

A High Efficiency, Dilute Gasoline Engine for the Heavy-Duty Market

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ABSTRACT

A 13 L HD diesel engine was converted to run as a flame propagation engine using the HEDGE™ Dual-Fuel concept. This concept consists of pre-mixed gasoline ignited by a small amount of diesel fuel - i.e. a diesel micropilot. Due to the large bore size and relatively high compression ratio for a pre-mixed combustion engine, high levels of cooled EGR were used to suppress knock and reduce the engine-out emissions of the oxides of nitrogen and particulates. Previous work had indicated that the boosting of high dilution engines challenges most modern turbocharging systems, so phase I of the project consisted of extensive simulation efforts to identify an EGR configuration that would allow for high levels of EGR flow along the lug curve while minimizing pumping losses and combustion instabilities from excessive backpressure. A potential solution that provided adequate BTE potential was consisted of dual loop EGR systems to simultaneously flow high pressure and low pressure loop EGR. The use of this blended EGR concept allows the high EGR engine to fit on the compressor map, although with high levels of pumping work compared to the baseline diesel.

The combustion experiments were carried out at 17:1 and 14.5:1 CR with gasoline levels varying from 60% to 90% of the total fuel. The results at 17:1 CR showed that, even with 50% EGR, the load was limited by knock and combustion noise. At 14.5:1 CR, the load was again limited by combustion noise and knock, but was much closer to the baseline engine with acceptable combustion noise and low engine-out emissions. The fuel economy potential of the engine was shown to be high, with results of 195 g/kWh total (gasoline + diesel) at the A50 condition, while the high levels of EGR resulted in NOx and PM below the 2010 HD standard of 0.27 g/kWh NOx and 0.1 g/kWh PM respectively. A comparison between the two different gasoline-diesel dual fuel operations - i.e. diesel micro-pilot operation and PPC-style operation shows the tradeoffs between the two combustion modes and challenges of each.

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INTRODUCTION

Diesel engine emissions regulations in the U.S. have become exponentially more stringent since the year 2002 and the most stringent ever U.S. On-Road Heavy Duty regulation went into full effect with the 2010MY. For off-road engines, the implementation for the new Tier IV Final regulations is 2014. Most members of the transportation community only expect emissions restrictions to increase over the coming years. To meet the ever stringent emissions regulations, the diesel engine industry has developed and applied new technologies such as diesel particulate filter (DPF) for PM

reduction, and selective catalytic reduction (SCR) devices or lean NOx traps (LNT) for NOx control along with exhaust gas recirculation (EGR) technology. Although the diesel industry has successfully employed the technology to meet the U.S. 2010 on-road heavy duty regulations, the cost increase due to the very complex aftertreatment system is high and it requires more expensive engine components, i.e. turbochargers, injection systems, power cylinders, etc, to maintain or improve the engine efficiency. Due to the increasing complexity of the system, the risk and warranty cost increase associated with diesel aftertreatment system are

