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FPGA Controlled Pneumatic Variable Valve Actuation

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ABSTRACT

A control system for pneumatic variable valve actuation has been designed, implemented and tested in a single cylinder test engine with valve actuators provided by Cargine Engineering AB. The design goal for the valve control system was to achieve valve lifts between 2 and 12 mm over an engine speed interval of 300 to 2500 rpm. The control system was developed using LabView and implemented on the PCI 7831. The design goals were fulfilled with some limitations. Due to physical limitations in the actuators, stable operation with valve lifts below 2.6 mm were not possible. During the engine testing the valve lift was limited to 7 mm to guarantee piston clearance. Different valve strategies for residual gas HCCI combustion were generated on a single-cylinder test engine.

INTRODUCTION

Valve lift, timing and duration have generally fixed values for conventional valvetrains. These fixed values are usually optimized for the engine speed range most frequently used. They depend on what purpose the engine is made for and represent a compromise between stabile idle running and high engine speed performance. The ideal solution is to fully control when and how the valves should open and close. With such degrees of freedom one would be able to optimize the gas exchange for all operating conditions.

One way to achieve variable valve control, VVC, is through the use of pressurized air. A Swedish company, Cargine Engineering AB, has developed a pneumatic valve actuation system that offers fully variable valve control (see Appendix for physical dimensions). Pneumatic valve actuation could be easily implemented on heavy-duty vehicles since they have an existing system for pressurized air. On passenger cars it would require the addition of a compressor

This paper describes a control system, developed at Lund Institute of Technology, for the abovementioned VVC system. Test results are presented from a test rig consisting of the VVC system installed in a cylinder head with valves but also from actual engine operation with the VVC system. The objective is to offer a valve lift from 2 to 12 mm and a duration between 0 and 360 Crank Angle Degrees (CAD) over a range of engine speeds between 300 and 2500 rpm. A cylinder head from a Scania 12 liter engine with four valves was used for the test rig. The valves have to be controlled independently from each other with the possibility to deactivate them if desired.

After the evaluation of the program the whole system was implemented on a single-cylinder test engine for further testing. The engine testing consists of HCCI (Homogeneous Charge Compression Ignition) operation with three different valve strategies – Negative Valve overlap, Late/Early IVC and Rebreathe. The valve lift height will be limited to 7 mm to guarantee clearance from the moving piston.

PNEUMATIC VARIABLE VALVE ACTUATION

The system developed and delivered by Cargine Engineering AB consists of an actuator, two solenoids[1], an actuator piston and logical channels inside the housing. The actuator itself has to be seen as a “black box” due to company secrecy, but the concept will be described for better understanding.

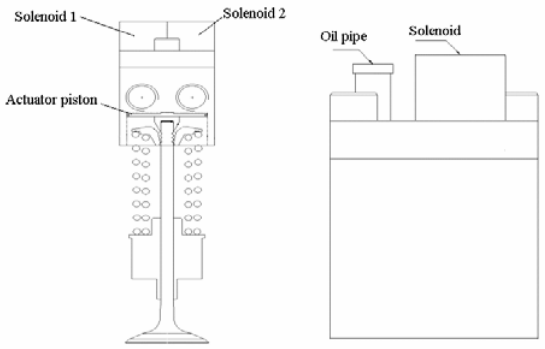

Figure 1 Sketch of the pneumatic valve actuator [2].

The right circle in Figure 1 is an intake port. Here the pressurized air should be connected. The left circle is the hole for outgoing air, which is released into

[1] The solenoids are Bicron SP1913P2410

theatmosphere. When Solenoid 1 is activated pressurized air can enter the actuator, Solenoid 2 stops the filling. Each actuator requires one or two electrical signals, depending on if one or two solenoids are used. This means that it is possible to operate the actuator with only Solenoid 1 activated and then the lift is only dependent on the pressure supplied. By the use of both solenoids instead of one, it is possible to vary the lift at a given pressure. In this investigation a pressure of 2.5 bar has been used all the time, since this is sufficient for the entire valve lift interval from 2 to 12 mm.

The actuator is equipped with a hydraulic brake, whose function is to slow down the valve before seating. At the top of the valve, there is a valve stem cap. Its function is to make it possible to adjust the spacing between the actuator piston and the actuator itself. The adjustment is made with shims, which are simply very thin plates with various thicknesses. These are put between the top of the valve and the valve stem cap as shown in Figure 2.

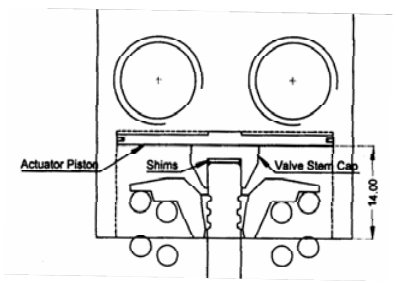

Figure 2 A close up of the pneumatic valve actuator [2].

If the clearance is too large, the hydraulic brake will not be able to slow down the valve enough, which leads to loud noise as the valve hits the seat. With insufficient clearance the pressurized air will not have full access to the actuator piston and the force will be insufficient to lift the valve.

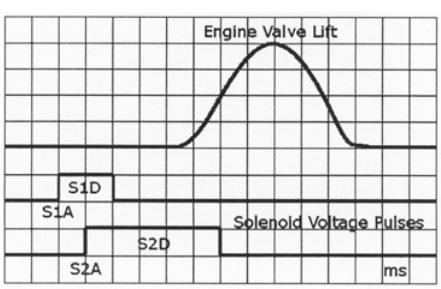

Figure 3 Schematic picture of solenoid voltage pulses and engine valve lift [2].

Figure 3 shows a schematically how the solenoid activation relates to the valve lift event..

Step 1 – Activation of Solenoid 1 (S1A) will open the inlet valve to the actuator[2] and thereby determine the starting point of the engine valve opening. The length of the activation time (S1D) determines the valve opening duration. Another function of Solenoid 1 is that when activated it turns on a hydraulic locking-mechanism that holds the valve at the desired lift where it dwells until the solenoid is deactivated.

Step 2 – Activation of Solenoid 2 (S2A) will close the inlet valve to the actuator. Hence the time difference between the two signals (S2A-S1A) will determine the engine valve lift height. S2D will affect neither the lift nor duration. However if S2D is chosen to end before S1D, this will result in another filling, which then leads to an incomplete closure of the valve. To avoid this, it is recommended to choose S2D to be as long as S1D.

Step 3 – The hydraulic brake will begin to slow down the valve about 3.0 mm from the end position during valve closing. In the interval 1.0 to 0.0 mm there is a ramp function, which means that the seating velocity is constant in that interval. The magnitude of this constant velocity is approximately 0.5 m/s according to the manufacturers. However this may change with different valve springs due to the precompression of the valve spring.

Figure 4 shows the actuators mounted on the test rig. Also the solenoids and the related hoses can be seen. Figure 5 shows the single cylinder test engine.

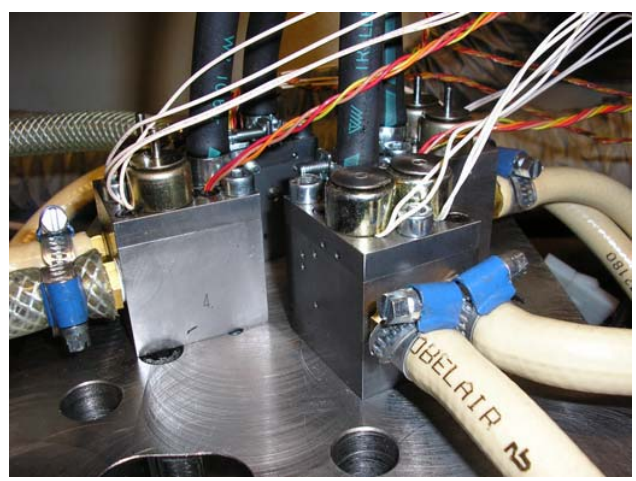

Figure 4 Valve test rig consisting of the actuators attached to a plate on the Scania D12 cylinder head.

[2] There is an inlet valve inside the actuator. This should not be mistaken for the engine inlet valve.

Figure 5 The Scania D12 single-cylinder test engine used in the study.

THE PROGRAM FOR VARIABLE VALVE CONTROL

INTRODUCTION

The valve control system has been developed in LabVIEW together with an FPGA device, both developed by National Instruments.

