

DigitalAir™ Camless FVVA System – Part 1, Valve Train Design, Capability and Performance

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ABSTRACT

This paper provides an overview of the analysis and design of the DigitalAir™ camless valve train including the architecture and design of the valve and head; the details of the electric valve actuator, and the flow characteristics of the valves and resulting charge motion in a motoring engine. This valve train is a completely new approach to fully variable valve actuation (FVVA), which allows almost unlimited continuously variable control of intake and exhaust valve timing and duration without the use of a camshaft. This valve train replaces conventional poppet valves with horizontally actuated valves located above the combustion deck. As the valves move, they open and close a number of slots connecting the cylinder with the intake and exhaust ports. The valve stroke necessary 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 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 results from the analytical models used to predict the performance of the valve train are presented and compared with experimental data (when available). JP SCOPE Inc. has been running engines with this valve train for several years and has successfully completed preliminary performance and durability tests. Part 2 of this paper [1] will present analytical and experimental data which confirms that the proposed FVVA system can meet the basic performance requirements of modern GTDI engines.

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INTRODUCTION

The cost/benefits tradeoffs of variable valve control are well understood [2] and in most cases proven to be positive as evidenced by the wide scale adoption of production variable valve train (VVT) systems. There are several production VVT systems that provide a wide variety of control including numerous permutations of variable timing, variable duration, variable lift and valve deactivation [3,4,5,6,7,8]. With a few exceptions, these systems are all cam based thus limiting the system flexibility. Researchers have done extensive analysis and testing evaluating the benefits that can be realized if the valve train has infinitely flexible valve timing. Those camless systems, often referred to as Fully Variable Valve Actuation (FVVA) systems, have been shown to offer significantly more advantage than cam based systems and enable operating modes that to date cannot be achieved without them. [2,9, 10, 11, 12, 13, 14, 15, 16]. Nevertheless, these camless systems have yet to make any significant penetration into the production marketplace despite being researched and prototyped for quite some time. The reasons for this include:

- High production cost associated with FVVA systems,
- High parasitic power consumption,

- Integration complexity
- Failsafe and durability concerns, especially in engines that have valve trains where piston and valve train interference is possible.

These issues are discussed in more detail below.

Cost - Most camless systems are expected to be relatively expensive to produce even in high volume. Hydraulic based systems require parts with very close tolerances and require a hydraulic power supply. Electrically (magnetically) actuated systems typically require expensive power electronics and additional electrical power generation capacity. Due to power and force limitations of the actuators, electric based systems have largely been limited to smaller SI engines that have short strokes and lower in-cylinder pressures than diesel engines.

Parasitic losses - Rapidly accelerating and decelerating a valve to achieve the required opening and closing speeds requires significant power. Fortunately, the inherent design of the conventional cam based systems with spring return recovers and reuses the vast majority of energy required to do this. Furthermore, these cam based systems have been in production for a long time and have become highly

optimized making the parasitic losses quite low, even at higher engine speeds. While hydraulic valve actuation systems can be designed to minimize power consumption, it is very difficult to come close to cam based power consumption at elevated engine speeds. At lower speeds, such as those on the FTP city cycle comparable and lower valve train power consumption is achievable by implementing valve deactivation when possible. However, the system has to be designed for the worst case scenario (highest engine speed and largest in-cylinder pressure) so hydraulic units need to be large and thus are typically not very efficient at low engine speeds and low flow output.

Integration Complexity - The addition of the hydraulic actuators, lines and hydraulic supply units to the engine and head leads to complexity in design and manufacturing and ultimately to higher manufacturing costs and the possibility of more manufacturing and durability issues. Integrating electrically actuated systems adds additional challenges and cost due to the additional electrical generation required, higher voltages and general integration required.

Fail-safe Operation and Durability - No mechanical linkage between the crank position and the timing of the valve motion means piston/valve collisions are possible. Piston valve collisions can destroy an engine. The mechanical impact of parts during the seating of valves and end of the actuator motion can lead to NVH and durability issues with the valve train.