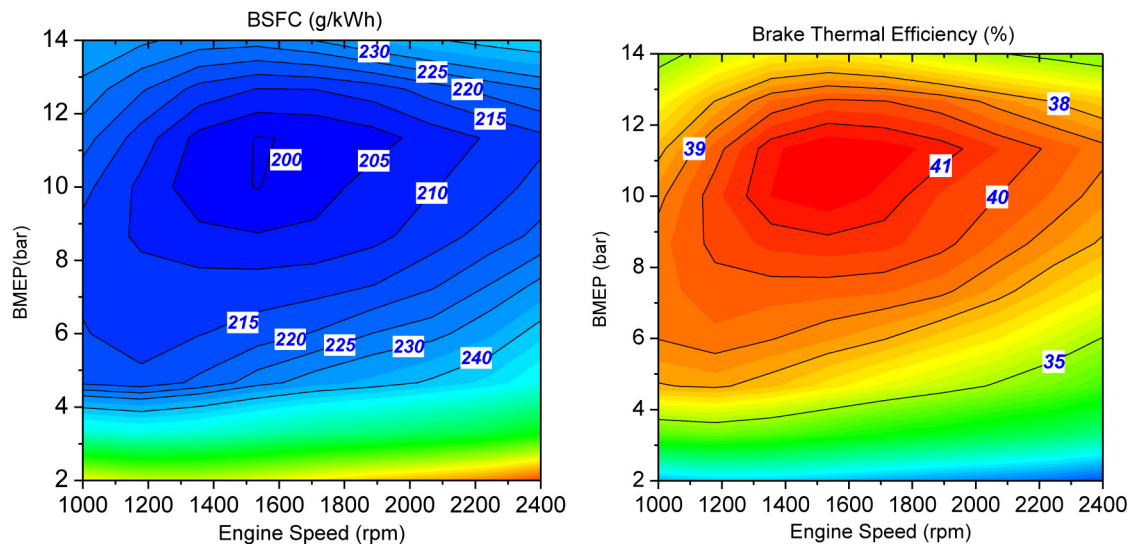


Figure 1. BSFC and BTE for the Dual-Fuel 4.5 L Engine

very high also. Because of the cost increase and negative impact on the engine's fuel efficiency associated with the complex aftertreatment systems, the diesel industry, especially heavy-duty diesel engine OEMs are enthusiastically seeking alternative technologies that provide high fuel economy and ultra low emissions with minimal cost increases.

Southwest Research Institute (SwRI) has actively investigated alternative solutions for the heavy duty industry through internal research programs and the High Efficiency Dilute Gasoline Engine (HEDGE™) Consortium. In these programs, SwRI has progressively demonstrated high fuel efficiency and low engine-out emissions using the dual fuel gasoline-plus-diesel approach. The first concept proof study was conducted on a single-cylinder engine [1]. The study showed the difference between using a traditional, relatively low energy ignition system and a high-energy diesel micro-pilot ignition system in a heavy-duty engine running on gasoline. The micro-pilot diesel injection ignition system led to much higher loads and efficiencies and a significant reduction in BSNOx. The main cause behind the improvements was an increase in dilution tolerance, from a peak of 30% with the spark ignition system to 40% with micro-pilot ignition. The potential indicated by this single cylinder study led SwRI to establish a second Internal Research Program in the heavy-duty gasoline engine area to investigate a multi-cylinder engine [2]. The goal of the study was to further investigate the potential for gasoline fuelled engines in the medium to heavy-duty market and to look at the effect of CR, fuel type and EGR delivery method on performance. The engine was tested in several different configurations: HPL (High Pressure Loop) EGR at 17.5:1 and 12.5:1 CR (Compression Ratio), and LPL (Low Pressure Loop) EGR and HPL EGR with external boost at 12.5:1 CR. Two different gasoline fuels were tested, 91 ON (Octane Number) and 103 ON ((R+M)/2) at several different speed

and load points. The study proved that the HEDGE concept has the potential to achieve nearly the same fuel economy performance as a diesel engine equipped to achieve MY2002/2004 emissions levels with ultra low tail pipe emissions and the potential to use simple aftertreatment.

The concept was investigated further in the HEDGE™ Consortium using a 4.5L, 4 cylinder engine platform. Results from that engine with a 106 mm bore are shown in Figure 1. The engine was configured with multi port gasoline injectors, stock direct diesel injectors, low pressure loop cooled EGR and squish enhanced 14.3:1 compression ratio bath tub style pistons. This engine achieved a minimum fuel consumption of 200 g/kWh, which equates to 42% brake thermal efficiency (BTE). For reference, the fuel consumption from the 2009 production version of this 4.5 L diesel engine was 214 g/kWh, or 39% BTE, at the same speed-load operating point. Typically, comparisons of fuel economy are only relevant when the engines are calibrated to equivalent tailpipe emissions, as fuel economy can almost always be improved if the engine is calibrated for higher emissions. However, in this case, it was not possible to compare the dual fuel engine and the production diesel engine at the same emissions because the dual fuel approach intrinsically had significantly lower engine-out and tailpipe emissions. Figure 2 shows that the engine-out NOx and PM levels were low enough to meet the US 2010 heavy duty emission standards without NOx or PM aftertreatment. The production diesel engine, on the other hand, was calibrated to approximately 10 times higher NOx emissions. Even with this seemingly unfair emissions comparison, the dual fuel engine still showed 7% lower fuel consumption. As such, the operating cost (i.e., fuel cost) of the dual fuel engine should be lower than the diesel engine. In addition, the lack of NOx and PM aftertreatment for the dual fuel engine should also result in a lower initial cost and reduced package size. As with any gasoline engine, the dual fuel engine will require a low-cost oxidation catalyst to clean