FPGA – Field Programmable Gate Array

A FPGA is a chip that consists of many unconfigured logic gates. It is possible to configure and reconfigure the FPGA for each application, unlike other chips with fixed functionality. This is a big advantage, since designing and manufacturing a new chip every time the application changes may be very costly.

The FPGA offers benefits such as precise timing, rapid decision making with loop rates up to 40 MHz and simultaneous execution of parallel tasks. FPGAs appear for example in devices such as consumer electronics, automobiles, aircraft and copy machines [3].

In this project a NI PCI-7831R (see Figure 6) device has been used. This device has 8 analog inputs, 8 analog outputs and 96 digital I/O [4].

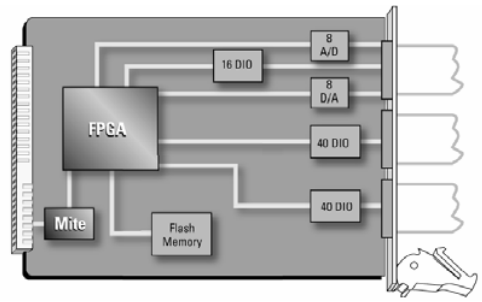

Figure 6 A schematic drawing of a NI PCI-7831R device [3].

THE PROGRAM

LabVIEW is an entirely graphical programming language. Instead of conventional programming, the user programs the algorithms with pre-programmed blocks. Any complete functional program made in LabVIEW is called a virtual instrument and is almost always referred to as a VI.

A LabVIEW VI consists of two “faces”. These are the front panel and the block diagram. The front panel is the face that the user of the system works with when executing his program. Here the user can change input data, read output data and monitor graphs. The block diagram is almost the backside of the front panel and can be compared with for example the inside of an oscilloscope.

When combining FPGA with LabVIEW there is a certain development flow that should be followed, see Figure 7. Therefore the main program consists of two modules, one that will be compiled to the FPGA and one that will control the FPGA program and is called the Host VI. The FPGA should only be used for the parts that require fast operations.

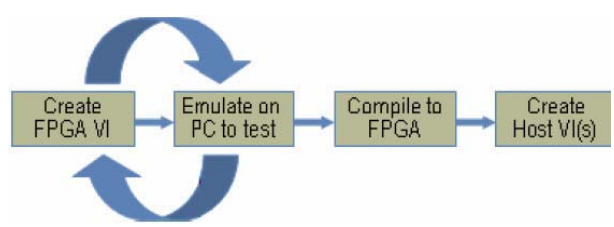

Figure 7 Application development flow [3].

The programs for each valve are quite similar with only minor differences and therefore the focus will be on describing the program for one inlet valve.

Host VI – Front panel

At the front panel the user has the possibility to choose engine speed, valve opening time, duration and valve lift, as seen in Figure 8. This can be done at any time, even while executing the program. There are some limitations in the program due to physical and programming constraints, and therefore there are indicators that for example indicate the smallest possible duration etc. Their task is to make the user aware of the limitations in the program and thus prevent the user from choosing unsuitable values.

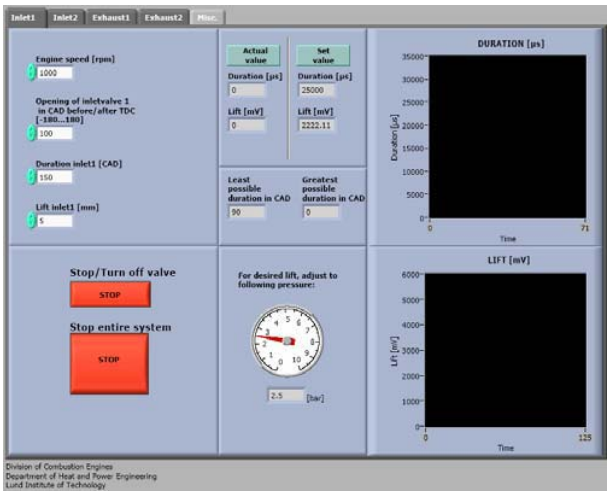

Figure 8 Front panel.

The upper part in the middle of the front panel contains four indicators that show the actual and the set values for duration and valve lift. Below them are two additional indicators. They indicate smallest and greatest possible duration. It is very important to stay in the interval suggested by the indicators, otherwise the system may not function as expected. The front panel is also equipped with a pressure gauge. It shows the pressure needed for the chosen valve lift. The right part of the front panel consists of two graph windows. They offer the user a possibility to monitor the cycle to cycle variation of the duration and valve lift as well as step responses when changes are made. At the top of the front panel there are 5 tabs, one for each valve and one miscellaneous tab where the user can change the integral constants and monitor the solenoid 1 activation duration, S1D, and the solenoid activation time difference, S2A-S1A. This tab should only be used by an advanced user that fully understands the system.

Host VI – Block diagram

As mentioned before the block diagram is the part of the VI where all the programming is done.

Figure 9 shows the block diagram of the host VI and will be described briefly below

Nbr 1 - Because the Host VI has to be connected to the FPGA VI, the following four functions are necessary. From the left there is a function that opens a specified FPGA VI, which means that it is activated for use in the Host VI. The following function invokes the FPGA VI and runs it. To be able to change input values to the FPGA VI, there is a need for another function called *Read/Write Control.* This control gives access to all controls and indicators on the FPGA VI. The left side is used for changing the controls while the right side is used for reading the indicators on the FPGA VI.

The FPGA works independently from the computer to avoid any possible disturbance. This means that even if the Windows operating system crashes, the FPGA will still run as long as it has power. Therefore there is a need for a final function that will stop the FPGA VI from running when wanted.

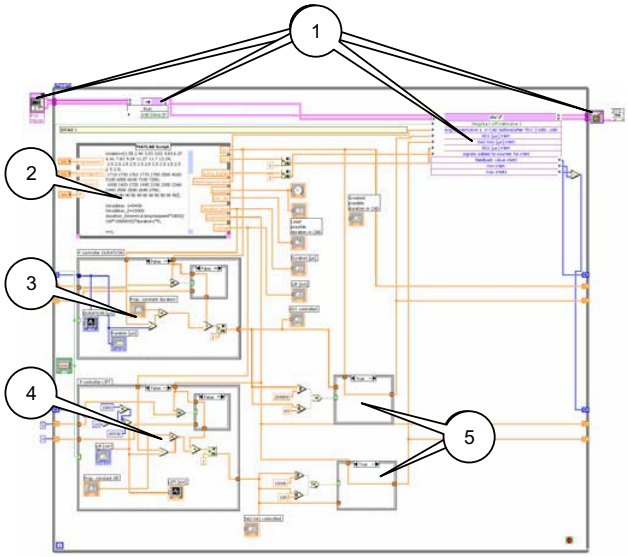

Figure 9 Block diagram for inlet valve 1¨

Nbr 2 - This section is called a *Matlab script* and as the name implies, it is used for connecting the Host VI to Matlab. Using Matlab in LabVIEW will simplify some calculations such as linear interpolation which has been used here.

The user sets the input values in the front panel, which are then processed in the Matlab script and calculated values are then passed on to other sections in the Host VI.

Nbr 3 - The Integral controller is a very important part of the system. It makes sure the actual valve lift duration is heading towards the set value. Without this, one can not be sure that the actual duration will be as desired. The following formula describes the integral controller.

$$U = \sum_{cycles} K(S - A)$$

where *U = control signal*

K = integral constant

S = set value

A = actual value

The proportionality constant, K, is not a pre-defined parameter, which means that the user can use a value that best suits the application in question. An increasing K will give a faster controller to a certain point. Beyond this point it will make the system unstable. The controller in Section 3 regulates the duration. It adds U to S1D and thus introduces feedback. As long as there is a remaining error, U will keep growing in the direction opposing the error.

Nbr 4 - The lift also needs to be regulated. This is done by another integral controller that works in the same way as the previous one. The only difference is that U will instead be added to S2A-S1A.

Nbr 5 - The purpose of this section is to avoid absurd controller values. Such values can occur if for instance, the pressure is too low and insufficient for the desired lift or duration. Then U keeps growing out of bounds which is called integrator wind-up. Therefore there are intervals that only allow reasonable values.

FPGA VI – Block diagram

Nbr 6 - This part of the program begins with a so-called sequence. It looks like the frames of an old-fashioned film. The sequence function is used when the programmer wants to be sure that the algorithm runs in the right order.

