

Scuderi Split Cycle Fast Acting Valvetrain: Architecture and Development

2011-01-0404

Published
04/12/2011

Riccardo Meldolesi, George Bailey, Clive Lacy, Ian Gilbert, Jean-Pierre Pirault and
Anthony Perkins
Southwest Research Institute

Copyright © 2011 SAE International

doi:10.4271/2011-01-0404

ABSTRACT

The Scuderi internal combustion engine is characterized by a split cycle that divides the four strokes of a conventional combustion cycle over two paired cylinders, one intake/compression cylinder and one power/exhaust cylinder, connected by a crossover port. This split cycle also has an additional high pressure “crossover” gas transfer phase versus the conventional 4-stroke cycle, during which the charge air is moved from the first to the second cylinder. The intake/compression, power/exhaust and crossover events are repeated every revolution, i.e. over two cycles, with a small phase angle between the two cylinders. The separate cylinders enable opportunities for improved combustion and the possibility for pneumatic hybridization of the engine. This paper describes the technical challenges posed by the actuation of the crossover valves in the Scuderi Split Cycle research engine. Fast actuation of these valves is a key requirement for successful combustion and for hybrid operation in the Scuderi and perhaps other pneumatic hybrid engines where the mass of air entering the cylinder is controlled by the operation of poppet valves. To successfully meet these demands design modeling, dynamic simulations of the valve opening event, and structural analysis on the components were performed concurrently. Experimental rig work was undertaken to verify the valvetrain performance and durability. The result is a valvetrain with an unusual architecture that meets the speed of actuation requirements for successful engine operation. The structural limitations in terms of mass, strength and stiffness of the major components are fully understood, as well as the dynamic behavior of the whole system. A set of verified analytical models was developed which will be used to tailor valvetrain design and performance as various Scuderi engine configurations are developed in the future.

INTRODUCTION

The Scuderi Split Cycle (SSC) engine concept was first conceived by Carmelo Scuderi, who filed the first patent for the concept on July 20, 2001 which was granted in 2003 [1]. The SSC concept divides the four strokes of a conventional combustion cycle over two paired cylinders. The first cylinder, referred to as the “compressor” (Figure 1), provides intake and compression strokes. The second cylinder, referred to as the “expander”, provides power and exhaust strokes. For the SSC prototype engine, the compressor top dead center (TDC) is phased 20° later than the expander TDC (TDC-e). Throughout this paper, crank angles are referred to in terms of degrees after TDC of the expander cylinder (ATDC-e). The SSC architecture can utilize any combination of normally aspirated, turbocharged, spark or compression ignition combustion strategies.

The two cylinders are connected by a “crossover port”, through which the high pressure gas is transferred from the compressor cylinder to the expander cylinder between the compression and power strokes. This crossover (Xovr) event must quickly transfer all the working fluid in typically 35° crankangle (CA) duration around TDC-e, essentially at compressor pressure, before combustion can commence. Outwardly opening Xovr valves (opening away from the pistons) are employed to enable this high mass flow while avoiding contact between the valves and pistons. The combination of valve opening period and engine rated speed results in a valve event duration of 1.25 milliseconds (ms). This, together with the required valve peak lift means very challenging valve acceleration values.

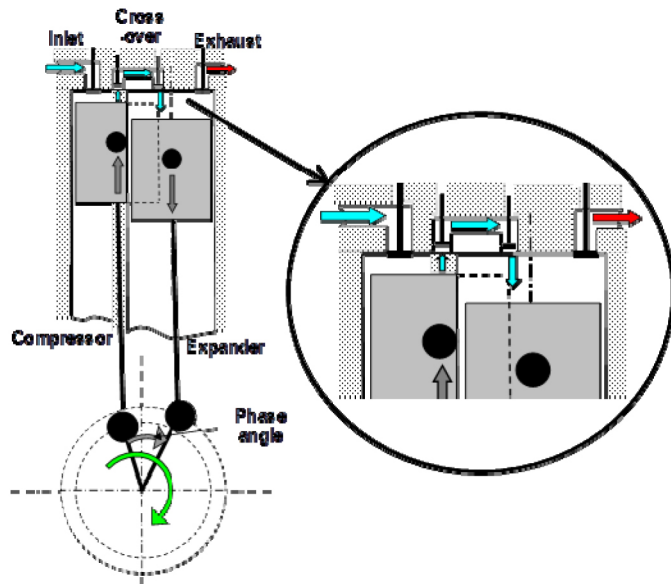


Figure 1. Principal Elements of the Scuderi Split Cycle Engine

Figure 2 compares the Scuderi Xovr valve events to the intake valve lift events of a standard production gasoline engine and of a Formula 1 engine. The comparison is done by calculating the pseudo acceleration by dividing the peak lift by the square of the event duration. To meet these challenging targets required a careful, CAE led design of the valvetrain and its components, together with experimental validation of the analysis tools and durability testing of the parts. This paper describes the architecture of the Xovr valvetrain, its analytical development, and experimental verification. For a detailed description of the design and operation of the SSC see [2].

