3- Cylinder SI Engine with fully variable Valve Train UpValve on Intake and Exhaust Side

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Abstract

UpValve is a fully variable valve actuation (VVA) system, allowing for continuous adaptation of valve opening duration and correlated peak lift throughout the complete engine map. This mechanical approach was already presented on this conference in 2015 [1]. Here an example of application is given, where both intake and exhaust side are equipped with UpValve. The target engine is dedicated for stoichiometric combustion of compressed natural gas (CNG) [2].

On the intake side the VVA system cuts the pumping work at part load thanks to an optimized intake valve closing (IVC) time. In addition, in the higher load area external compression and subsequent intercooling in combination with Miller valve timings reduce charge temperature and knock limitation, gaining the potential for higher BMEP, elevated compression ratio (CR) and/or more downsizing [3]. The exhaust side VVA system controls the flow split to the turbines of an innovative, two- stage boost system, supporting both steady-state torque as well as driveability, which are often concerned for heavily downsized engines. This engine and valve train concept was drafted, designed, procured and tested within a challenging time schedule. Currently the full potential is explored on the engine dyno. Subsequently, vehicle tests will complete the project.

After outlining the motivation for an innovative CNG application and presenting the project scope the paper will

- explain how UpValve works and is tuned for the actual application
- describe the necessary modifications on cylinder head and surroundings
- show how the challenging package constrains were met
- report results of VVA measurements and some initial engine testing.

1. Introduction: CO_2 Challenge and CNG Operation

Further reducing the CO_2 emissions of vehicle's powertrains is imperative. According to **Eq. 1** (see Appendix for details) the instant ratio of released CO_2 mass m_{CO2} per driving distance x from an internal combustion engine (ICE) is linked to fuel properties (a), engine efficiency (b) and the actual road load (c):

$$\frac{dm_{CO2}}{dx}=\left(\frac{44}{12}\right)\underbrace{\left(\frac{\xi_{C,F}}{H_U}\right)}_{a}\underbrace{\left(\frac{1}{\eta_e}\right)}_{b}\underbrace{\left(m[gf_R+a]+\frac{\rho_{Air}}{2}c_d Av^2\right)}_{c} \qquad \textbf{Eq.1}$$

Hence, the following measures can be taken to reduce the CO_2 output of an IC engine:

(a) Fuel: Change to a fuel with a low ratio of carbon content $\xi_{C,F}$ and calorific value H_u
(b) Engine: optimize the overall efficiency η_e of the powertrain
(c) Road Load: for a given vehicle speed v and acceleration a (according to the actual test cycle) reduction of vehicle mass m, coefficient of rolling resistance f_R and aerodynamic drag (drag coefficient c_d, frontal area A) are subject of optimization

Changing fuel from gasoline ($\xi_{C,F}/H_u = 0{,}84/(42\ MJ/kg) = 20\ g_{CO2}/MJ$) to CNG ($\xi_{C,F}/H_u = (0{,}63\ldots.. 0{,}74)/(41\ldots49\ MJ/kg) = 15{,}2\ g_{CO2}/MJ$) cuts CO_2 already by 24% (a) [4]. Moreover, in terms of engine efficiency (b), the high RON number of CNG (≥120 vs. 95) allows a significant increase in the compression ratio (CR) if the engine is dedicated for CNG only operation. In addition, CNG has favourable ignition characteristics, which improve combustion tolerance to very (EGR) diluted mixtures. According to these CNG advantages in (a) and (b) a CO_2 reduction of 30% can be assumed [5,6].

However, the road load is higher (c), as the CNG tanks are larger and heavier even if a reduction in vehicle range is accepted. Therefore the engine output has to be increased for identical driving performance to the cost of some efficiency losses.

In addition, the fuel effect on the engine output has to be considered. **Eq. 2** (see Appendix) shows that the fuel properties are mainly down to parameter (d), which stays almost constant (Gasoline: $H_u/L_{st} = (42\ MJ/kg)/14{,}3 = 2{,}9\ MJ/kg_{Air}$, CNG: $H_u/L_{st} = (41\ldots49\ MJ/kg)/(14{,}3\ldots.16{,}9) = 2{,}85\ MJ/kg_{Air}$).

$$P_e=\underbrace{\left(\frac{H_u}{L_{st}}\right)}_{d}\underbrace{\left(1+\left[\frac{1}{\lambda_V L_{st}}\frac{M_A}{M_F}\right]\right)^{-1}}_{e}\left(\frac{p_{In}}{R_A T_{In}}\right)\left(\frac{\eta_e\lambda_A}{\lambda_V}\right)(V_H in)=(BMEP)(V_H in) \qquad \textbf{Eq.2}$$

But in contrast to the gasoline DI version, gas injection into the ports displaces up to 15% of the trapped air (e), which causes an associated loss of torque. To cope with that, either the engine displacement (V_H) or the boost pressure (p_{In}) have to be increased with some drawbacks in fuel economy. Obviously direct injection of CNG into the cylinder after IVC is favourable.

Nevertheless, the outlined CO_2 reduction opportunities justify thorough investigations of highly sophisticated, monovalent CNG concepts. Within the GASON project (GAS-ONly internal combustion engine, GA No. 652816), founded by the EU in the frame of the

HORIZON 2020 initiative (H2020-GV3–2014) [7], the subproject "downsized CNG engine" should fully realize the CO_2 advantages of CNG [2]. The project scope is sketched in the next chapter.

