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ADVANCED COMBUSTION AND ENGINE INTEGRATION OF A HYDRAULIC VALVE ACTUATION SYSTEM

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Abstract: *An electro-hydraulic camless system has been installed on an EGR 11 litre multi-cylinder engine. This engine with fully flexible valvetrain is therefore a powerful and efficient tool to study new combustion concepts in the scope of US10/Euro6 emission regulations. The fully variable valve actuation also required to develop new engine braking strategies, which is one of the key features for heavy duty engines. The ability to pilot each valve individually provides many combustion feature possibilities like:*

- *Use of Miller effect by playing on intake valve duration*
- *Use of variable air motion by modifying intake valve lift separately from one port to another one*
- *Manage exhaust temperature by cylinder shut off and isolate the non-fired ones*
- *Injection at breathing top dead center to perform partially or fully homogeneous combustion.*

This presentaton highlights the potential benefits of Miller cycle on different engine speeds and loads. lt can be stated that both early and late Miller bring benefits on NOx/Soot trade-off on low load points, whatever the engine speed.

Keywords: Camless; pHCCI; Miller.

INTRODUCTION

In engineering, the Miller cycle is a combustion process used in a type of four-stroke internal combustion engine. The Miller cycle was patented by Ralph Miller, an American engineer, in the 1940s. In the Miller cycle the intake valve is left open longer than it normally would be: "Late" Miller effect. This is the "fifth" cycle that the Miller cycle introduces. As the piston moves back up in what is normally the compression stroke, the charge is being pushed back out the normally closed valve. Thus, as the piston travelled back up the cylinder, rather than compressing the fuel mix, it simply pushed it back out into the intake manifold. The compression phase only occurred from the moment the valve was closed to the moment the piston reached TDC; thus, by varying the timing on the intake valve, it is possible to effectively change the compression ratio of the engine, dropping it below the total ratio of the cylinder. By extension, when the intake valve is left open shorter than it normally would be is also called early Miller effect. In that case, there is no reverse flow into the intake manifold but it reduces the effective compression ratio. This cycle is called either Atkinson or Miller Cycle. Indeed, the first patent was written by Atkinson in 1882. This outcome reduced compression ratio enhances Partly Homogeneous Compressed Combustion Ignition in case of Diesel engines.

ENGINE DESCRIPTION

ENGINE DATA

The engine data as well as the main dimensions of the power unit are presented in Table 1 while Figure 1 shows a picture of this engine with its modified cylinder head.

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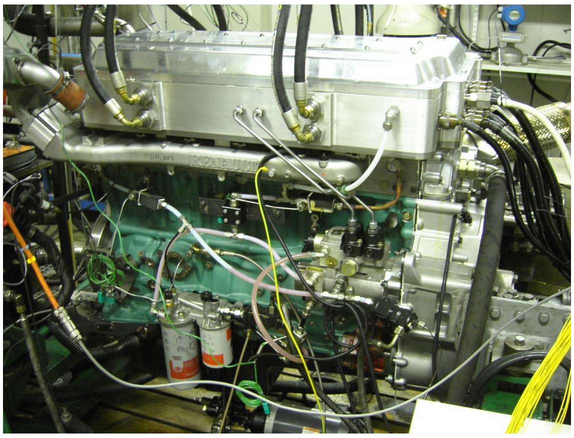

Figure 1. MD11 Camless engine.

Table 1. Engine performances and main dimensions.

Main dimensions	
Displacement (dm3)	10.837
Bore (mm)	123
Stroke (mm)	152
StrokeIBore ratio	1.24
Single-cyl Displacement (dm3)	1.806
Engine performance	360 Hp
Max power at speed	269.5 kW at 1800 rpm
Max torque at speed	1750 Nm at 1200 rpm
Compression ratio	16:l
Max cylinder pressure (bar)	220
Injection nozzle	
Valve in Flow	1.49 Ilmin Q 100 bar
Number of holes	6
Spray angle	140"

INJECTION SYSTEM

A Bosch CRSN-4 B-sample amplified common rail injection system was used in this study. The nozzle has 6 holes with a spray angle of 140° and a hydraulic flow of 735cc @ 100 bar.

ANALYSIS AND RESULTS

PERFORMED TESTS

Three types of tests are reported here. The main goal is to investigate potential benefits of Miller cycle with the following sequence:

- Variation of intake valve closing (IVC) (Equivalent to Inlet Valve Opening duration in the figures);
- EGR variation around the optimum Miller Setting;
- Torque increase and further Miller optimization to extend the area of pHCCl combustion.

The engine was first calibrated for 360 Hp / 1750 Nm. Figure 2 plots the real lifts using the Camless system by playing on intake valve opening duration. These lifts can be compared to the reference mechanical lifts. It can be stated that there is great latitude on the IVC time. Figure 2 shows the minimum intake valve opening duration tested (75 °CA duration, equivalent to 10.1 effective compression ratio) and the maximal intake valve opening duration (245 °CA duration, also equivalent to 10.1 effective compression ratio) to be compared to the reference duration 160 °CA (green curve, equivalent to 16 compression ratio). Figure 3 shows the influence of the intake valve opening duration on the effective Compression Ratio. On the exhaust side, the exhaust opening duration was kept constant to 200 °CA during all the first tests and the lift close to 10 mm, trying to be very close to classic mechanical valvetrain.

