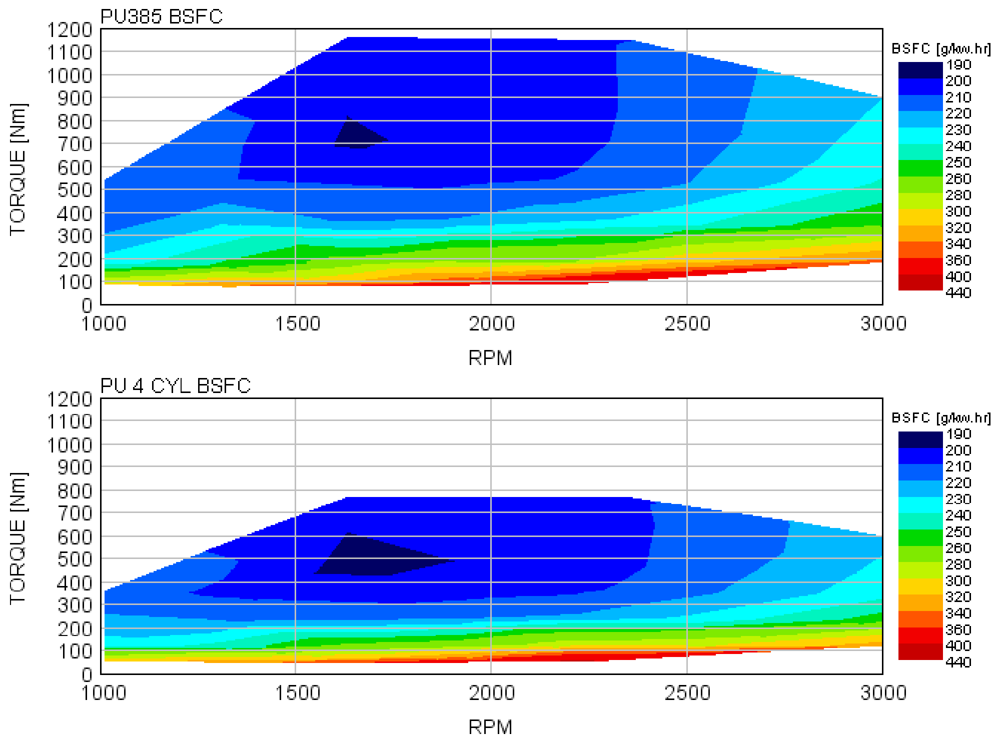


Figure B3.5b. BSFC comparison of 6 & 4 Cylinder ISB Versions

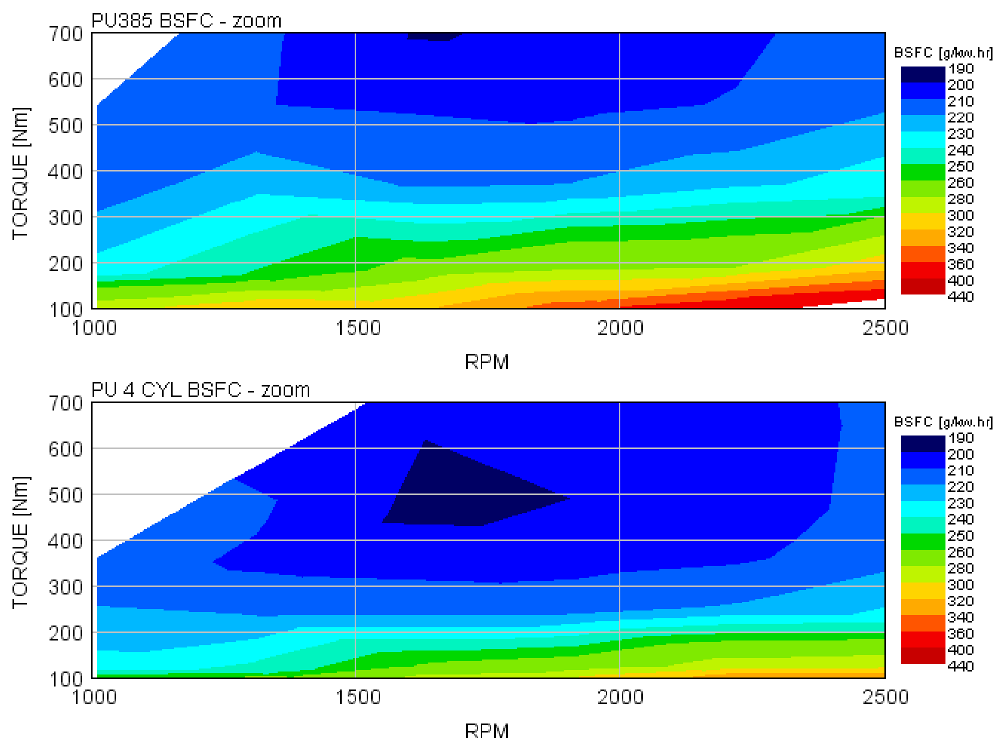


Figure B3.5c. BSFC comparison of 6 & 4 Cylinder ISB Versions

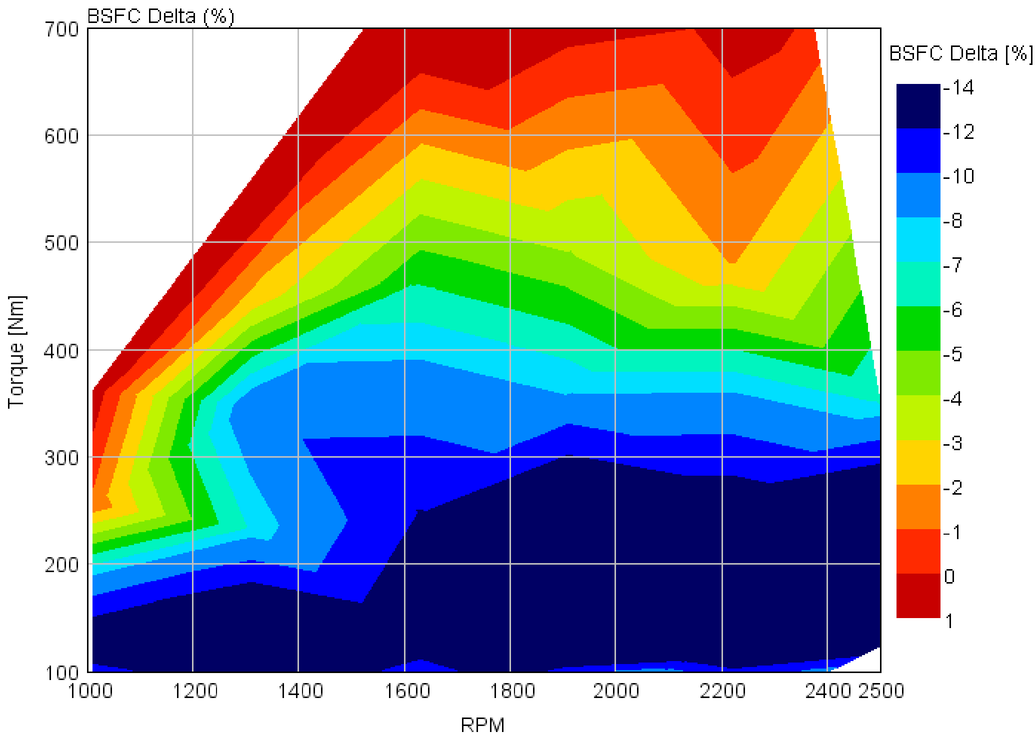


Figure B3.5d. BSFC Delta of 4 Cylinder ISB, Compared to 6 Cylinder

3.6 Explore FMEP Improvements

The baseline 385bhp engine model with a reduced friction map was run at the same conditions of as the base model. The friction (FMEP) was reduced on a speed-load relationship, with full load reductions of 10% and part load reduction of up to 35%. As expected, the largest benefits occur at light load and lower engine speeds.

Reducing engine friction can be achieved in several ways, such as low viscosity oils, smaller or dynamic bearings, ring/piston/liner interface improvements, a more efficient fuel pump, as well as smart controlled oil and water pumps. In this study, all the friction improvements where lumped together.

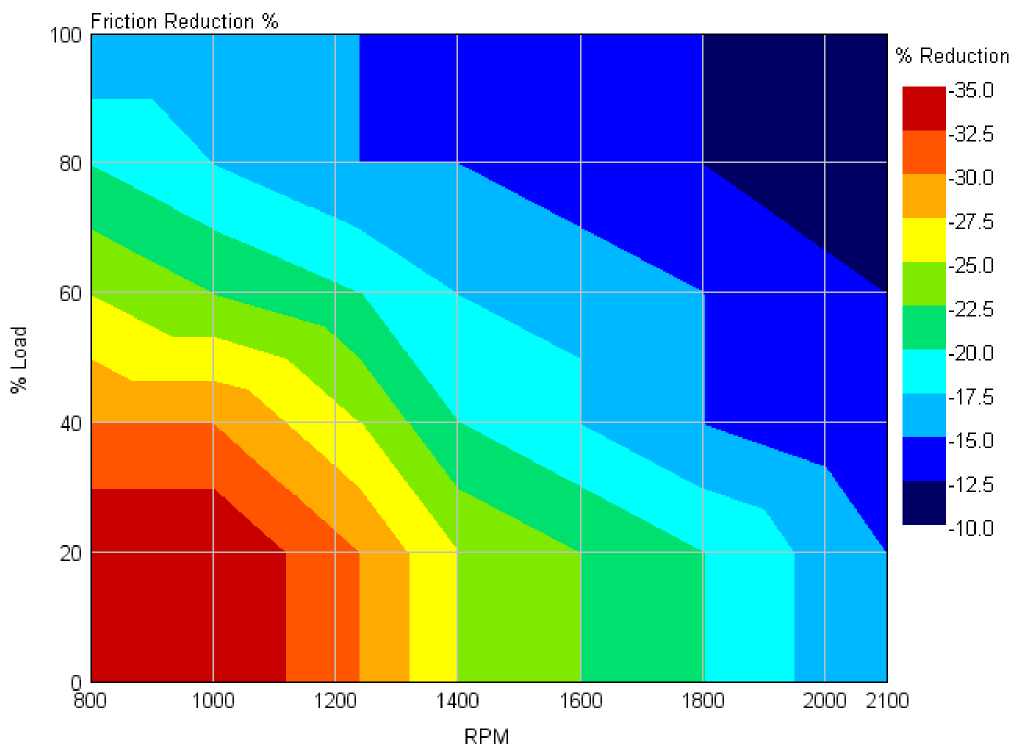


Figure B3.6a. Friction Reduction Assumptions

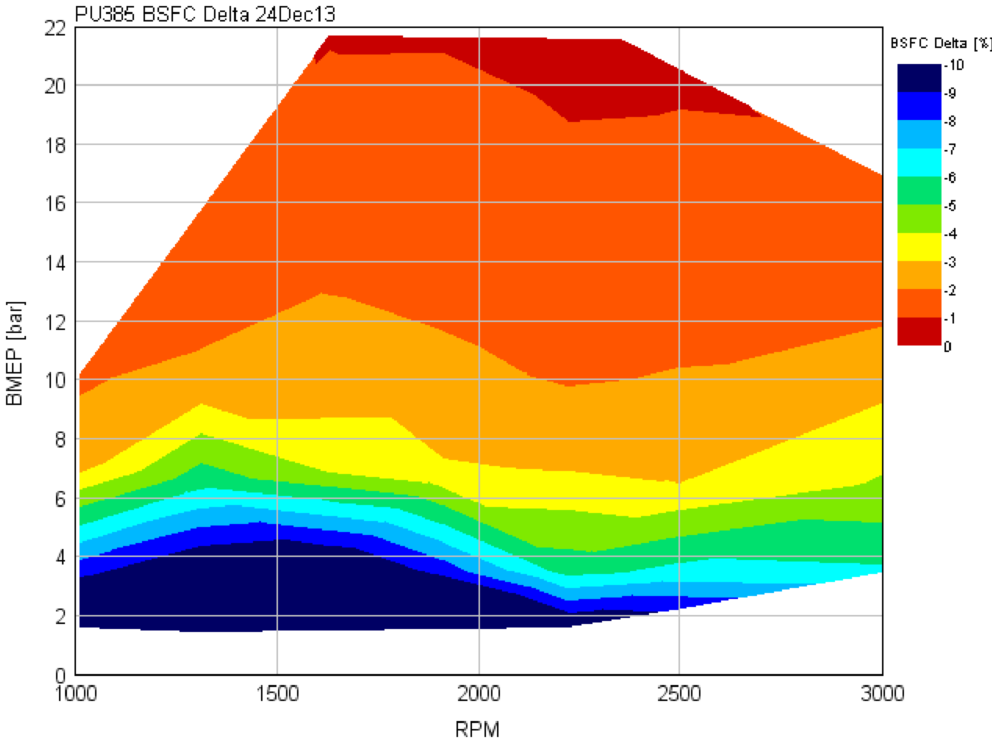


Figure B3.6b. Friction reduction BSFC Improvement over Baseline

APPENDIX C

VEHICLE SIMULATION AND VEHICLE TECHNOLOGIES

Vehicle Simulation and Vehicle Technologies

Objective: Simulate Future Vehicle and Engine Technologies to Demonstrate Fuel Economy Improvement Potential in Class 2b through 7 vehicles utilizing:

- **A range of engines and engine technologies**
- **A range of vehicle technologies**

1. Vehicle Modeling Approach:

1.1. Simulation Tool
1.2. Description of Baseline Vehicle Models
1.2.1. Ram Pickup Truck
1.2.2. Ford F-650 Roll-On Tow Truck
1.2.3. Kenworth T-270 Box Delivery Truck
1.2.4. Kenworth T-700 Tractor-Trailer Truck
1.3. Vehicle Drive Cycles

2. Vehicle Efficiency Technologies

2.1. Reduced Air Conditioner Power Demand
2.2. Reduced Aerodynamic Drag (Cd)
2.3. Reduced Tire Rolling Resistance (Crr)
2.4. Weight Reduction
2.5. Chassis Friction Reduction
2.6. 6X2 Axle Configuration
2.7. Road Speed Governor (Reduced Vmax)
2.8. Transmission Alternatives
2.8.1. 6-Speed, 10-Speed, and 18-Speed AMT Transmissions
2.8.2. 10-Speed Manual Transmission
2.8.3. 5-Speed, 6-Speed, and 8-Speed Torque Converter Automatic Transmissions
2.9. Engine Alternatives
2.9.1. 4.5 and 6.7 Liter Diesel (Pickup Only)
2.9.2. 6.7 and 8.9 Liter Diesel (Classes 4 – 8)
2.9.3. 3.5 V-6 and 6.2 V-8 Gasoline Engines (Classes 2b – 7)
2.9.4. 12.5 Liter and 14.8 Liter Diesel (Class 8 Only)

3. Examples of Vehicle Drive Cycle Operation

C1. Vehicle Modeling Approach

C1.1 Simulation Tool

Southwest Research Institute has developed a number of simulation tools for vehicle performance and fuel economy evaluation. The most popular ones are PSAT and RAPTOR. PSAT (Powertrain System Analysis Toolkit) was the first forward-looking (integral-approach) modeling tool developed in the MATLAB/Simulink environment. PSAT was developed for The United States Council for Automotive Research (USCAR) during the Partnership for a New Generation of Vehicles (PNGV) Initiative. SwRI provided a version of PSAT to Argonne National Lab, who utilized it for its DOE-funded programs and continued to upgrade and sell the tool to the rest of the automotive industry. RAPTOR, on the other hand, was co-developed by SwRI and Chrysler Corp for generating validated NHTSA Corporate Average Fuel Economy (CAFE) numbers on specific vehicle platforms. These two tools are still used in the industry today.

SwRI has also utilized other simulation tools and upgraded them to meet its client's needs. One of these tools is SwRI's Vehicle Simulation Tool (VST). This software is based on NREL's original Advisor vehicle simulator (see Figure C1 below) offered by NREL and utilized worldwide by hundreds of users. The main difference between Advisor and PSAT is that Advisor utilizes a backward-looking simulation algorithm that starts from the vehicle speed requirements during a cycle and then determines the powertrain demand, using a differentiation rather than an integration approach.

