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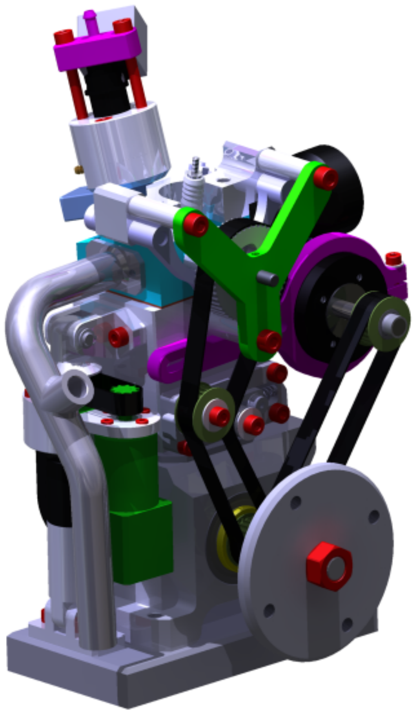

Development of fuel efficiency improvements for a single cylinder internal combustion engine

An investigation and implementation of five fuel efficiency improving technologies on an Eco Marathon engine

Bachelor thesis in Applied Mechanics

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CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover image: The figure shows the CAD-model of the finished technology demonstrator

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Chalmers Vera Team

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Abstract

This thesis describes the work of developing a technology demonstrating single cylinder engine for Chalmers Vera Team. The engine was to display five different technologies that could be of use to Chalmers Vera Team in the continued development of their fuel efficient engine.

The first technology investigated was the use of ceramics and low-friction materials in the combustion chamber in order to reduce heat and friction losses and hence increase the thermal efficiency.

Using the ideal thermodynamic Atkinson cycle was the second technology investigated as it has a higher thermal efficiency than a conventional Otto cycle engine and can be utilized by relatively small mechanical changes of the Chalmers Vera Team existing engine.

Replacing the conventional electric spark plug with a laser ignition system to achieve a more efficient combustion and lower the engine heat losses were investigated as a third fuel efficient technology.

The fourth technology was the use of variable compression in the combustion chamber to achieve an optimum compression ratio over the full speed operating range of the engine, in order to get a constant high thermal efficiency.

Direct injection was investigated as a fifth technology and a possibly more fuel efficient method of fuel injection, specifically the air assisted direct injection system, as the Chalmers Vera Team is limited, by competition rules, to a low pressure fuel system.

The different technologies were implemented into a complete technology demonstrating engine, where only the laser ignition never was actualised with a physical component. The different technologies never got tested on the engine due to lack of time, but they all showed potential to decrease the fuel consumption.

Keywords: Internal combustion engine, Atkinson cycle, Variable compression, Air assisted direct injection, Laser ignition, Ceramic materials, Low friction materials, Low heat rejection engine, Chalmers Vera Team

Sammandrag

Denna uppsats beskriver arbetet med att utveckla en teknikdemonstrerande encylindrig motor för Chalmers Vera Team. Motorns uppgift var att demonstrera fem olika teknologier som kan vara av intresse för Chalmers Vera Team i deras fortsatta utvecklingsarbete av sin bränsleeffektiva motor.

Den första teknologin som undersöktes var användandet av keramer och lågfriktionsmaterial i förbränningskammaren för att reducera värme- och friktionsförluster och följdaktligen öka den termiska verkningsgraden.

Användandet av den ideala termodynamiska Atkinsoncykeln var den andra teknologin som undersöktes då den har en idealt högre termisk verkningsgrad än den konventionella Ottocykeln samt att den kan implementeras genom relativt små mekaniska förändringar av Chalmers Vera Team:s befintliga motor.

Att ersätta det konventionella elektriska tändstiftet med ett lasertändningssystem var den tredje teknologin som undersöktes. Detta då lasertändningssystemet kan ge effektivare förbränning och minska värmeförlusterna i motorn.

Den fjärde teknologin som undersöktes var användandet av variabel kompression i förbränningskammaren för att kunna upprätthålla ett optimalt kompressionsförhållande över hela motorns arbetsintervall. Detta för att erhålla en konstant hög termisk verkningsgrad.

Direktinsprutning undersöktes som den femte teknologin då det kan vara en bränsleeffektivare metod för att spruta in bränsle i motorn. Specifikt undersöktes direktinsprutning m.h.a. tryckluft då Chalmers Vera Team är begränsade, av tävlingsregler, till lågt insprutningstryck.

De olika teknologierna implementerades i en komplett teknikdemonstrerande motor, där enbart lasertändning aldrig förverkligades som en fysisk komponent. De olika teknologierna testades aldrig på motor på grund av tidsbrist, men de visade alla på potential att minska bränsleförbrukningen.

List of Abbreviations

AADI	Air Assisted Direct Injection
AFR	Air Fuel Ratio
BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
CAD	Computer Aided Design
CAD	Crank Angle Degrees
CNC	Computer Numerical Control
CVT	Chalmers Vera Team
DI	Direct Injection
DLC	Diamond Like Carbon
EIVC	Early Intake Valve Closing
EVO	Exhaust Valve Opening
FEM	Finite Element Method
FMEP	Friction Mean Effective Pressure
HV	Hardness Vickers
IMEP	Indicated Mean Effective Pressure
ISFC	Indicated Specific Fuel Consumption
IVC	Intake Valve Closing
LHRE	Low Heat Rejection Engines
LI	Laser Ignition
LIVC	Late Intake Valve Closing
Nd:YAG	Neodymium-doped Yttrium Aluminium Garnet
NOx	Nitrogen Oxides
PMEP	Pump Mean Effective Pressure
PTFE	Polytetrafluoroethylene
SAE	Society of Automotive Engineers
SI	Spark Ignition
STEP	STandard for the Exchange of Product model data
TBC	Thermal Barrier Coating
VE	Volumetric Efficieny
Y-PSZ	Yttria Partially Stabilized Zirconia

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1 Introduction

The first section of this thesis describes the background, purpose and problem statement of the project as well as the scope.

1.1 Background

One of the most discussed topics in the environmental debate today is how to reduce emissions from vehicles. There is an ongoing change of energy sources used in vehicles, from almost completely being powered by fossil fuels to instead run on for example, electrical power or renewable fuels. Presently the main disadvantage of electric cars is that they are more expensive and do not have the same range as a car that runs on fossil fuels. Hence, there is still a need to further develop and improve the petrol powered engine to reduce the impact on the environment.

At Chalmers University of Technology there is an ongoing project called Chalmers Vera Team (henceforth CVT) competing in Eco Marathon competitions such as the Shell Eco Marathon in Rotterdam. The goal of Eco Marathon is to build a car that can run as far as possible on the equivalent energy of one litre of petrol. The name of the car is Vera and she can be seen in figure 1. Vera is a light-weight three wheel vehicle with a self-supporting carbon fiber chassis with low flow resistance. The current propulsion system in the car is a four-stroke, port injected petrol engine which has been developed and manufactured by CVT. The main focus for CVT when developing their engine is to improve the fuel efficiency by converting more of the fuel energy to mechanical work by lowering the losses in the engine.

CVT is a student project where the students manage everything from designing to manufacturing the car. This is done in their spare time, besides studying, which makes it hard for CVT to investigate all the different ideas they come up with. Due to lack of time and also unsureness of reliability they would like help to assess the potential of the ideas. This Bachelor project is therefore meant to help CVT to review some of the most promising ideas.

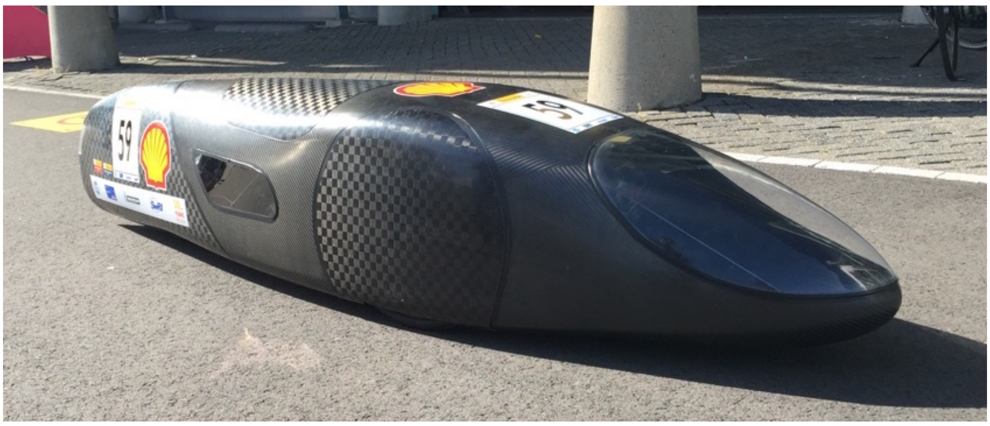


Figure 1: The Vera car.

1.2 Purpose

The purpose of this Bachelor thesis is to design a technology demonstrating engine where different fuel efficiency improving ideas are implemented and evaluated. This engine will also demonstrate how the ideas can be implemented in the Chalmers Vera Team engine.

1.3 Problem statement

To help CVT evaluate the ideas, the project aimed to see if they were realisable, if they could improve on fuel efficiency and if so, show with the use of a technology demonstrating engine, how they could be implemented in the CVT engine.

Five different ideas shall be evaluated with the above mentioned aim:

1. Use ceramics and low-friction materials in the combustion chamber of the engine. If possible, also examine if this can lead to an engine design with an open crankcase.
2. Running the engine with the Atkinson engine cycle.
3. Using a laser to ignite the fuel.
4. Utilise controllable variable compression.
5. Use direct injection of the fuel into the combustion chamber.

1.4 Scope

The budget for the project was decided by the Division of Combustion at the Department of Applied Mechanics in collaboration with the project group after completed literature survey and contact with potential suppliers. Some ideas from CVT have not been investigated due to lack of time or because they were deemed too complicated.

The project did not prioritise reducing the weight, emissions and outer dimensions of the engine in the design phase since improving the fuel efficiency of the engine was the main aim. However, assessing how the ideas could be implemented in the CVT engine was a crucial part of the project. The rules of the Shell Eco Marathon [1] competition restricted the development of the concepts since they were the rules that the CVT, and subsequently their engine, had to abide by.

Only simplified simulations, calculations and designs were performed since the aim of the project was to demonstrate how the technology works and not to optimise them for the CVT engine.

2 Method

The working method of this project was divided into four main phases:

1. Literature survey
2. Design
3. Manufacturing
4. Testing

The first phase consisted of gathering information about all of the different ideas from CVT. The material was collected from various sources, mainly automotive databases such as SAE. The ideas from CVT were then evaluated using the knowledge obtained in the initial literature survey. The ideas that were considered to have the most potential to improve fuel efficiency, as well as being possible to realise within the project scope were chosen for further development.

The project group was divided into three sub-project groups consisting of two members each to focus their work on some of the five ideas that were divided according to the list below:

Subproject 1	Use ceramics and low-friction materials in the combustion chamber of the engine.
Subproject 2	Running the engine with the Atkinson engine cycle.
	Using a laser to ignite the fuel.
Subproject 3	Have controllable variable compression.
	Use direct injection of the fuel into the combustion chamber.

A number of questions about each of these five ideas were specified before the extensive literature survey commenced. The questions regard the fuel efficiency improvement potential, the current applications of the technologies, and the durability of the technologies. They also concern the advantages and disadvantages of the different technologies, as well as explaining the theory behind them. These questions were used as the basis for the survey which was considered complete when all questions were answered. The aim of this phase was to be able to make qualified decisions on how to implement each idea and obtain the knowledge necessary for the design phase.