Influence of Miller cycle on A25 speed-load point

The best results were achieved on A25 speed load point (1 200 rpm - 25% Load - 5 bar BMEP) where significant NOx and soot reduction have been simultaneously achieved. The Main Injection Timing used is 3 °CA BTDC for all results presented in this section. Figure 4 shows the impact on Soot by modifying the intake valve opening duration while figure 5 presents the impact on NOx. The red diamond represents the reference inlet valve opening duration at 160 °CA duration equivalent to the reference IVC (540 °CA). There is a "hump" effect on soot. At the beginning of lntake valve duration swing, the soot remains constant in direction of Early Miller or increase a little in the direction of Late Miller. Less soot oxidation, due to cooler combustion products (decrease of effective compression ratio) and less excess of oxygen (decrease of mass of gas trapped within the cylinder) are probably the dominant causes of the increasing soot level. Beyond a certain limit, soot are very low with very long intake valve opening duration or very small intake valve opening durations. This soot decrease can be explained by the start of pHCCl combustion. In this pHCCl combustion, fuel is injected into the combustion chamber in the vicinity of the top dead centre for preventing fuel from adhering to the wall, and pre-mixture, which is formed shortly before ignition is burnt [2]. By decreasing the effective compression ratio, the ignition delay is increased thus leading to much longer time for the charge to become homogeneous. Due to the fact that most of the fuel is injected before start of combustion, there is less combustion at extremely rich local conditions favorable for soot formation; hence less soot is formed. It should be noted that even if the number of locally over-rich areas decreases, the global equivalence ratio increases significantly. The use of Miller cycle decreases indeed the mass of air trapped within the cylinder, but this effect is compensated by a better air-fuel mixing. NOx are also affected by this swing on intake valve duration and very late Miller (220" to 240 °CA duration) leads to low NOx level. This decrease in NOx cannot be explained by an increase in EGR (see figure 6) but by a decrease in oxygen content due to lack of fresh air. The decrease in oxygen content and the decrease in charge temperature due to lower effective compression ratio lead to a lower NOx formation. This decrease in EGR level is explained by the difference between exhaust manifold pressure and intake manifold pressure that decreases with both early and late Miller. Late Miller leads to lower NOx level because EGR keeps higher values with Late Miller than with Early Miller. HC and CO are almost constant with moderated Miller settings and then increase sharply with extreme Miller settings (see figure 7). Following these tests, two optimum can be selected: one called "Early Miller optimum" with 90 °CA intake opening duration instead of 160 °CA (IVC = 470 °CA instead of 540 °CA for the reference), and the second called "Late Miller optimum" with 240 °CA intake opening duration. (IVC = 620 °CA instead of 540 °CA for the reference).

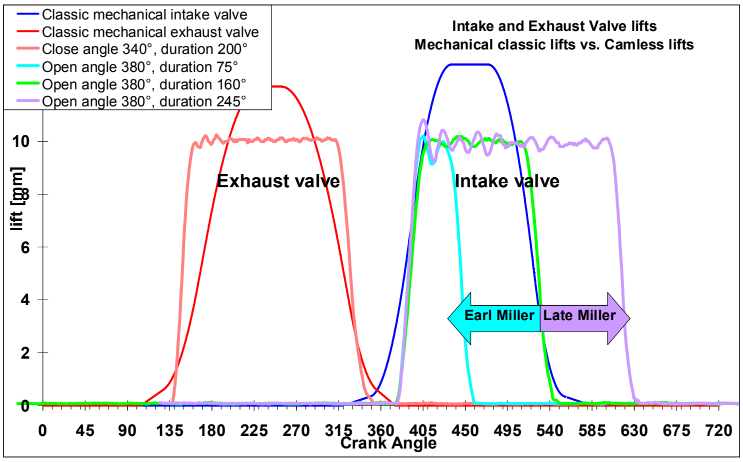

Figure 2. Comparison between mechanical lifts and HVA lifts.

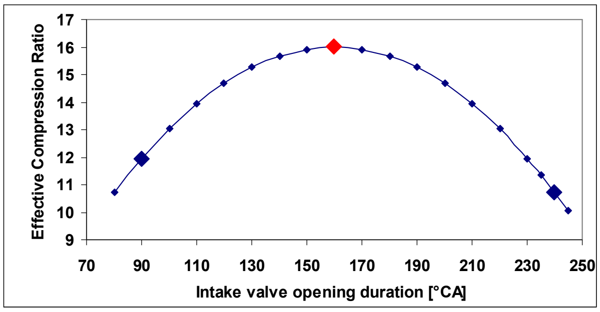

Figure 3. Influence on intake valve opening duration on effective Compression ratio at A25.

