

Diesel Engine Fuel Economy Improvement Enabled by Supercharging and Downspeeding

Sean Keidel, Philip Wetzel and Brandon Biller
Eaton Corp

Karen Bevan
Eaton Innovation Center

Aaron Birckett
FEV

ABSTRACT

In order to improve power density, the majority of diesel engines have intake manifold pressures above atmospheric conditions. This allows for the introduction of more fuel, which results in more power. Except for a few applications, these engines receive charged air from a turbocharger. The turbocharger develops boost by converting exhaust gas energy into power. This power is then used to compress the intake charge.

The medium and heavy-duty engine markets have both stringent regulatory targets and customer demand for improved fuel efficiency. Two approaches used to meet fuel efficiency targets are downspeeding and downsizing. Until now, the industry has adapted to the turbocharger lag experienced during a transient acceleration event. This performance deficiency is severely exaggerated when the displacement and speed of an engine are reduced. The solution proposed to improving fuel economy, while maintaining equivalent performance, is supercharging.

Eaton's TVS® supercharger is a positive displacement pump with significantly improved efficiencies over prior models. Unlike a turbocharger, which relies on exhaust gas energy to create boost, a supercharger is directly coupled to the engine. Therefore, the supercharger speed increases proportionally with engine speed. This greatly reduces the lag that is present in a turbocharged engine. Addition of a supercharger, as this paper will show, dramatically improves low speed engine performance. This enables greater engine downspeeding that was previously not feasible with a turbocharged engine.

This paper explores the fuel economy benefits of combining a supercharger with a turbocharger on a diesel engine. The fuel economy and performance improvements enabled by supercharging are dependent on application, drive cycle, and engine displacement. It also reviews the simulation results of three distinct engine sizes and applications exercised over vehicle appropriate drive cycles. The effect of boosting system configuration, downspeeding, exhaust gas recirculation (EGR) configuration, and supercharger clutching on fuel economy were examined. All of the simulations provided a fuel economy benefit over the baseline turbocharger-only engines. Fuel economy was improved, while maintaining or improving vehicle performance over the baseline. Efficiency improvements in excess of 14% were demonstrated over a transient drive cycle.

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INTRODUCTION

Medium-duty and heavy-duty diesel engines are almost exclusively turbocharged. The primary advantage of using a turbocharger is its ability to harness waste heat from the engine. This advantage brings with it the disadvantage of increased backpressure on the engine and lower quality heat

for the aftertreatment devices. The turbocharged engine also has a significant performance disadvantage as well. Since the turbocharger relies on exhaust gas energy, it struggles to produce adequate levels of boost at lower engine speeds. A transient load step, especially when starting from a relatively

low engine speed, can result in a delayed response from the engine, a phenomenon called “lag.”

The goal of this analytical study was to determine how an Eaton TVS® supercharger could improve the fuel economy of medium-duty and heavy-duty vehicles in real-world driving cycles. The scope of work investigated the effect of boosting system configuration, downspeeding, exhaust gas recirculation (EGR) configuration, and supercharger clutch on fuel economy. A market study was conducted for the medium- and heavy-duty North American vehicle markets to determine three representative applications. The applications selected for study were as follows:

1. Class 8 linehaul
 - a. 15 liter engine
 - b. 500 horsepower (373 kW)
 - c. 80,000 lb. (36,287 kg) tractor trailer
 - d. 13 speed automated manual transmission
 - e. 3.55 final drive ratio for the baseline; 3.24 final drive ratio for the supercharged vehicle
 - f. Variable-geometry turbocharger (VGT) with high-pressure loop EGR
 - g. Diesel Oxidation Catalyst (DOC) / Diesel Particulate Filter (DPF) / Selective Catalytic Reduction (SCR) aftertreatment
2. Class 8 vocational
 - a. 13 liter engine
 - b. 425 horsepower (317 kW)
 - c. 57,000 lb. (25,855 kg) dump truck
 - d. 10 speed automated manual transmission
 - e. 4.30 final drive ratio for all vehicles
 - f. VGT with high-pressure loop EGR
 - g. DOC/DPF/SCR aftertreatment
3. Class 6 pickup and delivery
 - a. 6.7 liter
 - b. 230 horsepower (172 kW)
 - c. 26,000 lb. (11,793 kg) box truck
 - d. 6 speed transmission
 - e. 4.11 final drive ratio for all vehicles
 - f. Twin-sequential turbochargers with high-pressure loop EGR
 - g. DOC/DPF aftertreatment

Description of Drive Cycles

Two drive cycles were selected for the linehaul application simulations. The first cycle, shown below in [Figure 1](#), is Eaton Linehaul Route 1. This drive cycle is representative of a vehicle accelerating to a highway cruise condition and trying to maintain speed. The grade percentage varies over the cycle to simulate a real-world situation. The data was derived from an experimentally-recorded drive cycle.