In the design phase, these ideas were turned into concepts that were designed based on the existing CVT engine. Existing CAD models of an older CVT engine, called MKVI, were available and new parts were designed to fit this engine. This phase also involved simplified calculations and the choice of materials for the parts that were to be manufactured. When simulations were deemed necessary they were carried out during this phase as well. The MKVI was used as a base for the engine in this project, and the different concepts redesigned parts of the MKVI that needed redesigning for the technology to work, and kept the rest of the engine intact. The aim of this phase was to be able to begin manufacturing, using models, drawings and chosen materials.

In the manufacturing phase all new parts were manufactured using the machines available in the prototype workshop at Chalmers. Some parts could be modified using spare parts from the MKVI instead of making new ones. When parts or components could not be manufactured on site, manufacturers were contacted, and if the price was within budget, the parts were acquired. The parts were then assembled into the technology demonstrating engine.

The testing phase was meant to test the different finished concepts on their own to assess their individual effects on the engine and potential to increase fuel efficiency. The entire engine was then supposed to be tested and optimised as a whole and the fuel efficiency to be evaluated. Unfortunately, the testing phase was never reached within the time scope of the project.

3 Engine Theory

The engine that the project has been based on is the CVT engine MKVI which is a single-cylinder 50cc port injected petrol engine of the four-stroke type with overhead camshaft and two valves.

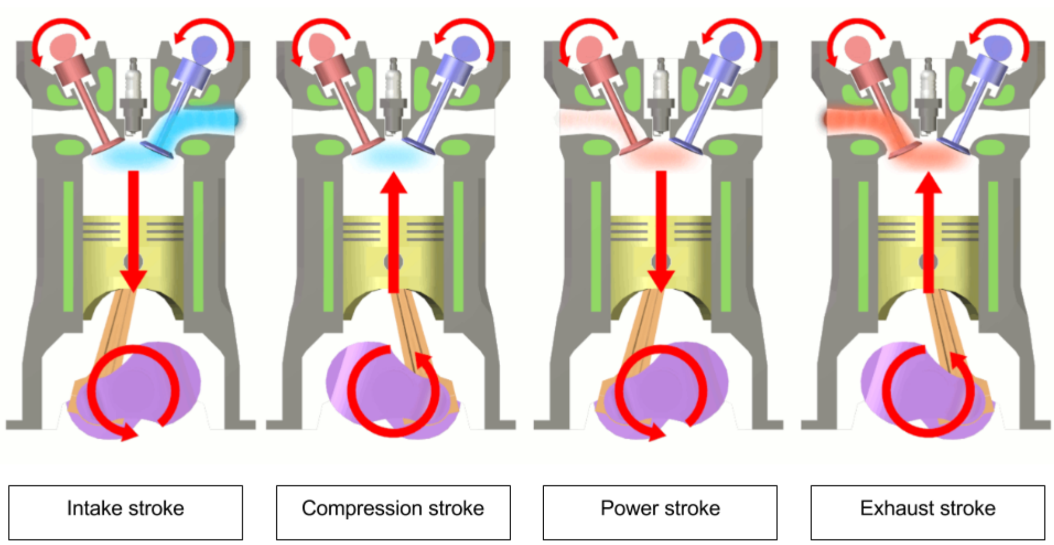
3.1 4-Stroke engine

The four stroke cycle consists of four subsequent phases; intake-, compression-, power- and exhaust stroke, shown in figure 2. During a stroke, the piston moves either to the top dead center, TDC, or to its bottom dead center, BDC. The movement between these endpoints corresponds to 180 crank angle degrees. The entire period then corresponds of $180 \cdot 4 = 720$ crank angle degrees, that is, two revolutions of the crankshaft.

During the intake stroke the intake valve is open and the piston moves down to the BDC creating a low pressure which causes air to flow into the combustion chamber. At the compression stroke, the valves are closed and the piston then moves up to the TDC, compressing the air. The fuel is injected into the cylinder at some point during the intake or compression stroke depending on the method of fuel injection. The compressed air and fuel is then ignited and combusted which creates a rise of temperature and, hence, pressure causing the piston to be pushed down to BDC. As the air and fuel is combusted exhaust fumes are formed and fills the cylinder. During the exhaust stroke the exhaust valve is open and the exhaust fumes are pushed out through the exhaust duct as the piston moves back up to the TDC completing the cycle and then starting at the intake stroke again.

3.2 Engine components

The engine consists of a number of key components. Excluding ignition system, fuel system and control system, they are; crankcase, cylinder, cylinder liner, crankshaft, conrod, piston, piston rings, cylinder head, camshaft and valves. Figure 3 shows a sectional view of the Vera MKVI engine where the visible parts are mentioned. Figure 4 shows the full



[2]

Figure 2: The strokes in a 4-stroke engine

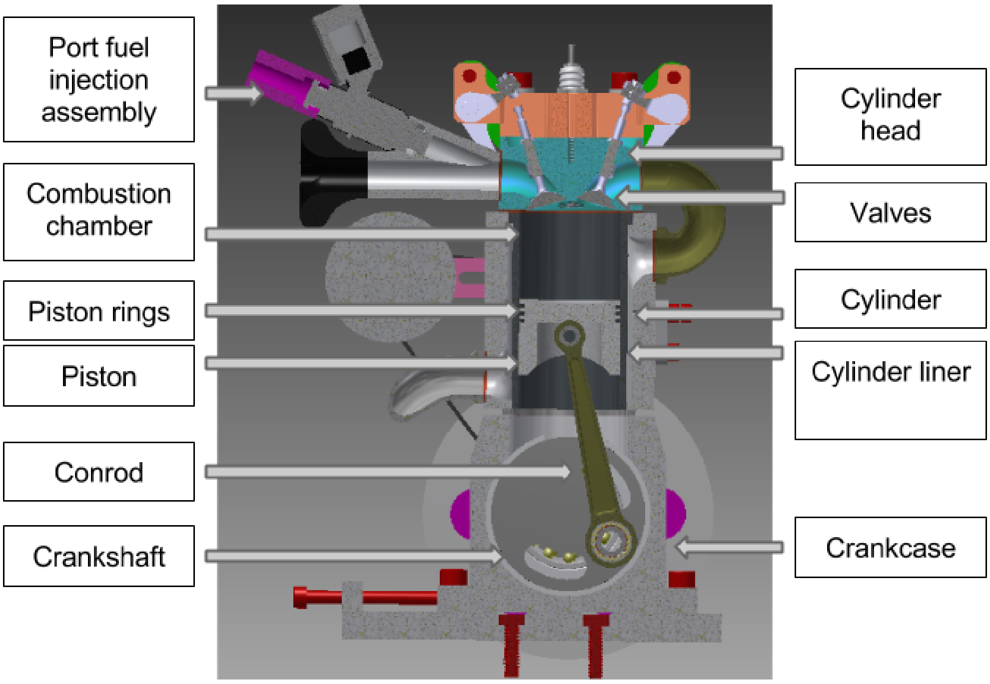


Figure 3: The MKVI engine in a half-section view

view of the engine, with visible components denoted. The base of the engine is the engine crankcase and the cylinder. The piston is connected to the conrod which in turn is connected to the crankshaft. The piston moves up and down in the cylinder and that linear movement is translated into rotational movement of the crankshaft which then powers the wheels of the CVT vehicle. To lubricate all moving parts, the crankcase contains oil that is spread around by the rotating crankshaft.

The piston is a short cylindrical part which seals one end of the cylinder, the combustion chamber, from the crankcase. The cylinder liner lines the cylinder to make the piston move smoothly inside the cylinder and is usually made of a softer material such as cast iron. To make sure the seal is tight, piston rings are fitted around the circumference of the piston to prevent gas from leaking into the crankcase from the combustion chamber, known as blow-by, and oil from the crankcase to leak into the combustion chamber.

The top of the cylinder is sealed with the cylinder head, and the bottom of the cylinder head is bowl shaped. The bowl shape, the crown of the piston, and the sides of the cylinder liner make up the combustion chamber where the combustion happens. In the cylinder head the intake and exhaust valves are mounted and their function is to open and close at different strokes, allowing gas to be both inhaled and emitted. The spark plug which ignites the air-fuel mixture in the combustion chamber is also mounted in the cylinder head. To make the valves open, the cam lobes are mounted on the camshaft which rotate and the cam lobes force the valves to open by pushing directly on them or by a lever mechanism. To decide when the valves open, the camshaft is connected to the crankshaft to time the opening of the valves with a stroke.

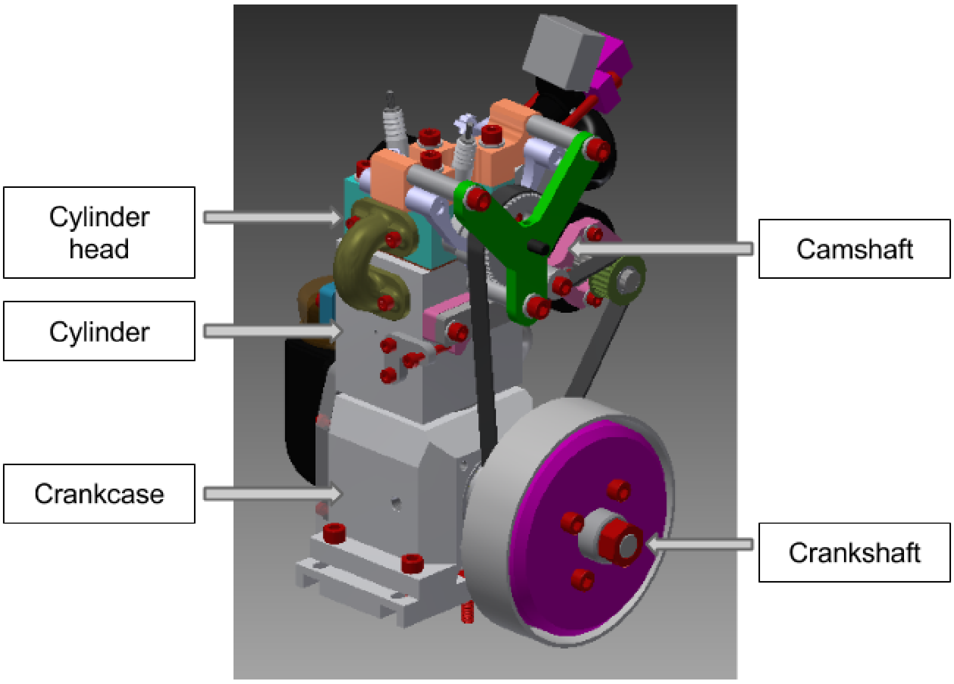


Figure 4: Full section view of the MKVI engine

3.3 Operating the engine

The power that is acquired from the engine corresponds to the load. A high load corresponds to for example, climbing a hill or accelerating, and it requires the engine to work at maximum capacity. A light load on the other hand can be going downhill or simply using a small share of the maximum engine power.

The CVT engine is designed to be as fuel efficient as possible. The mean speed for the Eco Marathon is 25 km/h and to be as fuel efficient as possible the engine runs at a high load at around 4000 rpm for a short interval to accelerate up to maximum velocity. After that the engine is shut off until the minimum velocity is reached again, the engine is restarted and the cycle repeats itself. This way of operating the engine means that the engine needs to be restarted several times. Therefore the engine is insulated to be kept warm when it is not running, as a cold-start is fuel consuming and should be avoided.

4 Ceramic materials

The walls enclosing the combustion chamber is normally made out of metal materials. Metals are easy to process and can resist the mechanical stresses occurring in an engine. However, metals are good heat conductors. They will thereby transfer a lot of heat energy away from the combustion chamber. This is known as in cylinder heat losses. Ceramic materials have extremely low heat conductivity. This subproject was therefore focusing on implementing ceramic materials in the combustion chamber to insulate the combustion chamber and hence lower the in cylinder heat losses.