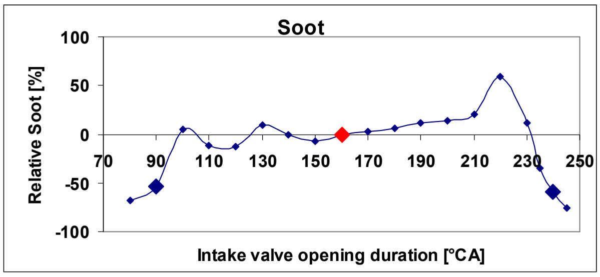

Figure 4. Influence on intake valve opening duration on soot emissions at A25.

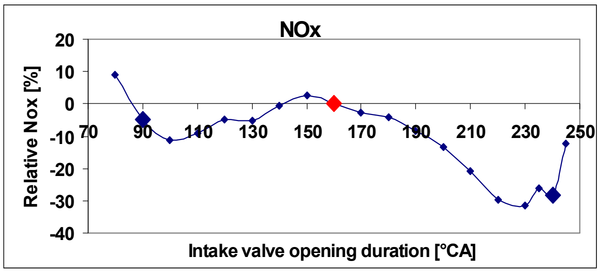

Figure 5. Influence on intake valve opening duration on NOx emissions at A25.

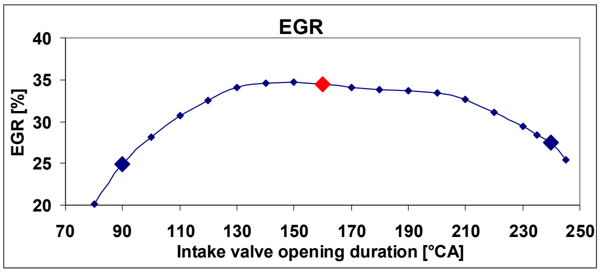

Figure 6. Influence on intake valve opening duration on EGR emissions at A25.

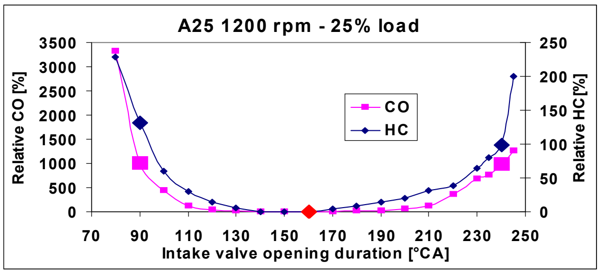

Figure 7. Influence on intake valve opening duration on CO and HC emissions at A25.

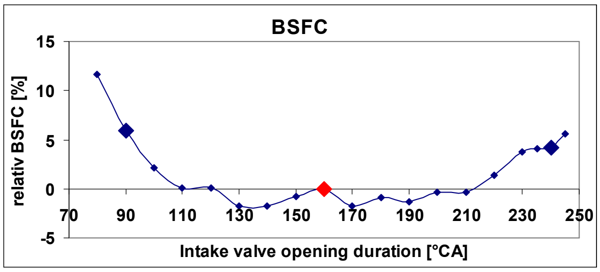

Figure 8. Influence on intake valve opening duration on relative BSFC at A25.

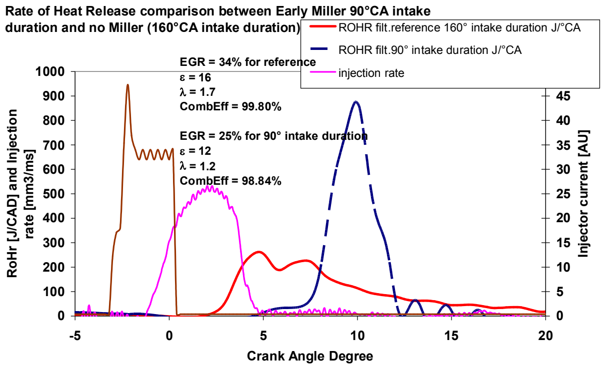

Figure 9. Rate of heat release with early and late Miller at A25.

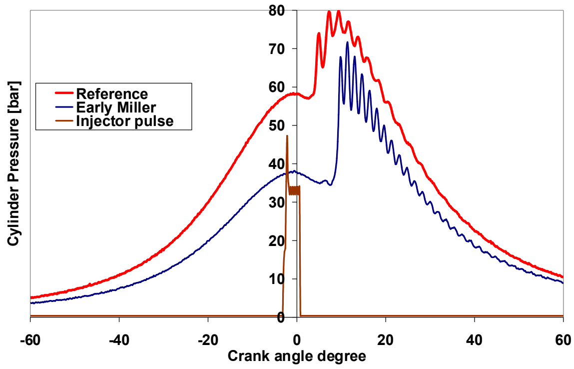

Figure 10. Cylinder pressure with early Miller compared to the reference combustion at A25.

Table 2 gives the benefits for these two optimum settings. On this speed load point, the optimum setting is obtained with late Miller which brings a 28% NOx decrease and 60% Soot decrease in comparison to the reference. However, there is a 4% penalty in consumption due mainly to the late combustion phasing. On both Early and Late Miller settings lambda is decreasing down to 1.2 (reference case = 1.7). Effective compression ratio is 10.7 with the Late Miller optimum setting and 12 with the Early Miller optimum setting. These conditions are very similar to the PPC combustion settings described in [1].

