

Development of a Compact Intake Porting Design for a 2-Stroke DI Outboard Engine

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ABSTRACT

Early implementations of direct injection technology were primarily adaptations of fuel systems to existing loop scavenge carbureted engines that did not leverage the strong interaction between the scavenge flow and fuel distribution in the cylinder. Emissions reduction techniques have been limited to engine calibration strategies using injection timing to minimize lost fuel at the expense of mixture preparation and power. This work focuses on manipulating the scavenging pattern to reduce lost fuel while improving mixture preparation and trapped oxygen.

The project goals were to design an intake porting specifically for a 3.4 liter V6 2-stroke DI outboard engine that meets EPA 3 star NTE emissions regulations while increasing power by 10 percent and increasing fuel economy 10 percent over production baseline. Project constraints on the radial space available for the intake ports limited traditional guidance provided by the port walls and required alternative methods to target the scavenge flow.

The multi-dimensional CFD code KIVA-3V2 was used to simulate and design the intake ports. The model setup is a single cylinder with pressure boundary conditions incorporating an outwardly opening hollow cone fuel injector. CFD results, both integrated and spatially resolved, are evaluated on the basis of scavenging, mixture preparation and lost fuel out the exhaust port.

Correlation of scavenging performance was achieved by comparing delivered oxygen levels calculated from emissions data to simulation results. For mixture preparation, a qualitative correlation was achieved by using experimental metrics including specific fuel consumption and power output sensitivity to injection timing.

Simulation and experimental results are presented for two compact porting designs that strike different balances between lost fuel and mixture preparation. One of these new configurations had a unique scavenge pattern that allowed for substantially increased fuel penetration while maintaining good fuel trapping. This penetration allowed for improved interaction with exhaust plug induced flow and an associated improvement in mixture preparation.

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INTRODUCTION

The evolution of the 2-Stroke outboard marine engine has typically been isolated to the method of fuel delivery and driven by increasingly stricter emissions regulations. The evolution of fuel delivery for 2-stroke outboard engines has progressed from carbureted, to fuel injection, and then to first and second generation direct injection. While direct injection technology has provided substantial gains in emissions and fuel economy across the entire operating range, to date little work has been done in optimizing the scavenging pattern to maximize fuel trapping without sacrificing mixture preparation. Traditional calibration strategies for reducing emissions in DI 2-stroke engines focus on timing the Start of Injection (SOI) to minimize the amount of lost fuel. Since

injection is occurring during cylinder scavenging, the path the fuel cloud takes is strongly influenced by the scavenging pattern produced by the intake ports. The traditional looping pattern that has provided good power performance in carbureted engines also convects the fuel spray of a DI engine to the exhaust port. The reduction of lost fuel is achieved by allowing less time for the scavenging flows to convect fuel to the exhaust port. However, this strategy also reduces the amount of time for the fuel spray to mix with the air delivered during scavenging. As a result, emissions can be decreased but so is power output.

Relying on engine calibration alone to achieve the emission targets is not sufficient for the next generation of emissions regulations. Currently, emissions are determined

Table 1. ICOMIA Five Mode Steady State Marine Emissions Test

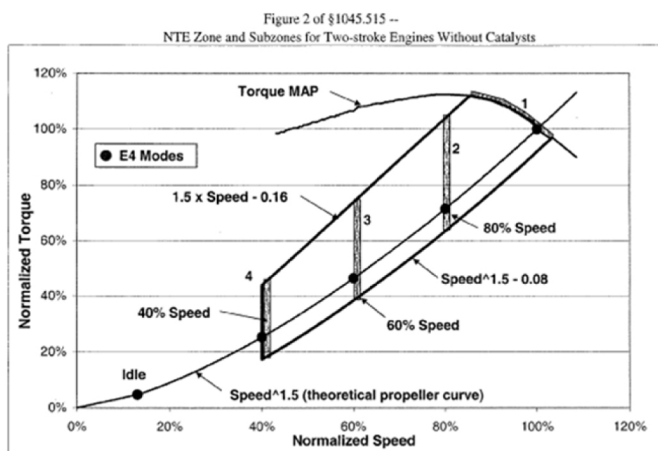
Mode	% RPM	% Torque	% Weight Factor
1	100	100	6
2	80	71.6	14
3	60	46.5	15
4	40	25	25
5	Idle	0	40

