

Thermodynamic Systems for Tier 2 Bin 2 Diesel Engines

Arvind Suresh, David Langenderfer, Clay Arnett and Michael Ruth
Cummins Inc.

ABSTRACT

Light duty vehicle emission standards are getting more stringent than ever before as stipulated by US EPA Tier 2 Standards and LEV III regulations proposed by CARB. The research in this paper sponsored by US DoE is focused towards developing a Tier 2 Bin 2 Emissions compliant light duty pickup truck with class leading fuel economy targets of 22.4 mpg "City" / 34.3 mpg "Highway". Many advanced technologies comprising both engine and after-treatment systems are essential towards accomplishing this goal. The objective of this paper would be to discuss key engine technology enablers that will help in achieving the target emission levels and fuel economy. Several enabling technologies comprising air-handling, fuel system and base engine design requirements will be discussed in this paper highlighting both experimental and analytical evaluations. Mostly, this paper will focus on steady-state emissions and fuel consumption results that are tied to operating duty cycle with actual vehicle comparisons wherever applicable. GT-Power 1D cycle simulation analysis was done to narrow the turbocharger and air-handling architecture evaluation. The importance of turbocharger matching is critical to achieving both power density and emission goals. This paper highlights the challenges involved in selecting a single stage variable geometry turbocharger capable of delivering a 75hp/liter power density in addition to meeting light duty drive cycle emission requirements. The benefits of low pressure [LP] and high pressure [HP] EGR systems for this light-duty engine architecture will be explored in detail along with outlining the possible strategy of dual loop EGR to reduce overall fuel consumption.

On the combustion system development, this paper will highlight the importance of compression ratio, variable swirl architecture, injector nozzle hardware and injection strategy optimization towards meeting the stringent emissions, fuel consumption and noise targets. Fundamental thermodynamic explanations relating the fuel consumption improvements back to closed cycle and open cycle work are also being presented. Also, the paper will discuss challenges involving unburned HC and CO emissions when operating in premixed/low temperature combustion like regimes involving higher EGR rates and early pilot injection timings. Potential solutions to mitigate such undesired emissions have been investigated both analytically and experimentally.

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INTRODUCTION

Future diesel emission standards applicable to light-duty pickup trucks as mandated by US EPA and CARB are provided in [Table 1](#) [Source: www.dieselnets.com].

In order to meet the stringent tailpipe emissions as shown in [Table 1](#), state-of-the-art engine and aftertreatment technologies will be needed. This paper will discuss and analyze critical engine technologies toward the goal of meeting above standards in addition to delivering class leading fuel economy when compared to a gasoline counterpart. Additionally based on CAFE fuel economy standards for light trucks, it is possible to meet a mandated fuel economy for a vehicle footprint of 65 - 70 square feet.

Table 1. US Light Duty Diesel Emission Standards

Tailpipe Emissions [gm/mile]	Emission Standards [Full Useful Life – 150,000 miles]	
	US EPA – Tier 2 Bin 2	CARB LEV III – SULEV20
NOx + NMOG	0.03	0.02
NOx	0.02	-
PM	0.01	0.01
CO	2.1	1.0
HCHO	0.004	0.004

BACKGROUND AND HARDWARE

The baseline engine architecture involves a Euro IV emissions compliant inline 4 cylinder 2.8 liter ISF engine. The engine specifications are highlighted in [Table 2](#).

Table 2. Base Engine Architecture and Specifications

Engine Displacement	2800 cc
Cylinders	Inline- 4, Diesel
Peak Power	160 hp at 3600 rpm
Peak Torque	265 lb.ft at 1800 rpm
Stroke	100 mm
Bore	94 mm
Connecting Rod Length	158 mm
Compression ratio	16.9:1
Number of Valves	4
Emission Control	High Pressure Cooled EGR with diesel oxidation catalyst
Turbocharger	Single Stage, Wastegate
Fuel System	Bosch CRI2.2 – 1600 bar
Injectors [holes x SA x flow]	7 x 145° x 480 cc/30sec

The engine test bed evaluation and development was simultaneously complemented with vehicle evaluation. A 2010 model year Nissan Titan crew cab 2 × 4 pickup truck as shown in [Figure 1](#) below was used for the baseline vehicle evaluation. The truck's 5.6L V8 - flex fuel engine was replaced with the 2.8L inline 4 cylinder diesel engine as specified above.