Figure 8 shows the impact of intake valve opening duration on fuel consumption. The consumption is roughly 5% higher on both Miller optimums.
This can be explained by several reasons:
- pHCCI combustion occurs later in the cycle (see figure 9).
- Compressor efficiency is lower in case of Miller settings (due to decrease in air flow)
- Combustion efficiency is lower due to increase in HC and CO emissions.

In figure 9, the premixed combustion starts more than 5 °CA later than the reference combustion at same injection timing. This difference cumulated with the decrease in compressor efficiency and decrease in combustion efficiency explains the 5% higher consumption. It can be stated that in both Early and Late Miller settings, injection is finished before the start of combustion. Therefore, the lower effective compression ration allowed increasing ignition delay, thus leading to a greater time for the charge to become homogeneous before start of combustion occurs ("pHCCI" effect). The effective decrease in compression ratio can be also stated on the cylinder pressure traces by the difference in cylinder pressure at top dead center and before the combustion starts (Figure 10). Early and Late Miller settings are not equivalent in term of volumetric efficiency and A/F. This means that these curves are not symmetric around the reference at 160 °CA intake valve opening duration. 1D gas exchange simulations performed with GT-Power confirm this statement, With Late Miller, the bock flow happens with air at higher temperature.

Table 2. Results on A25 speed-load point.

A25 – 1200rpm 438Nm	SNOx %	AVL439 soot %	BSFC %	Temperature after turbine °C
Reference 160°CA inlet valve opening duration	Reference	Reference	Reference	315
Early Miller 90°CA inlet valve opening duration	- 5	- 54	+ 6	419
Late Miller 240°CA inlet valve opening duration	- 28	- 59	+ 4	395

Therefore this back flow air has a lower density than the potential fresh air that does not enter in the cylinder in case of Early Miller. This leads to on air mass pushed back in intake manifold lower than the one non admitted in case of Early Miller. This statement is enhanced once calculating the volumetric efficiency including EGR flow because with Late Miller, both EGR flow and volumetric efficiency w/o EGR (fresh air only) increase. Finally, Miller settings bring higher exhaust temperature which is very interesting on low load points where SCR efficiency is not optimal. There is more than 100 °C temperature often turbine increase in case of Early Miller, and 80° increase for Late Miller. This is sufficient to go above the 270 °C threshold which is the minimum temperature required for an effective SCR behaviour (Figure 11). This increase in exhaust temperature is due to later combustion and lower thermodynamic efficiency.

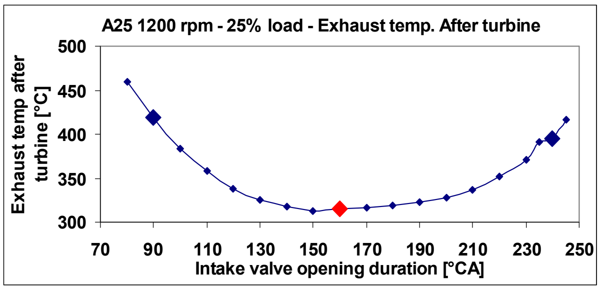

Figure 11. Intake valve duration impact on temperature after turbine at A25.

Influence of Miller cycle on all 25% load points

The same study has been carried out on B25 and C25 speed load points with success. Table 3 gives the benefits on B25 while Table 4 gives the benefits on C25. On B25, the optimum setting is obtained with late Miller which brings a 33% NOx decrease and 45% Soot decrease in comparison to the reference. However, there is a 4% penalty in consumption. On C25, two optimum setting are equivalent, the first one with early Miller which brings 8% NOx decrease and 98% Soot decrease in comparison to the reference with 3% BSFC penalty. The second one with late Miller that brings 88% Soot decrease at roughly same NOx in comparison to the reference with a 2% BSFC penalty. This very good results on C25 compared to B25 can be explained by the late timing used for the main injection on the reference setting and on the Miller settings (-3 "BTDC) to be compared to the early timing used on B25 (+4 "BTDC). Main injection timing retard plays a great role on ignition delay increase and therefore on pHCCl efficiency. This main injection delay compensates the fact there is less time at higher engine speed for the charge to become homogeneous.

Table 3. Results on B25 speed-load point.

B25 – 1500rpm 418Nm	SNOx %	AVL439 soot %	BSFC %	Temperature after turbine °C
Reference 160°CA inlet valve opening duration	Reference	Reference	Reference	301
Late Miller 240°CA inlet valve opening duration	- 33	- 45	+ 4	390

