HIGH-EFFICIENCY VCR ENGINE
with
VARIABLE VALVE ACTUATION AND NEW SUPERCHARGING TECHNOLOGY

FINAL REPORT

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1. EXECUTIVE SUMMARY

Program Objectives and Accomplishments

Higher efficiency engines are needed for full size pickup trucks, utility vans and medium duty trucks. The primary objective of the program was to demonstrate the potential for improving the fuel economy of a full-size pickup truck by 40 percent while producing at least 281 horsepower. Commercialization requirements include:

1. High torque values at low vehicle speeds: This capability is needed for accelerating a heavy vehicle from a standstill, accelerating uphill, towing or any combination of these factors.

2. High power output: This capability is needed for accelerating onto a highway or passing at higher speeds. The vehicle may be carrying a heavy load, going uphill and/or towing.

3. Compliance with tailpipe emission regulations: This requirement must be met, and is difficult or costly to attain with diesel engine technology.

4. Low fuel consumption: This capability is needed both for reducing operating costs and to reduce greenhouse gas (GHG) vehicle emissions. The capability is also essential for complying with fuel economy standards.

5. Affordability: The powertrain cost must be consistent with mass market pricing.

6. Engine size and weight: The engine must fit in the existing engine bays. Engine weight must also be minimized.

7. Favorable noise vibration and harshness (NVH): The VCR must not present unique NVH challenges.

The current program was successful. The efficiency target was achieved and the engine produced 300 horsepower. Figure 1 is reproduced from the original grant proposal. Project accomplishments have been added to the figure and include **Achieved** ● and **Projected** ■ brake specific fuel consumption (BSFC) values. The fuel economy improvement for a Ford F150 full-size pickup truck equipped with the VCR engine was projected to be between 35 and 43 percent.

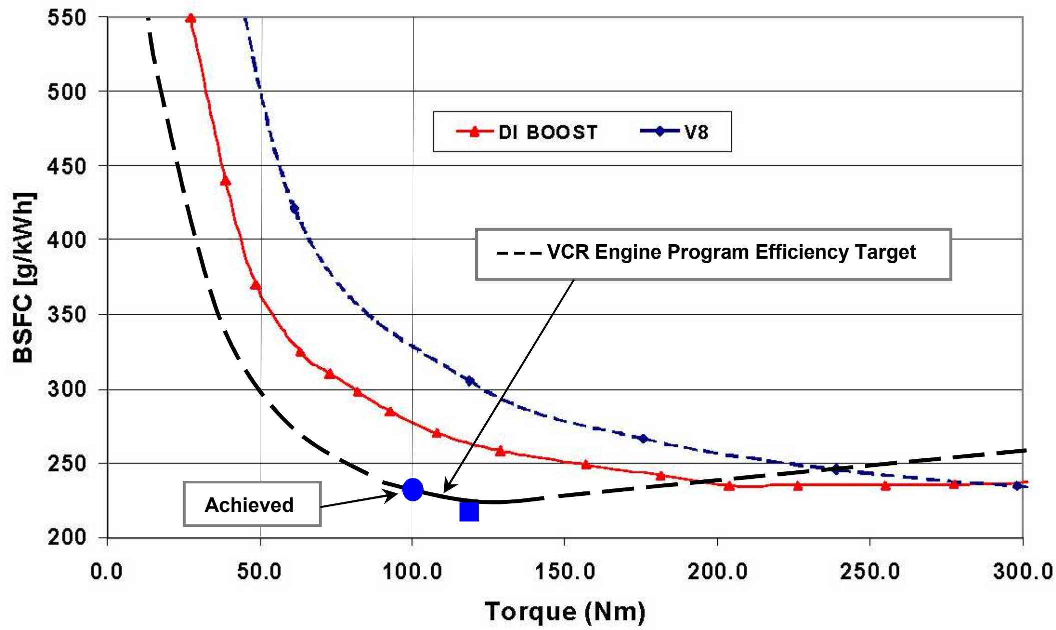


Figure 1. Fuel efficiency comparison of a 5.7L V8, a 3.6L Turbo-DI V6, and the 2.2L VCR 4-cylinder program engine efficiency target curve. **Achieved** ● and **Projected** ■ BSFC values attained under the current program.

Technological Approach

The approach for attaining high mileage was to combine aggressive engine down-sizing with high-efficiency Atkinson Cycle technology. A challenge is attaining power and torque requirements from the down-sized Atkinson Cycle engine. Technologies employed for attaining high power and torque requirements include:

- Cam profile switching for switching from Atkinson Cycle cams to boosted cycle cams
- Advanced boosting to increase engine power and torque, and
- Variable compression ratio to prevent detonation (engine knock).

Engine specifications are shown in Table 1. The engine was gasoline fueled and employed lambda 1 stoichiometric fueling for effective reduction of tail pipe emissions with proven 3-way catalytic converter technology.

TABLE 1
VCR Engine Specifications

Displacement	2.4 L
Bore	88.5 mm
Stroke	97.6 mm Nominal, minor shift with VCR
Bore center	96.0 mm
Bore offset	10.5 mm
Rod center length	166.0 mm
Main bearing diam	55.0 mm
Connecting rod bearing diam	50.8 mm
Compression ratio	17.6 Maximum
	8.2 Minimum
Cylinder head	GM 2.5L MY 2014 LKW
	DOHC
	Eaton cam profile switching, intake
	Dual cam phasers
Fuel injectors	PFI: Bosch EV14
Fuel	93 octane, pump grade with lab cert heating value
Oil	10w 30 Mobil 1
Ignition coils	AEM 103 MJ coil near plug
EGR	Cooled external loop
Oil pump	Barnes Systems, cog belt drive, wet sump
Water pump	Electric, shop driven

1A. VCR MECHANISM SELECTION

VCR mechanism options were evaluated to determine which VCR mechanism should be developed under the current program. General findings from the evaluation are as follows:

Crankshaft mounted in a crankshaft cradle

With this mechanism the crankshaft is mounted in a crankshaft cradle that can pivot +/- 15 degrees (shown in Figure 2). The crankshaft axis of rotation is offset from the cradle pivot axis so that pivoting the cradle moves the crankshaft closer or farther from the cylinder head for adjustment of compression ratio.

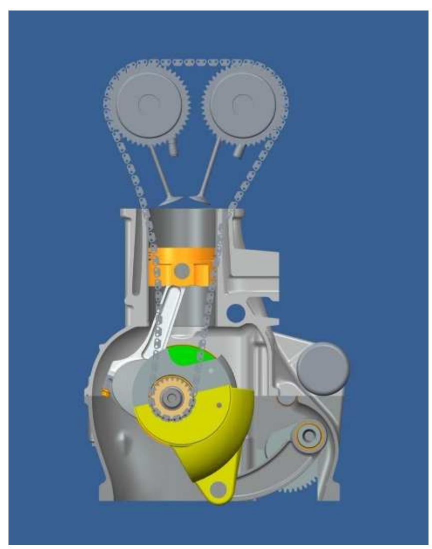


Figure 2. VCR mechanism having a crankshaft cradle

Several of these engines have been built by Envera, FEV and Caterpillar. The Envera VCR engines of this type (for which we have data) demonstrated VCR mechanism robustness and functionality.

A limitation of the mechanism is that the crankshaft axis changes location with respect to the transmission axis. A power takeoff coupling is needed to transfer power from the crankshaft to the transmission. Preferably a VCR mechanism can be selected for development that does not require a power takeoff coupling.

Adjustable pistons and connecting rods

With these mechanisms compression ratio is adjusted by changing the distance between the connecting rod big end bearing center and the piston flame face, either by changing the length of the connecting rod or by changing the compression height of the piston. An advantage of these mechanisms is potential for limited change of the core engine and a lower cost of implementation.

A limitation with these mechanisms is that the range of compression ratio adjustment is too small. For example, a VCR travel distance of 8.0 mm would be needed for the 2.5L Toyota TNGA engine (SAE Paper 2017-01-1021) to adjust compression ratio from 16.5:1 to 8.0:1 (6.9 mm is needed to adjust compression ratio from 14:1 to 8.0:1). Adjustable piston and connecting rod mechanisms are understood to provide only about half of this travel distance. For a number of automotive and R&D companies a development objective has been to identify a small VCR travel range (plus other engine settings) that will provide 50 to 60 percent of the benefit of a full range system. The alternative development objective is to identify settings that provide a 10 percent benefit on top of the full range VCR mechanism.

A second limitation with these mechanisms is increased mass. An engine development objective is to minimize piston and connecting rod mass.

Some of these mechanisms have only two settings, high and low compression ratio. A fully variable compression ratio adjustment capability is desirable.

Infiniti VC-T

The Infiniti VC-T variable compression ratio mechanism employs a large rocker arm (shown in Figure 3). A crankshaft journal bearing is located in the middle of the rocker arm. The piston is connected to one end of the rocker arm with a first connecting rod, and an eccentric control shaft is connected to the other end of the rocker arm with a second connecting rod. The eccentric control shaft can be rotated up to about 70 degrees with an electric motor. Rotating the eccentric control shaft moves the anchorage location for the second connecting rod resulting in a change of compression ratio. Compression ratio can be adjusted from 14:1 to 8:1.

The engine has been advertised as having low internal friction losses because balance shafts are not needed and because there is less side loading of the piston on the cylinder wall. Elimination of balance shafts does provide a friction reduction benefit however the benefit is relatively small when compared to new engine designs where friction from the balancer system has been minimized. Of note, smaller 4-cylinder engines generally don't have balance shafts, such as the Toyota 1.8L Prius engine. The 2L Infiniti VC-T also features an active engine mount vibration damping system (ATR), which is advertised by Infiniti as being a world's first in this class of cars. One question is why ATR is used with the VC-T engine? If the VC-T largely eliminates vibration it wouldn't seem necessary to use ATR. Another question is could ATR enable balance shafts to be eliminated from conventional 2L or larger engines?

The VC-T engine also has a more upright connecting rod, which is beneficial for reducing piston side load friction on the cylinder bore. New engines such as the Toyota Prius 1.8L and TNGA 2.5L have an offset crankshaft axis to provide a more upright connecting rod at peak cylinder pressure. The benefit that the VC-T achieves over engines having an offset crankshaft axis may be small. A major source of piston friction is from the ring pack. Ring pack friction does not benefit from a more upright connecting rod.

The Infiniti VC-T does have low-friction cylinder bores featuring a mirror bore finish. A coating is applied by plasma jet to the bores, which is then hardened and honed. Infiniti states that the finish contributes to a

44 percent reduction in cylinder friction. This surface finish can also be applied to other types of VCR engines.

The VCR linkage mechanism itself has higher friction levels than a standard cranktrain. Engine friction values for the VC-T engine are compared to that of a V6 engine in Figure 13 of SAE paper 2018-01-00371. At 2000 rpm the engines have almost identical friction toque (within 3 Nm), but the V6 is much larger and has a displacement of 3.5L in comparison to the smaller 2L VC-T. Accordingly, the friction mean effective pressure (FMEP) of the VC-T is about 1.75 times greater than that of the V6 (= 3.5L/2L). The high FMEP value detracts from the efficiency gains that are possible with VCR. Figure 14 of the SAE paper shows a peak efficiency value of 230 g/kWh for the VC-T (in very small print). With lower friction values lower fuel consumption values would be attained. The 40 percent brake thermal efficiency Toyota 2.5L TNGA engine was used as a benchmark for the current program (SAE paper 2017-01-1021). A goal of the current program was to develop a VCR mechanism that does not have high FMEP values that compromise efficiency gains attainable with VCR technology.

Concerns with the Infiniti VCR mechanism include engine friction relative to the higher baseline of newer engines; difficult engine assembly and engine cost; tolerance stack and if that is a reason for not having a higher maximum CR; limited engine speed capability, and engine size.

Figure 3. The Infiniti VC-T VCR engine.

Dual eccentric control shafts

With these VCR mechanisms the cylinder liners are formed in a cylinder jug casting (shown in Figure 4). The cylinder head is mounted on top of the cylinder jug. The cylinder jug is retained in the crankcase by two control shafts, one on either side of the cylinder bores. The control shafts can rotate in bearings formed in the cylinder jug casting and in the crankcase. The shafts are synchronized with gearing to rotate in the same direction. The journals formed on the control shafts for the crankcase are offset from the journals formed on the control shafts for the cylinder jug. Because of the offset journals, rotating the control shafts moves the cylinder jug relative to the crankcase resulting in a change of compression ratio. Any desirable range of compression ratio can be attained by specifying the offset distance between the eccentrics.

An advantage of this type of VCR mechanism is that a power takeoff coupling is not needed. Another advantage is that current production crankshaft, connecting rod and pistons can be used with no or only minimal change. Best practices for minimizing cranktrain mass and friction may be applied without restriction. Low friction values, high load capacities and high engine speeds are attainable. Additionally, there is no or only minimal increase of engine size this type of VCR mechanism.

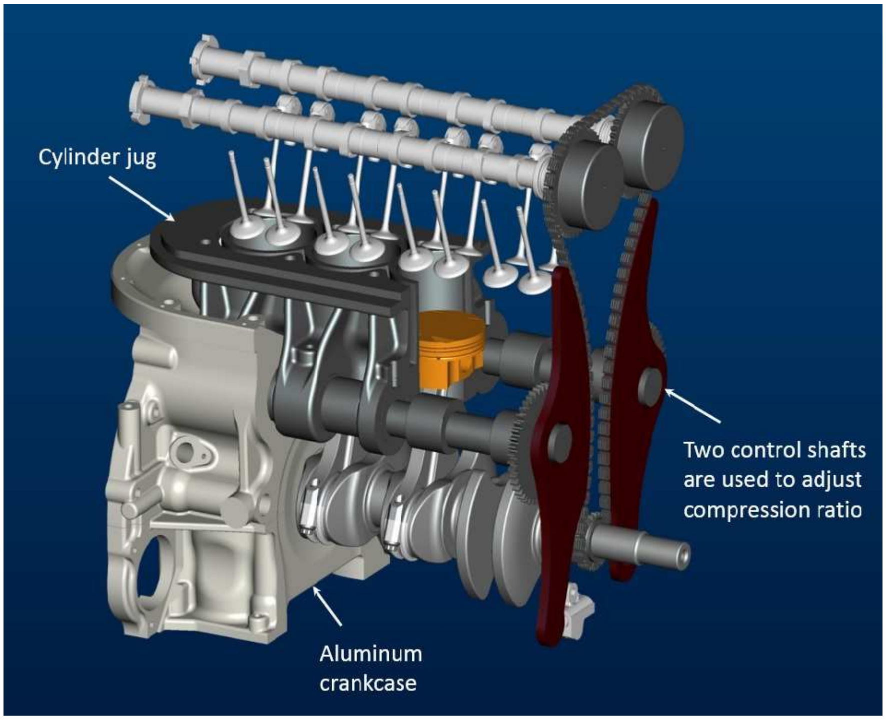


Figure 4. VCR mechanism having two eccentric control shafts.

Concerns with this type of mechanism include assembly of the cylinder jug casting to the crankcase with the control shafts; sealing between the crankcase and cylinder head assembly and the VCR actuator mechanism. Design solutions for the control shaft assembly and engine sealing are unlikely to compromise engine performance or efficiency. Actuator power consumption will detract from net engine efficiency. Data from earlier VCR engine programs and the current program indicates that the net loss of efficiency attributable to the VCR actuator will be relatively small.

VCR Mechanism selected for further development

The dual eccentric control shaft VCR mechanism was selected for development under the current program for the following reasons:

- A large enough CR range is provided for optimization of engine efficiency and performance
- Cranktrain friction and mass are not adversely effected
- Optimized cranktrain hardware can be carried over from non-VCR engines
- A power takeoff coupling is not needed
- The compression ratio is fully variable between minimum and maximum CR settings
- There is no or only minimal increase of engine size
- Outstanding design challenges do not appear to require large development undertakings
- Low cost manufacture / Low complexity system
- Durable and reliable structure

Valuable input and review on which VCR mechanism type to pursue was provided by automobile manufacturers involved with VCR development, DOE and NETL

1B. DUAL CONTROL SHAFT VCR ENGINE

The dual control shaft VCR engine shown in Figures 5 includes two control shafts that rotate 180 degrees in the same direction to adjust compression ratio from 17.6 to 8.2. Timing gears are used for rotation of the two control shafts in the same direction. The control shafts include two sets of journals. The first set of journals are concentric with the timing gears and are mounted in conventional bushings in the crankcase. The second set of journals are eccentrically located on the control shaft. The eccentricity is 4.0 mm for providing an 8 mm change of cylinder head height relative to the crankshaft. The eccentric journals are mounted in eccentric bushings. The eccentric bushings are mounted in the cylinder jug.

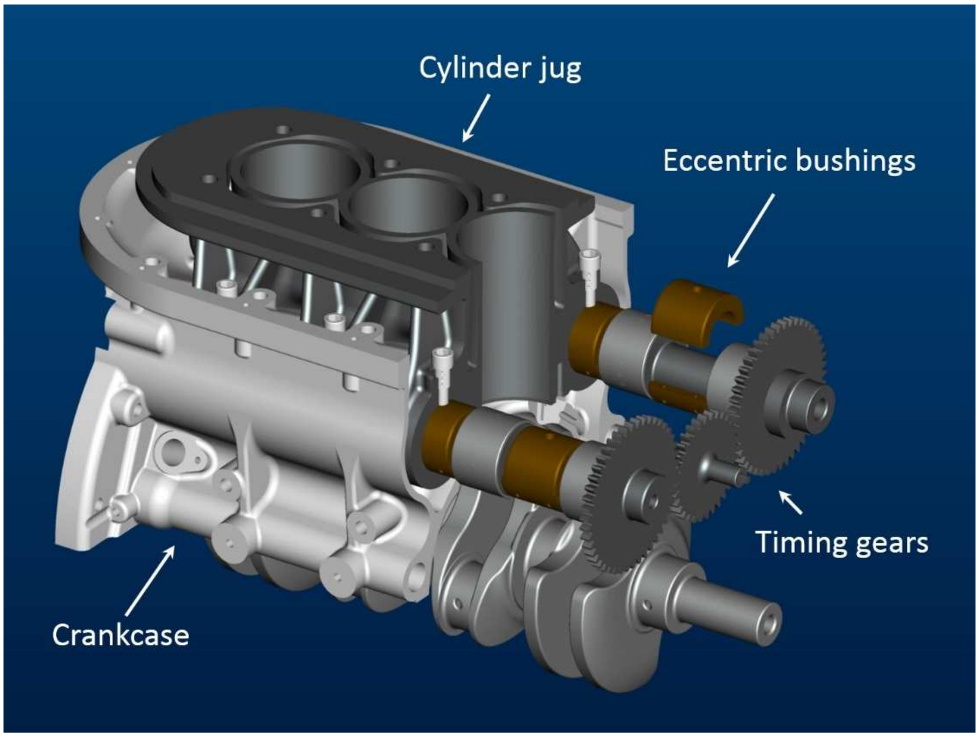


Figure 5. Rotation of the control shafts 180 degrees adjusts CR from 17.6 to 8.2

Figures 6 and 7 shown high and low compression ratio settings for the dual control shaft engine. The chain guides move with change of compression ratio to maintain a constant chain tension at all compression ratio settings. The left chain guide as shown below is supported by a journal located on the front of the control shaft. A conventional chain tensioner is located on the lower end of the chain guide. The right-side chain guide is pined to the crankcase at its base. A cam is located on the front of the right control shaft for moving the chain guide with change of compression ratio. The cam is formed to maintain constant chain tension at all compression ratio settings.

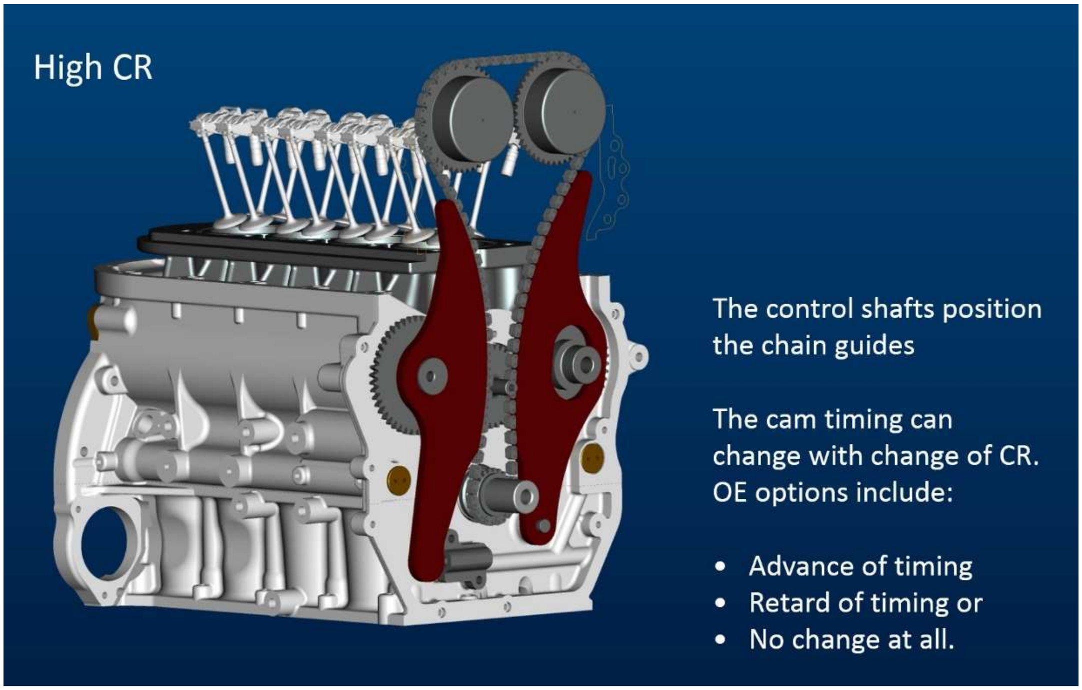


Figure 6. Dual control shaft VCR engine shown at a high compression ratio setting.

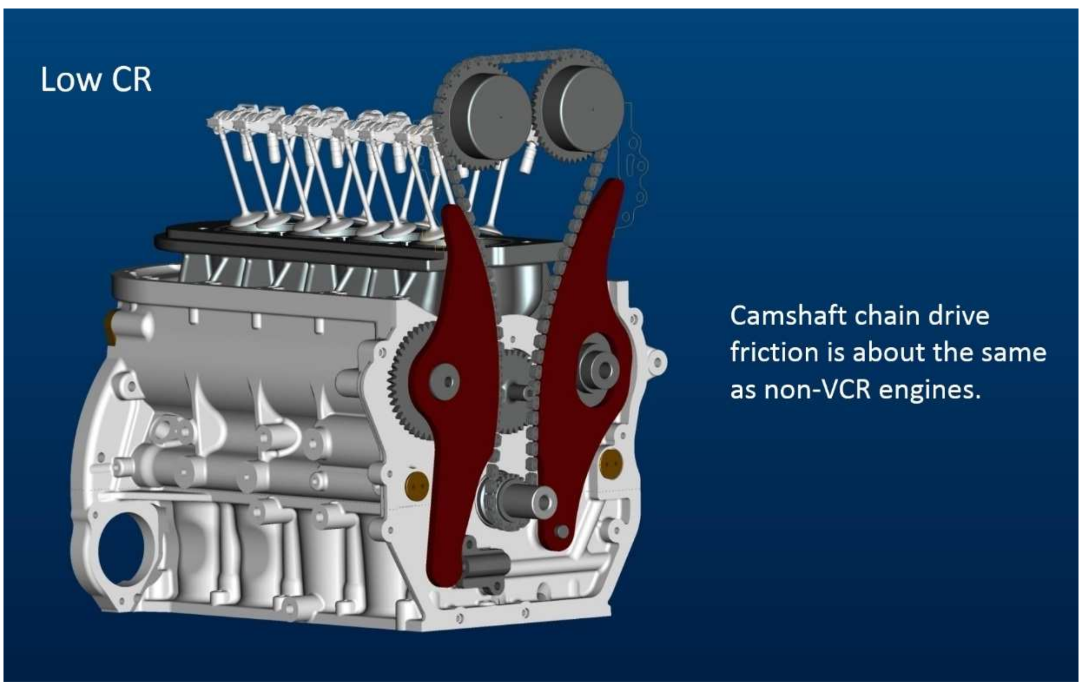


Figure 7. Dual control shaft VCR engine shown at a low compression ratio setting.

The cylinder jug, crankcase and other structural components were developed using finite element analysis (FEA). The cylinder jug was designed for a maximum of 0.03 mm distortion under a cylinder pressure of 2000 psi. (shown in Figure 8).

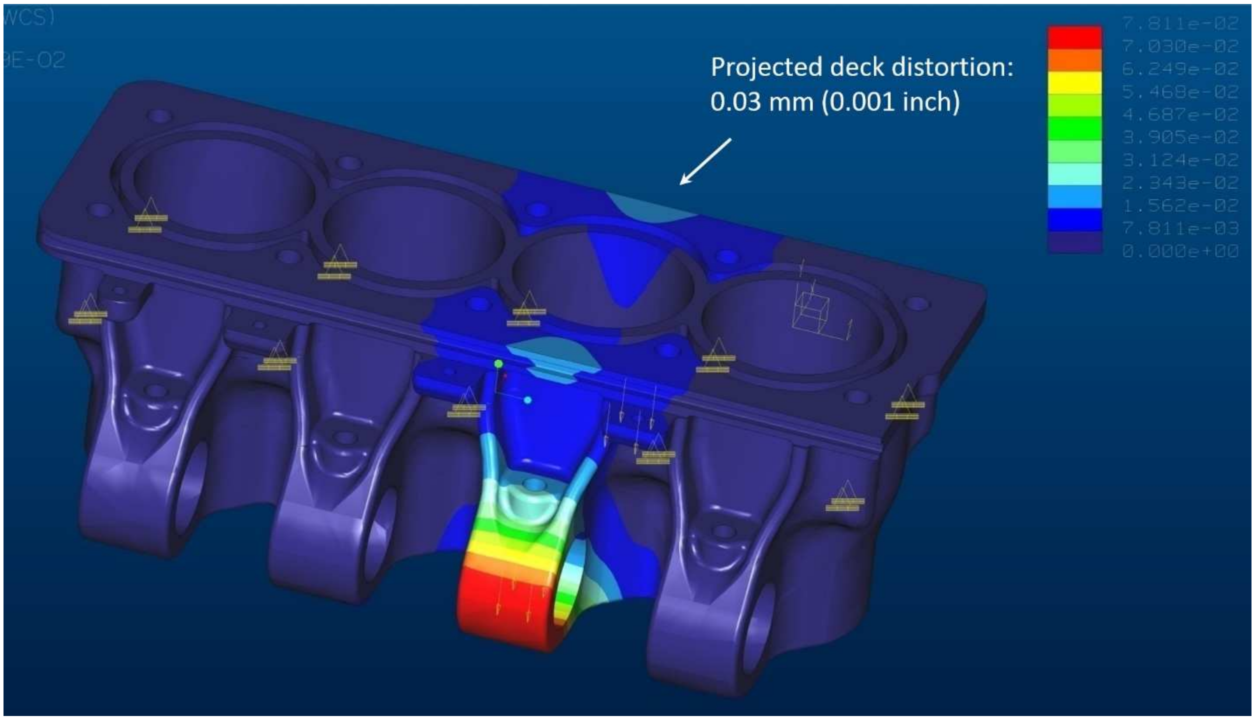


Figure 8. FEA analysis of the cylinder jug.

A domed piston was designed for attaining the high compression ratio combustion chamber. The piston dome, shown in Figure 9 has a reasonable form. A small central bowl was used to provide clearance around the spark plug.

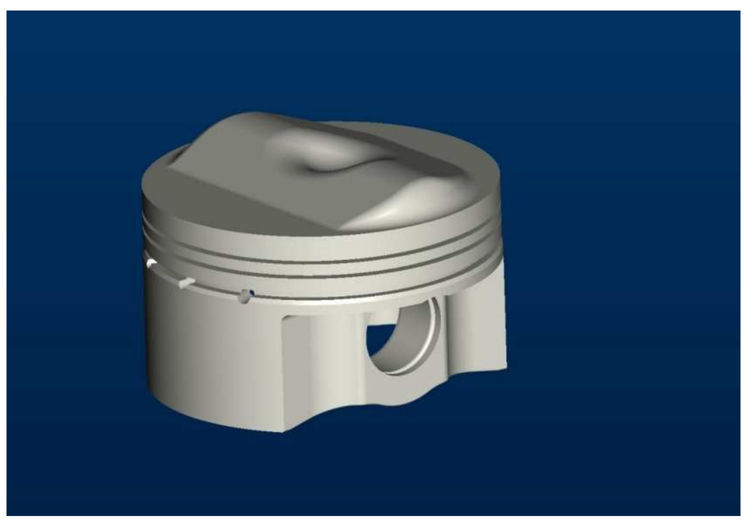


Figure 9. High compression ratio piston.

1C. VARIABLE VALVE CONTROL

The Atkinson cycle is used to achieve high engine efficiency at small engine power levels, and a conventional combustion regime is used for attaining high engine power levels. The Eaton switchable rocker arm mechanism shown in Figure 10 was used to switch between Atkinson cycle and conventional cam lobe profiles (SAE Paper 2018-01-0382). Figure 11 shows one of the cam lobe / valve lift designs. The Atkinson cams have the longer valve open duration to provide later intake valve closing. Three cam profile options were developed for testing, with Build B shown in Table 2 being the preferred configuration. The valve timing values were developed using GTPower software.

Figure 10. Eaton switchable rocker arm mechanism.

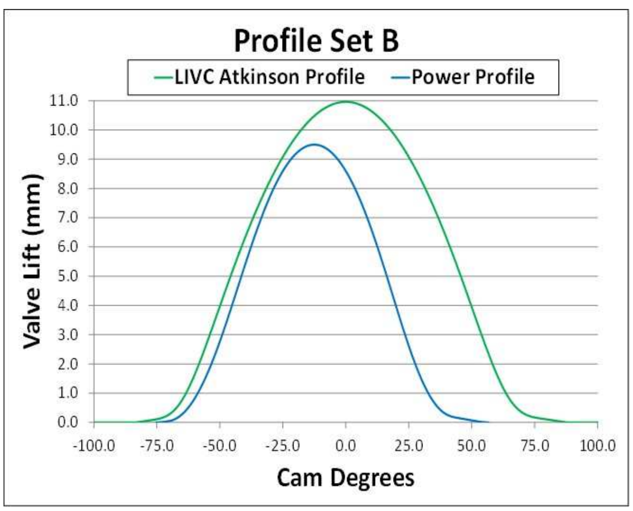


Figure 11. Representational Atkinson cycle and conventional valve lift profiles

The optimum amount of exhaust gas recirculation (EGR) to use in combination with the Atkinson Cycle was evaluated using GTPower. Minimum BSFC is shown in dark blue in the right side chart of Figure 12. The minimum BSFC value occurs with an exhaust valve closing angle (EVC) of about 390 crank angle degrees. The chart on the left shows a trapped exhaust gas residual mass fraction of about 12 percent at an EVC of 390 degrees. This finding indicates that minimum BSFC can occur with a modest EGR mass fraction, and more specifically the EGR mass fraction is too small to cause unstable combustion. This was a significant finding because it indicated that achieving stable, robust combustion would not be challenging under the current program. Figure 12 provides an early development finding. The later optimized cams for testing are shown in Table 2, with Build B being the preferred configuration.

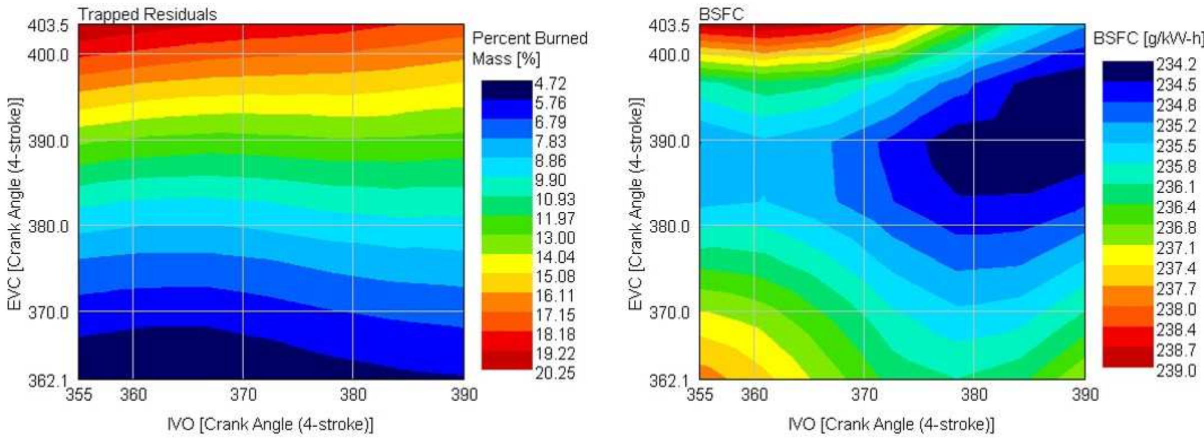


Figure 12. Optimum EGR concentration with Atkinson Cycle cam timings.

EATON - ENVERA VVL CAM TIMINGS
All values at 1.00 mm lift

CAMSHAFTS	STOCK		BUILD A		BUILD B		BUILD C	
	Light load	Torque	Atkinson	Torque	Atkinson	Torque	Atkinson	Power
Intake Cam	IN-S		IN-A		IN-B		IN-C	
Duration	**246.2**	**200.0**	**246.2**	**200.0**	**254.2**	**183.1**	**280.0**	**214.0**
IVO	378.3	334.0	378.3	334.0	380.6	346.2	378.3	334.0
IVC	624.5	534.0	624.5	534.0	634.8	529.3	658.3	548.0
MOP	501.4	434.0	501.4	434.0	507.7	437.8	518.3	441.0
Exhaust Cam	EX-S		EX-A		EX-B		EX-B	
Duration	**186.0**	**186.0**	**213.7**	**213.7**	**200.4**	**200.4**	**200.4**	**200.4**
EVO			176.0	175.0	180.5	199.5	180.5	199.5
EVC			389.7	388.7	380.9	399.9	380.9	399.9
MOP			282.9	281.9	280.7	299.7	280.7	299.7

TABLE 2: Cam timing values developed for testing.

1D. ADVANCED BOOSTING TECHNOLOGY

Turbocharging and/or supercharging is needed for attaining high power and torque output from the downsized VCR engine. For the 4-cylinder VCR engine to have the same drivability as a much larger V8, high torque values must be attained at low engine speeds with no or almost no perceptive time delay. Supercharging was selected for the current program over turbocharging to avoid turbo lag. A clutched supercharger was specified for the program in order to largely eliminate motoring losses from the supercharger during times when boost pressure is not needed.

Several advanced boosting options were evaluated using GTPower modeling, including a variable speed drive supercharger, combining turbo and supercharging, the clutched supercharger and turbocharging. Figure 13 shows a conventional supercharger and Figure 14 shows and advanced supercharging concept investigated under the program that includes a novel cooled air recirculation design.

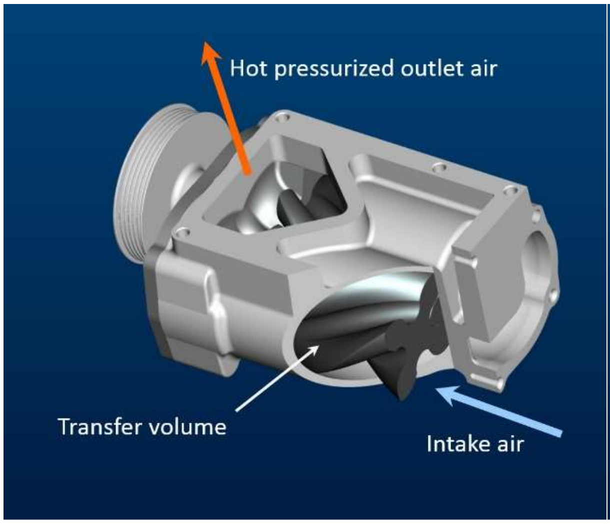


Figure 13. Conventional supercharger.

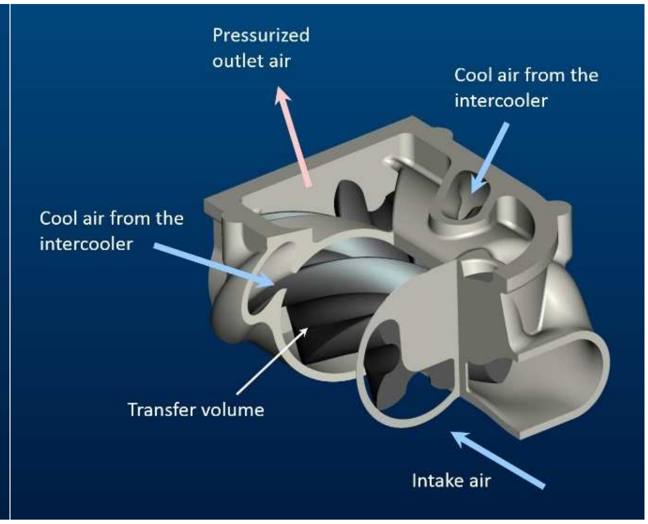


Figure 14. Cool-air recirculation supercharger.

1E. VCR ENGINE TEST SETUP

Engine build and startup took place at Hasselgren Engineering. The engine management system was made fully functional and initial calibration settings were dialed in. The engine was then shipped to MAHLE Powertrain for precision testing (Figures 15 and 16). Testing equipment included: AC Engine dynamometer; Kistler 6052 pressure transducers & amps; AVL IndiCom combustion analysis; AVL PUMA data compilation; and AVL AMA i60 emissions measurement.

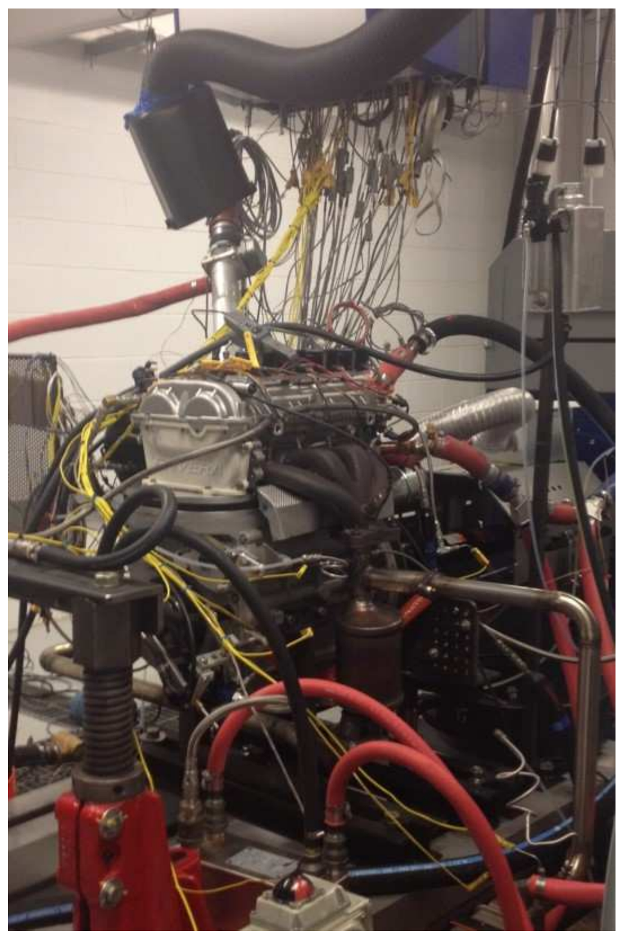


Figure 15. Envera VCR engine at MAHLE Powertrrain.

Figure 16. PFI fuel rail added to the stock GDI cylinder head.

1F. VCR ENGINE TEST RESULTS

Engine test results are shown in Figures 17, 18 and 19. Figure 17 shows 10 to 90 percent burn duration in crank angle degrees at 2000 rpm and with a compression ratio of 17.6. The engine is operated wide open throttle (WOT) with Atkinson cam timing controlling engine power output. Cam B with a maximum open point (MOP) of 502 degrees, and Cam A with an MOP of 501 degrees provided the fastest burn rates.