Figure 1. Test Vehicle

Methodology and Approach

The test vehicle upfitted with a Euro IV compliant diesel engine was recalibrated for an automatic transmission. Then the vehicle was tested on a chassis dynamometer to establish the baseline emissions, performance and fuel economy for FTP-75, LA-4 and HFET cycles. From this data, it was possible to characterize the duty cycles through steady-state modal points matching the cycle work of transient data. This

approach is very valuable and expedites the process of engine architecture screening and evaluation during the early development stages [1]. [Figure 2](#) shows actual duty cycle data representing LA-4 and HFET cycles in engine speed and torque space from the chassis test results. The dotted line represents the baseline engine torque curve. The duty cycles represent a vehicle test weight of 5500 lbs and resulted in cycle work of 5.1 bhp-hr and 5.5 bhp-hr for LA-4 and HFET cycles respectively. The 10 modal steady-state points capture the cycle work reasonably well as a function of % duty cycle time.

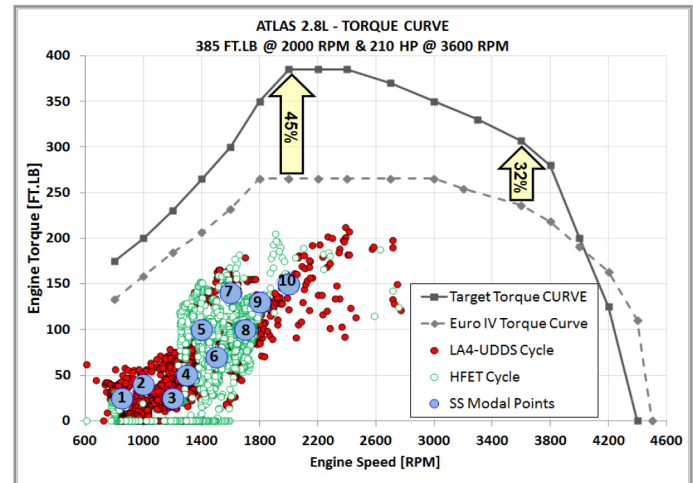


Figure 2. Duty Cycle, Modal Points and Torque Curve

The solid line in [Figure 2](#) represents the target torque curve for development engine designed to meet Tier 2 Bin 2 emission levels. Further details around power density development will be discussed later. Chassis lab emissions and performance results for the test vehicle with base engine configuration for different duty cycles are reported in [Table 3](#).

Table 3. Chassis Test Results with Base Engine

Emissions and Fuel Economy	FTP 75	LA-4	HFET
NO _x [gm/mile]	1.89	1.82	1.30
NMHC [gm/mile]	0.03	0.03	0.10
CO [gm/mile]	0.07	0.17	0.83
PM [gm/mile]	0.10	0.09	0.06
FE [mpg]	23.4	23.8	33.0

By comparing the emissions from the baseline engine and test vehicle in [Table 3](#) to actual Tier 2 Bin 2 targets shown in [Table 1](#), it can be seen very clearly that a significant reduction in emission levels is required by more than 99%, 90% and 67% for NO_x, PM and NMHCs respectively. This calls for significant advancements in both engine and aftertreatment technologies moving forward.

Base Engine Architecture Evaluation

The base engine architecture evaluation was carried out in a steady-state dynamometer test bed to prove the concept. The existing fuel system was upgraded to a state-of-the-art piezo common rail fuel system with 2000 bar capability. The advantages of a piezo common rail fuel system can be attributed towards improved combustion noise characteristics, reduced emissions and improved drivability as cited in [2] and [3]. The injectors were fitted with a nozzle configuration comprising of 8 holes, 146 deg SA and 410 cc/30 sec hydraulic cup flow. A general schematic of improved engine architecture aimed towards meeting Tier 2 Bin 2 goals is shown in [Figure 3](#).

