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The potential of increased efficiency and power for a turbocharged PFI-SI engine through variable valve actuation and DEP

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Thesis for the degree of Master of Science in Engineering
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The potential of increased efficiency and power for a turbocharged PFI-SI engine through variable valve actuation and DEP

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Abstract

New legislation involving emissions from internal combustion engines are pushing the manufacturers to develop new technology faster than ever before with the amount of greenhouse gases. To meet the standards new concepts need to be developed with lower fuel consumption and emissions. This thesis covers the implementation of a couple of methods to achieve this. These concepts are DEP with fully variable valves in a port fuel SI engine with high compression ratio (CR). The results show an increase in efficiency followed by lowered fuel consumption. The improvements in fuel consumption are mainly found to be the result of raising the CR and because of decreases in pumping losses due to de-throttling via the Miller-cycle. The reduction in pumping losses by implementing the DEP concept was not as great as expected. The results show a decrease of fuel consumption of 9.5% at part load and 5 % at high load. The main improvement with the DEP concept was the reduction of the in-cylinder residual gases at 40 CAD before top dead centre firing (TDCF). This could be enough to be able to use such high CR that otherwise just wouldn´t be possible.

The thesis reveals many of the difficulties involving combustion simulation and with no experimental work available in particular. The thesis would gain a lot from implementing a predicted combustion model to simulate EGR and the full capability of the DEP concept in terms of affecting the combustion, by for example changing the burn rate.

Nomenclature

AFR	Air to Fuel Ratio
BFSC	Brake specific fuel consumption
BMEP	Brake Mean Effective Pressure
CAD	Crank Angle Degrees
CR	Compression Ratio
DEP	Divided Exhaust Period
ECR	Effective Compression Ratio
EGR	Exhaust Gas Recirculation
EIVC	Early Intake Valve Closing
ExDEP	Externally Divided Exhaust Period
GTP	GT-Power
IC	Internal Combustion
IVC	Intake Valve Closing
LIVC	Late Intake Valve Closing
LIVO	Late Intake Valve Opening
NTC	Negative Temperature Coefficient
PMEP	Pumping Mean Effective Pressure
PPC	Partially Premixed Combustion
PR	Pressure Ratio
RGF	Residual Gas Fractions
ROHR	Rate Of Heat Release
TDC	Top Dead Centre
TDCF	Top Dead Centre Firing
TDSI	Turbocharge Direct Injected Spark Ignition
TPSI	Turbocharge Port Injected Spark Ignition
VTEC	Variable Valve Timing and Lift Electronic Control
VVT	Variable Valve Timing

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1 Introduction

The demands on engine efficiency and emissions is increasing fast and fully electrical cars still have a long way to go regarding battery capacity and charging time. This means the development of new fuel efficient solutions for internal combustion engines is more important than ever. Various hybrid concepts such as the Toyota Prius is able to reduce the fuel consumption but the trade-off is expensive electrical converters and in the consumers point of view a large decrease in resale values when batteries are worn out is a problem. Even though Tesla motors claims a range of 270 miles on one charge [1], approximately 432km, the possible reduction in carbon dioxide emissions highly depends on the electric mix in the country the car is charged. For the Nordic countries which have mostly hydro-, nuclear- and other CO_2 neutral energy production, electric cars is a good alternative. But for example the U.S who has mostly coal and natural gas based energy the extra energy conversion to electricity rather than directly from gas to work in a combustion engine will reduce or in some cases even worsen the overall efficiency.

The Otto engine was invented in 1876 and ever since there have been endless attempts on making the engine more efficient and powerful[2]. Big breakthroughs have been made in exhaust energy recovery, by the means of adding a turbocharger, and advanced engine control. There are still many more improvements to be made, both in terms of advanced combustion methods such as partially premixed combustion, PPC, and further downsizing. This master thesis will study a couple of them. For instance increasing the compression ratio (CR) for faster more efficient combustion, decreasing the pumping losses by implementing throttle-free load control via fully variable valve train and by implementing the relatively new concept called divided exhaust period, DEP.

The main focus in this master thesis is on investigating the potential of using DEP with fully variable valve timings to reduce BSFC at part load and knock issues for high loads in a 1.6l four cylinder port fuel injected spark ignited engine. Since no experimental engine is available the study will be conducted using 1D engine simulations in Gamma Technologies program GT-Power. Not only will DEP and valve timings be studied, but also the effect of intake and exhaust length via acoustic tuning. The latter is thought to help reducing knock. The engine model in GT-Power is based on a benchmark engine with potential for the typical downsizing feature of high boosting through turbocharging. The throttle-less operation will be achieved by applying the Miller-cycle i.e. early intake valve closing.

2 Variable valve trains

In a conventional engine the intake and exhaust valves are controlled by a cam mechanism which limits the ability to control valve lift, valve overlap etc. This in turn restricts what you are able to achieve with the combustion. To have full control of the combustion it is beneficial to be able to fully control the valve events.

2.1 Types of variable valve trains

The last couple of years downsizing and turbocharging has been the major part of achieving lower fuel consumption without torque/power reduction and thereby compromising drivability. In addition to this variable valve timing (VVT) and lift have the opportunity to reduce the pumping losses and increase volumetric efficiency and hence improve the brake specific fuel consumption (BSFC). Semi mechanical VVT-systems have been on the market for a while, e.g. Hondas VTEC system launched 1989[3] which is based on dual cam lobes with different lift and timing and a hydraulic shifter between the low speed and high speed lobe. Similar systems have been evaluated, such as in SAE-paper 2014-01-1699[4] were dual cam lobes and dual fixed rocker arms are used. The change between the two cam profiles is done by fixating the two rockers to each other via a sprint mechanism. The high speed profile is designed in such way that it is dominant to the low speed profile regarding lift. The rpm for changing from low to high speed cam profile can be changed. A system which gives a higher freedom of cam strategies is the Multiair system developed by Fiat. This system is based on a fixed camshaft solution with a gap filled with hydraulic oil above the valve. The hydraulic oil is regulated via a solenoid valve, as the solenoid valve is open the oil is pressed into an accumulator and thereby not transferring any force from the cam lobe to the valve. As the solenoid is closed the oil, which is incompressible, acts on the valve according to the fixed cam profile causing the valve to lift. If the valve is closed earlier than the cam profile the solenoid opens and releases the hydraulic oil, in order to slow down the valve as it approaches the valve seat a hydraulic break is used[5]. By slowing down the valve, noise is avoided as it settles towards the valve seat. To get a complete freedom of design a valve actuation system without mechanically fixed profiles is needed. For this electronic-, hydraulic- or pneumatic actuation can be used. With these type of systems independent valve lifts for each cylinder is possible e.g. in order to introduce swirl at low loads, cylinder deactivation and square shaped lift profiles. One system on the market that has these features is FreeValves system [6].

2.2 Throttle-free load control

Another benefit with having completely controllable valves is that you can eliminate the need of a throttle valve. The throttle valve increases the pumping losses in the engine and is thereby lowering the efficiency. Studies have been performed on this matter where one of them was made by Kaiserslautern Technical University, Germany [7]. Experiments were performed on a Turbocharged Port Injected Spark Ignited (TPISI) engine as well as on a Turbocharged Direct Injected Spark Ignited engine (TDISI). The purpose of the study was to examine the differences between the TDISI-engine and the port fuel injected one and also to see which is most suitable for the throttle free load application.

The engines were equipped with continuously variable inlet valves, which can vary the valve lift and duration continuously. However the lift and the duration cannot be varied independently in this case. This leads to very small and short inlet opening at low loads, which prevents backflow of the cylinder charge but can also cause the turbulence to be lowered. On full load, however, the prevented backflow of the cylinder charge is said to yield 20% higher torque rating for the engine [7].

The direct injected engine equipped with the variable intake valves achieves the highest torque curve in comparison with the standard equipped engine. These torque increases was achieved in the engine speeds of 1200-1500 rpm, which is an important engine speed range. The torque is about 10% higher than in the GPI-engine with the otherwise same valve setup. The TPISI-engine with variable intake valves produces higher torque than the TPISI-engine with standard inlet valves although the difference is less noticeable.

At low engine speeds the turbocharged engine is comparable to a naturally aspirated engine, due to the low turbocharger speed, this is where the shorter intake valve closing time achieve better cylinder filling and thus sends more mass flow towards the turbine and raises the turbo-charger speed.

The system also enables the use of wider valve spreads which would not be possible in a throttled engine due to the increase in residual gases in the cylinder. If the engine is throttle free the residual gas content can be controlled almost exclusively by varying the exhaust spread, which could easily be done with a fully variable valve system. Figure 1 explains the different types of valve spreads in an IC-engine.

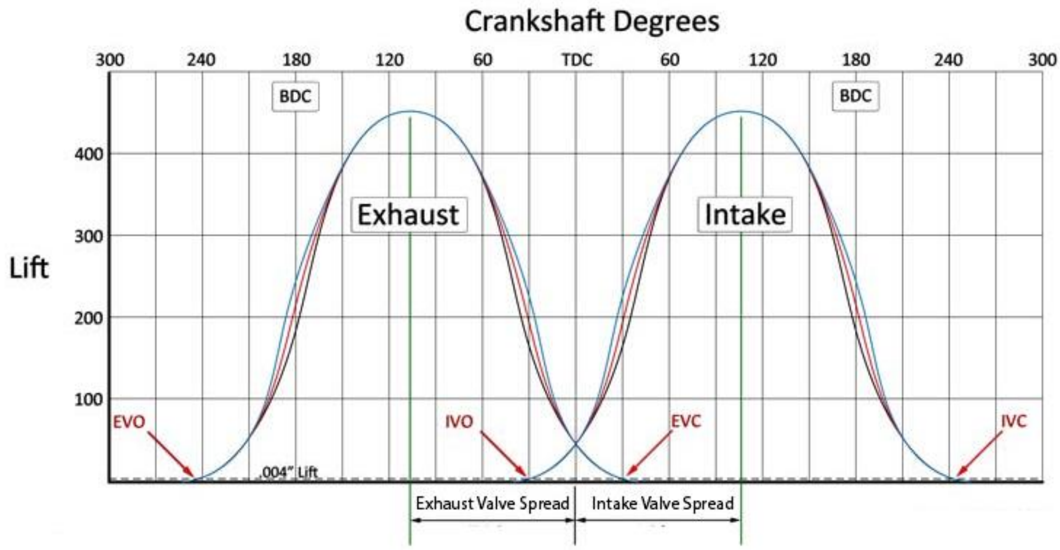

Figure 1. Exhaust spread explained

The particular study shows a reduction in pump losses of about 5 % [7]. Again this could probably be done even more efficiently with the addition of a fully variable exhaust valve. The lower pumping loss leads to a reduction of about 1.2 % in fuel consumption.

The improvement in fuel consumption due to less pumping losses is said to be more noticeable in the TPISI-engine than in the TDISI engine. The TDISI-engine show an improvement of 12 % in fuel consumption compared to the TDISI which only showed a 9% reduction. This is said to be caused by the use of a narrower spread between intake and exhaust which decreases the pumping losses further. Another reason for the higher fuel consumption is the higher friction caused by higher piston ring preload, higher piston mass and high pressure pump friction.

The torque, however, is greater in the TDISI- engine at low load were it exceeded the standard turbocharged engine with 10 percent. The writers of this study also makes the assumption of fuel saving of 20% using a downsized engine with fully variable intake valves and even higher when introducing variable exhaust valves.

2.3 Valve lift/phasing strategies

Throttle free control can be achieved with fully variable intake valves and there are multiple strategies for this. See the list below.

- Early intake valve closing (EIVC)
- Late intake valve opening (LIVO) + EIVC
- Pre-lift
- Multi-lift
- Combinations of the above with different lift heights
- High- and low speed static lift profiles

2.3.1 EIVC

By closing the intake valve early, Miller-cycle, a smaller charge than usual is present for the combustion. At the same time no backflow occurs but the effective compression ratio (ECR) drops. Another problem is after the IVC, when the piston is still moving down, the turbulent intensity is reduced. This results in slower flame propagation and thereby increasing the combustion duration. The latter problem have been discussed in the SAE-paper 2013-24-0057 [5], where the possible solutions of introducing extra tumble by masking one side of the intake valves or swirl by using asymmetric intake valve lifts have been tested. The result was a decrease in combustion duration from the baseline engine by around 10 crank angle degrees (CAD) with extra tumble and by 6 CAD with swirl. These improvements were achieved at 2000 rpm and 2bar BMEP. For higher loads or engine speeds the improvement decreases. At full loads the extra tumble masking resulted in a lowered turbulence intensity and hence a worsening in combustion and power. The 50% mass fraction burned (MBF 50%) were delayed by 8 CAD. For the extra swirl strategy there was no big difference from the baseline engine.

2.3.2 LIVO + EIVC

When a late intake valve opening is adopted the pumping losses increase drastically due to the syringe effect near top dead centre (TDC), as shown in SAE-paper 2014-01-1064 [8] the LIVO strategy have a higher ECR than the EIVC strategy. The big decrease in pumping losses results in a very small improvement regarding the brake specific fuel consumption (BSFC) compared to the baseline engine. To reduce the pumping losses this strategy can be combined with EIVC by using dual lifts. This will be further described under multi-lift.

2.3.3 Pre-lift

By opening the intake valve a small portion during the end of the exhaust stroke internal exhaust gas recirculation (EGR) is possible. As the piston moves towards TDC residual gases are pushed up in the intake port, causing large backflow, and in the beginning of the intake stroke these gases re-enter the cylinder. With the pre-lift strategy a high ECR is possible at the same time as the pumping losses are minimized, although the substantial EGR amount slows down the combustion. The latter effect has a bad influence on the BSFC but this could be limited by the use of multi-lift. The strategy with pre-lift and the later described multi-lift sets higher demands on the valve actuation system. With pre-lift the system needs to be able to lift the valve a small step and then hold it at that lift for multiple CAD and then continue the lift in a second step.

2.3.4 Multi-lift

With a multi-lift solution the advantages with EIVC regarding low pumping work is maintained at the same time as the turbulent energy is boosted by the second small lift, this results in a faster combustion and higher constant volume heat release. Without the internal EGR as with pre-lift, the ECR will decrease a bit but still a large decrease in BSFC can be achieved [8].

2.3.5 High- and low speed static lift profiles

The implementation of dual lift profiles, as in [4] or like Hondas VTEC system can offer some reductions in BSFC and increase in torque/power. By optimizing the lift profiles in GT-power simulations an average BSFC reduction of 1.1% and 6.22% average increase in power is achieved for an 110cc motorcycle engine with CR 9.5:1. These results were compared to a baseline engine. For low speeds the optimized intake valve timings used a moderate LIVO + EIVC strategy (13 CAD later opening and 8 CAD earlier closing) with 6mm lift while the high speed profile had 3 CAD earlier opening and 9 CAD later closing and a lift of 7.6mm [4]. While these systems have a limited freedom of design they are well tested and reliable.

2.3.6 Problems and solutions

If using variable valves for non-throttled load control a new problem is encountered. The problem is that by eliminating the conventional butterfly valve the gases can move more freely in the intake and pressure waves which generate noise appears. These pressure waves would have otherwise been dampened by the throttle [9]. This problem introduces a multi variable optimization problem were valve lift and valve timing amongst things like air-box design needs to be considered to not compromise the BFSC.

3 Divided exhaust period

We have yet to discover the full potential of fully variable valves. A study made by Borg Warner involves a concept for turbocharged engines. The concept involves separating the blowdown part of the exhaust stroke, which is the first part of the exhaust stroke which supply most of the work to the turbocharger, from the displacement part which is when the piston pushes out the last of the exhaust gases. What the study demonstrated was that by dividing the exhaust port into two and make one of them go straight to the turbine and the other one past by, this was called Divided Exhaust Period (DEP) and is demonstrated in Figure 2. In parallel with the practical experiments a model for simulation in GT-power was made and checked against the real results [10].

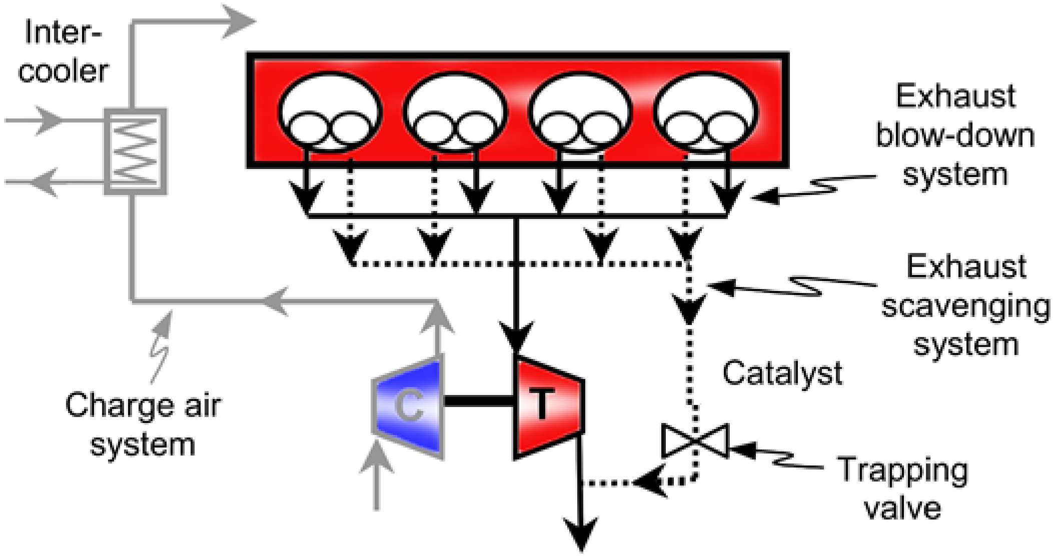

Figure 2. DEP-concept explained

3.1 The DEP-concept so far

There are some well-known issues encountered when turbocharging an engine. First there is the catalyst light-off time. The time it takes for the catalyst to reach working temperature. In a naturally aspirated engine this can be handled with ease by retarding the ignition and hence increasing the exhaust temperature. The problem with a turbocharged engine is that the turbine acts as a heat sink and much heat energy is lost through the turbine and its housing [10]. By bypassing the exhaust straight to a close coupled catalyst you minimize the heat loss that otherwise would have occurred through the turbocharger. One thing to watch out for when doing this is if there is an expectable amount of power left, since the turbocharger is not in use and also because the cam duration for the scavenging valve is much shorter than for a standard engine, the mechanical stress to the cam shaft is more severe [10].

To examine the light-off time for the catalyst from a cold start the amount of HC is measured downstream from the catalyst. Experiments show a decrease in temperature after the turbine in comparison to the standard engine. Before the turbine, however, the time it took to reach 300 degrees Celsius was reduced by 50% with the DEP [10]. Other measurement was performed to study the amount of HC after the catalyst and how long it would take for the concentration to be below 50 ppm. The experiments showed a 38% reduction in time [10].

Another goal with the DEP concept is to reduce the knock sensitivity and make it possible to increase the boost pressure. To accomplish this, the amount of residual gas fractions (RGF) left in the cylinder after the exhaust stroke needs to be reduced. This can be done with the DEP-concept thanks to its scavenging port which is used to flush the cylinder with fresh air and lowering the in-cylinder temperature. It comes at a price however, and that is the amount of unburned HC that passes straight through and out the exhaust. In a study made by Borg Warner the experiments showed that the HC-concentration was 3-5 times higher in the scavenge exhaust than in the blowdown exhaust [11]. To keep the HC from escaping through the exhaust the test engine was fitted with a “trapping valve”. This trapping valve was placed after the close coupled catalyst at the scavenging port and was used to regulate the back pressure of the scavenging port and improve the trapping ratio [10]. The trapping valve was also beneficial for increasing the boost pressure at low engine speeds and closing the trapping valve increased the boost by 40%. This is said to be caused by the exhaust scavenging system acting as a buffer at the end of the stroke [10]. The trapping valve increased the PMEP at lower engine speeds and if the phasing of the exhaust valves would have been variable the need of the trapping valve would be eliminating by varying the blowdown to scavenging overlap [10].

A simulation model showed a reduction in residual gas content with 14% at high speed and in the lower speed range the reduction was up to 60% [10]. No real testing was performed at this stage but pressure differences from the intake to the exhaust system at TDC, were the DEP concept showed a positive pressure difference up to 4500 rpm, give a hint that residual gas content in theory is reduced [10]. This in turn also gave a 6% increase in BMEP and 16% reduction in BFSC.

A few different combinations of scavenging and blowdown overlap have been tested and indicated increase of about 20% in available torque when widening the overlap between intake and exhaust scavenging valve [10].

3.2 The future of the DEP concept

Power in the DEP engine was limited by the allowed exhaust temperature due to material constraints, this temperature was reached at around 2000 rpm. This is said to be the cause of splitting up the relatively cold exhaust from the displacement phase from the blowdown exhausts and thus not cooling down the turbine. The experienced lower temperature after the turbine could however be beneficial for the aging of the catalyst placed after the close coupled one, since it´s exposed to lower temperature. In the future the issue with the high exhaust temperature pre-turbo must be addressed to allow higher loads for the DEP-concept to be successful.

The flow across the exhaust valves was choked in the experiments even though the original valves were replaced with larger diameter ones [11]. One solution to this is to keep the original valves and timing and instead mounting two extra valves in the exhaust which then divides the flow to a blow-down and scavenging pipe respectively. This prevents the choking of the cylinder exhaust valves at higher engine speeds [12]. The concept is called “Externally Divided Exhaust Period” or ExDEP and may be a subject for future development of the DEP-concept.

It was discovered that the use of a “trapping valve”, as seen in Figure 2, can aid with controlling the pressure difference over the cylinder and thus influence the blow through [10]. The scavenging system resembled a wastegate, but a much more efficient one and can assist when using a smaller turbocharger that otherwise would create high backpressure and poor scavenging. Another thing discovered is that the relation between the scavenging cam position and BMEP is close to linear, indicating that the boost could be easily controlled with the cam phasing for the scavenging valve and eliminate the need of a wastegate valve [13].

In the first studies of the DEP concept they used a standard camshaft and changed between different ones to see the effects different phasing had [10]. The authors of that study expressed the benefits of having a fully variable valve train instead, in which case the need for a trapping valve as well as a wastegate could be evaded and thereby eliminating the PMEP penalty and exhaust energy waste you otherwise would get.

The DEP-concept seems to have a lot of benefits to existing system and is a good tool in meeting the higher demands on efficiency and emissions.

4 High Compression ratio

To reach really low BSFC high compression ratio is needed, but with higher CR the in-cylinder temperature and pressure is increased before the combustion. This will for high CR (~13-14:1) result in problems with knock in SI engines. To avoid knocking the ignition angle is delayed or/with combination of a richer mixture i.e. lower lambda. The latter will hurt the BSFC and move a possible three way catalyst out of its operation range, which for obvious reasons is not wanted. To avoid this fuel with higher octane rating can be used, such as ethanol or methanol, or different valve strategies can be applied. One strategy can be to apply a large valve overlap in order to remove all residual gases hence decreasing the in-cylinder temperature during compression and with a delay of the injection timing no extra hydrocarbon emissions will be introduced. Higher boosting and use of Miller-cycle can also reduce the in cylinder temperature since more heat can be conducted through the intercooler.