Table 2. Development Engine Configurations

Configuration	Production	Proto 1	Proto 2
Displacement	3439 cc	3439 cc	3439 cc
Bore (mm)	98	98	98
Stroke (mm)	76	76	76
Connecting Rod (mm)	152	152	152
Compression Ratio (geometric)	10.3	10.9	10.9
Compression Ratio (trapped)	6.2	6.9	6.9
Exhaust Port Timing (BTDC)	89°	89°	89°
Intake Port Timing (BTDC)	121°	127°	118°

by testing engines on the ICOMIA five mode weighted, steady state test summarized in Table 1 [1].

Recently, the EPA has revised the test to include Not-To-Exceed (NTE) zones that are an additional set of test points defined by speed and load curves for propeller driving engines, shown in Figure 1 [2]. For two-stroke outboard engines, a traditional 5 Mode Certification is still required, but additionally engines must also pass a screening of points throughout the NTE zone for which a weighted analysis of any of the points must be less than 1.2 times the Federal Emissions Limit (FEL) for that engine family [3].

**Figure 1. Two-Stroke Specific NTE Zones and Subzones.**

In order to meet the new emissions requirements, a new intake port development effort was initiated to focus on manipulating the scavenging pattern to reduce lost fuel without excessively degrading mixture preparation and trapped oxygen. The goals for the program were to increase emissions compliant power output by 10% and improve fuel economy 10% over the current production second generation DI engine. The engine platform selected for development is outlined in Table 2 along with the two prototype designs generated in this study. It is based on a 3.4 liter V6 2-stroke DI outboard engine currently in production rated for 250 to 300 horsepower.

The project was constrained in several aspects of engine architecture. The bore, stroke and bore spacing were specified to remain the same as the production engine in order to meet cost, weight, commonality, and engine envelope targets. The new porting design was also constrained to be an unnested configuration. This imposed significant limitations on the flexibility for intake port designs as the unnested configuration combined with the same bore spacing greatly reduced the depth of the transfer ports and their ability to guide the jet stream exiting from the port during scavenging as Figure 2 shows. Please refer to Figures 3, 5, and 7 for port naming conventions.

The strategy for the project was to rely heavily on CFD simulation to develop the intake port designs prior to prototyping and testing. This allowed for very wide range of designs to be explored in a low cost, timely manner that proved to be effective and key in the development process. This process was based on an iterative loop starting with hypothesis generation and simulation. Once viable hypotheses were identified, the loop expanded to include

prototype hardware testing and results analysis. The loop was closed by providing feedback from experimental results to the original hypotheses, which could then start a new simulation loop. In this paper, simulation and experimental results are presented for three select configurations: production, prototype 1, and prototype 2. All are based on the same engine architecture with the main differences being in the intake porting.

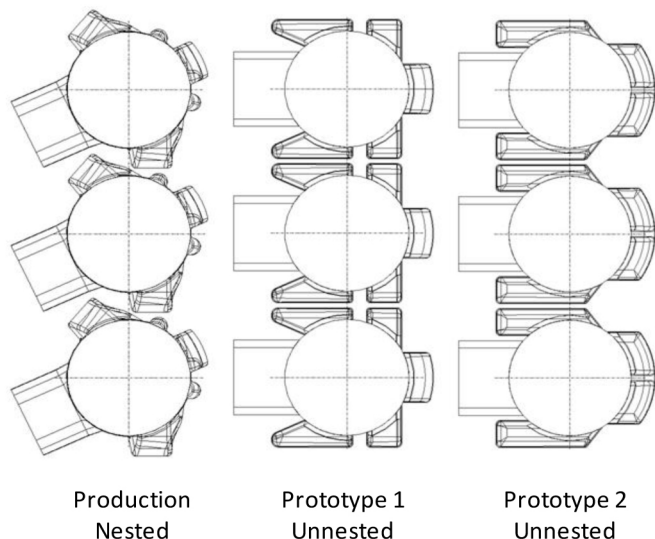


Figure 2. Nested vs Unnested Porting Layouts

The baseline production intake configuration is a conventional loop scavenged 3 port design with two main transfer ports, one boost port and two additional finger ports (Figures 3 and 4). It is based on an earlier carbureted engine. The ports are nested which allows for much more side wall guidance targeting of the transfer ports as compared to the unnested prototypes. Power was the focus of production intake port design more than fuel management via scavenge pattern.