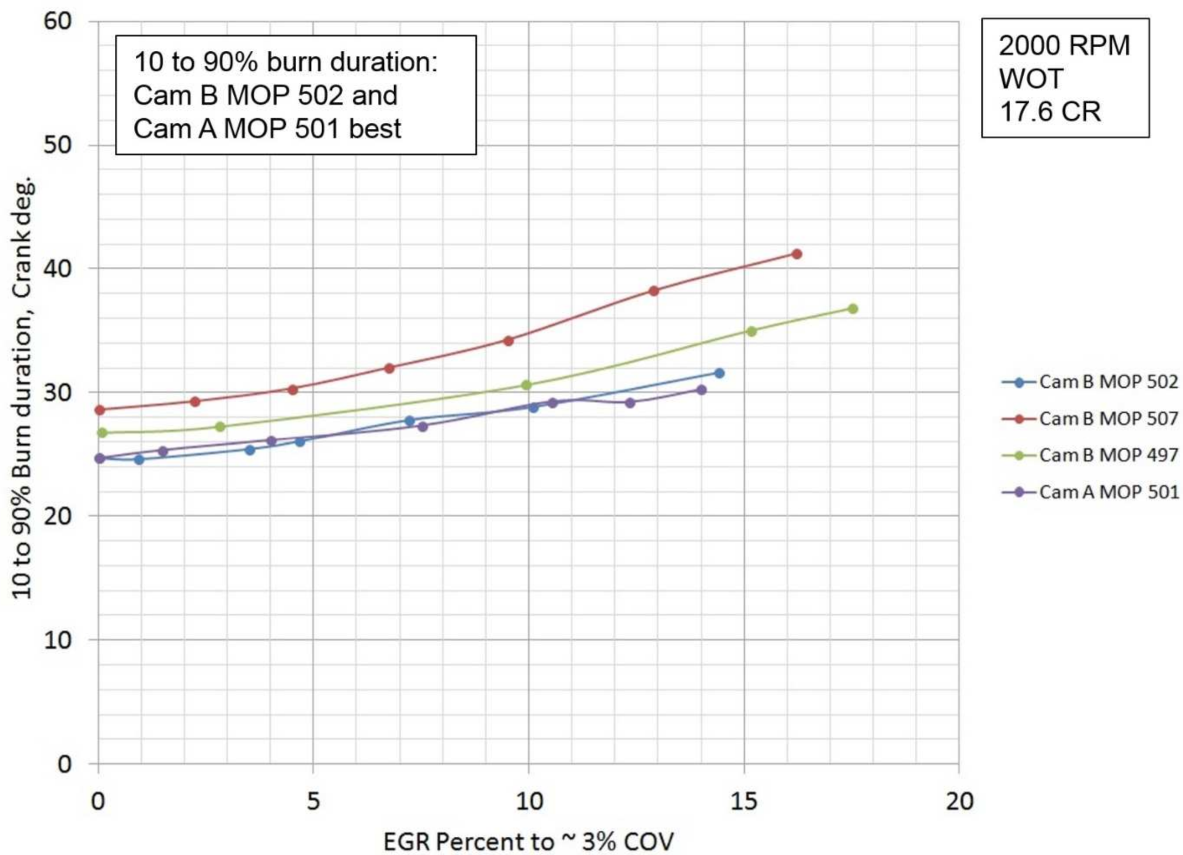


Figure 17. 10 to 90% burn duration in crank angle degrees.

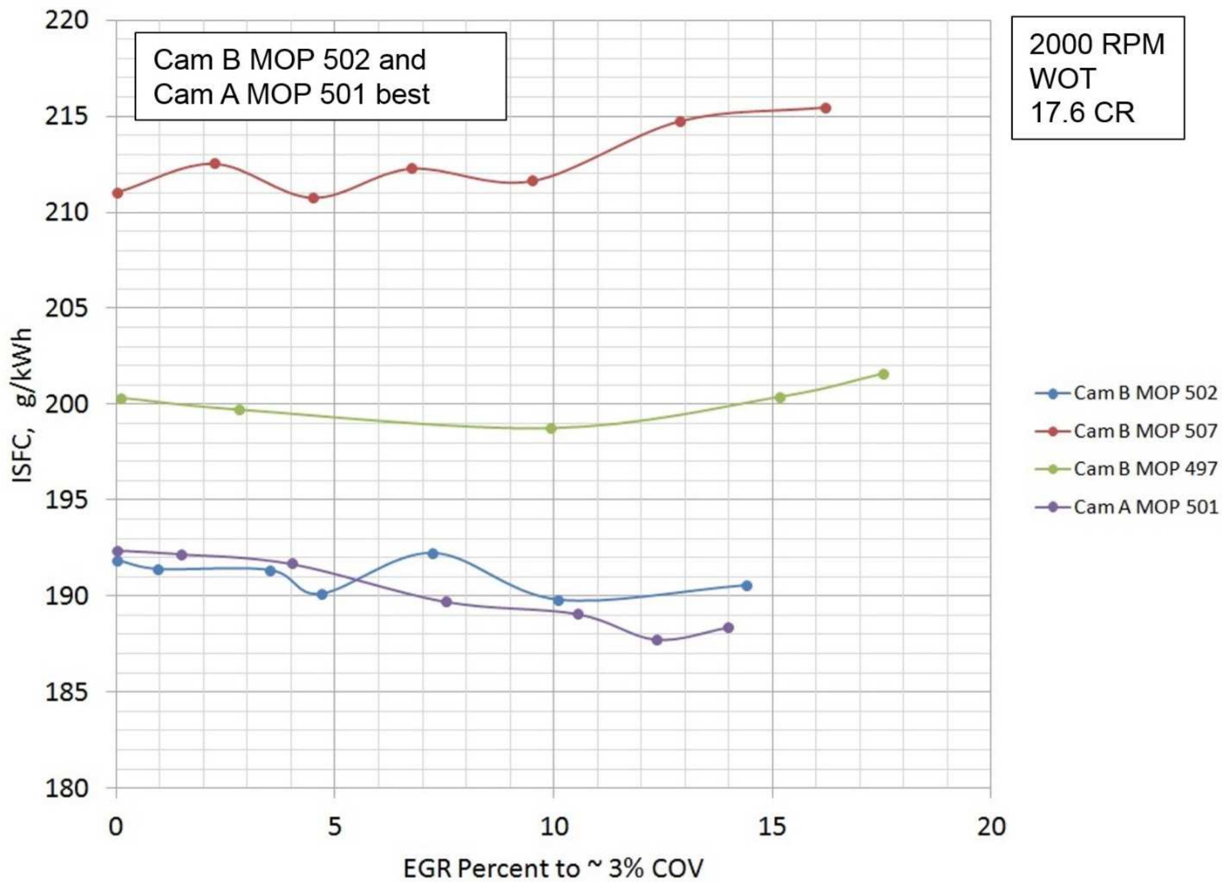


Figure 18. Indicated specific fuel consumption.

Figure 18 shows indicated specific fuel consumption, where it can be seen best results are with Cam B with an MOP of 502 and Cam A with an MOP of 501.

Figure 19 shows final results for Cam B at 2000 rpm and a compression ratio of 17.6. A maximum BMEP of 6.45 bar is achieved without EGR dilution. The 10 to 90 crank angle burn duration was 24.2 degrees. A BSFC value of 230.4 g/kWh was achieved at 5.36 bar BMEP, a value close to the target value of 230 g/kWh at 5.25 bar BMEP. The EGR dilution level was 9.5 percent, and the 10 to 90 percent burn crank angle degree duration was 34.3 degrees. These results were achieved during the second day of testing under the program. Engine tuning for attaining lower BSFC values is described in the following section.

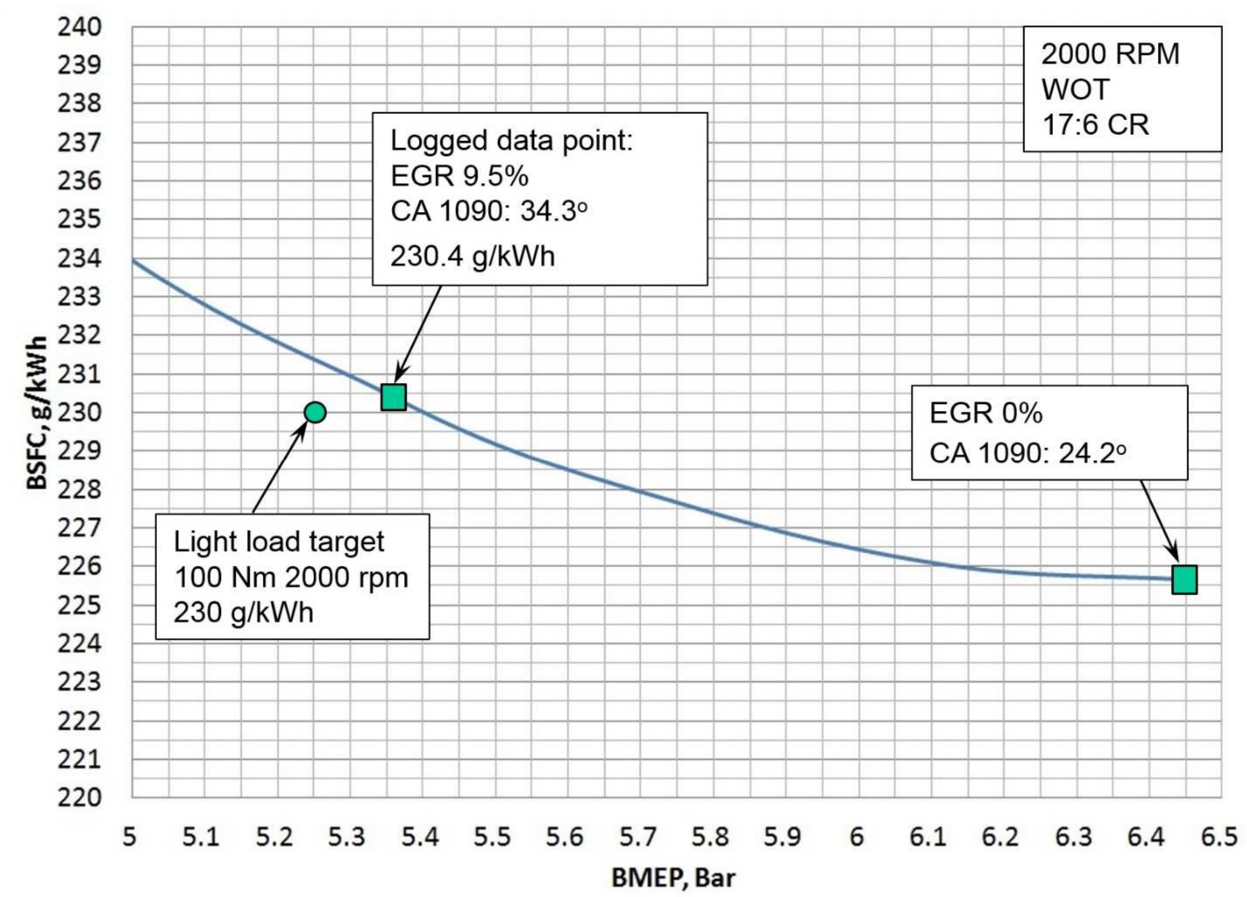


Figure 19. BSFC vs. BMEP test results at 2000 rpm with a 17.6 compression ratio and Atkinson Cycle cam timing.

1G. MILEAGE PROJECTIONS

The optimization process for maximizing vehicle fuel economy included a number of steps. The first step included development of a GTPower model for simulating and optimizing engine efficiency. Combustion burn rate for the actual engine was not known, so assumed combustion burn rate values were used. The GTPower model was used to design the camshafts and develop other engine settings for initial engine dynamometer testing.

The second optimization step involved dynamometer testing of the engine and recording actual combustion burn rate data. Testing was conducted using three different cam profile options and with various exhaust gas recirculation levels. Testing was conducted at MAHLE Powertrain. Engine build and high-power testing support was provided by Hasselgren Engineering.

The third step involved updating the GTPower model to include actual combustion burn rate values. The GTPower model was then re-tuned to further improve engine efficiency. A BSFC map was then created. The GTPower modeling was performed by the EngSim Corporation.

The forth step involved creating a GTDrive model of the Ford F150 full-size pickup truck. The initial version of the GTDrive model included the stock 3.5L V6 engine. This was done in order to calibrate and check the GTDrive model against known data. GTDrive modeling was conducted by EngSim Corporation.

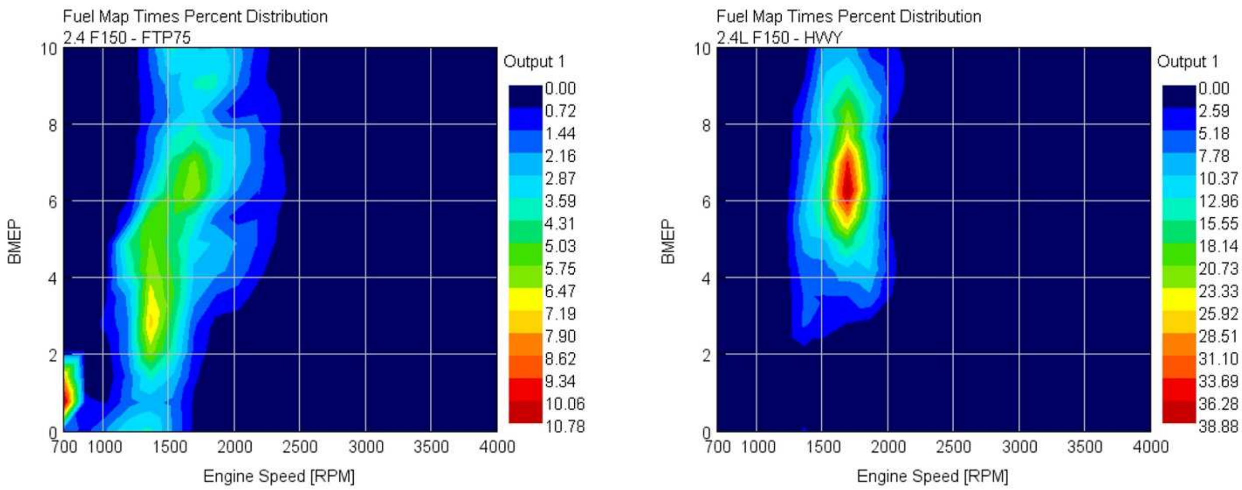


Figure 20. F150 fuel consumption over the FTP75 City and Highway drive cycles.

The fifth step involved replacing the 3.5L V6 engine in the GTDrive model with the 2.4L VCR engine, and then developing a transmission shift schedule for maximizing fuel economy and meeting performance needs. The F150 was assumed to be 200 pounds lighter due to the reduced engine size, but otherwise the same. Fuel economy values were then projected. Table 3 shows these "Optimization 1" fuel economy values. Optimization Level 1 was conducted by the EngSim Corporation.

The sixth step involved plotting fuel consumption on BMEP vs. engine speed charts for the city and highway EPA test drive cycles. These plots are shown in Figure 20, and show the engine load and speed conditions where most of the fuel is consumed. On the city FTP75 test cycle, high engine efficiency is needed most at 3.0 bar BMEP @ 1400 rpm and at the engine idle condition. For the highway drive cycle a high engine efficiency is needed most at 6.4 bar BMEP @ 1700 rpm.

The seventh step involved optimizing VCR engine efficiency where it counts most: The city load condition of 3.0 bar BMEP @ 1400 rpm and the highway load condition of 6.4 bar BMEP at 1700 rpm. Potential improvements in mileage where then estimated. Table 3 shows these "Optimization 2" fuel economy values. Optimization 2 fuel economy values assume variable valve control on both the intake and exhaust cam shafts, in addition to the VCR. Optimization Level 2 was conducted primarily by Envera.

FUEL ECONMOMY PROJECTIONS		**Baseline**		**Optimization 1**	**Optimization 2**
F150 PICKUP TRUCK *with*					**GT Modeling Only**
2.4L in-line 4-cylinder engine					
Variable Compression ratio		5.0L V8	3.5L V6	2.4L VCR-VVL	2.4L VCR-VVL
Variable Valve Control		6spd-a 2WD	6spd-a 2WD	6spd-a 2WD	6spd-a 2WD
Advanced Boosting		MY 2012-14	MY 2014		
Fuel economy projection method		EPA Guide	EPA Guide	Engine Testing &	GT Optimized
				GT Modeling	CTY & HWY
Mileage Projections					
City	mpg	15	16	22.4	24.0
Highway	mpg	21	22	24.3	25.4
Combined	mpg	17	18	23.2	24.6
Percent improvement over baseline	Percent	*Baseline*	6	35	43

Table 3. F150 pickup truck fuel economy projections

Results

The following results were achieved in Phase 3 of the program:

- Dynamometer testing of the VCR engine up to approximately 300 hp

- Achieved 230.4 g/kWh @ 2000 rpm and 5.36 bar bmep

- Projected 219 g/kWh @ 1700 rpm and 6.35 bar bmep
  (The highway load condition shown in Figure 20)

- High efficiency values were achieved using stoichiometric combustion for attaining low tail pipe emission values using proven 3-way catalytic converter technology

- Projected 35 to 43 percent fuel economy improvement for an F150 full-size pickup truck equipped with the VCR engine.

1H. FUTURE DEVELOPMENTS

The first-ever build of the Envera dual control shaft VCR mechanism was realized under the current program. To ensure first-build program success large engineering factors of safety were used. The crankcase and eccentric control shafts are far more robust than needed. Design of a second VCR engine has commenced and includes a number of upgrades, including design with production-intent factor of safety values for a lighter and more compact engine. Engine friction values for the dual control shaft VCR mechanism can be further improved. The next engine design will utilizing additional advanced friction reduction technologies such as piston cylinders with a mirror bore finish as well as a number of additional friction reduction technologies. Several advances will be incorporated into the cylinder head. The next engine will also include advanced turbocharging technology to realize both higher efficiency and increased torque.

Hybridization will become more widespread in the future. The next VCR engine is being developed for a hybrid application. Advantages of the Turbo-VCR Hybrid include:

- Ability to minimize the size, weight and cost of the electric machines and batteries
- Sustained higher power outputs *No range limitation under load*
- Higher vehicle mileage than hybrids without engine downsizing
- Uncompromised payload weight *No major payload loss due to battery weight*

The powertrain cost must be consistent with mass market pricing. Powertrain affordability requirements can be met with the Envera VCR engine.

1I. PROGRAM SUMMARY

The primary objective of the current program was to demonstrate the potential of improving the fuel economy of a full-size pickup truck by approximately 40 percent. This objective was successfully achieved through engine build and testing and computer modeling.

Advanced combustion technologies (HCCI, lean burn, extreme dilution, pre-chamber combustion) were not required for achieving the efficiency target. Therefore, no advances in emission control technology are required for complying with tailpipe emission standards.

The dual control shaft VCR mechanism was successfully demonstrated at power levels up to about 300 hp. The prototype is far more robust than needed. Future designs can be optimized for a smaller size and a lighter weight.

The approach for attaining high mileage is combining aggressive engine down-sizing with high-efficiency Atkinson Cycle technology. A challenge is attaining power and torque requirements from the down-sized Atkinson Cycle engine. Technologies employed for attaining high power and torque requirements include:

- Cam profile switching for switching from Atkinson Cycle cams to boosted cycle cams
- Advanced boosting to increase engine power and torque, and
- Variable compression ratio to prevent detonation (engine knock).

Hybridization will become more widespread in the future. The next VCR engine is being developed for a hybrid application. Advantages of the Turbo-VCR Hybrid include:

- Ability to minimize the size, weight and cost of the electric machines and batteries
- Sustained higher power outputs *No range limitation under load*
- Higher vehicle mileage than hybrids without engine downsizing
- Uncompromised payload weight *No major payload loss due to battery weight*

The powertrain cost must be consistent with mass market pricing. Powertrain affordability requirements can be met with the Envera VCR engine.

1J. ACKNOWLEDGEMENTS

We wish to acknowledge Roland Gravel, Program Manager from the U.S. Department of Energy, Ralph Nine, Program Manager from the National Energy Technology Laboratory, and David Genise, Austin Zurface, Scott Brownell, David Yee and Karen Bevan from the Eaton Corporation for their valuable contribution to this study. We would also like to acknowledge the valuable contribution to this study from Brad Tillock of the EngSim Corporation, Boris Kesil from ADEM LLC, Paul Hasselgren from Hasselgren Engineering and Hugh Blaxill and Matthew Williams of MAHLE Powertrain. We wish to thank the DOE Vehicle Technologies Office for major funding of this study, as well as the Eaton Corporation for their contributions.

2. VCR CRANKCASE DESIGN ENGINEERING AND PROCURMENT
Envera LLC

Design and engineering of the crankcase includes a number of major subsystems. Development order of the subsystems was important for obtaining optimum system integration and robustness. Major subsystem areas include:

- Piston and bore
- VCR jug base design: Water jacket; head bolt, head gasket and deck design
- VCR gasket design
- VCR control shaft assembly
- VCR jug control shaft interface
- VCR actuator
- Valve chain drive
- Connecting rod
- Crankshaft assembly
- Crankcase
- Front covers and oil pan
- Supercharger integration
- Oil pump

Primary documents that were produced for guiding and tracking development include:

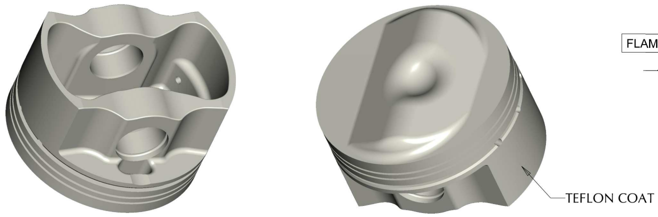
- Bill of materials (BOM) which lists all parts included in the engine
- Geometry tracker, an excel document containing primary engine geometry values
- Design guide, a word document used to track design details
- Complete set of machine drawings
- Complete CAD file of the engine, including all parts (Proe/Creo)

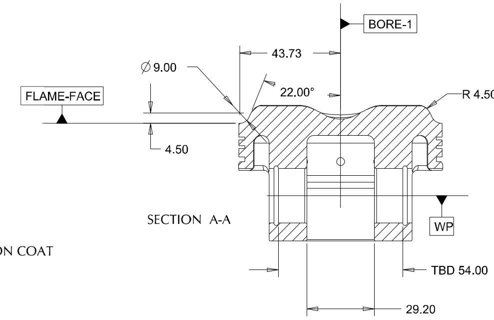
Areas requiring computer aided engineering (CAE) include:

- Cranktrain dynamics MDO, needed primarily for VCR actuator development
- Water jacket CFD
- Port and manifold CFD
- Jug and deck FEA
- Control shaft FEA
- Crankshaft FEA
- Crankcase FEA

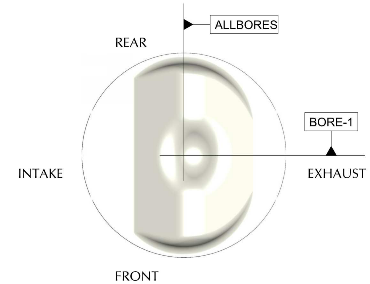
In general the design factors of safety were considerably higher than needed. The large factors of safety were used to insure robustness of the first-build prototype. A production engine would use smaller factors of safety to reduce overall weight.

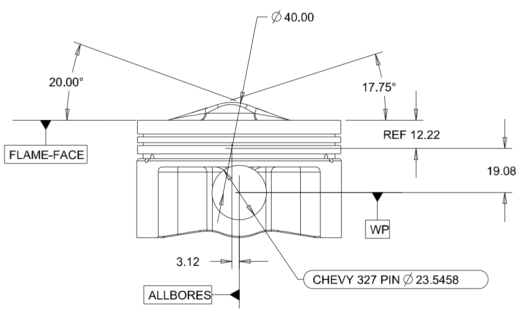
A practice used by Envera is to include engine-builder design details on the machine drawings. Example machine drawings are shown on the following pages. It can be seen that the piston machine drawing includes ring pack and piston pin specifications.

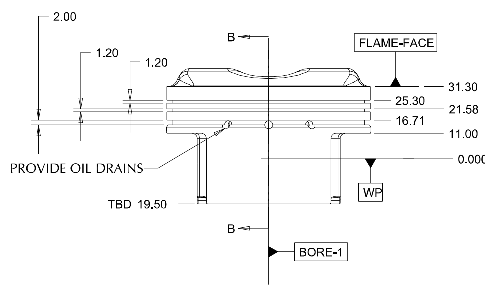


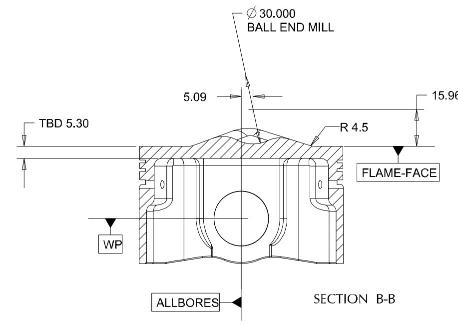


SECTION A-A

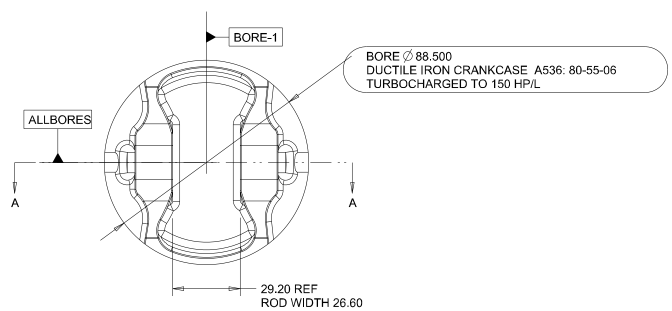








SECTION B-B



PISTON MATERIAL	
PISTON PIN	0.927 X 2.125 & 0.150 Wall - Tool Steel
LOCKS	Wire
TOP RING	1.2 mm 440 B Stainless - PVD applied CrN Total Seal #102104
SECOND RING	1.2 mm Cast Iron Total Seal # 205027
OIL RING	2.0 mm Total Seal Rails # 212508, Expander # 207277

CYLINDER HEAD	GM 2.5L EcoTech
BORE	88.50 mm
STROKE	97.50 mm
ROD CENTER LENGTH	166.00 mm
ROD SE OD	33.2 mm
DOME VOLUME	15.00 CC
COMPRESSION RATIO	17.5 max 8.2 min
ENGINE SPEED	7000 RPM
FUEL	Gasoline
TURBO BOOST	38 psig
POWER	380 hp

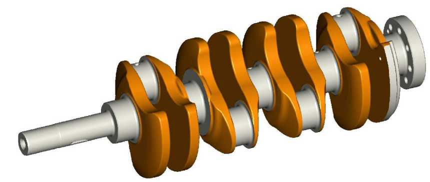
ENVERA LLC

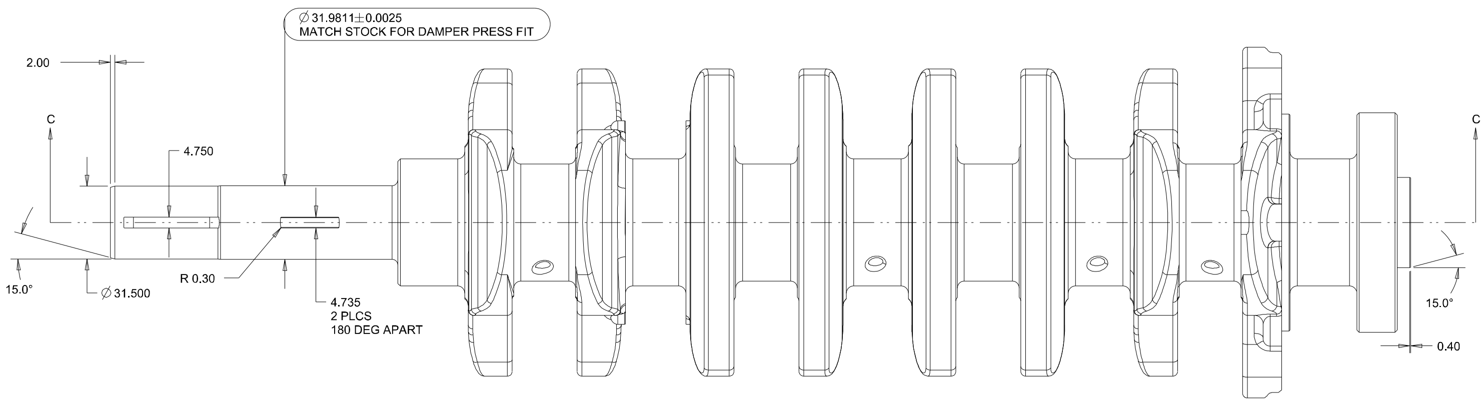
7 Millside Lane, Mill Valley, California 94941
Tel. 415 381-0560

PISTON		D4F-F04	
Part name		Part number	
Project Code	D4F	REV	A4
File Name	D4F-F04—Piston	Software	Proe WF 3.0
Drawn	C Mendler	Date	31 May 2016
Approved	C Mendler	Date	7 July 2016
Checked		Date	
Units	Millimeters	Scale	1 : 1
Unspecified Tolerance .XX = ±	0.06	Material	
Unspecified Tolerance .XXX = ±	0.026	Finish	
Unspecified Angular Tol ±	0.20 deg	Mass	0.374 kg est
Dimensioning & Tolerancing to		ASME Y14.5M: 1994	
Do not scale from drawings. If in doubt ask.			

CONFIDENTIAL INFORMATION

SHEET	1 OF 1	REV	A4





TOP VIEW

MATERIAL NOTE:

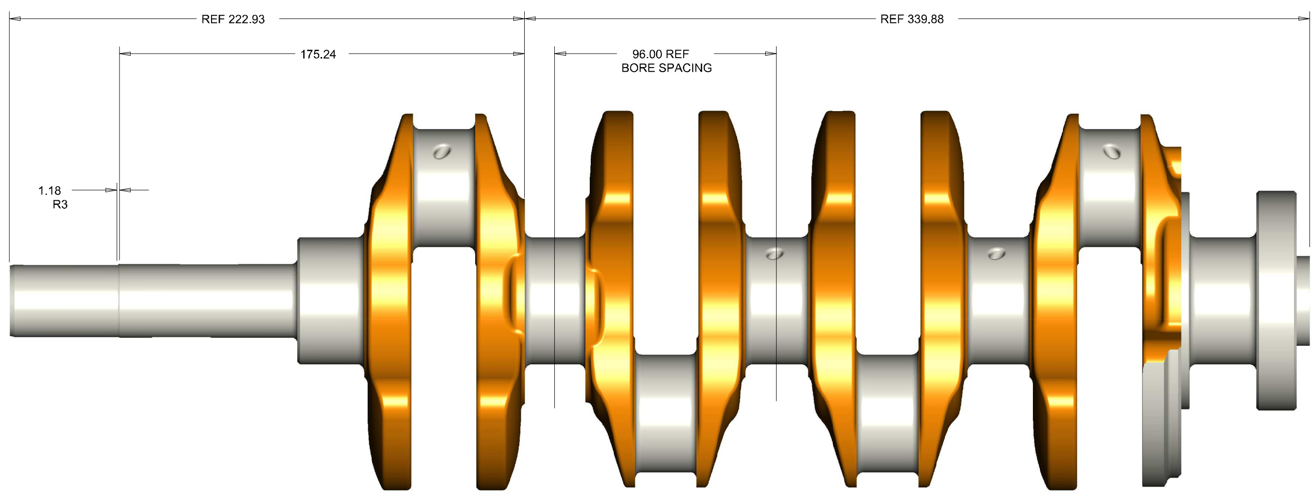
Normalize, quench and temper to RC 38 - 42 after rough machining. Ion-nitride surface harden to 0.003 - 0.005 inch after final grind. Polish main and connecting rod journals, thrust faces and front and rear oil seal raceway to Ra 6. The crankshaft has a standard clockwise rotational direction.

STP files available on request.

BALANCING NOTE:

Spin balance the crankshaft.

Connecting rod big-end mass 507.4 grams

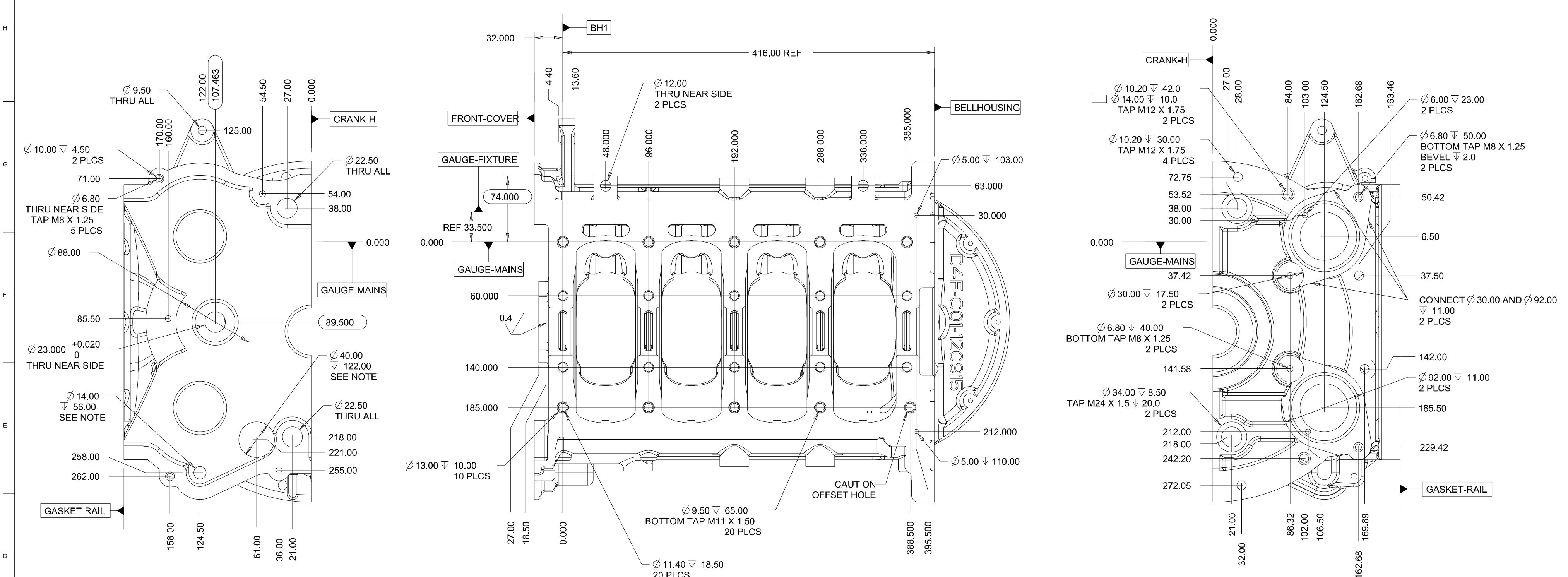


SIDE VIEW

ENVERA LLC

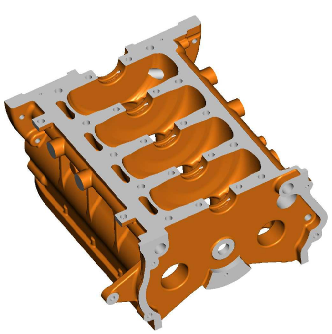
7 Millside Lane, Mill Valley, California 94941
Tel. 415 381-0560

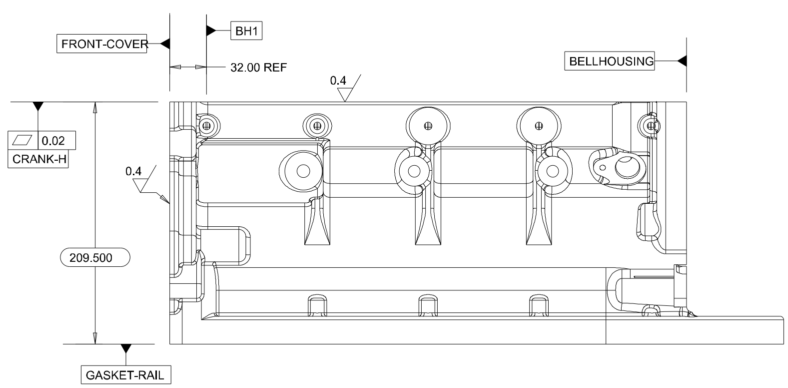
CRANKSHAFT (Part name)		D4F-F17 (Part number)	
Project Code	D4F	REV	A3
ProE file name	D4F-F17—CRANKSHAFT		
Drawn	C Mendler	Date	4 Feb 2016
Approved	C Mendler	Date	11 Feb 2016
Updated	C Mendler	Date	26 May 2016
Units	Millimeters	Scale	1 : 1
Unspecified Tolerance .XX = ±	0.06	Material	Steel 4340
Unspecified Tolerance .XXX = ±	0.026	Finish	AS NOTED
Unspecified Angular Tol ±	0.20 deg	Mass	21.0 kg
Dimensioning & Tolerancing to		ASME Y14.5M: 1994	
Do not scale from drawings. If in doubt ask.			
SHEET	1 OF 7	REV	A3

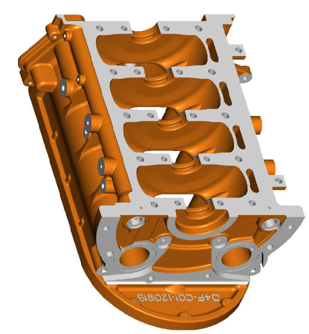


NOTE: ALL HOLE DEPTHS MEASURED FROM FRONT COVER DATUM PLANE

NOTE: ALL HOLE DEPTHS MEASURED FROM BELLHOUSING DATUM PLANE







ENVERA LLC

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Tel. 415 381-0560

CRANKCASE		D4F-C01	
Part name		Part number	
Project Code	D4F	REV	A4
ProE file name			
Drawn	C Mendler	Date	17 March 2016
Approved	C Mendler	Date	23 March 2016
Checked	C Mendler	Date	16 June 2016
Units	Millimeters	Scale	0.500
Unspecified Tolerance .XX = ±	0.06	Material	A 356
Unspecified Tolerance .XXX = ±	0.026	Finish	T6
Unspecified Angular Tol ±	0.20 deg	Mass	24.9 kg
Dimensioning & Tolerancing to		ASME Y14.5M: 1994	

Do not scale from drawings. If in doubt ask.

SHEET	1 OF 5	REV	A4

3. SUPERCHARGER MATCHING & ENGINE GEOMETRY FOR BEST EFFICIENCY
EngSim Corporation – GTPower Modeling

OBJECTIVES

1. Determine supercharger size and pulley ratio for attaining power and torque targets
2. Investigate compression ratio needs for peak torque
3. Determine bore offset distance for best part load efficiency
4. Determine connecting rod length for best part load efficiency
5. Investigate minimum BSFC settings at 100 Nm 2000 rpm.

RESULTS

1. An Eaton 1040 TVS supercharger with a pulley ratio of 2.9 was projected to provide approximately 430 Nm torque over the engine speed range.
2. Optimum compression ratio vs spark timing values are inconclusive at this time.
3. Increasing bore offset benefits engine efficiency at all engine speeds. The maximum bore offset that can be used is limited by connecting rod to cylinder liner clearance requirements. The GTPower study underestimates the benefit, as friction reduction benefits are not being fully accounted for.
4. Increasing connecting rod length reduces piston side force resulting in reduced fuel consumption and increased torque. Benefits are relatively small. A connecting rod center length to stroke ratio of 1.7 was selected for the project.
5. Minimum BSFC was investigated under two conditions at 100 Nm 2000 rpm:
   a. Stoichiometric & no external EGR ~235 g/kWh
   b. Lean 16:1 AFR ~228 g/kWh

   Cylinder deactivation was also investigated, but reported values were at a different (lower) torque value that doesn't support a direct comparison.

   The stoichiometric engine settings and BSFC projection established a starting point for engine optimization under the program.

ENGSIM REPORT

The EngSim Corporation report on supercharger matching is provided on the following pages.