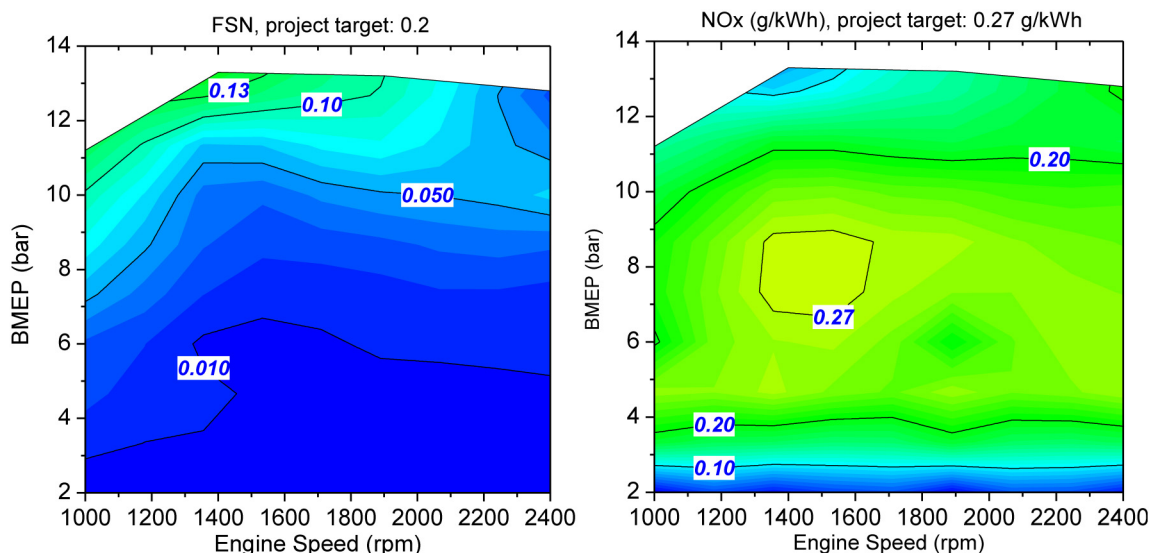


Figure 2. Emissions Results for the Dual-Fuel 4.5 L Engine

up the HC and CO emissions generated from premixed combustion. Although it increases cost associated with the gasoline fuel system i.e., gasoline injectors, fuel delivery and storage tank, significant overall cost reduction should be possible.

The same technology is applied to a larger bore engine in this study. The purpose is to investigate the feasibility of dual fuel combustion for a heavy duty application. The main issue that arises when applying the dual fuel approach to the larger bore sizes found in heavy duty applications is engine knock. The propensity for knock scales with the distance the flame has to travel, which is directly related to bore size. Bore size in these applications range from 125 to 170 mm, depending on engine displacement.

The main targets for this study are as below.

- Minimum fuel consumption of less than 200 g/kW-h BSFC
- NOx emissions lower than 0.27 g/kW-h BSNOx
- Smoke emissions less than 0.2 FSN
- Identify the maximum power density in dual fuel combustion mode

The potential of the dual fuel combustion has been recognized by other researchers. For example, Reitz et al studied the concept by simulation and recognized the potential of controlling the onset of combustion by blending two fuels that have different reactivity in kinetically controlled compression ignition combustion [3][4]. The follow on experimental studies proved the concept that combustion could be phased by blending diesel and gasoline in-cylinder [5][6][7].