Prototype 1 has a 5 port intake (Figures 5 and 6) with a focus on minimizing the lost fuel out the exhaust port and hence reducing the hydrocarbon emissions. As previously discussed, the ports are not nested due to project constraints. This combined with the tight bore spacing results in very little side wall for targeting guidance for all four of the transfer ports. As a result jet targeting is achieved through the balance of the momentum between the boost side and exhaust side transfers. The larger exhaust side transfers impinge on the smaller boost side transfers redirecting their flow towards the boost wall.

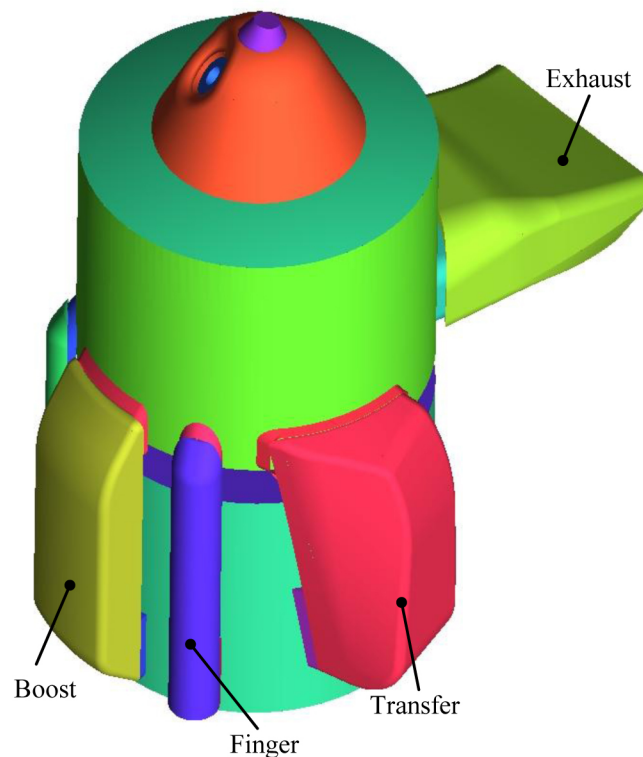


Figure 3. Production

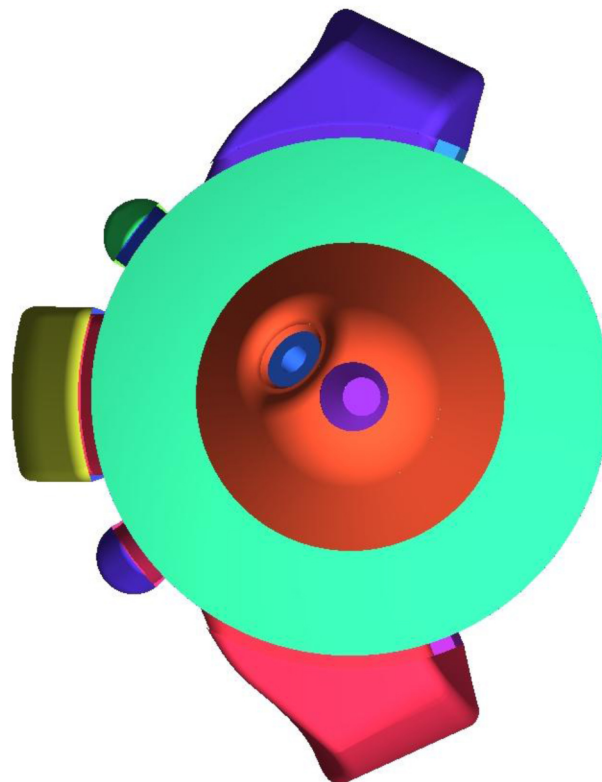


Figure 4. Production Top View

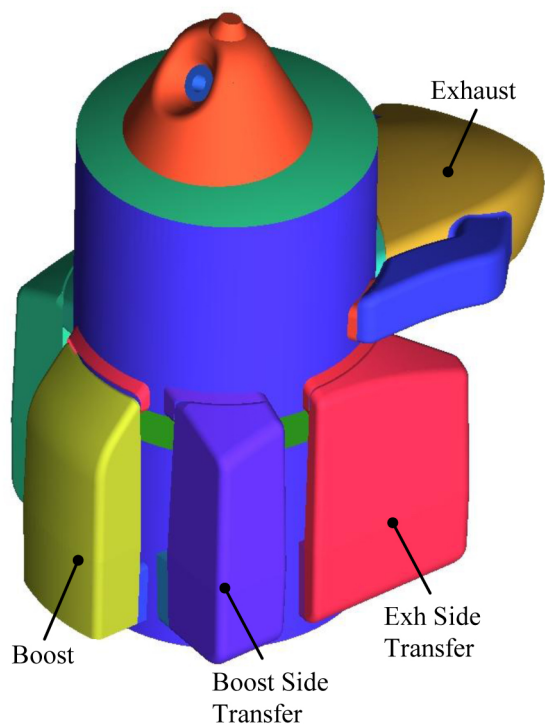


Figure 5. 5 Port Prototype 1

flow area centroid in the transfer port runner relative to the sleeve opening.