Envera Concept (2)

EngSim Corporation
Brad Tillock
December 10, 2014

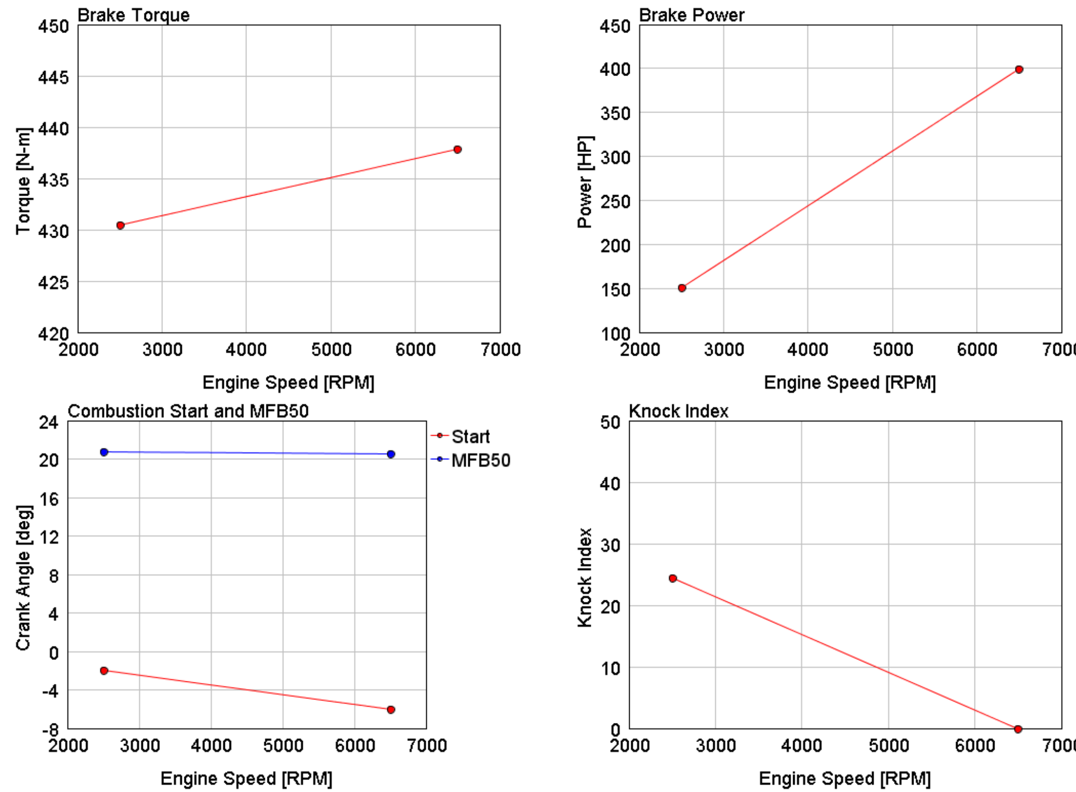
Model Updates

- EngSim and Envera have arrived at decisions regarding the supercharger and engine operation for the peak torque and peak power operating conditions
- This update will show detailed results for peak torque, peak power, and a part load efficiency point

Best Torque and Power

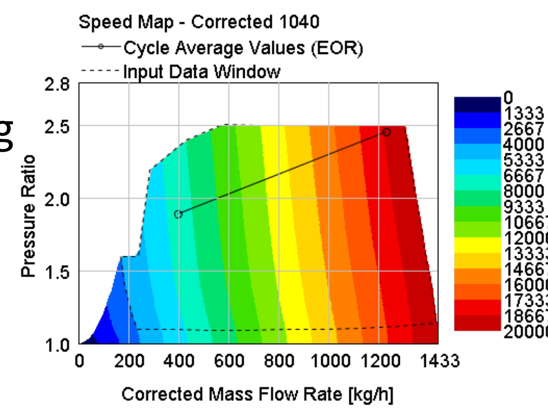
- The following cases have been run with the following conditions
  - 1040 supercharger
  - 2.9 gear ratio
  - Same valve durations for power and torque
  - Variable phasing
  - 10:1 compression ratio
  - Spark timing near the onset of knock

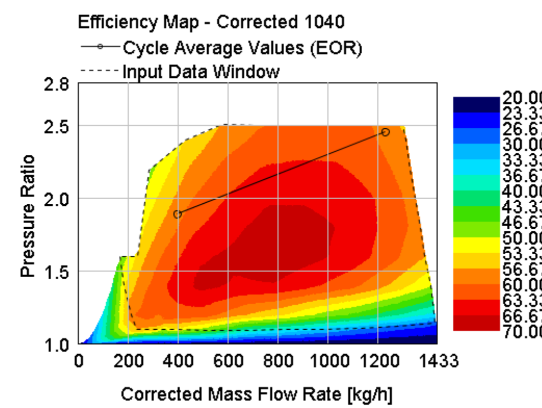
- Previous mappings were used to guide the tuning of the knock model



Best Torque and Power

- At 2.9 gear ratio, the supercharger can remain on the supercharger map with a small margin remaining
- It should be possible to maintain the nearly constant torque curve and remain on the linear line connecting the 2500 rpm point and the 6500 rpm point at WOT.



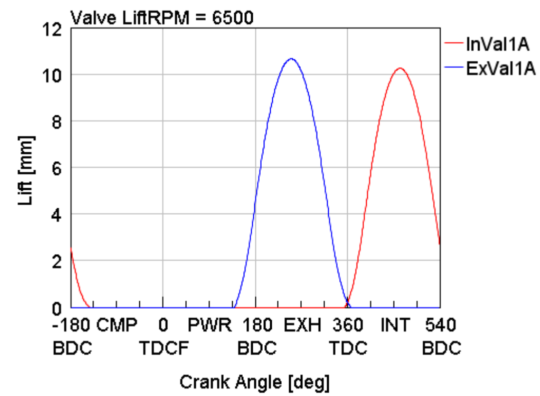


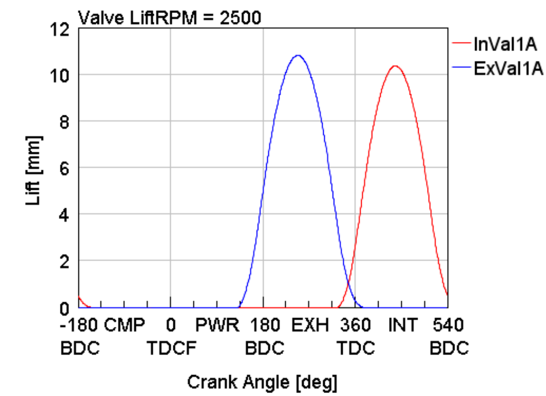
Best Torque and Power

- Valve phasing was allowed to vary for the peak torque and peak power points.

- The intake follows the expected behavior of delaying IVC with increasing engine rpm

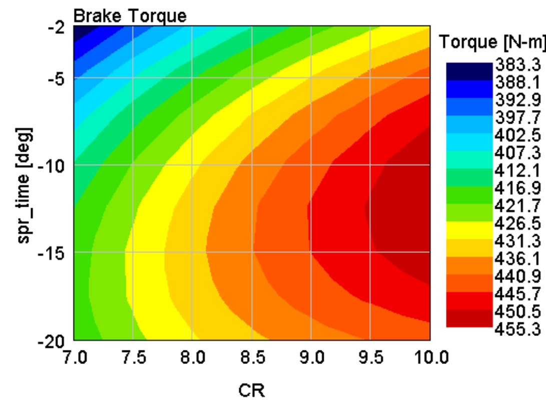
- Normally increasing engine rpm would also lead to advancing the exhaust valve but that did not happen in this case. EngSim will double check the valve timing ranges in the optimization. Maybe this behavior is reasonable for a supercharged engine.

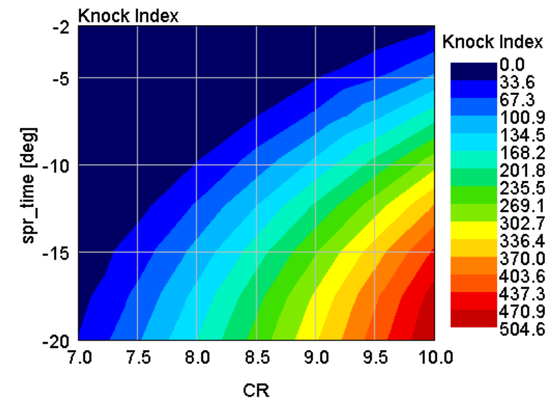




Compression Ratio and Spark Mapping - 2500

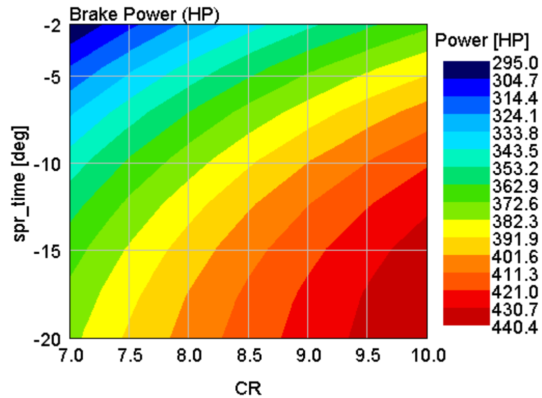
- Plots show the relationship of compression ratio, spark, torque, and knock
- There are multiple combinations of spark timing and compression that will allow for a brake torque of near 430 Nm
- The project has not set a precise limit on the knock index for what is designated as knock and what is acceptable. This is a little arbitrary at this stage in the project. Generally, the dark blue portions of the knock index match what is possible with production engines.

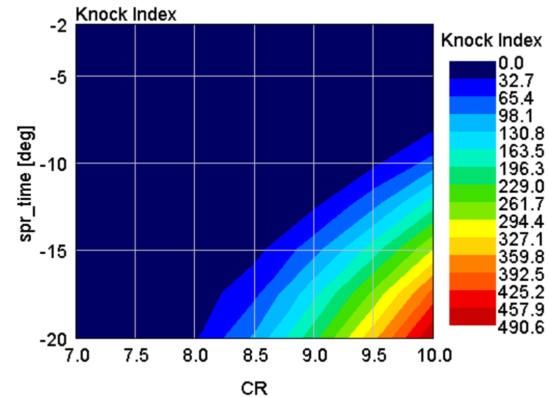




Compression Ratio and Spark Mapping - 6500

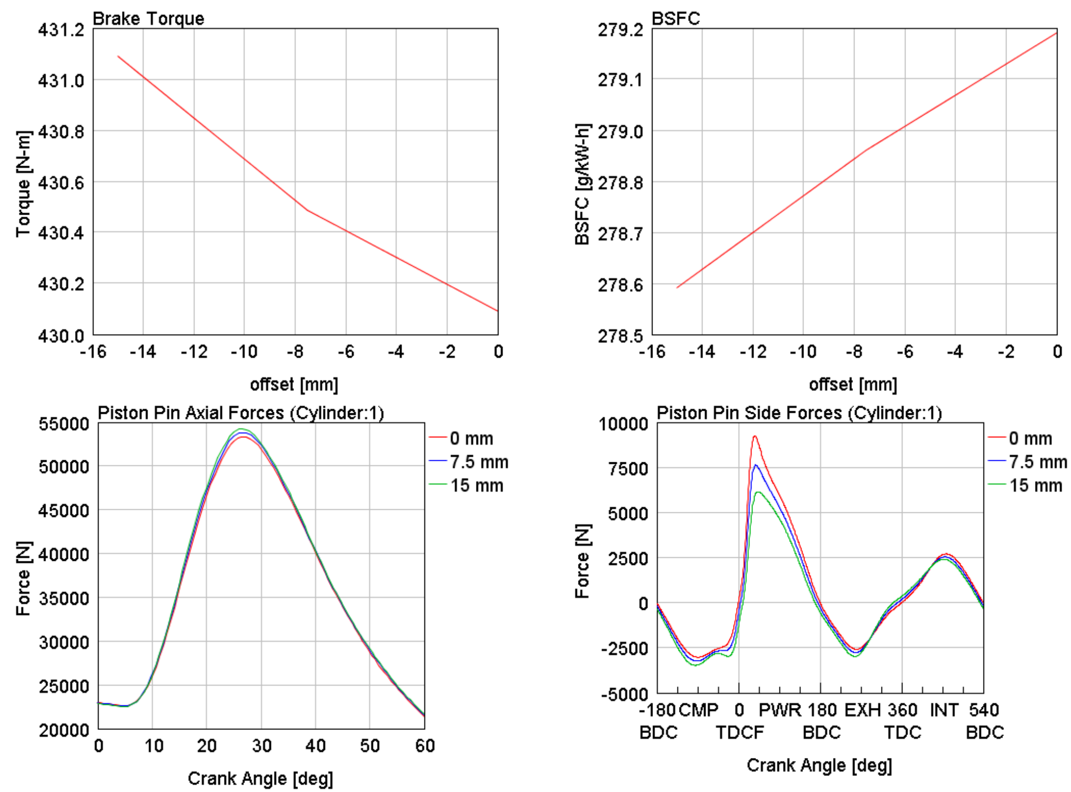
- As expected, the engine is much less likely to knock at the higher rpm
- The 360 HP power target should be possible





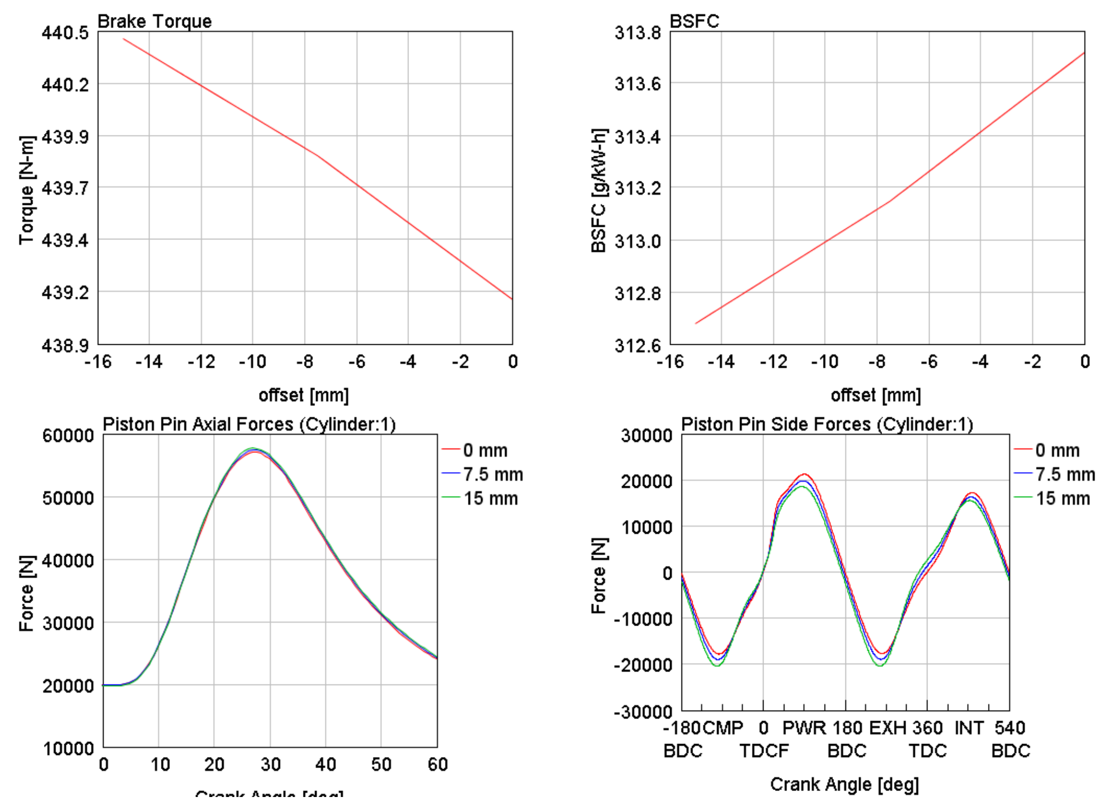
Bore Offset Test – 2500

- Pin offset run at 0, -7.5, and -15 mm
- When viewing these results please keep in mind that the offset is not being fed into any predictive friction model. Friction is constant for each offset.
- Change in thermodynamics from offset is very small
- Only significant difference is in the piston pin side force. This will have some unknown influence on overall crank-slider friction.



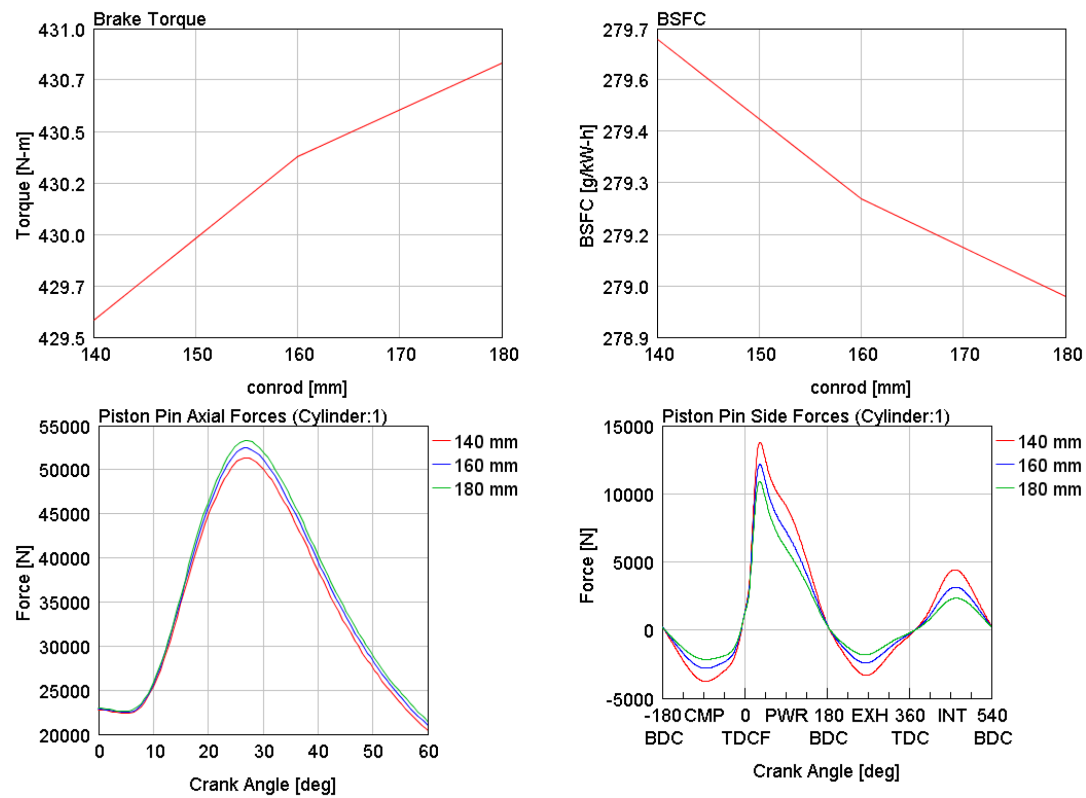
Bore Offset Test – 6500

- Pin offset run at 0, -7.5, and -15 mm
- Bore offset differences are less visible at the higher engine speed



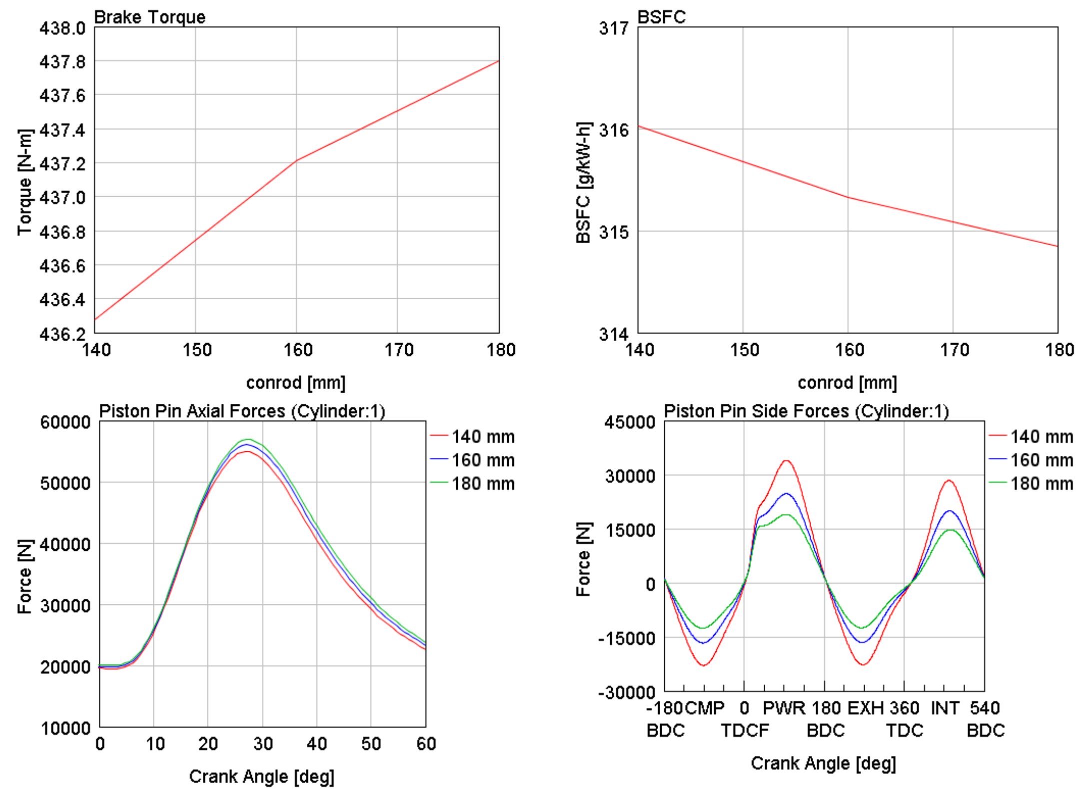
Connecting Rod Test – 2500

- Connecting rod run at 140, 160, and 180 mm
- Piston pin axial force increases with connecting rod length but the piston pin side force is more significantly reduced.



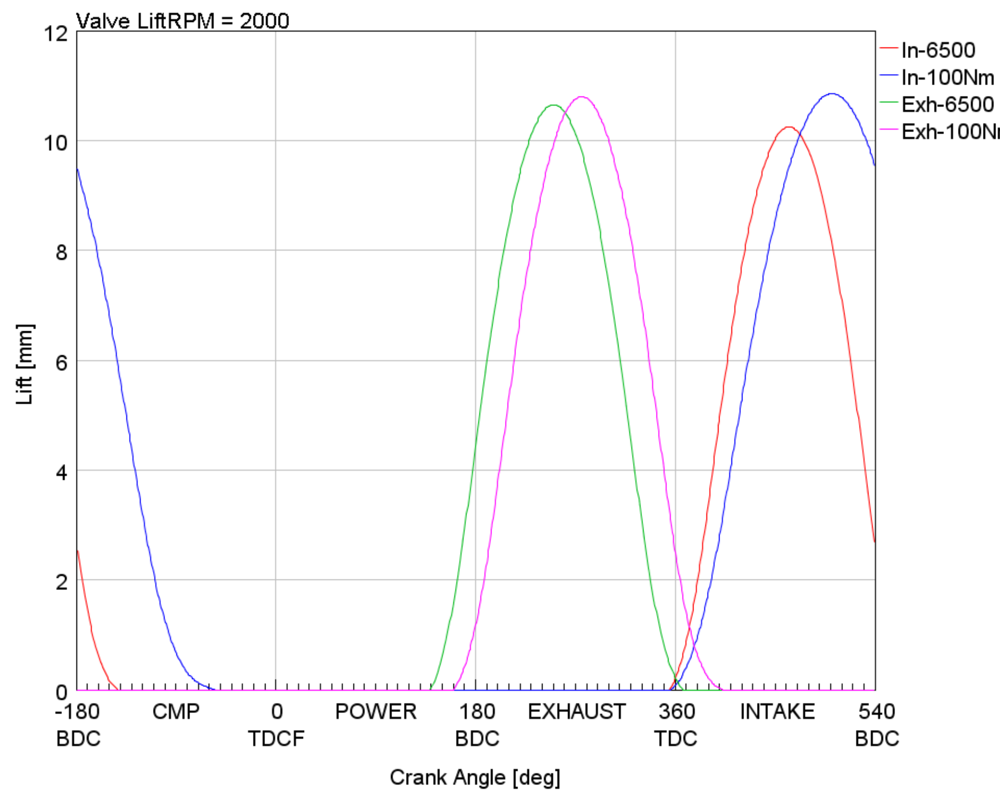
Connecting Rod Test – 6500

- At higher engine speed the effect of the connecting rod is more pronounced. There are significant differences in the piston side force throughout the cycle.



Part Load Results

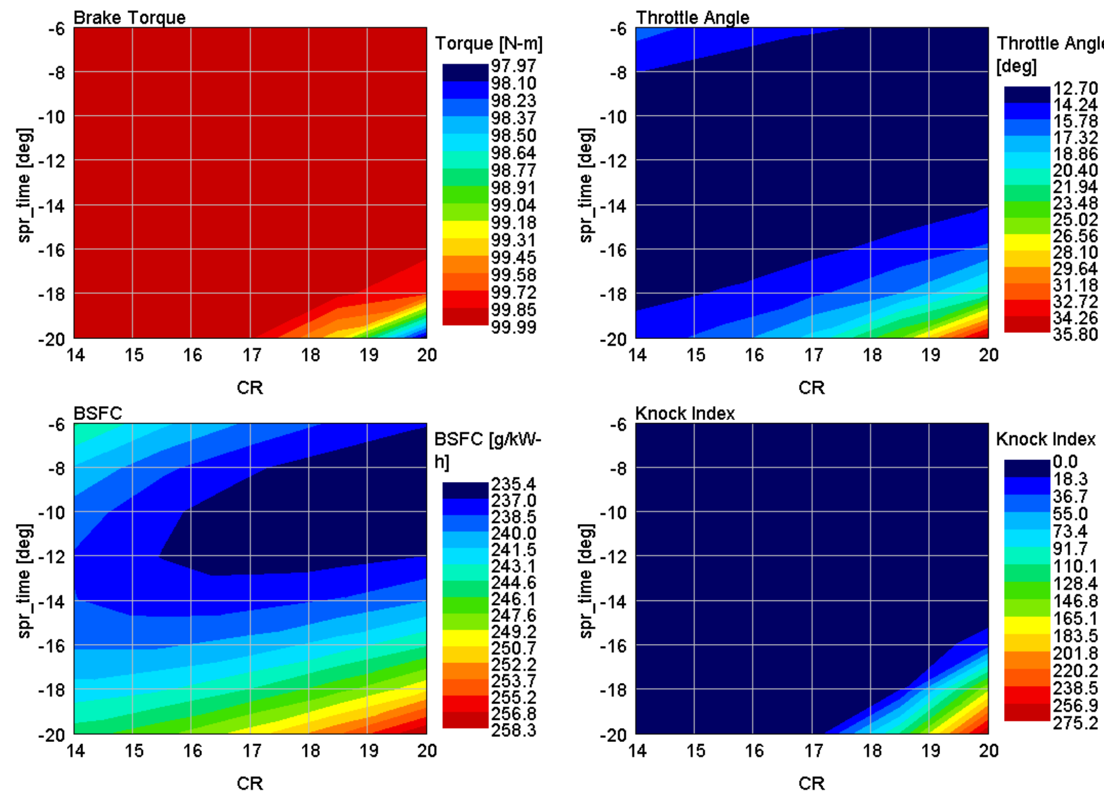
- Goal – Minimize the BSFC at 100 Nm, 2000 rpm
- EngSim used an iterative process to find the best combination of valve events, compression ratio, and spark.
- The optimal valve profiles for the part load condition are plotted along with the peak power valve profiles.
- When the part load profiles were allowed to vary completely, the intake did go to a very late IVC (as expected when minimizing BSFC at part load)



- Intake at part load has higher lift and longer duration – allowed based on Envera specifications

Part Load Results – CR and Spark Mapping

- Target torque was obtained at almost all combinations
- Keep in mind that CR refers to geometric compression ratio – not dynamic
- The engine is still throttled despite the very late IVC. The effects of this will be seen on the following slide.
- The small effect of the compression ratio is explained on slide 15

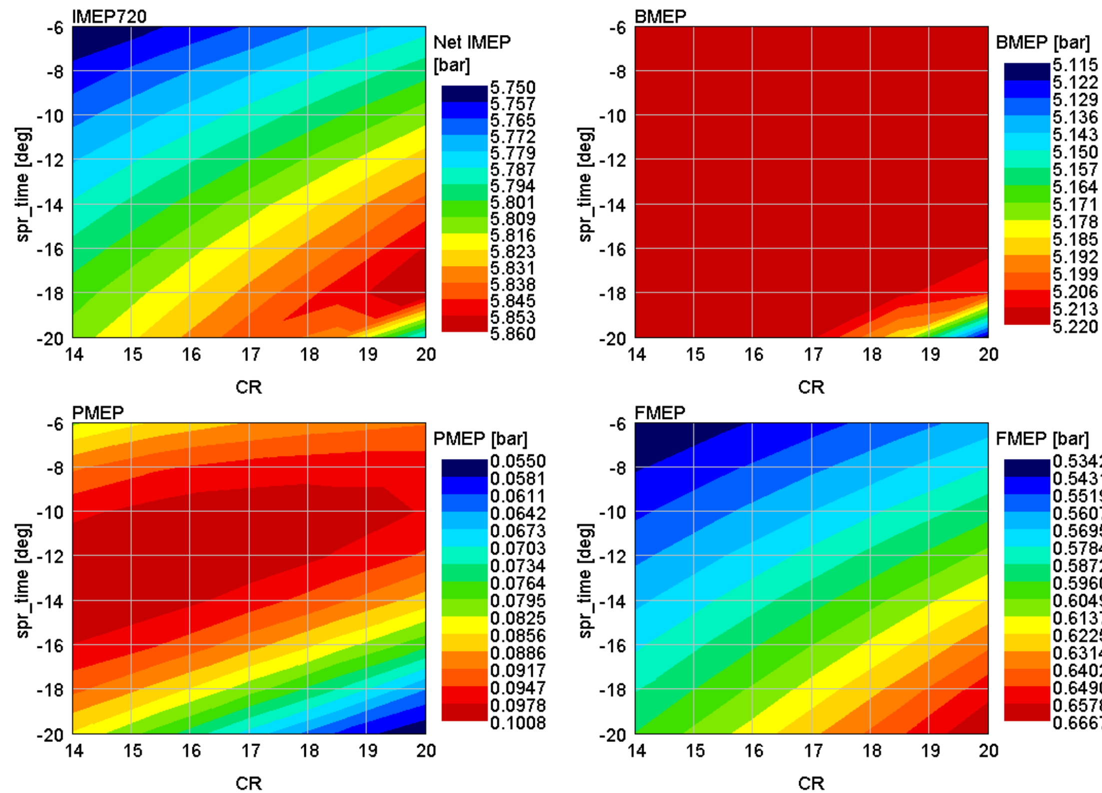


EngSim Corporation

Part Load Results – CR and Spark Mapping

- The contours to the right show the relative importance of FMEP and PMEP. Despite the throttling seen on the previous slide, pumping losses have been decreased to almost nothing.

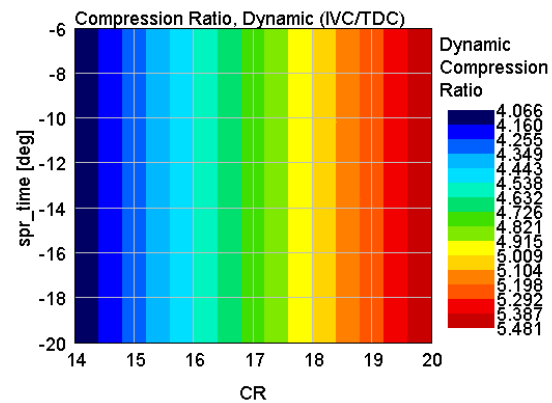
- The friction used in the model is optimistic but reasonable

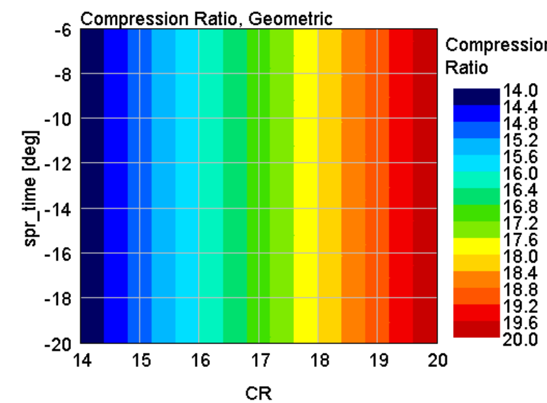


Part Load Results – CR and Spark Mapping

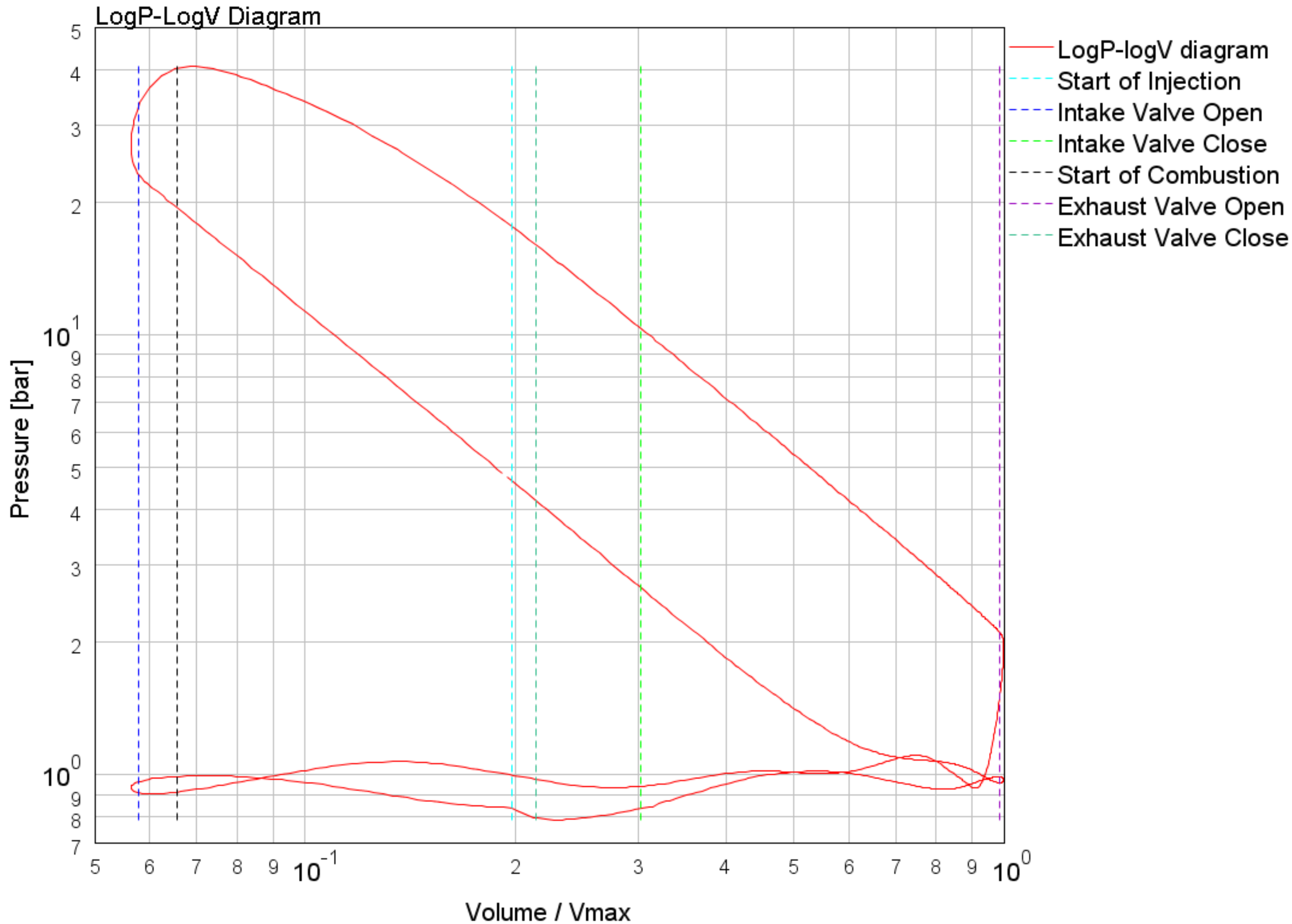
- The small dependence on compression ratio can be explained if we focus on the dynamic compression ratio. With the very late IVC a very large variation in geometric compression is dampened out.

- Expansion ratio is important from an efficiency standpoint but the starting point after compression matters as well.





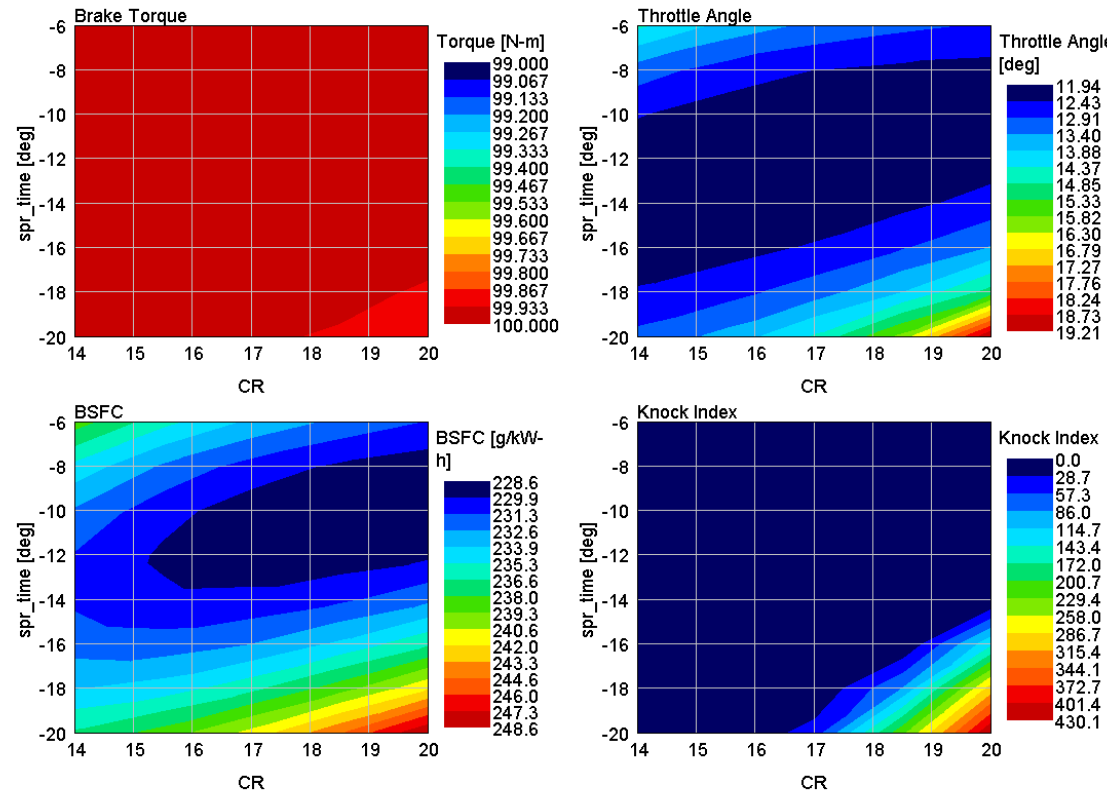
Part Load Results – Low BSFC Point



EngSim Corporation

Part Load Results – Lean (16:1)

- Target torque was obtained at almost all combinations
- EngSim ran a simple test to optimize the valves a second time after changing the fuel ratio from 14.6 to 16
- There was a small reduction in BSFC but EngSim will need to perform more post-processing to break down the efficiency differences



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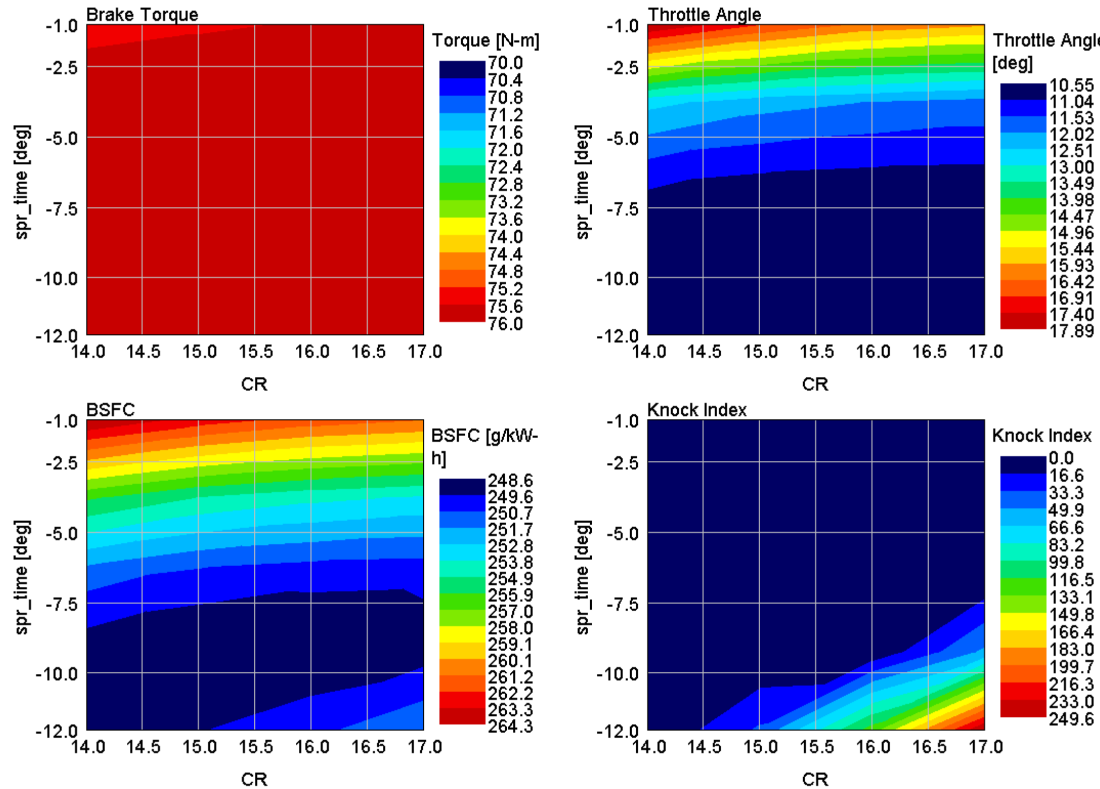
EngSim Corporation

Part Load Results – Deactivation Test

- EngSim ran a test looking at the Envera concept coupled with cylinder deactivation. The operating condition was changed to 2000 rpm, 76 Nm.
- The results show the general potential but we need a better designed test to draw conclusions.
- The proper test would be to run the same operating condition for conventional engine, Envera concept with all cylinders, and Envera concept with deactivation. This would give insight to the potential of cylinder deactivation coupled with late IVC and variable compression.



