Influence of turbulent jet ignition and variable valve lift technology on the performance of spark ignition engine

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BACHELOR'S THESIS

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Zagreb, 2023.

I declare that I wrote this thesis independently, using the knowledge I gained through my studies and from the citated literature.

I would like to thank Davide Villa for selflessly guiding me through the whole process of creating the simulations, analyzing the results and writing this thesis, Domagoj Šanić for his useful tips and my mentor Momir Sjerić for helping me complete this thesis.

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ZAVRŠNI ZADATAK

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Naslov rada na hrvatskom jeziku: **Utjecaj primjene pretkomore i varijabilnog otvaranja ventila na značajke rada Ottovog motora**

Naslov rada na engleskom jeziku: **Influence of turbulent jet ignition and variable valve lift technology on the performance of spark ignition engine**

Opis zadatka:

The application of modern engine technologies can improve fuel consumption and tailpipe emissions. The addition of pre-chamber in a spark ignition engines enables the combustion of lean mixtures in the main chamber due to higher ignition energy achieved by turbulent jets that flow from the pre-chamber. Maintaining the same power target, TJI adoption will allow to reduce exhaust temperature. On the other hand, the modification of valve lift profile can improve the engine volumetric efficiency at full load conditions, while the decrease of valve lift at part load conditions can improve fuel economy due to reduced pumping losses. Within this thesis it is necessary to perform simulations for steady state points in a commercial software GT Power™ to investigate the application of turbulent jet ignition and variable valve lift technology on SI engine performance. It is necessary to perform the following within the thesis:

- analyze the influence of turbulent jet ignition (TJI) on engine brake power at wide open throttle (WOT) conditions, on exhaust gas temperatures and brake thermal efficiency,
- simulate the engine performance with increased compression ratio when TJI is applied,
- analyze the most suitable valve profile for peak power and peak torque regions at WOT conditions,
- analyze the most suitable exhaust valve profile to maximize expansion work at part load conditions,
- calculate the decrease of pumping losses with the modification of intake valve lift profile at part load conditions,
- show and analyze achieved simulation results,
- present conclusions.

It is necessary to list all cited literature and received assistance.

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LIST OF ABBREVIATIONS

Abbreviation	**Description**
10-90MFB	Duration in crank angle degrees from the 10% fuel burned point to the 90% burned point
50MFB/AI50	Crank angle at which 50% of the total fuel in the cylinder has been consumed by combustion
BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Speciffic Fuel Consumption
CAD	Crank Angle Degrees
CFD	Computational Fluid Dynamics
CR	Compression Ratio
DI	Direct Injection
DoE	Design of Experiments
EIVC	Early Intake Valve Close
EVC	Exhaust Valve Close
EVO	Exhaust Valve Open
GDI	Gasoline Direct Injection
GMEP	Gross Mean Effective Pressure
ICE	Internal Combustion Engine
IMEP	Indicated Mean effective Pressure
IVC	Intake Valve Close
IVO	Intake Valve Open
NOx	Nitrogen oxides
OEM	Original Equipment Manufacturer
PFI	Port Fuel Injection
PMEP	Pumping Mean Effective Pressure
RPM	Rounds Per Minute
SA	Spark Advance
SI	Spark Ignition
TDC	Top Dead Center
TDCF	Top Dead Center Fire
TJI	Turbulent Jet Ignition
UMFKO	Unburned (fuel) Mass Fraction at Knock Onset
VVL	Variable Valve Lift
WOT	Wide-Open Throttle

SUMMARY

Turbulent Jet Ignition and Variable Valve Lift are relatively new technologies for improving the performance of internal combustion engines. Over the last years they have been a topic for many scientific researches. In this paper, the claimed performance gains are tested using a commercial simulation software GT-Power^{TM}. The benefits of those technologies are compared against the base engine model, created and calibrated using the experimental data from the test bench.

The TJI is evaluated by modifying the base engine model, where assumed benefits of using the system will be imposed. Firstly, brake torque and power are simulated and compared to the base model. Following that, the compression ratio of the engine is increased to further examine the usage of the prechamber combustion. The influence on the exhaust temperatures is also inspected and the found improvements are exploited by increasing the access air ratio.

VVL is examined by modifying the base model intake and exhaust valve profiles for different steady state points. The potential improvements in brake torque and power are examined at wide-open throttle conditions and again compared to the base model. For part load conditions, the reducement of pumping losses with the increasement of expansion work is targeted.

Key words: Spark Ignition Engine, Turbulent Jet Ignition, Prechamber Ignition, Variable Valve Lift, Engine simulations, GT-Power^{TM}

SAŽETAK

Izgaranje u pretkomori i varijabilno podizanje ventila relativno su nove tehnologije za poboljšanje značajki motora s unutarnjim izgaranjem. Proteklih godina bile su predmet istraživanja mnogih znanstvenih radova. Pretpostavke o poboljšanju značajki rada motora biti će ispitane koristeći komercijalni programski paket za simulacije GT-Power^{TM}. Prednosti spomenutih tehnologija uspoređene su sa osnovnim simulacijskim modelom motora koji je izrađen i kalibriran prema podacima s ispitne stanice.

Izgaranje u pretkomori ispitano je izmjenom početnog modela, gdje su pretpostavke o uporabi pretkomore određene parametrima. Najprije su ispitani i uspoređeni efektivni moment i snaga koji su dobiveni simulacijama početnog i izmjenjenog modela. Nakon toga, kompresijski omjer motora biti će uvećan kako bi se nadalje promatrali učinci korištenja pretkomore. Utjecaj na temperaturu ispušnih plinova također je promatran te su poboljšanja primjenjena povećanjem pretička zraka.

Varijabilno podizanje ventila ispitano je izmjenom profila podizaja usisnog i ispušnog ventila za različite radne točke. Poboljšanje iznosa efektivnog momenta i snage ispitano je u uvjetima punog opterećenja te također uspoređeno s osnovnim modelom. Za uvjete djelomičnog opterećenja cilj je smanjenje gibitaka tijekom izmjene radnog medija te povećanje rada prilikom ekspanzije.

Ključne riječi: Ottov motor, izgaranje u predkomori, varijabilno podizanje ventila, simulacije motora, GT-Power^{TM}

1. INTRODUCTION

The internal combustion engine is the most widely used power source for transportation and is also used in various industrial and power generation applications. The spark ignition (SI) engine is the most common type of internal combustion engine used in passenger vehicles due to its high-power output, low weight, and a simple, known design.

Looking at the recent years of engine development there has been a big focus on improving the performance of the internal combustion engines. New technologies are being developed, largely due to new emission standards and manufacturers trying to get ahead of the competition. Many of these new technologies are still in the development phase but some are trying to find their place in a series production vehicles.

The main challenges in designing SI engines are achieving efficient and knock-resistant combustion while providing high performance as well as keeping up with strict engine-out emission regulations.

One of the ways to improve combustion and efficiency in SI engines is using turbulent jet ignition (TJI). TJI uses a conventional electrical spark system to ignite a given amount of air-fuel mixture in a dedicated volume (pre-chamber) connected to the combustion chamber (main chamber) through a set of orifices, leading to faster and more homogeneous combustion. This results in improved knock resistance, reduced emissions, increased brake efficiency and opportunity for torque and power gains.

On the other hand, variable valve lift (VVL) technology is a mechanism that allows the valves in an internal combustion engine to change the amount of lift they provide during the intake and exhaust cycles depending on the operating conditions. This can be used to improve the performance of internal combustion engines by increasing the amount of air and fuel delivered to the combustion chamber and thus improving power and torque. Also, it can be used to decrease pumping losses and increase brake efficiency of the engine at part load conditions.

2. TURBULENT JET IGNITION

Turbulent Jet Ignition is a combustion enhancement technique that has gained significant attention in recent years for its potential to improve the performance and efficiency of internal combustion engines. This system, in general, creates a controlled combustion in a small volume located outside the main chamber (the prechamber) after which hot turbulent jets of partially combusted products initiate the combustion in the main chamber. This results in faster burn durations, increased thermal efficiency and performance at wide open throttle conditions. Also, this faster combustion is also favourable to mitigate the knock tendency, allowing an increasing compression ratio to further improve the engine thermal efficiency.
There are two principles of creating a turbulent jet ignition system:

2.1. Active prechamber

The active prechamber system uses an auxiliary fuel supply in the prechamber as well as the spark plug. The air-fuel mixture is precisely controlled in the prechamber so that it creates a stochiometric charge that can be easily ignited using a common spark plug. The main advantage of this system is the ability to have ultra-lean mixture in the main chamber ($\lambda>2$) that can be ignited with hot turbulent jets from the prechamber. That sort of lean combustion is expected to have very high thermal efficiency (greater than 42%) at specific conditions together with near zero engine out NOx emissions due to reduced combustion temperatures. There are also claims of going above 45% indicated net thermal efficiency with a CR increase to ~14 and a wall guided DI system [1].
However, the extra cost of the double injection system compromises its application to automotive applications, especially in passenger cars, in which the packaging and manufacturing costs are critical. Also, there are problems with the operation of a standard three-way-catalysts since they require a stochiometric operating conditions. Using ultra lean mixture as mentioned above would require significant change in exhaust aftertreatment systems.
For these reasons, the passive concept arises as a promising solution to overcome these limitations and will only be considered in further chapters.

2.2. Passive prechamber

The passive prechamber system doesn't use an auxiliary fuel injector, instead the fuelling is done from the main chamber during the compression stroke. In port fuel injection (PFI), engines air-to-fuel ratio is essentially equal in both chambers, while in direct injection (GDI) engines air-to-fuel ratio differences between chambers can be indirectly generated by adjusting the pre-chamber and injector relative spatial position up to some extent.

The key advantage of passive systems is its mechanical simplicity as the pre-chamber can be directly assembled into a conventional spark plug body, making the implementation straightforward in terms of packaging and costs. The three-way-catalysts can also maintain its normal operation keeping the engine out emissions under control.

Fuelling the passive prechamber is a known disadvantage of the system and will require more research, since the flow of air-fuel mixture through the orifices and the narrow channel of the prechamber is hard to predict. Also, the current systems are not capable of improving combustion at low engine loads. In fact, using solely pre-chamber ignition can lead to reduced engine performance and higher emissions in this operating range.

One of the leading automotive manufactures that have included this kind of technology in their engine development is *Maserati*.