The first frame in the sequence consists of a TDC detector. When TDC is detected, the program continues at the next frame.

Nbr 7 - This control makes it possible to turn off and on the valve during operation.

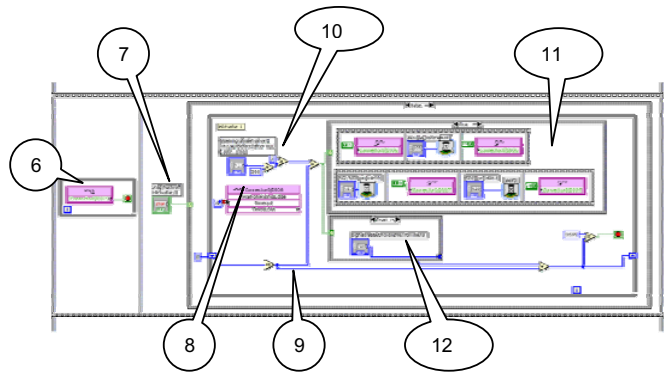

Figure 10 Controlling the solenoids.

Nbr 8 - An engine is assumed to be equipped with a crank pulse generator that provides 3600 pulses per cycle. To be able to open the valve at a desired crank angle the program needs a counter to count the pulses. LabVIEW has a function that makes the program wait until a rising edge of a digital signal is detected.

Nbr 9 - Every rising edge will be added to a counter. The sequence ends when the counter reaches 3595 pulses[3].

Nbr 10 - When the user sets the value for desired opening angle of the valve, the value is converted from CAD to a number of pulses. When the counter reaches the same value as the one that the user has set, the program will continue to Section 11.

Nbr 11 - Each solenoid is connected to a digital output. The first thing that will happen in Section 11 is that Solenoid 1 will be activated for a certain time. As mentioned before, this time will determine the duration of the valve actuation. While Solenoid 1 is activated, a wait function will delay the activation of Solenoid 2. This delay has also been mentioned before and it determines the valve lift.

Nbr 12 - While Section 11 is executing, the counter will stop counting. This means that when Section 11 ends and the counter continues counting the pulses, it will not take count of all the pulses that actually have been generated during this period of time. To avoid miscalculation of pulses, an extra algorithm has been made. It calculates the duration of Section 11, and with the help of the actual engine speed it converts this into pulses and in turn adds them to the counter.

EVALUATION OF THE PROGRAM

The purpose with this section is to show some typical curves obtained from an oscilloscope while executing

[3] The reason for using 3595 instead of 3600 pulses is to have a safety margin of 1 CAD which equals 5 pulses. This will avoid missing the TDC due to any lag of the program.

the program. The testing is performed on the test rig consisting of a Scania D12 cylinder head with the pneumatic valve actuators installed instead of the normal pushrods and rocker arms. Valve lift information is provided by lift sensors in the actuators. The sensors are optical and measure the amount of reflected light which decreases with valve lift. The voltage output from the sensor is highly nonlinear and has been linearized and calibrated to provide the correct lift in millmeters. The figures below show the behavior of one of the inlet valves during operation. The remaining valves show a similar pattern. Figure 11, Figure 12 and Figure 15 were obtained at an equivalent engine speed of 1000 rpm.

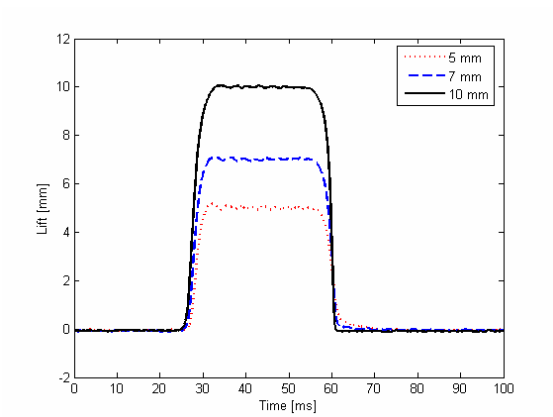

Figure 11 Three different valve lifts at a constant opening duration of 200 CAD.

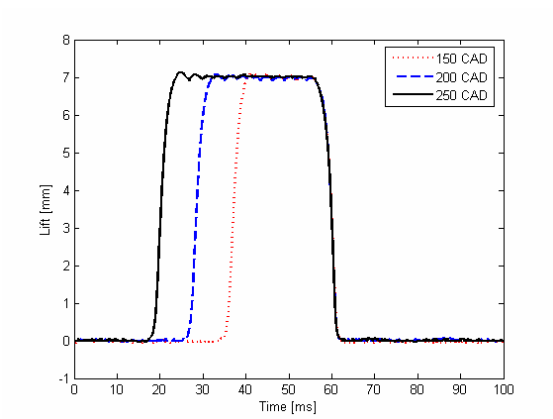

Figure 12 Three different opening durations at a constant valve lift of 7 mm.

Figure 11 shows that the duration remains constant when the valve lift is varied. At the valve closure, there is a slight difference between the valve lifts. This difference occurs because of the hydraulic brake. It seems like the braking is somewhat too effective at low valve lifts and the valve closure gets extended compared to at higher valve lift. The reason is that a higher valve lift has a higher returning velocity. Figure 12 shows how the lift remains unchanged for different opening durations.

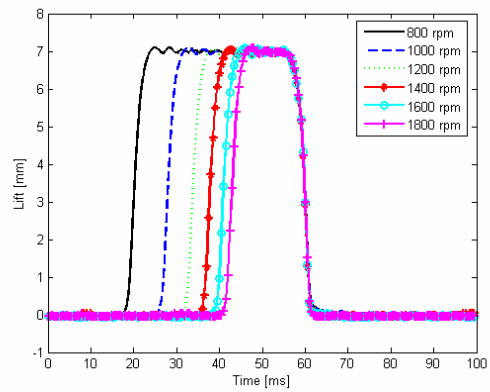

Figure 13 Six different engine speeds at a constant valve lift of 7 mm and a constant opening duration of 200 CAD.

Figure 13 shows how the valve lift is unaffected by the change of engine speed. The duration also remains unchanged, but this can't be seen in the figure. The reason for this is that duration in time varies with different engine speeds while duration in CAD remains the same.

Figure 14 shows how the valve lift remains stable at an engine speed of 2500 rpm, which was the upper limit, stated in the design goals.

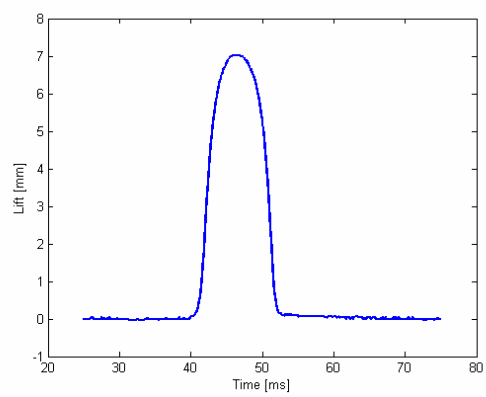

Figure 14 Valve lift of 7 mm at an opening duration of 200 CAD and an engine speed of 2500 rpm

The valve lift remains stable all the way down to 3 mm throughout the whole duration and engine speed interval. There exists a physical lower level limitation in valve lift, which is 2.6 mm. Below this value the system doesn't behave as expected and the lift tends to be very unstable. It is recommended to use 3 mm as a lower limit, thus providing a safety margin and avoiding any possible instability. Figure 15 shows a stable valve lift at the recommended minimum level.

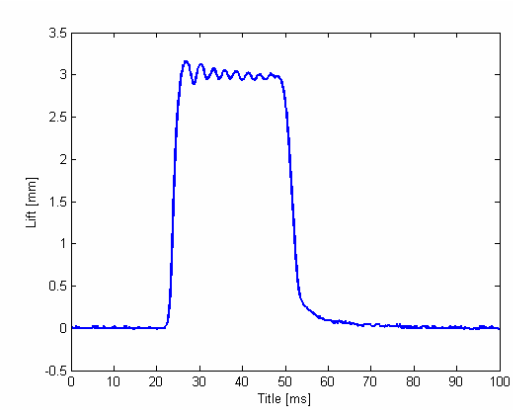

Figure 15 Valve lift of 3 mm at an engine speed of 1000 rpm and a valve opening duration of 200 CAD.