2. The GASON Engine: a dedicated CNG Concept

The baseline case is a generic Ford Grand C-MAX with a turbocharged (T/C), 1.6l I4 port fuel injected (PFI) CNG engine. From this reference, 20% CO_2 reductions are targeted, which should be achieved through a bundle of cutting- edge technologies, provided by different companies, **Fig. 1**. Some of these measures are:

- Downsizing the original 1,6l T/C I4 to an 1,0l T/C I3 engine with high CR (gas only operation) to fully exploit the superior knock resistance of CNG. Reinforcement of engine structure and components to stand the higher specific engine load and peak pressures while minimizing the friction drawbacks.
- Change to CNG Direct Injection system (CNG DI) to avoid the volumetric disadvantages of PFI as described above
- Introduction of a fully variable VVA systems on both intake and exhaust side for reduction of pumping work and advanced gas exchange control
- Development of an innovative parallel- sequential dual T/C system to save drive-away torque and responsiveness. This bi-turbo approach requires a modified head design with separated exhaust ports and manifolds. The exhaust VVA system is used to sequentially actuate the turbo chargers.

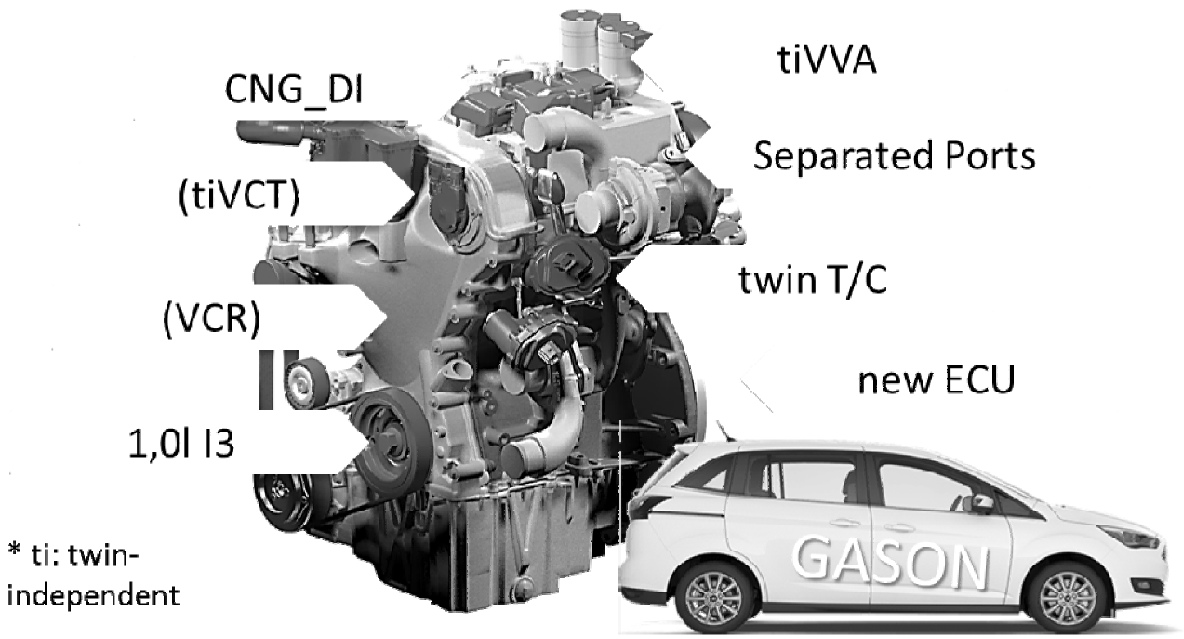

Fig. 1: GASON Project Overview

Engine targets are 240 Nm/l (30 bar BMEP) and 110 kW/l. The demanded boost pressure is significantly higher than for any known downsized and supercharged gasoline production engine. Therefore a novel twin-turbo system is proposed, **Fig. 2**.

The cylinder head features two integrated 3-to-1 exhaust manifolds. All right exhaust valves and ports per cylinder are linked to a first common head orifice, and all left ports to a second one. Each head outlet serves one turbine. In low up to mid mass flow conditions, only the exhaust valves associated to the permanent engaged T/C_1 are opened by the VVA system, while the other valve group is kept closed. Hence, T/C_1 can be tuned for low mass flows, enabling more low end torque and favourable transient response. A so called compressor shut off valve (SOV) is closed here to avoid charge loss through the non- active T/C_2.

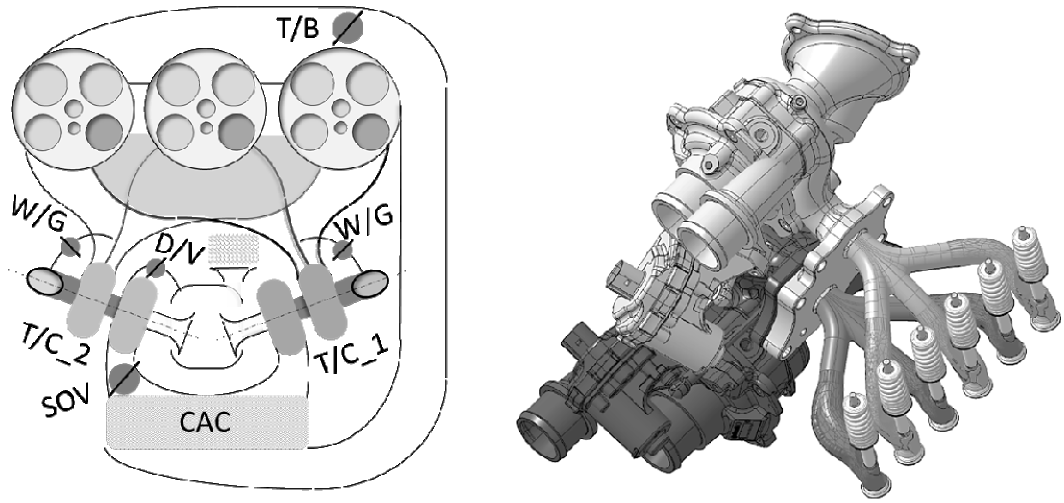

Fig. 2: Parallel-sequential Twin Boost System

If engine speed and load increase towards the choke limit of T/C_1, the VVA system starts to open the second group of exhaust valves. To help T/C_2 to spool up, initially the SOV is kept closed with open dump valve (D/V). As soon as the pressure across the shut off valve is balanced, the SOV is opened, and both compressors start to charge the engine.

