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DigitalAir™ Camless FVVA System – Part 2, Gasoline Engine Performance Opportunities

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Abstract

The paper describes a completely new approach to fully variable valve actuation (FVVA), which allows almost unlimited continuously variable control of intake and exhaust valve opening and closing events, and duration without the use of a camshaft.

DigitalAir replaces conventional poppet valves with horizontally actuated valves located directly above the combustion deck of the cylinder head, which open and close a number of slots connecting the cylinder with the intake and exhaust ports, Figure 1. The stroke of the valves to provide the full flow area is approximately 25% of the stroke of the equivalent poppet valve, thus allowing direct electrical actuation with very low power consumption. This design arrangement also avoids the risk of poppet valve to piston collision, or the need for cut-outs in the piston crown, since the valves do not open into the cylinder.

The paper will present analytical and experimental data which confirms that the proposed FVVA system can meet the basic performance requirements of modern GDI engines with respect to breathing characteristics across the speed range, throttleless operation at and above idle, opening and closing event optimization, cylinder deactivation, control of residual gas fraction / scavenging and exhaust thermal management.

Analytical results were developed using GT-POWER Cycle Simulation and CONVERGE computational fluid dynamics (CFD). Cycle simulation was used to study the system level performance, such as full load capability and transient response, and in particular to quantify the fuel consumption benefits of throttleless operation. CFD was used to better understand the opportunities for in-cylinder charge motion – tumble, swirl and turbulence.

JP SCOPE Inc. has been running experimental engines with DigitalAir for several years and has successfully completed performance and durability tests. The mechanical and thermal design of the cylinder head, and the design of the actuator will be covered in Part 1 of this paper [1].

Introduction

The benefits of FVVA for gasoline engines are well understood and well documented, for example by Schechter and Levin of Ford Motor Company [2]. With the development of turbocharged GDI technology, combined with downsizing, the opportunities for FVVA are perhaps even greater, especially given the regulatory pressure to reduce CO_2 emissions.

Fully variable valvetrain actuation may be defined as the complete freedom to command engine valve events directly from an engine control unit (ECU), for example as occurred with the development of common rail fuel systems. This would include the freedom to command the timing of intake and exhaust opening and closing events, and hence the duration. The definition of FVVA should also include fast response such that the valves may be commanded within one engine cycle.

The most significant impediment to achieving FVVA is the camshaft, which ultimately restricts the range of valve events that can be achieved, since the driving force for valve operation is provided by the cam lobe. Several hundred mechanical and hydraulic mechanisms have been proposed as a means to modify the physical connection between the cam lobe and the poppet valve in order to provide some degree of control over intake and exhaust valves [3,4]. Some of these are available today in production vehicles, for example the Schaeffler UniAir [5,6], Audi AVS [7,8] and BMW Valvetronic [9]. Cam phasing systems (VCP) are widely available and often combined with variable valve systems (VVA) to provide a range of timing, lift and duration flexibility.

The documented studies of FVVA have, almost without exception, involved the development of fast response electro-hydraulic actuators which act upon conventional poppet valves [2,10-20]. With the possible exception of Koenigsegg [20], such systems have failed to progress beyond the research stage despite intensive efforts to move them forward. Perhaps the most significant impediment to successful commercial application is the poppet valve itself, which requires a relatively long stroke, high opening forces (at least on the exhaust valve), and carries the risk of valve to piston contact. Overcoming these factors ultimately drives cost, complexity, high energy consumption and packaging challenges into the electro-hydraulic actuator which thus far have restricted their application to the engine research laboratory.

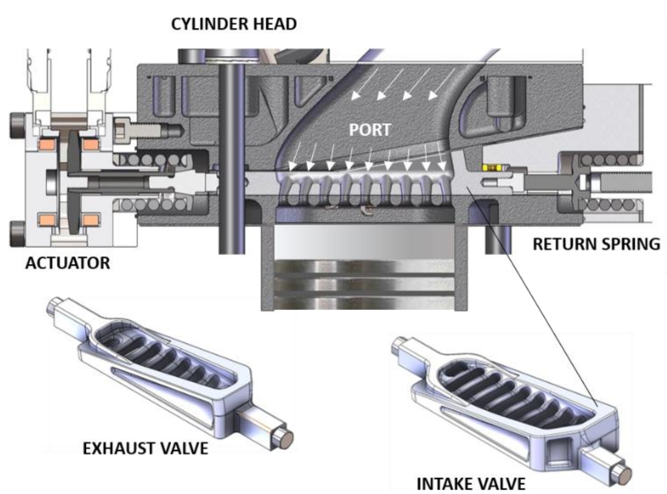

Figure 1 – Horizontally acting intake and exhaust valves with direct electrical actuation

DigitalAir addresses the limitations to FVVA posed by the poppet valve, and at the same time eliminates the need for electro-hydraulic actuation. Figure 1 shows a cross-section through the cylinder and cylinder head of the DigitalAir system. Also shown are intake and exhaust valves. The important performance features will be discussed in a later section, supported by analytical results from cycle simulation and CFD and experimental results from single cylinder engine testing.

Brief Review of Camless and Cam-Driven VVA Systems

Camless Valve Systems

There have been several notable attempts to develop cost-effective and efficient camless valve systems. Without exception, these have retained the poppet valve and sought to drive the intake and exhaust valves directly using electrical or electro-hydraulic actuation. Perhaps the first of these developments was that by Richman and Reynolds [10]. They developed a hydraulic actuator controlled by a fast acting servovalve. The system is shown in Figure 2.

The performance of this system was successfully demonstrated at engine speeds up to 1000 r/min. This system was used as a research tool to explore the benefits of valve control on engine performance and emissions.

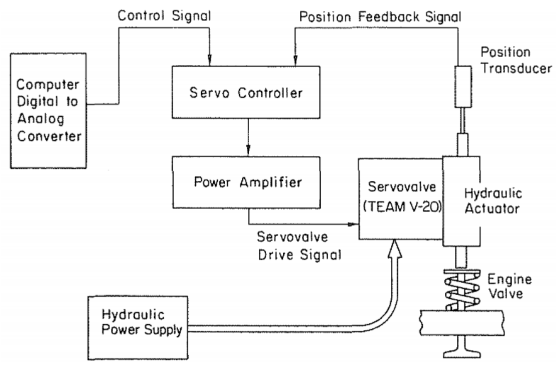

Figure 2 – Valve-actuator system block diagram [10]

Ford engineers Schechter and Levin [2] developed a camless engine, again using poppet valves and an electro-hydraulic actuation system. The authors referred to the system as an electro-hydraulic camless valvetrain (ECV). The design used a novel hydraulic pendulum, shown in Figure 3. In Part I of their paper, they outlined the potential benefits of a camless engine, for example:

- Freedom to independently schedule valve lift, duration and placement,
- Reduced throttling,
- Improved fuel consumption,
- Improved volumetric efficiency,
- Improved torque at low and high engine speeds,
- Lower idle speeds and idle stability – by better control of residuals,
- Ability to deactivate cylinders for variable displacement,
- Improved packaging,
- Influence over charge motion and turbulence,
- Control of effective compression and expansion ratios, and
- Ability to create and control internal EGR.

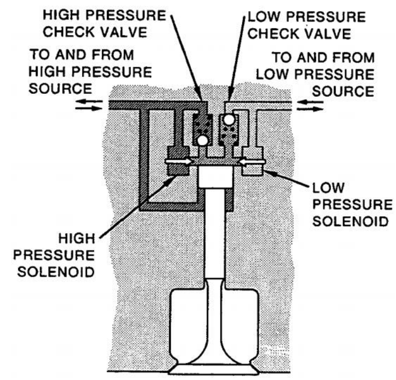

Figure 3 – Hydraulic pendulum used in the Ford camless engine [2]

Turner et al [11] (Lotus Engineering), and Turner, Kenchington (Lotus Engineering) and Stretch (Eaton Corporation) [12] have developed an electrohydraulic valve train (AVT), shown in Figure 4.