A problem Mazda faced when developing their high CR "skyactiv" engines was that the flame in the early propagation hit the piston top just below the sparkplug. This caused a big heat transfer away from the flame and thereby slowing down the combustion process, which resulted in torque limitation due to knock. The solution to this problem was the introduction of a small cavity in the piston head below the sparkplug, which allowed the piston to start moving down before the flame could hit the surface [14].

5 Engine modelling

5.1 Combustion models

When modelling the combustion it´s important to choose the right combustion model. This is often a matter of how much experimental result you have access to and how fast running you want your model to be. In some cases you have to choose a more advanced model to be able to predict certain parameters like knock or rate of heat release (ROHR). There are three categories of combustion models, which are described below.

5.1.1 Non-predictive

In a non-predictive combustion model you impose burn rate as a function of crank angle degrees (CAD). So the CA50 and CA10-90 is constant thus not affected by residual gas content or dilution. A model like this is adequate for optimizing intake runner length and looking at engine acoustics etc. [15] The non-predictive model have the advantage of running faster since the burn rate is not calculated. These models also have the advantage of not requiring as many input parameters as a semi-predictive and a predictive model needs and is thereby more suitable if experimental data are difficult to obtain.

Mainly two combustion models are used for non-predictive modelling of SI combustion, imposed combustion profile and the SI Wiebe model.

The imposed combustion profile is a predefined burn rate profile which can be useful if the cylinder pressure has been measured in experiments which then can be used to calculate the burn rate.

The SI Wiebe model imposes the burn rate using a Wiebe-function which mimics the typical shape of a burn rate in a SI-engine. This model is very useful when you do not have access to any measured values through lab experiments. The Wiebe equations look like below [15].

$$BMC = -\ln(1 - BM) \qquad \text{Burned Midpoint Constant} \qquad (1)$$

$$BSC = -\ln(1 - BS) \qquad \text{Burned Start Constant} \qquad (2)$$

$$BEC = -\ln(1 - BE) \qquad \text{Burned End Constant} \qquad (3)$$

$$WC = \left[\frac{D}{BEC^{1/(E+1)} - BSC^{1/(E+1)}}\right]^{-(E+1)} \qquad \text{Wiebe Constant} \qquad (4)$$

$$SOC = AA - \frac{(D)(BMC)^{1/(E+1)}}{BEC^{1/(E+1)} - BSC^{1/(E+1)}} \qquad \text{Start of Combustion} \qquad (5)$$

Where

AA = Anchor Angel
D = Combustion Duration
E = Wiebe Exponent (2.0 by definition)
BM = Burned Fuel percentage at Anchor angle
BS = Burned Fuel percentage at Duration Start
BE = Burned Fuel Percentage at Duration End

A typical Wiebe-function looks like in Figure 3

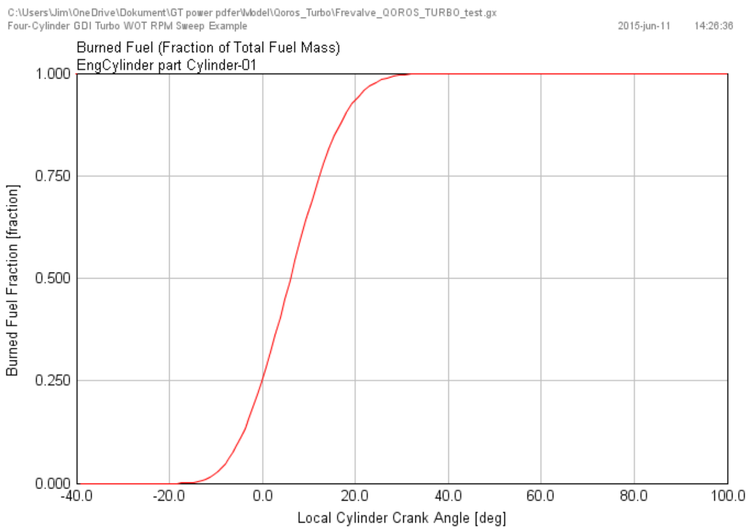

Figure 3. Typical Wiebe function

5.1.2 Semi-predictive

A semi-predictive model uses the advantage of the faster running non-predictive models thus also impose the burn rate with a Wiebe-function. However the parameters for the Wiebe-function are calculated from the input variables. This makes it dependent of other important parameters like intake pressure, spark timing and parameters which affect the burn rate. The semi predictive model predicts the three parameters used in the Wiebe-function as a function of variables that affect the combustion. For a SI-engine these variables are typically engine speed, trapped air mass at intake valve closing (IVC), temperature at (IVC), spark timing, air-fuel ratio (AFR) and trapped residual gas fractions [15].

A semi-predictive model is more accurate than the non-predictive models without the penalties of longer solution runtime.

5.1.3 Predictive

The last alternative is a predictive model. The predictive model can be used to model everything from knock to injection timing etc. the disadvantage with a predictive model is that it requires a lot more computational power for the solution to converge. A lot of parameters on the combustion is needed from experimental studies to calibrate the model, which are not always available and has a lot of influence on the combustion rate [15].

In terms of emissions the only way to calculate the amount of unburned hydrocarbon is to implement a predictive combustion model. An example of a model that can do this in GT-Power is the “EngCylCombSITurb” [15].

One way of speeding up the calculation in a predictive model is to use a Wiebe-function for the first couple of engine cycles and in that way having a start of value closer to the solution. This helps systems like the turbocharger and manifolds reach a near steady state before starting the real calculation with the predictive model [15].

5.2 Knock prediction model

In a modern SI-engine it´s getting more and more common to use higher downsizing factor and thus increasing turbocharger boost. It is also beneficial to raise the compression ratio and by that increasing the thermal efficiency of the engine. These two means of decreasing fuel consumption result in a higher tendency to engine knock. The strategy to run an engine close to knock condition means that extra care has to be taken not to ruin the engine. However there are some techniques in order to reduce knock like increasing the amount of EGR, latter the ignition [2] and also by reducing the amount of RGF after each combustion cycle [10]. To further investigate the influence things like exhaust- and intake-geometry or valve timing have on knock in a simulation environment, you first need an appropriate combustion model and second, a working knock model.

When it comes to predicting knock there is a couple of method available and implemented in the GT-suite software, *Douaud-Eyzat*, *Franzke*, *Worret* and the more recent one called *Kinetics-fit.* These are explained below.

5.2.1 Douaud-Eyzat

Douaud-Eyzat is an empirical model based on the Arrhenius equation [16]. Out of all available knock prediction models in GT-power the Douaud-Eyzat is the only one recommended for investigating pre-ignition knock. The Douaud-Eyzat is adequate for predicting knock but since the model does not take into account dilution it cannot be used properly when having more than 5% EGR [17]. As a result the Douaud-Eyzat knock model can´t be used when investigating the influence of EGR on knock. The Douaud-Eyzat model is capable of modelling negative temperature coefficients (NTC) and is influenced by temperature, pressure and octane number [18].

5.2.2 Franzke and Worret

The Franzke and Worret knock prediction models are very similar since the Worret model is based on the Franzke which in turn is based on the Douaud-Eyzat [15, 19]. Franzke and Worret´s Knock prediction models are based on the matching of empirical data to the Arrhenius function [19] to determine the speed of the chemical reactions taking place in the combustion chamber. The Worret model however is an improvement from the much older Franzke model (from 1981) with more empirical data based on high-pass filtered heat release to back it up. Neither of these models are recommended for predicting pre-ignition knock [15].

5.2.3 Kinetics-fit

The Kinetics-fit is a model developed by Gamma Technologies. Unlike the other knock models the Kinetics-fit is based on a triple Arrhenius equation. It is not recommended for predicting Pre-ignition knock however but it is capable of calculating NTC and are influences by the same parameters as the Douaud-Eyzat with the addition of air/fuel ratio and dilution by inert gases [18]. This model has limited validation against measured data and is not recommended to predict knock without validation against real data [15].

5.2.4 Alternatives

One alternative to using any of the above mentioned knock prediction models is the couple GT-power to a more advanced software for chemical modelling. The burn rate is then calculated from hundreds of chemical species and if properly done this can yield a more accurate result [8]. The GT-power model can also be coupled with CFD-calculation to study the in-cylinder flow and the effects of charge mixture. This would be preferable when studying the effect of pre-ignition knock that is sometimes caused by gasoline mixed with engine oil and influenced by for example injector position [20].

6 Problem formulation

6.1 Reducing fuel consumption

One of the goals with this thesis is to uncover any potential in fuel saving using fully variable valves, high compression ratio and DEP as tools. The goal is to see how much of improvement is possible to get by reducing the pumping work by the means of eliminating the throttle and using EIVC instead, but also by having the exhaust stroke divided into two parts.the exhaust is divided into one port that drives the turbine (blowdown) and one that by passes the turbine hence reducing pumping work (scavenge). That is the theory but to what extent the fuel consumption can be reduced will be investigated in this thesis. The overall thesis work was divided into two engine speed/load cases, namely the following.

2000 rpm	4bar BMEP
2000 rpm	20bar BMEP

6.2 Reducing Knock probability

To be able to run a compression ratio of 14:1 something needs to be done to prevent the engine from knocking. This is where the DEP-concept really plays its role. The thesis aim to find out how much you could reduce both pre- combustion cylinder temperature and burned residual gases left in the cylinder prior to combustion. Both of these play a part in causing engine knock.

6.3 Delimitations

When looking at a potential fuel savings some delimitation had to be done since the degree of freedom becomes larger in numbers when applying variable valves, where both the opening and closing timing angle of the valve can be adjusted freely as well as the lift height. For the first simulations focus was on investigating just the blowdown closing and the scavenge opening and keeping the blowdown opening and scavenge closing at a pre-set CAD. This made it easier when analysing the results due to the fact that 3D surface plots could be used and no fourth or fifth dimension would be needed, which otherwise would further complicate the analysis of the result. This approach was kept during the thesis, only changing a maximum of two variables at a time.

The thesis was focused on reducing fuel consumption at part load, 4 bar BMEP, at 2000 rpm, which is an important load point since it approximately represents driving on a highway road.

The reducing of the knock probability is the most important at low engine speed and high loads. For this the case 2000 rpm at a load of 20 bar BMEP was investigated. When it comes to predicting knock this thesis is limited to reduce some known reasons for knock, the cylinder temperature and the RGF. Due to a limited time and resources, no knock model was used since it would have obliged the use of predictive combustion model, of which measured values for making it accurate was non-existing. The alternative of using Logesoft to do the Chemical kinetics and knock was scrapped due to the limited amount of time.

Another criterion for the whole thesis is that lambda was kept at a constant value of 1.0 to ensure the function of a three way catalyst since no investigations was done on the emissions. These criterias also applies to the standard engine when comparison are made between standard and DEP-concept.

7 Method

7.1 Model

The building of the model in GT-power is critical for the whole thesis work. The model needs to be as correct as possible, without the use of experimental data. This puts a large pressure on getting the right setups for the different parts. Much of the model was originally borrowed from GT-Power´s own example library and with these came values on friction, flow coefficients etc. This would otherwise be hard to find, considering the lack of a physical engine in a test rig to extract these values from.

7.1.1 Geometries

The base geometries of the engine were given from the start and are shown in Figure 4. From the left it starts with a flow split to be able to measure the combined mass flow of the ports and inject the right amount of fuel. The flow split has an inlet diameter of 65 mm and two outlets of 36.6mm diameter each. The flow split is then connected to an inlet runner with the inlet diameter of 36.5 mm, outlet diameter of 32.5 and length, 100 mm which is then connected to a second inlet runner part with the outlet diameter of 26.5 mm and length, 90 mm. Next is the intake port which has an inlet diameter of 26.5 mm is 60 mm long and has an outlet diameter of 25.8 mm. Connected to one of the inlet ports is an fuel injector. In the end of the inlet port is the inlet valve which has a diameter of 29 mm. The cylinder object is then defined according to Table 1.

Table 1. Cylinder geometry

Cylinder bore	77 mm
Stroke	85 mm
Compression Ratio	14

The exhaust valves have the same geometry as the inlet valves. The port however is different with an inlet diameter of 25.8 mm but an outlet diameter of 30 mm connected to an exhaust runner with the uniform length of 30 mm and length of 200 mm.

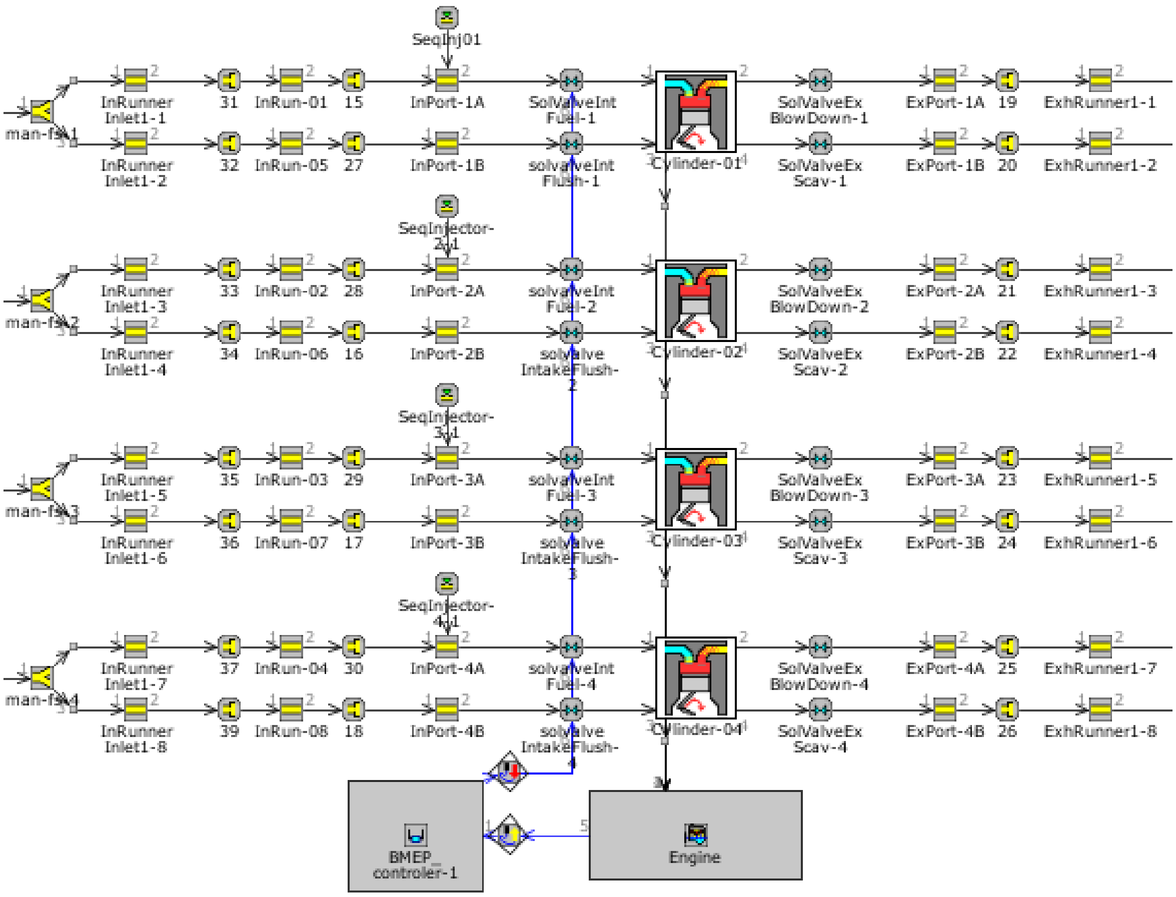

Figure 4. Base model

7.1.2 Valves

To simulate the pneumatic valves used in this thesis a solenoid valve object was used. The input needed for this object was coefficient of discharge, both forward and backwards, but also the lift curve divided into an opening array and a closing array. The lift curve was retrieved from the supervisors and hade the appearance shown in Figure 5.

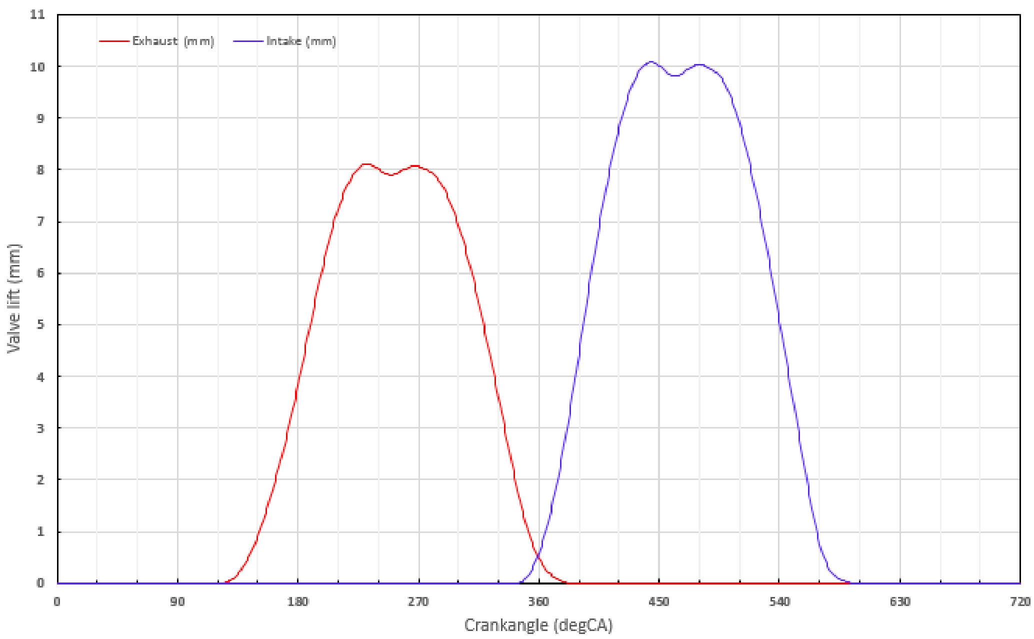

Figure 5. Lift curves at 6000 rpm

The lift curves were defined in lift at specified CAD but the GT-Power valve object was needed them to be specified in lift at specific time in milliseconds, since the lift depended on the engine speed with limited opening and closing speed. This needed to be accounted for by calculating the time corresponding to the given CAD in Figure 5, which corresponds to the lift at 6000 rpm. The lift per ms was then calculated by using equation 6

$$ms = \frac{1000}{6} * \frac{\theta}{n} \tag{6}$$

The curve then needed to be divided into an opening and closing curve as Figure 6 illustrates. The same curves where used for both intake and exhaust. The distance D was varied for different valve lift durations where the lift height was kept at top lift for the distance D to get the closing timing angle aimed for, the closing angle was then set in GT-Power.

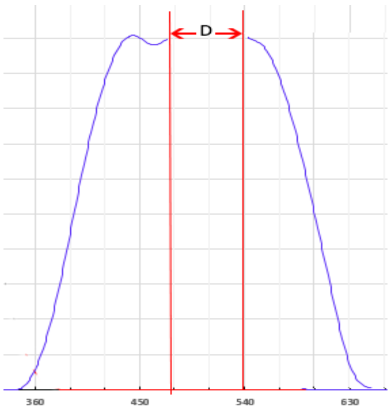

Figure 6. Opening and closing curve

7.1.3 Physical data

Because there are no experimental data to be found on this particular engine many of assumptions had to be made. One of the assumptions was regarding the combustion object. A non-predictive Wiebe model was used throughout the thesis work and typical values for CA50 and CA10-90 was given for full load from the supervisors of the thesis and can be seen in Table 2.

Table 2. Combustion data

Full load combustion data		
RPM	CA50	CA10-90
0	3	22
2000	6	23
4000	7.5	25
6000	9	27
8000	9	29

A delivery rate of 17 g/s was set to the fuel injector and the injection duration was then calculated by using the mass flow to reach the lambda target from a defined start of injection of 250 CAD before TDC. The fuel demand for the DEP concept was unknown and thus the standard value from the GT-Power example model was used. A fluid object of indolene combustion was chosen in the injector object representing gasoline combustion.

7.1.4 Simplified model

In order to get a fast start and to get familiar with the characteristics of the DEP concept a simplified model was imposed. This model was stripped down to only consist of inlet boundaries, intake runners and -valves, cylinders, exhaust valves and -runners and two different outlet boundaries. A schematic layout is depicted in Figure 7.

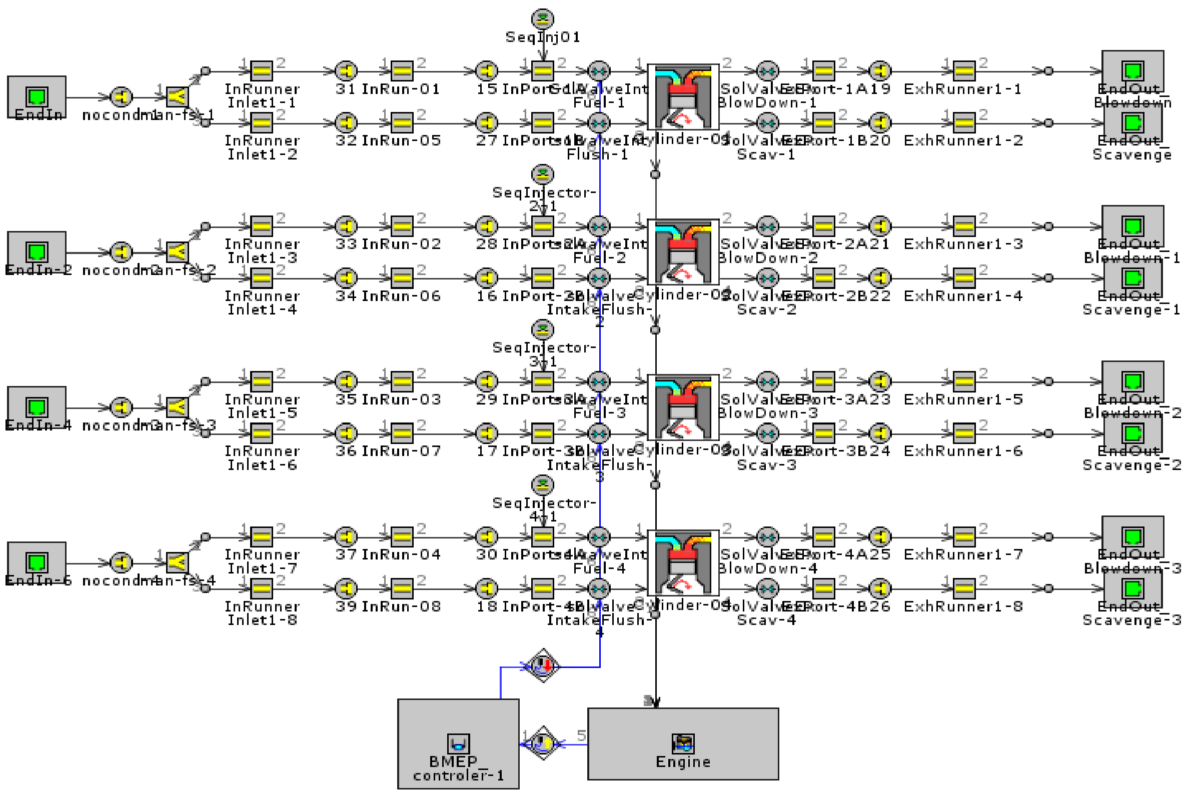

Figure 7. Simple GTP model

With such a simple model the simulation time was noticeably reduced, although it should be noted that the results will not be that accurate. A simple model has many limitations and it is important to understand the flaws the simplifications introduce. One of the most noticeable flaws when not simulating a turbo and only imposing a higher pressure at the blowdown boundary is the lack of penalty in boost pressure when bypassing the turbine and letting much of the exhausts go through the scavenging exhaust port. This since the boost pressure imposed in the inlet boundary was not coupled to the amount of bypassed exhaust gases.