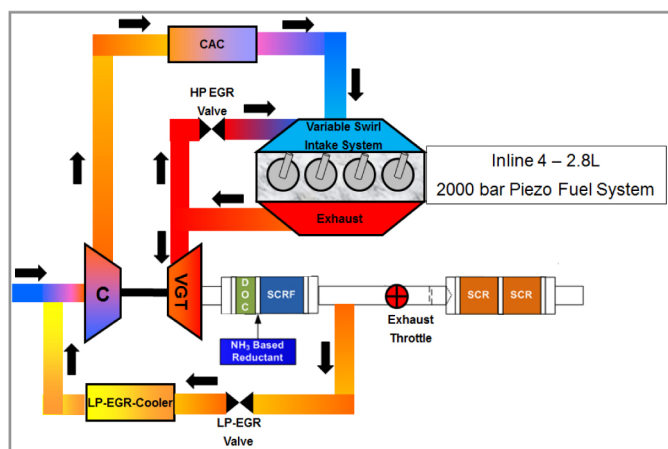


Figure 3. Concept Engine Architecture Layout

Air-Handling Architecture

The concept engine architecture layout as shown in [Figure 3](#) has significant improvements when compared to the base engine configuration. The single stage turbocharger has been upgraded to an efficient variable geometry turbine and a larger compressor wheel/housing to meet power density requirements as well as reduced overall fuel consumption. The Exhaust Gas Recirculation [EGR] circuit shown in [Figure 3](#) has both High Pressure [HP] and Low Pressure [LP] loops. The HP EGR loop features recirculation of exhaust from exhaust manifold to intake manifold and the flow is controlled by means of a HP EGR valve. It is important to note that there is no physical EGR cooler in the HP loop. The LP EGR loop runs from downstream of an integrated DOC-SCRF after-treatment system [26]. An exhaust throttle and a LP EGR valve work together to control the EGR mass flow

rate in the LP loop. Additionally, an EGR cooler in the LP loop helps to reduce the gas temperature before mixing with fresh air upstream of turbo compressor.

Low Pressure EGR

It is important to understand the benefits of LP EGR when compared to traditional HP EGR systems. The duty cycle and emissions requirements dictate the primary choice of selection. For the Tier 2 Bin 2 chassis certification emission targets, higher levels of EGR rate are needed. With increase in EGR rates, oxygen concentration in the intake manifold decreases resulting in reduced engine out NO_x emissions. Typical steady state modal analysis represented in [Figure 4](#) shows the advantage of LP EGR with respect to achieving even lower intake oxygen concentration values when compared to HP EGR system [3] [4] [7]. This analysis was done using GTPower DI-Jet combustion model with different air-handling actuator settings. The results were then optimized to achieve the desired A:F ratio and intake oxygen concentration values. The differences in potential to reduce pumping work as a function of intake oxygen concentration can be clearly seen between HP and LP loop. For a constant A:F ratio, with decrease in intake oxygen concentration, the LP EGR loop has a clear advantage of reduced pumping losses when compared to HP EGR loop.

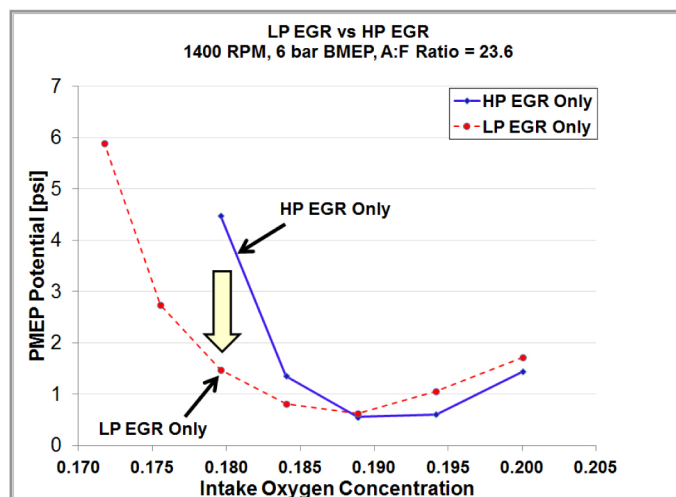


Figure 4. Pumping Loss Potential vs Intake Oxygen Concentration

[Figure 5](#) shows experimental data for a light duty steady state modal point at 1400 RPM, 6 bar BMEP as represented by Mode 5 in [Figure 2](#). The dotted line in [Figure 5](#) represents engine out smoke levels for a constant engine out BSNO_x value of 0.9 g/hp.hr. A significant magnitude reduction of smoke levels can be observed going from HP to LP EGR loop. By improving EGR and fresh air mixing in HP loop, this difference in gap can be reduced. However, the added benefit of fresh air and EGR mixing still exists in the LP loop due to the turbocharger compressor along with cooler charge temperatures in the intake manifold. The cooler charge

temperature inherently increases the charge air density and helps to reduce fuel consumption as shown by the solid line in Figure 5. Further optimizing the LP EGR combustion recipe, it was possible to reduce fuel consumption by approximately 16% when compared to the optimal HP EGR solution.