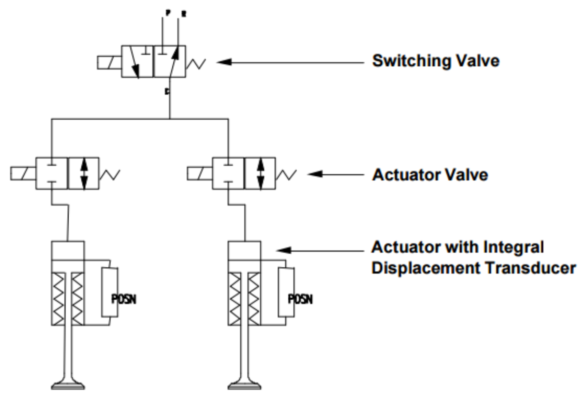

Figure 4 – Hydraulic circuit of the Lotus / Eaton camless valve train [11,12]

The authors claimed the system to be capable of

- Throttleless operation,
- Controlled auto-ignition (HCCI),
- Fast start,
- Variable firing order,
- Differential cylinder loading, and
- Ultimately air hybridization.

The targets for the system development included: up to 15 mm lift, unrestricted event timing and phasing, maximum engine speed 7000 r/min with a timing repeatability of 1 degCA.

The analytical results presented from a Simulink model indicated power consumption for the AVT system would be of the order of 5.5 kW at 6000 r/min for a European 2L 16V engine operating at full valve lift.

Turner and Babbitt et al [13] at Sturman Industries developed a two-stage electro-hydraulic valve actuation system, shown in Figure 5. They had previously developed a single stage electro-hydraulic actuator and found limitations in the ability to control seating velocity over a wide range of engine speed and oil temperature / viscosity. The two-stage system was developed as a means to provide variable geometry that could respond to inputs such as engine speed and oil temperature to better control seating velocity in a wide speed range gasoline engine.

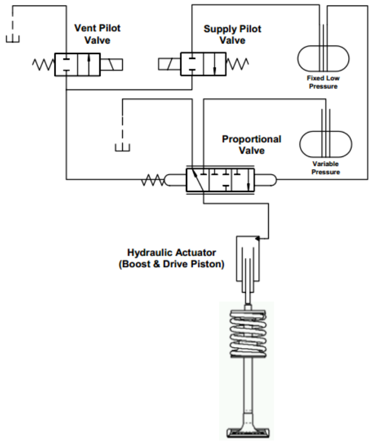

Figure 5 – Two-stage electro-hydraulic valve actuation system [13]

Cam Driven Variable Valve Systems

Many mechanical and hydro-mechanical variable valve systems have been proposed. The most complete reviews by Dresner and Barkan [3,4], while dated, provide a useful classification system. They concluded that the 800 or so patents and technical papers they reviewed could be reduced to only 15 basic concepts. Here we will briefly review a selection of those systems in current production engines.

UniAir [5,6]

Figure 6 shows the design of the UniAir VVA system for the intake side of a 4 cylinder engine [5,6]. Figure 7 shows the General layout of the UniAir variable valve train system.

Figure 6 – Design of the UniAir VVT system for the intake side of a four cylinder engine [5,6]

UniAir is a cam driven electro-hydraulic variable valve timing and lift system, first developed by Fiat as the *MultiAir* system, later licensed by Schaeffler. The major components of the UniAir system are shown in Figure 7, and comprise a fast acting solenoid valve, a high pressure reservoir and a piston pump operated by a roller finger follower and output cylinders that act directly on the engine valves. The available valve lift profiles are shown in Figure 8. Essentially this device is an electro-hydraulic lost motion device, which allows a range of lift profiles within the envelope of the cam lobe. The primary application of the UniAir system is to reduce or eliminate throttling as a means of load control in gasoline engines.

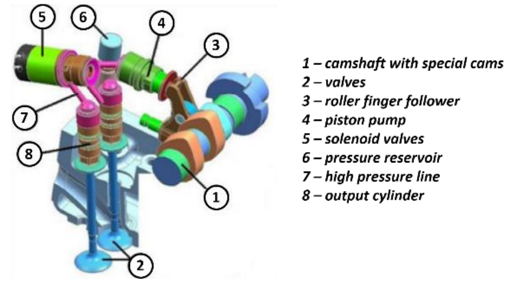

Figure 7 – General layout of UniAir variable valve train system [5,6]

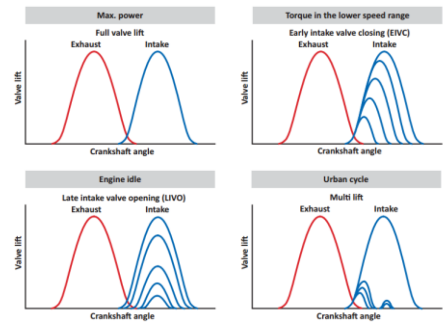

Figure 8 – UniAir valve lift modes for the intake side [5,6]

Audi Valvelift System (AVS) [7,8]

The Audi AVS system is essentially a cam profile switching system (CPS) combined with a hydraulic cam phasing system [7,8]. The cam profiles are designed to better optimize performance across the engine speed range. When combined with cam phasing AVS provides a means to significantly reduce throttling and to provide optimum torque and power. In some applications AVS is applied to both intake and exhaust valves. Figure 9 shows two operating modes of the Audi AVS system.

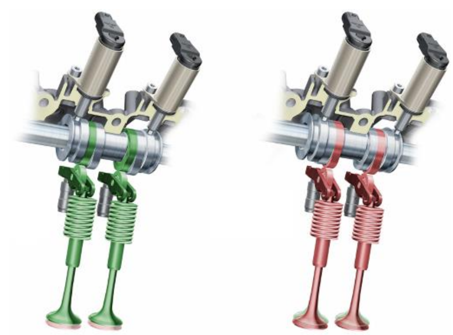

Figure 9 – Audi valvelift system for the 2.0 L TFSI engine [7,8]. At left is the 'small cam' for low engine speeds, and at right is the 'large cam' for high engine speeds.

BMW Valvetronic [9]

Figure 10 shows the BMW Valvetronic variable valve timing system, which provides variable intake lift and duration. It is typically combined with cam phasing. Valvetronic controls valve lift by modulating the fulcrum of a roller finger follower by means of an eccentric shaft, thereby delaying the engagement of the cam lobe with the follower and allowing control of valve lift. Valvetronic is combined with a cam phasing system to allow control of both lift and timing [9]. The wide range of valve lift control (0.18mm to 9.9mm) combined with cam phasing allows throttleless operation over a significant portion of the operating range of the engine.

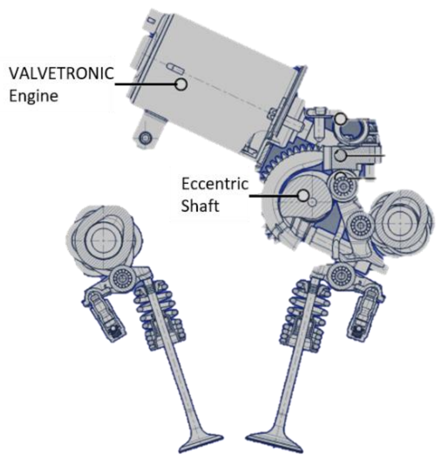

Figure 10 – BMW Valvetronic variable valve timing system [9], which provides variable intake lift and duration. It is typically combined with cam phasing.

The DigitalAir Concept

Figure 1 shows a cross-section of the DigitalAir system developed by JP SCOPE Inc. DigitalAir replaces the poppet valve with horizontally acting slotted valves, which provide the full flow area with a very short stroke compared with a poppet valve [21-25]. This approach provides advantages over electro-hydraulic camless poppet valve systems, such as:

- Short stroke from fully closed to fully open
- Direct solenoid actuation
- Fast opening – 2 ms
- High integrated effective flow area … $\int_{\text{Open}}^{\text{Close}} A.\text{Cd}$
- No possibility of valve-to-piston contact
- Low opening forces
- Low power consumption
- Control of seating velocity

As with any new technology, this approach does raise some obvious questions, which are being addressed and will be discussed further in this paper and in the companion paper [1], for example:

- Volume of slots added to the combustion chamber
- Sealing between cylinder and ports
- Surface area, heat transfer and cooling of the valve bridges
- Potential for quenching and hydrocarbons in the slots
- Wear on the mating faces due to debris

In this paper analytical and experimental results are presented from a variety of engines, Table 1.