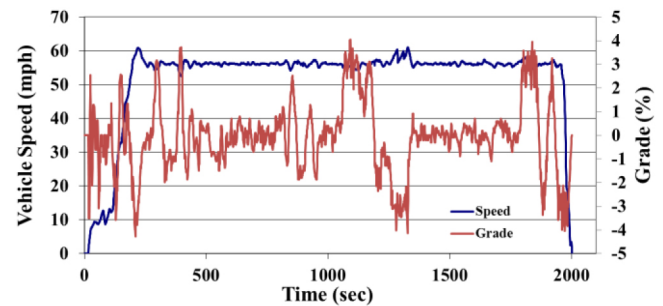


Figure 1. Eaton Linehaul Route 1 drive cycle vehicle speed and road grade profile.

The other drive cycle that was used for the linehaul analysis was HHDDT Cruise, shown below in [Figure 2](#) [1]. This drive cycle maintains a constant grade of zero percent for the entire cycle. The vehicle accelerates from a stop to highway speeds. The targeted vehicle speed then dithers about 55 miles per hour.

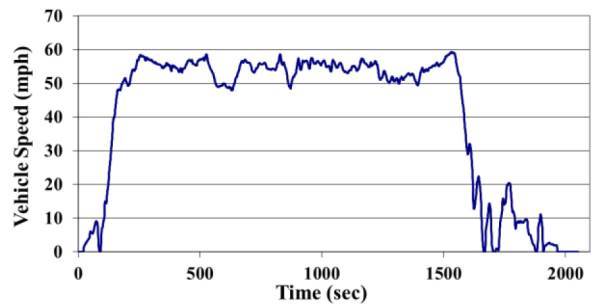


Figure 2. Heavy Heavy-Duty Diesel Truck (HHDDT) Cruise

Two different drive cycles were selected as representative for the vocation application analysis. [Figure 3](#) shows the HHDDT transient drive cycle [1]. This drive cycle is highly transient with several starts and then stops. This cycle is simulated at zero percent grade. The second drive cycle for the vocation application is Urban Dynamometer Driver Schedule (UDDS) [2], shown below in [Figure 4](#). This cycle also is a transient drive cycle. However, the UDDS has more aggressive acceleration events than the HHDDT transient drive cycle. It is also simulated with a constant zero percent grade.

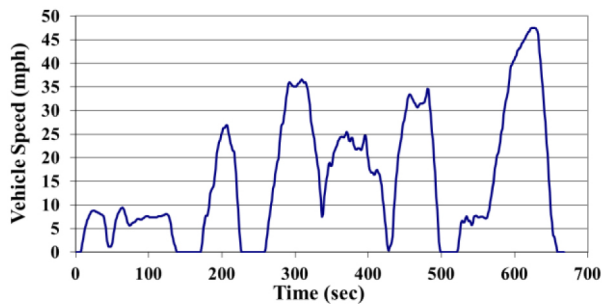


Figure 3. Heavy Heavy-Duty Diesel Truck (HHDDT) Transient

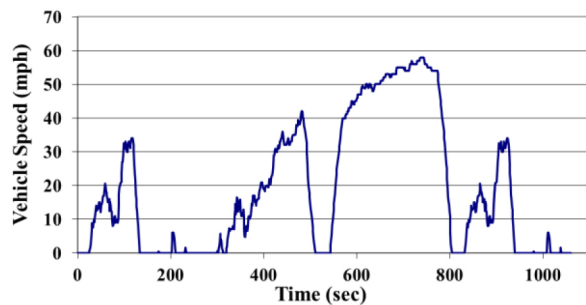


Figure 4. Urban Dynamometer Driving Schedule (UDDS)

The Class 6 application was exercised over the FTP 72 drive cycle, shown in [Figure 5](#) [3]. This cycle is a very transient drive cycle with numerous starts and stops. This cycle was chosen to represent a pickup and delivery application. Simulation over this drive cycle was run at zero percent grade.