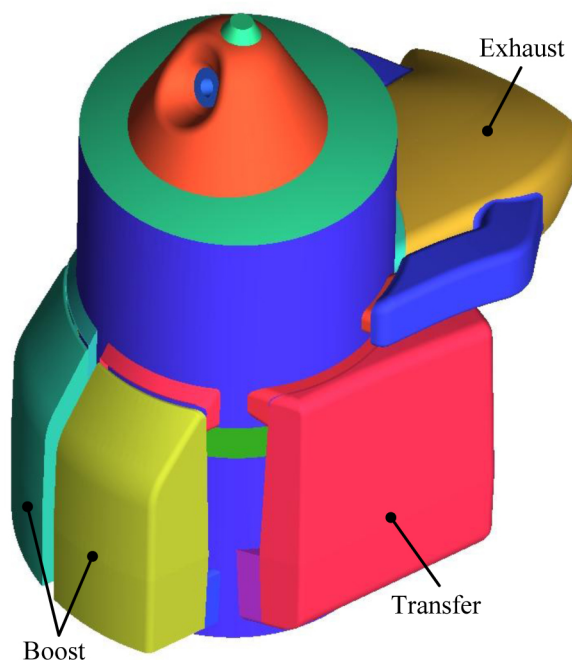


Figure 7. 4 Port Prototype 2

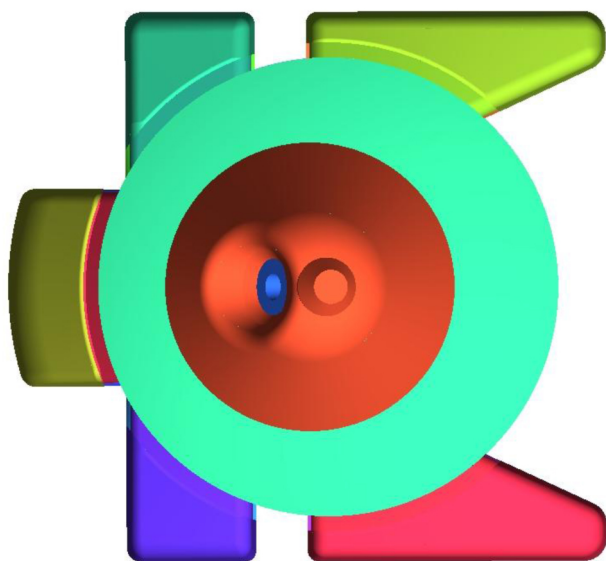


Figure 6. 5 Port Prototype 1 Top View

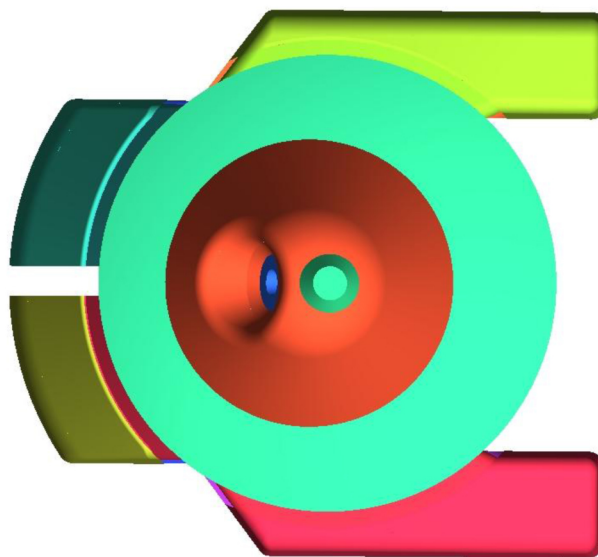


Figure 8. 4 Port Prototype 2 Top View

Prototype 2 has a 4 port intake ([Figures 7 and 8](#)) with a focus to balance the often competing goals of lost fuel out the exhaust port and adequate mixture preparation. The design was initiated after testing of prototype 1 indicated the possibility of mixture preparation issues. The ports are also unnested like prototype 1. The boost side transfers are eliminated since they appeared to be an obstacle for achieving boost wall targeting of the exhaust side transfer. Targeting is achieved by manipulating the location of the

MODELING APPROACH

The modeling focus was to predict the impact of intake port geometry on the cold flow portion of the cycle, specifically the interaction between the scavenged induced charge motion and the fuel cloud as it pertains to mixture preparation and lost fuel. Scavenge efficiency was also

considered as high levels of trapped oxygen are needed to achieve project power targets.

A modified version of the well established engine simulation code KIVA-3V2 [4] [5] was used for the CFD modeling in this study. The primary modifications to the base KIVA-3V2 code relevant to this effort were the addition of a 0-dimensional crankcase control volume and beam based reed model and the addition of a hollow cone injector break up model. All of these modifications have been discussed in previous papers and detail can be found in the references [6] [7].