Table 1 – Engine platforms included in this study.

Displacement	Bore x Stroke mm	Study	Comments
0.5 L	92 x 75	Performance, Durability	Single Cylinder Test platform
2.0 L	87.5 x 83.1	Cycle Simulation	Four Cylinder GTDI with VVA
0.57 L	87.5 x 94	CFD	Single Cylinder 4V & DigitalAir

Breathing Capability

Figure 11 shows the geometry of the intake and exhaust slots controlled by the movement of the valves placed above the combustion deck of the cylinder head as shown in Figure 1. Although the fully open geometric area of the DigitalAir valves is less than the typical 4V intake fully open geometric area, this is offset by a combination of rapid opening (2 ms), favorable discharge coefficient (>0.90) and the freedom to optimize event timing. As will be shown in a later section, this geometry is capable of delivering volumetric efficiency comparable with a conventional 4V poppet head.

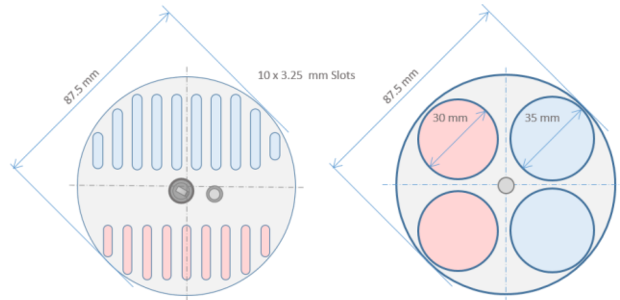

Figure 11 – Plan view of the prototype head based on the Mazda 2.3L engine. At left is the DigitalAir head showing the intake and exhaust slots. At right is the production head showing intake and exhaust poppet valves.

Figure 12 shows a sample of the flow bench fixtures used to quantify the discharge coefficient of intake and exhaust slots under steady flow conditions. A range of slot geometric parameters were studied as illustrated in Figure 13.

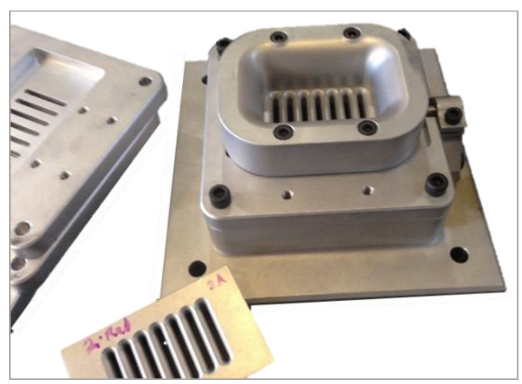

Figure 12 – Flow bench fixtures used to assess slot discharge coefficients.

The CFD study considered 38 different slot geometries at 3 pressure drops: 70 mbar, 172 mbar and 6.9 bar, with inlet pressures of 1.07 bar, 1.17 bar and 7.9 bar respectively. The fluid was assumed to be air at 300K. Figure 14 shows a sample of the CFD study where slot geometries were being screened for performance. The CFD and flow bench results correlated quite well with discharge coefficients exceeding 0.90.

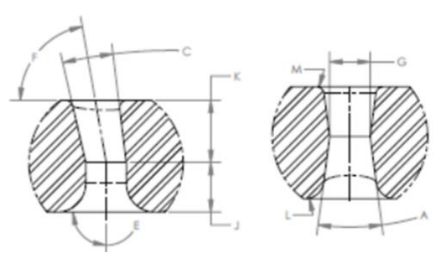

Figure 13 – Slot parameters studied on a flow bench and with CFD analysis.

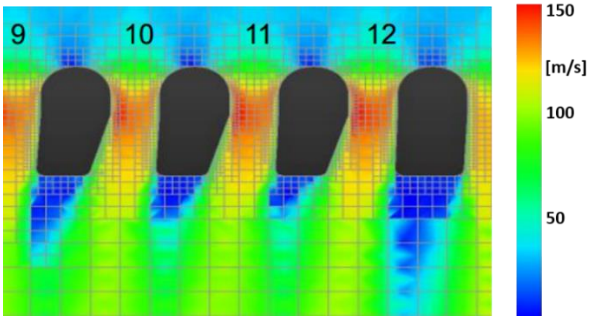

Figure 14 – A large number of slot geometry variations were studied using CFD to narrow the options for intake and exhaust valve design. The figure shows velocity contours for four variations – with 70 mbar pressure drop, 1070 mbar inlet pressure.

Range of Operation

The DigitalAir system allows intake and exhaust valves to be commanded open and closed almost without limitation. With 2 ms opening and closing ramps, valves can be opened fully in less than 20 degCA at 1000 r/min, and 84 degCA at 7000 r/min, see Figure 15.

Figure 16 illustrates the freedom to command valve events provided by the DigitalAir system. Since the valves can be opened in 2 ms, the valve profile resembles a square wave at low engine speeds and a trapezoid at higher engine speeds.

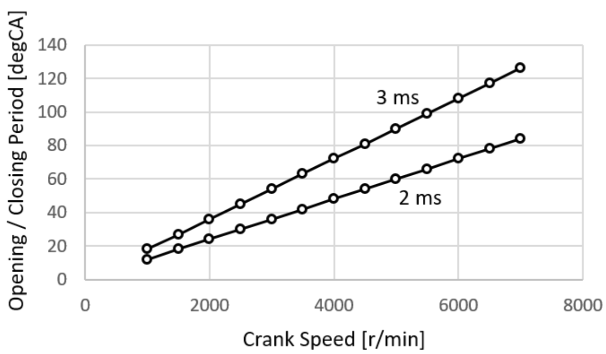

Figure 15 – Valve opening / closing period versus engine speed for 2 ms and 3 ms actuation rates. The system has been designed for, and achieves, 2 ms opening and closing periods.

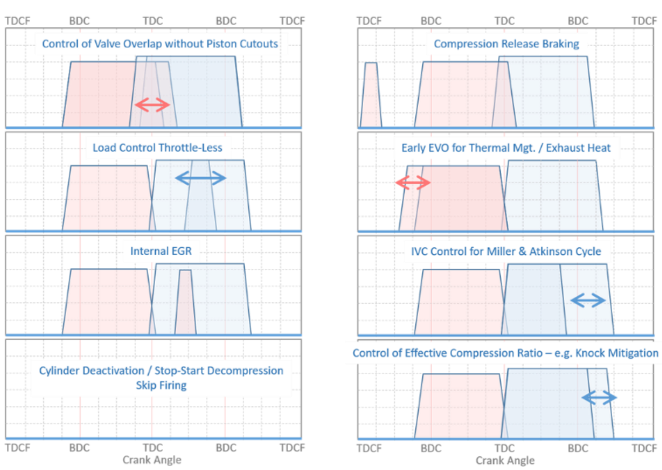

Figure 16 – Wide range of valve events provided by the DigitalAir system. Opening / closing period 2 ms at 1500 r/min.

Because the valves do not project into the cylinder, there is no opportunity for piston-to-valve contact. This allows unrestricted control of valve overlap, such that trapped residuals can be readily managed.

Combustion Chamber

The design and analysis results presented here are based on the Mazda 2.3L production engine. Table 2 shows the impact of the combined intake and exhaust slot volume on the total combustion chamber volume at TDC. Through attention to detail, the slot volume has been minimized at 11.7% of the volume at TDC with a compression ratio of 12:1, while maximizing the geometric flow area of the intake and exhaust valves.

Table 2 – Combustion chamber parameters with exhaust and intake slots, based on the 2.3L GDI base engine geometry.