Figure 16, Figure 17, Figure 18 and Figure 19 show the step response when changing duration and valve lift. When changing the duration from a low to a high value, an immediate change is seen and then a last adjustment is made to reach the desired level. The opposite, i.e. changing from a high to a low value, also has an immediate effect on the duration. Now the value is set a little too low by the program, but the integral controller will adjust the error and eventually the desired level will be reached, as seen in Figure 16 and Figure 17. The same line of argument can be applied to Figure 18 and Figure 19, which show the step responses when changing valve lift.

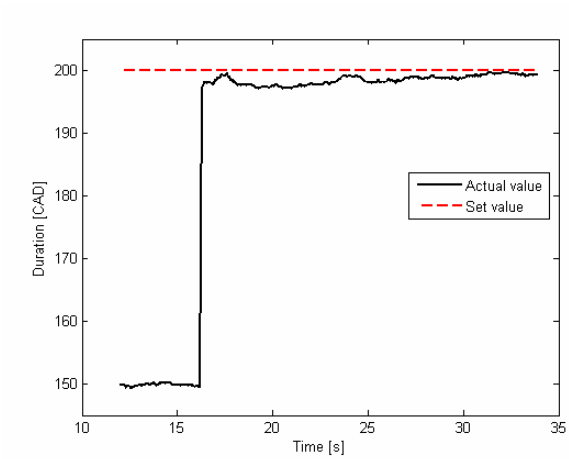

Figure 16 Step response when changing valve opening duration from 150 to 200 CAD.

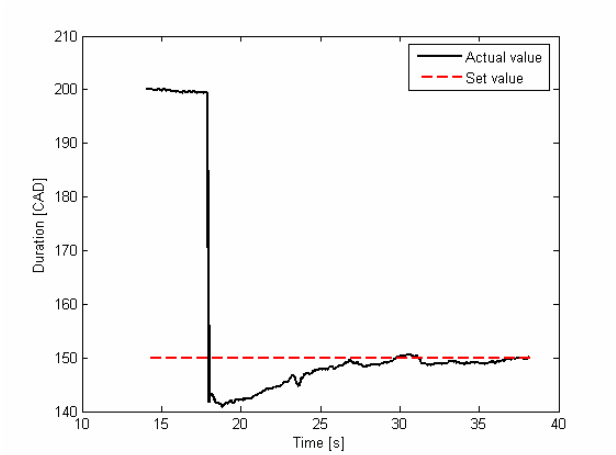

Figure 17 Step response when changing valve opening duration from 200 to 150 CAD.

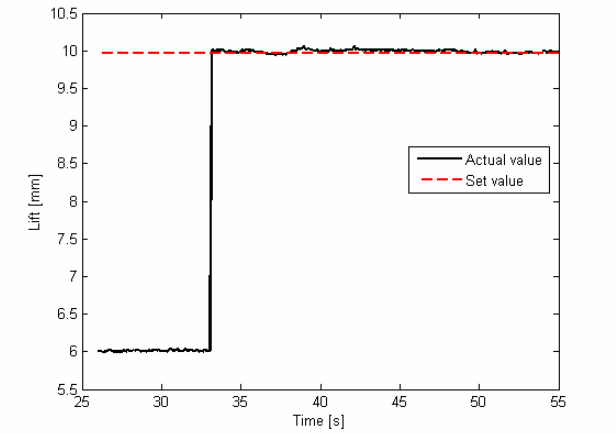

Figure 18 Step response when changing valve lift from 6 to 10 mm.

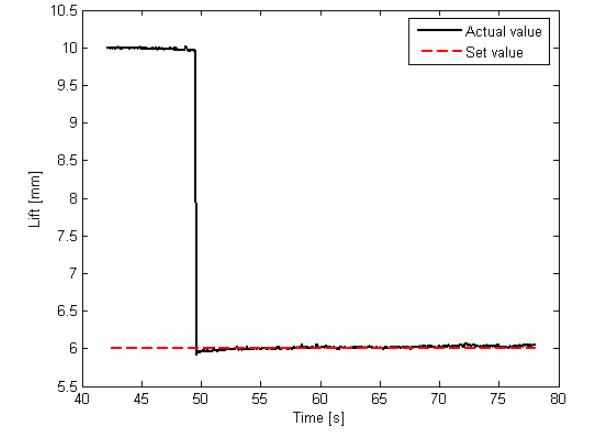

Figure 19 Step response when changing valve lift from 10 to 6 mm.

Figure 20 and Figure 21 show the cycle to cycle variation for duration and lift. It seems a little more dramatic than it actually is. The difference in percentage between the set value and the actual value is very small.

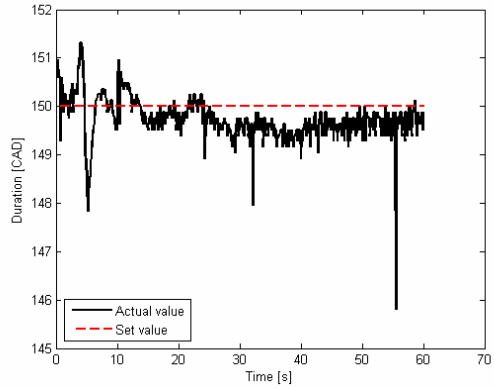

Figure 20 Cycle to cycle variations at a valve opening duration of 200 CAD.

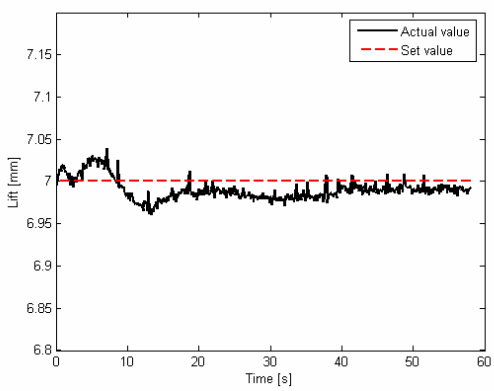

Figure 21 Cycle to cycle variation at a valve lift of 7 mm.

This study does not include any economic calculations, still it can be of great interest to investigate the consumption of pressurized air since this gives an indication how energy consuming the valve mechanism is. A simple rotameter was integrated into the system to measure the outgoing air flow from one actuator. The measurements were done at an ambient temperature of 20 degrees Celsius. It is assumed that the compressor uses 1 hp (736 W) to supply an air flow of 100 l_n/minute. This corresponds to a compressor efficiency of 40%. The power consumption of the compressor is subsequently used to compute a mean effective pressure, ValveMEP, which allows a comparison with the brake mean effective pressure produced by the engine.

$$ValveMEP = \frac{2P_{compressor}}{NV_d}$$

where N is the engine speed, V_d is the engine displacement volume and

$$P_{compressor} = \frac{Airflow}{100} * 736\ \mathrm{W}$$

Figure 22 shows how the valve lift affects ValveMEP at a constant opening duration and engine speed. Figure 23 shows ValveMEP as a function of engine speed at constant opening duration and valve lift. Both measurements are compared with simulations done at the same conditions by Cargine Engineering AB.

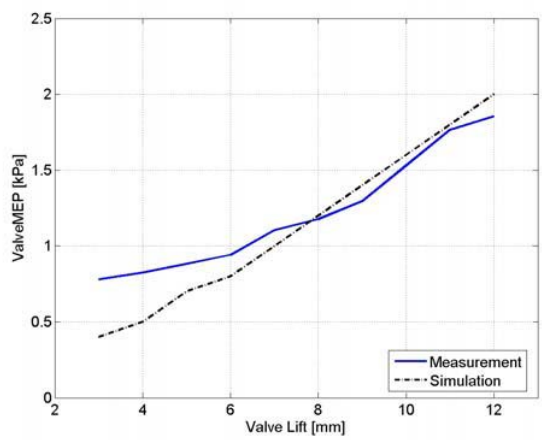

Figure 22 Valve mean effective pressure (single valve) as a function of valve lift at duration of 200 CAD and engine speed of 1000 rpm.

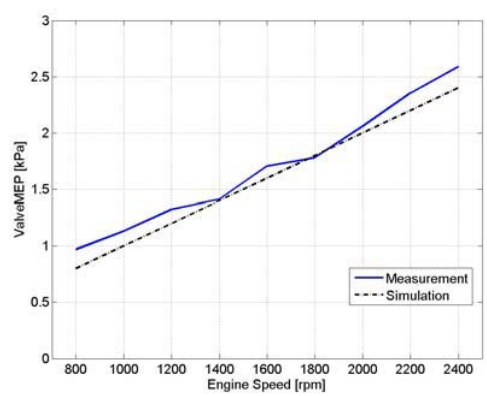

Figure 23 Valve mean effective pressure (single valve) as a function of engine speed at a duration of 200 CAD and a valve lift of 7 mm.