This camless valve train addresses many of the short-comings of the poppet-based FVVA system discussed above. With simple electric valve actuation, this system has the potential for low power consumption, low cost and relatively simple integration from a controls and packaging standpoint. This system has:

- Simple design with lower required tolerances results in lower cost
- Lower power consumption realized using short stroke and energy recovery via spring centered design. Also takes less power to open the valve against higher in cylinder pressures.
- Lower integration costs due to no hydraulic supplies needed and lower power electrical system
- Failsafe operation resulting from non-interference design

Additional advantages include:

- Lower head height
- Unique charge motion characteristics.

This valve train technology could also be implemented only for the intake valves, where the advantages of both variable valve timing and the unique charge motion are most evident. The full potential of this valve train still unknown as engine data from the latest design iteration is just now becoming available.

System Components and Operation Details

Figure 1 shows a cross-section through the cylinder and cylinder head of the DigitalAir™ camless FFVA system. As the valve moves

horizontally from a closed to open position, slots in the valve are aligned with slots in the head, connecting the cylinder to the intake or exhaust port. The actuation mechanisms are discussed in the actuator design section, and are comprised of springs and an actuator which includes two opposing electromagnetic coils. The timing of actuation is virtually unlimited with respect to piston position, and the duration of valve events is limited on the short end only by actuation speed.

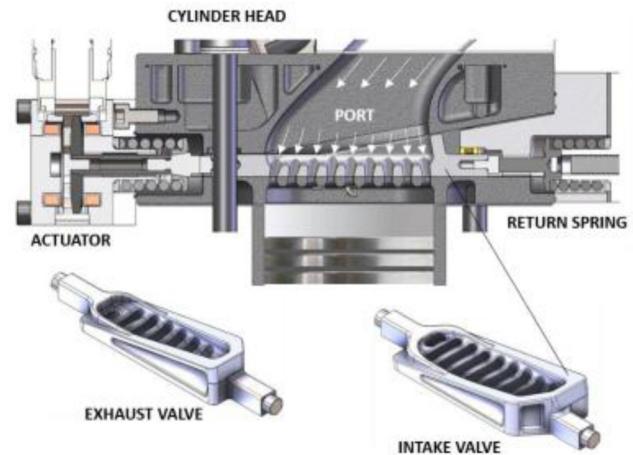


Figure 1. View of DigitalAir™ Hardware

The valve train hardware has been integrated into multiple single cylinder test engines including one that uses one cylinder of a Mazda 2.3 L engine. A test engine, based on a single cylinder ATV engine with the camless valve train, is shown in Figure 2



Figure 2. View of DigitalAir™ installed on engine

The remaining sections of the paper outline the multi-physics analysis approach used to design the latest revision of this valve train. Rigorous examination of the system design and operation identified the concerns listed below.

Mechanical concerns included:

- sealing of the valve,
- distortion in the head and valves,
- durability of the valves and head due to both thermal cycling and impact loading.

Actuator design concerns included:

- The actuator force needed to unseat and accelerate the valve
- The actuator force and valve speed needed to achieve repeatable motion, reliable sealing force and acceptable valve travel times.
- The current, voltage and resulting system power consumption of the actuator.

Engine flow concerns included:

- Volumetric efficiency
- Charge motion in the cylinder
- Flow and turbulence characteristics near TDC
- Heat transfer to the head.

DESIGN METHODOLOGY

A systems engineering process incorporating analysis-led design and extensive modeling was used for the design of the camless system. A summary of the analysis and modeling software used is provided below with detailed description following later in the paper.

