

A Simulation Method to Guide DISI Engine Redesign for Increased Efficiency using Alcohol Fuel Blends

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ABSTRACT

A turbocharged 2.0L 4-cylinder direct injection spark ignition (DISI) engine designed for use with gasoline is simulated using one dimensional engine simulation. Engine design modifications - increased compression ratio, 2-step valve train with dual independent cam phasing and fuel injection timing - are considered in an effort to improve fuel economy with gasoline and take advantage of properties of ethanol fuel blends (up to E85). This paper discusses a methodology to use the simulation to quantitatively evaluate the design modification effects on fuel economy. Fuel consumption predictions from the simulation for each design are evaluated. The goal is to identify the best design with the constraints of hardware physical limitations, engine residual tolerance and knock tolerance. The result yields a specification for a 2-step valve train design and phasing requirements that can improve fuel economy for each compression ratio design.

INTRODUCTION

Ethanol fuel blends have been viewed by many as a viable source of energy over the years [1]. There are several reasons for this view, among them that ethanol can be derived from renewable resources and it can be grown locally thus reducing the dependence on oil imports. Ethanol as a fuel for use in internal combustion engines is compatible with gasoline and can be presently found blended in small quantities (0 to 10% by volume) in gasoline stations throughout the US. Recent oil price escalations and global unrest have led proponents to believe the blends should contain much higher alcohol content, up to 85%, the blend called E85. When a fuel blend only contains 10% ethanol it behaves nearly like gasoline but when the alcohol is 85% it behaves much more like pure alcohol and performs quite different from gasoline in an IC engine. An engine that is E85

compatible may be able to run on that fuel but unless the engine is designed to operate on E85 the engine fuel economy is degraded because a gallon of E85 only has about 70% of the energy content of a gallon of gasoline. Through engine design modifications it is possible to take advantage of the fuel property differences in the alcohol fuels and improve the fuel economy using alcohol fuel blends. The engine is expected to operate on all blends in between gasoline and E85.

The design modifications that are considered for the engine are an increase in the cylinder base compression ratio, addition of variable valve lift for the intake valve and extended range intake valve phaser, fuel injector modifications. The engine control will require knowledge of the fuel content which can be sensed from an in tank sensor module and it is possible there will be a need for high energy ignition system under some operating conditions. The current program is to develop the hardware and control strategy virtually using simulation tools. Performance will be verified on an engine dynamometer with the intent of vehicle implementation.

There have been alternative concepts presented for engine design for the purpose of increasing performance when using alcohol fuels [2,3,4,5,6,7]. One concept is for a dual fuel strategy that uses gasoline and alcohol as separate fuels [3]. This paper documents work assuming that the gasoline and ethanol are blended together as a single fuel.

The concept proposed here is a three part strategy that involves an increase in geometric compression ratio (GCR) of the cylinder, a variable intake valve for air and residual control and improved fuel injection strategy. The modifications all work together in making the engine operation more efficient. The increase in GCR leads to an



increase in efficiency [8]. This works well with high alcohol content fuel that is much more resistant to knocking than regular gasoline. The variable intake valve strategy can be used to reduce the effective compression ratio and amount of exhaust residual gases trapped in the cylinder to control knock with gasoline. Controlling the amount of air entering the cylinder with the valve lift is more efficient than throttling to control load. Employing a dual independent valve phasing strategy allows for controlling the air charge inducted, the residuals that remained trapped. Controlling the intake valve closing angle gives additional variation in the effective compression ratio when the strategy is used with a short early intake closing valve and a long late intake closing valve. The cam phaser requirements for the phaser range of authority are coupled to the lift profiles when using this strategy for engine load control.

In addition to different energy densities and knock properties of ethanol and gasoline fuels, the fuels have different stoichiometry and heat of vaporization. A stoichiometric mix of E85 in air is 0.1 fuel/air ratio while gasoline is about 0.069. This means the fuel injectors need to provide a higher fuel flow. Alcohol has higher heat of vaporization than gasoline so the alcohol fuel has potential to provide more cooling to the air charge. Increased performance by virtue of higher charge density is therefore possible provided the fuel injection is timed properly. Direct injection allows flexibility in fuel injection targeting and timing that can be used to exploit the evaporative properties of the high alcohol content fuel.