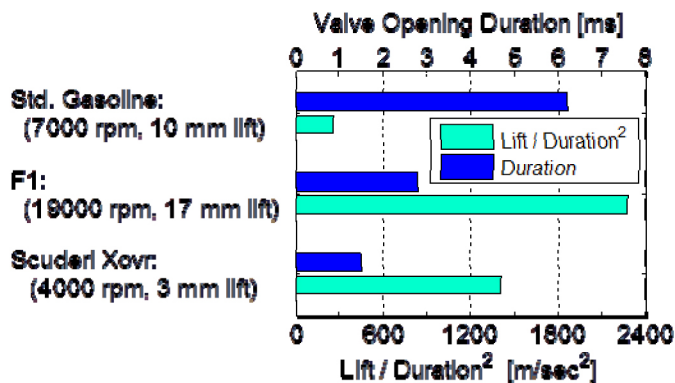


Figure 2. Typical valve operating parameters for various engine types at rated speed

DEFINITION OF CROSSOVER VALVETRAIN ARCHITECTURE

The performance modeling of the engine described in [2] defined the breathing requirements for the engine, as well as the size of the Xovr valves for both compressor (XovrC valves) and expander (XovrE valves), their opening timing, opening duration, and peak lift. Table 1 summarizes the valvetrain functional requirements.

Table 1. Summary Specification of the Scuderi Xovr Valvetrains

	XovrC	XovrE
Peak lift	2.8-3.0 mm	2.6-2.8 mm
Opening Duration	35-40° CA	30-35° CA
Rated speed	4000 RPM	
Spring type	Pneumatic	
Valve material	Titanium	
Valve adaptor material	High strength alloy steel, DLC coated	
Valve Collets material	Titanium	
Rocker material	High strength alloy steel, DLC coated	
Gas spring piston material	Aluminum, DLC coated	
Camshaft material	100Cr6	

From the start the driving approach to this valvetrain design was to deliver a highly reliable system which would enable the team to concentrate on combustion development work on the research engine [2] without concerns for the valvetrain durability.

As discussed in [2] the need for a high geometrical compression ratio (>30) in both the compressor and expander cylinders was recognized early on. This, together with the need to time the valve opening events near TDC, drove the architectural choice to open the valve outwardly, i.e. away from the piston. This therefore avoids altogether the risk of valve to piston contact as well as removing the need to machine pockets on the piston top surface, or to recess the valve seats in the cylinder head. As a result the valve head sealing surface faces inward toward the piston and the corresponding surface of the valve seat insert faces outward, as shown in Figure 3.

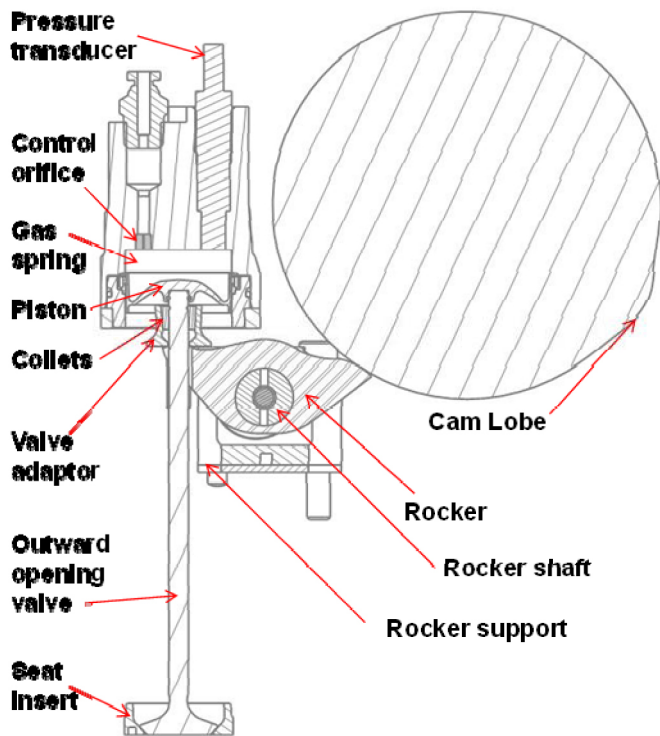


Figure 3. Xovr valve actuation elements for initial research engine

The selection of outwardly opening valves led to the adoption of a center pivoted rocker with a forked end to accommodate the valve stem and engage a valve adaptor from underneath. The valve adaptor is connected to the valve via standard design collets.

