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BRAKE THERMAL EFFICIENCY IMPROVEMENTS OF A COMMERCIALLY BASED DIESEL ENGINE MODIFIED FOR OPERATION ON JP 8 FUEL

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W56HZV-08-C-0678, Commercially Based, JP-8 Compatible Diesel Engines

ABSTRACT

The majority of commercial off the shelf (COTS) diesel engines rely on EGR to meet increasingly stringent emissions standards, but these EGR systems would be susceptible to corrosion and damage if JP-8 were used as a fuel due to its high sulfur content. Starting with a Cummins 2007 ISL 8.9L production engine, this program demonstrates the modifications necessary to remove EGR and operate on JP-8 fuel with a goal of demonstrating 48% brake thermal efficiency (BTE) at an emissions level consistent with 1998 EPA standards. The effects of injector cup flow, improved turbo match, increased compression ratio with revised piston bowl geometry, increased cylinder pressure, revised intake manifold for improved breathing, and piston, ring and liner designs to reduce friction are all investigated. Testing focused on a single operating point, full load at 1600 RPM. This engine uses a variable geometry turbo and high pressure common rail fuel system, allowing control over air fuel ratio, rail pressure, and start of injection. These parameters were optimized for several component combinations to provide an estimate of the best engine efficiency that could be achieved for various levels of engine modification. While the program goal is to have emissions consistent with 1998 EPA standards, testing was also conducted at higher emissions levels to determine the additional gain in BTE that could be possible if emissions were not a constraint.

INTRODUCTION

The majority of commercial off the shelf (COTS) diesel engines rely on EGR to meet increasingly stringent emissions standards, but these EGR systems would be susceptible to corrosion and damage if JP-8 were used as a fuel due to its high sulfur content. Starting with a Cummins 2007 ISL 8.9L 425 hp production engine, this program demonstrates the modifications necessary to remove EGR and operate on JP-8 fuel with a goal of demonstrating 48% brake thermal efficiency (BTE) at an emissions level consistent with 1998 EPA standards. The modifications considered for improved BTE include:

- *Increased peak cylinder pressure to 3200 psi max*
- *Increased compression ratio*
- *Modified piston bowl geometry*
- *Swirl*
- *Injector cup flow*
- *Port and manifold breathing improvements*
- *Improved turbo match*

Three Project Phases
Analysis, Performance Demonstration, Durability

The overall project is divided into three phases. Phase 1 consisted of analysis to develop and evaluate engine modifications necessary to meet project goals. Phase 1 also included limited engine testing for the purpose of calibrating analytical models. Phase 2 involved testing of hardware recommendations based on analysis results. Phase 3 is then a limited durability test of the final engine configuration determined by the Phase 2 test results.

This paper includes a limited discussion Phase 1 analysis results and selection of test hardware. The primary focus is then on actual Phase 2 test results

Operating condition for improved BTE

While the engine must be capable of running anywhere within its operating range with smooth transitions from one point to another, the focus for demonstrating 48% BTE is at a single steady state operating point. The modifications to improve BTE at this single point generally result in improved BTE across much of the operating range, but attaining a specific BTE target across multiple operating points or a particular duty cycle is beyond the scope of this work.

There are no defined criteria for selecting this operating point. Data collected from the Phase 1 engine test was used to identify operating points with reasonable BTE that would be a good starting point to focus efforts for further improvements. BTE was reasonably constant near 42% along the torque curve between 1300 RPM and 1700 RPM. Based on this, 1600 RPM full load (1148 lb-ft) was selected as the point for targeting 48% BTE. Other operating conditions are explored during Phase 2 testing, but the analysis and design efforts leading to Phase 2 test hardware were aimed at maximizing BTE at this point.

A few words on emissions targets...

1998 EPA standards only regulated transient emissions with a requirement that bsNOx not exceed 4 g/hp-hr over the FTP cycle. Since this project is limited to steady state testing only, some assumptions are required to determine emissions targets for this work. Current EPA standards limit both transient and steady state emissions. Steady state emissions must be measured over a cycle defined under supplemental emissions testing (SET) requirements consisting of 13 operating modes. Total emissions during this cycle must meet same limits as the transient FTP test. A not to exceed (NTE) region is then defined for steady state operation at points between the 13 mode test, and emissions in this NTE region must be less that 150% of the SET limit.

Applying this same rational to 1998 emissions standards leads to a steady state emissions requirement of 4 g/hp-hr bsNOx over an SET cycle and 6 g/hp-hr in the NTE region. Although a complete SET cycle is also outside the scope of this work, it is reasonable to set a target of 4 to 6 g/hp-hr for bsNOx at any point being tested. This establishes a target for NOx, but testing was also conducted over a much wider range to determine the additional benefit to BTE if emissions constraints were relaxed.

No actual limits for smoke or DPM are defined, however smoke levels should be reasonable and were targeted to be less than 1 FSN during testing.

ANALYSIS RESULTS & RECOMMENDATIONS

Several analytical studies were conducted and used to develop designs for the hardware tested in Phase 2. As discussed previously, this analysis focused on improving BTE at the selected operating condition of 1600 RPM full load (1150 lb-ft). KIVA was used to evaluate the benefit of increased peak cylinder pressure and to optimize compression ratio, piston bowl geometry, swirl, injector cup flow, and injector spray angle. Intake and exhaust manifold and port optimization was done with Fluent. GT Power was used to also consider the effects of increased cylinder pressure and compression ratio along with turbo and other air handling improvements.

Combustion System – KIVA Analysis Piston, Swirl, Injector, Peak Cylinder Pressure

DOE's were run using KIVA to evaluate various combinations of compression ratio, bowl geometry, swirl, injector cup flow, and rail pressure. Optimization of these results was done subject to constraints of fuel specific NOx of 30 g/kg (roughly equivalent to 5 g/hp-hr bsNOx) and peak cylinder pressure of 3200 psi. The KIVA analysis was done without EGR, so NOx control does become a challenging aspect.

The final recommendations from this process are listed in Table 1 and result in a predicted gain of 3.5% in BTE over the baseline.

Table 1 - Combustion System Recommendation

	Recommended for Improved BTE
Compression ratio	19.0
Swirl	1.3
Injector cup flow	145 pph
Cylinder Pressure Limit	3200 psi

Increasing compression ratio beyond 19 requires retarding injection timing to limit NOx, resulting in a net decrease in BTE. Injector cup flow of 145 pph is at the low end of the range considered, which in conjunction with rail pressure at the low end (1000 bar) increases combustion duration and reduces NOx. Injection timing can then be advanced with a predicted net gain in BTE at the target NOx level.

The two piston bowls with the best predicted performance are shown in Figure 1 along with the production bowl for reference.

CR19-3 is wider and shallower than the baseline bowl and had slightly better predicted BTE at the target NOx level. CR19-7 is the baseline bowl shape but simply scaled to reduce volume and increase compression ratio. It had slightly better predicted BTE at NOx levels above the target. Given the desire to explore performance over a range of NOx values and the fact that there is some uncertainty in KIVA predictions, it was decided to test both of these.

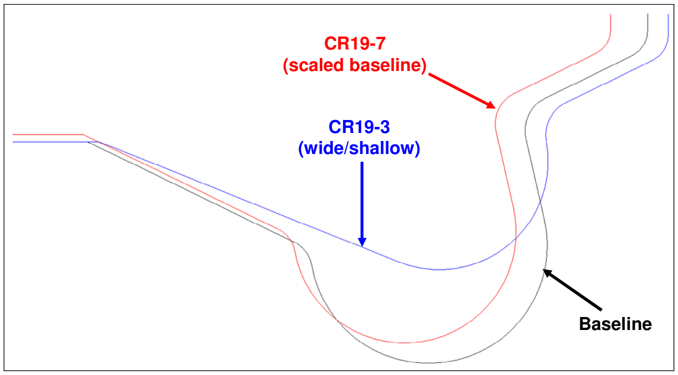

Figure 1 - Piston Geometries

Intake Manifold and Ports

Intake port and manifold designs were evaluated using Fluent CFD with a goal of reducing pressure drop while hitting the swirl target. With a completely redesigned manifold and ports, this analysis showed pressure drop could be reduced by 32%. Designing and procuring a completely new head is beyond the scope of this project, but the analysis results can be used to guide modifications to the production head and still gain a significant benefit.

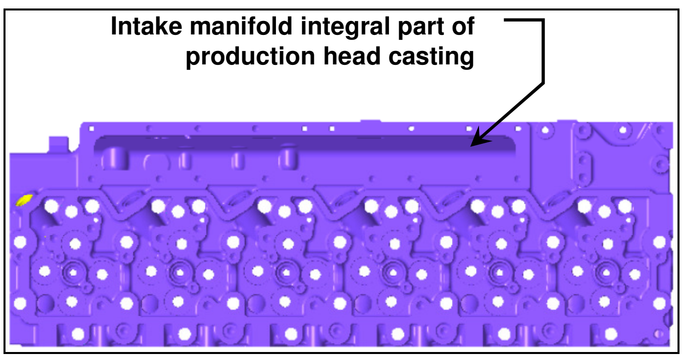

Figure 2 - Production Head with Intake Manifold

Intake flow passages of the production head are an integral part of the casting as seen in Figure 2. For the engine test, this air box was milled off and an aluminum casting made that bolted to the head, illustrated in Figure 3. The fuel rail was repositioned and new fuel lines made to accommodate this change. Pressure drop and swirl were measured on a flow bench and the intake ports modified with a hand grinder until the target swirl of 1.3 was achieved for each cylinder.

Flow bench data showed an 18% reduction in pressure drop compared to the production head, a little over half of what could be possible with a completely new head.

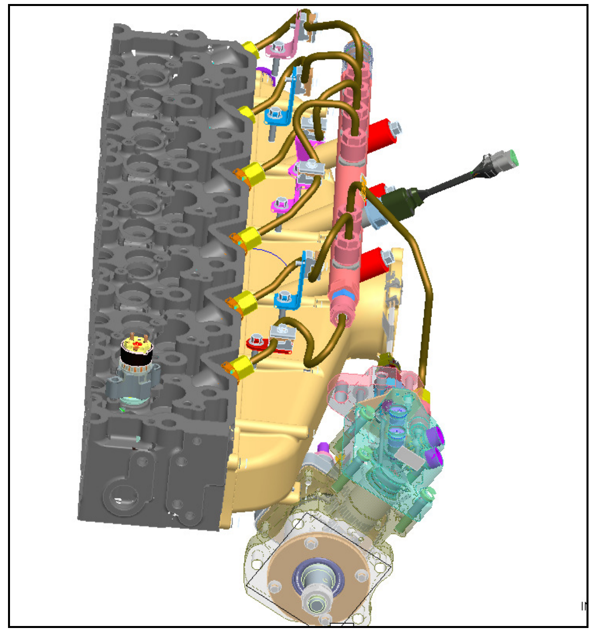

Figure 3 - Modified head with bolt on intake manifold

TEST HARDWARE

The test hardware is described in the following sections with a summary presented in Table 2.