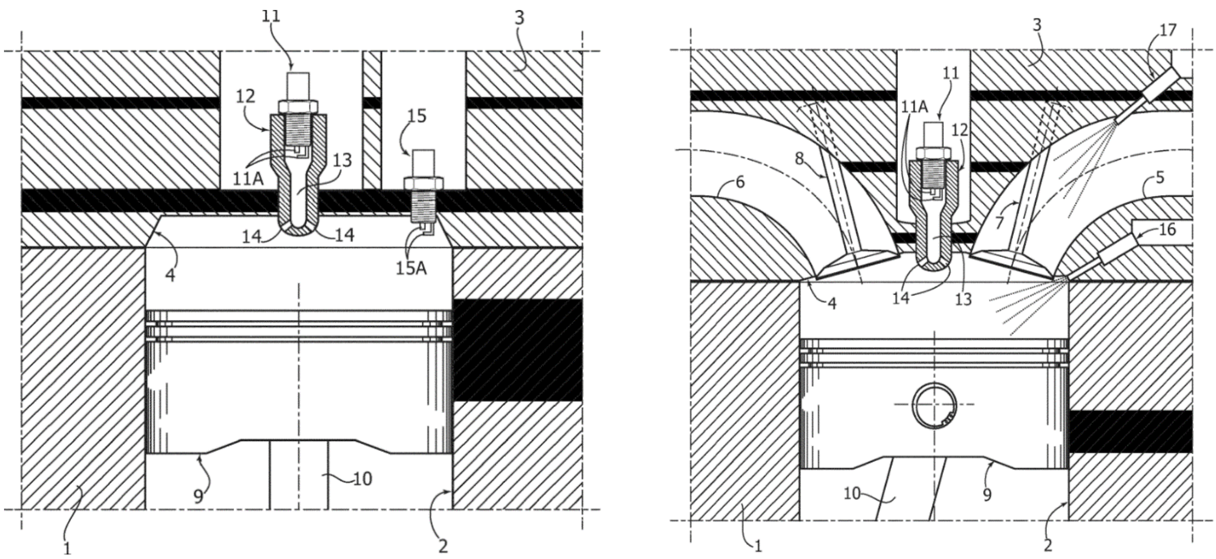

Figure 2.1. Maserati TJI system [2]

The figure 2.1. shows two views of the combustion chamber. The section on the left shows the position of the prechamber (13) with the integrated main spark plug (11). The second spark plug (15) is initiating the combustion at low and medium engine loads in order to reduce the combustion rate and allow normal operation at said conditions.

The side view reveals the fuel supply to the cylinder. Direct injection (16) allows reduction of charge temperature that further improves detonation resistance at high loads, while port-fuel injection (17) allows better preparation of the charge, ensuring near stochiometric conditions at all locations in the cylinder.

3. SIMULATION TJI

GT-Power^{TM} is the industry standard engine performance simulation software, used by all major engine manufacturers and vehicle OEMs. It is used to predict all kinds of engine performance quantities using its multiphysics computations.

The starting simulation model is based on the experimental engine data from the test bench. That means that the combustion will be imposed, rather than modelling it with a standard Vibe heat release function. This is done by defining a non-predictive combustion profile (a combustion model where the burn rate is directly imposed as a simulation input) and therefore does not depend on variables such as residual fraction or cylinder pressure. The fuel and air will simply burn at the prescribed rate. This will in turn reproduce a measured cylinder pressure trace. Both the combustion profile and combustion start are unique for each cylinder and vary depending on the engine speed.

The imposed combustion rate for one cylinder at maximum engine speed is shown in Figure 3.1.

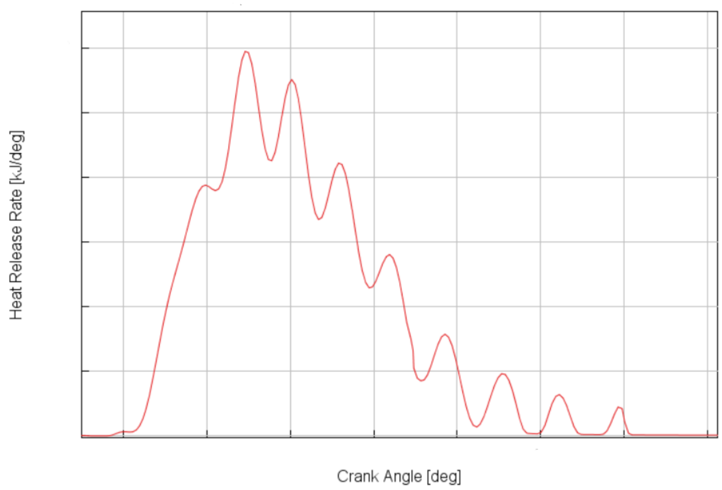

Figure 3.1. Imposed combustion rate

Figure 3.2. presents how this simulation model compares with the experimental data by comparing the average maximum in-cylinder pressures and pressure traces. The absolute difference in simulation and experimental data is 4%.

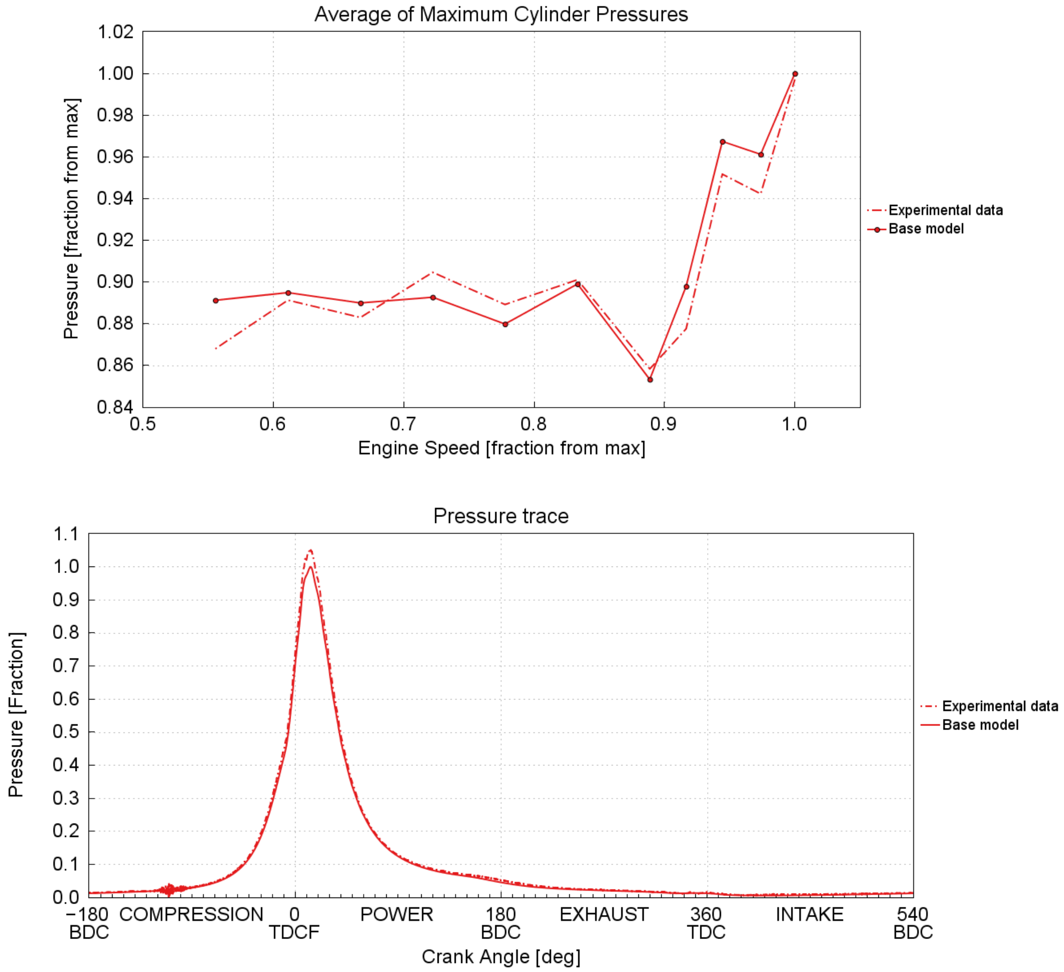

Figure 3.2. A comparison between experimental and simulated data

In this paper, only the influence on combustion by using the prechamber will be considered. Therefore, no specific prechamber design will be inspected since it would require 1D and 3D CFD simulations that are not in the scope of this thesis.

The possible modifications of the base model to act as if the prechamber is used were examined by reviewing the literature [3][4]. The conclusions were drawn by analysing the operating conditions of the experimental/simulation engines from the literature and comparing the engine geometry and associated systems.

By using a passive prechamber, the combustion duration (10-90MFB) can be shortened by at least 20% at WOT conditions. Also, from research [3] it was concluded that 50% of burned fuel point (50MFB/AI50) will be advanced by at least 4° compared to base combustion model.

The knock occurrence will be monitored by examining how much of the fuel mass fraction remains unburned at knock onset (UMFKO). The simulation uses Douaud&Eyzat Single-Zone knock model. The prediction of knock in GT Power^{TM} is based on empirical induction time correlations. The induction time integral, when written in terms of crank angle, depends on engine speed, fuel octane number, cylinder pressure and unburned gas temperature. For the Douaud&Eyzat model, knock is predicted to occur at the crank angle at which the induction time integral attains a value of 1.0.

The Single-Zone parameter means that the induction time integral will be calculated using the bulk unburned gas temperature, rather that creating multiple zones through the cylinder and taking into account the cylinder wall temperature for the zones in contact with it.

Since the fuel has not been characterized for this simulation, the base model knock limit can be different from what's predicted by the Douaud&Eyzat model. That is why there will be unburned mass fractions higher than 0 at knock onset point.

3.1. Improvements on the base engine model

Firstly, the influence of shortened burn duration and spark advance gain was examined. To set the modified burn duration on the base model, the burn duration multiplier was introduced and set to 0.8, making the new combustion 20% shorter. The Figure 3.3. shows base burn duration and the adjusted burn duration that represents the use of TJI system.

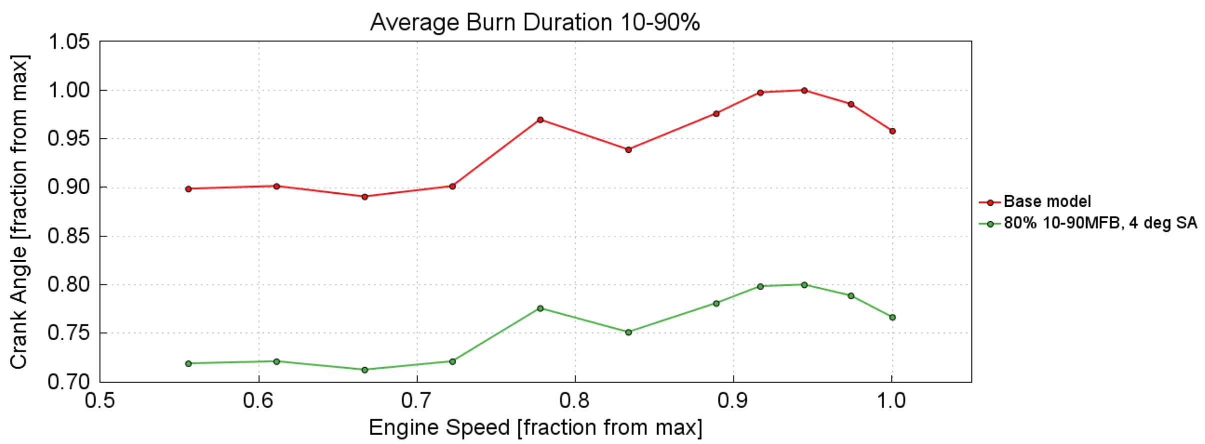

Figure 3.3. Average 10-90 burn duration

The Figure 3.4. shows the compared 50MFB points for the base model and for the adjusted model that are advanced by 4° of crank angle.