Hex mesh of the geometry was generated using ICEMCFD (Figure 9). The base mesh size is 2 mm square with additional elements added as required to capture geometric features. The average mesh count is 550k elements. Runtimes on single processor Zeon 5570 are on the order of one day.

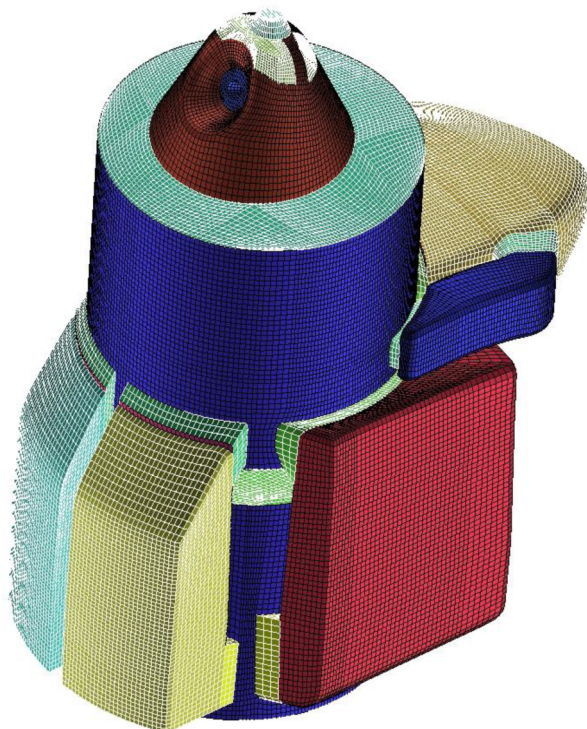


Figure 9. Typical Hexmesh

The CFD model is based on a single cylinder of the V6-engine. The top cylinder was selected as being representative of the V6 engine and used in all modeling although it should be noted that some cylinder to cylinder variation exists in the actual hardware due to exhaust system design especially for the production configuration. The inlet boundary condition is placed at the top of a cylindrical volume located below the piston at BDC simulating a portion of the crankcase volume. This places the boundary condition a distance away from the actual intake runners entrance thus avoiding an unphysical over constraint of the static pressure at this high gradient entry region. The size of the 0-dimensional volume is adjusted

to make the summation of the combined crankcase and intake runner mesh volume, and 0-dimensional volume equal the physical volume. The outlet boundary condition is located at a cross sectional plane in the exhaust runner downstream of the exhaust port. Its position is a compromise of the experimental pressure measurement location and the need to prevent unknown constituent outlet boundary backflow from entering the cylinder during exhaust plugging and thus altering the predicted purity. The distance from the exhaust port to the outlet boundary condition plane is kept consistent for all configurations.

