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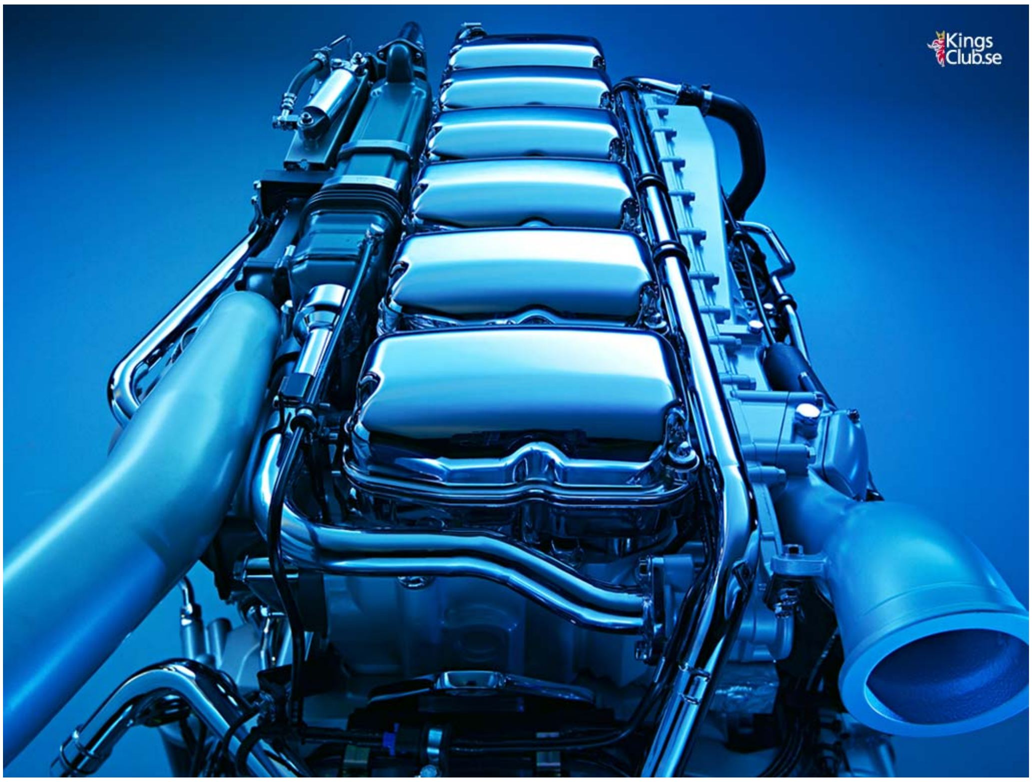

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1. Sammanfattning

Denna report beskriver ett examensarbete inom ämnet förbränningsmotorteknik och är utfört på Scania CV AB i Södertälje i sammarbete med Kungliga Tekniska Högskolan i Stockholm.

Fordonsindustrin står inför nya utmaningar när en större fokusering på koldioxid (CO_2) utsläpp från väg-gående fordon efterfrågas, vilket är direkt kopplat till fordonets bränsleförbrukning. Detta skall klaras av med lägre övriga emissioner i form av kväveoxider (NO_x) och rök som är de två stora problemområdena för en dieselmotor. Scania vill därför undersöka potentialen med Miller cykling och om detta koncept skulle kunna implementeras på en lågeffektsmotor. Idén är att byta hög effekt mot verkningsgrad på motorer i motorprogrammet som inte utnyttjar hela grundmotorns effektpotential.
Med Miller cykling menas senare- eller tidigarelagd stängning av insugsventilen och på så sätt öka graden expansionsarbete i förhållande till kompressionsarbete, dvs. öka den indikerade verkningsgraden. Då detta medför en minskning i effektiv slagvolym då cylindern inte fylls med luft helt måste kompensering i laddtryck göras för att motorn skall få samma luftmängd. En del kompressionsarbete görs då alltså utanför cylindern som belastar turbosystemet och inte motorn. En ny turbomatchning måste således göras då förflyttningar i turbomappen görs på ett mindre fördelaktigt sätt.

Arbetet inleddes med motorprov av en encylindrig dieselmotor som har samma cylinder, kolv och cylinderhuvud som en D12 produktionsmotor där förbränningsdata samlades in för senare simuleringar i GT-power. Två olika kompressioner, tidig resp. sen stängning av insugsventil och EGR provades i testcellen. Det visade sig att generellt sjunker NO_x-emissionerna vid Millring tillsammans med en högre verkningsgrad på framförallt höglast. Vid jämförelse mellan standardkompression 17.3:1 CR standard ventilprofil och 23:1 CR Millerprofil åstadkoms en ökning (i procentenheter) av bromsad verkningsgrad på mellan 1.79 % till 3.3 % vid encylinderprov utan EGR. Den annorlunda kolvgropen vid 23:1 CR ökar rök-emissionerna, kolvgropsutformning bör alltså optimeras för att sänka röken. Med EGR är den maximala ökningen i bromsad verkningsgrad (också i procentenheter) mellan 0.93 % och 3.43 %. Vid låglast ökar avgastemperaturen med upp till 100°C vid 50 kW effektuttag på en motsvarande DLC motor vid 1250 rpm. Detta är till fördel vid eventuell avgasefterbehandling. Vid fullmotorsimuleringar med matchad turbo ökar den bromsade verkningsgraden från 43.5 % (standard motor 17.3:1 CR) till 45.2 % (Miller motor 23:1 CR) på bästa punkten i ESC cykeln.

Miller tillsammans med höjd kompression kan vara ett koncept som är användbart i vissa effektvarianter i en motorfamilj där en högre avgastemperatur på låglast och en högre verkningsgrad på höglast efterfrågas. Det som bör undersökas närmare är att få ned rök emissionerna vid en ökning i kompression.

2. Abstract

This report describes the master thesis work in combustion engines carried out at Scania CV AB, Södertälje and in association with Royal Institute of Technology, Stockholm.

The engine industry faces new challenges now that the focus on CO_2 emissions means that greater demands are placed on engine fuel efficiency. These goals must be met at the same time as other emissions such as NO_x and smoke which are the diesel engines weak spots must be reduced even further. Scania were therefore interested in examining the possibility of using the Miller cycle on a heavy duty diesel engine. The idea is to trade performance with fuel economy on some of the lower power engines in their range which do not take advantage of the engines full potential.

The Miller cycle is defined as earlier or later closing of the intake valves which increases the effective expansion ratio in relation to the compression. This means that the indicated engine efficiency should be increased. The earlier or later closing of the inlet valves means that the effective swept volume is reduced and must therefore be compensated by increasing the intake pressure with a turbo to make sure the same air mass is found in the cylinder when compression takes place. This is in effect an outsourcing of work from the cylinder to the turbo. This outsourcing will place greater demands on the turbo system, a result which means that a new turbo must be matched to the engine.

The work commenced with testing of a single cylinder diesel engine which had the same cylinder, piston and head as a production engine (in effect 1/6 of a Scania 6 cylinder lorry engine). The results of the tests were then used to calibrate simulation models made in GT-power. Two different compression ratios, early and late closing of the inlet valves and with / without EGR were tested in the engine test cell.

It was found that generally NO_x levels were reduced and engine efficiency increased especially for the higher load points. A comparison between the standard engine with 17.3:1 CR standard valve lift profile and a CR of 23:1 together with a Miller cycle gave an increase of between 1.9 and 3.3 % (percent units) in brake efficiency when the single cylinder test engine was run without EGR. It is worth mentioning that the bowl of the 23:1 CR piston was not optimized for smoke emissions, more work needs to be carried out in this area at a later date. With the use of EGR it was possible to increase the brake efficiency by between 0.93 and 3.43 %. When the engine was run at low load (1250 rpm and 50kW for a full engine) the exhaust temperature rose by up to 100 °C. This would lead to benefits in the after treatment of exhaust gases.

A simulation model of a full engine was calibrated and run in GT-power. Results showed that the best brake efficiency for a standard full engine when taken through the ESC cycle was 43.5 %. This could be increased to 45.2 % by increasing the compression ratio to 23:1 and using a Miller cycle.

The summary of the report concludes that a combination of increased compression ratio and running the engine with a Miller cycle is a viable concept for some of the engines in Scanias engine program. The benefits include a higher exhaust temperature under low loads and increase in brake efficiency for higher loads. Further work needs to be carried out on the optimization of the piston geometry with the aim of reducing smoke for the higher compression ratios.

3. Acknowledgments

During the autumn term 2008 we have had the privilege of doing our Master thesis for Scania CV AB in Södertälje, Sweden for the advanced combustion (NMPF) department. It has been a pleasure to spend time at such a professional company with the knowledgeable staff that have help us whenever help was required. Some special thanks to our supervisors Stefan Olsson at Scania and Hans-Erik Ångström at Royal Institute of Technology for excellent guidance throughout the work.

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Clive Lewis and Henrik Dembinski, January 2009

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4. Nomenclature

CR = Compression Ratio
HR = Heat release
IVC = Inlet Valve Close
LIVC = Late Inlet Valve Close
EIVC = Early Inlet Valve Close
IVO = Inlet Valve Open
EVC = Exhaust Valve Close
EVO = Exhaust Valve Open
TDC = Top dead centre
BDC = Bottom dead centre
ABDC = After bottom dead centre
SOI = Start of injection (start of electric signal)
EOI = End of injection (end of electric signal)
SOC = Start of Combustion (CAD between injected fuel to start of comb)
MBF50% = Mass burned fraction 50 %
IMEP = Indicated mean effective pressure over the entire engine cycle, 720° CAD
IMEPH = Indicated mean effective pressure, High pressure part 360° CAD
PMEP = Pumping mean effective pressure = IMEP-IMEPH
FMEP = Friction mean effective pressure
BMEP = Brake mean effective pressure
FSN = Filter smoke number
BSN = Bosch smoke number
br.eff = Brake efficiency
CFD = Computational Fluid Dynamics
CO_2 = Carbon dioxide
CO = Carbon monoxide
NO_x = Nitrogen oxide
HC = Hydrocarbon
ESC = European Stationary Cycle
Lambda = Air/fuel ratio
EGR = Exhaust gas recirculation
CAD = Crank angle degrees
MBT = Maximum Brake Torque
P-max = Maximum cylinder pressure

5. Background

The Scania engine program consists of a range of engines with different power outputs. To reduce production and development costs only three different base engines are manufactured with each engine being tuned to different power levels. The engines which are de-tuned have the possibility with the use of the Miller cycle to improve their potential, in effect trading performance with efficiency.

The Atkinson cycle was defined by James Atkinson in 1885 as a four stroke cycle where the effective expansion is greater than the effective compression. In the Atkinson cycle engine all four strokes occur during one rotation of the crankshaft, the expansion stroke being physically longer than the compression stroke. The Miller cycle builds upon the Atkinson cycle and is named after Ralph Miller. During the 1940’s he discovered that by closing the inlet valve late or early (on the compression stroke) and thereby reducing the effective compression ratio together with increasing boost levels he could improve engine performance. In this type of cycle the engine draws a full charge into the cylinder but pushes part of it back into the inlet manifold. Due to the shorter effective opening time of the inlet valve it is necessary to increase the air mass in the cylinder to be able to reach the same level as a normally operated engine. This is achieved by turbo charging. If the energy used by the turbo system is less than the energy it would normally take to compress the charge to the same state then there is the possibility to improve the efficiency of the engine. A similar way to achieve the same goal is to close the inlet valve earlier during the intake stroke and let the charge expand and compress until it reaches the same state as if the inlet valve had been closed late.

One of the advantages of the Miller cycle is the reduced temperature in the cylinder which leads to lower NO_x emissions. Another advantage is that lambda can be reduced in the low power output cases and the exhaust temperature is thereby increased which is extremely useful for exhaust aftertreatment systems. A Variable Valve Timing (VVT) mechanism could be used to achieve the Miller cycle where its benefits at specific load condition are highest. Another way is to have a set Miller cam profile for low power output engines.

6. Literature Study

Miller-PCCI (Premixed charge compression ignition) combustion in a high speed direct injection diesel engine were simulated and tested at Waseda University [1]. The Miller strategy was achieved by VVT and late inlet valve closing. By use of VVT the effective compression ratio could be controlled while keeping the expansion ratio constant. The maximum gas temperature could be lowered by reducing the compression ratio through later closing of the inlet valve. The lower temperature at the start of combustion led to a reduction in NO_x emissions. This longer ignition delay premixed all the fuel that could be part of the combustion. This gives benefit in soot emissions because the fuel has time to vaporise in the combustion camber before ignition. The ignition is still compression ignition. Lower concentrations of fuel rich areas are created compared to classical diesel combustion which gives benefits in soot emission levels. At the same time NO_x could be reduced by applying a large amount of EGR (Exhaust gas recirculation). However, low combustion efficiency was altered due to the reduced level of vaporised fuel in the piston cavity, this was a result of early fuel injection into low gas density and temperature conditions in the combustion camber. The solution was to inject the fuel nearer top dead centre (TDC) to suppress HC-emissions.

At IMEP 7.7 bar the control of late inlet valve closing combined with EGR reduced NO_x and soot emissions while maintaining the fuel consumption performance. Miller-PCCI combustion gave increased exhaust gas temperature with reduced exhaust gas flow rate which is beneficial when catalytic gas aftertreatment is used. Miller-PCCI could be expanded up to an IMEP of 13 bar. After that even higher IMEP could be achieved with standard diesel combustion combined with a Miller cycle.

The potential of dual stage turbo charging and Miller cycle for heavy duty diesel engines was analysed by Millo and Mallamo [2]. A 12.8L straight six diesel engine was both simulated in GT-Power and tested in a laboratory. The test engine in the laboratory was fitted with a single VGT turbo and the tests were carried out at 1500 rpm and at full load. The simulations showed that with an early inlet valve closing strategy whereby the inlet valve was closed at 20° BBDC (before bottom dead centre) and dual turbo it was seen that BMEP increased by 5-6 %. NO_x levels were reduced by about 10 % and even the specific fuel consumption was lowered by 2 %. The dual turbo system included a wastegate to control the boost level on the HP (high pressure) turbine and just one after cooler downstream of the HP turbocharger compressor.

The Otto engine uses a slightly different valve timing strategy, the conventional closing time for the inlet valve is usually found around 50-60 degrees ABDC. In a report from Nissan Motor Co. [3] it was found that even an Otto engine can benefit from the Miller cycle. It was possible to reduce fuel consumption by 11.2% by using a VVT to achieve the Miller cycle. The decrease of the effective compression ratio did however lead to a lesser reduction of fuel consumption under light operating conditions.

At TU Graz [4] a comparison between Early Inlet Valve Closing (EIVC) and Late Inlet Valve Closing (LIVC) was made on a turbocharged Diesel engine. Both simulations and single cylinder test were carried out. The engine was a 5-cylinder passenger car engine equipped with a two stage serial-sequential turbo charging system. Both cycle concepts can be seen as an outsourcing of compression from the cylinder to the external charger. It is thereby possible to put the same amount of air in the cylinder with a lower temperature. This means that some of the air in the combustion chamber is ejected out through the inlet valve before it closes and the amount of air is reduced.

The potential of the system is significantly dependent on the potential of the turbocharger. This is due to more compression work is being done by the turbo compressor. Due to the higher inlet air pressure from the same amount of inlet air, limitations can come from surge limit and rotational speed for the compressor. LIVC cycle has a higher compression temperature at the end of the compression stroke than the EIVC cycle. The LIVC cycle has also a higher air temperature in the end of the extended inlet stroke. This is because some of the air is pushed back through the inlet valve and due to the higher air movement a greater wall heat flow occurs. Other effects include the pulsations in the intake manifold and the inertia of the air. Inertia effects usually bring an advantage for the LIVC cycle, whereas pulsations can affect both strategies in a beneficial or non beneficial way depending on the valve strategy. The valve closing in the EIVC cycle can cause problems because of high valve accelerations. This due to the valve open time is shorter and to have the same valve lift as the LIVC cycle high valve accelerations is unavoidable if no limitations in valve lift are use. This gives on the other hand higher flow resistance over the valve.

At TU Graz [4] a CFD-simulation was done on the flow conditions of the intake stroke. The distribution of turbulent kinetic energy and velocity at TDC compared to conventional diesel, see Figure 1 where Miller is earlier closing and Atkinson is later closing of the inlet port. The turbulent kinetic energy is much weaker at EIVC cycle than conventional diesel cycle. The LIVC cycle does not lose much in kinetic energy and velocity. EIVC can there by have bigger problems with the after oxidation of soot emissions.

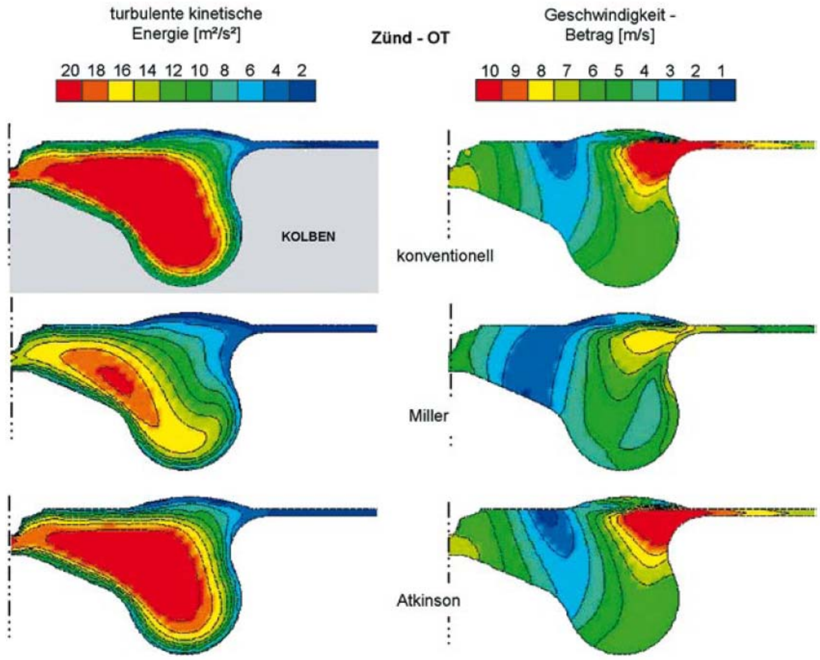

Figure 1 Distribution of kinetic energy and velocity at TDC. CFD-simulation

In the single cylinder testing that was carried out at TU Graz, the settings for boost pressure, exhaust back pressure and particularly intake air temperature were taken directly from simulations. This was to ensure the boundary conditions corresponded to a real engine. Most interesting measurement result is the reduction in NO_x emissions. With conventional diesel cycle a NO_x level of 3.4 g/kWh could be lowered to 2.4 g/kWh with alternative cycling. Smoke and CO remain nearly constant and HC dropped significantly for the EIVC cycle. Comparison of EIVC and EGR showed that CO and soot emissions are higher for the same NO_x emissions when EGR is used instead of EIVC cycle.

So EIVC is a way to reduce NO_x without having high soot emissions. The potential of EIVC cycling is thereby almost exclusively limited by the capabilities of the charging system.
TU Graz [4] came to the conclusion that an EIVC or LIVC cycle on a diesel engine did not make sense at that time, if an EGR system is applied.

At Ricardo [5] investigations have been done on EGR engines with external EGR and Internal EGR (iEGR). This was done by using an engine simulation code (WAVE) on a Mercedes OM501 LA V6 engine. Miller cycling was used to lower the start temperature in combustion when iEGR was used. iEGR was created by an extra exhaust valve lift and different lift strategies were tested. High boost pressures and a new turbo matching must be done to fully investigate Miller cycle qualities. That has not been done in this report, but calculations showed that up to 6 bar boost pressure was needed to maintain air fuel ratio together with EGR. This means a dual turbo charged system needs to be fitted to the engine if full Miller cycle is used.

7. Problem

This master thesis aims to examine the potential of Miller cycling for a turbo charged diesel engine. Scania CV wants to investigate if the Miller cycling strategy can be a way to lower the fuel consumption and to maintain the emission levels. At high loads when Miller cycling is used, the reduced effective swept volume needs to be compensated with higher boost pressure to assume the same performance. This means that it must be a trade off between the level of Miller cycling, over boost performance, efficiency and compression ratio. This relation is interesting to investigate. The turbo system is thereby influenced and the movement in the compressor map is going to take place in an adverse way compared to standard valve profile. This means that a new turbo match needs to be done to compensate the higher boost pressure for the same air mass that the engine demands. The same will apply to the turbine when it experiences other exhaust temperatures and masses. The turbo match is done on the basis of the single cylinder tests in the cell and simulated tests in GT-power [6].

The master thesis will investigate the previously described problem with following limitations;

A single cylinder diesel engine with a Lotus AVT free valve system is available for a limited time. This is used to collect measurements and investigate the correlation between air-fuel relationships, combustion, inlet and exhaust pressures and exhaust temperatures. The limitations in the test cell are, lambda min 1.3, EGR between 0 % and 30 %, two different compression ratios, standard 17.3:1 and one investigated CR. Maximum 250 bar cylinder pressure, limitations in fuel rail pressures, no pilot injections. Simulations will be done in GT-power where investigation on how a Miller strategy would affect engine efficiency and engine turbo performance. In GT-power a built in combustion model shall be used or an own model created on the basis of measured test data.

The work will answer the following questions;

What advantages can Miller cycling give in the EGR and non EGR cases?

Which fixed Miller level should be used and which compression ratio for low power output engines are the best compromise?

Which variable Miller level gives the highest efficiency?

What will the turbo system look like to manage the different over boost strategies?

8. Methods and equipment

To investigate the Miller concept, tests were done on a test engine and simulations were carried out in GT-power. The engine tests were done in a single cylinder test cell described under the headline “Engine test cell”. The simulation tool GT-power was used before, parallel and after the engine testing to generate simulated data.

8.1. GT-Power simulation tool

GT-Power is a powerful simulation tool used by engineers worldwide. The underlying principle is to model the single cylinder test-engine, calibrate the model with the help of data from the test cell and then use a calibrated model of a full engine to predict engine performance. Because of the large number of variables that can affect the performance of a combustion engine it was necessary to define some of them as constants, thereby reducing the number of test cases and the complexity of the test process.

8.1.1. Single cylinder model in GT-power

To simulate the single cylinder test engine and a full six cylinder DF12 engine, two models were built up in GT-power. The purpose of the single cylinder model was to investigate which Miller cycle strategy should be tested in the test cell and which higher compression ratio would be optimal to increase engine efficiency.

The GT-power model that was received at the beginning of this project was created by Olof Erlandsson at Scania CV AB who made it for an ethanol DI engine that had been tested in the test cell earlier. Some modifications were made so the model was more suitable for a diesel Miller engine.

How GT-power works and how to build models can be read in [7]. In Figure 2 the single cylinder engine can be seen in the GT-power environment. In this view the outer gas exchange system can be seen together with injector, cylinder and crankshaft. The outer gas exchange system is the same as in the old ethanol model and it describes the test cell environment well.

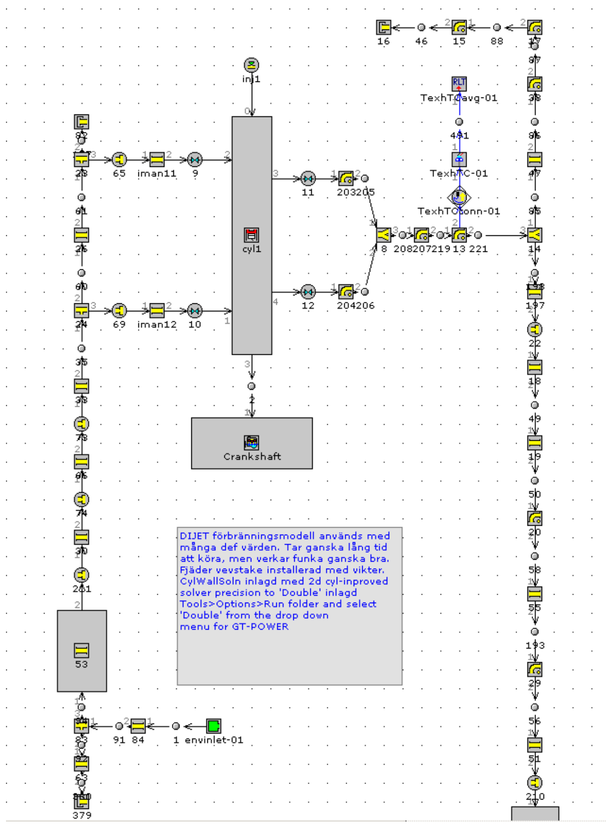

Figure 2 Model on the single cylinder

The friction model used was the Chen-Flynns model [8]. Equation 1 for FMEP is a simplification using actual engine test data and that has been shown to be accurate for Heavy duty diesel engines in experimental tests.

$$\mathrm{FMEP} = A_1 + A_2 \cdot p_{max} + A_3 \cdot C_{pis} + A_4 \cdot C_{pis}^2 \qquad \textbf{(1)}$$

The input data to the Chen-Flynns model:

A_1 = Constant part of FMEP

A_2 = Peak Cylinder Pressure factor

A_3 = Mean piston speed factor

A_4 = Mean piston speed squared factor

p_{max} = Peak cylinder pressure

C_{pis} = Mean piston velocity

All of the A_x are factors that need to be set in GT-power. In this first simulation this data was not changed from the ethanol model.

The model was modified with new physical parameters for example piston mass, connecting rod mass and so on, see Figure 3. This was done to have a more accurate model, many of these parameters were not set before. The con-rod stiffness was set to 742 MN/m in GT-powers built-in crank-slider system. This is due to the fact that the con-rod behaves like a spring under high loads. The cylinder head also acts in similar way so to make the model less complicated the cylinder head stiffness is added to the con-rod stiffness. This however was done later when cylinder pressure data was available after the single cylinder tests.

Attribute	Unit	Object Value
Piston Mass	g	5084
Connecting Rod Mass	g	4759
Connecting Rod Inertia	kg-m^2	0.0667
Conrod Center of Gravity	mm	80.26

Figure 3 Inertia data for piston and connecting rod

8.1.2. Combustion

GT-Power has different built-in combustion models. The combustion model that was used in the ethanol model case was the HRprofile. This model uses real measured cylinder pressure trace from real engine tests and calculates the burn rate. The shape of the heat release is then used in GT-power to calculate the combustion. Ethanol combustion data was not used and diesel combustion is different so the CombDIJET combustion model included in GT-power was used. This model calculates the combustion from the injector and how it delivers fuel to the combustion camber. The injection model needed to be modified first.

The injector model describes the injector behavior. The injector used in the test engine is described in this report in chapter “Engine test cell”. In the model the nozzle hole diameter, depth of the injection holes, injector sack and so on are specified. In GT-power the amount of fuel is specified in the model, but how the fuel is delivered must also be specified by an injection rate. An RLT object that describes the fuel pressure, the injection duration on the basis of amount injected fuel was set up in the model. The injector sack pressure versus CAD is described in the injector object.

CylTWallSoln is a cylinder wall temperature solver in GT-power that calculates the temperature through finite element model of the combustion chamber. The inputs of the model include geometry of the combustion chamber, boundary conditions of the coolant temperature, boundary conditions of heat transfer and initial surface temperatures. CylWallSoln is needed to calculate the heat transfer. Three different heat transfer models and two user input heat transfer models can be chosen between in GT-Power.

Possible models in GT-Power:

- **Woschni**

$$h_c\left(W/m^2 \cdot K\right) = 3.26B^{-0.2}p^{0.8}T^{-0.55}w^{0.8} \tag{2}$$

h_c = heat transfer coefficient
w = Average cylinder gas velocity (m/s)
T = Average temperature of the cylinder gas (K)
p = cylinder pressure (kPa)
B = bore (m)
More on how the Woschni model works can be found in [8] or [9]

- **Hohenberg**

Hohenberg formula, Equation 3, that has been shown to be accurate for direct injected diesels [10].

$$h_{Hohenberg} = 130V^{0.6}p^{0.8}T^{-0.4}\left(\overline{S}_p + 1.4\right)^{0.8} \quad \textbf{(3)}$$

$h_{Hohenberg}$ = heat transfer coefficient
T = Average temperature of the cylinder gas (K)
p = Cylinder pressure (kPa)
V = Instantaneous cylinder volyme
$\overline{S}_p$ = Mean piston speed

- **Flow**

GT-power built in heat transfer model, formula Equation 4 are more explained in GT-power documentation [6].

$$h_g = \frac{1}{2}C_f \rho U_{eff} C_p P_r^{-\frac{2}{3}} \quad \textbf{(4)}$$

h_g = heat transfer coefficient
C_f = friction coefficient
ρ = density
U_{eff} = effective velocity outside boundary layer
C_p = specific heat
P_r = Prandtl number

The choice of heat transfer model will be investigated in capital 9.8. calibration of the GT-Power model. The next step is to examine the model for the valve lift profiles.

8.1.3. Variable Valve Timing

To be able to model the camshaft profiles for different inlet valve closing times the lift profile from a D12 engine was used. Engines in the Scania engine program have different camshafts but they all share the same lift profile. The lift values are stored in mm and are coupled to the crank angle in crank angle degrees. To create late closing profiles the maximum lift was retained for an appropriate number of crank angles. Start of lift remained constant. It was then possible to create a table where the lift in mm was stored for different closing angles. This table was then imported into GT-Power to be used to model VVT.

Seven different cam profiles were tested including the standard cam profile. The valve profiles with late valve closing have standard valve ramp for opening and closing and only the opening duration was changed. The valve trains on HD diesels have restrictions in valve acceleration due to the relatively high masses of the valve train components. Therefore most of the tests were late valve closing with longer opening duration, EIVC places too high demands on the valve train. Figure 4 shows the lift profiles for the different late inlet valve closing times.

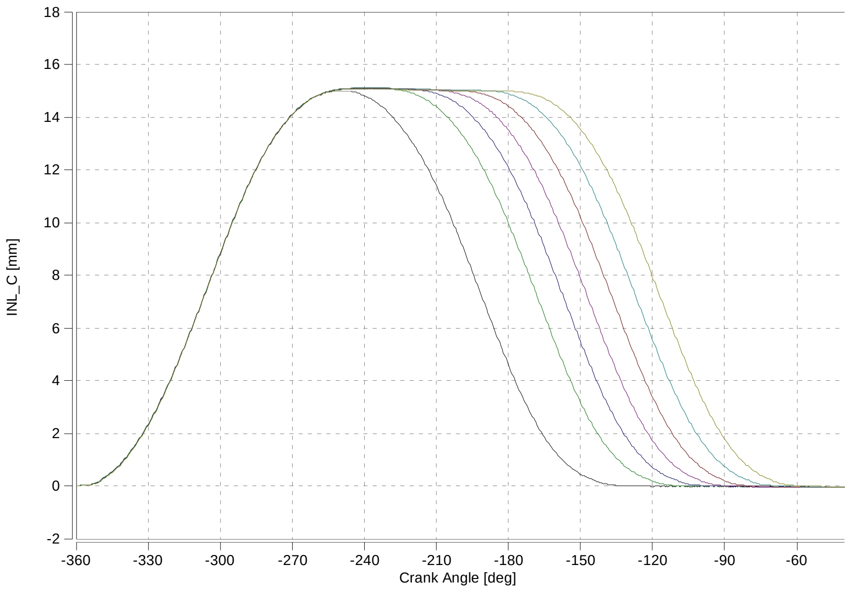

Figure 4 Late inlet valve closing, the black line is the standard valve profile.

In Figure 5 the early valve closing profiles that were tested are shown. The standard valve profile is the black one. The valve acceleration is higher in these valve profiles because otherwise the maximum lift height was too low. The opening and closing ramp have therefore a greater slope than the standard valve profile.

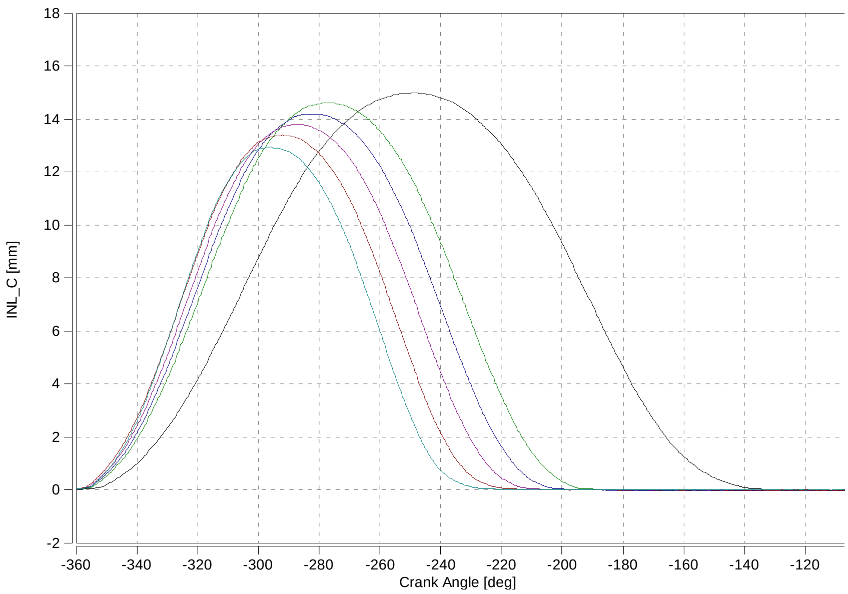

Figure 5 Early inlet valve closing, the black line is the standard valve profile

The first test with a Miller cam profile was now done on the single cylinder model in GT-power. All boundary conditions were held constant for all cases, for example injected fuel mass, SOI, inlet air pressure etc.

At this time only late inlet valve closing was investigated. Restrictions in valve acceleration due to the relative high masses of the valve train components meant that studies with early closing of the inlet valve have been omitted. To understand more and to see how much profit the Miller cycling can give without having boost system limitations, the GT-power model of the single cylinder engine was used. Some modifications were made to the model such as a constant air flow and a feedback system for exhaust pressure, see Figure 6.

This was made so lambda was constant independent of which Miller cycle is used. The feedback backpressure was linked from the inlet pressure and the backpressure was set similar to the inlet pressure. Due to the constant inlet air flow the inlet pressure is thereby compensated automatically depending upon which Miller cycle is used.

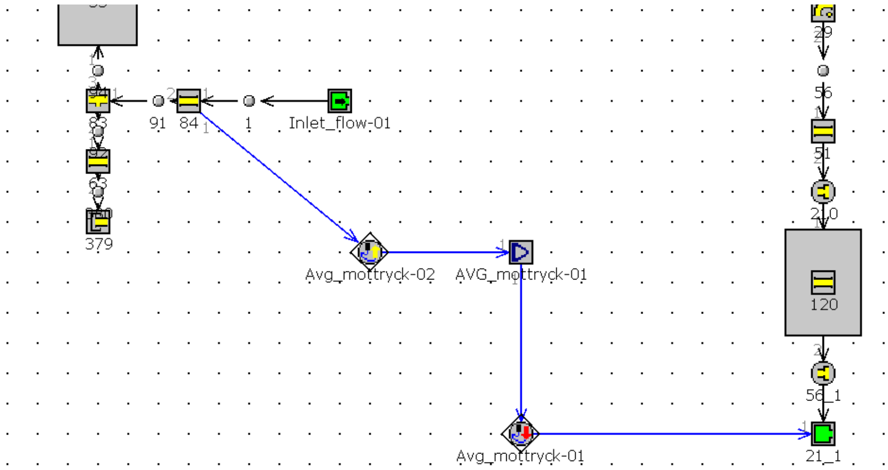

Figure 6 How the constant inlet air flow and the feedback back pressure was made in GT-power

This means that lambda was nearly constant; it fluctuated +/-0.015 when run with different Miller cycles and compressions. Now the investigation with different Heat transfer models could be done. When other compression ratios were investigated the Heat transfer was a very important parameter.

The difference between the Heat transfer models can be seen in Figure 7, Figure 8 and Figure 9. They are all tested with the same boundary conditions, 1500 rpm, 200 mg delta, -3° SOI and 1.85 lambda without EGR. The compression ratio together with Miller level, late inlet valve closing, were tested at different settings, PMEP and BMEP were plotted.

In all the different Heat transfer models the BMEP and PMEP behaved in similar way. The compression that gave the highest BMEP varied between the heat transfer models. Therefore it was important to investigate which heat transfer model gave the most accurate results and can be used later in the simulations. Later a higher compression ratio that would be tested needed to be determining on the basis of the GT-power simulations.

The Hohenberg uses time-averaged gas velocity which is proportional to the mean piston speed. The Woschni separates the gas velocity into two parts, the unfired gas velocity that is proportional to the mean piston speed and the time-dependent combustion induced gas velocity that is a function of the difference between the motoring and firing pressures.

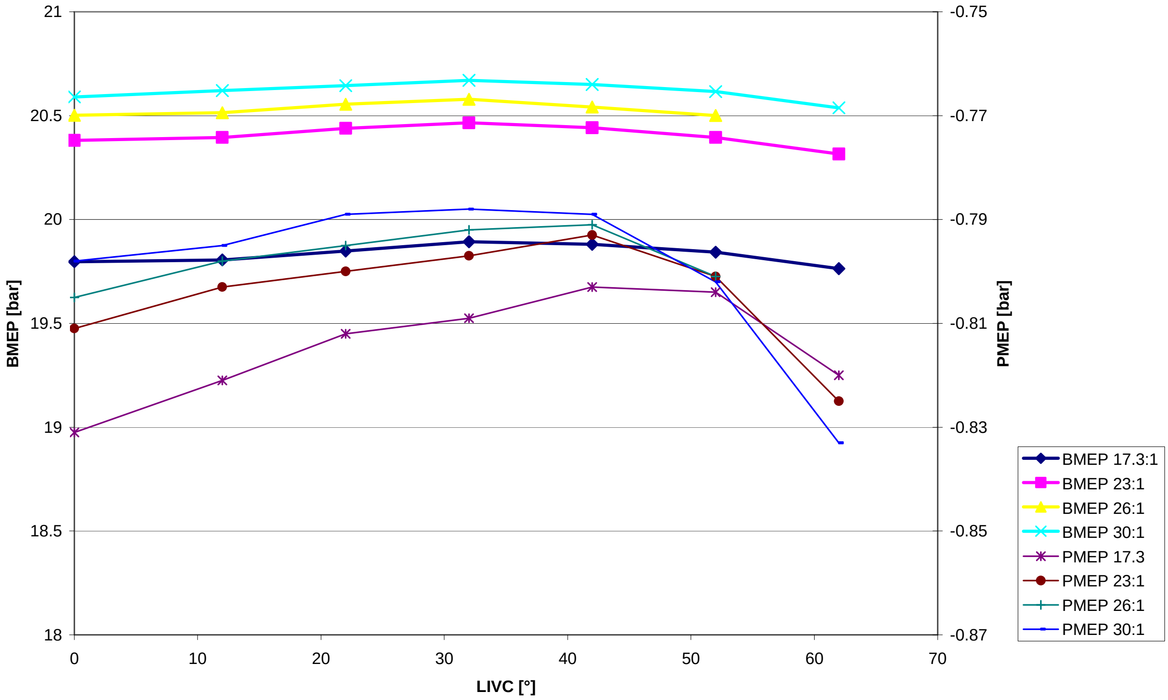

Figure 7 BMEP & PMEP with different Miller cycling and compression, Heat transfer model: Hohenberg

In a SAE report by Chang et.al. [11] a comparison between different Heat transfer models was done. The conclusion was that Woschni was inadequate for HCCI engine.

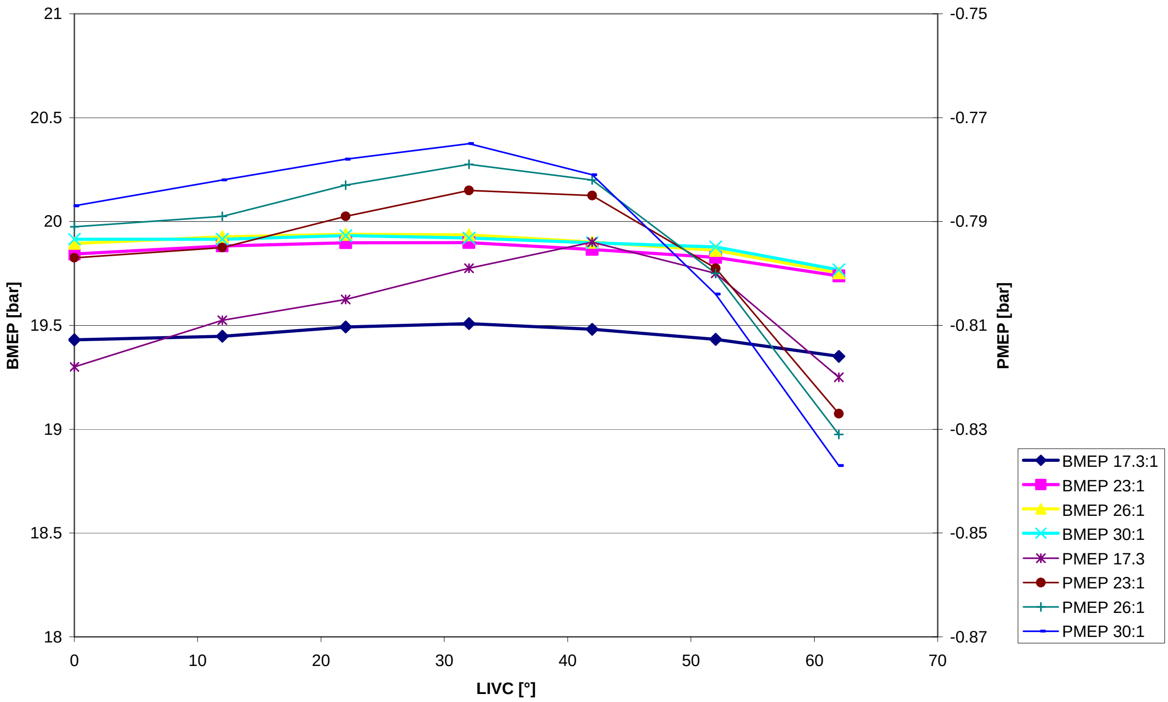

Figure 8 BMEP & PMEP with different Miller cycling and compression, Heat transfer model: Woschni

The biggest difference in the Heat transfer models were found between Hohenberg and Woschni/Flow seen in the figures. The Hohenberg had the highest BMEP which increased

with a higher compression ratio. The Woschni and flow models had a maximum BMEP output at compression ratios 23:1 to 26:1. Even higher compression ratio gave lower BMEP, the two models showed the same behaviour. This means that the conclusion of which C.R. to choose is dependant on which heat transfer model that is used. The Woschni model had generally a higher BMEP output than the flow model.

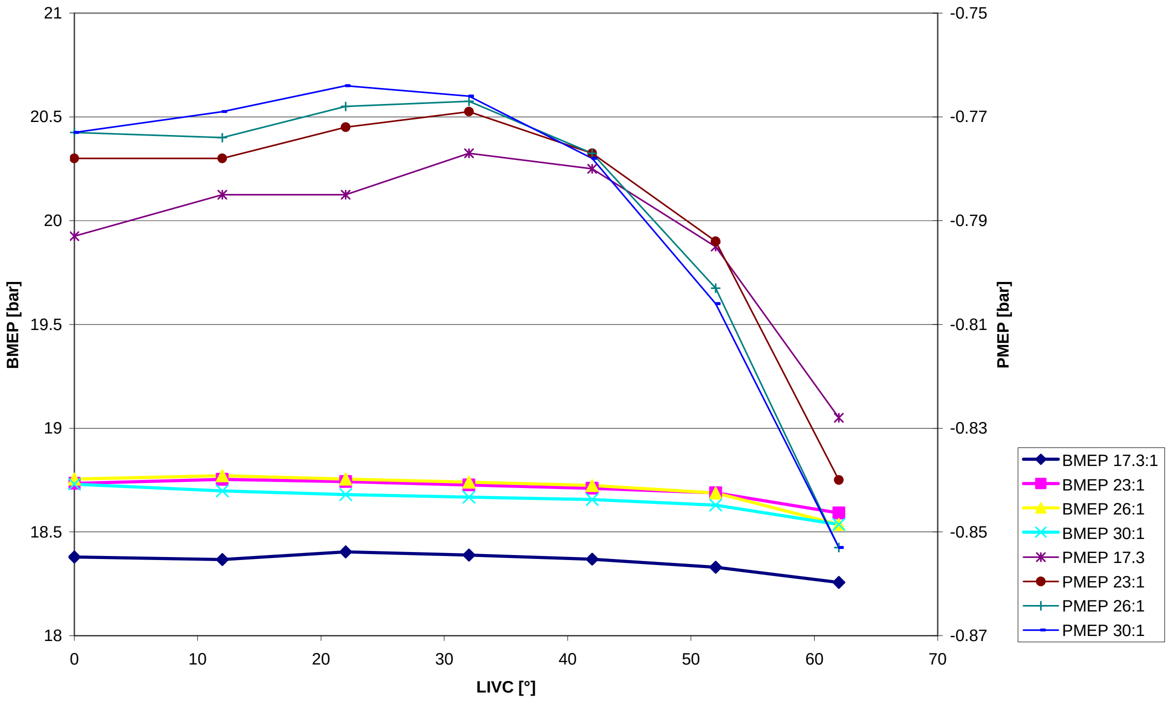

Figure 9 BMEP & PMEP with different Miller cycling and compression, Heat transfer model: Flow

It was also interesting to see that the BMEP trend between the different compression ratios at different Miller levels was similar. The BMEP curves did not cross at higher Miller levels as might be expected, Figure 9 shows this. The geometric compression ratio controls the efficiency not the effective compression ratio. The brake efficiency can however be increased with Miller cycling. PMEP was lower with higher Miller levels and increased with higher Miller levels. This is because the piston speed and the air velocity out of the cylinder are higher at high Miller levels and throttle losses arise over the inlet valve.

The chosen Heat transfer model was the Flow model that was also recommended by GT-Power for this application. The model calculates air movements more accurate than the Woschni model. This is because it uses the swirl data and calculates the boundery layer moments of inertia. The Woschni and Hohenberg uses approximations as described earlier where Hohenberg uses the most simplified correlations.

Identify the range of the compression sweep

To investigate where the optimal compression ratio range is for a diesel engine being operated with a Miller cycle, a sweep of compression ratios was carried out in GT-power. This was done with the single cylinder engine model and a standard lift profile. The standard lift profile was then used in the simulations with different heat transfer models and showed that higher Miller levels did not need a higher geometric compression ratio for higher efficiency. This important result means that the effects of using a Miller cycle are completely independent of CR Maximum cylinder pressure, Pmax, was ignored, only the BMEP at a specific injected fuel mass, 150 mg/cycle, SOI -4° and 1500 rpm was examined. To find the highest BMEP a compression ratio sweep was done from 17:1 to 70:1 in CR. The highest BMEP was found at a compression ratio of around 20:1-30:1, see Figure 10.