4.1 Theory

A great amount of heat energy is released when the air-fuel mixture is combusted in the combustion chamber. This energy increases the temperature of the gases which will heat up the surrounding surfaces in the combustion chamber. The surfaces transfers some of the heat energy away which results in heat losses. A higher heat conductivity of the wall materials lead to higher heat transfer and thereby more heat losses. A material with low heat conductivity is thus desired. The released energy of the fuel can be divided into three parts: heat transfer and friction losses, mechanical work and energy in the exhaust gases. The principle of using ceramic materials, with low heat conductivity, was to insulate the combustion chamber and minimize the heat transfer losses and instead gain more mechanical work from the same amount of fuel.

To contain as much heat energy as possible in the gases of the combustion chamber an insulating thermal barrier between the combustion chamber and the engine block is needed. This barrier had to be made out of a material which could withstand the thermal and mechanical stresses and have a low heat transfer conductive coefficient. Ceramics have these properties and is therefore the material type used in such thermal barriers. The surfaces which are enclosing the combustion chamber are the piston crown, the cylinder liner and the cylinder head, denoted in Figure 3.

These surfaces could either have inserts, or whole parts, made out of ceramics or regular metal parts with a ceramic coating to make the thermal barrier. The use of coated surfaces is the simplest and most common way to create a thermal barrier. These coatings are called Thermal Barrier Coating (TBC) and are commonly used in gas turbines. The turbine blades are coated with a TBC to withstand higher temperatures and to lower the heat transfer into the material of the blades [3]. The same concept could be used in internal combustion engines to minimize the heat losses.

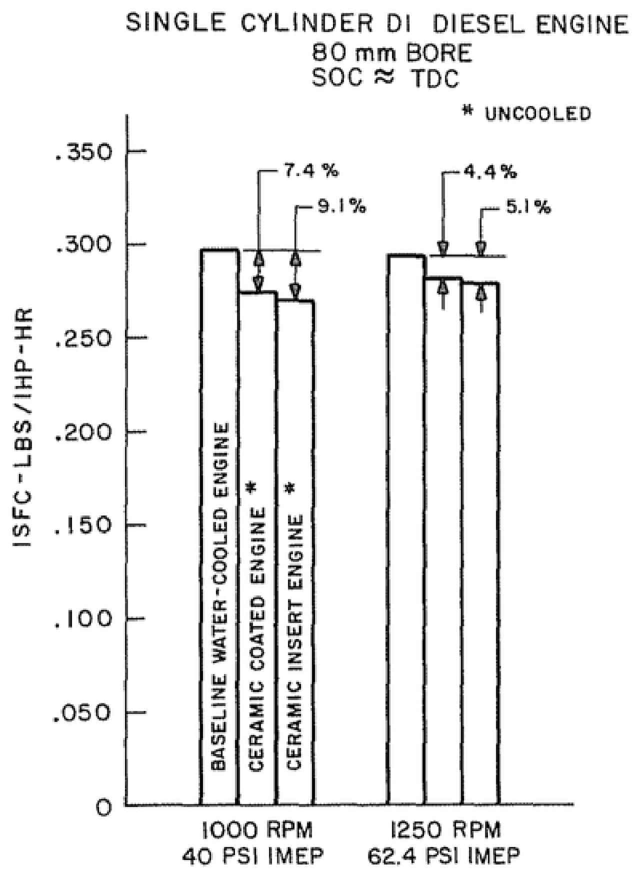
Engines using thermal barriers, either with coatings or inserts, have combustion chamber surfaces which are only rejecting a small amount of heat and are therefore known as Low Heat Rejection Engines (LHRE). These engines only exist as research test engines at the moment, but are expected to improve the fuel efficiency in conventional internal combustion engines as well as to eliminate the need for engine block cooling.

4.1.1 Advantages

Ceramics have many desirable properties for usage in the combustion chamber. They are resistant to high temperatures and are chemically stable. In addition, technical ceramics are resistant to wear and can withstand high compressive stresses with a desired low heat conductive coefficient. This is why technical ceramics are being used for creating thermal barriers. Many investigations has been carried out on LHRE and the results are inconsistent. This has lead to other reports comparing the investigations to find out why their results differ [4, 5].

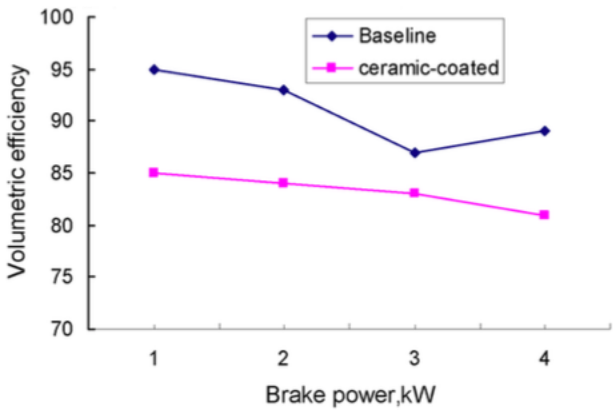
The main goal of these investigations has often been to completely remove the water cooling system but also to develop a more efficient engine. Therefore it was hard to compare a LHRE with a conventional engine in a correct way. The most accurate way [4] was nevertheless to control the Air Fuel Ratio (AFR), see section 8.1.1, so that the combustion is as similar as possible. With this setup the LHRE are often more fuel efficient [6].

In 1986, Havstad tested both the use of ceramic inserts and ceramic coatings. Both showed improvements in fuel consumption, with a small upper hand for the ceramic inserts. The test results on fuel consumption from Havstad's investigations are shown in Figure 5 [6].



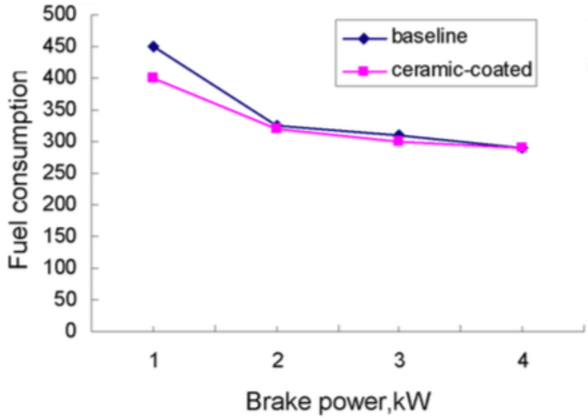
[6]

Figure 5: Comparison on ISFC, Indicated specific fuel consumption, between diesel engines with ceramic inserts, ceramic coatings and a baseline engine. (Lower value is better)



[9]

Figure 6: Volumetric efficiency of ceramic coated engine and baseline engine.



[9]

Figure 7: Fuel consumption of ceramic coated engine and baseline engine.

However, other investigations suggest that LHRE instead increases the fuel consumption, due to increased losses [7, 8]. These losses are explained as increased friction due to lack of lubrication between piston rings and cylinder liner at higher temperatures.

Another explanation why LHRE are less efficient in some test was the decreased volumetric efficiency (VE) due to increased temperatures of the combustion chamber walls. If the walls have a higher temperature when new cold gases are inhaled through the intake valve, the gases will heat up and therefore expand more. When the gases expand, less air and less oxygen molecules will be inhaled into the combustion chamber, which decreases the VE. Tests of the decrement in VE due to higher wall temperature of the combustion chamber are displayed in Figure 6 [9].

This effect is confirmed in many reports [4, 5]. But the investigations also confirms that the thermal efficiency was increased with the use of LHRE. The thermal efficiency explains how much of the heat energy in the fuel that is converted to mechanical work. This leads to a trade off between the volumetric and thermal efficiency, but when LHRE was compared with conventional engines at the same AFR the LHRE:s showed decreased fuel consumption, shown in Figure 7 [9]. This way of comparing the engines was regarded to be the most fair comparison as the combustion properties would be equal.

The best estimate of fuel consumption improvement with LHRE was by about 2 % - 10 %. Although others proposed that naturally aspirated engines has an increment in fuel consumption (0 % - 10 %), and that turbocharged engines uses the higher exhaust gas energy and thus has a decreasing fuel consumption (0 % - 10 %) [4]. Most investigations in general suggest that LHRE should be combined with a turbocharger for the best result, but that naturally aspirated engines still could improve the fuel efficiency.

4.1.2 Problems

The research of today on LHRE is mainly focusing on implementing ceramic materials in diesel engines. Diesel engines are compression ignited and the increment in temperature due to ceramic surfaces in the combustion chamber only effects the combustion process in a positive manner.

This project, on the other hand was focusing on implementing ceramic materials in a small petrol engine. A petrol engine is spark ignited, where an increment in temperature can result in spark knock. Spark knock occurs when small pockets of combustion gases self-ignites at the edge of the combustion chamber, which have a destructive effect on the engine. The knock phenomena is described in more detail in section 7.1. The compression ratio is often lowered to avoid knock. Implementation of ceramic material in an spark ignited engine can therefore lead to the need of lowering the compression ratio, which reduces the thermal efficiency. This is shown by equation 1 on page 25.

The benefit of ceramic materials in spark ignited engines is hence less straightforward compared to compression ignited engines, but there is nevertheless room for improvements in terms of fuel efficiency for spark ignited engines.

The gain in diesel engines is greater and that is why thermal barriers are not used as often in petrol engines. However, the small CVT engine that this project was focusing on only runs for approximately 10 seconds with minutes of interval. The engine is therefore very cold in comparison with regular petrol engines and the increment in gas temperature may only be beneficial. In addition, a smaller engine have higher ratio between combustion chamber surface area to displacement volume.

This makes a small engine less fuel efficient compared to larger engines, since the displacement volume determines the possible heat energy release from the fuel and a larger surface area gives higher heat transfer losses. This explains the reason why implementation of ceramics in large spark ignited engines is considered inefficient. However, this project is aiming to implement ceramics in a small engine which runs for a short period of time and this makes the implementation of ceramics interesting with a potential of increasing the fuel efficiency.

If the cylinder liner is chosen to be made in ceramics another issue with the mating material in the piston rings occurs. The hardness ratio between the materials in the cylinder liner and the piston rings is matched to minimize wear on the piston rings and still keep the friction as low as possible. The material in the piston rings has to be increased a lot if the same ratio is used with the much harder cylinder liner in ceramics. This topic will be discussed in more detail in chapter 4.1.5.

Table 1: Material properties of some technical ceramics.

Material:	Melting temp.: [K]	Conductive heat coefficient: [W/mK]	Elasticity module: [GPa]	Fracture toughness: [Mpa$^{1/2}$]	Hardness: [HV]
SiO_2	500	1.4	66-72	0.5	650
Al_2O_3	2050	16	230-260	4.5	1400
ZrO_2	2700	1.7	170-200	4	1200
SiC	300	62.0	390-410	3.4	2800
Si_3N_4	1900	19.8	300-320	5	1500
TiO_2	1850	5.2	275-290	2.7	980

[10]

4.1.3 Technical ceramics

A ceramic material consists of a metal and a non-metal, like alumina (Al_2O_3). The use of ceramics in internal combustion engines is well known and tested in the research society. Table 1 shows the properties of some technical ceramics. Interesting properties when choosing materials for LHRE are primarily conductive heat coefficient, melting temperature and fracture toughness.

The first ceramics to be used in LHRE were alumina (Al_20_3) and silicon carbide (SiC) since these are well known and accessible. Other ceramics such as zirconia (ZrO_2) and titanium oxide (TiO_2) was soon adapted in the research engines. Zirconia and titanium oxide both have low heat conductive coefficient and high melting temperature, this is preferable for usage in LHRE. However zirconia has higher fracture toughness and lower conductive heat coefficient than titanium oxide, two properties very crucial in a LHRE. The elasticity module is higher for titanium oxide, but this property is adequate for zirconia to withstand the forces which occur when the engine is operating. Hence, the project focused on implementing zirconia in the technology demonstrator.