pHCCI further optimization and limits

With the optimum settings found for B25 speed- load point (Late Miller cycle with IVC=230°, effective Compression Ratio 12 instead of 16), the injection timing was varied from 10 °CA BTDC to -3 °CA BTDC (reference case at 3 °CA BTDC). Figure 12 presents the effect of this Main lnjection Timing swing on the Soot emissions. This figure shows that soot increases when injection timing is varied from 10 "BTDC to 4 "BTDC. Beyond this limit, the soot emissions are decreasing. This could be explained by the decrease in ignition delay at the beginning of the main timing swing followed by an increase of the ignition delay in the second part of the swing. The bigger is the ignition delay, the better the charge becomes homogeneous thus leading to less local rich area and lower soot emissions. lnjection is completely finished before combustion occurs with late Miller setting especially with late main timing. It's not the case for the reference combustion without late Miller. Figure 13 shows this phenomenon. It’s clear that with very early injection timing and Late Miller setting, the combustion starts just before the end of injection. With the reference setting without Miller (green line), the injection is not finished at all when the premixed combustion peak happens and therefore the diffusion flame peak is still noticeable. With the Late Miller setting and very late injection timing, it becomes obvious that the injection is completely finished when the premixed combustion starts. Figure 14 shows the effect on NOx of this Main Timing swing with late Miller setting. It can be stated that NOx are decreasing by delaying the Main lnjection timing until main timing is 2 °CA BTDC. Beyond, with a retarded main timing, NOx is increasing again.
This NOx increase can be explained by a combination of two factors:

- Increase in O2 concentration due to increased boost pressure and increase A/F due to more energy given to the turbine, thanks to late combustion.
- Decrease in EGR level thanks to decrease in difference between exhaust manifold pressure and intake manifold pressure.

Figure 15 shows this Main Injection Timing swing in NOxl Soot axis followed by EGR swing (turquoise line). With the EGR level increase, NOx are decreasing while the soot remains at very low values. These very low soot values can be explained by the big ignition delay that is increasing with increasing EGR. Furthermore, if combustion temperature under-runs the critical soot generation temperature, the soot formation stops. The decrease in NOx can be explained by decrease in O2 concentration and decrease in combustion temperature (Table 5). There 64% NOx decrease and 82% Soot decrease (+14% BSFC) after complete emission optimization. In all cases, it was not possible to define some settings with pHCCl combustion and BSFC and equivalent to the reference normal Diesel combustion. Once these huge benefits in NOx and Soot were achieved, the same process was carried out on B50. The target was to use several features to shorten the combustion and to increase ignition delay. An injection pressure increase has been performed (red line with diamond in Figure 16) from 750 bar amplified 2.2 times to 900 bar amplified 2.2 times (injection pressure from 1650 bar to 2000 bar). This injection pressure increase is followed by a main timing swing (dark blue line with diamonds) from 10 "BTDC to 5 "ATDC and finally an EGR swing (light blue line with crosses). On B50, it can be stated on Figure 16 that it cannot be possible to reach 'no soot1 no NOx" conditions even with high EGR level, late main timing and maximal injection pressure. The last EGR swing gives nevertheless a better NOx/Soot trade-off as the reference setting without Miller. Figure 17 gives the cylinder pressure trace, injection rate and Rate Of Heat Release of the last measurement point with late injection, highest injection pressure; high EGR and heavy Late Miller setting.

Table 4. Results on C25 speed-load point.

C25 – 1800rpm 358Nm	SNOx %	AVL439 soot %	BSFC %	Temperature after turbine °C
Reference 160°CA inlet valve opening duration	Reference	Reference	Reference	289
Early Miller 110°CA inlet valve opening duration	- 8	- 98	+ 3	365
Late Miller 230°CA inlet valve opening duration	- 3	- 88	+ 2	356

Table 5. Results on B25 speed-load point after pHCCI optimization.

B25 – 1500rpm 418Nm	SNOx %	AVL415S soot %	BSFC %	Temperature after turbine °C
Reference 160°CA inlet valve opening duration, main timing 2°BTDC	Reference	Reference	Reference	282
Late Miller 230°CA inlet valve opening duration after pHCCI optimization, main timing 3°ATDC	- 64	- 82	+ 14	393

It's quite clear on this figure that injection is not finished when the combustion starts, and therefore the combustion keeps a diffusion flame which was not the case on 25% load. This explains why "no Soot1 no NOx" conditions cannot be reached on B50 without other potential features like increasing EGR level or cooling
even more the EGR.

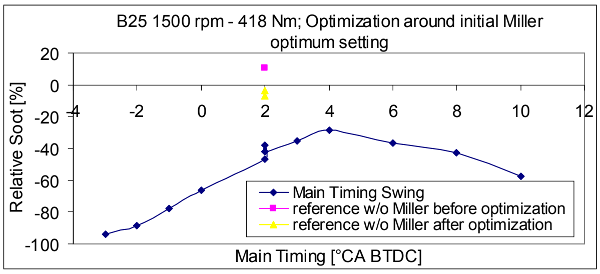

Figure 12. Effects of Main Timing swing on Soot emissions at B25 with late Miller setting.