Boosting is a challenge in a high efficiency dilute engine because of the low exhaust temperature and high intake manifold pressure requirement associated with high levels of dilution either by EGR or excess air [8]. A 13L, 6 cylinder

heavy duty engine with two stage turbocharging that utilized high levels of EGR to control NOx emission was chosen as a test platform considering the potential for boosting challenges. A significant amount of one dimensional simulation was carried out before the engine tests to examine the boosting and EGR architecture options.

1-D SIMULATION FOR EGR ARCHITECTURE

A baseline diesel engine model (GT-Power) was provided by the engine manufacturer. After the validation of the base model with standard diesel combustion, the following modifications have been made to convert the base model to a HEDGE dual fuel combustion model.

- Reduced compression ratio to 14.5:1. The compression ratio was selected based on SwRI's experience of dual fuel combustion in 4.5L 4 cylinder medium duty HEDGE engine.
- Apply cumulative heat release rate taken from the 4.5L 4 cylinder HEDGE engine data across all speeds
- Modify start of combustion (SOC) to realize approximately 10 deg aTDC CA50 location
- Add central gasoline injection (Single Point Injection)

The first set of simulations determined if the stock HPL EGR system is able to provide sufficient flow at full load conditions. The required EGR flow at full load was estimated at 50% based on the previous experience with the 4.5L 4 cylinder engine. The results showed that the stock HP loop EGR system was not able to provide the required EGR flow. The model estimated the maximum EGR flow was 45% at the rated speed of 1900 rpm and 41% at 1200 rpm. Also, the predicted BSFC as shown in Figure 3 with the stock HPL

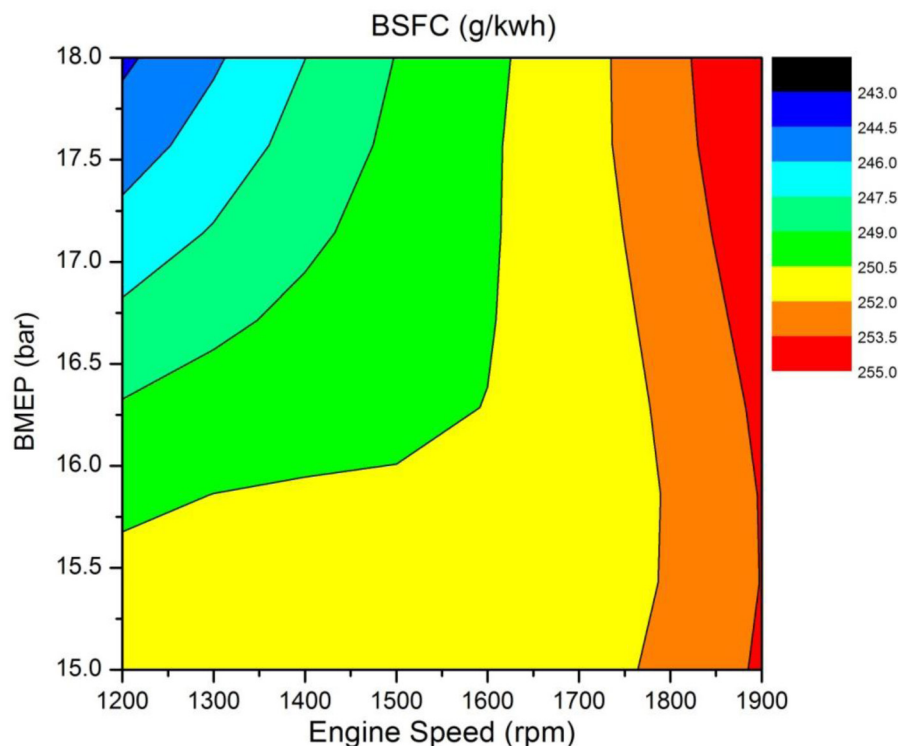


Figure 3. Predicted Brake Specific Fuel Consumption with a HPL EGR Configuration. GT-Power Results

EGR was substantially higher than modern diesel engines because of the excessively high pre-turbine pressure.