ENGINE TESTING AND EVALUATION

The system has been implemented on a single-cylinder test engine to investigate if it is functioning as expected in a real application. The engine used is a Scania D12

engine where cylinders 1-5 have been deactivated and only cylinder 6 is operating. The operating parameters for the tests can be seen in Table 1

Table 1 Operating parameters

Actuator intake pressure	3 bar
Hydraulic brake pressure	4 bar
Valve spring constant	11.5 N/mm
Valve spring precompression Maximum valve lift	100 N 7 mm
Compression ratio	18:01
Engine speed	1200 rpm
Fuel	Isooctane
Fuel energy per cycle	0.75 KJ
Displacement volume	1.96 l/cylinder

Each valve strategy began with valve timings according to Table 2.

Table 2 Initial valve timing

IVO	0 CAD ATDC
IVC	180 CAD ATDC
EVO	0 CAD ABDC
EVC	180 CAD ABDC

A new and promising combustion concept is HCCI (Homogeneous Charge Compression Ignition), which promises high efficiency, and low NO_x. Retained or rebreathed residual / exhaust gas can be used to increase the charge temperature and achieve HCCI combustion. Below, the variable valve system is used to enable these strategies. Only low loads have been tested in this study. The maximum load during the tests was 2.2 bar net indicated mean effective pressure (IMEPn) which corresponds to about 35% of the maximum achievable load with HCCI and naturally aspirated conditions. It is however only about 10% of the maximum load achievable with turbocharged Diesel operation.

NEGATIVE VALVE OVERLAP

With negative valve overlap, NVO, the amount of trapped residual gases can be changed and can be seen as “internal EGR”. The basic idea with NVO is that the exhaust valve should close early and the intake valve should open late which leads to a higher fraction of residual gas. Figure 24 shows how the exhaust valve closing and the inlet valve opening vary with NVO. It can also be seen that with increasing NVO the pressure during the gas exchange increases. Observe that the right valve lift is the exhaust valve and the left valve lift is the intake valve.

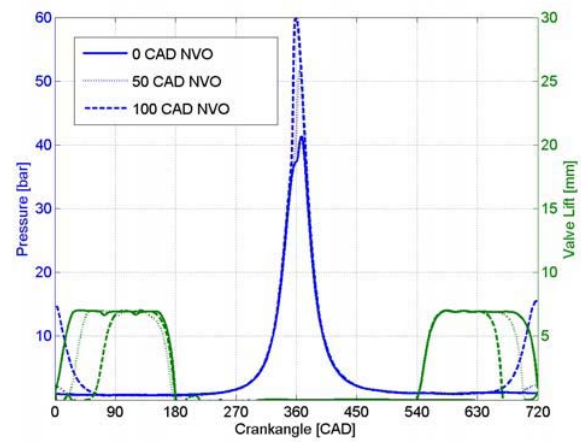

Figure 24 An illustration of how the NVO works.

As mentioned the amount of residual gas increases with increasing NVO. With more residual gas the in-cylinder temperature increases and therefore the combustion will start progressively earlier.

Figure 25 shows how CA50 varies with NVO. CA50 is the crank angle where 50% of the energy from combustion has been released. This parameter is very useful combustion timing indicator.

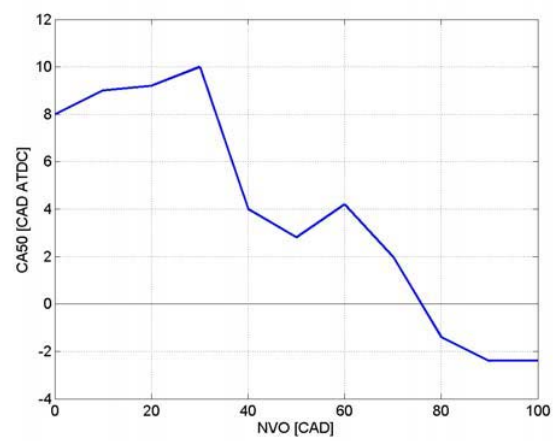

Figure 25 CA50 as a function of NVO

It can also be seen that there is an insufficient charge temperature for NVO less than 30 CAD and therefore the combustion is delayed. The explanation for this behavior is probably that the combustion is very poor. When NVO is increased beyond 30 CAD, the combustion improves and combustion timing advances with increasing NVO.

In Figure 26 it is clearly visible that the combustion efficiency is very bad between 0 and 40 CAD NVO. Both IMEPg (gross indicated mean effective pressure) and IMEPn (net indicated mean effective pressure) show low values in this interval.

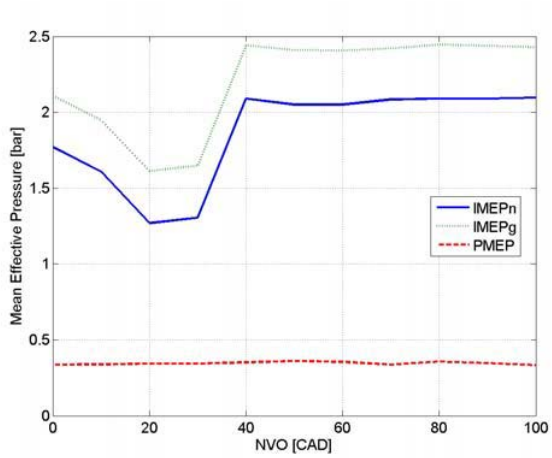

Figure 26 IMEP and PMEP as a function of NVO

As the combustion improves, IMEPg and IMEPn increase and after 40 CAD of NVO they are almost constant at a value of about 2.4 bar and 2.1 bar respectively. The Pumping Mean Effective Pressure (PMEP) seems to be constant throughout the whole interval.

NVO displaces some of the air in the combustion chamber and thus the air/fuel ratio should decrease. Figure 27 shows exactly this behavior.

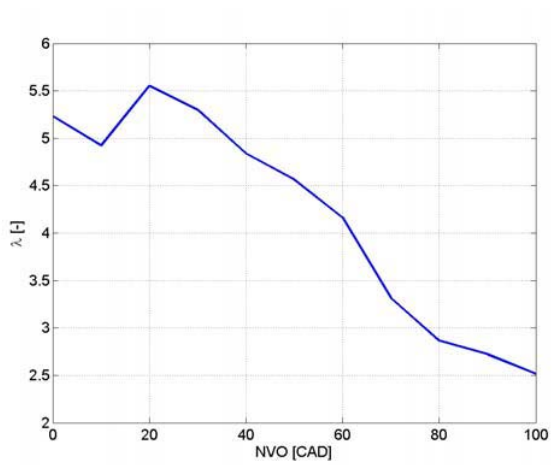

Figure 27 Lambda as a function of NVO

The poor combustion in the interval from 0 to 40 CAD NVO can once again be seen in Figure 28. The hydrocarbon levels in this interval are fairly high compared to when the combustion is more effective.

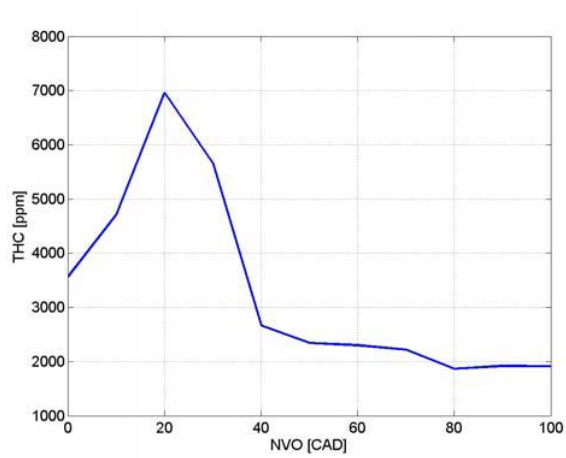

Figure 28 Total Hydrocarbon emission (THC) as a function of NVO

Better combustion due to higher temperature and lower air/fuel ratio leads to lower hydrocarbon emissions. Combustion of some of hydrocarbons in the retained burned gas also contributes.

NOx and NO are below measurable levels (<1 ppm) and therefore they are not considered in this paper. This also applies to the rest of the valve strategies.