Summary and Next Steps

- Basic optimizations have been run to maximize performance at 2500 and 6500 rpm and maximize efficiency at 2000 rpm, 100 Nm
- At WOT, Envera and EngSim now have a good understanding of the relationship between boost, compression, spark, and torque
- At part load Envera and EngSim now have a good understanding of the relationship between intake duration, intake closing, compression ratio, spark, and efficiency
- EngSim will still check a few items with the optimizations but the results should be accurate given the assumptions we made.
- EngSim will also run a test to evaluate the port fuel injection. To this point all cases have been run with direct injection.

4. TURBO-VCR INVESTIGATION
EngSim Corporation – GTPower modeling

OBJECTIVES

1. Investigate power and torque performance of the 4-cylinder VCR engine with a single turbocharger.

2. A regulated 2-stage (R2S) turbocharging system was then investigated. R2S includes two turbochargers for increasing engine torque at low engine speeds.

RESULTS

1. Projected values for the single turbocharger configuration:

RPM	2500	5700	
BMEP	25.3	20.4	
Power, kW		233	(312 hp)
Torque, Nm	485	391	

2. With the single turbocharger torque falls rapidly below 2000 rpm.

3. The R2S shows potential for attaining over 25 bar bmep from 1200 rpm to 5000 rpm. These results are not conclusive because a number of design and computer modeling factors are not yet known, such as combustion quality.

ENGSIM REPORT

The EngSim Corporation report on turbocharging is provided on the following pages.

Engine Analysis for Turbocharged, 4 cylinder, 2.4L, SI Engine

Update 2

EngSim Corporation
Brad Tillock

July 14, 2015

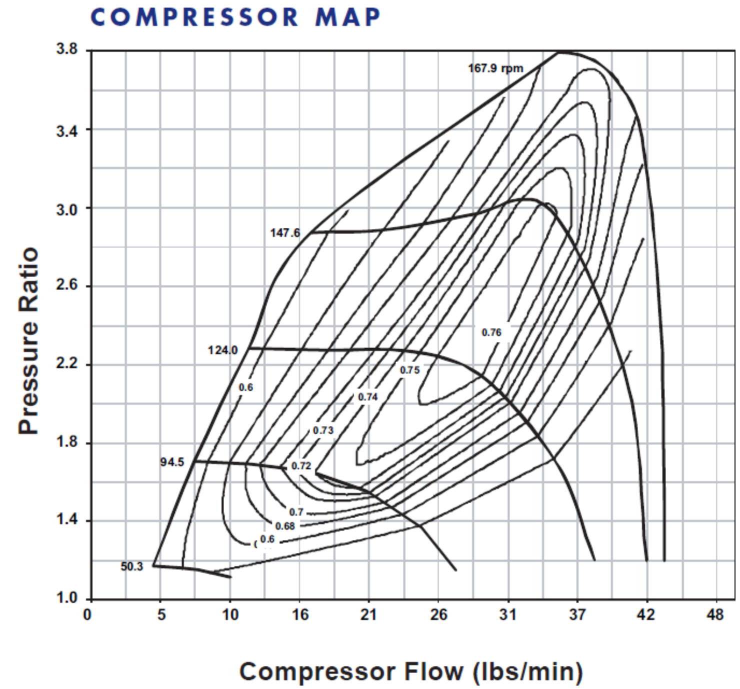
EngSim Corporation

Peak Power

- The first simulation studies will be done to size a single stage turbocharger and find optimal valve lift duration and phasing for peak power
- Peak power is to occur at approximately 5700 rpm and the target is 360 hp (268 kW)
- This update will show optimization results for varying target boost and some simple turbocharger scaling

Turbo 1

- The first turbo used in the study is shown below. Compressor map can be digitized from the BW catalog.

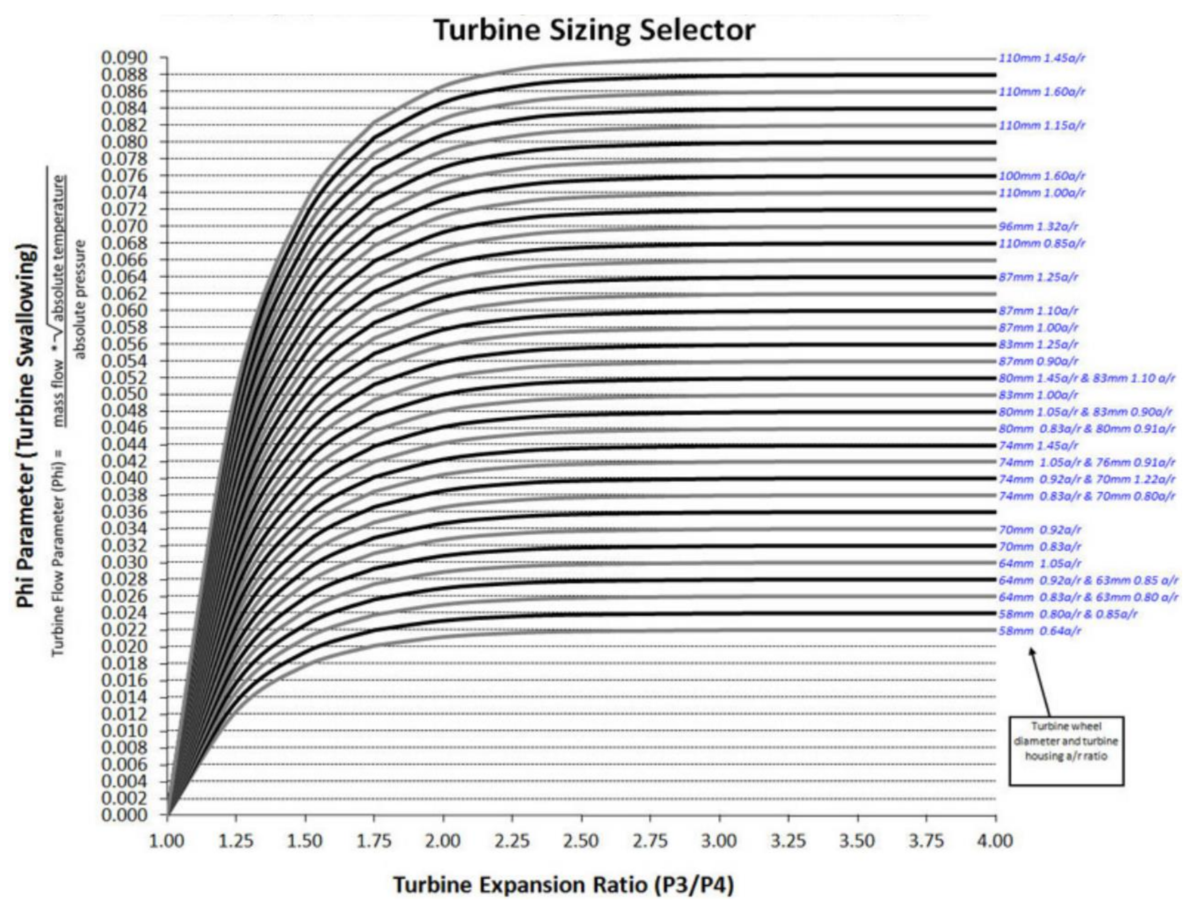


Turbo Part #	Turbo Frame Size	Super Core Part #	Comp. Wheel Outer Dia. (mm)	Comp. Wheel Inducer Dia.	Turbine Wheel Outer Dia.	Turbo A/R	Inlet Flange Config.
179150	B1	179140	62	49	58	.64	T25

Turbo 1

- The EFR series has a chart to help users pull off the correct swallowing line for each turbo. In our first case we are taking the line for the 58mm wheel and 0.8 A/R.

- The swallowing line requires a little additional work to use in GT. It is necessary to find the turbine speed at each pressure ratio in order to fall on the line. EngSim did this by varying the speeds at each pressure ratio until the BSR was approximately 0.7.



Optimization Studies

- The engine will need to be wastegated at peak power. The goal will be to find a desirable boost pressure that will meet the power requirement as well as maintain reasonable levels for turbine inlet pressure and peak cylinder pressure.

- Study 1: Optimize valve lifts duration and phasing for the following cases (the max lift was carried over from the supercharger work – 1.029 multiplier on intake and 1.078 multiplier on exhaust)
  - Target boost pressure of 2.1 bar
  - Target boost pressure of 2.2 bar
  - Target boost pressure of 2.3 bar
  - Target boost pressure of 2.3 bar with 20% increase in turbine mass map

- For this study the compression ratio was fixed at 9 and the spark timing was carried over from the previous supercharger work. Combustion parameters will be reported.

Optimization Studies

- For each case there were approximately 200 variations
- The optimal valve profile settings are not changing much with target boost or even with a 20% increase in turbine size. The exhaust duration is unchanged from the baseline but the intake wants to be shorter duration.
- Each increase in target boost leads to higher power, higher back pressure, and higher peak cylinder pressure
- An increase in turbine size lowers back pressure and increases power – this happens at every target boost although just 2.3 bar is shown

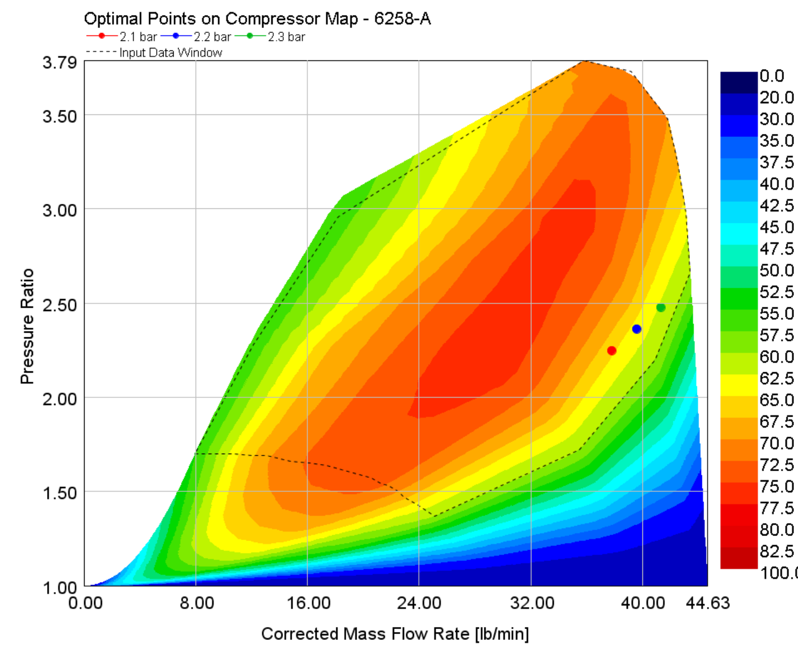
	2.1 bar boost	2.2 bar boost	2.3 bar boost	2.3 bar boost + 20% turbine
Power [HP]	363	380	396	407
Exh Scale	1.00	1.00	1.00	1.00
Int Scale	0.930	0.930	0.930	0.930
Exh Time [deg]	251	250	250	250
Int Time [deg]	470	470	470	468
Turbine Inlet Pressure [bar]	2.88	3.10	3.29	2.89
Peak Cylinder Pressure [bar]	95.1	99.8	104	106

Optimization Studies

- Each case was run with the optimal valve settings and some additional results are shown in the table
- The average operating points are also displayed on the compressor map to show the margin available between the peak power point and the choke line
- The good thing about this compressor is that pressure ratio can be maintained at 1/3 the mass flow and still avoid surge
- The 2.1 bar case is wastegating 35% (this can be supported)

Varying Target Boost

	2.1 bar - base turbo	2.2 bar - base turbo	2.3 bar - base turbo
Brake Power [HP]	364	381	397
Turbo Speed [RPM]	141000	147000	155000
Boost Pressure [bar]	2.11	2.21	2.31
Turbine Inlet Pressure [bar]	2.89	3.09	3.31
Peak Cylinder Pressure [bar]	95.4	100	105
Knock Index	0.00	0.00	0.00



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Optimization Studies (2)

- Study 2: Show the differences with turbine and compressor scaling at fixed target boost
  - Base
  - +20% mass multiplier on turbine only
  - +20% mass multiplier on turbine and compressor

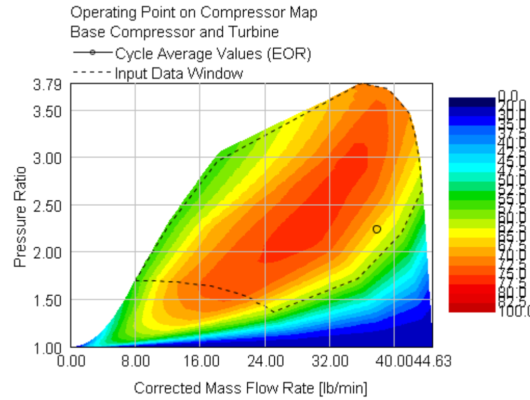
	2.1 bar - base turbo	2.1 bar +20% turbine only	2.1 bar +20% turbine and comp
Brake Power [HP]	364	369	372
Turbo Speed [RPM]	141000	141000	130000
Boost Pressure [bar]	2.11	2.11	2.10
Turbine Inlet Pressure [bar]	2.89	2.71	2.61
Peak Cylinder Pressure [bar]	95.4	95.9	96.4
Knock Index	0.00	0.00	0.00

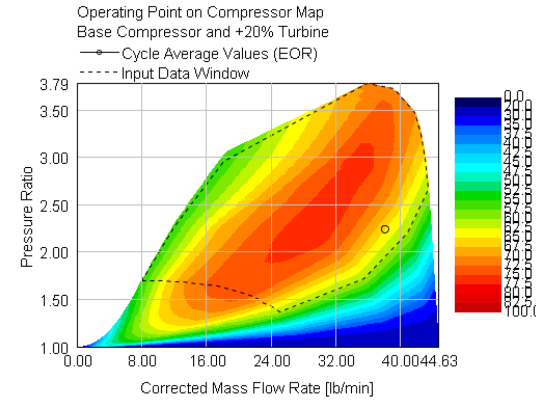
- Increasing the mass flow at given pressure ratio and speed helps to produce more engine power and reduce turbine inlet pressure

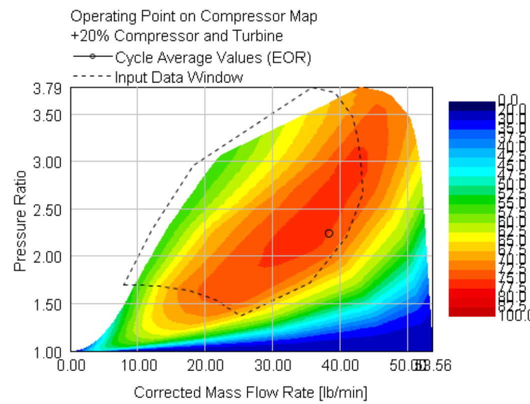
Optimization Studies (2)

- With the 20% increase to the turbine map there is no obvious drawback except it will be harder to generate boost at lower engine speeds

- With the multiplier to the turbine AND compressor (bottom) we see that you will be sacrificing some pressure ratio at lower mass flows – surge is more of an issue when trying to get peak torque







First Summary

- The first turbocharger is sized approximately correct and the system should be able to hit the target engine power
- The optimal valve durations and phasing do not change much with varying target boost or varying turbo scaling. Actually, the valve settings changed very little from the supercharged variation.
- The baseline turbo should allow for high compressor pressure ratios at low mass flows. This might allow for a fixed geometry turbine to reach the project's target torque.

Peak Torque

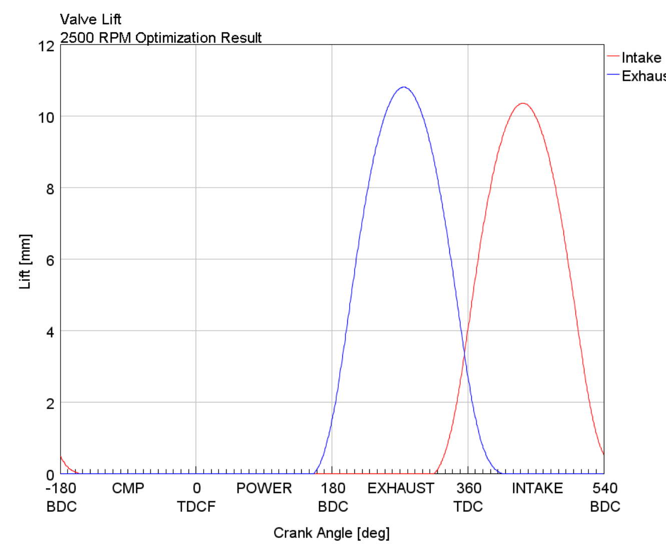
- The second simulation study will be done to find the optimal valve settings for peak torque at 2500 rpm
- The first optimization will be to run with compression ratio and spark timing fixed at 9 and -5. The valve durations and phasing will be varied and the lift multipliers will again be carried over from the supercharged result – we can vary this later but we were trying to reduce variables.
- After the valve settings are determined, the engine will be mapped as a function of compression ratio and spark timing in order to see the knock model behavior.
- EngSim will select one point off the compression and spark timing mapping and call that the optimum point for torque

Optimization Studies

- The optimal valve events for 2500 rpm are shown in the table to the right. In a later slide the 2500 rpm and 5700 rpm valve settings and curves will be displayed together

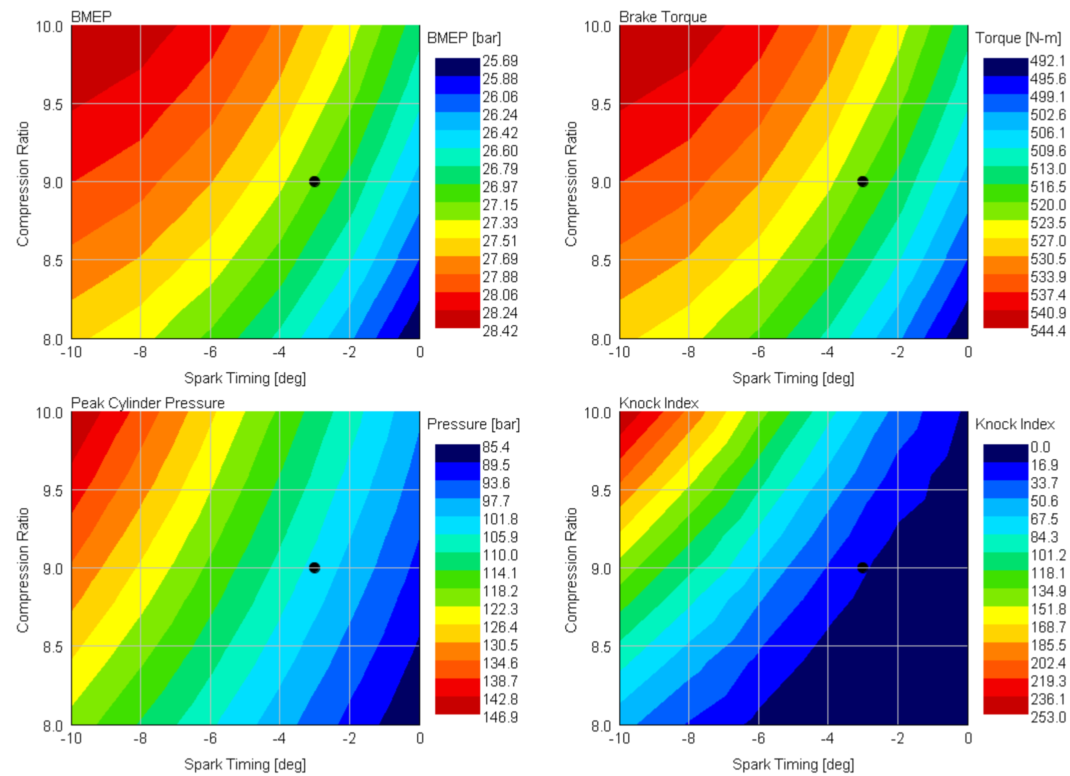
- These settings were then used in a compression ratio and spark timing mapping and those results will be shown on the following slides

	Opt Valve Settings for Torque
Boost Pressure [bar]	2.10
Turbine Inlet Pressure [bar]	1.97
Exh Open (1mm)	175
Exh Close (1mm)	376
Int Open (1mm)	336
Int Close (1mm)	532
Exh Duration Mult	0.974
Int Duration Mult	0.949



Mapping for Knock Behavior

- The knock index plot is to serve as a guide to where operation is likely to be ok.
- If we assume the first contour line is safe operation, we can say that a compression of 9 and spark timing of -3 degrees is optimal (shown as black dot), but really anywhere along that line would be ok
- Simulation shows a brake torque of 520 Nm. This is not the final number for the project because EngSim still has to make some checks on pressure drops in the system and FMEP is still very low compared to similar engines.

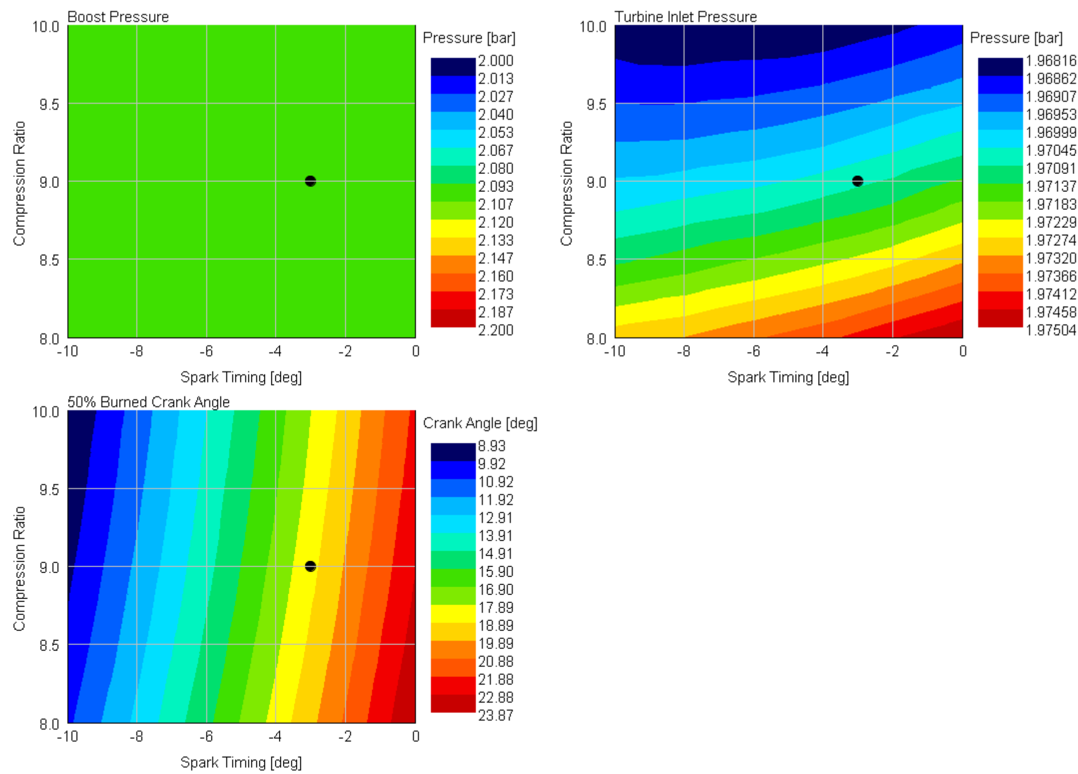


Mapping for Knock Behavior

- The boost pressure contour is shown to confirm that boost pressure is being set correctly using the wastegate controller

- Turbine inlet pressure changes very little with compression ratio and spark timing. The good news is that the match at this condition is much better than peak power. Now back pressure is lower than boost.

- The 50% burn point indicates the burn profile is retarded about 12-14 degrees from optimal timing



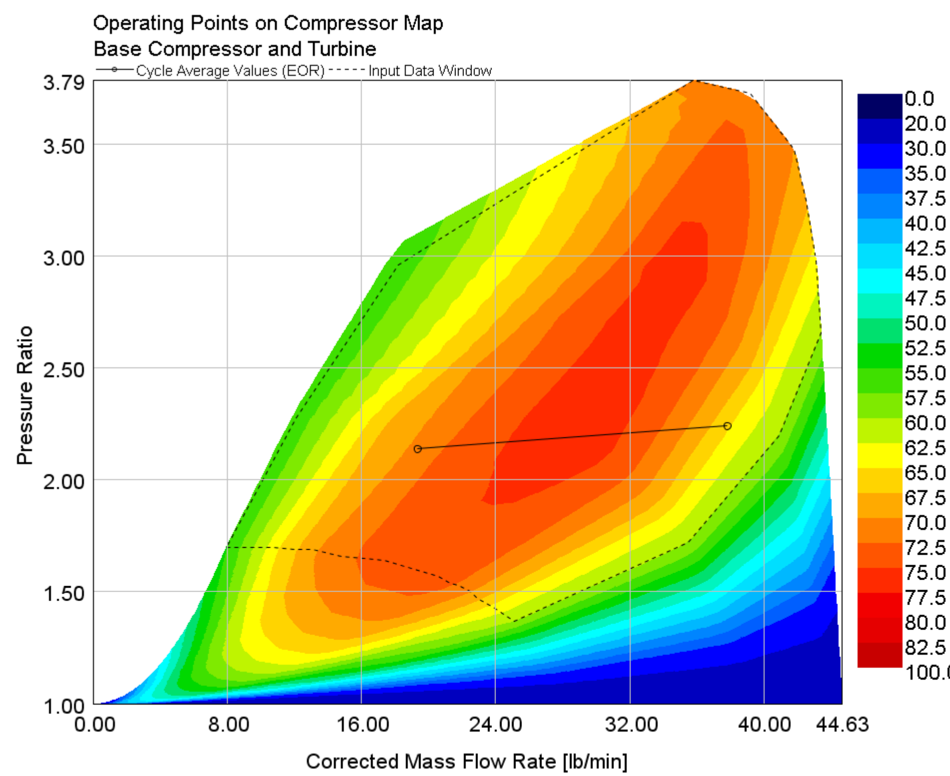
Overall Torque and Power Summary

- The following slides will show optimal valve settings and results for both the peak power and peak torque points

Operation on the Compressor Map

- The operating points on the compressor map are quite encouraging. The elevated boost pressure could be carried to lower rpm and still have some surge margin.

- There is also pressure ratio margin above the points. If intake losses cause the compressor pressure ratio to rise, we will still be in the map.

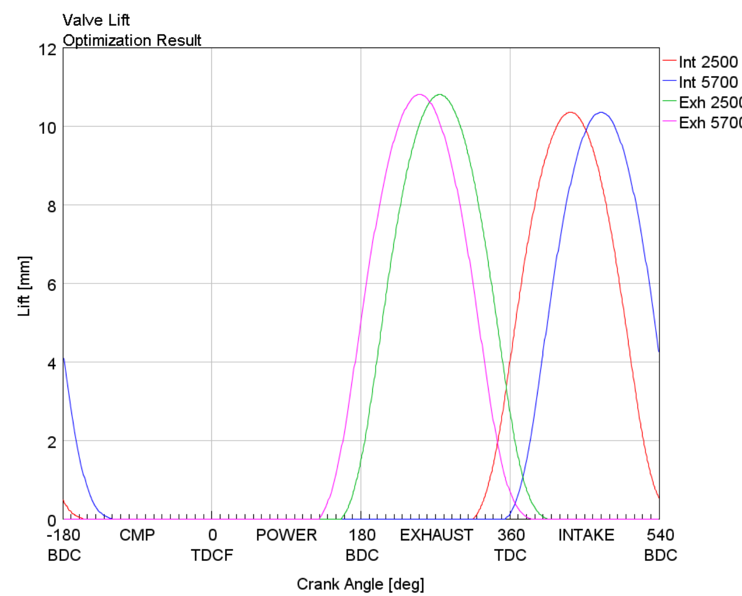


Valve Profiles and Settings

- The optimal valve settings are shown in the table to the right and the full profiles are plotted below

- In terms of phasing, the exhaust needs about 24 degrees of shift and the intake needs about 38 degrees of shift

	Opt Valve Settings for Torque	Opt Valve Settings for Power
Boost Pressure [bar]	2.10	2.10
Turbine Inlet Pressure [bar]	1.97	2.89
Exh Open (1mm)	175	148
Exh Close (1mm)	376	355
Int Open (1mm)	336	375
Int Close (1mm)	532	567
Exh Duration Mult	0.974	1.00
Int Duration Mult	0.949	0.930



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Overall Results

- Some of the important engine quantities are shown to the right

- The next step will be to check to make sure the compressor inlet pressure and turbine outlet pressure are still reasonable

- After some model checks, EngSim will run the runner study

	Opt Valve Settings for Torque	Opt Valve Settings for Power
Engine Speed [rpm]	2500	5700
BMEP [bar]	27.2	23.7
Brake Power [kW]	136	271
Brake Torque [Nm]	521	455
Boost Pressure [bar]	2.10	2.10
Turbine Inlet Pressure [bar]	1.97	2.89
Wastegate Diameter [mm]	11.2	20.5
Peak Cylinder Pressure [bar]	105	95.3
Knock Index	18.8	0.00

Model Modifications

- EngSim has made some changes to the base model in order to make the model more realistic
- Combustion – The spark timing has been delayed in order to match the MFB50 numbers from the Ford DOE Program Review. They are reporting MFB50 between 24 and 27 degrees for the full load sweep.
- The charge cooler is assumed to be air-to-air with an effectiveness of 0.85 and slave side air temperature of 90 F
- Losses in the exhaust system have been tuned to create a 1.33 bar static pressure at the turbine outlet at peak power. We can adjust this if needed.
- Losses in the intake have been tuned to result in 0.9 bar total pressure at the compressor inlet
- The charge cooler pressure loss has been modified to result in 5 kPa pressure loss at peak power – this might have to be increased
- Changed intake system pipe lengths to be more realistic

Overall Torque Results - New

- The original presented results are shown along with the modified results. The modifications are listed on the previous slide. Additionally, EngSim ran a case with the friction model inputs changed to what is typical for ic engines
- The model is still targeting 2.1 bar boost so engine output can be increased by increasing boost, if needed
- The modified result is at 206 Nm/l

Original

	Opt Valve Settings for Torque
Engine Speed [rpm]	2500
BMEP [bar]	27.2
Brake Power [kW]	136
Brake Torque [Nm]	521
Boost Pressure [bar]	2.10
Turbine Inlet Pressure [bar]	1.97
Wastegate Diameter [mm]	11.2
Peak Cylinder Pressure [bar]	105
Knock Index	18.8

Modified

	Modified	Modified w/friction change
Engine Speed [rpm]	2500	2500
BMEP [bar]	25.8	25.3
Brake Power [kW]	130	127
Brake Torque [Nm]	495	485
Average Pressure [bar]	2.10	2.10
Turbine Inlet Pressure [bar]	1.95	1.95
Wastegate Diameter [mm]	11.2	11.2
Peak Cylinder Pressure [bar]	93.2	93.2
Knock Index	1.76	1.76
FMEP	0.834	1.36

Overall Power Results - New

- The original presented results are shown along with the modified results. The modified engine is still producing 100 kW/l.
- For reference, this condition is wastegating 34.5% which is acceptable for the EFR series

Original

	Opt Valve Settings for Power
Engine Speed [rpm]	5700
BMEP [bar]	23.7
Brake Power [kW]	271
Brake Torque [Nm]	455
Boost Pressure [bar]	2.10
Turbine Inlet Pressure [bar]	2.89
Wastegate Diameter [mm]	20.5
Peak Cylinder Pressure [bar]	95.3
Knock Index	0.00

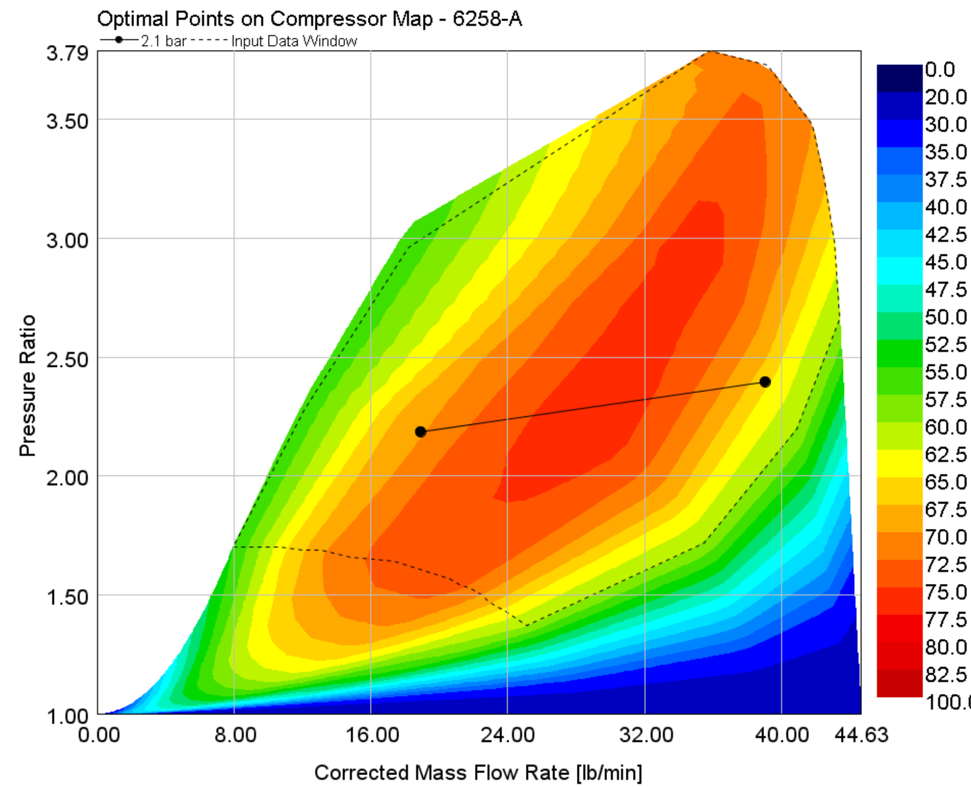
Modified

	Modified	Modified w/friction change
Engine Speed [rpm]	5700	5700
BMEP [bar]	21.3	20.4
Brake Power [kW]	243	233
Brake Torque [Nm]	408	391
Average Pressure [bar]	2.10	2.10
Turbine Inlet Pressure [bar]	2.93	2.93
Wastegate Diameter [mm]	19.8	19.8
Peak Cylinder Pressure [bar]	74.7	74.7
Knock Index	0.00	0.00
FMEP	1.41	2.29

Modified Results on the 6258 Map

- The modified results still show a little margin for the peak power point and the ability to carry the elevated boost pressure to lower rpm

- It is not certain if the choke margin is enough for production. The engine will have some issue maintaining that boost at engine speeds higher than 5700 rpm.



Modified Model Speed Sweep – 6258-A

- A speed sweep was run from 1500 rpm to 5700 rpm using the modified model

- The cam settings were interpolated between the 2500 rpm and 5700 rpm optimized values. The cam settings for 2500 rpm were used at rpms below 2500 rpm.

- The wastegate controller was modified to target BMEP. The goal was to obtain 25 bar BMEP at all engine speeds below 5700 rpm.

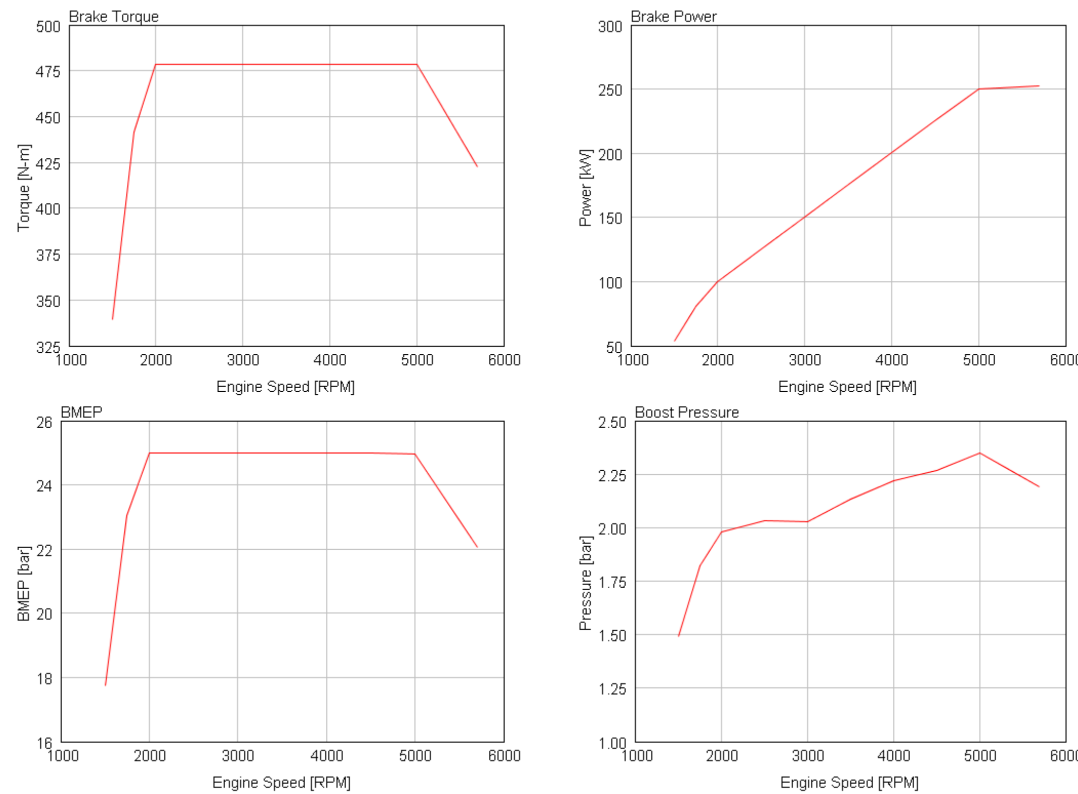
- A 25 bar BMEP target at 5700 rpm gets too close to the choke line

Modified Results

- The speed sweep results are shown on the next few slides

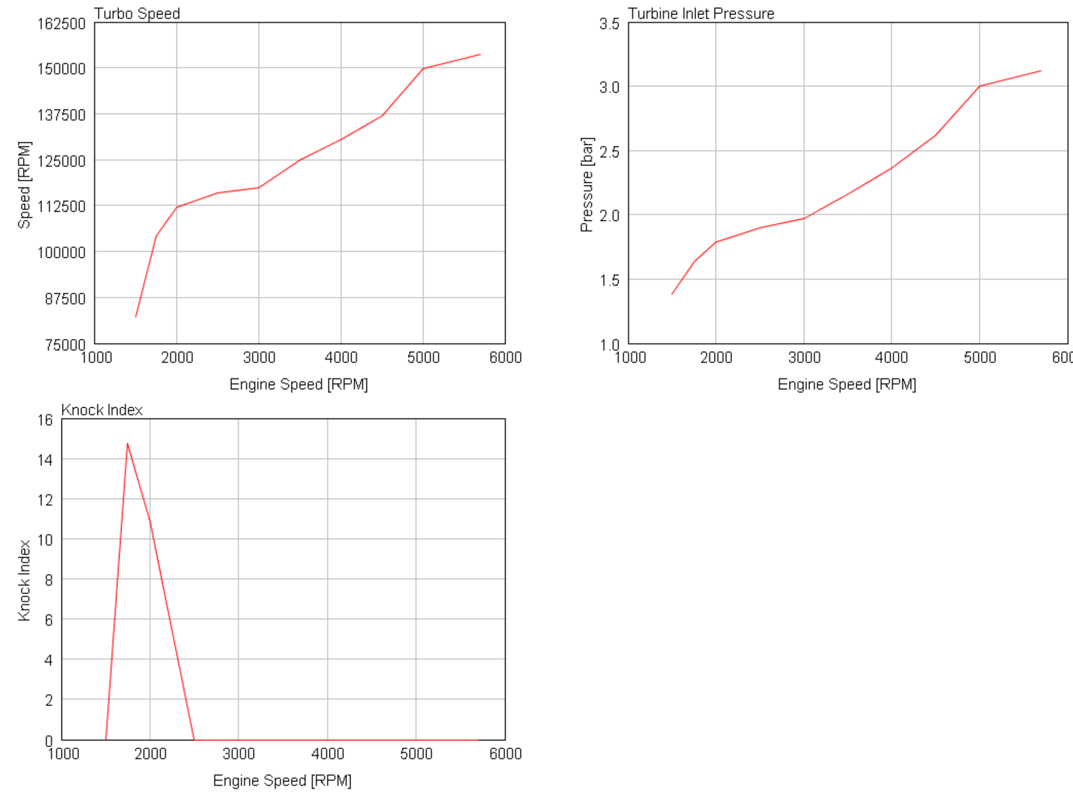
- The brake torque could be increased some at 5700 rpm. We would just have to be careful how close we get to the choke line (slide 26).