7.1.5 Model with turbocharger

With a better understanding of how the different parameters affected the characteristics of the system, a more advanced model including a turbo was implemented. To be able to model a turbo an intake distribution pipe, i.e. plenum, was built and a simple intercooler modelled. For simplicity the intercooler was modelled as a bundle of 200 identical rectangular pipes with a fixed wall temperature. This will result in a big heat sink that lets the user specify the intercooler outlet temperature. This is a fast and easy way to model this. Since there is no room for designing and optimizing an intercooler in this thesis this is a good approximation. For the turbocharging a compressor was connected to the incoming air. The compressor was connected via a shaft to the exhaust turbine that connects to the blowdown valve collector. In order to stabilize the turbo in the simulation start a high inertia multiplier was set to the shaft connecting the turbine and compressor to keep the initial speed. This multiplier was set very high during the first three iterations, 2000 times the normal inertia, and then it faded down to the actual inertia finally set as unity. For this case one big difference from a regular cam-driven valve-actuated turbocharged engine is that no wastegate is used. Instead the exhaust energy that is not needed to achieve the wanted boost pressure is passed by the turbine via the scavenging port. Alternatively the intake valve is closed earlier (Miller cycle) which allows for a higher pressure ratio for the compressor for a given amount of in-cylinder air. A schematic picture is shown in Figure 8.

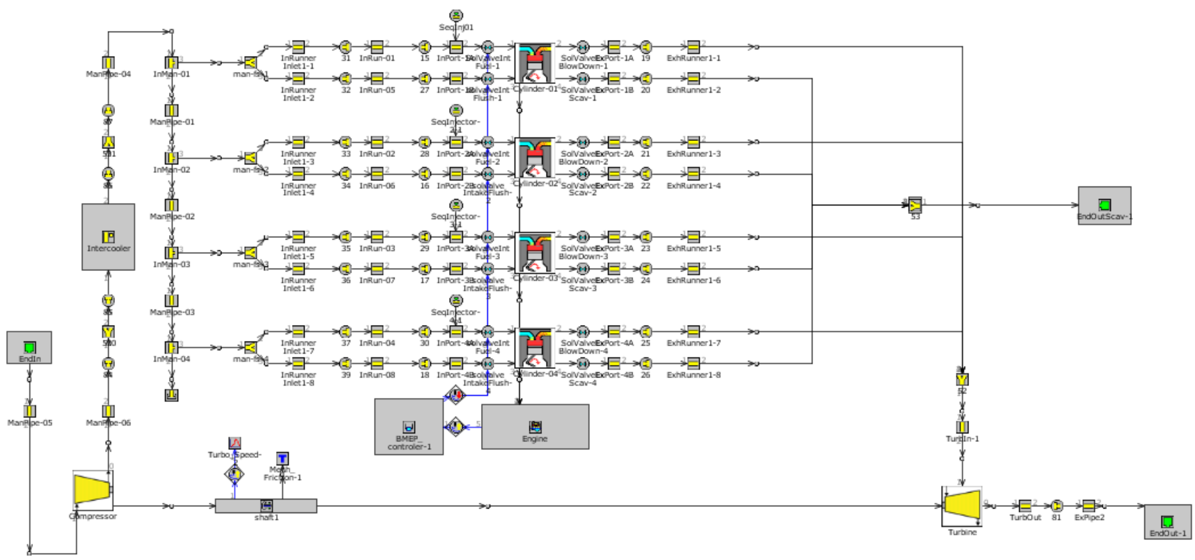

Figure 8. Turbo model

At first a turbine and compressor map from a GTP tutorial was used, but since these maps had unrealistically large high efficiency zones an alternative map was used in the last simulations and comparisons were made between the two. The second turbocharger was taken from the benchmark engine and needed to be adjusted to fit the requirements of this model. The requirements where that the turbocharger should be able to produce 20 bar BMEP at 2000 rpm and 24 bar BMEP at 6000 rpm. However this proved to be difficult with the real turbocharger map since it was narrower than the one originally used. To make it work the load limit at 6000 rpm was decreased to about 22 bar BMEP. The compressor performance maps for the two Turbochargers are illustrated in Figure 9 and Figure 10

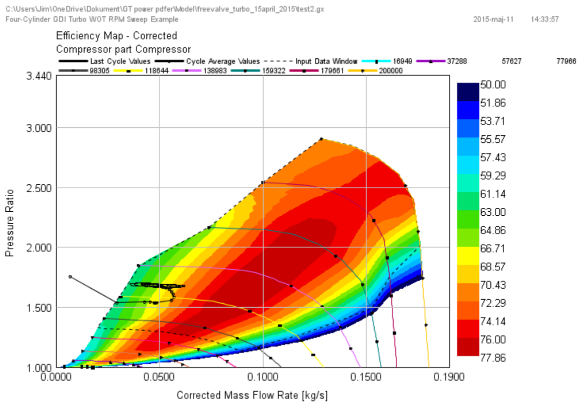

Figure 9. GT-Power compressor map

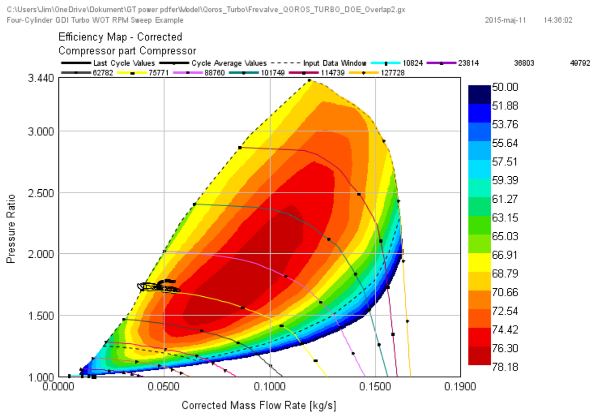

Figure 10. more realistic compressor map

7.2 Design of Experiments

In order to easily get a good overview of the problem, DOE's were used to change different parameters within a specified interval and later plot them in 3D graphs for analysing. The varied parameters was from the beginning the closing of the blowdown valve and the opening of the scavenge valve, while the blowdown opening and scavenge closing was fixed at 150 CAD and 370 CAD. The intake closing angle was controlled via a PI-controller to meet the specified BMEP target and thus varied between 390-560 CAD.

By running these relatively large setups of approximately 150 cases the trends are a lot easier to spot then by varying a parameter independently and in single cases by hand. When doing so it is easy to forget the changes and a trend cannot be displayed in e.g. a 3D- or contour plot.

7.2.1 Amount of Scavenging

The main experiment consisted of figuring out how the amount of scavenging affected the residual gases and pumping work. The blowdown closing and scavenge opening valve timings were varied with different overlaps, an overall timing DOE is seen in Table 3. The intake opening, blowdown opening and scavenge closing was locked in this experiment according to Table 4. The experiment was performed both with imposed backpressure in the simple model and with the more advanced turbo model.

Table 3. Scavenge DOE

	min	max	# levels
Blowdown closing	180	360	11
Scavenge opening	180	360	11

Table 4. Locked parameters

	Timing angle [CAD]
Intake opening	350
Blowdown opening	150
Scavenge closing	370

Figure 11 illustrates how the valve timings were varied. The way which the timings were varied presents a problem when the valve-overlap between the blowdown closing and the scavenge opening is negative. So called negative DEP-overlap, which is undesirable.

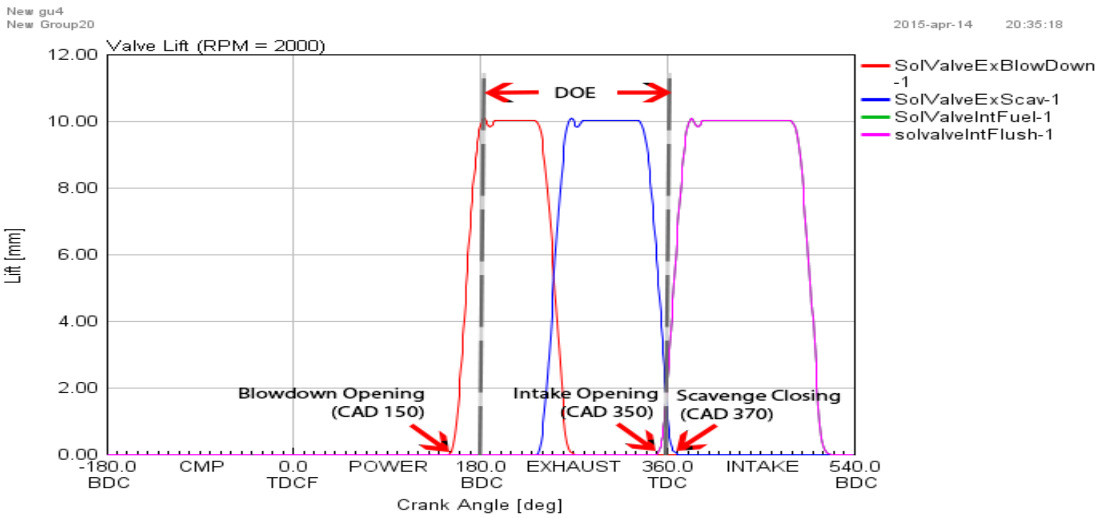

Figure 11. Scavenging DOE

7.2.2 Timing for the lowest BSFC

The DOE of the scavenging gave a bigger picture to what could be accomplished with the DEP-concept and this knowledge was then used to further investigate any kind of improvements in BSFC at both 4 and 20 bar BMEP at 2000 rpm, with emphasis on the 4 bar load point. The best points regarding BSFC was picked out for further investigation of the parameters that was formerly locked to see if there was any improvements when varying these.

For the case with 20 bar BMEP a DOE was designed for varying the intake open and scavenge closing as illustrated in Table 5.

Table 5. DOE valve overlap

	Min CAD	Max CAD	# Levels
Intake opening	330	360	6
Scavenge closing	350	380	6

Since the model had not been calibrated against experimental data it was compared to an identical benchmark engine with the exception of the DEP-concept and the variable valves. This to get some notion of whether there was any improvements when implementing DEP, variable valves and high CR and if so, how much.

7.2.3 Timing for lowest residual gas content and cylinder temperature

With the first scavenge DOE as starting point the minimising of residual gases and in-cylinder temperature was performed in similar fashion as with the minimising of BSFC. A compromise of the points with the least amounts of residuals left in the cylinder and the point with the lowest in-cylinder temperature was chosen from the first DOE and was then further analysed. For further analysis the intake- and exhaust valve overlap was varied for the 20 bar BMEP load point, this time with an even greater overlap and a smaller increment for more detail as illustrated in Table 6. This was performed with the two different turbocharger performance maps but focus will be on the latter one which represents a more realistic turbocharger.

Table 6. Minimizing RGF, DOE

	Min CAD	Max CAD	# Levels
Intake opening	320	370	11
Scavenge closing	355	400	11

7.2.4 Intake and exhaust lengths

From the beginning the geometries concerning the lengths of the intake runners was given but to see if there was any dynamic effect that could be used for better scavenging or/and lower fuel consumption a DOE was performed on pipe lengths. The DOE consisted of varying the intake-, blowdown- and scavenge lengths. For this the intake runner which earlier consisted of two separate runners divided into two sub-parts was merged to only two separate runners with no sub-parts as presented in Figure 12.

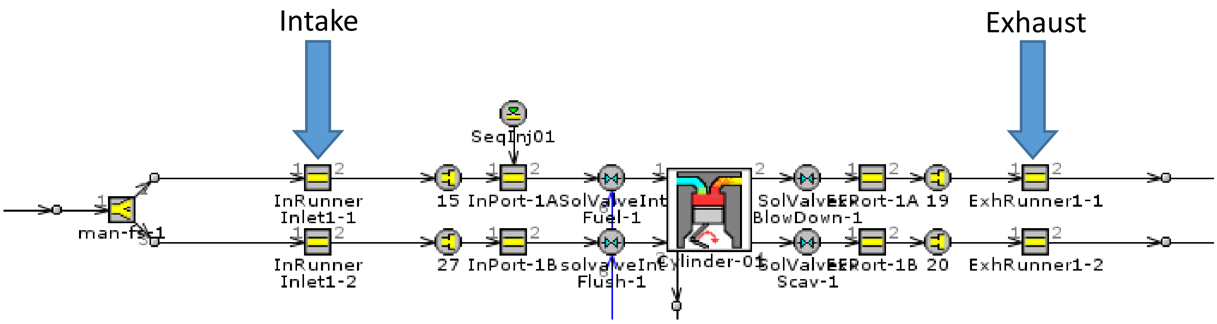

Figure 12. Model for pipe lengths DOE

The DOE were constructed as illustrated in Table 7. The exhaust runner would have to be varied separately which would have increased the CPU time exponentially and the main purpose of this experiment was to reduce RGF which the blowdown exhaust was believed to have little to do with. Therefore only the scavenging length was varied.

Table 7. DOE pipe lengths

	Min mm	Max mm	# Levels
Intake length	75	300	11
Scavenge length	150	400	11

7.2.5 Internal EGR

In order to further reduce the BSFC on part load the possibility of using internal EGR was tested. The test was divided in to two different strategies. Internal EGR through re-breathing, i.e. by letting the exhaust gases pass out into the exhaust system and then be sucked back into the combustion chamber. This is achieved by having the exhaust valves open past the TDC, or by opening the intake valves a large portion before TDC. This lets the exhausts go up in the intake runners and then sucked back into the cylinders. The other strategy used negative valve overlap, NVO, where the exhaust valves are closed before TDC and intake valves opened after TDC. In this way residual gases are trapped in the cylinders. By closing the exhaust valves at different angles the amount of internal EGR is easily controlled. The different strategies are depicted with their characteristic valve timings in Figure 13 *and* Figure 14. When using re-breathing the valve lift was reduced in order to have valve to piston clearance.

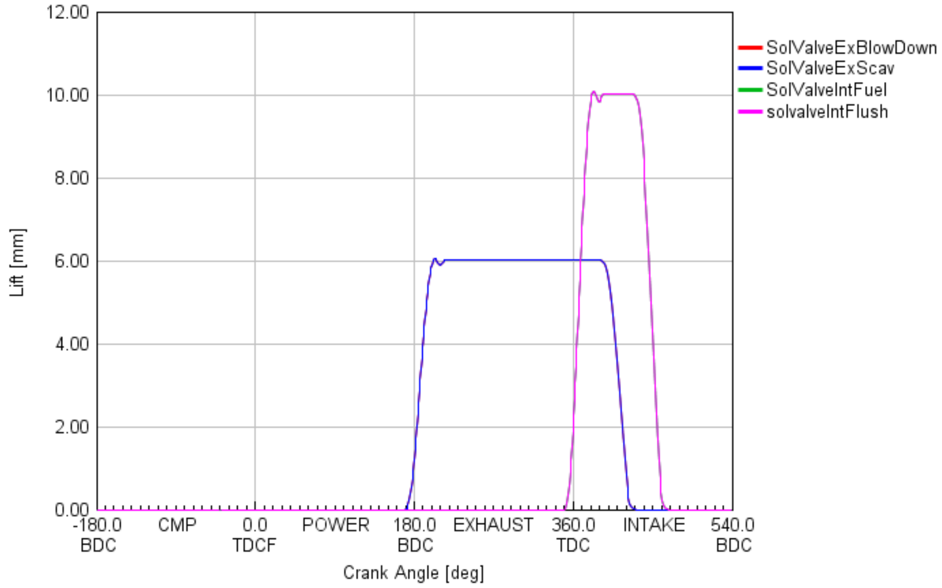

Figure 13 EGR through re-breathing

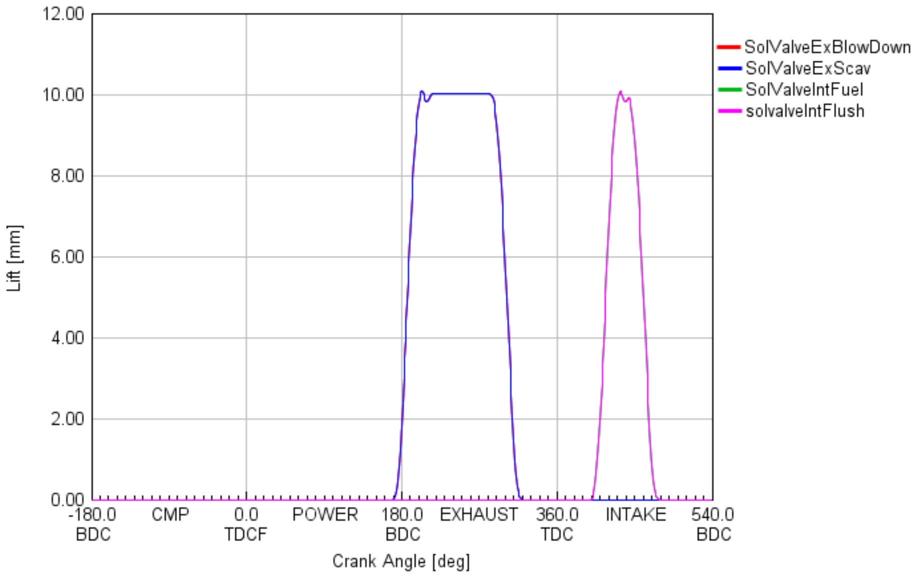

Figure 14 EGR through NVO

7.3 Post Processing

The post processing was performed in GT-post and MATLAB. When processing larger DOE simulations the extension GT-post HUGE was used. Since GT-post HUGE allocated more memory at start-up, this was needed when processing large DOE-files.

In order to easier describe the results of the DOE simulations the data was exported from GT-post and plotted in MATLAB, this was due to the ease of more freely choosing the ascending order of the axis and plotting the actual data point in order to see the inter- and extrapolated data. This was done using a downloaded plot routine for MATLAB called *Contoureplot.m*, see Appendix. The difference between the GT-post and MATLAB plots is shown in Figure 15.

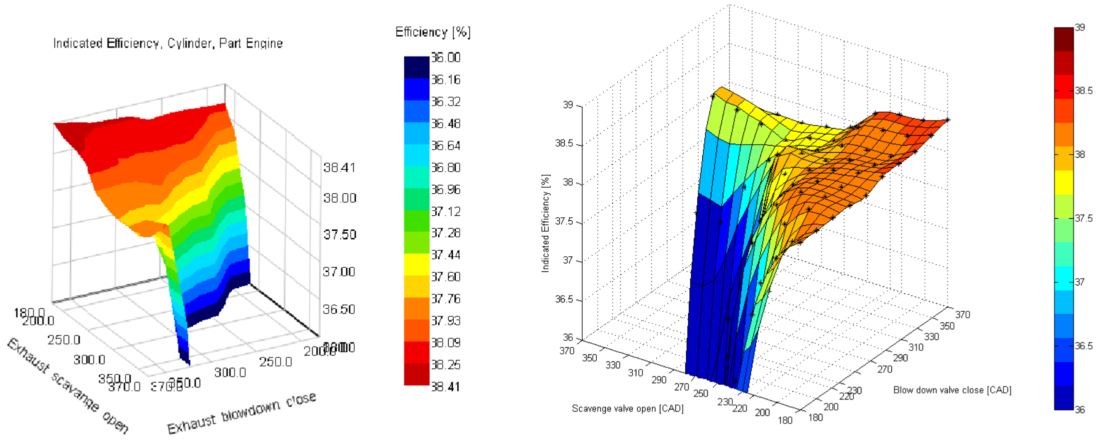

Figure 15 Plots in GT-post (left) and MATLAB (right)

As seen in Figure 15 when plotting in GT-post the axis numbering is interfering and the ascending order feels flipped when the highest values is present at the front and vice versa. This problem is solved by using MATLAB. Furthermore the chequered contour gives more depth to the plot which helps to interpret the difference in values.

8 Result

8.1 Amount of Scavenging

The amount of scavenging was examined with the simple model as well as the model with the turbocharger. The data points marked with a star (*) is simulated data points. Cells with no stars are inter- or extrapolated in the plot routine. This goes for all of the upcoming 3D surface plots.

8.1.1 Result with simple model

First off is the result for when no turbo model was used and shows why negative DEP overlap has a negative effect on engine efficiency.

Figure 16 is an illustration of what happens when the exhausts are trapped due to a negative DEP valve overlap. A pressure rise at the end of the exhaust stroke is the result. This pressure in later released when the intake valves opens, and thus the energy used for compressing is lost.

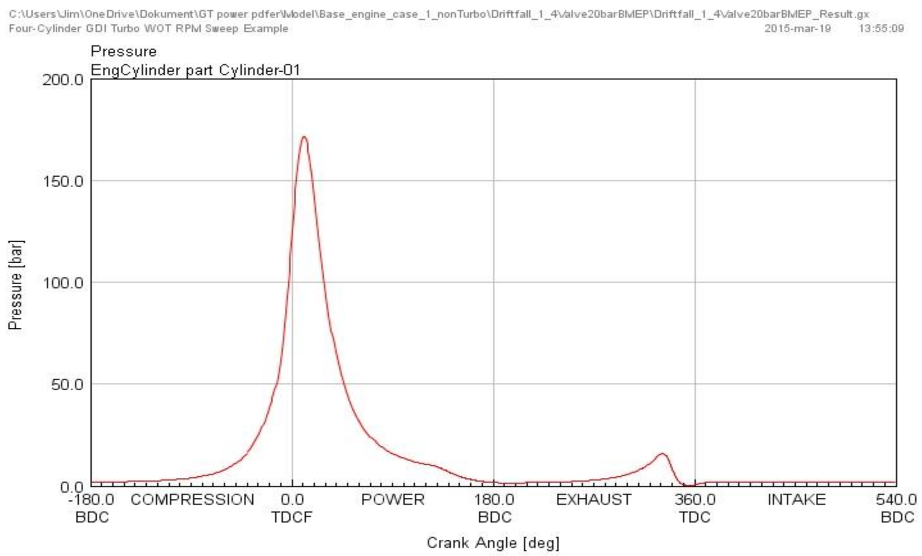

Figure 16. Illustration of what happens when negative DEP overlap is applied

8.1.1.1 Part load

With the simple model, tests were run with the use of less than four valves. These setups are presented in Table 8.

Table 8 Valve setup

Intake	Exhaust
1	1
1	2
2	1
2	2

Only the case with 1-1 and 2-2 valve setup is presented below, this is due to the similarities between the cases 1-1 and 1-2 resp. 2-1 and 2-2. The results for case 1-2 and 2-1 are located in the appendix. The result for using one intake- and one exhaust valve is shown in Figure 17 and Figure 18**Error! Reference source not found.** below. The DOE was set to examine the overlap between the intake- and exhaust valves at TDC. It is seen that the lowest BSFC is located at the point with largest valve overlap. This is also the point where the lowest pumping work is. The optimum is in this case located at an extremity, it would be interesting to simulate even larger valve overlaps. But this was not possible due to valve to piston clearance. This is a reoccurring problem. The valve timings for the lowest BSFC, 275 g/kWh, are presented in Table 9.