To comprehend the coupling of the air delivery and fuel delivery interactions on performance and efficiency a one dimensional model is used. Engine performance and efficiency are computed and compared for a range of engine geometric compression ratios, valve lifts, phases and fuel injection strategies over a range of engine speeds. Fuel efficiency is quantified using a vehicle model with computed engine performance and fuel consumption values. The process that was used to select engine configuration, valvetrain hardware and injection timing, and the impact on efficiency is demonstrated.

MODELING APPROACH

Preliminary work is done using a 1-D Gamma Technologies Inc. GT Power [9] model of the engine, associated engine sub-systems and vehicle. Simulation tools are used to make the design process more efficient and reduce the time and cost associated with testing hardware iterations [10]. This section contains a description of the engine model and the use of the model results to identify the engine systems to be constructed and tested.

STEADY-STATE ENGINE MODEL

The primary tool used throughout the analysis is a steady-state engine model of the 4 valve, 2.0 L, L4 turbocharged engine with direct fuel injection. Initially a model was calibrated to match steady-state speed-load performance test results measured on a production engine with a geometric compression ratio of 9.2 from 800 rpm to 6500 rpm. Engine airflow, imep and FMEP are matched at each operating condition ranging from throttled to boosted at each engine speed. Combustion characteristics, heat release duration and phasing, were determined over the operating range as well and fit to a Wiebe function for use in the modeling. The calibrated model constructed to match the performance of the production engine was then used as a baseline for all further development for improved ethanol fuel performance. This baseline model was modified to reflect changes in base engine configuration, geometric compression ratio, and breathing and fuel injection system hardware changes to enable the performance enhancements.

BREATHING STRATEGY

Direct fuel injection architecture allows for flexibility in controlling the engine airflow and fuel flow separately. Numerous valve lift profiles were evaluated using the model to determine the most efficient profiles to entrain the desired amount of air to best operate at the desired speed and load condition. Both intake valve profiles and exhaust valve profiles were evaluated for their contribution to efficiently control the amount of air and amount of retained residual gas trapped in the cylinder. In addition to evaluating the lift profiles of the valves the phasings of the valves were independently varied to determine the best position for each lift profile.

INTAKE VALVE LIFT

Intake valves play a significant role in controlling the amount of air admitted into the cylinder. In order to improve engine operating efficiency it is desired to have the valve opening appropriately sized to the amount of flow area required and phased so the air enters during the cycle when it is most efficient. The scope of this project is to use a mechanical valvetrain so maximum acceleration and other operating limits exist and must be considered. Such constraints prohibit square lift profiles, high-lift/short-duration profile families that could be possible with electrohydraulic or electromagnetic valvetrain mechanization, but still permit a wide variety of lift profiles to be considered. [Figure 1](#) shows some examples of profiles that were considered using the model. Three lift values are shown in this figure, and are referred to by their maximum lift values of approximately 6 mm, 8 mm and 10 mm. As can be seen from the two 6 mm profiles plotted in the figure profiles having the same maximum lift and different durations were considered in

order to select profiles for their lift and duration profiles somewhat independently.

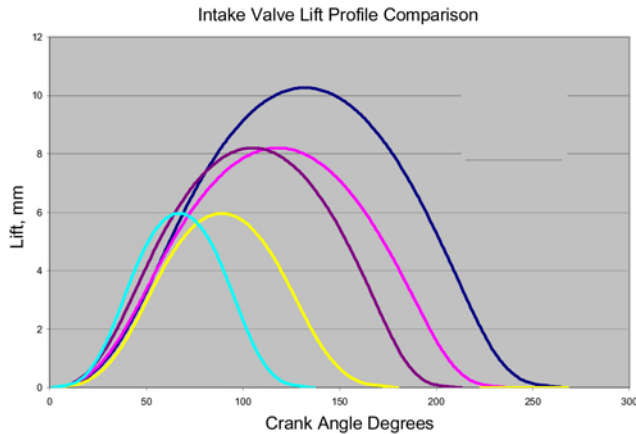


Figure 1. Intake valve lift profiles comparison from 10 mm to 6 mm with various durations

Phasing variation of the profiles was considered in addition to lift and duration evaluation of the profiles. Both early intake valve closing (EIVC) and late intake valve closing (LIVC) strategies were evaluated.