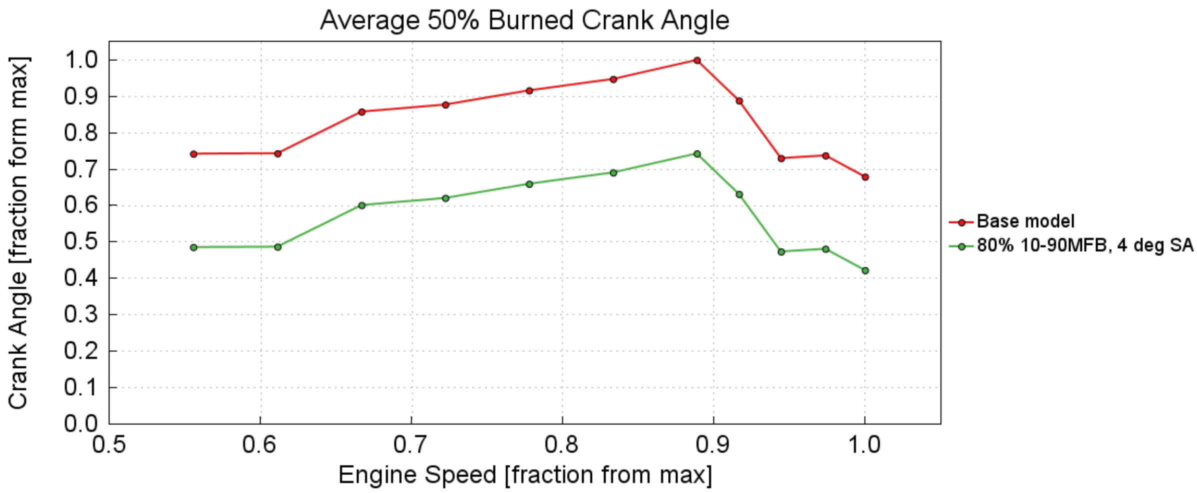

Figure 3.4. 50% burned crank angle

The Figure 3.5. shows the difference on the knock limit indicator (UMFKO) between the base model and the model with shorter combustion duration and included spark advance. This green curve will be the new benchmark for further steps in the analysis of TJI.

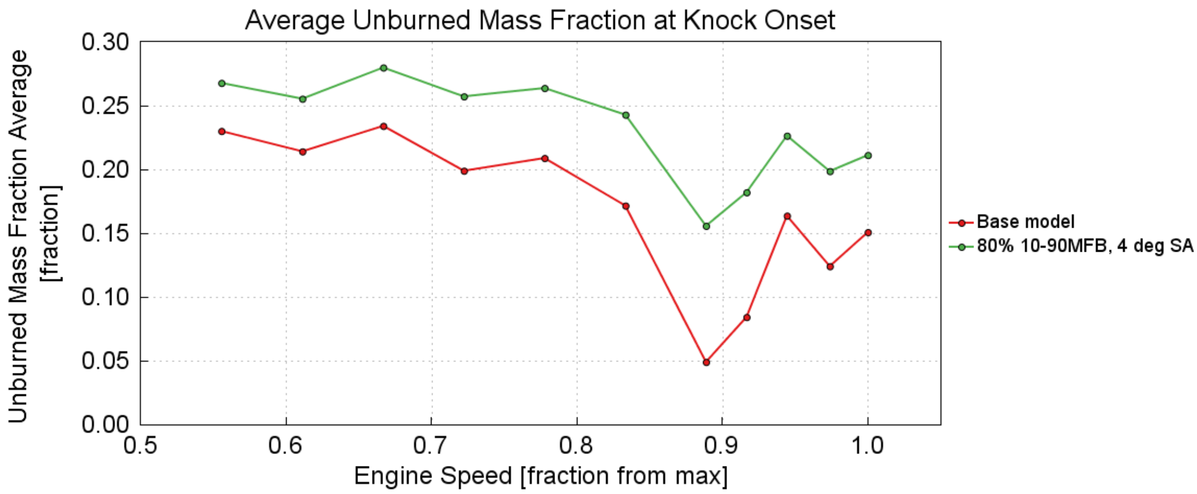

Figure 3.5. Unburned mass fraction at knock onset

In the Figure 3.6. it is shown how TJI technology improves brake torque and power.

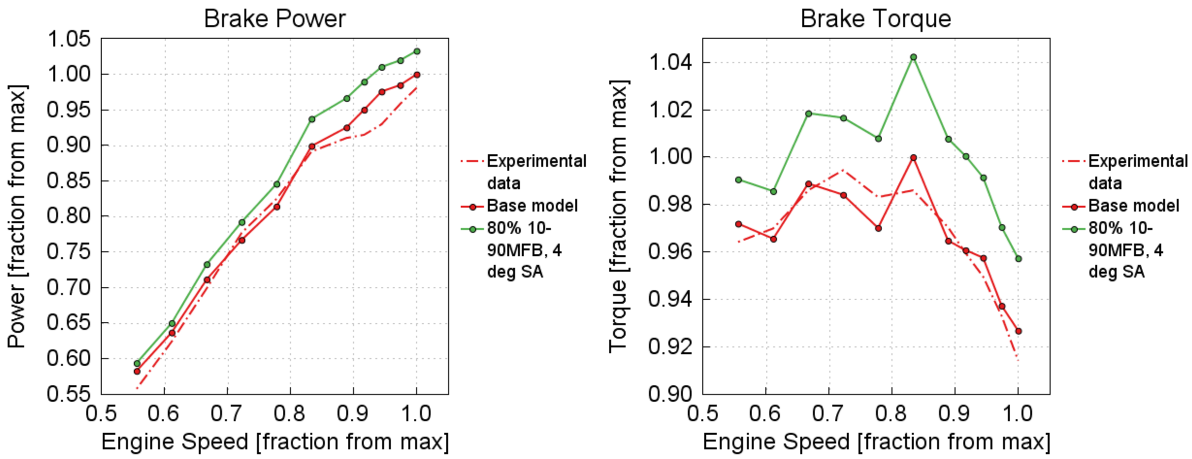

Figure 3.6. Brake power and torque

The Table 3.1. shows improvement in both power and torque with impact of combustion length and spark advance gain.

Table 3.1. Power and Torque increase

RPM, fraction	Torque and power increase, %
0,555	1.92
0,611	2.08
0,667	3.00
0,722	3.30
0,778	3.90
0,833	4.25
0,889	4.46
0,917	4.14
0,944	3.54
0,974	3.54
1	3.30

Brake cylinder efficiency is also influenced by TJI technology as shown in the Figure 3.7. There is also a considerable impact on the average maximum cylinder pressures, pushing the boundaries of in-cylinder stresses.

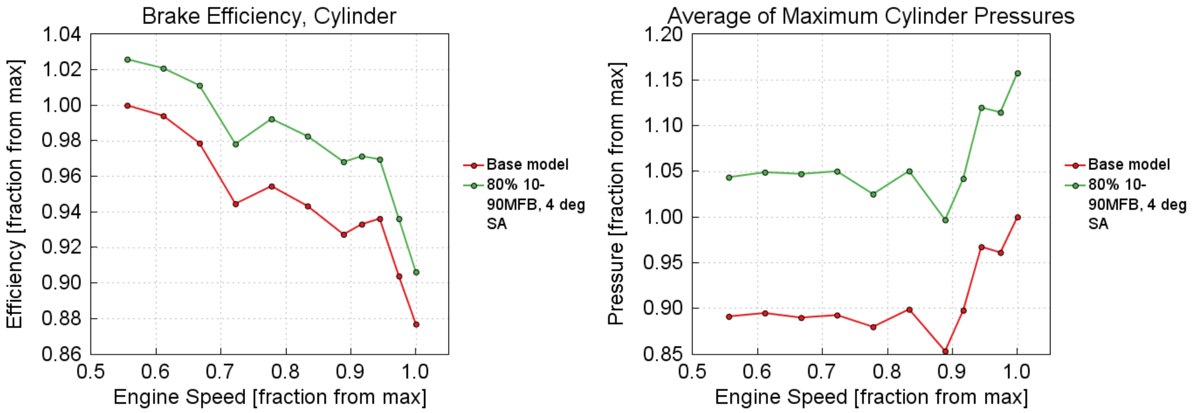

Figure 3.7. Brake in-cylinder efficiency and average of maximum in-cylinder pressure

By having a quicker, anticipated and more efficient combustion process, there is an impact on exhaust port temperatures as shown in Figure 3.8.

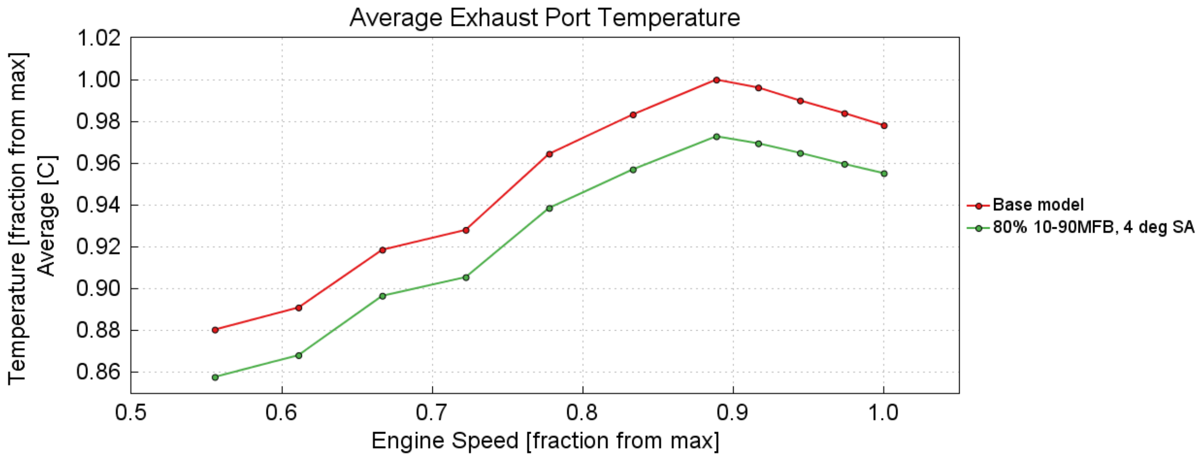

Figure 3.8. Exhaust port temperatures

3.2. Compression ratio increase

In this paragraph the impact of compression ratio increase will be evaluated. The key parameters will again be the data from unburned mass fraction at knock onset and the point of 50% burned fuel, which shows how point of spark ignition timing and beginning of combustion is affected.