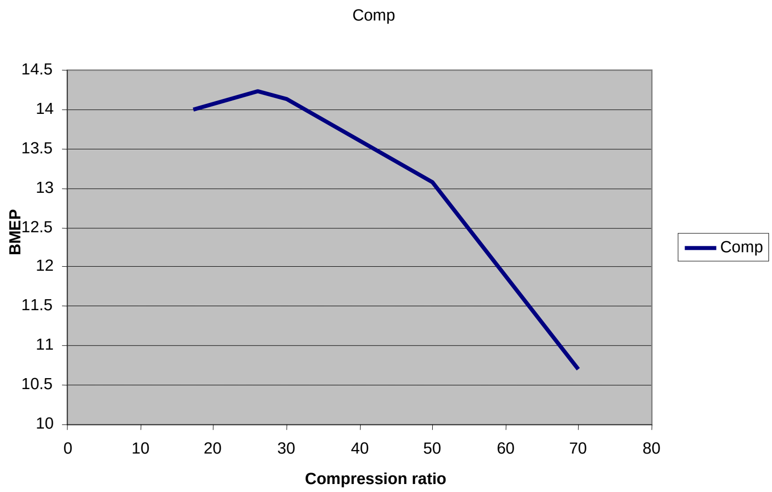

Figure 10 Compression ratio sweep between 20:1 and 70:1

Now the load was varied by setting the injected fuel amount, delta, from 100 mg to 300 mg. The engine speed was decreased to 1250 rpm and the same test was done with the single cylinder model as when the heat transfer model was investigated. The compression ratio was set as before, 17.3:1, 23:1, 26:1 and 30:1. Lambda was held constant over the Miller levels and the inlet manifold pressure increased with the Miller level. In the case with 100 mg delta BMEP was higher at high Miller levels, but PMEP increased also, see Figure 11.

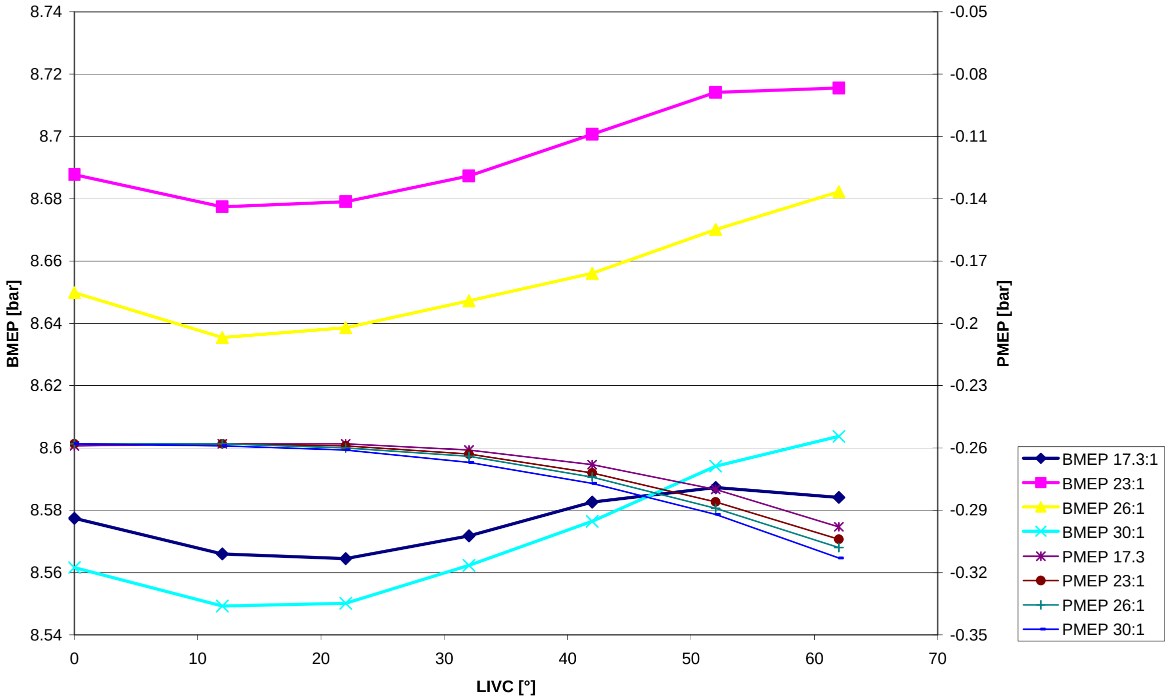

Figure 11 BMEP and PMEP at 1250 rpm, 100 mg delta

In the 200 mg delta case 23:1 still has the highest BMEP output as in the 100 mg delta case. The BMEP output is still higher at high Miller levels, PMEP behavior is still the same, Figure 12.

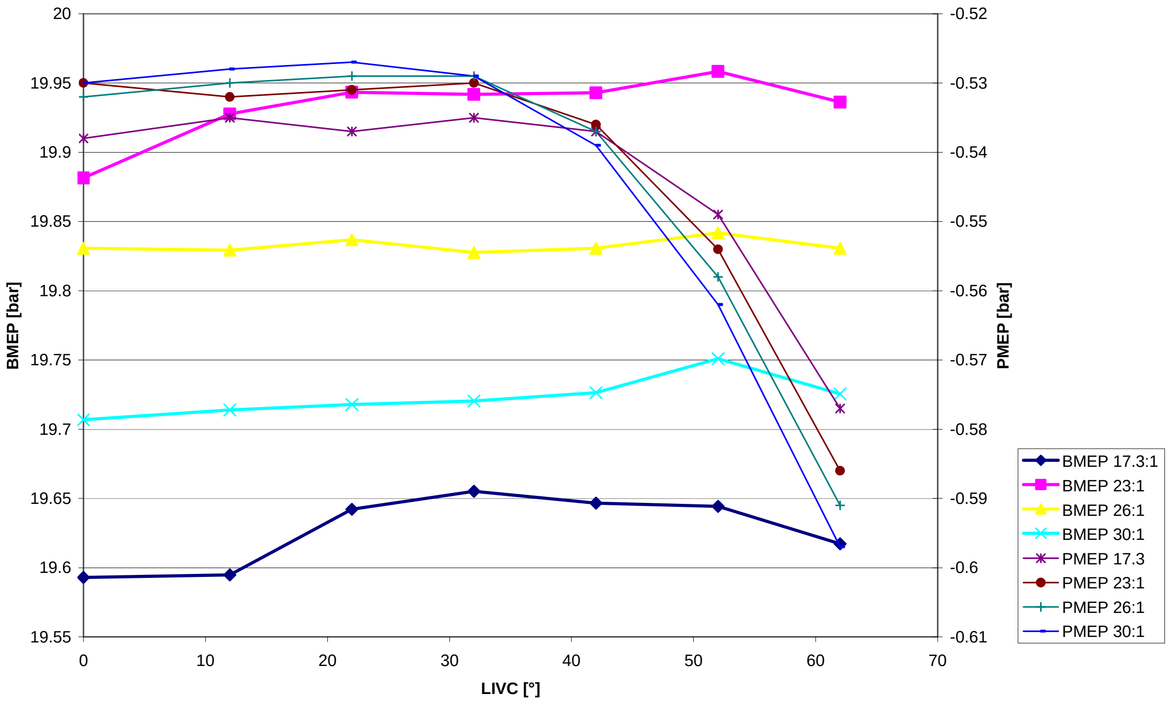

Figure 12 BMEP and PMEP at 1250 rpm, 200 mg delta

When delta was increased to 300 mg at high load, the highest BMEP hade 23:1 in CR case, see Figure 13. However the maximum cylinder pressure was too high at 30:1 in CR, so this CR is not of any interest in reality and not plotted. 23:1 CR had the highest BMEP was at Miller level of 32° later IVC than standard IVC, see Figure 13. The PMEP behavior is little different than before. An optimum Miller level at 32° LIVC was found where the pump work (PMEP) was the lowest possible. Values for PMEP are negative therefore the peak will be the optimal point.

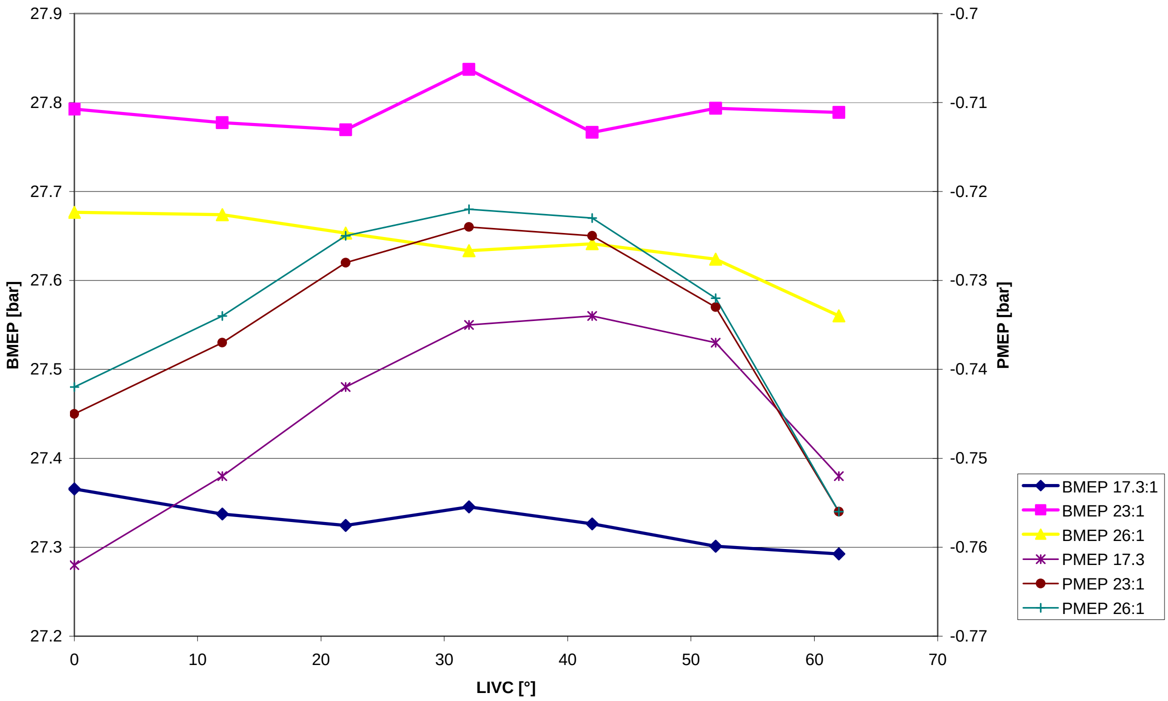

Figure 13 BMEP and PMEP at 1250 rpm, 300 mg delta

A CR of 23:1 had the best efficiency in this test. The CR was also investigated in the full engine GT-power model.

8.1.4. Full engine modell in GT-power

When the single cylinder engine model was complete a full engine model could be built up. The model will be used to investigate the behaviour of the turbo system when the engine is run with Miller cycling.

A full engine model was received from Johan Forss NMPF Scania CV AB. This model was then modified the same as the single cylinder model. In Figure 14 the engine model can be seen in GT-power. The greatest difference between this model and the single cylinder is the turbo boost system with VGT turbine, EGR loop and an engine control system.

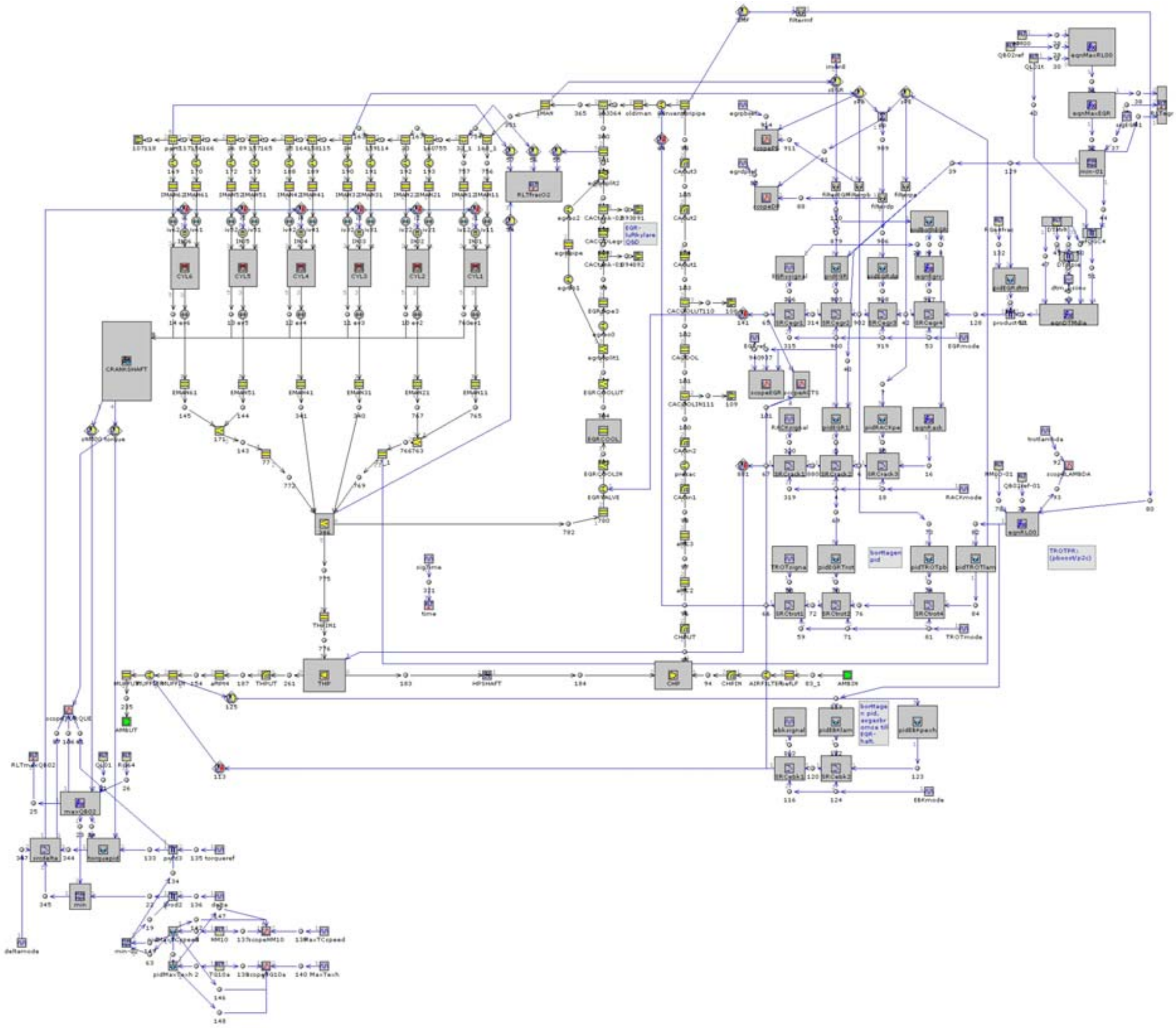

Figure 14 Full engine model

The model is originally a Scania DLC6 engine with a maximum power output of 480 hp. The model is calibrated with combustion data from real engine test measurements. The SOI, EGR amount and so on were taken from data earlier recorded in Scanias test cells for full engines, see Figure 15.

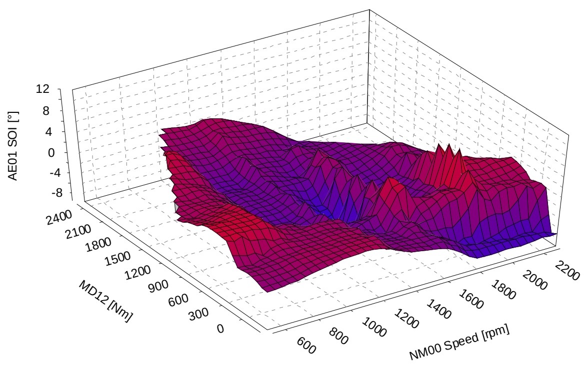

Figure 15 SOI for DC1307, 480 hk, plotted for different load and speed

The DLC6 engine from Scania that the GT-power model is based on can be delivered to the customer in four different power outs, 360, 400, 440 and 480 hp. The compressor/turbine is from Holset with number: HE531Ve - B98K66RBA / A84M81 SH29LX-V12V.

The same compressor and turbine is fitted for all power outputs, the only difference between the engine models is the engine control strategy.

The engine control system can control VGT position, EGR valve, exhaust brake and delta (specific torque output). The amount of EGR can be controlled in combination or separately with the VGT position and/or EGR valve. The EGR valve can also be steered by setting the pressure difference over the valve. The same can be made for the VGT position for the turbine and inlet throttle. The pressure difference may be the only way to stabilize the control system in certain operational points so the different regulators do not counteract with each other.

The real engine had earlier been tested in the standard mode for the 13 ESC cycle points [13. Appendix 1 - ESC cycle], the data was compared with the results from the full engine simulations. This was done to calibrate the model so it performed like the real engine. The combustion model was modified as in the single cylinder engine model. DiJet combustion was used. This method was chosen because different compression ratios together with different levels of Miller give different heat releases than the measured standard engine HR. Other injection parameters like the single cylinder were inserted. The first test that was set up for the full engine model was aimed at finding a more exact highest efficiency.

The model was set at 265 kW (360 Hp) and 1600 rpm. The compression was varied from standard 17.3:1 to 30:1 in CR in steps of one. In Figure 16 every other CR are present with engine energy balance. With a higher compression ratio the heat transfer part in the energy balance increases together with the friction losses and the energy in the exhaust decreases. A compression of 23:1 gave the highest efficiency. The reason efficiency does not increase with compression is due to the heat transfer and friction losses increasing. The piston bowl gets smaller with compression and increases the risk for wall wetting. It is also harder to get the air to mix with the fuel spray when there are smaller clearances in the combustion camber between the walls and the spray. Lower lambda areas are formed with soot emission problems as a result.

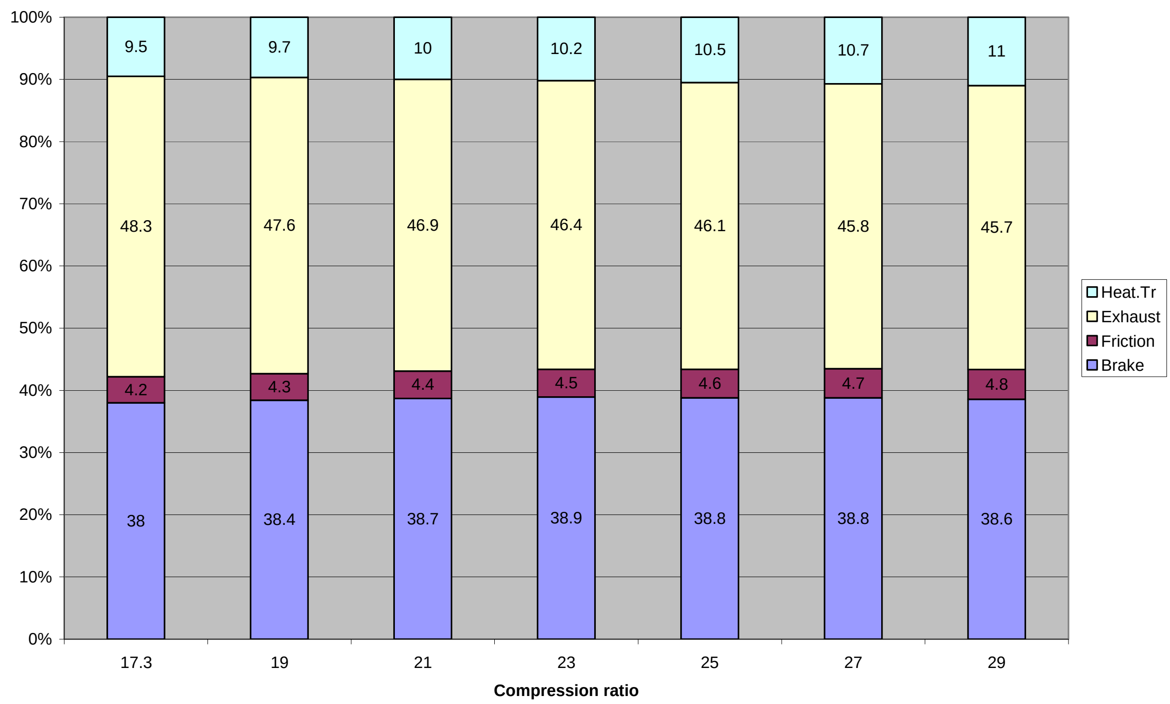

Figure 16 Engine energy balance on a DLC6 with different compression ratio at standard valve profile.

The full engine model was run for all the 13 ESC cycle points with fixed Miller cycles. When the engine speed was increased a higher exhaust mass was produced, the inlet mass was also higher. This gave a better operation point for the compressor, inlet pressure increased. Table 1 shows a higher power output for the full engine model with a standard camshaft in the low engine speed range and high load. With high and low engine speeds and loads the Miller engine with the higher compression ratio has the highest power output for the same injected fuel per cycle. When a higher level of Miller (later closing of the inlet valves) is used BMEP decreases. The engine demands a better turbosystem to deliver the same BMEP. More about this later in the chapter on Turbo matching.

Table 1 Difference in engine power between different strategies during a ESC cycle. The yelow field is the highest BMEP for the specifc load point. The yellow cells are for the higher BMEP outputs.

		BMEP		
Speed [rpm] - Load [%]	Standard Cam 17.3:1	Miller 17.3:1 IVC +32°	Miller 23:1 IVC +32°	Miller 26:1 IVC +32°
500 - 0	0	0	0	0
1250 - 100	21.5	17.3	15.6	14.9
1600 - 50	8.7	8.67	9.05	9.15
1600 - 75	13.9	13.8	14.2	14.2
1250 - 50	10.9	10.8	10.9	10.9
1250 - 75	16.2	15.8	15.3	14.3
1250 - 25	4.73	4.7	4.85	4.91
1600 - 100	18.4	18.2	18.5	18.4
1600 - 25	4.03	4.05	4.2	4.25
1950 - 100	12.9	13	13.7	13.8
1950 - 25	0.71	0.794	0.344	0.395
1950 - 75	9.18	9.15	9.65	9.81
1950 - 50	5.51	5.51	5.76	5.85

The chosen compression ratio for the new piston design was 23:1. This is a balance of CR that has shown the highest efficiency and what is physical possible with the combustion camber and injector limitations.

8.2. Engine and test cell

The engine that was used for testing was a single cylinder research engine built on a Scania D12 unit cylinder. The cylinder had 127 mm bore, 154 mm stroke which gave 1.95 litres in swept volume. The cylinder head was a modified standard cylinder head from a DL engine with 130 mm bore. This cylinder head was chosen because the Lotus AVT valve actuators fits these cylinder heads better and only smaller modifications were needed to install this system on the head. The Swirl ratio of the cylinder head was also modified to 1.4 this had been done earlier for another test setup. After discussions it was concluded that the lower swirl ratio wouldn’t have a great influence on planed tests, standard swirl is 2.1.

Two different pistons where used during these tests. The standard piston was a Euro 4 which had the same piston bowl geometry as a euro 5 but scaled to fit the 127mm cylinder with 17.3:1 in compression ratio, and one prototype piston that was manufactured after simulations in GT-power, see Figure 17. A compression ratio of 23:1 was chosen. The piston bowl design is the same as that found on a standard 17.3:1 piston. The same injector with a 16° spray angle was used in both tests. This gives limitations in the design of the bowl. The radius of the ridge

that splits the diesel sprays in two half when the sprays meets the piston lines up to the centerline of the spray from the injector in both the standard piston bowl and the modified piston bowl. The highest point in the piston bowl, right under the injector was raised by 3 mm and the lowest point in the bowl was also raised without altering the geometry of the rest of the bowl. This was done to limit the decrease in the total radius of the bowl to prevent wall wetting. The total radius however needed to be decreased to go up in compression ratio. If another spray angle was possible then the bowl could be designed with a larger radius and a shallower bowl, this would prevent wall wetting.

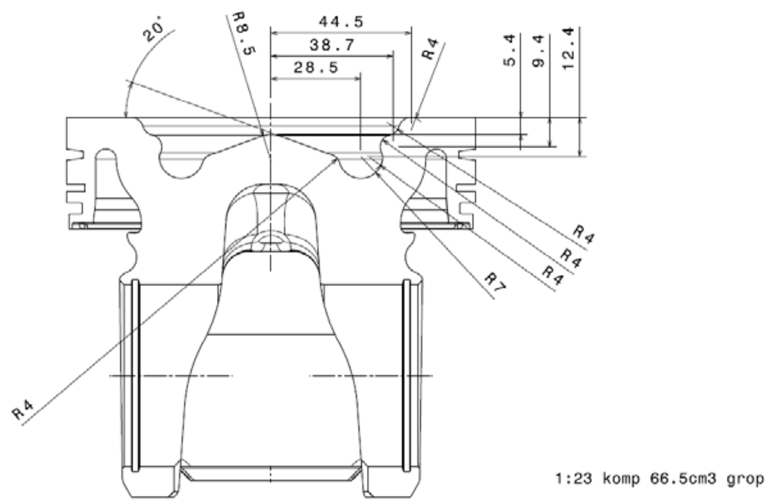

Figure 17 Prototype piston with 23:1 in compression ratio

Scania XPI injection system was used with a crankshaft driven injection pump from a full engine. This meant that when a higher injection pressure was required the pump power consumption increased which decreased the delivered torque to the test cell dynamometer. The injector used was a 207 pph with a 8 hole nozzle, and spray angle of 16°. A standard fuel rail was used and outlets to the other injectors were plugged. This gave some problems with fluctuating rail pressure when the total fuel flow rate was lower than a full engine with all injectors connected. Adjustments of the PID-regulator in the engine management system reduced the problem somewhat. The fluctuations that still arose were between +/- 10 to 15 bar but due to the measurement time of 90 seconds this did not have any significant effect on the results.

To control the different Miller valve lift profiles a fully variable valve lift profile system was used, the Lotus AVT. This opened the possibility to fully design the inlet valve opening profile in duration, lift height, valve acceleration etc. A camshaft controlled the exhaust valves.

The Lotus AVT is a hydraulic system that drives the inlet valve directly with hydraulic pressure, see Figure 18. The system replaces a conventional camshaft with hydraulically operated actuators and servo valves. One double acting hydraulic actuator per valve sits on the cylinder head. Co-axial with the actuator is a hydraulic piston mounted on top of a standard poppet valve. To accurately record the position of the valve a displacement transceiver is situated on top of the piston. A hydraulic pump together with an oil conditioning system delivers the high pressure oil to the actuator valves. The actuators have double working cylinders and no return spring is needed.

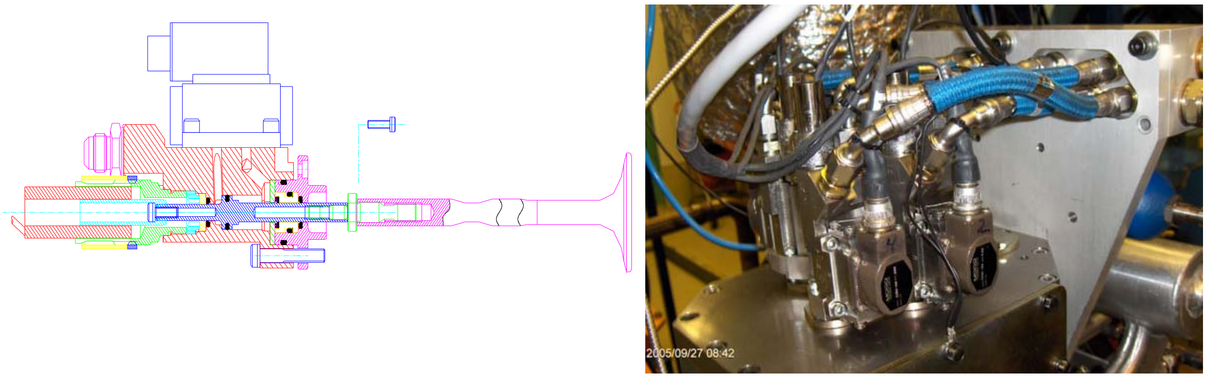

Figure 18 The Lotus AVT valve actuator with inlet valve(on left) and the actuators on the cylinder head (on right).

The Lotus Active Valve Train (AVT) system has been in use since 1990. The electronically controlled and hydraulically operated system controls valve lift by a digital signal processor based controller. This controller even includes a closed loop feedback system to achieve the desired valve lift. The lift profiles can be trapezoidal, conventional polynomial, double lift or user defined. These profiles are then used during a four stroke cycle over 720 degrees. Profiles can be modified through the use of the GUI and stored for later use. The profiles can even be modified whilst retaining the original acceleration and velocity characteristics.

The valve profile can be generated by GT-crank train or Lotus own Matlab program, engine valve profile generator (EVPG). EVPG generates a series of valve opening profiles that are stored in a map structure. From this map the valve profiles can be downloaded and used by the system by just giving a location number in the map. Valve profiles can be shifted whilst the engine is running. Control parameters of the system are accessed through the user interface program which runs in a Matlab environment on a PC. Figure 19 shows some of the most important dialog boxes which are used to define and control the AVT system.

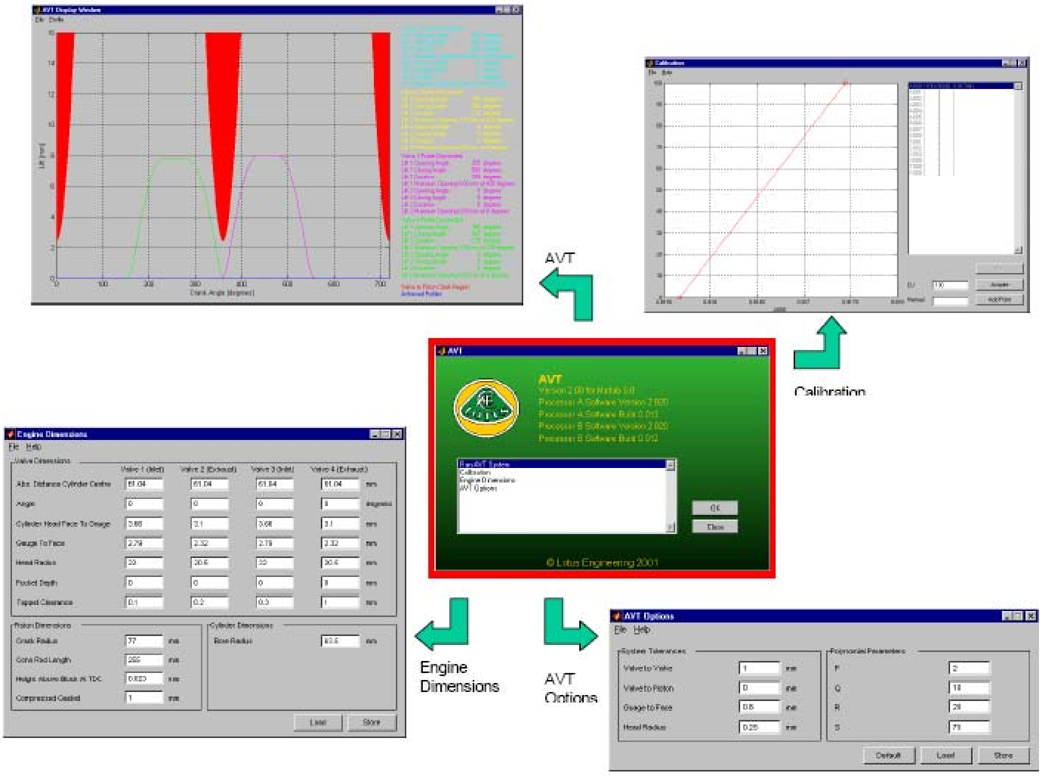

Figure 19 Dialog boxes in the lotus AVT system

Figure 20 gives a graphic description of the relationships between hardware and software.

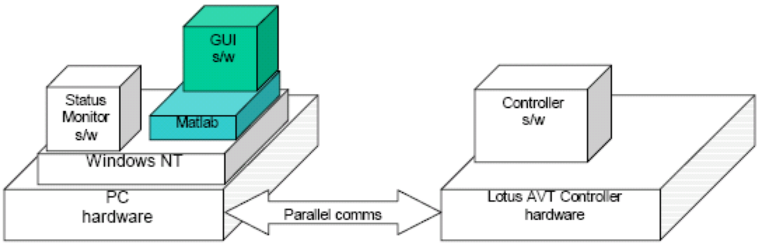

Figure 20 Diagram of hardware and software

8.2.1. Measurement Equipment

The cell computer system that recorded all the slow channels and controlled the test cell is called PDP, the software was developed by Scania CV. PDP logs all of the measured signals at 10 Hz. A mean value over 90 seconds is then calculated and stored in the PDP computer for every signal. From the test cell a protocol with the measured data is generated with every measured test point. The protocols are then used to analyze the test run. High temperatures are measured with type K- thermo couple and lower temperatures where a higher resolution is required was measured with PT-100 probes. Pressure is measured with different pressure transducers depending on its purpose. Fast pressure measurements are done with Kistler piezo transducer with an operating range up to 250 bar. The signal is logged by AVL Indiset cylinder pressure measurement system. The system also logs the valve position from the AVT system. The fast measurements have a resolution of 0.1 CAD and 100 cycles are logged. Calculations of IMEP, PMEP, Pmax average and so on are sent to PDP and stored in the protocol. Fast measurement files are also stored in the Indiset with pressure trace for all 100 measured cycles and named after PDP protocol number.

Further calculations and presentation of the fast measurements are done in AVL's Concerto. Calibration of the cylinder pressure is carried out with oil pressure scales and the entire measurement chain is thereby calibrated from pressure transducer via amplifier and A/D transformer to the measurement system. This is done regularly. Calibration of the test cell is done every year according to Scania UTTC calibration routine.

AVL smoke meter type 415 is used for smoke measurements. This is a filter type smoke meter for soot content in exhaust gas. The result is displayed in FSN (filter smoke number), the range are from 0 to 10 FSN.

Emission measurements were done with two different but similar emission measurement systems from Horiba. The first test run with 17.3:1 in CR was measured with Horiba MEXA-9200. After the test engine was modified with the new compression 23:1 a Horiba MEXA-7100DEGR was installed and used for measurements. The biggest difference between the two Horiba is that 7100DEGR has a separate calibration gas for CH4 and high respective low span gas for HC and CO. This will give higher accuracy in the emission measurements. However, a problem with the Horiba installation resulted in different emission measurements levels which were discovered after the second test run was completed and a reference measurement was done with 17.3:1 CR and the new Horiba. To be able to compare the measurements with different CR (and different Horiba measuring equipment), a correlation factor for the difference in emission levels was calculated by re-running all the load points

with a standard valve lift profile and comparing to the old values. This was done because there was not enough time left in the project to re-run all of the load points with all of the Miller profiles again.

The EGR loop was cooled to 80°C and an EGR pump delivered the exhaust to the inlet manifold. The amount of EGR was measured with the Horiba emission measuring equipment by recording the CO_2 emissions in the inlet manifold and comparing them with the CO_2 emissions in the exhaust manifold.
To ensure the emission data was correct with regards to the amount of injected fuel and air mass, a carbon balance was calculated. Measurements from the fuel and air mass consumed were compared with the emission data and the correlation between these values can be seen in PDP protocol data number RL15. If everything is correct RL15 will show 0 % error, and typically +/- 2 % error is acceptable before suspicion in measurement error can be assumed.

8.3. Method of testing

A test protocol was made (see Table 2) with different load points for the single cylinder test engine run in the test cell. All load points were run with and without EGR. A case is a load point where EGR may or may not be used, for example case 1 is load point 1 without EGR and case 2 is load point 1 with EGR etc. The test points were chosen to cover the entire engine operation range and where limitations were expected. Load point 1 is at idle and
500 rpm. It was of interest to examine the exhaust temperatures at this point due to the reduced air mass retained in the cylinder when the inlet valves where held open during the compression stroke. All the EGR cases were set to 30 % EGR to reduce the quantity of test points.

The next load point 2 is a low power output, typically 50 kW at full engine operation and 1250 rpm. This was an interesting load point to test since the low power running of the engine where problems with low exhaust temperature leads to difficulties with SCR light off and Oxidation catalytic. Load point 3, 100 kW 1250 rpm (full engine operation) is a load point many haulage trucks run at for longer periods on highways for example.

The 240 kW 1250 rpm point 4 was investigated because this is the full power output for the lowest specified engine in the DL series. One idea that was interesting to investigate is to have a fixed Miller strategy for a low power engine in the DL series. The 300 kW and 330 kW are the two highest power outputs in the engine series (load points 5 and 6 both 1250 rpm). To design a boost system to a “full power” engine, what boost pressure would be required? Another aspect was to see if it is any efficiency profit at high power output. The last load point 7 is at a higher engine speed (1600rpm) and 70 kW. The chosen load points for the full engine were recalculated to BMEP. The single cylinder test engine was then runned at these BMEP targets.

Table 2 The tested load cases

Load condition	Rail press. [bar]	Speed [rpm]	Valve strategy	EGR [%]	Fixed lambda	BMEP [bar]	Full engine power: [kW]
Load 1	400	500	Std to late	0	No	0	Idle
	800	500	Std to late	30	No	0	Idle
Load 2	700	1250	Std to late	0	No	3.77	50
	1200	1250	Std to late	30	No	3.77	50
Load 3	1400	1250	Std to late	0	No	7.53	100
	1800	1250	Std to late	30	~1.84	7.53	100
Load 4	1400	1250	Std to late	0	~1.45	18.08	240
	2400	1250	Std to late	30	~1.38	18.08	240
Load 5	1400	1250	Std to late	0	~1.4	22.6	300
	2400	1250	Std to late	30	~1.36	22.6	300
Load 6	1600	1250	Std to late	0	~1.49	24.86	330
	2400	1250	Std to late	30	~1.34	24.86	330
Load 7	700	1600	Std to late	0	No	4.12	70
	1200	1600	Std to late	30	No	4.12	70
Load 8	1800	1250	Std to early	30	~1.8	7.53	100
Load 9	1400	1250	Std to early	0	~1.6	18.08	240
Load 10	700 bar	1250	Std to early	0	No	3.77	50

The fuel rail pressure was set to 1400 bar without EGR and 2400 bar with EGR, however this parameter was changed under the test period so the rate of increase in cylinder pressure was kept in check on the one hand and the smoke emissions were limited on the other hand. The valve profiles included a standard valve profile followed by at least 5 different Miller levels (see Table 3). Every load point was tested with a SOI angle similar to a standard production engine and with a SOI for MBT (Maximum brake torque).

BMEP calculations were used to find the same load conditions in the test cell. When the right parameters were set the on-time was held constant under all measurements at that load point. No pilot or/and post injection was used, this was to reduce the complexity in the testing and the amount of tuning parameters.

For the low load points where boost pressures are not needed, lambda values were allowed to drop at higher Miller levels. When boost pressure was required the lambda was held constant by means of a higher inlet pressure to be able to compensate for the lower volumetric efficiency.

Without EGR the exhaust pressure was set to the same as the inlet pressure (1:1). When EGR was used the exhaust pressure was set 200 mbar higher than the inlet pressure. This was done to correctly model a full engine.

Start of injection, SOI used in this report is the start of the electrical signal that is sent to the XPI injector. In Figure 21 the heat release is plotted together with the injector signal (electrical signal to the injector) and the fuel based signal when the actual injection happens. The SOI delay between electrical signal and the proper fuel injection at 1250 rpm is typically 4.4° CAD for 1400 bar common rail pressure and for 2400 bar 3.5° CAD. For 1400 bar the delay is 5.5° CAD and for 2400 bar 7.2° CAD. This means that in the case, 1250 rpm, 1400 bar, SOI -5° and ontime 1.09 ms start of injection is around -0.5° CAD and end of injection 8° CAD.

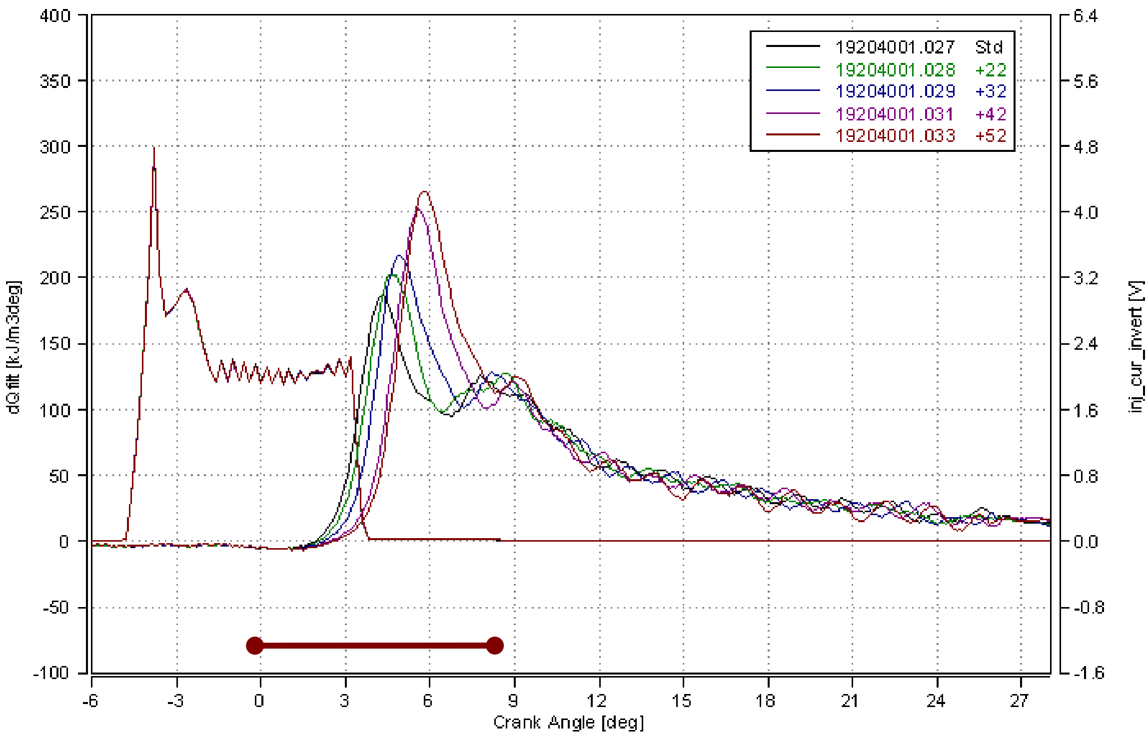

Figure 21 Heat release and injection signal without EGR and 100kW

The needle lift signal is not measured because of technical difficulties, especially when the injector needle is hydraulic controlled and the surrounding pressure is very high. Actual injector rate measurements have however been done before on this type of injector and this data was used to calculate the actual injector period

The indicated cylinder pressure is compensated in the cylinder pressure indicating equipment (AVL Indiset advanced). This was done because the piezo cylinder pressure transducer does not have any fixed zero charge, it drifts in charge capacity by the inperfect isolation between the transducer and amplifier. The transducer operates under dynamic pressure with thermal fluctuations that influence the measured data. The Indiset compensates for this behaviour when the inlet valve is closed and the compression stroke begins. Adiabatic compression is assumed between -110° and -90° CAD before TDC. The pressure difference is calculated. Due to the adiabatic compression it is assumed that there is no heat transfer and Equation 5 can be applied. The pressure is then calculated from the geometric difference in volume between this two CAD points and then compared with the measured cylinder pressure. An offset is calculated and included in the following readings. The calculation method is described by A.L. Randolph in [12]. This leads to some problems when LIVC is used (Miller cycling). The intake valve is still open under these CAD points, the compression is lower due to the airflow out from the valve port and the adiabatic assumption can not be used in this interval. To correct this error the same assumption is made but with different CAD points.

$$p \cdot V^{\kappa} = \text{constant} \quad \textbf{(5)}$$

V = Volume
p = Pressure
κ = Specific heat ratio

In Concerto this is done in CalcGraf, see Figure 22. The inlet valve lift profile is read and the CAD for the valve shutting is calculated with the module "Injection timing". The CAD value is then used to determine a new start CAD for the adiabatic compression that will be calculated. In this file a software filter is inserted for the cylinder pressure.

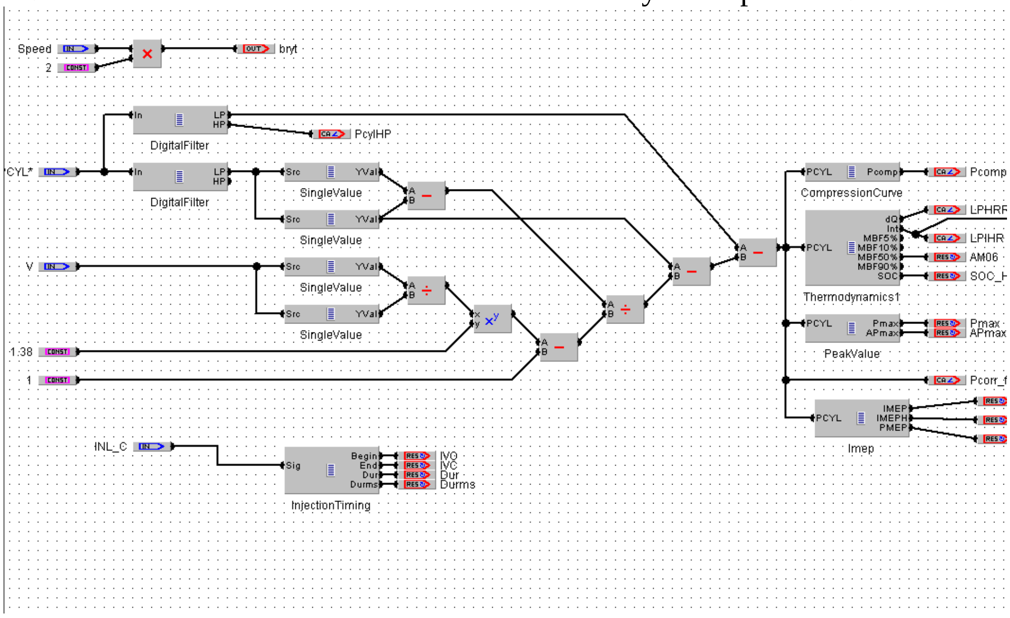

Figure 22 Concerto CalcGraf with the cylinder pressure compensating tool, software filter and HR calculation

The adiabatic compensation calculation was made between 36° and 41° after IVC (under 0.7 mm lift), and is carried out at different CAD depending on the level of Miller used. 36° was used because of the interruptions that arise when the valve is closing against the valve seat.