Displacement	2.3	L
Bore	87.5	mm
Stroke	94.0	mm
Swept Volume	2.261	L
CR	**12:1**	
Clearance Volume	51.4	cm3
Intake + Exhaust Slot Volume	6.01	cm3
Slot Volume - %	**11.7%**	
Intake Valve Geometric Flow Area	806	mm2
Exhaust Valve Geometric Flow Area	543	mm2
Intake / Exhaust Area Ratio	**1.48**	
Ratio - Chamber Surface Area at TDC with Slots / Baseline 4V Head	**1.28**	

The intake to exhaust geometric area ratio selected for this study was 1.48 based on cycle simulation analysis. The valve slots add surface area to the combustion chamber. The combustion chamber surface area at TDC was 28% greater than the baseline Mazda 2.3L engine.

The design of the DigitalAir cylinder head provides a land through the center of the head as shown in Figure 11. In the initial design, this area is used to locate the centered spark plug and injector. This design offers a degree of freedom to locate those components and a cylinder pressure sensor for optimum combustion and control.

Charge Motion and Turbulence

Detailed CFD analysis results will be shared in a later section. The design of the valves, having vertical slots covering one half of the cylinder, naturally creates tumble. Furthermore, if the slots in the intake valve are angled to the vertical by say 20 degrees, a significant swirl component is induced, allowing charge motion to be tuned to some extent.

Power Consumption

Power consumption is discussed in detail in the companion paper [1]. The actuator has been designed to minimize power consumption by adopting a center-biased approach, wherein the neutral position of the valve is the mid positon between fully open and fully closed, see Figure 17. Opening and closing coils are used to pull the valve to its end stop upon start-up, and to hold the valve at either end of its travel during operation. Valve opening and closing energy is provided by springs located either side of the valve. This approach allows energy expended to open a valve to be recovered and used to close the valve. Additional force may be applied by the solenoids as needed to overcome friction or other losses in the system.

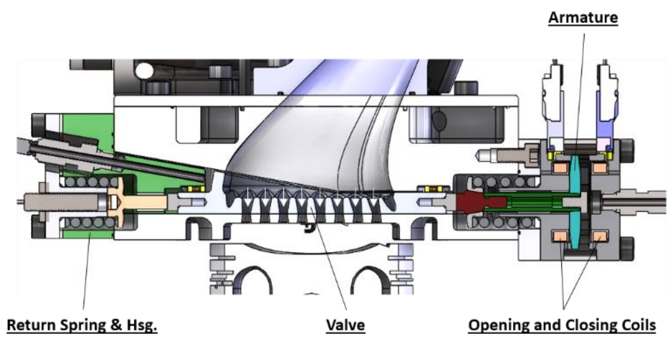

Figure 17 – Cross section showing the center-biased solenoid actuation approach. Energy stored in the opening and closing springs is recovered with each cycle for efficient operation.

Analytical Studies

Cycle Simulation

Cycle simulation was used to study the benefits of the DigitalAir system on a 4 cylinder turbocharged GDI engine of 2.0L displacement, Table 1. In the following discussion the baseline case is represented by simulation results from the 2.0L 4 cylinder turbocharged GDI engine with 4 poppet valves per cylinder. The baseline case assumed variable cam phasing.

In order to simulate DigitalAir slotted valves in the cycle simulation model, the cam-driven poppet valves were replaced by valves with opening and closing rates, flow area and discharge coefficients as a function of crank angle to represent the effective flow area of the DigitalAir valves.

The baseline model was calibrated against test cell data and has been validated under steady state conditions. The model includes closed-loop control of the wastegate to achieve commanded pre-throttle boost pressure and closed-loop control of the throttle to achieve commanded brake mean effective pressure (BMEP).

The exhaust and inlet valve timings for the baseline 4V engine are based on variable cam phasing with phase angles provided in lookup up tables as a function of engine speed and load.

Although the baseline cycle simulation model was extensively calibrated to experimental data from the base 2.0L 4V engine, application to the DigitalAir version of the engine lacked such rigorous validation. The following are areas where assumptions had to be made which should be noted:

- Combustion model uses fixed burn rate profiles unchanged between the baseline and DigitalAir simulations,
- Heat transfer in the cylinder head was not adjusted, and
- The Chen-Flynn friction mean effective pressure (FMEP) model was applied to baseline and DigitalAir simulations.

Future work will address these assumptions as experimental data from the latest generation of DigitalAir hardware becomes available.

Full Load Results

Figure 18 shows the results of the cycle simulation at wide open throttle (WOT) with valve timing optimized for BMEP across the engine speed range. The figure shows the results of two different optimizations of the DigitalAir system compared with the baseline engine full load curve. In this analysis, wastegate control is active and the same boost pressure calibration is used for all simulations, which results in higher turbine bypassing in the cases with higher scavenge flow, in order to maintain the same boost pressure.

Figure 18 shows the DigitalAir system optimized for maximum BMEP at matched or reduced residual gas faction (RGF). This gives the highest BMEP performance at low engine speed, but comes at the cost of high brake specific fuel consumption (BSFC) as shown in frame 2. The RGF plot (frame 3) shows how this parameter is minimized, particularly at low speeds, which is the main factor behind the improved BMEP.

The trapping ratio is the ratio of air trapped in the cylinder at IVC to total air flow through the intake port, where lower values indicate more scavenging. Frame 4 illustrates the high level of scavenging taking place at low speeds. This scavenge flow is responsible for clearing the cylinder and reducing RGF, and it also improves turbocharger performance.

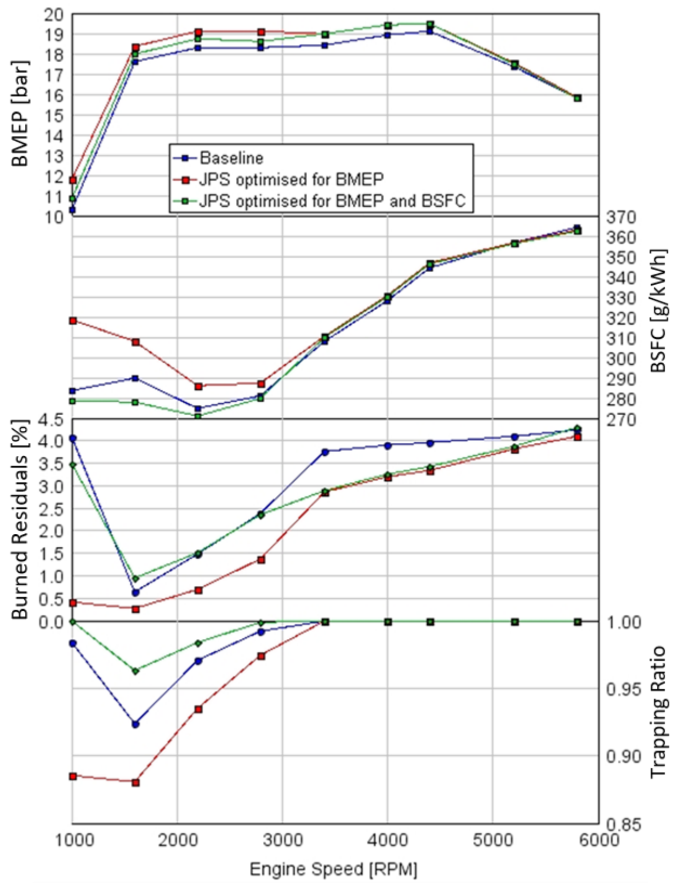

Figure 18 – Full load performance of the DigitalAir system compared with the baseline 4V engine.

Figure 18 also shows the case where timing is optimized for BMEP and BSFC, with trapping ratio limited to the same or higher than the baseline case. The result is a closely matched RGF at low speed but with less scavenging, and an improvement in both BMEP and BSFC compared to the baseline, though slightly lower BMEP than in the higher scavenge case.

For engines where the exhaust gas lambda is less critical, for example diesel engines, the scavenge behavior can be exploited to deliver substantial improvements to low speed BMEP. The DigitalAir system enables this by allowing independent control of the overlap timing whilst maintaining optimal IVC and EVO.

Figure 19 shows the full load performance for a range of valve opening and closing times. It should be noted that in this study the 1ms and 3ms cases have not been fully optimized, whilst the 2ms example is optimized for BMEP at matched or improved RGF and trapping ratio. Even with this partially optimized set of results, it is clear that the differences in performance with valve travel time are small for this engine. The limitations of the 3ms valve travel time only become apparent at the highest engine speed, where optimization was unable to match the BMEP, thus boost pressure would have to be raised here to compensate.