Brief summary of analysis tools used:

1. Simulink – dynamic models of the valves and actuator
2. GT Power – engine performance
3. Converge – valve flow characteristics and motoring engine simulations
4. ANSYS Multiphysics – mechanical and thermal stress/distortion, thermal modeling
5. FEMM – magnetic modeling of actuator solenoid
6. Solidworks – CAD and FEA

This camless valve train provides some distinct advantages over other camless valve trains, as outlined in the Introduction. However, the unique geometry and operation of this system, which uses slotted valves that open perpendicular to the cylinder, raised some concerns that needed to be evaluated analytically and through testing. Some of the notable concerns were:

1. Valve sealing
2. Valve and head durability
3. Valve actuator power consumption

4. Engine breathing
5. Effects on combustion
 - a. Charge motion effects
 - b. Combustion chamber design limitations

The three primary analysis areas presented below attempted to address these areas of concern.

1. Mechanical design of the valve and engine head, primarily focused on valve sealing and controlling stresses in the head and valves
2. Dynamic and magnetic modeling of the actuator design, primarily focused on meeting performance metrics for valve motion and low power consumption.
3. CFD and engine performance modeling, primarily focused on improving the engine breathing (volumetric efficiency) and quantifying the charge motion in the cylinder.

Mechanical Design and Analysis

Simulation Objectives

Simulations were used throughout the design process to improve product performance, cost, and development time. Although physical testing has and will continue to serve a critical role, using models in parallel to analyze system characteristics enables a more targeted and informed path to final hardware. Analyses were used to quantify characteristics of the existing proof-of-concept system, inform the new design, and ultimately evaluate that design against established metrics.

Design Criteria

The long-term goal for the FVVA system is to meet competitive performance metrics within a GTDI platform. However, intermediate goals were set for the current prototype iteration, which is meant to serve as a development platform for advancing the technology towards the long-term goals. Given that, the development team, in conjunction with JP SCOPE and industry consultants, determined that a 2-year design life is reasonable for a prototype iteration. Within the design life, the system should be capable of operating under naturally aspirated conditions without significant valve leakage or component failures. It is also the intention that the system can operate at limited turbo-charged conditions, that may not necessarily include the entire power range for extended periods. Sealing effectiveness was evaluated using simulated contact gap, which is the distance between sealing surfaces of the valves and the head. Short of performing extensive CFD on the deformed shapes, contact gap is the best available indication, albeit relative, of leakage. The previous proof-of-concept demonstrated adequate valve sealing and simulations of that system indicated that contact gap on sealing surfaces ranged from 0-0.003 inches. Thus, the requirements for the current design is that simulated contact gap shall not exceed 0.003 inches.

Component failures were delineated by the onset of cracks, typically a concern on surfaces exposed to cyclic loading. The following life-cycle was considered when evaluating against the two-year life:

- 5 cold assemblies per year
- 1465 thermal cycles per year (warm up to peak temperature 4 times/day)
- 220 million full engine cycles per year (8hrs/day @ 2500RPM)

Risk of crack initiation was evaluated using Miner's cumulative damage model, in combination with the Augmented Modified Goodman criterion.

Analysis Methodology

The analytical process for this iteration began with modeling the existing proof-of-concept system to generate baseline results and correlate simulation parameters against measured test data. The baseline results were then used to inform initial concept designs of the valves and head, which were then similarly analyzed and iterated upon until the acceptance criteria were met. The modeling workflows for both the proof-of-concept and new design are shown in Figure 3.

As shown in Figure 3, five primary loads were applied to the head assembly: cold assembly, thermal strain, valve closing, combustion pressure, and valve opening (additional details provided in Table 1, Figure 4, Figure 5, and Figure 6). Along with available proof-of-concept test data, the loads were informed with several supplemental modeling streams, such as CFD combustion simulations and MATLAB dynamic models. Compacted Graphite Iron (CGI) was selected as the material for both the cylinder head and the valves; see Table 2 for properties. Once the loads were established, contact behavior, material properties, and mesh controls were defined, and the simulation was executed. The primary outputs of the thermomechanical simulation were stress and deflection, which were both used to evaluate the system against the failure criteria.