- Below 2000 rpm, the turbine wheel would have to be reduced to generate more boost. There is some surge margin to increase boost.



Modified Results

- Turbo speed, turbine inlet pressure, and knock index are shown for reference



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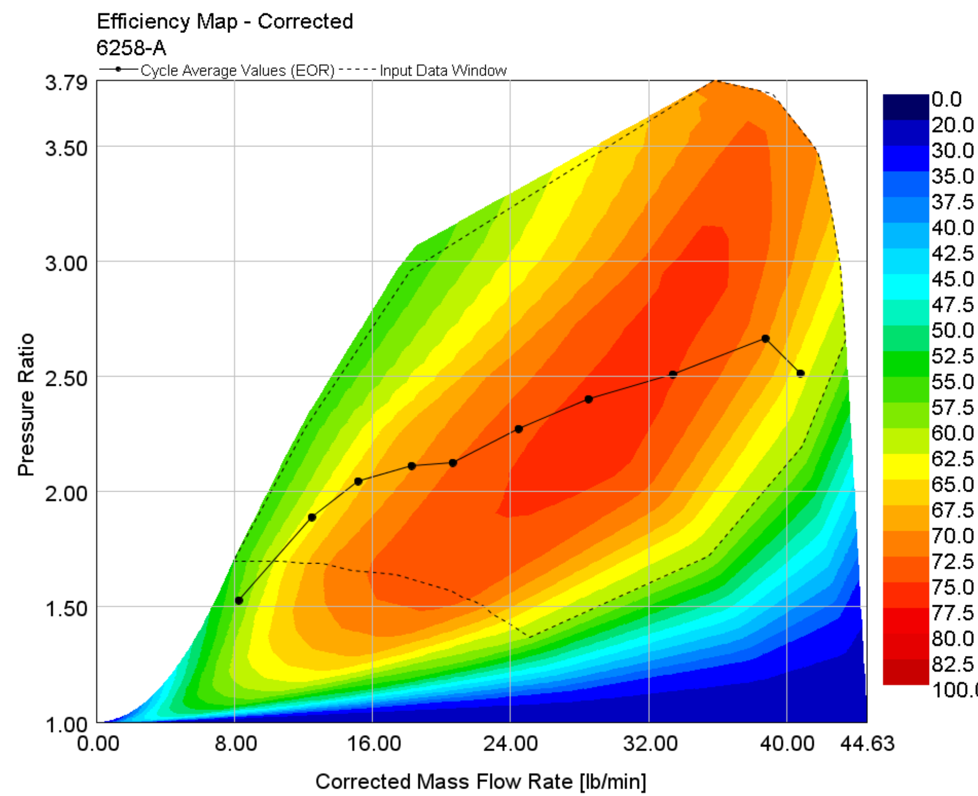
Modified Results on the 6258 Map

- The speed sweep is shown on the 6258-A compressor map

- The engine operation for most mid to high speed points is not limited by the boosting system. Target pressures could go higher if the system can deal with the pressures and avoid knock.

- The boosting system will limit the engine operation at speeds greater than 5700 rpm



R2S Design – Initial Tests

- The 58mm turbine with A/R=0.8 can generate enough boost at all speeds above 2000 rpm. The wastegate controller can target 22 bar BMEP at 5700 rpm with 35% wastegate flow percentage. That BMEP equates to approximately 105 kW/l.

- At speeds between 2000 rpm and 5000 rpm, the wastegate controller can target 25 bar BMEP. There is plenty of margin to go higher in BMEP if the engine system can tolerate the increased peak pressures and avoid knock. The boosting system is not limiting the peak torque.

- Below 2000 rpm, the turbine would have to be downsized to generate more boost (wastegate is closed)

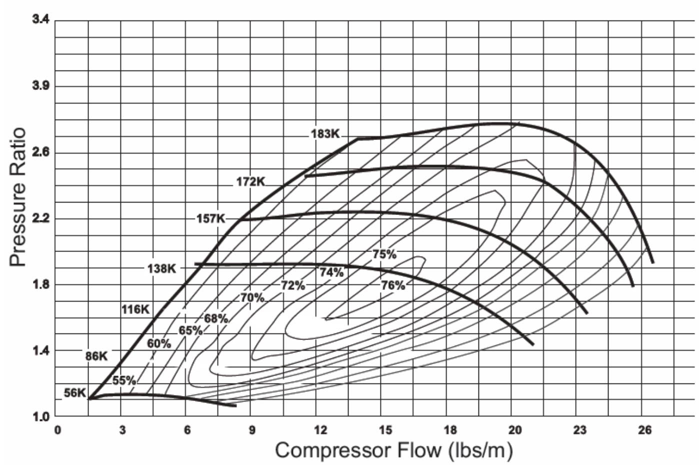
- A smaller, high pressure stage turbo could generate more boost and provide more surge margin for engine speeds below 2000 rpm

R2S – Proposed High Pressure Stage

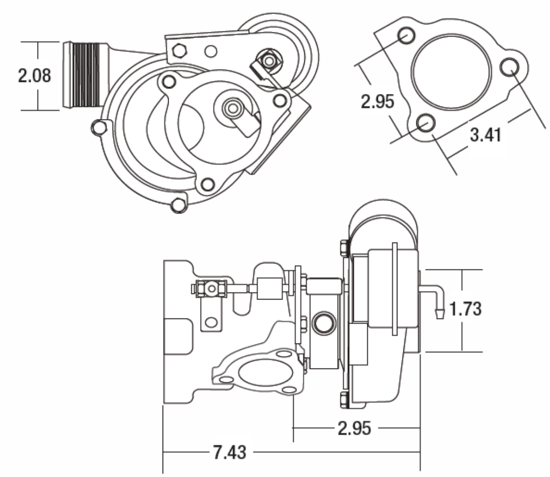
- The first high pressure stage to be tested is the K03-2075, using a 42mm turbine wheel. The compressor map is digitized from the BW catalog.
- The turbine swallowing curve will have to be estimated

K03-2075

COMPRESSOR MAP



TURBO FRAME DIMENSIONS



Turbo Part Number	Comp. Wheel O.D	Comp. Wheel Inducer Dia.	Comp. Wheel Inducer Dia. (mm)	Turbine Wheel OD	Turbine Wheel Exducer	Turbine Wheel Exducer (mm)	Turbine A/R	Cartridge Assembly	Service Kit
5303 988 0073	2.00	1.50	38	1.81	1.65	42	5²cm	5303 710 0521	

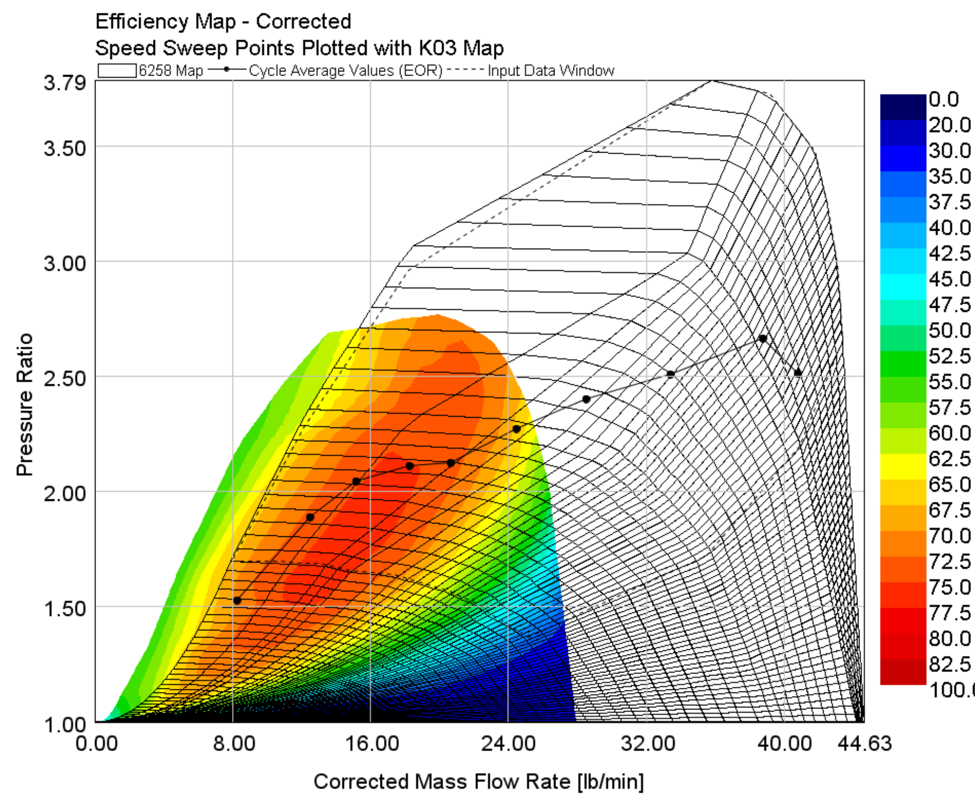
EngSim Corporation

Map Comparison

- One interesting view is to look at the single stage speed sweep operating points in relation to both the 6258 and K03 compressor maps. The contour shading had to be turned off for the 6258 map in order to see everything.

- The smaller turbine wheel should allow for more boost generation below 16 lbm/min

- It would be necessary to bypass the high pressure stage between 3000 and 3500 rpm



R2S First Sweep

- The real R2S development will take much longer than the 2 or 3 days available in this study but some good results have been generated

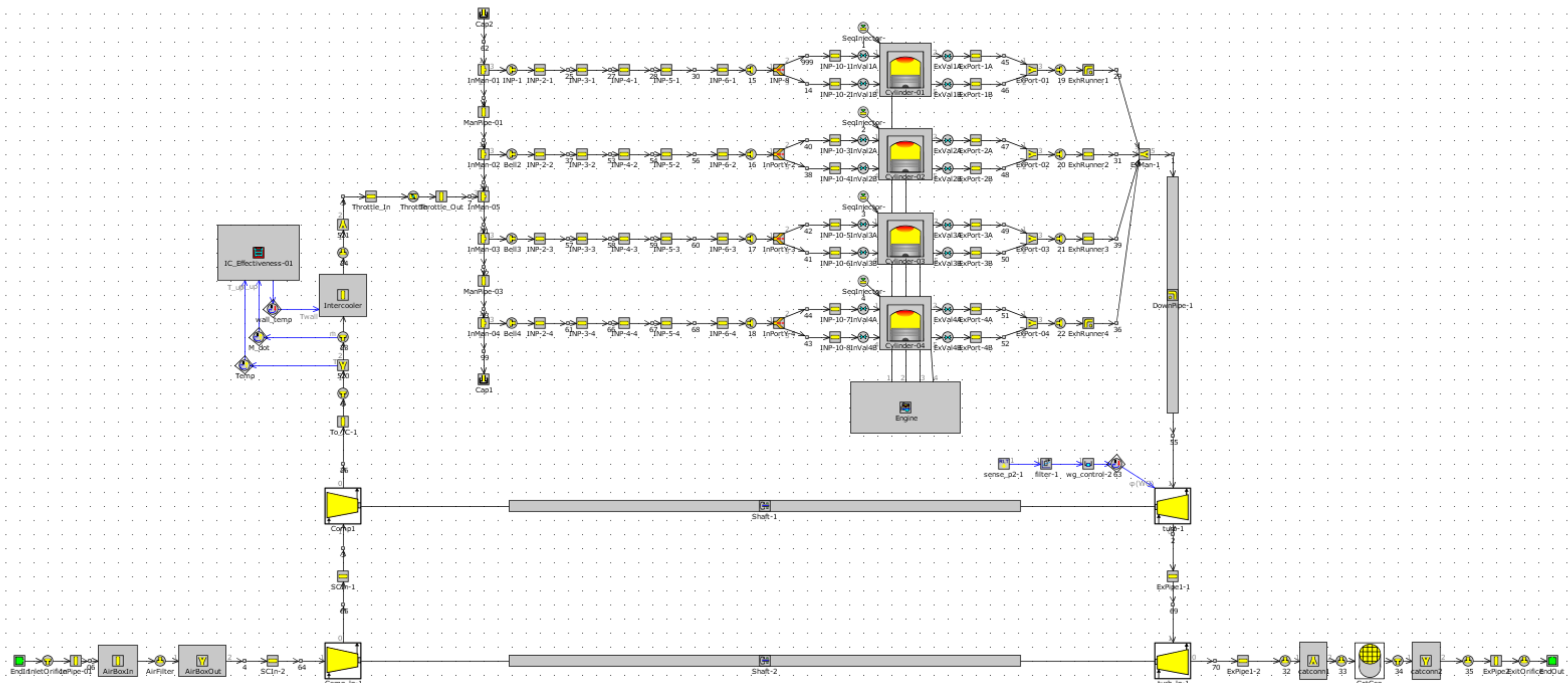
- The K03-2075 is used as the high pressure stage and the 42mm swallowing curve was estimated to be about 72% of the 58 mm swallowing curve

- The low pressure stage started out as the 6258-A compressor with 58 mm wheel. After some running it made sense to make the low pressure stage a little bigger so that it did not have to be wastegated so much at the low engine speeds simulated. Envera and EngSim will have to check on the correct operation.

- The 6258-A compressor and turbine maps were scaled up 20%. There will be a turbocharger that is approximately 20% bigger than 6258-A.

R2S First Sweep

- A simplified 2-stage system has been built in GT-SUITE. In the figure below the turbine bypasses are internal to the components and the compressor bypass has not been added because the model will not be run in the compressor bypass portion of the map.



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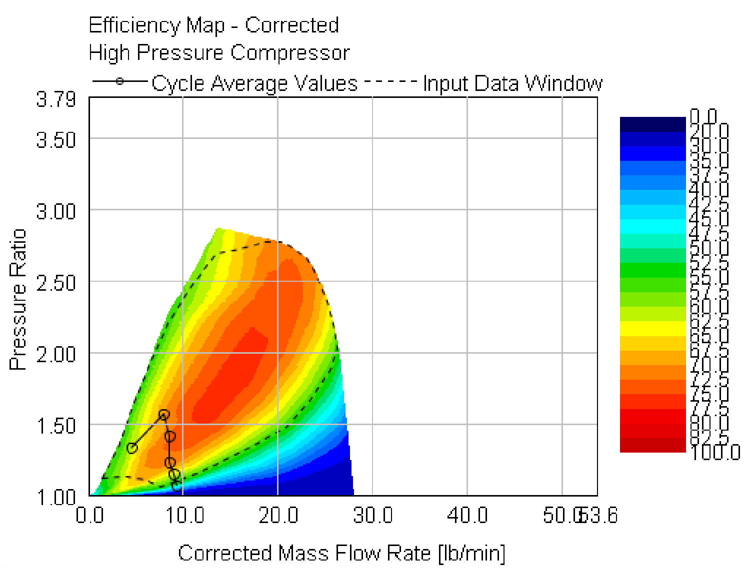
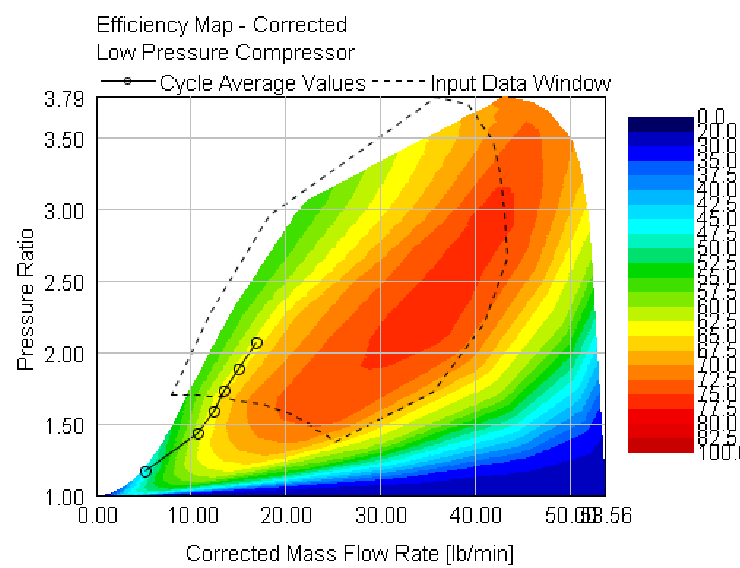
R2S First Sweep

- The model was run from 1000 rpm to 2250 rpm. The low pressure turbine wastegate was closed for all cases. Somewhere around 2500 rpm, the model would completely bypass the high pressure stage and the boost control would be done by the low pressure wastegate instead of the high pressure wastegate.
- With the current control strategy the high pressure bypass was already transitioning out. The real operation might wastegate the low pressure stage some at these rpms.

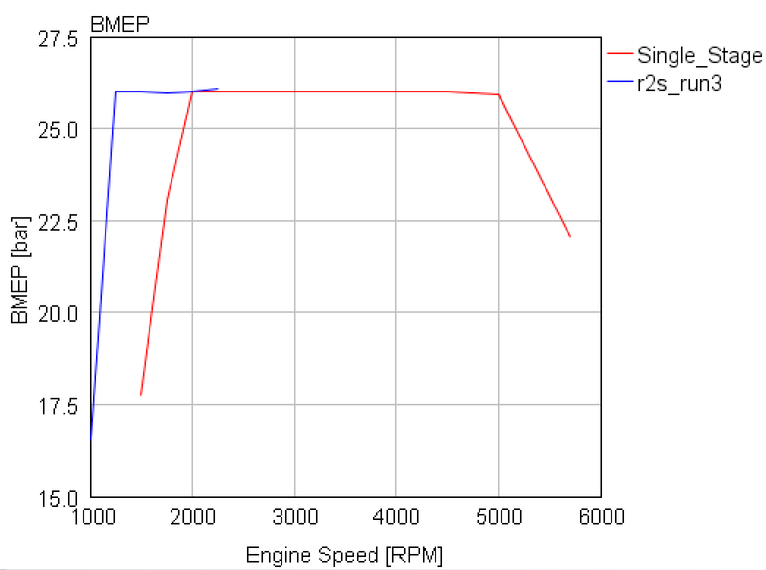


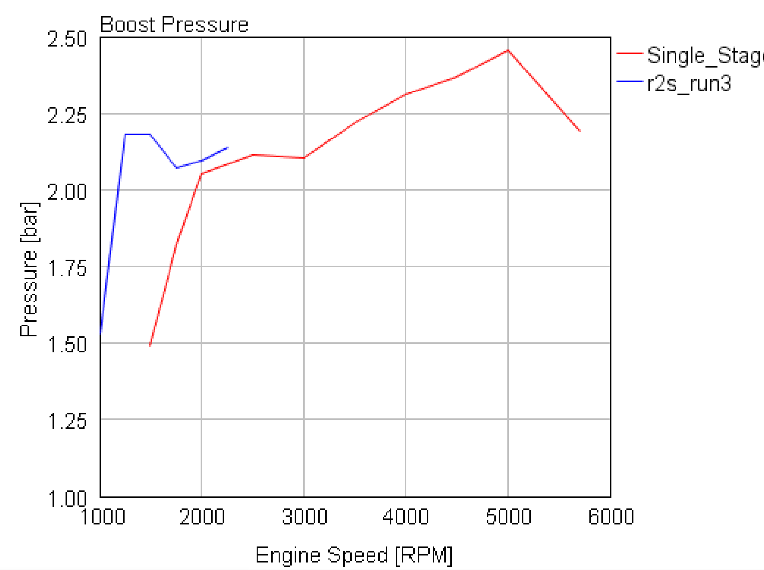


R2S First Sweep

- The figures below show the R2S modifications to the single stage BMEP and boost curves

- The big unknown is the combustion quality at these lower rpms. The model is using an uncorrelated SI combustion model. The model results below depend on a quality combustion system. It appears the boosting system can be fine tuned to create the boost needed.





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5. TURBOCHARGER TIME-TO-TORQUE (TTT)
EngSim Corporation – GTPower modeling

OBJECTIVES

1. Investigate turbocharger TTT, commonly known as turbo lag for a single-turbocharger engine build. Ford recommended evaluating TTT at 1600 rpm commencing from 5 bar bmep and measuring the time required to reach 90 percent of peak bmep.

RESULTS

1. The baseline TTT value at 1600 rpm starting from 5 bar bmep was approximately 1 second.

FINDING

Time-to-torque is not the primary challenge for turbocharged engine performance. The primary challenge is the ability of the turbocharger to attain high boost pressures at low engine speed with a turbocharger sized large enough for attaining maximum power requirements.

Turbo lag implies a slow turbocharger response. The single stage turbocharger that was evaluated does not have an inertial lag problem. However, a turbocharger large enough to attain 300 to 360 hp is unable to generate high boost pressures at low engine speeds due to the surge limitations of the turbocharger. The vehicle is experiencing lag because the turbocharger lacks performance capability at low engine speeds. Vehicle lag is primarily due to lack of boost pressure, not turbocharger inertial response.

ENGSIM REPORT

The EngSim Corporation report on turbocharger TTT is provided on the following pages.

Time to Torque Baseline and Inertia Effects

Brad Tillock
EngSim

Time-To-Torque and Max dP

- Envera wants to get estimates for the baseline time-to-torque as well as understand some variables that might influence both time-to-torque and the maximum rate of change of pressure in the cylinder.

- A throttle step transient was set up at a constant 1600 rpm. A throttle controller targeted 5 bar BMEP before the step. At the time of the step, the throttle jumped to full open. The time required to reach 90% of peak engine BMEP will be recorded.

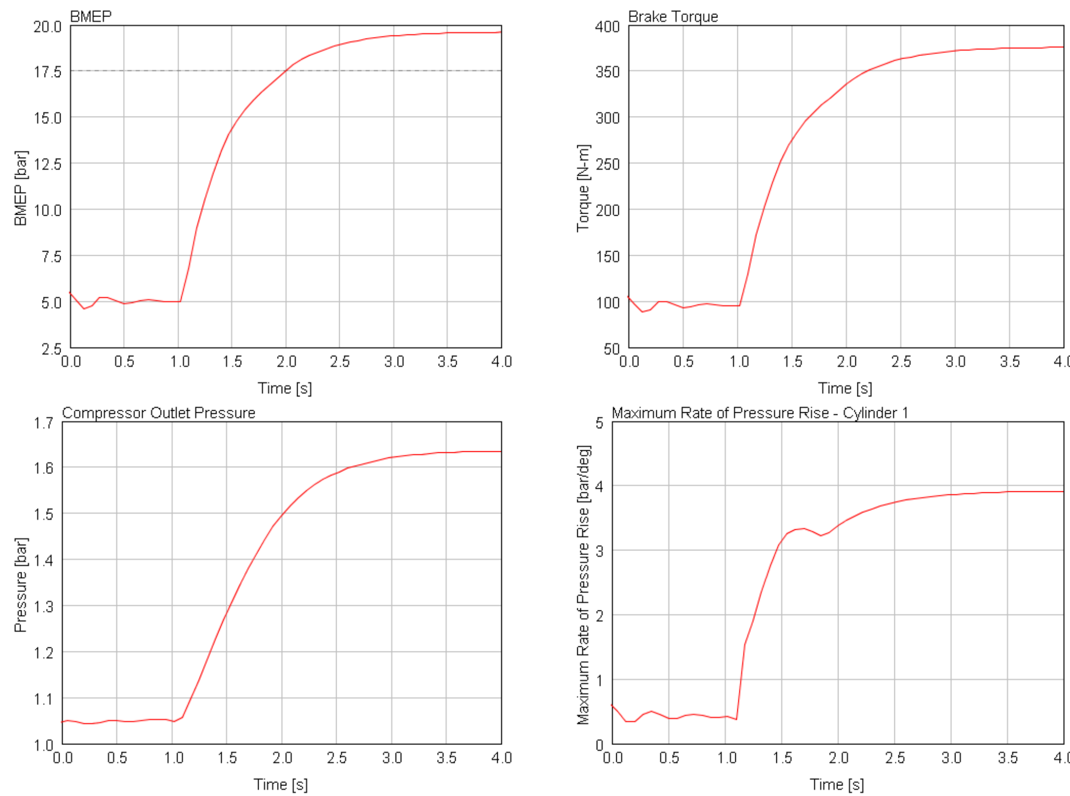
- These cases are all run at a constant compression ratio of 9 and a valve timing/strategy for best peak torque. AFR and spark timing vary based on the current engine load.

- In addition to baseline values for time-to-torque and max dP, the report will also show the effects of varying turbo inertia and varying spark timing.

Baseline

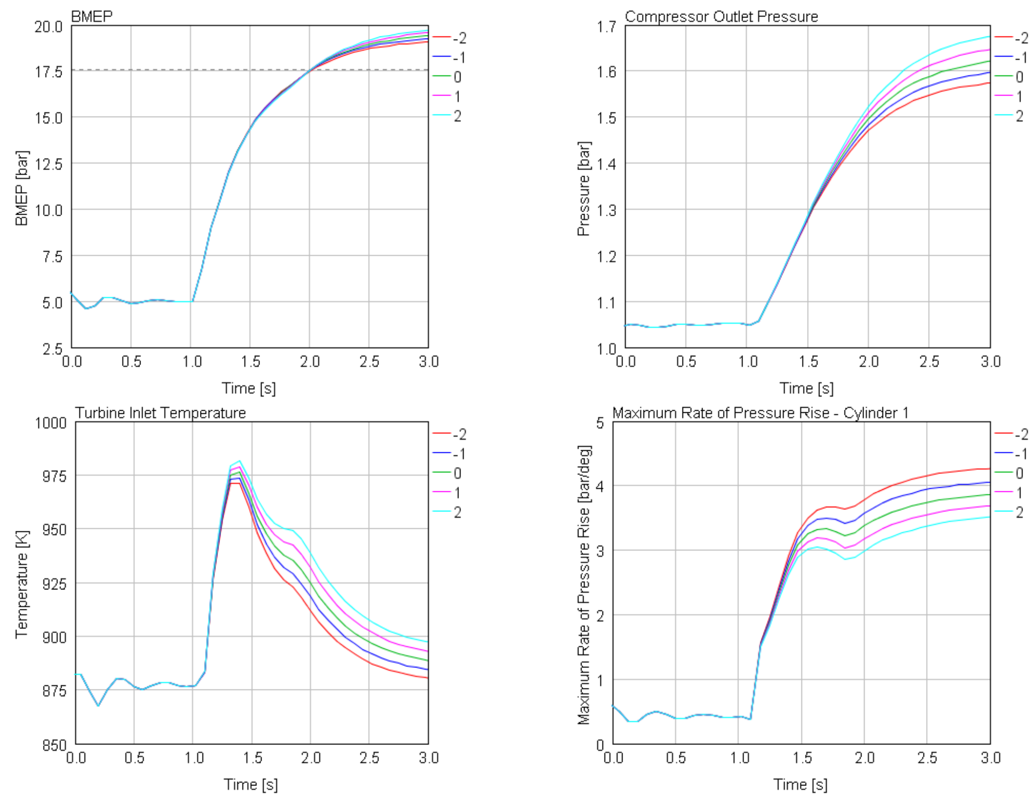
- The baseline time-to-torque for this engine at 1600 rpm is almost exactly 1 second

- The baseline maximum pressure rise is 3.9 bar/deg



Spark Sweep Effects on TTT and Pressure

- For these tests spark timing was set to -10 before the throttle step. After the throttle step the spark timing varied linearly from -10 to -2, -1, 0, 1, 2
- Spark timing itself did not have much effect on the time to torque but it does affect the final BMEP value as well as the max rate of change of pressure in the cylinder.
- Later spark is leading to more energy at EVO, leading to more boost, and then more BMEP
- Spark is also a good means to control the maximum rate of change of pressure. In our sweep max values ranged from about 4.25 to 3.5 bar/deg.

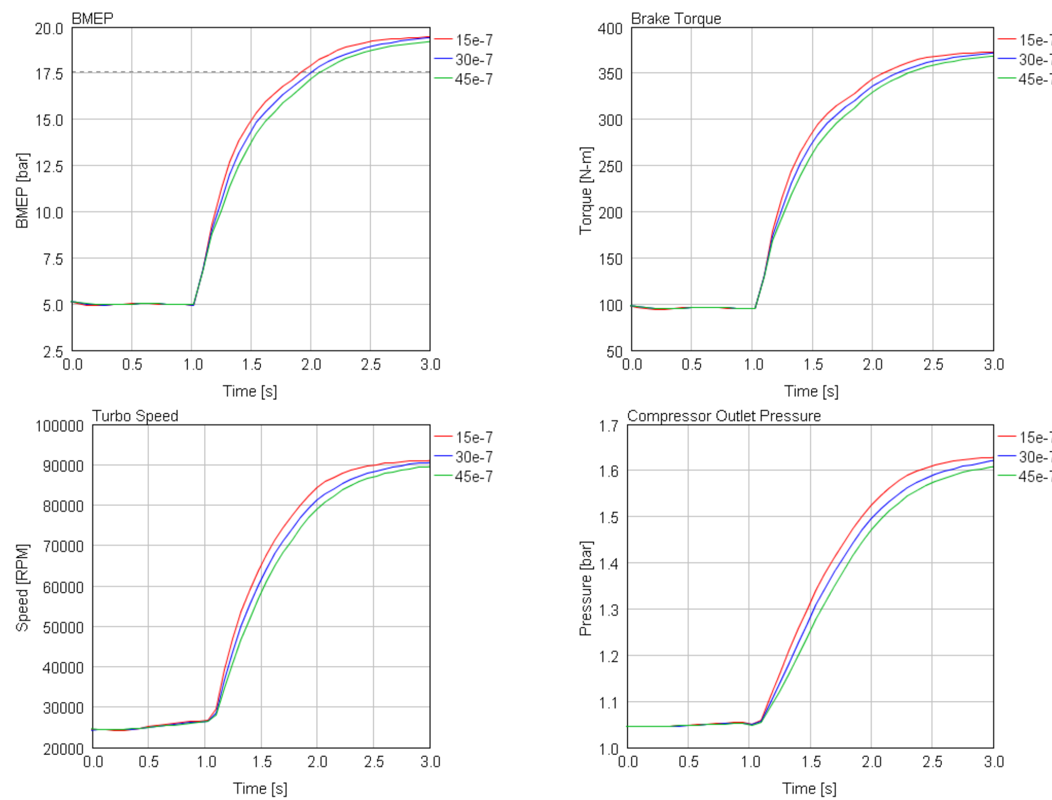


ESEngSim

Effect of Turbo Inertia

- In order to test the sensitivity of the engine to turbo inertia, EngSim ran with the baseline inertia value as well as + and – 50%

- The time-to-torque values for -50%, baseline, and +50% were 0.93 sec, 1.0 sec, and 1.1 sec



Summary

- Report shows baseline transient response of the engine as well as the effects of spark timing and turbo inertia

- Spark timing is a good way to control the maximum rate of change of pressure in the cylinder. The results indicate a follow-up study might be in order. Perhaps a jump to very delayed spark timing might be a way to reduce the time-to-torque if there is a large torque request.

- We may need to revisit spark timing at low engine speeds where the wastegate is closed. There may actually be benefit in running a further delayed spark in order to create more turbine power and boost.

- Compression ratio sweeps were also run but the result is counter-intuitive. Reduction of compression ratio is resulting in higher max dP. Compression of 8.2 causes a delayed/slower combustion, more energy at EVO, and therefore more boost. The feedback effect of the turbo causes compression pressure to be higher with the lower compression ratio. We can discuss how to handle this.

6. BOOSTING SYSTEM OPTION ASSESMENT & DOWN-SELECT
Envera LLC

The objective of this task was to assess boosting options and down select the best option for the program. Boosting systems that were considered include:

- Variable speed CVT supercharger
- Combined electric / mechanical drive supercharger
- Clutched supercharger
- Supercharger with advanced cooling
- Turbocharging and supercharging combined
- R2S regulated 2-stage turbocharging
- Single stage turbocharging

The variable speed CVT supercharger and the electro/mechanical supercharger did not provide large enough benefits relative to their cost and complexity. The key issue is that the size of the supercharger is driven by maximum power requirements. Therefore the variable speed drive is not an enabler for using a smaller supercharger that has higher efficiency at low engine speeds.

The clutched supercharger was selected for the current program. The ability to clutch out the supercharger is beneficial for minimizing friction FMEP and pumping PMEP losses when the supercharger is not in use. The supercharger can also be engaged very quickly. A good clutch is needed.

A supercharger with advanced cooling was investigated, and is shown in Figure 14 on page 14. This technology may be beneficial for future engine builds. Pursuit of the technology was beyond the scope of the current program.

Combining turbo and supercharging provides good performance but the system is somewhat complex and costly.

Regulated 2-stage (R2S) turbocharging is also somewhat complex and costly.

Single stage turbocharging does not have an inertial lag problem. However, a turbocharger large enough to attain 300 to 360 hp is unable to generate high boost pressures at low engine speeds due to the surge limitations of the turbocharger.

7. INTAKE HIGH-LIFT CAM AND ROCKER DEVELOPMENT
Eaton Corporation

ATKINSON CYCLE POPPET VALVE DEVELOPMENT

The Envera VCR engine employs a switchable rocker arm mechanism to enable switching between an Atkinson Cam lobe profile and Power Cam lobe profile. The Atkinson cam has a long duration enabling the intake valves to close late. The Atkinson cam profile must have a higher lift than the power cam for mechanical clearance reasons. The development effort under this task was for the high-lift Atkinson Cycle cam and rocker arm development.

Target cam timing values were developed by EngSim Corporation and Envera using GTPower engine development software. These values were then provided to Eaton for design and development of the poppet valve assembly. Three different cam duration values were developed for testing. The final valve lift profiles were then entered back into GTPower.

OBJECTIVES

1. Intake high-lift kinematic analysis and development of Atkinson cycle cam profiles.

2. Procure camshaft blanks

3. Produce machine cam grinding drawings for the camshafts

RESULTS

All objectives were satisfactorily met for the current program. The Intake High Lift Kinematic Analysis report number 2015-23 from Eaton addressed the following items:

- Cam design
- Rocker arm study
- Spring design study
- Spring load requirements
- Valve tip stresses
- Contact stresses
- Lash adjuster loads

8. INTAKE LOW-LIFT CAM AND ROCKER DEVELOPMENT
Eaton Corporation

POWER CAM POPPET VALVE DEVELOPMENT

The Envera VCR engine employs a switchable rocker arm mechanism to enable switching between an Atkinson Cam lobe profile and Power Cam lobe profile. The Atkinson cam has a long duration enabling the intake valves to close late. The power cam profile must have a smaller lift than the Atkinson cam for mechanical clearance reasons. The development effort under this task was for the low-lift power cam and rocker arm development.

Target cam timing values were developed by EngSim Corporation and Envera using GTPower engine development software. These values were then provided to Eaton for design and development of the poppet valve assembly. Three different cam duration values were developed for testing. The final valve lift profiles were then entered back into GTPower.

OBJECTIVES

1. Intake low-lift kinematic analysis and development of the power cam profiles.
2. Procure camshaft blanks
3. Produce machine cam grinding drawings for the camshafts

RESULTS

All objectives were satisfactorily met for the current program. The Intake Low Lift Kinematic Analysis report number 2015-24 from Eaton addressed the following items:

- Cam design
- Rocker arm study
- Spring design study
- Spring load requirements
- Valve tip stresses
- Contact stresses
- Lash adjuster loads

9. ENGINE MANAGEMENT SYSTEM
Envera LLC

OBJECTIVE

1. Procure a fully functional closed-loop engine management system (EMS) for running the engine.

2. Produce required support documentation for operation of the engine.

3. Benchtop preprograming of the ECU

RESULTS

1. The following documents were produced for operation of the EMS

    - Motec First-Fire manual
    - Engine Management Build Sheet
    - EMS Bill of materials (BOM)
    - Loom
    - Wiring harness
    - Pin I/O connections
    - Spark timing – Startup table
    - Fuel injection – Startup table
    - Engine brake-in test schedule

2. An intermediate manifold including port fuel injection was designed and procured to enable port fuel injection and/or port fuel injection plus direct fuel injection. Fuel injector selection and targeting was performed.

3. All items on the BOM were procured.

4. The Motec ECU was benchtop preprogramed and readied for delivery to Hasselgren Engineering for wiring harness assembly and engine startup.

10. ENGINE BUILD AND FIRST FIRE
Hasselgren Engineering

OBJECTIVE

1. Engine Build

2. Engine startup including mechanical and EMS shake down

3. Engine teardown inspection and rebuild

RESULTS

All tasks were completed.

11. TESTING AT MAHLE POWERTRAIN
Mahle Powertrain North America

OBJECTIVES

1. Record actual combustion burn rates and friction fmep values, as needed for calibrating the GTPower model.

2. Through testing find the optimum combination of variables for maximizing brake thermal efficiency at 2000 rpm and ~100 Nm torque. Variables include: Cam timing, spark timing, EGR mass flow percent and compression ratio. *Optimization Level 1*

TEST EQUIPMENT

- AC Engine dynamometer
- Kistler 6052 pressure transducers & amps
- AVL IndiCom combustion analysis
- AVL AMA i60 emissions measurement.
- AVL PUMA data compilation

RESULTS

1. Combustion burn rate and fmep values were recorded. The GTPower model was then calibrated for further optimization of engine settings.

2. Engine load conditions were light enough for the engine to not be detonation limited with a compression ratio of 16.8:1 and a brake torque of ~120 to 90 Nm

3. High levels of EGR were not needed for attaining minimum fuel consumption at ~100 Nm. With only modest levels of EGR used combustion COV values were favorably low.

4. FMEP values were higher than state of the are production engines. These values were high due use of a high flow oil pump and other factors. A high flow oil pump was used to increase reliability and robustness of the first build prototype engine. Approximately a 5 percent improvement in BSFC can be realized in future builds by reducing fmep from ~1.034 bar to ~0.73 bar (Optimization Level 3).

5. An Optimization Level 1 BSFC value of 230.4 g/kWh was achieved at 5.36 bar BMEP, a value close to the target value of 230 g/kWh at 5.25 bar BMEP. Representational test data is provided on the following page.

   Optimization Level 2 includes GTPower optimization at target load conditions using the calibrated GTPower model. Optimization Level 3 includes lowering fmep values through oil pump and other fmep related improvements.