Base Engine

The base engine is a production ISL 8.9L engine compliant with 2007 EPA emissions standards rated at 425 hp with peak torque of 1200 lb-ft.

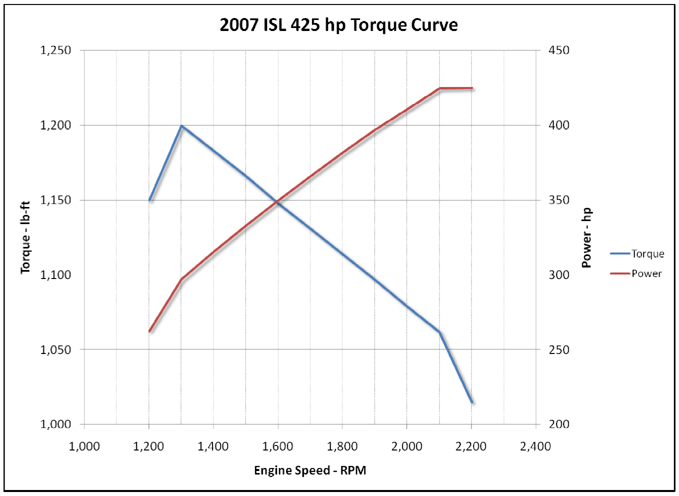

Figure 4 - 2007 ISL 425 hp Torque Curve

Features of this engine include cooled exhaust gas recirculation, a variable geometry turbocharger and a high pressure common rail fuel system.

Peak Cylinder Pressure

One of the areas of interest is operation at higher peak cylinder pressure. Design and analysis studies were conducted of modifications required to allow cylinder pressure up to 3200 psi. A review of those results is outside the scope of this paper, however it is sufficient to say that the modifications were significant enough that they could not be practically incorporated into the test hardware. Testing was conducted however at cylinder pressures up to 3200 psi. This exceeds the design limit for this engine, but could be allowed for the limited test time of this program.

Injectors

KIVA pointed towards low injector cup flow for best performance in the target 4-6 g/hp-hr bsNOx range – low cup flow increases duration, reducing NOx and allowing more advanced SOI with a net gain in BTE. As discussed previously though, there is also an interested in understanding the additional improvement in BTE at higher NOx levels, and here, KIVA predicted a better BTE with higher cup flows.

Injector cup sizes are identified by flow rate in pounds per hour (pph). Cup flows of 149, 165, and 180 pph were selected for testing. 149 pph is slightly larger than the KIVA recommendation of 145 pph, but was available in existing hardware and considered to be close enough to the KIVA recommendation.

Pistons

The pistons tested were the baseline production piston, and the 19:1 compression ratio pistons with bowls 1 and 3 shown in Figure 1.

Cylinder Heads

Cylinder heads tested were the baseline production head and the modified head for improved breathing and reduced swirl shown in Figure 3.

Turbochargers

Turbochargers were assembled from available components, mixing and matching to attain the best combination of turbine efficiency, compressor efficiency, and size to match the flow requirements of this engine without EGR – which proved to be somewhat of a challenge. Improving one end of the turbo (e.g. compressor with improved efficiency) required use of components on the other end that were less than an ideal match in size or efficiency. Some compromises had to be made, but the turbos selected do provide improvements over the baseline production turbo in terms of both efficiency and match to the flow requirements.

Turbo A	Larger turbine for operation without EGR
Turbo B	Improved compressor efficiency (Reduced turbine efficiency)
Turbo C	Improved turbine efficiency (Reduced compressor efficiency)
Turbo D	Improved compressor efficiency Improved turbine efficiency (Larger flow capacity than desired)

Table 2 - Test Hardware Summary

Injector Cup Flow	149 pph 165 pph 180 pph
Pistons	Production piston 19:1 CR19-3 (wide/shallow bowl) 19:1 CR19-7 (scaled production bowl)
Peak Cylinder Pressure	3200 psi (above design limit but acceptable for limited test time)
Cylinder Head	Production Modified with reduced swirl
Turbochargers	4 turbo variations A Larger turbine for operation without EGR B Improved compressor efficiency (Reduced turbine efficiency) C Improved turbine efficiency (Reduced compressor efficiency) D Improved compressor efficiency Improved turbine efficiency (Larger flow capacity than desired)

PERFORMANCE WITH JP-8

Baseline fuel maps were run on the production engine with both diesel and JP-8 fuel to characterize the effects of JP-8 on engine performance. The only difference in the test configuration with JP-8 was use of a fuel filter with a lubricity additive. There were no modifications made to the engine, its calibration, or EGR levels (with the short time needed to complete this test, there was little risk of damage or failure of EGR components from the sulfur in the JP-8 fuel).

JP-8 Properties

No special requirements were placed on the JP-8 purchased for this testing, only that it meet the specification MIL-DTL-83133E. Properties of both the diesel fuel and JP-8 used in this testing were measured for comparison though with results presented in Table 3.

Table 3 - Measured Fuel Properties Diesel Fuel #2 Compared to JP-8

Property	Units	DF #2	JP-8
Cetane Number		46.6	46
Hydrogen Content	% wt	13.99	15.02
Carbon Content	% wt	86.01	84.98
Nitrogen Content	% wt	< 0.50	0
Sulfur Content	ppm	1	75
Heat of Combustion [gross]	BTU/lb	19711	19884
Heat of Combustion [net]	BTU/lb	18492	18514

The most notable difference is in the sulfur content. At 75 ppm, the JP-8 is above the 10 ppm limit for today's on highway fuels, but well below the maximum allowable of 3000 ppm.

Effect of Lubricity Additive

To avoid excessive wear in the fuel system due its lower lubricity, a fuel filter with a lubricity additive was used when testing with JP-8. To verify the lubricity additive does not affect performance or emissions, a torque curve was run using diesel fuel with the lubricity additive to provide a back to back comparison using the same fuel but with and without lubricity additive.

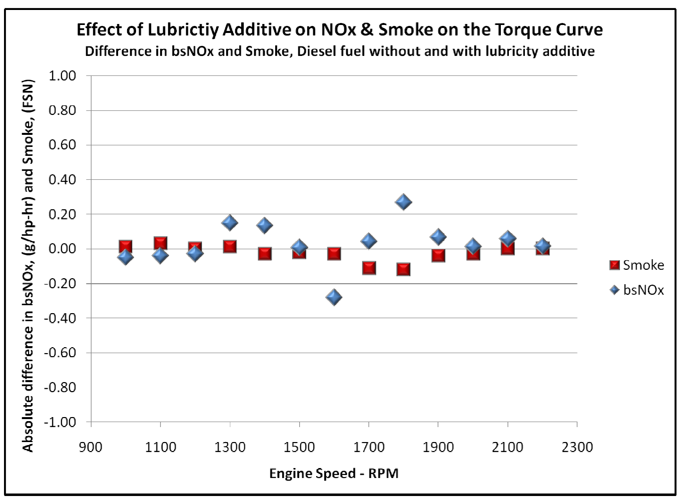

Figure 5 - Difference in bsNOx and Smoke Diesel fuel with and without lubricity additive

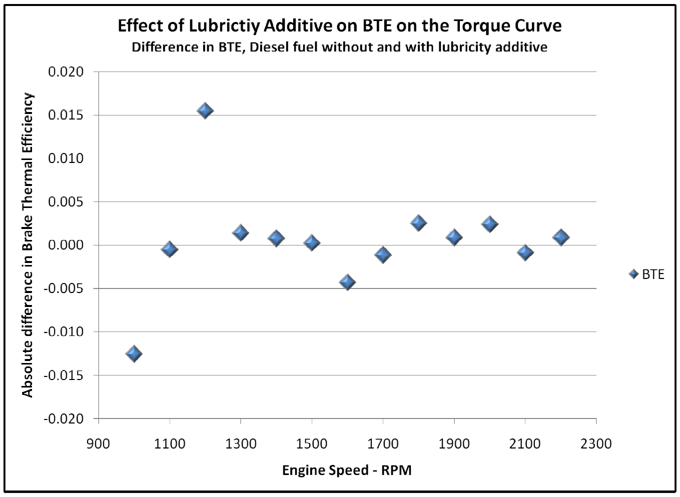

Figure 6 - Difference in Brake Thermal Efficiency Diesel fuel with and without lubricity additive

Figures 5 and 6 show the absolute difference in bsNOx, smoke and BTE arrived at by subtracting the results with lubricity additive from those without. While there is a little scatter in the data, it can be seen the effect of lubricity additive is negligible. Any difference in performance when using JP-8 fuel can therefore be attributed to the fuel itself and not the influenced by the lubricity additive used with it.

Performance Comparison, JP-8 to Diesel

JP-8 was found to have an insignificant effect on engine power, brake thermal efficiency, and NOx. The most notable difference was that smoke was lower with JP-8, however JP-8 did require slightly more throttle for the same fuel rate.

The fuel map was run holding speed constant by the dyno and throttle adjusted under test cell control until target torque values were met. Figure 7 shows the throttle command along the advertised torque curve. Fuel rates along the torque curve are nearly identical for both diesel and JP-8, but it can be seen the commanded throttle is 1-2% higher with JP-8. Another way to look at this is at the same throttle, fueling and hence power would be 1-2% lower for JP-8.