An investigation into the duration which adiabatic compensating could be used was done. A length of just 5° CAD gave almost the same result as with 20° CAD when tested with standard valve lift profile. The shorter length was used because when a large degree of Miller that are used a relatively short CAD before TDC gives larger heat transfer interference. The difference between indisets cylinder pressure and the compensated cylinder pressure in Concerto is seen in Figure 23. The signal was also filtered in Concerto.

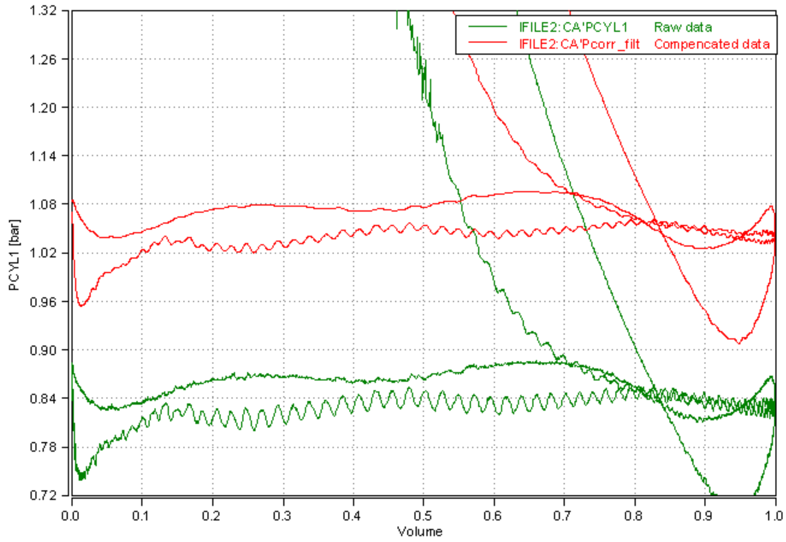

Figure 23 Uncompensated and compensated cylinder pressure in pump stroke

To verify calculations for compensations in cylinder pressure a comparison between results from GT-Power and compensated measured pressure was made. The GT-Power model was tested with the same boundary conditions as in the test cell. The air mass, delta and backpressure were set in the model after the measured data. In Figure 24 the measured and simulated cylinder pressure is plotted in a P-V diagram. A standard valve lift profile together with Miller profile 42° LIVC can be seen in the figure. At BDC the cylinder pressure is lower for the Miller case despite the inlet pressures which are set at the same level. The correlation between the measured data that is compensated and the simulated data can clearly been seen at BDC. The model to compensate the pressure when late valve closing are used can thereby be assumed to be accurate.

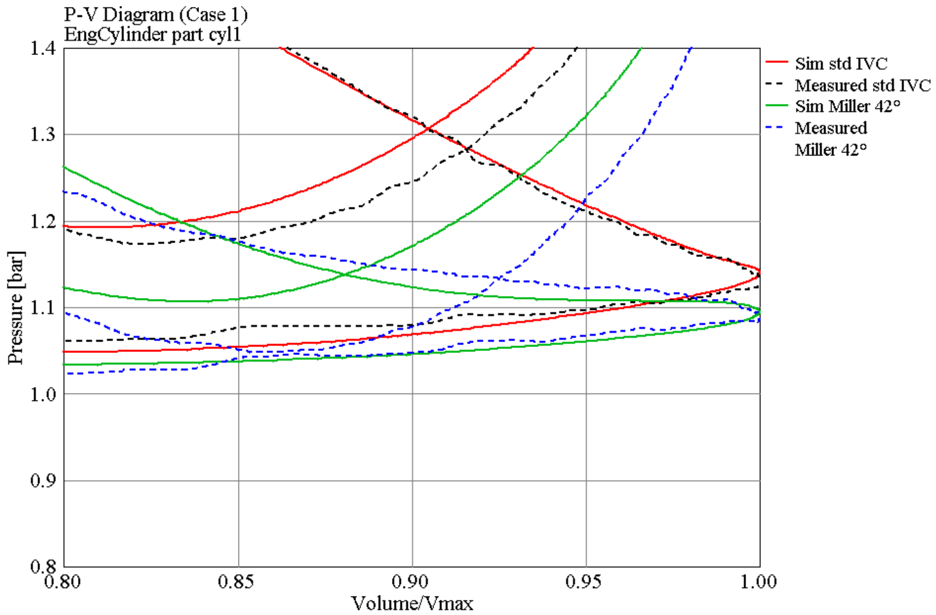

Figure 24 P-V diagram on a part of pumpcycle for measured and simulated data

9. Result

Initial tests were carried out with the standard compression ratio of 17.3:1. The Lotus AVT system was loaded with a file which contains all of the lift data for the standard lift profile plus all of the possible Miller profiles. It was thereby possible to choose any number of degrees which the inlet valve should be held open. It was possible to alter the closing time without the need to stop the engine, this was important because it was now possible to maintain relatively similar pressure, temperature and EGR levels from case to case. In Table 3 a description of the Miller level that is used in following diagram are presented.

Table 3 Valve profile nomenclature for the diagrams

Valve profile:	LIVC:	EIVC:
1	Std	Std
2	22°	-55°
3	32°	-65°
4	42°	-75°
5	52°	-85°
6	62°	-94°
7	72°	

9.1. Load point 1

The idle quality was investigated with and without EGR together with different valve profiles. The rail pressure was set so that the smoke emissions were within a reasonable range for the 17.3:1 CR case and were subsequently held at the same level for the 23:1 CR case, see Table 4. Start of injection (SOI) was set to -2.6° and some reference points with different SOI were also measured. Inlet air temperature and intake pressure were held constant at ambient conditions.

In the column "Miller Strategy" the range of tested Miller levels can be seen. Std stands for standard valve profile, the same valve profile that is used in production engines today. When a Miller valve profile is used the inlet valve is held open 10° longer between different Miller valve levels except for the first profile which is held open for 22° (ie +22°, +32°, +42° etc). The amount of injected fuel was held constant by using the same ontime and rail pressure between the different Miller levels and compression ratio. The ontime was set by using standard valve profile, 17.3:1 CR and SOI -2.6° so that the torque output was 0-1 Nm. The torque was subsequently logged for the different Miller cases. Typically higher torque outputs were measured with higher CR.

Table 4 Test parameters for Idle

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	500	0.83	400	-2.6	24.9 to 26.1	0.1	0.15	92 - 121	8.3 - 5.38	0	Std to 62° later IVC
23.0:1	500	0.83	400	-2.6	25.7 to 26.9	0.1	0.15	96 - 111	8.6 - 6.7	0	Std to 62° later IVC
17.3:1	500	0.59	800	-2.6	24.6 to 25.8	0.1	0.3	110 - 121	8.3 - 5.38	30	Std to 52° later IVC

The exhaust temperature increases with a higher Miller level. The combustion quality is on the other hand reduced with the higher Miller levels because of the lower effective compression with the 17.3:1 CR cases. This is seen in the emissions, both CO and HC increase with higher Miller level, see Figure 25. The higher exhaust temperature and the larger amount of HC and CO can be a benefit if a combined CRT and SCR catalyst is used. The HC and CO levels are reduced in the oxidation catalyst raising the exhaust temperature. This can make the HC injector in the SCR catalyst system (if fitted) unnecessary.

The amount of NO_x decreases with a rise in Miller level for 17.3:1 in CR which means that the catalytic efficiency of the SCR does not need to be as high in the Miller cases as with a standard valve profile. This can be a benefit under transient cycling when the exhaust system is heated. A problem can arise though during longer periods of idling, the catalyst has a light off temperature of 200°C and the exhaust temperature is lower than this.

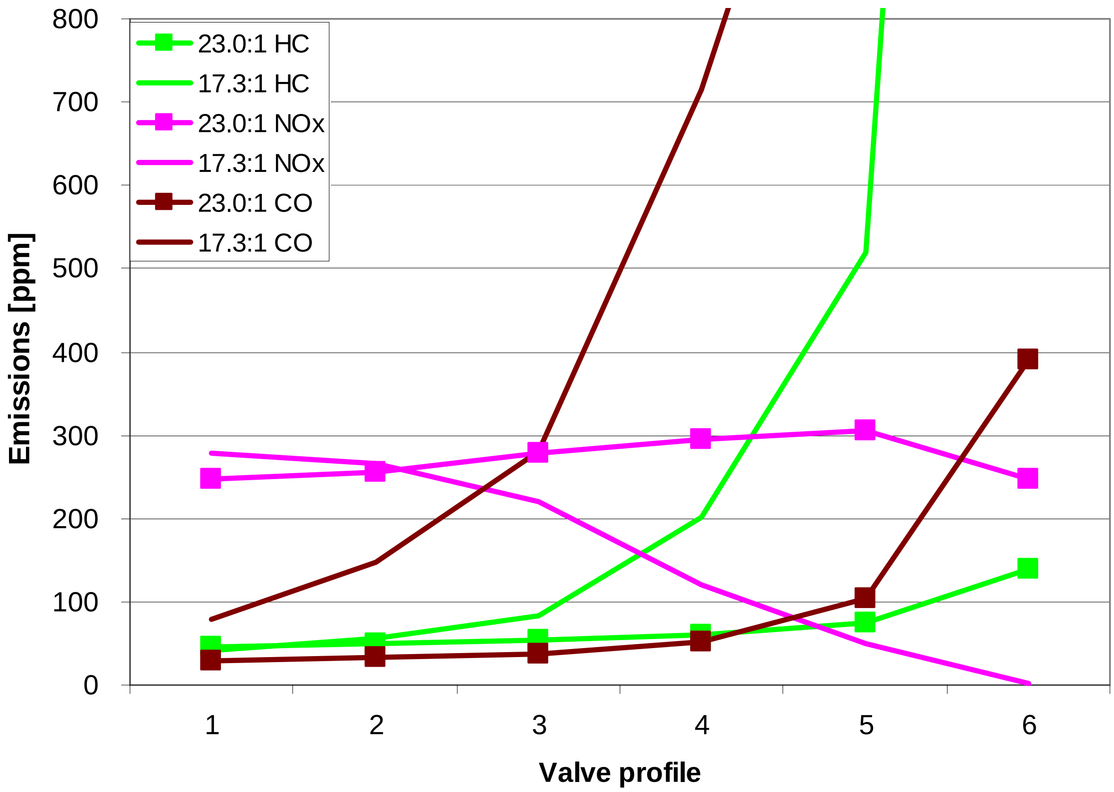

Figure 25 Emissions at 500 rpm, idle without EGR, 23:1 and 17.3:1 in CR

The difference in emissions at idle with higher compression ratio is obvious, see Figure 25. At high Miller levels the HC and CO emissions are much lower with a higher compression ratio. The problems with missfire at very high Miller levels are no longer found with the higher CR. NO_x emissions are lower with a lower CR at high Miller levels. This can be explained by the problems with missfire that arise with 17.3:1 in CR. The exhaust temperature is just little higher with CR 23:1. The reason exhaust temperature increases at higher Miller levels and CR 17.3:1 is the slower HR and a lot of after oxidation during the expansion stroke. The SOI angle needed to be moved earlier to avoid missfire with a combination of 17.3:1 CR and valve profile 6.

The smoke emissions are higher with the higher CR, see Figure 26. This can be explained by the smaller piston bowl of the 23:1 CR piston compared to the 17.3:1 piston. The same rail

pressure, 400 bar, was used in both cases. A higher rail pressure gives a smaller penetration depth in the combustion camber. To reduce the number of variables the rail pressure was held constant. The rail pressure was not set higher because the increase in cylinder pressure with the standard CR should not be too great. Pilot injection was omitted to reduce the number of tuning parameters.

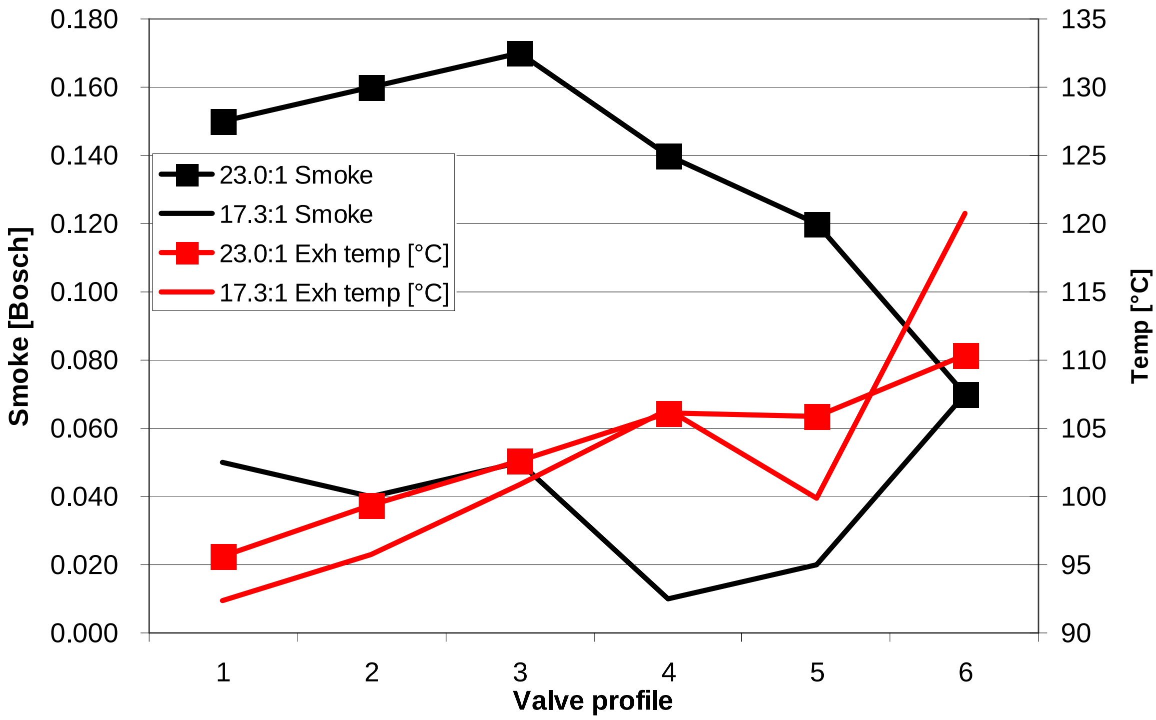

Figure 26 Smoke and br.eff. at 500 rpm, idle, without EGR, 23:1 and 17.3:1 CR

The same phenomenon in emissions that was seen in the case without EGR appeared in the case with EGR and 17.3:1 CR, see Figure 27. The exhaust temperature, HC and CO increased with higher Miller levels. In this case if SCR is not used the higher emission levels are no longer of any benefit. Measurements were only taken for EGR with 17.3:1 CR.

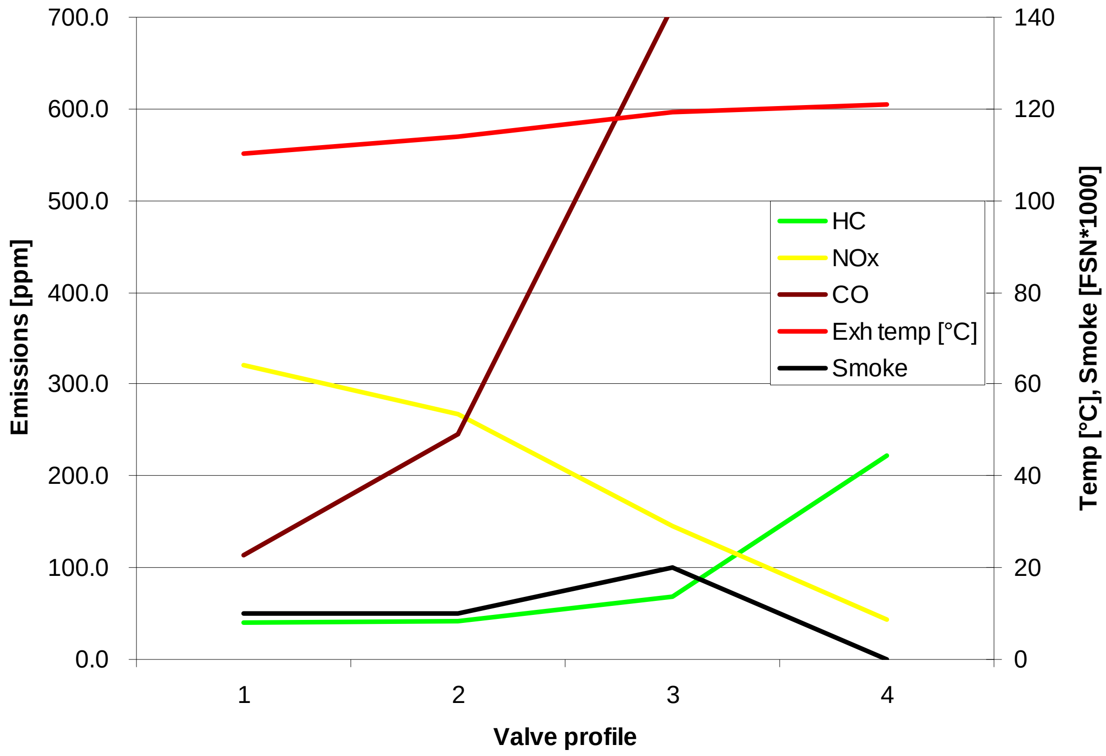

Figure 27 Emissions at 500 rpm, idle, with EGR and 17.3:1 in CR

9.2. Load point 2

Equal to a 50 kW power output for a full engine with and without EGR

Similar to the idle load point the test parameters were set as seen in Table 5. When a Miller profile was used no compensation in inlet air pressure was made, the exhaust temperature was seen to increase at higher Miller values.

Table 5 Test parameters for 50kW full engine power output

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1250	1.14	700	-5	26.7 to 27.8	0.1	0.15	308 - 388	1.94 - 2.71	0	Std to 62° later IVC
23.0:1	1250	1.14	700	-5	26.1 to 27.8	0.1	0.15	305 - 413	2.07 - 2.93	0	Std to 62° later IVC
17.3:1	1250	0.84	1200	-5	29.5 to 36.3	0.1	0.3	319 - 447	1.26 - 1.90	30	Std to 72° later IVC
23.0:1	1250	0.84	1200	-5	25.7 to 26.9	0.1	0.3	307 - 424	1.51 - 2.17	30	Std to 72° later IVC

At low load conditions the Miller strategy with late closing inlet valves gives higher brake efficiency (versus the standard valve profile) at 22° Miller level only to reduce for later Miller profiles, see Figure 28. However, a higher exhaust temperature arises with higher Miller level, the CO and NO_x emissions also increase for the 17.3:1 CR cases. The smoke on the other hand decreases with higher Miller levels. A higher exhaust temperature at this load point can be interesting because SCR catalytic converters have problems with the lower exhaust temperature. The NO_x emissions were higher for the 17.3:1 CR cases, one explanation can be

a longer ignition delay and greater premixed combustion that gives a greater heat release together with a lower air mass in the cylinder. This leads to higher local temperatures that gives higher levels of NO_x emissions, see Figure 28.

In the 23:1 CR cases the NO_x emissions decrease only to increase later with higher Miller levels. The ignition delay is shorter with the higher CR for all Miller cases were the air in the cylinder is compressed with the higher compression ratio. This combined with the fact that the combustion flame is nearer the piston walls at the higher CR which takes away much of the high temperature gradient at the boundary of the combustion flame where thermal NO_x emissions are created.

In comparison with the standard valve profile and 17.3:1 CR an increase in exhaust temperature, brake torque, HC and NO_x - emissions can be made by increasing the CR to 23:1 and using the valve profiles nr: 5 and 6. The exhaust temperature can be increased by 100°C whilst increasing brake torque and retaining the same NO_x emissions, see Figure 28 and Figure 29. Gains in brake efficiency of 1.71% were measured for this load point.

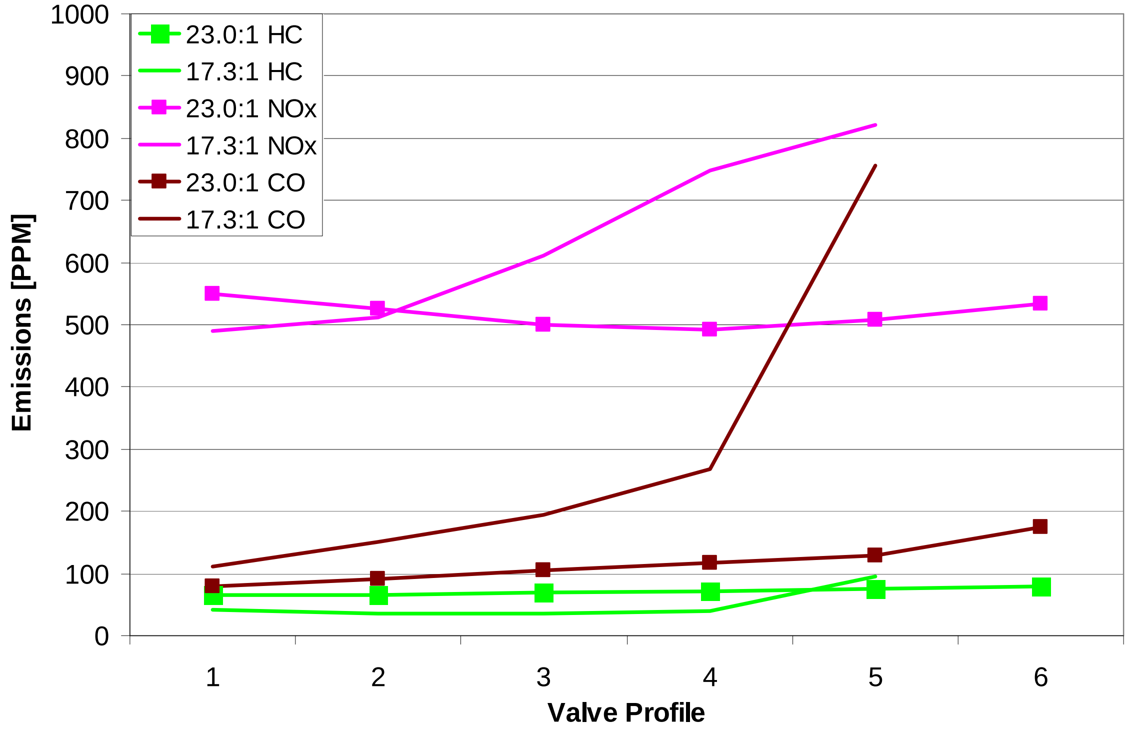

Figure 28 Emissions at 1250 rpm, 50 kW, without EGR, 23:1 and 17.3:1 in CR

Lower smoke emissions arise with higher Miller levels for the 17.3:1 CR cases, see Figure 29. This can also be explained by higher ignition delays and more premixed combustion that give fewer fuel rich zones. For the 23:1 CR cases the soot emissions increase with higher Miller levels until valve profile 4 and decreasing thereafter.

The increases in soot levels can be explained by the smaller piston bowl which was not optimised for low soot emissions in any way. The spray probably hits the piston walls when a lower amount of air is captured in the combustion camber at higher Miller levels. The difference in soot between the two CR cases is obvious. When the SOI angle was altered the NO_x emissions and the power output increased. When NO_x emissions increased with the amount of Miller and the same SOI as the standard inlet profile.

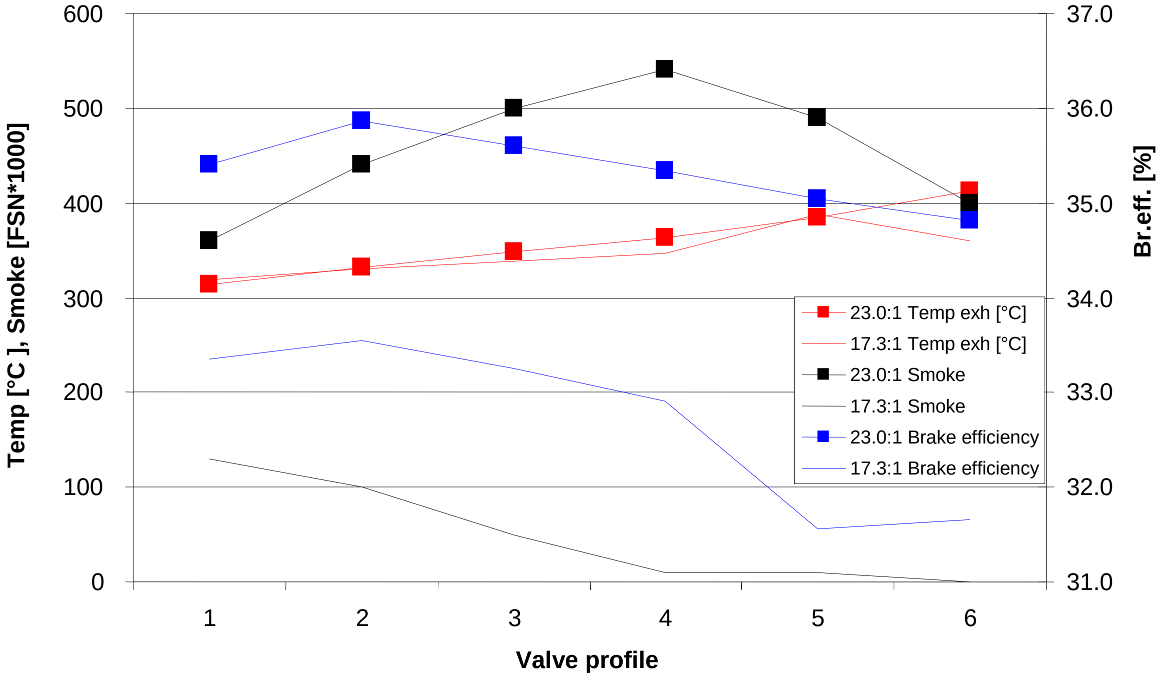

Figure 29 Br.eff, exh. temp. and smoke at 1250 rpm, 50 kW load, without EGR, 23:1 and 17.3:1 in CR

The maximal cylinder pressure decreases with higher Miller levels whilst retaining a similar torque output. The standard valve profile and 17.3:1 CR was compared with 23:1 in compression and the Miller profile (profile nr: 5, 52° later IVC) that gave the same max compression pressure (or nearly the same), see Figure 30. In the 23:1 CR case the torque output is 64.6 Nm and the 17.3:1 CR case had 59.8 Nm with the same SOI, 4.8 Nm difference. The NO_x, CO and HC were nearly the same for both cases, the smoke however was higher and the exhaust temperature 73°C higher. The higher exhaust temperature depends on the lower lambda since a smaller amount of air is captured in the cylinder with Miller.

The heat release is comparable between the two cases although there is nearly the same ignition delay for both CR case, the higher premixed combustion in 17.3:1 case can be explained by the piston bowl. The bowl is smaller in the 23:1 CR case and the combustion flame are thereby nearer the wall and the oxygen access is thereby smaller for the 23:1 case and a smaller premixed combustion arise.

The lower lambda in 23:1 case gives also a little longer combustion. The interesting part is that the compression pressure is lower under the compression stroke with the exception of the end of the stroke when the higher CR gives a higher pressure rise. Less power is needed to do the compression work, see Figure 30. Table 6 shows that IMEP is higher and Pmep lower for the 23:1 CR case which confirms why higher torque output is possible with higher compression ratio and Miller.

Table 6 Indicated mean pressure for the two cases

[bar]	IMEP	PMEP	Pmax
std, 17.3:1 CR	4.84	-0.31	59.33
52° Miller, 23:1 CR	5.25	-0.26	61.21

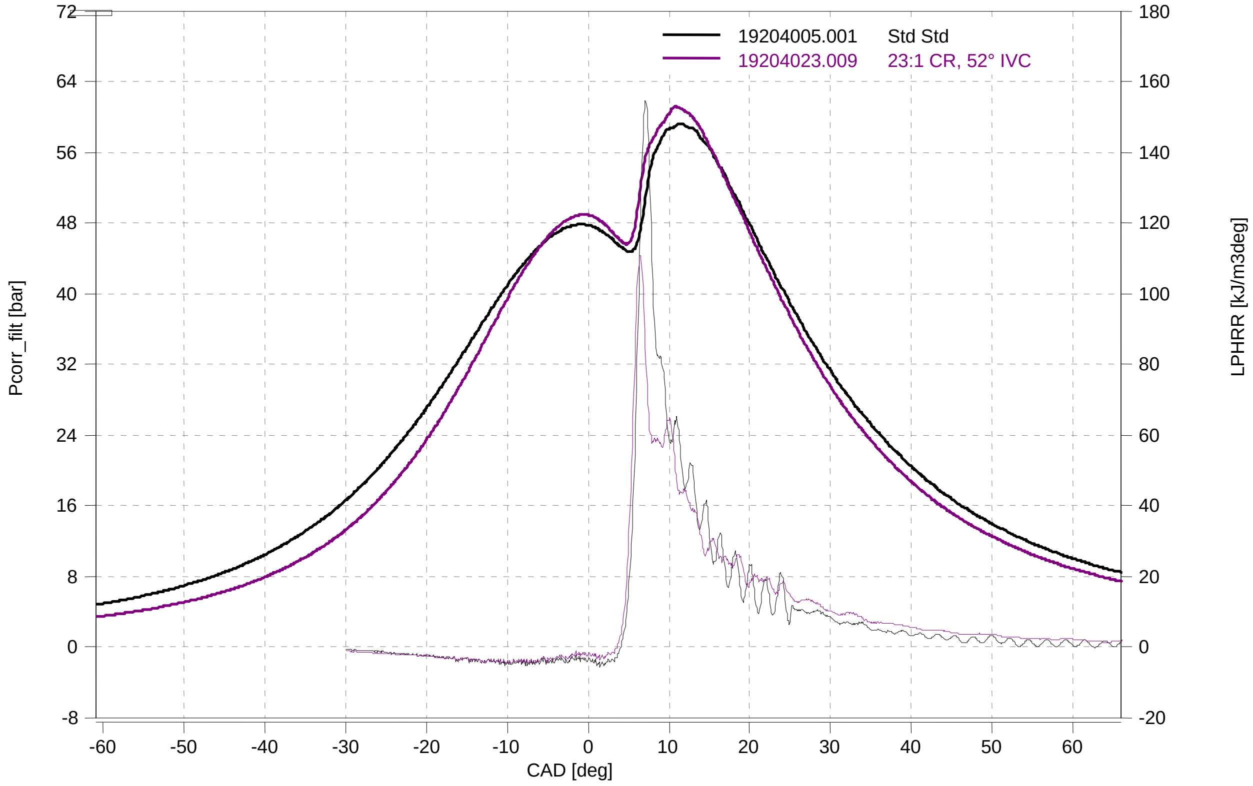

Figure 30 Cylinder pressure and HR at 1250 rpm 50 kW, without EGR, 52° Miller 23:1 and std cam profile 17.3:1 in CR

In the P-CAD diagram (see Figure 30) the pump trace is plotted for the standard case and the 23:1 Miller case. The inlet and backpressure are the same in both cases. What is interesting is that the exhaust stroke pressure in the 23:1 CR case goes down to the same level as the inlet stroke and thereby reducing the pumping loses compared to the 17.3:1 CR case. The standard engine has a greater drop in pressure from the exhaust cycle to the intake cycle, but the pressure level is still higher for the whole inlet stroke compared with the Miller engine. Generally a lower inlet and exhaust pressure for the 23:1 CR case give the lower PMEP, see Table 6.

The inlet valve profiles are plotted in the P-V diagram, Figure 31. When the Miller valve closes the piston speed is higher than in the standard valve profile case this means the air flowing through the valve port also has a greater velocity. The cylinder pressure starts to rise and soon critical flow arises in the valve port just before the valve closes. Some throttle losses are unavoidable. A higher valve velocity could be one part of the solution to reduce these throttle losses.

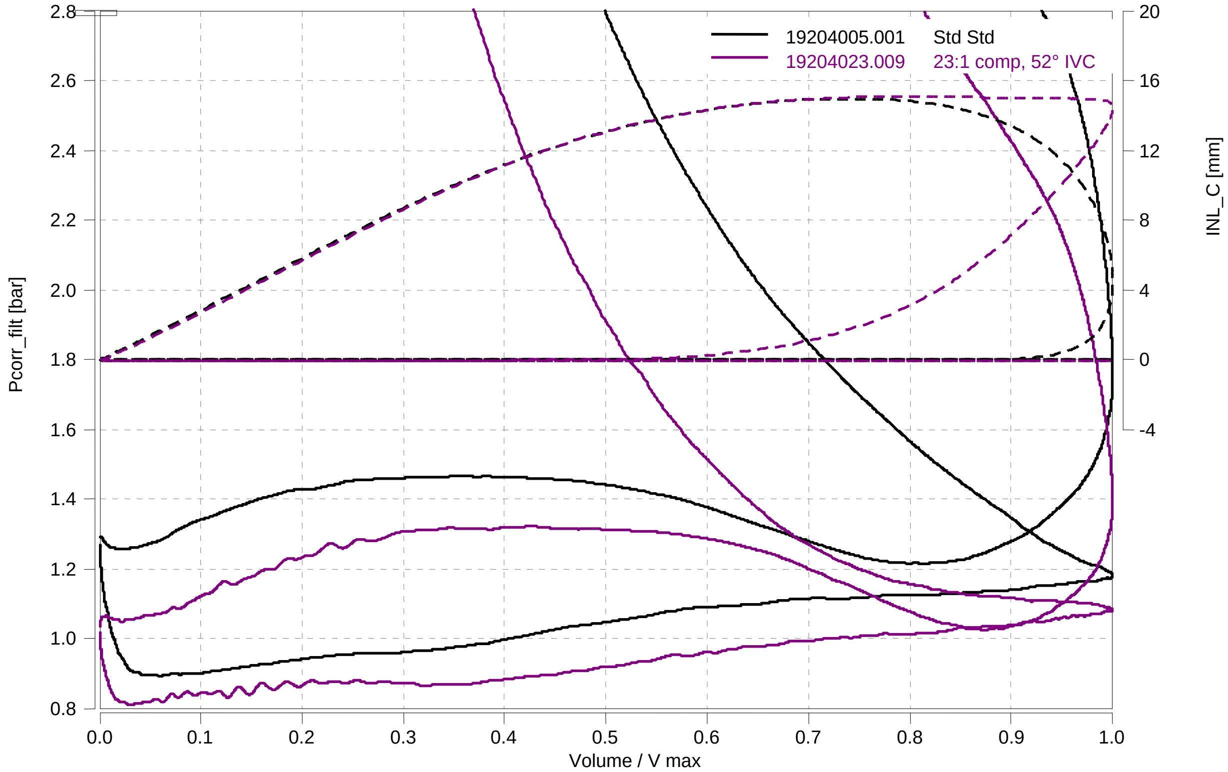

Figure 31 P-V plot for pump cycle with valve profile curves at 1250 rpm 50 kW, without EGR, 52° Miller 23:1 and std cam profile 17.3:1 in CR

In the cases where 30 % EGR is used the fuel rail pressure is increased to reduce the smoke emissions that are naturally higher when EGR is in use. EGR however gives greatly reduced NO_x emissions see Figure 32. NO_x emissions together with the smoke decrease with higher Miller levels for the 17.3:1 CR case. With 23:1 in CR the NO_x emissions are lower compare to 17.3:1 in CR and they decrease with higher Miller levels. That NO_x is lower here in the EGR and 23:1 CR case can be explained by the fact that the walls in the piston bowl are closer and that gives cooling effect to the diffusion flame. The power output decreases with higher Miller level. In the case where SCR is not exclusively used the higher exhaust temperature and the higher HC and CO emissions do not give any benefit to the after treatment system except maybe if a particle filter is used in combination with EGR and SCR.

With Valve profile 5 and 17.3:1 in CR the SOI was changed from -5° to -7° to maintain combustion quality and prevent high CO and HC emissions. In the 23:1 CR case the SOI was kept constant to -5°, the combustion quality was still good. The CO and HC emissions however increased.

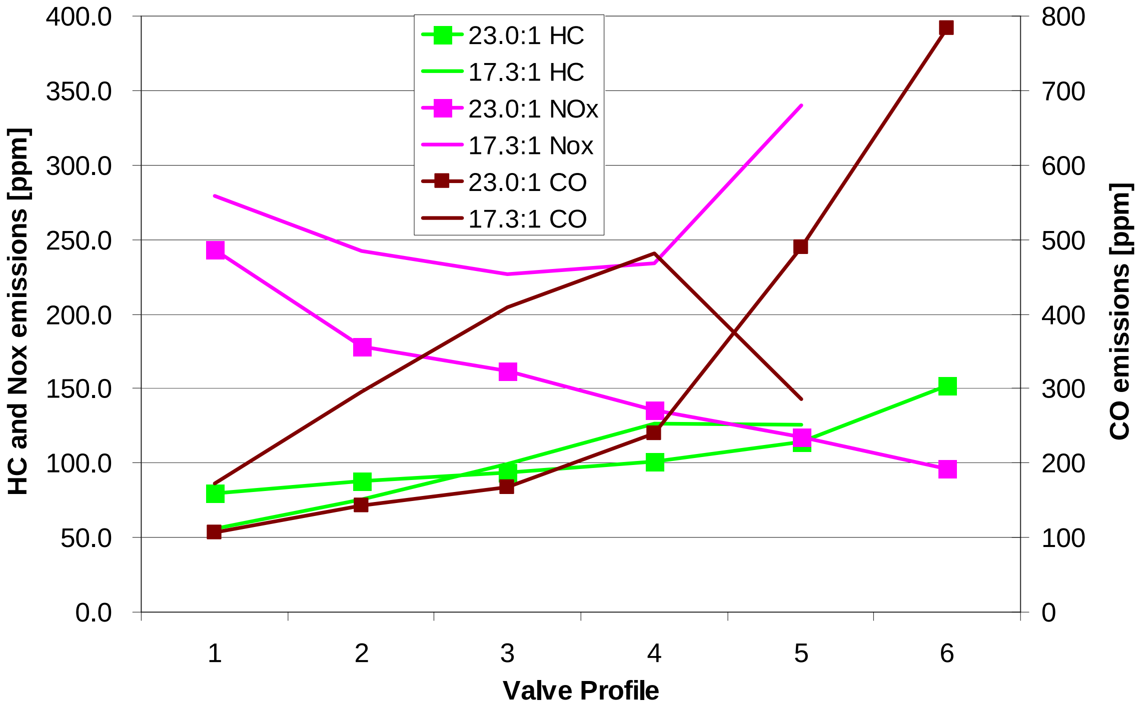

Figure 32 Emissions at 1250 rpm, 50 kW with EGR, 23:1 and 17.3:1 in CR

The exhaust temperature increases with higher Miller levels and can be increased by as much as 100 °C, Figure 33. The difference in exhaust temperature between the two CR cases is very small. Compression ratio does not have a large effect on heat transfer at this load point. Torque is higher with the higher compression ratio dropping when Miller valve profiles are used. The smoke levels are significantly higher for the 23:1 compression ratio.

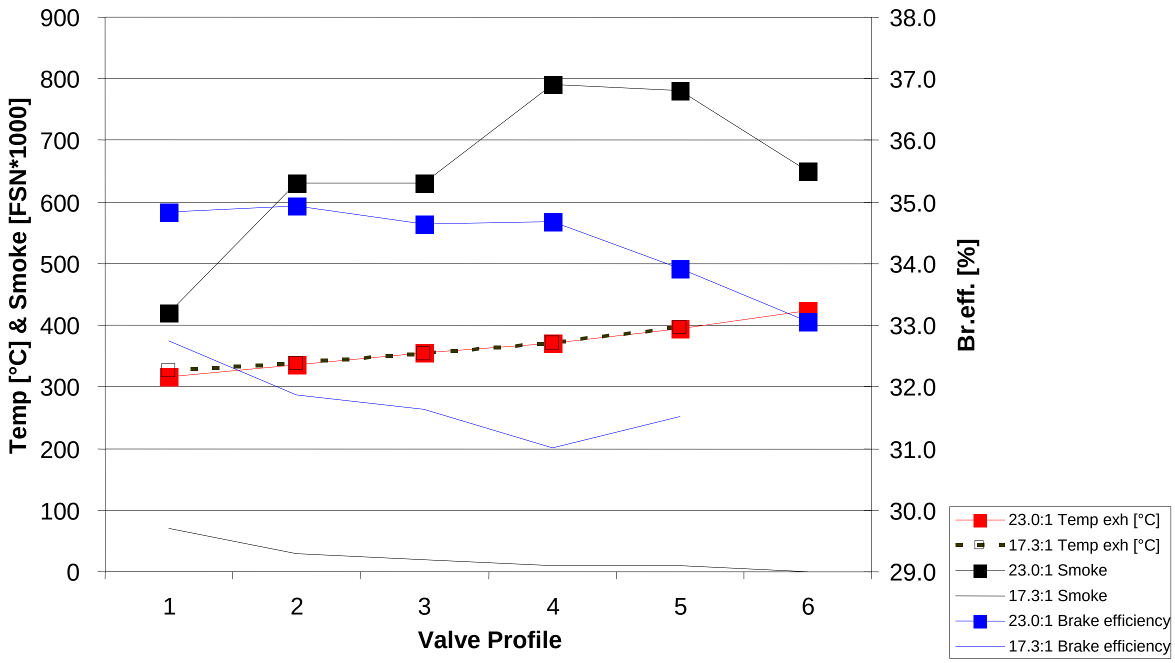

Figure 33 Brake efficiency, exhaust temp and smoke at 1250 rpm, with EGR, 23:1 and 17.3:1 in CR

The problem with the higher smoke emissions from the new piston bowl was investigated to see if it was possible to decrease the smoke levels with a higher rail pressure. In Table 7, the rail pressure was changed from 700 bar to 1600 bar; on-time was reduced so that all cases hade the same torque output. The smoke was quickly reduced with the higher rail pressure, but the NO_x emissions were higher. CO decreased and HC increased with the new rail pressure. It was interesting to see that eta for the engine [%] was better with the higher rail pressure even though the fuel pump was driven by the crankshaft.

Table 7 Difference in emissions with different rail pressures

Rail pressure [bar]:	700	1000	1200	1400	1600
Smoke [FSN]	0.510	0.170	0.110	0.080	0.070
HC emissions [ppm]	68.6	78.0	83.2	87.1	92.3
NO_x emissions [ppm]	513.1	691.5	810.8	946.6	1067.7
CO emissions [ppm]	112	75	65	57	51
Eta engine [%]	35.0	35.3	35.4	35.3	35.5

9.3. Load point 3

Load point three equal to 100 kW power output for a full engine DLC6 was chosen to investigate the cruising power output for a long haulage truck. This point is therefore very interesting because any increase that can be made in efficiency will have a large impact on the fuel economy of the vehicle. The test parameters can be seen in Table 8. The cases without EGR did not have any boost pressure in the first Miller levels. Reductions in lambda were experienced with higher Miller levels. When lambda dropped under 1.3 at Miller level 5 a higher boost pressure was used to lower the smoke emissions.

Table 8 Test parameters for 100kW full engine power output

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1250	1.09	1400	-5	26.5 to 30.2	0.1 to 0.5	0.15 to 0.5	480 to 545	1.59 to 1.30	0	Std to 62° later IVC
23.0:1	1250	1.09	1400	-5	25.9 to 30.1	0.1 to 0.8	0.15 to 0.8	456 to 551	1.43 to 1.71	0	Std to 72° later IVC
17.3:1	1250	0.93	1800	-5	33.1 to 42.7	0.65 to 1.7	0.85 to 1.9	366 - 396	1.70 - 1.75	30	Std to 72° later IVC
23.0:1	1250	0.93	1800	-5	30.2 to 37.8	0.65 to 1.35	0.85 to 1.55	357 - 378	1.86 - 1.9	30	Std to 62° later IVC

In the 100 kW power output case, no benefits in engine efficiency were found at late IVC when standard SOI is in use. The HC and CO emissions however did not increase as much as in the low power and idle cases with 17.3:1 CR when a higher Miller level was used. The emissions were not higher with the later IVC in this case because the boost pressure was raised giving an improved combustion quality.

The NO_x concentration decreased with a higher Miller level, see Figure 34. Of interest is that the NO_x emission level is lower for the 23:1 CR even at this load point, at the higher load points the same trend is observed. The smaller piston bowl together with the lower ignition delay gives the benefit in NO_x emissions. CO emissions increased to the point where the boost pressure is increased. The difference in CO emissions between 23:1 CR and 17.3:1 CR can be

explained by the higher cylinder pressure in the 23:1 CR case that gives a better combustion quality. HC was low with nearly the same emission levels between the two compression ratios.

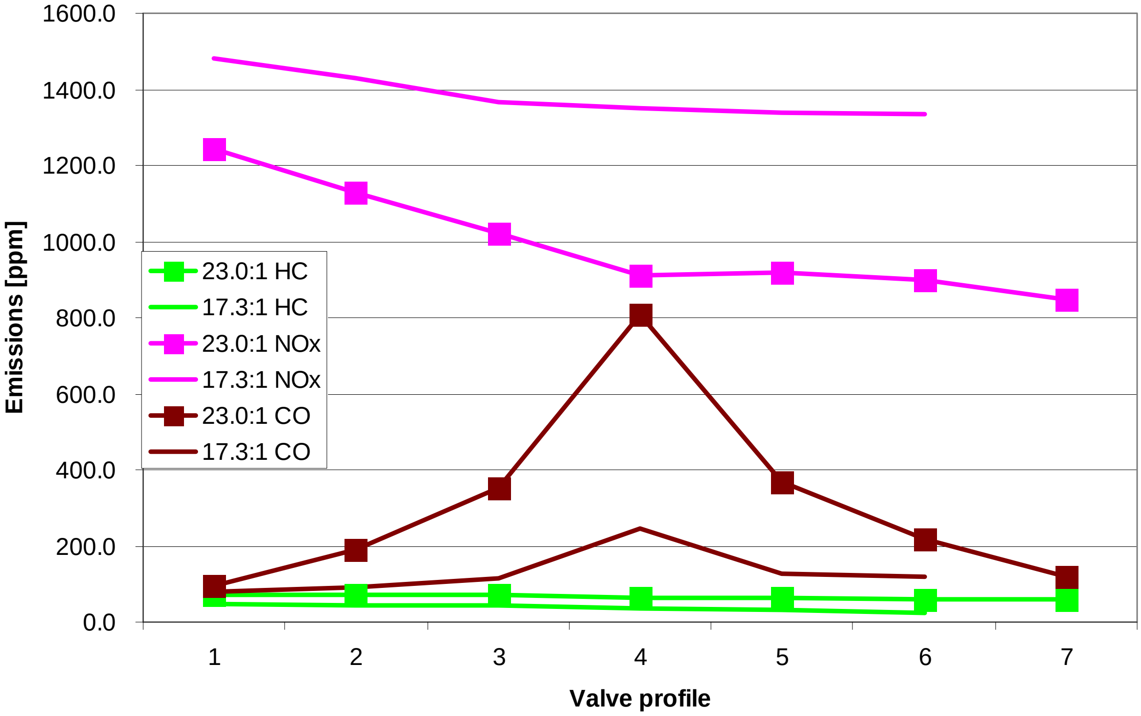

Figure 34 Emissions at 1250 rpm, 100 kW without EGR, 23:1 and 17.3:1 in CR

The exhaust temperature increases with higher Miller levels, the change in the trend in the exhaust temperature curve in Figure 35 is explained by a higher boost pressure with a higher lambda that gives the lower temperature. The exhaust temperature does not deviate between the two compression ratios so the heat transfer to the cylinder wall is not that much higher with 23:1 in CR. The smoke emissions are much higher with 23:1 CR at the point with lowest lambda, valve profile 4. The smaller piston bowl gives this higher smoke level when lambda decreases as the penetration depth of the injection spray increases and reaches the walls of the piston. When the boost pressure was increased the smoke levels were lowered at the same time even though the Miller level is increased, see Figure 35.