EXHAUST VALVE LIFT

Exhaust profiles were varied throughout the modeling effort in much the same way the intake valve profiles were varied. Lift and durations of exhaust lift profiles were independently varied as well as the timing relative to the engine cycle. Exhaust valve closing (EVC) is varied from early to late closing timing for each profile.

BREATHING RESULTS

Model results were obtained for a set of intake and exhaust cams at a given engine speed over a complete set of phase combinations as shown in the example in [Figure 2](#).

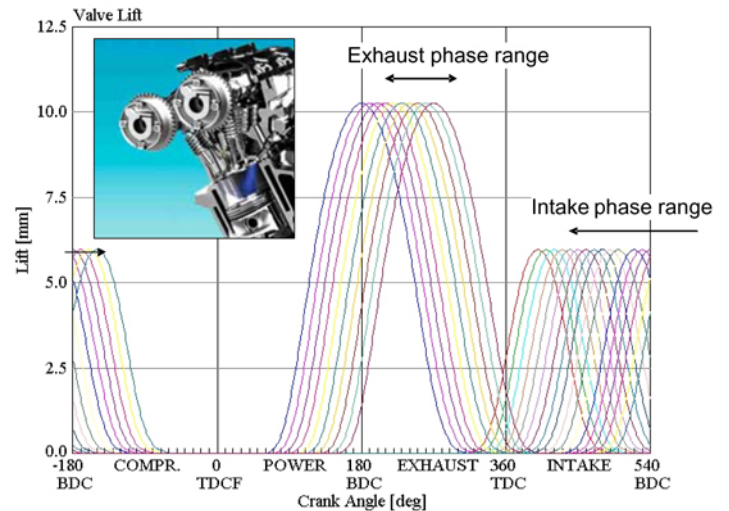


Figure 2. Range of intake and exhaust valve phasings evaluated in the simulation study and hardware implementation of phasers.

Results are presented as a contour map as shown in [Figure 3](#) where the axes are intake valve opening angle on the horizontal, sweeping from early to late from left to right, and exhaust valve closing sweeping from bottom to top on the vertical axis. This makes the lower left corner of the plot represent the early IVO and EVC region, the upper right is late IVO and EVC region. At the upper left region is early IVO and late EVC so this is the maximum overlap area and the lower right corner is where the EVC and IVO directions are apart and opposite, so this represents the negative valve overlap direction of phasing. The range of phase combinations modeled for intake and exhaust valves is chosen to be larger than a phaser may be able to physically achieve. This is done with the purpose to determine the phaser authority requirements.

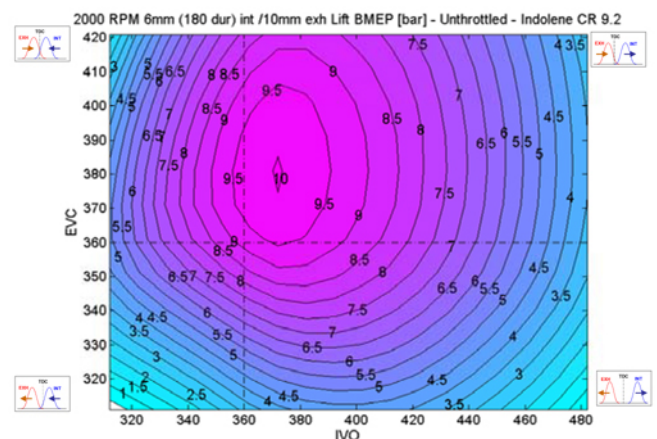


Figure 3. Brake mean effective pressure (bar) contours for range of intake valve opening and exhaust valve opening (CAD) at unthrottled operation.

The contours shown in Figure 3 are of cycle BMEP for an unthrottled engine. This plot is a steady-state result for one engine speed (2000 rpm) with a chosen exhaust profile (10 mm, 250 crank angle degrees duration), chosen intake profile, (6 mm lift, 178 crank angle degrees duration), unthrottled engine operation (1.0 bar MAP) and using indolene fuel. This BMEP contour in Figure 3 shows the BMEP can be controlled using these valves from 10 bar down to 0.5 bar in 0.5 bar increments. The contour plot shows the BMEP surface to be a maximum for a slightly retarded from TDC firing EVC and IVC.