Thanks to the exhaust VVA this advantageous arrangement does not need further flaps on the challenging, hot engine side, which are often a concern in terms of robustness and tightness. Moreover, a very precise and responsive T/C management is realized through the applied VVA system, described in the next chapter. The demonstrator system is equipped with waste gates (W/G) on both turbines. Investigations are scheduled to understand whether the W/G function could be partly shifted to the exhaust VVA.

3. UpValve: a continuously variable VVA System

The VVA UpValve system enables infinitely variable, on-demand control of the valve lift from valve shut off up to full lift [8]. The system is based on a low-friction roller finger follower (RFF) with hydraulic lash adjuster (HLA), **Fig. 3.** The camshaft ⟨N⟩ moves outward, coming into contact with the cam follower roller ⟨RN⟩ in the UpValve rocker ⟨H⟩, which, via the roller

⟨RS⟩, also rests on the circular guide ⟨K⟩ and control shaft ⟨S⟩, (b). The operating contour of ⟨K⟩ follows a circular path around the axis of the RFF roller when the valve is closed, (a).

The rocker ⟨H⟩ itself carries a working lobe ⟨A⟩ that acts on the roller ⟨R⟩ of the RFF. It consists of a base circle concentric to the rocker pivot point ⟨RS⟩ and a subsequent lift zone. The geometry of circular guide and rocker base circle provides the system's inherent valve shut off capability. Together with the valve spring and the HLA a hairspring securely maintains all contacts.

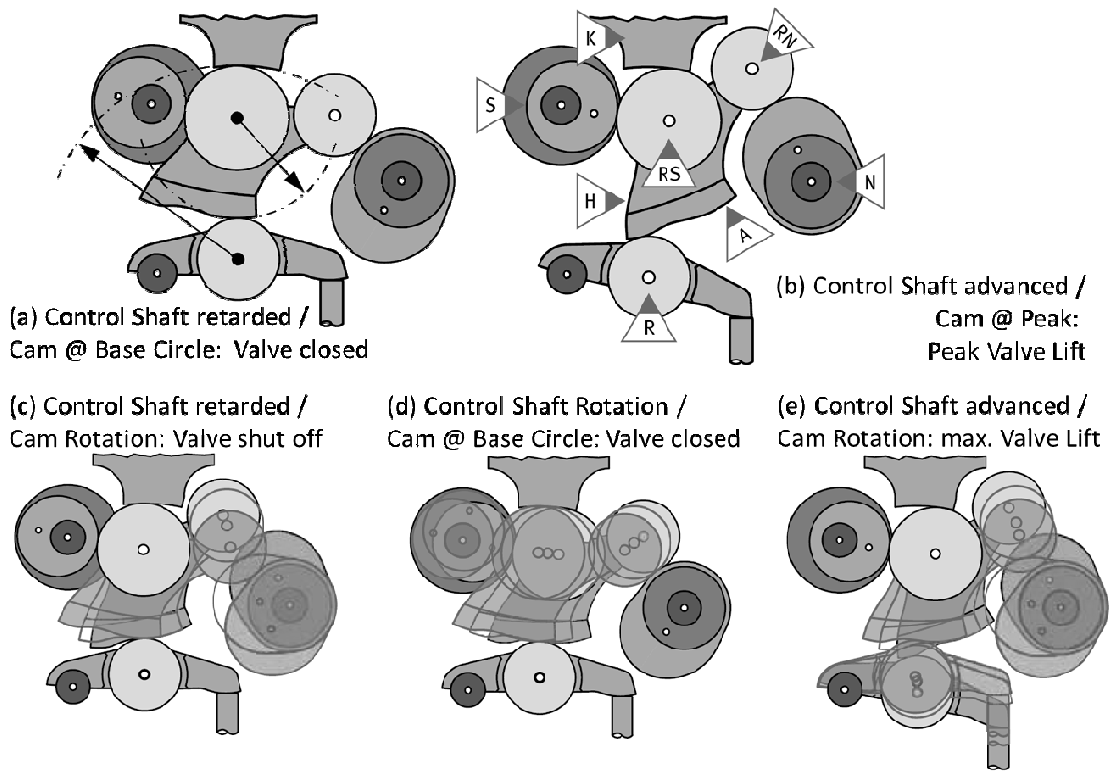

Fig. 3: UpValve Architecture and Operation

The rotating camshaft imparts a pure oscillation of the UpValve rocker around the stationary, roller axis ⟨RS⟩, (c). A rotation of the control shaft shifts the pivot point along the circular guide (d), so that different areas of the working lobe are traversed during cam lift (e). As a result, the valve lift is infinitely adjustable. Hence, the VVA system can be regarded as a serial arrangement of rotating ⟨N⟩ and oscillating ⟨H⟩ cam, where their correlation and thus the lift section effectively engaged on ⟨A⟩ is infinite variable.

In **Fig. 4** (a,b) $H(\alpha)$ depicts the required full lift, while $s(\alpha)$ shows the lift of the cam follower. To start an iteration, the opening side of $s(\alpha)$ is initially assumed. Together with the first half of $H(\alpha)$ this already determines the (lift zone of the) rocker lobe. Subsequently, the closing flank of $H(\alpha)$ provides the second halve of ($s(\alpha)$ and) the cam shape, as the pre-defined rocker lobe is traversed again in rearward direction. At that point in time the complete lift family down to zero lift is already defined. Obviously, it is still possible to tune the opening side of the cam. This degree of freedom is used to improve the correlation between peak lift and opening duration for all lift curves [1]. **Fig. 4** (a,b) reveal these optimized lift and acceleration sequences for selected angular positions of the control shaft.