The brake efficiencies (br.eff.) were higher for the 23:1 CR case (for the same amount of injected fuel), both cases showed the same behaviour at higher Miller levels. An interesting point is that the br.eff. was highest for the 17.3:1 CR at the standard valve profile . One explanation why the br.eff. decreases is because lambda is lowered, see Table 9. A constant lambda can give a lower to zero drop in brake efficiency. When the boost pressure is increased a trend is seen in higher br.eff., see Figure 35.

Table 9 Lambda at the different Miller levels 100kW without EGR

Miller Level		1	2	3	4	5	6	7
23.0:1	Lambda	1.71	1.61	1.52	1.43	1.54	1.63	1.73
17.3:1	Lambda	1.59	1.49	1.41	1.32	1.44	1.49	

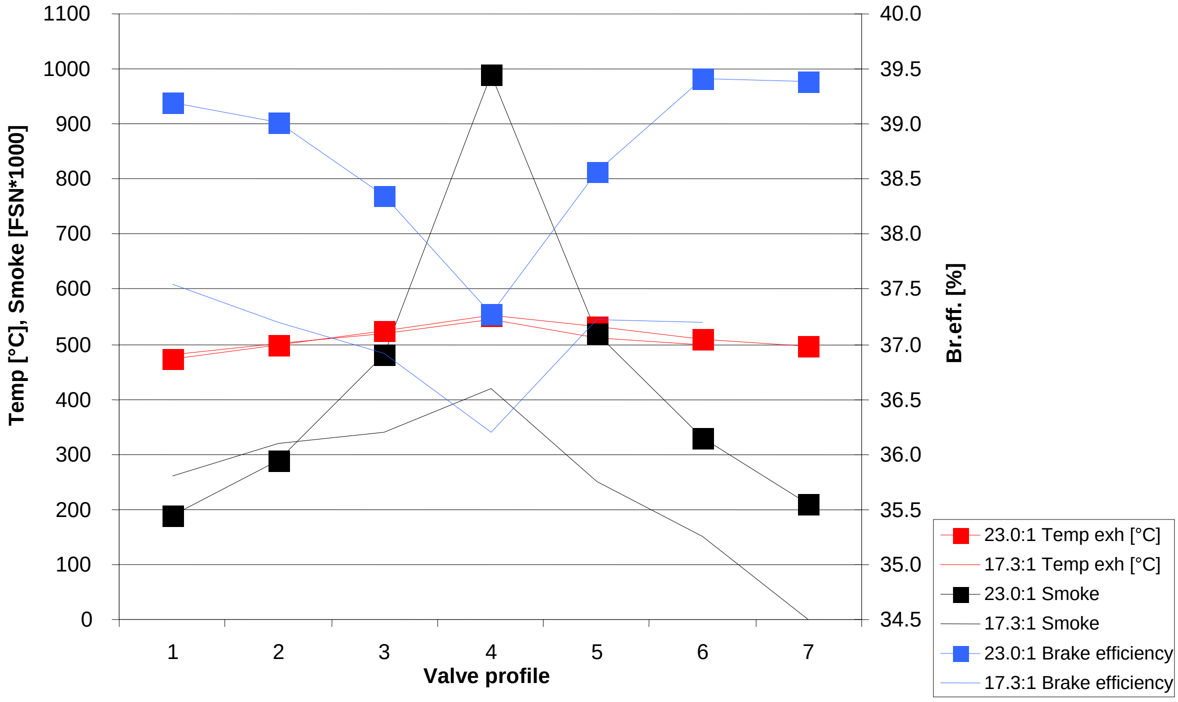

Figure 35 Br.eff., exh. temp. and smoke at 1250 rpm, 100 kW, without EGR, 23:1 and 17.3:1 in CR

Why was the Miller levels compared with the same SOI? A better way to do this would be to have the same combustion phase for all of the tests. One way would be to have 50 % mass burned fraction (MBF50%) at the same crank angle. The reason why this option was rejected can be seen in Table 10 where the start of combustion (SOC) together with MBF50% is presented for the 17.3:1 CR case. SOC is later with more Miller cycling, but 50 % mass burned fraction is earlier with higher Miller levels. This means a higher heat release rate and shorter duration with high Miller cycling, see Table 10. So if the same MBF50% should be used the SOI needed to be set later. The torque output will then be decreased. A comparison with SOI at maximum brake torque (MBT) is however done and presented later in this report.

Table 10 with and without Miller cycle, no EGR 100kW

Miller [°] (Profile)	IMEP [bar]	PMEP [bar]	MBF50% [°]	SOC [°]
STD (1)	8.65	-0.27	10.7	4.25
22 (2)	8.69	-0.22	10.3	4.5
32 (3)	8.69	-0.22	10.05	4.7
42 (4)	8.70	-0.22	10.2	5.3
52 (5)	8.72	-0.25	9.45	5.4

In Figure 36 the cylinder pressure and HR are plotted for a standard valve profile with 17.3:1 CR and two Miller valve profiles with 23:1 and 17.3:1 in CR. If we compare the standard and Miller valve profile HRs with 17.3:1 the longer ignition delay together with the higher premixed combustion peak can easily be seen. The longer ignition delay in the Miller case can be seen in the figure and the pressure trace for the respective cases. When a higher CR is used all the pressure levels increase, when Miller is used the pressure levels are decreased again. Figure 36 shows also that the higher pressure level in the Miller case with 23:1 gives a shorter ignition delay than the standard valve profile case with 17.3:1 in CR. Even if a Miller profile increases the ignition delay, the cylinder pressure together with the cylinder temperature shortens the ignition delay.

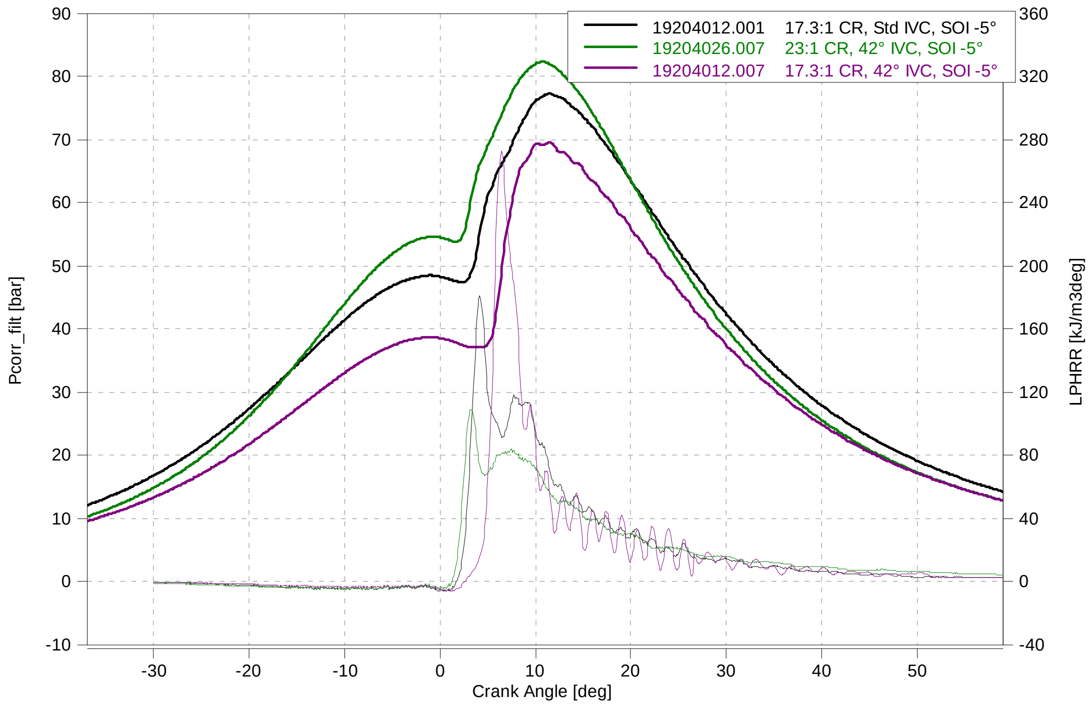

Figure 36 Cylinder pressure and HR at 1250 rpm, 100kW, without EGR, Miller 23:1, Miller and std 17.3:1 in CR

The high smoke emissions at 23:1 CR case was investigated to see if it could be lowered by raising the fuel rail pressure and decreasing the penetration depth on the injected spray.

Table 11 shows the same phenomena as in the 50kW case, different (higher) fuel rail pressures make the smoke level go down and the NO_x emission go up at the same time. The high pressure fuel pump is driven from the crank shaft and for the higher rail pressures the fuel pump needs more power. The total engine efficiency tells us that there is a trade-off in higher efficiency by the higher fuel pressure and the extra energy that the fuel pump needs for the higher pressures. Between 1600 to 1800 bar gives the highest efficiency in this case, see Table 11.

Table 11 Different fuel injetion pressure at 100kW load without EGR

Rail pressure [bar]:	1400	1600	1800	2000	2200	2400
Smoke [FSN]	0.230	0.150	0.090	0.070	0.060	0.040
HC emissions [ppm]	60.0	63.9	67.6	66.4	65.8	63.0
NO_x emissions [ppm]	1090.4	1201.0	1341.3	1467.6	1607.4	1724.8
CO emissions [ppm]	155	108	85	79	68	66
Eta engine [%]	39.4	40.1	40.1	40.0	39.7	39.4

With EGR

In the EGR case with 30 % EGR a constant lambda was held from the standard valve profile to the last Miller valve profile. The emission trends are easier to follow than the cases without EGR. NO_x levels decrease with higher Miller levels and are lower for the higher compression ratio, see Figure 37. HC is relative constant, CO increases with Miller level. Even in this case with EGR the CO level is higher for the 17.3:1 CR than the 23:1 CR.

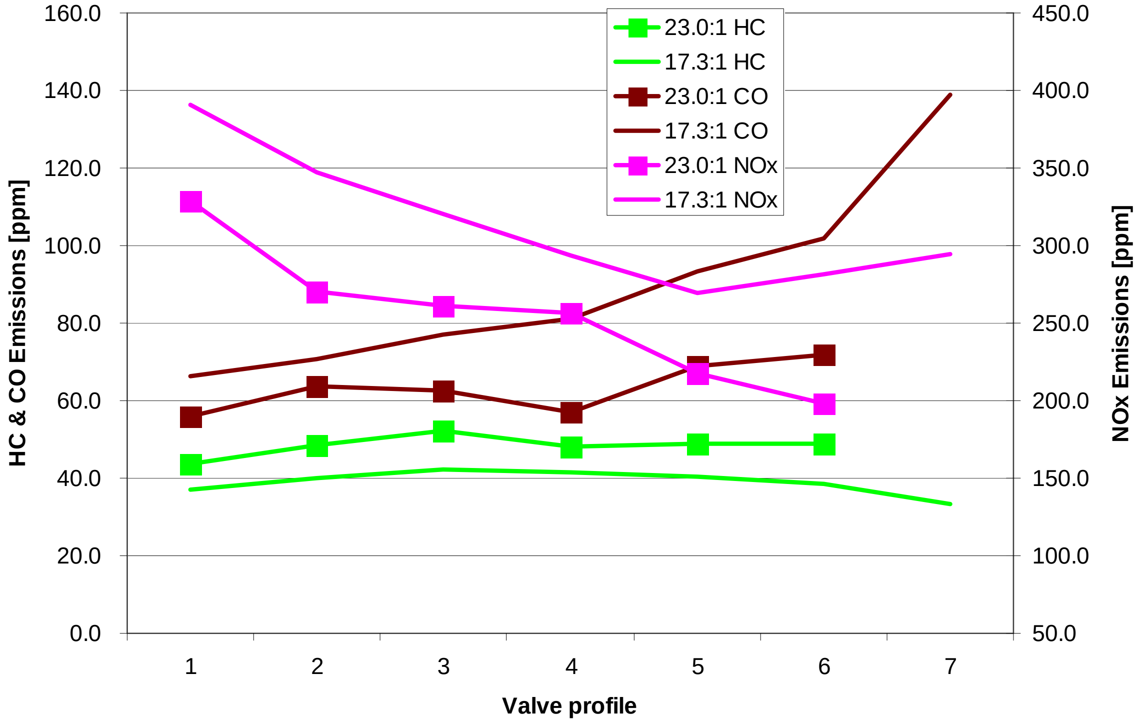

Figure 37 Emissions at 1250 rpm, 100 kW, with EGR, 23:1 and 17.3:1 in CR

The smoke emissions are nearly the same for the standard valve profile in both the two CR cases, see Figure 38. At higher Miller levels when the smoke goes down for the 17.3:1 CR case the smoke increase for the 23:1 CR case. In the 17.3:1 CR case the longer ignition delay that arises at higher Miller levels, thanks to the lower compression temperature with the same amount of air and EGR in the cylinder, gives a more homogenous fuel mixture before combustion and lower concentration of low lambda areas.

In the 23:1 and Miller cases the ignition delay was only slightly longer than with the standard lift profile but because the piston bowl wall is closer to the injector this delay meant that the spray front reaches the piston bowl, this results in the fuel hitting the piston bowl walls which generates higher levels of smoke. The exhaust temperature is little bit higher for the 17.3:1 CR case, see Figure 38. Larger heat transfer rates arise with higher cylinder pressures that come from a higher CR.

The brake efficiency is higher for the 23:1 CR, peaking at valve profile 4. Benefits in higher brake efficiency, lower smoke and lower NO_x than the standard case with 17.3:1 CR are made. If SOI can be altered whilst retaining the same NO_x, torque benefits can be even greater. This will be discussed later on in the report.

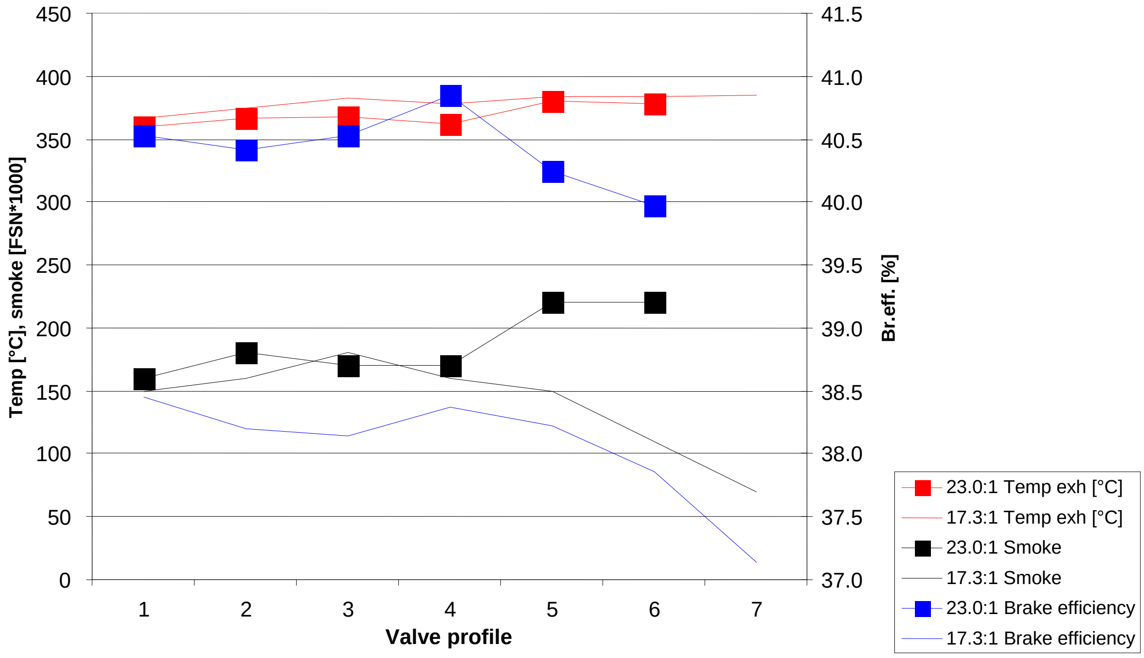

Figure 38 Exh. temp., smoke and br.eff. at 1250 rpm, 100 kW, with EGR, 23:1 and 17.3:1 in CR

In Figure 39 the cylinder pressure is plotted. Heat release and injector signal traces are plotted against CAD at CR 17.3:1. Three different cases are plotted, first standard valve profile with SOI -5°, blue curve. Second case is Miller cycled valve profile at 72° LIVC and SOI -5°, green line. The third case is Miller cycle with 72° LIVC and one pilot injection before the main injection at 9.29°.

The lower compression pressure with the Miller cases can easily be seen in the Figure 39. The heat release shows the longer ignition delay and the greater increase in cylinder pressure with full Miller cycling.

The case with pilot injection gives a smaller increase in cylinder pressure and a smaller premixed heat release. The pilot injection significantly reduces this rapid increase in cylinder pressure and the noise levels are reduced. The heat release rate is smoother and lower than the other cases.

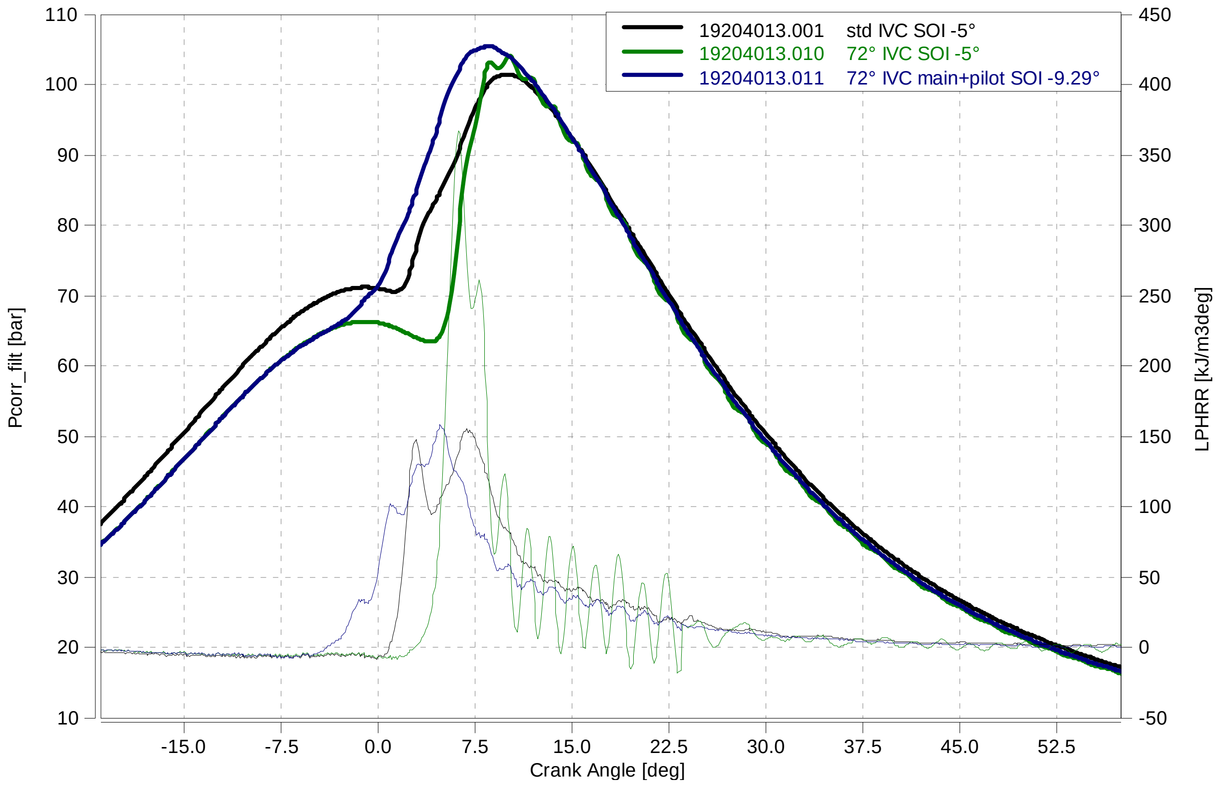

Figure 39 Cylinder pressure, heat release and injection signal at 1250 rpm, 100 kW, with EGR and 17.3:1 in CR.

When the compression ratio was increased the cylinder pressure also increased, see Figure 40. In this case where the same standard valve profile is used together with SOI at -5° the compression pressure increased 20 bar. The difference in ignition delay and premixed combustion is also seen in the same figure.

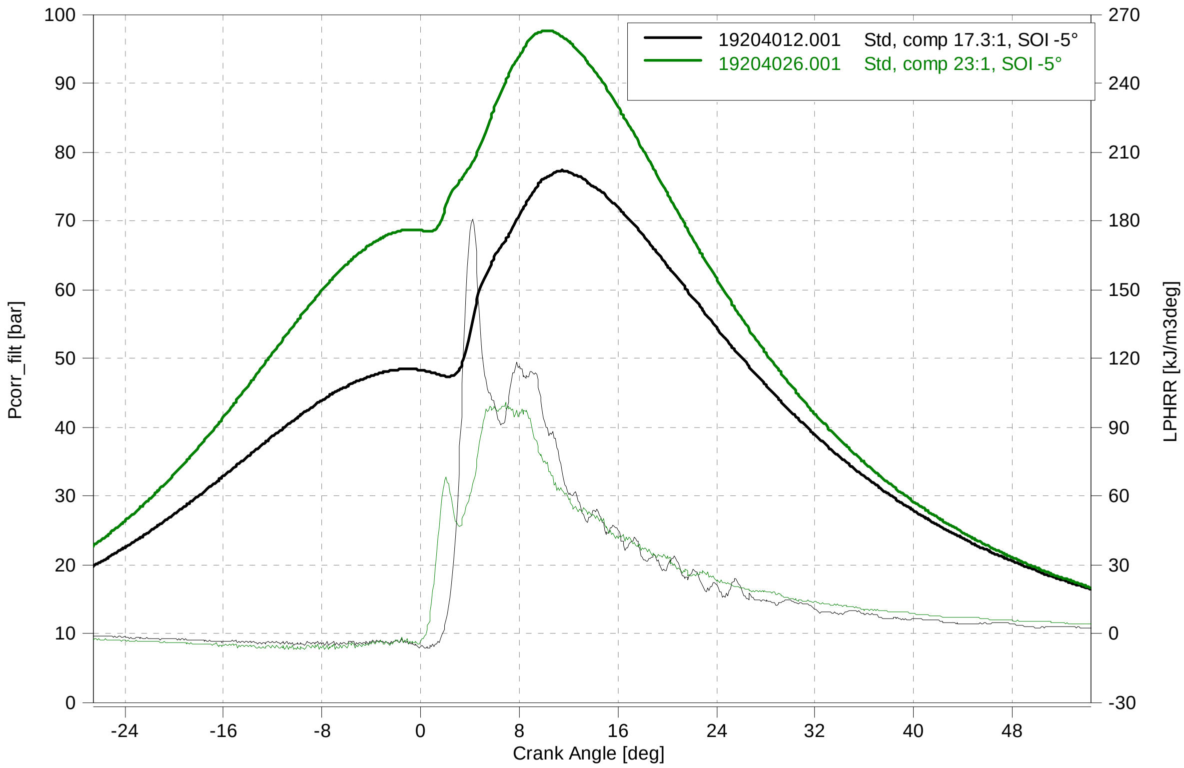

Figure 40 Pressure diagram at 1250 rpm, 100 kW, with EGR, std cam profile, 23:1 and 17.3:1 in CR

In Table 12 the same phenomena with a later SOC and an earlier MBF50% as in the case without EGR can be seen. This phenomenon is therefore not connected to the cases without EGR. PMEP is also shown in Table 12, typically a trade off during Miller cycling with the lower compression work on the one hand and the higher throttle losses when the valve closes on the other hand. This gives a profile that has the lowest PMEP. The same phenomena were seen in the simulations and the optimal Miller strategy changes with case and load.

Table 12 With and without Miller cycle, EGR 100kW and 17.3:1 in CR.

Miller [°] (Profile)	IMEP [bar]	PMEP [bar]	MBF50% [°]	SOC [°]
STD (1)	9.14	-0.50	8.4	2.75
22 (2)	9.16	-0.49	8.4	3.05
32 (3)	9.25	-0.53	8.4	3.15
42 (4)	9.04	-0.53	8.2	3.8
52 (5)	9.03	-0.58	7.85	4.15

SOC is earlier in the EGR case in comparison with no EGR. This can be explained due to the higher inlet manifold pressure in the EGR case that gives a higher pressure at SOC, see Table 10 and Table 12.

9.3.1. EIVC with EGR

To compare the late valve closing strategy, an early valve close strategy was also tested on certain points. The case with 30 % EGR was run with an EIVC. In Table 13 the test parameters for this load point are described.

Table 13 Test parameters for EIVC with EGR 100 kW power output

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1250	0.93	1800	-5	35.5 to 40	0.65 to 1.15	0.85 to 1.35	363 to 396	1.75 to 1.65	30	Std to 94° earlier IVC
23.0:1	1250	0.93	1800	-5	33.4 to 37.7	0.65 to 1.25	0.85 to 1.45	356 to 382	1.87 to 1.94	30	Std to 94° earlier IVC

The test setup was as similar as possible to the case with LIVC so comparisons can be made. The emission behaviours and levels are the same as the late IVC concept. NO_x are lowered with higher Miller levels and HC is constant. CO did not increase as much as in the LIVC case. One explanation can be that the highest inlet pressure in EIVC is lower than the LIVC inlet pressure at same the lambda. The Miller level is thereby higher in the LIVC case and the bad combustion quality is thereby creating higher CO emissions, see Figure 41.

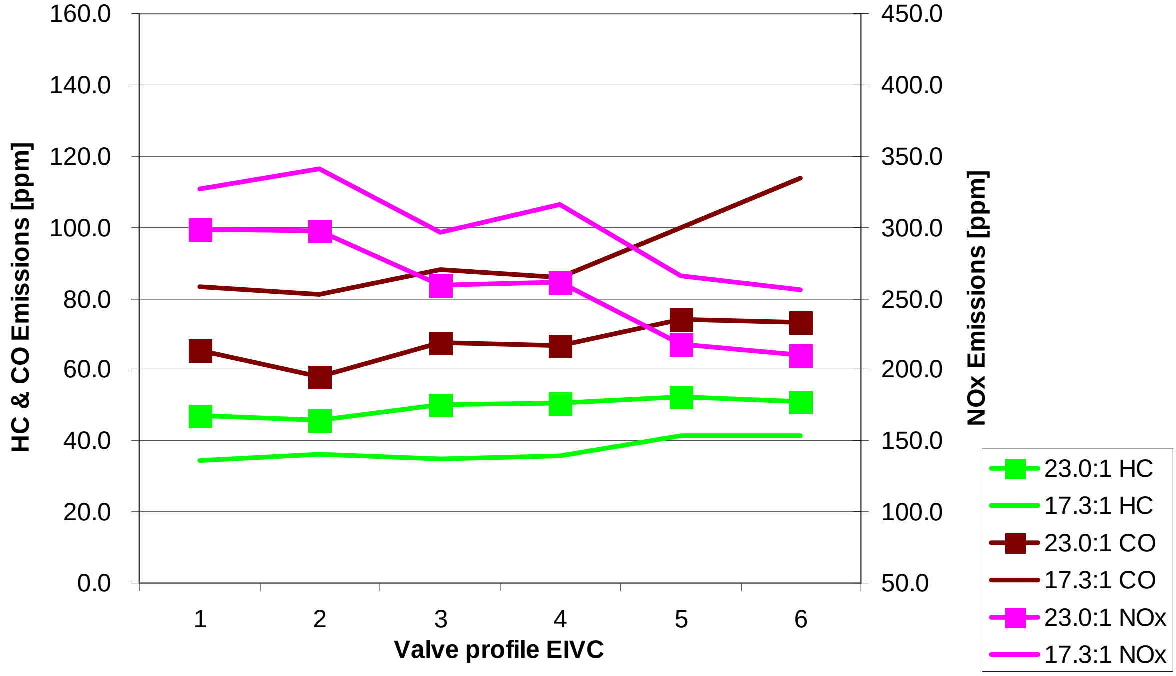

Figure 41 Emissions at 1250 rpm, 100 kW, with EGR, with EIVC, 23:1 and 17.3:1 in CR

The brake efficiency is highest with Miller level 32° and 23:1 in CR, see Figure 42. Smoke levels are about the same as the LIVC. In the literature study a calculation in the difference between early and late IVC claimed that the EIVC could result in lower swirl thereby producing higher soot emissions [4]. In this case the fuel injection pressure was so high that the air movement in the combustion chamber was created mostly by the fuel injection, not the swirl ratio. The exhaust temperature was shown to be higher with increased Miller levels and the difference between the two CR was not so large, see Figure 42.

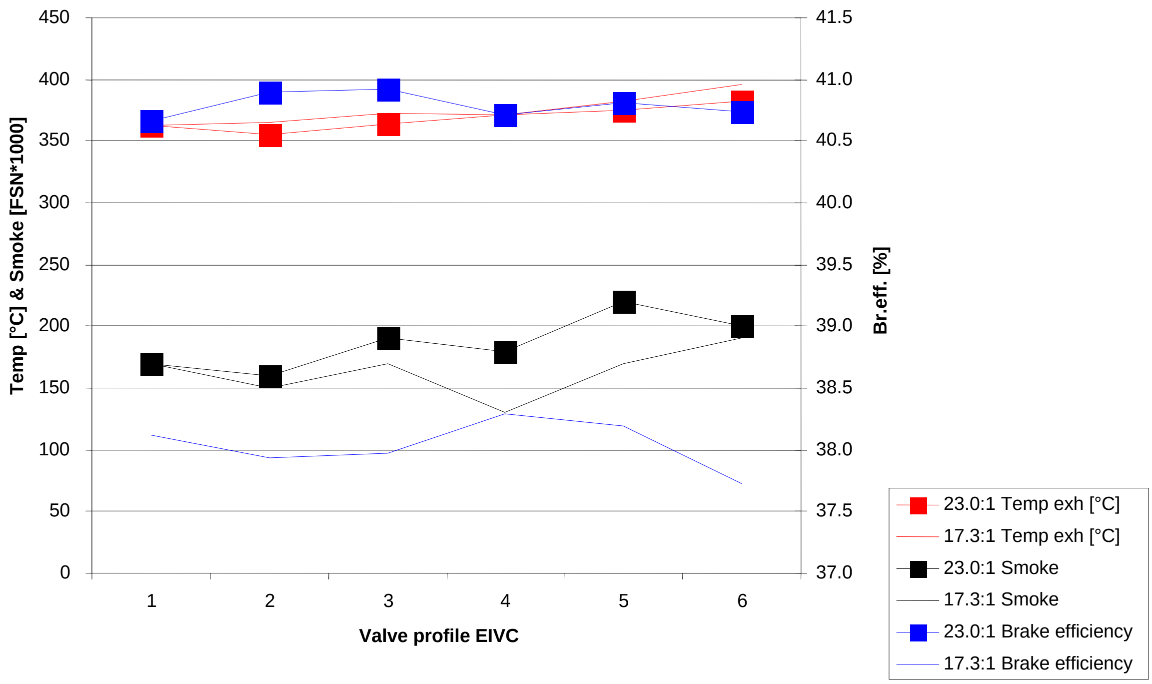

Figure 42 Br.eff., smoke and exh. temp. at 1250 rpm, 100 kW, with EGR, with EIVC, 23:1 and 17.3:1 in CR

In Figure 43 a comparison between the two Miller strategies can be seen. First the standard valve profile is plotted with 23:1 in CR and then the late together with the early Miller cases that have the same pressure trace can be seen. The HR behaves in the same way in both Miller cases and no large differences were found. In both cases the ignition delays are a little longer than in the standard case. Tabell 14 shows measured data for a comparison between EIVC and LIVC.

Tabell 14 Data for EIVC - LIVC for 100 kW, EGR and 23:1 CR

	NO_x [PPM]	Br.eff. [%]	Smoke [Bosch]	HC [PPM]	CO [PPM]
19204040-006 EIVC	210.6	40.7	0.2	51.0	73.0
19204027-009 LIVC	218.0	40.2	0.2	48.8	69.0

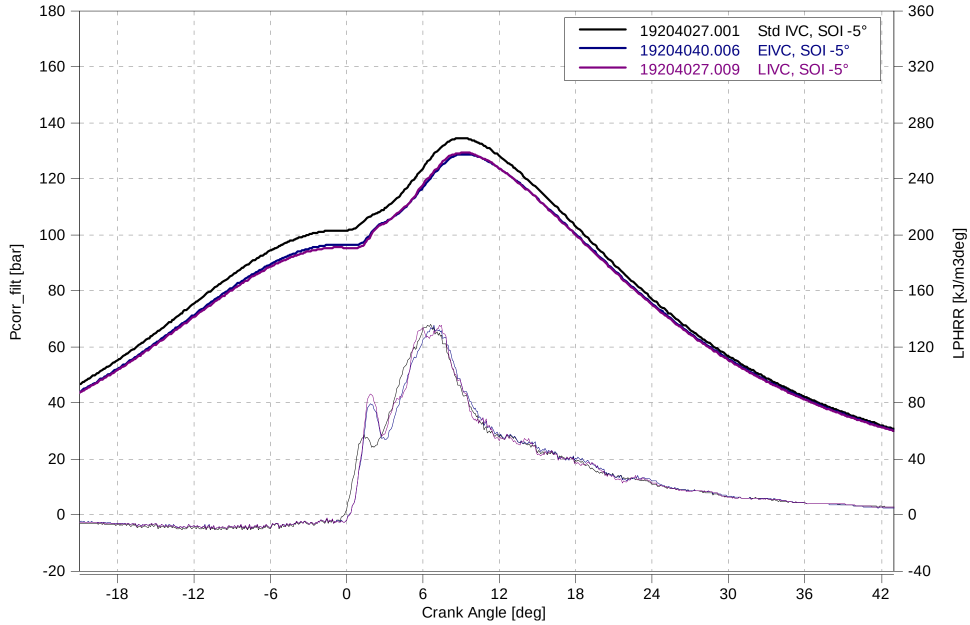

Figure 43 Cylinder pressure and heat release at 1250 rpm,100 kW, with EGR, EIVC, LIVC and 23:1 in CR

In the P-V diagram, Figure 44, the difference in the two valve strategies can bee seen. The EIVC has a lower pressure trace compare to the LIVC. The biggest difference occurred when the inlet valve closed. In the EIVC case the pressure dropped when the valve had closed and the piston was still expanding the gas in the cylinder. In the LIVC case the pressure level never dropped, when the piston reached BDC and was on the way up again the pressure increased even though the valves were still are open. This means that throttle losses occur when the cylinder gases are being expelled from the cylinder.

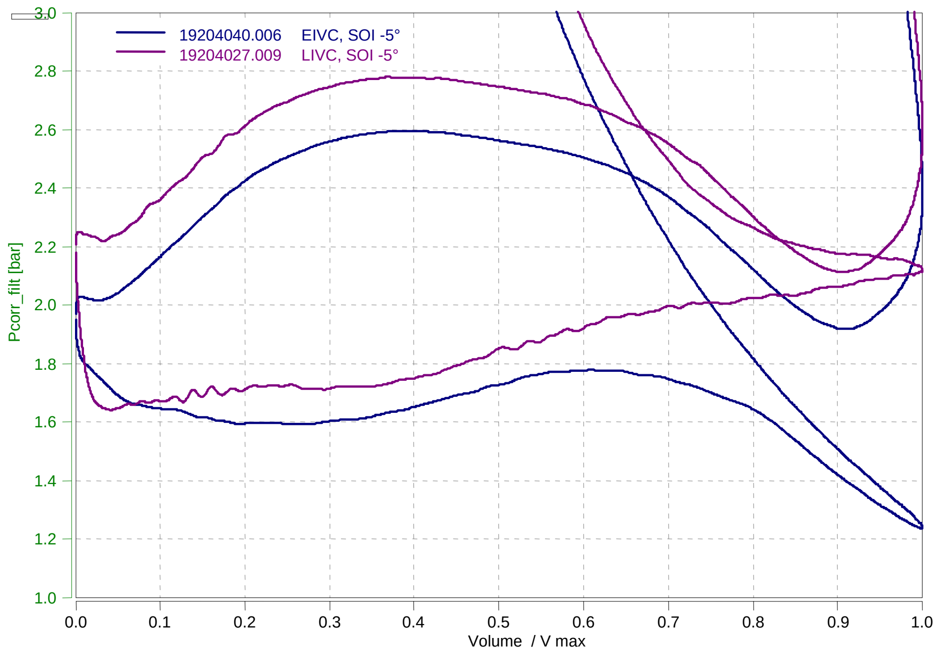

Figure 44 P-V diagram at 1250 rpm, 100 kW, with EGR, EIVC, LIVC and 23:1 in CR

The pump losses are lower for the EIVC than the LIVC. In Figure 45 the IMEP and PMEP are plotted for the two Miller strategies. Generally the IMEP for LIVC is lower than for EIVC.

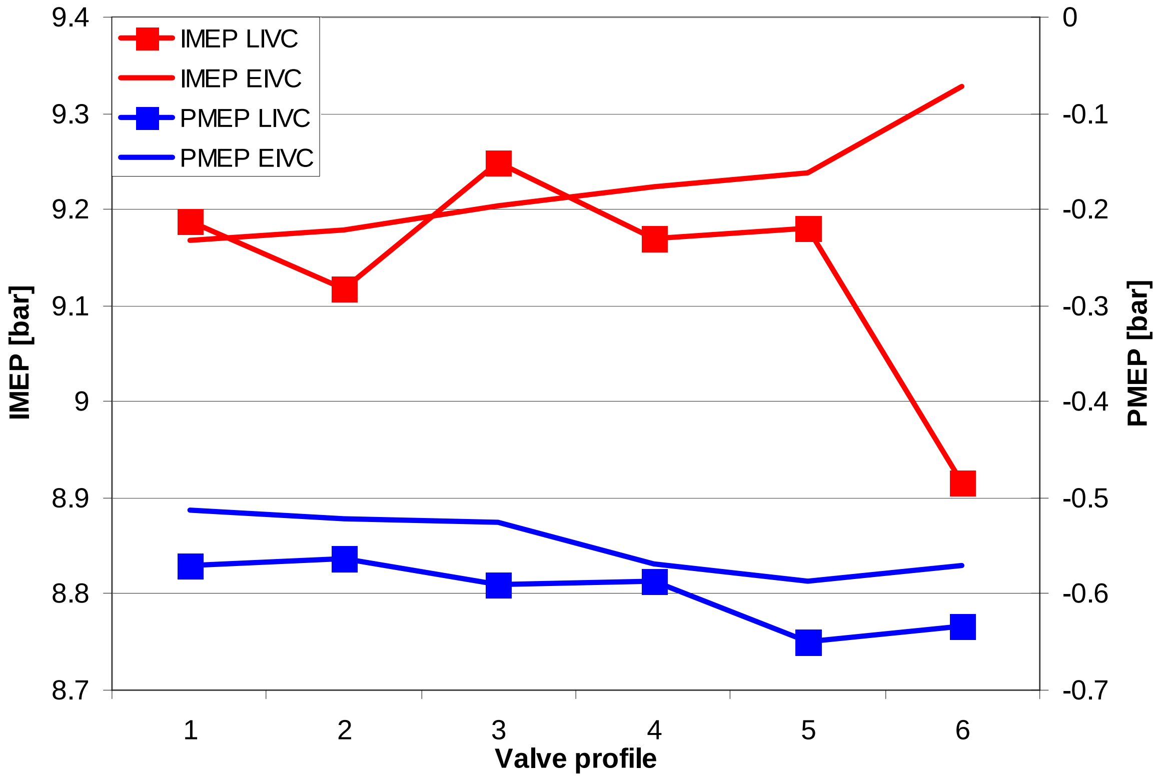

Figure 45 IMEP and PMEP at 1250 rpm, 100 kW, with EGR, EIVC and LIVC, 23:1 and 17.3:1 in CR

9.4. Load point 4

Load point 4 is equal to 240kW power output for a full engine with and without EGR at 1250 rpm. The test parameters for this load can be seen in Table 15. All the tests were carried out with boost pressure and therefore the lambda could be adjusted to the same level for all the Miller cases.

Table 15 Test parameters for 240kW full engine power output

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1250	2.2	1400	-5	27.3 to 31.3	1.1 to 2.35	1.1 to 2.35	583 to 649	1.32 to 1.4	0	Std to 72° later IVC
23.0:1	1250	2.2	1400	-5	26.3 to 30.6	1.1 to 2.4	1.1 to 2.4	569 to 643	1.43 to 1.48	0	Std to 72° later IVC
17.3:1	1250	1.54	2400	-5	45.6 to 50.3	1.9 to 2.95	2.1 to 3.15	489 to 546	1.30 to 1.36	30	Std to 62° later IVC
23.0:1	1250	1.54	2400	-5	46.7 to 53.5	2.0 to 3.3	2.2 to 3.5	482 to 518	1.39 to 1.43	30	Std to 62° later IVC

At this load point an increase in br.eff. (brake efficiency) was recorded (1,5 % br.eff.) with the same NO_x level as the standard valve profile, SOI at -5° and 17.3:1 in CR. The Miller profile of 32° was used together with an earlier SOI (-6.63°) for the same NOx emissions without EGR. The emissions at this load point without EGR are typically lower at higher Miller levels, see Figure 46. The NOx emissions decreased with higher Miller levels, NOx were still lower in the 23:1 CR case than in the 17.3:1 case. HC also decreased with the Miller level, CO did not increase much at high Miller levels even though this trend was observed in the lower power output cases.

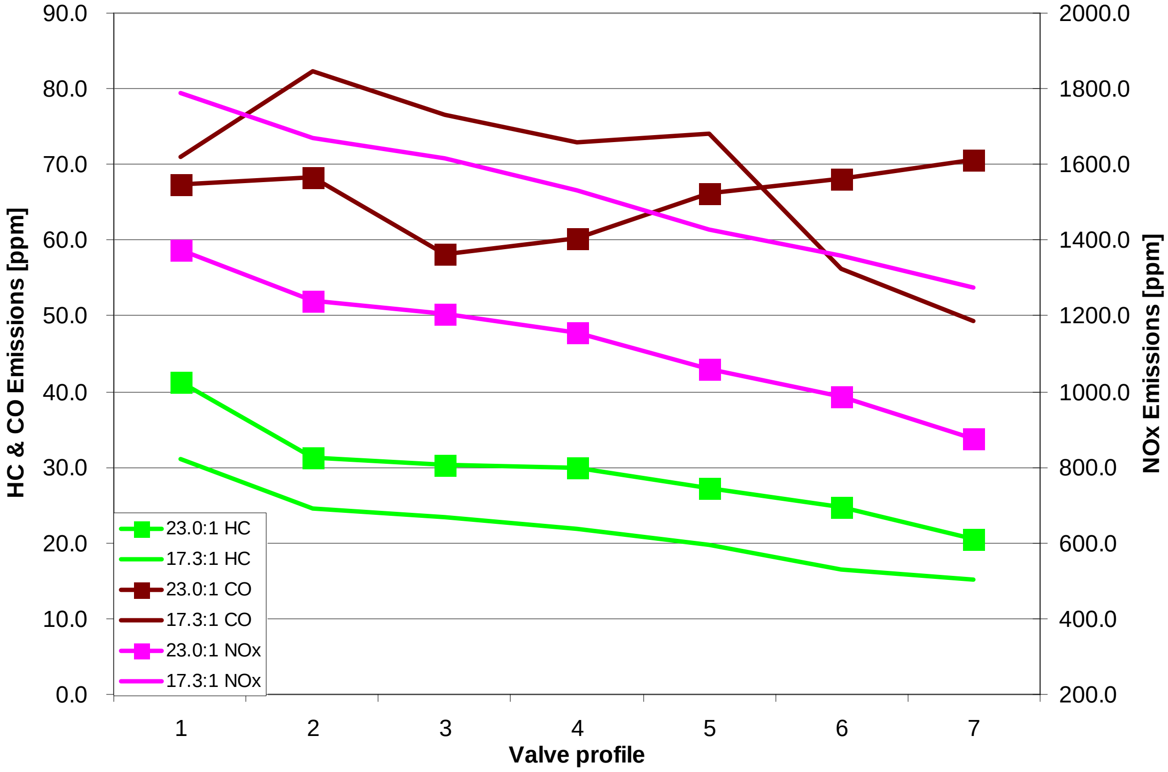

Figure 46 Emissions at 1250 rpm, 240kW, without EGR, 23:1 and 17.3:1 in CR

The exhaust temperatures increased with higher Miller levels, the 17.3:1 CR had a higher exhaust temperature, see Figure 47. The heat transfer rates are lower here (with 17.3:1 CR) than with 23:1 CR. The brake efficiency decreased in both cases with the same SOI. The lower NO_x emissions make it possible to have an earlier SOI that gives a higher br.eff. output. Smoke emissions for the different CR lie at the same level in some cases and lower in 17.3:1 CR at higher Miller levels. The smaller piston bowl of the 23:1 CR gives some disadvantage with spray penetration depth. Higher Miller levels have shown a lower maximal cylinder temperature and pressure for the same amount of air retained the cylinder. In the 17.3:1 case the longer penetration depth can be used to create smaller areas with low lambda which gives lower smoke emissions.