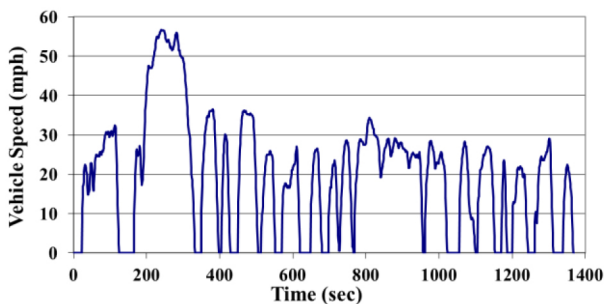


Figure 5. Federal Test Procedure (FTP) 72

Boosting System Configuration

Current turbocharger-only boosting systems that are being envisioned for medium-duty and heavy-duty vehicles are challenged by the need to balance engine performance with fuel efficiency. With turbochargers, the exhaust enthalpy content can be too low at low load conditions to drive the turbocharger. Under these conditions, when the accelerator pedal is depressed the vehicle will respond slowly due to turbocharger “lag” as the turbocharger accelerates up to

speed. A common approach to improve transient performance at low engine speeds is to use a smaller turbocharger turbine with less inertia. The improved transient performance, however, comes at the expense of poorer fuel efficiency at high speeds since the smaller turbocharger turbine causes higher exhaust restriction. Superchargers, in contrast, use energy from the engine crankshaft. As a result, superchargers and specifically roots type superchargers such as Eaton's TVS® Superchargers, can provide air-on-demand for effective air-fuel ratio control. The ability to maintain high air-fuel ratios during transients leads to superior transient performance, since more fuel can be injected to produce more power with reduced smoke production. These advantages from the supercharger, however, typically come at the cost of slightly higher fuel consumption versus a comparably-sized turbocharger due to the mechanical losses of the supercharger and its inability to recover waste heat energy.

Two potential boosting system configurations were candidates for this analysis, Super-Turbo and Turbo-Super. The two system configurations are shown in [Figure 6](#). The Super-Turbo configuration uses the supercharger as the low-pressure boosting device and the turbocharger as the high-pressure boosting device. Alternatively, the Turbo-Super [4] uses the turbocharger as the low pressure device and the supercharger as the high pressure device. Super-Turbo and Turbo-Super boosting system configurations can harness the inherent complementary benefits of supercharging and turbocharging to provide high fuel efficiency with high transient engine performance. For example, the turbocharger can be sized to meet the full boost requirements at higher engine speeds and loads. Further, the turbocharger design can be optimized in a manner that cannot be achieved in a turbocharger-only system. The turbocharger can be sized appropriately for maximum efficiency at the desired operating points without having to compromise for transient response. The supercharger is then selected to supplement conditions when the turbocharger cannot provide sufficient boost, such as at low speeds and during transient events. For these simulations, the turbocharger was sized to meet all steady-state lug curve points. Since the supercharger was able to nearly eliminate the lag and aid in driving EGR, the turbocharger could be sized for high efficiency operation. The analysis demonstrates the ability to use a FGT, instead of a VGT, and still meet both performance and EGR requirements, while downspeeding for peak fuel economy. [Figure 7](#) shows the compressor map that was used for the baseline VGT engine. [Figure 8](#) is the turbocharger compressor map that was used in the Turbo-Super configuration. Finally, [Figure 9](#) is the efficiency map for the supercharger used in the Turbo-Super configuration. All of these maps were for the 13-liter vocational application.