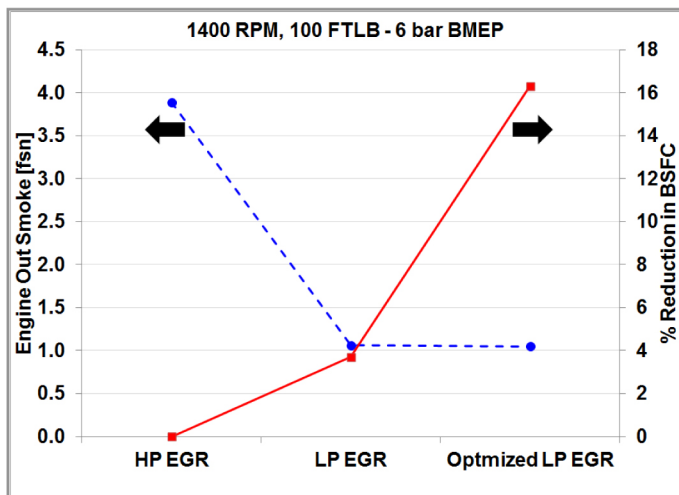


Figure 5. HP EGR vs LP EGR at Mode 5

The LP EGR solution resulted in improving closed cycle combustion work [IMEP_g] when compared to the HP EGR combustion recipe. Further optimizing the LP EGR solution, it was possible to significantly reduce the pumping losses due to increased turbine efficiency and thus realize a higher net indicated work [IMEP_n]. For the Mode 5 condition represented in Figure 5, pumping work was almost cut by 50% resulting in overall reduced fuel consumption. Figure 6 below illustrates the fundamental thermodynamic chart in Log P - Log V coordinates highlighting the reduction in pumping losses [PMEP] with the optimized LP EGR solution at Mode 5.

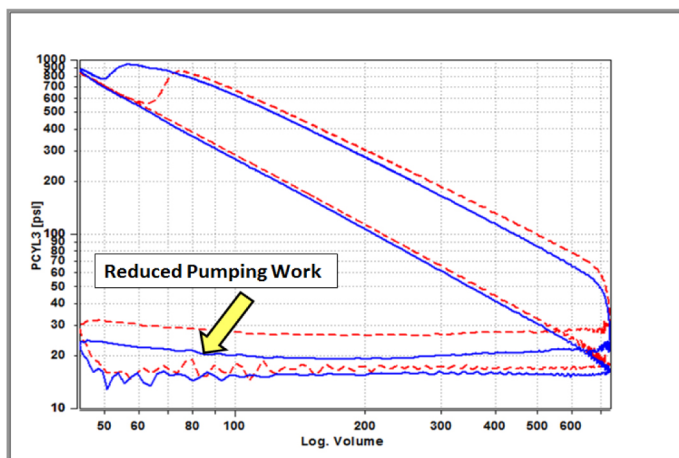


Figure 6. Comparison of Log P- Log V Plot of LP EGR vs Optimized LP EGR

The dotted line in Figure 6 represents the LP EGR solution and the solid line represents the optimized combustion recipe with LP EGR. Air-handling and combustion system optimization provides additional control levers to minimize pumping work and reduce BSFC in order to meet the specified emission targets.

Clearly LP EGR has a distinct benefit at Mode 5 based on the results as discussed above. However, there is also an associated risk of condensation in the charge air cooler when running in LP loop due to cooler temperatures and the presence of exhaust gas in charge air cooler. Hence, careful temperature management of charge air cooler cooling loop is required to avoid condensation [5]. One possible way to achieve this applying the laws of thermodynamics would be to run a higher sink temperature [cooling fluid] at the charge air cooler outlet to maintain the mixture or gas-fresh air above a certain dew point value. The alternative option would be to explore a dual loop strategy involving both HP and LP EGR loops simultaneously to avoid condensation by limiting EGR mass flow rates in the LP loop. It would be also valuable and interesting to explore the turbine efficiency and its impact on fuel consumption when operating in dual loop.