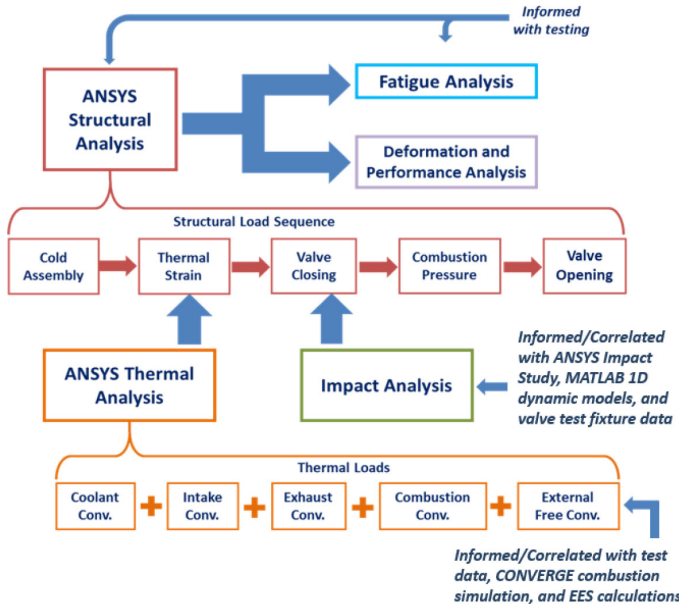


Figure 3. Mechanical modeling and simulation workflows

Table 1. Mechanical load summary

Load Description	Value		Load Step					
	Lbf, psi	N, Bar	1	2	3	4	5	6
Primary Head Bolt Preload	21,204	94,318	ON	LOCK	LOCK	LOCK	LOCK	OFF
Secondary Head Bolt Preload	3,996	17,774	ON	LOCK	LOCK	LOCK	LOCK	OFF
Thermal Loads	Mapped Globally		OFF	ON	ON	ON	ON	ON
Exhaust Valve Close	27, 48	120, 214	OFF	OFF	ON	ON	ON	OFF
Intake Valve Close	27, 53	120, 236	OFF	OFF	OFF	ON	ON	OFF
Combustion Pressure	1,900	131	OFF	OFF	OFF	OFF	ON	OFF
Intake Valve Open	663	2,950	OFF	OFF	OFF	OFF	OFF	ON
Exhaust Valve Open	445	1,980	OFF	OFF	OFF	OFF	OFF	ON

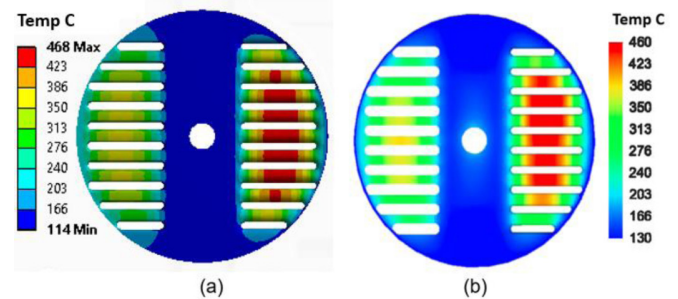


Figure 4. Comparison of temperature profiles between ANSYS (a) and CONVERGE (b) in the combustion chamber roof. Note that although color scales are clipped at different max and min values, the intermediate color bars are identical. Profiles between the two simulations are very similar, indicating good correlation between models. Although the correlation is strong, there are inherent limitations in reproducing the dynamic CONVERGE simulation statically in ANSYS, which explains the relatively minor temperature discrepancies.