The high valve acceleration rates created by the target lift and periods drove the requirement for minimum moving masses together with a stiff kinematic chain capable of transmitting the motion from the cam profile to the valve and to overcome the loads generated by the high differential pressures on the valve head with the minimum amount of vibrations. One obvious choice was to use titanium valves. The adoption of a pneumatic spring in place of a more conventional high strength steel coiled spring achieved both the objective of very low mass and low vibrations. The pneumatic spring piston sits directly on top of the valve stem, resulting in a direct load path giving the most lightweight solution. Pneumatic springs also offer the ability to modulate the spring pressure as a function of engine speed allowing lower pressures to be used at low engine speeds, resulting in reduced friction [3,4,5].

The requirement for high reliability drove the choice, for the camshaft, rocker, and valve adaptor, of adopting high strength steel (UTS > 2000 MPa), capable of withstanding high contact stresses in operation. The rocker and valve adaptor were also coated with a diamond-like coating (DLC)

layer to reduce the dry friction coefficient and improve their tribological behavior in boundary elasto-hydro-dynamic (EHD) lubrication experienced during operation.

Another striking feature of this valvetrain in its current design is the large size of the cam lobe. The base circle radius of the cam lobe is in excess of 40 mm. The combination of the 2-stroke operation, short valve opening periods, high accelerations and low rocker ratio (for high stiffness), conspire to the need for the cam in this valvetrain to be larger than conventional automotive engines. This is mainly because of high contact stresses at the cam nose due to small radius of curvature, as well as small radii in the concave part of the cam profile which impose manufacturing challenges and high values of pressure angles. The “reliability first” approach also led to the adoption of a cam of diameter larger than necessary, which results in substantial safety margins on lubrication and on the contact stresses the materials are capable of withstanding. This left open the possibility in the future to design profiles with more aggressive accelerations without the need for major design changes.

The rocker shaft is mounted eccentrically in its supports to enable lash adjustment. Oil ports are provided to the rocker shaft to supply pressurized oil to the rocker bore.

PRESSURE INDUCED LOADS ON THE CROSSOVER VALVE

Figure 4 shows that the XovrC valve has to operate with a pressure differential of 16 bar resisting valve opening, and then with 5 bar assisting the valve closing. The pressure resisting valve opening arises because there is residual pressure in the Xovr port from the last revolution, and the valve must begin to open before the compressor cylinder reaches Xovr pressure or it will not be sufficiently open when mass transfer needs to begin. In the case of the XovrE valve, there is a differential pressure of 27 bar acting resisting the valve opening, and 9-15 bar (depending on exact valve closing point) resisting the closing. It is the task of the valve spring, as well as to force the valve to follow the motion imposed by the cam, to push the valve against its seat and seal it against the excess of compression or combustion pressure in the cylinders over Xovr port pressures.

The XovrE valve has the shortest opening period, experiences the highest pressure differential at the beginning of its opening event and needs to be kept shut against the combustion pressure. The XovrE valvetrain was therefore selected for the experimental verification.

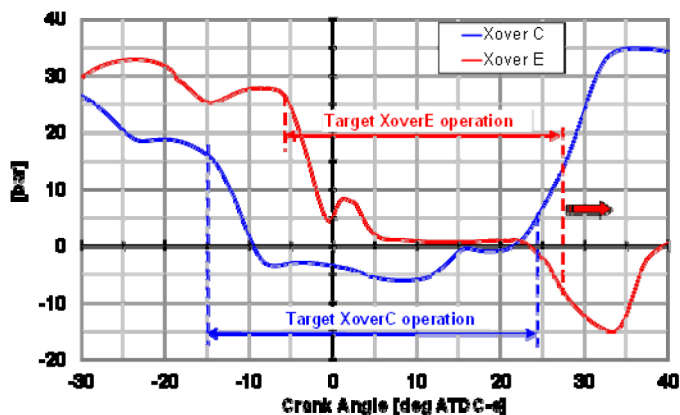


Figure 4. Pressure differentials acting on XoverC and XoverE valves 4000RPM full-load