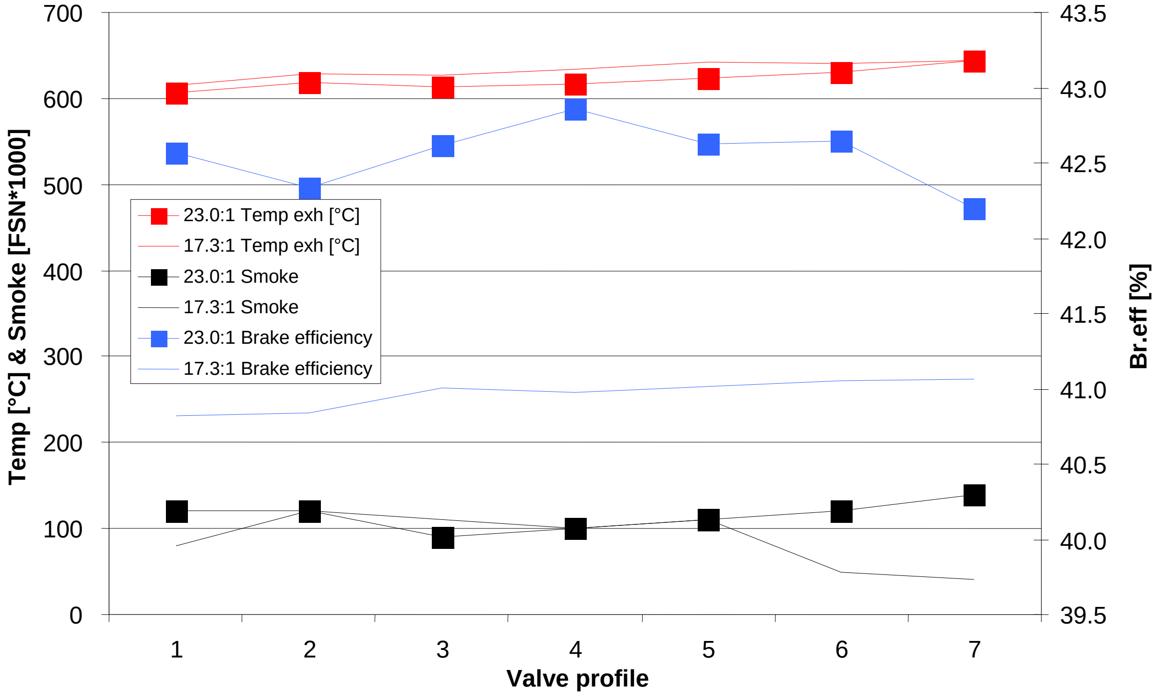

Figure 47 Br.eff., smoke and exh. temp. at 1250 rpm, 240 kW, without EGR, 23:1 and 17.3:1 in CR

In Figure 48 the pump work can be seen together with the valve lift profile plotted against volume. The black line and the green dotted line are the pressure and standard valve profile lift traces. The red and pink dotted lines are from the Miller cycle. The rake on the pink line tells us that the volume is changing fast in the cylinder and a difference in pressure is building up with the air mass transport out of the cylinder. This creates high throttle loses when the pressure difference over the valve is going towards critical flow. So there is a trade off for the level of Miller where one part is the higher critical flow losses and the other higher compression losses.

Other factors that influence the efficiency are the lower temperature at start of combustion. More of the charge is cooled outside the engine by the intercooler. The same amount of air is trapped in the cylinder (comparing standard profile and Miller). The difference is that the boost pressure is higher for the Miller strategy and the start temperature when the inlet valve is closed and compression has started is nearly the same. The compression stroke in the Miller case is effectively shorter due to the fact that the crankshaft has already started to move up towards TDC when the inlet valve closes. This gives the lower temperature at TDC. The lower temperature gives longer ignition delays and more premixed combustion. The NO_x decreased with higher Miller levels due to the lower start temperature at SOC, the soot emissions were lower due to the longer ignition delay with the 17.3:1 CR case. One other interesting phenomenon is the difference in the exhaust strokes. The Miller cases had a clearer exhaust pulse in the beginning of the exhaust stroke. The cylinder pressure is even under the inlet pressure when the piston is in BDC. This means that some positive work is done in the pumping cycle (just a little, the total PMEP is still negative work). The pulse can be interesting under transient engine operation when the boost pressure needs to increase. If the right tuning is used in the exhaust manifold, this can be of benefit.

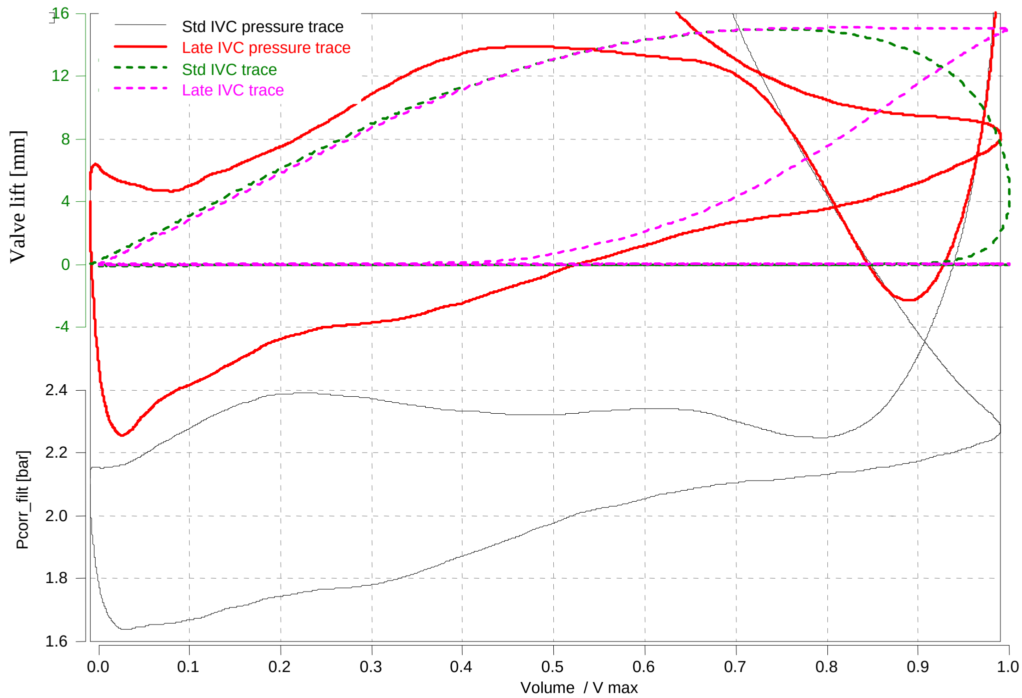

Figure 48 Pump stroke at 1250 rpm, 240 kW, without EGR 17.3:1 in CR

With EGR

When EGR was used all the NO_x levels decreased compared to the cases without EGR, see Figure 49. The NO_x levels decrease at higher Miller levels as in previous cases, NO_x levels were lower for the 23:1 CR case. HC are on the same level as before, but CO for 23:1 CR was significantly higher. The EGR gives naturally a longer heat release at the same injection pressure thanks to the low concentration of oxygen in the cylinder. In this case the injection pressure was set 1000 bar higher when EGR was in use to compensate for this. In the 23:1 CR case this increase would not be enough. The smoke emissions in Figure 50 confirm this. The longer ignition delay gives an longer penetration depth. A greater penetration depth means the spray hits the piston wall at higher Miller levels with higher CO and smoke as a result.

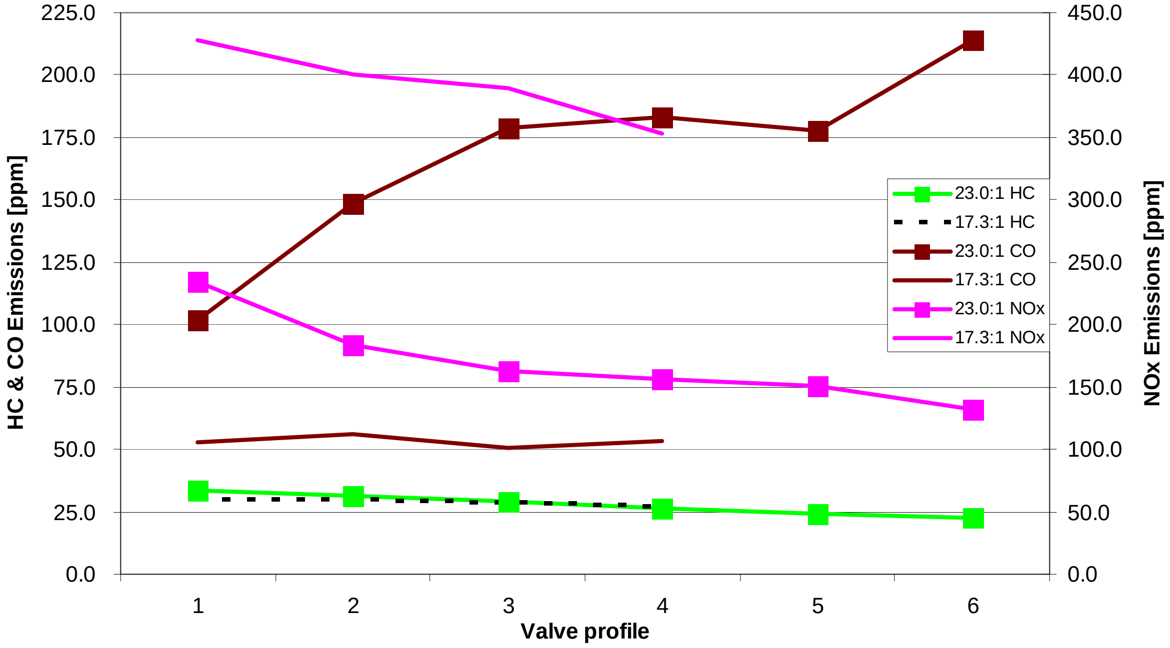

Figure 49 Emissions at 1250 rpm, 240 kW, with EGR, 17.3:1 and 23:1 in CR

Even if the smoke emissions are higher for this case the torque output was also higher for the 23:1 CR case, see Figure 50. A maximum br.eff. output at valve profile 5 shows that the Miller strategies are giving a greater benefit at higher cylinder pressures. The exhaust temperatures are higher for the 17.3:1 CR case, as seen in earlier cases.

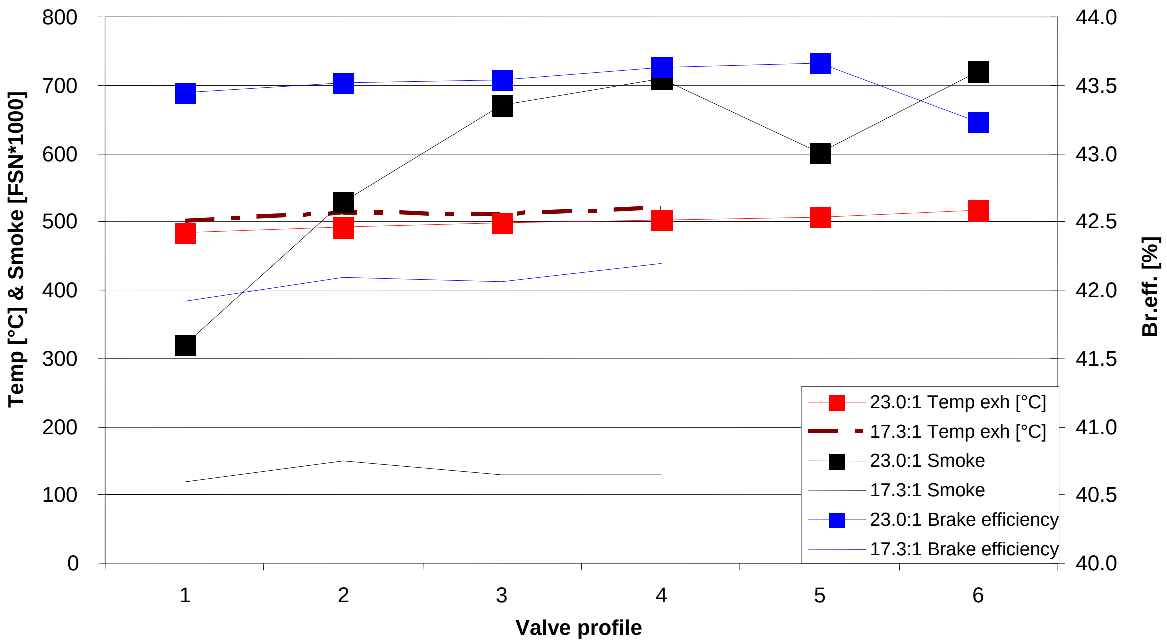

Figure 50 Br.eff., exh. temp. and smoke at 1250 rpm, 240kW, with EGR, 23:1 and 17.3:1 in CR

In Figure 51 the cylinder pressures and heat releases are plotted for the standard valve lift profile and Miller valve lift profile, 17.3:1 CR. The inlet pressure for the standard case is 1.1 bar (rel) and 2.35 bar for the Miller case. This is to maintain the same amount of air in the cylinder. On the cylinder pressure curve a difference in pressure can be seen between Miller and standard. This was caused by the lower temperature in the cylinder under compression.

The heat release trace looks nearly the same for the two cases, but the Miller case has a longer ignition delay and oscillates, the lower temperature and pressure at SOC are the main reasons for this. Nearly the same torque is delivered from both cases, but the Miller case has a lower maximal cylinder pressure (P max). This can be a benefit if the engine has P max limitations, for example when run with high EGR content and high boost pressures. The Miller strategy can thereby be used to lower the maximum cylinder pressure or trade it for an earlier SOI.

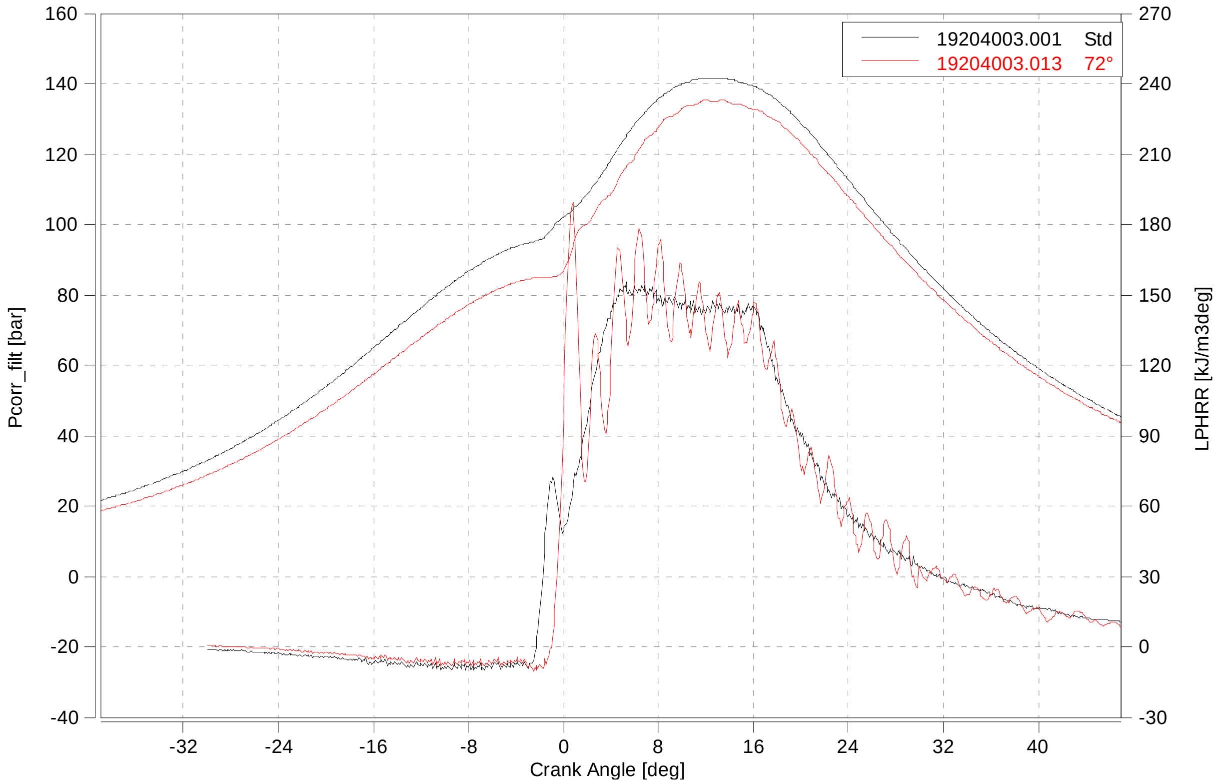

Figure 51 Heat release and cyl. press. at 1250 rpm, 240 kW,with EGR, std and Miller, 17.3:1 in CR

9.4.1. EIVC without EGR

At this load point a test with early inlet valve close was also done. The test parameters can be seen in Table 16.

Table 16 Test parameters for early inlet valve close without EGR at 240kW power output

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1250	2.2	1400	-5	27 to 27.8	1.3 to 2.05	1.3 to 2.05	568 to 602	1.55 to 1.50	0	Std to 94° earlier IVC
23.0:1	1250	2.2	1400	-5	33.4 to 37.7	1.3 to 2.1	1.3 to 2.1	554 to 579	1.64 to 1.61	0	Std to 94° earlier IVC

The NO_x emissions for the 23:1 CR early IVC have the same trend as with late IVC, see Figure 52. One explanation can be that the Miller levels were lower with EIVC than with LIVC. The other emissions behave similarly to the case with LIVC.

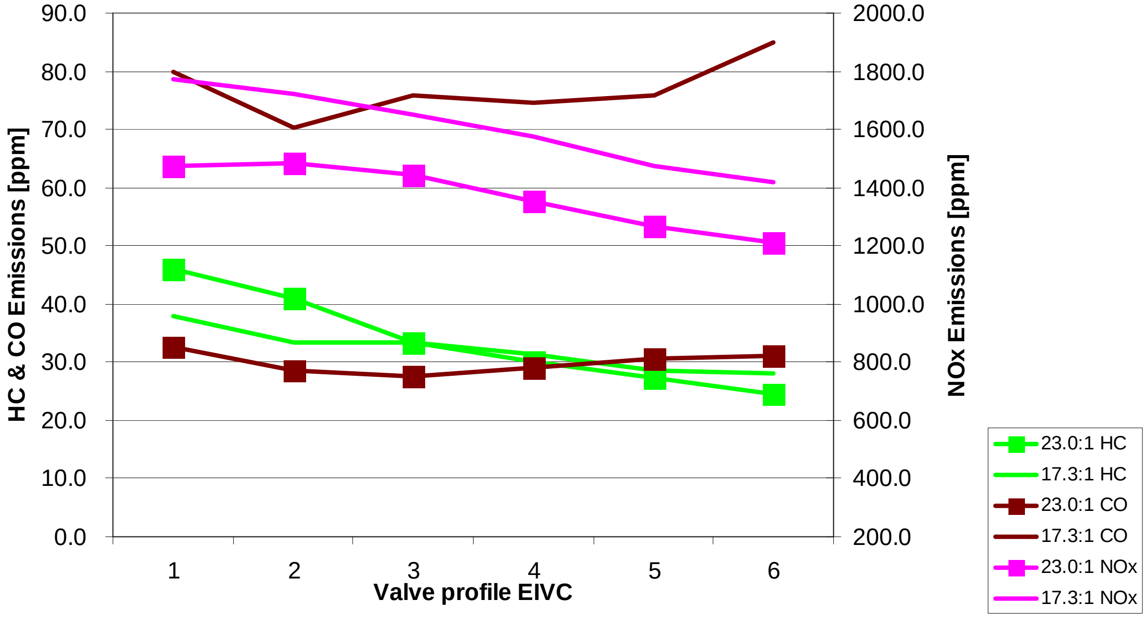

Figure 52 Emissions at 1250 rpm, 240kW, without EGR, EIVC, 23:1 and 17.3:1 in CR

The exhaust temperature is higher for the 17.3:1 CR case, see Figure 53, the smoke levels are constant. The brake torque for 17.3:1 CR case has a maximum at valve profile 3 and the 23:1 case at valve profile 2. The differences are not so large so it is difficult to say if it can depend on the compression ratio or not.

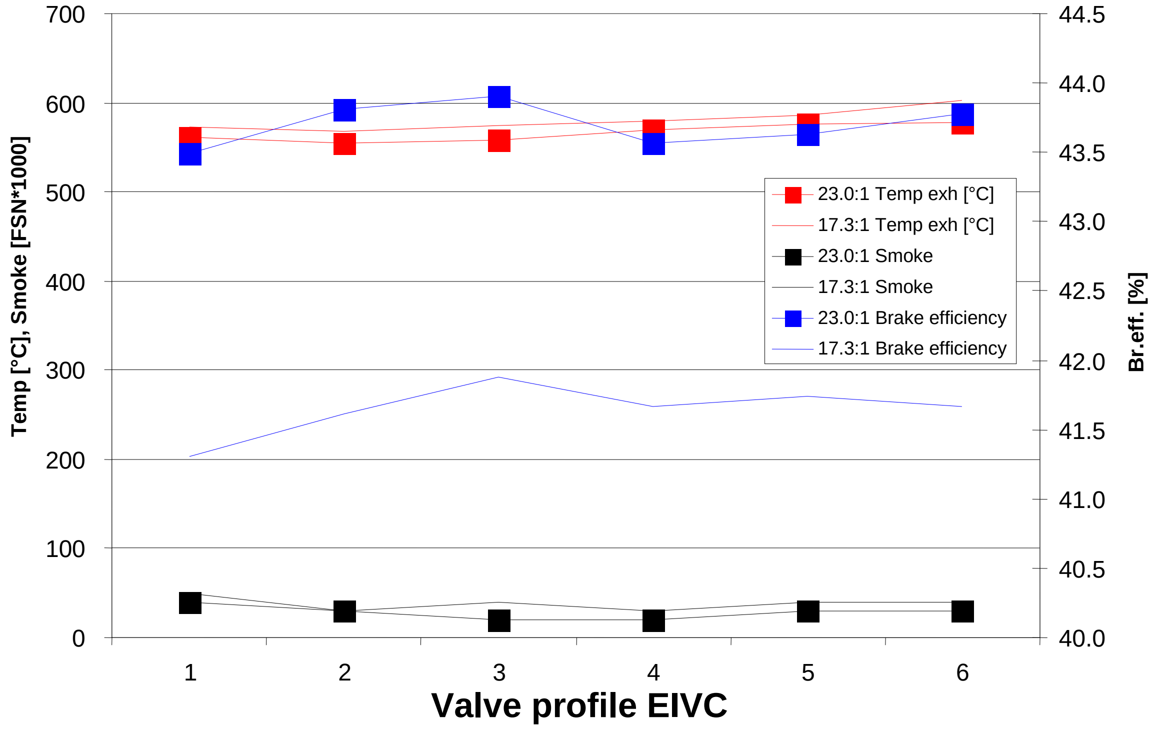

Figure 53 Smoke, br. eff. and exh. temp. at 1250 rpm, 240 kW, without EGR, EIVC, 23:1 and 17.3:1 in CR

9.5. Load point 5

Equal to 300 kW power output for a full engine with and without EGR, the test parameters can be seen in Table 17. This power output is interesting from the point of view that it is a full load point for a 400 hk engine. Table 17 shows that a very high boost pressure was necessary at high Miller levels and EGR to keep lambda above 1.3, this was one of the boundary conditions in the test setup. The entire test was carried out at the same lambda. The 23:1 CR case had a little higher lambda to compensate for the higher smoke emissions.

Table 17 Test parameters at 1250 rpm, 300kW full engine power output, 23:1 and 17.3:1 in CR

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1250	2.85	1400	-12 to -16.71	26.9 to 28.8	1.5 to 2.5	1.5 to 2.5	616 to 663	1.30 to 1.36	0	Std to 62° later IVC
23.0:1	1250	2.85	1400	-12 to -13.6	26.3 to 29.1	1.5 to 2.7	1.5 to 2.7	617 to 640	1.39 to 1.43	0	Std to 62° later IVC
17.3:1	1250	2	2400	-5 to -11.09	50.1 to 55.5	2.6 to 3.9	2.8 to 4.1	517 to 578	1.24 to 1.31	30	Std to 62° later IVC
23.0:1	1250	2	2400	-3 to -4.41	50.5 to 58.2	2.7 to 4.3	2.9 to 4.5	514 to 554	1.31 to 1.34	30	Std to 62° later IVC

The NO_x emissions went down with the level of Miller and NO_x was lower for the 23:1 CR case, see Figure 54. The lower NO_x at higher CR can be explained by the smaller and hotter premixed combustion where lambda was lower. The HC emissions are lower at higher Miller levels and CO for the 23:1 CR case are higher together with higher smoke emissions than the 17.3:1 CR case.

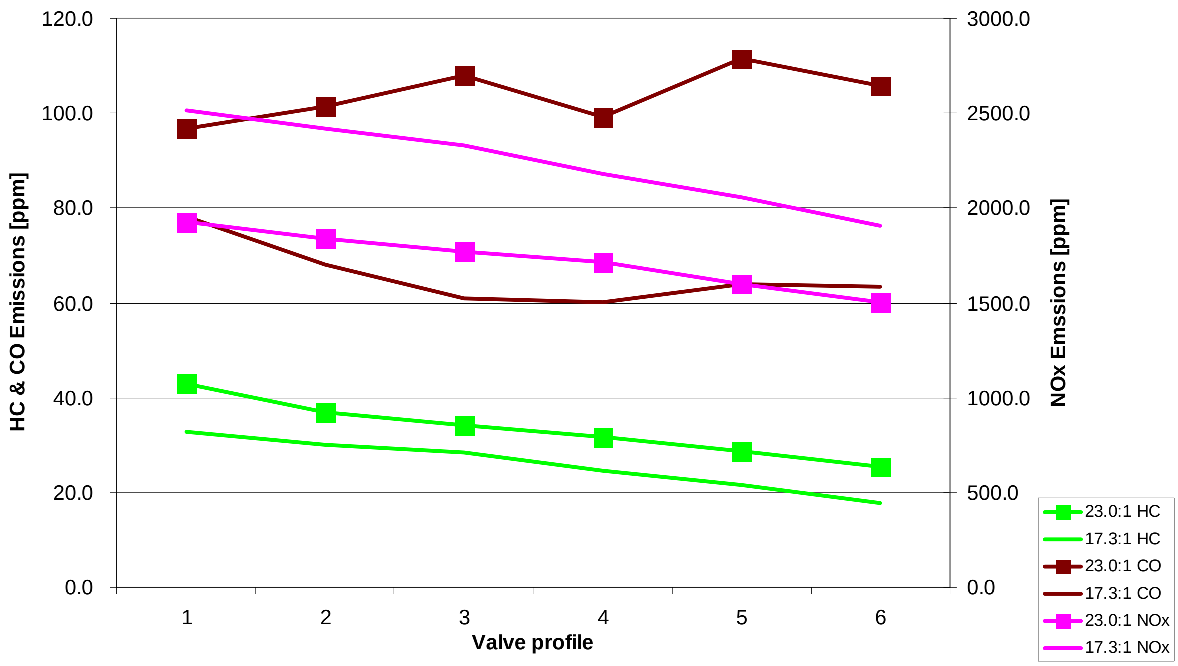

Figure 54 Emissions at 1250 rpm, 300kW, without EGR, 23:1 and 17.3:1 in CR

The exhaust temperature are little bit higher for the 17.3:1 CE case due to the lower heat transfer to the cylinder wall, see Figure 55. Brake torque are still higher for the 23:1 CR case with a maximum brake torque output around valve profile 4 and 5.

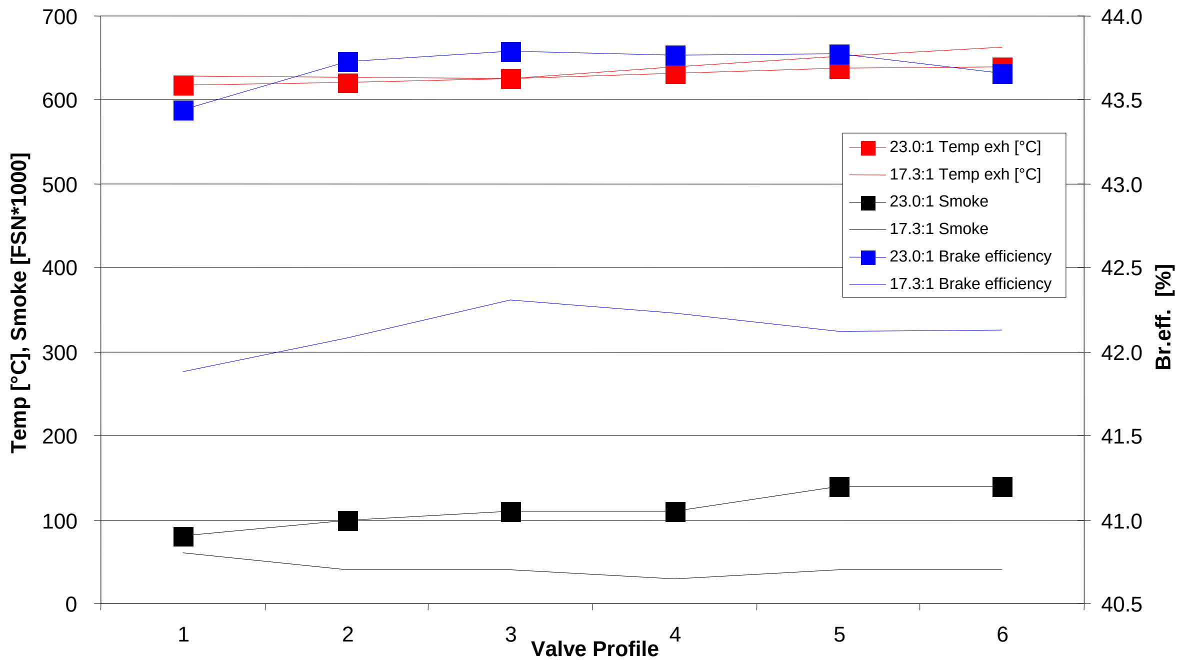

Figure 55 Smoke, br.eff. and exh. temp. at 1250 rpm, 300 kW, without EGR, 23:1 and 17.3:1 in CR

In Figure 56 the cylinder pressure and HR are plotted for the two different compression ratios and standard valve profile together with Miller 62° LIVC. When a Miller valve profile is used the cylinder pressure levels decreases. The ignition delays did not increase as much as in the low power output cases when higher Miller levels were used. The biggest difference in ignition delay is found in the 17.3:1 CR cases where the premixed combustion peaks were higher for the Miller case.

The HR rates were higher for the 17.3:1 CR case, the amount of fuel injected into the cylinder were lower than the 23:1 CR case. The amount of fuel is lower even if the same rail pressure and ontime on the injector are used. Measured delta for 17.3:1 CR case are 244.7 mg and for 23:1 CR case 240.3 mg. When the injector is a hydraulic activated injector the actual injection period is affected by the rail pressure and the difference in pressure over the injector. Therefore the actual amount of injected fuel can be different for different cylinder pressures.

Why the HR shapes are different between the two CR can be explained by the smaller piston bowl in 23:1 CR case. The combustion flame has a shorter distance to the surrounding walls and the area that air can be delivered in to the combustion flame is thereby reduced by the tight piston bowl. This results in a longer combustion duration for 23:1 CR case.

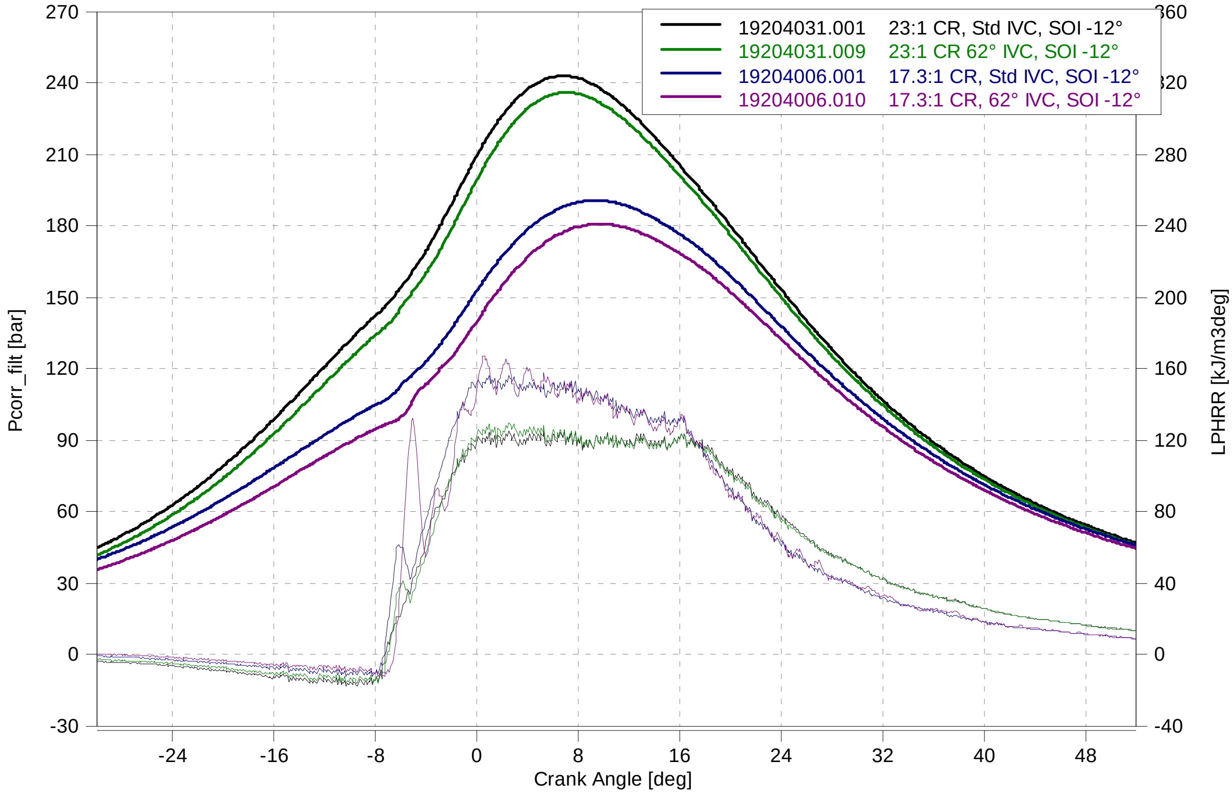

Figure 56 Cyl. press. And heat release at 1250 rpm, 300kW, without EGR, 23:1 and 17.3:1 in CR

In Figure 57 the P-V diagram is plotted for the pressure traces in Figure 56. The higher inlet pressure in the Miller cases are easily seen in the P-V diagram. The exhaust pressure pulse after the exhaust valves open go under the inlet pressure in both Miller cases that we have seen before in the lower load outputs. In the standard valve lift case this does not happened. This pulse can depend on the higher expansion ratio in the Miller case compared to the effective swept volume of the Miller set-up.

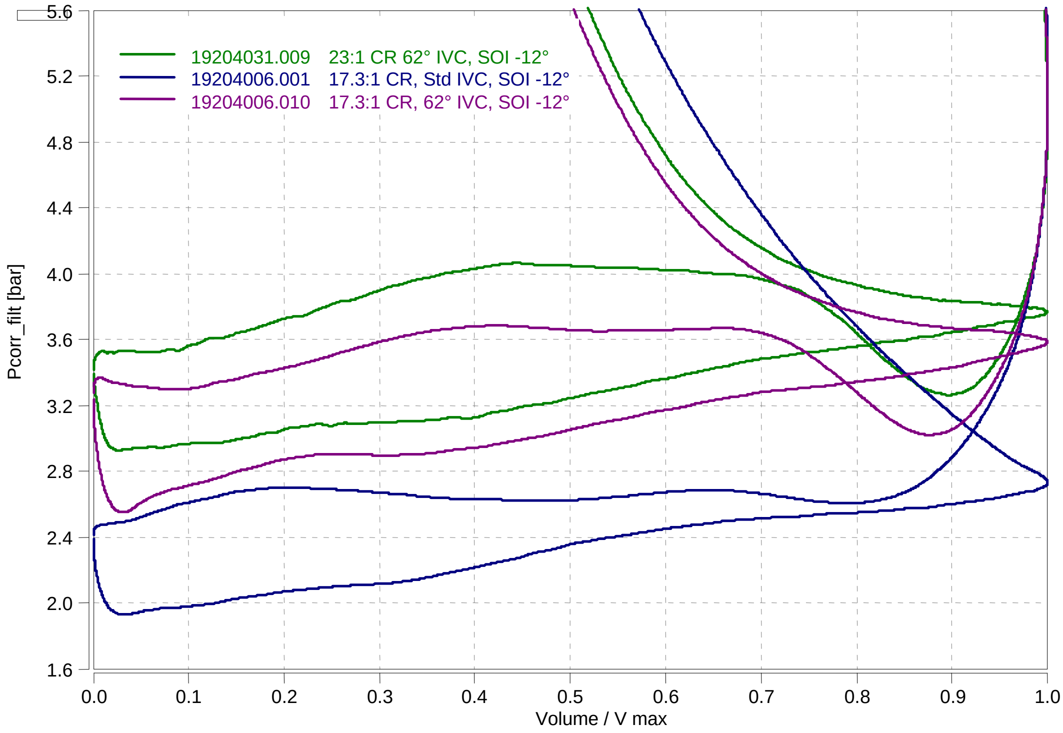

Figure 57 P-V diagram at 1250 rpm, 300kW, without EGR, 23:1 and 17.3:1 in CR

With EGR

In the EGR case very high boost pressure was needed to maintain lambda above 1.3 at high Miller levels. To make this pressure in a real engine a two stage turbo boost system would be required. The emission trends are similar to all previous cases, NO_x is lower for 23:1 CR case and decreasing with Miller level, see Figure 58. CO and smoke increase with Miller level and are higher in the 23:1 CR case. HC levels are low with just a small decrease at higher Miller levels.

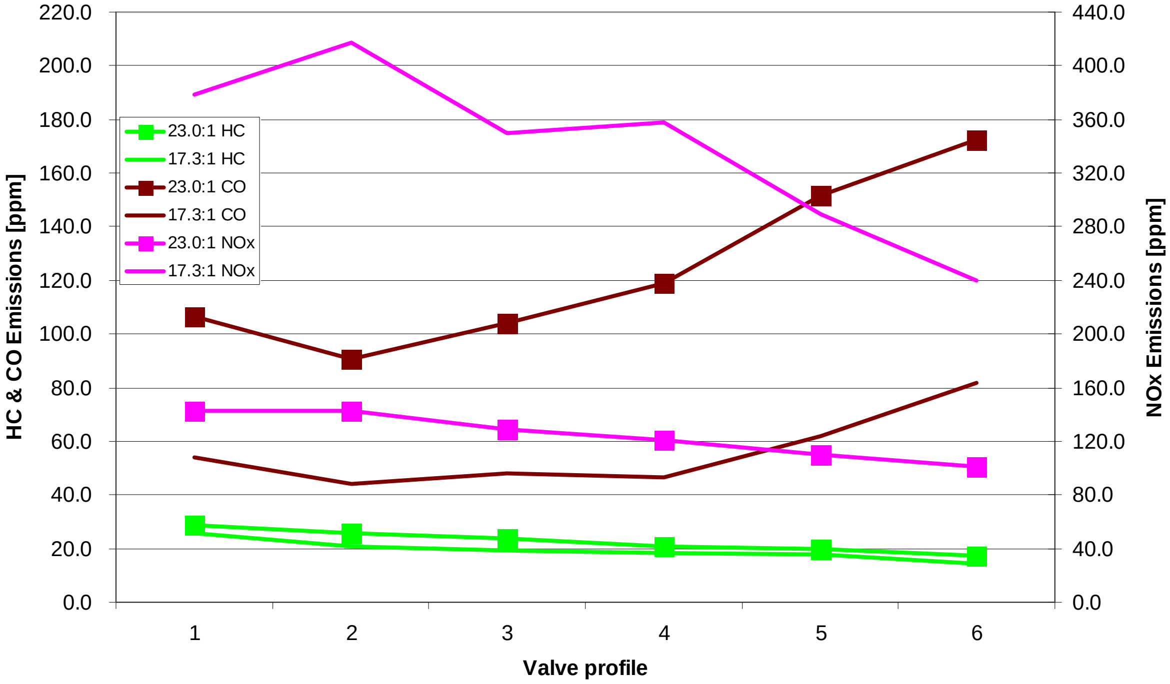

Figure 58 Emissions at 1250 rpm, 300kW, with EGR, 23:1 and 17.3:1 in CR

Exhaust temperature increases with higher Miller levels and are a little higher for 17.3:1 CR case, Figure 59. The br.eff. output was highest at 22° LIVC for 23:1 CR and 42° LIVC for the 17.3:1 CR case. The SOI was set to -5° for 17.3:1 CR and retarded to -3° for 23:1 CR cases to reduce maximum cylinder pressure. A 250 bar cylinder pressure maximum was set in the boundary conditions for the test, to come under this boundary SOI was retarded. Therefore the torque curve can not be directly compared. It was of interest to see that the exhaust temperatures were still higher with 17.3:1 in CR. Smoke also increased with higher Miller levels, the penetration depth for the spray was greater at the lower pressure and cylinder temperature in the Miller case.

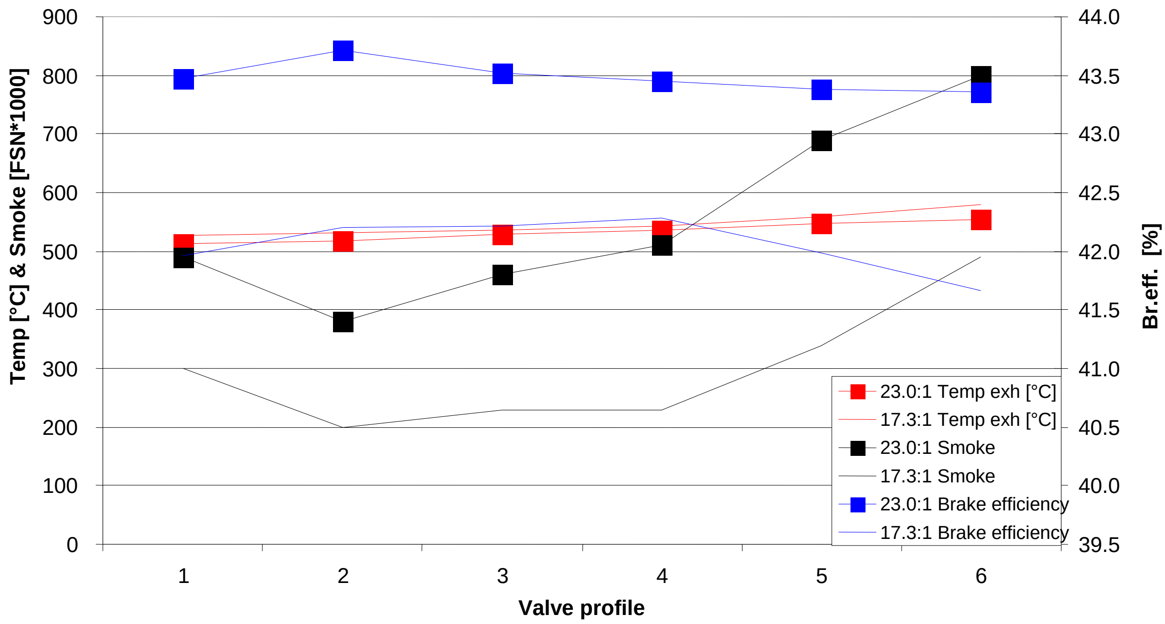

Figure 59 Br.eff., smoke and exh.temp. at 1250 rpm, 300 kW, with EGR, 23:1 and 17.3:1

The cylinder pressure that reached the maximum pressure limits can be seen in Figure 60. When run with a Miller profile the cylinder pressures are lower than with the standard valve profile for the same SOI, this gives the opportunity to advance the SOI angle to a more suitable position and a higher torque output can be achieved. In this case when EGR is used the ignition delays compare to the case with a standard valve profile. The ignition delays with the Miller profiles case do not change.

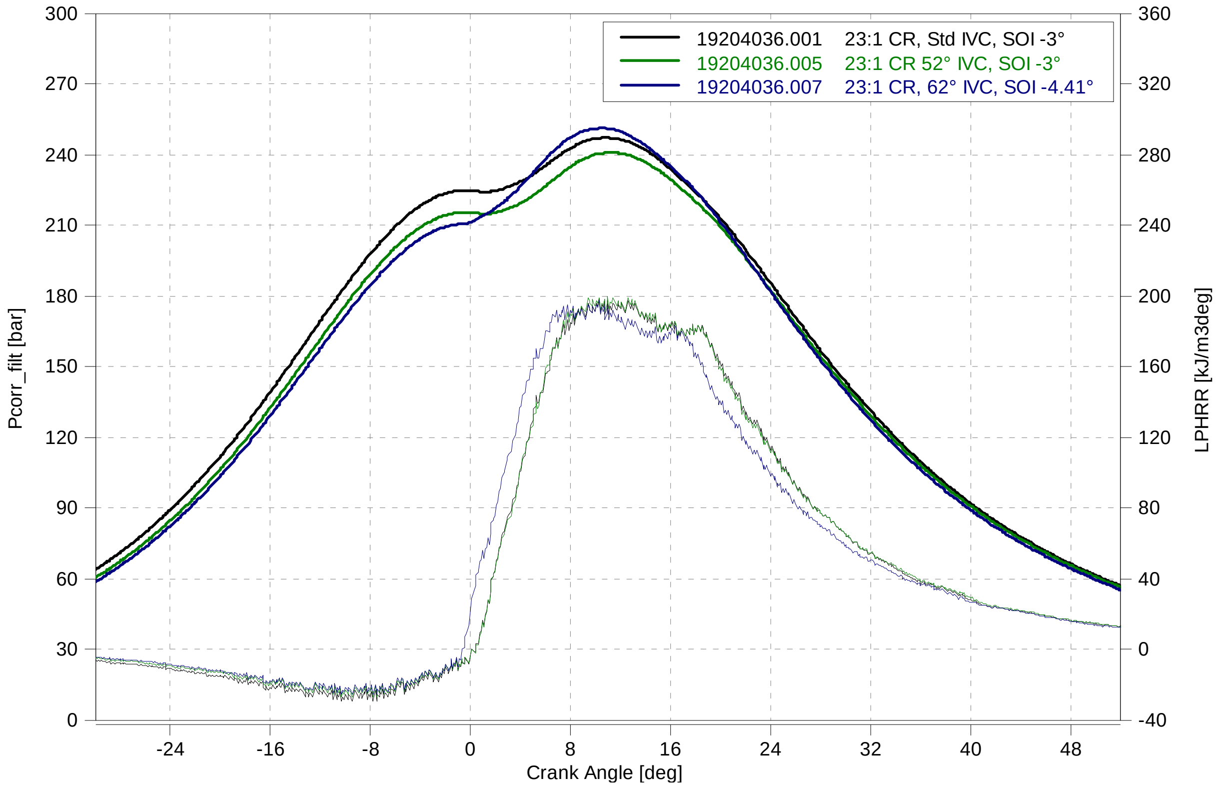

Figure 60 Cyl.press. and heat release at 1250 rpm, 300kW, with EGR and 23:1 in CR

The higher and shorter HR that we have seen in the 17.3:1 CR case without EGR can also be seen in the 23:1 CR case with EGR, see Figure 61. The heat releases are higher for the lower CR case, even if this comparison is not exactly correct because the SOI is different for the cases. The SOI needed to be retarded 2° from -5° for 17.3:1 CR case to -3° for the 23:1 CR case. This was done to reduce the maximum cylinder pressure to less than 250 bar.