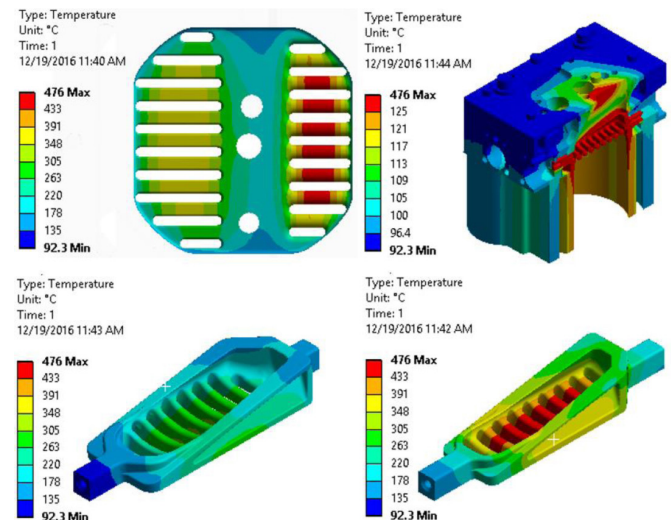


Figure 5. Final simulated temperatures. Color scales are similar except for top-right image which was modified for clarity.

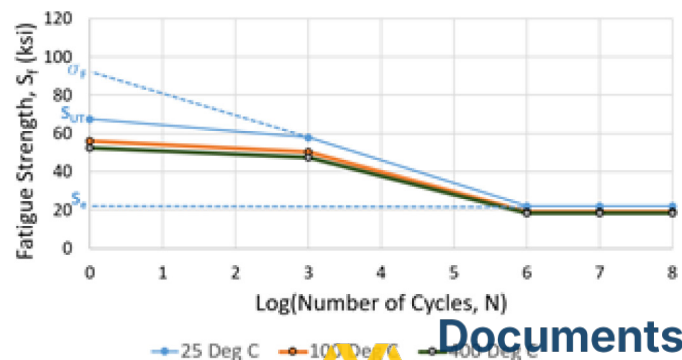


Figure 6. Generated SN curve for valves and head

Table 2. Material properties for SinterCast Compacted Graphite Iron GJV 450 [18]

Property	Units	25°C	100°C	400°C
Ultimate Tensile Strength	Mpa	450-525	425-500	350-425
0.2% Tensile Yield Strength	MPa	315-365	290-340	265-315
0.2% Compressive Yield Strength	MPa	410-440	-	350-390
Elastic Modulus	Gpa	145-155	140-150	135-145
Elongation	%	1.0-2.5	1.0-2.0	0.5-1.5
Endurance Ratio	-	0.45-0.50	-	-
Thermal Conductivity	W/m-k	38	37	36
Thermal Expansion Coefficient	μm/m-K	11.0	-	12.5
Poisson's Ratio	-	0.26	0.26	0.27
Fatigue Strength Reduction Factor	-	1.2-1.6	-	-
Density	g/cc	7.0-7.2	-	-
Brinell Hardness	BHN	207-255	-	-

Analysis Results and Conclusions

Sealing effectiveness, as measured by contact gap, in the exhaust valve meets the acceptance criteria under naturally aspirated conditions throughout the entire power range, and under turbo-charged peak BMEP conditions: on sealing surfaces contact gap is typically about 0.001" and ranges from 0-0.0023". As shown in Figure 7 and Figure 8, contact gap is best in the center of valve, and is worst at the front of the valve. Sealing effectiveness on intake valve narrowly achieves the acceptance criteria under the same conditions: on sealing surfaces contact gap is typically about 0.0015" and ranges from 0-0.003".