To start with, compression ratio was increased by 0.5 and then by 1 point. This immediately shows the increase in power and torque but also a substantial amount of unburned mass fraction at knock onset.

To avoid knock occurrence, spark advance has been reduced in order to match reference UMFKO. By adjusting the point of spark discharge, it was possible to make knock level of the model with increased CR same as the model of the starting compression ratio.

This is shown again in UMFKO diagrams in Figure 3.9. where the adjusted models clearly match the new benchmark.

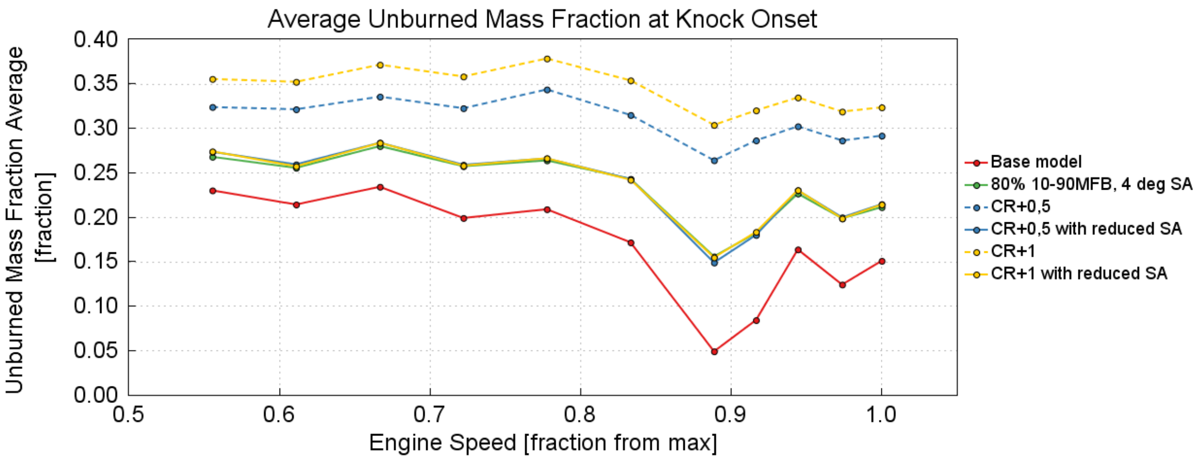

Figure 3.9. Unburned mass fraction at knock onset with increased CR

The 50MFB points are also presented in the Figure 3.10. to show how the combustion start was moved in time by retarding the spark advance.

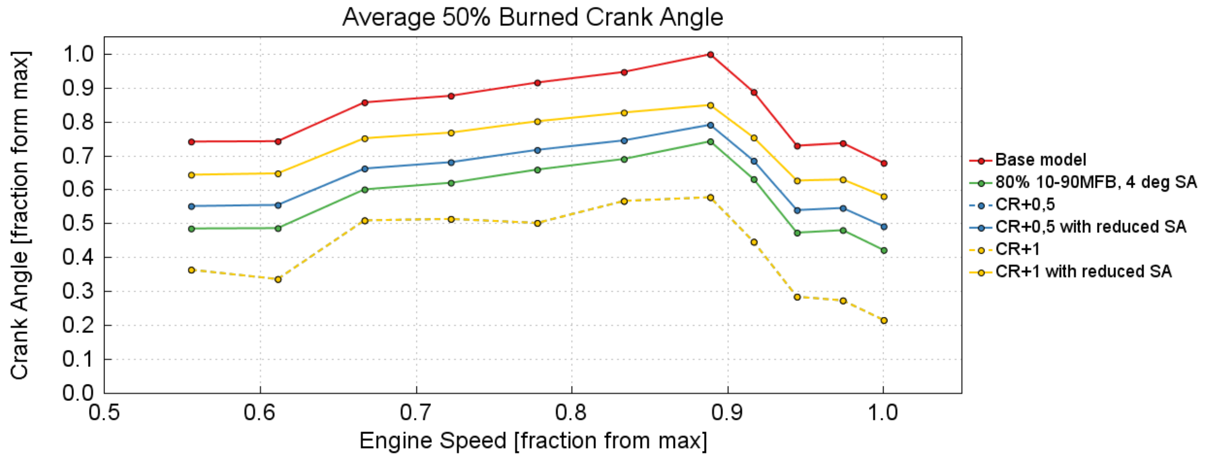

Figure 3.10. 50% burned crank angle with increased CR

This resulted in similar brake power and torque as the model with base compression ratio.

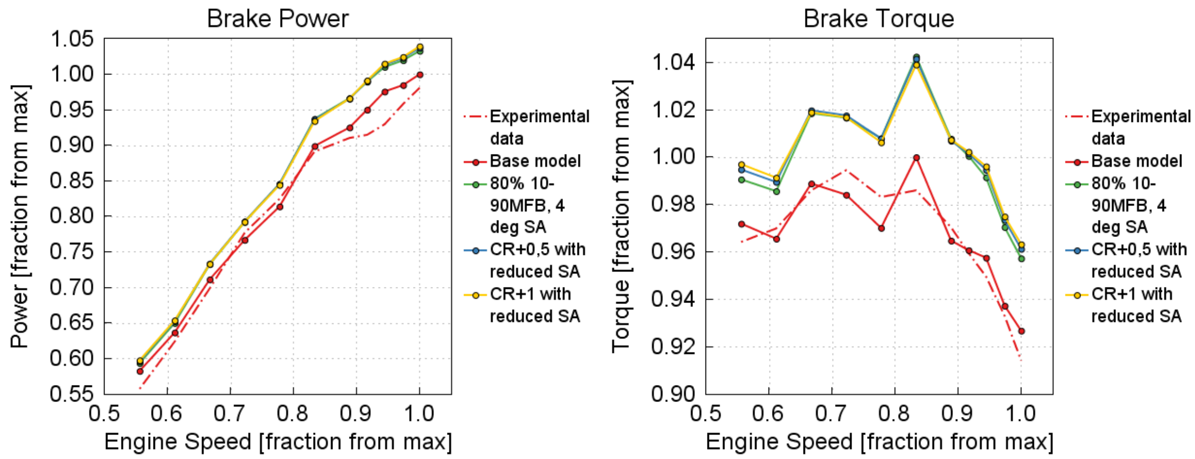

Figure 3.11. Brake power and torque with increased CR

Heat rejection due to engine cooling is presented in Figure 3.12. The results are very comparable to the base model.

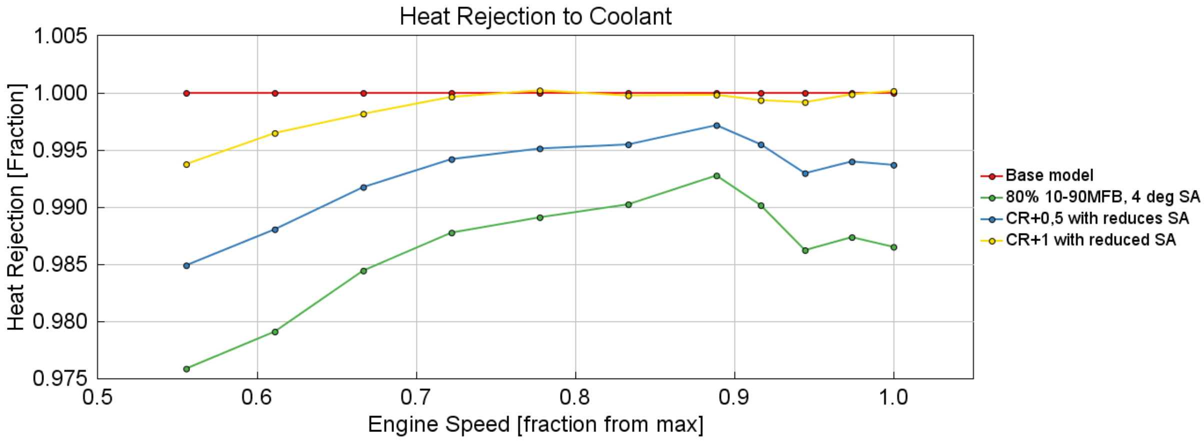

Figure 3.12. Heat Rejection diagram for TJI implementation

3.3. Lambda increase

The advanced combustion timing (with regards to the base engine model) and higher combustion speed both develop lower temperatures at exhaust ports. This can be exploited by increasing the air-to-fuel ratio, making the combustion more efficient and reducing fuel consumption. The starting point will be the model with CR increased by 1 point. The lambda value will gradually increase by 1 point until the exhaust port temperatures of the baseline are reached.

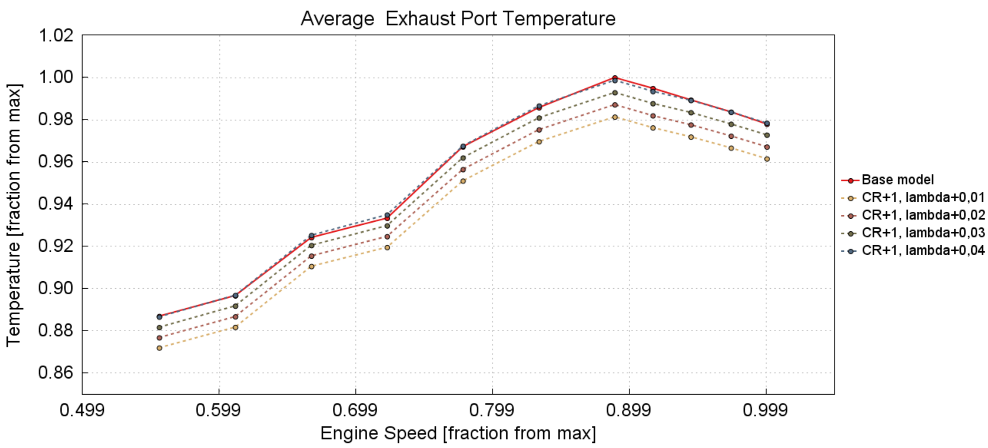

Figure 3.13. Lambda increase

The results show that lambda can be increased by a maximum of 4 points before reaching the initial exhaust temperatures.

The influence of lambda value increase is shown by the brake specific fuel consumption and brake in-cylinder efficiency in the Figure 3.14.

The BSFC falls about 2,25% while efficiency improves by the same amount for every step of lambda increase, with a total change of about 9%.