4.1.3.1 Zirconia

Zirconia is one of the most studied ceramics in the engineering field, due to its low electric and thermal conductivity. The property which differs the most from other ceramics is the thermal expansion coefficient, which tells how much the material expand when subjected to heat. The thermal expansion coefficient for zirconia is almost as high as the for metals. For example, conventional cylinder liners are often made of cast iron, which has almost an identical thermal expansion coefficient as zirconia. Hence, the same thermal expansion could be expected if a cylinder liner in zirconia could be made with the same dimensions as the conventional cast iron cylinder liner. In the case of normal cast iron liner with zirconia coating the cast iron liner and the coating would expand equally and therefore lower the risk of expansion fractures in the coating layer. The different expansion coefficients are shown below, in Table 2 where zirconia is compared to the most commonly used metals in engines.

The main drawback with pure zirconia is the shift in molecule structure at 1200°C, which increases the volume by approximately 3 % [4]. A shift in volume can lead to blow-by, when the combustion gases blows by the piston rings and enter the crankcase. Zirconia have three molecular structures: Monoclinic structure, tetragonal structure and cubic structure which are shown in Figure 8.

Table 2: Thermal expansion coefficient.

Material:	Thermal expansion coefficient $[K^{-1}]$
Zirconia	$11.0 \cdot 10^{-6}$
Cast iron	$10.8 \cdot 10^{-6}$
Steel	$12.0 \cdot 10^{-6}$
Stainless steel	$16.0 \cdot 10^{-6}$
Aluminium	$22,2 \cdot 10^{-6}$

[10]

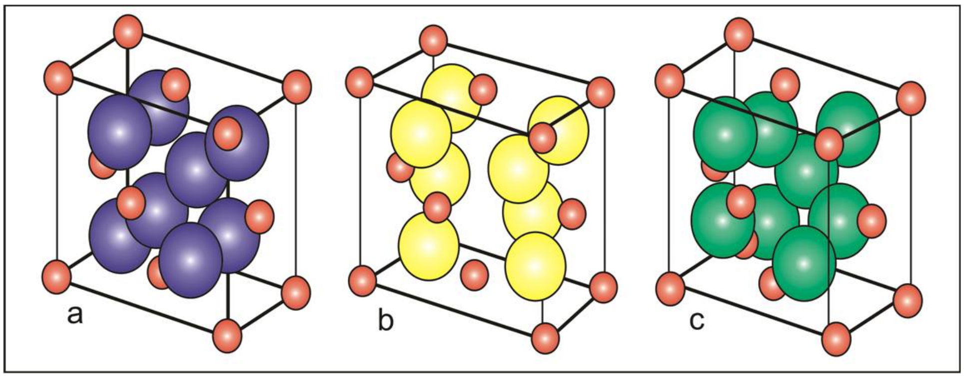


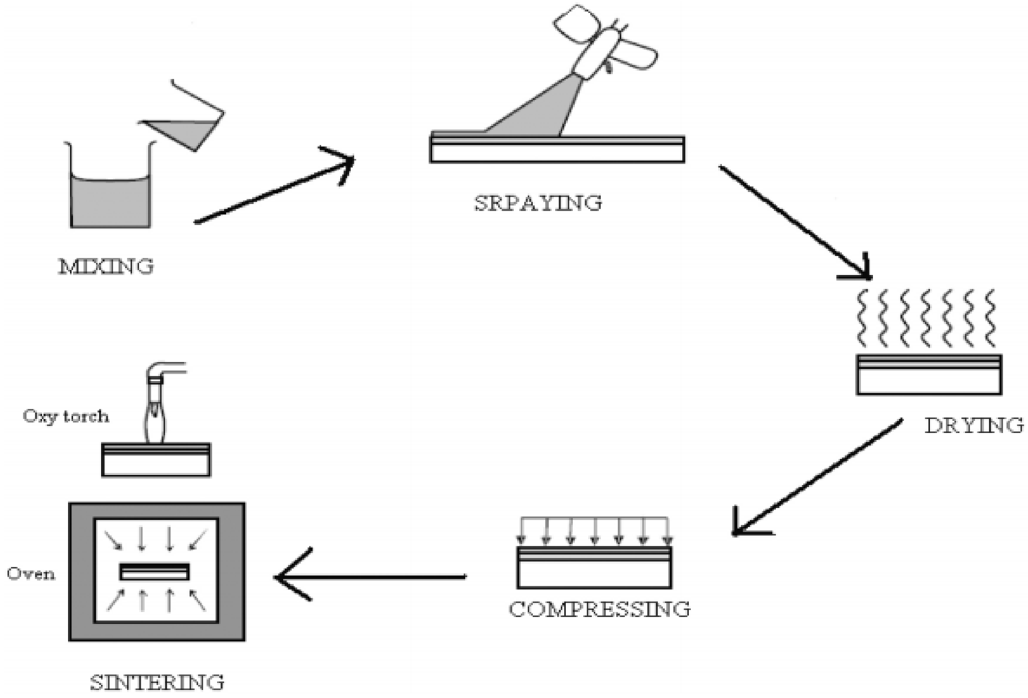
Figure 8: Zirconia molecular structures; a) Monoclinic, b) Tetragonal and c) cubic

The first shift between monoclinic to tetragonal structure takes place at 1200°C which is a normal temperature of the gases in the combustion chamber during the combustion. Hence, there was a need to stabilize the pure zirconia. When the zirconia is stabilized, it stays in the monoclinic structure at higher temperatures. To stabilize it, another ceramic material is mixed with zirconia, like CaO, MgO or Y2O3. The last one is called Yttria Partially Stabilized Zirconia (Y-PSZ) . It is possible to fully stabilize the zirconia, but this is seldom necessary. Instead, zirconia is mostly partially stabilized, Partially Stabilized Zirconia (PSZ) [4].

4.1.4 Solid parts or coatings?

Solid parts and coatings have different pros and cons. Coatings are generally easier to implement since they can seize the mechanical structure of the metal part. A coating is on the other hand much thinner than the thickness of a part made of solid ceramics. The conductive heat resistance depends on the thickness of the thermal barrier. A solid part in ceramics will thereby have a greater thermal resistance which decreases the heat losses more than a coating would.

Few investigations has been carried out comparing solid ceramics and ceramic coatings, except Havstad in 1986 [6]. Havstad concluded that solid parts in ceramics had the upper hand. Despite this, all subsequent investigations used coatings, probably due to its simplicity and robustness. The technique to manufacture a solid part in ceramic material is described in section 4.3.



[9]

Figure 9: The slurry spray technique (SST) manufacturing process.

4.1.4.1 Coating techniques

The most common way to create a ceramic layer of coating is to use plasma spraying, which requires expensive tools but generates the best results.

Other simpler coating techniques is the Slurry Spray Technique (SST) which could be used instead. The results of this treatment was not expected to be as good as for plasma spraying, but could still give a sufficient result. The SST have many advantages in terms of simplicity. SST utilizes spraying of wet powder which consists of ceramic particles. These particles are mixed with a binder to form the slush which is then sprayed on to the surface using a regular spray gun. The wet surface dries in atmospheric air and is then put under pressure to enhance the contact between the ceramic particles. Lastly, the coated part is heated in an oven and put under an oxyacetylene torch to bind and sinter the coating [9]. All steps are showed in Figure 9. The whole process is rather simple and is not in need of expensive equipment.

4.1.5 Piston rings

The purpose of the piston rings is to seal the combustion chamber to prevent combustion gases from entering the crankcase and to prevent crankcase oil to enter the combustion chamber. The piston rings controls the oil balance where the optimum is to keep the oil consumption as low as possible and still lubricate the cylinder liner and the piston rings. The piston rings also transfer 20%-60% of the absorbed heat of the piston surface to the cylinder liner [11]. These are numbers from a conventional engine without ceramics materials in either the cylinder liner or on the top of the piston.

Figure 10: Piston ring.

The piston rings are designed with a small gap, showed in Figure 10, this gap will be closed as the engine has reached working temperature and pressure. It is therefore crucial to design the piston rings with the correct strength and elastic behavior.

The oil from the crankcase creates a lubricating oil film between the cylinder liner and the piston rings. Engine speed and load determines the friction. The hydrodynamic friction, which depends on the oil viscosity, is a greater part of the total friction at higher engine speed. Where the mechanical friction is higher at lower engine speed.

The implementation of ceramic materials strongly depends on the piston rings which abuts the cylinder liner. Friction and wear is initiated as the piston moves up and down in the cylinder. To implement ceramic materials, a mating material of the piston rings needs to be found. The mating material should keep the wear rate on the piston rings down and minimize the friction losses.

4.1.5.1 Low-friction

The friction in the piston ring-piston liner system stands for approximately 40 % of all mechanical losses in the propulsion system, which relates to 5-10 % of the fuel consumption [12]. A decrement in piston ring friction will therefore improve the fuel efficiency of the engine.

The project aimed to find several solutions which decreased the piston ring friction and still made the other engine improvements possible. One option was to use self-lubricated low-friction rings made out of PTFE (Polytetrafluoroethylene).

Self-lubricated low-friction rings made out of PTFE (Polytetrafluoroethylene) has the lowest friction coefficient. In addition a self-lubricating piston ring is not in need for lubrication by oil which enables the use of an open crankcase. An open crankcase would decrease the pumping losses which occurs when a pressure builds and air is pumped in and out the oil trap up as the piston moves up and down.

The drawbacks with PTFE is that it cannot withstand higher temperatures than 250°C. On a large diesel LHRE, the upper piston ring can reach temperatures of up to 550°C[13]. But, as mentioned earlier, the CVT engine is colder than usual diesel LHRE engines and the aim was therefore to still test these PTFE piston rings.

4.1.5.2 Mating material

Piston rings and cylinder liner are designed to minimize wear on the piston rings, because if the piston rings wears down, too much blow-by will occur or the piston rings will break down completely. Instead the piston rings should wear down the cylinder liner, however this wear should of course be kept as low as possible. To achieve the desired wear between the cylinder liner and the piston rings, a ratio in hardness is used. The piston rings should be harder than the cylinder liner to get the desired effect.

Conventional cylinder liners in cast iron has a hardness of roughly 250 HV (Hardness Vickers) and should be combined with piston rings of about 900-1000 HV. With the use of ceramics on the cylinder liner surface, the hardness will be around 1200 HV and conventional piston rings will quickly wear down. Therefore harder piston rings is usually used in LHRE, but often with the result of increased friction [13], even though there is generally no direct correlation between hardness and friction.

A ceramic coating on the piston rings would harden them, but the coating is difficult to retain due to their elastic behavior. Thus, the hardness of the piston rings has to be increased in some other way. The most effective way to make harder piston rings is with DLC (Diamond Like Carbon) treatment [13].

4.2 Design

The overall design of the combustion chamber aimed to test out a complete enclosure of ceramic materials in the combustion chamber. This means that the cylinder liner, the piston crown and the cylinder head all should have ceramic surfaces. The aim was also to compare ceramic coatings with solid parts in ceramic materials.

4.2.1 Cylinder liner

The cylinder liner is one of the fixed parts in the combustion chamber. The combination of this and its simple geometries led to the decision to design one cylinder liner completely in Y-PSZ. Y-PSZ had higher compressive strength than cast iron, but the fracture toughness is lower. This makes the material brittle, which is a typical material property of ceramics. The ceramic cylinder liner was designed to be slided in to the cylinder, instead of being pressed into place which is the normal fastening technique for cylinder liner in cast iron. The ceramic cylinder liner was designed with a flange ring which should be pressed between the cylinder and the cylinder head in a screw joint when mounted. Simple FEM analysis were performed to ensure that the ceramic cylinder liner would not crack when mounted as well as the existence of high pressure during combustion was performed in the design of the cylinder liner. To prevent fracture, the surfaces in the cylinder which the cylinder liner is slided in against, had to be smooth. More material properties of the selected Y-PSZ are found in Table 3.