REBREATHE STRATEGY

The idea with the rebreathe strategy is to open the exhaust valve a second time a short while after it has closed the first time which can be seen in Figure 29. It is very similar compared to NVO. The amount of rebreathed exhaust gas increases with increasing duration of the rebreathe opening and the difference is that with the rebreathe strategy the burned gas is not compressed during the gas exchange.

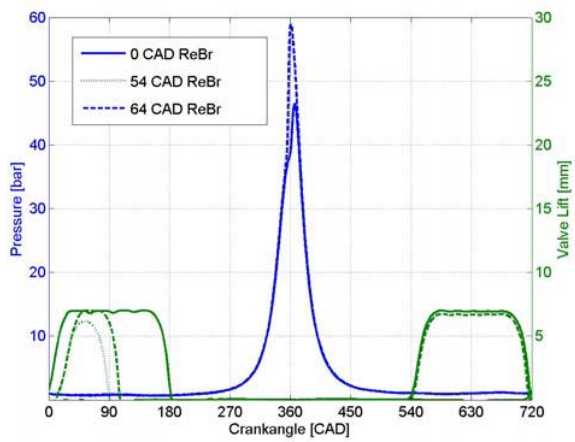

Figure 29 An illustration of how the rebreathe strategy works. The dashed and dotted valve profiles in the 0-90 CAD interval represent the rebreathe opening of the exhaust valves

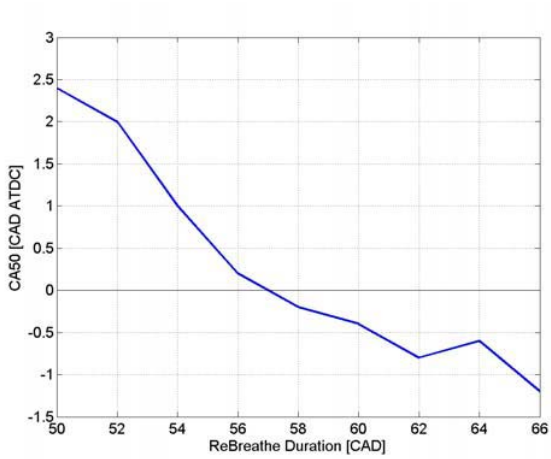

Figure 30 CA50 as a function of rebreathe duration

The behaviour of combustion timing in Figure 30 with respect to rebreathe duration is as expected. Both the decrease in air/fuel ratio and the increase in charge temperature contribute.

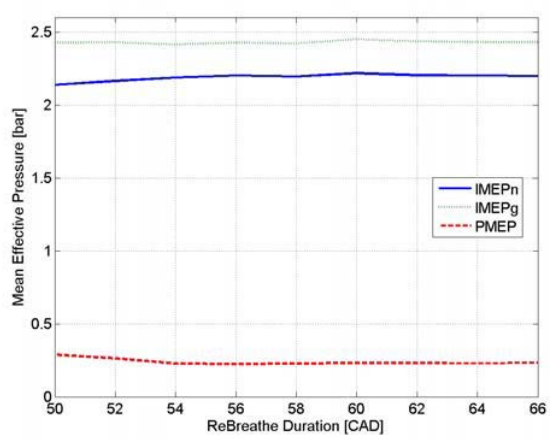

Figure 31 IMEP and PMEP as functions of the rebreathe duration

The IMEPg in Figure 31 shows that the combustion is good since it is essentially constant throughout the whole interval. It should also be noted that the pumping work is lower with rebreathe than with negative valve overlap since there is no recompression of burned gas.

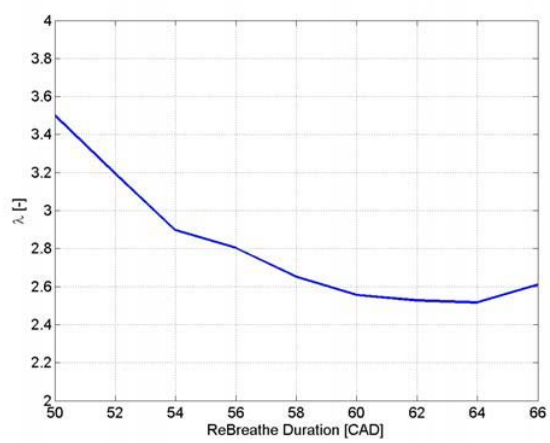

Figure 32 Lambda as a function of rebreathe duration

Rebreathing of exhaust displaces some of the air in the combustion chamber and thus the air/fuel ratio decreases which is clearly visible in Figure 32.

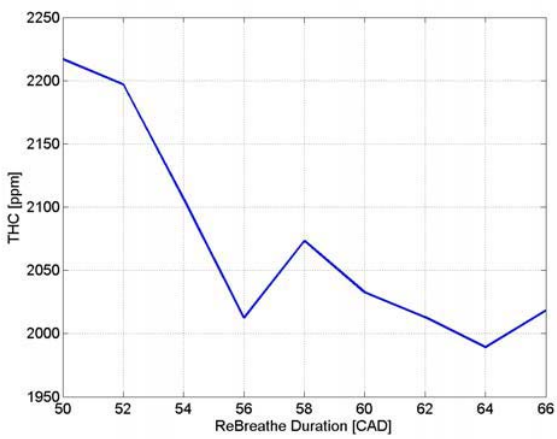

Figure 33 THC as a function of rebreathe duration

Figure 33 shows that the combustion is, as with NVO, better due to higher temperature and lower lambda which leads to lower hydrocarbon emissions. Also here, combustion of some of the rebreathed hydrocarbons contributes.

MILLER STRATEGY

With the Miller strategy the effective compression ratio can be varied by either closing the inlet valve early or late. The decrease of the compression ratio, and thus the compression work, leads to a decrease in charge temperature after compression. Figure 34 shows how the Miller strategy affects the in-cylinder pressure.

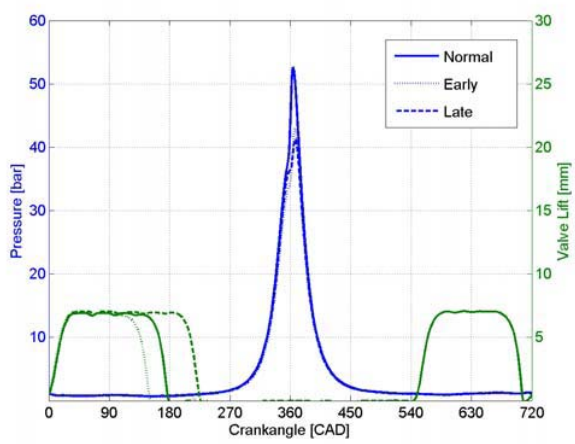

Figure 34 An illustration of how the Miller strategy works

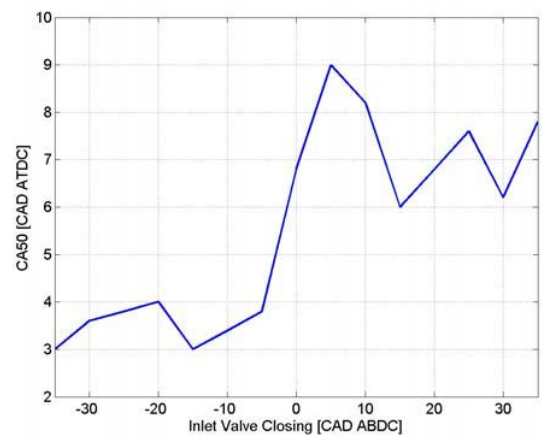

Figure 35 CA50 as a function of IVC angle

Late IVC gives higher volumetric efficiency due to tuning effects. This in combination with somewhat cooler charge causes poor combustion, which can be seen in Figure 35. With early IVC, combustion is satisfactory and there are two effects competing to decide the combustion timing. Decreased air/fuel ratio causes earlier combustion whereas decreased effective compression ratio causes later combustion. The result is that combustion timing stays relatively constant independent of IVC.