Knowing that the HPL EGR could not provide sufficient EGR at high load conditions, a LPL EGR system was added to the model. The idea was that the stock engine was designed to provide high levels of EGR along with a lean A/F ratio typical of diesel combustion, therefore LPL EGR (post low pressure turbocharger turbine to pre low pressure turbocharger compressor) was installed to replace the excess air which should increase the total EGR flow.

Introducing LPL EGR in addition to the HPL EGR yielded the required level of EGR. [Figure 4](#) shows the effect of HPL EGR ratio (HPL EGR / Total EGR, %) to BSFC in this blended EGR concept. The total EGR in this simulation was controlled to 45% up to 8 bar BMEP and to 50% above 8 bar BMEP. The engine torque was controlled by the HP turbocharger waste gate at high load, or by the HP intake throttle at low load. The LPL EGR valve position was swept while the HPL EGR valve position was controlled to meet the target EGR level (50% at max load, 45% at light load). At the given engine speed, load, target EGR, and LPL EGR valve position, the LPL EGR level was determined by valve position and pressure differences while HPL EGR controller adjusted its valve position to match the overall EGR target. In [Figure 4](#), as we observed in the previous simulations, HPL EGR could only provide a limited level of EGR. More importantly, the BSFC consistently decreases with increasing HPL EGR regardless of the engine speed and load. The reason is attributed to the

reduction of PMEP with increasing HPL EGR as it bleed off excessive pre-turbine pressure. The decrease of PMEP with increasing HPL EGR is shown in [Figure 5](#).

From 2 bar BMEP to 6 bar BMEP where the engine was throttled to maintain the targeted BMEP and equivalence ratio of 0.9, the PMEP was rapidly improved with increasing BMEP due to the less throttling as expected. From 8 bar BMEP to 18 bar BMEP when the engine is boosted, PMEP gets worse with increasing BMEP due to high pre-turbine pressure especially when the HPL EGR was low. As we bled the high pre-turbine pressure by increasing HPL EGR, the PMEP was improved rapidly. As a conclusion, it was found that utilizing as much HPL EGR as possible while providing sufficient boost for a given load was optimal for the best BSFC.

[Figure 6](#) and [Figure 7](#) show the final simulation results of the blended EGR strategy. The blended EGR concept reduced the predicted BSFC by ~ 50 g/kWh at high load compared to the HPL EGR only configuration and satisfies the fuel consumption target of below 200 g/kWh BSFC. The best BSFC estimate of 188 g/kWh took place at 1200 rpm 18 bar BMEP. The promising BSFC was predicted with 45% total EGR below 8 bar BMEP and 50% total EGR above 8 bar BMEP and the HPL EGR to total EGR ratio was roughly 40% ~ 45% as shown in [Figure 7](#). Because of the success of the blended EGR strategy, it was implemented in the engine tests.