To compare a solid part of ceramics against ceramic coating another cylinder liner was designed in cast iron with a thermal barrier layer of Metco 450NS + Metco 233A YSZ which is a coating from the company Oerlikon. This coating was another type of Y-PSZ. The thickness of the layer was selected to be divided into different layers:

- 100-150 µm bond coat
- 250-300 µm top coat

These recommendations of thicknesses for the layers was also received from the company Oerlikon [15].

Table 3: Properties of Y-PSZ (KZY 8%)

Reference	Units	Y-PSZ KZY 8%
PHYSICAL PROPERTIES		
Melting point	°C	2700
Max. operating temperature	°C	1000
Density	g/cm³	5,9
MECHANICAL PROPERTIES		
Polished surface quality (Ra)	µm	0,04
Hardness Vickers 20°	GPa	13
Bending strength (4 points)		
20°C	MPa	1000
800°C	MPa	270
Tension strength	MPa	420
Compressive strength		
20°C	MPa	4000
800°C	MPa	2000
Young's Modulus 20°C	GPa	210
Fracture toughness 20°C K_{1c}	MPa.√m	6,8
THERMAL PROPERTIES		
Thermal expansion coefficient	$\times 10^{-6}$/°C	11,5
Thermal shock resistance	ΔT°C	
Thermal conductivity		
20°C	W/m°K	1,7
500°C	W/m°K	1,7
1000°C	W/m°K	1,7
Specific heat 20°	J/Kg/°K	650
ELECTRICAL PROPERTIES		
Electrical resistivity		
20°C	Ω.m	
1000°C	Ω.m	
Dielectric strength	KV/mm.	
Dielectric constant 25°C		
1 KHz		
1 MHz		
1 GHz		

[14]

4.2.2 Piston crown and cylinder head

The areas that were selected to be coated with a thermal barrier layer is the parts that are included in the combustion chamber. These are the surfaces of the valves, cylinder head, the piston head and the cylinder liner wall. The thermal barrier layer and its thickness that is used here is the same as for the second cylinder liner.

4.2.3 Piston rings

The difficulty in having a hard cylinder liner in zirconia and still keep the friction and the wear on the piston rings as low as possible, lead to the decision that several designs of the piston rings should be tested. The PTFE piston rings with the lowest friction coefficient had the greatest possibility to minimize the friction losses. But, they did not have the specified properties to withstand the high temperature in the combustion chamber. Although tests were still considered useful to conduct to see what effect these piston rings would have on friction losses.

Due to the uncertainty of the reliability of the PTFE ring, conventional piston rings were the most realistic choice. The possible risk with these could be higher friction and substantially more wear on the piston rings. The operating time for the CVT engine is nevertheless very short and wear might not be a problem, therefore this was decided to be tested out. DLC treatment was a good way of making the piston rings harder, but this treatment could not be afforded within this project.

4.3 Manufacturing

One cast iron cylinder liner along with the piston crown and the cylinder head was designed to be coated with a thin layer of Y-PSZ. The manufacturing process for this treatment could be done in several ways where the best results was given by plasma spraying. Plasma spraying is used for coating the blades in gas turbines and also the most common way of creating a TBC in LHRE. However, this coating technique was expensive and was unfortunately not affordable within this project. Another coating investigated coating technique was the Slurry Spray Technique described in section 4.1.4.1.

The ceramic liner is one of the parts that could not be manufactured on site in Chalmers prototype workshop, in this case it was necessary to consult external manufacturers. Via the Swedish company KG-Fridman a company named Nanoker Research in Spain was contacted and a ceramic cylinder liner based on a specified drawing was manufactured. The material which was chosen to be used was zirconia KZY-8% and it is a form of Y-PSZ.

When ceramic parts are manufactured, there are many important factors to be taken into account and, depending on the geometry and desired tolerances there may be additional steps in the manufacturing process. As ceramics are generally very brittle and complicated to manufacture, there is a high risk of workpiece damage during the process, therefore at least one extra piece of raw material is always made to use as a backup. The manufacturing process is visualized in Figure 11.

[16]

Figure 11: Production Process when ceramics are used

The first step in the manufacturing process of the ceramic cylinder liner was to compress the ceramic powder mixture by a process called "cold-isostatic pressing" (CIP). Thereafter a solid steel cylinder was produced with dimensions which took account of the measurement processing and shrinkage of the material. Around the steel cylinder a "polyurethane sock" is attached and the space that exists between the cylinder and the sock is then filled with the ceramic powder mixture. To further compact the powder around the steel cylinder, the powder is subjected to a pressure of about 2000 bar in a pressure chamber. At this stage in the manufacturing process the powder has taken the shape of a cylinder and the material is then in what is called the green stage, which means that it can be processed with conventional tools, but it is still very fragile. In order to further process the material a new, slightly larger steel cylinder was made, which served as a holder when the flange and the other outer shapes of the cylinder liner was processed. In all these above mentioned steps in the manufacturing process the account of the shrinkage rate has to be calculated and for the selected material it is 21% in all directions.

Next step was to sinter the workpiece in a furnace at a temperature of about 1500°C for a holding time of two hours. After this step, the part has become hard and then it can only be machined with diamond tools. It was during this processing, that the final dimensions and surface tolerance requirements were achieved.

The last step in the manufacturing process was polishing the surfaces that require good surface finish with diamond paste, which in this case were the interior surfaces of the cylinder liner.

4.4 Results and discussion

The purchased cylinder liner in Y-PSZ was delivered at the end of the project. The quality of the liner seemed promising and future test were planned after the end of the project. Figure 12 shows the ceramic cylinder liner alongside the cast iron cylinder liner in Figure 13. No external order of ceramic coating afforded in the project but the SST coating technique was regarded as doable, but could not be done within the time frame of this project.

Implementations of ceramic materials in the CVT engine is considered to be highly achievable. This can be done both as solid ceramic parts and as ceramic coating. Coating techniques such as SST could be performed with inexpensive tools and without external assistance. However, the effects of implementing ceramic materials in the CVT engine still has to be investigated. Previous tests shows mixed results, but these test cannot be directly applicable to the CVT engine because of its size and drive cycle [4].

Implementing ceramic materials will most likely result in higher exhaust gas temperatures. Therefore the use of the Atkinson cycle or a turbocharger would utilize more of the potential in ceramic materials. Implementation of ceramic materials in engines are best suited for diesel engines, since an increased combustion chamber gas temperature can increase the risk of knock for petrol engines. However, the CVT engine drive cycle might benefit from higher gas temperatures since the engine is substantially colder than conventional engines. This leads to the conclusion that ceramic materials can be implemented in the CVT engine and has the possibility to improve the fuel efficiency of the engine.

The ceramic cylinder liner was designed to withstand the compressive stresses of the preload between the cylinder head and the cylinder. The initial plan was to preload the liner and run the engine to test the design and durability of the liner subjected to the high pressure peaks occurring during the combustion. This test could not be carried out due to longer manufacturing time of the engine technology demonstrator than planned.

Figure 12: The manufactured cylinder liner made of ceramics

Figure 13: The manufactured cylinder liner made of cast iron.

In future work the effect on fuel consumption, friction and wear are recommended to be tested. An interesting test would be to compare an engine with parts made of ceramic material with a conventional engine at the same AFR and measure the cylinder pressure and the exhaust temperature. In this way it is possible to track the released energy of the fuel and to compare energy split between friction and heat losses, mechanical work and energy in the exhaust gases.

A comparison between solid ceramic parts and coated parts are suggested to be carried out, partly in terms of fuel efficiency but also in aspects of durability, wear, manufacturing simplicity and cost.

The impact on piston rings with a ceramic surface on the cylinder liner is also an important aspect. First, testing the possibility to combine a ceramic liner with low-friction PTFE piston rings could decrease the friction losses considerably if these piston rings can withstand the temperature and pressure with long enough life span. Wear and friction rates on conventional piston rings abuts the cylinder liner are also suggested in future work.

5 Atkinson cycle

This section gives an overview of the theory behind the use of the thermodynamic Atkinson cycle. Also different implementations of the cycle are described, and an analytical way of optimizing an Otto engine for conversion to Atkinson cycle operating is discussed.

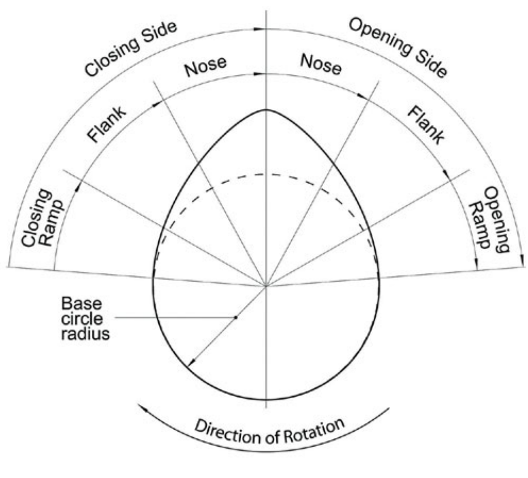
5.1 Theory

During one revolution, a piston engine both inhales and emits gases in a process called gas exchange. This process is of most importance when designing the engine to meet the desired characteristics. The gas exchange is mainly dependent on engine geometry, rotational speed, intake and exhaust ducts and valves, together with the valve timing.

During the intake stroke in a naturally aspirated 4-stroke petrol engine, air is sucked in through the intake duct and intake valve. The intake duct is in most types of engines of a fixed design while the valves opening and closing timing is controlled by a rotating camshaft with attached cam lobes.

The valves used in 4-stroke engines are mostly of a poppet valve design whose movements are determined by the rotation of the cam lobes. The rotating cam lobes either directly pushes upon or uses a rocker arm lever mechanism to open the spring loaded poppet valves. The spring then pushes the valves back to their closed position.

The cam lobes design determines when, how long and how much the valves will open at each cycle. Instead of measuring the duration and timing in time units it is usually expressed in terms of crank angle degrees, CAD, since the duration in time will depend directly on the rotational speed of the engine. The crankshaft, in a 4-stroke engine, rotates two times for every rotation of the camshaft and hence the attached cam lobes. This means that for the valve to be open during the full intake stroke, which is 180 CAD, the cam lobe outer profile, shown in figure 14, should be greater than the base circle during at least 90 camshaft degrees.



[17]

Figure 14: Cam lobe designations

In reality though, the opening and closing of the valve usually happens before and after the intake stroke respectively. Both to utilize gas flow dynamics to get an even better gas exchange process, but also not to open and close the valve too rapidly in order to decrease the accelerations and hence the forces acting on the spring, valve and lobe as much as possible.

The rate of change in acceleration during the valve opening is called jerk and is another important parameter to control in order to reduce the wear of the cam lobes. The acceleration should, in order to reduce the jerk, be as constant as possible and the transition between the base circle, shown in figure 14, and the opening ramp should therefore be as smooth as possible.

In many conventional Otto cycle engines, the focus when designing the gas exchange system is to get as much power output from the engine as possible. In order to do that the engine has to get as much air into the cylinder as possible before the intake valve closes and compression begins. A greater air charge can provide greater combustion and hence give more power output per cycle.

The amount of normal air, i.e. air at normal atmospheric conditions, that the engine inhales before the intake valve closes compared to the amount that it could inhale if the whole cylinder is filled is called the engines volumetric efficency, VE. A volumetric efficiency of over 100% can be achieved if the intake system is properly designed, and tuned, to utilize the momentum and pressure waves in the exhaust and intake air ducts to push more air into the cylinder.