Fig. 4 (c) explains how the peak lift (obtained during one full revolution of the cam shaft) and the associated opening duration (for 1mm and 0,1 mm reference lift) is correlated to the angular position of the control shaft. This characteristic is hard coded into the shape of the shaft lobe and can be different for each individual valve. Constrains are drawn by the curvature of this lobe and the Hertzian pressures. In addition, steep gradients provoke high torques on the shaft and unfavourable controllability.

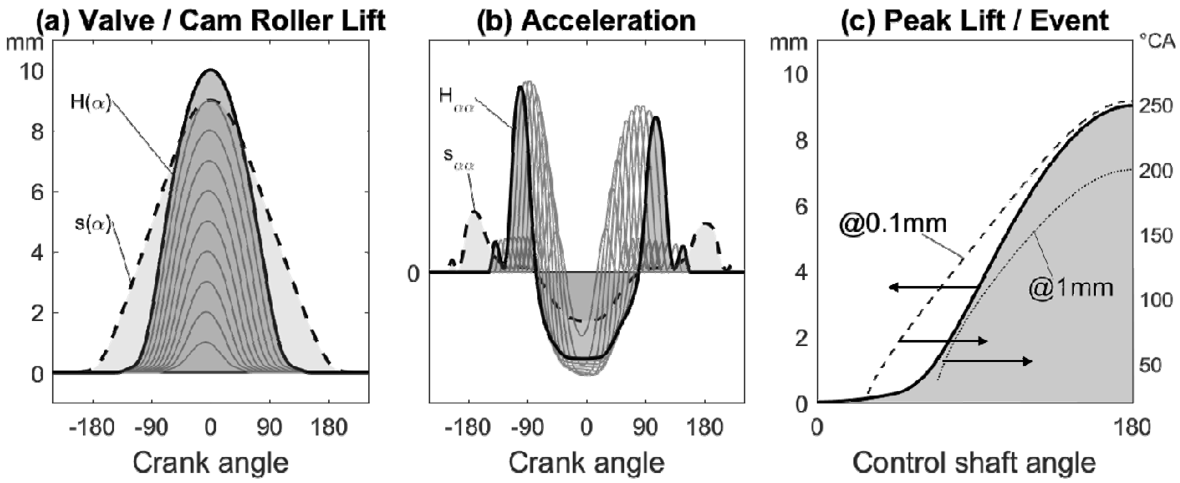

Fig. 4: UpValve Valve Lift Shapes and Control Characteristics

On its own, UpValve keeps the peak lift timing almost constant. The cam phaser (VCT) is used to optimize the actual correlation to the piston travel. Due to the valve- individual set up also adjacent cams assigned to one cylinder may be differ in terms of shape and peak position. Beside noise advantages (offsetting the valve seating) this asymmetrical layout offers some gas exchange opportunities.

Fig. 5 exhibits the whole, generic UpValve system. A brushless actuator ⟨A⟩ and a worm gear are installed in order to adjust the needle bearing-mounted control shaft ⟨S⟩. The end of the control shaft is fitted with a magnet for position reporting via a Hall sensor ⟨H⟩. Sensor and actuator are linked to the valve control unit (VCU), which receives its control shaft position targets from the ECU and powers the actuator accordingly.

4. The GASON VVA System: dual UpValve System in a structural Head Cover

The UpValve rocker, circular guide and control shaft must be added to the valve train. Moreover, worm drive and actuator have to be packaged. Cam shafts and associated phasers will be shifted, which gains impact on the timing drive.

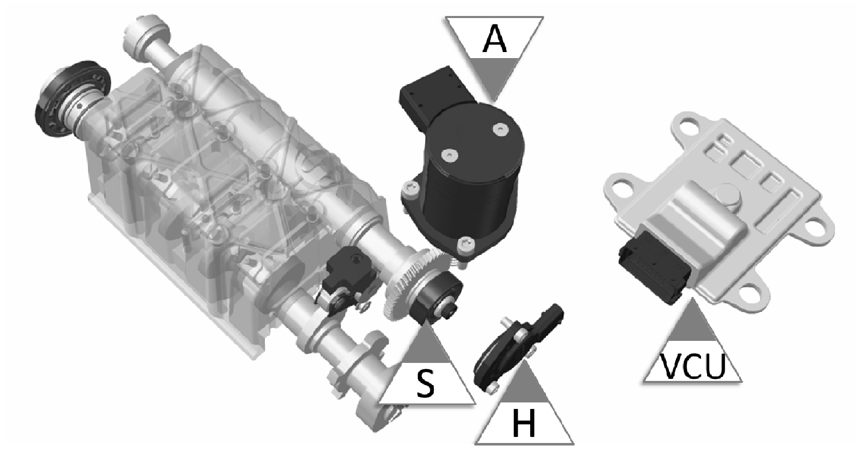

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Fig. 5: UpValve System Overview

The cylinder head is equipped with roller finger follower (RFF) on intake and exhaust side with outside hydraulic lash adjusters (HLA). Like for conventional RFF valve trains, UpValve can be drafted for pushed or pulled RFF. This conceptual decision as well as the actual arrangement and scaling of all VVA components is mostly driven by numerous package constrains and the structural integrity of the complete valve train [8]. Additional attention must be payed to the bearing concept of cam and control shafts, as this also contributes to the system stiffness. Further on, accessibility and serviceability are concerned

During this optimization process the best layout was identified for inside control shafts, pushed RFF and outside cam shafts, **Fig. 6**(a). All UpValve specific valve train components are included in a structural cover and will be introduced from the underneath cylinder head face, (b). Hence, installation windows are prevented, resulting in a very solid and compact structure. The cover is protected for high pressure die cast in mass production. The cylinder head itself carries the valves, the HLAs and the RFFs.