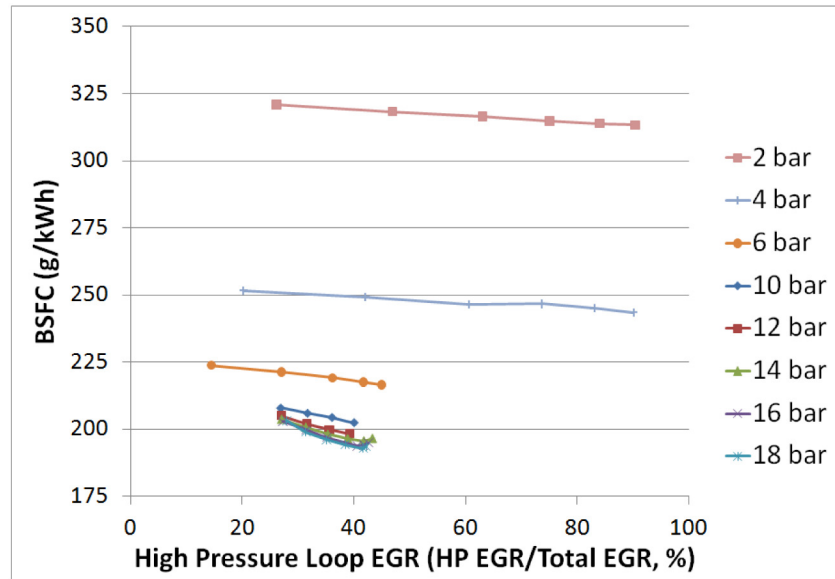


Figure 4. The Effect of HPL EGR % to BSFC at 1500 rpm with Various BMEPs (bar). GT-Power Simulation Results

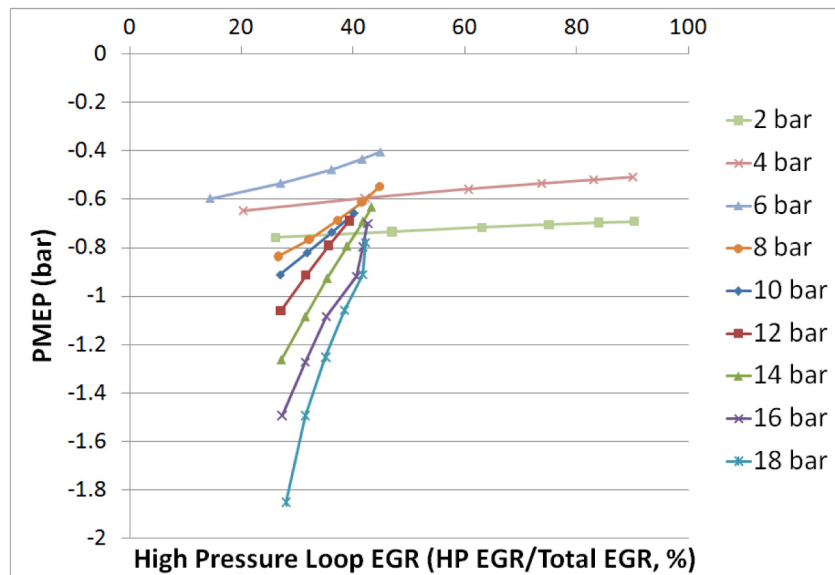


Figure 5. PMEP at 1500 rpm with Varying HPL EGR % at Various BMEP (bar). GT-Power Simulation Results

EXPERIMENTAL SETUP

A Navistar Maxxforce 13 production engine (2009MY), as shown in [Figure 8](#), was used for this study. Detailed specifications are shown in [Table 1](#). The engine was instrumented with a standard complement of pressure, temperature, and flow measurement equipment. The instrumentation includes 6 cylinder pressure sensors, MicroMotion flow meters for each fuel (gasoline and diesel

for fuel consumption calculations), RTD temperature sensors on the coolant side of the coolers for heat rejection measurements, a coolant flow meter, EGR cooler pressure restriction sensors (delta pressure) and standard gas pressure sensors and temperature sensors at the key locations on intake and exhaust system. An AVL smoke meter was used for FSN measurement.