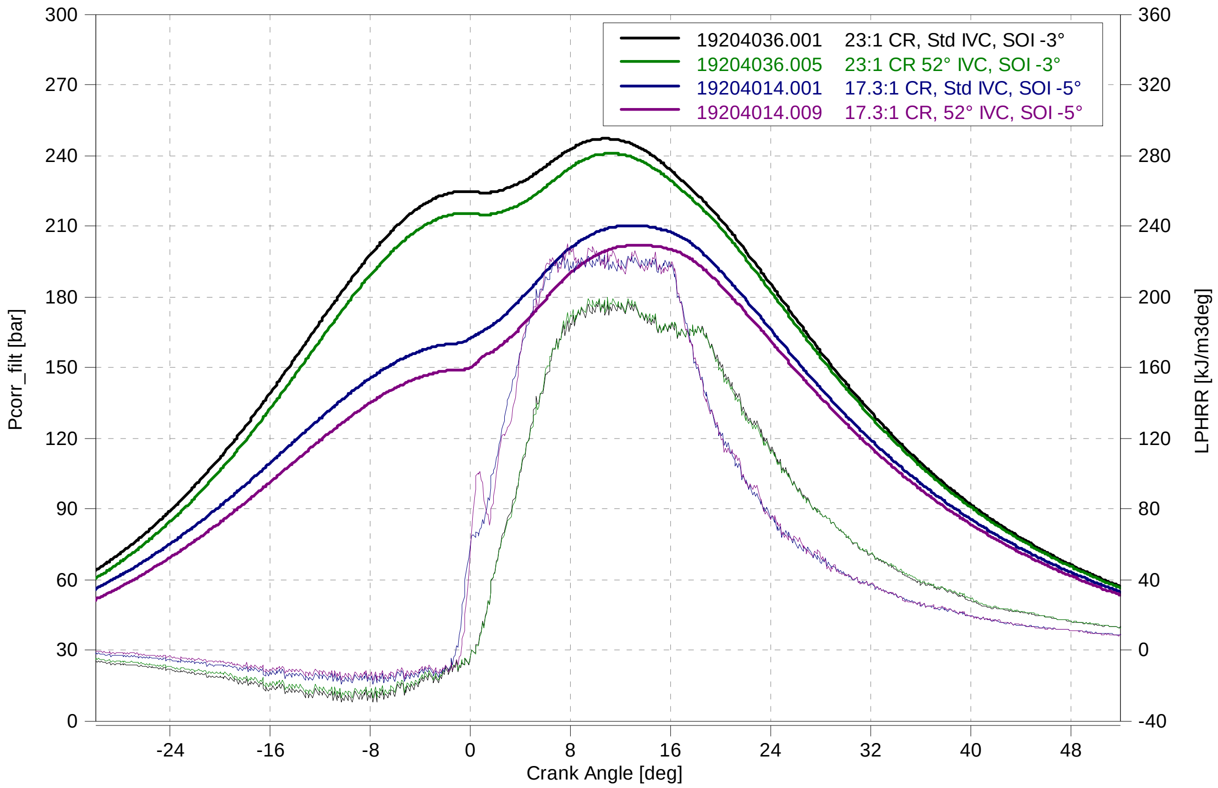

Figure 61 Cyl.press. and heat release at 1250 rpm, 300kW, with EGR, 23:1 and 17.3:1 in CR

9.5.1. EIVC without EGR

A comparison between early and late inlet valve closing was carried out on the load point without EGR. The test parameters for EIVC can be seen in Table 18.

Table 18 Test parameters for early inlet valve close 300kW without EGR

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
23.0:1	1250	2.85	1400	-12	26.1 to 27.4	1.5 to 2.35	1.5 to 2.35	612 to 639	1.38 to 1.43	0	Std to 94° earlier IVC

In Figure 62 the emissions for early and late IVC cases are plotted with the same CR. NO_x reductions look similar. HC emissions are at same levels also. The CO emissions however are a little higher for early IVC, but the smoke remained at same level in both cases. The higher CO can depend upon the lower swirl levels in the combustion camber, but on the other hand smoke emissions should have increased.

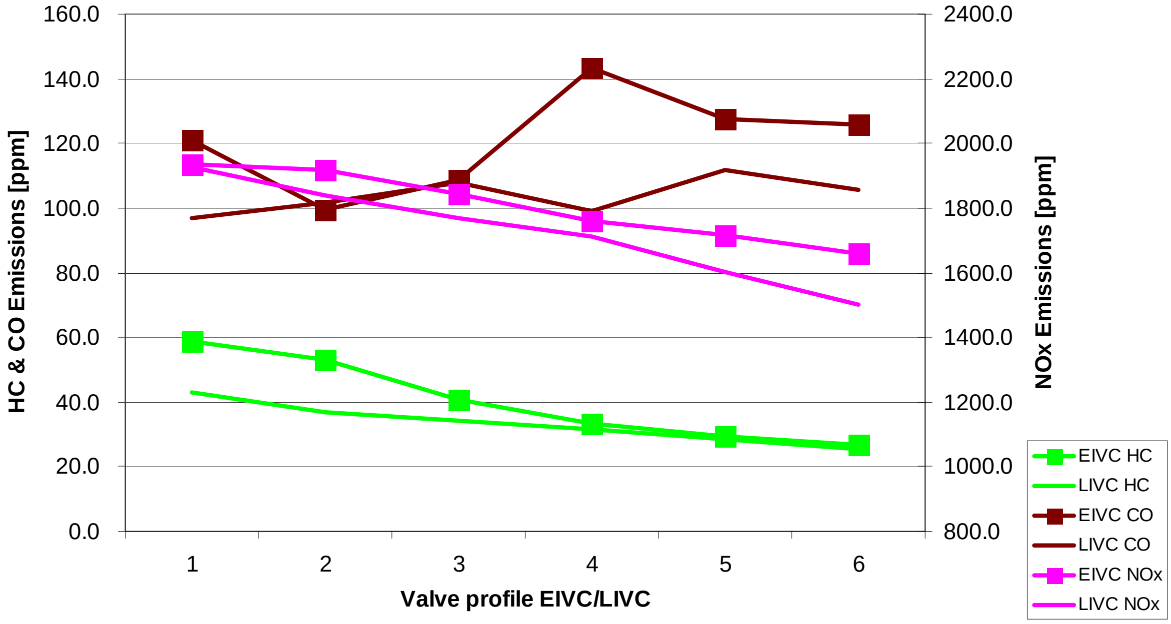

Figure 62 Emissions at 1250 rpm, 300 kW, without EGR, early and late IVC 23:1 in CR

In Figure 63 the exhaust temperatures and smoke emissions lie at the same level. The br.eff. curve looks little bit strange with EIVC, the variations can depend upon measurement errors as the differences are so small. With late IVC the torque curve looks familiar with a maximum at valve profile 4 and 5.

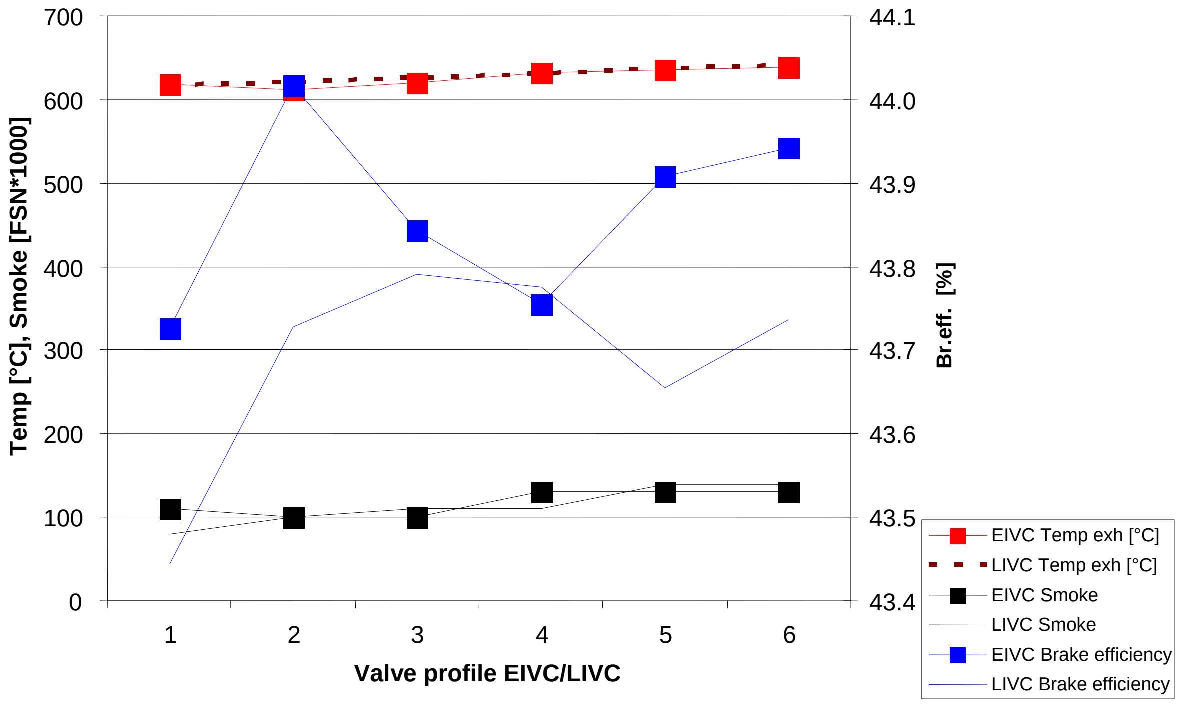

Figure 63 Exh. temp., smoke and br.eff. at 1250 rpm, 300 kW, without EGR, late and early IVC 23:1 in CR

9.6. Load point 6

Equal to 330 kW power output for a full engine with and without EGR, the test setup is shown in Table 19. This load point is a full load point and very high boost pressures are required at high Miller levels to keep lambda above 1.3. In the last case the inlet air pressure was set to 5.9 bar abs, which was the limit in the test cell. At the higher compression ratio cases the cylinder pressure limitations at 250 bar forced an adjustment of the SOI angle. The higher CR gave once again a higher torque output. The rail pressure in the cases without EGR was increased from 1400 to 1600 bar to reduce smoke emissions and to compare the differences in rail pressure with the 300 kW cases without EGR which is a load point fairly near the 330 kW load point.

Table 19 Test parameters for 330kW full engine power output

CR	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1250	2.95	1600	-10 to -15.16	27 to 30.4	1.9 to 3.6	1.9 to 3.6	610 to 666	1.37 to 1.43	0	Std to 72° later IVC
23.0:1	1250	2.95	1600	-7.34 to -9.68	26.5 to 30.7	2.0 to 3.9	2.0 to 3.9	598 to 647	1.43 to 1.5	0	Std to 72° later IVC
17.3:1	1250	2.25	2400	-5 to -8.98	50.5 to 56.5	2.9 to 4.5	3.1 to 4.7	538 to 580	1.25 to 1.28	30	Std to 62° later IVC
23.0:1	1250	2.25	2400	-1.17 to -2.11	53.9 to 60	3.1 to 4.9	3.3 to 5.0	543 to 563	1.28 to 1.32	30	Std to 62° later IVC

The same trend in emissions as previously, NO_x lower with higher Miller levels and lower with 23:1 CR than the 17.3:1 CR, see Figure 64. An interesting point here is that the CO emissions are higher for the 17.3:1 CR case, not much higher, but the lambda in this case was lower. This can have caused the higher CO emissions. The trend with lower HC with higher Miller levels is still valid.

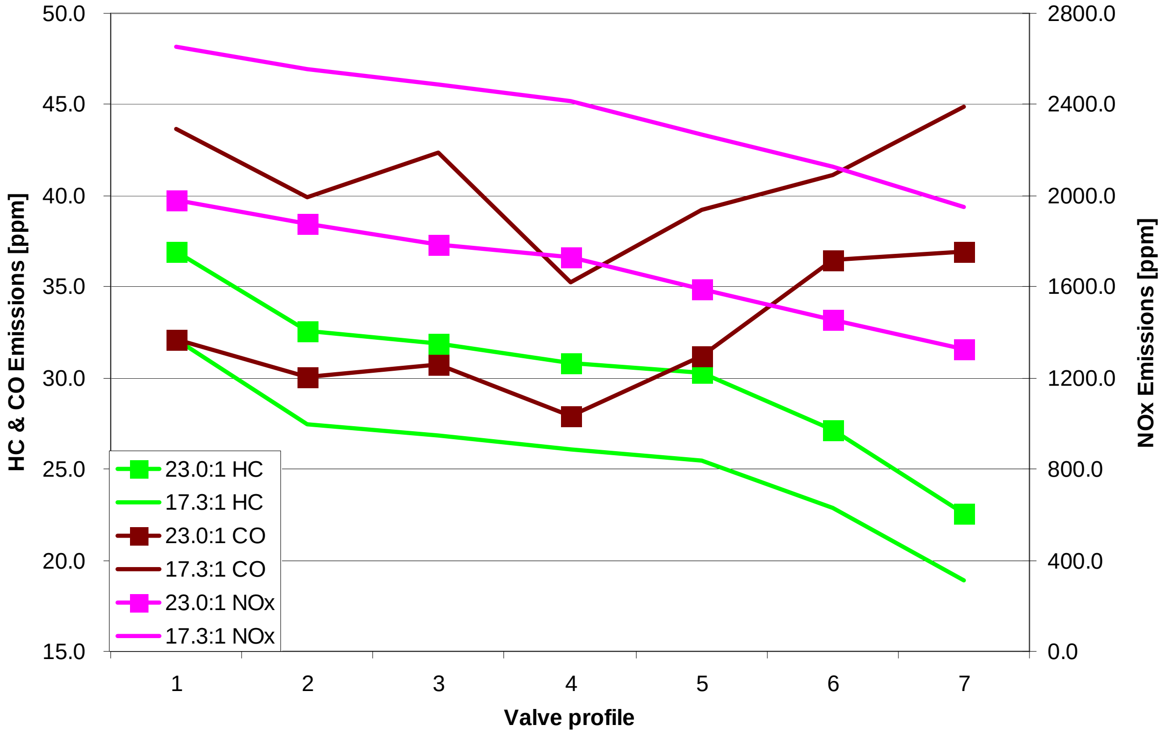

Figure 64 Emissioner at 1250 rpm, 330kW, without EGR, 23:1 and 17.3:1 in CR

Figure 65 shows the exhaust temperatures are higher in the 17.3:1 CR cases even if the SOI is retarded from -10° in this case to -7.34° in 23:1 CR cases. The exhaust temperatures increased with Miller level, a possible explanation could be that the exhaust backpressure increases with the inlet air pressure and thereby the exhaust temperature. The br.eff. output has a maximum at valve profile 4 in both CR cases, the 23:1 CR still gives the highest br.eff. with the reduced SOI. The smoke emissions are at the same level in both cases and decrease slightly at higher Miller levels.

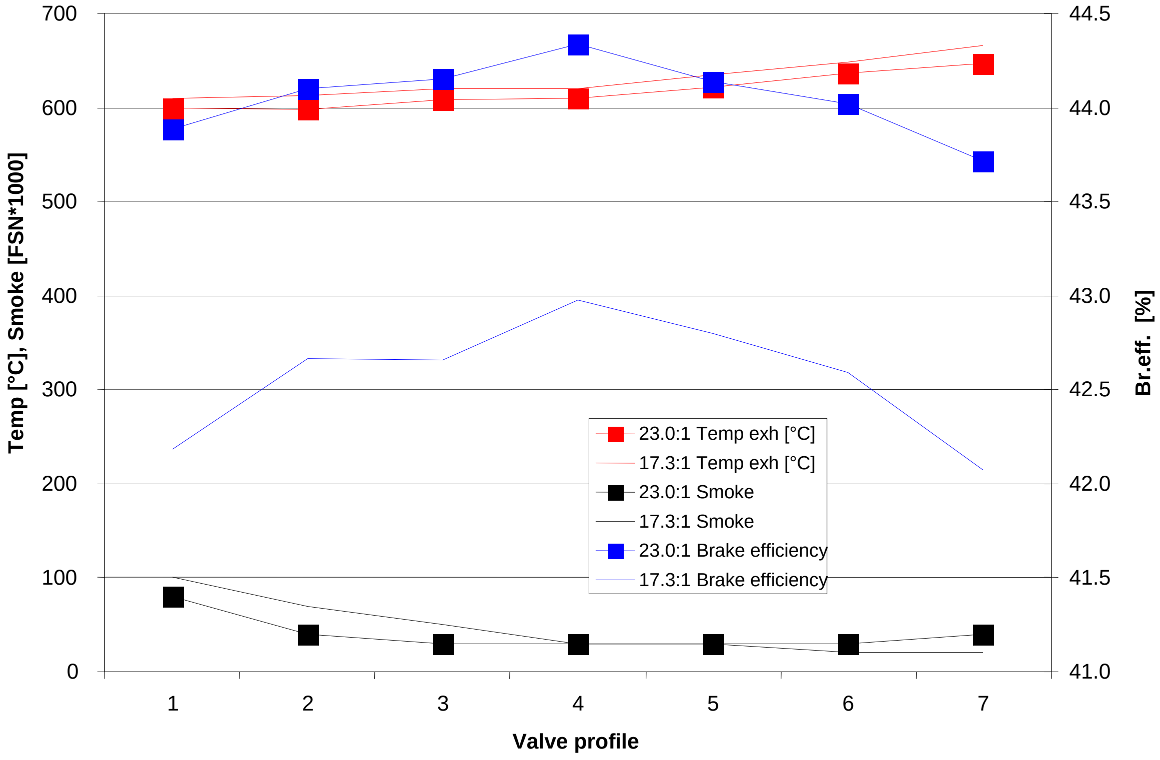

Figure 65 Br.eff. smoke and exh.temp. at 1250 rpm 330kW, without EGR, 23:1 and 17.3:1 in CR

With EGR

In the EGR cases very high cylinder pressures arise. The SOI was retarded for the 23:1 CR case from -5 deg for 17.3:1 CR to -1.17 deg, see Figure 66. This was to keep the maximum cylinder pressure under 250 bar.

NO_x decreases with higher Miller levels and is once again lower for 23:1 CR. The CO is higher for the 17.3:1 CR case, as in the cases without EGR. The smoke emissions
(see Figure 67) are lower for this case, so wall hit of the piston can be excluded. Why CO is higher in this case depends upon the lower lambda at 17.3:1 CR. The inlet air pressure was set a little higher at 23:1 in CR to compensate for the higher smoke emissions. The high cylinder pressure together with the cylinder temperature makes it difficult for HC to survive in the combustion chamber.

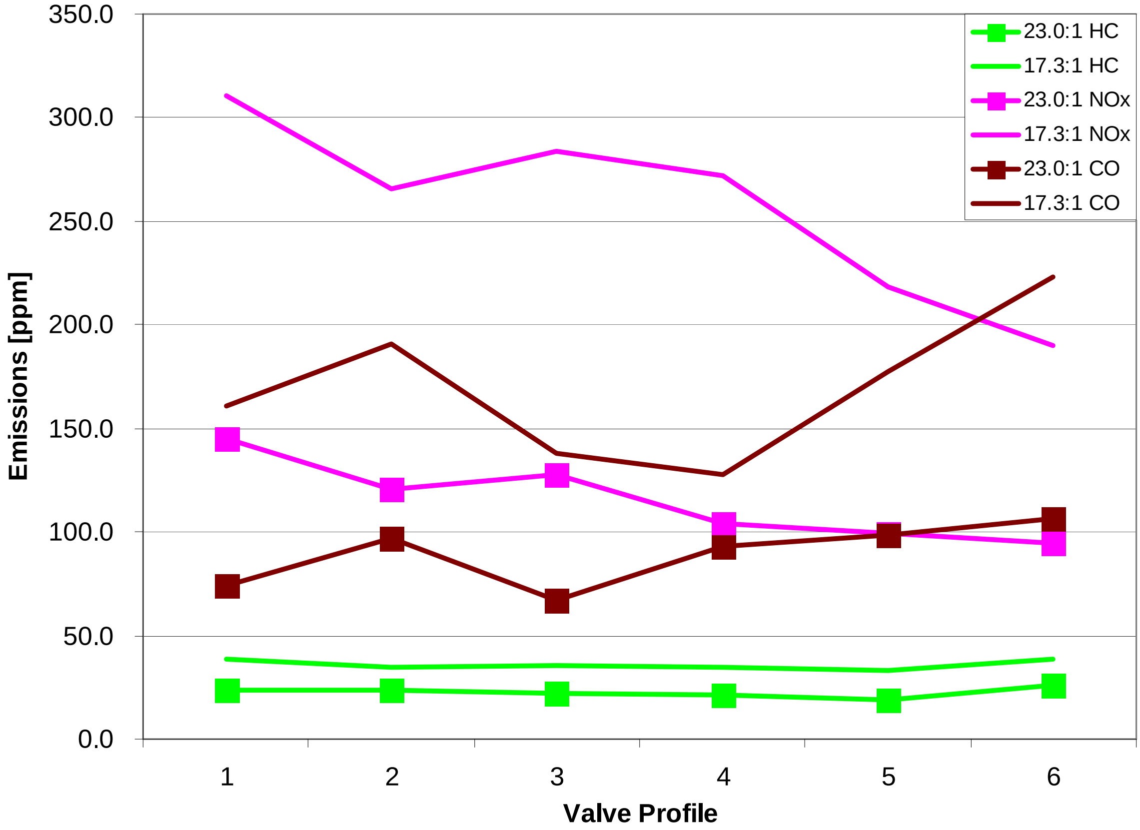

Figure 66 Emissions at 1250 rpm, 330 kW, with EGR, 23:1 and 17.3:1 in CR

The exhaust temperatures are a little higher in the 17.3:1 CR case, like before, see Figure 67. The br.eff. output is higher for the 23:1 CR case even when the SOI was adjusted to get the cylinder pressure under 250 bar. Smoke emissions are higher for the 23:1 CR case even when the inlet pressure was increased to maintain a reasonable level of smoke emissions. The later SOI can be one explanation of the higher smoke.

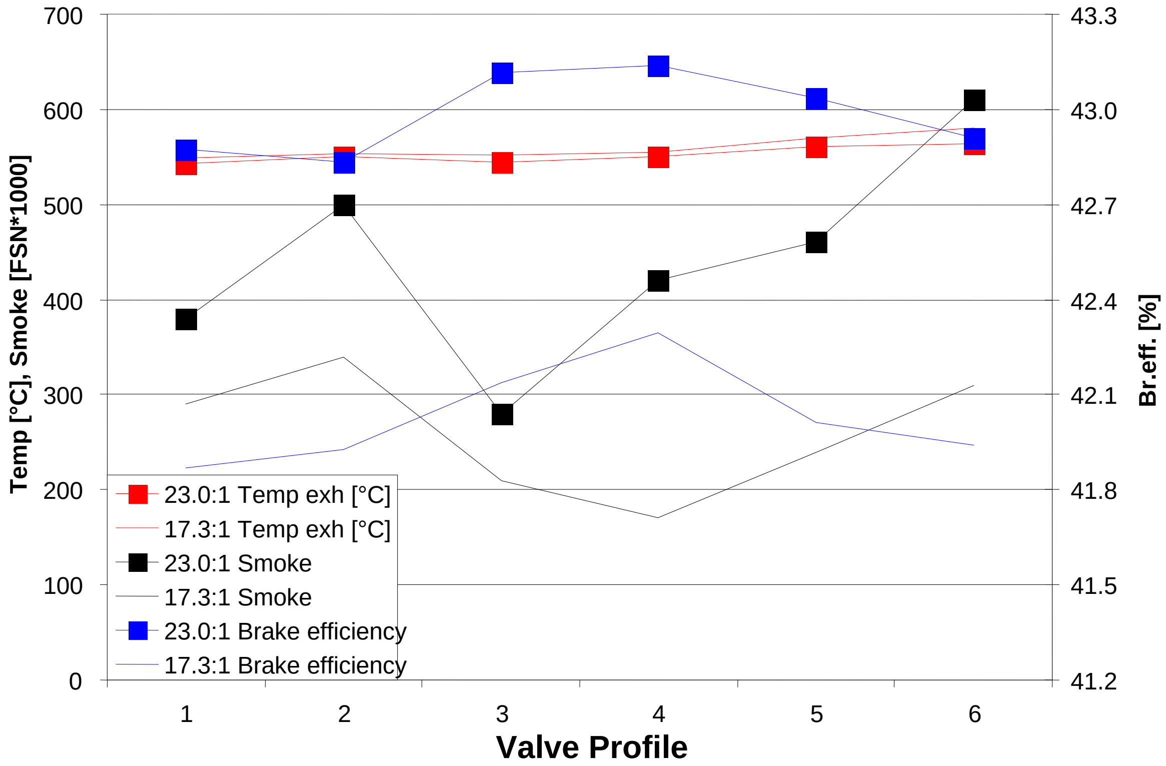

Figure 67 Br.eff., smoke and exh. temp. at 1250 rpm, 330 kW, with EGR, 23:1 and 17.3:1 in CR

In Figure 68, the cylinder pressure and HR are plotted for two different compressions and with/without a Miller valve profile. All tests are P-max limited, SOI are set so the cylinder pressure does exceed over 250 bar. The MBT SOI is also set where and when an increase in torque was found. The SOI was a lot different between the two compressions; the higher compression has essentially a later SOI. The torque output is higher for all 23:1 comp cases compared to 17.3:1 comp cases.

With these high loads the ignition delay is short and just a small difference in delay can be seen at higher Miller levels. The green line is the standard valve profile and the compression pressure has reached 250 bar, thereby SOI can not be as early as desired. The black curve is a Miller profile and the SOI can be set a little earlier or Miller can be used to reduce P-max. The blue and magenta lines are cylinder pressure and or MBT points for 17.3:1 CR. A little premixed combustion can be seen in the Miller case. The heat release looks nearly the same in all the cases.

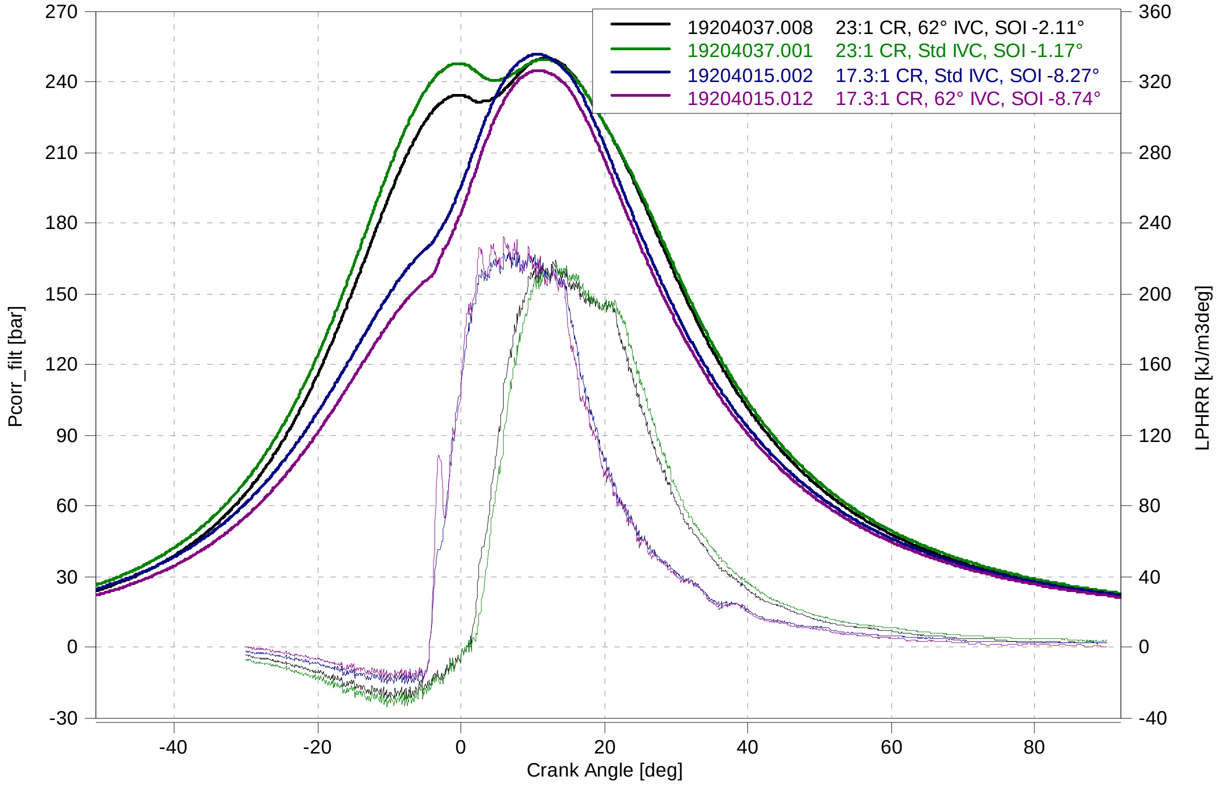

Figure 68 Cyl.press. and heat release at 1250 rpm, 330 kW, with EGR, 23:1 and 17.3:1 in CR

In Figure 69, the P-V diagram is plotted for the same cases that are plotted in Figure 68. The Miller cases have a much higher pressure level in the pumping cycle then the standard cases. The exhaust pulse that goes under the inlet pressure in the Miller cases that we have seen in lower load points is also shown here. The standard cases did not have this pressure dip. The pressure dip arises because of the longer expansion stroke than the inlet stroke with a Miller cycle.

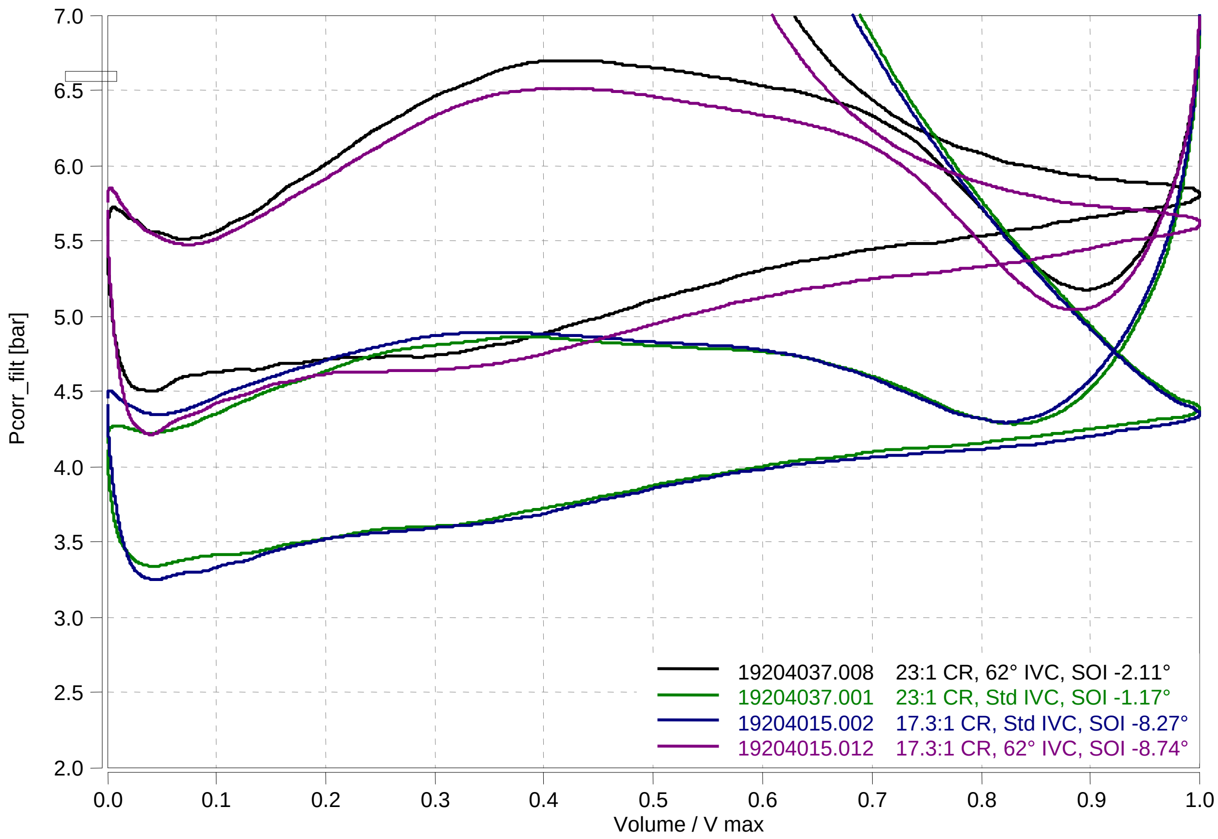

Figure 69 P-V diagram at 1250 rpm, 330 kW, with EGR, Miller and std cam profiles, 23:1 and 17.3:1 in CR

9.7. Load point 7

Equal to 70 kW full engine power at 1600 rpm. At this load point the Miller strategy was tested with high engine speed and low load. Throttle losses when a Miller cycle is used are assumed to increase with higher engine speed. In Table 20 the test setup can be seen. Boost pressures were not required for any of the cases, lambda was allowed to drop at higher Miller levels to increase the exhaust temperature.

Table 20 The testdata for 70kW and 1600 rpm loadpoint

Comp	Speed [rpm]	Ontime [ms]	Rail Press [bar]	SOI [°]	Inlet Temp [°C]	Inlet press [bar] rel	Exh press [bar] rel	Exhaust temp [°C]	Lambda	EGR [%]	Miller Strategy
17.3:1	1600	1.21	700	-12 to -19.29	26.1 to 28.9	0.1	0.15	346 to 449	2.53 to 1.84	0	Std to 62° later IVC
23.0:1	1600	1.21	700	-12 to -18.3	26.7 to 29.7	0.1	0.15	344 to 477	2.72 to 1.93	0	Std to 62° later IVC
17.3:1	1600	0.89	1200	-12 to -22.1	31.4 to 36.4	0.1	0.3	367 to 461	1.8 to 1.28	30	Std to 52° later IVC
23.0:1	1600	0.89	1200	-12 to -18.3	30 to 34.3	0.1	0.3	350 to 415	2.01 to 1.62	30	Std to 42° later IVC

The emissions behave a little different from earlier cases. In the 23:1 CR case NO_x levels decrease with higher Miller level, see Figure 70. The 17.3:1 CR case showed that NO_x increases with Miller level and the smoke decreases. The longer ignition delay gives smaller

areas with low lambda that leads to lower smoke emissions but longer ignition delays give a faster and greater premixed heat release that gives higher NO_x emissions. The NO_x are higher in the beginning for the 23:1 CR case. A trade off appears between cylinder temperature and ignition delay that increases NO_x emissions. HC increased a little at higher Miller levels, CO increased a lot in both cases. The increase in CO can be explained by lower compression temperature at high Miller levels. In the 23:1 CR case the smoke emissions rise, see
Figure 71, wall hit can not be excluded. In the 17.3:1 CR case the smoke decreases, deeper penetration of the spray and colder combustion gave higher CO. The smoke are reduced by the fact that a longer ignition delay gives a more premixed combustion with less low lambda zones that gives high soot emissions.

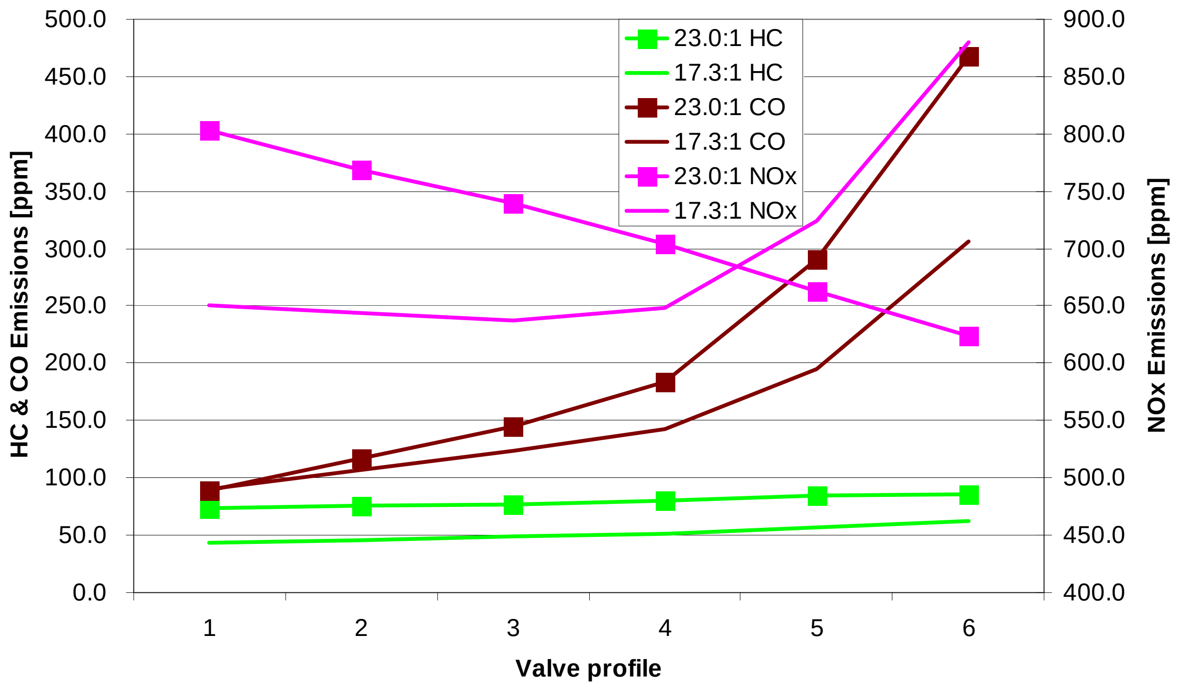

Figure 70 Emissions at 1600 rpm, 70 kW, without EGR, 23:1 and 17.3:1 in CR

The large differences in smoke emissions seen in Figure 71 are a sign of wall wetting on the piston for the 23:1 CR case. The exhaust temperature increased with Miller level, there was no difference between the two CRs. The br.eff. output is higher for 23:1 CR case decreasing with Miller level. Miller could be used in this case to raise the exhaust temperature and reduce lambda.

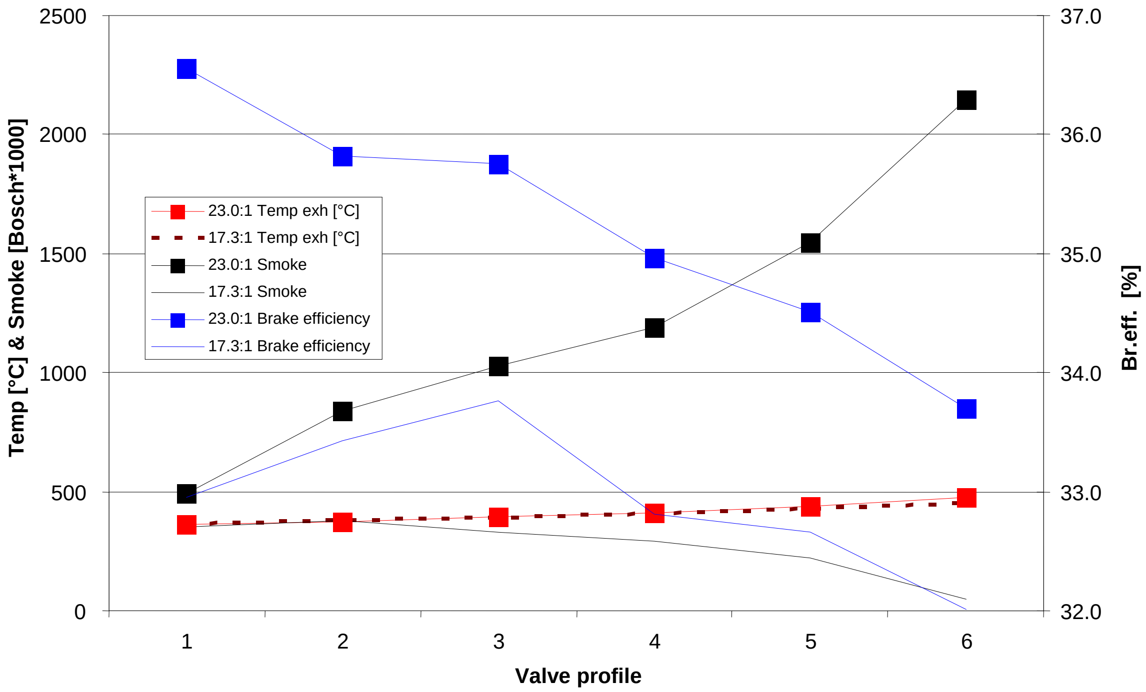

Figure 71 Br.eff., smoke and exh. temp. at 1600 rpm, 70 kW, without EGR, 23:1 and 17.3:1 in CR

In Figure 72 the cylinder pressures and heat releases for the two different CR and valve profiles are plotted for the same SOI. At this load point where the engine speed was increased to 1600 rpm it was necessary to advance SOI to ensure an effective combustion. The highest cylinder pressure was recorded with the 23:1 CR case along with the standard valve profile. The ignition delays were shorter for this case compared to the other plotted cases and the cylinder pressure rise was the lowest.

An inspection of the pressure traces for the standard valve profile 17.3:1 CR compared to the 62° Miller case and 23:1 CR shows that the peak pressure increases and HR are similar.
23:1 CR has a lower compression pressure and a lower expansion pressure, the higher torque output in 23:1 CR case is a result of these pressure conditions. The ignition delay in the 17.3:1 CR case decreases with the higher Miller levels, at the same time a higher premixed combustion can be seen.

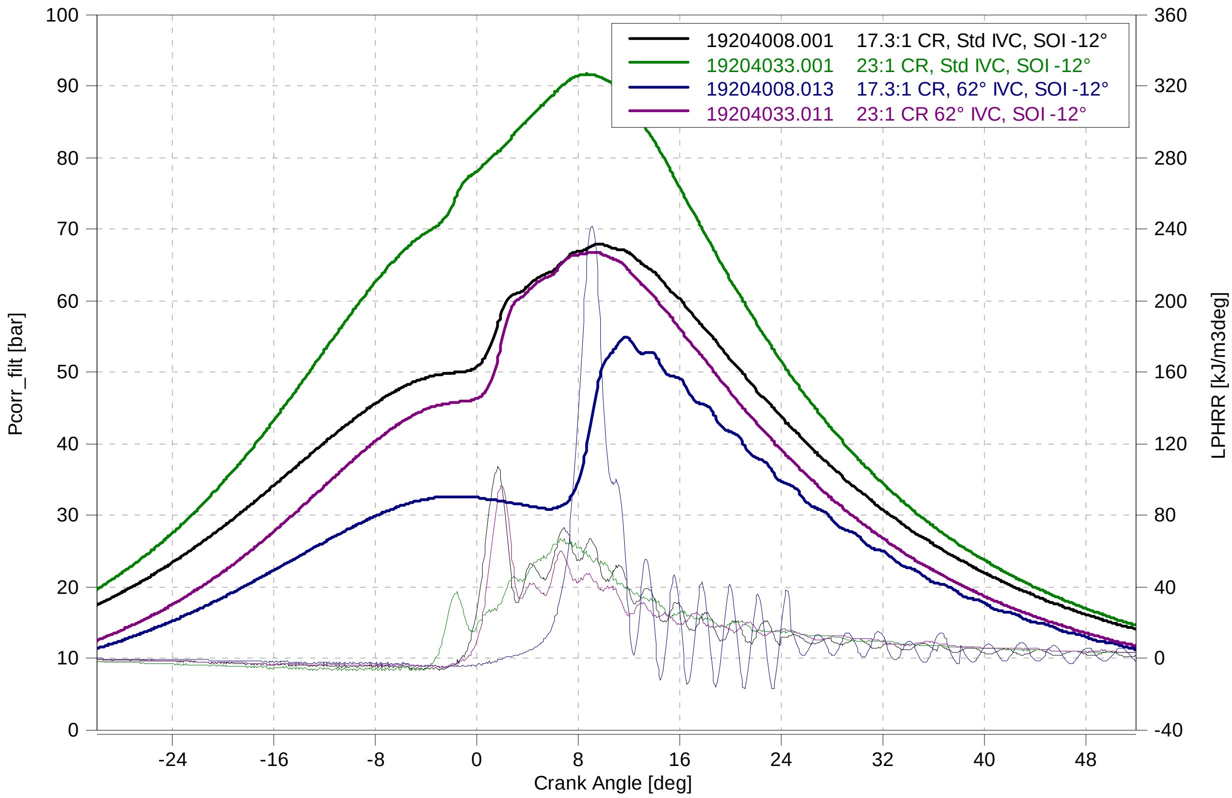

Figure 72 Cyl.press. and heat release at 1600 rpm, 70 kW, without EGR, 23:1 and 17.3:1 in CR

In Figure 73 the P-V diagram is plotted for the standard valve profile and Miller valve profile with 23:1 and 17.3:1 in CR. With this higher engine speed the exhaust pressure pulse does not go down under the inlet pressure in the Miller case like the cases at lower engine speed. Still this pulse is nearer the inlet pressure than the standard valve profile case. The exhaust pressure is lower for all the cycle. This because of the longer expansion ratio compared to the compression ratio and the smaller amount of exhaust gases in the Miller case. The lower amount of exhaust gases depends on the lower lambda when no boost pressure is in use in this case.