Table 9 Valve timings for the case 1-1 valves

	[CAD]
Fuel intake opening	340
Flush intake opening	-
Intake closing (PI regulated)	458
Blowdown opening	160
Blowdown closing	380
Scavenge opening	-
Scavenge closing	-

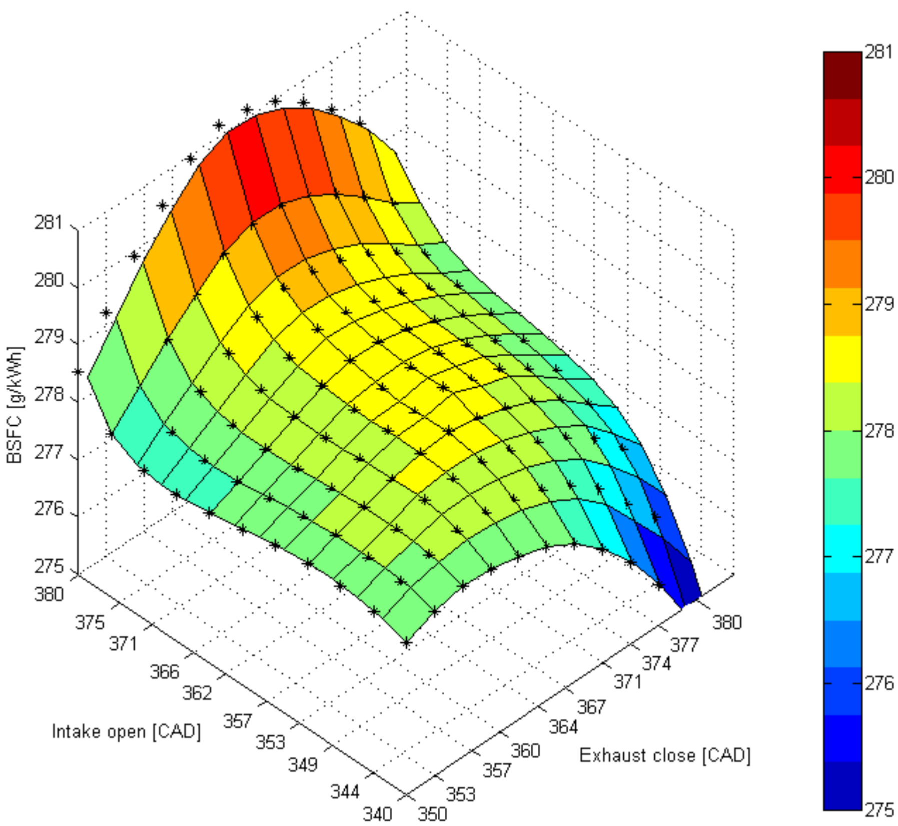

Figure 17 BSFC for 1-1 valve setup where all exhausts go to the turbine

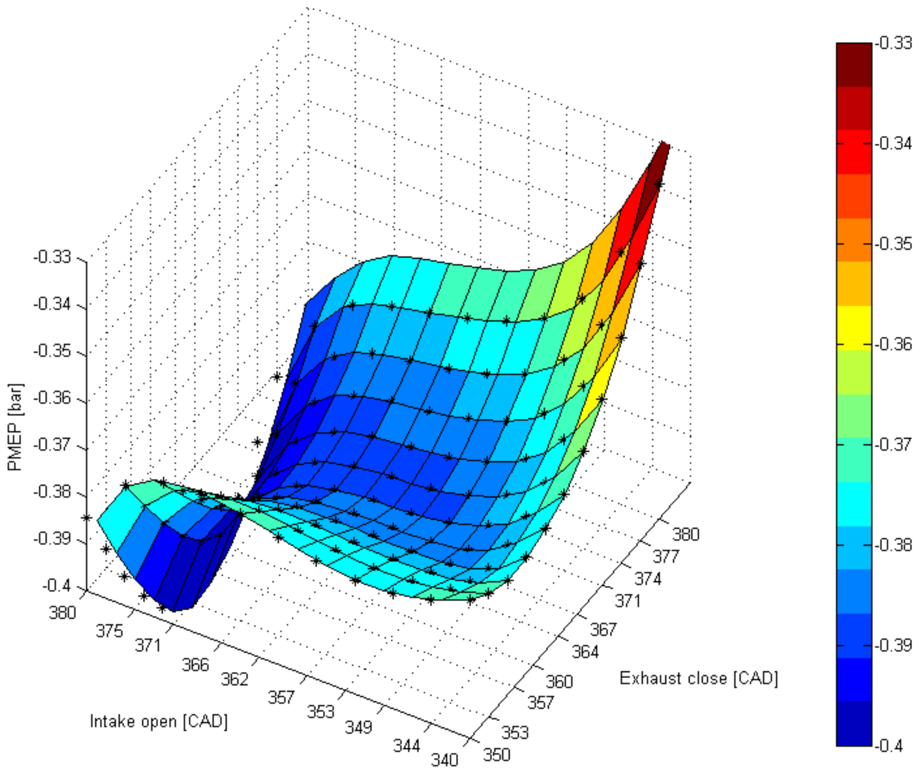

Figure 18 PMEP for 1-1 valve setup where all exhausts go to the turbine

The results for the four valve setting is presented in Figure 19 and Figure 20, where the lowest BSFC, 268 g/kWh, is achieved with the valve timings presented in Table 10. The lowest BSFC is achieved when both exhaust valves is used as much as possible, this is due to the fact that when bypassing the turbo the pumping losses decreases. Since the outlet to the turbo has a higher pressure, but at low loads this pressure is not that high. This results in a benefit in using both valves fully instead of closing the valve towards the turbine. The BSFC for 4 valves per cylinder is lower than for using only two valves, this could be explained by the possibility of spinning the turbine with the first exhaust gases with a higher pressure coming early from the cylinder and then close off the turbine to reduce pumping work. In the case with only two valves operating all the gases go to the turbine and a higher pumping work is needed.

Table 10 Valve timings for case 2-2 valves

	[CAD]
Fuel intake open	360
Flush intake open	360
Intake closed (PI regulated)	442
Blowdown open	160
Blowdown close	320
Scavenge open	180
Scavenge close	370

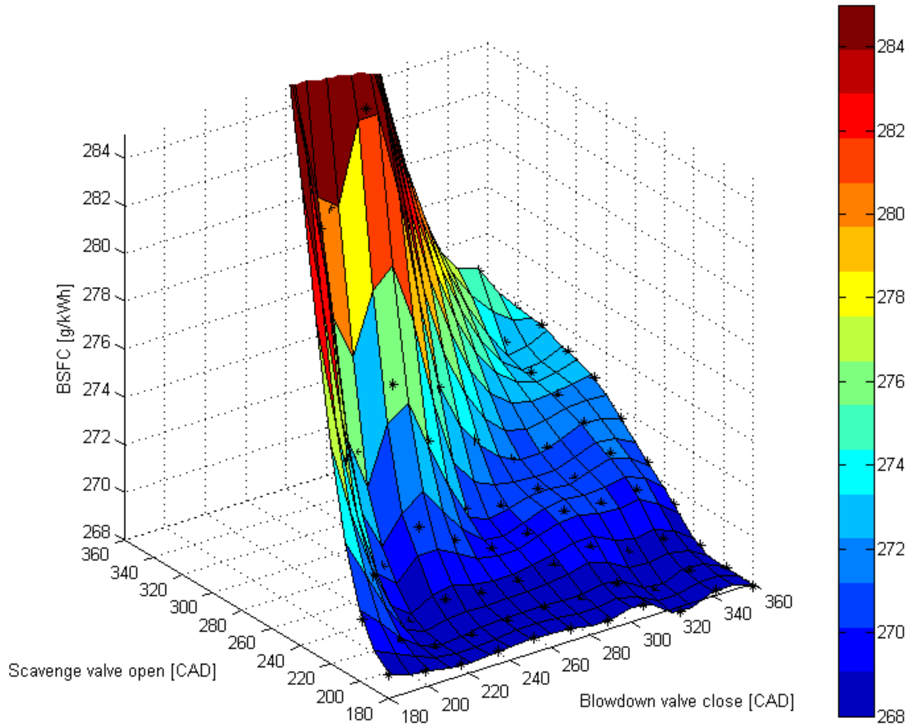

Figure 19 ***BSFC for 2-2 valve setup with amount of turbine bypassing***

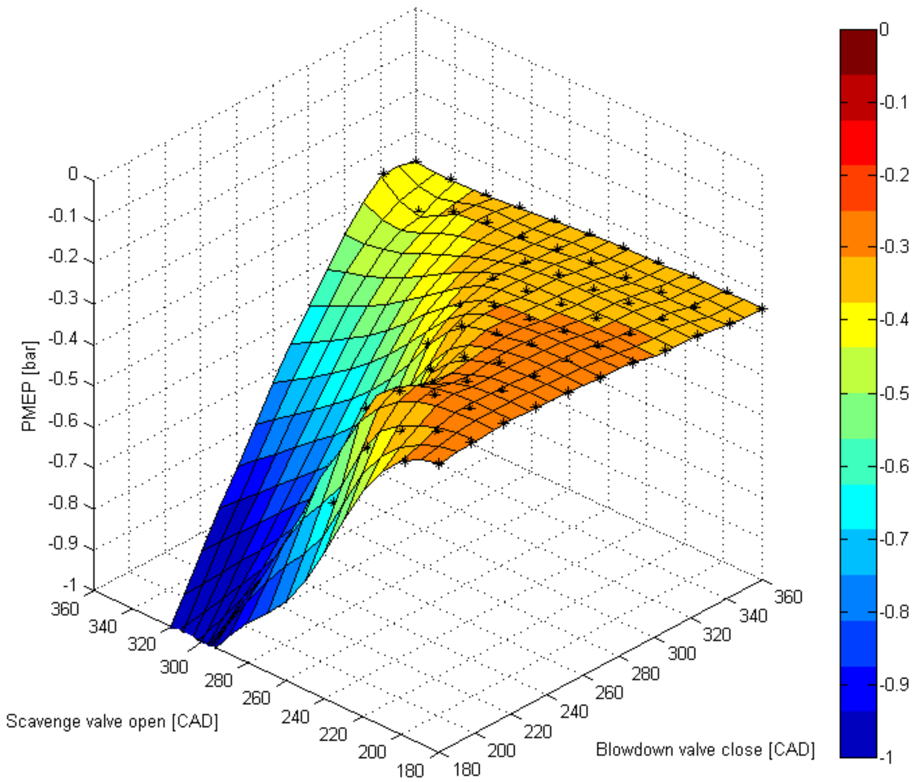

Figure 20 PMEP for 2-2 valve setup with amount of turbine bypassing

8.1.1.2 Full load

Figure 22 illustrates the efficiency when the blowdown closing and scavenge opening timing angle are varied with an imposed intake- and exhaust pressure through the end environments. The result show the highest efficiency when both of the valves are open as long as possible, with the longest overlap between blowdown and scavenge. This is further explained by the PMEP which shows the least pumping losses when the valves are open the longest, see Figure 22.

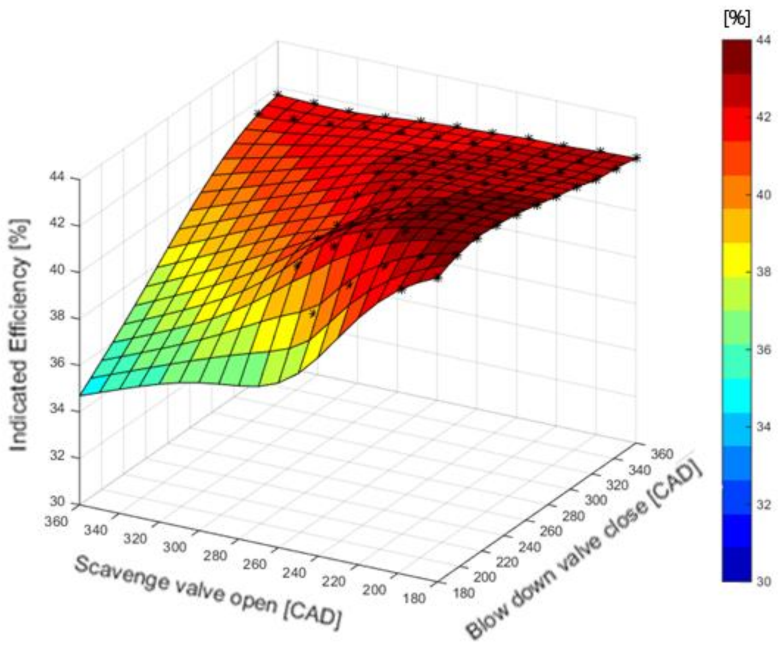

Figure 21. Indicated efficiency, DOE blowdown closing and scavenge open at 20 bar BMEP 2000 rpm

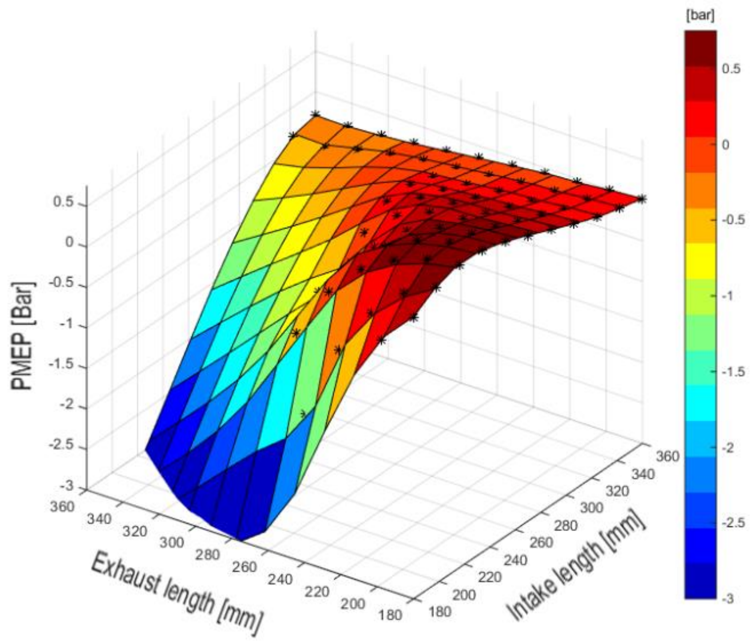

Figure 22. PMEP, DOE blowdown closing and scavenge open at 20 bar BMEP 2000 rpm

8.1.2 Result with turbo model

8.1.2.1 Part load

The turbo model is divided into two setups where the difference is the turbine and compressor maps. Results with standard maps from GT-Power are presented in Figure 23, Figure 24 and Figure 25. In this case the blowdown valve opening was fixed at 160 CAD, the scavenge valve closing at 380 CAD, the intake fuel opening at 380CAD and the intake flush opening at 350CAD. Lowest BSFC was achieved with the valve timings presented in Table 11. The optimal performance is achieved with as long exhaust durations as possible, this is due to the low load, 4bar BMEP, and thus the turbo does not make a large obstacle for the passing gases and only using one valve would make a larger obstacle. This is more accurate than the results from the simple model, since there the pressure representing the turbine was set as constant and the gain in boost pressure was the same regardless of the amount of gases passing the turbine.

Table 11 Valve timings for turbo model with GTP maps

	[CAD]
Fuel intake opening	380
Flush intake opening	350
Intake closing (PI regulated)	444
Blowdown opening	160
Blowdown closing	380
Scavenge opening	160
Scavenge closing	380

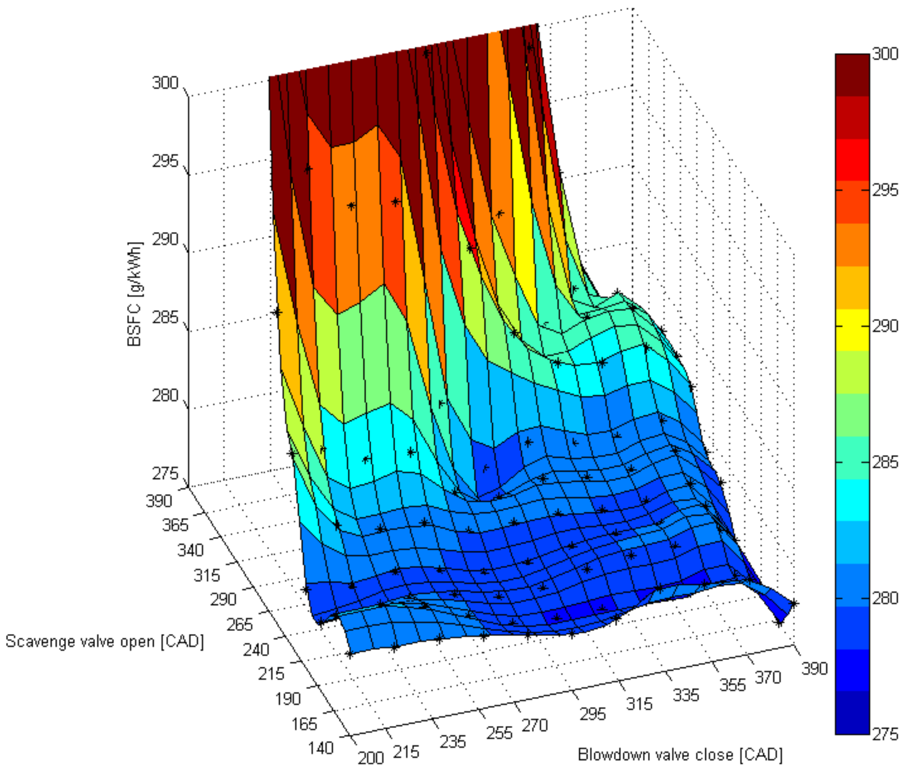

Figure 23 BSFC for turbo model with GTP maps

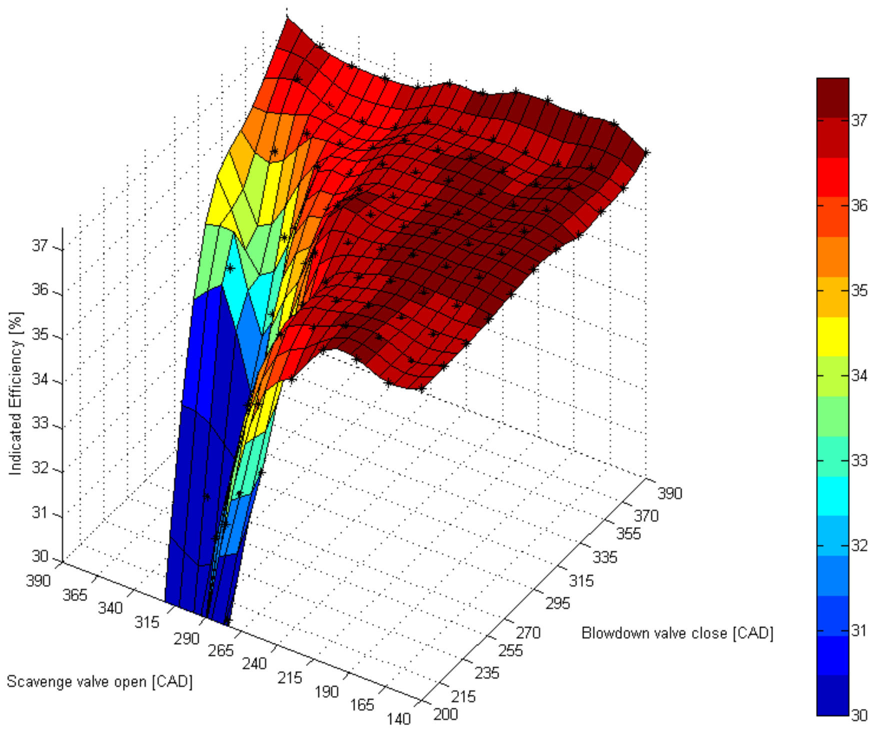

Figure 24 Indicated efficiency for turbo model with GTP maps

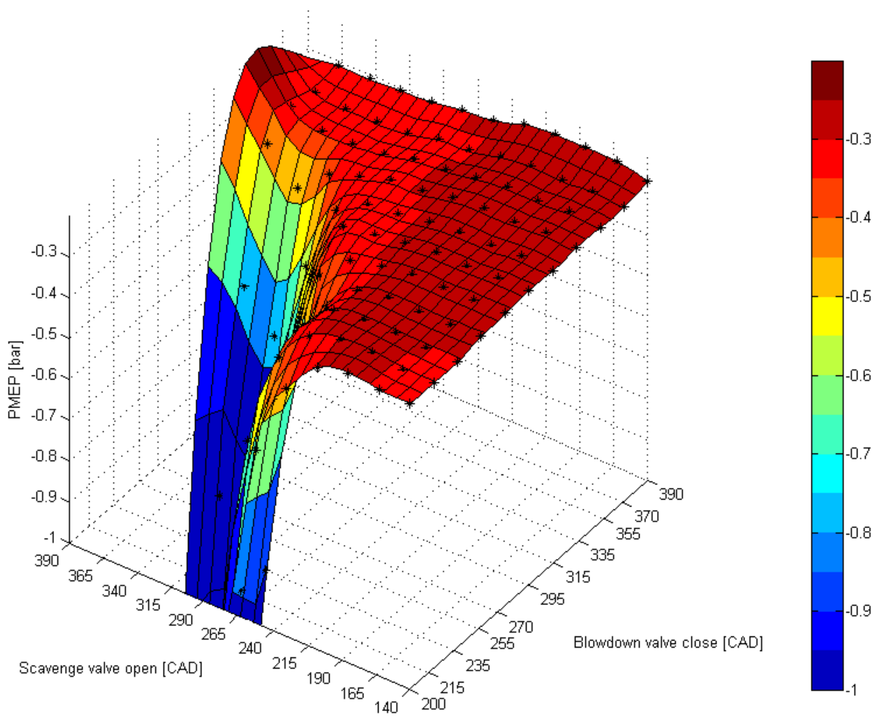

Figure 25 PMEP for turbo model with GTP maps

The results with turbo maps from the benchmark engine are presented in Figure 27, Figure 28 and Figure 29. In this case the blowdown valve opening was fixed at 160 CAD, the scavenge valve closing at 380 CAD, the intake fuel opening at 380CAD and the intake flush opening at 350CAD. Note that the resulting BSFC at such low load is not influenced by the turbo map since the operating point is very far down to the left in the map. This is seen in Figure 26.

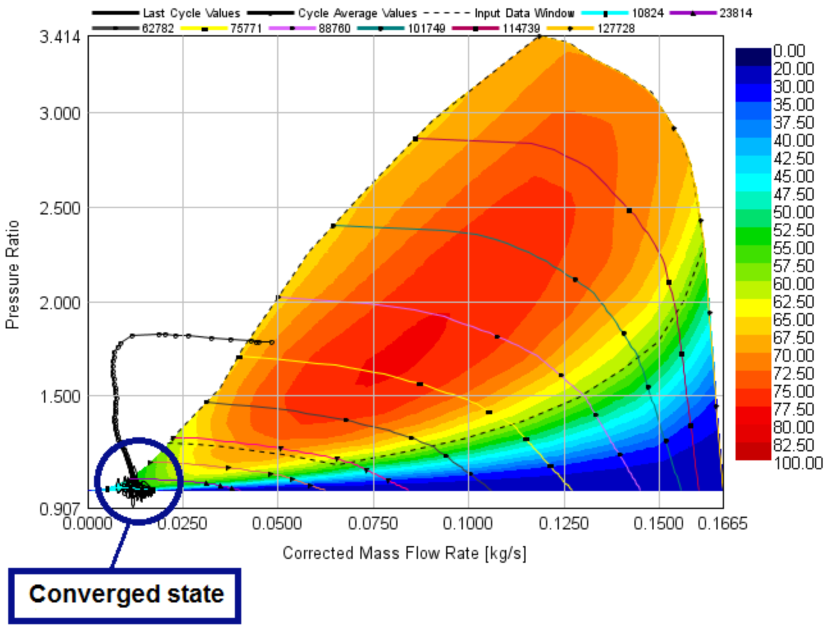

Figure 26 Operating point in the turbo map at 4bar BMEP

Lowest BSFC was located with the valve timings in Table 12. The result is very similar to the case with the GT-Power turbo maps, hence the same interpretation of the results is done.