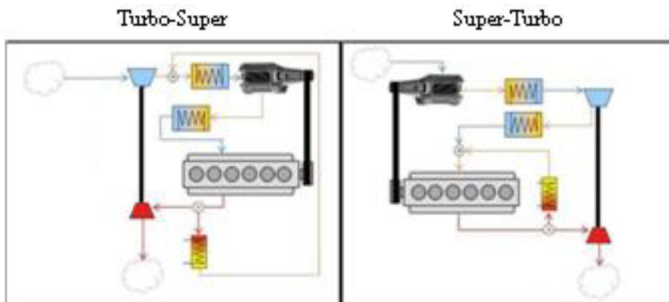


Figure 6. Supercharged Boosting System Configurations

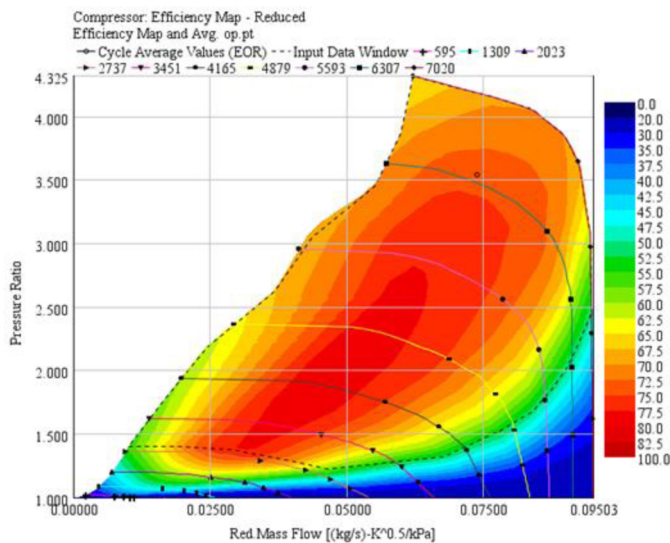


Figure 7. Variable Geometry Turbocharger Compressor Efficiency Map

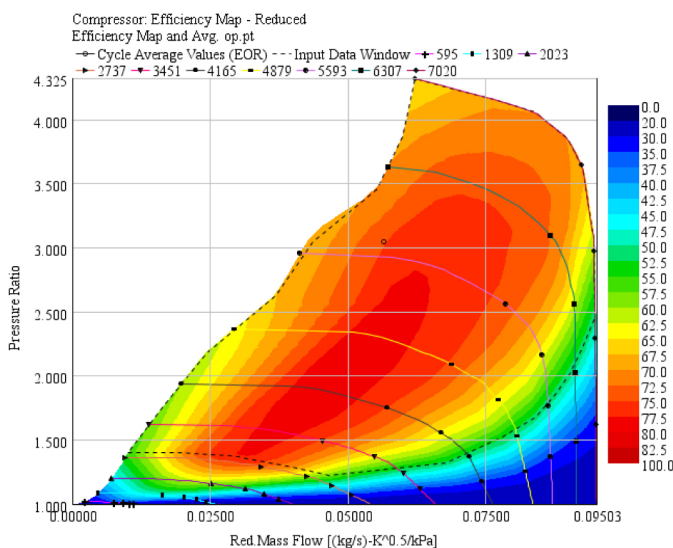


Figure 8. Fixed Geometry Turbocharger Compressor Efficiency Map

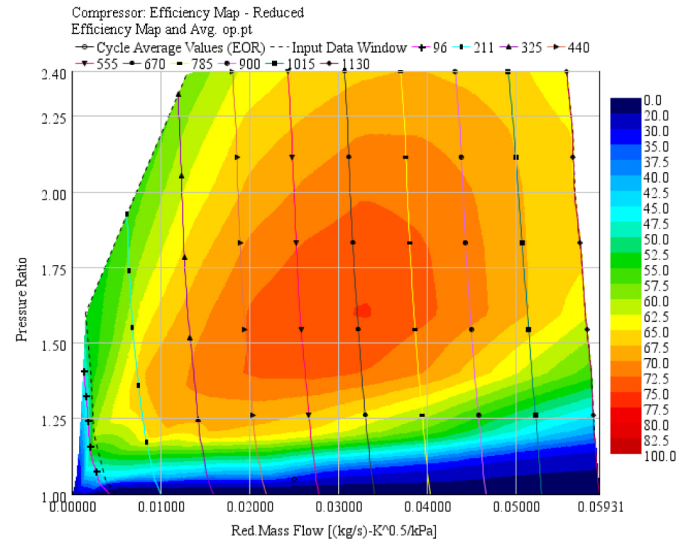


Figure 9. Supercharger Efficiency Map

The advantages of the Super-Turbo configuration over the Turbo-Super configuration are the ability to eliminate interstage cooling, require less input power for a comparable pressure ratio from the supercharger, and slightly improve transient response. The Turbo-Super configuration has the advantages of improved fuel economy, smaller supercharger size, ability to have mid pressure loop EGR (shown in [Figure 10](#)), and potential acoustic benefits since the turbocharger decrease supercharger inlet air pulsations. Previous work has demonstrated that the Turbo-Super system provides the greatest potential increase in fuel economy, as compared to a turbocharged baseline engine [16, 17, 18].

EGR System Configuration

Three EGR system configurations were evaluated as part of this study: high pressure loop (HPL), mid pressure loop (MPL), and low pressure loop (LPL). The system layouts and a brief description of each EGR configuration can be found in [Figure 10](#). The HPL EGR route transfers exhaust gas from the exhaust manifold to the intake manifold. The MPL EGR route transfers exhaust gas from the exhaust manifold to the intake system between the low pressure stage turbocharger compressor and the high pressure stage supercharger. The LPL EGR route transfers exhaust gas from a post DPF source to a pre low pressure stage in the intake system. The HPL and LPL EGR configurations can be used with either Super-Turbo or Turbo-Super boosting configurations. The MPL EGR configuration is recommended for use with the Turbo-Super boosting configuration only since the supercharger is tolerant of unfiltered cooled EGR, whereas the turbocharger compressor is typically not tolerant [4, 5, 6, 7, 8]. This configuration can eliminate the need to run a negative pressure gradient across the engine to flow EGR, thus reducing engine pumping work (PMEP) [15] and decreasing the backpressure demands of the turbocharger's turbine wheel. This can allow for lower backpressure designs or even