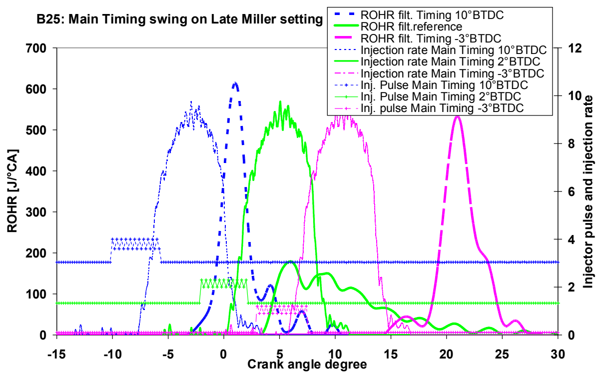

Figure 13. Relative position between end of injection and start of combustion at B25 with late Miller setting.

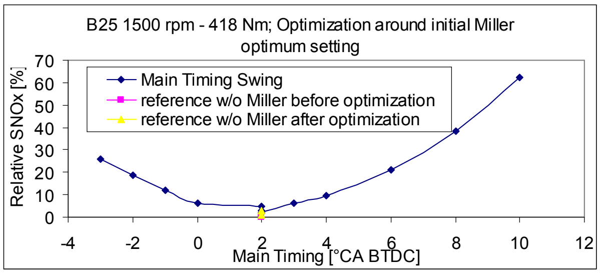

Figure 14. Effects of Main Timing swing on NOx emissions at B25 with late Miller setting.

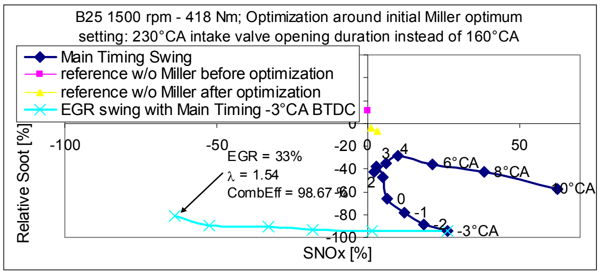

Figure 15. NOx/Soot trade-off at B25 with late Miller setting.

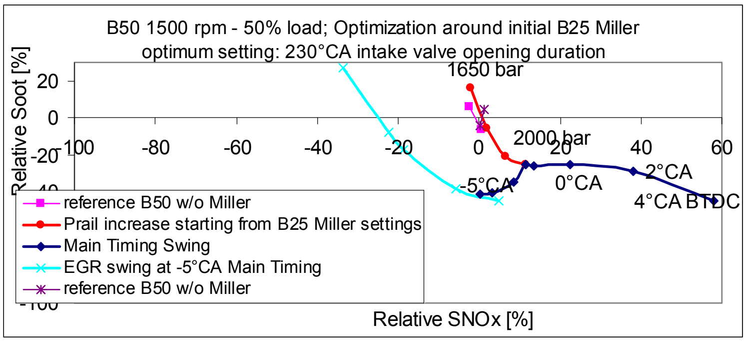

Figure 16. NOx/Soot trade-off at B50 with late Miller setting.

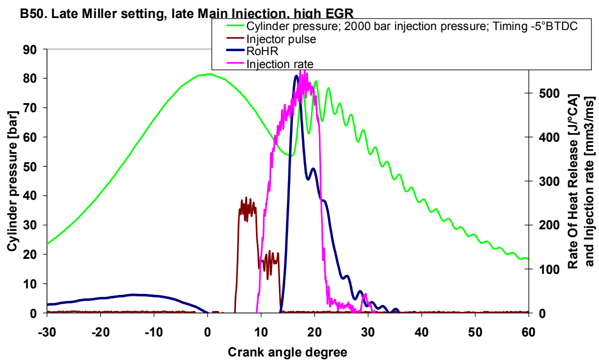

Figure 17. Cylinder pressure, Rate of Heat Rejection, injector pulse at B50 with late Miller setting.

HYDRAULIC VALVE ACTUATION SYSTEM FEATURES

SYSTEM ARCHITECTURE

The camless system is a patented electro-hydraulic system developed by Sturman Industries named HVA 4A with two-stage actuation [3]. The system is built with a specific oil circuit independent from the standard engine oil circuit (Figure 18). The two-stage actuation is run with three levels of oil pressure (100 to 210bar for power, around 35 bar for control and 1 bar for bock loop). The direct electronic control of the two digital valves (vent pilot valve and supply pilot valve) moves the 3-way proportional valve for engine valve opening, locking or closing events. The current system is developed with a separate hydraulic pump out of the engine. For the HVA system, a dedicated Engine Control Unit is used, Each valve motion is measured and controlled separately. Calibration is performed frequently to convert sensor signal to engine valve lift. During each valve event, the valve trace is recorded. Between two cycles, the signal is filtered, converted and then compared to the commanded parameters for opening and closing timing, lift height and landing velocity. Close loops are used with feedback/feedforward controllers to modify the commands to the digital valves on the next cycle.

ENGINE INSTALLATION

The HVA 4A system has been installed on an EGR 11 liter multi-cylinder engine. The geartrain has been reduced at the top to remove gears leading to the valvetrain. The cylinder head has been modified to prevent engine oil from being in contact with HVA oil. The HVA modules supporting the medium and high pressure circuits as well as the 24 independent actuators are screwed on a transfer plate located above the cylinder head. The valve springs rate and preload have been reduced in order to save power for valve opening. Additionally, it gives lower closing flanks as valve return is mechanical and not hydraulic.