The engine architecture layout in Figure 3 shows an exhaust throttle. Conventional HP EGR systems typically will have an intake air throttle as an additional means to increase EGR rates and achieve lower NO_x emissions. Preliminary analysis was carried out using GTPower to understand the benefits of intake throttle over exhaust throttle during this study. The analysis showed that compressor mass flows decreased substantially when using intake throttle to drive LP EGR resulting in higher fuel consumption. The analysis using exhaust throttle resulted in higher compressor mass flows leading to higher turbo machinery efficiency & reduced fuel consumption. Based on these findings and with the intention of reducing controls complexity, exhaust throttle was preferred over an intake throttle for this architecture.

Dual Loop EGR

In order to meet Tier 2 Bin 2 emission levels, higher % EGR rates will be required during the chassis certification drive cycles. The LP EGR certainly provides advantages as discussed earlier in this paper. This paper also addresses the need for dual loop EGR in certain areas of the drive cycle to help improve fuel economy [6][7][9]. Figure 7 represents a steady state modal point Mode 4 [1300 RPM, 50 FT.LB] as illustrated in Figure 2 and shows contours of BSNO_x, reduction in BSFC, increase in turbine efficiency and intake manifold temperature as a function of HP EGR valve position and exhaust throttle position. The LP EGR valve was fixed at 100% wide open and the VGT position was also fixed at 70% closed for this particular study.

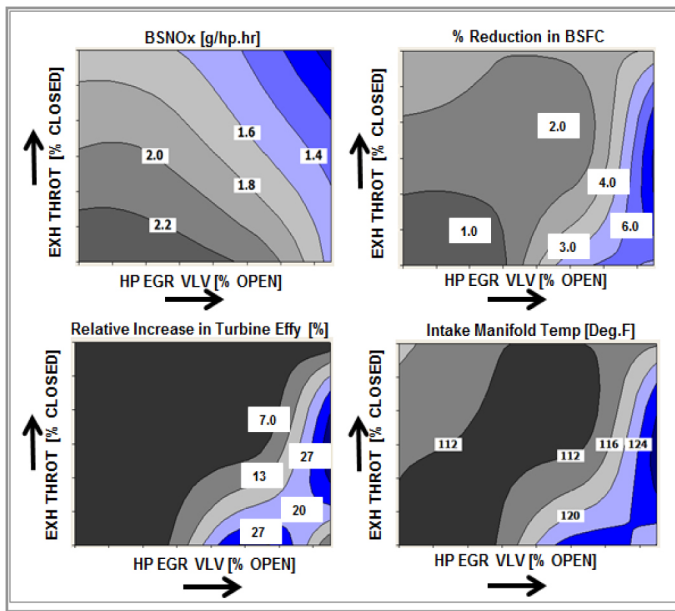


Figure 7. Dual Loop EGR Results at Mode 3 [1300 RPM, 50 FT.LB]

A design of experiments study was performed by varying the HP EGR valve position and exhaust throttle position. By closing the exhaust throttle and by opening the HP EGR valve, higher EGR rates can be achieved simultaneously in dual loops. The contours of BSNOx reflect the dual loop operation and the reduction in BSFC contours show the improvement in dual loop operation when compared to the reference LP EGR only mode of operation. The improvement in BSFC is a result of increase in turbine efficiency as shown by the contour plot in Figure 7.

Due to dual loop operation and the absence of EGR cooler in the HP loop, the intake manifold temperature as shown in the contour plot will be hotter when compared to reference LP only mode of operation. Typically, an increase in charge temperature results in an increase in fuel consumption due to reduced charge density. However at this particular operating mode, the increase in turbine efficiency with dual loop operation is still significant enough to improve the overall fuel consumption even at higher charge temperatures.

An interesting analysis with dual loop architecture was carried out at 2200 RPM, 190 FT.LB to simulate the effects of higher BMEP levels during engine operation. The objective was to understand the impact of dual loop with and without the presence of EGR cooler [4]. Figure 8 shows the plot of predicted fuel specific NOx [FSNOx] vs relative change in fuel consumption at steady-state operation. Cooled EGR in both loops resulted in best lowest fuel consumption over the range of engine out NOx levels. The reason being, cooler charge temperature helps to increase charge density resulting in higher closed cycle combustion work [IMEP_g].