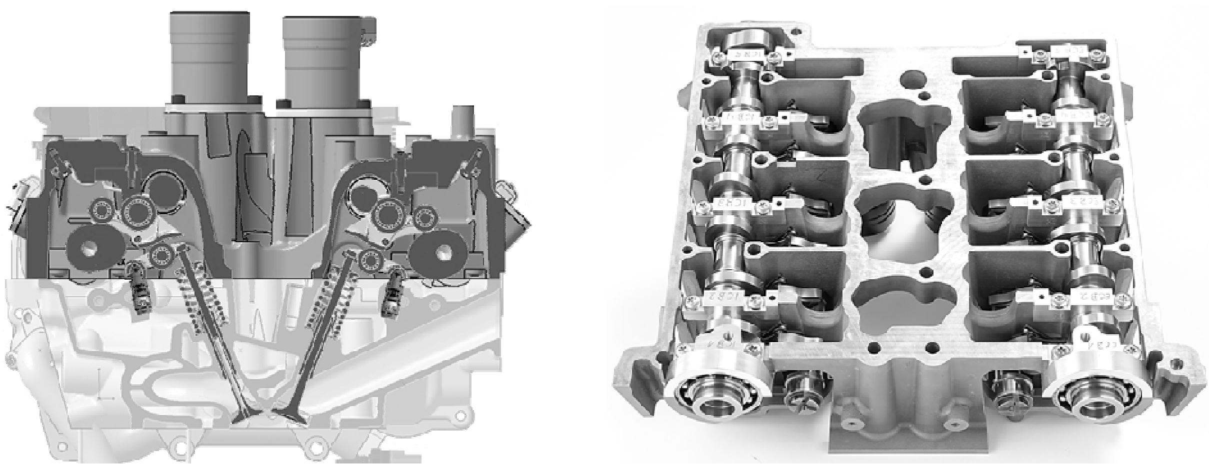

Fig. 6: Structural Valve Cover (CAD Design with Cylinder Head and Hardware)

The advantages of this architecture are e.g.

- cover with a high structural integrity to take all direct and lateral forces of the valve actuation, contributing to the desired, high stiffness of the valve train
- no access to head bolts through the VVA system necessary, thus less compromised layout
- unbroken responsibilities for design, procurement and (pre-)testing of the VVA system, as all VVA specific parts are contained in the structural cover
- no need for any additional covers

To ensure a high VVA response, the control shaft has needle bearings with split cages. It can be introduced from one end of the cover without any need for bearing caps. This design is compact and supports the cover stiffness additionally. There are three radial bearings at cylinder center, one at phaser end, and a thrust bearing close to the actuator, **Fig. 7**.

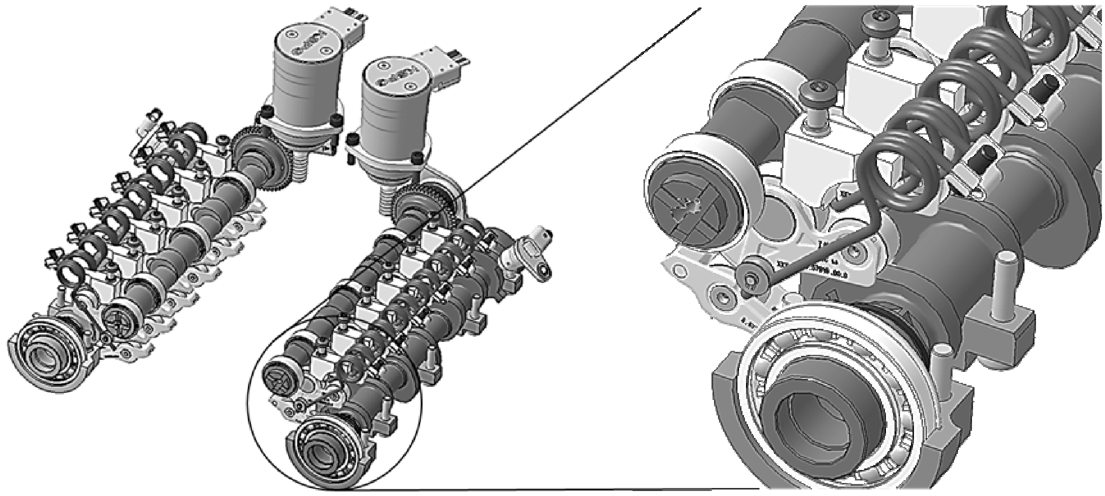

Fig. 7: Arrangement of Shafts and Bearings

Both camshafts have plain bearings with caps, which are bolted down to the upper cover. Once again, the bearings are located on cylinder center. The first bearing towards timing drive is rollerrized to cope the pretension of the belt.

Fig. 8 presents the valve train internals in more detail. The RFF design is slightly modified to tune the rocker ratio. Each UpValve rocker carries two rollers, the first (bigger) one in contact to circular guide and control shaft lobe, the second (smaller) one to follow the cam, both equipped with needle bearings. While the small roller is permanently driven by the cam, the large roller does rotate in respect to its contact points only, if the control shaft is actuated to visit a changed peak lift. Therefore, the large roller does not contribute to the rocker inertia.

Intake and exhaust rockers share common raw parts, while their individual working lobes differ. The raw parts are produced by metal injection moulding (MIM) in near net shape quality. Only the working lobe and both eyes for the roller pins are machined.