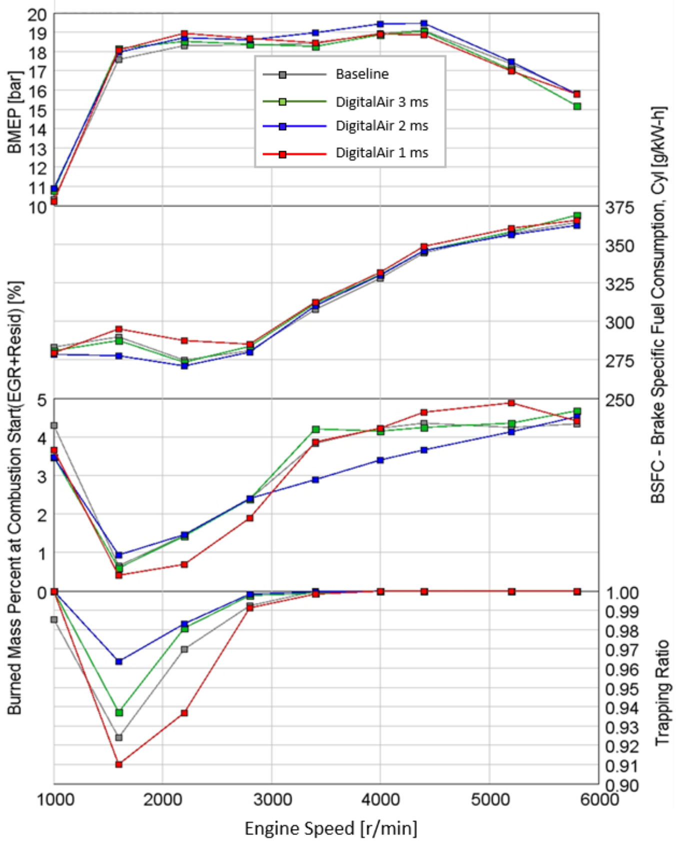

Figure 19 – Effects of valve opening and closing time at full load.

With the trend for engine downspeeding, downsizing and turbocharging, design for higher speed operation may be of reduced importance, thus 3ms valve travel time may be sufficient for applications other than high-performance engines. However, another consequence of this trend is that turbocharging will cause higher airflows at the same engine speeds, thus the limiting factor of the design may be the effective flow area rather than travel speed.

During the exhaust stroke the blowdown pulse is stronger for all DigitalAir cases, due to the rapid exhaust valve opening event. If an appropriate pulse energy recovery scheme is used this could potentially result in improved turbocharger efficiency and performance. For the single entry turbine scheme employed on this engine, the larger pulses will result in stronger interactions between each cylinder during the exhaust stroke, with potentially negative impact on RGF and scavenge.

Part Load Throttleless Limit

The limit of throttleless operation, where early intake valve closing (IVC) is used to control the engine down to part load, is shown in Figure 20.

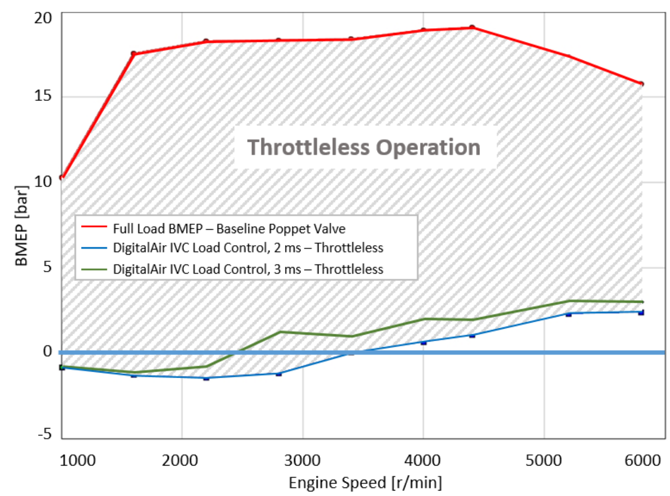

Figure 20 – Throttleless operation limit using early IVC.

Figure 20 also shows the lower limit for IVC control of engine load for valve opening and closing times of 2 ms and 3 ms. For this engine, throttleless operation with early IVC is effective across the majority of the range of engine operation. The range is reduced slightly for a slower valve travel, though this can be covered using throttle control. Importantly, this illustrates that throttleless operation using IVC control is effective down to zero load at low engine speeds, which means it can potentially be used to improve fuel consumption at the low speed and low load operating points critical for drive cycle performance.

Part load IVC control

Throttleless operation using early IVC is a proven technique for improving fuel consumption for gasoline engines. To quantify the effect using DigitalAir on this engine, IVC sweeps were performed at fixed engine speed and load with throttle control active. Figure 21 and Figure 22 summarize the predicted change in BSFC with IVC timing relative to the baseline case, at 2 bar and 5 bar BMEP.

As IVC is advanced BSFC is reduced substantially, compared with the baseline, due to the throttle opening and decreasing pumping losses. The same phenomenon occurs if IVC is retarded, as the intake mixture is being pushed back into the intake manifold, though this is less preferable from a control point of view as in-cylinder air/fuel ratio (AFR) estimation and control becomes more complex.

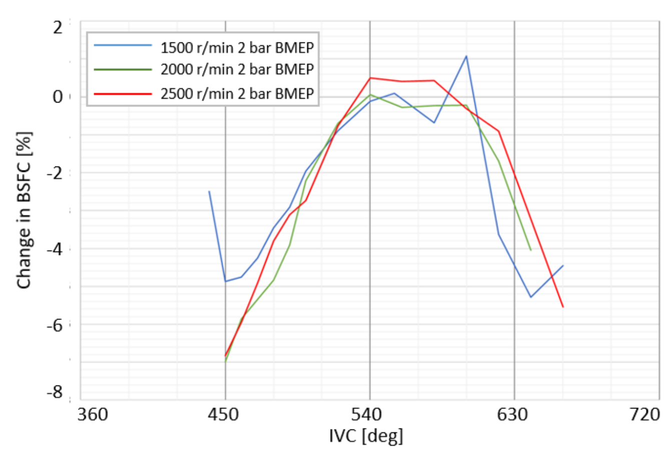

Figure 21 - Change in BSFC compared to baseline for 1500, 2000, 2500 r/min 2 bar BMEP IVC sweep

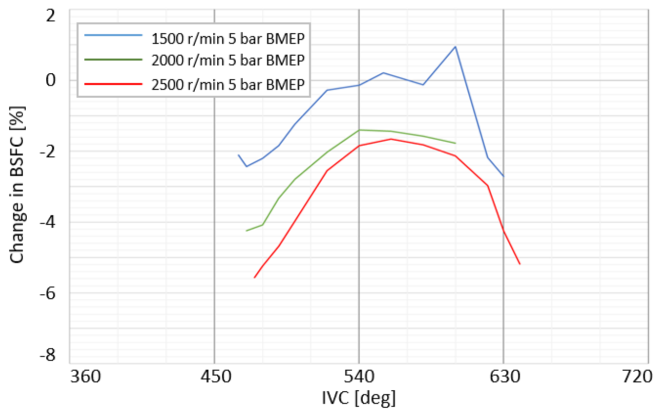

Figure 22 – Change in BSFC compared to baseline for 1500, 2000, 2500 r/min 5 bar BMEP IVC sweep

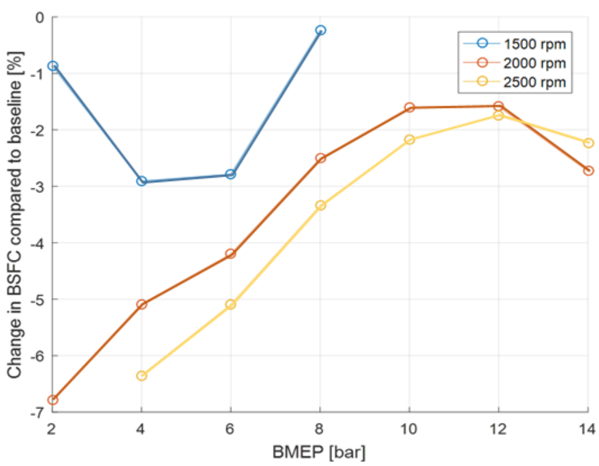

Figure 23 – Summary of BSFC performance of IVC load control relative to throttle load control at 1500-2500 r/min BMEP sweep.