TEST RESULTS FILE NO:	Formula, Data Source	Units	TEST RESULTS			EGR Sweep at 2000 rpm				
Cam B 2000rpm MOP 502	or Notes		Cam B 2000rpm MOP 502							
Test Data Identifier										
Data log number			1	2	3	4	5	6	7	8
Test Cell Variables										
RPM		RPM	2000	2000	2000	2000	2000	2000	2000	2000
Compression ratio			16.8	16.8	16.8	16.8	16.8	16.8	16.8	16.8
Spark timing	Crank angle	deg	-25.4	-26.4	-29.23	-30.4	-32.38	-37.4	-46.2	
Air Fuel Ratio	Lambda Lean: λ > 1		1.0164	1.014	1.0143	1.0143	1.0077	1.0119	1.0074	1.0057
Air Throttle Setting			WOT	WOT	WOT	WOT	WOT	WOT	WOT	WOT
EGR mass percent total flow		percent	0.02	0.95	3.52	4.68	7.23	10.1	14.4	19.29
Brake Data										
Torque	Measured	Nm	123.32	121.98	120.02	117.55	114.08	110.66	104.23	96.27
Power		kW	25.83	25.55	25.14	24.62	23.89	23.18	21.83	20.16
		hp	34.62	34.25	33.70	33.00	32.03	31.07	29.26	27.03
BSFC		g/kWh	226.37	226.11	226.62	226.03	229.51	227.57	230.56	239.16
Indicated Data										
IMEP		bar	7.583	7.510	7.408	7.285	7.100	6.915	6.573	6.140
PMEP		bar	-0.128	-0.13	-0.125	-0.12	-0.12	-0.113	-0.108	-0.105
NMEP		bar	7.455	7.383	7.285	7.165	6.980	6.803	6.463	6.040
FMEP		bar	1.027	1.0245	1.029	1.038	1.034	1.0345	1.0295	1.022
BMEP		bar	6.428	6.356	6.254	6.127	5.946	5.768	5.436	5.013
ISFC		g/kWh	191.88	191.41	191.35	190.11	192.25	189.80	190.56	195.43
Indicated Power		kW	30.331	30.041	29.631	29.141	28.401	27.661	26.291	24.561
Indicated Torque		Nm	144.815	143.43	141.473	139.133	135.6	132.067	125.526	117.265
Combustion Data										
Spark timing	Crank angle	deg	-25.4	-26.4	-29.23	-30.4	-32.38	-37.4	-46.2	
Burn duration: 0 to 10%	Crank angle	deg	23.4	24.0	26.0	27.1	29.1	33.2	41.0	
50% burnt	Crank angle	deg	9.25	8.77	8.30	7.45	9.00	8.46	8.25	9.54
Burn duration: 10 to 90%	Crank angle	deg	24.68	24.6	25.43	26.06	27.81	28.87	31.64	36.94
Max cylinder pressure		bar	50.82	51.38	51.67	50.9	49.16	49.16	48.47	45.53
Max cylinder pressure cank angle	Crank angle	deg	12.3	11.9	11.3	11.2	11	10.4	9.4	8.7
R-Max	Max Δbar/crank deg	bar/deg	2.32	2.39	2.42	2.33	2.23	2.22	2.21	2.09
COV	IMEPcov	Percent	1.1	1.2	1.4	1.4	1.7	1.7	2.4	5.1

12. GTPOWER MODEL CALIBRATION
EngSim Corporation

OBJECTIVES

1. Using test cell dynamometer data acquired at Mahle Powertrain, calibrate the GTPower model.

RESULTS

1. The GTPower model was successfully calibrated. The calibration includes the effects of external EGR. BSFC error is generally <1% at EGR values below about 17 percent.

ENGSIM REPORT

The EngSim Corporation report on GTPower model calibration is provided on the following pages.

High Efficiency VCR Engine GT-SUITE Model Update

Latest Model: D4F-N-050317-ModelCalibration.gtm

Brad Tillock

EngSim

Summary

- The engine has been run on dyno at Mahle and new test data has been provided to EngSim. The engine tested has a compression ratio of 17.6 and outputs ~100Nm at 2000 rpm.

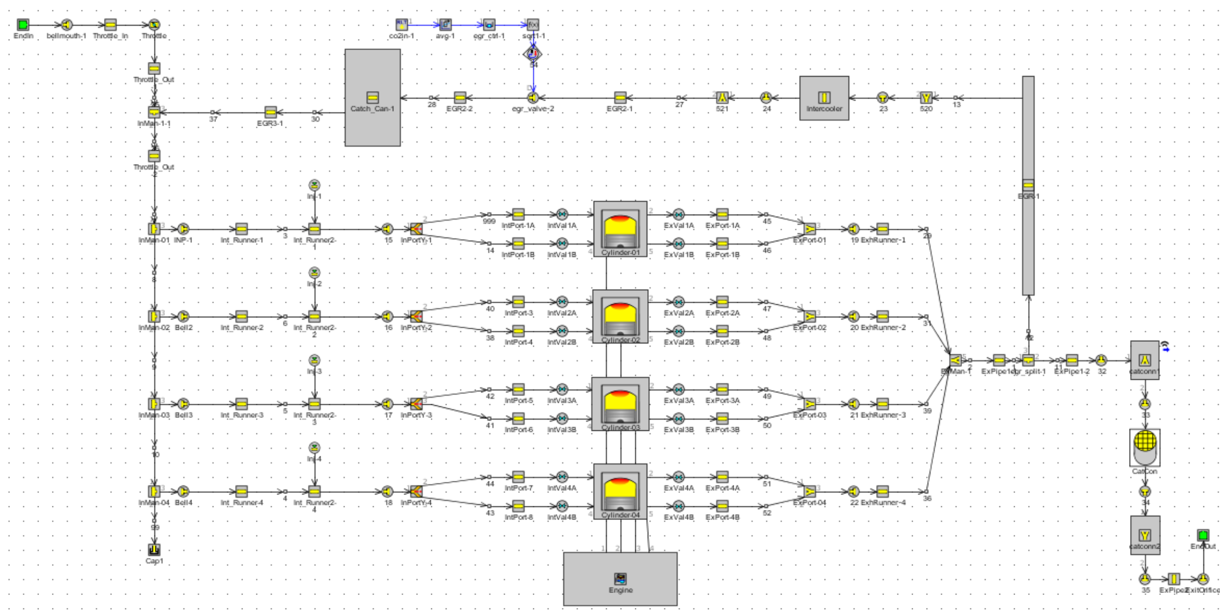
- The data used for the correlation is an egr sweep with CamB at 2000 rpm

- This update covers the changes made to the model and the quality of the latest correlation

EGR Sweep 2000rpm WOT CamB MOP 507

Model Updates

- Envera has provided updated geometry data for the intake and exhaust systems
- Mahle has provided rough measurements of the egr system
- New geometry and egr system added to the model. EGR controller added to target the required egr%.



ESEngSim

Correlation Data

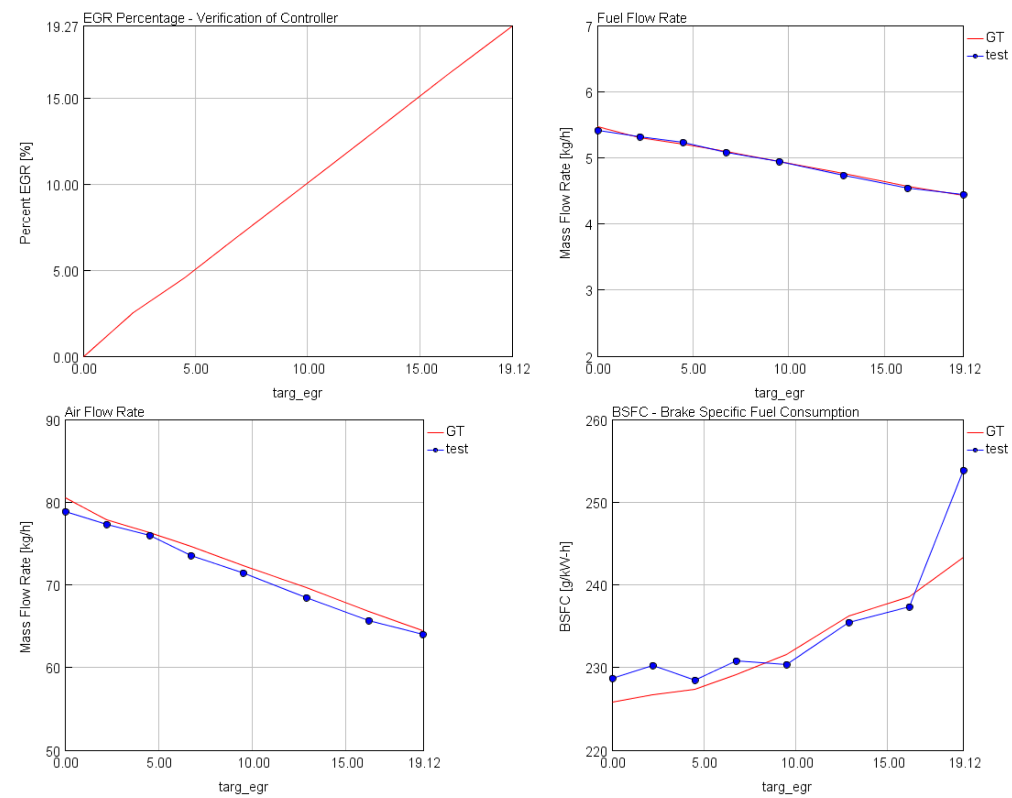
- The defining variables in this correlation sweep are shown to the right. Mahle swept EGR% at 2000 rpm and ~5 bar BMEP

- Mahle provided transient cylinder pressure data for all of the cylinders at each operating condition. EngSim averaged the pressure over 35 consecutive cycles and created burn curves to use directly in the model.

Engine Speed	BMEP	BSFC	EGR
rpm	bar	g/kWh	%
2000	5.924	228.692	0.03
2000	5.777	230.2034	2.24
2000	5.734	228.422	4.5
2000	5.504	230.8042	6.74
2000	5.36	230.4104	9.51
2000	5.032	235.4694	12.89
2000	4.78	237.3236	16.2
2000	4.38	253.8505	19.12

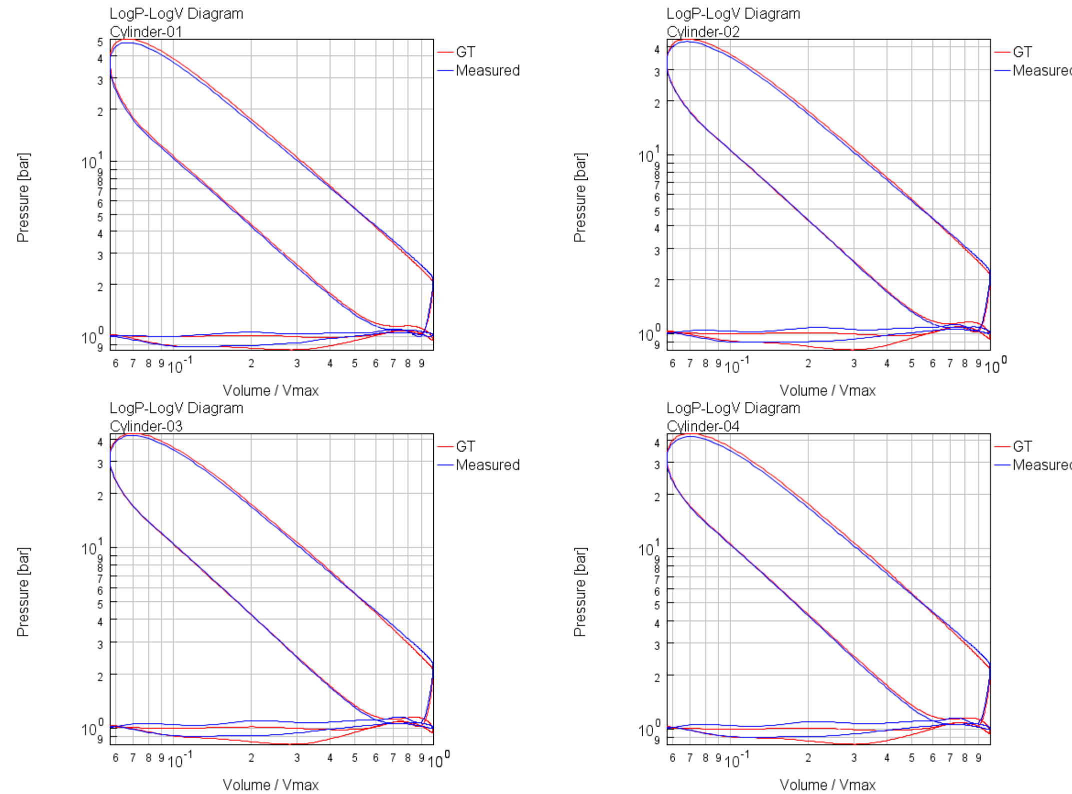
Correlation Results

- The top left figure is in place for verification of the EGR controller used in the model
- Fuel flow and air flow error are approximately 1-2%
- BSFC error is generally < 1% (except at highest egr) and shows the correct trend. Envera suggested the highest egr point may not be accurate due to combustion instability.



Correlation Results

- The figures to the right show the LogP-LogV comparisons for each cylinder for the ‘no-egr’ case
- There is a small difference near BDC before compression that is probably related to intake system wave dynamics. Perhaps a closer inspection of the system could improve this.
- There is also a small variation near the end of the expansion stroke as the simulation appears to curve away slightly. This could indicate a small error in the compression ratio.

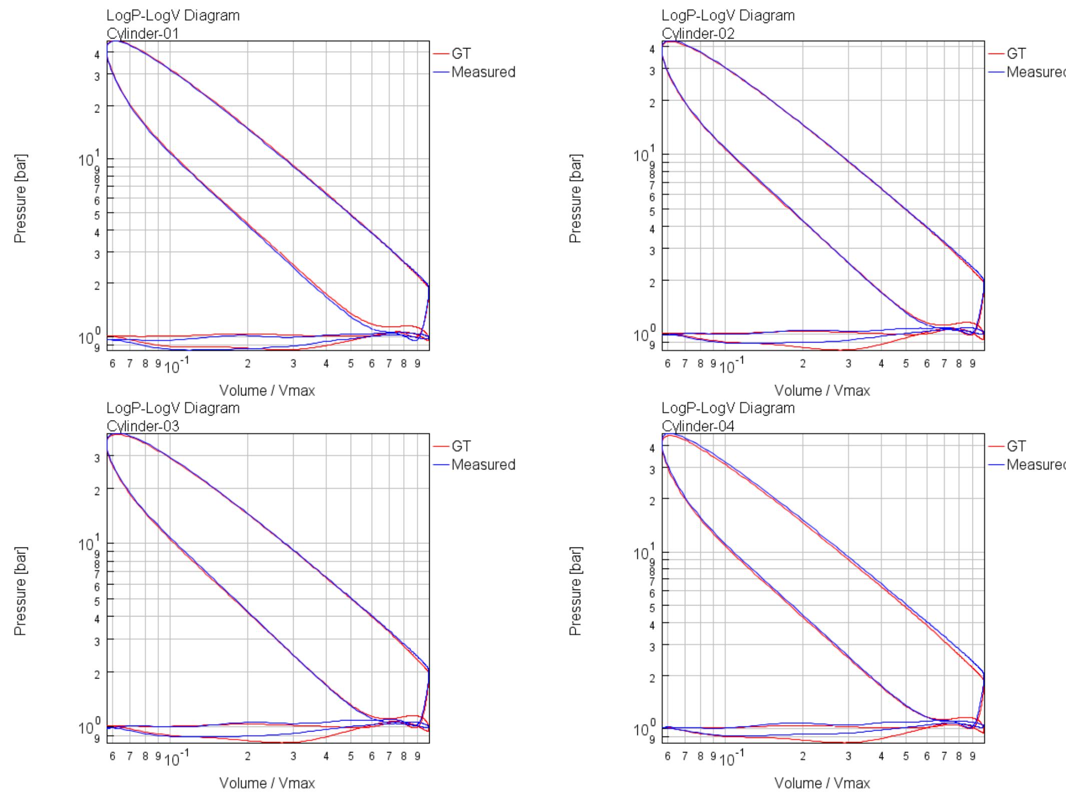


Correlation Results

- The figures to the right show the LogP-LogV comparisons for each cylinder for the 16% egr case

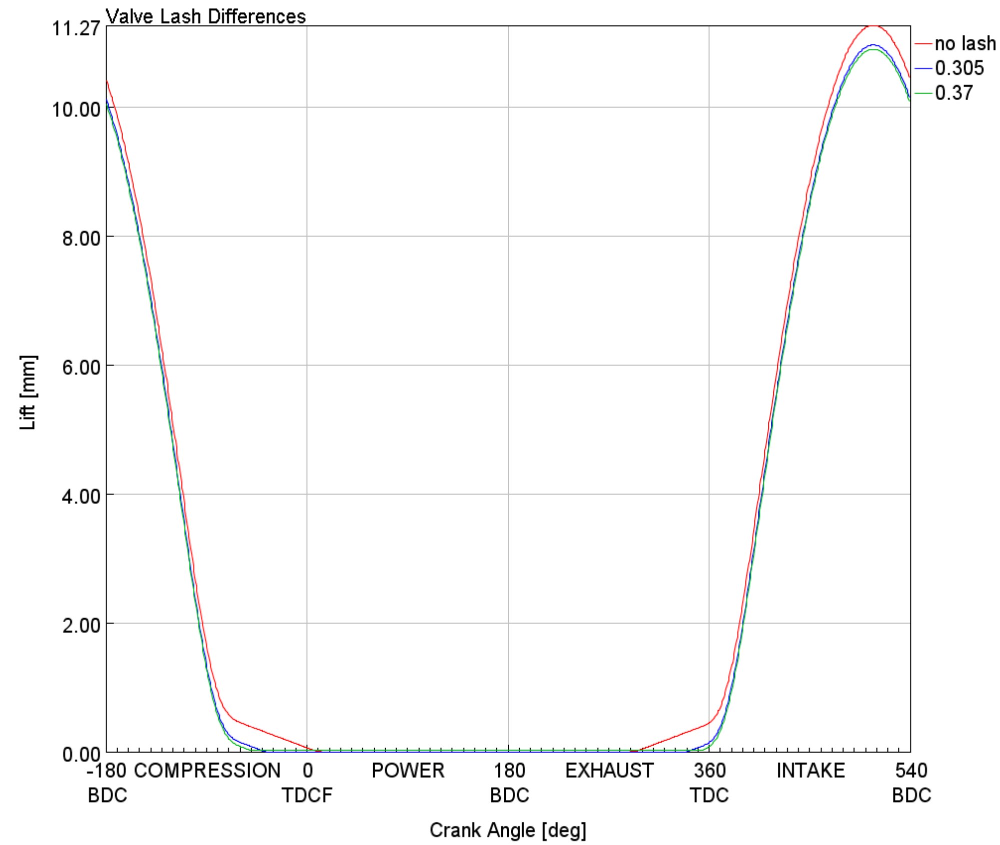
- Correlation is very good for this case as well

- Again, we might be able to improve wave dynamics during intake stroke



Calibration

- The valve system in place carries a constant lash of 0.305mm and that was the starting point in the simulation. With that value there was some error in the compression slopes.
- Intake valve lash was increased to 0.37mm for best result
- The Eaton B cam valve lift is shown as the red curve. The same curve with 0.305mm and 0.37mm lash are shown in blue and green. Additional lash closes valve about 6 degrees earlier.
- For the moment, we will treat the additional 0.065mm lash as a dynamic flexing in the system



Summary

- GT-SUITE model has been correlated to test data collected at Mahle. Overall engine results and LogP-LogV diagrams match well.
- Model will be used for further optimizations and verification at 2000 rpm-100Nm, peak brake efficiency, and peak power
- In the next update EngSim will show predictive combustion model tuning, new optimizations, and verification of peak power target

13. ENGINE OPTIMIZATION USING THE CALIBRATED GTPOWER MODEL
EngSim Corporation

OBJECTIVES

1. Investigate CR, Spark and EGR settings for minimum BSFC at 100 Nm 2000 rpm
2. Investigate CR, Spark and EGR settings for peak efficiency

RESULTS

1. BSFC projections at 100 Nm 2000 rpm:
   a. 10% EGR, 17.6 CR and -40 spark timing: ~231 g/kWh
   b. Re-optimized cam ~228 g/kWh
   c. 8% EGR, 17.6 CR and -37 spark timing: ~229 g/kWh
2. Peak efficiency at 2000 rpm
   a. 0% external EGR, 12.5 CR and -38 spark timing ~38 percent BTE

FOLLOW UP

1. Investigate benefits of EGR on peak efficiency

ENGSIM REPORT

The EngSim Corporation report on engine optimization using the calibrated GTPower model is provided on the following pages.

High Efficiency VCR Engine
GT-SUITE Model Update 2

Latest Model: D4F-N-062017-ModelCalibration_Predictive_Combustion.gtm

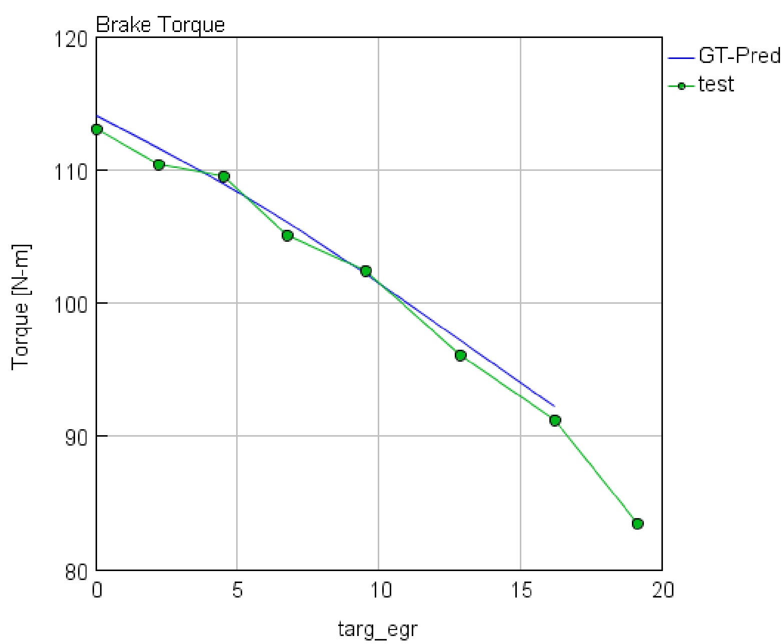
Brad Tillock
EngSim

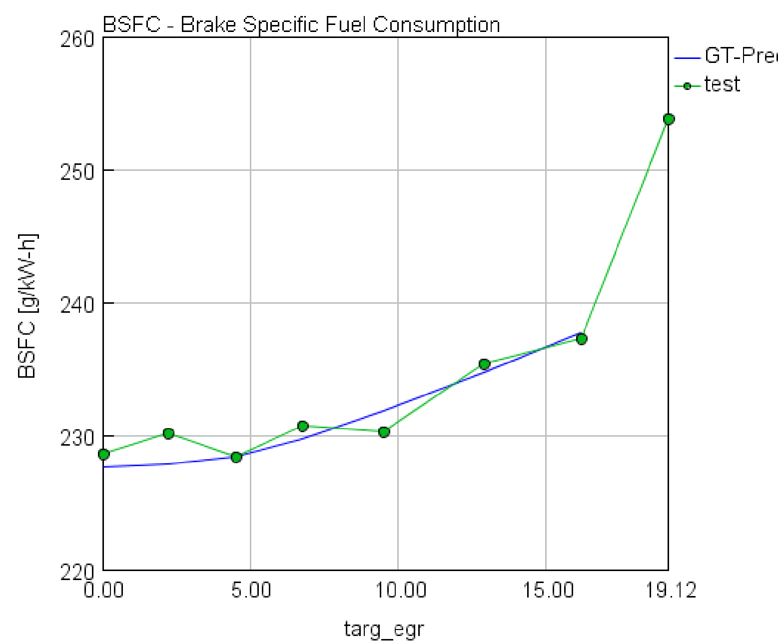
Summary

- Using the Mahle test data EngSim correlated the SI predictive combustion model. Many assumptions were made in the input data. EngSim used a simplified head and piston geometry and no information was available for the head swirl and tumble coefficients.
- New correlated combustion model used for CR and spark mapping of original setup
- New optimization at 2000 rpm, 100 Nm
- New optimization that spans a load range at 2400 rpm. Goal is to find the best possible efficiency.
- CR and spark mapping for new optimized cases

Predictive Combustion

- Predictive combustion is necessary to have some assessment of possible combinations of compression ratio and spark timing
- Model was tuned at 0% EGR and then tested at other EGR points. 4 basic attributes are used to tune the SI predictive model.





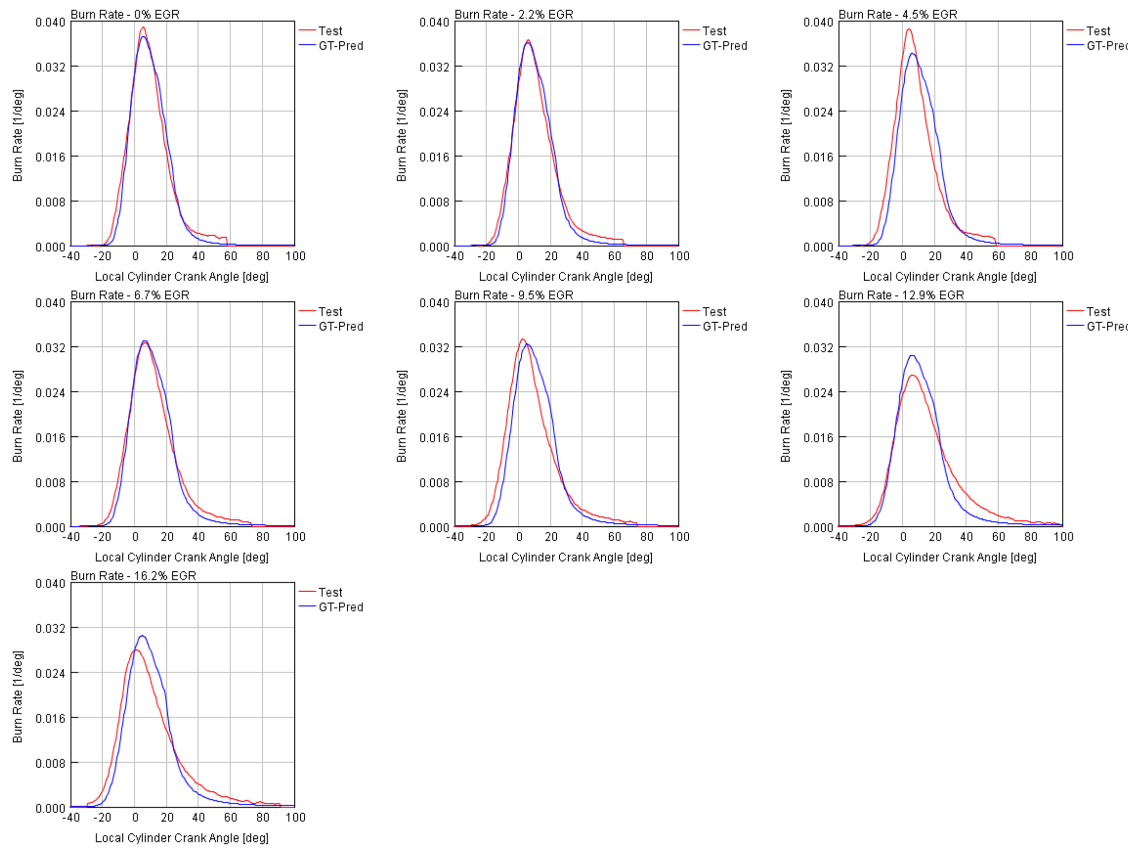
ESEngSim

Predictive Combustion

- Figures show cylinder burn rates for the test and predicted model. As EGR increases the burn rate peak drops and the burn duration increases.

- Match is not perfect but model captures the general trends we see in the test data

- Without knowing other information concerning knock, EngSim tuned the model to be on the edge of knocking for each of the Mahle test points



D4F-N-050317-ModelCalibration_Predictive_Combustion.gtm

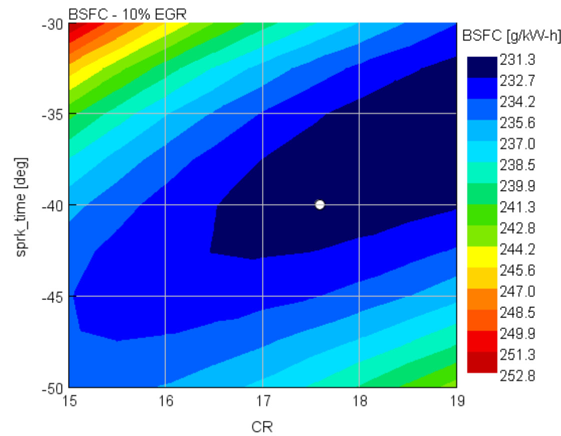
EngSim

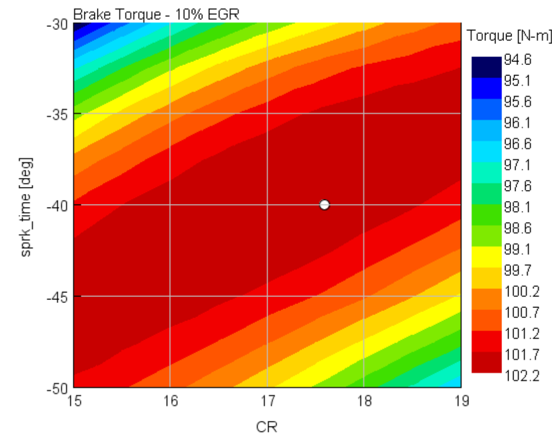
Predictive Mapping

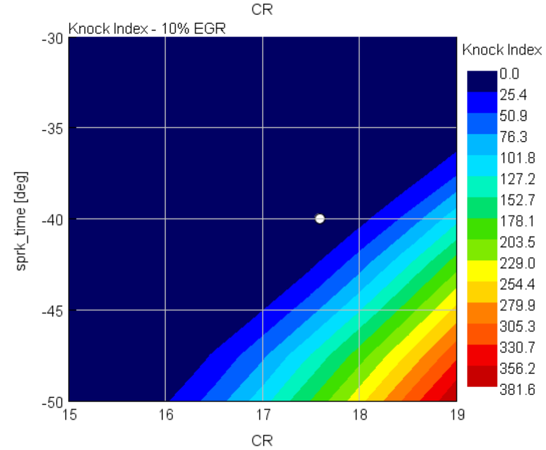
- Using Mahle test configuration and the predictive combustion model, EngSim performed a compression ratio and spark mapping at 10% egr. These mappings will now be possible with any new valve profiles.

- Mahle and Envera did find the optimal point for this particular setup

- Future mappings will show Envera the efficiency tradeoffs with CR and spark







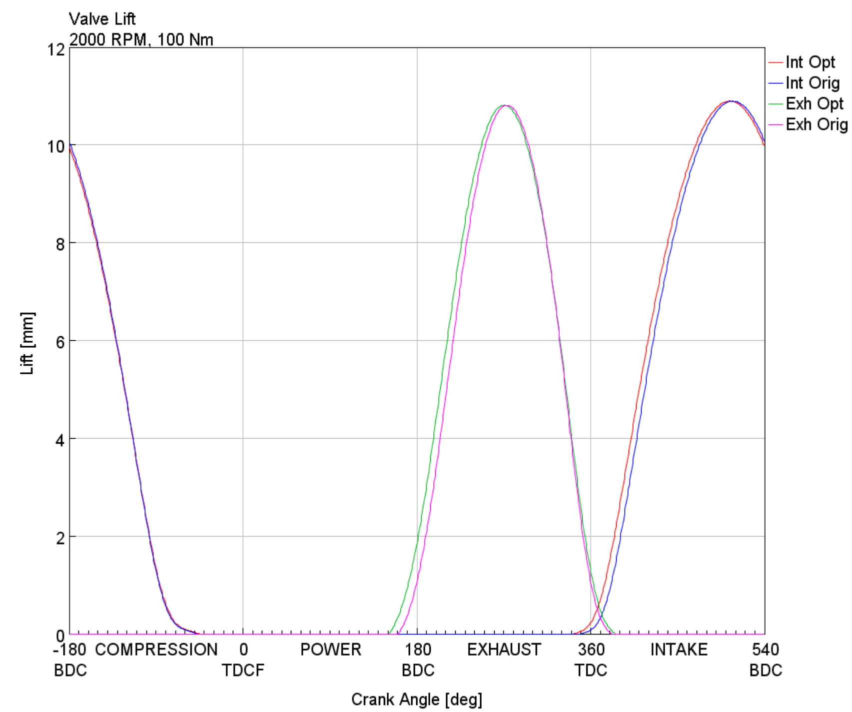
Optimization – 2000 rpm, 100Nm

- Envera requested an additional round of valve optimization for the low load point. EngSim set up an optimization which allowed the timings and durations of the valves to change. A controller was set up to automatically adjust the intake closing to maintain exactly 100 Nm. The throttle was left in the full open position

- The new optimum only changed the valve profiles slightly. This resulted in a predicted BSFC improvement of 1.3%. BSFC reduced from 231.5 to 228.4 g/kWhr.

- The slightly longer exhaust should allow for better breathing at high load conditions - system only has one exhaust profile.

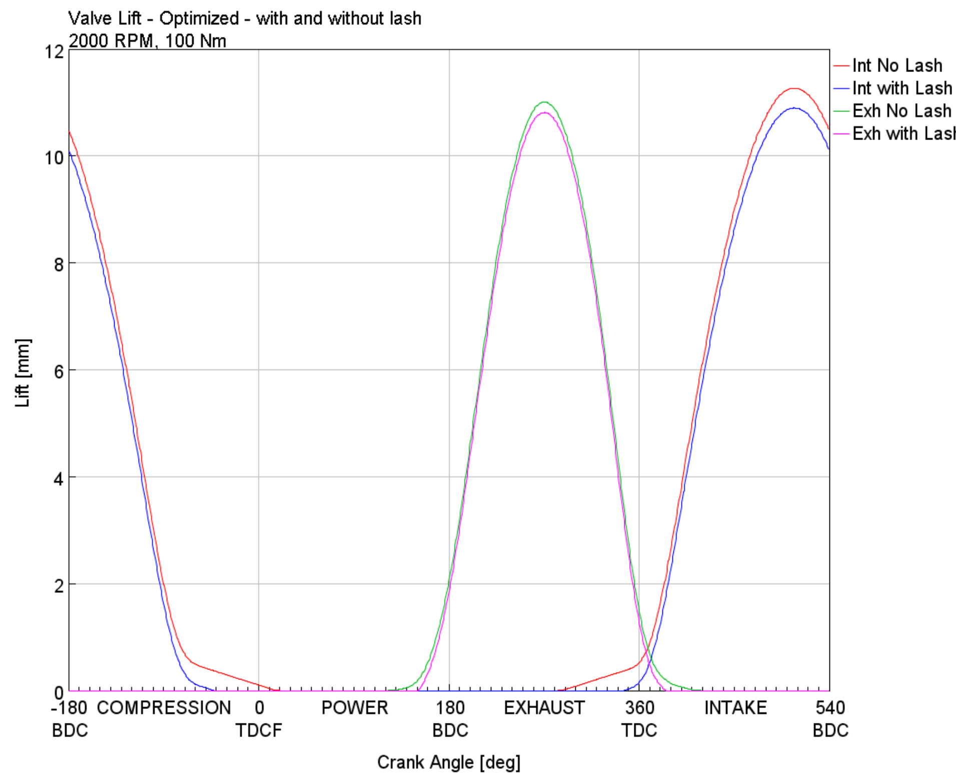
- Note – Optimizations were done using the imposed combustion profiles from test. This keeps the optimizations focused on the cams only. Future optimizations will use the predictive combustion model.



EngSim

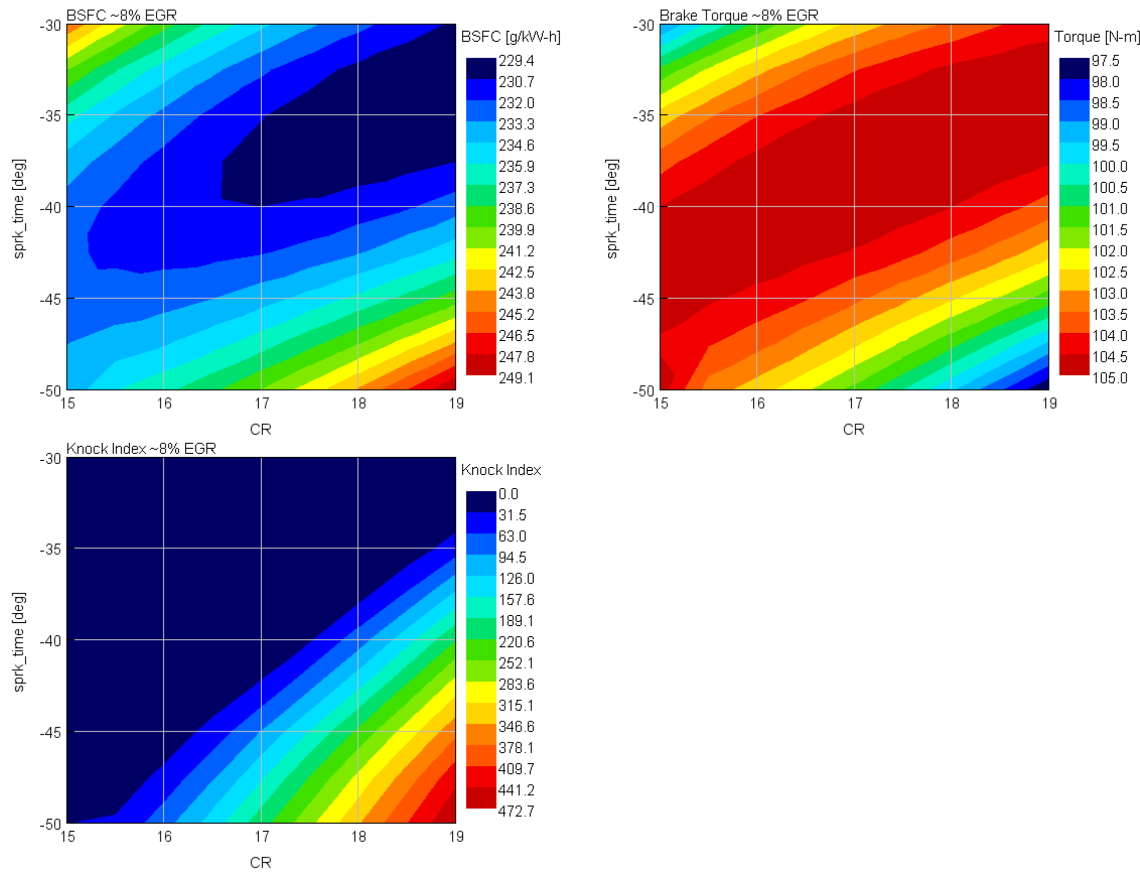
Optimization – 2000 rpm, 100Nm

- EngSim is providing the valve lift profiles to Envera in Excel format. Information is needed on the valvetrain layout in order to precisely get that back to a follower lift or cam lift (Mahle doing this?).
- Referring back to the first project report, EngSim used an intake lash value of 0.37mm in order to get a close match for the compression slopes. 0.305mm of that lash was specified from mechanical design and 0.065 was attributed to dynamic flexing of valvetrain. The curves provided to Envera will be provided with **no lash**.
- Figure shows what optimized lift profiles look like with and without lash



Predictive Mapping

- EngSim took the second optimization results and ran the spark and CR sweeps using the predictive model.
- Hopefully Mahle will see a similar behavior when they run the new profiles
- The maps don’t quite show the BSFC reduction down to 228.4 g/kWhr. This is attributed to the predictive combustion model not being exactly the same as the Mahle test burn curves. Also, this mapping does not have IVC load control.
- The trends should be correct



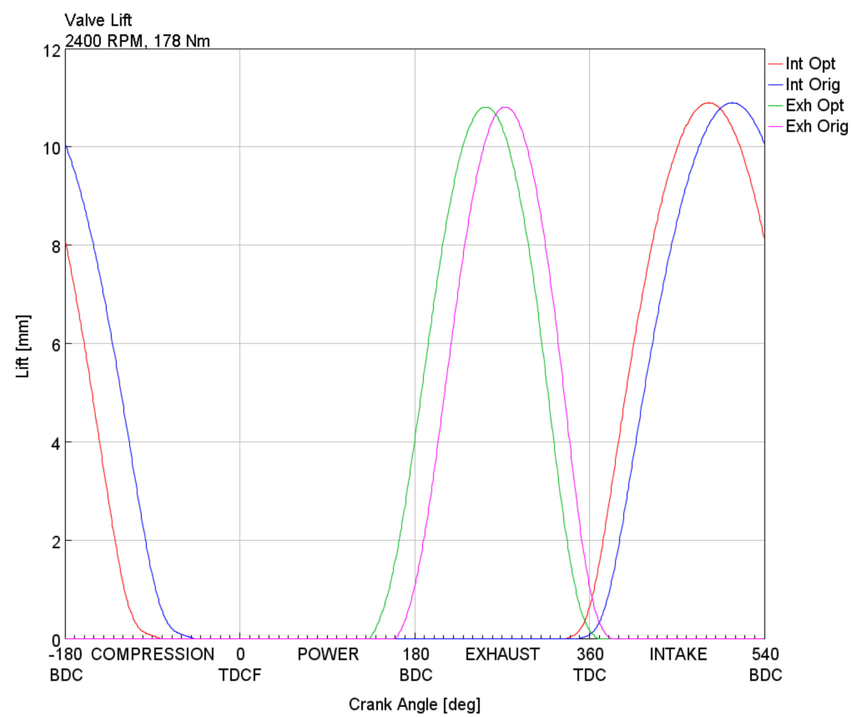
Optimization – Peak Efficiency

- Envera requested that EngSim search for a peak efficiency point. This was first done using the non-predictive combustion model. The combustion was modified slightly from the data collected at Mahle for 2000 rpm. Simple Wiebe curve was used.