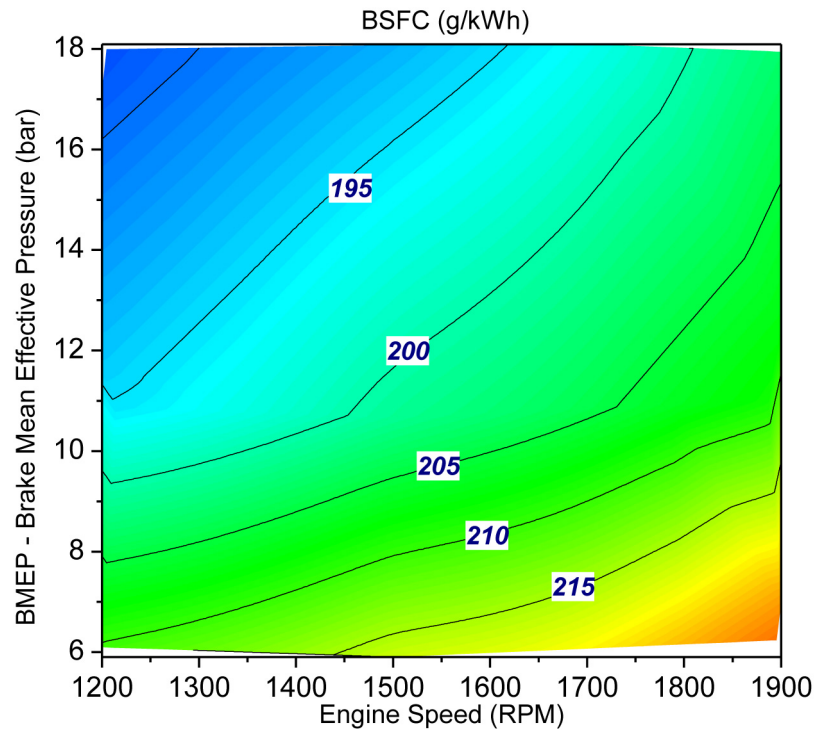


Figure 6. Best BSFC Estimate. GT-Power Simulation Results

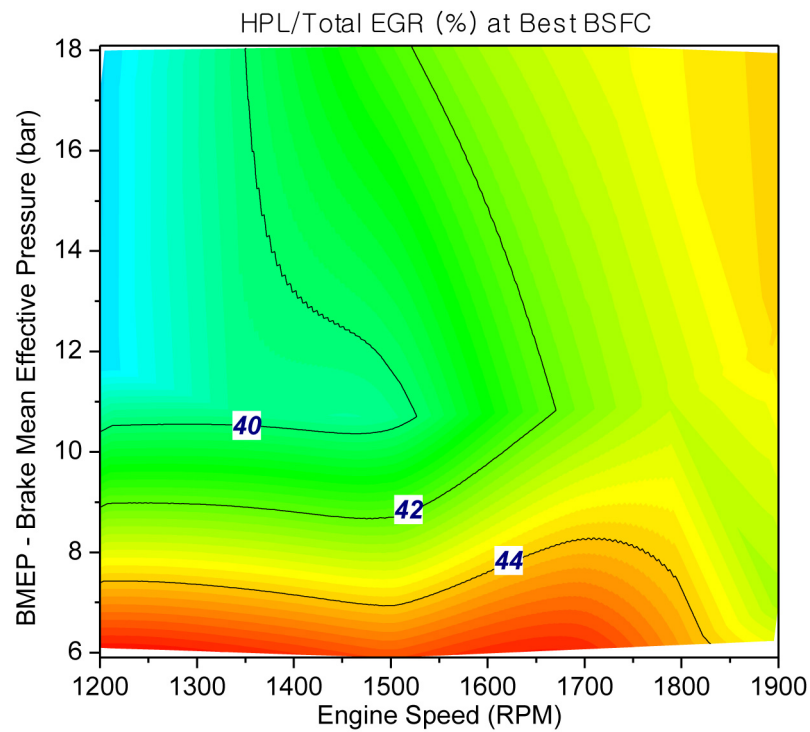


Figure 7. HPL EGR Ratio (HPL/Total EGR) at the best BSFC. GT-Power Simulation Results