Figure 23 summarizes the change in BSFC for a series of BMEP sweeps performed at fixed engine speed with IVC load control active. The vertical axis represents the change in BSFC compared to the baseline case at the same BMEP. This clearly shows that at low engine speeds, the improvement in BSFC is significant with throttleless operation using IVC control. In general, the improvement increases as load decreases, however in the 1500 r/min case the benefit appears to reduce at lowest loads. It is thought that this is due to the overlap timing not being fully optimized, as the RGF is notably lower at this point with the DigitalAir valves.

EVO Timing for Part Load Exhaust Energy Management

A variety of EVO timing scenarios were explored as a means to increase exhaust gas temperature for faster catalyst light off, and improved turbocharger response. Throttle and wastegate control loops were active during these simulations, thereby maintaining the target BMEP and boost pressure as the EVO timing changed.

Figure 24 summarizes the results of an EVO sweep at 1500 r/min and 2 bar BMEP for the DigitalAir system. The minimum BSFC corresponds to a very late EVO of 175 deg ATDC, which illustrates the advantages of rapid valve opening – in this case allowing an extended expansion stroke. At 1500 r/min, the valve opens fully in less than 20 degCA.

Figure 24 shows the engine indicated efficiency, and the percentage of the total fuel energy available in the exhaust after indicated power and in-cylinder heat transfer are deducted. As may be expected, advancing EVO increases the first law energy to the exhaust and decreases engine efficiency, though there is a plateau where no great change occurs as EVO is advanced, between approximately 140 and 160 degCA ATDC.

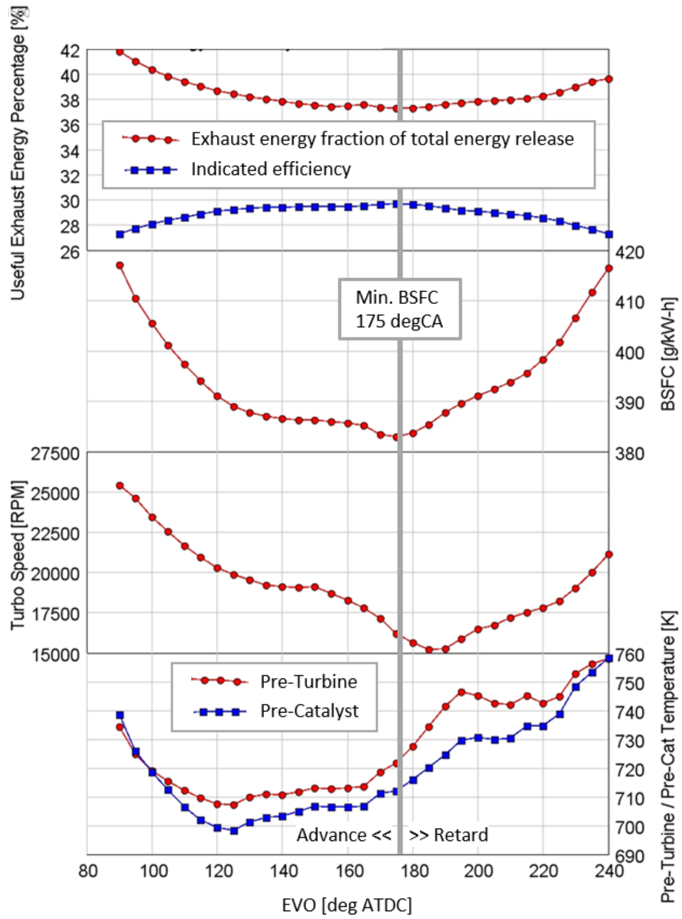

Figure 24 – DigitalAir EVO Sweep at 1500 r/min, 2 bar BMEP.

In Figure 24 it can be seen that advancing EVO from the optimal BSFC point initially causes a decrease in temperature until very early EVO settings. Whereas retarding EVO increases exhaust gas temperature significantly more than advancing it. Earlier EVO timing causes the in-cylinder gas temperature to fall earlier in the power stroke, and the result is a lower gas temperature through the exhaust stroke. At the same time RGF reduces and mass air flow rate increases - as a result of reduced RGF, and because the reduced expansion work will require the throttle to open to maintain BMEP. The two effects combine to create the fall then rise in exhaust gas temperature seen with early EVO.

In order to assess the potential improvement in turbocharger response with early EVO, a transient load step simulation was performed, the results of which are shown in Figure 25. The figure shows the BMEP response during a tip-in from 5 bar BMEP to WOT at fixed 1500 r/min engine speed. The lower curve represents fixed optimal EVO timing, the upper curve shows EVO initially set to early timing, then switched to optimal timing at the start of the tip-in. An improvement in response can be seen due to the improved initial condition for the turbocharger, from a driveability perspective this will be noticeable as better throttle response. The discontinuities in the response curves are likely due to some parameter transition between lookup tables in the model based on engine speed and load; since the model has not been optimized for transient testing therefore this is to be expected.

The drawback to this method is that by using an early EVO calibration at the part-load point, the BSFC will suffer. This can however be compensated for using early IVC to recover the BSFC at part load by operating throttleless as has been demonstrated earlier.

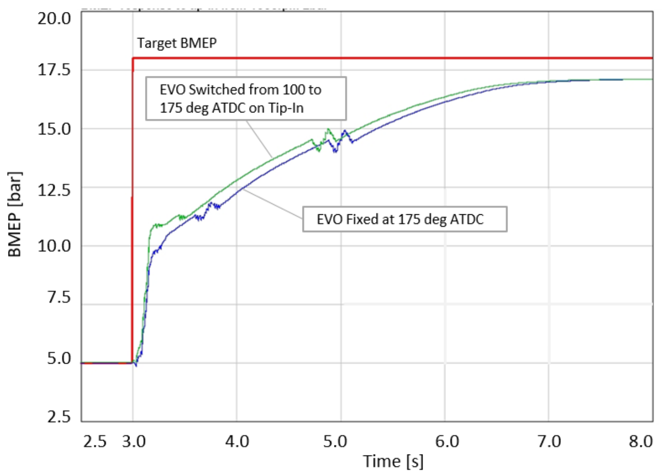

Figure 25 – Tip-in response from 1500 r/min 5 bar BMEP with different EVO strategies.

CFD Model and Findings

CFD simulations were performed to better understand the flow and turbulence characteristics of the DigitalAir system under motored conditions.

The software used for the CFD modeling allows the computational mesh to be redefined for every time step and to adapt in regions of high velocity, temperature or species gradient which commonly occur when valves open and close, when fuel is injected or during combustion. Re-meshing at every iteration also allows robust and accurate modeling of moving surfaces – such as valves and pistons.

The base grid used for this model had 4 mm cubic cells, however, the entire cylinder was embedded so that the refined grid consisted of 1 mm cubic cells. The volume around the spark plug was refined by 5 levels so that the grid had 0.125 mm cubic cells. The volume around the intake and exhaust valves were embedded so that the base grid around the valve surfaces used 1 mm cubic cells. In addition, another 3 levels of adaptive grid refinement were added to the model to refine the grid in any areas of large velocity or temperature gradients. The number of cells in the model varied from time step to time step based on the automatic refinement, and the maximum number of cells during an engine cycle was limited to 300,000.

The turbulence model used was a Reynolds Average Navier-Stokes (RANS) k-epsilon model. Grid embedding and automatic grid refinement were used to resolve the boundary layers near the walls and a law of the wall formulation was used to calculate the heat transfer coefficients and shear at the walls.