- The new valve lift profiles are show to the right. The only constraint was the exhaust duration was fixed to the duration found in the 100 Nm optimization. Throttle left at full open.

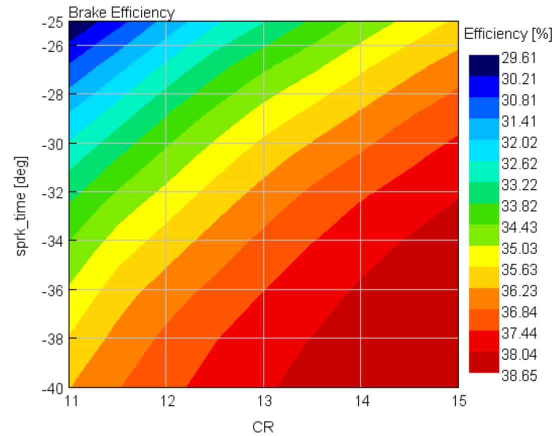
- The model predicted almost 40% brake thermal efficiency.

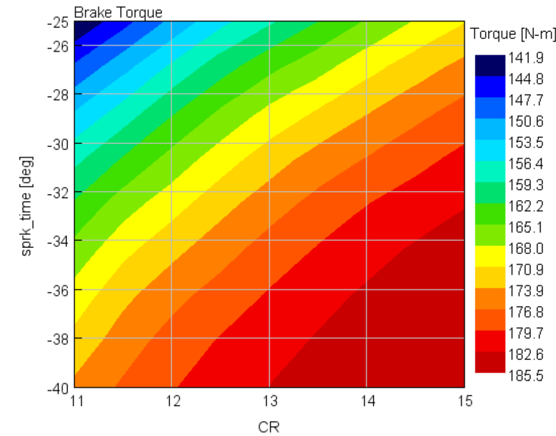
- A full CR and spark mapping using the predictive combustion model will give a more accurate efficiency number

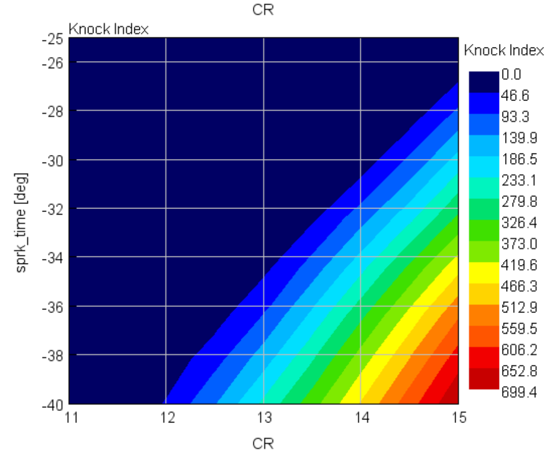


Predictive Mapping – Peak Efficiency

- EngSim ran the spark and CR sweeps using the predictive model and the best ‘Point 2’ valve profiles
- The maps suggest ~38% brake efficiency possible before knock onset

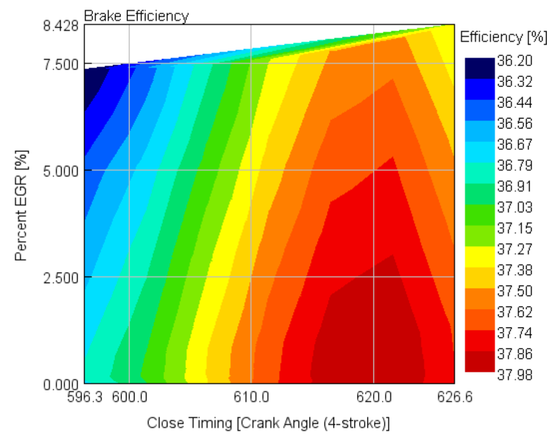


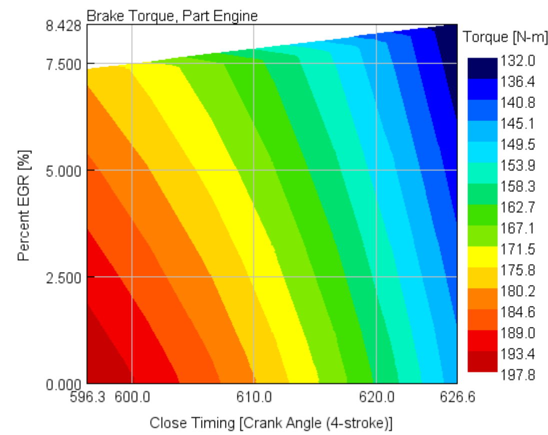


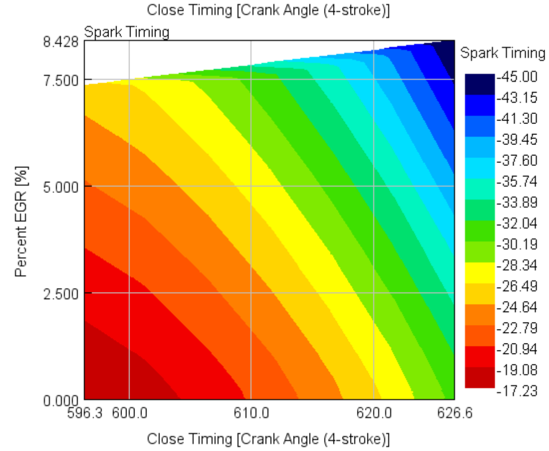


Optimization 2 – Peak Efficiency

- Since the predictive model is now available, EngSim ran a partial optimization of the intake profile using an alternative technique. With a fixed CR of 15, the simulation swept IVC and egr rate. A spark controller was put in place to automatically adjust spark to the onset of knock. This was not done in the first valve profile optimization for peak efficiency.
- With knock model in place to account for rpm, compression ratio, and spark timing, the results change. Now the intake valve prefers to close about 18 degrees later, making more of an Atkinson cam.



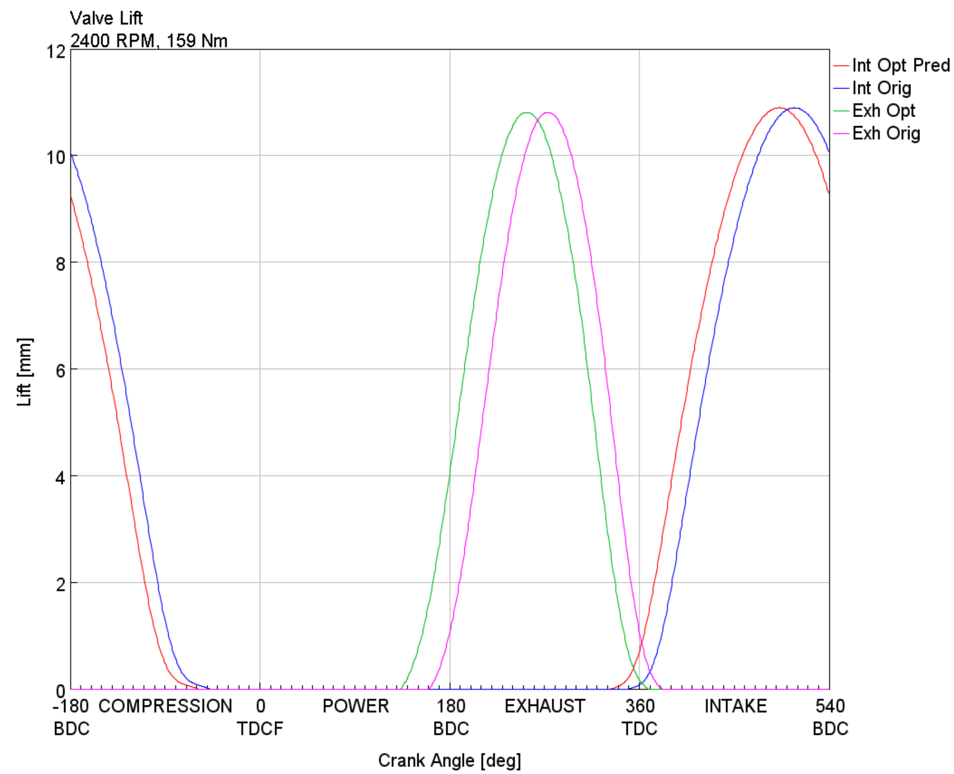




Optimization 2 – Peak Efficiency

- The peak efficiency point uses an intake that is about the same duration as the Eaton B cam but advances about 20 degrees. This creates both a higher load condition and an Atkinson type expansion.

- The very late IVC may be good for efficiency but the sacrifice is low end torque capacity. Envera will have to decide the proper objective for the shorter duration cam.



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Summary

- Predictive combustion model has been correlated to test data. Needs to be verified at varying speeds and loads but should work well.
- Model can now be used for CR and spark sweeps at varying EGR levels
- Envera requested a second optimization at 2000 rpm, 100 Nm. This was completed and the gain was around 1%. <u>Main change is a 6% increase in the exhaust duration</u>. There was a minor change in the intake valve as well. If there is uncertainty in applying the mechanical and dynamic lash correctly, Envera can stay with the existing long duration intake profile. Model is very sensitive to the IVC and very small changes to IVC affect the attainable BMEP for the long duration cam. We already know the current IVC gives a good BMEP level for the long duration cam.
- Envera requested that EngSim try to find the optimal efficiency point for the engine. This mostly centered on mid-high load sweep at 2400 rpm. EngSim used a couple different search methods and found a point at 2400 rpm that produced 38% brake efficiency.
- A well-tuned predictive combustion model is crucial in evaluating systems like this where knock is an important factor

Next Steps

- Envera can review presentation and EngSim and Envera can meet Monday morning to discuss.
- If all looks ok, EngSim will send Excel files with valve lift profiles
- EngSim will run a few more mappings with the optimized Point 2 configuration to see if we can improve on the 38% number.

14. DRIVE CYCLE ANALYSIS OF A VCR-POWERED F150 PICKUP TRUCK
EngSim Corporation

OBJECTIVES

1. Create a full BSFC map of the 2.4L Envera VCR engine using the calibrated GTPower model and test data from Mahle Powertrain.

2. Create a GTDrive vehicle model for the F150 Ford pickup truck

3. Validate the F150 GTDrive vehicle model using a known 3.5L Ecoboost engine map and known FTP75 and HWY drive cycle mileage values from Argonne National Laboratory testing.

4. Replace the 3.5L Ecoboost engine with the 2.4L Envera VCR engine, then optimize transmission shift schedule and project FTP75 (City) and HWY F150 mileage.

5. Identify the speed/load engine conditions where most of the fuel is consumed, so that those load conditions can be optimized for further improving fuel economy.

RESULTS

1. An optimized VCR engine BSFC map was created having the following performance values:

a.	Peak efficiency	~215 g/kWh
b.	Peak torque	~445 Nm @ 4000 rpm
c.	Torque at 2000 rpm	~340 Nm @ 2000 rpm

2. The BSFC map contains four distinct areas of optimization:

    - Atkinson cam at latest IVC timing – throttled control of load
    - Atkinson cam phasing for load control – open throttle
    - Naturally aspirated with intake power cam
    - Supercharger engaged – bypass for boost control

3. A GTDrive vehicle model was created of the F150 having a Ford 3.5L ecoboost engine. The baseline model predicts 19.5 and 25.3 mpg for the FTP75 and HWY cycles. The Argonne tests resulted in 19.0 and 28.2 mpg for the UDDS and HWY cycles. The UDDS cycle is used for comparison as it is essentially the same as the FTP75. Shift strategy and torque converter lockup are not known for the actual vehicle. The mpg comparison is reasonable but not perfect due to these and other factors.

4. The current program objective is to demonstrate potential for attaining a 40 percent improvement in fuel economy relative to a V8 powered F150 baseline. The following vehicle mileage values were projected:

Projected test cycle mpg values Optimization Level 1	FTP75	HWY
EPA Guide V8 F150	16.7	26.9
VCR 4-cylinder F150 projections	24.9	31.2

EPA Guide mileage values shown on the vehicle window sticker include a test cycle adjustment factor, and are shown below:

Projected EPA Guide mileage values Optimization Level 1		City	Highway	Combined
EPA Guide V8 F150	*Actual*	15	21	17
VCR 4-cylinder F150 projections		22.4	24.3	23.2
Percent improvement				~35%

A 35 percent improvement in mileage was projected for Optimization Level 1.

5. Fuel consumption for the FTP75 and HWY drive cycles was plotted on a BMEP vs. rpm charts. These charts indicate that most of the fuel is consumed at about 3 bar BMEP @1400 rpm for the city FTP75 drive cycle, and 6.4 bar bmep @ 1700 rpm for the highway HWY drive cycle. Under engine Optimization Level 2 the VCR engine was tuned for improved fuel economy at these load conditions.

   Level 2 optimization was performed by Envera. The projected EPA Guide mileage values are shown below:

Projected EPA Guide mileage values Optimization Level 2		City	Highway	Combined
EPA Guide V8 F150	*Actual*	15	21	17
VCR 4-cylinder F150 projections		24	25.4	24.6
Percent improvement				~43%

ENGSIM REPORT

The EngSim Corporation report on vehicle mileage projections using GTDrive is provided on the following pages.

Drive Cycle Analysis of the F150

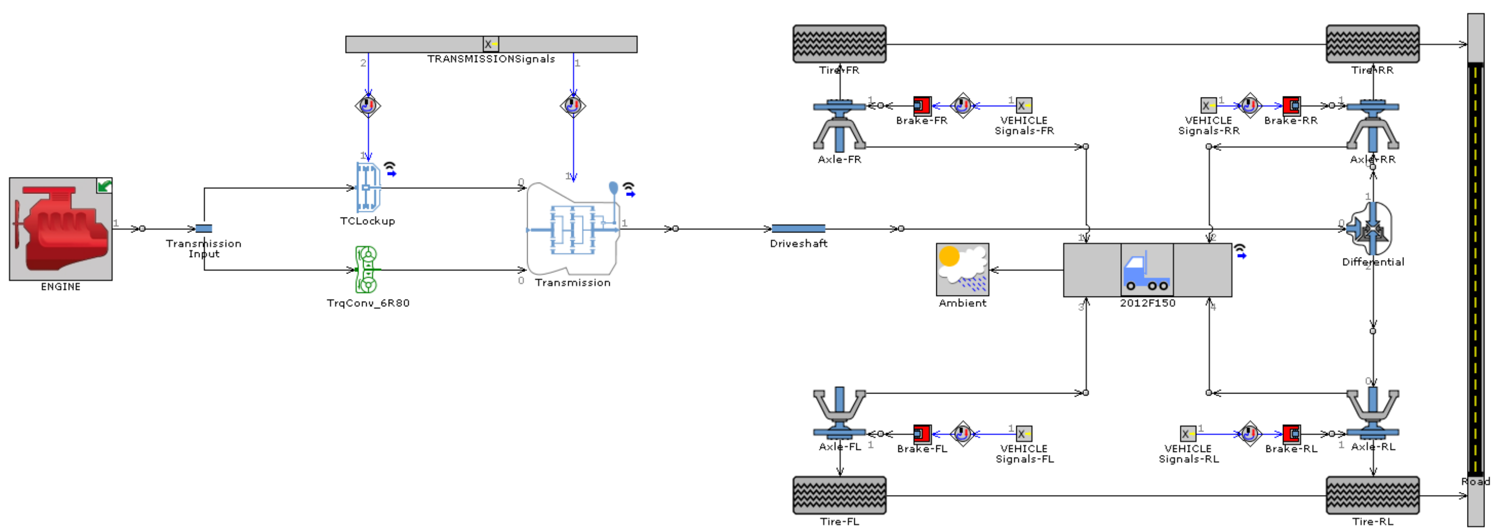
Brad Tillock
EngSim

Goals for This Stage

- Create a vehicle model for the Ford F-150
- Create a full map of the supercharged version of the 2.4L engine
- Run baseline drive cycles for the F-150 using the 3.5L EcoBoost engine – FTP75 and HWY cycles
- Replace the 3.5L engine maps with the supercharged 2.4L maps and rerun the drive cycles
    - Keep the 6R80 transmission
    - Change to the 10R80 transmission – requires an optimization of the shift strategy

F-150 Vehicle Model

- The basic layout of the F-150 GT-SUITE vehicle model is shown below. This consists of a map-based engine, transmission, and vehicle.
- The engine and vehicle data for the model come primarily from a benchmark investigation at Argonne
- The next 5 slides document the data available from Argonne



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Snapshot of Available Data for F-150

- Vehicle specifications and drive cycle results

Vehicle Information				
Car		2012 Ford F-150 w Eco-Boost		
Type		Truck		
Engine		3.5 liter twin-turbocharged, direct-injected EcoBoost V6		
Compression ratio		10.0:1		
Horsepower		365 @ 5000		
Torque		420 @ 2500		
Gear Ratio	1st	4.17		
	2nd	2.34		
	3rd	1.52		
	4th	1.14		
	5th	0.86		
	6th	0.69		
	7th	-		
	8th	-		
	Final Drive	3.15		
Wet or Dry Clutch				
Labeled EPA Fuel Economy	HWY	17		
	Urban	23		
	Combined	19		
0 to 60 time [s]		6.1		
2 Whl Dyno GUI table	Test weight	6000		
	a	44.26		
	b	0.5405		
	c	0.03654		
Fuel Info				
Fuel Name:	Tier II EEE HF437	Density:	0.74	[g/ml]
CWF:	0.8618	Net HV:	18344	[BTU/lbm]

Test Cycle	Test I.D.	Fuel Usage Summary Table			
		Date	Distance [mi]	FC [l/km]	FE [mpg]
Weighted UDDS		08/17/12	7.49	13.27	17.9
UDDS Cold	71208052	08/17/12	7.48	14.38	16.5
UDDS Hot	71208053	08/17/12	7.49	12.53	18.9
UDDS Hot2	71208054	08/17/12	7.49	12.49	19.0
		Cold Start Penalty**		14.76%	12.86%
Hwy	71208055	08/17/12	10.32	8.42	28.2
US06	71208056	08/17/12	8.05	14.06	16.9
US06 City Portion	71208056	08/17/12	1.78	21.24	11.2
US06 HWY Portion	71208056	08/17/12	6.27	12.02	19.8
NEDC	71208039	08/15/12	6.92	11.93	19.9
NEDC Urban	71208039	08/15/12	2.54	15.02	15.8
NEDC Extra Urban	71208039	08/15/12	4.38	10.14	23.4
LA 92	71208040	08/15/12	9.84	13.94	17.0
JC08	71208038	08/15/12	5.09	13.47	17.63
WLTP	71208045	08/16/12	14.46	12.31	19.30
WLTP Phase 1	71208045	08/16/12	1.93	16.37	14.5
WLTP Phase 2	71208045	08/16/12	2.96	12.12	19.6
WLTP Phase 3	71208045	08/16/12	4.42	10.61	22.4
WLTP Phase 4	71208045	08/16/12	5.15	12.34	19.2

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Snapshot of Available F-150 Data

- Engine data comes from:

https://www.nhtsa.gov/sites/nhtsa.dot.gov/files/812146-commercialmdhd-truckfuelefficiencytechstudy-v2.pdf

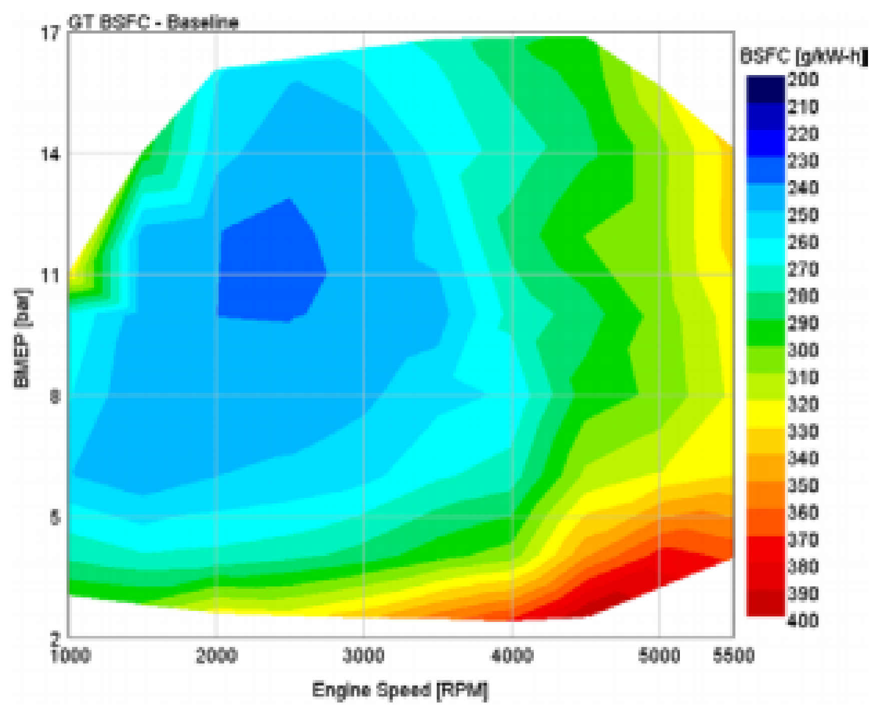
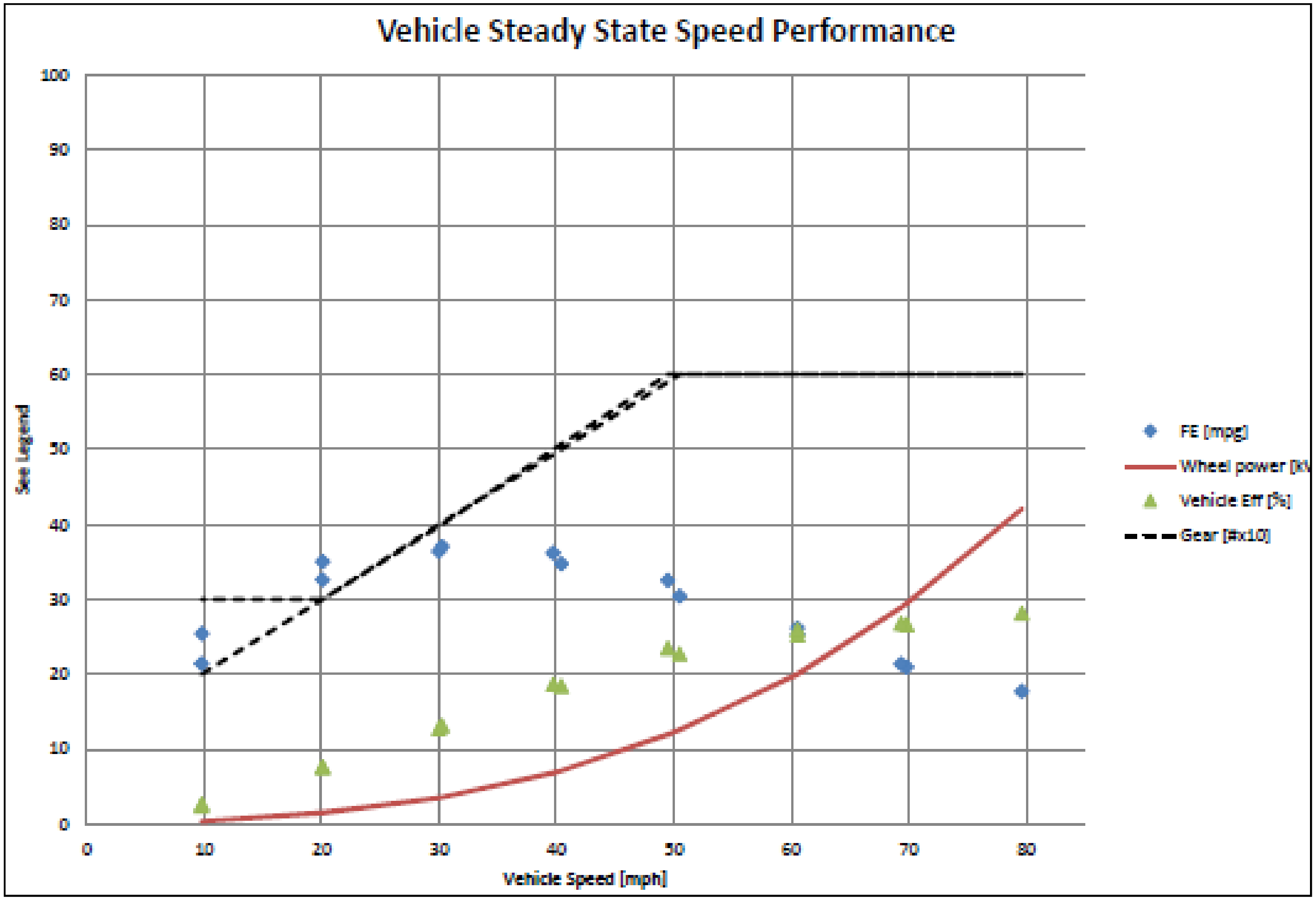


Figure A 6. Baseline Model BSFC

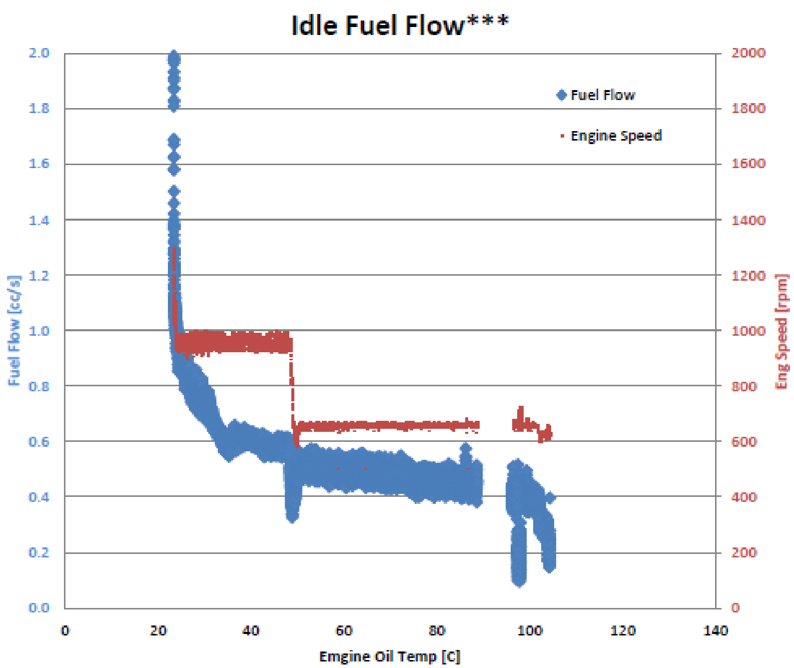
Snapshot of Available Data

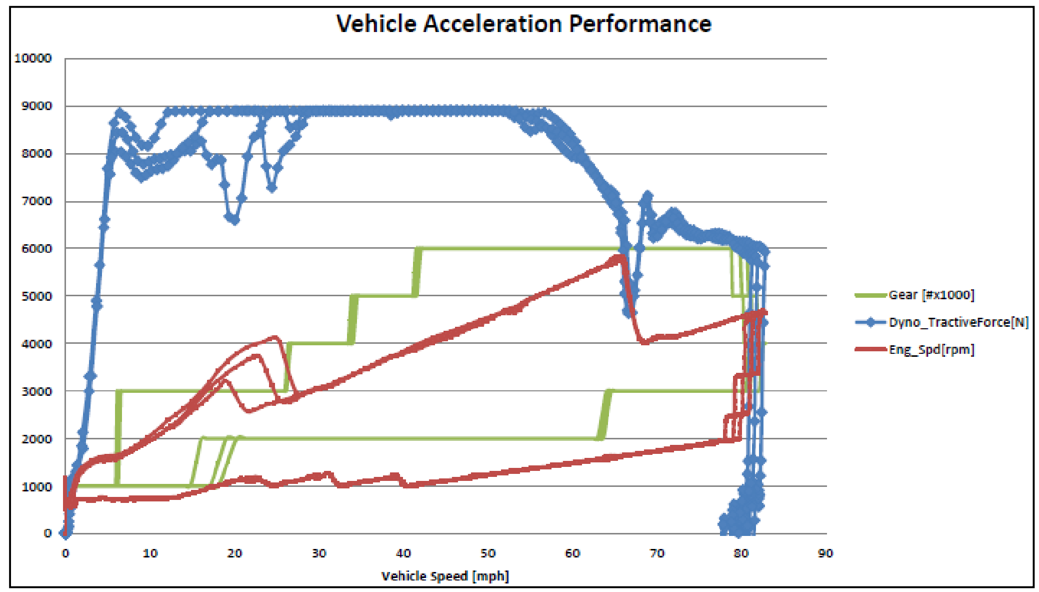
- Steady-state fuel economy



Snapshot of Available Data

- Steady-state idle and acceleration





Snapshot of Available Data

- Added in dyno force coefficients. Put vehicle mass at 6000 lbs in order to be in line with the available vehicle test data.

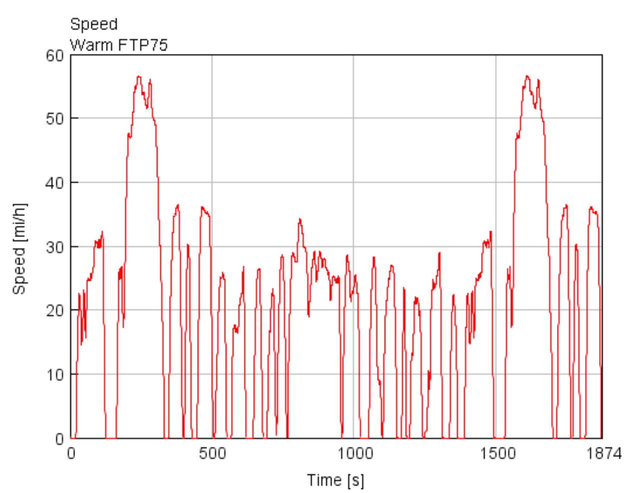
	Attribute	Unit	Object Value
	Vehicle Aerodynamic Drag/Retarding Force Specification Option		
○	DRAG/LIFT COEFFICIENTS		
●	RETARDING FORCE COEFFICIENTS (F=A+BV+CV^2)		
	Retarding Force Coefficient A	lbf	44.24
	Retarding Force Coefficient B	lbf-h/mi	0.5405
	Retarding Force Coefficient C	lbf-h^2/mi^2	0.03654
○	COASTDOWN DATA		

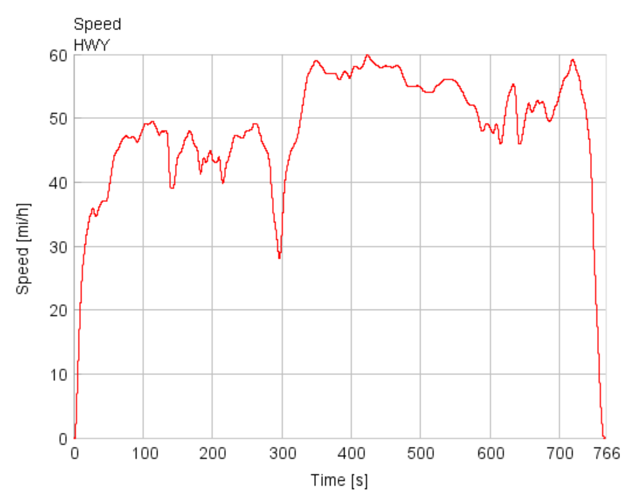
- The torque converter lockup strategy is to lock and remain locked at third gear and higher. The converter does not unlock until vehicle speed drops below 10 mph.

F-150 Results

- The baseline model predicts 19.5 and 25.3 mpg for the FTP75 and HWY cycles. The Argonne tests resulted in 19.0 and 28.2 mpg for the UDDS and HWY cycles. The UDDS cycle is used for comparison as it is essentially the same as the FTP75.

- The comparison is not perfect but there are still many things unknown in the actual vehicle (shift strategy, torque converter lockup). The baseline should be close enough for an A-B comparison when switching out only the engine map.



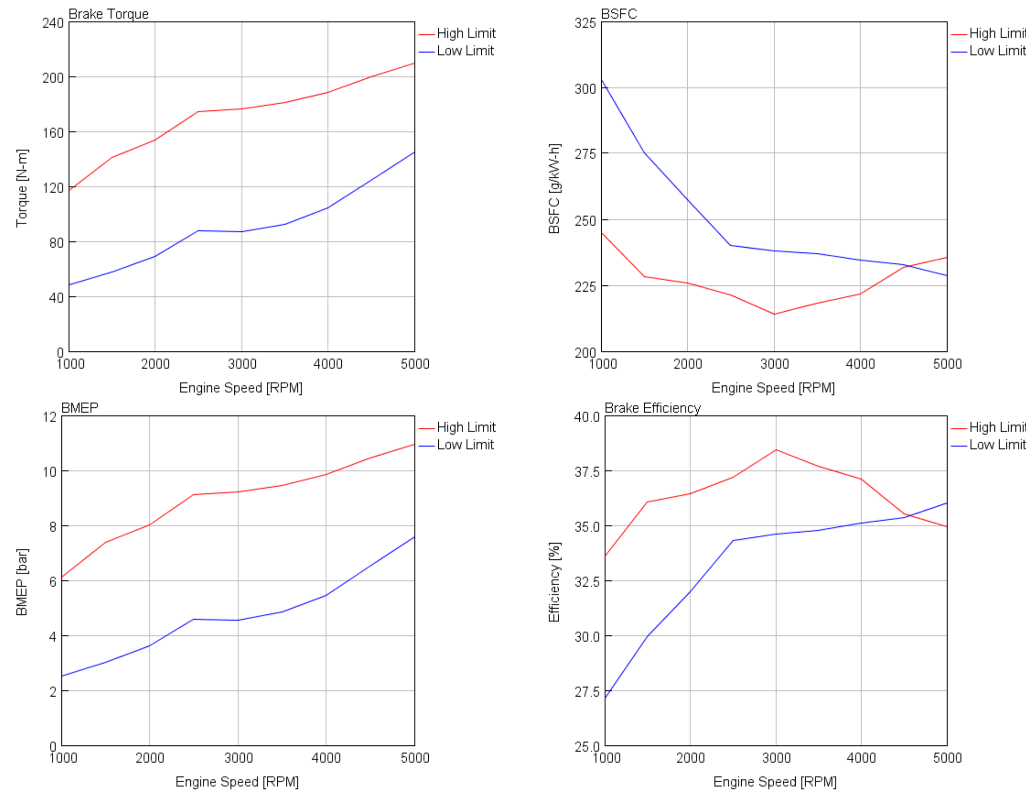


Supercharger Map Creation

- There are 4 distinct operating modes to the supercharged engine
  - Atkinson cam at latest IVC timing – throttle control of load
  - Atkinson cam phasing for load control – open throttle
  - Naturally aspirated with intake power cam
  - Supercharger engaged – bypass for boost control

Map Creation

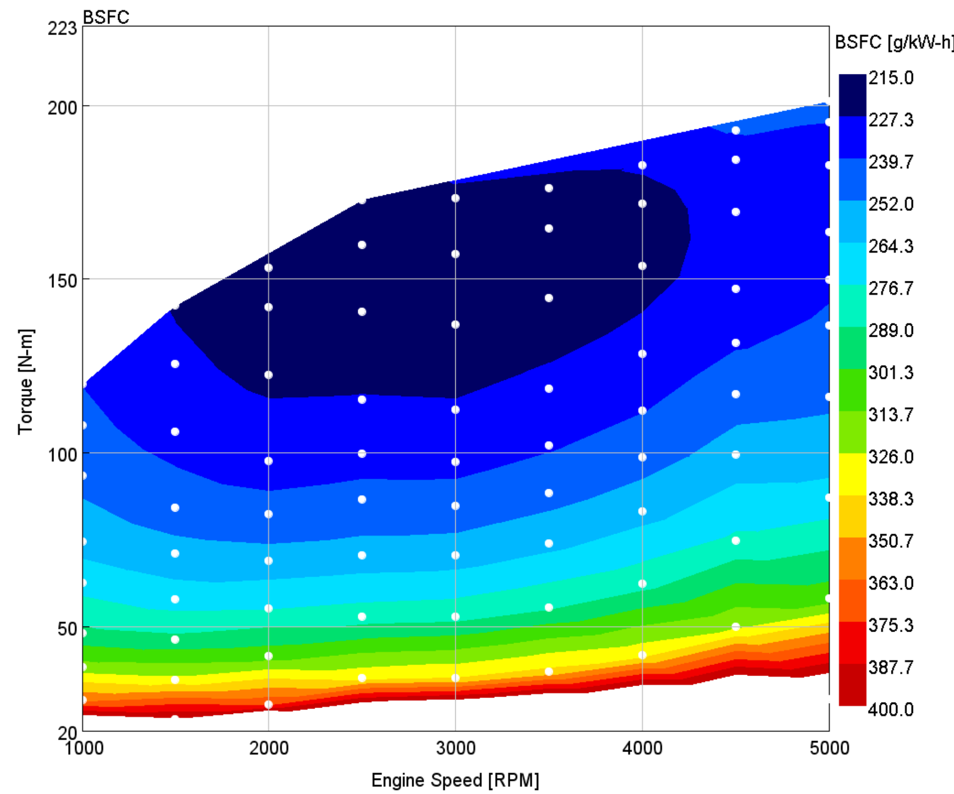
- The first step to the map creation is to find the limits of each operating mode. The figures to the right show a couple of these limits.
- The blue curves represent the latest IVC allowed by EngSim and Envera. All loads below the blue line must be controlled with the throttle.
- All loads between the blue and red line will be controlled with Atkinson cam phasing
- All loads above the red line require the switching to the power intake cam



Map Creation (Atkinson Cam Portion)

- Creating the BSFC map required an optimization at each speed and load point. The figure to the right shows the optimization result.

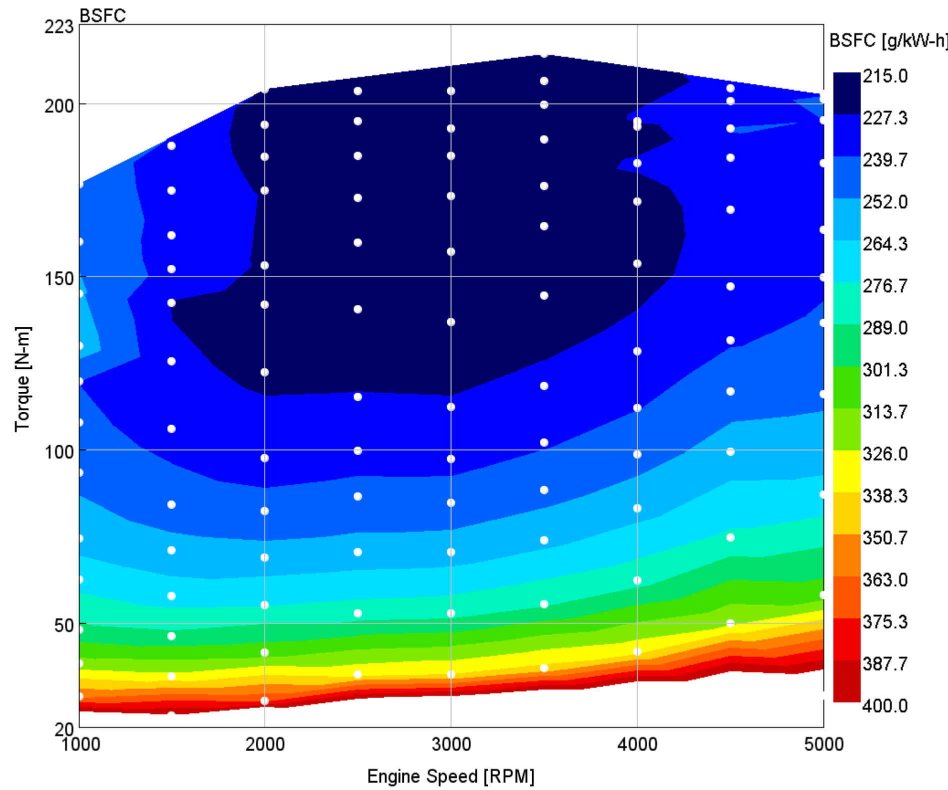
- At each of the white dots, spark advance was adjusted for best efficiency. In some cases the spark timing was limited by knock



Map Creation (+Naturally Aspirated)

- Above torque levels shown in slide 12, the engine must switch to the power intake cam.