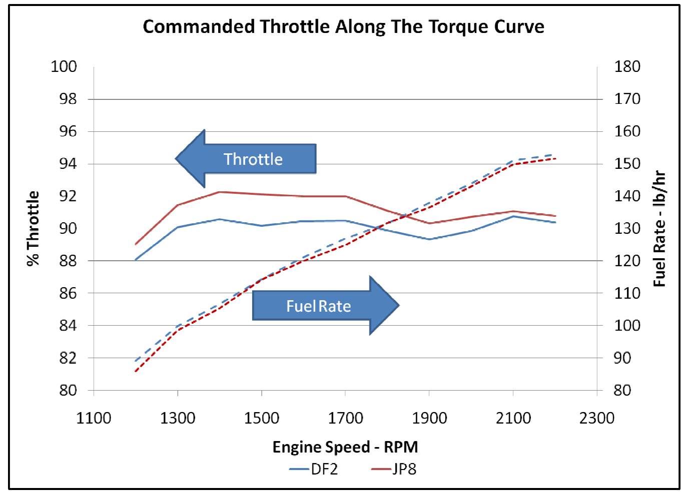

Figure 7 - Fuel Rate & Commanded Throttle Along the Torque Curve

The graphs in Figures 8-10 show the difference in brake thermal efficiency, bsNOx and smoke arrived at by subtracting results with JP-8 from those with diesel over the operating range of this engine. BTE with JP-8 is within .005 points of that with diesel over almost the entire operating range, a small enough difference to be outside the ability to measure accurately. Similarly, there is no measurable difference in bsNOx. The two small regions with a difference of 1 g/hp-hr are a result of missing NOx data at those points and do not represent an actual performance difference.

Smoke however is lower with JP-8 over the entire operating map, with the largest reduction being 0.3 FSN around 2000 RPM, 400 lb-ft. This is the area with the highest smoke levels for this engine and hence the largest reduction was observed here.

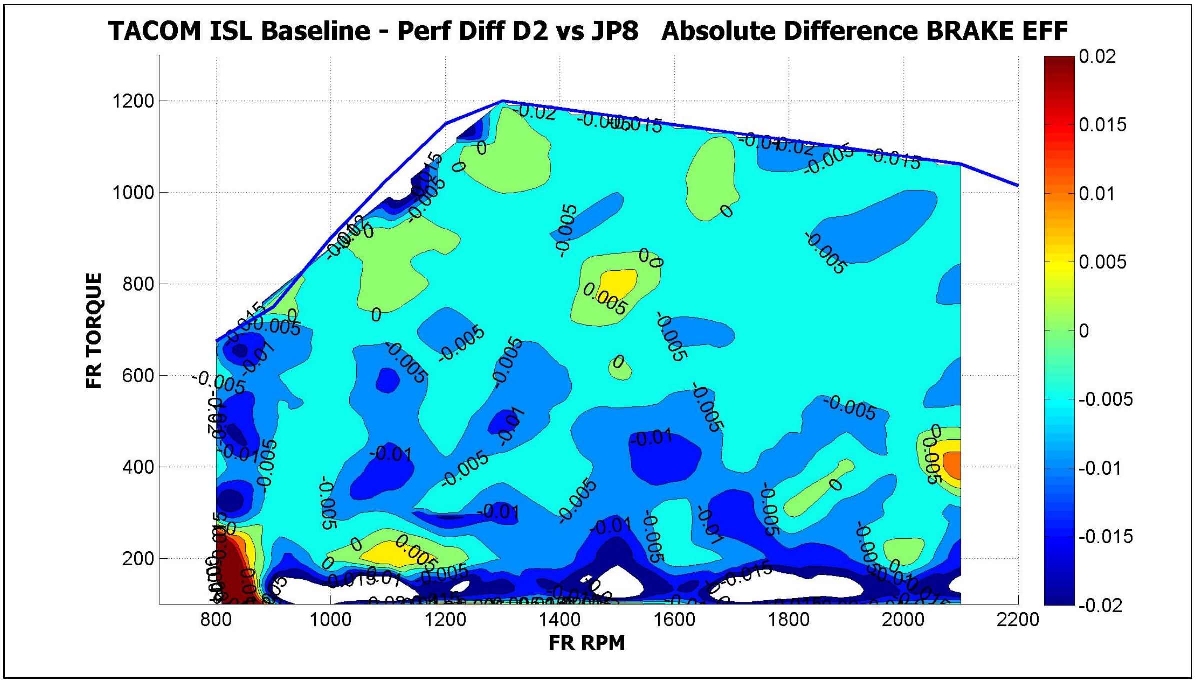

Figure 8 – Change in Brake Thermal Efficiency Due to Operation on JP-8

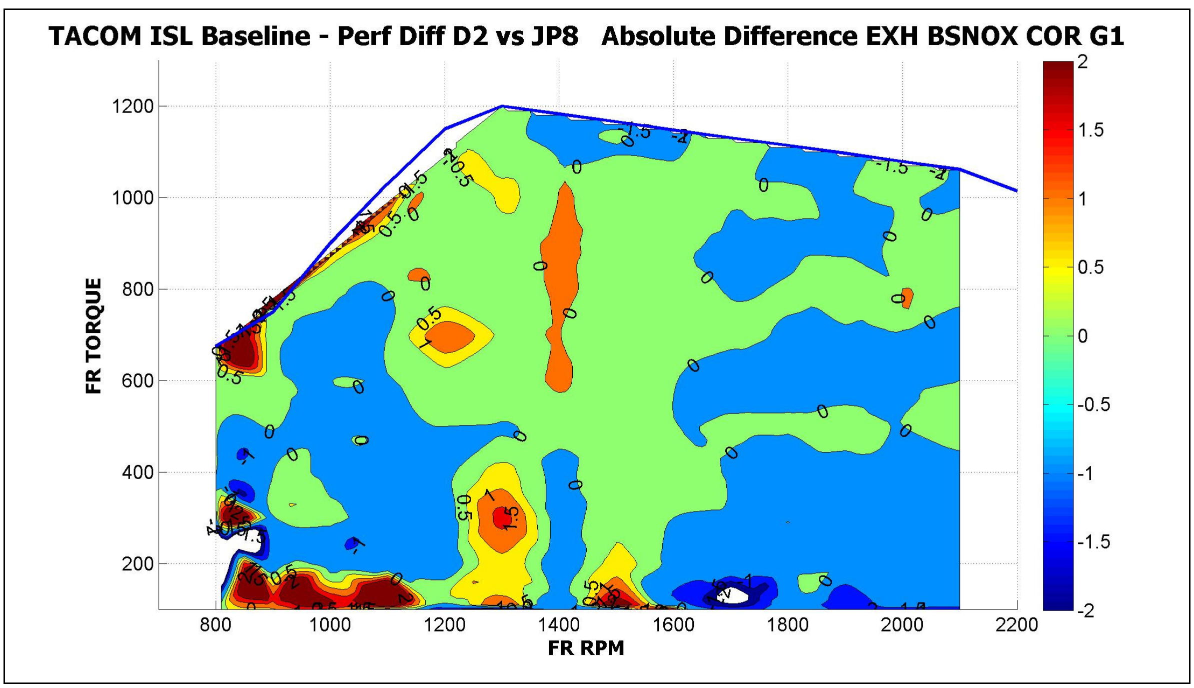

Figure 9 - Change in bsNOx (g/hp-hr) Due to Operation on JP-8

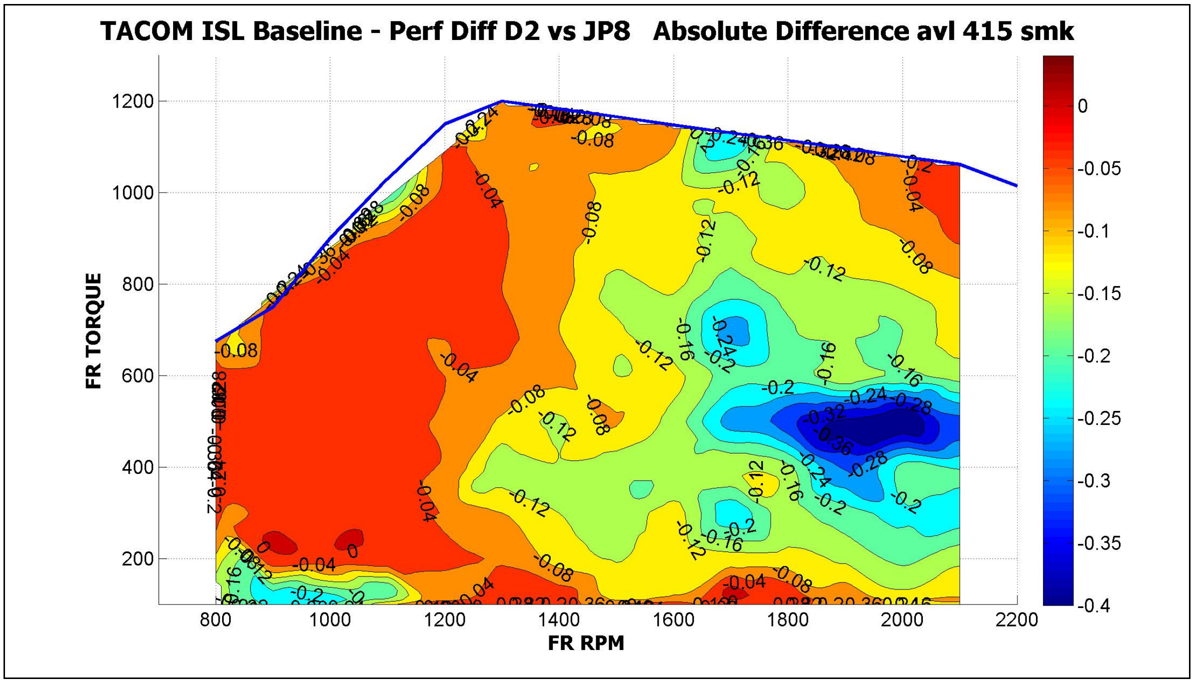

Figure 10 - Performance with JP-8 Compared to Diesel, Difference in Smoke (FSN)

MODIFICATIONS FOR IMPROVED BTE TEST SETUP & PROCEDURES

Sequence of hardware tests

Recognizing the desire for minimum modifications to operate on JP-8 without EGR, tests were generally conducted by stepping from minimum modifications but with minimum expected gain in BTE to progressively more significant modifications with larger expected improvements to BTE.

Operating Conditions

As discussed previously, 1600 RPM, full load (1148 lb-ft) was selected as the point to for analysis and design work to improve BTE, and is the primary focus of test work. From the baseline fuel map data, there was a region of high negative PMEP around 1600 RPM, 850 lb-ft which offers an opportunity for improvement, so testing was also conducted there.

Other speeds both higher and lower than 1600 RPM at both full and part load were also initially tested. These did not yield significantly better BTE and relative hardware comparisons at those other operating conditions led to the same general conclusions– e.g. the injector that gave the best performance at 1600 RPM full load also gave the best performance at other operating conditions. Results presented here will be based on testing at the 1600 RPM 1148 lb-ft and 850 lb-ft points.