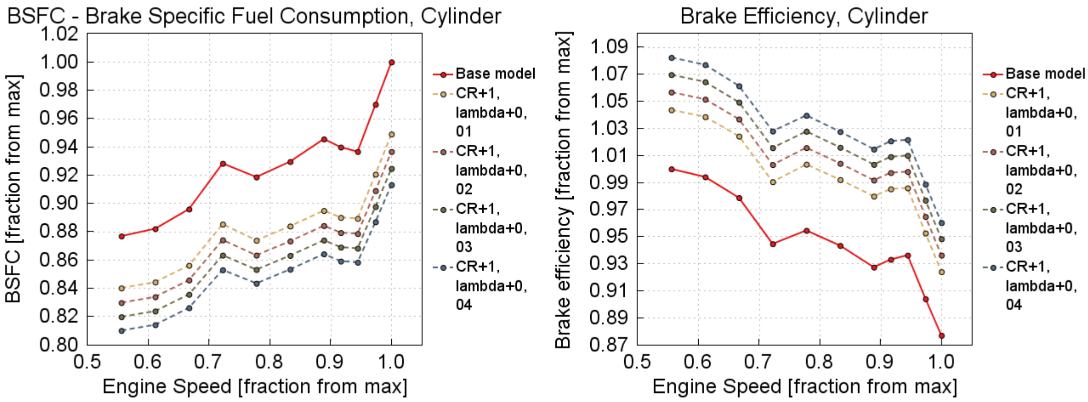

Figure 3.14. Brake specific fuel consumption and brake efficiency with lambda increase

4. VARIABLE VALVE LIFT

Variable valve lift (VVL) technology is used to vary the lift of the intake and exhaust valves depending on the operating conditions of the engine. In a commercial internal combustion engine, the valves will be controlled mechanically by a fixed camshaft profiles that describe a constant lift and timing profile of the valves through entire engine speed range. This design will usually result in a compromise solution between engine power, torque, and variables such as engine emissions and fuel consumption.

VVL system provides optimization for a variety of operating conditions, meaning it can assure maximum power at high speeds and maximum torque at lower speeds, as well as efficiency gains at part load conditions, all depending on the different lift profiles.

There are many different methods of designing the VVL system. The most common one is by having a camshaft with a several different lobe size that determine the amount of lift and timing. For the purpose of this paper only the effects of the modified valve profiles will be examined and not the method of doing so.

4.1. Maximum torque

For maximum power, the lift profile will usually be wide, in order to exploit the tuned dynamics, and provide more air (and therefore more fuel) into the cylinders. For maximum torque regions, the profile will be narrower, in order to reduce the backflow at Intake Valve Close (IVC).

The assumption done is that the valvetrain kinematics will be kept similar, and therefore also the acceleration on valves. Lift will be scaled according to the square of the width scaling coefficient.

The base model valve profiles, together with mass flow, for maximum brake power and maximum brake torque are presented in the Figures 4.1. and 4.2.

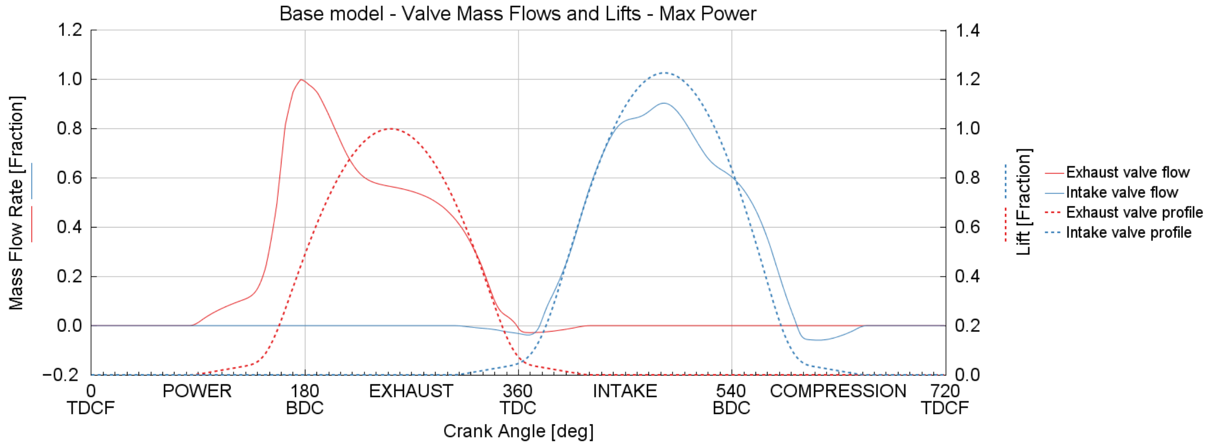

Figure 4.1. Intake and exhaust profile for maximum power on the base model

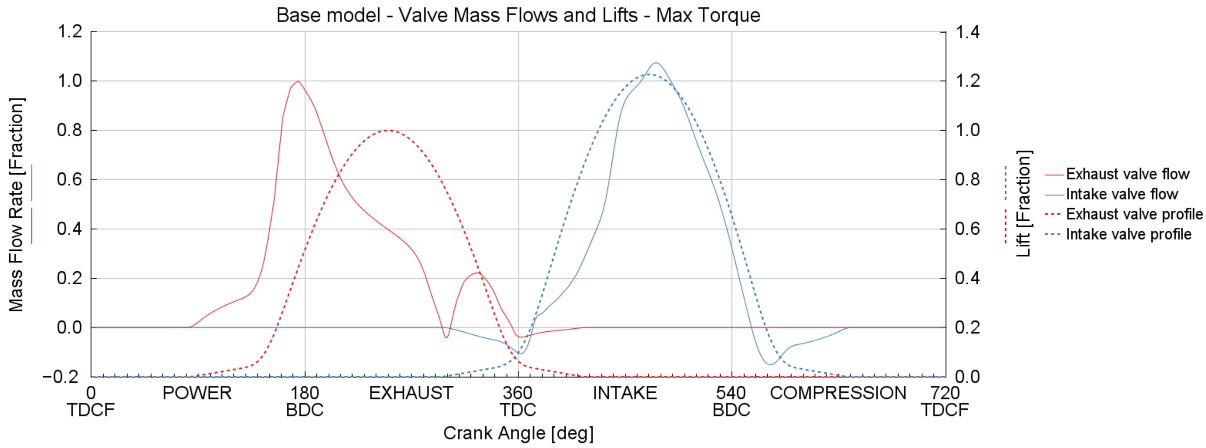

Figure 4.2. Intake and exhaust profile for maximum torque on the base model

Since the base model already has valve profiles optimised for maximum power at the highest engine speed, the valve profiles for maximum torque will now be examined.

To get the different profiles, a width and lift multipliers will be used. The width of the profile will be scaled with a factor k, and the lift with k^2.

Design of Experiments (DoE) is a special kind of simulation run and will be used for finding the most suitable valve profiles. A DoE is a table or matrix of simulations that will run for each active case in Case Setup. It is configured by moving one or more parameters to the DoE folder within Case Setup. When a parameter is varied in a DoE, it's referred to as a factor.

An experiment is a specific combination of factor values that are varied in the DoE. The number of experiments is defined by multiplying the ranges of the factor values (# of Levels) and then

multiplying it with the number of cases in the Case Setup (cases in this model represent different engine speeds). The number of experiments ultimatively defines the duration of the DoE run. Since the maximum torque is the target value, it makes sence to run only the case where the maximum torque point is expected to be, rather than simulating all possible engine speeds.
The Table 4.1. shows how the DoE is set up for finding the optimal valve profiles for maximum torque. Again, valve lift multipliers will directly be calculated from the width multipliers, so there is no need for them to be included in the DoE factors. The range between 0.8 and 1.2 was chosen because there are possible factors in the engine model, such as intake geometry or exhaust runners, therefore good practice is to keep the range wide.

Table 4.1. DoE setup with varying width multipliers for maximum brake torque

Design of Experiments

Parameter	Unit	Description	Min	Max	# of Levels
Angle_mult_Int (DOE)		Intake width multyplier	0.8	1.2	9
Angle_mult_Exh (DOE)		Exhaust width multyplier	0.8	1.2	9

This DoE setup led to a total of 81 experiments. The results are presented with a 2D plot in the Figure 4.3. The base model brake torque was given the value of 1 so the results show the comparison with that value.

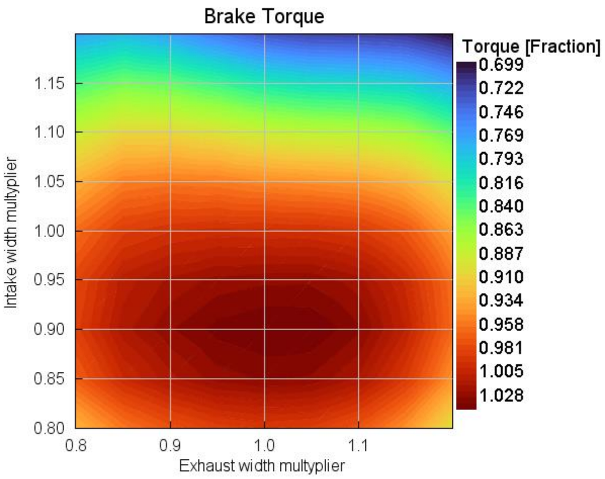

Figure 4.3. 2D plot for intake and exhaust multipliers vs maximum brake torque

It is concluded from the Figure 4.3. that the multiplier of 0.88 for intake and 1.02 for exhaust produce the best profile for maximum brake torque for the examined case. Next, the timing shift will be examined by running another DoE simulation, now with fixed values for the width multipliers.

The Table 4.2. shows the setup for finding the optimal timing shift form the modified valve profiles.

Table 4.2. DoE setup with varying profile shift for maximum brake torque

Design of Experiments

Parameter	Unit	Description	Min	Max	# of Levels
ECL_shift (DOE)		Shift of Cam Timing	-8.0	2.0	6
ICL_shift (DOE)		Shift of Cam Timing	-4.0	8.0	7

The final results for achieving maximum torque are shown in Figure 4.4.

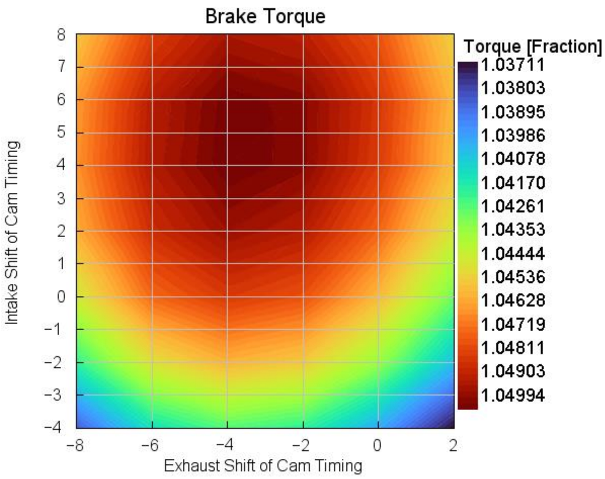

Figure 4.4. 2D plot for timing shift vs maximum brake torque

It is concluded from the Figure 4.4. that the timing shift of 4° for intake and -4° for exhaust produce the best profile for maximum brake torque for the examined case.

The optimised valve profiles result in a 5% increase in brake torque. Final intake and exhaust valve profiles for maximum torque are presented in Figure 4.5.