ANALYTICAL MODELING

VALVETRAIN DYNAMIC SIMULATION

To ensure that the performance demands on the Xover valvetrain would be met, a lumped mass and stiffness dynamic model of the valvetrain was constructed using GT-VTRAIN software [6]. The model comprised all the moving parts of a single valvetrain as well as the rocker shaft and camshaft supports. The model was at first populated with estimated mass and stiffness values and exercised to define the combination which could achieve the target valve lift and timings whilst maintaining control of the valvetrain vibrations. This was a very valuable exercise as it drove the decisions on design and materials, and on the adoption of the pneumatic springs instead of coil springs.

Figure 5 shows the simulation results of a version of the valvetrain using a conventional progressive coil spring sized to meet the load requirements defined previously and operating within acceptable static stress range. The natural frequency of such spring was 2000 Hz, very high by normal automotive standards. Despite this the severe accelerations experienced during the valve opening event induced very high vibration amplitude of the spring mid coil (also called surge) at speeds greater than 2000 RPM. This in turn resulted in excessively high fatigue stress ranges in the spring wire, prohibitive for even the best spring materials and treatments and which would lead to failure. Ultimately the problem with this type of spring could be summarized in relatively high mass (the mass of the spring considered was 58 grams) and relatively low internal damping, even when using progressive coils.

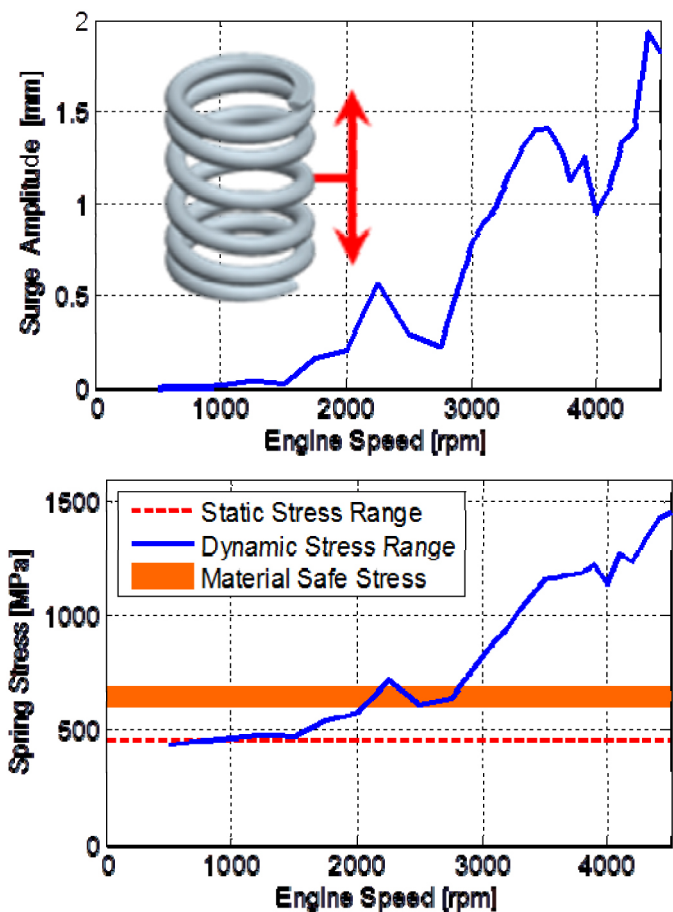


Figure 5. Simulation of coil spring dynamics. Top: Valve spring surge amplitude, Bottom: Maximum valve spring stress amplitude

Pneumatic valve springs on the other hand have intrinsically high progressivity as shown by the spring frequency chart in Figure 6 because the gas pressure in them follows a substantially adiabatic transformation. This allows tuning the spring progressivity by varying the spring initial volume. Furthermore the extremely low mass of the DLC coated aluminum spring piston means that the critical damping of the pneumatic spring is circa 7 times smaller than the critical damping of the coil spring. The damping in the pneumatic springs is generated by the friction of the spring piston against the seal. The seal is made of a grade of filled PTFE, a very low friction material. The sealing action is achieved by allowing the gas inside the spring to push the seal against the spring piston and this generates enough friction force to dampen any resonance of the pneumatic spring itself.