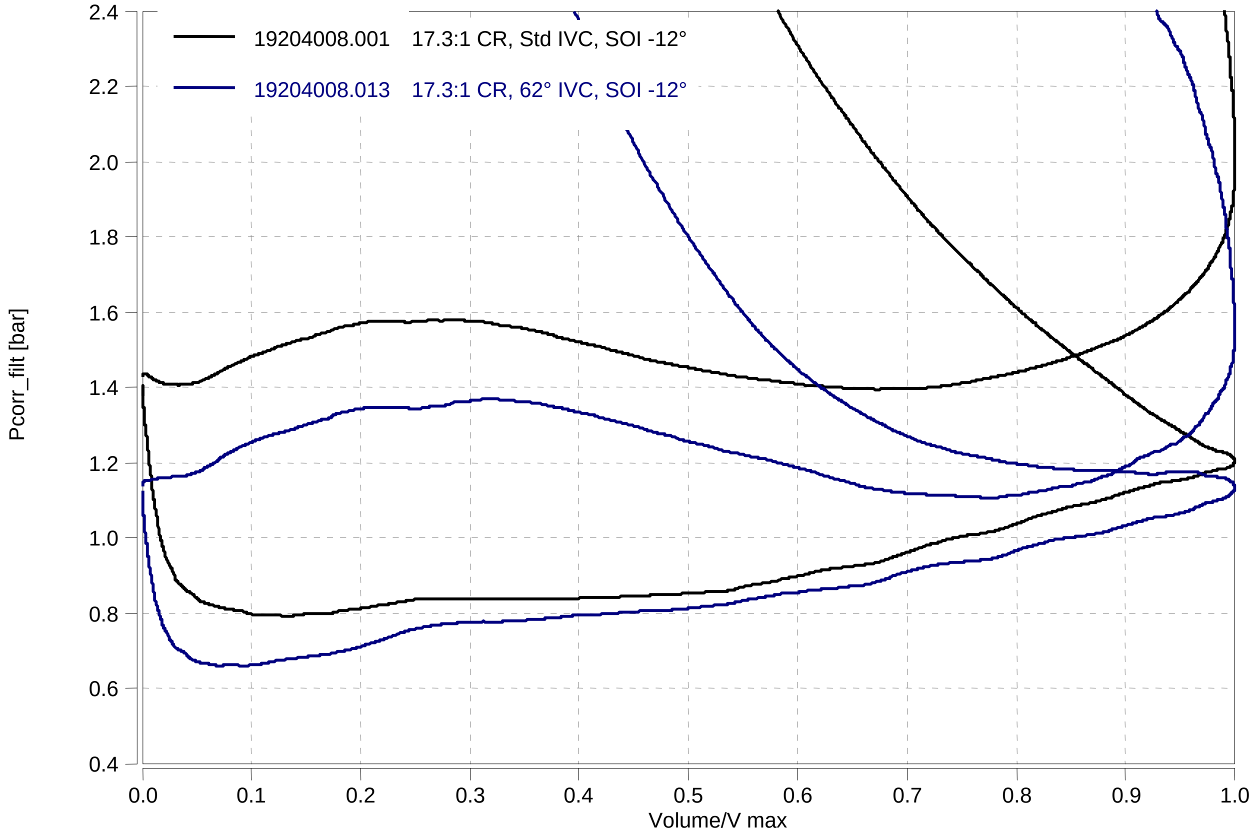

Figure 73 P-V diagram at 1600 rpm, 70 kW without EGR and 17.3:1 in CR

With EGR

In the EGR case no boost pressure was needed, but to get EGR from the exhaust manifold over to the inlet manifold the exhaust pressure was set to 200 mbar higher than the inlet pressure. In this case the NO_x and HC emissions were nearly the same for the two CR, see Figure 74. The CO emissions increased with EGR. CO levels were still higher for the 23:1 CR cases. The higher CO was caused by lack of oxygen when EGR is used.

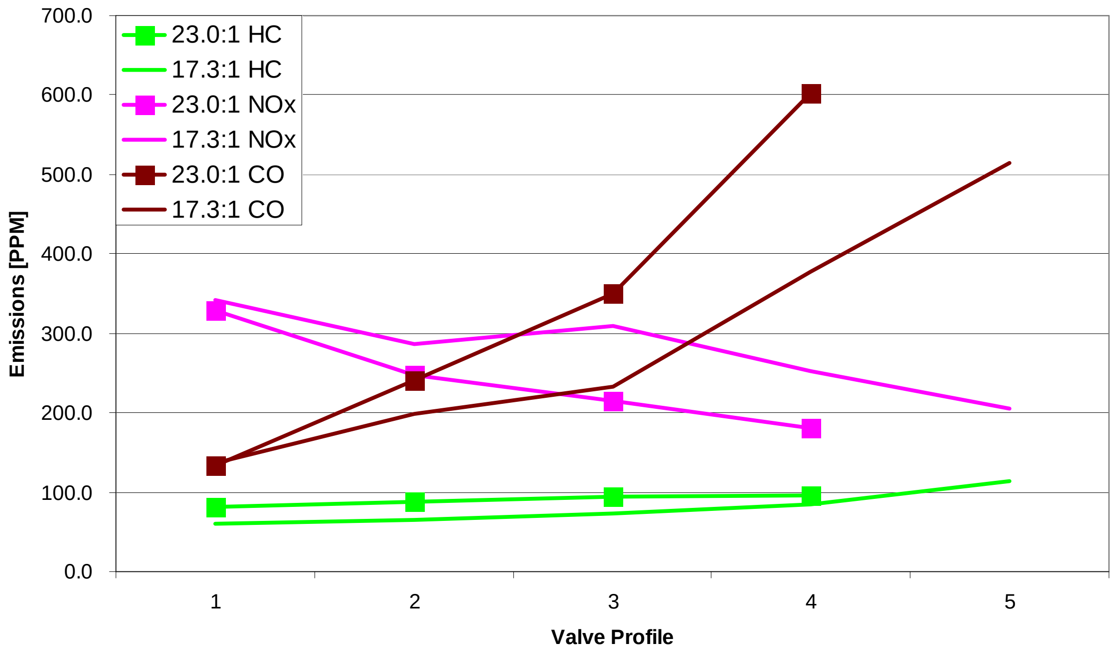

Figure 74 Emissions at 1600 rpm, 70 kW, with EGR, 23:1 and 17.3:1 in CR

The increase in injection pressure leads directly to lower smoke emissions, see Figure 75, the last measurement point in the case with 23:1 in CR. The exhaust temperature increases with Miller level and are a little higher with 17.3:1 in CR. The br.eff. output is higher with 23:1 CR and decrease with Miller level. For the 17.3:1 in CR the highest br.eff. is at 22° later IVC.

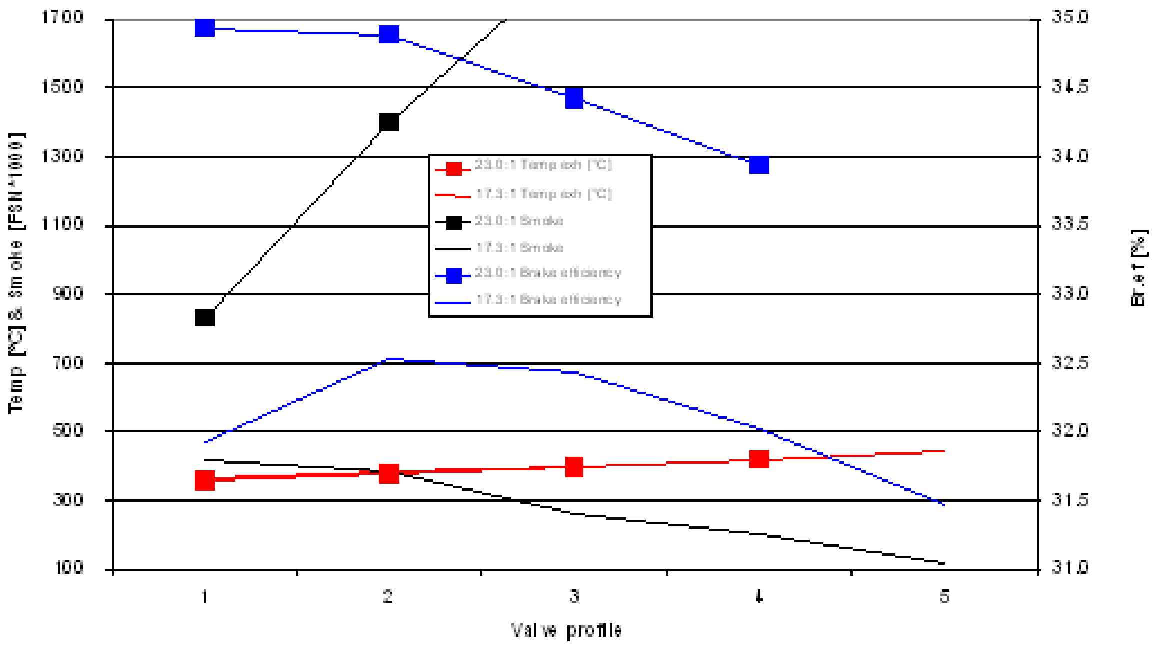

Figure 75 Br.eff., Smoke and exh. temp. at 1600 rpm,70kW, with EGR, 23:1 and 17.3:1 in CR

9.8. Calibration of the GT-power Model

The single cylinder model that was used to investigate the compression and Miller strategies before the test engine was run was now calibrated using data from the single cylinder engine which had been run in the test cell. The logged cylinder pressure trace together with the data for the fuel/air consumption, temperature and pressure were used to adjust the GT-power model.

First step in the calibration process was to set the outer boundary conditions of the engine. The engine speed was set to 1250 rpm and low, medium and high loads were calibrated. The outer pressure of the outer gas exchange system needed to be set to the right level. In this case the Endenvironment box in GT-power was changed to Endflowinlet, see Figure 76.

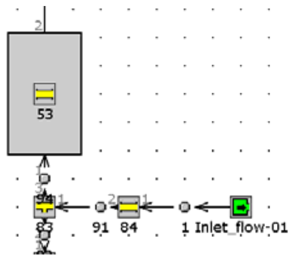

Figure 76 The intake system changes

In this function the amount of air in kg/min can be set so that the model uses the same as in the test cell engine. The inlet pressure was then adjusted so the inlet air mass was correct. The back pressure that was used in the test cell was then set in exhaust end environment. The inlet temperature was then checked in the model. An inlet temperature sensor was located in the inlet manifold on the single cylinder in the test cell. The heat transfer and wall temperature of the inlet manifold was adjusted so the temperature was in the same range. The heat transfer in the inlet manifold and wall temperatures were important when late inlet valve closing was used compared to a standard or early IVC. This is because air was pushed back out from the cylinder at the end of the inlet stroke when the piston was on its way up from BDC and the inlet valves were still open. In the model another inlet manifold was installed compared to the one in the test cell, the inlet manifold dimensions were thereby changed so it could be computed accurately.

The heat transfer model was calibrated in GT-power by measured data. To make the calibration a skip fire test was done in the single cylinder test cell. The engine was run with combustion and load for some time so that all the boundary temperature conditions were constant. Then the fuel injection was stopped and the pressure trace was logged directly after for the following cycle. In Figure 77 the pressure trace for the combustion cycle can be seen together with the skip fire cycle that was measured immediately after. The idea of this test was to maintain the same temperature and boundary conditions in the cylinder that it has during combustion and compare this pressure trace with the simulated pressure trace in GT-power. The heat transfer could then be adjusted so that the two pressure traces coincided.

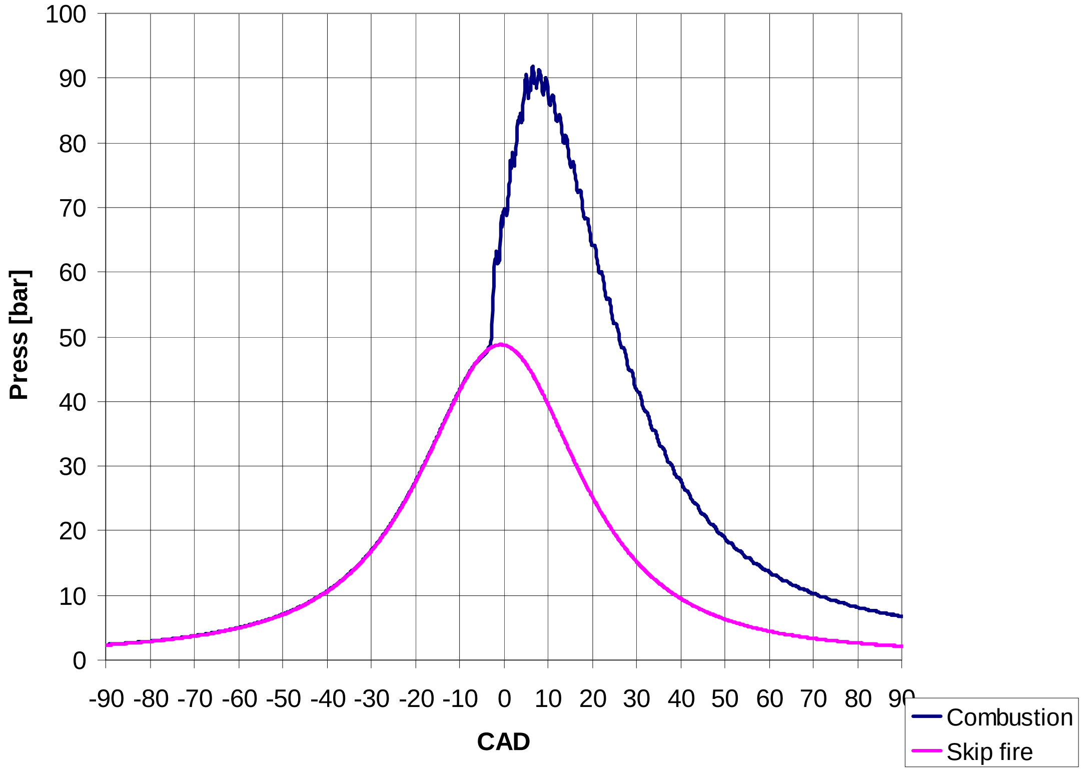

Figure 77 Skip fire at 1250 rpm for 17.3:1 in CR

Before the heat transfer model could be adjusted the correct boundary conditions during the pumping cycle needed to be set. In Figure 78 the P-V diagram is plotted for the measured and the simulated pressure traces. The same skip fire pressure trace is used for this investigation. To match this pumping trace to the measured pressure, an adjustment of the valve timing was needed. The exhaust valve timing was moved -7° to calibrate the model to the fixed profile of the camshaft for the exhaust valves. Tappet clearance was adjusted in the model. In the real engine tests only the inlet valve profile was possible to adjust. This was because the lotus AVT system was only used on the inlet valves not the exhaust valves. The exhaust valves were controlled using the standard camshaft with standard valve mechanism.

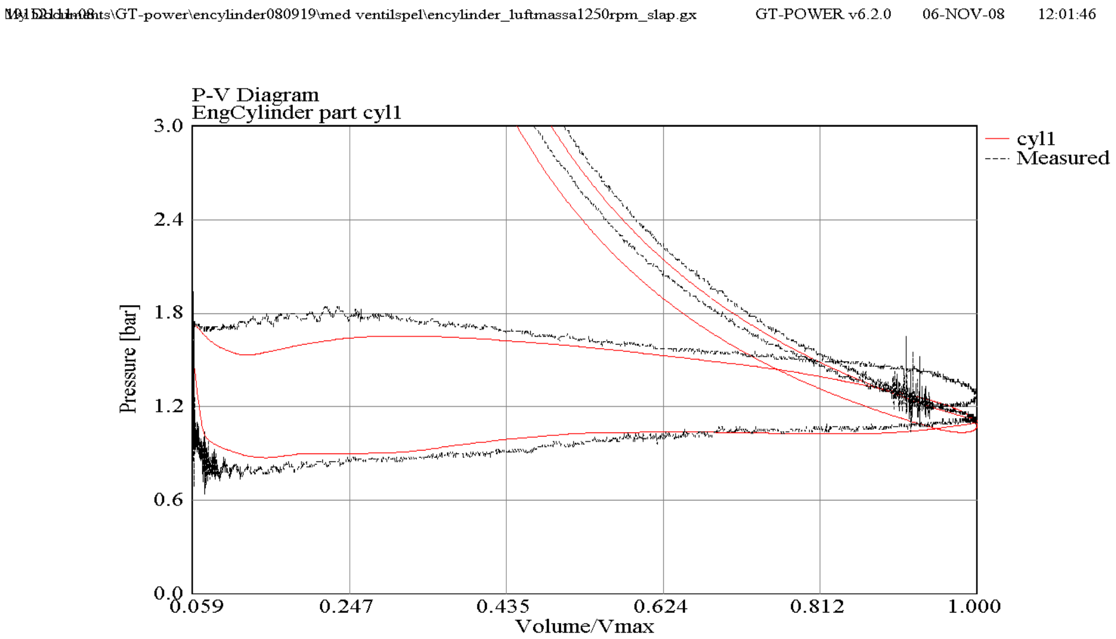

Figure 78 P-V diagram with measured and simulated pressure traces for the single cylinder GT-power model.

Now the Heat transfer model could be calibrated. A test with the different heat transfer models was carried out, see Figure 79. All of the parameters were set like for all of the heat transfer models to be able to correctly compare them. The measured cylinder pressure was compared to the built-in heat transfer models in GT-power. The flow model was the one that best modelled the measured cylinder pressure after all the boundary parameters in GT-power were set correctly.

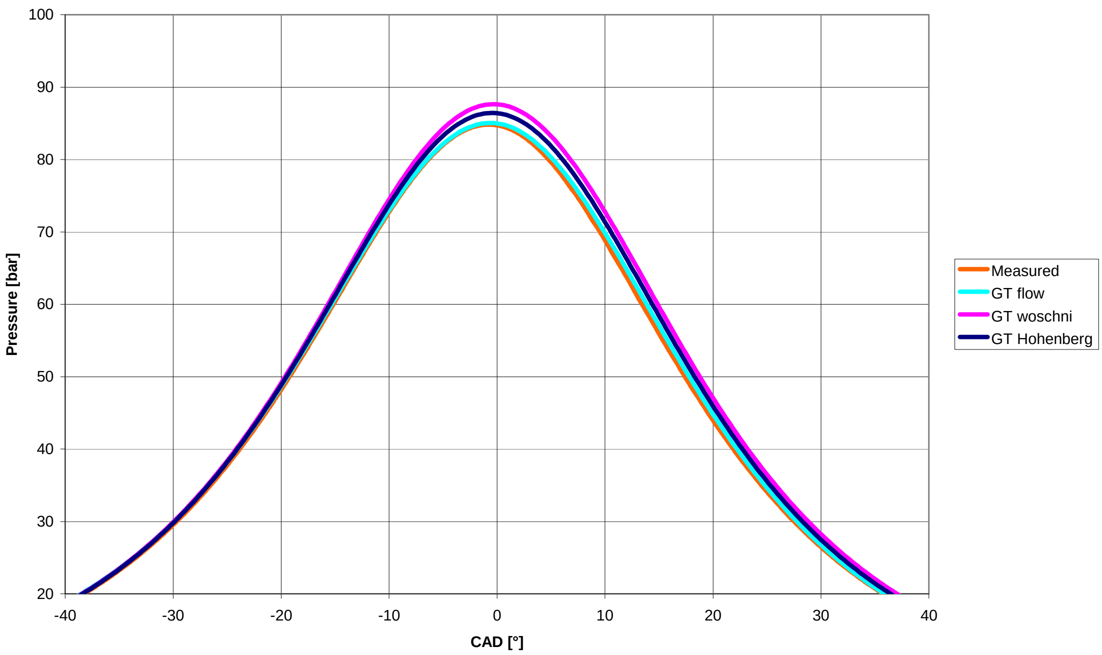

Figure 79 Measured cylinder pressure compared with simulated cylinder pressure with different Heat transfer models at 1000 rpm and 1000 mbar inlet and outlet pressure

The chosen Heat transfer model was the flow model that is available in GT-power and goes by the name EngCylFlow. This object calculates the swirl, turbulence and tumble used by the combustion model EngCylCombDIJet and/or the heat transfer model option in EngCylHeatTr. If another combustion model should be used the swirl and tumble that are specified in the calculations will not be used by the flow object. In the model a HR profile combustion model was used and the Swirl was specified.

In the flow object the piston bowl needs to be specified. In Figure 80 the piston measurements that needed to be specified in the model can be seen. The piston bowl shape is a little different to the actual piston bowl in the real engine. A compromise had to be reached in the accuracy of the measurements of the cylinder geometries. They are as close as possible in respect to swirl, piston surface area, piston depth, and clearance heights etc. Swirl values were store in matrixs for each load point.

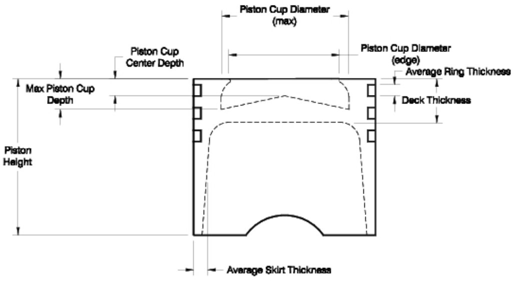

Figure 80 Piston cup measurements that needed to be specified in GT-power

The temperature zones in the combustion chamber need to be defined in the flow model. The wall temperature object, EngCylTWallSoln, calculates the wall temperature in the combustion chamber. The calculation is done once per cycle and is based on a finite element model (FEM) in the combustion chamber, see Figure 81. The cylinder temperature zones are divided into three different zones. To calculate the temperature in these zones the cylinder is specified by material, material thickness, heat transfer coefficients, water jacket etc. The piston is also divided in three different temperature zones and in the same way the material and heat transfer coefficients are specified. Models are then used to calculate the wall temperature in the piston bowl with the heat release and combustion temperature. The cylinder head has also three temperature zones, the valve temperatures are calculated separately. Note in the picture that the injector nozzle is calculated separately too.

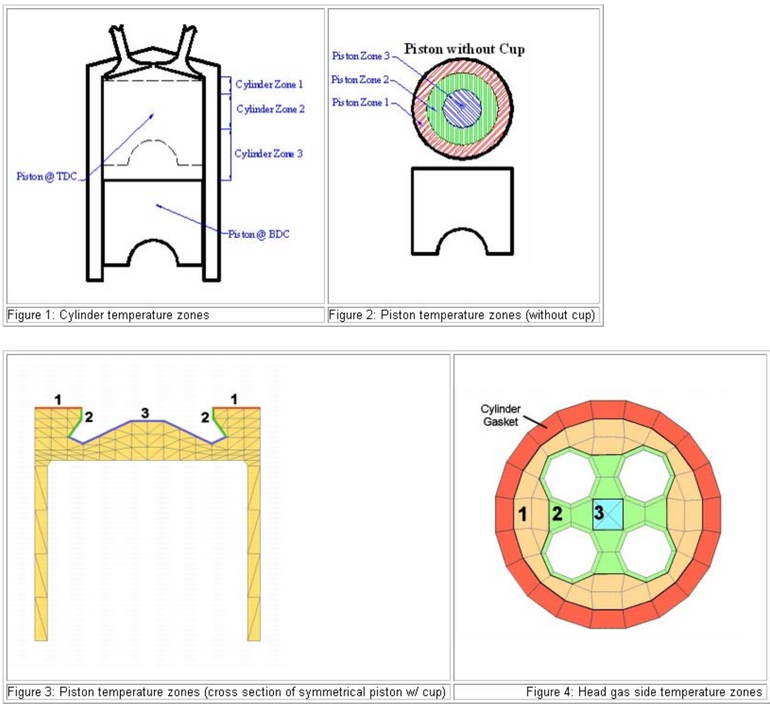

Figure 81 The temperature zones in GT-power

The skip fire pressure trace together with the simulated pressure trace is plotted in Figure 82. In GT-power documentation the calibration of cylinder pressure traces should be done with an accuracy of 5%. To fulfil that condition some adjustments of the cylinder wall heat transfer coefficients were made to closer resemble the measured pressure trace.

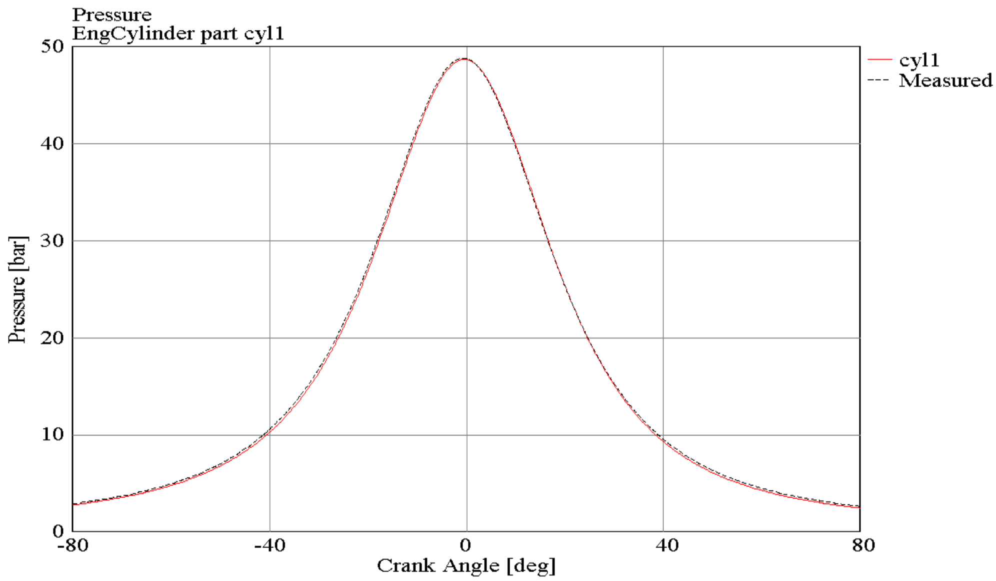

Figure 82 Simulated and measured pressure trace for first cycle after combustion was stopped

After calibration with the skip fire cycle a pressure trace with combustion was set in the model. The heat release from the measured data of the test engine was inserted in the GT-power model and the pressure traces were compared again, see Figure 83 and Figure 84.

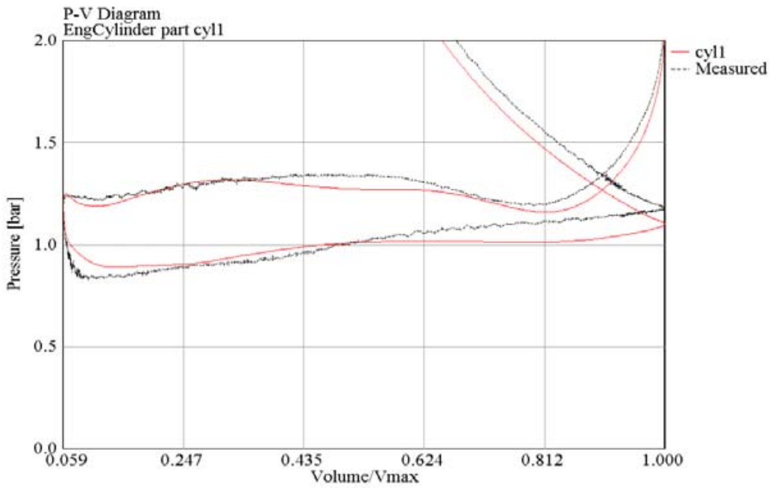

Figure 83 P-V diagram for measured and simulated data at 1250 rpm with combustion

The HR rate was shortened to better reflect the measured pressure trace. This must sometimes be done when the measured HR ends in fluctuations and it is difficult to see were the HR ends. A CAD value where the HR has finished is specified in GT-Power. The HR amplitude is then calculated by GT-Power according to specified injected fuel amount and combustion efficiency. Only the HR shape is used in GT-Power. The con-rod stiffness was also modified. This was done to get a better pressure trace match at high cylinder pressure. Not only the con-rod is deformed under high cylinder pressure, a lot of other components also change. To take these deformations into the calculations the con-rod stiffness was reduced. The actual con-rod stiffness was set to 742 MN/m.

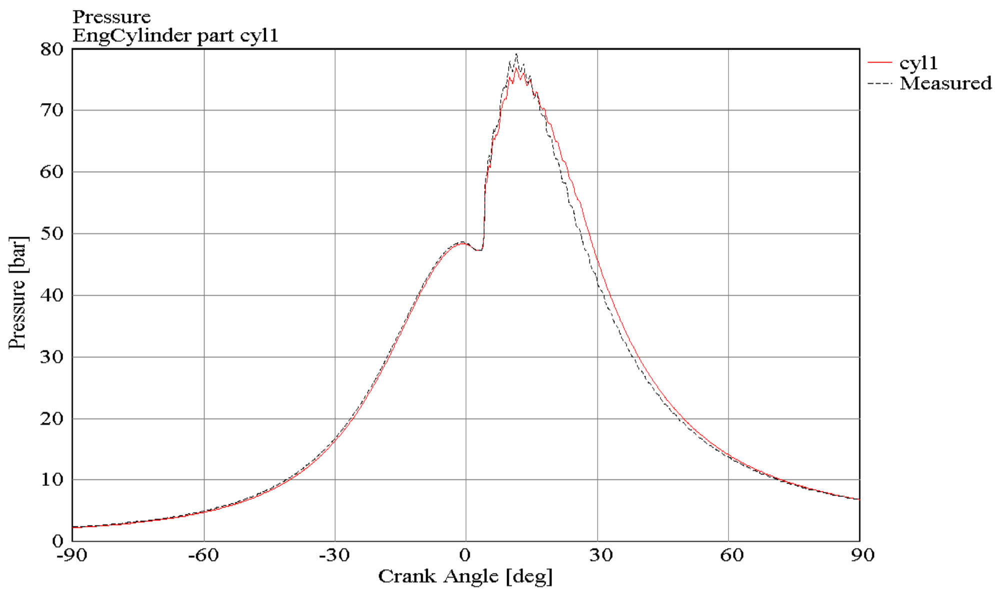

Figure 84 Cylinder pressure for measured and simulated data at 1250 rpm with combustion

The model was then run at different loads and compared to the corresponding measured pressure traces to confirm that the calibrated model was accurate for more than just the calibrated load point.

9.9. Turbo simulations

To investigate the possibility of having a full scale Miller engine, a full engine with turbo was simulated in GT-power. To choose a turbo that could give the engine the required performance three load points were tested. These load points were the full torque output at 1000 rpm, 1250 rpm and 1900 rpm and define the desired torque output curve. The single cylinder test engine was run at the same engine speed and power output (recalculated so the same BMEP was used as the full engine) and data was recorded. This was done with and without EGR for the different compression ratios and power outputs. The turbo pressure was set at reasonable levels to help control smoke emissions and lie within reasonable limits of what a real turbo system could deliver. The back pressure was set at the same level as the inlet pressure in non EGR cases and with an increase of 200 mbar in the EGR cases (to help the EGR go into the inlet manifold). The minimum for lambda was set to 1.3 and EGR to 30 %.

The measured combustion data was now used in the GT-power model, the model is described earlier in this report. Some modifications were made to calibrate the model correctly. The combustion model was modified and measured HR from the test cell was used. In Figure 85 and Figure 86 the measured cylinder pressures and HRs can be seen for the non EGR cases. In Figure 85 the HR is for 17.3:1 CR and a standard IVC and Figure 86 for 23:1 CR and 32° LIVC. The HR profiles are quite similar for the two cases at this high loads.

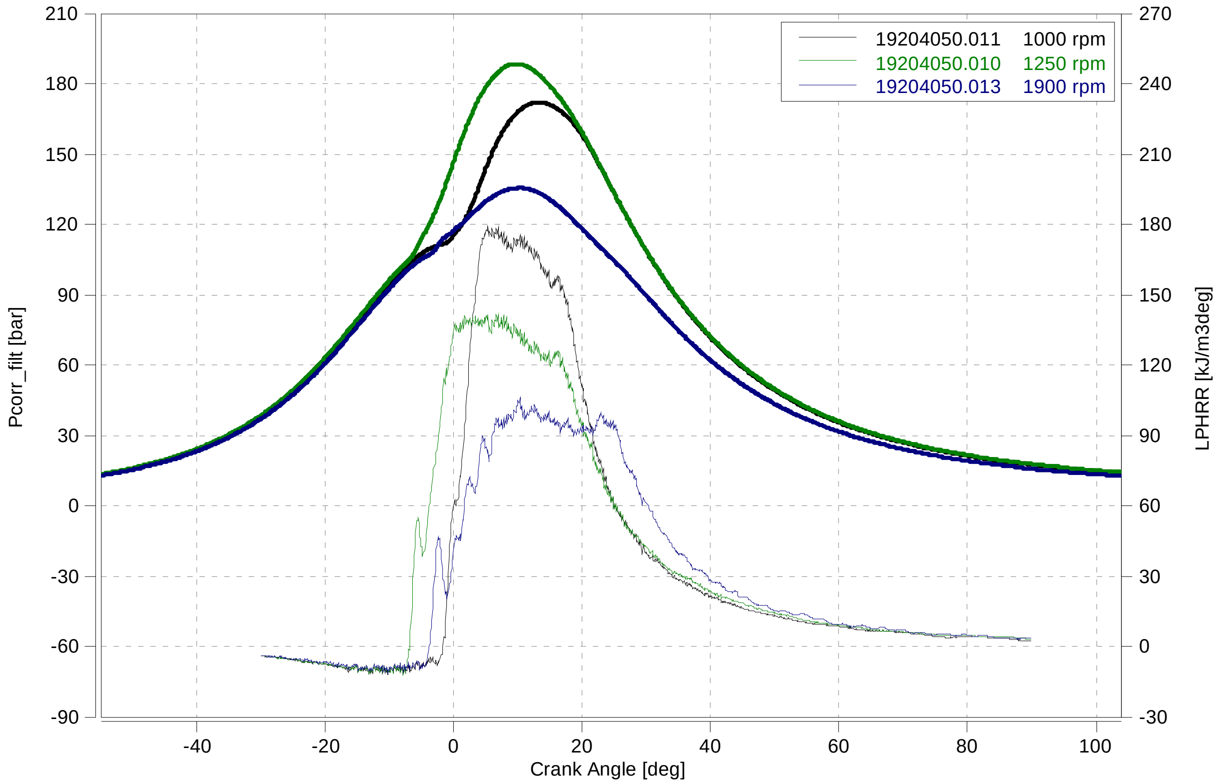

Figure 85 Turbo match points at 1250 rpm ,300 kW, without EGR std. valve profile and 17.3:1 in CR

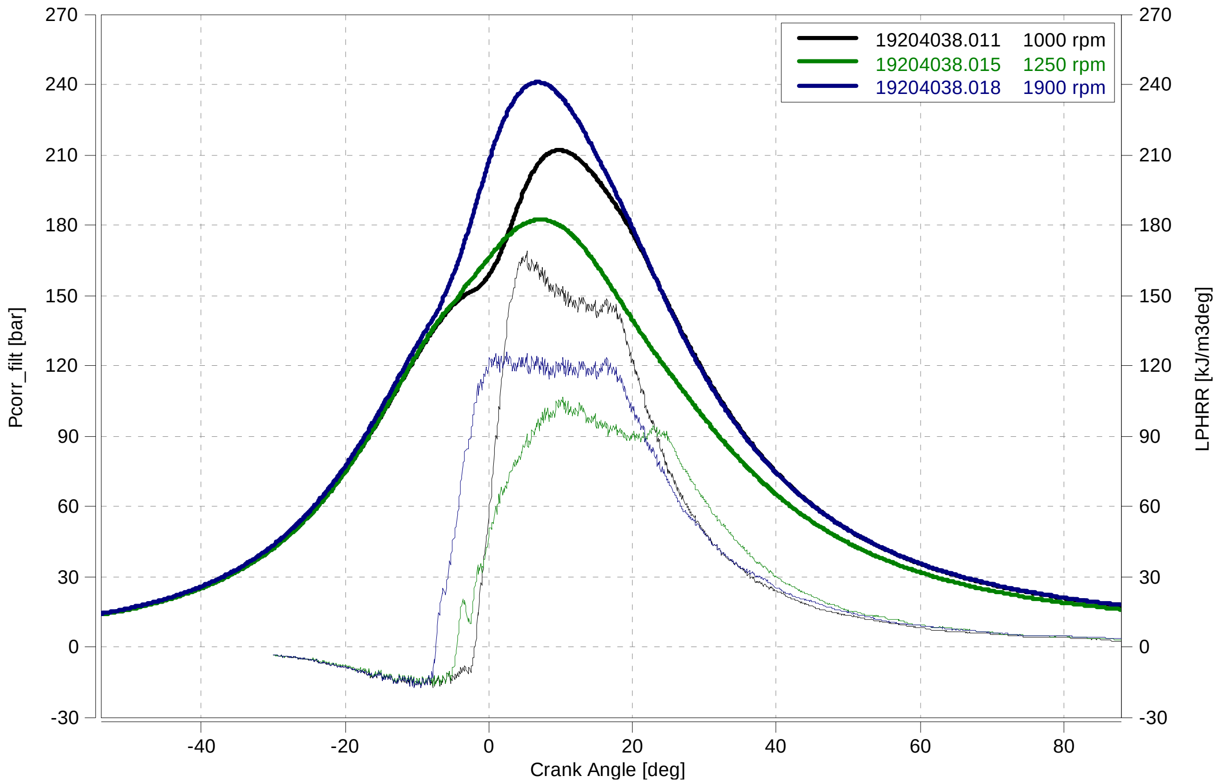

Figure 86 Turbo match points at 1250 rpm ,300 kW, without EGR std. valve profile and 23:1 in CR

The GT-Power model was also modified, different turbine and compressors were used and tested.

9.9.1. Turbo match with EGR

The GT-Power model has an EGR circuit with a control system for VGT position, EGR valve position and throttle valve position. At first when full power output was required for the three test points the throttle valve and EGR valve remained fully opened and the EGR was controlled only by the VGT position. At the first load point (full load at 1000rpm) the standard engine provided good results in GT-power where standard 17.3:1 CR and standard valve profile was used. Some problems were experienced when trying to get the right EGR quantity, with the standard turbo EGR quantity was reduced from 30 % to ca: 18 %. This was done to maintain good total engine efficiency and keep the throttle fully open. The two other load points, 1250 rpm and 1900 rpm, did not experience this problem and 30 % EGR was achieved. In Figure 87 the operation points for the standard and Miller engines can be seen in the compressor map. Lines connect the three operation points, the three points to the right in the figure are for the standard engine. The Miller engine had 32° delayed IVC and a higher compression ratio of 23:1. The engine was run with the same boundary conditions, 18 % EGR at 1000 rpm and 30 % EGR at 1250 rpm and 1900 rpm. The same control metod for EGR with VGT was used as with the standard engine model. Figure 87 clearly shows the problem with Miller cycling, the points in the compressor map are moving towards the limits. Despite this the torque output for the engine with Miller is higher than in the standard case.

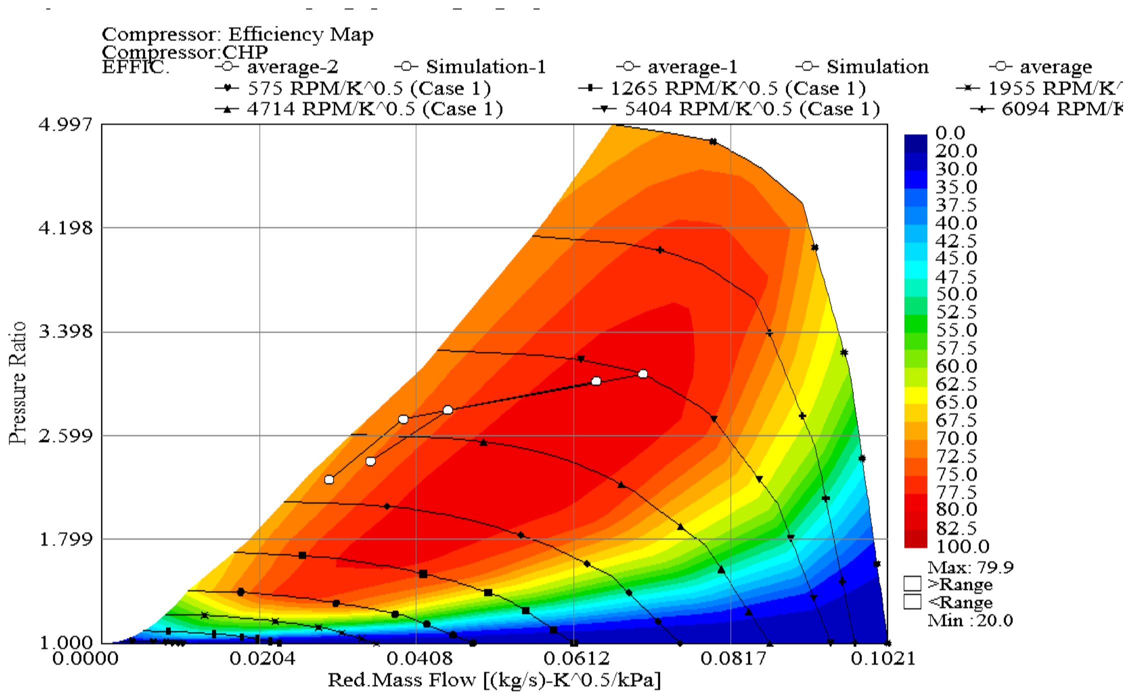

Figure 87 Standard turbo compressor with standard valve profile and 17.3:1 CR and 32° Miller 23:1 CR

When the compressor runs at a lower efficiency, lambda and pressure ratios fall. The turbine, (VGT turbine) is too large and cannot deliver enough power to the compressor, the pressure ratio drops. In Figure 88 to Figure 90 the turbine data is plotted at the operating points for the standard and the Millered engine. Different rack positions make it difficult to plot the operating points in the same turbine map. The Miller cases have a more closed VGT rack position than the standard cases. The rack position was never fully opened, only to 36 %, the turbine was too big. A comparison between the graphs shows that we have operation points at lower pressure ratios and lower exhaust mass ratios in the Miller cases, that gives a lower power output at the turbine shaft.

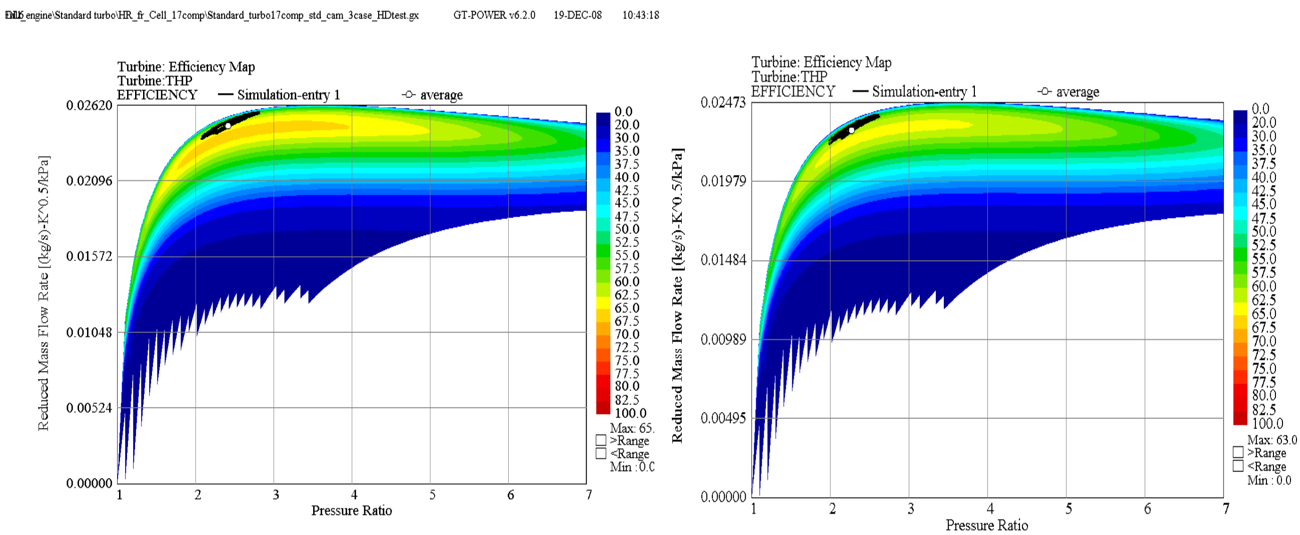

Figure 88 (left) case 1 std valve profile, 17.3:1 CR and standard turbo (right) case 1 Miller 32° valve profile, 23:1 CR and standard turbo

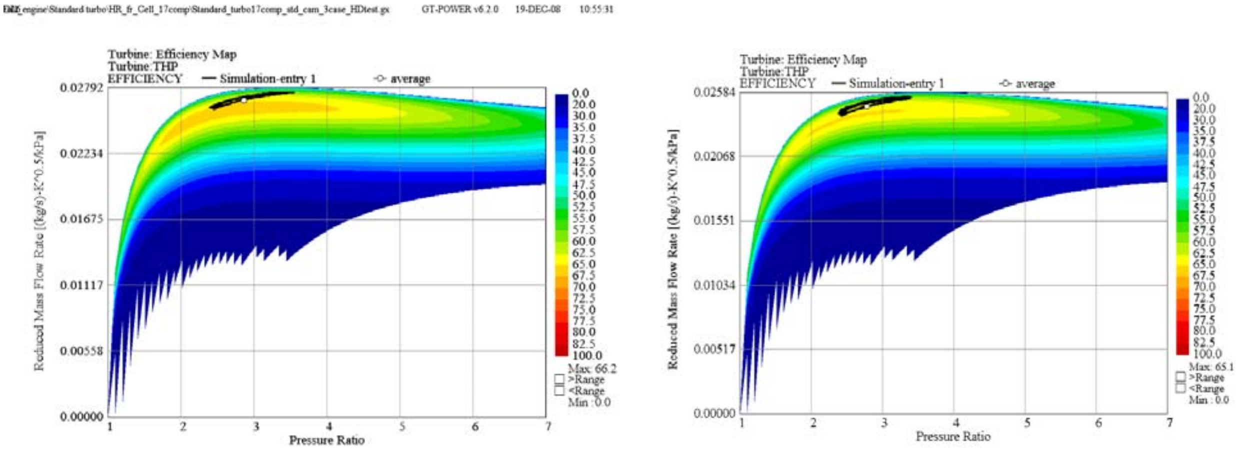

Figure 89 (left) case 2 std valve profile, 17.3:1 CR and standard turbo (right) case 2 Miller 32° valve profile, 23:1 CR and standard turbo standard turbo

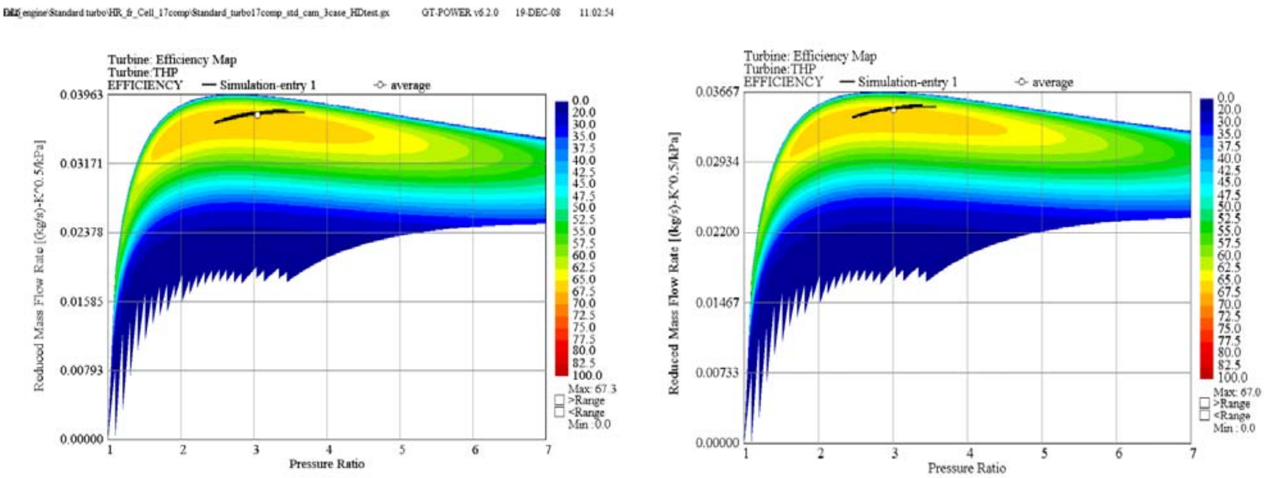

Figure 90 (left) case 3 std valve profile, 17.3:1 CR and standard turbo (right) case 3 Miller 32° valve profile, 23:1 CR and standard turbo

The aim is for the turbo system to increase the pressure ratio for the same mass flow over the compressor when Miller cycling is used. This is to compensate the loss of air in the cylinder when the inlet valve is closed earlier or later than in the standard cases.