- This was done and the map to the right shows the new extended BSFC map for all conditions up to the supercharger engagement.



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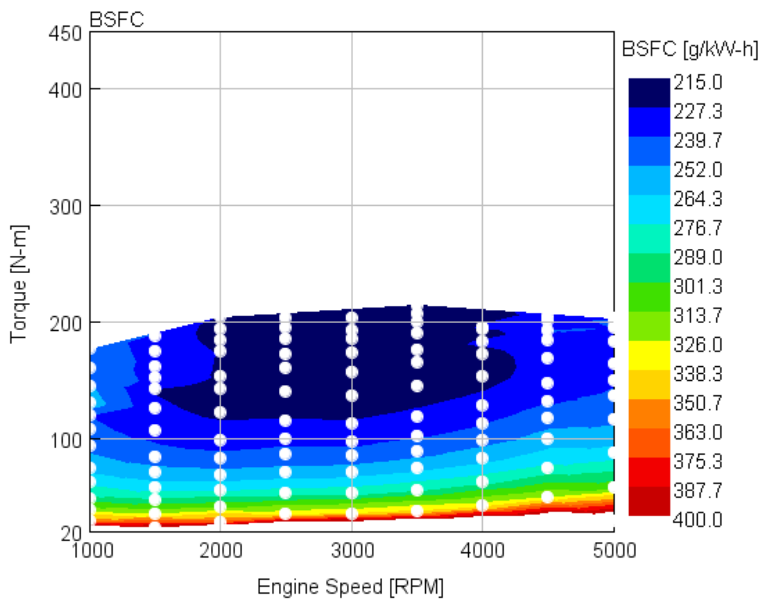
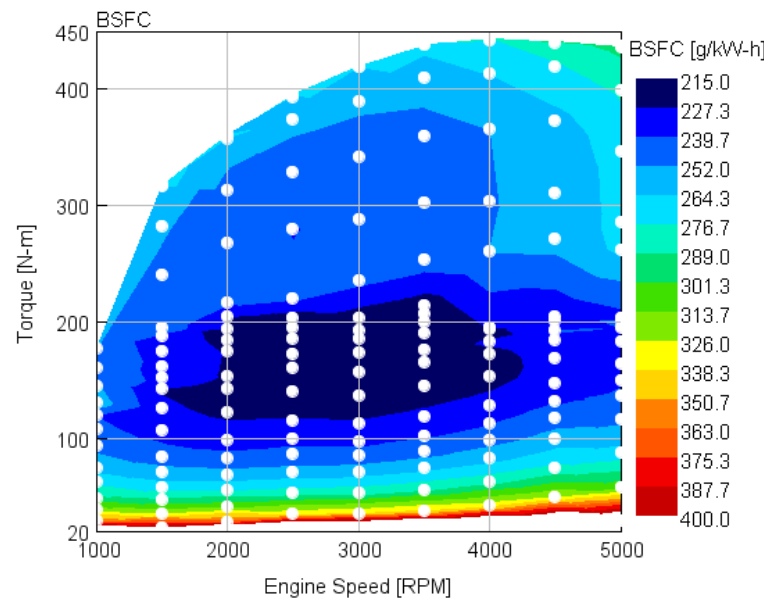
Map Creation – Full Map

- Left figure shows the naturally aspirated map and the right figure shows the full BSFC map. There are some assumptions regarding fuel enrichment and load control which will have to be confirmed in later project stages.
- Right map assumes an unthrottled engine with bypass control of the 1040 supercharger boost

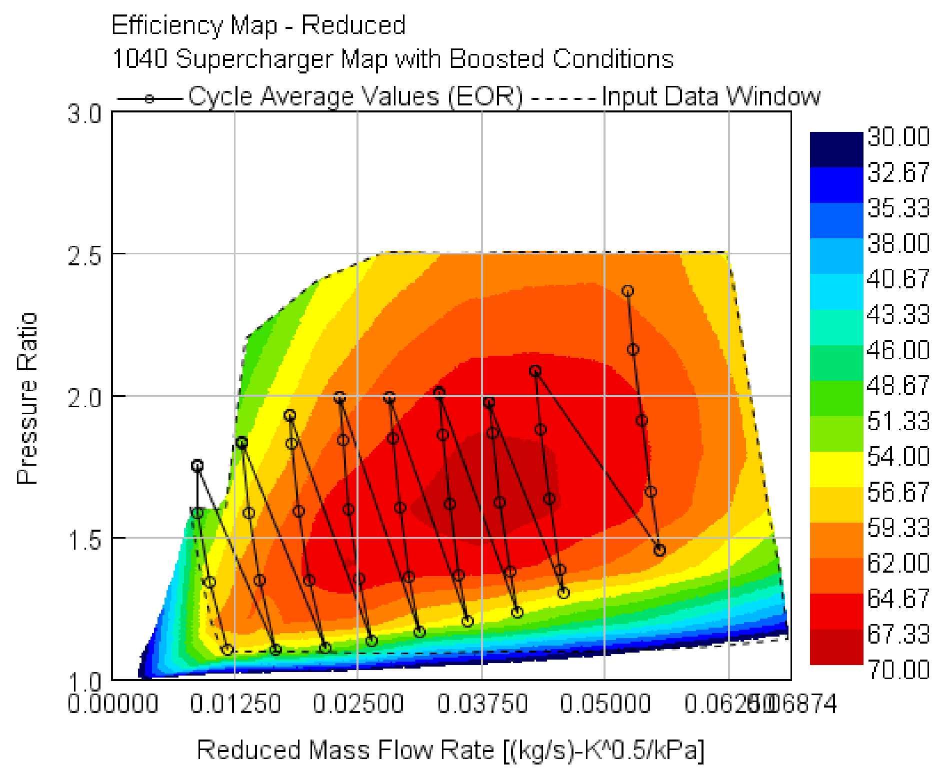




Map Creation – Boosted Conditions

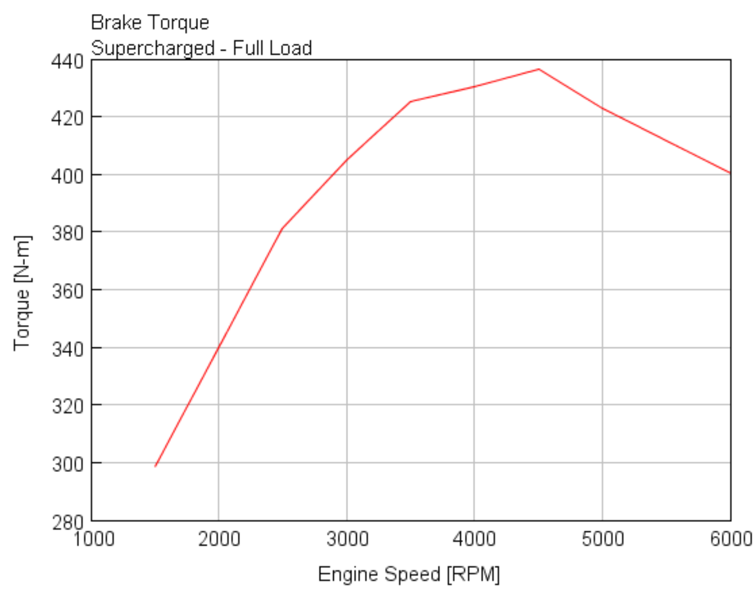
- Figures shows the boosted map points on top of the 1040 supercharger map

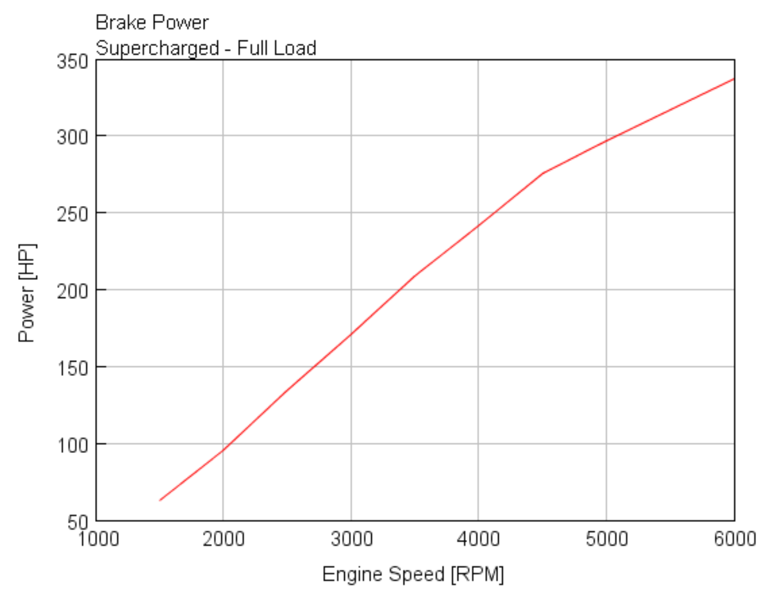
- The placement of points on the map depend on the exact system back pressure, head flow performance, and the pulley ratio. Several variables can be adjusted in getting to the peak performance.



Full Load

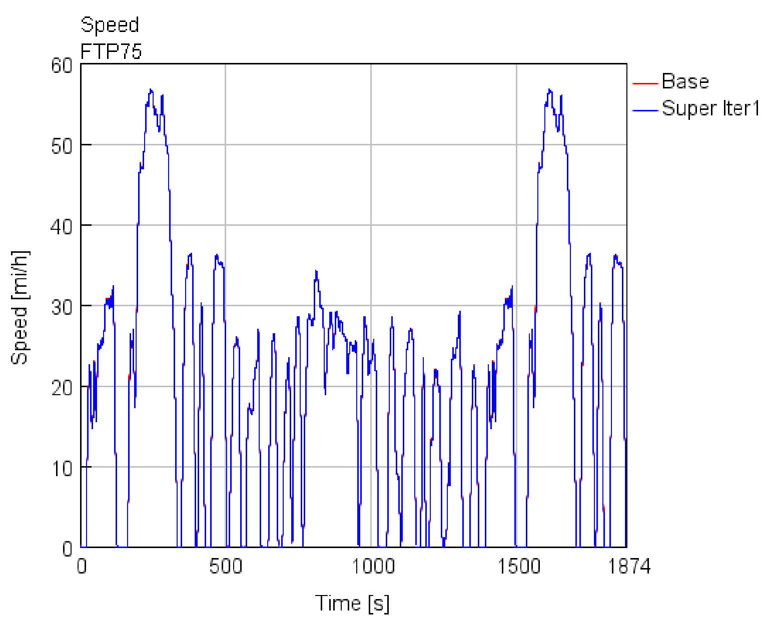
- The current model is ending up at about 340 HP. In order to get the high performance in boosted mode, EngSim had to make radical changes to the head flow performance and open up the exhaust system to reduce back pressure. The engine will need those types of changes as the baseline engine cannot flow air required for these power numbers.

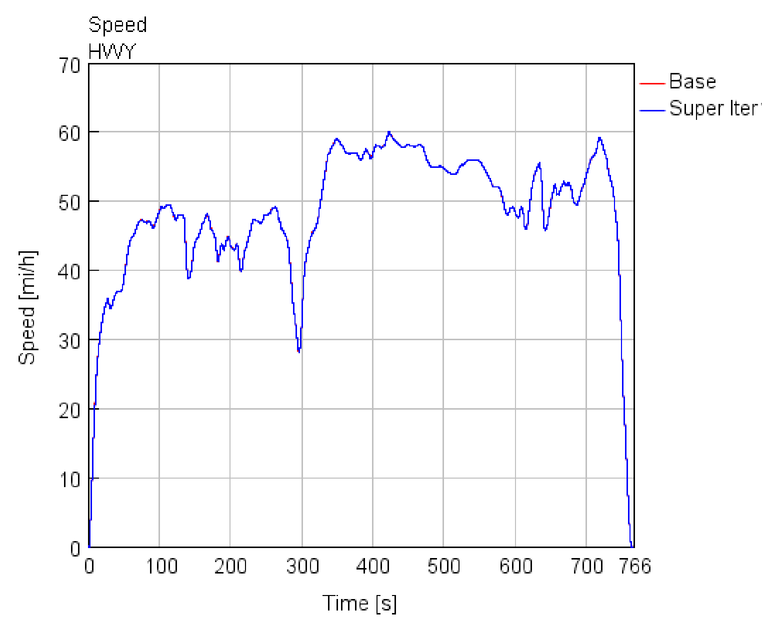




New Map with the 6R80 Transmission

- The old 3.5L map was replaced with the 2.4L supercharged map and the drive cycles were run
- In the first iteration, the 6R80 transmission and shift map were unchanged
- One interesting note, only the map from slide 13 is needed to run the FTP75 and HWY drive cycles. Plots below plot the baseline and new map speed profile on each other. This is to show the new engine has the power to follow the drive cycle.

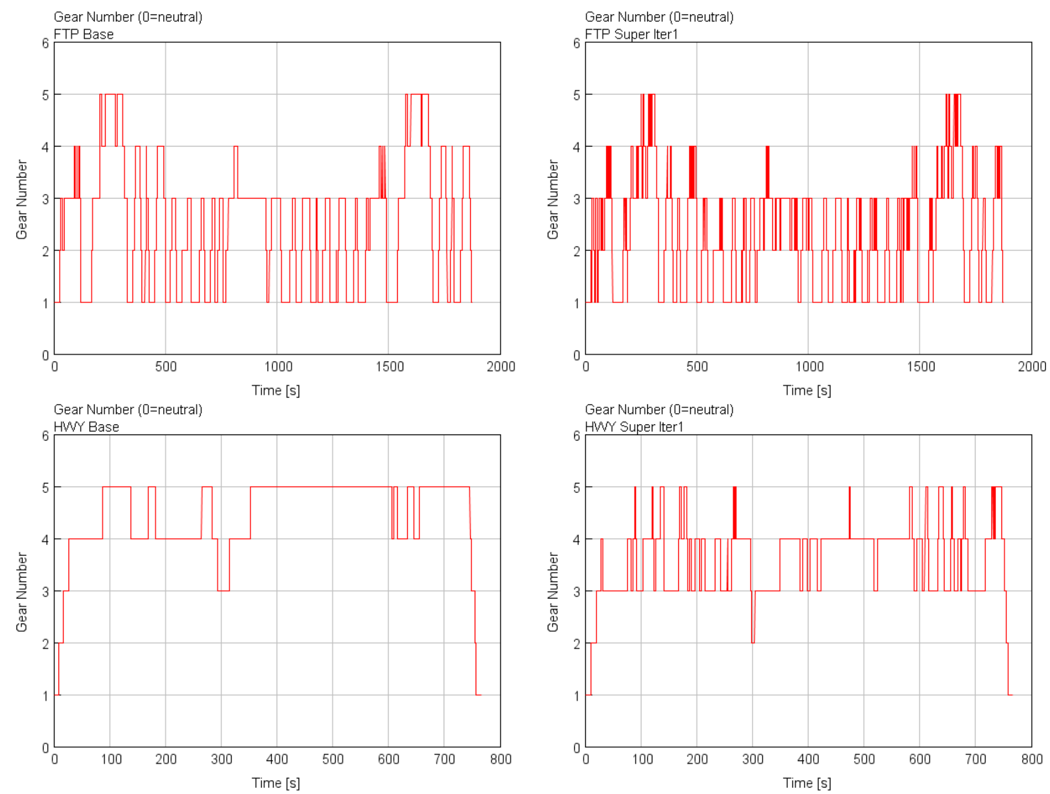




New Map with the 6R80 Transmission

- The new engine map improved the FTP75 and HWY cycles by **20.8%** and **18.9%** respectively. The only change other than the engine fuel map was a 200 lb reduction in the overall vehicle weight.

- The gear number plot comparison for both drive cycles is shown to the right. There are numerous shifting errors in the supercharger setup (right plots) that, once fixed, should improve the fuel economy numbers further.



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6R80 Shift Optimization

- There were many problem in the shift map optimization for the 10R80 transmission so we reverted to optimize the 6R80 shift strategy to work out the issues.

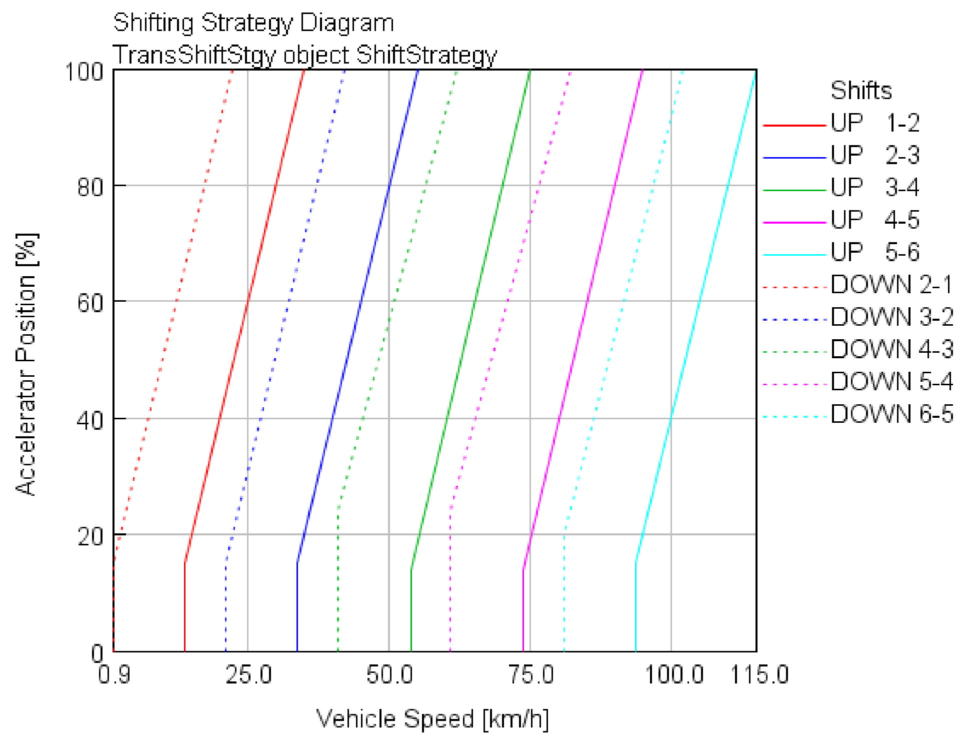
- After considerable time we found that units were converted improperly when running the GT-SUITE DOE. This has been resolved.

- The upshift and downshift have all been tied together using Case Setup formulas so the shift strategy could be optimized be varying only a few parameters

gear_num		Initial Gear Number	1
clutch	%	Normalized Clutch Actuator Position	100
start	mi/h	Y Data	12.9
start2	mi/h	Y Data	=[start(mi/h)]
end	mi/h	Y Data	48
startoff	mi/h	Y Data	=[start(mi/h)]-12.9
startpff2	mi/h	Y Data	=[start(mi/h)]-12.9
endoff	mi/h	Y Data	=[end(mi/h)]-12.9
gear2	mi/h	Y Data	=[start(mi/h)]+[off1(mi/h)]
gear22	mi/h	Y Data	=[start(mi/h)]+[off1(mi/h)]
gear2end	mi/h	Y Data	=[end(mi/h)]+[off2(mi/h)]
gear2off	mi/h	Y Data	=[gear2(mi/h)]-12.9
gear22off	mi/h	Y Data	=[gear2(mi/h)]-12.9
gear2endoff	mi/h	Y Data	=[gear2end(mi/h)]-12.9
gear3	mi/h	Y Data	=[gear2(mi/h)]+[off1(mi/h)]
gear32	mi/h	Y Data	=[gear2(mi/h)]+[off1(mi/h)]
gear3end	mi/h	Y Data	=[gear2end(mi/h)]+[off2(mi/h)]
gear3off	mi/h	Y Data	=[gear3(mi/h)]-12.9
gear32off	mi/h	Y Data	=[gear3(mi/h)]-12.9
gear3endoff	mi/h	Y Data	=[gear3end(mi/h)]-12.9
gear4	mi/h	Y Data	=[gear3(mi/h)]+[off1(mi/h)]
gear42	mi/h	Y Data	=[gear3(mi/h)]+[off1(mi/h)]
gear4end	mi/h	Y Data	=[gear3end(mi/h)]+[off2(mi/h)]
gear4off	mi/h	Y Data	=[gear4(mi/h)]-12.9
gear42off	mi/h	Y Data	=[gear4(mi/h)]-12.9
gear4endoff	mi/h	Y Data	=[gear4end(mi/h)]-12.9
gear5	mi/h	Y Data	=[gear4(mi/h)]+[off1(mi/h)]
gear52	mi/h	Y Data	=[gear4(mi/h)]+[off1(mi/h)]
gear5end	mi/h	Y Data	=[gear4end(mi/h)]+[off2(mi/h)]
gear5off	mi/h	Y Data	=[gear5(mi/h)]-12.9
gear52off	mi/h	Y Data	=[gear5(mi/h)]-12.9
gear5endoff	mi/h	Y Data	=[gear4end(mi/h)]-12.9
off1	mi/h	off1	16
off2	mi/h	off2	24.15

6R80 Shift Optimization

- A new shift map was optimized for the FTP cycle but the optimization can also be optimized for HWY
- With the new shift map the cycle improvement for FTP and HWY has been increased to **27.5%** and **23.6%**
- Optimized 6R80
  - FTP75 – 24.9 mpg
  - HWY – 31.2 mpg



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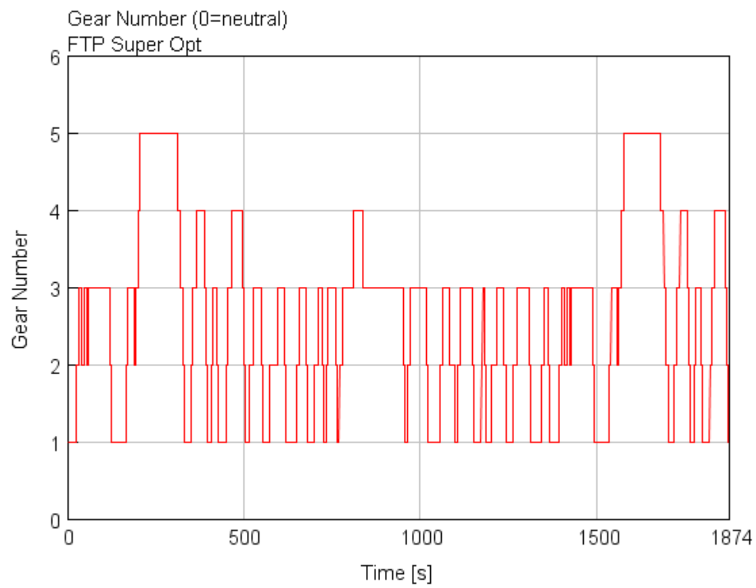
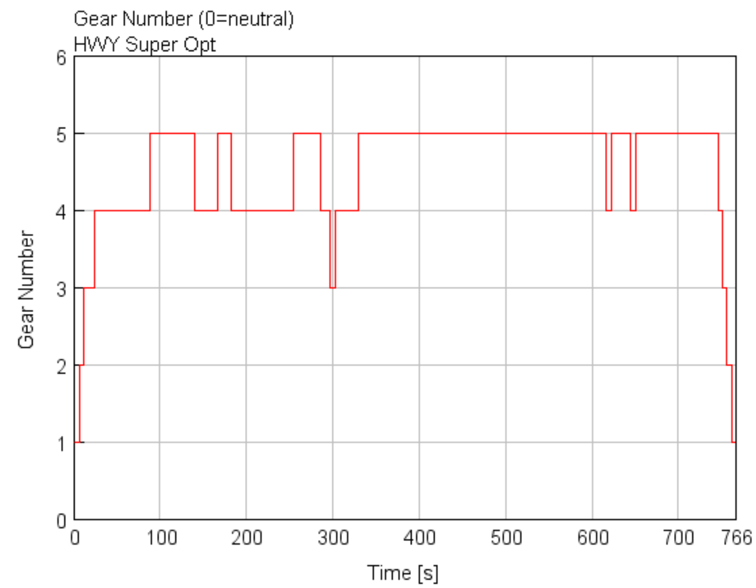
6R80 Shift Optimization

- Now the shifting is much cleaner than the supercharger gear plots shown in slide 15





10R80 Shift Optimization

- The 6R80 optimization technique was carried over to the 10R80
- The 10R80 optimizations did not improve on the 6R80 drive cycle results
- The different transmission gear ratios are shown below. It seems like there should be an improvement and EngSim is still trying a few more techniques for shift strategy.

Forward Gears | Neutral Gear | Reverse Gear

Attribute	Unit	Gear #1	Gear #2	Gear #3	Gear #4	Gear #5	Gear #6	Gear #7	Gear #8	Gear #9	Gear #10
Main											
Gear Ratio		4.69	2.98	2.14	1.76	1.52	1.27	1	0.85	0.68	0.63
In-Gear Efficiency		0.97	0.97	0.97	0.97	0.97	0.97	0.97	0.97	0.97	0.97

Forward Gears | Neutral Gear | Reverse Gear

Attribute	Unit	Gear #1	Gear #2	Gear #3	Gear #4	Gear #5	Gear #6
Main							
Gear Ratio		4.17	2.34	1.52	1.14	0.86	0.69
In-Gear Efficiency		0.97	0.97	0.97	0.97	0.97	0.97

Summary

- The F150 vehicle model was built in GT-SUITE and baseline drive cycles produced reasonable results

- The supercharger map was constructed from several different models running in different operating modes
  - Atkinson throttled
  - Atkinson timing controlled
  - Naturally aspirated with power cam
  - Boosted with power cam

- The supercharger map was put into the F150 vehicle model using the 6R80 transmission. After the 6R80 shifting was optimized, the vehicle saw drive cycle improvements of 27.5% and 23.6% for the FTP75 and HWY cycles.

- The drive cycle improvements are attributed to both engine downsizing and Atkinson, VCR operation.

Re-Examine the Drive Cycle Points

- The following two slides show the locations of maximum fuel usage when using the 3.5L and 2.4L engines in the drive cycle analysis

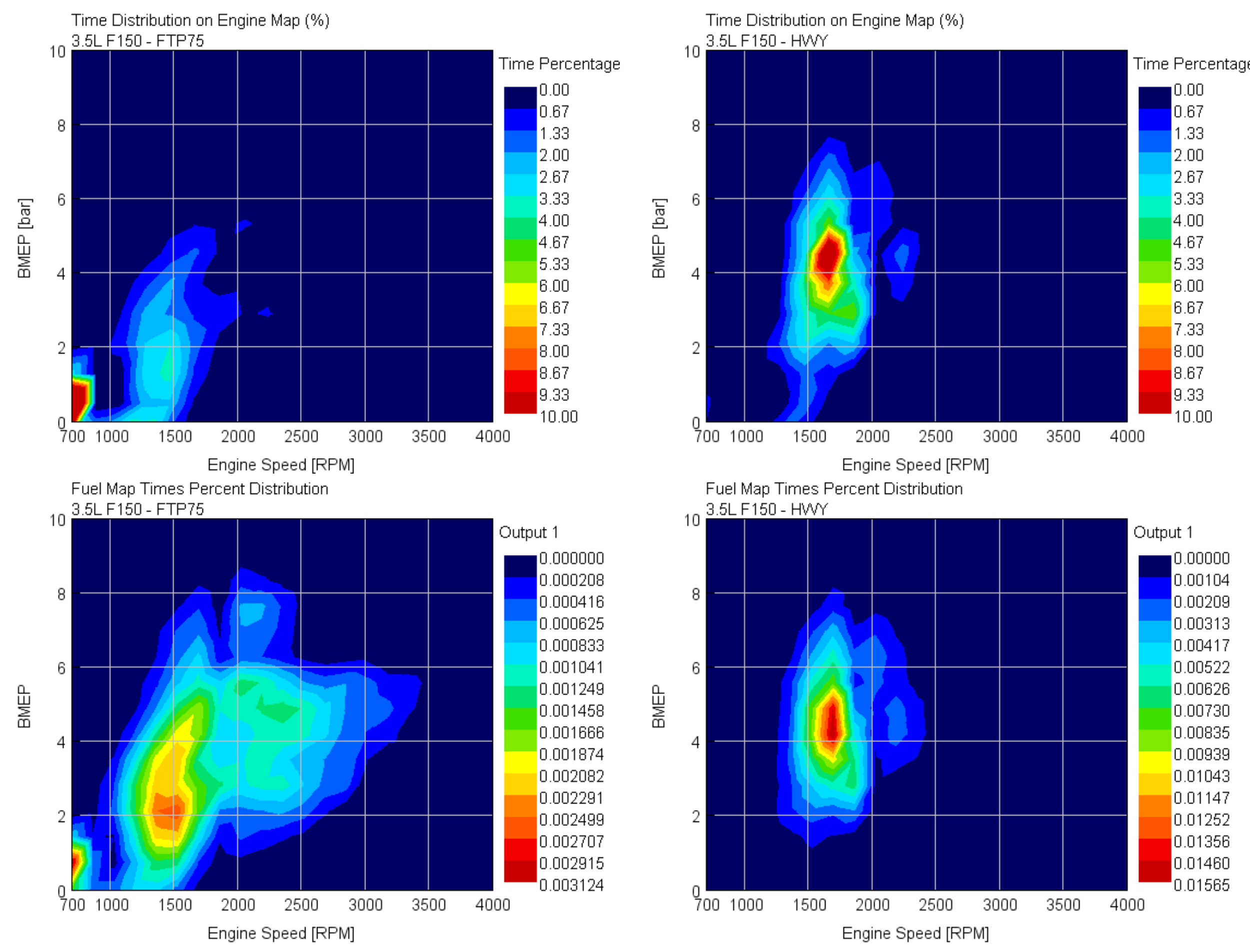
- Envera suggested to use the 2.4L fuel usage data to attempt one additional optimization

- The optimization was to adjust the valve events, as needed, to get the most efficiency from the maximum fuel usage points shown on slide 26
  - FTP – 1400 rpm, 3 bar
  - HWY – 1700 rpm, 6.5 bar

- Previous timing optimization was completed so additional gains should be marginal

Support Slide – Fuel Mapping for Cycles – 3.5L

- This slide attempts to relate where the most fuel is used for the FTP and HWY cycles
- Top figures are a time distribution on the BMEP and RPM map
- Bottom figures show the time distribution times the fuel map – this is a fuel rate weighting of the time distribution
- Bottom figures show where the most fuel is being used for the 3.5L engine



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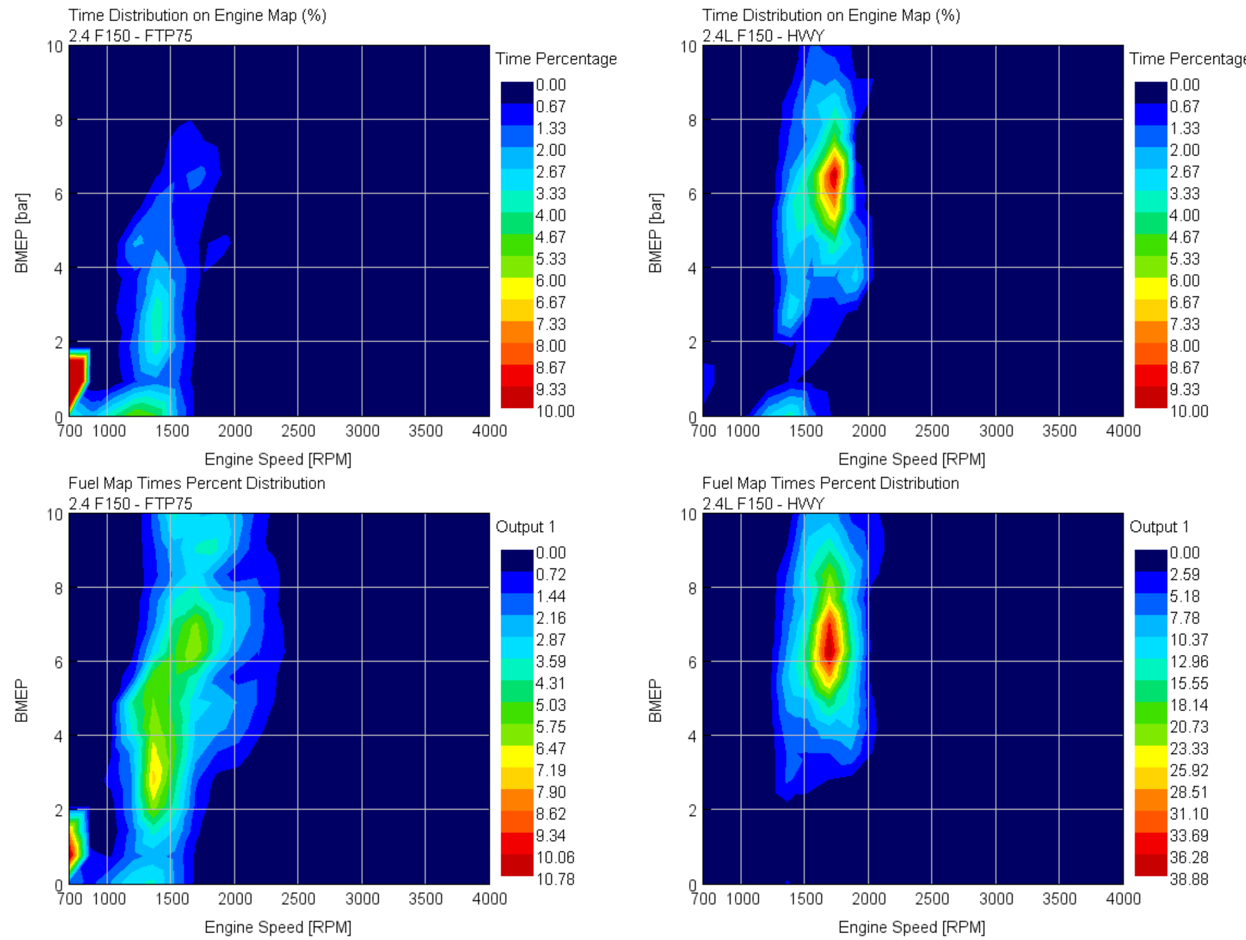
Support Slide – Fuel Mapping for Cycles – 2.4L

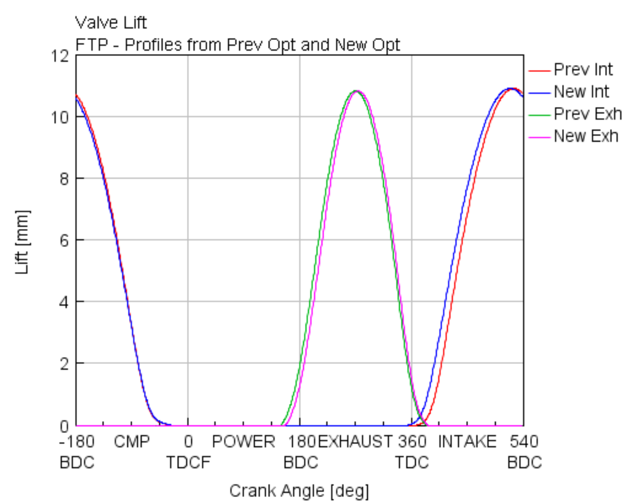
- This slide is similar to the previous slide except it is for the 2.4L engine
- This is the new time and fuel distribution after optimizing the 6R80 transmission shifting
- The change in engine displacement means the average BMEP is scaled up about 40%
- The engine speed is similar between the two engines. You can toggle between the two slides to see the specific differences.

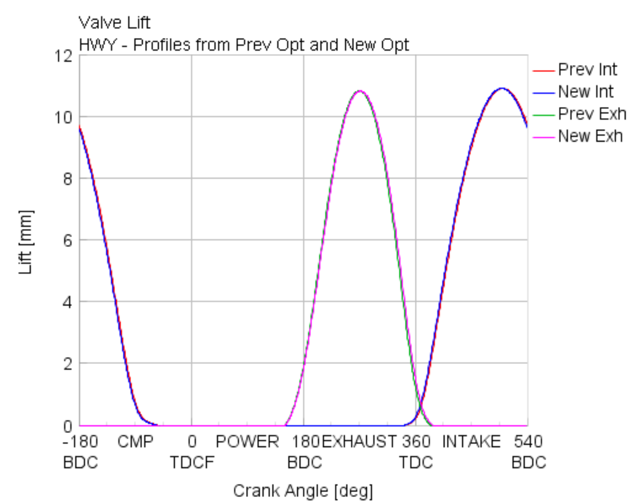


Re-Examine the Drive Cycle Points

- EngSim made previous efforts in finding the optimal timings to find the best BSFC map for the 2.4L engine. Making new optimizations for similar rpms and higher loads should not result in significantly different profiles – and we see that. New and old profiles are shown below with short explanations.

- IVC – Controlled to get the load we want. Comparisons at different loads are not valid.
- IVO – Least important valve event but change a little from the base Atkinson cam
- EVO – At low speeds the EVO tries to be very close to BDC
- EVC – Was constrained by valve-piston interference concerns





Re-Examine the Drive Cycle Points

- The table shows the new valve events (1mm lift) when optimizing for the maximum fuel usage points

	New Opt FTP	New Opt HWY
IVO	388	373
IVC	-69	-96
EVO	180	174
EVC	363	363

- Table below shows the new optimization improving the BSFC for the maximum fuel usage point by about 1.5% over the initial mapping for FTP. Drive cycle results may improve slightly.

	FTP	HWY
BSFC from Mapping	275.1	229.2
BSFC New Optimization	271.1	229.1

Additional Data for Cycle Points

- The table shows additional information for the drive cycle points

	RPM = 1400	RPM = 1700
Engine Speed [RPM]	1400	1700
Brake Torque [N-m]	57.9	123
Brake Power [HP]	11.4	29.3
BSFC [g/kW-h]	271	229
IMEP720 [bar]	3.37	6.81
BMEP [bar]	3.03	6.40
FMEP [bar]	0.344	0.411
Air Flow Rate [kg/h]	33.7	73.2
Fuel Flow Rate [kg/h]	2.30	5.00

A1 ABBREVIATIONS

Argonne National Laboratory	ANL
Bill of materials	BOM
Brake mean effective pressure	BMEP
Brake thermal efficiency	BTE
Brake specific fuel consumption	BSFC
Coefficient of variance	COV
Compression ratio	CR
Computational fluid dynamics	CFD
Computer aided design	CAD
Computer aided engineering	CAE
Continuously variable transmission	CVT
Direct fuel injection	DI
Engine control unit	ECU
Engine management system	EMS
Environmental Protection Agency	EPA
Exhaust gas recirculation	EGR
Exhaust valve closing	EVC
Exhaust valve opening	EVO
Federal test procedure	FTP
Finite element analysis	FEA
Friction mean effective pressure	FMEP
Gasoline direct fuel injection	GDI
Greenhouse gasses	GHG
Highway	HWY
Indicated mean effective pressure	IMEP
Intake valve closing	IVC
Intake valve opening	IVO
Maximum opening point	MOP
Miles per gallon	MPG
National Energy Technology Laboratory	NETL
Noise vibration and harshness	NVH
Port fuel injection	PFI
Pumping mean effective pressure	PMEP
Regulated 2-stage turbocharging	R2S
Society of Automotive Engineers	SAE
US Department of Energy	DOE
Variable compression ratio	VCR
Variable valve lift	VVL
Wide open throttle	WOT