Test Procedures

Hardware evaluation was typically done by running a DOE with a series of randomly generated combinations of air fuel ratio (A/F), rail pressure, and start of injection (SOI). A statistical model was fit to the test results and optimization done to maximize BTE subject to constraints for bsNOx, PCP, and smoke. The constraint for PCP and smoke were always held to 3200 psi and 1 FSN max. Optimization was done starting with the NOx constraint at a low value in the target range of 4-6 g/hp-hr and progressing upwards until there was no longer a benefit to BTE or until the boundaries of the data set were reached. This results in a BTE/NOx tradeoff for a given set of hardware with optimized values for A/F, rail pressure and SOI at each NOx level These conditions were then run in the test cell to validate the results.

Given the performance transparency of JP-8 fuel when compared to diesel, this development work was done using diesel, occasionally switching to JP-8 as a check that it did not produce a different result.

EGR REMOVAL & EFFECT ON IMT

EGR hardware was physically removed from the engine. The EGR cooler was taken off, cooling lines plugged, the EGR crossover tube and valve removed, and flow passages capped. Without EGR, more fresh air flow is needed to maintain similar charge flow levels so the turbocharger was changed to a larger one that is used on a production 11L engine.

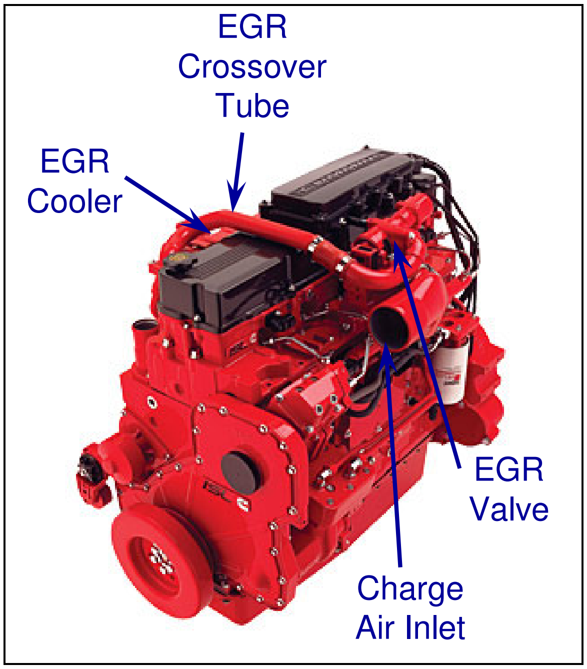

Figure 11 - 2007 ISL EGR Components

In the test cell, charge air from the turbocharger flows through a water cooled heat exchanger. Water flow is controlled to maintain an air discharge temperature 125 °F, simulating a typical air to air charge air cooler used in a vehicle. When mixed with the cooled EGR in the intake manifold, the intake manifold temperature was in the range of 140 °F to 150 °F. After removing EGR, the intake manifold temperature was the same as the charge air cooler outlet temperature, 125 °F.

INJECTOR CUP FLOW WITH PRODUCTION PISTON

The first step toward improving BTE was to test different injector cup flows. This is one of the easiest changes that can be made and the expectation from KIVA was that lower cup flow would extend duration with a reduction in NOx, allowing more advanced SOI and a net improvement in BTE.

Production pistons and cylinder head were used with turbo "A" (larger turbine stage). DOE's were run at 1600 RPM, full load and optimization was done with each injector at various NOx levels.

Table 4 - Engine Configuration for Injector Testing

Injector Cup Flow	149 pph 165 pph 180 pph
Piston	Production
Cylinder Head	Production
Turbocharger	"A" Larger turbine for operation without EGR

The results for BTE vs. bsNOx shown in Figure 12 did not follow the expected result. The lowest cup flow, 149 pph gave the worst performance and with the two higher cup flows, 165 pph and 180 pph being very similar.

Smoke shown in Figure 13 is below a value of 0.5 FSN and is very acceptable for all three injectors, but tends to increase as cup flow decreases.

Similar peak cylinder pressure was attained for all three injectors, seen in Figure 14. It is worth noting that in the target NOx range, there is no need for high cylinder pressure capability. The system becomes constrained by NOx and the operating conditions required to stay within target NOx levels do not create in high cylinder pressures. It is not until higher NOx levels are allowed that there is a benefit from higher cylinder pressure.

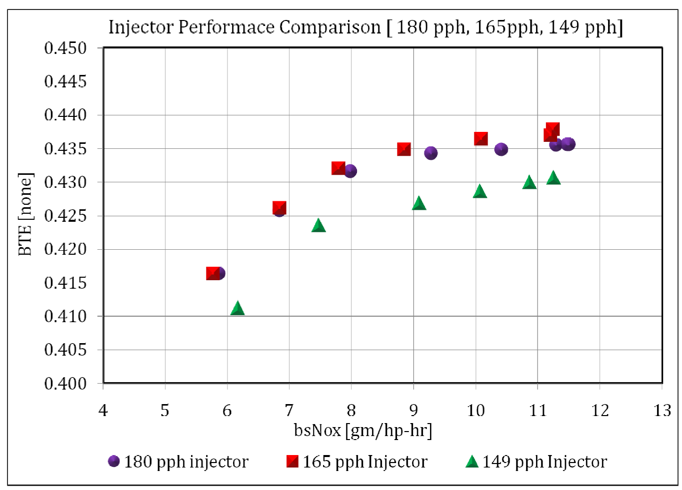

Figure 12 - Injector Comparison, BTE vs bsNOx

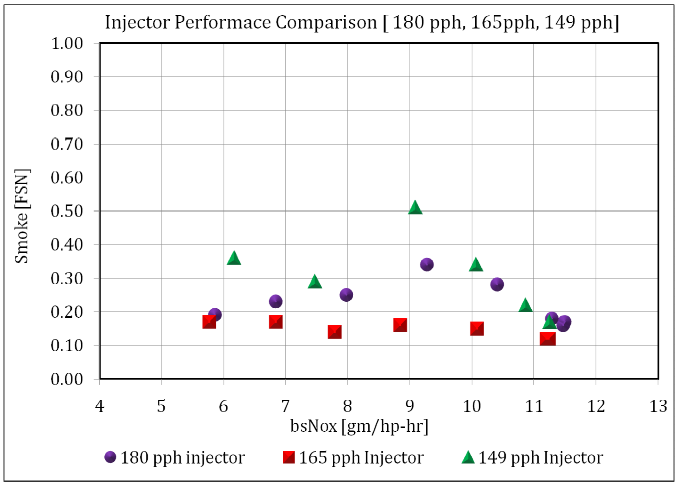

Figure 13 - Injector Comparison, Smoke vs bsNOx

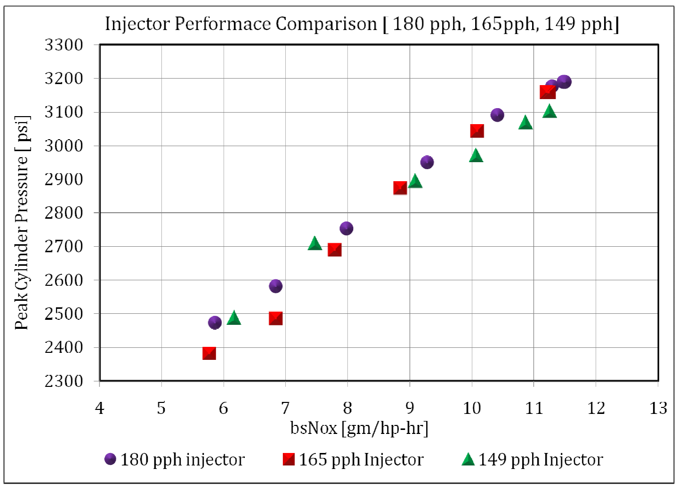

Figure 14 - Injector Comparison
Peak Cylinder Pressure vs bsNOx

It is also interesting to compare the values for A/F, SOI, and rail pressure arrived at through the optimization process for each injector (Figures 15-17). The optimum air fuel ratio was very similar for both the 165 pph and 180 pph injectors, however the 149 pph injector optimized at a somewhat lower value.

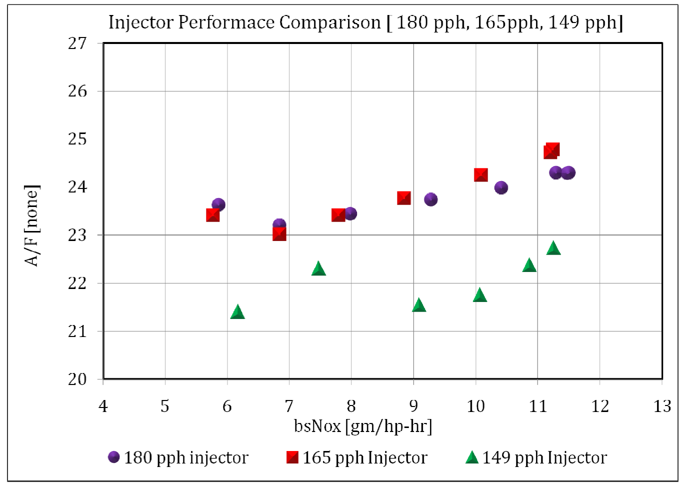

Figure 15 - Injector Comparison, A/F vs bsNOx

The expected general trends of increasing rail pressure and advancing SOI as NOx increases are evident, and over most of the NOx range, the 149 pph injector had more advanced timing than the other two.

Looking at the details of this though, SOI for 165 and 180 pph injectors are very similar, with the 180 pph injector being slightly more advanced. At a NOx level of 10 and above, the 149 pph actually starts to back off in timing. These trends can be explained by looking at the rail pressure. While the 180 pph injector is inherently shorter duration, it optimized at a lower rail pressure that the 165. The 149 pph injector jumps to higher rail pressure at higher NOx levels, which requires less advanced timing.