The Engine Model

The engine platform used for the CFD study was the Mazda 2.3 L turbocharged GDI engine, Table 1, with standard production poppet valves and DigitalAir valves. The extent of the CFD model is shown in Figure 26 for the DigitalAir case and the operating condition is provided in Table 3.

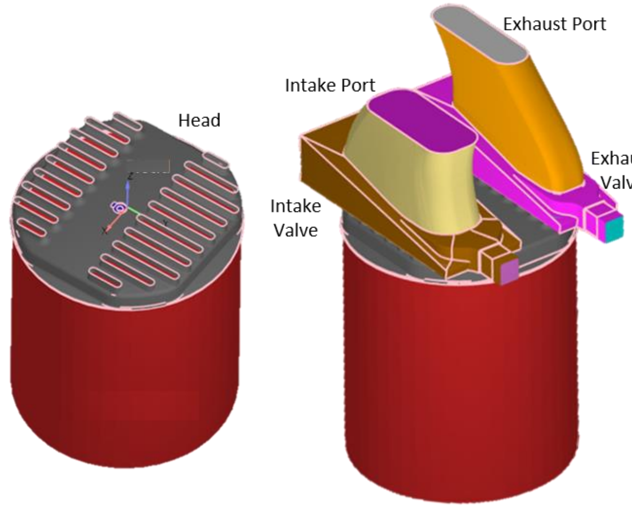

Figure 26 – CFD model geometry of DigitalAir FVVA applied to the Mazda 2.3 L engine.

Table 3 – Engine operating c/onditions for the CFD study.

Displacement	2.3L
Engine Speed	3000 r/min
Intake Manifold Pressure	1.9 bar
Intake Manifold Temperature	330 K
Exhaust Manifold Pressure	1.9 bar
Intake Valve Timing	IVO / 356 IVC / 563
Exhaust Valve Timing	EVO / 132 EVC / 363
Valve Opening / Closing Time	2 ms
Intake Valve Geometric Flow Area	806 mm2 (DigitalAir)
Exhaust Valve Geometric Flow Area	543 mm2 (DigitalAir)

Motored Engine CFD Results

Simulation of the flow for the motoring case with DigitalAir valves was performed to quantify and visualize the resulting charge motion in the cylinder. Figure 27 shows the streamlines in the engine at 85 deg ATDC of the intake stroke. The geometry of the slotted valves directs the intake flow vertically into the cylinder below the intake valve. This flow sets up a strong tumble motion due to the offset vertical nature of the incoming charge.

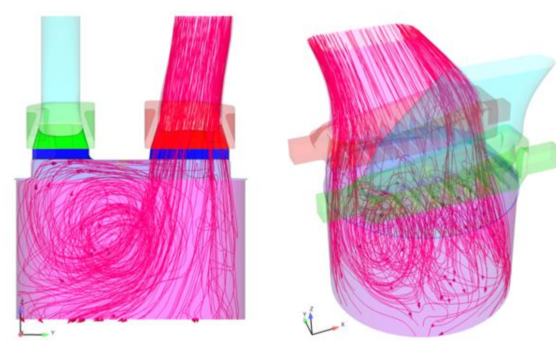

Figure 27. Streamlines during intake stroke for the DigitalAir motoring case at 3000 r/min, 85 deg ATDC.

The predicted charge motion tumble ratio and turbulence kinetic energy (TKE) are shown as a function of crank angle in Figure 28. The maximum tumble ratio is 2.5 which occurs near the peak piston velocity. Turbulence is generated in the cylinder during the intake stroke and is generally maintained through the compression stroke, peaking at 30 deg BTDC. As the tumble starts to break down at the end of the compression stroke, it generates turbulence in the cylinder, Figure 28.

With vertical slots in the intake valve, the swirl ratio is effectively zero. However, as discussed in Part 1 of this paper [1], the swirl in the cylinder may be tuned by tilting the slots to the vertical, for example by 20 degrees.

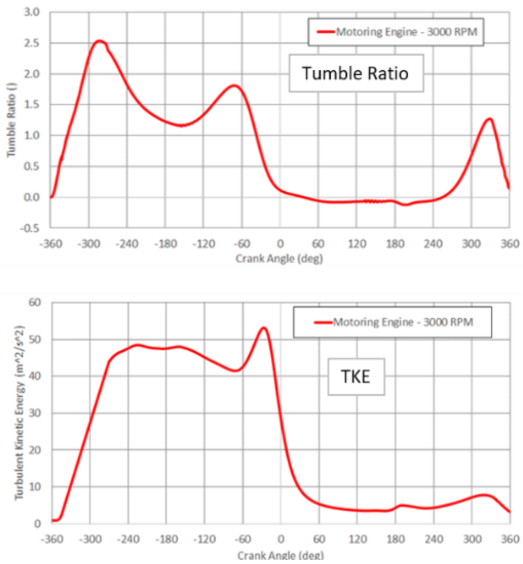

Figure 28. Predicted charge motion and turbulence during motoring with DigitalAir valves

Experimental Studies

Experimental setup

A single cylinder Polaris 0.5L EFI engine was fitted with a DigitalAir cylinder head as shown in Figure 29. This generation of head is fitted with spring-biased actuators, which use a solenoid to open and a return spring to close each valve. This design has limitations in full lift accuracy and power consumption. The latest design described in the companion paper [1] and shown in Figure 17, uses a center-biased actuator giving tighter control and energy recovery for low power consumption.

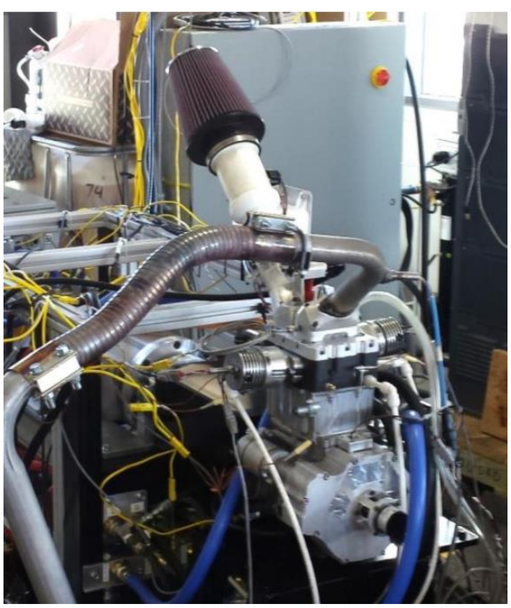

Figure 29 - View of DigitalAir installed on a Polaris EFI engine.

The test setup uses a Parker electric motoring / absorbing dynamometer, closed-loop control of oil, fuel and coolant temperatures, and National Instruments data acquisition. Data is post-processed using Matlab-Simulink scripts.

Test results

Intake Valve Load Control

As described earlier, fully variable valve timing may be used in many different ways to improve gasoline engine fuel consumption, performance and emissions. Here we will present initial results from a study where intake valve closure was advanced at constant engine speed and load from normal throttled timing to early unthrottled timing, as shown in Figure 30. It is well understood that load may be controlled through timing of the intake valve closing event. The advantage of this approach over intake throttling is that pumping loop work may be reduced significantly, since intake manifold vacuum is reduced or eliminated. Table 4 shows the key parameters for the intake valve closure study.

Table 4 – Key data for the intake valve closure study.

Engine Displacement	0.5L
Engine Speed	1230 r/min
Intake Condition	Naturally Aspirated
Intake Manifold Temperature	20 degC
Torque	14 Ft-lb (19 Nm)
BMEP	4.8 bar
Spark Timing	22 deg BTDC
IVO	1.6 deg BTDC
EVO	158 deg ATDC
EVC	7.8 deg ATDC

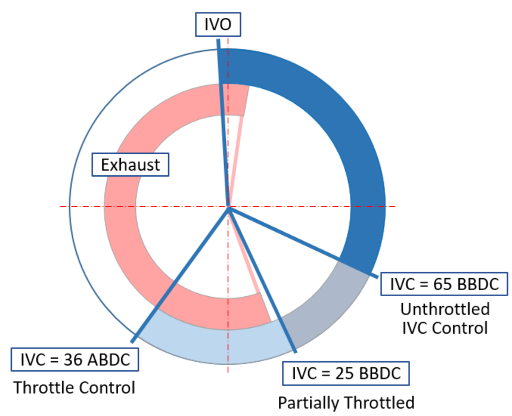

Figure 30. Valve timing events for the IVC sweep study. Exhaust events: EVO 16 deg BBDC, EVC 6 deg ATDC.