The port geometry is represented as accurately as possible within the snapping and structured mesh constraints of KIVA-3V2 software. Some compromises to the port opening curvature were necessary but not believed to be significant. Although not typically done in KIVA due to meshing difficulties, a simplified version of the spark plug and retaining boss geometry was included to improve the fidelity of the fuel cloud prediction.

For consistency, all models are run using the exact same boundary conditions, injection parameters and initial conditions. This approach isolates geometric effects without the added complication of different boundary conditions. All models are run at WOT and 5500 rpm.

Pressure boundary conditions are used at the inlet and outlet to drive the model as shown in Figures 10 and 11. Because changes are being made to the intake port geometry and timing it is important to have the inlet pressure boundary react to the geometric changes. This is done by using a 0-D crankcase control volume model to provide the inlet pressure boundary condition. The crankcase pressure at the beginning of the solution (50 deg ATDC) is pegged to the experimental data. The outlet pressure boundary condition is provided by experimental data. Alternatively, if experimental data was not available either predicted values from a 1-D gas dynamics code like Optimum Power or from a multi-cylinder Kiva prediction could be used.

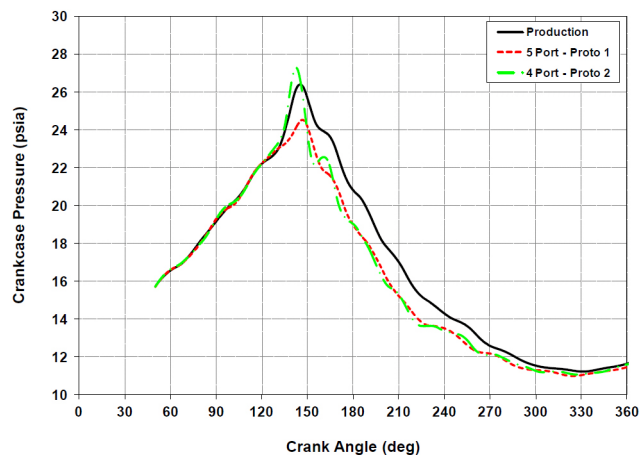


Figure 10. 0D Model Based Inlet Pressure Boundary Condition

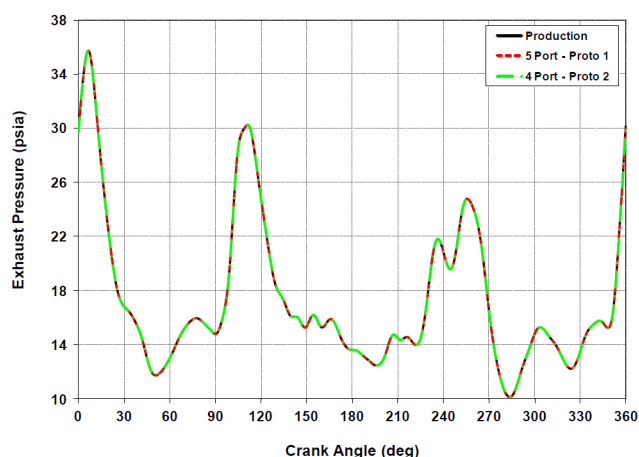


Figure 11. Exhaust Outlet Boundary Condition

The model is initialized at 50 deg ATDC based on the experimental boundary condition pressures, measured incylinder pressures, and temperatures provided by our 1-D gas dynamic code Optimum Power. Velocities are initialized to zero because the blowdown event drives the velocity distribution in the cylinder prior to the beginning of scavenging. Consequently only one cycle is run during the simulations.