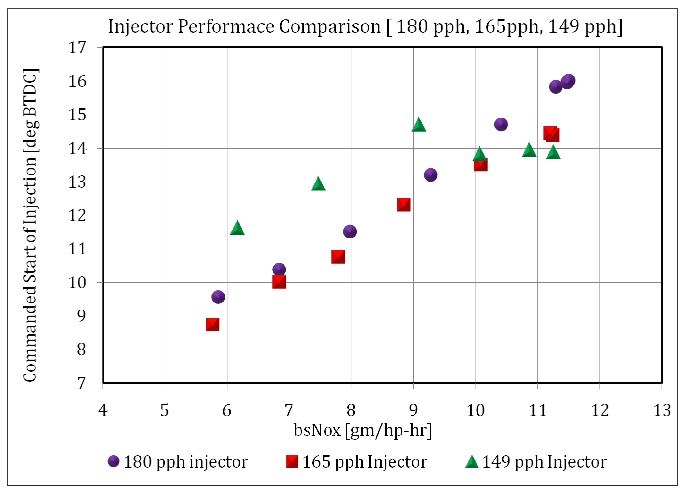

Figure 16 - Injector Comparison
Commanded Start of Injection vs bsNOx

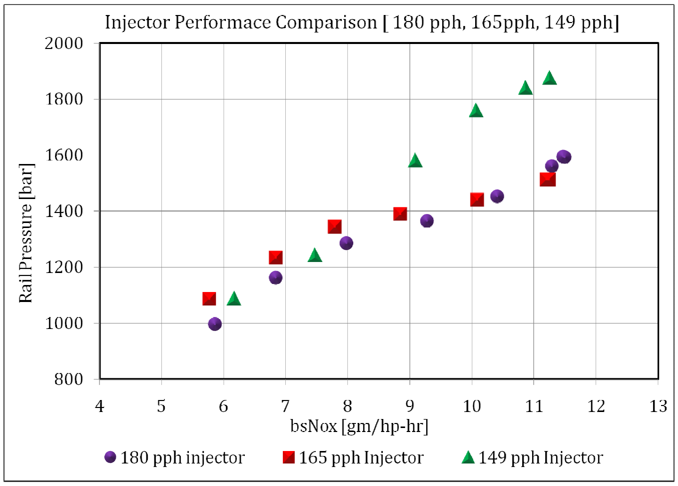

Figure 17 - Injector Comparison, Rail Pressure vs bsNOx

The effect on heat release can be seen in Figure 18. The 180 pph injector with lower rail pressure produces almost identical apparent heat release to the 165 pph with higher rail pressure. As expected, the 149 pph injector at about the same rail pressure as the 165 has longer duration. The earlier SOI for 149 pph is also evident.

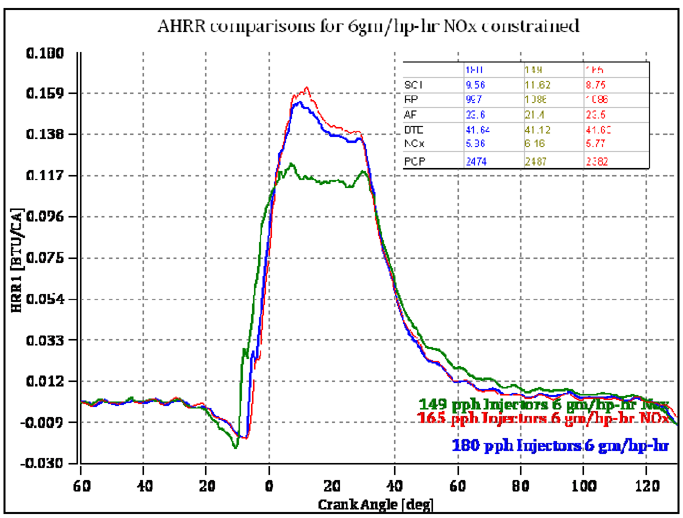

Figure 18 - Apparent Heat Release at 6 g/hp-hr bsNOx 149, 165, & 180 pph Injectors

Comments on Optimization Procedure

Discussion of the optimizer can help explain why some results of the results aren't necessarily what might be expected. The optimizer marches through the solution space from a starting point towards the objective of maximum BTE subject to NOx, PCP and smoke constraints. Rail pressure and SOI can trade off against each other such that higher rail pressure with more retarded timing produces similar performance as lower rail pressure with more advanced timing – i.e. multiple solutions can exist for the same BTE/NOx point.

Solutions were found for the 180 pph injector with slightly more advanced timing but lower rail pressure than the 165 pph, however it is possible that other solutions exist with the 180 pph injection at more retarded timing but similar rail pressure to the 165 pph injector. For the 149 pph injector, solutions were found with more advanced timing until high NOx levels were reached where the optimizer moved to a solution space with higher rail pressure and less advanced timing

The fact that multiple solutions may exist to maximize BTE at a given NOx level does not change the conclusions of which injector is preferred based on this test data. If desired, further investigation could be done to identify other possible solutions. It would then be up to the judgment of the engineer to select one based on other criteria such as smoke, desire for low rail pressure, or perhaps combustion noise.

Conclusions from Injector Testing with Production Head and Pistons

Considering this as the minimum modification to allow operation with JP-8 and no EGR, the recommendation would be to use a 165 pph injector, yielding a BTE of 41.6% at 6 g/hp-hr, at the upper end of the target NOx range, and there is need for increased peak cylinder pressure capability. If the NOx constraint were removed, this configuration is capable of a maximum BTE of 43.7% at a bsNOx of 11.2 g/hp-hr, however this does require in peak cylinder pressure capability to 3200 psi.

INCREASED COMPRESSION RATIO

The next step in engine modification was to change to higher compression ratio pistons. Pistons CR19-3 with a wider, shallow bowl (ref. Figure 1) were installed with the production head and 149 pph injectors. These injectors were chosen because they were the recommendation from KIVA for use with this piston, even though they were not preferred with the production piston. Other injector cup flows with this piston are presented later in this paper.

Table 5 - Engine Configuration for 19:1 CR Piston Test

Injector Cup Flow	149 pph
Piston	CR19-3 (wide/shallow bowl)
Cylinder Head	Production
Turbocharger	"A" Larger turbine for operation without EGR

Test Method

As before, DOE's were run and an optimization of A/F, SOI and rail pressure done at various NOx levels subject to constraints of 3200 psi max PCP and 1 FSN smoke. The engine was run at the optimized conditions and results compared to those previously generated with the production pistons.

Test Results

At the target NOx level of 6 g/hp-hr, increasing compression ratio to 19:1 improved BTE by 2% as seen in Figure 19. It can also be seen that while there is still a benefit at higher NOx levels, it starts to diminish. At 8 g/hp-hr bsNOx, the cylinder pressure limit of 3200 psi was reached and because of this, there no additional improvement in BTE at higher NOx levels. The conditions required to stay within 3200 psi cylinder pressure (lower rail pressure, less advanced SOI) caused BTE to start falling off at higher NOx.

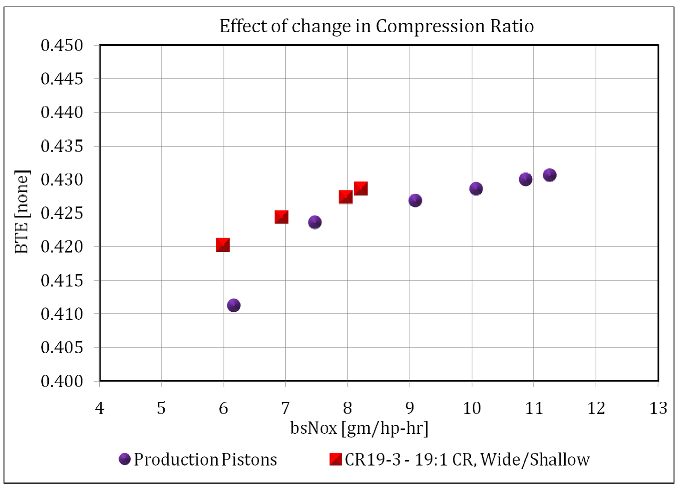

Figure 19 - 19:1 Wide/Shallow Piston, BTE vs bsNOx

Note that peak cylinder pressures shown in Figure 20 are generally higher with 19:1 compression ratio, which is not a surprise. Taking advantage of efficiency gains from higher compression ratio requires higher peak cylinder pressure capability, however in the target NOx range, it does not need the full 3200 psi.

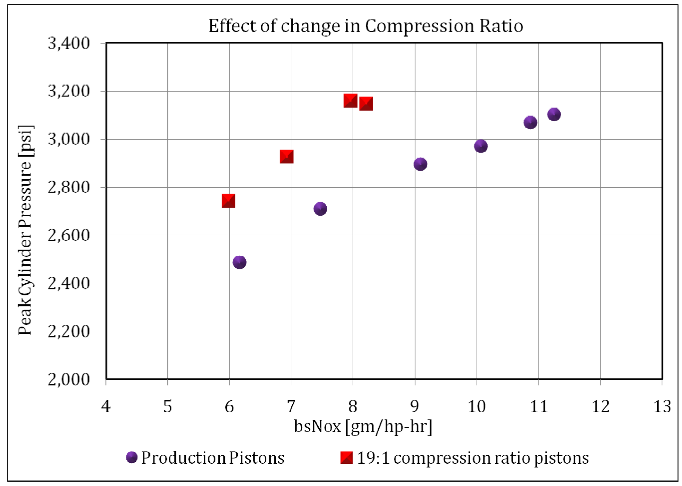

Figure 20 - 19:1 Wide/Shallow Piston, PCP vs bsNOx

Conclusions from Testing with Higher Compression Ratio Piston

Increasing compression ratio to 19:1 with the production cylinder head improved BTE by approximately 2% at 6 g/hp-hr bsNOx, the upper end of the target NOx range. To take advantage of this requires an increase in peak cylinder pressure capability to 2800 psi. The BTE benefit diminishes at higher NOx levels, and requires still higher cylinder pressures up to 3200 psi at 8 g/hp-hr bsNOx. Above this NOx level the system becomes constrained by cylinder pressure and no further benefit was found.

EFFECT OF BACKPRESSURE DUE TO DPF

In an on-highway application, the baseline engine requires use of a diesel particulate filter (DPF) to meet 2007 EPA standards for particulate emissions. A DPF would not be required for 1998 EPA emissions levels though and operation without a DPF will reduce backpressure on the engine and improve BTE.

Engine Configuration and Test Conditions

The backpressure testing was done with the modified cylinder head, CR19-7 pistons (scaled production), 180 pph injectors, and a turbocharger intended to provide improved efficiency. This is jumping ahead a couple steps in terms of engine configuration, but review of the effect of backpressure will provide a background for discussion of the effect of the head change in the next section.