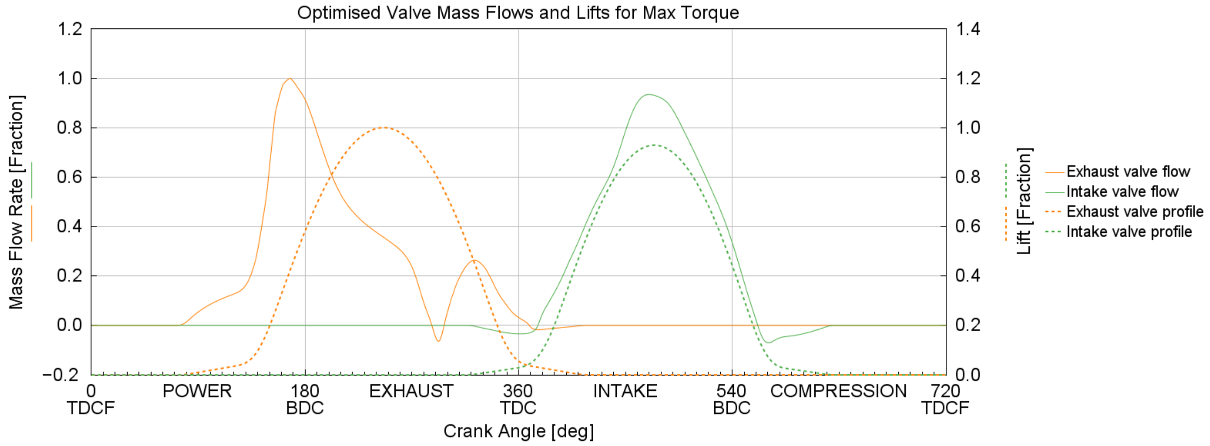

Figure 4.5. Optimisvalve profiles for maximum brake torque

The Figure 4.6. shows the comparisson between the base and the optimised model. It can be seen that the new profiles have aproximatelly the same accelerations as the base profiles.

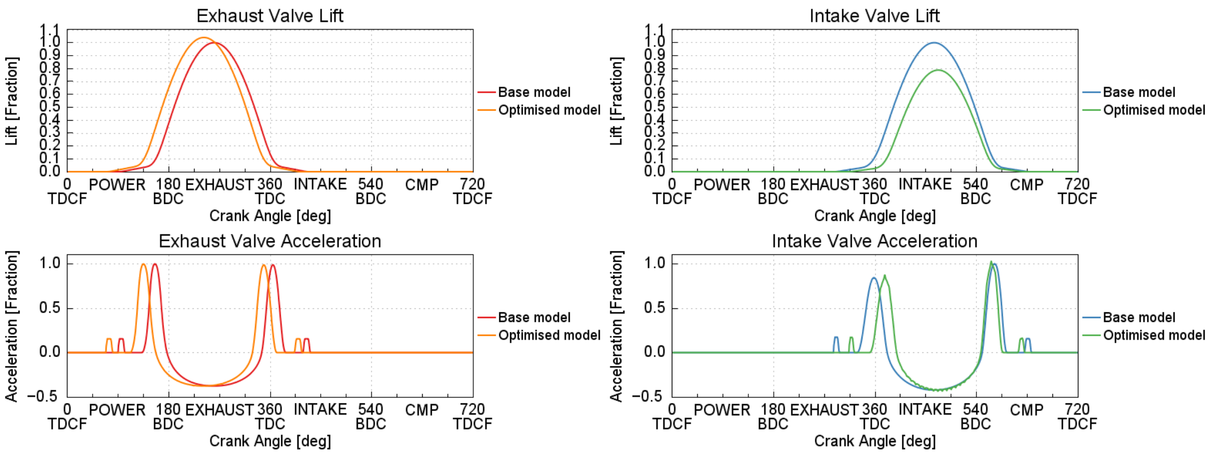

Figure 4.6. Comparison between base model and optimised model for maximum torque

4.2. Decrease of pumping losses

Pumping losses are the losses in power and efficiency that occur in an internal combustion engine due to gas exchange process in the low-pressure part of the cycle. During the intake stroke the piston must pull the air from the intake manifold into the combustion chamber which creates losses in performance. During the exhaust stroke the piston pushes the gases out of the chamber that also creates negative work.

The losses are represented by the value of Pumping Mean Effective Pressure (PMEP), but can also be visualized by examining *p,v* diagrams.

The common strategy for decreasing the pumping losses is varying the intake valve profile, specifically making it shorter and narrower. The point of Intake Valve Close (IVC) is moved back in phase, naming this method Early Intake Valve Close (EIVC), also known as the Miller cycle.

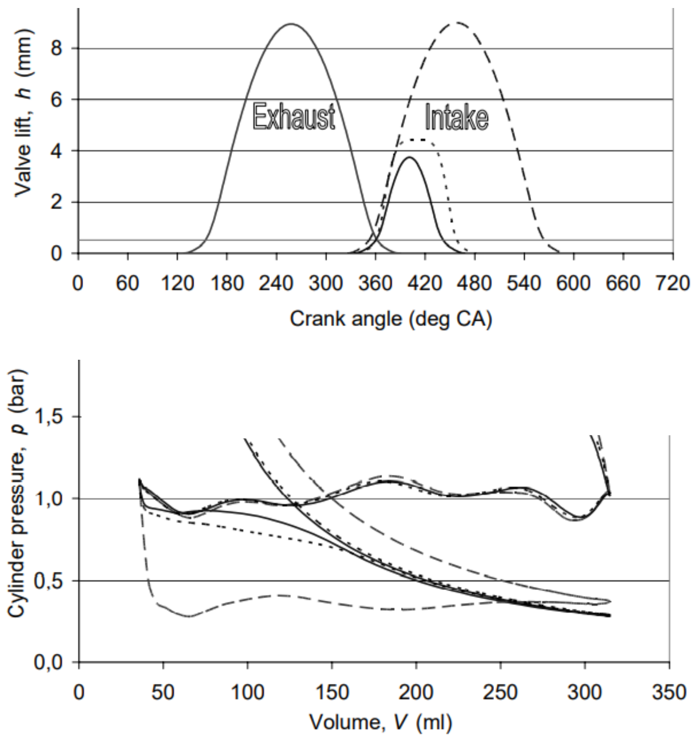

Figure 4.7. Early Intake Valve Close strategy [5]

This strategy will be implemented in part load conditions. Part load modeling in GT Power™ is done by adding a throttle controller in the main model space. This controller is based on PI regulator that takes the desired outcome values and by controlling the throttle angle tries to replicate the „desired" result. That way it will be clear how much the pumping losses decrease for the same conditions. In this case, brake torque from the base part load model will be targeted, but the controller can also be modified to target BMEP, power, airflow, etc.

The operating point of 2000 rpm and 2.3 bar BMEP was selected first, as it represents one of the most common part load conditions. The resulting valve profiles from this operating point will be used as an input to the next steady state points at part load conditions.

Figure 4.8. shows the intake and exhaust valve profiles for base model at operating point of 2000 rpm and 2.3 bar BMEP.

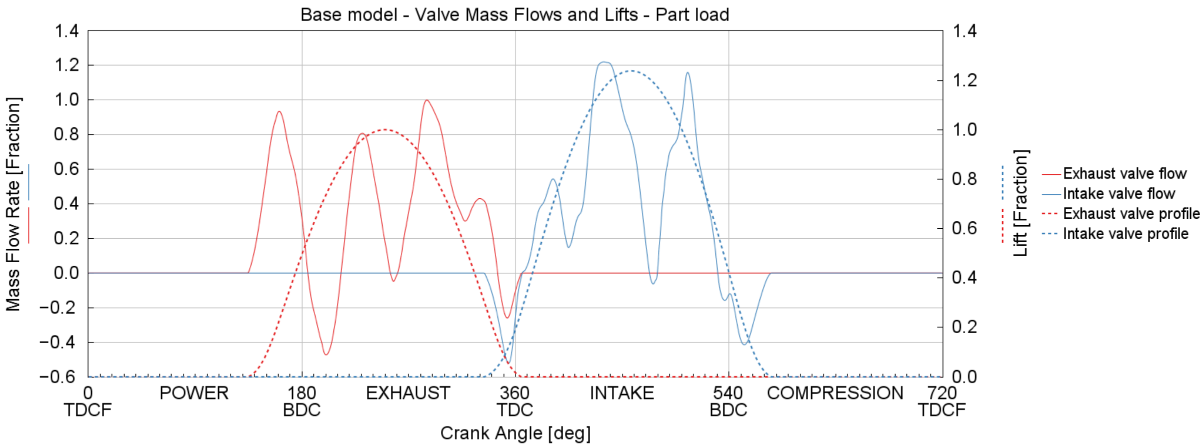

Figure 4.8. Part load intake and exhaust profiles

Log p, log v diagram in Figure 4.9. visually shows the current state of pumping losses. The area of gas exchange is expeced to decrease with the optimised valve profiles.

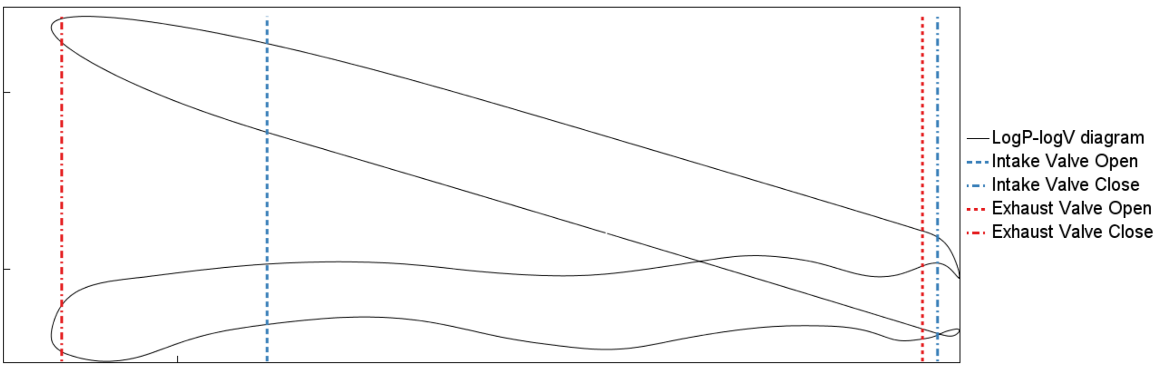

Figure 4.9. Base model *log p, log v* diagram

DoE will again be used for finding the optimal intake and exhaust profiles by taking a range of width multipliers and simulating every possible combinations of factors.

The DoE setup is shown in Table 4.3.

Table 4.3. DoE setup with varying width multipliers for pumping loss decrease

Design of Experiments

Parameter	Unit	Description	Min	Max	# of Levels
Angle_mult_Int (DOE)			0.6	0.9	4
Angle_mult_Exh (DOE)			0.7	1.3	7

The results from the DoE run are shown in Figure 4.10. The PMEP from the base model was given the value 1 so every result shown in Figure 4.10. is in comparison with that value.