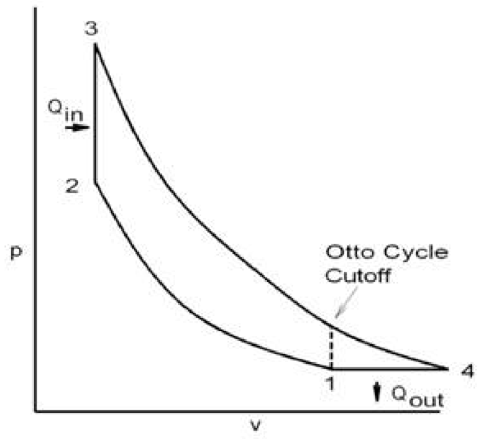
The volumetric efficiency is also dependent on the speed of the engine, since higher speeds require the intake air charge to get sucked in during a shorter period of time which can result in a smaller air charge and the volumetric efficiency therefore often varies over the engine's operating speed range.

Since the cam lobes mostly are of fixed design it is difficult to optimize the gas exchange over the full range of the engine's operating speeds and the engine is therefore often optimized at some speed interval where the engine is supposed to run the most or where the power is needed the most.

5.1.1 The Ideal Atkinson cycle

The conventional thermodynamic process used in today's 4-stroke petrol engines is often idealized as the Otto cycle, which has the same length for the compression and expansion stroke. It has a relatively low efficiency due to the inevitable heat losses in the exhaust gases.

One way to convert more of this energy loss into mechanical work, and thereby increase the thermal efficiency, and decrease the BSFC, is to have a longer expansion stroke compared to the compression stroke, which is idealized as the thermodynamic Atkinson cycle. In this process more of the generated in-cylinder overpressure will be converted to mechanical work, through the piston and crankshaft, and less released to the exhaust gases. The area to the right of the Otto cycle cutoff in figure 15 is the extra work that can be derived compared to the Otto cycle to the left of the line.



[18]

Figure 15: The ideal atkinson cycle

By finding an optimum ratio between the cylinder volume used for compression and expansion, ideally all of the overpressure in the cylinder could be utilized so that when the piston gets to BDC, and the exhaust valve opens, there is atmospheric pressure in the cylinder, point 4 in figure 15, and hence the exhaust heat losses, Q_{out}, would be lowered as much as possible.

In practice though, the engine will have a friction resistance called FMEP, a gas flow resistance called PMEP and in most cases some kind of load called BMEP that can be thought of as working in a negative direction of the in-cylinder pressure, IMEP, pushing down against the piston head area. If the in-cylinder pressure is lower than required to accelerate the piston enough to overcome the resistance and to keep a steady average angular speed over multiple full strokes, the engine will eventually come to a stop.

However, this in-cylinder pressure, at the time of the exhaust valve opening, EVO, should be just high enough to keep the engine turning at the desired load in order to achieve as low thermal losses as possible with the exhaust gases.

5.1.2 Thermal Efficiency

If an Otto cycle engine is converted only by change of valve timing to work according to an Atkinson cycle it will have to use a lesser cylinder displacement volume instead of a greater expansion volume. This will result in a lower compression ratio, which is the ratio of the cylinder volume before and after compression, that will lower the cycles thermal efficiency. It can be shown thermodynamically with the idealised Otto cycle that the thermal efficiency will depend solely on the compression ratio according to equation 1.

$$\eta_{Otto} = 1 - \frac{1}{r_k^{\gamma-1}} \tag{1}$$

where r_k = the compression ratio and $\gamma = 1.4$ for air.

To achieve the same compression ratio with less cylinder displacement volume, the top cylinder and/or cylinder head compression volume will have to be proportionally smaller. This can be a problem to achieve if the compression volume already is minimized so that lowering the cylinder head will make e.g. the valves collide with the piston.

On the Vera MKVI engine the calculated dynamic compression ratio is about 10:1 with an assumed volumetric efficiency of 80% at 3000 rpm which is quite normal on an engine with not perfectly tuned valve timings. According to equation 1 this compression ratio ideally gives a thermal efficiency of about 61% working according to the conventional Otto cycle.

The ideal thermal efficiency of the Atkinson cycle working with the same compression ratio and an expansion until you reach atmospheric pressure would give almost 72% thermal efficiency according to equation 2 but would in this case require the expansion volume to be almost 4 times greater than the compression. This would decrease the volume used for compression to only about 27% of the total cylinder volume and lowering the power output to, in the best case, the same percentage.

$$\eta_{Atkinson} = 1 - \gamma \frac{r_e - r_k}{r_e^{\gamma} - r_k^{\gamma}} \tag{2}$$

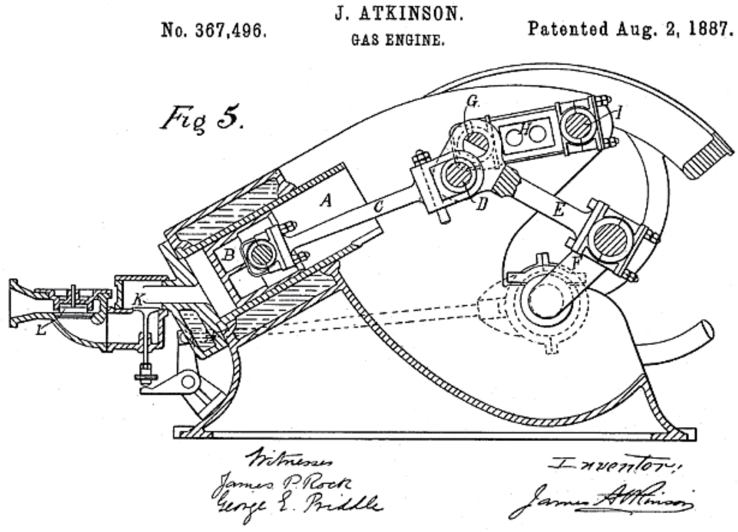
where r_k = the compression ratio, r_e = the expansion ratio and $\gamma = 1.4$ for air.

Since the absolute friction losses during one cycle in this type of Atkinson converted Otto cycle engine would approximately be the same as a conventional Otto engine, they would constitute a bigger proportion of the energy balance since the combustion energy is decreased with a smaller charge. This would, in the non-idealized cycle, mean that the thermal efficiency decreases with increased use of the Atkinson cycle.

5.1.3 Advantages and disadvantages

An overall disadvantage involved using the Atkinson cycle is that the engine power output per cylinder volume unit will be lower since only a part of the cylinder volume will be used for compression. A lesser amount of compressed air results in a smaller combustion compared to a conventional Otto cycle engine. This means that for an Atkinson cycle engine to have the same power output, it will have to be larger and heavier. A larger engine will also have greater friction losses compared to a smaller one, with the same power output, which will decrease its efficiency and increase its BSFC.

On the other hand, the Atkinson cycle engine will convert a larger part of the combustion energy into mechanical work and will hence require less fuel for the same amount of work, lower BSFC, compared to the idealized Otto cycle engine. The increase of BSFC due to greater friction and the decrease due to a more effective conversion of thermal energy counteract each other and the result can be either lower or higher BSFC depending on implementation, engine overall efficiency and calibration etc.



[19]

Figure 16: The original Atkinson engine patent

5.2 Design

To implement the Atkinson cycle, different modes of implementations were evaluated with advantages and disadvantages. The chosen type of design was then discussed and modelled.

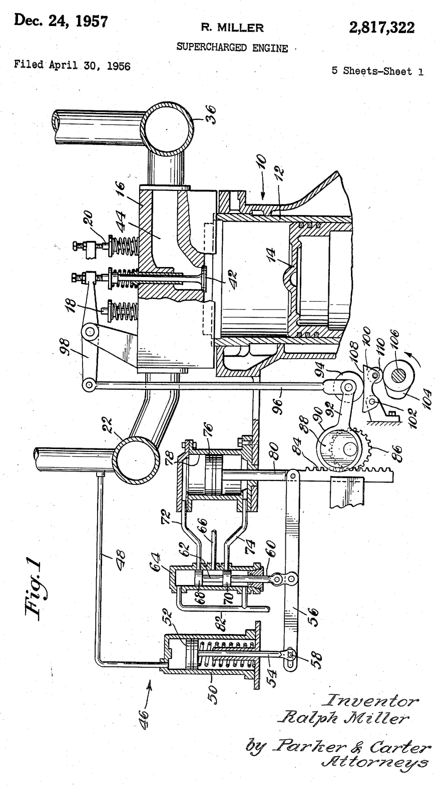
5.2.1 Implementations

There are several ways to realize the Atkinson cycle. The oldest is the concept that James Atkinson used in his Atkinson engine, shown in figure 16, patented in 1887, where he used a multiple linkage design to achieve a longer expansion than compression stroke [19]. Even though this engine was more effective than the conventional Otto cycle, it delivered less power and were far more mechanically complex than an equally sized Otto engine and hence very seldom used [20].

However, Honda in 2011 released their new Exlink engine using the principles of the Atkinson engine with a multiple linkage system to achieve a difference of the stroke lengths. In addition to this, a better connecting rod angle during the expansion stroke of only about 2.4° gives significantly less piston side forces acting on the cylinder and hence reduced friction losses and increased the mechanical work done [21].

Another, more common, implementation of the Atkinson cycle is a modified Otto cycle invented by Ralph Miller in the 1940s [20]. It used, in addition to the conventional intake and exhaust valves, a compression control valve, shown in figure 17, to let part of the trapped intake air out of the cylinder again during the compression stroke and hence achieve a shorter effective compression than expansion stroke, i.e. a lower VE [22]. The design also included a supercharger and intercooler for an increased density of the intake air to compensate for the lesser air volume trapped during compression.

The simplest and far most common implementation of the Atkinson cycle is the modified Miller cycle which, instead of an additional compression control valve and its regulating mechanism, uses a modified intake cam lobe to either close the intake valve well before or after the piston has reached BDC [20]. This technique is usually called early or late intake valve closing, EIVC or LIVC. This way, the air flow through the intake channel will either



[22]

Figure 17: The original Miller engine patent

be stopped before the cylinder is filled or be reversed into the intake air duct and thereby partly emptying the cylinder during compression. In both cases only part of the cylinder volume will be used for compression and since the exhaust cam lobe is unmodified the full cylinder volume will be used for expansion.

5.2.2 Choice of design

Well aware that the result might vary from either lower or higher BSFC, and due to the limited time in the project, the implementation method for the technology demonstrating engine was chosen to try and run the engine according to a modified Miller cycle. This meant only designing new cam lobes and not manufacture a complete new engine with some sort of multiple linkage crankshaft design nor use the Miller patent design with a compression control valve including intercooler and supercharger which was thought of as to complicated and time consuming.

The chosen method was also found to be used in Toyota's hybrid car engines where the engine can be optimized for this type of cycle in a shorter engine speed interval since the electric motor can take over at the unfavourable speed intervals [23]. The engine then only operates in this optimized interval which gives a higher thermal efficiency than a conventional engine but would have an overall lower thermal efficiency compared over the whole engines speed range. This type of engine operating is somewhat similar to the normal operating mode used by CVT where they try to run the engine in intervals at between 2000 - 4500 rpm to keep the car going at a mean speed of about 25 km/h.

The modified Miller cycle can be implemented to different extents depending on how much air that is trapped in the cylinder after IVC. So the focus of the continued work was to make a theoretical base for how the optimum VE ratio could be found on the finished engine and to manufacture different cam lobes to evaluate on the finished engine.

5.2.3 Optimization modelling

The calculation of optimum VE requires very precise engine test data. The project has been unable to obtain these but a Matlab model was developed to make a good assumption and to be easy to calibrate with accurate test data if/when possible, see appendix A. The model could then be used in further development work.

To effectively compare the thermal efficiency of a conventional Otto cycle engine with an modified Miller cycle engine, which is converted to use only part of the cylinder displacement volume for compression i.e. utilise a lower VE, the compression ratio has to be the same. So in the model it was assumed that the compression volume, the cylinder head volume and cylinder volume above TDC, could be made sufficiently small to achieve the same compression ratio even if a smaller air charge is trapped during intake.

A FMEP value of 1 bar, for a similar engine, was assumed to be able to make a comparison of different VE ratios [24]. After test bench measurements of the finished engine this value should be updated and the optimization modelling iterated.