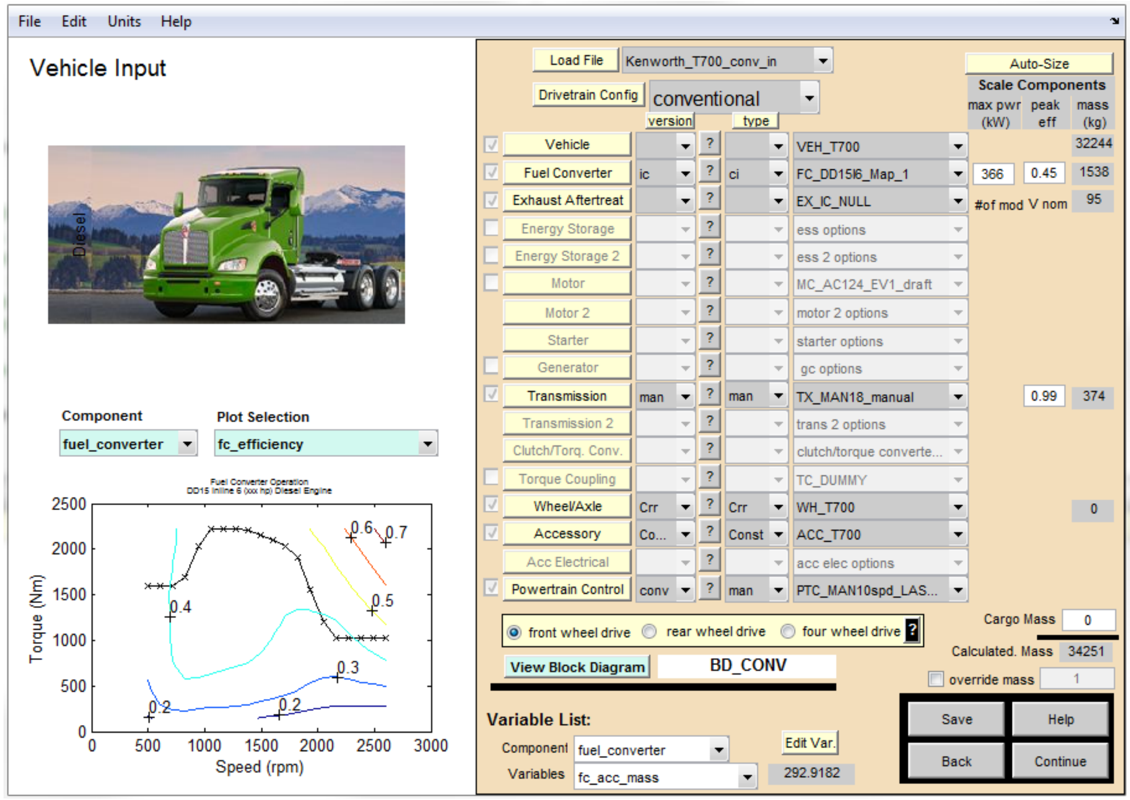


Figure C1. Screenshot of SwRI's VST User Interface

The advantage of using the VST's differentiation approach is that each vehicle variant that is simulated is forced to precisely run the cycle in the same manner, regardless of the vehicle configuration. An integration approach (like PSAT) must utilize a driver model to modulate the accelerator pedal and brakes to follow the cycle. Even though this approach can be more representative of a real driver's behavior, one cannot guarantee that every vehicle configuration will follow the drive cycle in precisely the same way. When small variations in vehicle or powertrain parameters need to be evaluated, which is the case in this NHTSA program, a backwards looking approach is better suited for such simulation requirements. VST uses a backward looking algorithm. In addition, the execution of backward-looking representations can offer other benefits as well, such as an execution speed that is, at least, one order of magnitude faster than forward-looking representations. Since thousands of simulation runs were necessary to conduct this fuel economy study, SwRI chose to use VST for all configurations and analyses described in this report.

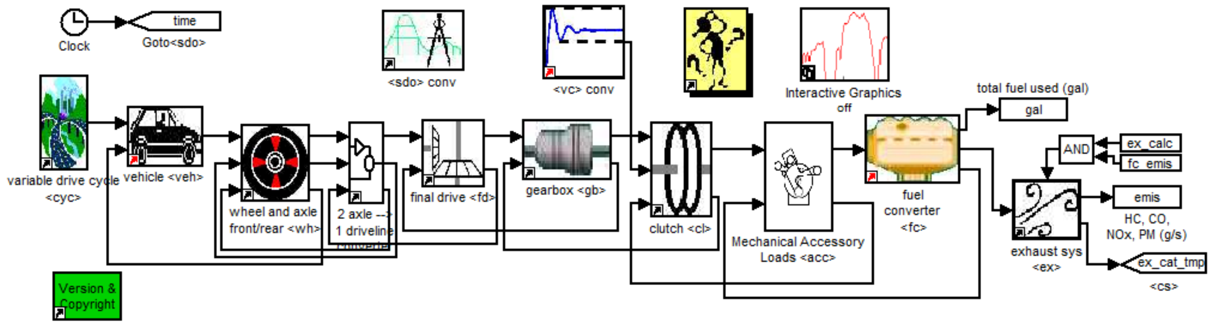


Figure C2. Top-Level Simulink Diagram of the Vehicle Model

SwRI's VST tool incorporates improvements to the original NREL component models, and enhanced functionalities in ways that allow the user to define each component of the vehicle. VST can simulate and post-process the results in an efficient and timely manner. Each component's set of parameters is defined in a MATLAB scripting format that is used in conjunction with a Simulink model shown in Figure C2 above.

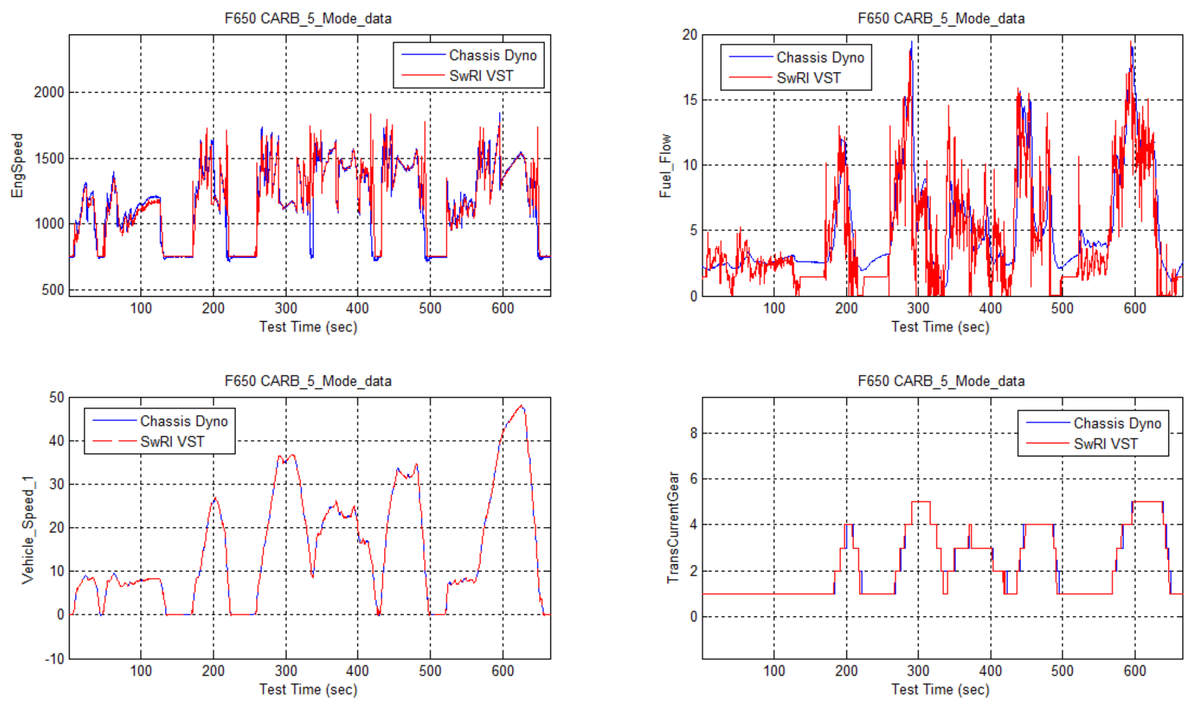


Figure C3. Validation of SwRI's VST Simulation Tool against Chassis Dyno Data for a Representative Vehicle Similar to one of the Vehicles Evaluated in this Study

SwRI's VehSim tool was validated against available chassis dynamometer test data as shown in Figure C3. These results are for the baseline Ram pickup truck with the diesel engine.

The results showed good agreement in following the desired speed trace and transmission shift schedule. As a result, the two engine speed traces are also very similar. The fuel flow trace on the upper right of the figure shows much more fluctuation for the simulation than for the chassis dyno test, but this result is misleading. The chassis dyno fuel flow results are effectively low pass filtered (smoothed) by the fuel flow measurement system. The total quantity of fuel used for fuel economy computations, however, was measured by separate, more accurate volumetric means.

As part of this study, SwRI performed minor custom modifications to the simulation tool in order to accommodate some specific requirements for this study. The most significant modification was the introduction of an adaptive driving cycle algorithm to guarantee that the vehicle drives the same distance regardless of the severity in road grade and powertrain limitations. This was particularly important when assessing the fuel economy of the vehicle on the NESCCCAF cycles with grade. An example of results from the resulting algorithm (see Figure C4) shows the original and the adapted cycle traces. The drive cycle includes periods of 3% grade from around 2900 seconds to 3800 seconds. Because the vehicle does not have enough power to maintain speed on the 3% grade, the actual speed falls short of the target. The algorithm automatically compensates by adding additional time to ensure that the vehicle covers the same distance, regardless of the speed achieved.

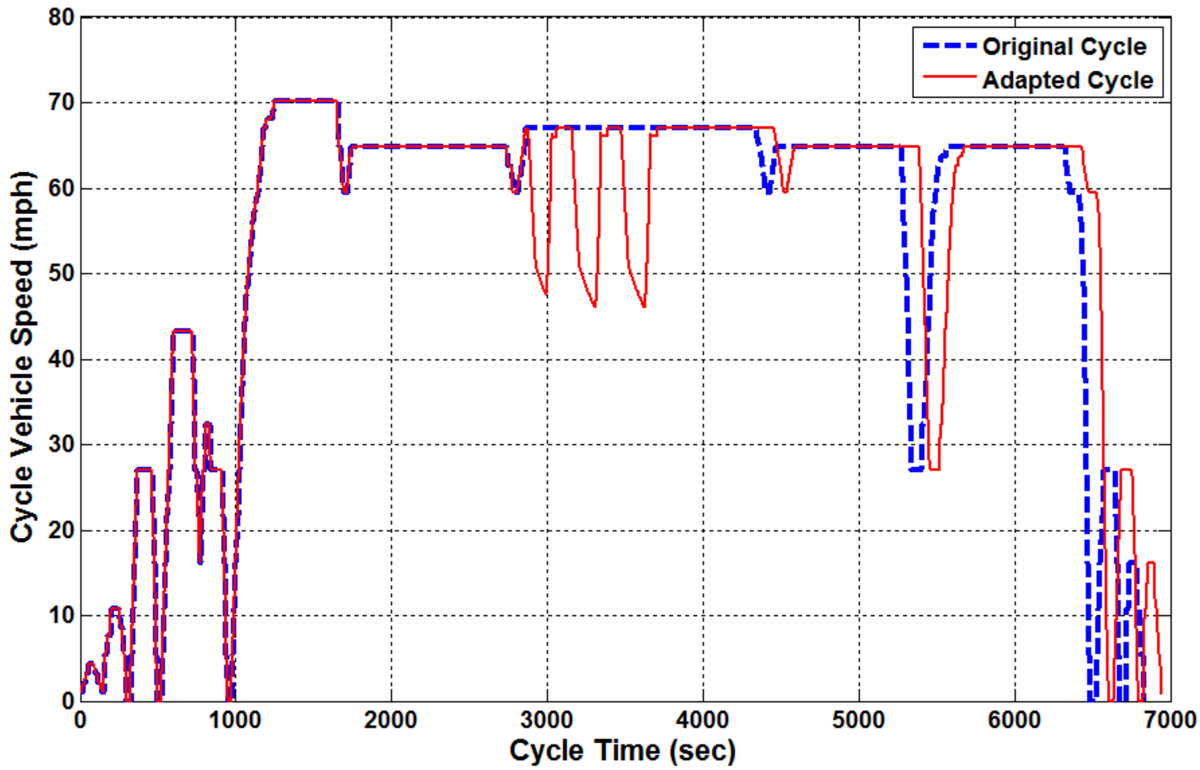


Figure C4. Resulting Cycle from SwRI's New Adaptive Algorithm

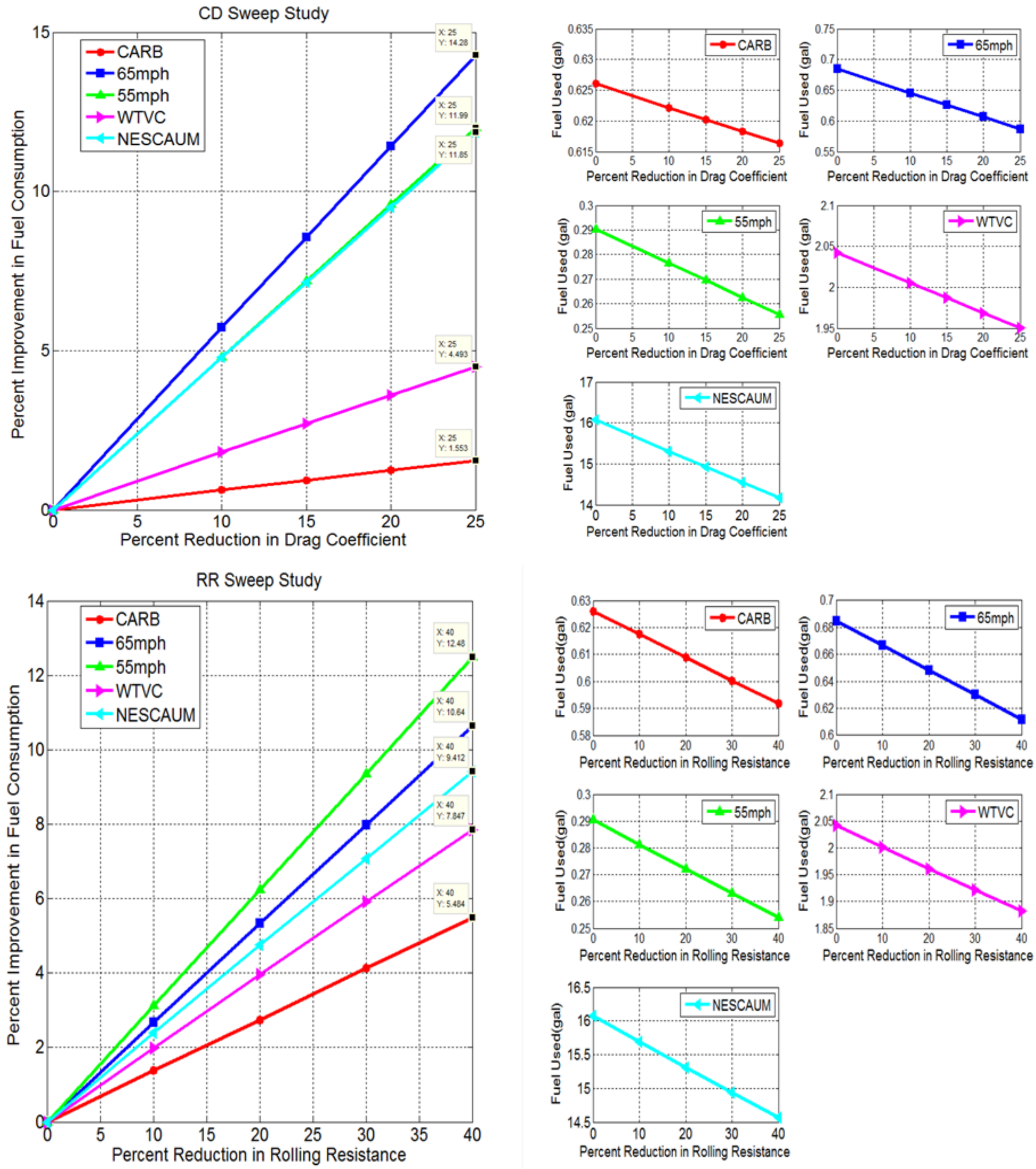


Figure C5. Special Parameter Sweep Studies on the Kenworth T-700 Tractor-Trailer Truck

Finally, some special analyses involving parameter sweeps for selected vehicles were conducted, as shown by the examples above. Figure C5 shows the effect of sweeping aerodynamic drag (Cd) and tire rolling resistance (Crr) over a range of values. These particular parameters produce results which are very linear.

C1.2 Baseline Vehicle Models

This section provides the baseline input parameters that are used for each of the four vehicle models that are covered in this report.

C1.2.1 Ram Pickup

Table C.1. shows the baseline vehicle simulation input parameters of the Ram Pickup model. This model is used to represent Class 2b and 3 vehicles. Note that axle efficiencies are listed in Table C.11.

Table C.1. Input parameters for the Dodge Ram Pickup Truck Model.