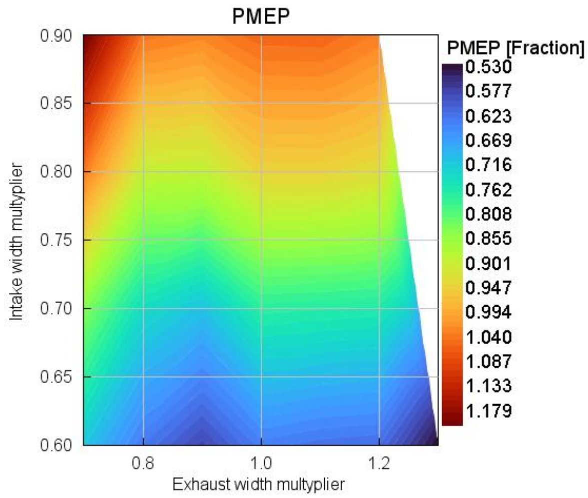

Figure 4.10. 2D plot for intake and exhaust multipliers vs PMEP

Although the biggest decrease in pumping losses is with the lowest intake and highest exhaust multipliers, the value of 0.6 for intake and 0.9 for the exhaust multipliers was selected. This is because that values generate most feasible *p,v* diagrams.

It is also nessesery to observe how much residual gases are present in the chamber at combustion start. This is a consequence of having an overlap of intake and exhaust valve profiles near TDC. In order to have a stable combustion, it is assumed that in the optimised model there will be the same amount of of exhaust gasses in the cylinder as in the base model.

For this, a 2D plot will be presented in Figure 4.11. where the percentage of the residual gasses in the base part load model is set to 1.

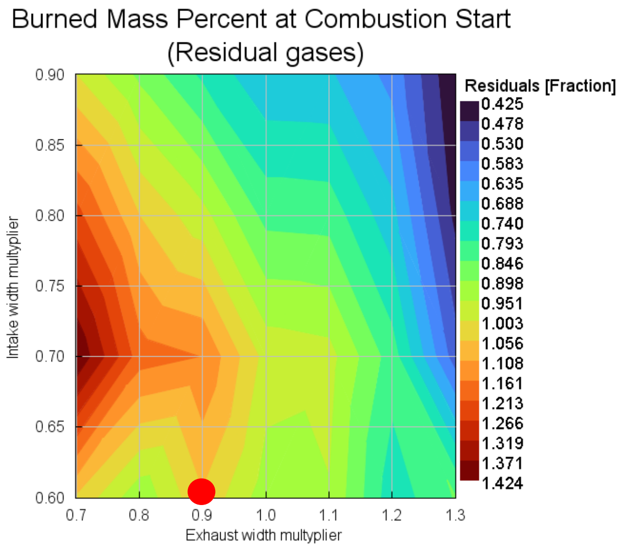

Figure 4.11. 2D plot for intake and exhaust multipliers vs residual gases

It is concluded that the selected combination of width multipliers (red point in Figure 4.11) will produce optimised valve profiles that assure the same amount of residual gasses at combustion start.

Before showing the final optimised profiles, the second DoE will be run to explore the benefits of adjusting the phase of intake and exhaust profiles.

The Table 4.4. shows the DoE setup for adjusting the timing of intake and exhaust profiles.

Table 4.4. DoE setup with varying profile shift for pumping loss decrease

Design of Experiments

Parameter	Unit	Description	Min	Max	# of Levels
ECL_shift (DOE)			-10.0	10.0	9
ICL_shift (DOE)			-10.0	10.0	9

The final results for achieving minimum pumping losses are shown in Figure 4.12.

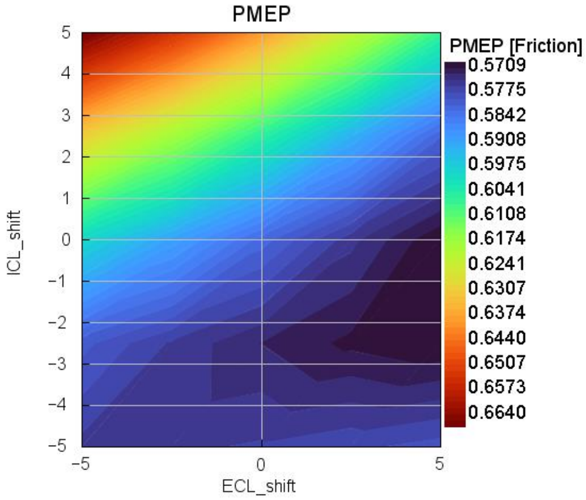

Figure 4.12. 2D plot for timing shift vs PMEP

Only the timing shift of the exhaust profile by 5° decreases the pumping losses, the intake profile is already at the optimal point from the base model. This retarding of the exhaust valve will be examined further in the next chapter when analysing the expansion work.

It is also nessesery to observe how much residual gases are present in the chamber at combustion start by varying the shift coefficients. This again is a consequence of having an overlap of intake and exhaust valve profiles near TDC.

A 2D plot will be presented in Figure 4.13. where the percentage of the residual gasses in the base part load model is set to 1.

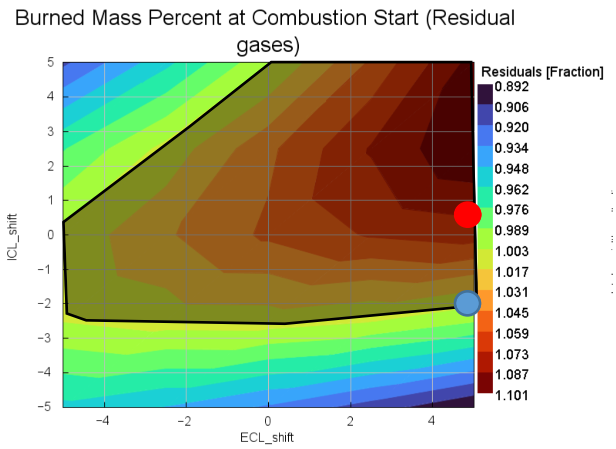

Figure 4.13. 2D plot for timing shift vs residual gases

The chosen shifts of 0° for intake and 5° for the exhaust provide 6% more residual gases in the combustion chamber compared to the base model (the red point in Figure 4.13.). Despite of that, the shift of -2° on intake valve was not chosen (the blue point in Figure 4.13.) because it performs slightly worse in terms of decreasing the pumping losses. The shaded area describes all possible points that would satisfy the concentration of residual gases present at combustion start.

It is concluded that the optimised valve profiles reduce the pumping losses by aproximatelly 43%.

Optimised valve profiles for minimum pumping losses are presented in the Figure 4.14.

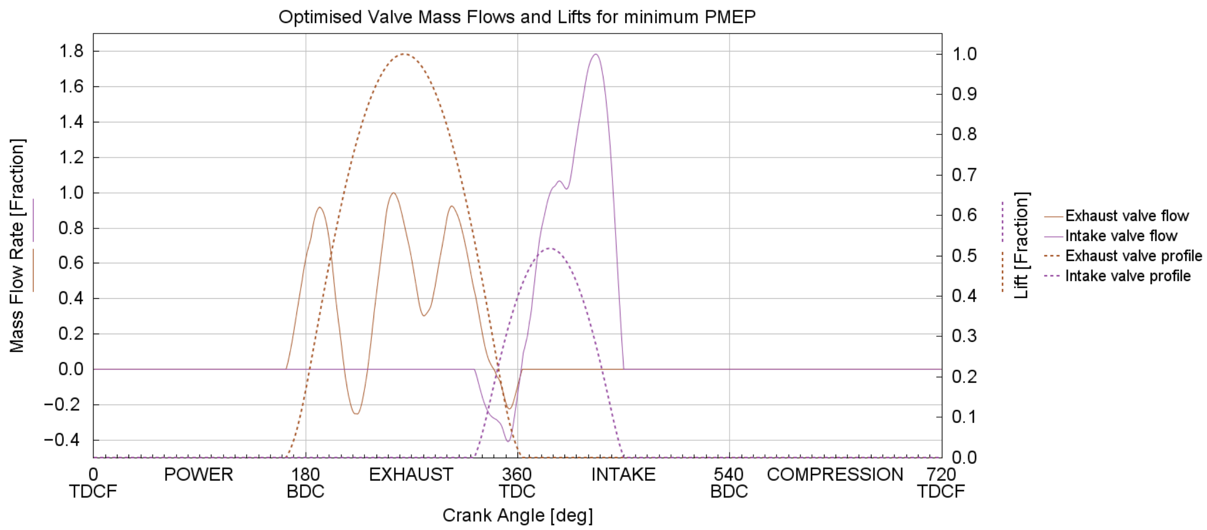

Figure 4.14. Optimised valve profiles for minimum pumping losses

The optimised *log p, log v* diagram is shown in Figure 4.15. and visually presents the decrease of pumping losses. Also, behind the point of IVC the cylinder pressure drops as expected, creating a characteristic shape of the Miller cycle.

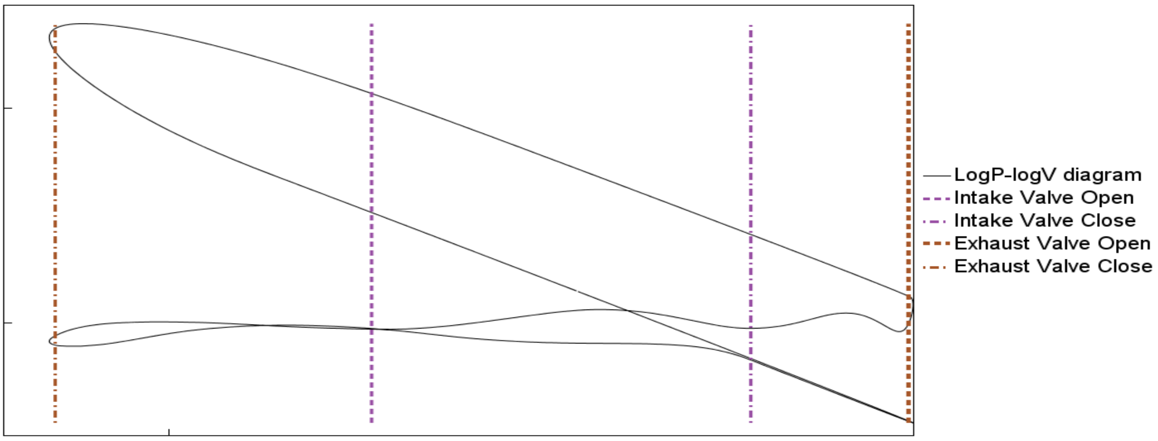

Figure 4.15. Optimised model *log p, log v* diagram

The Figure 4.16. shows the comparisson between the base and the optimised model.