SYSTEM PERFORMANCES

The HVA-4A system gives a high level of flexibility on valve motion:
Timing: opening and closing angles are fully variable providing valves will not interfere with pistons (Figure 19). An algorithm has been developed to prevent any contact between valve and piston for all engine speeds, Some timing restrictions can also take place for post-processing time between two cycles.

- Lift: valve lift is fully variable between 1,5 to 10 mm (Figure 20);
- Landing: Valve landing is set with knee height and landing velocity (in m/s) fully variable (Figure 20);
- Events: for each valve, two events can take place per cycle for internal EGR or brake applications. The secondary event has the same level of flexibility as the primary one (Figure 21);
- Intake valve opening: for air management purposes, intake valves can be opened below the piston before the top dead center. The valve lift is limited to 1 mm to prevent any contact with piston (Figure 22);
- Opening flank: the valve opening flank velocity is directly linked to the oil pressure in the high pressure circuit (Figure 23). Depending on engine speed, the opening flank can be sharp (low engine speed and very high oil pressure) or smooth (high engine speed and reduced high oil pressure);
- Valve-to-valve command: as each of the 24 valves is separately commanded, valve profiles can be easily modified between two intake or exhaust valves within the same cylinder, or between cylinders.

The system accuracy and repeatability have been measured on the engine, Accuracy is defined to be the difference between the mean value over 100 consecutive cycles and the commanded value. Repeatability is defined to be three times the standard deviation over the mean value:

- Opening angle: accuracy is +I-2 °CA and repeatability is 3 °CA.
- Valve lift: accuracy is +/- 0,2mm for command below 3,5mm and +I0,-5 mm for command above. Repeatability is 0,8mm for standard lift and 0,5mm for the 1 mm intake early opening plateau.
- Closing angle: accuracy is +I-3 °CA and repeatability is 3 °CA.

As far as opening (with standard high oil pressure level) and closing flank velocities are concerned, they are comparable to mechanical lift profiles at maximum engine speed.

SYSTEM POWER CONSUMPTION

The HVA-4A system is a first prototype which has not been fully integrated on the engine and whose power consumption has not been considered.
It is important to highlight that such system power consumption has to be compared with systems providing similar functionalities in terms of flexibility. Even if one can say that hydraulic valve actuation is more power consuming than traditional mechanical valvetrain without any flexibility, such combustion settings as previously described requires a certain level of flexibility where engine manufacturers need to find the optimal trade-off between combustion features, packaging, power, weight, cost,…

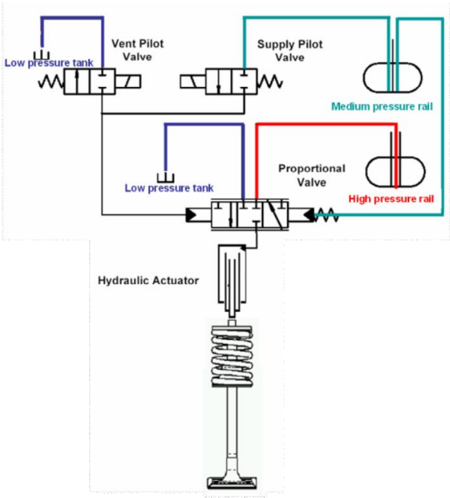

Figure 18. HVA 4A hydraulic scheme.

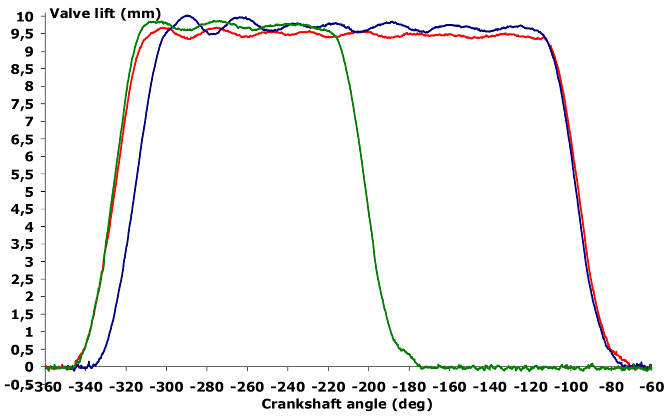

Figure 19. Intake and exhaust valve opening/closing flexibility.

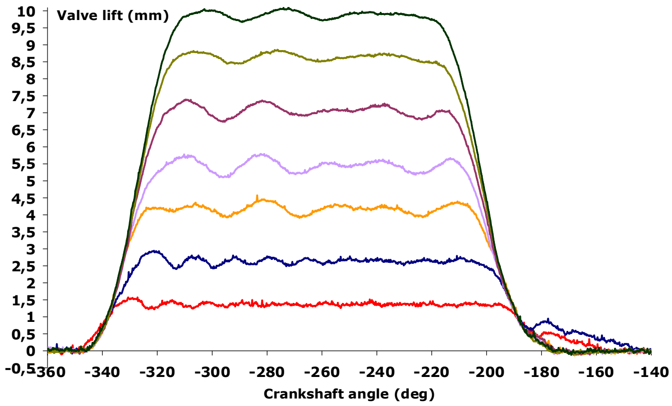

Figure 20. Intake and exhaust valve lift flexibility.