	Component	Vehicle 1	Vehicle 2	Vehicle 3	Vehicle 4	Vehicle 5	Vehicle 6
Basic Parameters	Vehicle Name	Dodge RAM	Dodge RAM	Dodge RAM	Dodge RAM	Dodge RAM	Dodge RAM
	Vehicle Type	Class 2b/3 Truck	CL 2b/3 Truck	CL 2b/3 Truck	CL 2b/3 Truck	CL 2b/3 Truck	CL 2b/3 Truck
	Engine	Baseline 6.7L I6	Diesel 4-Cylinder	Baseline 6.7L I6	Gasoline 3.5L V6	Gasoline 3.5L V6 Downsped	Gasoline 6.2L V8
	Transmission	6-Speed Auto	6-Speed Auto	8-Speed Auto	6-Speed Auto	6-Speed Auto	6-Speed Auto
	Transmission Controller	6-Speed Auto Diesel Schedule	6-Speed Auto 4-Cylinder Diesel Schedule	8-Speed Auto Diesel Schedule	6-Speed Auto Gasoline Schedule	6-Speed Auto Downsped Gasoline Schedule	6-Speed Auto Gasoline Schedule
	Engine/ Transmission Coupling	Torque Converter (stall speed of 2,800 rpm)	Torque Converter (stall speed of 2,420 rpm)	Torque Converter (stall speed of 2,800 rpm)	Torque Converter (stall speed of 2,325 rpm)	Torque Converter (stall speed of 2,090 rpm)	Torque Converter (stall speed of 2,450 rpm)
	Engine RPM (Top Gear at 65 mph)	1,592	1,820	1,595	2,500	2,000	2,500
	Final Drive	3.42:1	3.91:1	3.21:1	5.37:1	4.27:1	5.37:1
	Wheel Radius	0.3726	0.3726	0.3726	0.3726	0.3726	0.3726

C1.2.2 Ford F-650 Tow Truck

Table C.2. provides the baseline input parameters for the Ford F-650 roll-on tow truck. This Class 6 model is one of two trucks used to represent medium-duty vocational vehicles.

Table C.2. Input parameters for the F-650 tow truck model.

	Component	Vehicle 1	Vehicle 2	Vehicle 3	Vehicle 4	Vehicle 5	Vehicle 6	Vehicle 7
Basic Parameters	Veh. Name	F650	F650	F650	F650	F650	F650	F650
	Vehicle Type	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck
	Engine	Baseline 6.7L I6	Baseline 6.7L I6	Baseline 6.7L I6	Baseline 6.7L I6	Gas 3.5L V6	Gas 3.5L V6 Down sped	Gas 6.2L V8
	Transmission	5-Speed Auto	6-Speed AMT	10-Speed AMT	8-Speed Auto	5-Speed Auto	5-Speed Auto	5-Speed Auto
	Transmission Controller	5-Speed Auto Diesel Schedule	6-Speed AMT Diesel Sched	10-Speed AMT Diesel Sched	8-Speed Auto Diesel Sched	5-Speed Auto Gas Sched	5-Speed Auto Down sped Gas Sched	5-Speed Auto Gas Sched
	Engine / Transmission Coupling	Torque Converter (stall speed of 2,560 rpm)	Clutch	Clutch	Torque Conv (stall speed of 2,560 rpm)	Torque Conv (stall speed of 2,330 rpm)	Torque Conv (stall speed of 2,090 rpm)	Torque Conv (stall speed of 2,445 rpm)
	Engine RPM (Top Gear at 65 mph)	2,073	2,072	2,072	2,073	3,300	2,576	3,300
	Final Drive	4.33:1	3.08:1	4.22:1	4.62:1	6.89:1	5.38:1	6.89:1
	Wheel Radius	0.4125	0.4125	0.4125	0.4125	0.4125	0.4125	0.4125

C1.2.3 Kenworth T-270 Box Delivery Truck

The baseline input parameters for the Kenworth T-270 box delivery truck are given in Table C.3 below. This Class 6 model is one of two trucks used to represent the medium-duty vocational vehicle segment.

Table C.3. Input parameters for the T-270 box truck model.

	Component	Veh 1	Veh 2	Veh 3	Veh 4	Veh 5	Veh 6	Veh 7
Basic Parameters	Veh. Name	T270	T270	T270	T270	T270	T270	T270
	Vehicle Type	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck	Class 6 Truck
	Engine	Baseline 6.7L I6	Baseline 6.7L I6	Baseline 6.7L I6	Baseline 6.7L I6	Gasoline 3.5L V6	Gasoline 3.5L V6 Down sped	Gasoline 6.2L V8
	Transmission	5-Speed Auto	6-Speed AMT	10-Speed AMT	8-Speed Auto	5-Speed Auto	5-Speed Auto	5-Speed Auto
	Transmission Controller	5-Speed Auto Diesel Schedule	6-Speed AMT Diesel Sched	10-Speed AMT Diesel Sched	8-Speed Auto Diesel Sched	5-Speed Auto Gas Sched	5-Speed Auto Down sped Gas Sched	5-Speed Auto Gas Sched
	Engine/ Transmission Coupling	Torque Converter (stall speed of 2,560 rpm)	Clutch	Clutch	Torque Conv (stall speed of 2,560 rpm)	Torque Conv (stall speed of 2,330 rpm)	Torque Conv (stall speed of 2,090 rpm)	Torque Conv (stall speed of 2445 rpm)
	Engine RPM (Top Gear at 65 mph)	2,071	2,068	1,514	2,070	3,300	2,937	3,300
	Final Drive	5.29:1	3.76:1	3.76:1	5.64:1	8.42:1	7.5:1	8.42:1
	Wheel Radius	0.5044	0.5044	0.5044	0.5044	0.5044	0.5044	0.5044

C1.2.4 Kenworth T-700 Tractor-Trailer Truck

The Kenworth T-700 is used to represent long-haul high-roof sleeper tractor-trailer combination trucks. The baseline vehicle simulation input parameters are provided in Table C.4 below.

Table C.4. Input parameters for the T-700 tractor-trailer model.

	Component	Veh 1	Veh 2	Veh 3	Veh 4	Veh 5	Veh 6	Veh 7
Basic Parameters	Vehicle Name	T700	T700	T700	T700	T700	T700	T700
	Vehicle Type	Class 8 Tractor-Trailer	Class 8 Tractor-Trailer	Class 8 Tractor-Trailer	Class 8 Tractor-Trailer	Class 8 Tractor-Trailer	Class 8 Tractor-Trailer	Class 8 Tractor-Trailer
	Engine	Baseline DD15	Down-spd A	Down-spd B	5-Cyl	Baseline DD15	Baseline DD15	Baseline DD15
	Trans-mission	10-Speed AMT	10-Speed AMT	10-Speed AMT	10-Speed AMT	18-Speed AMT	10-Speed Manual	10-Speed 6x2
	Transm Control	10-Speed AMT Shift Sched	10-Speed AMT Down-spd A Sched	10-Speed AMT Down-spd B Sched	10-Speed AMT Sched	18-Speed AMT Sched	10-Speed Manual Sched	10-Speed AMT Sched
	Engine/ Transm. Coupling	Clutch	Clutch	Clutch	Clutch	Clutch	Clutch	Clutch
	Engine RPM (Top Gear at 65 mph)	1,368	1,209	1,050	1,368	1,368	1,368	1,368
	Final Drive	3.36:1	2.97:1	2.58:1	3.36:1	3.36:1	3.36:1	3.36:1
	Wheel Radius	0.4975	0.4975	0.4975	0.4975	0.4975	0.4975	0.4975

C1.3 Vehicle Drive Cycles

A total of eleven vehicle drive cycles were used in the study. Each cycle is described in this section. The NESCCAF cycle was developed to represent a typical line haul type of operation. There are brief urban/suburban sections at the beginning and end of the cycle, to represent getting out to the highway. The main portion of the cycle consists of five cruise segments at speeds of 65 to 70 MPH. One of the cruise segments has an alternating +/- 1% grade, and a second cruise segment has an alternating +/- 3% grade. The NESCCAF cycle is the only cycle that includes grades. Because tractor-trailer trucks cannot normally maintain cruise speed on a 3% grade, the cycle was given a feature that extended the drive time to force the vehicle to complete the required distance. Figure C6 shows the speed vs. time trace, and Figure C7 shows the grade vs. time trace. This cycle was only used for tractor-trailer trucks.

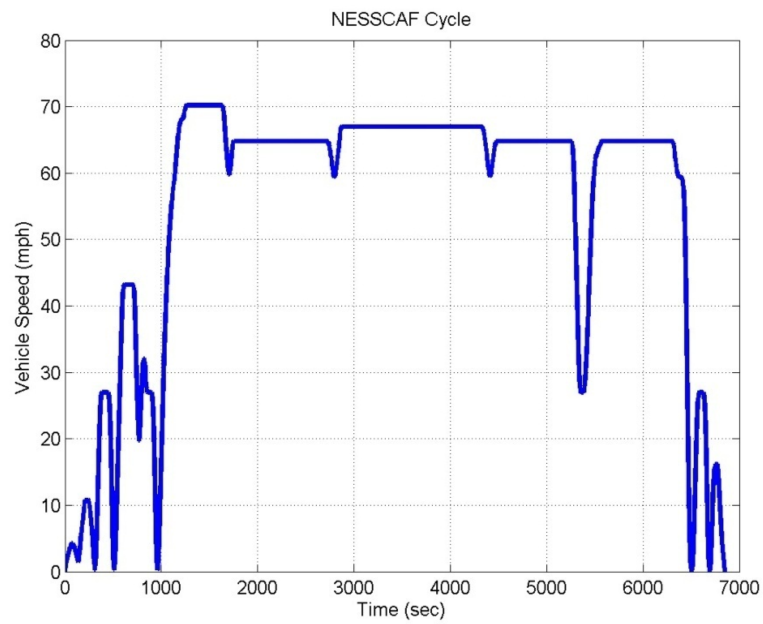


Figure C6. Speed vs. time trace for the NESCCAF cycle.

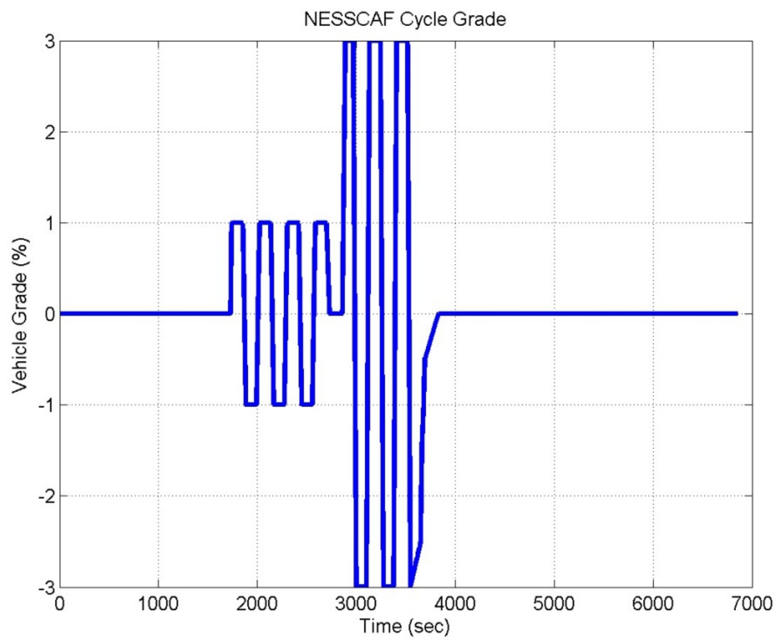


Figure C7. Grade vs. time trace for the NESCCAF cycle.

Figure C8 shows the World Harmonized Vehicle Cycle. This cycle is intended for medium and heavy-duty trucks, and includes urban, suburban, and highway segments. The cycle assumes that a road speed governor is used to limit speed to about 53 MPH, which is required in Europe. The WHVC was used for all vehicles, to provide a way to compare results across different vehicle types.

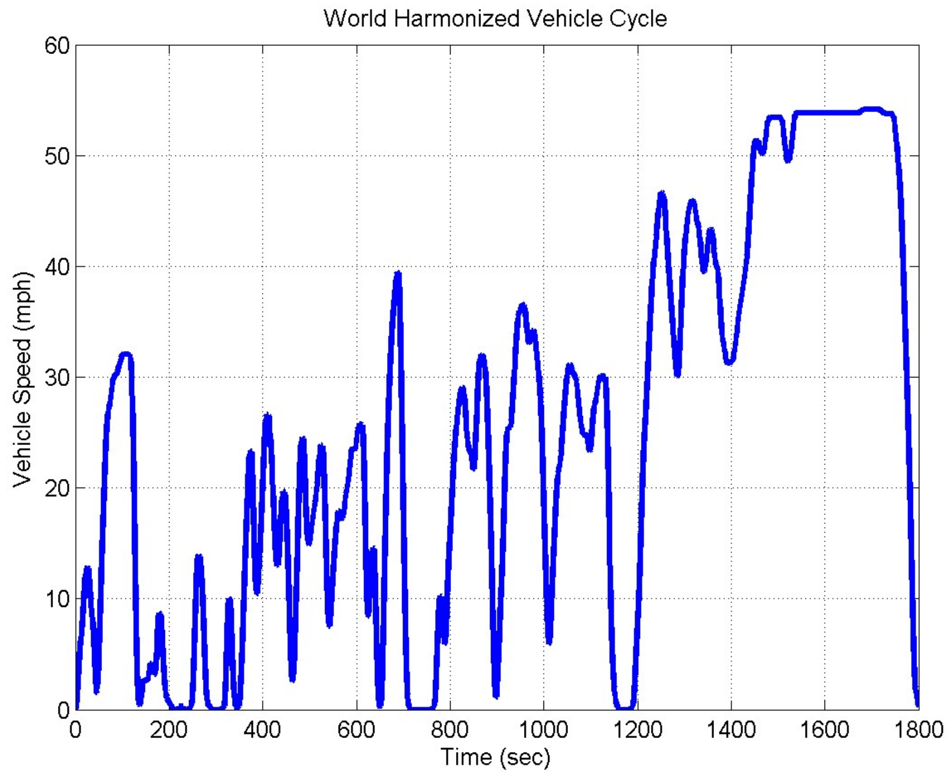


Figure C8. World Harmonized Vehicle Cycle (WHVC)

The FTP-Highway cycle shown in Figure C9 has been used for many years to evaluate light duty vehicle fuel economy. This cycle was developed in the time of the 55 MPH speed limit, so speeds are low compared to typical modern highway driving, and accelerations are gentle. This cycle was only used for the pickup truck.

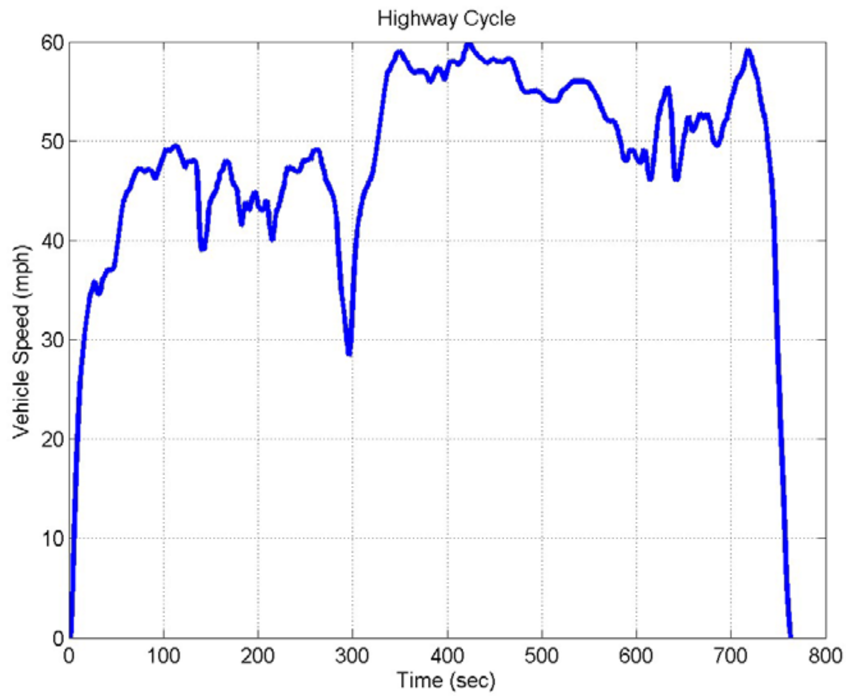


Figure C9. FTP-Highway drive cycle.

The FTP-City cycle shown in Figure C10 has been used for many years to evaluate light duty vehicle fuel economy. The cycle uses relatively gentle accelerations compared to what is found in typical city driving. This cycle was only used for the pickup truck.

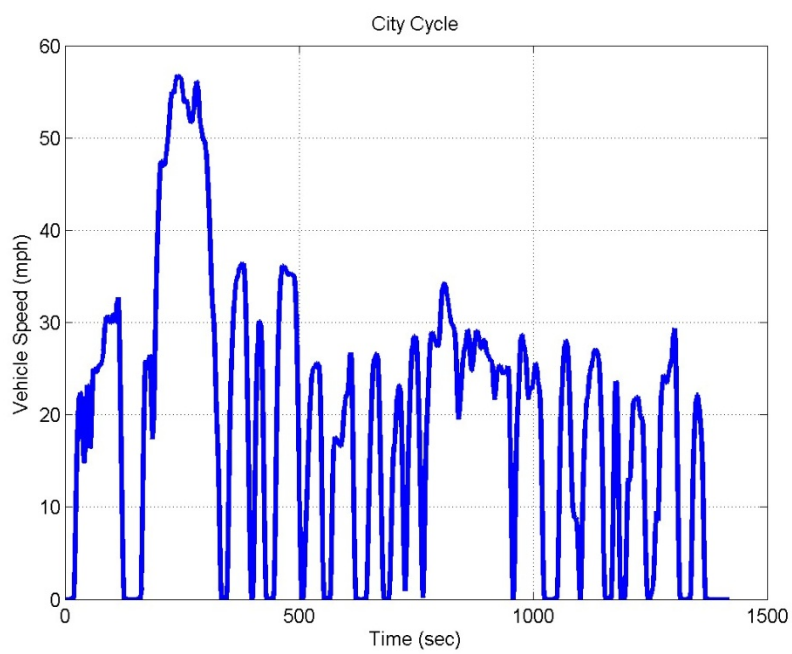


Figure C10. FTP-City drive cycle.