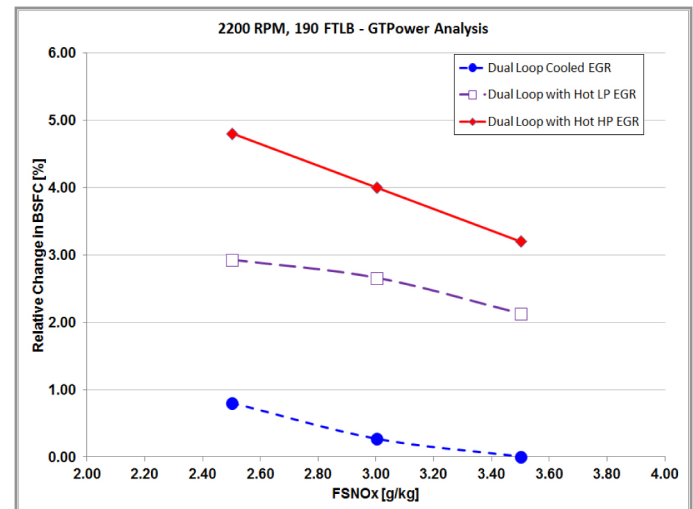


Figure 8. Predicted FSNOx vs Normalized BSFC - GTPower

Running hot EGR in LP loop bypassing the LP EGR cooler and running cooled EGR in HP loop results in approximately 2% increase in fuel consumption when compared to the earlier case for a FSNOx value of 2.5 g/kg. Finally, as outlined in architecture layout per Figure 3, running cooled LP EGR and hot HP EGR results in approximately 4% increase in fuel consumption when compared to cooled EGR architecture in dual loops for a FSNOx value of 2.5 g/kg.

Hence the dynamics of dual loop EGR varies over the engine operating map driven by duty cycle and emission requirements. There is no single strategy that will work consistently across the engine map and hence several iterations of optimization will be required both analytically and experimentally to arrive at the best possible solution. Several factors like air-handling thermodynamics, turbocharger selection, emission requirements during drive cycle and cold start have a significant effect on calibration development and optimization.

Hot EGR using HP loop will help provide combustion stability during cold starts and avoid misfire issue [2]. Emissions control during cold bag involving transient engine warm-up from idle would be a challenging task and the short routing length in the HP loop would further help in this tuning process without compromising transient engine response behavior.

Figure 9 represents a preliminary overview on the EGR split strategy based on analysis results from modal points. Primarily, the emissions certification cycle and drive cycle will be controlled in dual loop mode, idle and warm-up regions would be HP hot EGR. LP EGR can be used in mid to high load areas maintaining higher turbine efficiency. However near the torque curve region, HP EGR will be used to reduce engine out NOx emissions. LP EGR can still be an option near the torque curve, but needs to be carefully

managed without violating the thermal limits of the turbocharger and charge air cooler components.

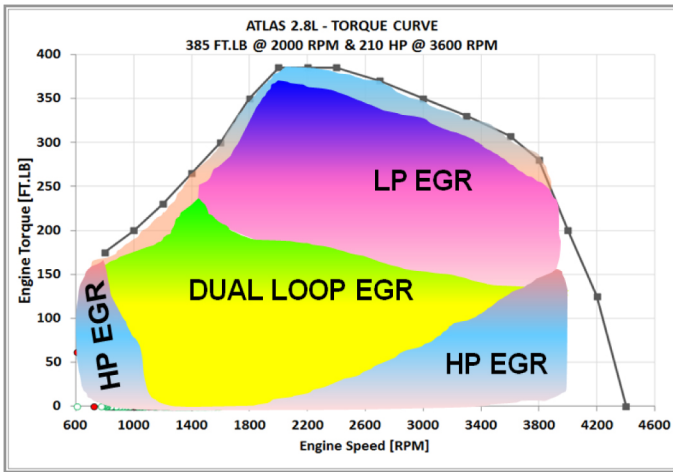


Figure 9. Schematic of EGR Split Strategy

EGR Flow Estimation

When EGR is operational in dual loops, it is possible to use additional set of physical sensors that either directly measure the flow or measure the pressure drop across a venturi or an orifice. Theoretically, it is possible to estimate the EGR flow rate in both loops by applying the 1st law of thermodynamics energy balance with certain assumptions [6]. Assuming a perfect EGR mixer as the control volume, HP EGR gas, LP EGR gas and fresh air enters the mixer and the uniformly mixed charge leaves the mixer. Detailed equations outlining the approach are described under the [Appendix](#) section. The final expression for EGR mass flow rate in each of the loops can be expressed as,