The injector geometry is based on an outwardly opening hollow cone ETEC design. The injection event is modeled using a prescribed injection normalized velocity and lift vs time as described in [7]. The injection timing is selected to be consistent with good emissions not maximum power.

SIMULATION RESULTS

The simulation results are evaluated both objectively using integrated parametric values and subjectively using 3-D visualization.

The process begins by objectively reviewing scavenge related integrated results as shown in Figures 12, 13, and 14 which include delivered oxygen, trapped oxygen and purity. The simulation predicts a significant improvement in delivered oxygen for both prototypes relative to the production design (see Figure 12). The improved gas exchange with the cylinder is a result of the increased time area of the intake ports and exhaust port. The larger blowdown area of the exhaust port also fosters earlier flow into the cylinder by a reducing the residual backpressure on the intake ports.

The predicted trapped oxygen is shown in Figure 13. Consistent with the delivered oxygen trends, both of the prototypes have higher trapped oxygen than the production configuration. Although the trend is the same, the percent of improvement is less for the trapped oxygen than the delivered. Improvements in trapped oxygen are much more difficult to achieve than delivered. In many cases a significant improvement in delivered oxygen will result in little or no

improvement in trapped because trapped is driven by both purity and trapped pressure.

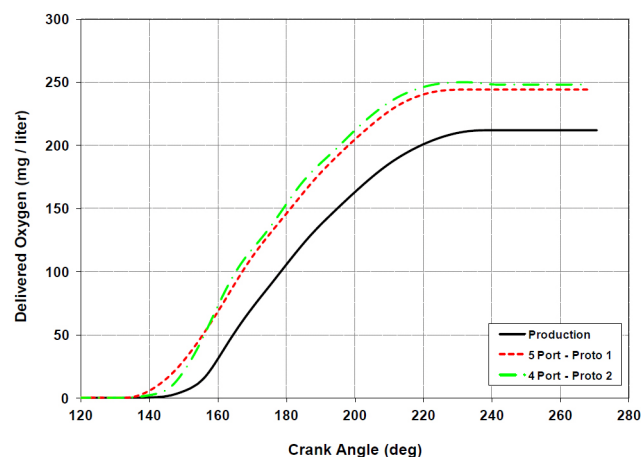


Figure 12. Delivered Oxygen Per Liter

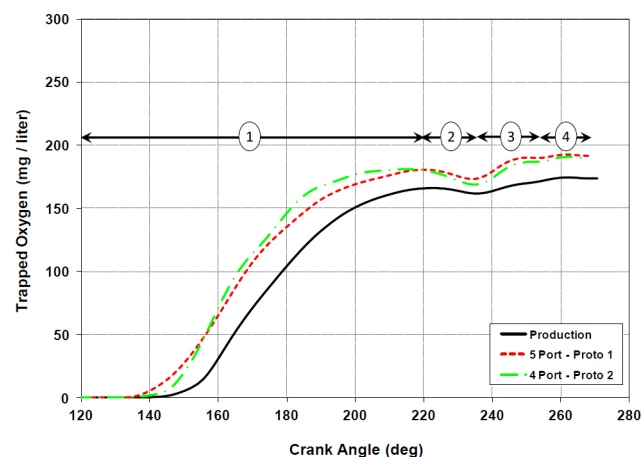


Figure 13. Trapped Oxygen Per Liter

To better understand Figure 13, the results are examined based on four distinct phases that comprise the whole scavenge event. Phase 1 is the “initial scavenging” which is strongly linked to the level of delivered oxygen. In terms of geometry, it is mainly controlled by the time area of the intake porting, ability of the exhaust system to fully lower the cylinder residual pressure prior to exhaust port opening and the effectiveness of the intake port targeting. Both prototypes perform better than the production during this phase because they have additional intake and exhaust time area which drives up the delivered oxygen. The second phase is the “bleed down”. During this portion of the cycle oxygen is lost from the cylinder. The piston is moving up reducing the in cylinder volume which causes the oxygen to flow out the exhaust during a time when the crankcase pressure is low and intake flow into the cylinder cannot offset the exhaust outflow. Contrary to phase 1, additional intake and exhaust time area aggravate this issue in the prototypes. The increased intake time area drops the crankcase pressure faster and