As the phasings of the intake and exhaust valves are changed the load changes. This is due to the volumetric efficiency changing as well as the pumping work differences for different phasings. Variation in volumetric efficiency for different phasings, results in a corresponding fueling map over this domain for these operating conditions. The assumption made throughout the analysis is fueling remains at stoichiometric for the fuel at all operating conditions. Figure 4 shows the fueling map, fuel quantity used in mg per cycle.

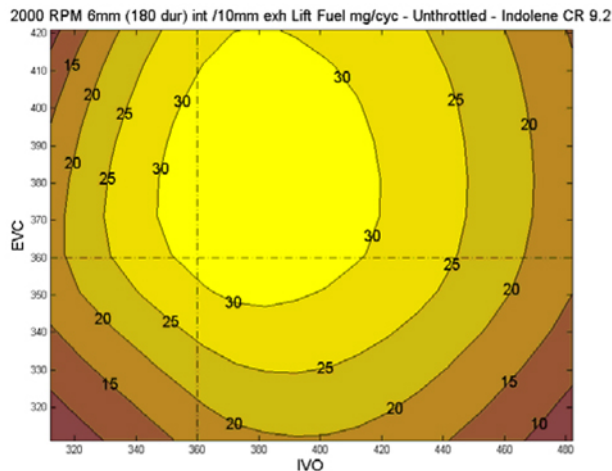
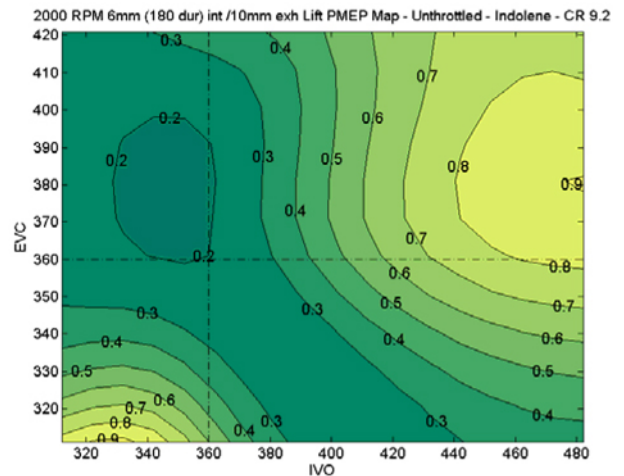


Figure 4. Fuel contours (mg per cycle) for range of intake valve opening and exhaust valve opening (CAD) at unthrottled operation

Controlling the load with the phasing can be more efficient than a conventional throttled engine where the amount of air admitted into the cylinder is done using a manifold depression and incurs a pumping penalty. Figure 5 shows the PMEP for the operating conditions in Figure 3. Here it can be seen that there is a preferred direction to phase the valves in order to minimize engine pumping work. The PMEP contours are lowest along the diagonal line where EVC and IVO are symmetrically spaced about TDC firing position. Early EVC causes the high spot in the lower left corner and when the IVO is late throttling occurs across the intake valve as indicated by the high levels in the right side of the plot.



Log P – Log V for IVO Variation with EVC at 385 CAD

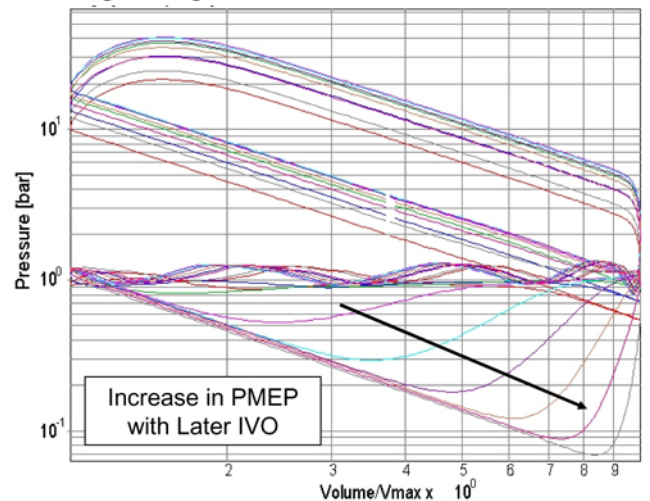


Figure 5. Pumping mean effective pressure, bar, contours for range of intake valve opening and exhaust valve opening at unthrottled operation and log P - log V plot showing pumping increase with late IVO for fixed EVC.