Table 6 – Engine Configuration for Testing Effects of Reduced Backpressure (no DPF)

Injector Cup Flow	180 pph
Piston	CR19-7 (19:1 CR, Scaled Production bowl)
Cylinder Head	Modified, 1.3 Swirl
Turbocharger	"D" Improved compressor efficiency Improved turbine efficiency (Larger flow capacity than desired)

Test Procedure

Tests were run at full load and part load at conditions established from the first optimization done after EGR was removed. The intent was to isolate the effect of backpressure alone by running at the same conditions with both high and low backpressure.

A DPF is not actually used in the test cell, but the backpressure it would create represented by adjusting a valve in the test cell exhaust stack. The engine is run at its rated condition (2100 RPM, 425 hp) and the valve position adjusted until the turbine outlet pressure is 10 in Hg and held in that position for all other operating conditions. To represent an exhaust system without a DPF, the same procedure is followed however the turbine outlet pressure is set to 2 in Hg, representative of an exhaust system without a DPF. Actual backpressure achieved at the test conditions is shown in Figure 21 below.

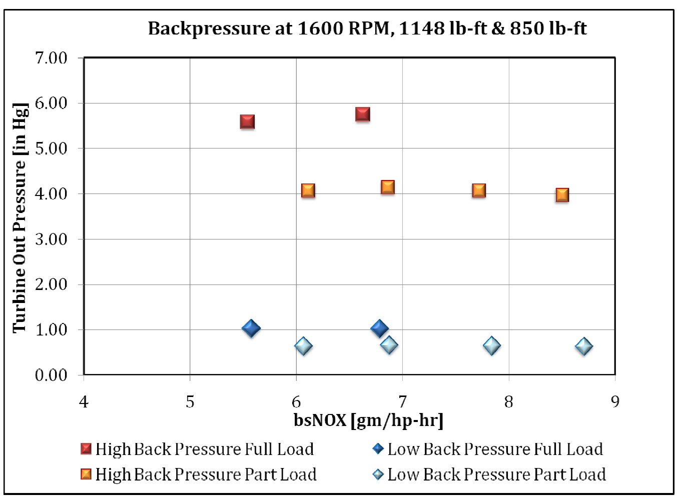

Figure 21 - Backpressure at 1600 RPM 1148 lb-ft & 850 lb-ft

Backpressure Reduction - Test Results

Figures 22 and 23 show BTE vs. bsNOx and the percent difference between the low and high backpressure data. Testing at full load could only extend to 7 g/hp-hr bsNOx before the cylinder pressure limit was reached, but both the full load and part load results show fairly consistently that reduced backpressure due to removal of a DPF improves BTE by 1.6% across the NOx range tested.

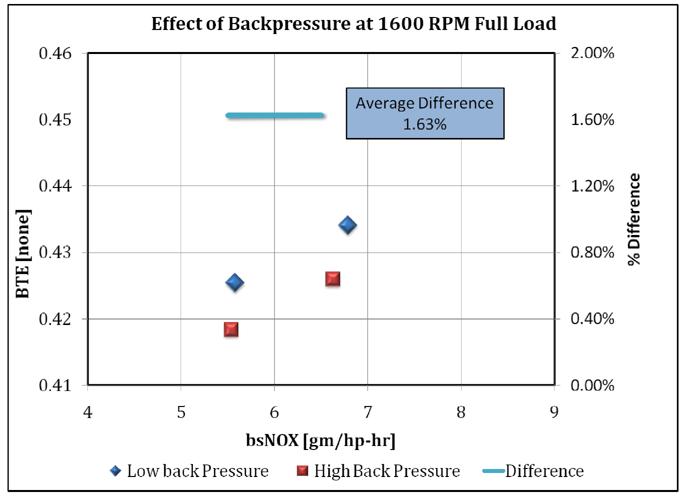

Figure 22 - Effect of Backpressure on BTE 1600 RPM, 1148 lb-ft

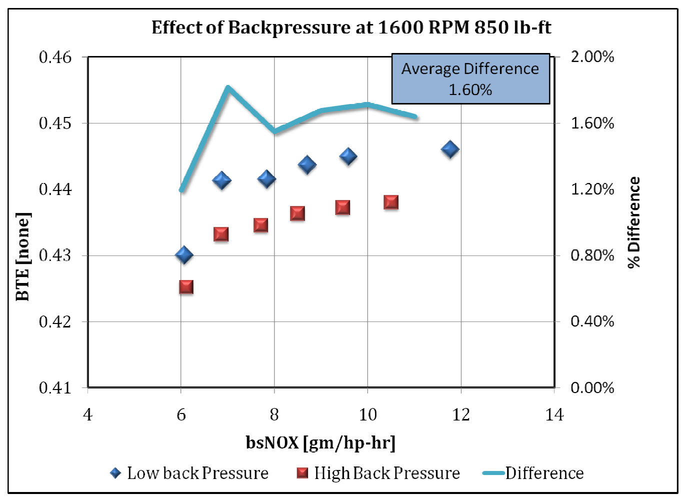

Figure 23 - Effect of Backpressure on BTE 1600 RPM, 850 lb-ft

Final Conclusions, Backpressure Reduction from DPF removal

Reduction in backpressure associated with removal of a diesel particulate filter improves brake thermal efficiency by 1.60% to 1.63%.

MODIFIED CYLINDER HEAD

Now back to hardware comparisons. Next was installation of the modified head with reduced swirl and pressure drop. Pistons were CR19-3 (wide/shallow bowl) with 149 pph injectors, and results compared to the production head with these same pistons and injectors.

Backpressure & Turbo Change

There were two other changes made along with the head change. The first was a reduction in engine backpressure. Up to this point, all testing with the production head had been done with a backpressure to simulate a DPF. Testing with the modified head was done at backpressure levels consistent with no DPF. Also, testing started with the turbo used with the production head, turbo A, but that turbo experienced damage before testing was finished and it was replaced with turbo B. Comparison between the two heads will include some effect of the turbo change, however is it seen in a later section that turbo A and B had similar performance in back to back tests.

Table 7 - Engine Configuration Modified Head Testing

Injector Cup Flow	149 pph
Piston	CR19-3 (19:1 CR, wide/shallow bowl)
Cylinder Head	Modified, 1.3 Swirl
Turbocharger	"B" Improved compressor efficiency (Reduced turbine efficiency)

Test Method – Full DOE & Optimization

Full DOE's and optimization of A/F, SOI and rail pressure were done at 1600 RPM 1148 lb-ft and 850 lb-ft, with results compared to the optimized performance with the production head.

Modified Head Test Results Volumetric Efficiency

To compare the breathing of the two heads, volumetric efficiency was calculated using compressor outlet pressure as a reference rather than intake manifold pressure. Since intake manifold pressure is measured downstream of some of the head improvements, a vol eff calculation based on that does not capture all the effects of the improvements. Using compressor outlet pressure as a reference, the new head improved vol eff at 1600 RPM from 0.885 to 0.90 at both full and part load.

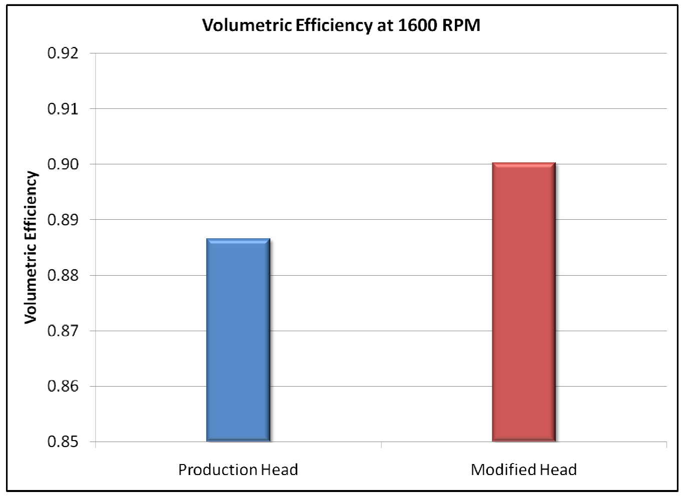

Figure 24 - Volumetric Efficiency Production Head & Modified Head at 1600 RPM

Modified Head Test Results – BTE

The combined effect of the head modification along with reduced backpressure significantly improved BTE, with a 3.8% improvement at full load and a 3.2% improvement at part load (Figures 25 and 26). In the previous section, it was shown that reducing backpressure improved BTE by 1.6%. That result can be applied here to estimate BTE of the modified head with high backpressure, leading to an estimate that the head change by itself is worth a 2.1% improvement in BTE at full load and 1.8% at part load.

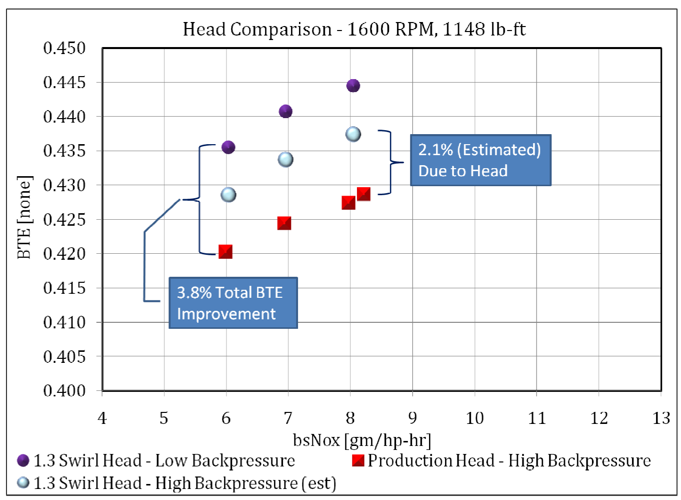

Figure 25 - BTE Comparison, 1600 RPM 1148 lb-ft Production Head & Modified Head

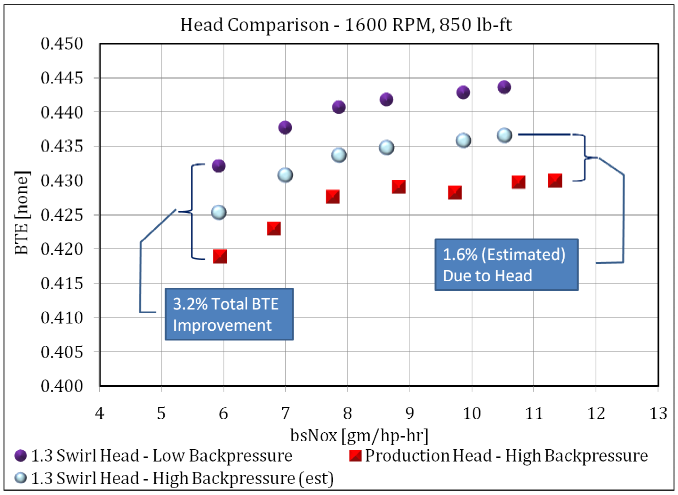

Figure 26 - BTE Comparison, 1600 RPM 850 lb-ft Production Head & Modified Head

It's really not possible to determine how much of this improvement is due to better breathing and how much is from reduced swirl. The original KIVA analysis showed a sensitivity to swirl. It is reasonable to speculate that much of the improvement seen here is from finally reaching the complete combustion system recommendation of compression ratio, bowl geometry, injector cup flow and swirl.