A Matlab program was written to plot the desired operation points from the single cylinder test engine in the compressor map. When this was done different compressors could be compared to see if the operation points are located correctly in the compressor map.
In Figure 91 blue dots are the working points for Miller cycled engine with 32º later IVC. The red dots are for the standard engine at the same torque output, engine speed and lambda. The problem with 30 % EGR and standard IVC can also be seen here. The compressor does not work satisfactory at this point and in the Miller case two points are outside the compressor map.

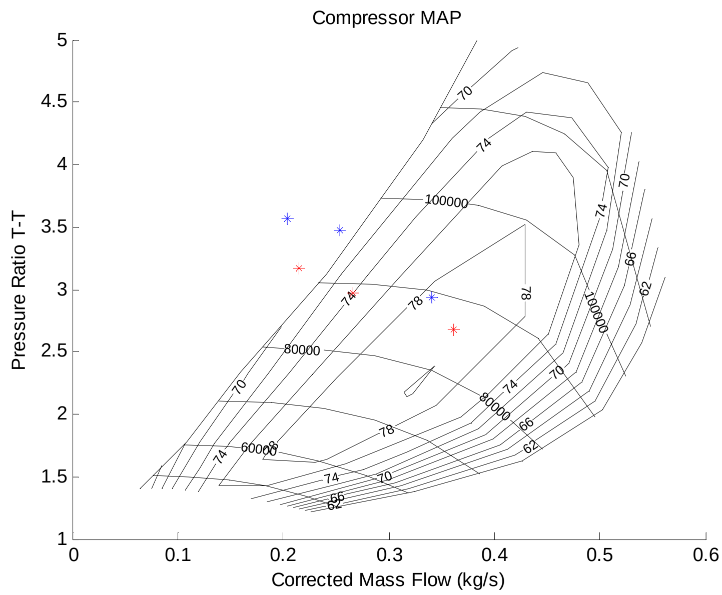

Figure 91 Compressor map with working points at 1250 rpm, 240 kW, with EGR, with and without Miller and 17.3:1 in CR

A smaller compressor was chosen that would operate at the points seen in Figure 92 that are better operation points.

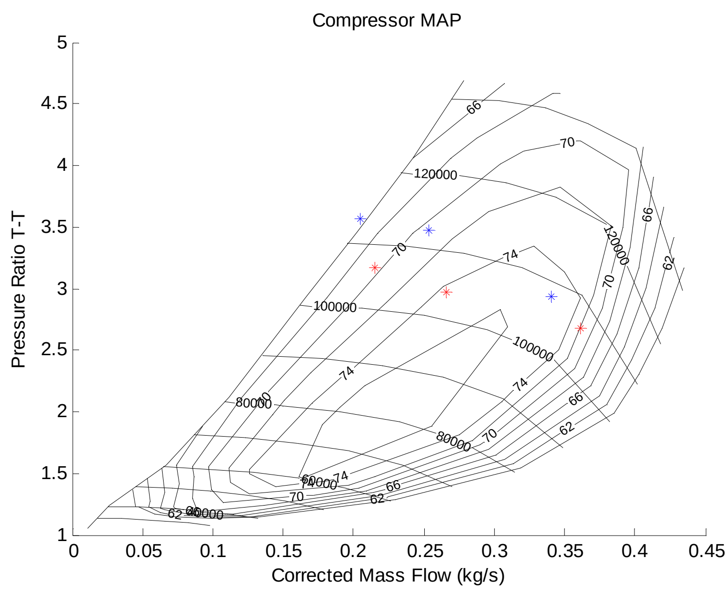

Figure 92 The new turbo match compressor at 1250 rpm, 240 kW, with EGR, with and without Miller and 17.3:1 in CR

When the new compressor was chosen a new turbine that could deliver the right power needed to be found. The required compressor power was first calculated by formula Equation 6. The shaft efficiency was included and the required turbine power output was calculated using formula Equation 7.

$$P_{Compressor} = \dot{m}_{air} \cdot T_{inlet} \cdot \left(\left(\frac{p_{comp\_out}}{p_{comp\_in}} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right) \cdot c_{p\_comp} \cdot \frac{1}{\eta_{comp}} \tag{6}$$

$$P_{Turbine} = P_{Compressor} \cdot \eta_{shaft} \tag{7}$$

c_{p_comp} = Specific heat capacity compressor
$P_{Compressor}$ = Compressor power
η_{shaft} = Shaft effiency
$\dot{m}_{air}$ = air mass flow in to the compressor
$\dot{m}_{exh}$ = exhaust mass flow in to the turbine
η_{comp} = efficiency compressor
T_{inlet} = temperature inlet compressor
$T_{inlturbine}$ = Temperature inlet turbine
p_{comp_in} = pressure before compressor
p_{comp_out} = pressure out of compressor
$p_{inlturbine}$ = pressure before turbine
$p_{outturbine}$ = pressure out of turbine
$P_{Turbine}$ = power output turbine
$M_{exhaust}$ =mol mass exhaust 29.03 kg/Kmol
R_0 = 8314 J/K*Kmol
κ = Specific heat ratio

Now the required turbine efficiency, $\eta_{turbine}$, could be calculated with formula Equation 8 were Equation 9 calculates c_p. This was then used to find a turbine that fulfilled the requirements with the specified exhaust mass flow and temperature.

$$\eta_{turbine} = \frac{P_{Turbine}}{\dot{m}_{exh} \cdot c_p \cdot T_{inlturbine} \left(1 - \left(\frac{p_{outturbine}}{p_{inlturbine}} \right)^{\frac{\kappa-1}{\kappa}} \right)} \tag{8}$$

$$c_p = 762 + \frac{T_{inlturbine}}{1000}(900 - 702) + \frac{R_0}{M_{exhaust}} \tag{9}$$

The turbo configurations that satisfied the conditions and which had been tested in the GT-power model had the following matches:

With EGR
Compressor match 1
Compressor match 2
Turbine 1

Without EGR
Standard compressor DLC6
Standard turbine DLC6

The new turbo match, compressor 1 and turbine 1, were tested under the same conditions as before in the Miller case and 23:1 in CR. The operation points in the compressor map can be seen in Figure 93. From the operation point on the right the compressor appears to be a little too small. This point is 1900 rpm and is a little too far from the optimal operation point. The pressure ratio is lower at this point compared to the standard turbo and standard IVC. The standard IVC had a high lambda, lower lambda is not a problem at this point.

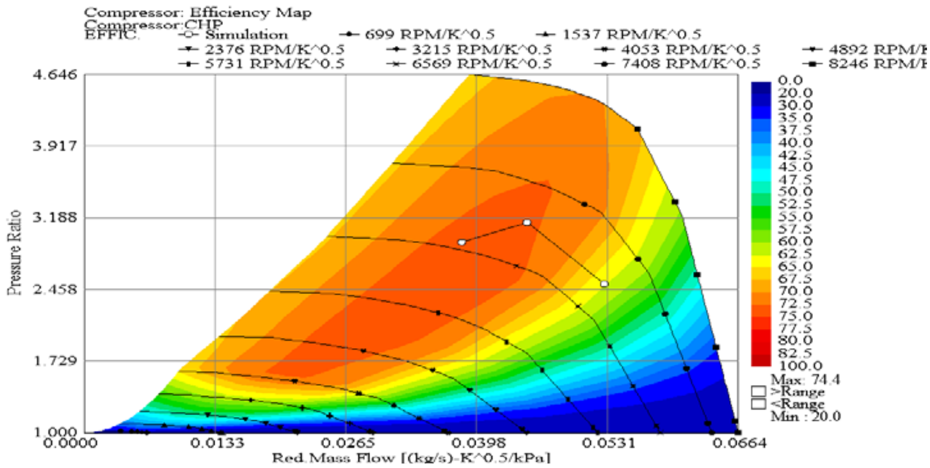

Figure 93 turbo match 1, Miller 32° with 23:1 in CR

The smaller turbine that can be seen in Figure 94 to Figure 96 works at better points in the turbine map. The rack positions are more open and in last case the rack position is nearly completely open. The EGR valve was now used to reduce the EGR quantities if required. In the last case a bigger turbine would have been desirable, but engines are rarely used at this point for longer periods. The drop in efficiency is outweighed by the higher efficiency levels at lower engine speed.

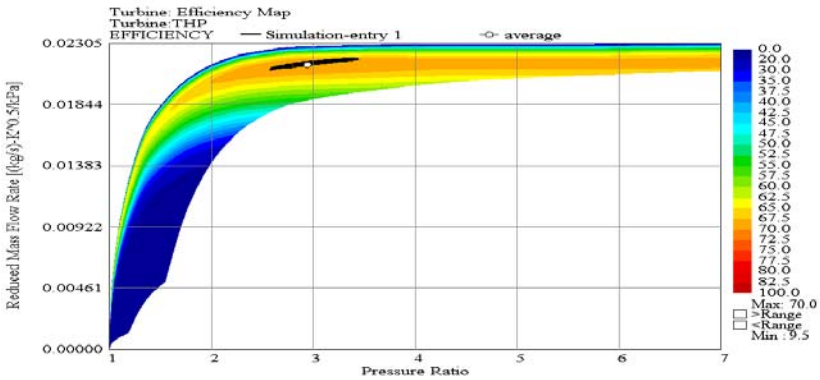

Figure 94 case 1 1000 rpm turbo match 1 with smaller compressor and turbine, Miller 32° with 23:1 in CR

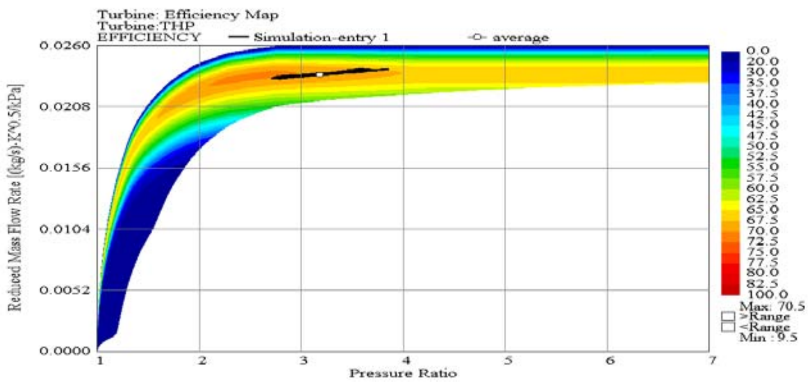

Figure 95 case 2 1250 rpm, turbo match 1 with smaller compressor and turbine, Miller 32° with 23:1 in CR

The VGT rack is nearly fully open in Figure 96.

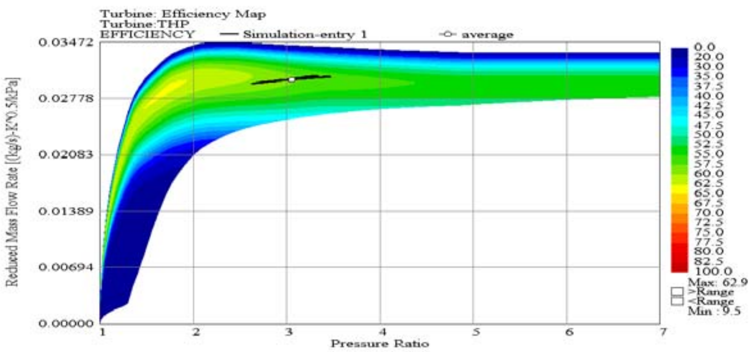

Figure 96 case 3 1900 rpm turbo match 1 with smaller compressor and turbine, Miller 32° with 23:1 in CR

Compressor 2 was now tested with the same turbine. The compressor worked in a better area, see the compressor plot Figure 97, compared to the smaller compressor 1.

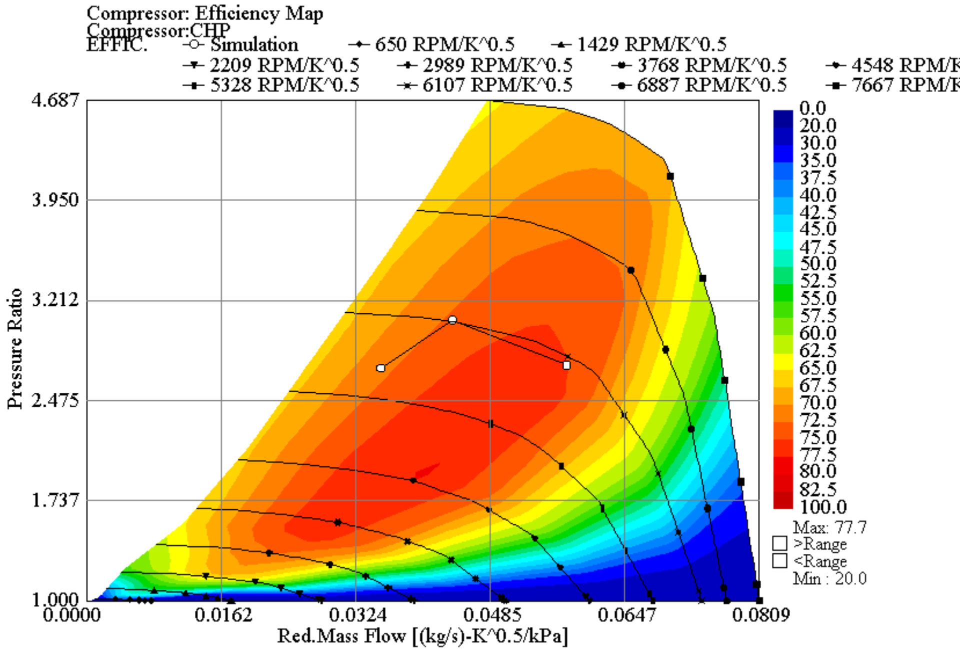

Figure 97 turbo match 2 with a bigger compressor than match 1 and turbine, 32°LIVC and 23:1 in CR

In Figure 98 the brake torques are plotted for the three turbo configurations that have been tested. The tests were performed with constant delta quatity. The highest torque output was found with turbo match 1 at 1000 rpm, turbo match 2 at 1250 rpm and standard turbo at 1900 rpm. The torques shown are for a 360 hp engine, the torque output for the Miller case has a power output of 390 hp. A maximum power output with fixed Miller profile and single turbo configurations is thereby limited to 400 hp otherwise the pressure ratios are too high for a single turbo and the outlet temperature from the compressor would be too high for a standard aluminium compressor wheel. The standard engine is also plotted in Figure 98, all the tested turbo matches including the standard turbo with Miller have a higher torque output for all the tested load points.

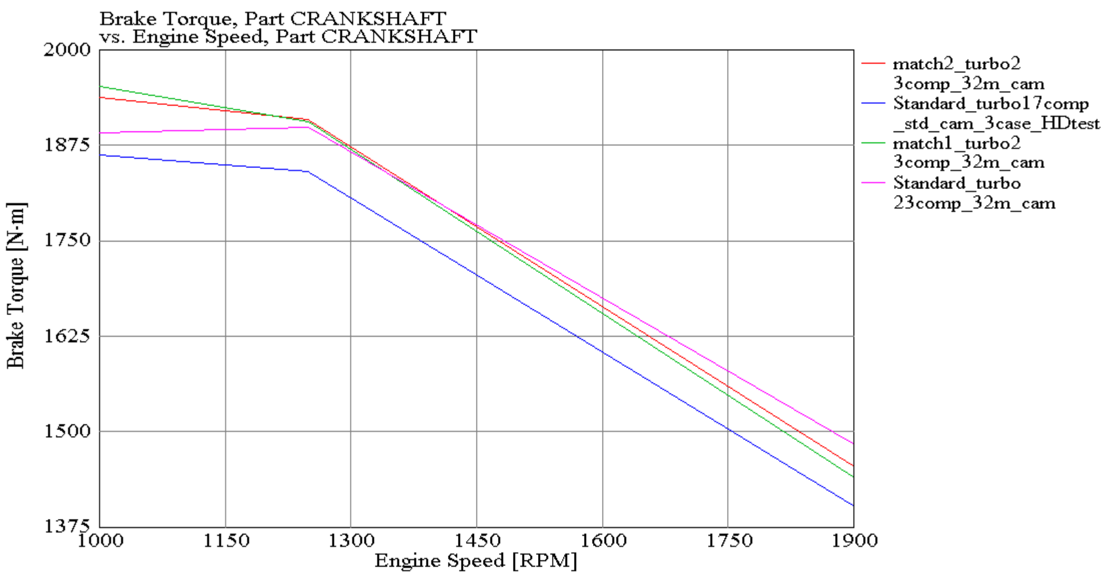

Figure 98 Power output for the different turbo matches at same delta quantity

In Table 21 the Miller engine with a standard turbo has less turbine power than the standard engine with the same turbo. The yellow cells in the previously mentioned table represent the highest air-fuel ratio and turbine shaft power for the Miller cycle tests. When a smaller turbine was used more power was delivered to the compressor wheel. The Air-Fuel ratio is higher for the Miller cases with the smaller turbo than the standard engine, but not for the 1900 rpm point. The EGR quantity can be increased at 1000 rpm with turbo match 1 and 2 if desired, that is not possible in the cases with standard turbo.

The rack position gives us a hint how large the turbine is. If the rack position is nearly closed at high power outputs, like in the standard cases, the turbine is probably too big. A turbine which is too large means the compressor takes longer for the boost pressure build up and torque will be compromised under transient engine running (for example acceleration). The amount EGR can be increased in turbo match one and two for the first case at 1000 rpm but not for the standard turbo cases. The rack position is nearly fully closed at 19 % EGR. To have the opportunity to compare the cases with each other, the EGR was helt constant.

Table 21 Data for different turbo cases

			17.3:1 standard turbo	
Speed	Rpm	1000	1250	1900
Percent EGR	%	18.88	30.06	30.63
Air-Fuel Ratio		19.80	20.47	25.38
Turbine				
Average Power	kW	23.62	35.57	58.09
Rack Position		0.17	0.20	0.42
			Miller standard turbo	
Speed	Rpm	1000	1250	1900
Percent EGR	%	17.82	29.81	30.65
Air-Fuel Ratio		17.06	18.26	23.53
Turbine				
Average Power	kW	18.82	30.76	52.27
Rack Position		0.15	0.16	0.36
			Miller match1 turbo	
Speed	Rpm	1000	1250	1900
Percent EGR	%	18.69	30.29	30.14
Air-Fuel Ratio		22.12	20.79	19.46
Turbine				
Average Power	kW	32.86	41.72	43.69
Rack Position		0.34	0.42	0.93
			Miller match2 turbo	
Speed	Rpm	1000	1250	1900
Percent EGR	%	18.37	30.07	30.18
Air-Fuel Ratio		20.49	20.39	21.26
Turbine				
Average Power	kW	28.29	39.61	44.21
Rack Position		0.35	0.41	0.97

9.9.2. Higher Miller levels

Higher levels of Miller were examined to see how the turbo system reacted. First turbo match 1 with the smaller compressor was tested. In Figure 99 the operation points for 32º, 42º, 52º and 62º later IVC was tested. The arrow in the diagram shows how the operation points move away from optimal The same EGR quantity was used as before. The operation points moved with the higher pressure ratio compared to the standard turbo where the air mass flow was only reduced when Miller was used. Some drop in air mass flow however was also experienced here, as in the standard case. The turbine was better suited to the engine. The last operation point at 1900 rpm now moved in the right direction to work with better efficiency.

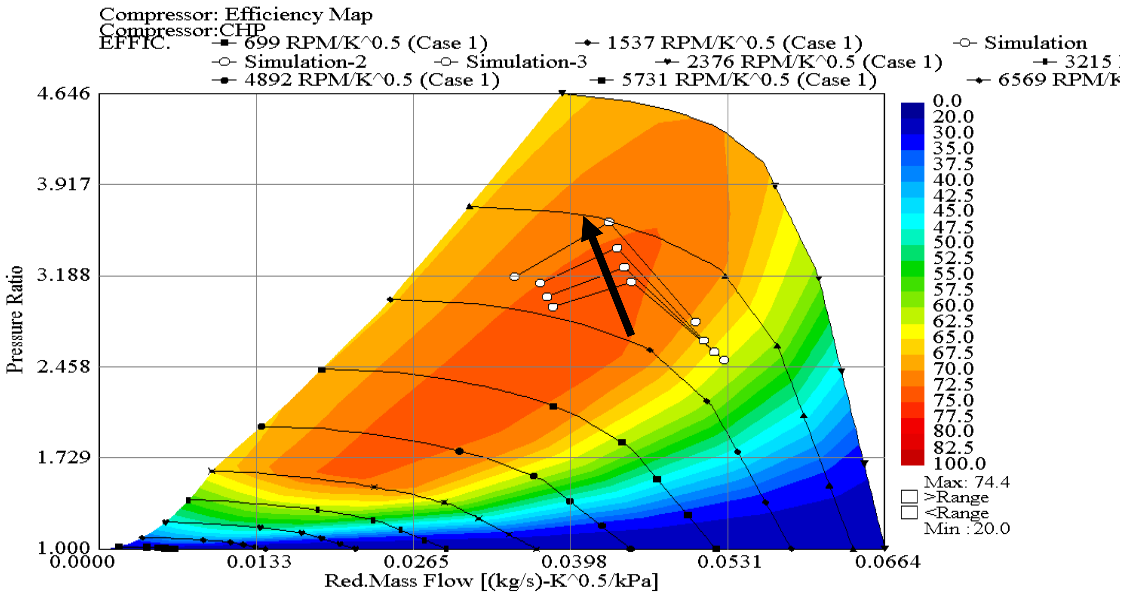

Figure 99 turbo match 1 Miller with 23:1 in CR

In Figure 100 the torque outputs at the different Miller levels are plotted. In the lower engine speed cases the 32° LIVC is the best choice and at higher engine speed a later IVC gives benefits.

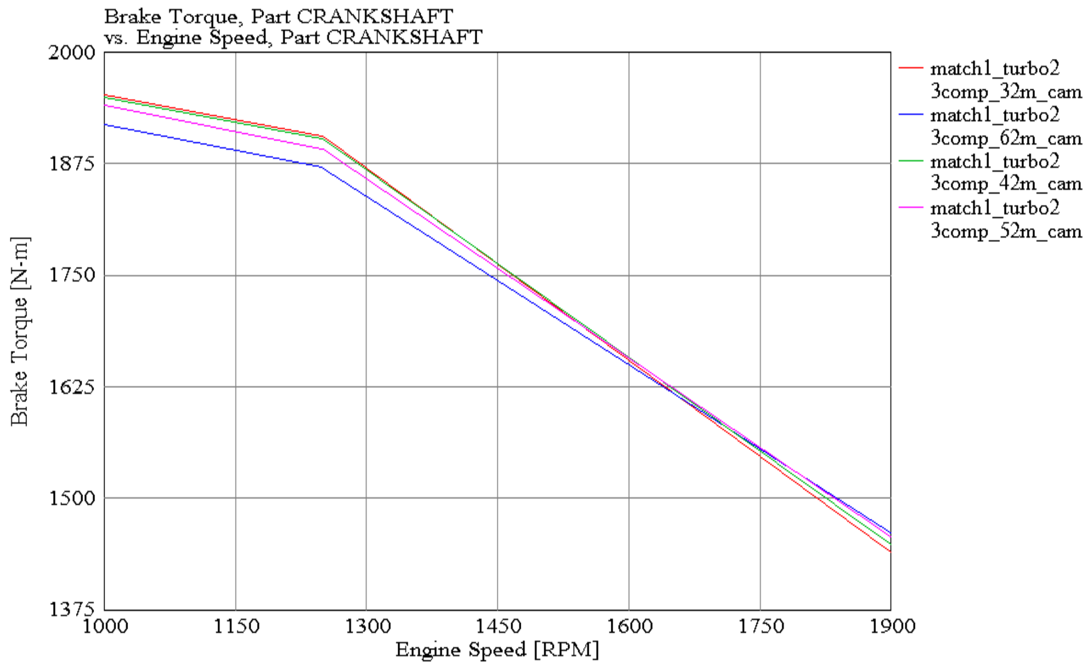

Figure 100 Torque output turbo match 1

The same Miller test was carried out with the turbo match 2. The same behaviour as previously can be observed in Figure 101. The 1900 rpm operating points are on a good compressor efficiency line, but at high Miller levels the lower operation points start to go near the limits. If the EGR quantity needed to be increased this could be a problem.

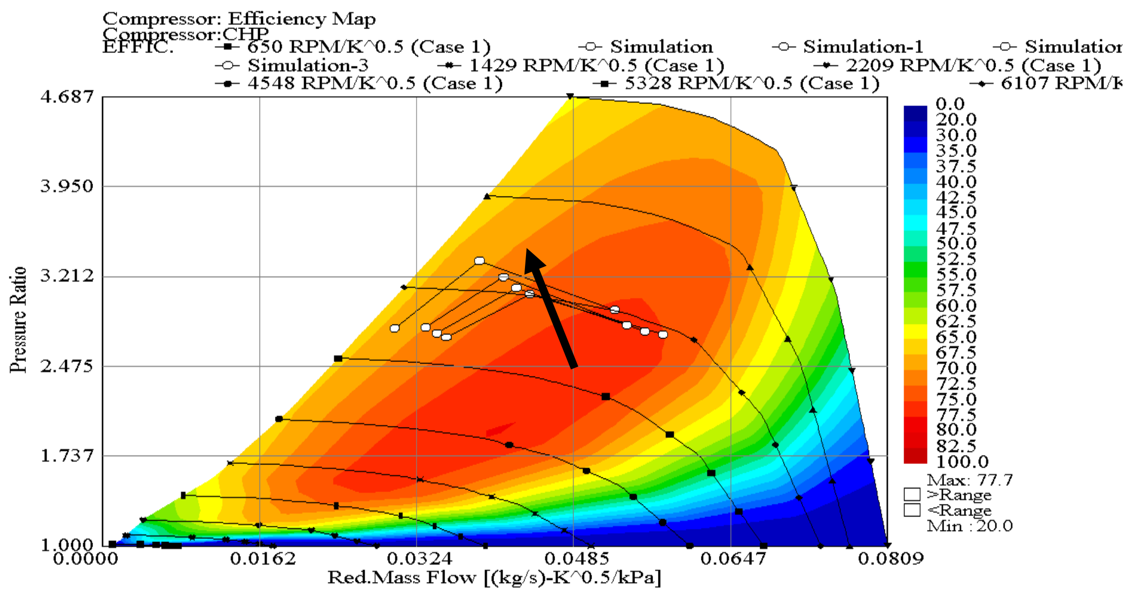

Figure 101 turbo match 2, Miller with 23:1 in CR

In the simulations the maximal temperature out of the compressor 199°C was seen with 62° later IVC. The boundary temperature was set to 200°C out of the compressor to prevent compressor wheel damage. One way to raise this limit would be to change the material of the compressor wheel. The torque outputs for the different Miller levels can be seen in Figure 102. Higher Miller levels are good at high engine speeds, at low speeds the lower Miller level are better.

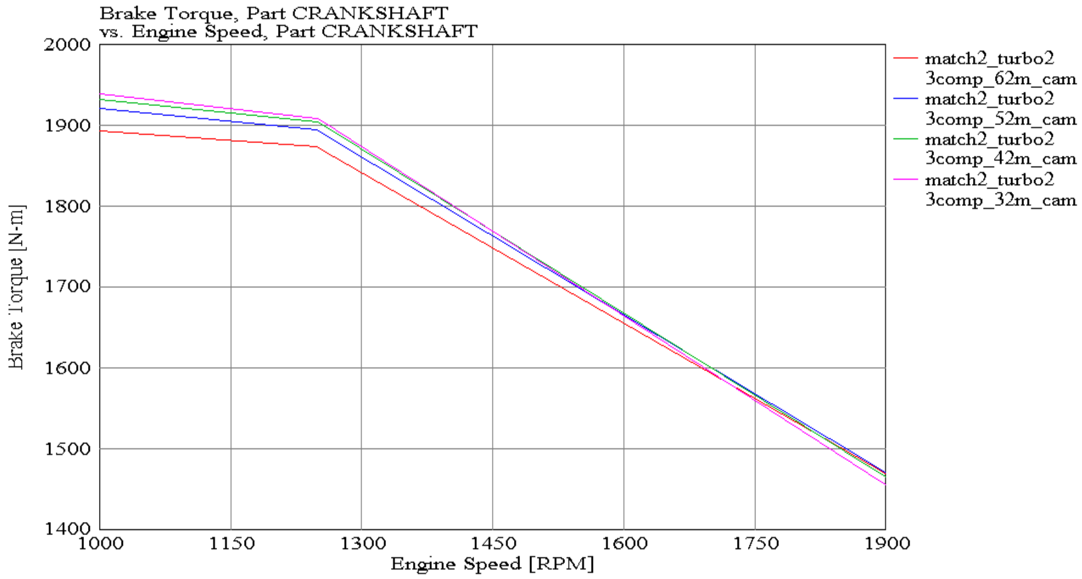

Figure 102 Brake torque at different Miller levels with turbo match 2

9.9.3. Turbo match without EGR

In the non EGR case the power output of the engine can be increased. This is because the turbo pressure does not need to be as high as the EGR case. The single turbo configuration is suitable for more of the higher power output engines in the range. As in the EGR case combustion data was recorded for the single cylinder test engine and used in GT-power. The test point was also plotted in Matlab and the compressor plot, see Figure 103. Blue dots are the working points for Miller cycled engine with 32º later IVC. The red dots are for the standard engine at the same torque output, engine speed and lambda. In this case the standard turbo appears to be a good choice as all the operation points are within acceptable limits.

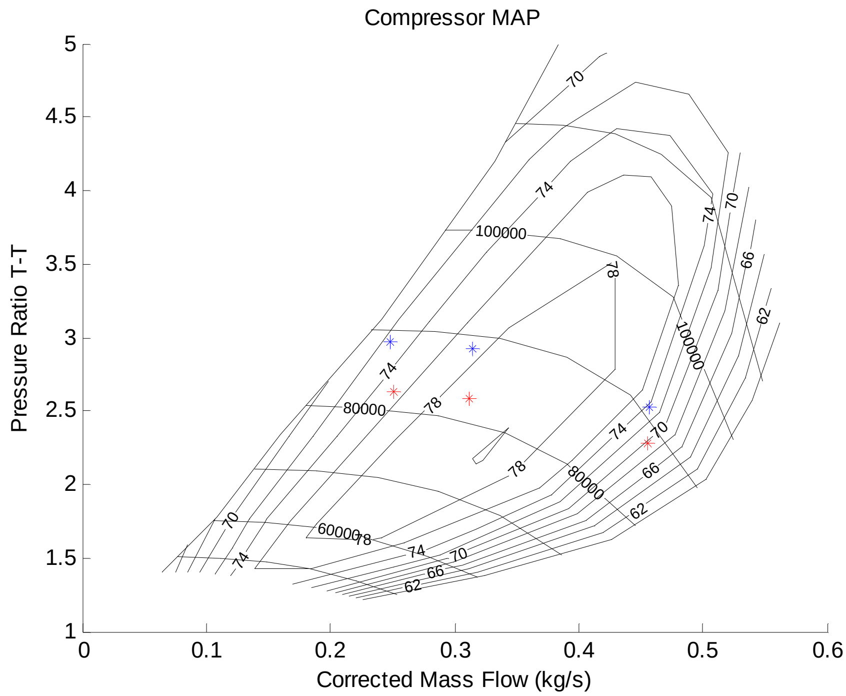

Figure 103 compressor map for working points for 300kW without EGR with and without Miller

The GT-power model was now run to compare two set-ups, one with the standard turbo, IVC and 17,3:1 in CR, the other 32° later IVC with 23:1 in CR, see Figure 104. Some differences arose between the Matlab and GT-power plots. The explanation is because the turbine is not calculated in Matlab.
The two low speed points move at higher pressure ratios and in the 1250 rpm case the mass flow rate is even higher.

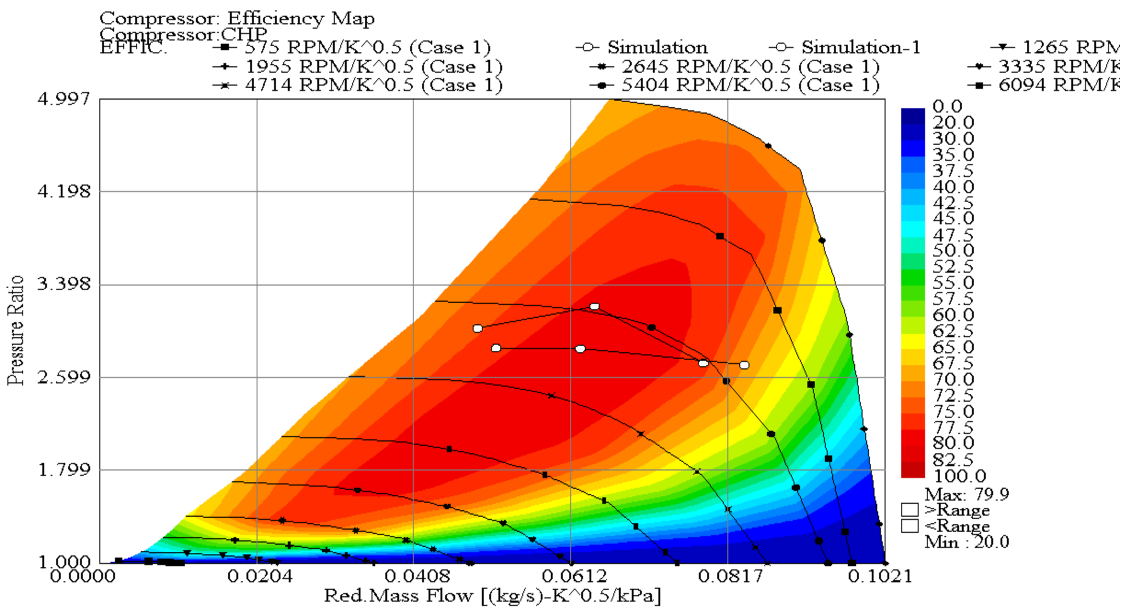

Figure 104 std. turbo, Miller 23:1 and 17.3:1 in CR

The torque output is higher for all the load points in the Miller case, see Figure 105. Investigations with higher Miller levels like the one in the EGR case were not carried out since the results were similar.

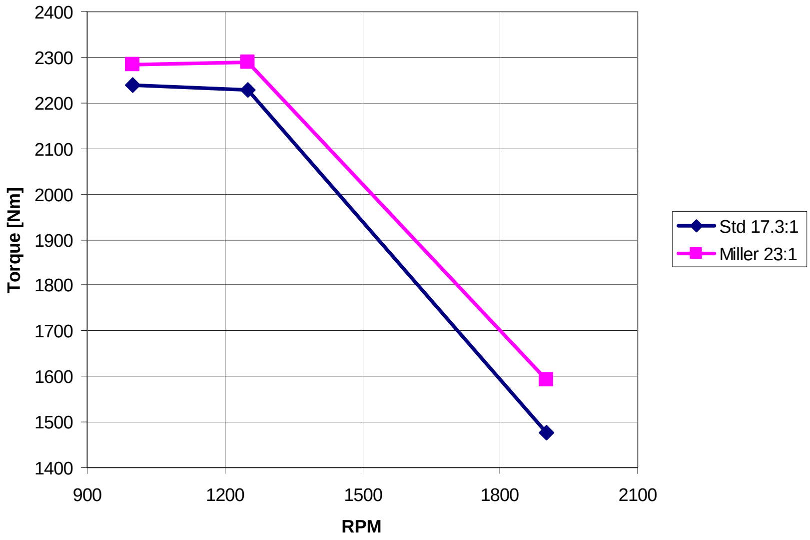

Figure 105 Torque output for standard engine and Miller engine

11. Discussion and summary

To summarize the work it can be said that the idea of increasing engine efficiency with the use of a Miller cycle together with higher compression ratio is a good one giving additional benefits in reduced NO_x emissions for all load points and a higher exhaust temperature when the engine is run at low load points. This statement is based upon tests carried out on a single cylinder test engine which mirrors a full production engine together with computer simulation of a full engine in GT-power.

The Miller cycle places larger demands on the turbo system. An engine with full power output can not be built with single turbo. It demands an two stage turbo configuration with two inter cooling stages (one inter cooler after every compressor). A type DLC6 engine with EGR is estimated to have maximum 400 hp output with single turbo configuration and retain the same torque/power quota as the standard engine.

GT-power showed itself to be an extremely important development tool with which it was possible to save a large amount of time and money by avoiding being dependent of the test cell. There are several parts of the simulations which could have been improved if there had been more time. The first is that the GT-power model of the single cylinder test engine did not have a fuel pump included. Even if a model was available for a fuel pump and common rail it would still have to be modified to resemble the plugged version that was fitted to the test engine (plugged because only 1 of 5 or 6 injectors were connected). Had a pump been included it would have been easier to compare the brake efficiencies from the model and the real engine. This can however be included in the friction model, but there was no time to investigate the pumps power demand.

The emission measurement equipment was changed half way through the test series as well. This interruption in testing was not really optimal but this is reality. Despite these small details the test results were most satisfactory. It was interesting to see that trends in the GT-power model held for the research engine which meant that the results of the full engine simulations could be assumed to be valid.

It was seen that with lower loads and no EGR, the exhaust temperatures were raised by nearly 100 °C which would help considerably with the light off of a catalytic converter. Continuing to analyse the data for the non EGR cases it was observed that the increase in brake efficiency was highest for the higher load points at 1250 rpm. But the greatest increase was found at a higher engine speed, 1600 rpm. This engine speed was only tested at low load, 70 kW for a full engine, with an increase in brake efficiency of 3.30 % was measured. The increase is from standard CR with standard IVC and the higher CR and the best Miller point.

The VVT system used to independently control each valve also demands power. This system was driven externally, thus the gains in having the freedom to control the valve lift must be offset by the cost in power to drive the system. If a camshaft which had a longer opening duration could be used the average absolute increase in brake efficiency of the tested load points without EGR was calculated to 1.99 %. Using a complete VVT gave a somewhat higher average gain of 2.27 % for the same load points.

When EGR was introduced the trends in increase in br. eff. were not so obvious. EGR use demands a higher boost pressure which have an effect on br.eff. The test engine was not fitted with a turbo system, the inlet pressure was created externally instead. The full engine simulations on the other hand included turbo charging and after calibration with measured data through the single cylinder test they showed promising results.

The test engine when run with EGR had a slightly smaller increase in performance compared to the non EGR case. The average total increase for a fixed profile camshaft and a full VVT was 1.95 and 2.12 % respectively.

Despite the good results shown by the study there still remains work to be done. The hardware (piston, injector e.t.c.) used in the 23:1 compression tests were not optimized for smoke emissions, further development is recommended. Turbo matching was carried out for the 13 ESC stationary load points. If a smaller turbo is fitted it would be a suitable match. This smaller turbo will give benefits in boost pressure build up, but the engine torque without boost pressure will decrease when the swept volume is restricted. A smaller amount of air is retained in the cylinder. This means that a fast boost pressure build up is more important than earlier, a turbo transient investigation should be done to reduce smoke emissions under engine torque build up.

The Miller cycle strategy together with a higher compression ratio and a new turbo match can be a way to offer more fuel efficient low power engines.

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13. Appendix 1 - ESC cycle

ESC Cycle, European stationary cycle
The ESC test cycle (also known as ACEA cycle) has been introduced, together with the ETC (European Transient Cycle) for emission certification of heavy-duty diesel engines in Europe. The ESC is a 13-mode, steady-state procedure with load points that are chosen on the basis of the engine performance like in the Figure 106. In Figure 107 the torque and power output that is the base for calculations of load points for an DLC6 360 hp engine. Emissions are then measured in engine test cell at the specified load points. The measured data are then weighted according to Table 22 and an average emission output from the engine is calculated.

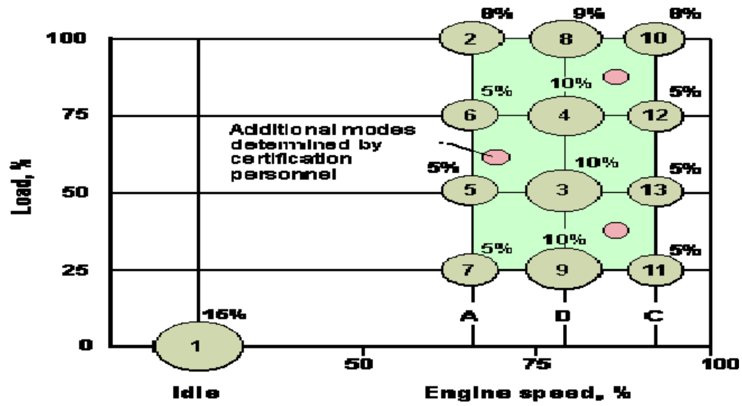

Figure 106 The ESC cycle load points

Table 22 ESC cycle load points with weight factors

ESC-cycle

Mode	Engine Speed	% Load	Weight factor, %	Duration
1	Low idle	0	15	4 minutes
2	A	100	8	2 minutes
3	B	50	10	2 minutes
4	B	75	10	2 minutes
5	A	50	5	2 minutes
6	A	75	5	2 minutes
7	A	25	5	2 minutes
8	B	100	9	2 minutes
9	B	25	10	2 minutes
10	C	100	8	2 minutes
11	C	25	5	2 minutes
12	C	75	5	2 minutes
13	C	50	5	2 minutes

The engine speeds are defined as follows:

1. The high engine speed (nhi) is determined by calculating 70% of the declared maximum net power. The highest engine speed where this power value occurs (i.e. above the rated speed) on the power curve is defined as nhi.
2. The low speed (nlo) is determined by calculating 50% of the declared maximum net power. The lowest engine speed where this power value occurs (i.e. below the rated speed) on the power curve is defined as nlo.
3. The engine speeds A, B, and C to be used during the test are then calculated from the following formulas:
   A = nlo + 0.25(nhi - nlo)
   B = nlo + 0.50(nhi - nlo)
   C = nlo + 0.75(nhi - nlo)
   The ESC test is characterized by high average load factors and very high exhaust gas temperatures.

In Figure 107 the torque and power output for the investigated DLC6 360hp engine can be seen. Max torque output 1850 Nm, max power output 264 kW at 1900 rpm.

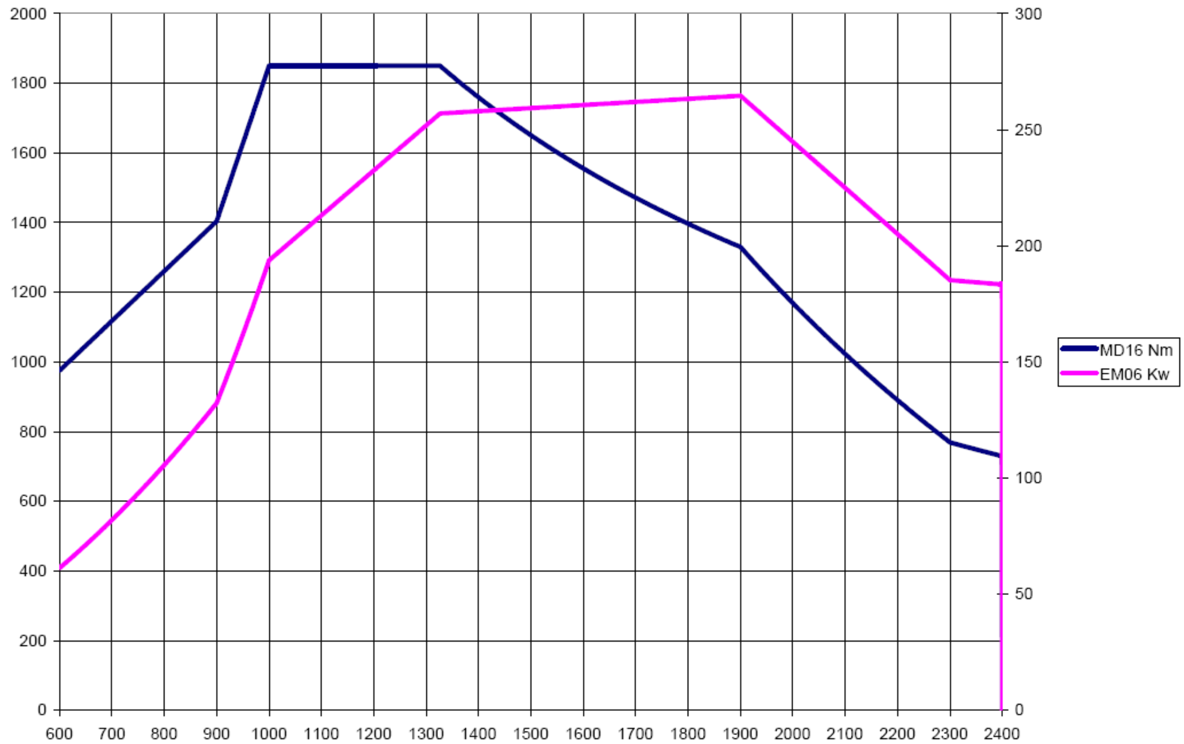

Figure 107 Torque and power output fr DLC6 360 HP