Knowing the fuel usage and brake output from the engine allows for the plotting of a specific fuel consumption map over the phase domain. Again, for the same operating conditions described, Figure 6 shows the brake specific fuel consumption.

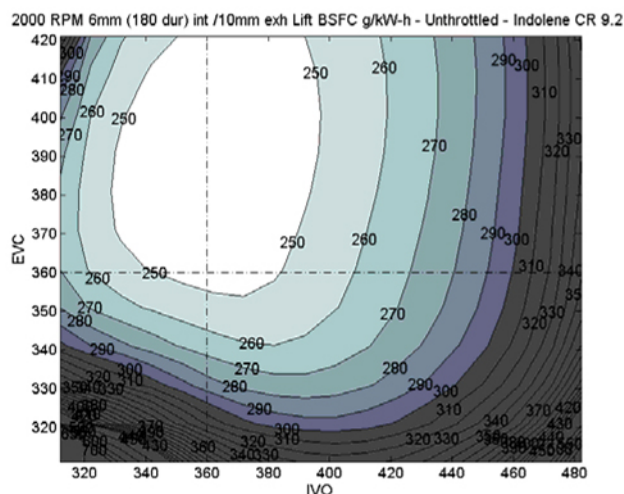


Figure 6. Brake Specific fuel consumption, g/kWh, contours for range of intake valve opening and exhaust valve opening at unthrottled operation

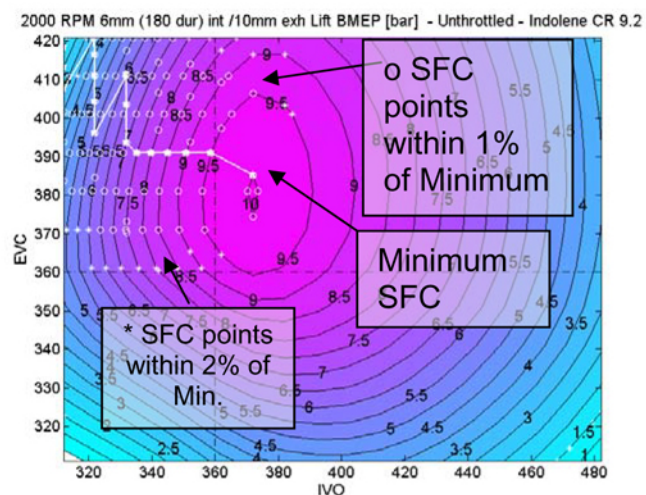


Figure 7. Load contours for phasing inputs with most efficient operating points and load path identified.

Combining results from [Figures 3](#) and [6](#) it becomes evident that there is a most efficient operating phasing at each load and it is possible to identify a most efficient path to control the load under these operating conditions. [Figure 7](#) shows the most efficient operating point at each load level connected by path overlaid on the load contour plot (line connecting starred points). To decrease the load in the most efficient manner starting from the maximum load operating point would require advancing the intake cam and retarding the exhaust cam in the example shown. The numerical method used to identify the optimum path looks for the absolute minimum fuel consumption and sometimes indicates a path that requires large phase fluctuations when the minimum fuel consumption is only insignificantly less than the adjacent points. For this reason a locus of points within 1 percent (circles) and 2 percent (asterisks) of the minimum are identified on the load contours. This indicates acceptable guidelines for smoothing the phasing fluctuations as the load is controlled.

Implicit in this analysis is the assumption that that the combustion is possible over the entire mapped domain and will occur with the same duration and efficiency of the calibrated point at the unthrottled condition with the full lift production valve lift profiles. A consequence of varying the valve phasings is fluctuation in the amount of internal residual gases trapped in the cylinder over the range of the phase maps. [Figure 8](#) shows the exhaust residual gas mass fraction trapped in the cylinder at the closing of the intake valve. The residuals vary from a minimum of near 5% at the maximum load phasing to over 30% in the extreme corners of the phase map at these operating conditions. The limits of residuals that the engine can tolerate depend on operating point, cylinder design, fuel injection system, ignition system, and fuel chemistry. These dilution limits are not known beforehand so some caution must be used when interpreting the model results.