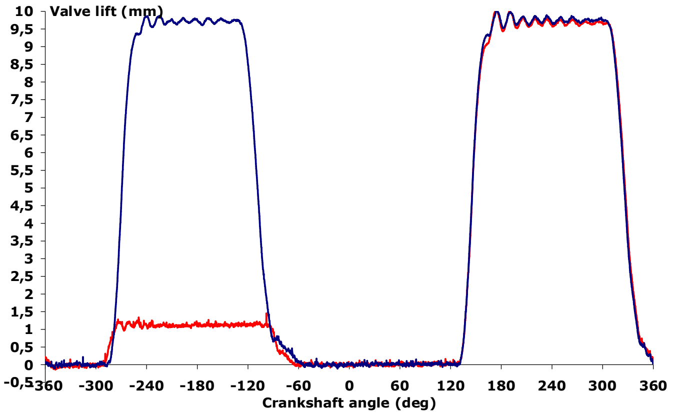

Figure 21. Intake and exhaust valve event flexibility.

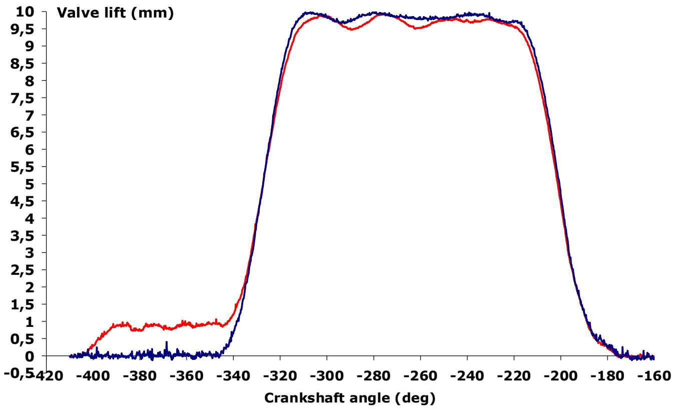

Figure 22. Intake valve opening flexibility.

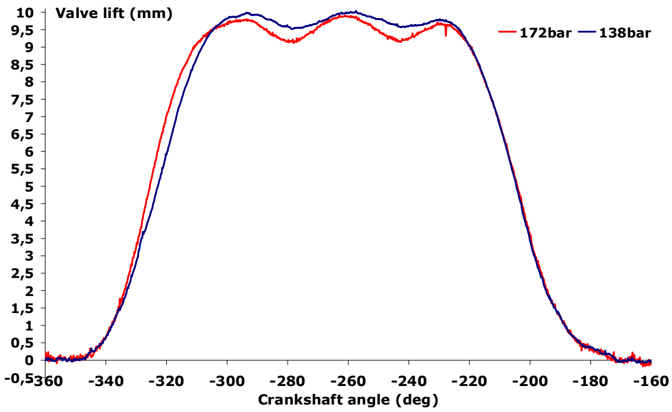

Figure 23. Intake and exhaust valve opening flank flexibility.

CONCLUSION

The combination of low effective compression ratio thanks to Miller effect, relatively high EGR level and operation close to stoichiometric conditions provides the possibility of simultaneous NOx and soot reduction while suffering a decline in engine efficiency and a rise in CO and HC. The combustion in these low NOx/Soot operating points is characterized as partly Homogeneous Compressed Combustion Ignition (pHCCI) It has been possible to run at 25% load in pHCCl combustion with premixed combustion only and without diffusion flame. The simultaneous soot and NOx reduction seems to be a combination of good premixing, due to longer ignition delay, and low local combustion temperature due to lack of oxygen. This pHCCl combustion is also very interesting for after treatment systems and especially SCR because temperature after turbine is drastically increased which improves SCR efficiency on low load points. Such "No Soot / No NOx" conditions cannot be reached on 50% load even if NOx-Soot trade-off is improved once comparing with the normal reference Diesel combustion without Miller.

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Glossary

°CA, crank angle degree
AF, Air Fuel ratlo
ATDC, After top dead centre
AU, Arbitrary Unlt
BSFC, Brake Speclflc Fuel Consumptlon
BTDC, Before top dead centre
CO, Carbon Monoxlde
EGR, Exhaust gas reclrculatlon
ESC, European Statlonary Cycle
HC, Hydrocarbons
Hp, British Horsepower = 0,746 kW
HP, High Pressure
HVA, Hydraulic Valve Actuatlon
IVC, Intake Valve Closing in crank angle degree
IVO, Intake Valve Opening in crank angle degree
NOx, Nitrogen Oxides
pHCCl, Partly Homogeneous Compressed Combustion lgnition
= PCI, Premixed Compression lgnition
= PPC, Partially Premixed Combustion
SCR, Selective Catalytic Reduction
SNOx, Brake specific NOx
TDC, Top dead Centre
VGT, Variable geometry turbine turbocharger