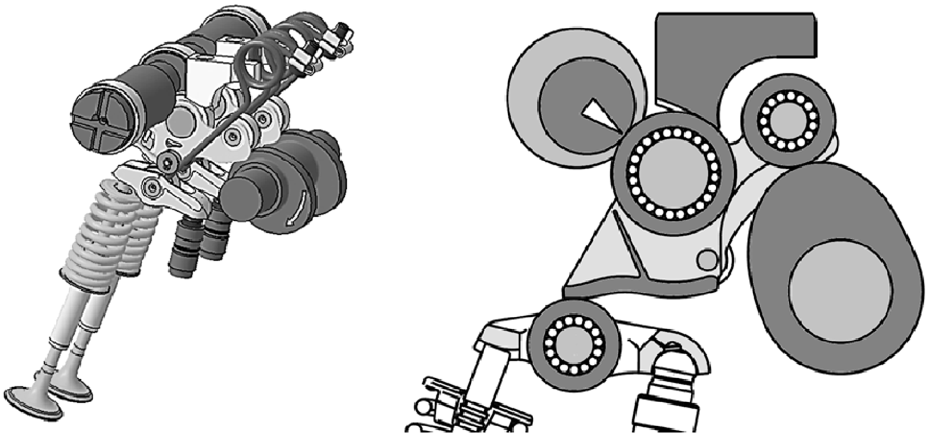

Fig. 8: Valve Train Components and Rocker Details

The circular pads are introduced from underneath and clamped to the cover top deck by through- bolting. The UpValve hairspring closes all clearances and keeps the assembly in place, if the valve spring is not yet engaged (at zero and small valve lifts).

Fig. 9 shows the peak lift characteristics for intake (a) and exhaust (b) currently used in this project. The intake side is designed to enable steady- state engine dyno investigations of early IVC load control in both single valve (swirl mode, (a1)) and parallel valve (tumble mode, (a3)) operation. The final design will be derived from these results and could have smooth transitions from single valve (low load) over differential lift (mid load) to parallel lift (full load).

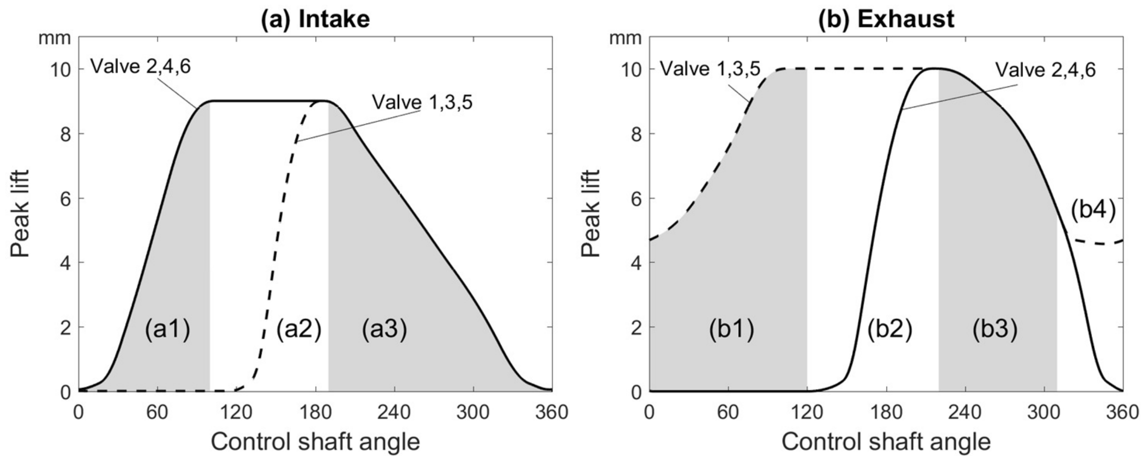

Fig. 9: Peak Lift Characteristics Intake (a) and Exhaust (b)

The exhaust side is shaped to control the twin-boost system. In section (b1) one exhaust valve is kept closed, and the complete exhaust mass flow is directed to the permanently engaged T/C. In this area both exhaust valve opening (EVO) and closing (EVC) can be optimized independently by VVA and VCT controller. (b2) indicates the transition area from single to twin turbo operation. Section (b3) allows for EVO and EVC control at dual T/C operation. Finally, (b4) is used to close the lobe shapes.

Together with offset cams a separation of blowdown and overlapping exhaust gas streams would be possible. However, this opportunity is more relevant for an I4 application with a single-scroll T/C, more suffering from the cross talk of blowdown and overlap events.

5. GASON Test Results: Valve Train Rig and Engine Dyno

After checking the dimensional integrity of the valve train by measuring the valve lift vs. cam and control shaft angle at elevated idle speed an essential testing output is the valve lift as a function of engine speed. For a fast evaluation, the lift curves $h(\alpha,n)$ during a continuous speed sweep are acquired keeping the control shaft in fixed position. The crank angle correlated deviations $\Delta h(\alpha,n)= h(\alpha,n) - h_L(\alpha,n_L)$ of the lift curves vis-à-vis the values h_L at idling speed n_L are depicted in color-coded fashion in a plane formed by crank angle α and the speed n, **Fig. 10** (a). The valve lift ⟨A⟩ and acceleration ⟨B⟩ serve as orientation.

As the speed increases, inertia forces initiate compressions ⟨C⟩ and load releases ⟨D⟩ at constant angle positions. In contrast to this, time-constant effects show up along inclined lines ⟨E⟩. From this the natural frequencies of the valve train are derived, forming the basis for frequency-filtered lift curves.