$$\dot{m}_{hpegr} = \frac{\dot{m}_{air}(T_{ch} - T_{air}) - \dot{m}_{egr}(T_{lpegr} - T_{ch})}{(T_{hpegr} - T_{lpegr})} \quad (1)$$

$$\dot{m}_{lpegr} = \frac{\dot{m}_{air}(T_{ch} - T_{air}) - \dot{m}_{egr}(T_{hpegr} - T_{ch})}{(T_{lpegr} - T_{hpegr})} \quad (2)$$

Equations (1) and (2) can be used for theoretically estimating respective EGR mass flow rates when operating in a dual loop fashion. The accuracy of these estimates need to be verified on engine with real measurements.

Turbocharger Selection

As both LP EGR and dual loop EGR tend to increase turbine efficiency resulting in fuel consumption improvement, turbocharger selection plays a vital role in the air-handling architecture. In order to achieve a good transient low end torque response as well as meet emissions and power density targets, a very efficient turbocharger is a critical element. Achieving a good match on both compressor and turbine selection through a single stage configuration is a

challenging task. The details behind the turbo trim selection process are outside the scope of this paper and cited in references [5] [8]. However, several analysis iterations were involved in down selecting the appropriate compressor and turbine hardware for engine evaluation. Figure 10 shows the peak torque and rated power conditions from engine testing on the final compressor hardware selection. The heart of the compressor map shows a peak efficiency of 74% based on the flow stand mapped data. The engine data demonstrated 62% efficiency at peak torque and 65% efficiency at peak power conditions. The selected compressor has a wider map width and provides excellent surge margin as well as increased choke flow capabilities.

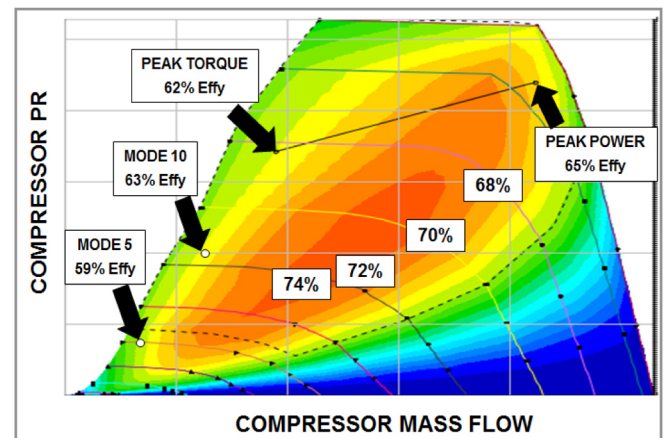


Figure 10. Compressor Map with Engine Data

The modal points Mode 5 and Mode 10 representative of duty cycle are also included in the compressor map. Ideally, it would be desired to have these points operate in the heart of the map. But with single stage architecture, it was challenging to achieve this especially having to meet power density and reduce system complexity.

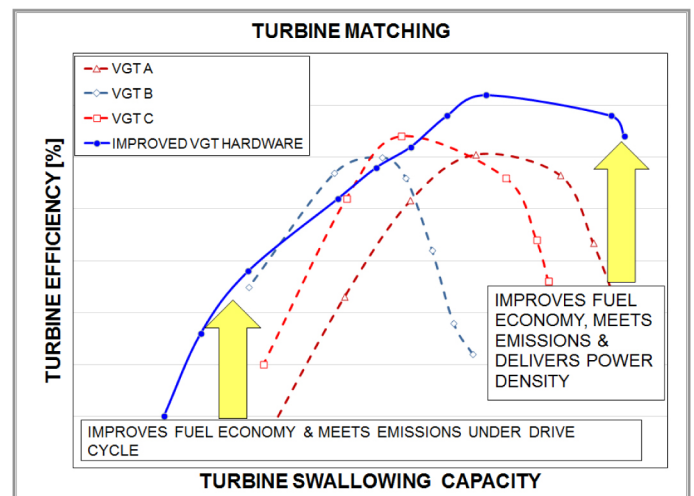


Figure 11. Turbine Efficiency vs Swallowing Capacity