Table 12 Valve timings for turbo model with benchmark maps

	[CAD]
Fuel intake opening	380
Flush intake opening	350
Intake closing (PI regulated)	444
Blowdown opening	160
Blowdown closing	380
Scavenge opening	160
Scavenge closing	380

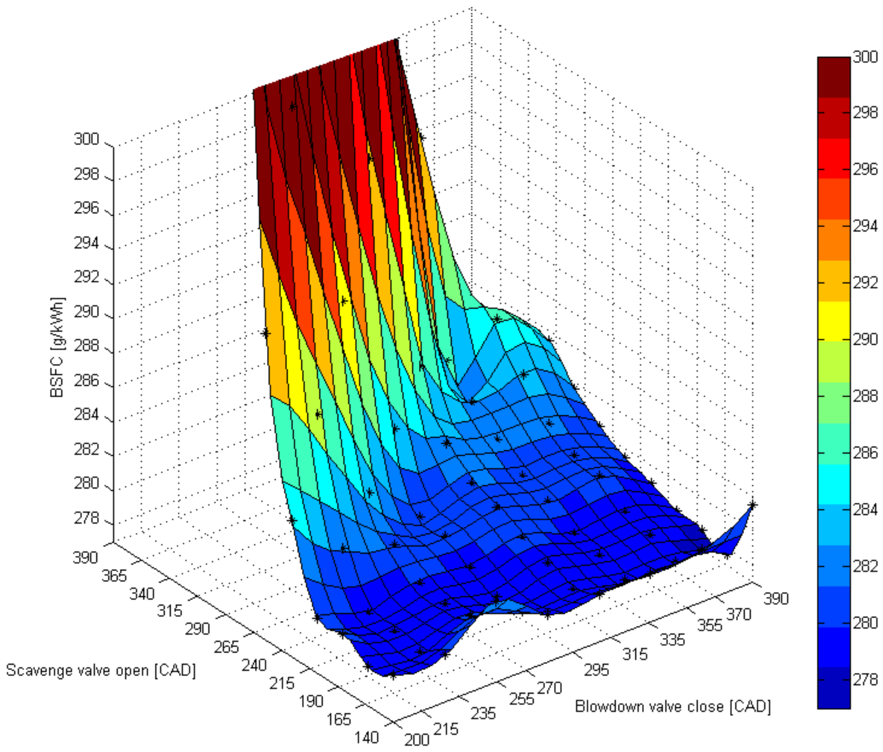

Figure 27 BSFC for turbo model with benchmark maps

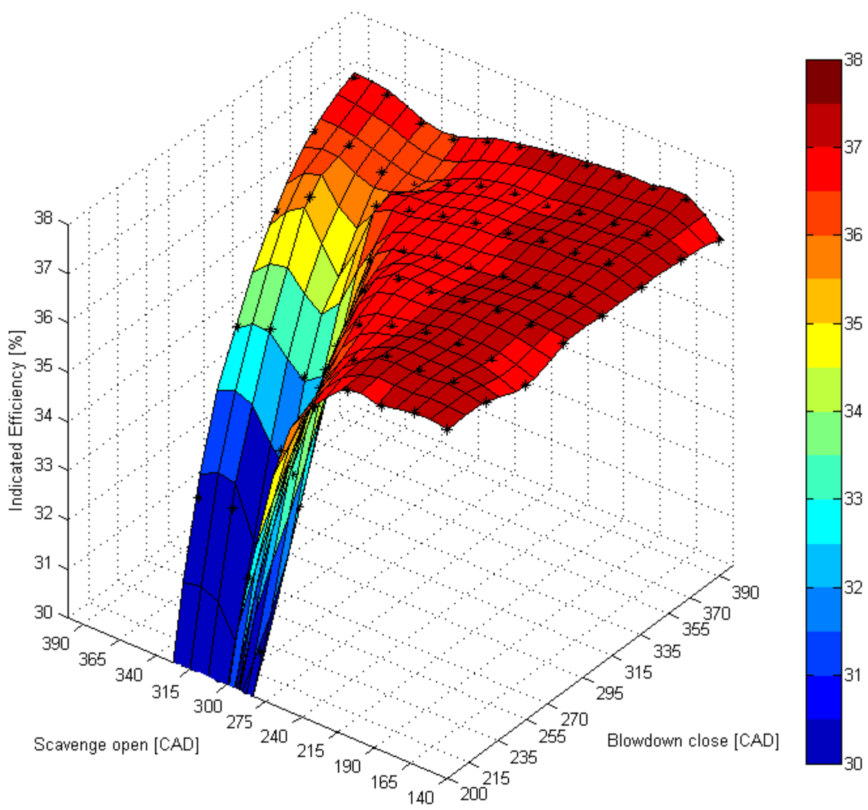

Figure 28 Indicated efficiency for turbo model with benchmark maps

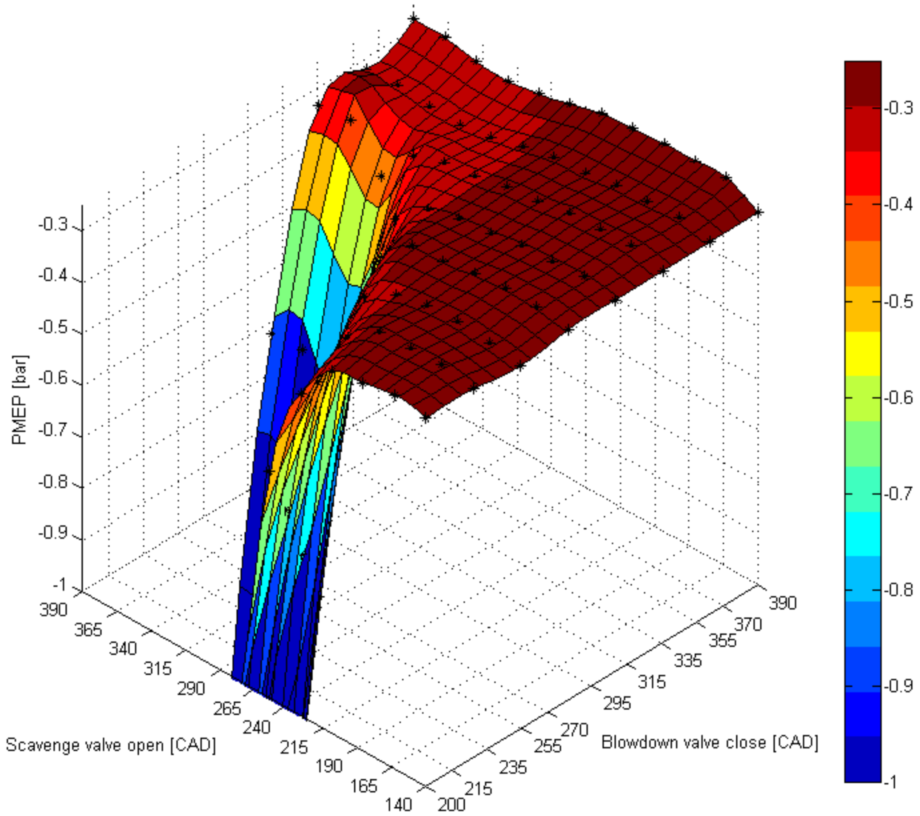

Figure 29 PMEP for turbo model with benchmark maps

8.1.2.2 Full load

When running the DOE for the full load case with modelled turbocharger the valve timing also effects if the model is able to reach the load target of 20 bar BMEP. In Figure 30 the load is illustrated and show it is unable to reach the load target when the scavenge valve opens earlier than 240 CAD and also when there is a negative DEP valve overlap.

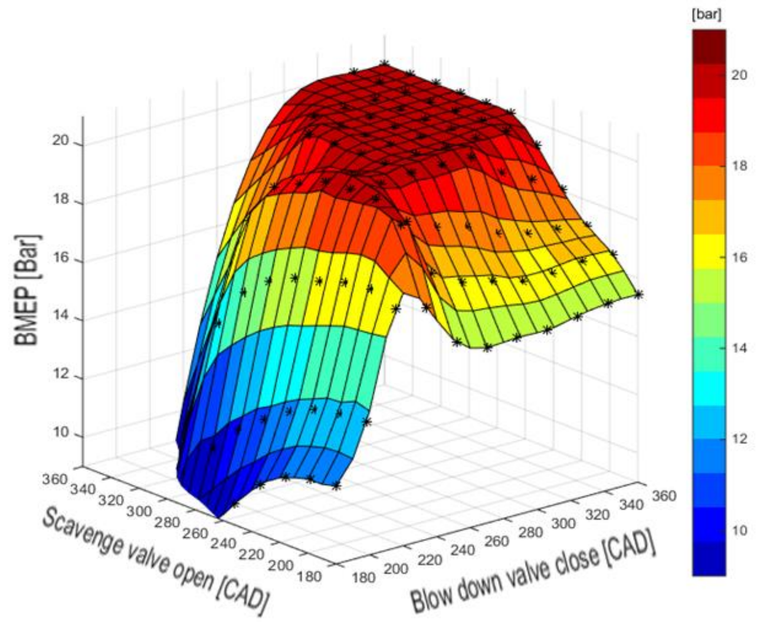

Figure 30. BMEP, full load turbo model

A problem with analysing the result is that the intake closing angle is constantly varying and thus affecting the result. In Figure 31 the intake closing angle is illustrated. As can be seen the intake controller is bottomed out when the model is unable to reach the load target and closes very early for the cases where the scavenge valve opens late.

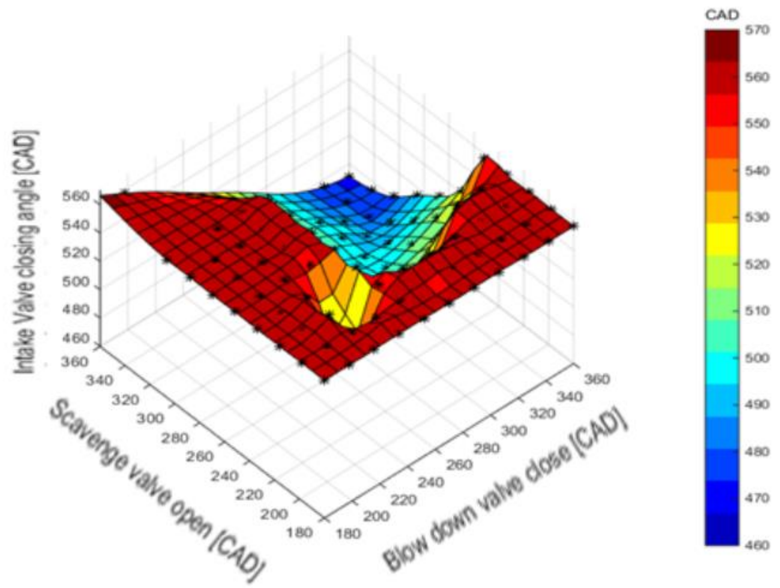

Figure 31. Intake closing angle. full load turbo model

The PMEP is depicted in Figure 32. The results are most favourable when the overlap between blowdown and scavenge is about 30-40 CAD and as predicted the result of negative DEP overlap is a high amount of pumping losses.

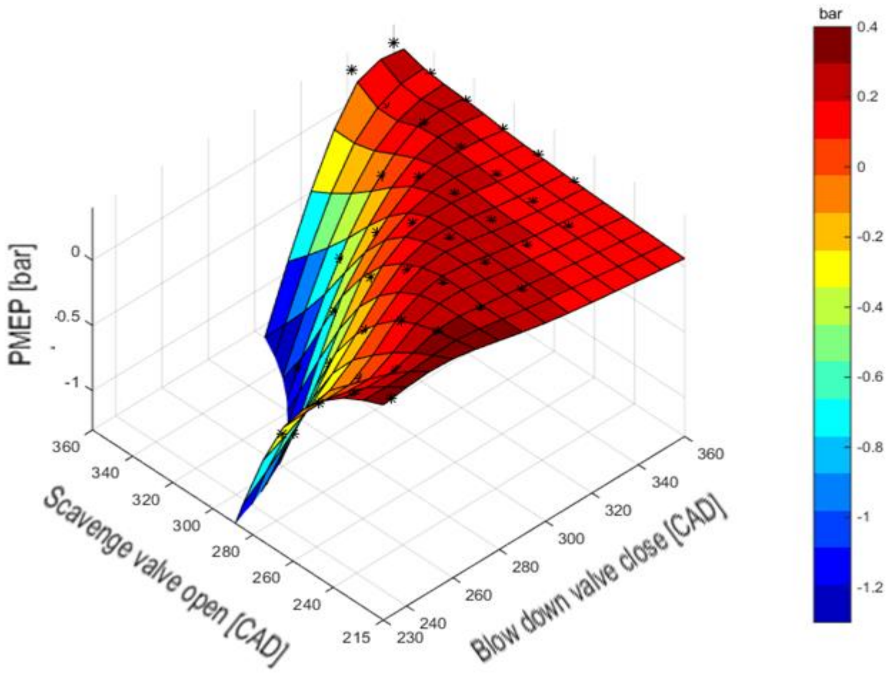

Figure 32. PMEP, full load turbo model

The resulted indicated efficiency is depicted in Figure 33. The highest efficiency is achieved when the blowdown valve closes at around 280-290 with the scavenge overlap of around 40 CAD at 240-250 CAD. At this point the scavenge open as early as possible without compromising the load target.

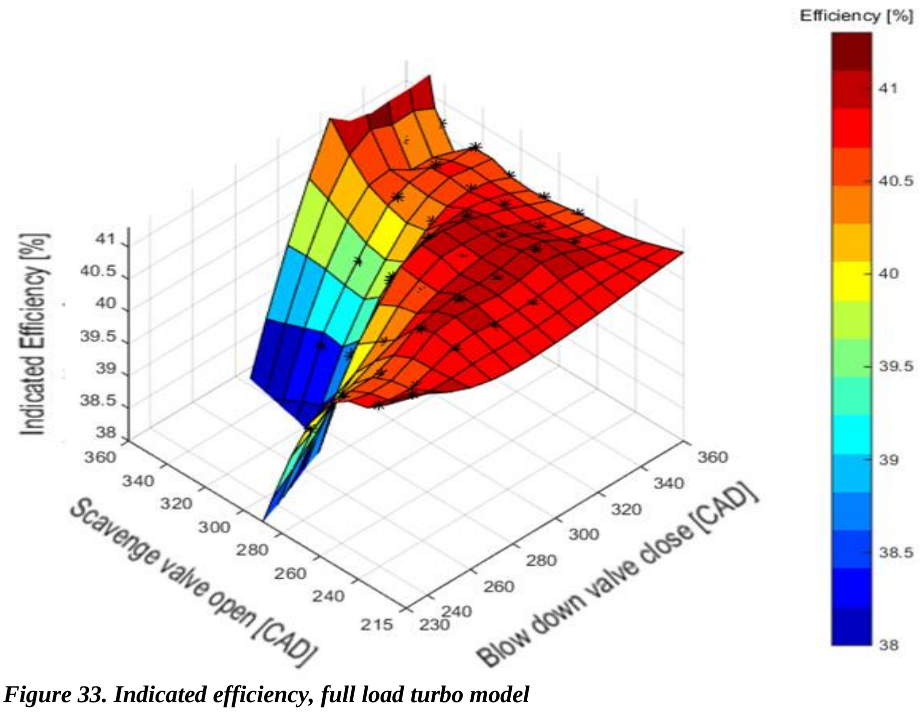

Figure 33. Indicated efficiency, full load turbo model

The goal in this thesis is BSFC reduction and thus a plot of the resulted BSFC is depicted in Figure 34. The point with the lowest BSFC coincides with the highest efficiency as predicted.

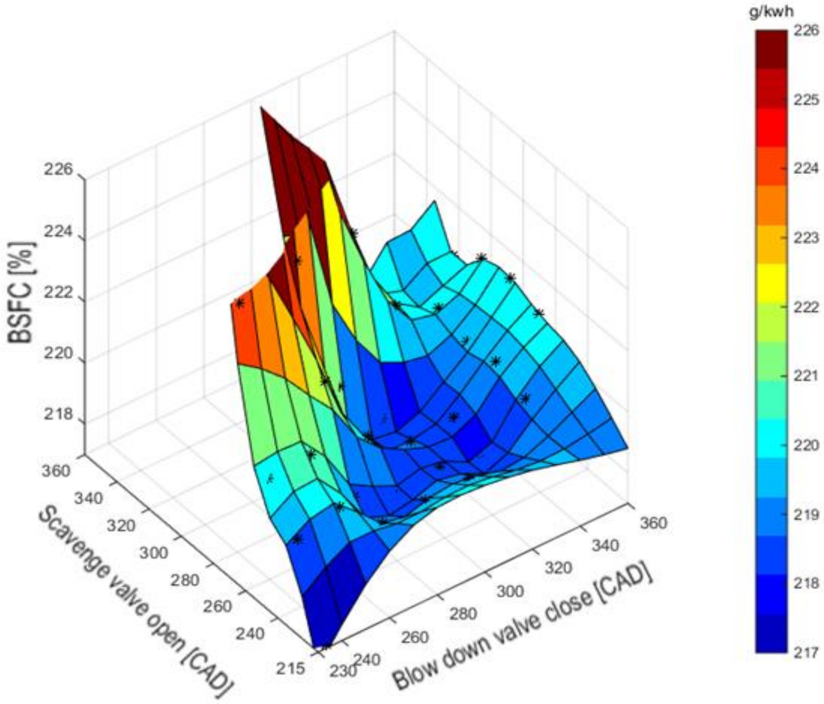

Figure 34. BSFC, , full load turbo model

Something that also is of interest is if scavenge timing also plays a role in the in-cylinder temperature just before combustion. Figure 35 illustrates this temperature with the lowest temperature when the scavenge valve is opened very late. A low in-cylinder temperature before combustion is good for reducing the probability of knock.

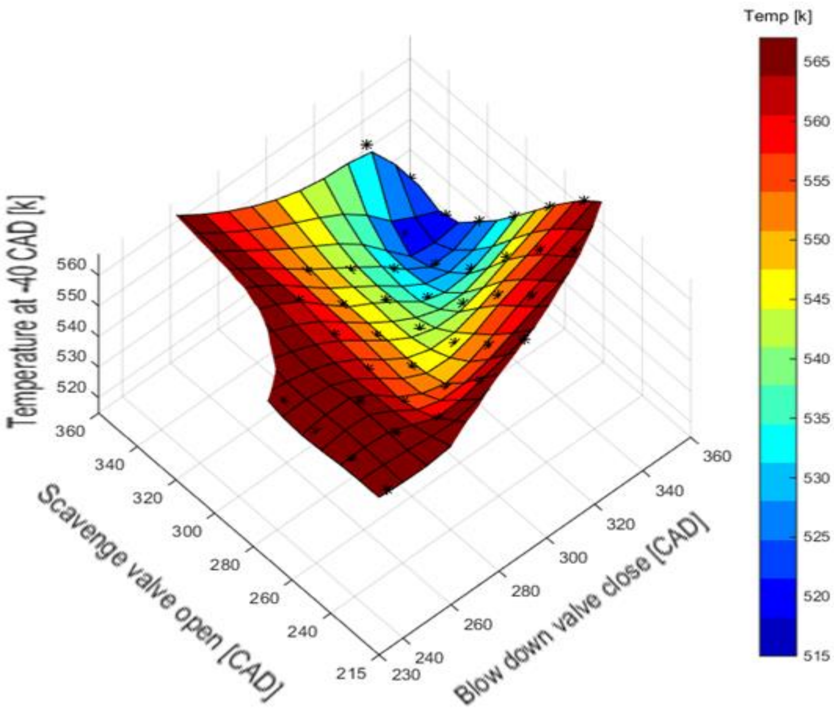

Figure 35. Temperature at 40 CAD before TDCF , full load turbo model

Further investigations were made on the residual gas content in the cylinder 40 CAD before TDCF. This is illustrated in Figure 36. It appears there is a valley with low RGF when the scavenge valve opens at 306 CAD with the lowest RGF appearing at the longest negative DEP overlap shown in Figure 37. This is likely some dynamic in-cylinder effect.

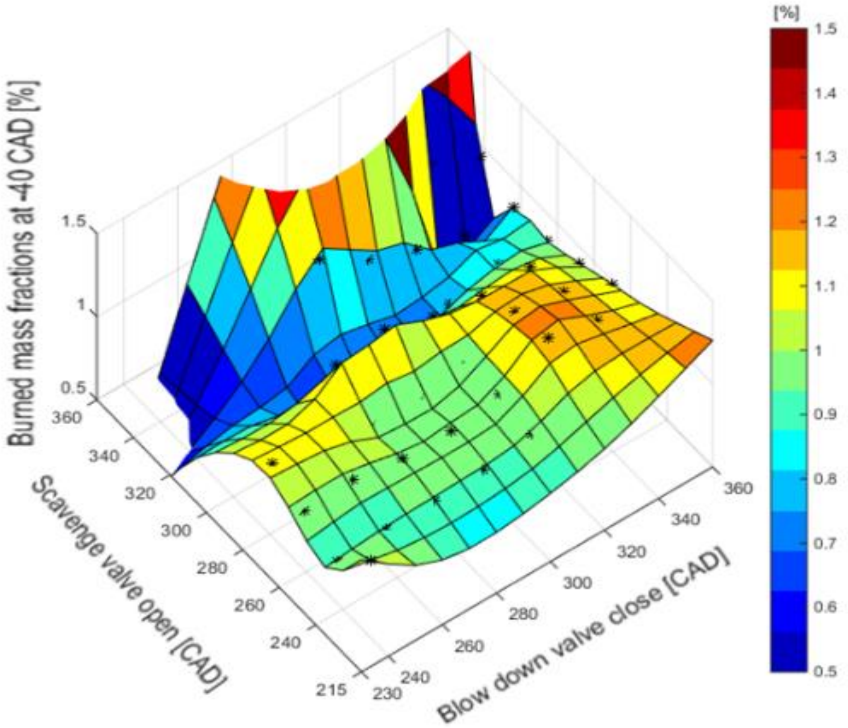

Figure 36. Burned mass fractions at 40 CAD before TDC firing, full load turbo model

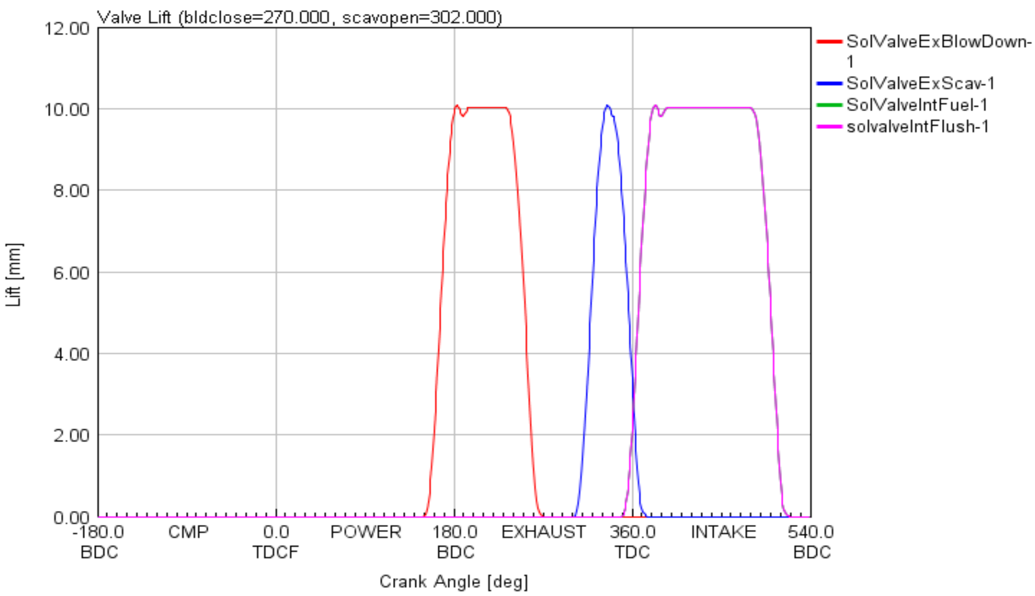

Figure 37. Valve timing at lowest RGF, full load turbo model

The cases with lowest residuals, temperature and BSFC all occurs at different valve timings with different pumping loops as shown in Figure 38. Where the red curve corresponds to the lowest amount of RGF, the blue curve corresponds to the lowest BSFC and the green corresponds to the lowest temperature. The red curves derives from the valley that could be seen in Figure 36 and shows an increase in pumping work but also some dynamic effects right before the overlap that causes an increase in pressure right before the overlap which could cause a lot of RGF escaping through the exhaust. For the green curve the pressure different between cylinder and exhaust, as could be seen in the Figure 38, is higher than for the other cases and it so happens that the RGF are low even in this case.