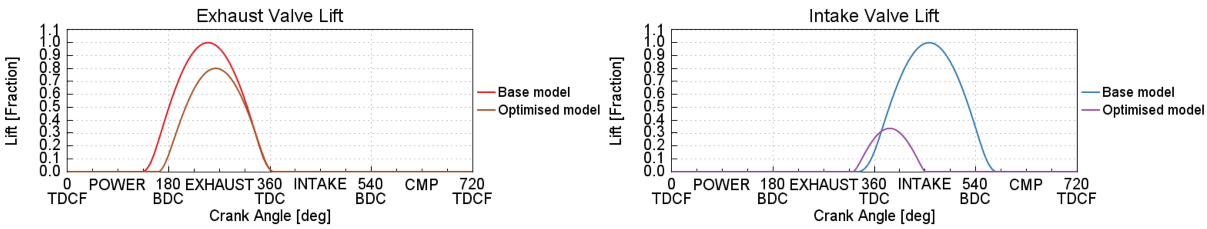

Figure 4.16. Comparrison between base model and optimised model for minimum PMEP

4.3. Increase of expansion work

By optimizing the intake and exhaust valve profiles for minimizing the pumping losses, there was also an impact on expansion work due to late Exhaust Valve Opening (EVO) point. This can visually be inspected by overlapping the *log p, log v* diagrams of the base and the optimized model and then zooming in to the end of expansion.

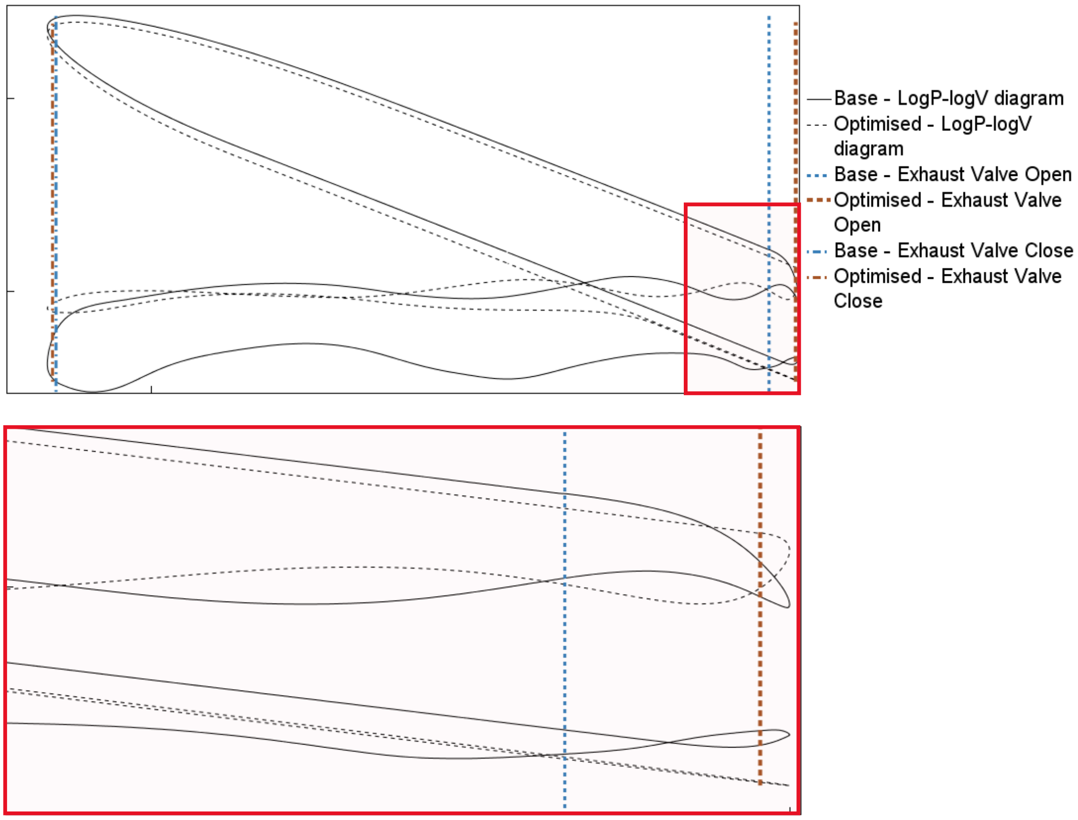

Figure 4.17. Expansion work increase in *log p, log v* diagram

The amount of increased expansion work and decreased pumping losses can also be shown by analysing the Gross Mean Effective Pressure (GMEP). It is a parameter that directly shows how much positive work is being done in the high pressure area of the *p,v* diagram. Knowing the values of PMEP from the simulations, and the IMEP that is set to be the same as the base model by using a throttle controller, the GMEP can be calculated by substracting the absolute value of PMEP from IMEP. Compared to the base model, the optimised model has a relative increase of 9% in GMEP.

Because of the increased GMEP there is also a 9% relative increase in brake cylinder efficiency at 2000 rpm and 2.3 bar BMEP compared to the base model.

To confirm that the opimized valve profiles have benefits on the whole part load conditions, another two steady state points will be analyzed.

4.4. Part load - 2500 rpm, 3 bar BMEP

The simulation results show 27% decrease in PMEP, and almost 4% relative increase in brake cylinder efficiency and GMEP. The concerning result is almost 50% decrease in residual gases present at combustion start compared to the base model. In conclusion, valve profile optimized for one part load point does not present the global optimum. The *log p*, *log v* diagram is presented in Figure 4.18.

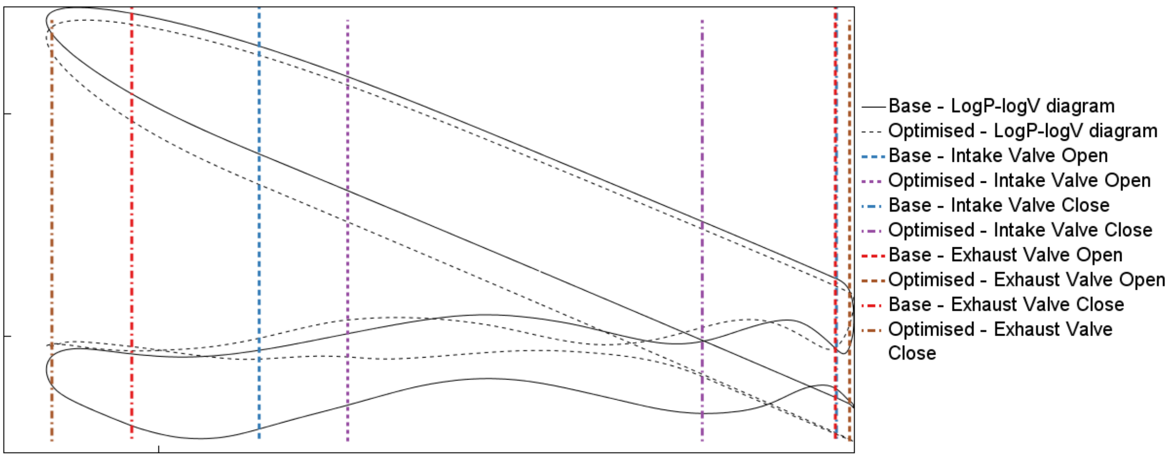

Figure 4.18. *log p, log v* diagram for 2500 rpm and 3 bar BMEP

4.5. Part load - 1500 rpm, 3.4 bar BMEP

The simulation results show 48% decrease in PMEP and almost 6% relative increase in brake cylinder efficiency and GMEP. The amount of residual gases at combustion start increased by around 3% which is not a big increase compared to the base model. This shows that optimised profiles for 2000 rpm and 2.3 bar BMEP show similar improvements in this part load condition. The *log p, log v* diagram is presented in Figure 4.19.

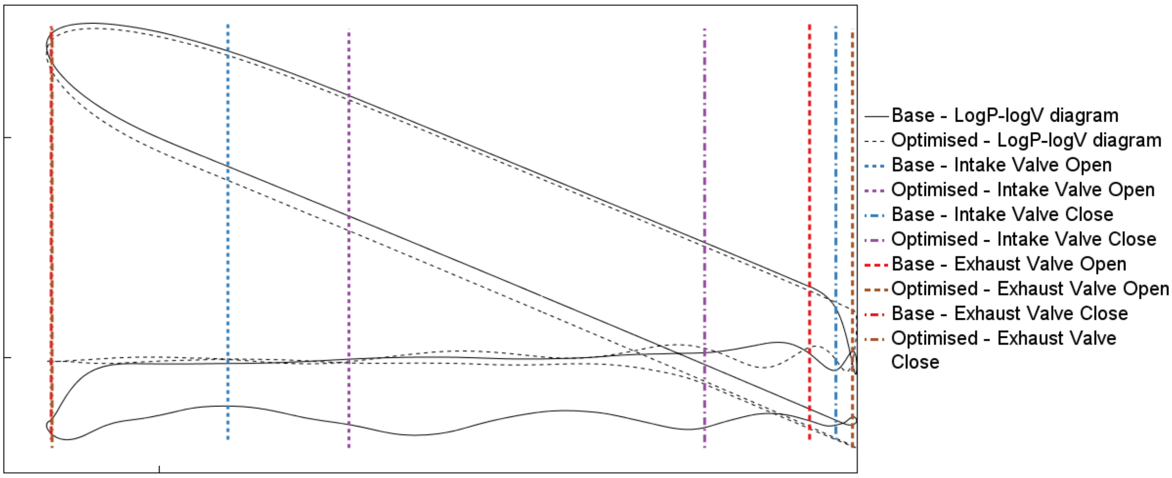

Figure 4.19. ***log p, log v*** **diagram for 1500 rpm and 3,4 bar BMEP**

5. CONCLUSION

In conclusion, both the passive prechamber ignition and variable valve lift improve the performance of spark ignition engines.

The prechamber ignition focuses mainly on the wide-open throttle conditions where it can improve brake torque and power, as well as reduce the exhaust temperatures. The simulation results show about 4% improvement in brake torque and power, 2% of relative increase in efficiency and 3% decrease in exhaust temperature compared to the base model. Furthermore, the temperature advantage can be exploited by reducing the enrichment, achieving a total of 9% relative increase in brake efficiency for lambda increase by 4 points. It is also concluded that increased compression ratio does not lead to a greater improvement in maximum torque and power since it is necessary to retard the combustion to avoid knock occurrence. The increased compression ratio is expected to bring improvements in brake efficiency at part load conditions, where second spark plug will be initiating the combustion.

The variable valve lift technology gives much needed optimization of the valve profiles, giving the engine more performance at wide open throttle conditions as well as improving the efficiency at part load conditions. The simulation results present a total of 5% increase in maximum torque at WOT conditions by having an adapted valve profiles. For the part load conditions, the pumping losses decreased by 43% at 2000 rpm and 2.3 bar BMEP while brake efficiency relatively increased by 9%. At other part load conditions, the benefits were not so noticeable since the optimization of the valve lifts and timing was done for 2000 rpm and 2.3 bar BMEP. Fully variable valve lift system would lead to a global optimum by varying the valve profiles for every engine speed and load case.

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