Final Conclusions – Modified Head

The test data directly showed the head modifications resulted in a BTE improvement of 3.8% at full load and 3.1% at part load, but this includes a benefit due to reduced backpressure. Accounting for the effect of backpressure discussed in the previous section, it is estimate that the head itself provided a benefit of 2.1% and 1.8% at full load and part load respectively.

19:1 CR BOWL PROFILE & ADDITIONAL INJECTOR CUP FLOW TESTING

To gain an understanding of the effect of bowl profile, pistons CR19-7 (production bowl shape scaled to increase compression ratio – ref. Fig. 1) were tested. The expectation based on KIVA was that these pistons would be preferred at higher NOx levels and would perform best with 180 pph injectors, however there was an interest to see how they performed with all three injector cup flows.

19:1 Scaled Production Bowl Test Method Timing Swings, Not Full Optimization

Since these pistons were expected to provide only a limited benefit at higher NOx and no benefit in the target NOx range, in the interest of time and schedule, it was decided to evaluate these pistons with timing swings at fixed A/F and rail pressure. Full DOE's and optimization require significant test time, and it has been seen in this project that relative comparisons between different hardware sets remain valid even when they are not individually optimized. In other words, it is generally valid to compare BTE vs NOx trends based on timing swings. Full optimization would yield improvements for each hardware set but not change comparison of one set of hardware relative to another.

Engine Configuration & Test Conditions

The engine had the modified cylinder head for 1.3 swirl and turbo A with larger flow capacity than the production turbo. Timing swings were done with rail pressure of 1000 bar and 1500 bar and air fuel ratio held constant at 26.

Table 8 - Engine Configuration for 19:1 CR Piston Test with CR19-7 Scaled Production Bowl

Injector Cup Flow	180 pph 165 pph 149 pph
Piston	CR19-7 (19:1 CR, scaled production bowl)
Cylinder Head	Modified, 1.3 Swirl
Turbocharger	A Larger turbine for operation without EGR

19:1 Scaled Production Bowl Test Results

At both 1000 bar and 1500 bar rail pressure, performance of the 165 and 180 pph injectors is similar at lower NOx levels. At high NOx, there is a preference for the 180 pph injector, however this is more evident at 1000 bar rail pressure. The 149 pph injector is worse across the NOx range, however at the higher rail pressure of 1500 bar, its performance becomes similar to the 165 and 180 pph injectors as NOx increases.

This is an interesting result in that it agrees with the general conclusions from testing these injectors with the production piston. With that piston, the 165 and 180 pph injectors had similar performance also, with a slight preference for 180 pph at high NOx, and 149 pph had the worst BTE across the entire NOx range. The original expectation was that lower cup flow with longer duration would reduce NOx, allowing more advanced SOI and always produce a net BTE benefit. This data strongly indicates an interaction between the plume and bowl that cannot be ignored. The 149 pph injector is simply not a good match for this bowl geometry.

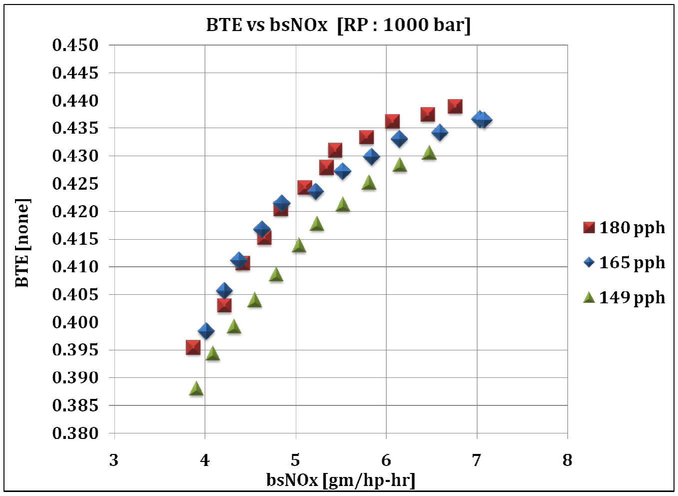

Figure 27 - 19:1 Scaled Production Piston BTE vs bsNOx, 1000 bar Rail Pressure

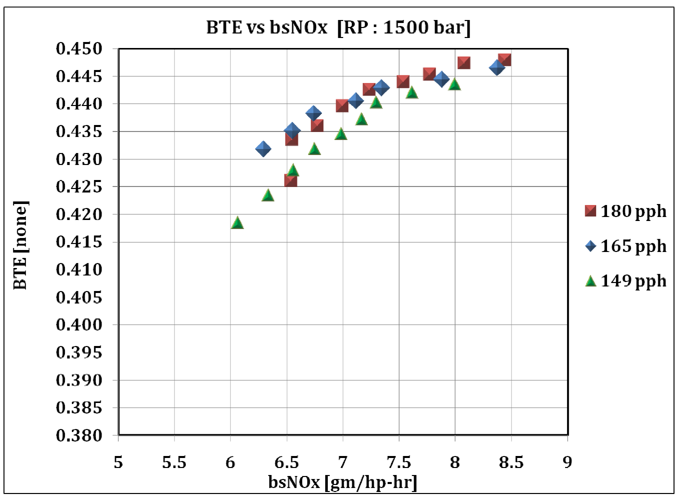

Figure 28 - 19:1 Scaled Production Piston BTE vs bsNOx, 1500 bar Rail Pressure

19:1 Wide/Shallow (Piston CR19-3) Revisited

With the results from piston CR19-7 (scaled production bowl) clearly indicating the significance getting the right match between the injector and piston bowl, it was decided to revisit injector testing with CR19-3 pistons (wide/shallow bowl). In the interest of time and schedule again, only the largest injector, 180 pph was tested, but since it would be compared to the previous data from an optimization with the 149 pph injector, a full DOE and optimization was done with the 180 pph injector.

Table 9 - Engine Configuration for 19:1 CR Piston Test with CR19-3 Wide/Shallow Bowl

Injector Cup Flow	180 pph 149 pph
Piston	CR19-3 (19:1 CR, wide/shallow)
Cylinder Head	Modified, 1.3 Swirl
Turbocharger	“C” Improved turbine efficiency (reduced compressor efficiency)

19:1 Wide/Shallow Test Results

The expectation from KIVA was that the 149 pph injector would be preferred with the wide/shallow bowl and would provide the best BTE in the target NOx range of 4-6 g/hp-hr. This is in fact proven out by the test results in Figure 29, but only by a small margin. Performance with the 180 pph injector is nearly identical to that with the 149 pph.

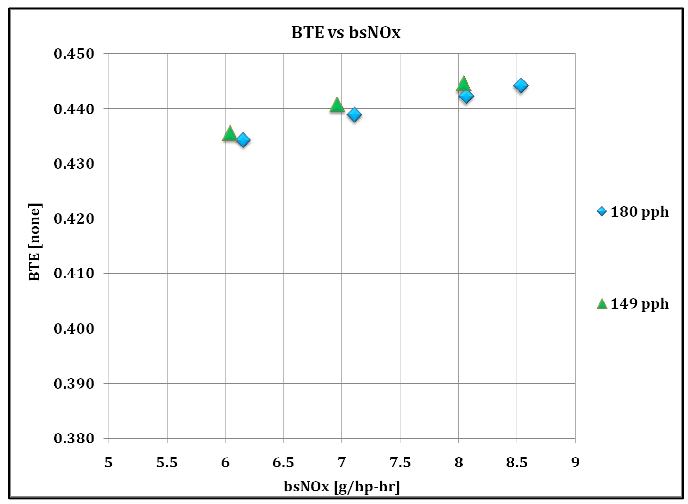

Figure 29 - 19:1 Wide/Shallow Piston, BTE vs bsNOx

Comparing these results to those from the timing swing with the 180 pph injector and scaled production bowl, at 6 g/hp-hr bsNOx, both combinations result in a BTE of about 43.5%. At higher NOx, 180 pph with scaled production bowl had better BTE than 149 pph with the wide/shallow bowl. Additional improvements in BTE with the 180 pph injector and scaled production bowl could be expected with optimization, making a preferred choice over the 149 pph wide/shallow bowl.

Final Conclusions, Piston & Injector Combination

The recommendation is to use the largest injector cup flow tested, 180 pph and 19:1 CR piston with scaled production bowl. To take advantage of increased compression ratio, increased peak cylinder pressure capability is required though. While this conclusion is different than the expectation from KIVA that the 149 pph cup flow with the wide/shallow bowl (CR19-3) would have been preferred, there is uncertainty in KIVA analysis and KIVA predicted only a marginal (1%) difference in performance with these two bowl.

IMPROVED TURBOCHARGER EFFICIENCY

The turbochargers tested were limited to models that are either currently used on other engines or by mixing and matching components available in existing hardware. Developing prototype turbomachinery was outside the scope of this program, however unfortunately this meant compromises had to be made in selecting hardware. Compressors that offered improved efficiency required use of a turbine stage that was not well matched to the target operating conditions and vice versa. Some insight to the importance of turbocharger efficiency can be gained from this testing though.

Test Method

Comparison at the Same Operating Conditions

Performance of the turbochargers was compared at the operating conditions established in the very first optimization. This particular test has proven to be useful for back to back comparisons of hardware, and while further improvement is likely through additional optimization for specific hardware combinations, experience through this program has shown values of A/F, SOI and rail pressure optimized for a specific hardware combination are not too far off from those found during the first optimization.

Engine Configuration

Turbos A, B, & C were all tested with 19:1 compression ratio, wide/shallow bowl pistons (CR19-3), 149 pph injectors, and the modified cylinder head. Turbo D which was selected based on the results from testing of the other three turbos was run with 19:1 compression ratio, scaled production pistons (CR19-7) and 180 pph injectors. This was intended to be the best combination of hardware based on all previous testing.