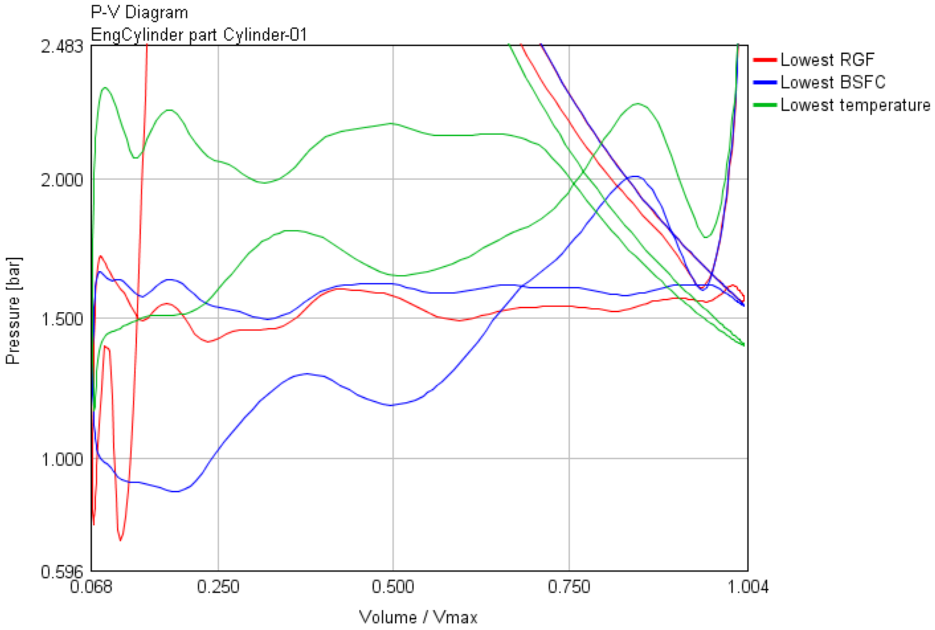

Figure 38. Pump loop PV-Diagram. full load turbo model

8.2 Minimizing of BSFC

To determine the effects of different DEP valve timings on the BSFC a set of DOEs was run with first the simple model and later with the more complete turbo model. These DOEs where run with a load of 4- respectively 20 bar BMEP at 2000 rpm.

8.2.1 Turbo model 4bar 2000 rpm

With the benchmark turbo map the BSFC was further reduced by testing different amounts of intake- and exhaust valve overlaps and exhaust openings, the starting point was the best point from the amount of scavenging DOE. The valve timings for minimum BSFC, 273 g/kWh, is presented in Table 13 and showed as valve curves in Figure 39.

Table 13 Valve timings for optimized BSFC

	[CAD]
Fuel intake opening	345
Flush intake opening	345
Intake closing (PI regulated)	443
Blowdown opening	155
Blowdown closing	380
Scavenge opening	155
Scavenge closing	380

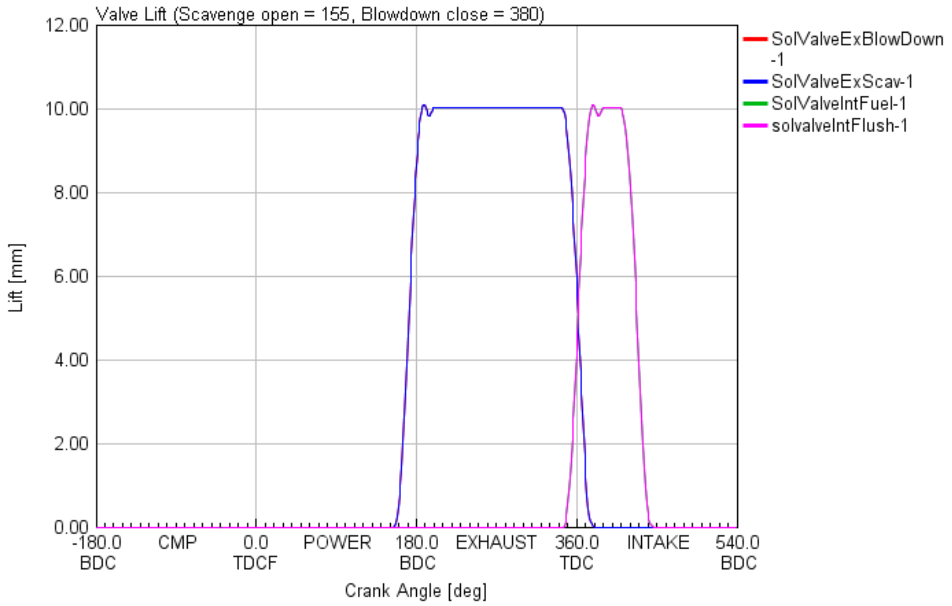

Figure 39 Valve curves for optimized BSFC

The lowest BSFC was achieved by reducing the pumping losses with more efficient valve timings and more suitable intake and exhaust lengths. This can be seen in the Pv-diagram in Figure 40, the pumping loop has been zoomed in. Here it is seen that the work needed to pump the gases in and out of the cylinders is greater in the case from the DOE. As the optimized valve timings results in a pressure closer to ambient during the intake stroke. The shift in pulsation during the exhaust stroke is due to different lengths of the exhaust manifold.

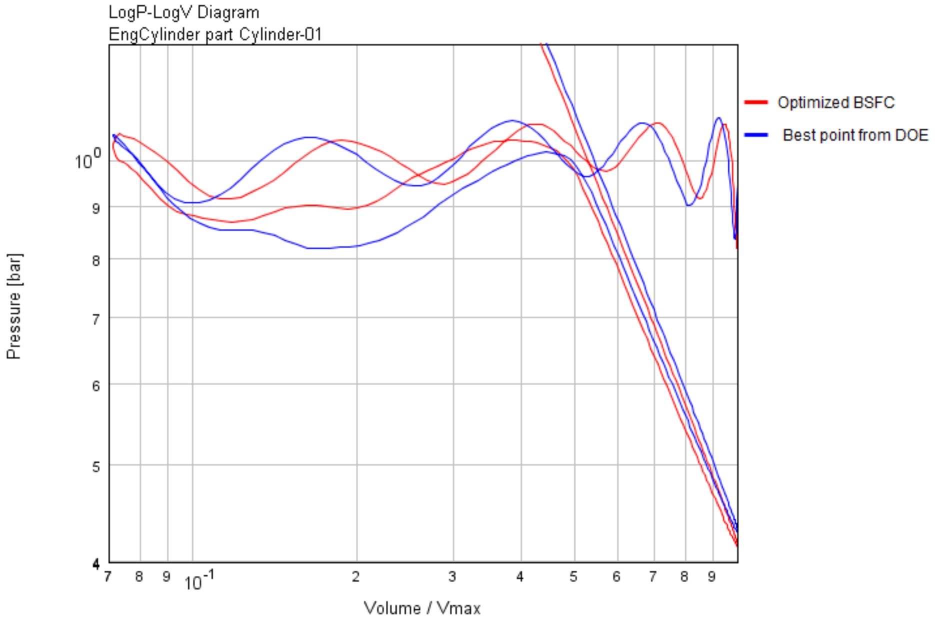

Figure 40 Pumping loop in the Pv-diagram, optimized- and DOE BSFC

8.2.2 Turbo model 20 bar 2000 rpm

When looking at minimizing BSFC the first approach was to use a DOE but after tinkering around with the different variables manually a better BSFC could be achieved. The scaveng-ing point, were the blowdown closes and scavenge valve opens. From the earlier scavenge DOE was chosen and the inlet opening angles, blowdown opening and scavenge closing was changed manually. It was discovered that opening the fuel port later was necessary for mini-mizing BSFC due to fuel otherwise escaping out the exhaust or having to open the intake later and increasing pumping work. An asymmetric opening of the intake valves proved most effi-cient. The resulted timing angles are shown in Table 14. The resulting lowest BSFC was 210 g/kWh with the turbocharger performance map found in GTP

Table 14. Optimal valve timings, full load turbo model

	[CAD]
Fuel intake opening	375
Flush intake opening	360
Intake closing (PI regulated)	501
Blowdown opening	147
Blowdown closing	290
Scavenge opening	250
Scavenge closing	360

The resulted pumping loop is illustrated in Figure 41. It shows a decrease in pressure during the exhaust stroke but the main reason for the decrease in BSFC is because the overlap be-tween the scavenging and fuel port valve opening is eliminated.

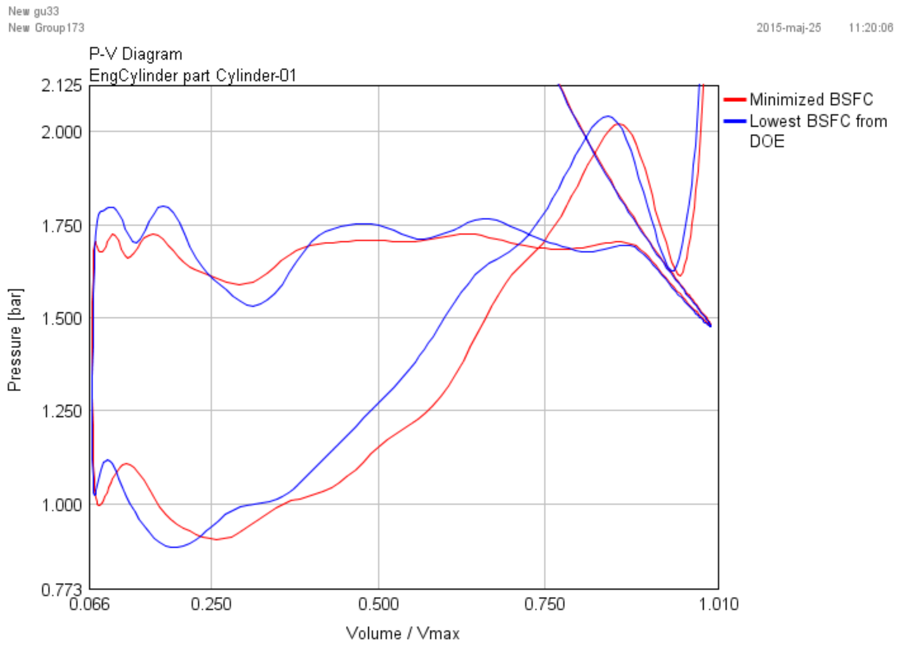

Figure 41. Pump loop PV-Diagram lowest BSFC, full load turbo model

Figure 42 shows the lift curves for the optimal BSFC valve timings. With a overlap between scavenge and blow down of around 40 CAD. The intake with the injector is also opened later to be able to increase the intake and exhaust overlap without letting fuel out the exhaust.

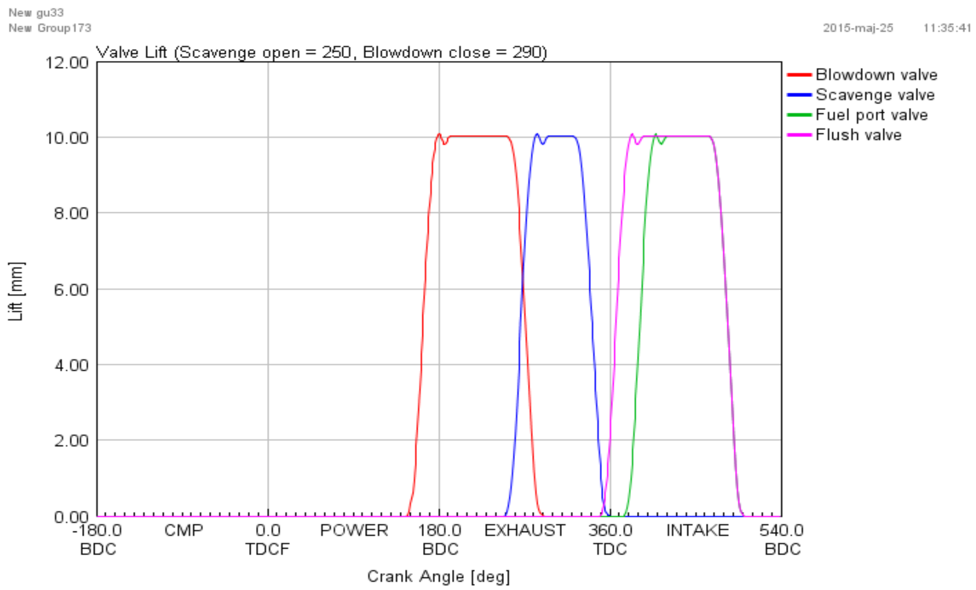

Figure 42. Lift curves for lowest BSFC, full load turbo model

It is known from literature that the DEP concept is likely to have choked flow over the exhaust valves and so is the case for the blowdown valve, as can be seen in Figure 43. This is not good since it indicates that a bigger valve area is needed for the gasses to flow sufficiently. However the choking is not that large and only appears in the beginning of the valve lift, this is common for most engines. Especially for higher loads.

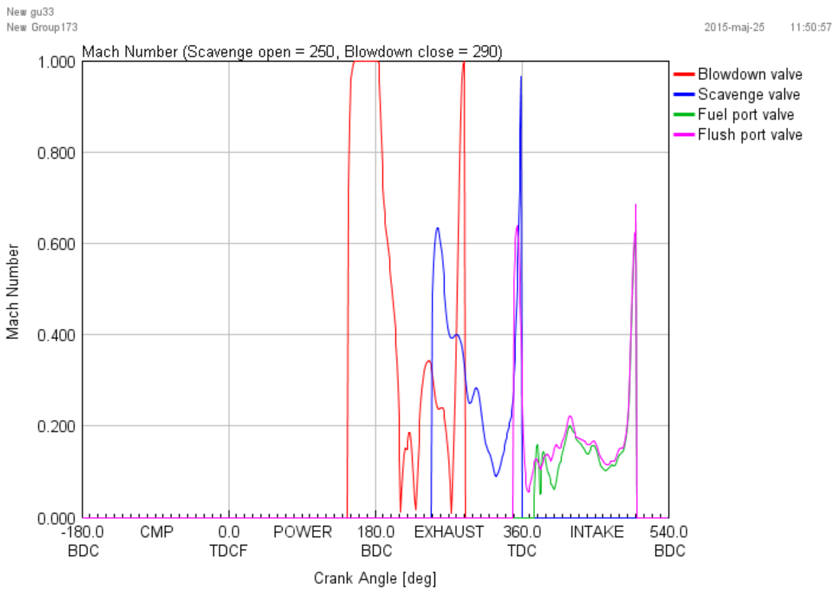

Figure 43. Mach-number over valves, full load turbo model

8.3 Minimizing of RGF and in-cylinder Temperature

When minimizing the residual gases in the cylinder the overlap was varied giving the lowest temperature at an overlap of 40 CAD as can be seen in Figure 44.

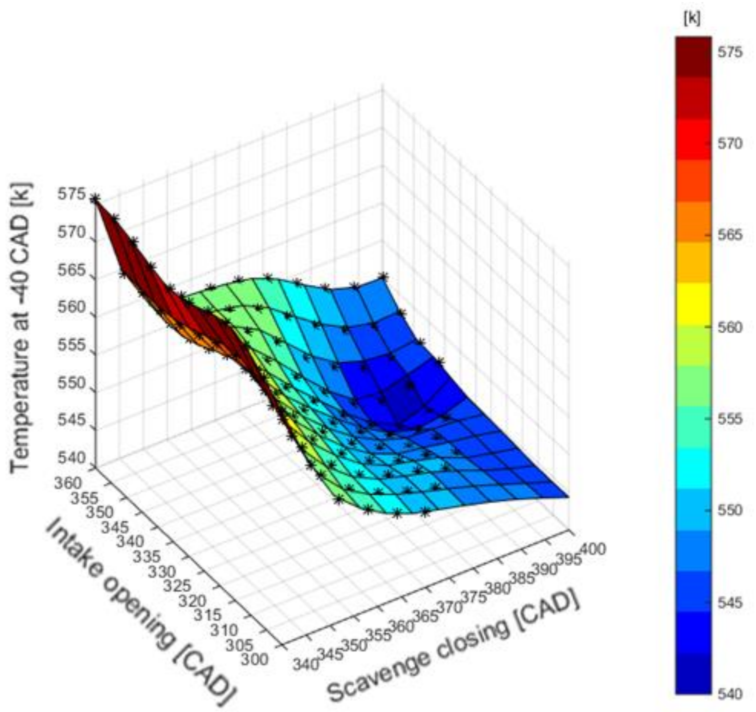

Figure 44. Temperature at 40 CAD before TDCF. Full load turbo model overlap DOE

The burned mass fractions left in the cylinder are as expected lower with a greater overlap as Figure 45 shows. As seen the residuals is almost entirely eliminated with exaggerated overlap between intake and exhaust. But there is also a sweet spot as can be seen toward the back corner of the contour plot.Figure 36

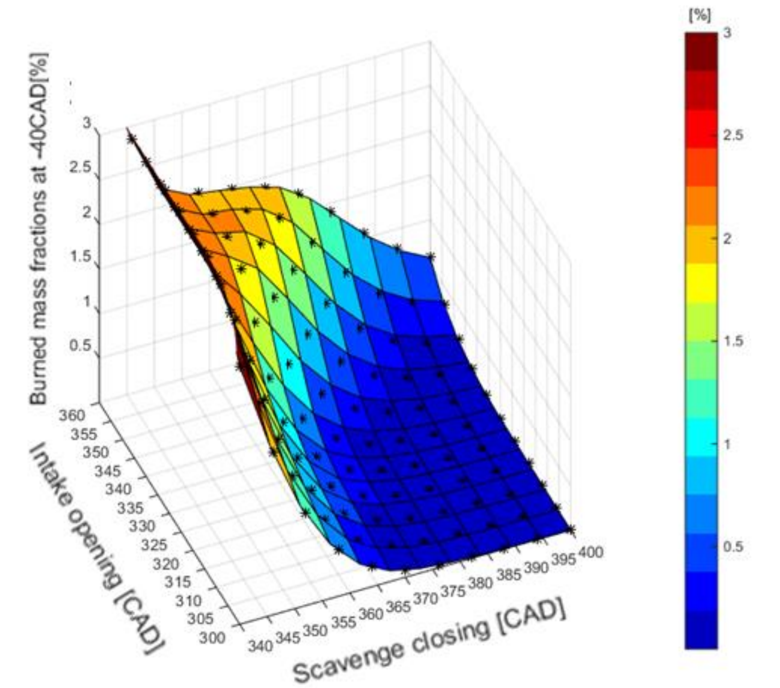

Figure 45. RGF at 40 before TDCF. Full load turbo model overlap DOE

The peak cylinder pressure is not so trivial however as Figure 46 illustrates. The lowest cylinder pressure can be found near the 40 CAD overlap which also has the lowest temperature. The pressure did not vary more than 5 bars or 3%

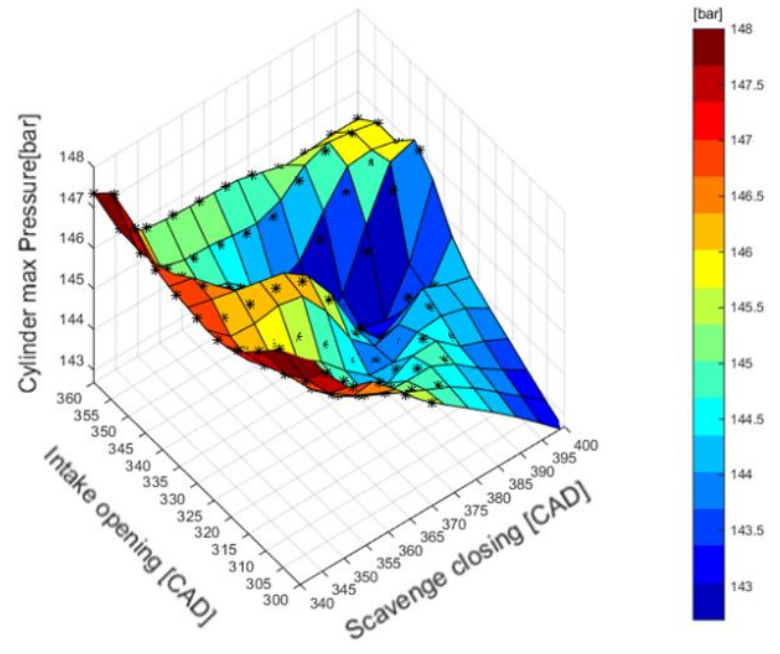

Figure 46. Peak cylinder pressure. Full load turbo model overlap DOE

With a greater overlap more fuel escapes through the exhaust as expected but a compromise could be found near the point with the lowest pressure and temperature to get as much RGF decrease without compromising too much on fuel consumption.