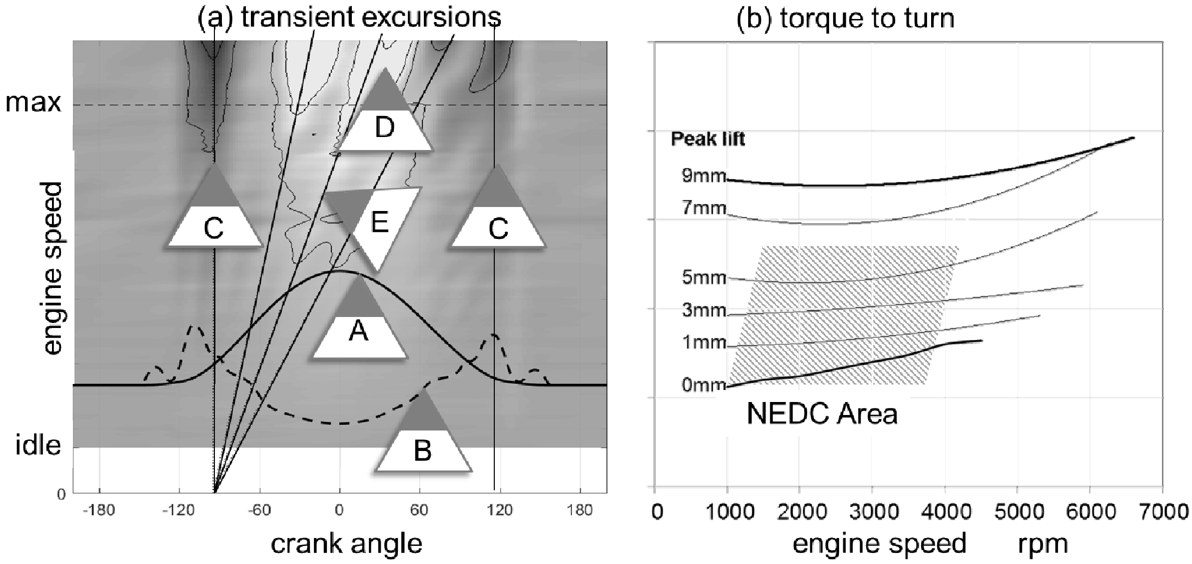

Fig. 10: Evaluation of Speed Sweep and Torque to Turn

Reducing these dynamic excursions was a central development task. Along with the sound valve train arrangement and the robust cover design, structurally rigid components and contacts as well as the lift curves developed in the frequency plane result in very low deformations below 0,15 mm. This favourable behaviour is also proved by seating velocities under 0,8 m/s. Moreover, at partial load and low speeds, a superior NVH is recognized.

The torque to turn the camshaft is revealed with **Fig.10**(b). At maximum lift, the friction is somewhat higher compared to a standard RFF type valve train, as more parts are engaged. But reducing the peak lift even a torque advantage is obtained, caused by the lower spring and inertia forces introduced into the system. Thus, in the main operating areas of the engine map, a friction gain adds up to the gas exchange advantages.

Initial dyno results suggest the high fuel saving potential of the engine concept. As a lot of parameters have to be tuned in each mapping point for best brake efficiency (e.g. spark and injection timing, valve timings and lifts on intake and exhaust side, boost mode, single/dual valve mode) a huge DOE program was performed. Fig. 11 shows the intake peak lift (controlled by the VVA) for single and dual valve opening as recommended by the DOE response surface.

Obviously a wide area of the VVA authority is used, demonstrating that the continuous adjustment is highly valuable. As expected, the single mode requires higher lifts, and starts to choke the engine towards higher mass flows. From this, a smoothed strategy for differential lift will be derived in the next step

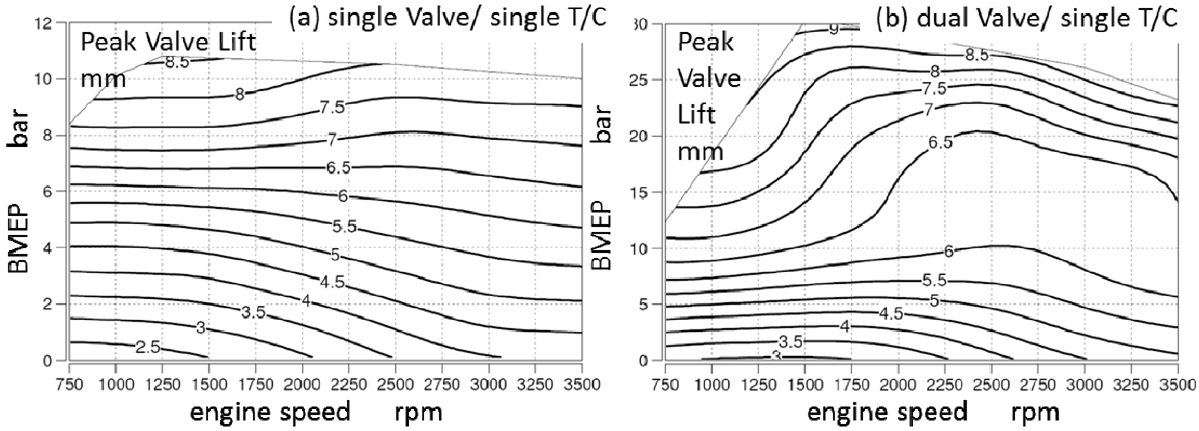

Fig. 11: Peak Lift Settings for best Efficiency in single and parallel Lift Mode

6. Conclusions

It is demonstrated, that an integration of the VVA system UpValve even into a small, downsized I3 engine on both intake and exhaust side is feasible. Rig tests confirm a high speed capability of the variable valve trains, combined with low friction losses and favourable response. Moreover, for low lifts a superior NVH behaviour is observed.

It is widely accepted, that a VVA application on the intake is reasonable. This study shows, that an additional exhaust side VVA allows for a robust and responsive control of a twin charger system without additional flaps. In a subsequent step, the function of waste gates might be partly covered by the VVA system.

Transition to CNG fuel reduces the CO_2 emissions significantly due to the lower carbon content and high calorific value. A dedicated CNG engine fully utilizes the high knock resistance of the fuel and allows, together with a sophisticated gas exchange and injection system, for massive downsizing. Thus further CNG benefits can be gained. In total, compared to today's best in class, bivalent CNG derivatives, further 20% CO_2 reduction are expected.

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Acknowledgments

This work was performed as part of a research project GASON assigned by the European Commission within the Horizon 2020 framework, supervised by Massimo Ferrera (CRF).

Publication of this paper by courtesy of Ford, Continental, Delphi, FEV and Schaeffler.