Table 10 - Engine Configuration for Turbo Comparisons

Configuration for Turbos A, B, C		
Injector Cup Flow	149 pph	
Piston	CR19-3 (19:1 CR, wide/shallow bowl)	
Cylinder Head	Modified, 1.3 Swirl	
Turbocharger	A	Larger turbine for operation without EGR
	B	Improved compressor efficiency (Reduced turbine efficiency)
	C	Improved turbine efficiency (Reduced compressor efficiency)
Configuration for Turbo D		
Injector Cup Flow	180 pph	
Piston	CR19-7 (19:1 CR, scaled production bowl)	
Cylinder Head	Modified, 1.3 Swirl	
Turbocharger	D	Improved compressor efficiency Improved turbine efficiency (Larger flow capacity than desired)

Test Results – Turbocharger Comparison

Turbo B shows the expected improvement in compressor efficiency, and turbo D delivered impressive efficiency up to 77.5% (Figures 30 & 31). A and C had very similar compressor efficiency.

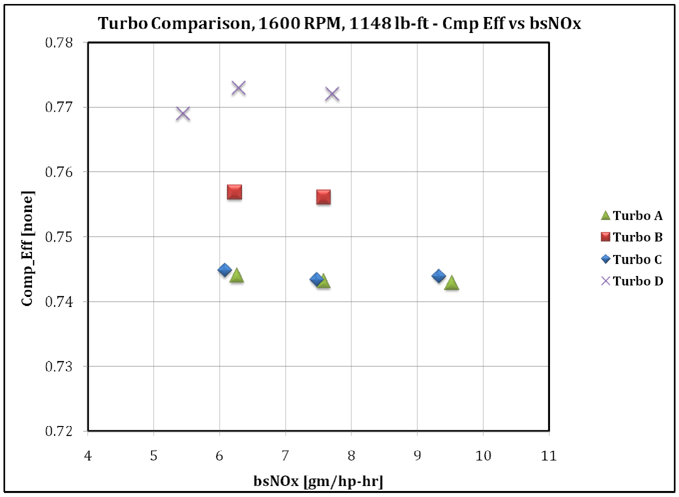

Figure 30 - Compressor Efficiency vs bsNOx 1600 RPM, 1148 lb-ft

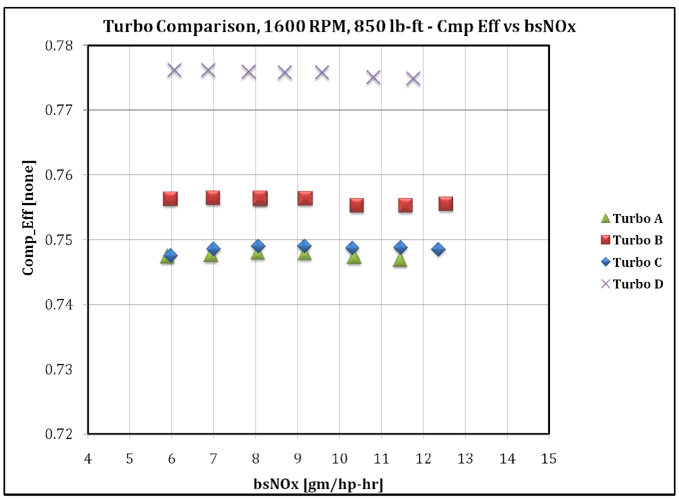

Figure 31 - Compressor Efficiency vs bsNOx 1600 RPM, 850 lb-ft

Turbo C did deliver some improvement in turbine efficiency as expected (Figures 32 & 33). Unfortunately turbo D did not deliver the higher turbine efficiency that was hoped for and the two turbos with the best compressor efficiency had the worst turbine efficiency.

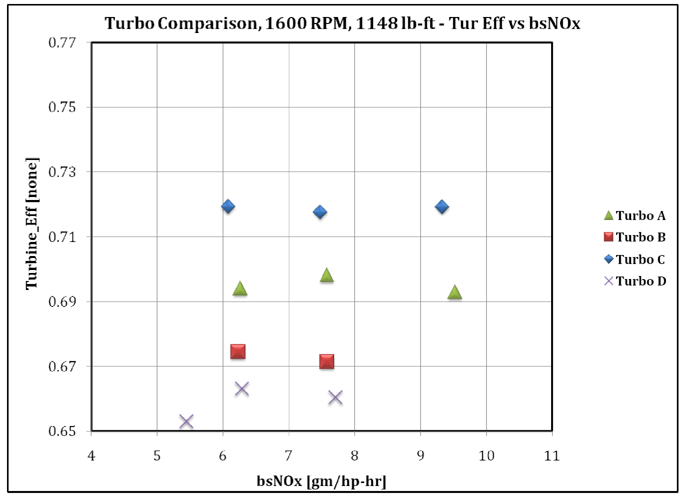

Figure 32 - Turbine Efficiency vs bsNOx 1600 RPM, 1148 lb-ft

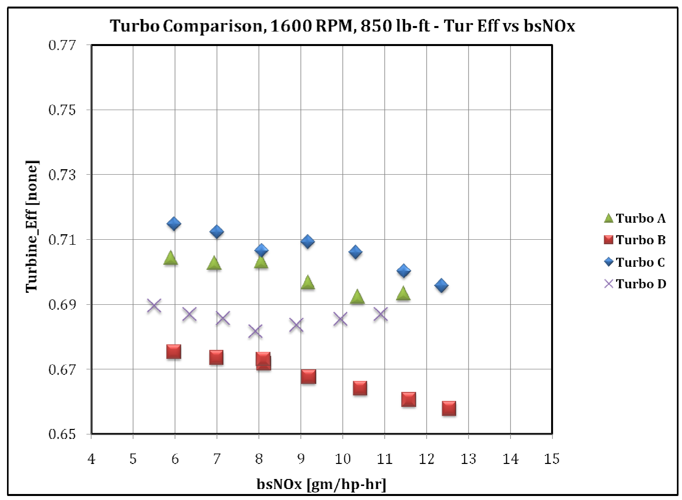

Figure 33 - Turbine Efficiency vs bsNOx 1600 RPM, 850 lb-ft

BTE at full load and part load in Figures 34 and 35 show mixed results. Turbo C which had the same compressor efficiency as Turbo A but with 1 to 2 points better turbine efficiency produced roughly a 1% (0.4 points) improvement in BTE when compared to turbo A. The improved compressor efficiency on turbos B and D was offset by poor turbine efficiency with no improvement in BTE seen.

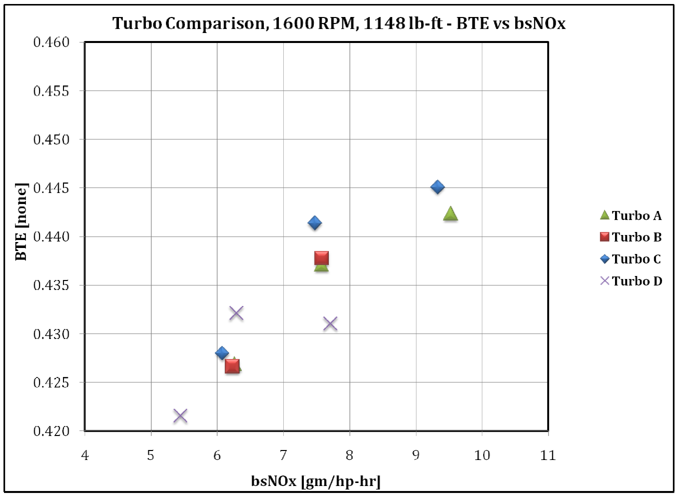

Figure 34 - BTE vs bsNOx
1600 RPM, 1148 lb-ft

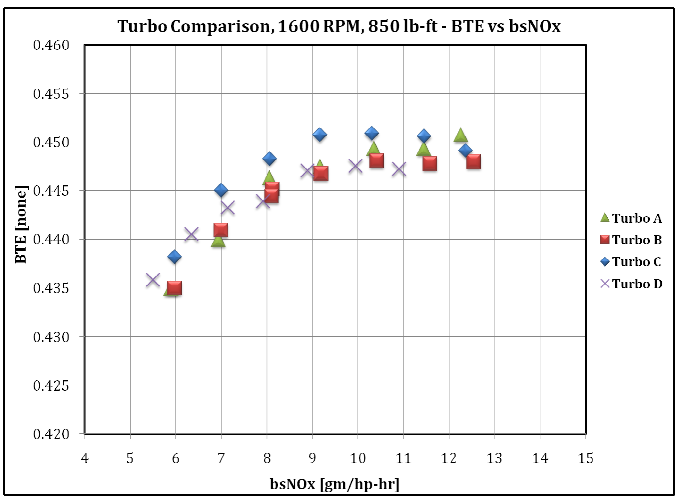

Figure 35 - BTE vs bsNOx
1600 RPM, 850 lb-ft

Final Conclusions – Turbo Testing

Turbocharger efficiency clearly has an impact on brake thermal efficiency and making improvements in either compressor or turbine efficiency was possible. The compromises required to make improvements by mixing and matching available components did not result in any significant BTE gain though. Additional BTE improvement would be expected if a turbocharger were developed specifically for this engine's operating conditions.

FINAL CONCLUSIONS

Little difference was found in engine power output, emissions, or fuel economy when running on JP-8. The engine can easily be converted to run on JP-8 without EGR at an emissions level consistent with 1998 EPA standards. Doing so only requires removing the EGR components and changing to a turbo more suitable for operation without EGR. The only "easy" modification would be to change from the production injectors, but there was no benefit found from that. A 1.6% improvement in brake thermal efficiency can be expected from operation without a diesel particulate filter, which would not be required for this engine to meet 1998 EPA standards.

A 2% improvement in BTE can be realized by increasing compression ratio to 19:1, however to take advantage of this, peak cylinder pressure capability of the engine must be raised above its current production limit. This would require substantial design changes to all structural components (block, head, crankshaft, etc.), so is not a trivial modification. An additional 2% improvement was found through head modifications which improve breathing and reduce swirl.

The highest brake thermal efficiency actually demonstrated at target emissions levels was 43.8% - short of the goal of 48%, but still a significant improvement over the baseline engine. If emissions constraints are completely removed, a BTE of 45.2% is possible.

Efficiency of available turbomachinery was a limiting factor in BTE improvement. Performance was certainly acceptable with available hardware, but today's turbochargers are optimized for engines with EGR and running somewhat off design when applied to an engine without EGR. If further BTE improvements are desired, the next place to look would be development of a turbo specifically for the operation of this engine without EGR.