Figure 31 shows three valve motion profiles corresponding to load control with normal throttled intake valve closure (36 deg ABDC), partially throttled intake valve closure (25 deg BBDC) and advanced intake valve closure with WOT (65 deg BBDC). The valve events are highly repeatable in the timing of opening and closing, within less than 1 deg CA standard deviation. This first generation spring-biased solenoid actuation leads to bounce and excessive variation when fully open. This should improve significantly with the next generation center-biased solenoid actuator under development [1]. However, it appears that the accurate and consistent event timing is the more important characteristic at this operating condition.

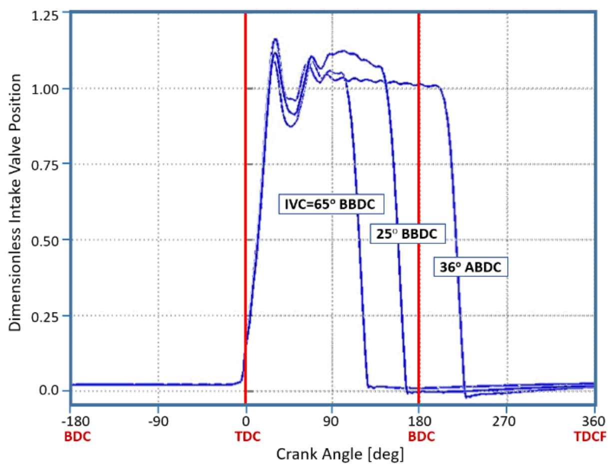

Figure 31. Intake valve motion curves for the IVC swing study. Spring-biased solenoid actuator, showing very sharp and repeatable opening and closing events, with oscillations at the fully open position.

Figure 32 shows log PV plots for each of the intake valve timings, all other event timings were held constant, along with speed and load. The figure illustrates the improvement in pumping loop work as IVC timing is advanced from throttled to unthrottled / WOT.

Figure 33 shows the change in brake specific fuel consumption with the timing of intake valve closure. At this speed and load point the data in Figure 33 indicates a fuel consumption improvement of 3.5% by switching to intake valve from intake throttling for load control. This is broadly in line with the cycle simulation results shown in Figure 23.

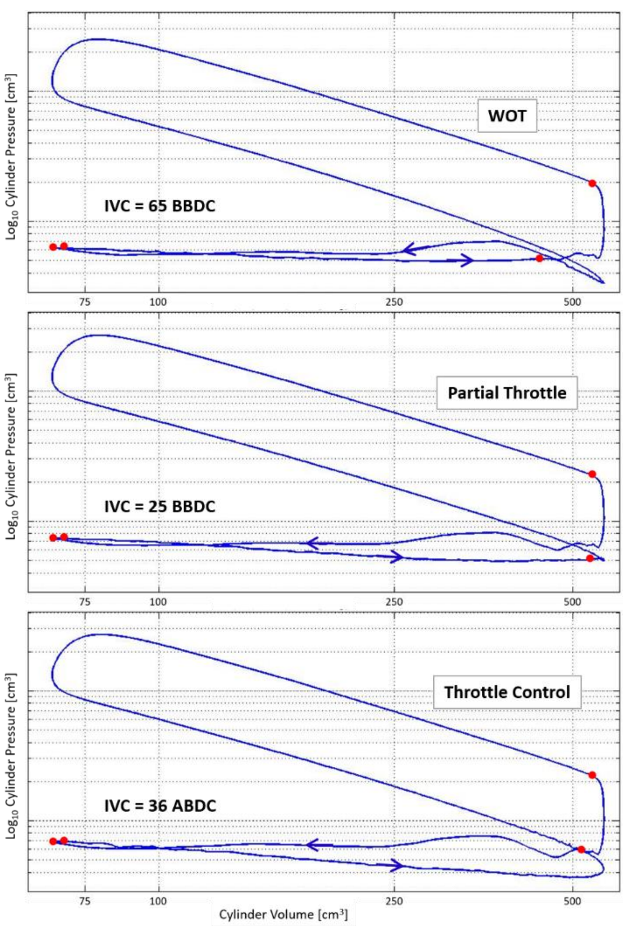

Figure 32. Log P / Log V diagrams showing change in the pumping loop with advancing intake valve closure. BMEP= 4.8 bar at 1230 r/min.

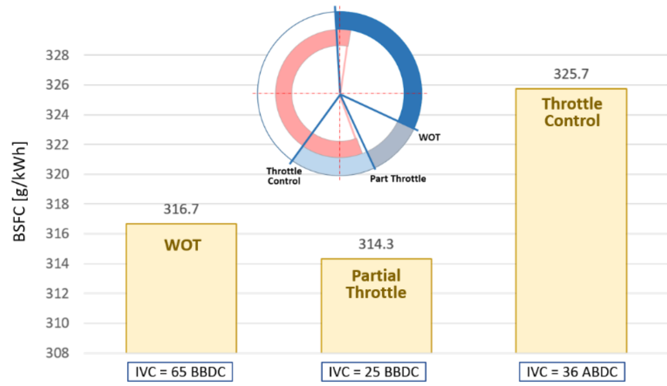

Figure 33. Measured BSFC results with intake valve closing event timing, BMEP= 4.8 bar at 1230 r/min.

Conclusions

This paper and its companion paper [1] have described an innovative camless FVVA system. The DigitalAir system dispenses with camshaft and poppet valves in order to achieve fully variable valve actuation, with direct electrical actuation and low energy consumption.

The DigitalAir FVVA system brings significant design and operational degrees of freedom for intake and exhaust valves:

- Freedom to command valve opening and closing events within an engine revolution,
- Freedom to command valve durations as short as 40 degCA at 1500 r/min,
- Freedom to command valve overlap without risk of contact with the piston,
- Freedom to command multiple events within an engine cycle,
- Freedom to deactivate valves as required,
- Flexibility to tune in-cylinder charge motion between tumble and swirl through the geometry of the valve slots, and
- Greater freedom to locate the spark plug and GDI injector.

These new degrees of freedom in turn lead to performance opportunities, for example:

- Improved fuel consumption by eliminating the throttle for load control over a wide speed and load range, including idle,
- Increased BMEP across the engine speed range, through optimized valve events,
- Variable effective compression and expansion ratios for Miller or Atkinson cycles,
- Lower idle speeds and idle stability – by better control of residuals,
- Variable displacement through cylinder deactivation and skip firing,
- Ability to create and control internal EGR, and
- Improved thermal management.

Other potential benefits include improved packaging, particularly reduced engine height, and ease of service – since valves may be removed complete with actuator without removing the cylinder head.

As with any new technology, this approach does raise some obvious questions, which are discussed further in the companion paper [1], for example:

- Sealing between cylinder and ports
- Wear on the mating faces due to debris

- Surface area, heat transfer and cooling of the valve bridges
- Volume and surface area of slots added to the combustion chamber
- Potential for quenching and hydrocarbons in the slots

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Definitions/Abbreviations

AFR	Air Fuel Ratio
AVS	Audi Valvelift System
ATDC / BTDC	After TDC / Before TDC
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
CFD	Computational Fluid Dynamics
CPS	Cam Profile Switching System
ECU	Electronic Control Unit
ECV	Electro-Hydraulic Camless Valvetrain
EVO / EVC	Exhaust Valve Opening / Closing
FVVA	Fully Variable Valve Actuation
FVVT	Fully Variable Valve Train
IVO / IVC	Intake Valve Opening / Closing
PID	Proportional, Integral, Derivative Controller
VVA	Variable Valve Actuation
VCP	Variable Cam Phasing
RGF	Residual Gas Fraction
TDC	Top Dead Center
TKE	Turbulence Kinetic Energy
WOT	Wide Open Throttle