The CARB urban cycle shown in Figure C11 is used in the GEM regulatory model for medium and heavy-duty vehicles. This cycle simulates urban driving for trucks. The CARB cycle was used for all vehicles simulated, except the pickup truck.

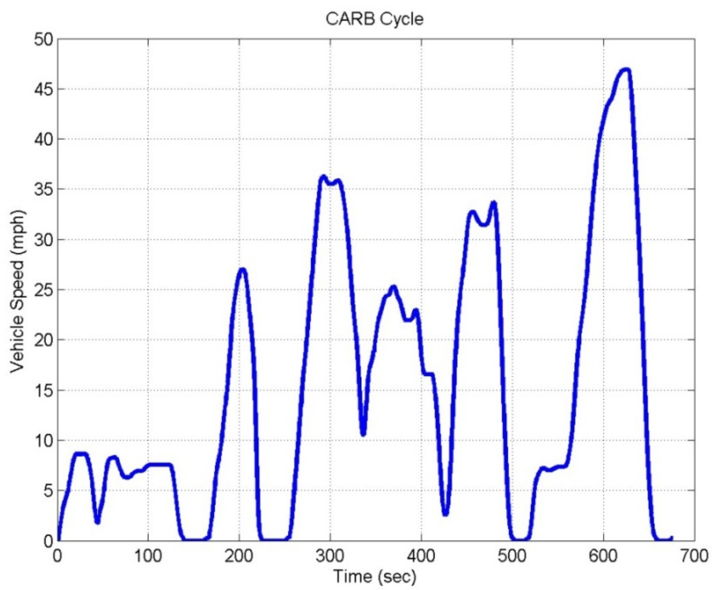


Figure C11.   CARB urban truck driving cycle.

The US06 cycle shown in Figure C12 is an aggressive drive cycle for light duty vehicles. It was introduced to help compensate for the overly optimistic fuel economy values generated by the older FTP city and highway cycles. This cycle was only used for the pickup truck.

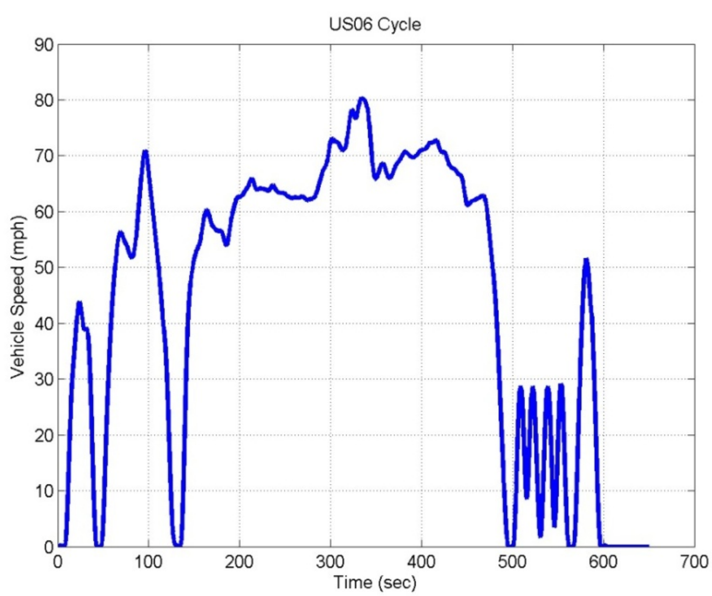


Figure C12.   US06 aggressive light duty drive cycle

The SC03 cycle shown in Figure C13 was developed to evaluate the effect of air conditioner use in hot conditions on fuel economy. In this study, the air conditioner was run on all drive cycles, which is not the standard approach. This cycle was used only with the pickup truck.

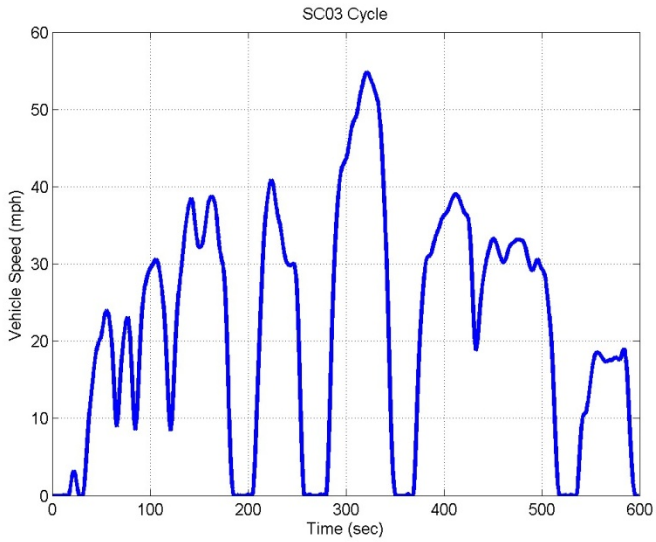


Figure C13. SC03 air conditioner drive cycle

The Parcel cycle shown in Figure C14 was developed to model the drive cycle of a parcel delivery truck. About 50% of the drive cycle time is at idle. This causes fuel consumption at idle to be significant, especially with an automatic, where the torque converter load is significant. The Parcel cycle was only used for the T270 and F-650 medium trucks.

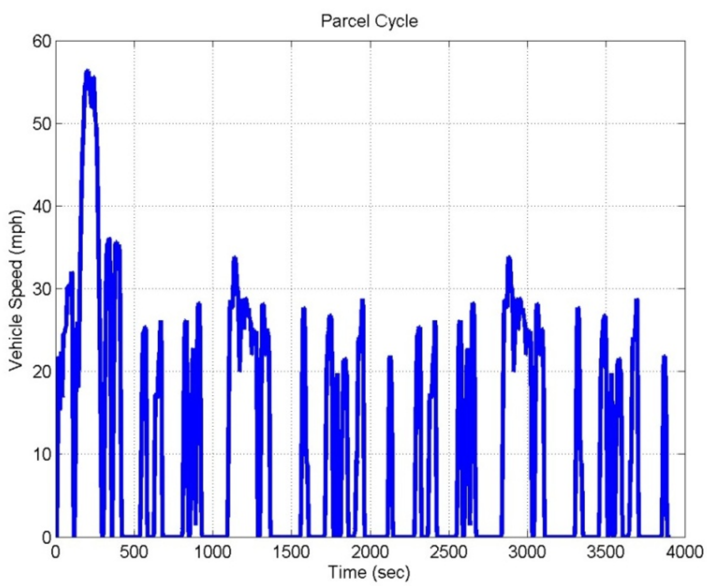


Figure C14. Parcel delivery drive cycle

The Combined International Local and Commuter Cycle (CILCC) shown in Figure C15 was developed to simulate urban driving. The general approach, with steady, very gentle accelerations, steady speed operation, and then gradual deceleration, is similar to the European NEDC cycle. This cycle results in very light loads on the engine. The CILCC cycle was only used for the T270 and F-650 medium trucks.

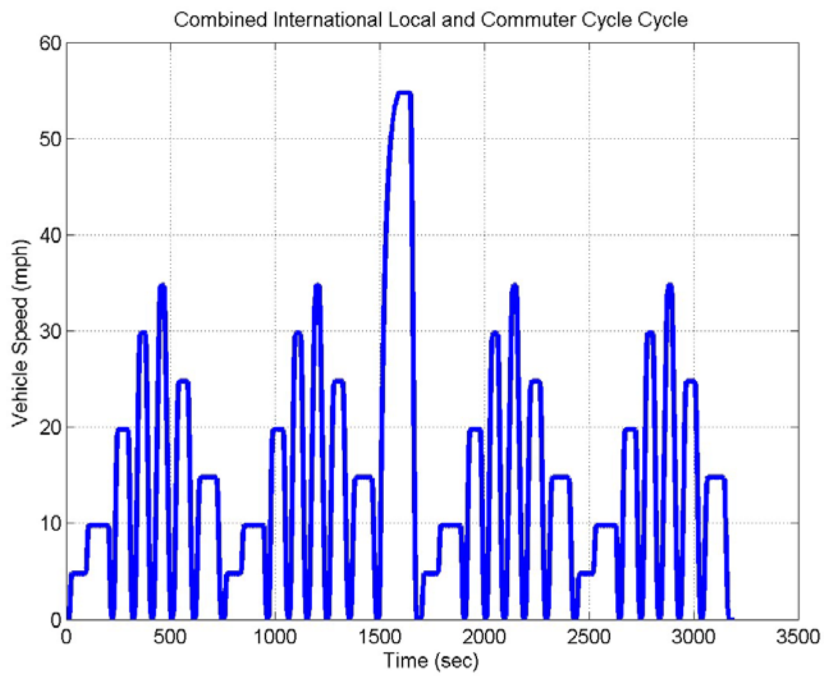


Figure C15. Combined International Local and Commuter drive cycle (CILCC)

There are two remaining drive cycles that were used: the 55 MPH and 65 MPH cruise cycles. These cycles are strictly steady state, with no grades or other changes in load, so they effectively only operate the engine at one speed/load point. These two cycles are part of the GEM vehicle certification model. The 65 MPH cruise was used for all vehicles in the study, while the 55 MPH cycle was used for all vehicles except the pickup truck.

C2. Vehicle Technologies

C2.1 Reduced Air Conditioner Power Demand

Accessory power demand is one of the contributors to overall vehicle power demand. Accessories can be defined as power absorbing devices that are not necessary to run the engine. These include the alternator, power steering pump, and air conditioner. In this report, devices that are required in order to run the engine, such as the water pump, oil pump, and fuel pump, are treated as part of the engine friction. As a representative accessory power demand reduction, the air conditioner system was evaluated. The air conditioner is operating on all vehicles during all drive cycles that are covered in this report. The power demand of the air conditioner system (including compressor, evaporator fan, and condenser fan) was modeled as 1,500 watts. This is a typical power demand for steady-state operation in a hot (90 to 95F) environment

An improved air conditioner system can incorporate features such as a more efficient compressor (such as a scroll design), the use of less reheat, or the use of additional cab insulation. For this project, a reduction in air conditioner demand of 40% was simulated, from a baseline of 1,500 watts to 900 watts. No specific technology approaches were considered, so the only effect on the vehicle is the reduction in power demand.

For all other accessory loads, assumptions were made for the average power demand. The assumptions are listed below Tables C.5 – C.7.

Table C.5 Dodge Ram accessory power demands.

Dodge RAM	air cond pwr	other mech_pwr[1]	acc_mech_pwr
base	1500	2950	**4450**
scale factor	60%	100%	**60%**
offset	0.00	0.00	**0**
number of variations	2	1	**2**
Min(Value)	900	2950	**3850**
Max(Value)	1500	2950	**4450**
Increment Value	600	0	**600**
Variant 1	900	2950	**3850**
Variant 2	1500	-	**4450**
Variant 3	-	-	**-**
Units	**W**	**W**	**W**

[1] Assumptions: Cooling fan = 1 kW, Alternator = 1 kW, Vacuum Pump = 250 W, Power Steering = 700W

In the drive cycle simulation runs, these average power demands were continuously applied during the entire cycle, including at idle. In practice, many of these loads are quite variable, and may be engine speed and ambient temperature dependent. Because no detailed data for actual accessory power demands on each vehicle was available, SwRI made engineering assumptions for average power demand levels.

Table C.6 F-650 and T-270 accessory power demands.

Ford F650/KW T270	air cond pwr	other mech_pwr[2]	acc_mech_pwr
base	1500	4250	**5750**
scale factor	60%	100%	**60%**
offset	0.00	0.00	**0**
number of variations (max of 3)	2	1	**2**
Min(Value)	900	4250	**5150**
Max(Value)	1500	4250	**5750**
Increment Value	600	0	**600**
Variant 1	900	4250	**5150**
Variant 2	1500	999	**5750**
Variant 3	999	999	**999**
Units	**W**	**W**	**W**

[2] Assumptions: Cooling fan = 2 kW, Alternator = 1 kW, Vacuum Pump = 250 W, Power Steering = 1 kW

Table C.7 T-700 accessory power demands.

KW T700	air cond pwr	other mech_pwr[3]	acc_mech_pwr
base	1500	5150	**6650**
scale factor	60%	100%	**60%**
offset	0.00	0.00	**0**
number of variations (max of 3)	2	1	**2**
Min(Value)	900	5150	**6050**
Max(Value)	1500	5150	**6650**
Increment Value	600	0	**600**
Variant 1	900	5150	**6050**
Variant 2	1500	999	**6650**
Variant 3	999	999	**999**
Units	**W**	**W**	**W**

[3] Assumptions: Cooling fan = 2 kW, Alternator = 1.25 kW, No Vacuum Pump, Power Steering = 1.25 kW, Air Compressor = 650 W

C2.2 Reduced Aerodynamic Drag (Cd)

Aerodynamic drag can be a substantial contributor to overall vehicle power demand, particularly at higher road speeds. Depending on the vehicle type, there is a range of potential for improvement in Cd. Table C.5 below shows the baseline Cd values and potential Cd improvements for each vehicle type. Results shown in Section 5.3.2.2 show that fuel consumption improvements are very linear with changes in Cd value. Therefore, the reader can interpolate or extrapolate from the benefits shown in Section 5 on any vehicle, using any given level of Cd improvement

In this study, changes in Cd are not associated with any specific hardware changes to the vehicle. For the purposes of the study, it was assumed that the OEM would select a hardware set that provides the specified Cd reduction.

Table C.8. Aerodynamic drag baseline values and reductions for each vehicle type.

Vehicle	Drag Coefficient Information					
	Baseline Cd	**Baseline CdA**	**Reduced Cd**	**Reduced CdA**	**Cd w/ Trailer**	**Cd Reduced w/ Trailer**
Dodge RAM	0.400	1.505	0.36 (10%)	1.354 (10%)	0.600	0.540 (10%)
Ford F650	0.619	3.151	0.557 (10%)	2.836 (10%)	--	--
KW T270	0.514	5.033	0.437 (15%)	4.278 (15%)	--	--
KW T700	--	6.481	--	4.861 (25%)	0.639	0.479 (25%)

C2.3 Reduced Tire Rolling Resistance (Crr)

Tire rolling resistance is a major contributor to overall vehicle power demand. For all of the vehicles involved in this study, a 30% reduction in tire rolling resistance was assumed. The specific tire features required to obtain this change in rolling resistance, and any trade-offs with other aspects of tire performance, were not considered. As shown in Section 5.3.2.3, the fuel consumption benefit of a reduction in Crr is very linear. As a result, the reader can interpolate or extrapolate from the benefits shown in Section 5 on any vehicle, using any given level of Crr improvement or increase.

The tire rolling resistance data used for all vehicle simulations comes from coastdown testing. In coastdown tests, there is no way to separate tire rolling resistance from driveline friction (transmission output shaft, driveshaft and axle). Axle efficiency test data was used to quantify the contribution from axle friction, while engineering judgment was used to split the remaining rolling resistance between tires (96%) and the transmission output and driveshaft

(4%). The coastdown testing was run on tires that were broken in but near the full tread depth. Lab data on the tires used in the coastdown testing is not available.

For the tractor-trailer vehicle, separate Crr values were used for the steer tires, drive tires, and trailer tires. For the medium-duty trucks, a single Crr value was used for the steer and drive tires. When the simulation models were run, only the average Crr value was used, so the effect of weight distribution at different payloads was not taken into account. These Crr values came from truck coastdown test runs and were divided between steer, drive, and trailer values using engineering judgment, so they will not match with values required by any particular standard, such as SmartWay or the 2014 GHG standards. Table C.9 provides the baseline Crr values and the improvement assumptions for each vehicle type. The Crr values for each vehicle were obtained from coast-down test results.

Table C.9. Tire rolling resistance baseline values and reductions for each vehicle type.

Vehicle	Rolling Resistance				
	Baseline				Reduced
	Steer	Drive	Trailer	Average	
Dodge RAM	N/A	N/A	N/A	0.007800	0.054600
Ford F650	N/A	N/A	N/A	0.010068	0.007047
KW T270	N/A	N/A	N/A	0.010967	0.007677
KW T700	0.00535	0.00650	0.00482	0.005608	0.003926

Table C.10 below shows the effect of payload on the distribution of tire load and rolling resistance for a tractor-trailer truck. The average Crr value shown in Table C.9 was calculated at 50% payload.

Table C.10. Effect of payload on weight and rolling resistance distribution.