8.4 Intake and Exhaust length

Figure 47 - Figure 49 illustrates the results from the exhaust/intake length DOE

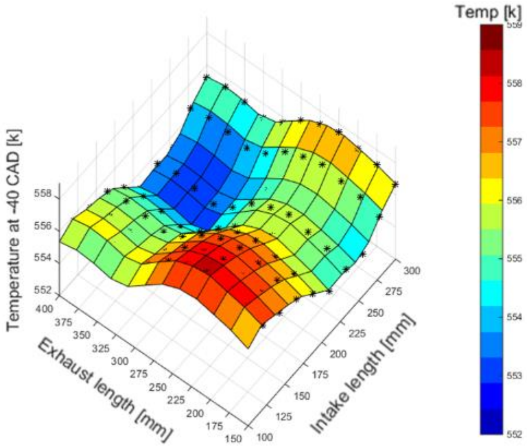

Figure 47. Temperature before TDCF.
Full load turbo model, intake and exhaust length DOE

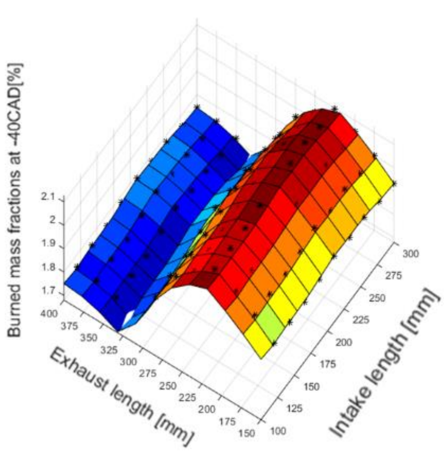

Figure 48. Residuals. Full load turbo model, intake and exhaust length DOE

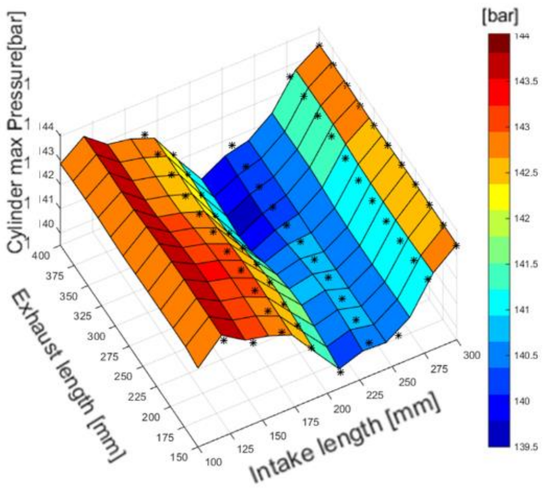

Figure 49. Peak cylinder pressure. Full load turbo model, intake and exhaust length DOE

From the DOE the best compromise between low cylinder temperature and pressure and RGF in the cylinder was chosen and match with the optimal scavenging point and overlap for lowest RGF. The configuration can be seen in Table 15.

Table 15. Lowest RGF. Full load turbo model, intake and exhaust length DOE

	[CAD]
Fuel intake opening	390
Flush intake opening	340
Intake closing (PI regulated)	501
Blowdown opening	147
Blowdown closing	290
Scavenge opening	240
Scavenge closing	390
	[mm]
Intake length	212
Exhaust length	350

The difference in the pumping loop is illustrated in Figure 50. During the exhaust stroke they are very similar but the case with the low RGF suffers from extensive blow by which causes poor cylinder filling. A lower cylinder pressure during intake confirms this.

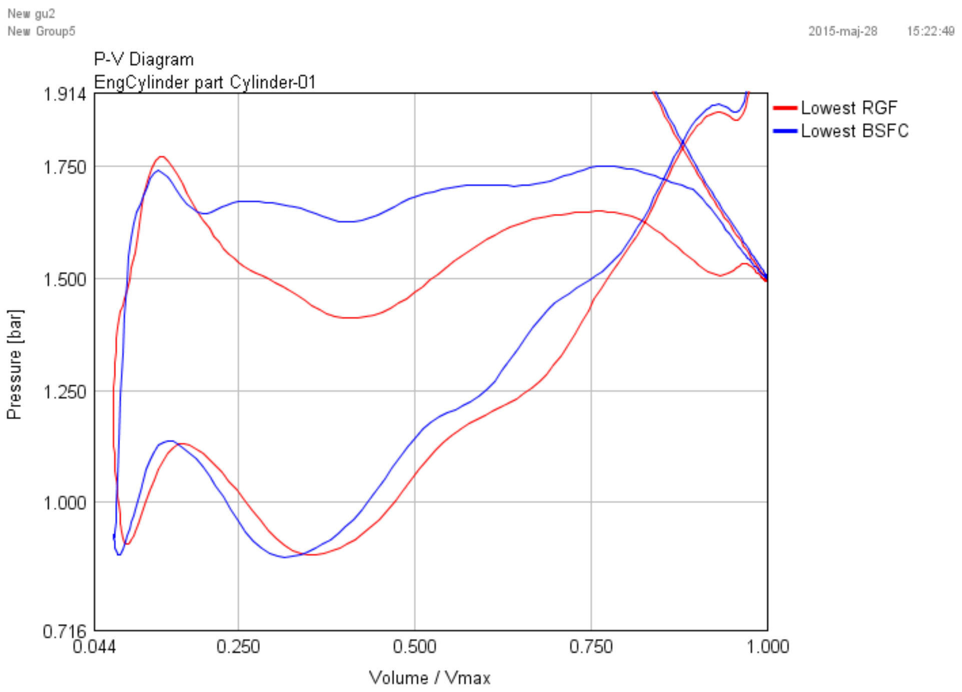

Figure 50. PV-diagram comparison low RGF vs. Low BSFC. full load turbo model

8.5 Internal EGR

By introducing different amounts of internal EGR with the purpose of reducing BSFC at low loads the amount of residuals at cycle start, BSFC and PMEP was analysed. Here only the case with NVO is presented since it showed a more stable control of the amount of residuals and an overall lower BSFC. The results shows a decrease in BSFC with a higher amount of internal EGR, but the BSFC reduction follows the decrease in pumping work with higher accuracy, see Figure 51 - Figure 53. At this realisation a re-check of the characteristics of the used Wiebe-function showed that the combustion is not taking EGR into account. Due to this no conclusion regarding internal EGR can be done. The results are normalized and shown in Figure 54.

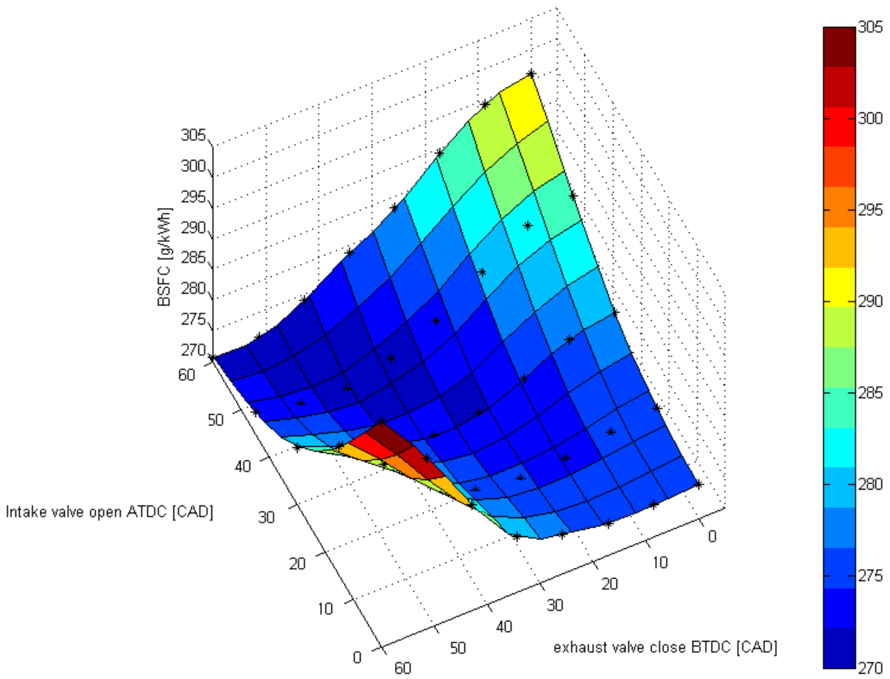

Figure 51 BSFC with varying NVO

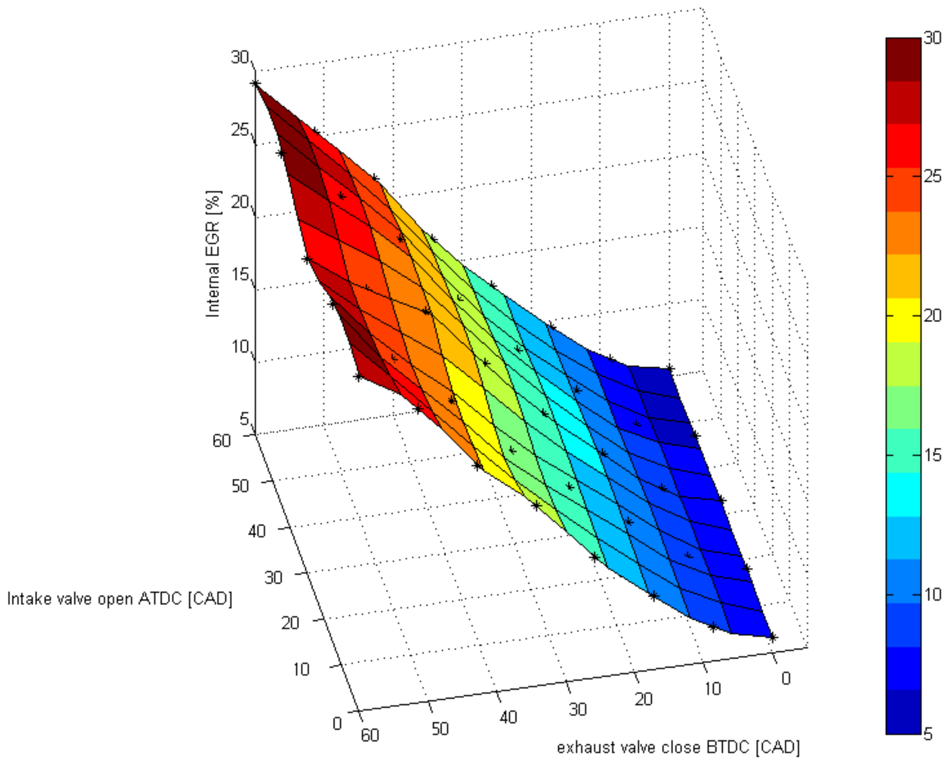

Figure 52 Internal EGR with varying NVO

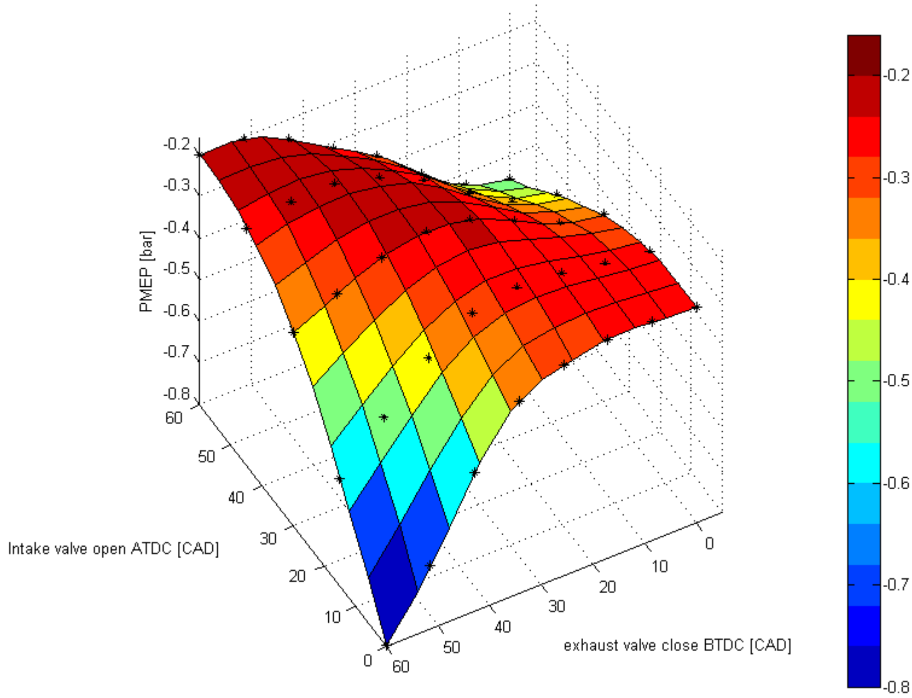

Figure 53 PMEP with varying NVO

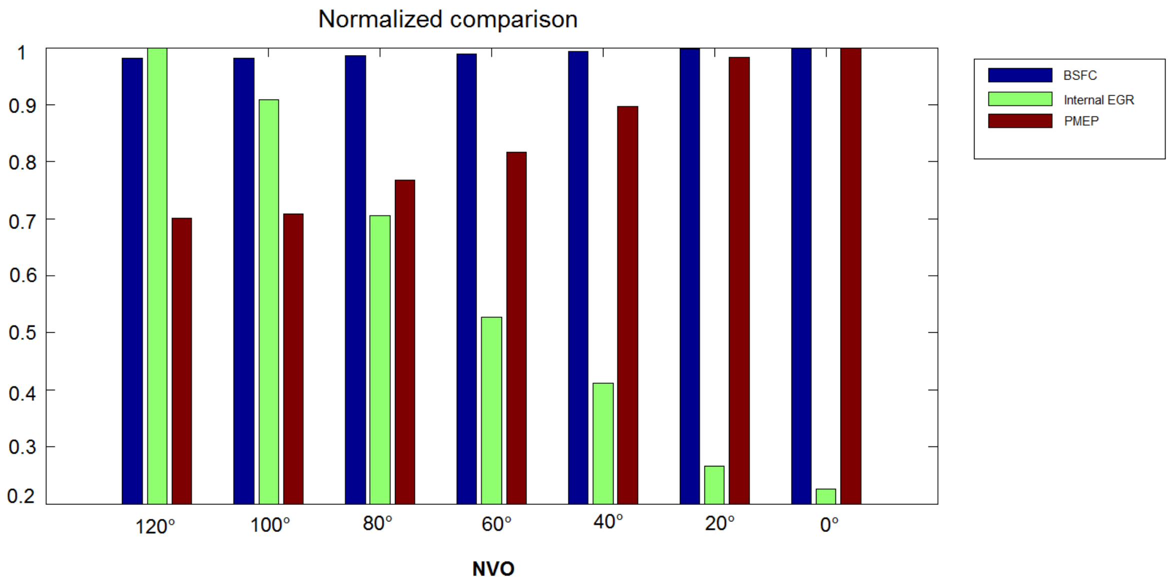

Figure 54 Normalized comparison of PMEP, BSFC and internal EGR for different amounts of NVO

8.6 Comparison with and without DEP

To evaluate where the BSFC reduction compared to the benchmark came from, the model was modified to run without DEP and the CR was varied between 9.5:1 and 14:1. The resulting test was as seen in Table 16 Case comparison with and without DEPTable 16.

Table 16 Case comparison with and without DEP

With DEP	**Without DEP**
CR=14:1	**CR=14:1**
4bar BMEP	4bar BMEP
20bar BMEP	20bar BMEP
CR=9.5:1	**CR=9.5:1**
4bar BMEP	4bar BMEP
20bar BMEP	20bar BMEP

The simulations had valve timings for DEP as in Table 17.

Table 17 Valve timings for DEP concept

	4bar [CAD]	20bar [CAD]
Fuel intake opening	370	365
Flush intake opening	350	355
Intake closing (PI regulated)	443	504@CR14 and 490@CR9.5
Blowdown opening	160	145
Blowdown closing	370	290
Scavenge opening	160	250
Scavenge closing	370	365

Without DEP the valve timings were as presented in Table 18.

Table 18 Valve timings for non-DEP concept

	4bar [CAD]	20bar [CAD]
Fuel intake opening	380	370
Flush intake opening	350	350
Intake closing (PI regulated)	442	518@CR14 and 514@CR9.5
Blowdown opening	160	145
Blowdown closing	380	370
Scavenge opening	160	145
Scavenge closing	380	370

The results are depicted in Figure 55 below. It is seen that the increase in CR gives a higher BSFC reduction than the DEP concept. For the low load this is logical since the DEP concept is not used as much, as the turbo is not a large pressure obstacle. But for the higher loads the use of DEP should in theory show a greater improvement, but the challenge to produce the required boost pressure and thus opening the blowdown valve early. This reduces the expansion work and the overall improvement in BSFC is damaged. It can be summarised as at low loads DEP works but is not needed and at high loads, where it is needed the sufficient valve area is not possible within the cylinder geometry.

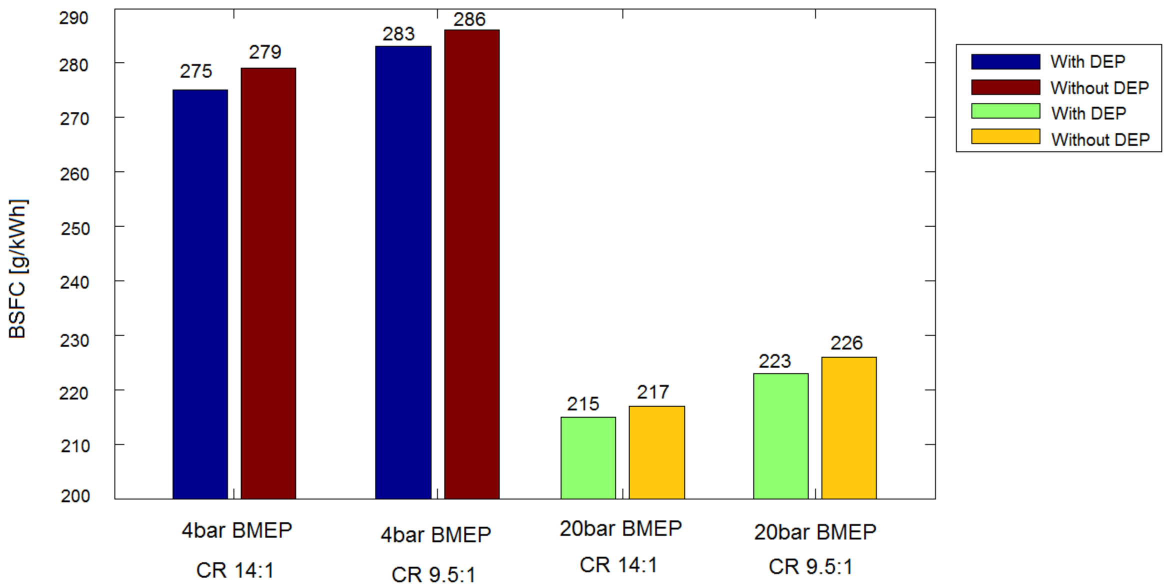

Figure 55 BSFC differences between DEP and non DEP concept with varying CR(14:1 and 9.5:1)

8.7 Comparison with benchmark

The goal with this thesis is to explore any advantages with the DEP and free valves compared to a benchmark engine. Comparison was made of BSFC, RGF and cylinder temperature. The benchmark engine was the same engine but with standard cam lobes, a throttle and a turbo with wastegate. The benchmark engine had a simulated BSFC of 302 g/kWh compared to its experimental value of 299 g/kWh.

8.7.1 Part load

A comparison was made between the simulated benchmark engine and the optimized BSFC with DEP-concept and high CR. The result is seen in Figure 56.

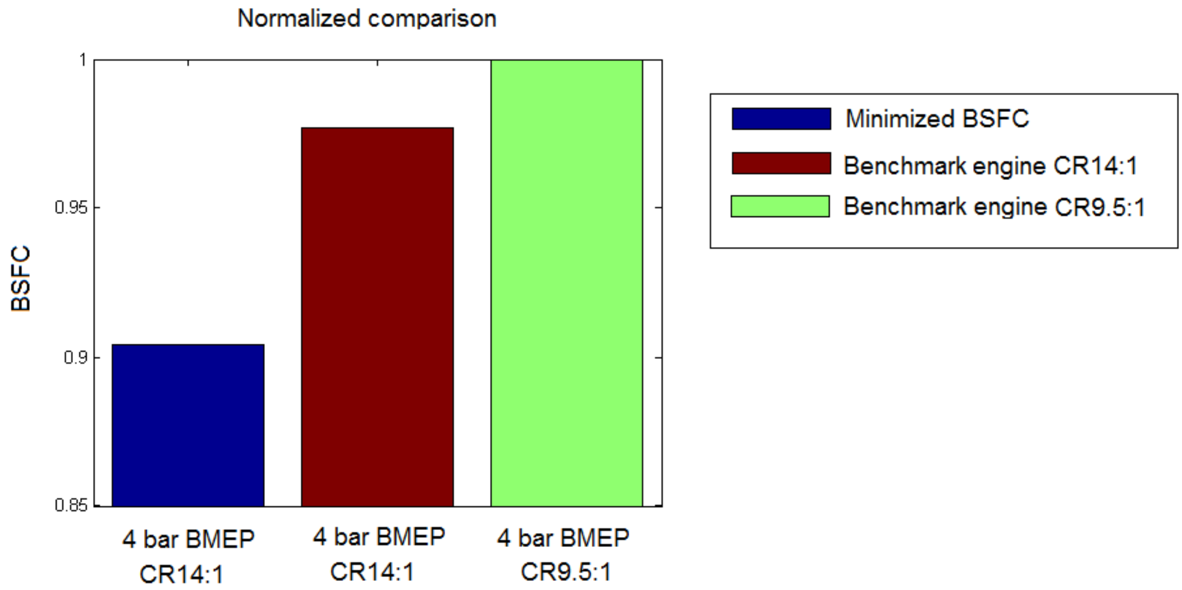

Figure 56 Comparison of the optimized BSFC and the benchmark engine

The large reduction in BSFC compared with the benchmark engine is the reduction in pumping losses, due to the non-throttling featured by using fully variable valves and the application of the Miller-cycle. These can be seen in Figure 57 below. As shown earlier in Figure 55 the DEP concept did not improve the BSFC as much.

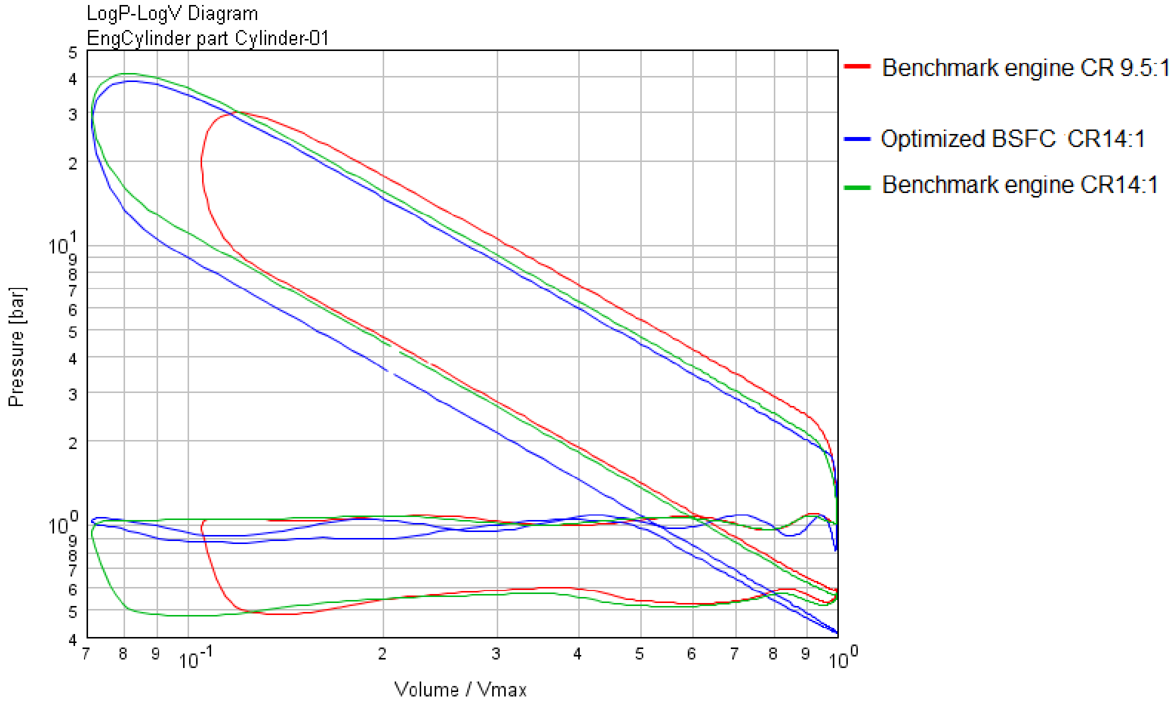

Figure 57 Pv-diagram for benchmark engine and optimized engine with DEP-concept

8.7.2 Full load

At 20 bar BMEP the difference is illustrated in Figure 58. In the figure all the values has been normalized so the standard engine is represented as 100% and the DEP concept as percentage of the standard engine. The results a show decrease in fuel consumption of 5%, a decrease in RGF of 96% (when compromising slightly on fuel consumption).