With these assumptions an idealized Atkinson cycle with the CVT engine MKVI characteristics and the FMEP losses above were modelled and a comparison of the total thermal efficiency could be made at different VE.

5.2.4 Cam lobe and air flow modelling

An optimized design of the cam lobes will be a compromise between reducing the wear and forces, and a desired gas exchange for the engine. During the design of the cam lobes in this project though, the focus have only been to achieve a desirable gas exchange and not reducing the wear on the components since the engine is not designed for endurance and already have tight service intervals with the possibility to exchange worn out parts.

To achieve a desired VE by redesigning the cam lobes, another model was set up to estimate the intake air charge with different cam lobes, see appendix B. As a base for the model the design of the MKVI engine intake system with rocker arms and cam lobes was used.

The base profile of the cam lobe was set to be the same as the one on the MKVI engine but to simplify the simulations of the opening and closing profile, a fourth-degree polynomial was used which have the same gradient as the base circle at the intersections between the base circle and the outer profile. It also was set to intersect the point of the desired maximum lift right between the opening and closing point and to have a zero slope there.

When the profile was determined, the valve lift at all CADs could be calculated an a model for the air flow into the cylinder be established. To compensate for the non ideal flow through the intake duct and valve, discharge coefficients were taken from an older, similar, engine that was experimentally tested in 2011 [25].

The air flow calculations were made by compressible flow formulas, for air, compensated with the discharge coefficient mentioned above and effects of choked air flow [26, eq. 9.46a & 9.47]. No considerations have been made to the effects of gas flow momentum or pressure waves in the intake and exhaust ducts nor the effects of any exhaust gas recycling.

An assumption in the model of heat transfer by convection from the cylinder walls to the intake air charge have been made with a convective coefficient $h = 100\frac{W}{m^2K}$ and an assumed in-cylinder constant wall temperature of 200 °C. This assumption should also be calibrated when test result from the test bench and/or temperature measurements from the engines can be made.

With a calculated intake air charge for a chosen cam lobe profile, the VE of that profile could be established. In this way, a profile could be found to match the calculated optimum VE found in the optimization model above.

Since all the assumptions and simplified calculations in the air flow model makes the models results very rough, and probably somewhat far from reality, calibration of the model was deemed necessary. So instead of just manufacture a cam profile that would give the calculated optimum VE it was decided to manufacture completely different ones to compare and calibrate with.

Four different cam lobe profiles were then designed using EIVC, LIVC, a normal duration but slightly reduced maximum lift and one was designed according to the original MKVI design.

5.3 Manufacturing

The four designed profile could then be modeled with the same fourth degree polynomial profile, as in the Matlab model, in AutoDesk Inventor Professional and be exported as a STEP file for manufacture in the CNC milling machine using the processing program MasterCAM X8. The result is shown in figure 18.

The manufactured profiles had the following specifications with duration in CAD from TDC at the intake stroke:

EIVC: -25 / 180 CAD, maximum cam lobe lift 8 mm.
LIVC: -25 / 250 CAD, maximum cam lobe lift 8 mm.
Reduced lift: -20 / 200 CAD, maximum cam lobe lift 4.6 mm.
Original Design: -40 / 220 CAD, maximum cam lobe lift 8 mm.

These cam lobes were then intended to be tested in the finished engine for evaluation of the effects and calibration of the cam lobe design and air flow model to be closer to reality and useful for further development.

Figure 18: The manufactured cam lobes

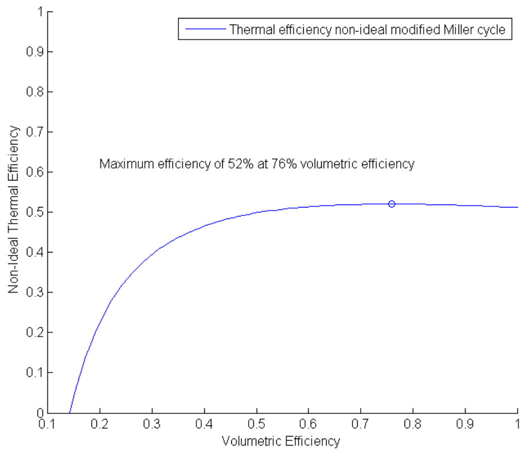


Figure 19: Thermal efficiency non ideal modified miller cycle

5.4 Results and discussion

From the optimization model a plot, figure 19, of thermal efficiencies at different VE showed that the optimum VE was 76% and gave a thermal efficiency of about 52%.

Since the assumption of 80% VE, at 3000 rpm, on the original engine is very close to the optimum VE of 76% the improvement of the thermal efficiency would be very low but on the other hand that was just an assumption and the aim of the work was to establish a foundation for further development when the finished engine could be tested and calibrated. Also the uncertainty of the friction and pump losses in the optimization model and the required power output of the engine will have to be taken into consideration before a true optimum VE ratio can be determined.

The cam lobe profiles and in-cylinder pressure and temperature results from the model, to be compared with the real testing results when done, for the different cam lobes is shown in figure 20, 21 and 22.

The VE for the EIVC, LIVC, reduced lift and the original design of the cam lobes were calculated to 72%, 92%, 39% and 86% at 3000 rpm respectively.

As seen in figure 21 the different cam lobes gave rise to relatively different pressure curves, which is visualized especially during the intake stroke between 0 and 180 CAD in the right plot of figure 21. Without any heat transfer from the cylinder walls, the temperature plots in figure 22 would be organized in the same way as the pressure plots. But as the cylinder walls were set to be at a higher constant temperature a smaller air charge gets hotter faster since the heat flux from the cylinder wall to the air is assumed only to depend of the temperature difference and not the density of the air. This can explain why the smallest air charge, achieved by the reduced lift cam lobe, still reaches the highest temperature at 360 CAD even if the end pressure for that lobe reaches the lowest value of the four lobes and hence in an adiabatic compression process also would have the lowest temperature.

The main continuous work to be carried out is to try and fit these pressure and temperature curves to the real test curves when the engine is tested. Then the starting and end pressure points of the curve can be set to match the model and the discharge coefficients of the air flow and heat transfer coefficients from the cylinder wall be redefined so the curve behaves as the real test curve in between the endpoints.

Other aspects such as blow-by between the cylinder and piston, exhaust gas recycling and effects of air momentum and pressure waves in the air ducts may also be of interest to include in the model if it is considered having significant impact on the results.