Description	Steer Axle Weight, lb	Drive Tandem Weight, lb	Trailer Tandem Weight, lb
0% Payload case, Total vehicle weight 33960 lb.			
Load distribution of empty tractor-trailer*	**11000**	**15460**	**7500**
% weight carried by axle or tandem	**32%**	**46%**	**22%**
Jan 2015 SmartWay thresholds for Crr tires, kg/T (ISO 25850)	**6.5**	**6.6**	**5.1**
Rolling resistance contribution of axle to Crr(veh), kg/T	**2.11**	**3.00**	**1.13**
Total effective vehicle rolling resistance coefficient, Crr(veh), kg/T		**6.24**	
% Contribution of axle, tandem to total effective vehicle Crr(veh)	**34%**	**48%**	**18%**
100% Payload case, Total vehicle weight 80000 lb.			
Load distribution of 46040-lb payload tractor-trailer	**12000**	**34000**	**34000**
% weight carried by axle or tandem	**15%**	**43%**	**43%**
Jan 2015 SmartWay thresholds for Crr tires, kg/T (ISO 25850)	**6.5**	**6.6**	**5.1**
Rolling resistance contribution of axle to Crr(veh), kg/T	**0.98**	**2.81**	**2.17**
Total effective vehicle rolling resistance coefficient, Crr(veh), kg/T		**5.95**	
% Contribution of axle, tandem to total effective vehicle Crr(veh)	**16%**	**47%**	**36%**
*** Assuming steer axle weight does not vary greatly due to engine weight on axle.**			

C2.4 Weight Reduction

Vehicle mass (weight) has two effects. First, it takes energy to accelerate the vehicle mass up to a desired speed. The energy put into accelerating the vehicle is largely lost as brake heat when the vehicle slows down. The second consequence of vehicle mass is that rolling resistance is proportional to mass. If vehicle mass can be reduced, less power is required to accelerate the mass, and less power is required to overcome rolling resistance. Mass reduction can be achieved in a number of ways, including:

- Material substitution, such as replacing steel with aluminum or a composite
- Design changes to eliminate mass that is not required to achieve the target capability and durability

Unfortunately, many fuel saving, emissions, and safety technologies add mass. Maintaining a given truck empty weight takes a consistent weight reduction effort to offset new features that added for both regulatory and vehicle performance reasons.

The effect of vehicle mass on light duty vehicle fuel efficiency is significant. Large fuel economy benefits can be obtained by reducing light duty vehicle mass. Considered from the point of view of freight efficiency, light duty vehicles are extremely inefficient. A 4,000 pound light duty vehicle transporting a 200 pound person has a “cargo” constituting less than 5% of the total vehicle mass. If the empty weight of our example light duty vehicle is reduced 10% (400 pounds), the total mass of vehicle plus cargo is reduced by 9.5%. Heavy duty trucks, on the other hand, may have a cargo mass greater than that of the vehicle. Take for example a tractor-trailer with an empty weight of 34,000 pounds and a loaded weight of 80,000 pounds. If the empty weight is reduced by 10%, the total mass of truck + cargo is reduced by only 4.25%, half as much as in our light duty example.

Table C.11 shows the empty weights and payload weights for each vehicle in the project. The specific technologies or features added to achieve the weight reduction were not specified. The assumption was made that the OEM would provide a set of changes required to achieve the target weight reduction. The following weight reductions were applied to each vehicle to evaluate the effect of a reduction in empty weight:

Vehicle	Weight Reductions
Ram Pickup	500 Pounds
F-650 Tow Truck	1100 Pounds
T-270 Box Truck	1100 Pounds
T-700 Tractor-Trailer	2200 Pounds, 4400 Pounds

These weight reductions were applied to the tare weights shown in Table C.11 below.

Table C.11. Vehicle empty weights and payloads used in simulations. "Tare" weight is equivalent to "curb weight".

Weights in Pounds	RAM diesel			RAM gasoline		
	Tare	Payload	Total	Tare	Payload	Total
0% payload	6876	0	6876	6376	0	6376
ALVW	6876	1562	8438	6376	1562	7938
100% GCW	6876	18124	25000	6376	18124	24500

Weights in Pounds	F650 diesel			F650 gasoline		
	Tare	Payload	Total	Tare	Payload	Total
0% payload	15640	0	15640	15139	0	15139
50% payload	15640	3180	18820	15139	3180	18319
100% payload	15640	6360	21999	15139	6360	21499

Weights in Pounds	T270 diesel			T270 gasoline		
	Tare	Payload	Total	Tare	Payload	Total
0% payload	17141	0	17141	16640	0	16640
50% payload	17141	4430	21571	16640	4430	21070
100% payload	17141	8860	26001	16640	8860	25500

Weights in Pounds	T700 diesel		
	Tare	Payload	Total
0% payload	33960	0	33960
50% payload	33960	23020	56980
100% payload	33960	46039	79999

The percentage of fuel consumption reduction that is provided by a given percent weight reduction is a function of several parameters:

- Drive cycle
- Coefficient of rolling resistance
- Vehicle payload

A given percent weight reduction will provide a larger percent fuel savings on a drive cycle with frequent stops and starts. This is because inertia loads represent a higher portion of the total power demand on a highly transient cycle. If the coefficient of rolling resistance is high, there will be a larger benefit from a weight reduction, since rolling resistance will be a larger portion of overall power demand. As the payload increases, a given percentage weight reduction on the empty vehicle becomes a smaller percentage of the test weight, so the percent fuel savings decreases. Refer to the results in Section 3 to see how percentage weight reductions translate into different fuel savings for the four vehicles, over a range of drive cycles and payloads.

C2.5 Chassis Friction Reduction

Chassis friction includes losses in the axle, U-joints, and wheel bearings. An improvement in axle efficiency, for example, can lead to a reduction in overall chassis friction power demand. In this study, no effort was made to evaluate individual chassis friction technologies, except for the use of a 6X2 drive axle configuration in the tractor-trailer truck (See Section C2.6 below). Table C.12 shows the percentage of vehicle power demand accounted for by chassis friction at a steady speed of 65 MPH, along with the level of improvement that has been simulated for each vehicle type.

Table C.12. Chassis friction levels at 65 MPH, and friction reduction assumptions

Chassis Friction	Dodge RAM	Ford F650	KW T270	KW T700
Driveline Spin Losses (N/N)	0.000197	0.000488	0.000606	0.00023
Axle Loaded Torque Efficiency (%)	97.5	97.5	97.5	97.5
Driveline Spin Losses Reduction (%)	30%	30%	30%	20%

C2.6 6X2 Axle Configuration

One way to reduce chassis friction on a tractor-trailer truck is to reduce the number of drive axles from the standard value of two, down to one. A two drive axle configuration is referred to as a 6X4, while a single drive axle configuration is referred to as a 6X2. When two drive axles are used, there are two sets of spin losses that need to be overcome, rather than only one. Note that the single axle needs to have a higher torque capacity than the tandems, so it will typically have a higher spin loss than the rear axle in a tandem pair. Another source of losses in the tandem is that the front axle in a tandem needs to split the power between the two axles. This involves additional gear sets and oil seals, which are losses that do not occur with a single drive axle. In the tandem axle case, it is assumed that the input torque is split equally between the front and rear drive axles. The front drive axle is assumed to be 95% efficient while the rear is assumed to be 97.5% efficient. In the single drive axle case, all of the torque is transmitted to the wheels with an efficiency of 97.5%.

The single drive axle has a downside: reduced traction. The standard legal weight limit on a tandem axle set is 34,000 pounds, and the limit for a single axle in a tandem pair is 17,000 pounds. So, for a fully loaded tractor trailer at 80,000 pounds, there is only 17,000 pounds, or 21.25% of the total vehicle weight, on the drive axle of a 6X2 configuration. This can lead to problems in low friction environment surfaces, such as snow, ice, sand, or even wet pavement. Trucks that need to go off-road, or which frequently operate in snow and ice, are not good candidates for a 6X2 configuration. One way to deal with the traction issue is to use the air suspension to temporarily lift the non-driven axle in situations where traction becomes a problem.

In this project, the 6X2 configuration is compared to the standard 6X4 configuration on the Kenworth T-700 tractor-trailer truck.

C2.7 Road Speed Governor (Reduced V_{max})

Road speed governors are widely used by large truck fleets as a fuel saving technology. The governor limits the road speed to a value set by the owner. Road speed governors can also be used to gain credits under the GEM model, but in this case the governor has to be set at the factory either permanently or for a defined number of miles, and cannot be changed by the owner.

In this study, road speed governors were evaluated only on the Kenworth T-700 tractor-trailer. Results can be found in Section 5.3.2.7.

C2.8 Transmission Alternatives

A wide range of transmissions have been evaluated in this program. Section C1.2 shows which transmissions were evaluated in each vehicle model. This section describes the individual transmissions in detail.

Table C.13 Transmission Parameter Tables for All Vehicle Configurations

	Dodge RAM	Veh 1	Veh 2	Veh 3	Veh 4	Veh 5	Veh 6
Transmission Parameters	Num of Gears	6	6	8	6	6	6
	1	3.231	3.231	4.696	3.231	3.231	3.231
	2	1.837	1.837	3.130	1.837	1.837	1.837
	3	1.410	1.410	2.104	1.410	1.410	1.410
	4	1.000	1.000	1.667	1.000	1.000	1.000
	5	0.816	0.816	1.285	0.816	0.816	0.816
	6	0.625	0.625	1.000	0.625	0.625	0.625
	7	-	-	0.839	-	-	-
	8	-	-	0.667	-	-	-
	TC Stall K Factor	9.35	9.35	9.35	11.55	8.74	11.55
	TC Stall Torque Ratio	1.74	1.74	1.74	2.16	1.7	2.16

	F650	Veh 1	Veh 2	Veh 3	Veh 4	Veh 5	Veh 6	Veh 7
Transmission Parameters	**Num of Gears**	5	6	10	8	5	5	5
	1	3.102	9.01	12.796	4.696	3.102	3.102	3.102
	2	1.8107	5.27	9.251	3.13	1.8107	1.8107	1.8107
	3	1.406	3.220	6.761	2.104	1.406	1.406	1.406
	4	1.000	2.040	4.901	1.667	1.000	1.000	1.000
	5	0.7117	1.36	3.579	1.285	0.7117	0.7117	0.7117
	6		1.000	2.611	1.000	-	-	-
	7	-	-	1.888	0.839	-	-	-
	8	-	-	1.38	0.667	-	-	-
	9	-	-	1	-	-	-	-
	10	-	-	0.73	-	-	-	-
	TC Stall K Factor	10.26	-	-	10.26	11.55	8.74	11.55
	TC Stall Torque Ratio	2.71	-	-	2.71	2.16	1.70	2.16

	T270	Veh 1	Veh 2	Veh 3	Veh 4	Veh 5	Veh 6	Veh 7
Transmission Parameters	**Num of Gears**	5	6	10	8	5	5	5
	1	3.102	9.01	12.796	4.696	3.102	3.102	3.102
	2	1.8107	5.27	9.251	3.13	1.8107	1.8107	1.8107
	3	1.406	3.220	6.761	2.104	1.406	1.406	1.406
	4	1.000	2.040	4.901	1.667	1.000	1.000	1.000
	5	0.7117	1.36	3.579	1.285	0.7117	0.7117	0.7117
	6	-	1.000	2.611	1.000	-	-	-
	7	-	-	1.888	0.839	-	-	-
	8	-	-	1.38	0.667	-	-	-
		-	-	1	-	-	-	-
		-	-	0.73	-	-	-	-
	TC Stall K Factor	10.26	-	-	10.26	11.55	8.74	11.55
	TC Stall Torque Ratio	2.71	-	-	2.71	2.16	1.70	2.16

Transmission Parameters	T700	Veh 1	Veh 2	Veh 3	Veh 4	Veh 5	Veh 6	Veh 7
	Num of Gears	10	10	10	10	18	10	10
	1	12.796	12.796	12.796	12.796	14.4	12.796	12.796
	2	9.251	9.251	9.251	9.251	12.29	9.251	9.251
	3	6.761	6.761	6.761	6.761	8.56	6.761	6.761
	4	4.901	4.901	4.901	4.901	7.300	4.901	4.901
	5	3.579	3.579	3.579	3.579	6.050	3.579	3.579
	6	2.611	2.611	2.611	2.611	5.16	2.611	2.611
	7	1.888	1.888	1.888	1.888	4.380	1.888	1.888
	8	1.38	1.38	1.38	1.38	3.74	1.38	1.38
	9	1	1	1	1	3.2	1	1
	10	0.73	0.73	0.73	0.73	2.73	0.73	0.73
	11	-	-	-	-	2.29	-	-
	12	-	-	-	-	1.95	-	-
	13	-	-	-	-	1.62	-	-
	14	-	-	-	-	1.38	-	-
	15	-	-	-	-	1.17	-	-
	16	-	-	-	-	1	-	-
	17	-	-	-	-	0.86	-	-
	18	-	-	-	-	0.73	-	-

C2.8.1 6-Speed, 10-Speed, and 18-Speed AMT Transmissions

The 6-speed and 10-speed AMT transmissions were evaluated as alternative transmissions in the medium duty applications (Kenworth T-270 and Ford F-650). The 10-speed AMT was the standard transmission in the long haul Kenworth T-700 tractor trailer truck. The 18-speed AMT was used as an alternative in the T-700. Proprietary SwRI test data was used to quantify the efficiency of each gear as a function of input torque. Eaton provided proprietary shift schedules for the 6- and 10-speed transmissions. SwRI modified these schedules slightly to achieve smooth transitions during the drive cycles – in other words, to avoid situations of excessive gear hunting. SwRI developed a shift schedule for the 18-speed transmission, based on the 10-speed schedule. Table C13 above details the gear ratios and other characteristics of these transmissions.

Table C13 above shows the transmission ratios for the 6-speed AMT transmission. The data can be found in the Veh 2 column for both the T-270 and F-650 portions of the table.

Table C13 above shows the transmission ratios for the 10-speed AMT transmission. The data can be found in the Veh 3 column for both the T-270 and F-650 portions of the table. The same data is also shown for the T-700 tractor in every column *except* Vehicle 5. Table C13 above also shows the transmission ratios for the 18-speed AMT transmission in the T-700 section in the column Vehicle 5.

C2.8.2 10-Speed Manual Transmission

The 10-speed manual transmission is mechanically identical to the 10-Speed AMT. The only changes are the deletion of the automated shifting system, and the use of a shift map with increased upshift speeds to represent a more "typical" driver. SwRI does not have extensive vehicle data available, which would be required to develop a validated shift schedule. As a result, it was decided to increase upshift RPM by 200 RPM at full load as an approximation.

C2.8.3 Automatic Transmissions

A variety of torque converter automatic transmissions were used for the Ram pickup and the medium duty trucks (T-270 and F-650). The standard transmission for the pickup truck simulation model was the Aisin 6-speed automatic. This is the transmission used in the 2014 Ram with the 385 HP rating of the Cummins ISB engine. SwRI does not have access to factory efficiency data, torque converter match, and shift schedules, so SwRI used existing data and engineering judgment to develop the required parameters. Alternative torque converter matches were developed for the 3.5 liter V-6 and 6.2 liter V-8 gasoline engines. Table C13 provides data on the 6-speed Aisin transmission.

Table C13 above shows the transmission ratios for the 6-speed torque converter automatic transmission in the Dodge Ram section in all columns *except* Vehicle 3.

For the T-270 and F-650 medium-duty trucks, the standard transmission is a 5-speed Allison 2000 Series. This is actually a 6-speed box, but for many applications, 6th gear is not used. The explanation for this is that 6th gear is a tall overdrive ratio, which causes the driveshaft speed to be well above engine speed. High driveshaft speeds lead to two potential problems. One issue is that the driveshaft length must be limited, in order to avoid driveshaft whirl. The fix for this issue is to use a multi-piece driveshaft with carrier bearings, so that each driveshaft segment is short enough to avoid whirl.

The second issue with a tall overdrive ratio is that it increases the frequency of any imbalance forces that may occur in a driveshaft. Unbalance forces in the driveshaft excite the powertrain bending resonances of the engine/transmission assembly. If the frequency of driveshaft rotation matches the lowest powertrain bending resonance somewhere in the operating range of the engine, mechanical failures of the flywheel housing are likely. Fixing this issue requires an increase in powertrain bending frequency. This is normally obtained by optimization of the flywheel housing structure. Occasionally, the transmission housing may also need to be stiffened. In some cases, it is not possible to achieve the target powertrain bending resonance frequency with conventional stiffening measures. In these cases, external brackets which tie the engine block directly to the transmission housing may be required. Some example values:

Maximum engine speed: 2,800 RPM
Overdrive ratio: 0.62:1
Maximum driveshaft speed: 2,200/0.62 = 4,516 RPM = 75.3 Hz
Target powertrain bending frequency with 15% margin: 86.6 Hz

Powertrain bending excitation normally becomes a problem when the vehicle is going downhill and the driver is using the engine for braking. Under these conditions, the engine can be motored above its normal maximum speed. This causes high driveshaft speed, and thus high rotating imbalance force frequency. Shaft whirl is also more likely to be a problem in downhill operation with engine overspeed.