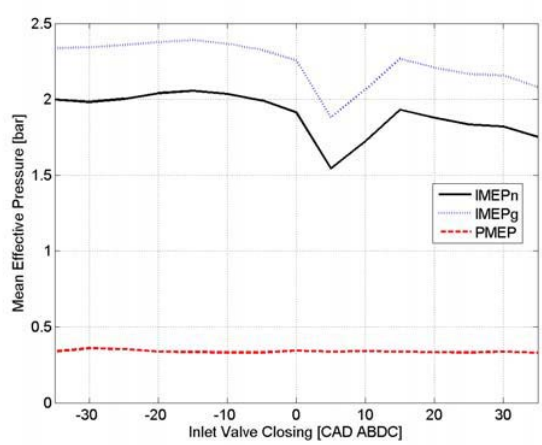

Figure 36 IMEP and PMEP as a function of IVC

On the right side of BDC in Figure 36 it is clearly visible that with inlet valve closing at 5 CAD ABDC the combustion is really poor and after that point is seems like the combustion improves a little but for IVC later than 15 CAD ABDC combustion deteriorates again.

With inlet valve closing at 5 CAD ABDC in Figure 37 the lambda value is much higher than in the rest of the interval. This can be explained by a higher volymetric efficiency in this point or that the poor combustion causes an increase in the lambda measurement.

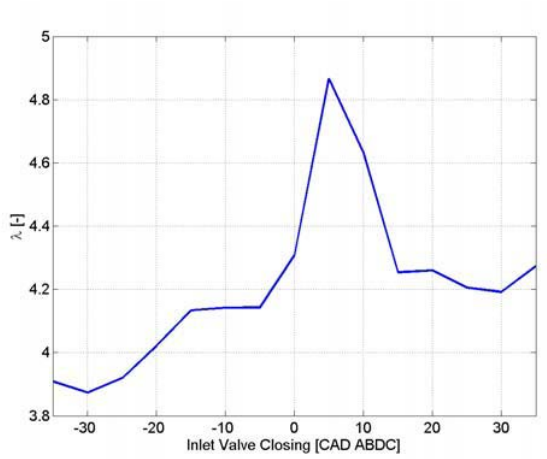

Figure 37 Lambda as a function of IVC

On both sides away from this point lambda decreases because less air is sucked into the cylinder.

Figure 38 shows high hydrocarbon levels with late intake valve closing and low levels with early closing which indicates better combustion with early valve closing than with late. There are two contributing reasons for this. One reason is that the charge temperature is lower with late closing since more intake air enters the combustion chamber and cools the charge. The other reason is that there is a certain tuning effect with late closing which increases the volumetric efficiency and thus the air/fuel ratio. This is one of the reasons why there is always a positive valve overlap on production engines.

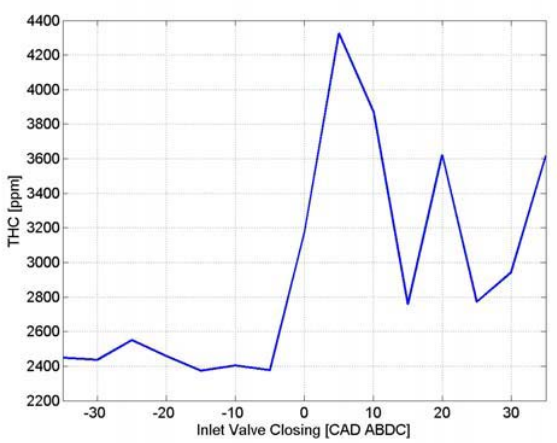

Figure 38 THC as a function of IVC

FUTURE WORK

The next logical step is to test the system for a range of engine speeds and loads combined with various valve strategies. The variable valve system can then also be used for closed-loop HCCI engine control.

CONCLUSIONS

The results achieved during this project clearly show the potential with pneumatic variable valve control. The actuators have proven to be well developed, with very little oil leakage and good reliability during long test runs. There are only remarks regarding time-consuming work such as air drainage and finding the right shims level.

The design goal to control the valves between 2 and 12 mm throughout an engine speed interval between 300 and 2500 rpm has been essentially fulfilled. The only exception is that valve lifts below 2.6 mm are unstable due to physical limitations of the actuators. In the engine tests the valve lift was limited to 7 mm in order to guarantee piston clearance.

Test runs with various valve strategies for HCCI combustion control on a single-cylinder test engine have shown that the system works as well in an engine as in the test rig. The engine tests were performed at a fairly low load level in order to prevent equipment damage in this early test stage. Negative valve overlap, rebreathe of exhaust and Miller cycle were all possible to achieve with the variable valve control system. It was seen that the NVO and rebreathe strategies had a large impact on the HCCI combustion timing.whereas the Miller strategy had less impact due to the competing effects of charge temperature and dilution.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

A	Actual Value
ABDC	After Bottom Dead Center
ATDC	After Top Dead Center
BMEP	Brake Mean Efficiency Pressure
CA50	Crank Angle of 50% mass fraction burned
CAD	Crank Angle Degree
FPGA	Field Programmable Gate Array
HCCI	Homogeneous Charge Compression Ignition
HP	Horse Power
IMEP	Indicated Mean Effective Pressure
IMEPg	Gross IMEP (compression and expansion only)
IMEPn	Net IMEP (full cycle)
K	Integral Constant
NVO	Negative Valve Overlap
PMEP	Pumping Mean Effective Pressure
RPM	Revolutions Per Minute
S	Set Value
S1A	Solenoid 1 Activation
S2A	Solenoid 2 Activation
S1D	Solenoid 1 Duration
S2D	Solenoid 2 Duration
TDC	Top Dead Center
THC	Total Hydrocarbons
U	Control Signal
VI	Virtual Instrument
VVC	Variable Valve Control

APPENDIX

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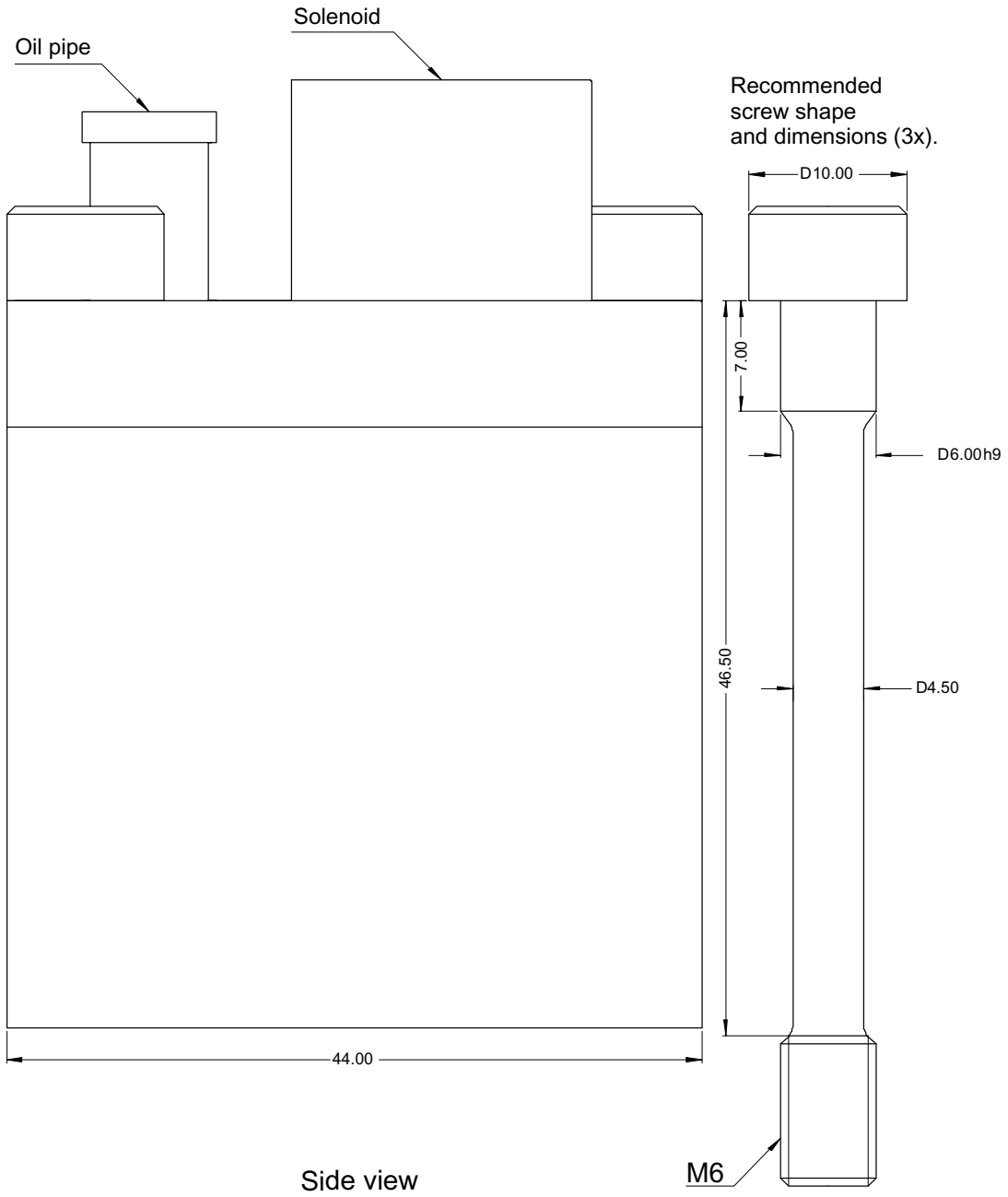