Appendix

Eq. 1: CO_2 emission per km. The released CO_2 mass m_{CO2} per driving distance x is

$$\frac{dm_{CO2}}{dx} = \left(\frac{dm_{CO2}}{dt}\right)\left(\frac{dt}{dx}\right) = \frac{\dot{m}_{CO2}}{v}$$

if v is the vehicle speed. The CO_2 mass flow itself can be expressed as

$$\dot{m}_{CO2} = \left(\frac{\dot{m}_{CO2}}{\dot{m}_{C,F}}\right)\underbrace{\left(\frac{\dot{m}_{C,F}}{\dot{m}_F}\right)}_{\xi_{C,F}}\underbrace{\left(\frac{\dot{m}_F H_u}{P_e}\right)}_{1/\eta_e}\left(\frac{P_e}{H_u}\right) = \underbrace{\left(\frac{\dot{n}_{CO2}}{\dot{n}_{C,F}}\right)}_{=1^*}\underbrace{\left(\frac{M_{CO2}}{M_C}\right)}_{=44/12}\xi_{C,F}\left(\frac{1}{\eta_e}\right)\left(\frac{P_e}{H_u}\right)$$

where

$\dot{m}_{CO2}$	CO_2 mass flow	$\dot{n}_{CO2}$	CO_2 mole flow
$\dot{m}_{C,F}$	carbon mass flow with fuel	$\dot{n}_{C,F}$	carbon mole flow with fuel
$\dot{m}_F$	fuel flow	M_{CO2}	mass of CO_2 molecule
P_e	engine brake power	M_C	mass of C atom
η_e	brake thermal efficiency	$\xi_{C,F}$	carbon mass fraction of fuel
H_u	lower calorific value of fuel	1^*	for complete combustion

Thus, the CO_2 rate per km is

$$\frac{dm_{CO2}}{dx} = \left(\frac{44}{12}\right)\left(\frac{\xi_{C,F}}{H_U}\right)\left(\frac{1}{\eta_e}\right)\left(\frac{P_e}{v}\right) = \left(\frac{44}{12}\right)\left(\frac{\xi_{C,F}}{H_U}\right)\left(\frac{1}{\eta_e}\right)\left(m[gf_R + a] + \frac{\rho_{Air}}{2} c_d A v^2\right)$$

where

m	vehicle mass	f_R	coefficient of rolling resistance
a	vehicle acceleration	c_d	coefficient of aerodynamic drag
A	vehicle frontal area	ρ_{Air}	density of air

Eq. 2: engine brake power. By plain introduction of sensible ratios the effective output can be written as

$$P_e = \underbrace{\left(\frac{P_e}{\dot{m}_F H_u}\right)}_{\eta_e} H_u \underbrace{\left(\frac{\dot{m}_F}{\dot{m}_{A,st}}\right)}_{1/L_{st}} \underbrace{\left(\frac{\dot{m}_{A,st}}{\dot{m}_A}\right)}_{1/\lambda_V} \left(\frac{\dot{m}_A}{\dot{m}_A[+\dot{m}_F]}\right) \underbrace{\left(\frac{\dot{m}_A[+\dot{m}_F]}{\rho_{A[F]} V_H in}\right)}_{\lambda_A} \underbrace{\left(\frac{p_{In}}{R_{A[F]} T_{In}}\right)}_{\rho_{A[F]}} V_H in$$

where

L_{st}	stoichiometric AFR	$R_{A[F]}$	gas constant of air [air/ fuel mixture]
λ_V	relative AFR (>1: lean)	p_{In}	pressure upstream intake valves
λ_A	volumetric efficiency	T_{In}	temperature upstream intake valves
$\dot{m}_A$	air mass flow	V_H	engine displacement
$\dot{m}_{A,st}$	stoichiometric air flow	i	power strokes per engine rev. (=0,5)
		n	engine speed

The terms in [brackets] only apply, if the fuel is injected into the ports (PFI), and a homogenous, gaseous mixture is trapped. The actual gas constant upstream the intake valves is

$$R_{A[F]} = \frac{\dot{m}_A}{\dot{m}_A\left[+\dot{m}_F\right]} R_A + \left[\frac{\dot{m}_F}{\dot{m}_A + \dot{m}_F} R_F\right] = \frac{\dot{m}_A}{\dot{m}_A\left[+\dot{m}_F\right]} R_A \left(1 + \left[\frac{1}{\lambda_V L_{st}} \frac{M_A}{M_F}\right]\right)$$

where M_A, M_F are the weights of air and (gaseous) fuel molecules. Thus, the engine output is

$$P_e = \underbrace{\left(\frac{H_u}{L_{st}}\right)}_{a} \underbrace{\left(1 + \left[\frac{1}{\lambda_V L_{st}} \frac{M_A}{M_F}\right]\right)^{-1}}_{b} \underbrace{\left(\frac{p_{In}}{R_A T_{In}}\right)}_{c} \underbrace{\left(\frac{\eta_e \lambda_A}{\lambda_V}\right)}_{d} \underbrace{\left(V_H in\right)}_{e} = BMEP\left(V_H in\right)$$

and is determined by fuel properties *(a)*, the mixture composition in the intake system *(b)*, a boost factor *(c)*, specific *(d)* and external *(e)* engine parameters.

Abbreviations

BMEP	brake mean effective pressure	I3/ I4	inline 3/ inline 4 (cylinder engine)
BSFC	brake specific fuel consumption	ICE	internal combustion engine
CAC	charge air cooler	NVH	noise, vibration, harshness
CR	compression ratio	PFI	port fuel injection
DI	direct injection	RFF	roller finger follower
DOE	design of experiments	RON	research octane number
D/V	dump valve (@ T/C)	SOV	shut off valve
ECU	engine control unit	T/B	throttle body
EGR	exhaust gas recirculation	T/C	turbo charger
EVC	exhaust valve closing (time)	ti	twin independent
VCT	variable cam timing (phaser)	VCR	variable compression ratio
EVO	exhaust valve opening (time)	VCU	valve control unit
FMEP	friction mean effective pressure	VVA	variable valve actuation
HLA	hydraulic lash adjuster	W/G	(T/C) waste gate