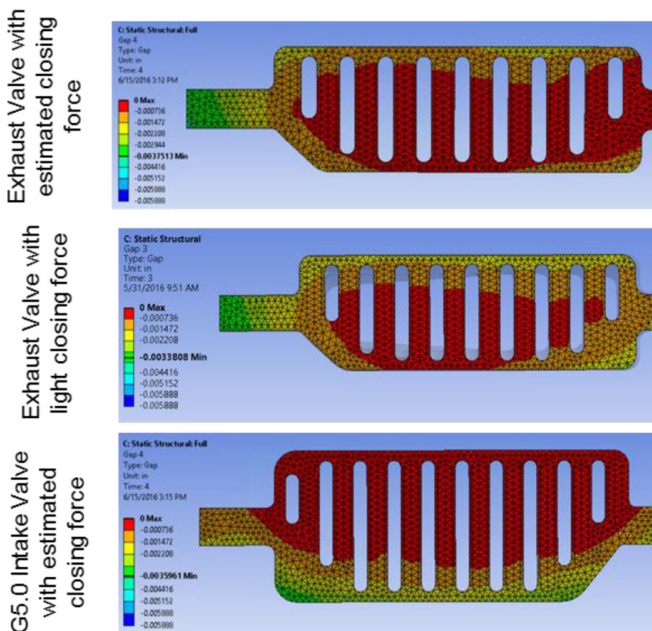


Figure 7. Contact gap on valve sealing surfaces after valves are closed but prior to combustion pressure

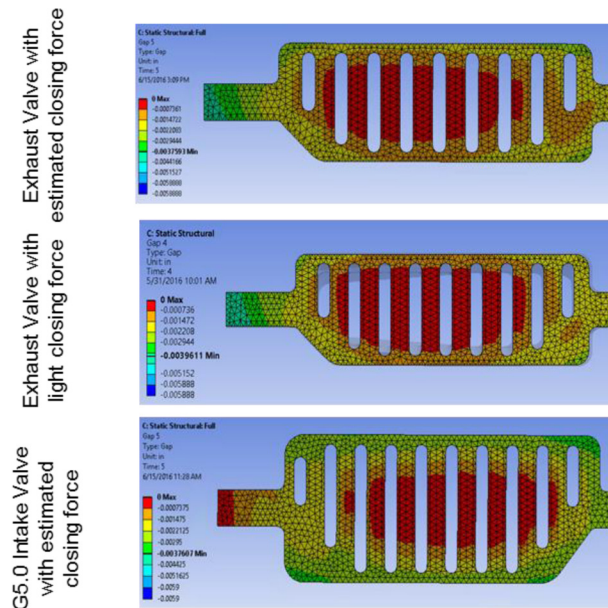


Figure 8. Contact gap on valve sealing surfaces after combustion pressure is applied

Both the intake and exhaust valve are predicted to exceed the minimum 2-year design life under naturally aspirated conditions throughout the entire power range, and under turbo-charged peak BMEP conditions. The cylinder head exhibited compressive stress concentrations beyond the maximum for a 2-year life under certain naturally aspirated conditions (peak power). However, stress concentrations of even greater magnitude, in similar locations, were observed in the proof-of-concept baseline simulation; given that proof-of-concept has proven robust in the areas of concern, it was decided that the compressive stress concentrations were an acceptable risk as they do not appear to contribute to the onset of cracks; this phenomenon is corroborated by the highly bimodular nature of the material used, which was simplified in the fatigue calculations. Under turbo-charged peak BMEP conditions, the cylinder head life appears borderline. On the whole, the results from the thermomechanical fatigue analysis were sufficiently assuring for the design to be manufactured for the proposed testing.

Actuator Design and Analysis

Initial prototypes of the camless slotted valve were constructed with an architecture that was coil-opened and spring-closed. In this configuration, the valve is opened by coil force, and closed by spring force when the coil is turned off. This design was useful as a proof-of-concept to demonstrate characteristics such as sealing, valve wear, and travel time, but suffered from fast, uncontrollable seating velocities and higher than desired power consumption. The next generation of actuator uses a centered design to address both of these concerns. As shown below in Figure 10, in the centered design the valve body is centered between two springs, which provide the majority of the motive force to open and close. The two actuators provide holding force at either end of the valve stroke and also can be used to assist valve motion as needed against variable and unknown factors, such as friction. While many centered designs are possible, the chosen configuration places both actuators on the same side of an armature that is not attached to the valve. This provides both