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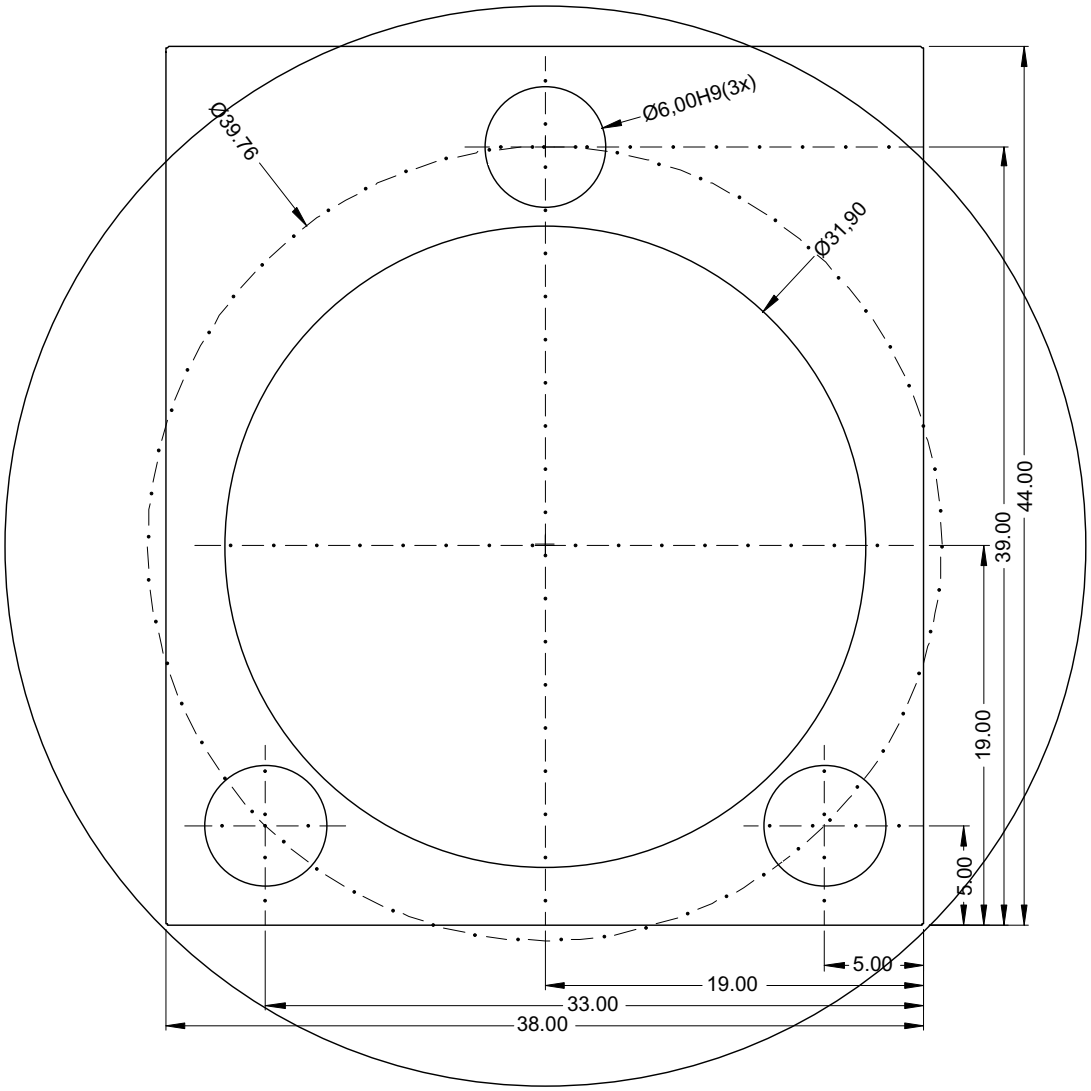

Bottom view

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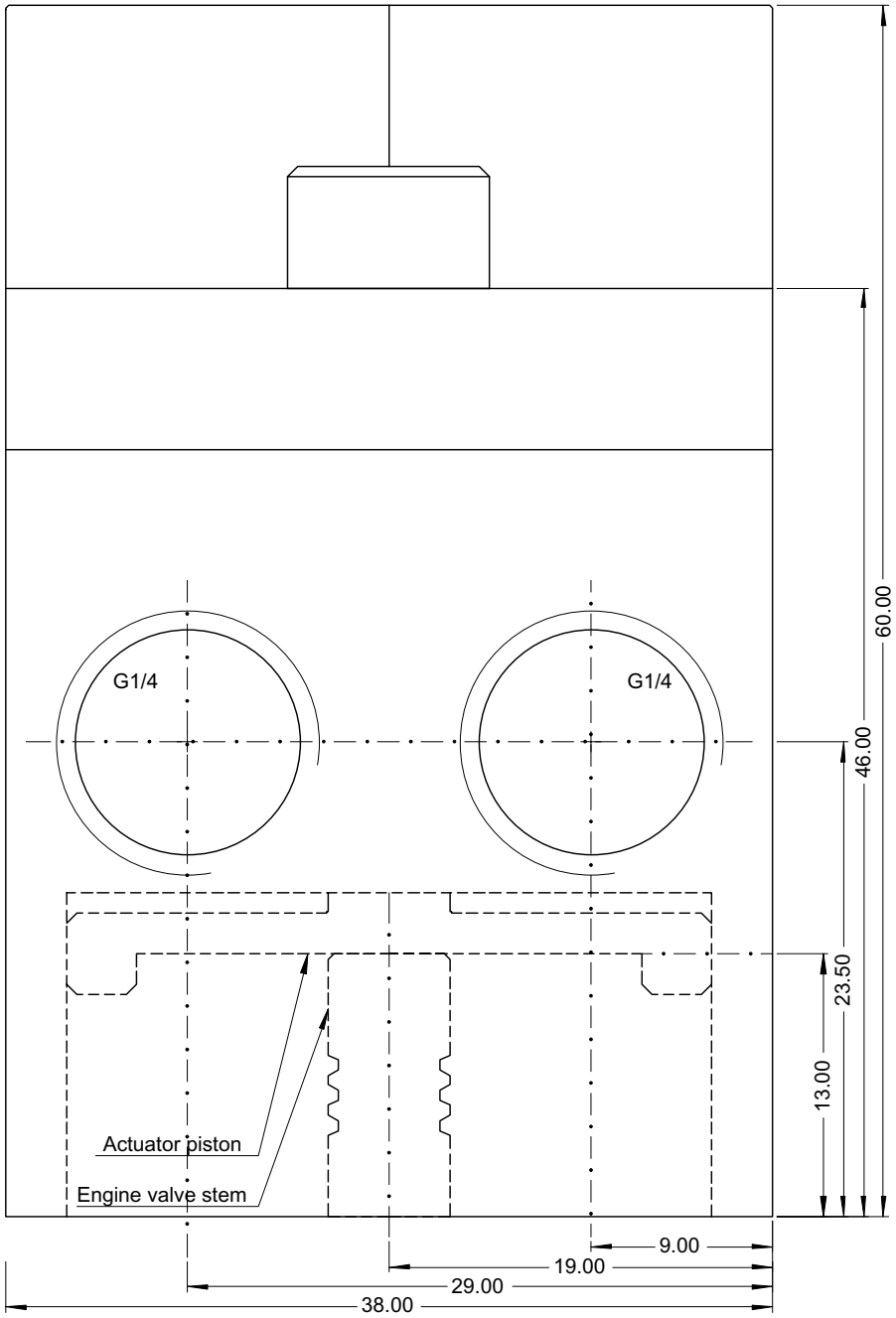

Front view