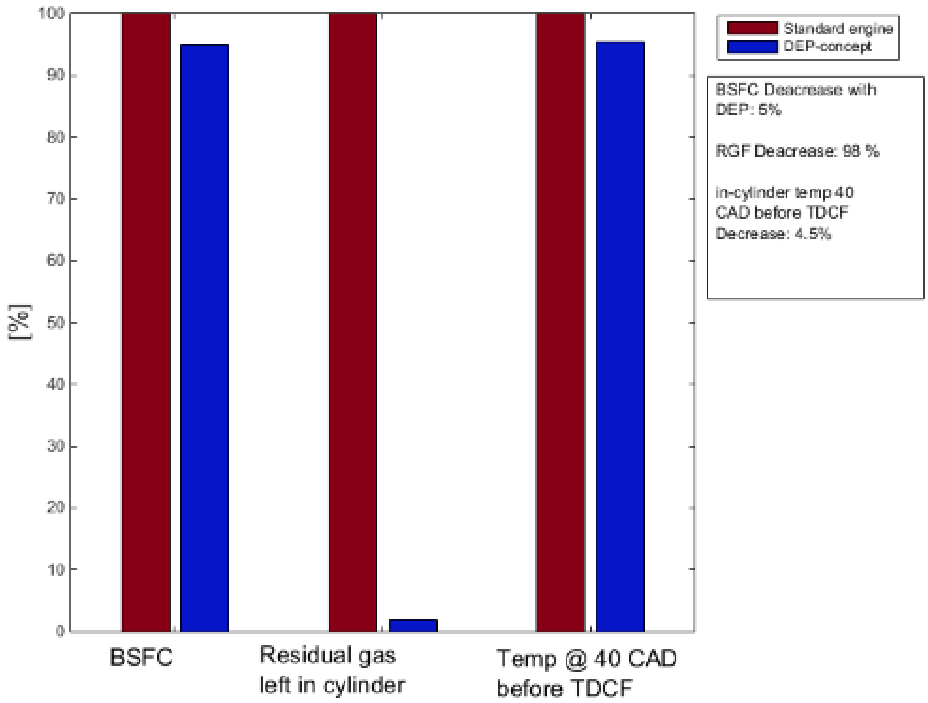

Figure 58. Comparison with benchmark. full load turbo model

To get a better understanding of the differences, a PV-diagram of the two is illustrated in Figure 59. As can be seen the pressure is much lower during the exhaust stroke and thereby causing less pumping losses. The pressure difference is also greater during the scavenging which helps getting rid of unwanted residual gases and thus reducing the knock probability.

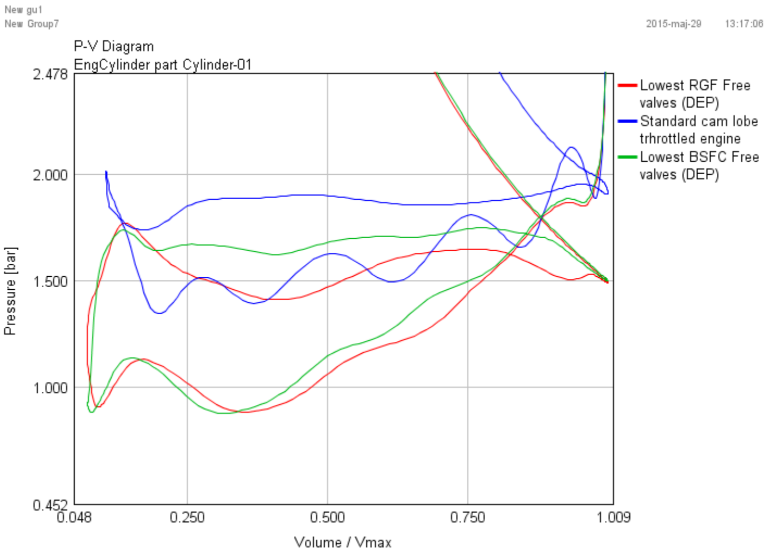

Figure 59. PV Pumping loop comparison with standard engine

9 Discussion

9.1 Part load

At first a simple model was used which gave a good idea of the system, since the boost pressure and backpressure set in the inlet- and outlet boundaries were both very low at part load. This is seen if a comparison is made between Figure 19 and Figure 27. It is already in Figure 18 seen that the pumping work, or PMEP, can differ 0.1 bars between different exhaust to intake overlaps while the DEP overlap is not that important, at least not at part load. After simulating part load condition with the turbo model it was clear that the amount of exhaust gases delivered to the turbine and the amount of bypassed exhausts did not play much role in reducing the BSFC, only a small difference could be seen when bypassing all exhausts or feeding equally to the turbine and bypass. The latter is explained by the fact that feeding the turbine some exhausts will reduce the pressure loss that otherwise exists over the compressor on the inlet side. By spinning the turbo and not dragging it along with the intake air is beneficial for the fuel consumption.

As EGR could help reduce BSFC by improve the evaporation of the fuel as it is injected, internal non-cooled EGR. And thus improving the combustion, this was simulated. Although the simple wiebe-function used for modelling the combustion process in this master thesis does not account for EGR. A reduction in BSFC is however obtained, see Figure 51. This reduction can as stated before not be linked to the increase in internal EGR. But the large similarities between the PMEP in Figure 53, and BSFC results in the different amounts of NVO explains the BSFC reduction. A cam of low pumping losses and hence low BSFC is found on the diagonal, the diagonal represents the line where the NVO is symmetric around TDC. In the case of symmetric NVO the compression work needed before TDC is re-gained as expansion work after TDC, except for the heat losses that will occur during this time.

By comparing the turbo model without the DEP-concept and testing both 14:1 and 9.5:1 in compression ratio the actual benefits with DEP in regards of BSFC is evaluated. As seen in Figure 55 the reduction in BSFC when applying DEP is not that big and the influence of the CR is greater and should probably reduce BSFC even more in reality. This since the imposed burn rate is not affected by the CR, which in reality would speed up the burn rate. At these low loads there is no problem with choking of the exhaust valves as stated earlier for higher loads, but instead the theoretical reduction in PMEP with DEP is at low loads too small to give a noticeable fuel reduction. Instead most of the exhaust gases are needed to spin the turbo and reduce the pressure drop over the compressor.

The optimized DEP-engine setup was further compared to a simulated benchmark engine, the simulated benchmark engine performed a BSFC at 2000 rpm and 4bar load off 302 g/kWh while this engines experimentally tested BSFC was 299 g/kWh. This shows that the model in GTP is fairly accurate, at least for low loads, but it should not be forgotten that an imposed burn rate via the wiebe-function would be better if the burn rates were imposed related to engine speed, -load and amount of residual gasses. The overall reduction in BSFC is seen in Figure 56, where the improvement in BSFC is around 10%. The reduction can be split in to two major parts, CR increase and de-throttling. The CR increase lowered the BSFC in the benchmark engine with around 3% and a further decrease of 7% is possible due to the reduced pumping losses as no throttle is used. Instead the Miller-cycle is applied. This is the main BSFC gain with fully variable valve timings. The pumping loop is shown in Figure 57 and the increased pressure during the intake stroke is clearly visible.

Another way of reducing the BSFC which has not been tested within this master thesis is cylinder deactivation. This will reduce the BSFC through a kind of dynamic downsizing by deactivating the injectors and fully close the valves. With the right amount of air trapped in the cylinder the heat losses could be minimized.

9.2 Full load

The simple model without the turbocharger gave a notion of how the system responds at full load but did not tell the whole story. The problem was that the boost pressure was not affected by the amount of scavenging since the inlet pressure was imposed. This meant opening the scavenging port earlier reduced the pumping losses, but there was no penalty for not supplying mass flow towards the turbine. The simple model did however confirm at an early stage that negative DEP overlap was a bad thing to have, as shown in Figure 21 and Figure 22.

When running the DOE of the scavenging, the simulation was not able to reach the load target that was imposed in some of the cases. Some measures was made to be able to reach higher loads at 2000 rpm but this resulted in having to close the intake valves earlier at the points with later scavenging. A better approach to this would have been to use a bigger turbocharger, optimized for a specific amount of scavenging and hence a more efficient load point for the turbocharger. Another approach could be to regulate the scavenging in parallel with the intake closing to be able to actively control the turbo performance and reduce pumping losses, kind of like variable turbine geometry. To really analyse the result considerations have to be made of the intake closing since this affects the result dramatically (see Figure 31). It could be argued that closing the intake valves later and regulate the load with earlier scavenging instead would have favoured the PMEP and resulted in higher efficiency. But it would also mean that less exhaust energy was recovered by the turbine. The first part of the exhaust contains most of the energy, while the last part contains a smaller amount of energy and is usually displaced by the piston. This part could preferably be wasted through the scavenge port with lower pressure.

The scavenging DOE also revealed a boundary with lower in-cylinder temperature at a scavenge opening at 306 CAD. The explanation is that the pressure difference between the exhaust and cylinder becomes greater with the pressure build up in the cylinder caused by the negative DEP overlap. The fact that there is a boundary when this happens implies that here are some dynamic effects and pulses could be seen in pressure in Figure 38.

The highest improvement of the BSFC came from opening the intake valves asymmetrically, with the port which contained the injector opening later. This improvement came from not wasting fuel that otherwise would blow by the cylinder and out the exhaust during the overlap. The improvement was expected to be higher at this point with DEP. One reasons for this could be the choked flow past the blowdown valve, seen in Figure 43. The exhaust flow normally experiences some amount of choking but with the DEP this condition was aggravated even though the valve diameter was increased to prevent it. Most part of the BSFC improvement in this thesis seems to be the result of increasing the CR and applying free valves. The CR increase would however maybe not be possible if DEP was not implemented, due to potential knock limitations.

It would have been interesting at this point to examine the impact that EGR would have had on fuel consumption. This was not possible since the simple combustion model used meant the burn rate was imposed and thus not affected by EGR. If there would have been experimental result available it would have been interesting to look at dedicated EGR from one cylinder or from the scavenging valves.

The biggest improvement from using DEP was the reduction of residual gases in the cylinder at 40 CAD before TDCF. The results however, showed that a compromise between really low RGF and fuel consumption had to be made. Generally the fuel consumption is less of concern at high loads and thus not the focus in this part of the thesis.

The result showed a very high peak cylinder pressure and to lower this pressure, measures like retarding the ignition could be used. Retarding the ignition was not possible in this combustion model however and would also decrease the efficiency. The case with the lowest BSFC had 60% less RGF than the standard engine, which is quite the improvement.

The result in the exhaust length DOE showed that the exhaust length has major influence on removal of RGF but not so much on cylinder temperature and pressure. The temperature and pressure could be affected by the intake length. The results showed a sweet spot considering RGF when the exhaust was around 350 mm long, where there was a negative acoustic pulse. This could however raise some issues with packaging in a production car where you usually want to keep them short to decrease the temperature loss between the port and the turbo but also to make a smaller package. To get a lower cylinder pressure and temperature the goal was to get a negative expansion pulse just before closing the intake valve, which would cool the charge. This was found with the length of the intake kept short.

The DEP proved very useful of reducing the RGF and not so efficient of decreasing pump losses, as expected, but then again the increase of CR may only be possible with the reduction of RGF. All the cases in this thesis were run with a lambda of 1 and this is far from possible with the standard engine at full load with a CR of 14. It is however more plausible with DEP and the use of free valves.

10 Conclusions

At low loads the gain of using DEP is very limited as shown by the simulations, but the implementation of fully variable valves has a potential for substantial fuel savings through throttle less operation and Miller-cycle. The increase in compression ratio is also confirmed to increase fuel efficiency. These results is captured with both a very simple model and the more advanced turbo model, so for more extensive simulations of the acoustic length of the intake- and exhaust pipes the simple model can be used to reduce the simulation time. Although this high increase in CR will cause problems with knock at higher loads, this is where DEP comes into play. When applied correctly the residual gases and hence the in-cylinder temperature before ignition can be lowered. The same can be achieved with acoustic tuning of the intake lengths in addition. The DEP helped decrease the residuals by over 90% which could be crucial for running with higher CR but it did not give the expected decrease in pumping losses. The results also showed a benefit of having 30-40 CAD overlap between blowdown and scavenge with scavenging as early as possible, without compromising turbocharger performance and load.

For part load the BSFC has potential to be lowered even further by introducing internal EGR though NVO. This is not captured by the combustion model, but it can be reasoned that the BSFC would be reduced with some internal EGR. By the fact that the rate of combustion would be increased with some hot residuals in the cylinder, to help the fuel evaporate.

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13 Appendix

13.1 MATLAB plot routine

```
function [ BMEPplot ] = Contourplot(
xmax,xmin,ymax,ymin,zmin,zmax,xtick,ytick,Zlabel,FileName,BMEPMin,BMEPfilen
ame)
M=dlmread(FileName);
% Turn the scanned point data into a surface 12.8571
figure
hold on
gx=xmin:12:xmax;
gy=ymin:12:ymax;
g=gridfit(M(:,1),M(:,2),M(:,3),gx,gy);
colormap(jet(16));
surf(gx,gy,g);

axis([xmin xmax ymin ymax zmin zmax zmin zmax])
colorbar
xlabel('Blowdown close [CAD]')
ylabel('Scavenge open [CAD]')
zlabel(Zlabel)
% set(gca,'DataAspectRatio', [12 12 1])
set(gca,'XTick', xtick)
set(gca,'YTick', ytick)
B=dlmread(BMEPfilename);
BMEPplot=[];
for i=1:length(M)
       if B(i,3)>=BMEPMin
          BMEPplot(i,:)=M(i,:);

       end
end
plot3(M(:,1),M(:,2),M(:,3),'*','Color',[0,0,0])
grid on
hold off
end
```

13.2 Extra results

When using two exhaust valves the DOE was set to vary the amount of exhausts passing by and through the turbine, the results are seen in Figure 60 and Figure 61.

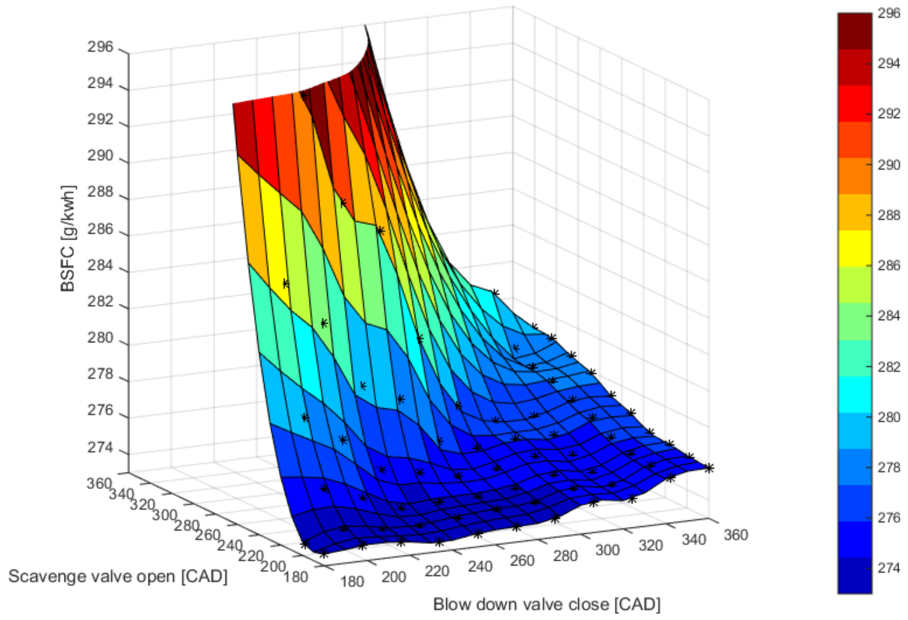

Figure 60 BSFC for 1-2 valve setup, where the amount of bypassing is examined

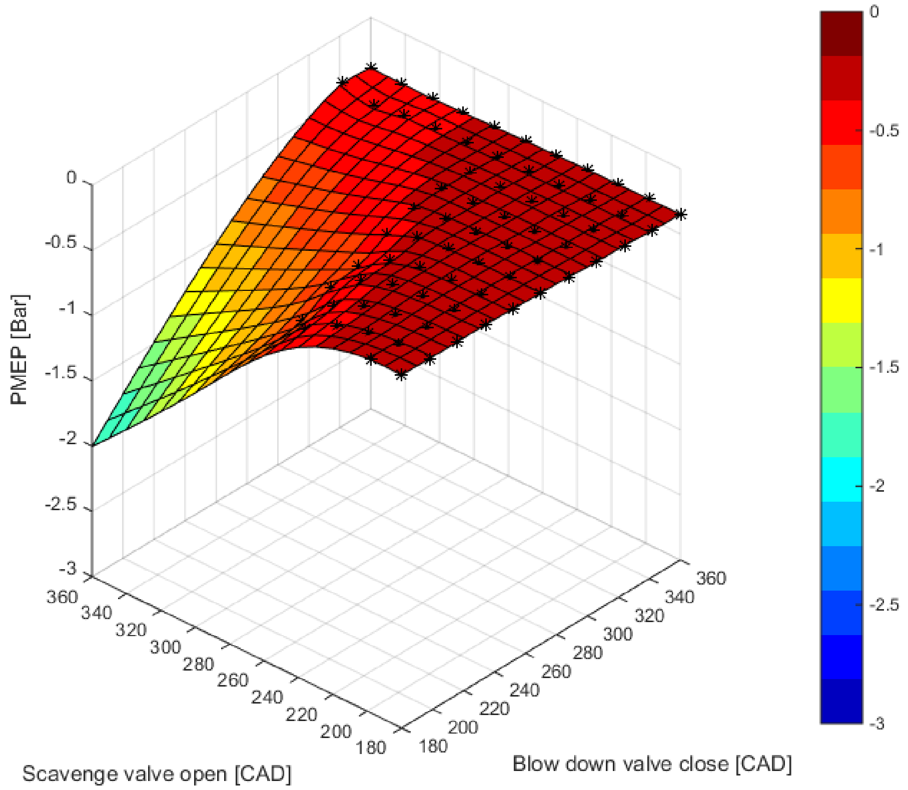

Figure 61 PMEP for 1-2 valve setup, where the amount of bypassing is examined

For the varying of overlap with two intake valves and one exhaust valve see Figure 62 Figure 63.

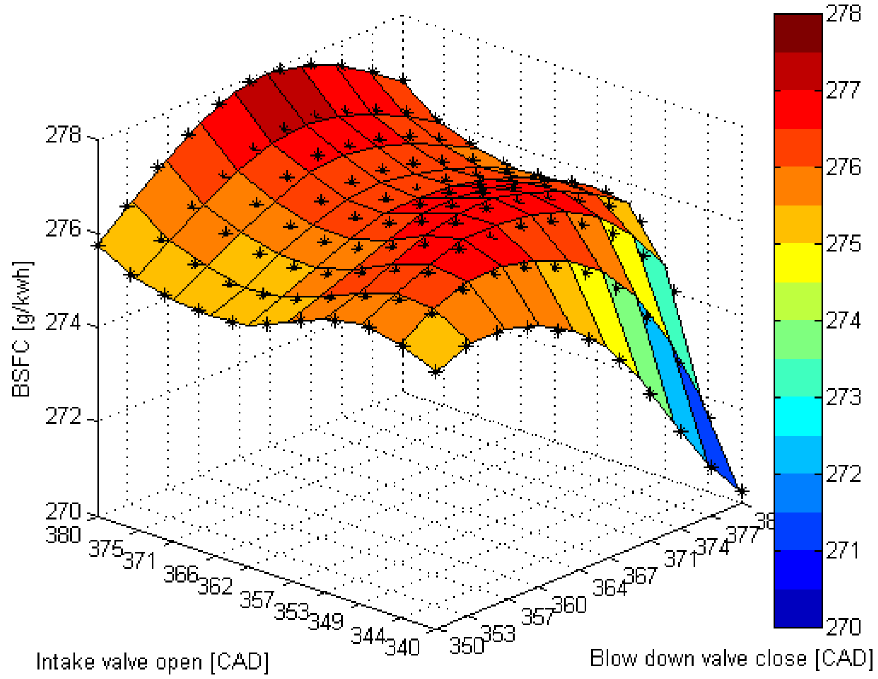

Figure 62 BSFC for 2-1 valve setup with TDC overlap variation

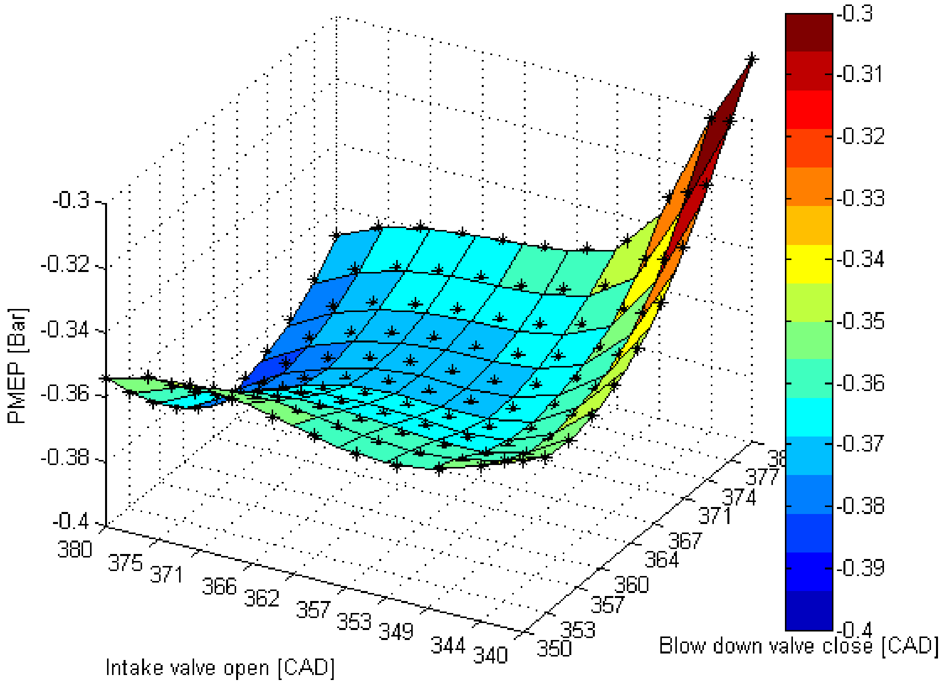

Figure 63 PMEP for 2-1 valve setup with TDC overlap variation