The 2012 model T-270 and F-650 trucks that were modeled in this project both lock out 6^{th} gear. As of 2014, Ford has modified their truck to make use of 6^{th} gear. Kenworth now offers both 5 and 6-speed versions of the Allison transmission, depending on the vehicle specification. The trend appears to be moving towards drivelines that can accommodate taller overdrive ratios.

Different torque converter matches were selected for the ISB diesel and the two gasoline engines. Allison provided proprietary efficiency data, which was input to the vehicle simulation model. Details of the 5-speed torque converter automatic transmission used in the T-270 and F-650 trucks are shown in Table C13 above, in columns Veh 1, Veh 5, Veh 6, and Veh 7.

As an upgrade to the 5 and 6-speed automatics, an 8-speed unit was evaluated. The advantages of an 8-speed over the baseline 5 and 6-speed units include closer ratios and a wider ratio range. The biggest advantage of the 8-speed is that it has a higher mechanical efficiency than the baseline transmission. SwRI used proprietary efficiency test data from the most efficient 8-speed light truck transmission now available, and scaled it to match the requirements of this project.

By scaling, we mean the following. Assume that the transmission subjected to physical testing has a torque limit of 500 Nm, but our engine provides 1000 Nm. Thus, the transmission's performance needs to be scaled up by a factor of 2. If the tested transmission has an efficiency of 97% at 300 Nm, we would then input an efficiency of 97% at 600 Nm for the simulation.

There are three primary sources of benefit for the 8-speed in vehicle operation:

1. The wider ratio range allows for a taller top gear for cruise
2. The closer gear spacing should allow the engine to be kept in a more efficient part of its operating map
3. The improved mechanical efficiency should reduce fuel consumption during all types of operation

In practice, advantage 2 proved to be very minor. The 5 and 6-speed transmissions actually do a very good job of keeping the engine in an efficient part of the map. Advantage 3 proved most significant, since the 8-speed is about 2% more efficient across much of the operating range.

Table C13 above shows the transmission ratios for the 8-speed torque converter automatic transmission in the Dodge Ram section in column Vehicle 3. The same data is also shown in the T-270 and F-650 sections in column Vehicle 4.

C2.9 Engine Alternatives

C2.9.1 4.5 and 6.7 Liter Diesel for Pickup (Class 2b/3)

The baseline engine for the pickup was a 385 HP 6.7 liter diesel. Several engine technologies were explored on this engine, and a 4.5 liter 4-cylinder variant was developed to explore downsizing. Details are provided in Appendix B.

C2.9.2 6.7 Liter Diesel for Classes 4 – 7

The baseline engine for the T-270 and F-650 medium-duty trucks was a 300 HP rating of the 6.7 liter diesel. Several engine technologies were explored on this engine. In addition, an 8-cylinder 8.9 liter derivative was created to cover heavier duty Class 8 vocational applications. This engine model will be used in future simulations. Details of the medium-duty diesel engines are provided in Appendix B.

C2.9.3 3.5 V-6 and 6.2 V-8 Gasoline Engines for Class 2b – 7 Vehicles

Gasoline engines were explored as alternatives to the diesel on the Ram pickup and also in the T-270 and F-650 vocational trucks. The smaller V-6 represents a downsized, turbocharged, direct injection alternative. The 6.2 V-8 is a more traditional naturally aspirated, port injected engine. Several technologies were explored on each engine type. Details of the gasoline engines and their technologies are provided in Appendix A.

C2.9.4 12.5 and 15 Liter Diesel Engines for Class 8

The baseline engine for the Kenworth T-700 long-haul tractor-trailer truck is a 485 HP rating of the 15 liter diesel. A wide range of technologies have been applied to this engine, including both downsizing (to 12.5 liters) and downspeeding. Details on these engines and technologies are provided in Appendix B.

C3 Examples of Vehicle Drive Cycle Operation

The figures below provide examples of how the engine operates on various vehicle drive cycles and payloads. These figures are very useful in a number of ways:

1. Verification of transmission shift points
2. Evaluation of drive cycle / payload aggressiveness
3. Evaluation of how well the powertrain setup (transmission and final drive ratio) keep the engine in its most efficient operating area, for a given power demand

These plots are rather complex, so some description is required. All plots have engine speed on the X-Axis, and engine torque on the Y-axis. The thick black upper curve is the engine torque curve, and the thick black lower curve (below zero torque) is the engine motoring curve. This is the torque required to spin the engine when fuel is shut off. The curved lines with numbers (205, 210, 215, etc.) are lines of constant brake specific fuel consumption, or BSFC. The engine is most efficient when it operates in the area of lowest BSFC. The blue circles show where the engine operates during the drive cycle, with the size of the blue circle indicating how much time the engine spends at or near that speed/load point. The blue circles give a good understanding of how hard the engine has to work on a given drive cycle, and how efficiently it works.

Figures C.6 and C.7 compare the NESCCAF and WHVC drive cycles. Note that the primary engine cruise point is at higher speed and torque on the NESCCAF cycle. This is because the NESCCAF has cruise speeds of 65 to 70 MPH, while the WHVC is a European cycle with a cruise speed around 53 MPH. The WHVC includes more low speed, light load city and suburban driving than the mostly-highway NESCCAF cycle. There is also more full load operation on the NESCCAF cycle, because of the +/- 1% and +/-3% grades imposed by this cycle.

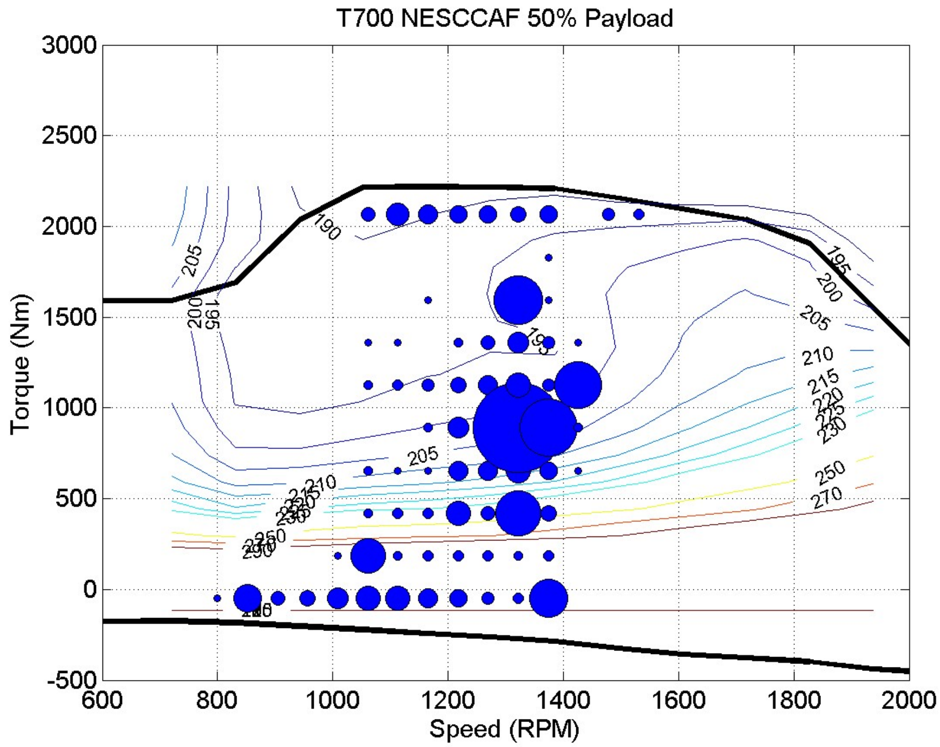


Figure C6. Engine operating map for the T-700 truck on the NESCCAF drive cycle at 50% payload.

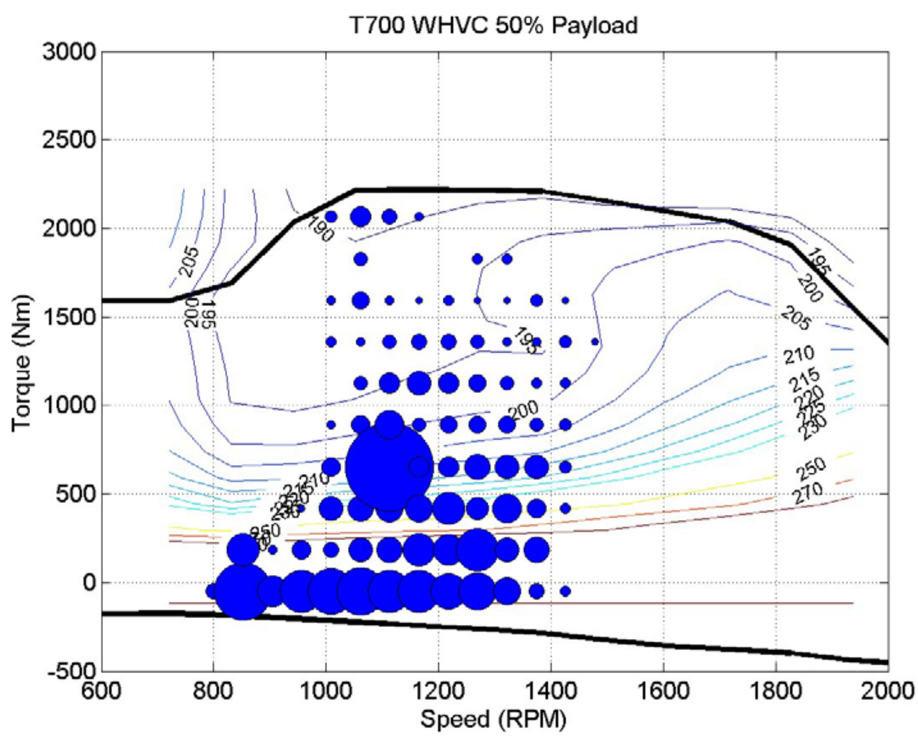


Figure C.7 Engine operating map for the T-700 truck on the WHVC drive cycle at 50% payload.

The next pair of figures compares the 65 MPH cruise operating point for the baseline engine in the T-700 tractor-trailer against the Downspeed 2 configuration. Downspeed 2 has a higher torque curve and a taller axle ratio, which reduces engine speed at 65 MPH.

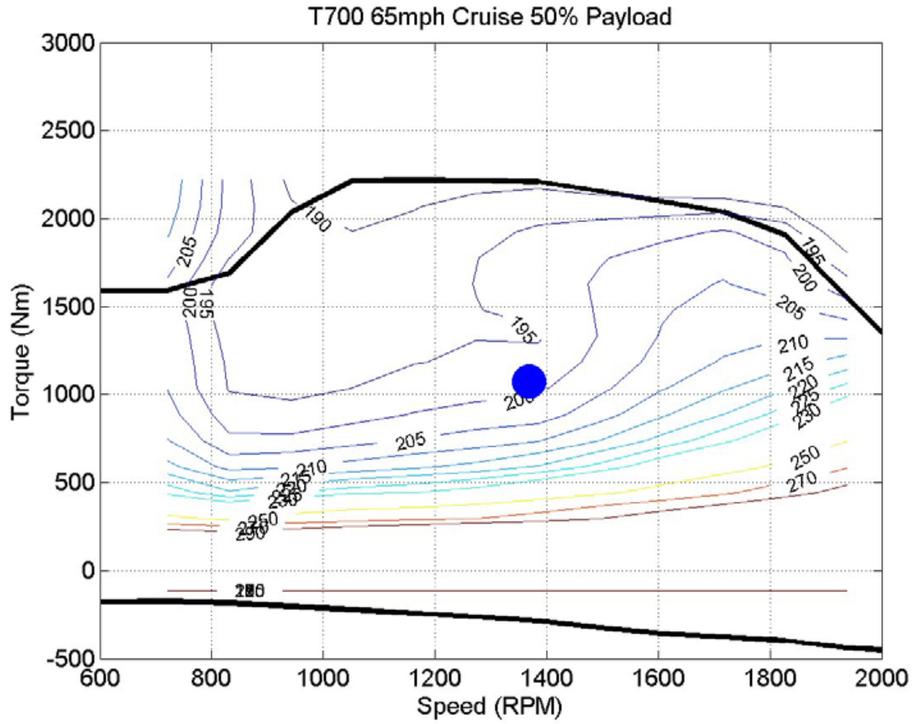


Figure C.8 Engine map for the T-700 baseline truck at 65 MPH and 50% payload.

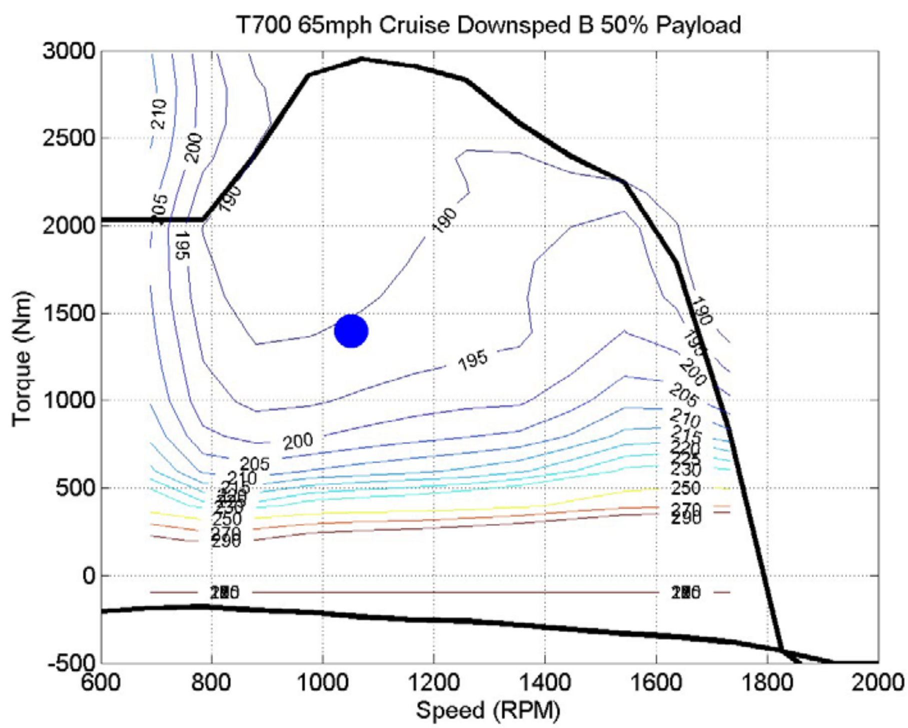


Figure C.9 Engine map for the T-700 Downspeed 2 truck at 65 MPH and 50% payload.

Note in comparing Figures C.8 and C.9 that the engine efficiency at the cruise load point improves with downspeeding. The baseline engine in Figure C.8 is operating at a point just under 200g/kW-hr, while the downspeed version is well under 195 g/kW-hr. There is about a 4% difference in engine efficiency at these two operating points.

The next four figures look at different engine types in the T-270 box delivery truck, all on the WHVC. First, Figures C.10 and C.11 compare the 6.7 liter diesel against the 6.2 liter gasoline V-8. The main cruising point for the diesel engine is around 1600 RPM and 450 Nm of torque, while the gasoline V-8 cruises at about 2650 RPM and 250 Nm of torque. Despite the big difference in engine operating point, both engines are operating near their sweet spot at cruise, and both have sufficient torque headroom to deal with some grade or with a desire to accelerate, without requiring a downshift.

Figures C.12 and C.13 compare the baseline 3.5 V-6 and the downspeed + EGR version of the 3.5 V-6 in the same vehicle and on the same drive cycle (T-270 box truck on the WHVC at 50% payload). The baseline 3.5 V-6 cruises at about 2650 PM and 250 Nm of torque, just like the V-8 in Figure C.11. The BSFC of the baseline 3.5 V-6 at the cruise point is about 240 g/kW-hr. The downspeed + EGR version of the 3.5 V-6 cruises at about 2200 RPM and 300 Nm, where the engine has a BSFC of about 225 g/kW-hr, a 6% to 7% improvement over the baseline. Both the lower cruise speed and EGR contribute to the improved efficiency of the downspeed + EGR engine.

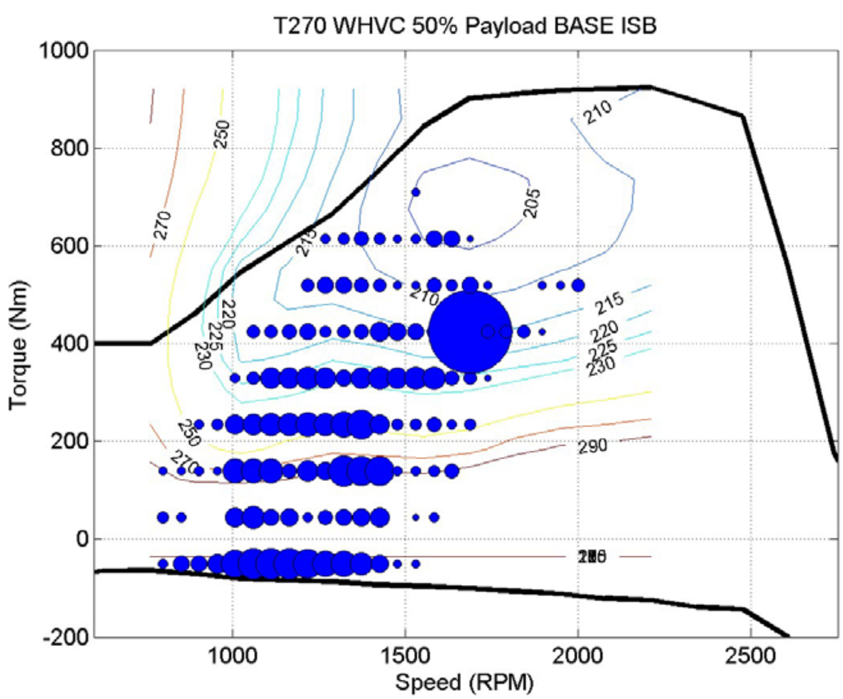


Figure C.10 Engine operating map for the T-270 truck with the baseline 6.7 liter diesel on the WHVC at 50% payload.