Figure 8. 2009 Navistar Maxxforce 13

Table 1. Base engine Specifications

Engine Type	Direct Injection Diesel, 4-cycle
Configuration	Inline 6-cylinder
Displacement	12.4 L
Bore and Stroke	126 mm x 166 mm
Compression Ratio	17:1
Min. Fuel Consumption	205 g/k Wh
Turbocharger	Sequential 2-stage turbochargers (high pressure turbocharger is wastegated)
EGR	2 separate cooled EGR loops (one for front 3 cylinders and one for rear 3 cylinders)

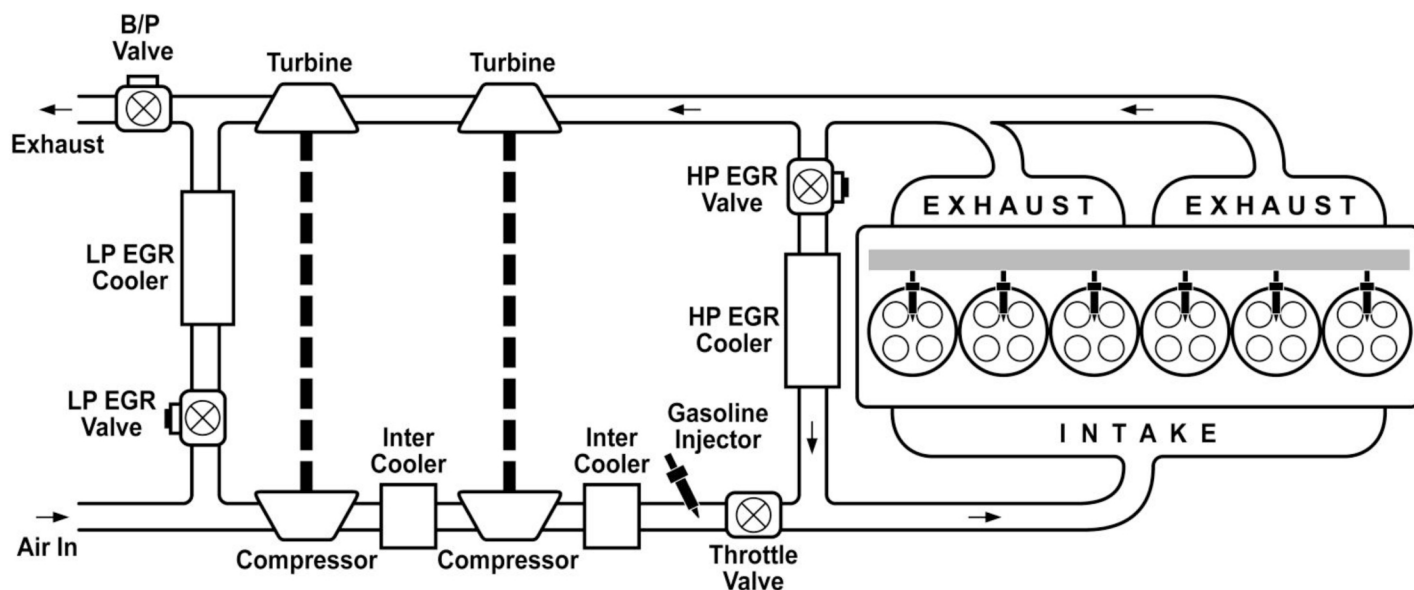


Figure 9. Schematics of modified engine for dual fuel combustion

Engine Modifications

The main modifications required for dual fuel combustion are listed below.

- Addition of low pressure loop EGR (LPL EGR) to utilize the blended EGR concept
- Central gasoline injection system
- Lower compression ratio pistons (17:1 to 14.5:1, squish area maintained, bath tub style bowl)

The stock engine has high pressure loop EGR which routes exhaust gas from the exhaust manifold to just before

the intake manifold. To supplement this EGR as determined by simulation, the LPL EGR routes exhaust from after the turbocharger and cools it via a heat exchanger then introduces into the intake just before the compressor, as shown in [Figure 9](#) and [Figure 10](#).

CHARACTERIZATION OF DUAL FUEL COMBUSTION

There are four different combustion modes possible with gasoline port injection combined with direct diesel injection.