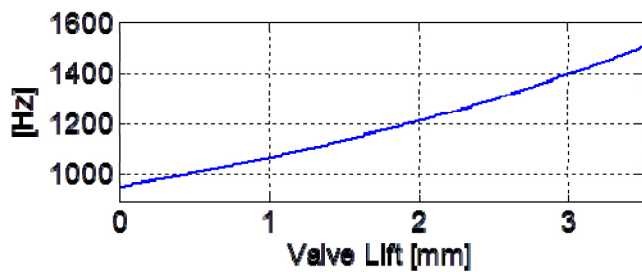


Figure 6. Pneumatic spring natural frequency. The curve indicates non-linear stiffness due to the non-linear nature of polytropic compression

The dynamic model was also used to determine the loads and stresses on components, help develop the valve lift profile and study the dynamic loads and motions, including the valve seating behavior. As more detailed information on component mass and stiffness became available during the design and testing process the dynamic model was updated and new simulations produced.

The bending stiffness of the rocker was the controlling stiffness of this valvetrain. Its effect is felt especially during the valve seating portion of the valve opening event. Simulations were performed to define the value of this stiffness which would be needed to achieve acceptable valve seating velocities in the whole speed range. Figure 7 shows the effect on the valve seating velocity of a rocker with the design target stiffness and of another one with 30% lower stiffness. It can be seen that with low rocker stiffness the valve seating velocity becomes excessive at speeds just above rated (4000 RPM), giving very little margin for error in the overall valvetrain design.

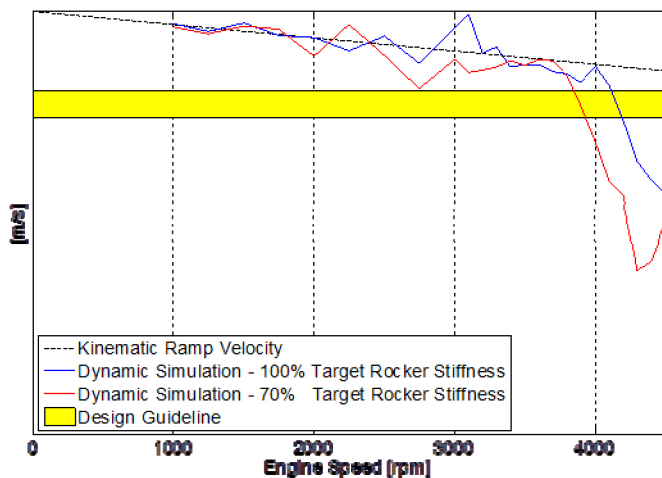


Figure 7. Simulation of the effect of rocker stiffness on valve seating velocity

Figure 8 shows the relative contribution, for the design which was made and tested, of the mass and compliance of the individual valvetrain components on the natural frequency which is in the region of 4000 Hz. The natural frequency is influenced primarily by the flexibility of the rocker and poppet valve. The poppet valve constitutes the payload, whereas all the other components have the function of controlling the motion of the valve. The rocker shaft and the camshaft supports are not part of the moving elements of the valvetrain (although the hole for the rocker shaft does reduce the rocker bending stiffness) and were therefore sized to add minimum compliance to the overall valvetrain. Their target stiffness was chosen to be about twice the rocker stiffness. In the following section the design process for the valve adaptor and for the rocker will be discussed.

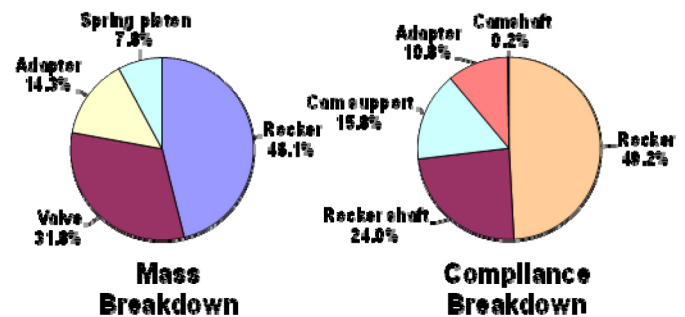


Figure 8. Mass and Compliance breakdown contributing to valvetrain natural frequency

VALVE ADAPTOR DESIGN AND ANALYSIS

The design of the valve adaptor focused on two aspects. The first was to insure that the closing impacts of the valve head against the valve seat at high engine speeds would not unseat the collets from the adaptor and the other was the need to make the bottom face of the adaptor flexible so that it could make flat contact against the forked end of the rocker, allowing for minor misalignment. The final design is shown in Figure 9.