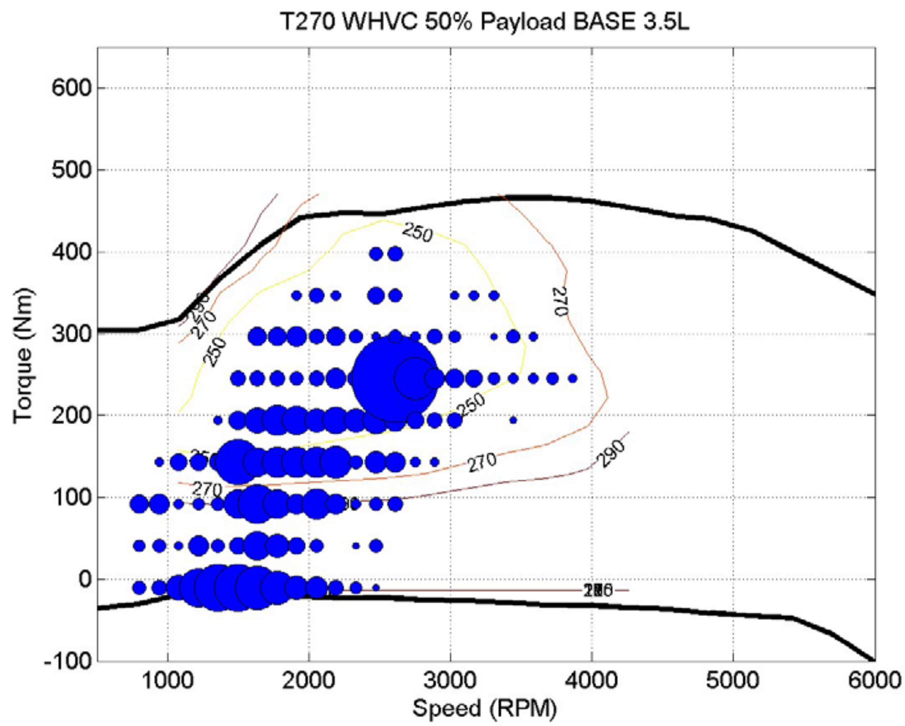


Figure C.11 Engine operating map for the T-270 truck with the baseline 6.2 liter V-8 gasoline engine on the WHVC at 50% payload.

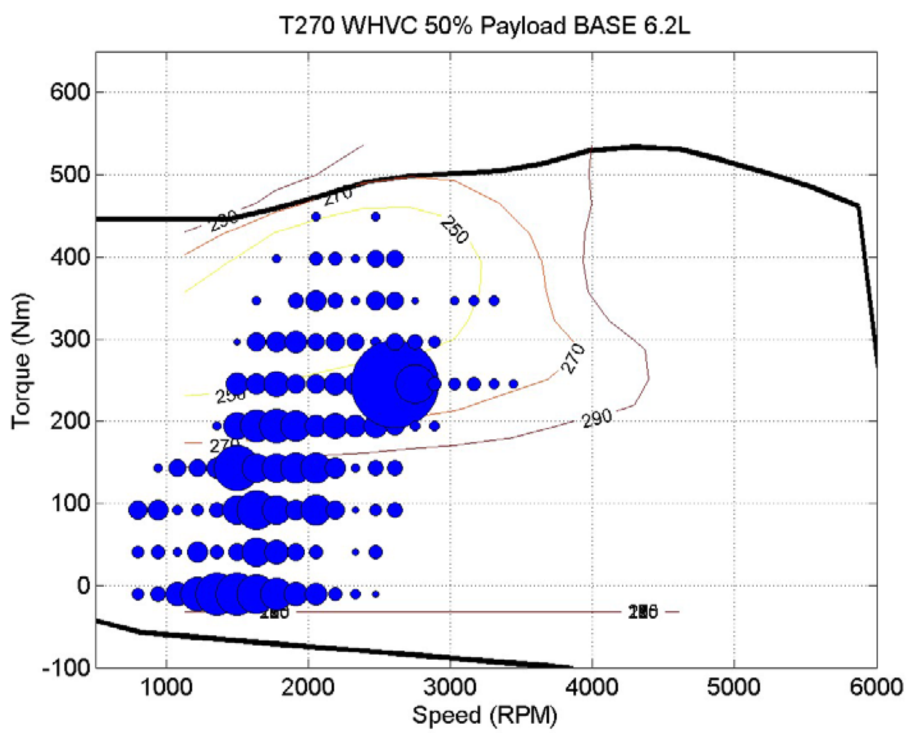


Figure C.12 Engine operating map for the T-270 truck with the baseline 3.5 V-6 on the WHVC at 50% payload.

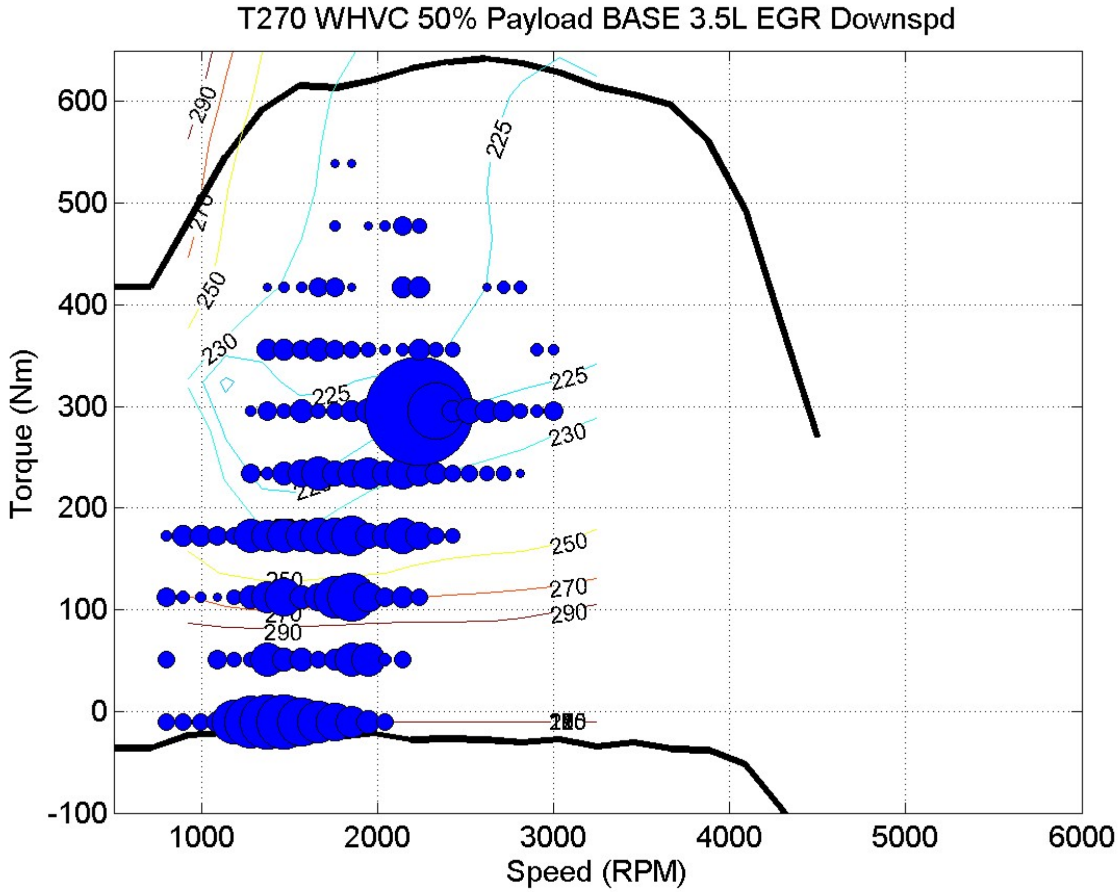


Figure C.13 Engine operating map for the T-270 truck with the Downspeed + EGR 3.5 V-6, on the WHVC at 50% payload.

The next figure provides results on the Ram pickup truck. Figure C.14 shows the Ram with the baseline diesel on the WHVC at ALVW (50% of the payload that can be loaded in the truck bed, but no trailer). This compares with Figure C.10 above, where the same basic engine is operating on the same test cycle, but in a larger and heavier vehicle. The T-270 truck (Figure C.10) has a cruise point at about 1650 RPM and 425 Nm. In the smaller Ram pickup, the cruise point is around 1250 RPM and 200 Nm. In fact, for the Ram, the engine never exceeds 1700 RPM or 400 Nm, even though the rating for the pickup has both a higher speed range and higher torque than the medium truck version of the diesel engine. For the pickup truck, rated speed is 3000 RPM, and peak torque is 1150 Nm. Note that for the pickup truck on this gentle drive cycle, the engine load never reaches the most efficient range. This suggests that a downsized engine could provide large benefits, which indeed matches the results shown in Section 3.3.6.2.

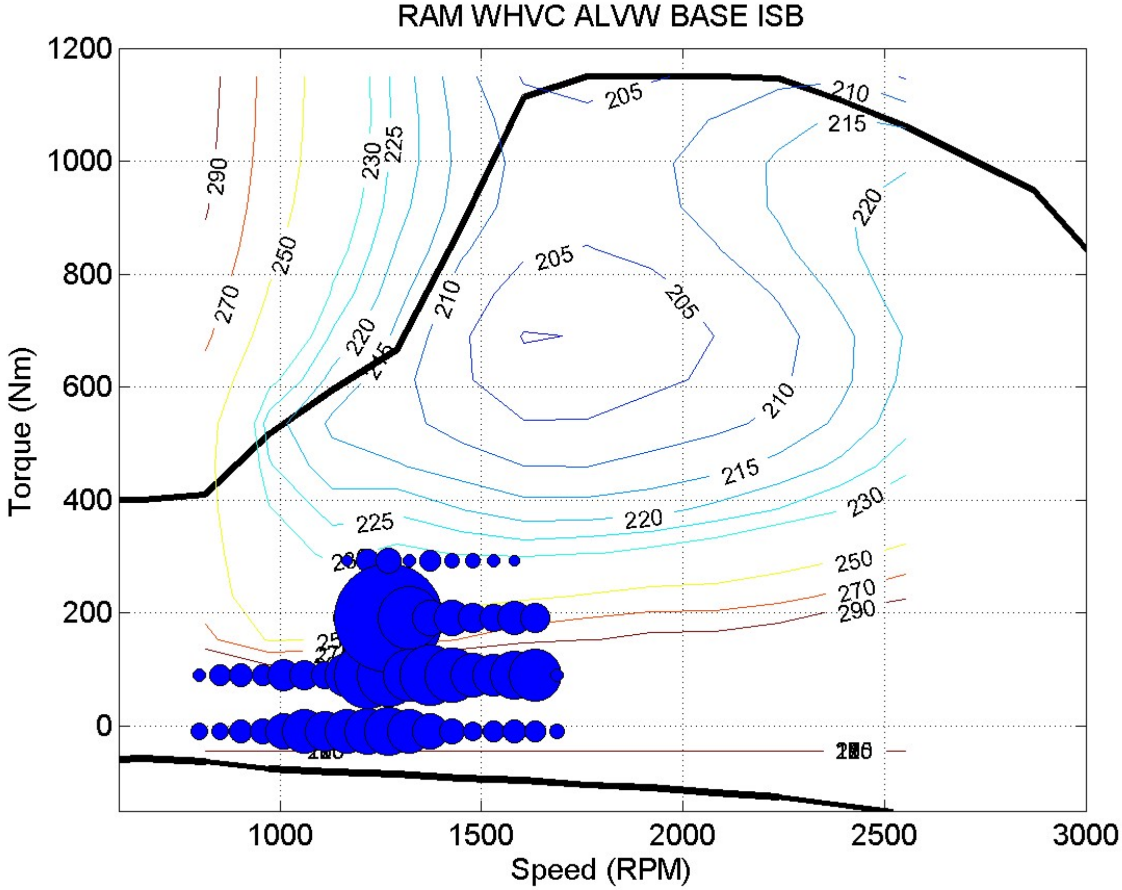


Figure C.14 Engine operating map for the Ram pickup truck with the baseline 6.7 liter diesel, on the WHVC at ALVW (8,438 pounds vehicle test weight).

The final pair of figures shows the baseline Ram diesel on the much more aggressive US06 drive cycle. Figure C.15 shows the results at zero payload (6,876 pounds), while Figure C.16 shows the results at full GCW (25,000 pounds). At zero payload, the engine reaches 2000 RPM and up to 900 Nm. Full load is never quite reached, and the engine operates below the most efficient area on most of the cycle. At GCW, the vehicle is unable to follow the drive cycle about 10% of the time, so the engine is at full load a significant portion of the time, as the vehicle tries to catch up with the drive cycle requirements. The maximum engine speed is around 2900 RPM, with significant time at the peak torque of 1150 Nm. Note that even on this severe duty cycle, the transmission does a good job of keeping the engine near its most efficient operating range whenever possible.

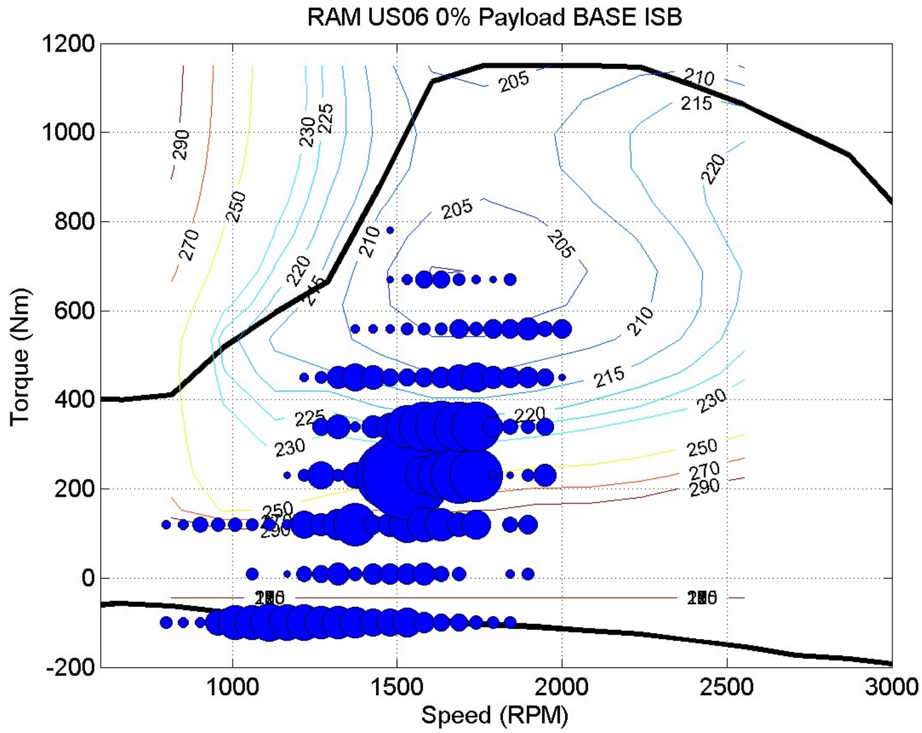


Figure C.15 Engine operating map for the Ram pickup truck with the baseline 6.7 liter diesel, on the US06 at zero payload.

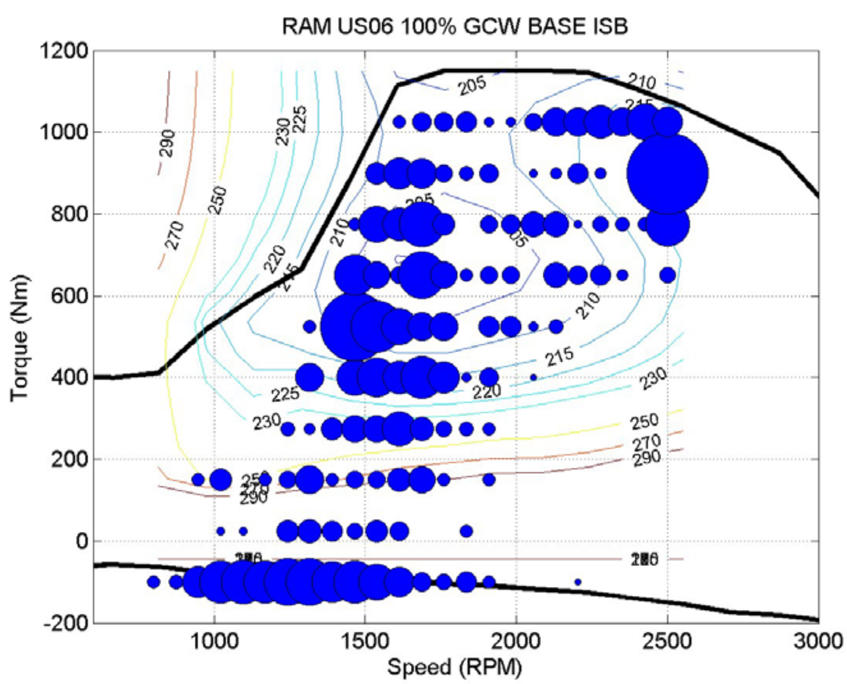


Figure C.16 Engine operating map for the Ram pickup truck with the baseline 6.7 liter diesel, on the US06 at full GCW (25,000 pounds).

APPENDIX D

BOTTOMING CYCLE MODEL DESCRIPTION

Bottoming Cycle Model Description

The Bottoming Cycle is one form of waste heat recovery that may be applied to internal combustion engines. The objective is to take heat that is otherwise "wasted" or rejected to the environment and convert as much of it as possible to useful shaft work. In an engine, the most common and useful sources of this heat are the exhaust stream, the EGR cooler (as applicable), and possibly the coolant and aftercooler systems. The conversion from heat to work is done using the Rankine Cycle. The Rankine cycle has been used for many decades as a primary steam power plant, fueled by coal or wood or virtually any heat source. The bottoming cycle is so named because it is taking heat from the "bottom" of the engine cycle and converting it to additional work.

The Rankine cycle, as shown schematically in Fig D1, is composed of four primary items:

1. Pump
2. Evaporator or boiler
3. Expander (turbine or piston)
4. Condenser

The cycle fluid is selected to have the desired properties, namely a boiling point in the range of the cycle's heat source. The fluid comes out of the condenser as a liquid and is pumped to high pressure in the pump. The fluid is then boiled to vapor phase in the evaporator. The heat for this heat exchanger is supplied from one or more of the waste heat streams. This hot, high pressure vapor fluid is then expanded through the expander, which is the stage where energy is extracted from the cycle. The shaft work produced through the expander can be used to generate electricity or in the vehicle bottoming cycle case, can also be geared back into the engine shaft output. After the expansion, the vapor will go through the condenser, where it rejects heat to ambient, allowing the vapor to condense to a liquid so that the pump can be used to re-pressurize it.

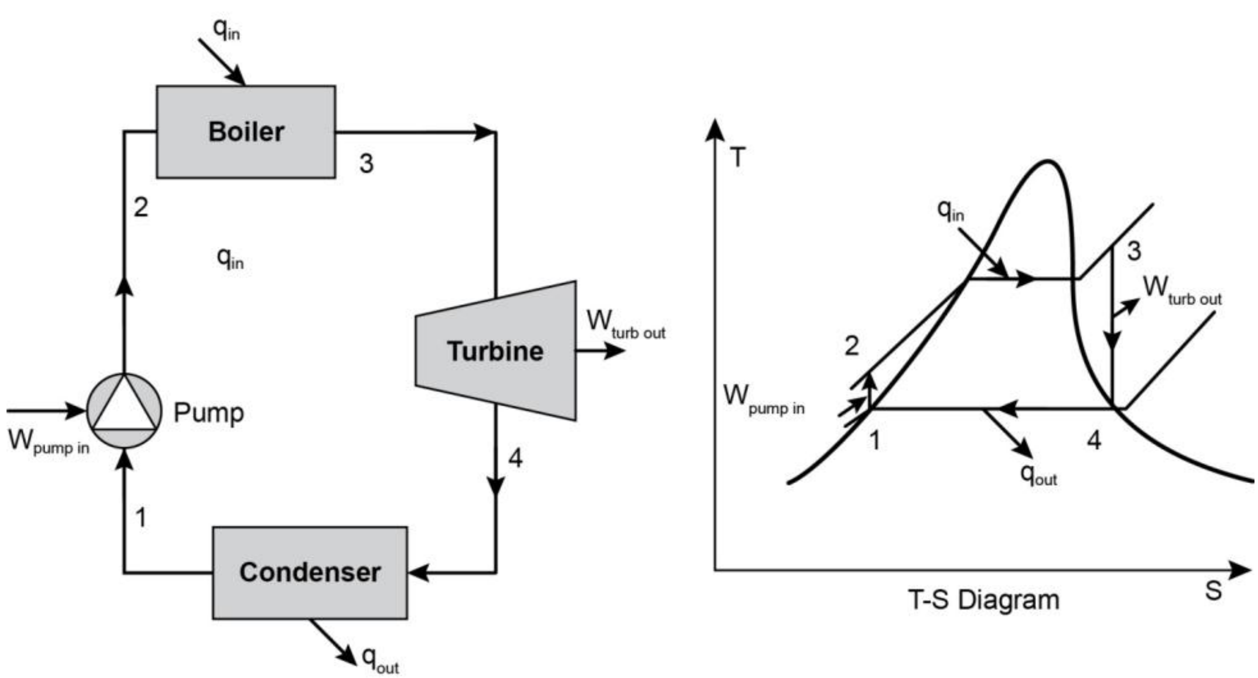


Figure D1. Concept sketch of a Rankine cycle waste heat recovery system.

Because the pump operates on an incompressible liquid, the work to compress the liquid is much less than the work expanded from the hot vapor, providing a positive net work output. The thermal efficiency of the cycle is defined as net work output / heat input. In general terms, this is the most important efficiency-type grading of a thermodynamic cycle. However, in the bottoming cycle, the heat input is low cost. The main "cost" of operation of the Rankine cycle is providing for the heat rejection. In a mobile application, this requires heat exchangers, which tend to increase the frontal area of the vehicle and reduce opportunities for aerodynamic improvement, and/or require higher fan power to assist the heat exchangers in rejecting the heat. Therefore, a more pertinent "efficiency" value for the mobile bottoming cycle might be net work output / rejected heat.

There are many variations upon the basic Rankine cycle, including regenerators. The regenerator applies part of the rejected heat from the condenser to the pump outlet liquid in a pre-heater, which both reduces the heat that must be rejected to ambient as well as reduces the amount of heat input required in the evaporator, for the same cycle conditions.

In our approach to the Rankine bottoming cycle, the following simplifications are made:

1. The pump is 100% efficient and can provide flowrate desirable to the Rankine cycle without regard to engine speed.
    a. A sensitivity study showed that for R245fa, a drop to 50% pump efficiency reduced the net work out of the Rankine cycle by 7-8%. So for example, if with 100% pump efficiency assumption the cycle was putting out 5 kW to add to the engine's 200 kW, with the 50% pump efficiency, it would drop to 4.6 kW. For the water cases, the pump work is a much smaller portion and the 50% efficiency change is minimal, dropping the Rankine cycle output by ≤ 0.5% in all cases.
2. The evaporator heat exchange process is governed by a minimum required temperature difference (ΔT) from one fluid to the other at any point in the heat exchange process. Unless otherwise noted, this ΔT is set to 14°C (25°F).
3. The expander is assumed to have a 70% isentropic efficiency, regardless of flow or pressure ratio. It is not constrained in any way by the speed of the diesel engine. This is consistent with the approach of electrically coupling the expander and the engine.
4. The power coupling between the expander and the engine is electrical, consisting of a generator attached to the expander and an electric motor attached to the engine output where it can add power. The assumed efficiency of each of these devices is 91%, meaning that the composite efficiency of transmission is 91% squared, or 82.8%. This is again, independent of speed or power level, but applied as a percentage of the power to be transmitted.
5. The condenser outlet temperature is specified as an input, according to good engineering judgment of the analyst. Unless otherwise noted, this value is set to 50°C (122°F), which implies that the condenser is ambient-air cooled.
6. The control of the system is such that the condenser outlet condition is maintained at saturation pressure of the fluid at the condenser outlet temperature.
7. The pump is controllable to provide desired outlet pressure, up to and including 35 bar-absolute (507 psia).
8. To maintain reasonable cost and packaging, the condenser heat output is limited. This has the effect of providing full theoretical performance up through roughly 80% engine load, above which the bottoming cycle output is basically fixed due to this limitation. Operating time above this level is minimal for most driving cycles, and to size the system for full engine power output requires a significant increase in cost of the condenser as well as packaging issues and high

parasitic losses for such a large cooling system. Unless otherwise noted, this limit is set at 80 kW.

Two fluids were selected for initial analysis: water and R245fa. Water has thermodynamic property advantages but is subject to freezing if the vehicle is parked in a cold climate. It also optimizes to very low flow rates, which may have an impact on component efficiencies. It is also a benign substance with regards to any leaks that may occur and with regards to servicing. There may be issues with its corrosiveness however.

R245fa is stable and has relatively good thermodynamic properties suitable for use as a bottoming cycle fluid. The main drawbacks to using R245fa are that its usable temperature range is limited to <250°C due to thermal degradation concerns. This reduces the potential cycle efficiency while using this fluid. R245fa also has a high global warming potential (GWP) of around 1000, although alternatives are being developed with lower GWPs which are otherwise very similar in thermodynamic performance to R245fa.

For a bottoming cycle applied to an engine, the potential sources of heat to the bottoming cycle include:

- Engine coolant
- Aftercooler
- Exhaust stack
- EGR cooler.

These are listed in order from the lowest to the highest temperature heat source. After some initial investigation, engine coolant and aftercooler were eliminated as being too low temperature of a source (engine coolant) or not being consistently high enough temperature (aftercooler). The exhaust stack and EGR cooler are considered as heat sources for this study.

Using the EGR cooler as a heat source is dependent on whether the engine is equipped with EGR. Since some of the other engine technologies that are studied include removal of EGR, this is a concern. But if the engine does have EGR, there are several advantages to using the EGR cooler as a heat source:

1. It is the highest temperature heat source, being placed before the temperature reducing effect of the turbocharger turbine
2. The EGR heat exchanger already exists, reducing the marginal cost of adding heat exchangers to support the bottoming cycle
3. The heat that is being brought into the system was already a heat load on the cooling system that had to be rejected

Drawbacks to the EGR cooler are more minor, but include the fact that the bottoming cycle, depending on configuration, may not be able to accommodate all of the EGR cooler's heat load, so that a supplemental EGR cooler may be required in addition to the bottoming cycle's EGR cooler.

The exhaust stack has another set of advantages and disadvantages. The main advantage is that is it the largest heat source on the engine. It is also relatively accessible, although not quite as physically close to the rest of the bottoming cycle as one might desire. Due to the concern of keeping the aftertreatment system as hot as possible during operation, the heat exchanger is placed downstream of the aftertreatment, requiring some longer-than-desired plumbing runs. The other negative to the exhaust stack as a heat source is that heat is brought into the system that used to carry itself away through the tailpipe. Keeping in mind that the bottoming cycle's efficiency is on the order of 10-20% means that roughly 80-90% of the heat brought in via the exhaust stack heat exchanger will now have to be rejected in the condenser, which is problematic as discussed earlier.

Both the EGR cooler and the exhaust stack heat exchanger must operate in hot, corrosive environments, and as a result must be made of high quality, premium materials.

The bottoming cycle described thus far is considered the *simple* bottoming cycle. Various improvements and embellishments are available including a recuperator, which is examined in this study. The recuperator constitutes a heat exchanger placed between the turbine outlet vapor and the pump outlet liquid. This allows recovery of a portion of the heat that would otherwise be rejected in the condenser by the pump outlet water. This both increases the thermal efficiency of the bottoming cycle and reduces the heat rejection.

The base configuration studied for this report was a simple cycle with the heat addition in two pieces: first from an exhaust stack heat exchanger (pre-heater) and then from the EGR cooler (superheater). Alternatives include a configuration with no EGR cooler and a configuration with a recuperator.

Shown in Figure D2 below is a schematic of the engine/bottoming cycle system for the base configuration studied. The numbers refer to bottoming cycle stations. Table 1 provides temperatures and pressures at those stations, as well as some of the performance values for one engine operating point with the base configuration with water as the bottoming cycle fluid for reference. This point is over the 80 kW condenser heat rejection limit that is utilized for most of the study, but is provided as one of the higher output, higher temperature and pressure points studied.

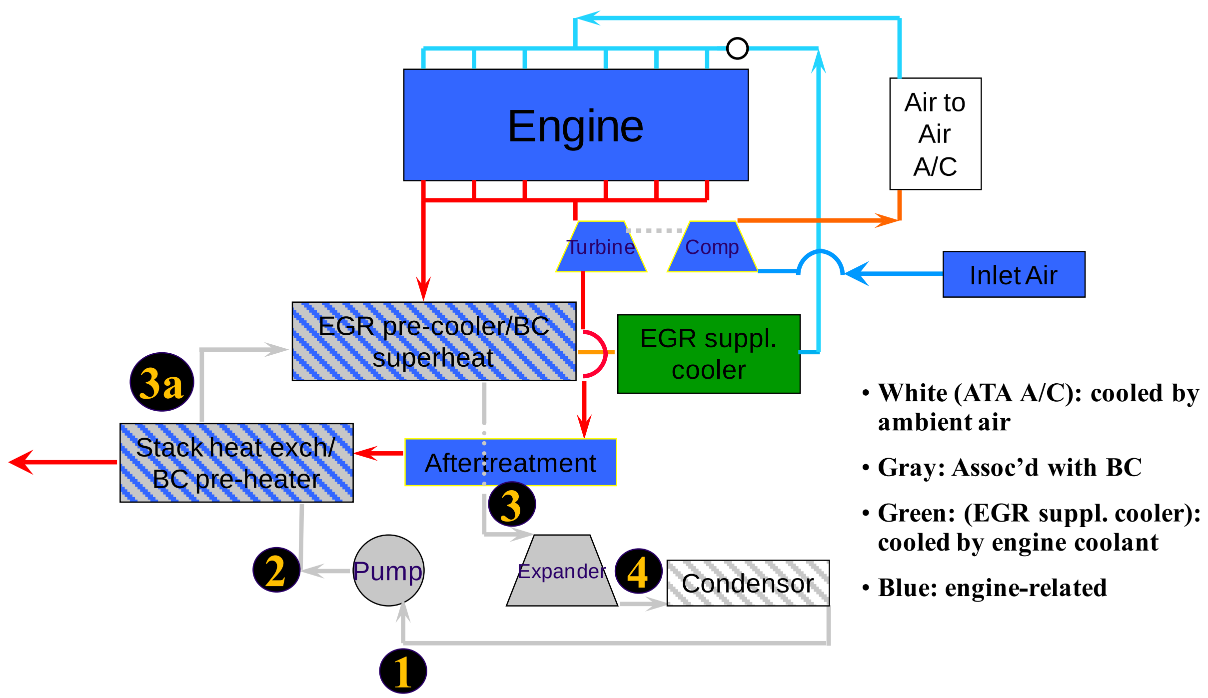


Figure D2: Engine/Bottoming Cycle Schematic

In Table 1 below, T1 and P1 represent the pump inlet conditions, while T2 and P2 are the pump outlet conditions, assuming a 100% pump efficiency. T3a and P3a are the outlet conditions of the exhaust stack heat exchanger, which serves as the pre-heater for the bottoming cycle circuit. T3 and P3 are the outlet conditions of the EGR heat exchanger, which serves as the superheater for the bottoming cycle. T3 and P3 also represent the inlet conditions for the expander, which extracts work from the bottoming cycle fluid. T4 and P4 are the outlet conditions of the expander, after work has been extracted.

The next column shows the condenser heat rejection. In this particular example, the value is shown in red, because it exceeds the self-imposed limit of 80 kW. This problem is fixed by either reducing the fluid mass flow in the bottoming cycle system, or by limiting the amount of heat input. The most effective approach to meeting the heat rejection requirement is a function of the fluid type. The next column shows the total heat input from the two heat exchangers. The following three columns provide the thermal efficiency of the bottoming cycle, the percentage of input heat that has to be rejected by the condenser, and the ratio of bottoming cycle power to engine power.

The bottoming cycle is assumed to drive an electric generator, and then an electric motor geared to the engine crankshaft, so the final two columns show the net electric power that is provided to the engine crankshaft, and the ratio of electric power input to engine power. This final column represents the overall benefit of the bottoming cycle at a given operating point.

Table 1: Bottoming Cycle Example Operating Point

T1	P1	T2	P2	T3a	P3a	T3	P3	T4	P4	**Cond Heat Rej**	Net Work	Gross Heat Input	Eff	% heat input rejected	Wbc / Wengine	Net Elec Work	Wbc_elec/ Wengine
°C	MPa	°C	MPa	°C	MPa	°C	MPa	°C	MPa	kW	kW	kW	%	%	%	kW	%
50	0.012	50.1	**3.47**	242.1	3.47	528	3.46	126.9	0.022	**95.6**	29.27	125	23.4	77	9.1	24.24	7.5

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