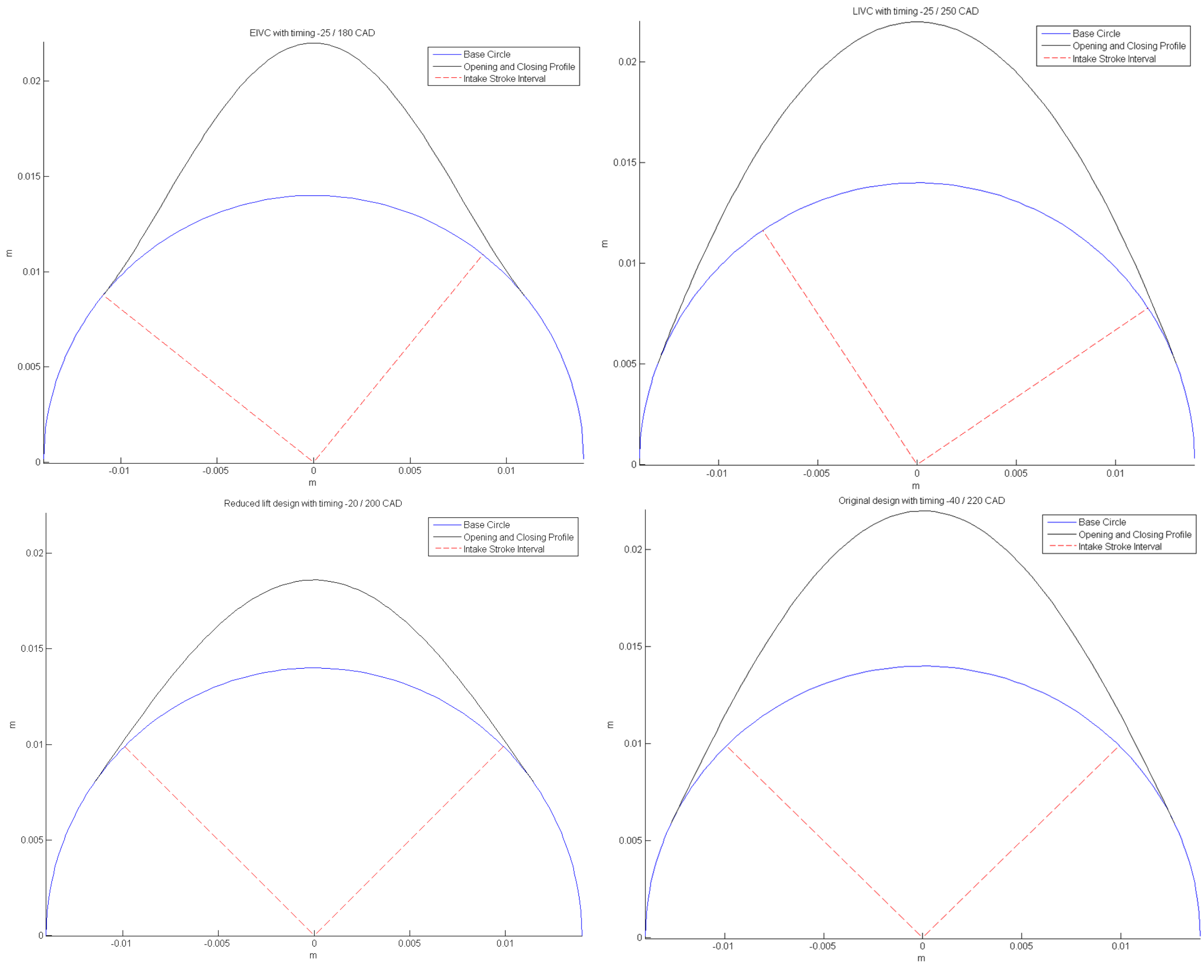


Figure 20: EIVC, LIVC, Reduced lift and Original design cam lobe profiles

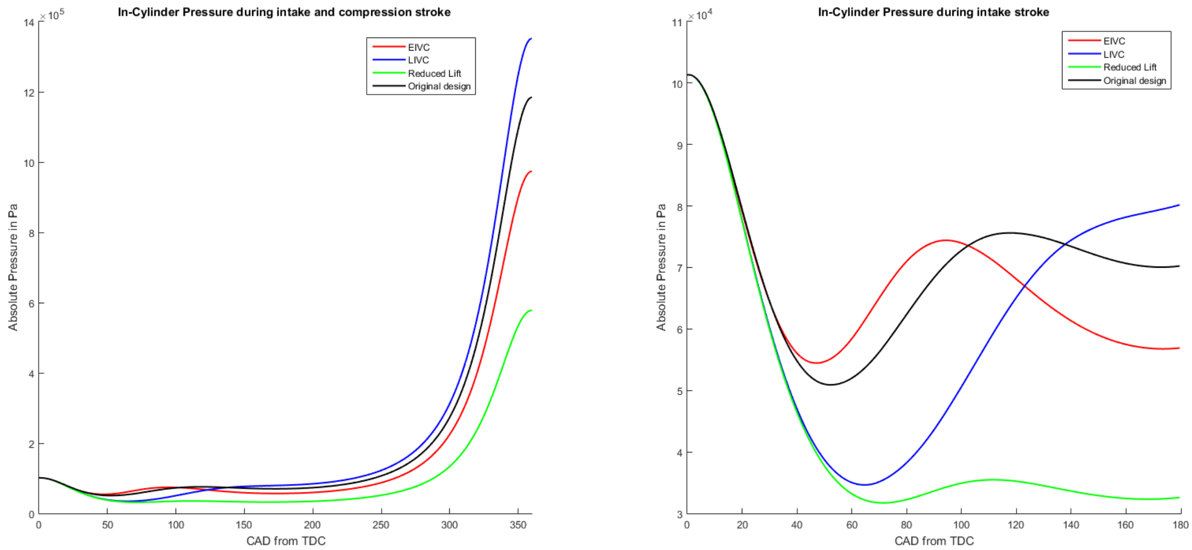


Figure 21: In-Cylinder pressure curves

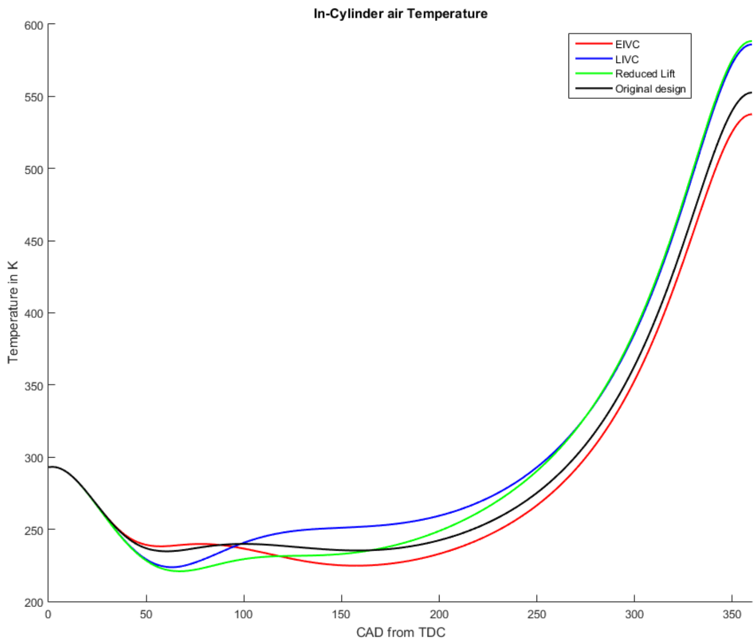


Figure 22: In-Cylinder temperature curves

6 Laser ignition

This section describes the theory behind laser ignition, and the possibilities to implement it in the technology demonstrator.

6.1 Theory

Conventional spark-ignited (SI) engines utilizes spark plugs to ignite the fuel mixture. Spark ignition has been used since the invention of internal combustion engines but modern spark plugs still affect combustion performance adversely. Because of their location inside the combustion chamber, spark plugs absorbs a lot of heat from the combustion process.

If the electrodes of the plug becomes too hot the engine may ignite prematurely [27]. To avoid this, spark plugs are designed to transfer heat from the combustion chamber to the outside. This increases the heat losses in the engine and reduces thermal efficiency.

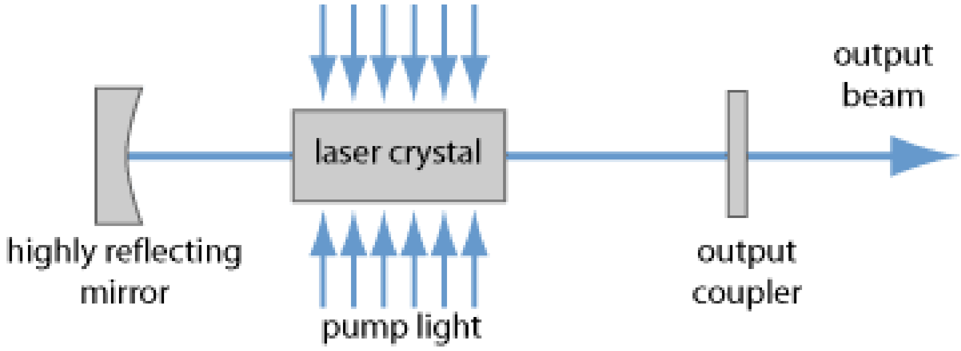
The electrodes that create the spark are also in the way of the flame during the initial part of the combustion stroke. The relatively cold surface of the electrodes may quench the flame during the early stage of ignition, causing a misfire [28]. To counteract this, engines need to be operated on a richer (lower air-to-fuel ratio) fuel mixture than otherwise necessary. A too rich fuel mixture leads to less complete combustion and more fuel is wasted as it passes through the engine without burning.

A laser ignition (LI) system replaces the spark with a focused laser beam. The laser creates a small plasma inside the combustion chamber which ignites the fuel mixture. The system has been previously realised by placing the laser source apart from the engine and guiding the beam into the combustion chamber using mirrors and lenses. The beam typically enters the combustion chamber through a small, non-reflective window [29, 30, 31, 32, 28].

The first documented attempt to use laser ignition in an engine was made in 1978 and it was successful. The engine used in this experiment achieved greater fuel efficiency and also better performance due to faster combustion than with a conventional ignition system [30]. Since then, new types of laser equipment has been developed which are smaller and more energy efficient. As of today, laser ignition yet remains an experimental technology and no commercially available engine or aftermarket kit exist.

6.1.1 Basic laser theory

Figure 23 provides a very basic description of the operation of a laser. The highly reflecting mirror and the output coupler make up the boundaries of the laser system. The output coupler is a semi-reflective mirror through which some light can escape. During operation of the laser, light oscillates between these two mirrors. Light that escapes the output coupler becomes the actual output of the laser (the laser beam) [33].



[33]

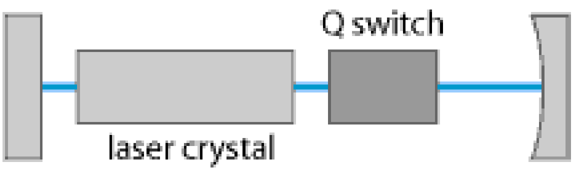
Figure 23: A simple model of a solid-state laser

The light oscillating inside the laser is amplified by a so-called gain medium. The gain medium can be a crystal as in this example, but may also consist of a gas (as in a CO2 or Helium-Neon laser) or a liquid solution (such as dye lasers). The gain medium in turn gets its energy from light sources located on the side. Common light sources are flash lamps and laser diodes. The process of energising the gain medium using other light sources is known as optical pumping [34].

Lasers can be divided into categories based on the type of gain medium used. Those lasers that use a solid gain medium such as crystals or glasses are known as solid-state lasers. Combining a solid-state gain medium with laser diodes as a pump light allows for a laser system that is compact, energy efficient and reliable [35]. A diode-pumped solid-state (DPSS) laser would therefore be the optimal choice for use in vehicles.

6.1.1.1 Q-switching

An optical resonator such as a laser may be described by its Q-factor. In terms of energy storage, the Q-factor describes the ratio of stored energy to dissipated energy in the resonator. The Q-switch is a device that allows for modulation of this factor during operation of the laser. The Q-switch is located inside the laser as seen in figure 24 [37].



[36]

Figure 24: A model of a Q-switched laser

When resonator losses are kept at a high level (low Q-factor) no lasing (emission of laser light) can occur and most of the energy that is pumped into the gain medium stays there. The Q-switch can then create a sudden rise in Q-factor, corresponding to a drop in resonator losses. This in turn allows the resonator to charge up quickly and emit a strong pulse of laser light. Q-switched laser usually have pulses with a duration of a few nanoseconds [36].

There are several ways to construct a Q-switch, and the most important distinction is whether the switch is active or passive. Active Q-switches are triggered by an external controller while passive Q-switches always pulse when the energy stored in the gain medium reaches a certain level. Active Q-switching allows for better control over parameters such as pulse energy, pulse duration and repetition rate, whereas in a passively Q-switched laser the output energy and pulse duration stays constant. The repetition rate of a passively Q-switched laser can be regulated by varying the intensity of the pump light [38].

Most experiments with laser ignition (LI) in engines have been carried out using actively Q-switched lasers. It is however possible that passive Q-switching may be sufficient, or even beneficial, for vehicle applications. There is usually no need to regulate pulse energy as long as it is high enough to ignite the fuel, and the relative simplicity of the passive Q-switch may be useful in reducing size and complexity of the system.

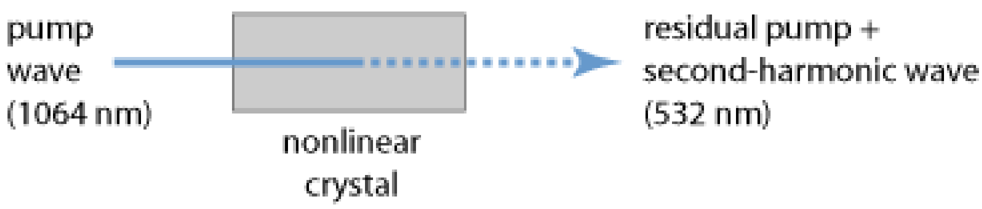
6.1.1.2 The Nd:YAG laser

The neodymium-doped yttrium aluminium garnet (Nd:YAG) laser is one of the most versatile lasers in use today. It has commercial applications in various fields such as medicine, materials processing and military equipment. The name Nd:YAG refers to the laser crystal used as a gain medium. A major reason for its popularity is that the same crystal can be used in lasers with completely different characteristics. Nd:YAG lasers can be operated as continuous-wave (CW) or be pulsed with repetition rates ranging from a few Hz to hundreds of kHz. The Nd:YAG crystal can be optically pumped by either flash lamps or laser diodes depending on the application and the power required [39].

6.1.1.3 Wavelength and frequency doubling

Another important property of lasers is the wavelength of the emitted light. The Nd:YAG laser has a natural wavelength of 1064 nanometers. This wavelength is part of the near-infrared spectrum, a spectrum of infrared light relatively close to the that of visible light (approximately 400 to 700 nm).

If visible light output is needed from an Nd.YAG laser this can be achieved through frequency doubling.



[40]

Figure 25: The basic principle of frequency doubling

In short, a certain crystal is pumped by the original laser beam, and in turn generates a beam with half the original wavelength (see figure 25. In the case of Nd:YAG, the frequency doubled (second-harmonic) beam will be visible green light with a wavelength of 532 nm. Frequency doubling has limited efficiency, but if the crystal and the incoming pump beam is both of good quality the efficiency may be 50% or higher [40].

6.1.2 Improvements

LI offers several benefits compared to conventional ignition systems. The main reasons for improvement are the higher ignition power delivered by the laser, less heat transfer from the combustion chamber, and the absence of electrodes inside the combustion chamber.

6.1.2.1 Higher ignition power

Previous research conducted on LI engines has shown that the more intense pulse provided by an LI system compared to conventional SI makes it possible to ignite leaner fuel mixtures without risk of misfiring. Lean fuel mixtures (more air per unit of fuel) allow for a more complete combustion and thus better efficiency. One study reports that the lambda value of an LI engine could be increased from 1.0 to 1.1 without affecting engine stability and performance adversely. Others have achieved similar results [31].

6.1.2.2 Absence of electrodes

Removing the spark plug's electrodes from the combustion chamber has benefits similar to the increase in ignition power. As described in the introduction to this chapter the electrodes may quench the flame and cause misfiring of the engine, especially when running with a lean mixture. This issue becomes more pronounced during cold starts [31].

6.1.2.3 Less heat transfer

One final benefit of laser ignition is the improved heat retention due to the lack of spark plugs. The issue with heat loss caused by spark plugs has also been mentioned in the introduction to this chapter. Good heat retention means less energy lost as waste heat, and one of the concepts evaluated in this project is a low heat rejection engine.

Due to the unconventional running characteristics of the CVT engine good heat retention and cold start performance are important parameters. Since LI helps to improve both these parameters while also offering improved efficiency it was selected as a promising concept to be evaluated in this study.

6.1.3 Drawbacks and implementation issues

The main issue with laser ignition systems today is the complexity and fragility of the system itself. Only recently has it been possible to build lightweight and power efficient laser heads thanks to the use of laser diodes instead of flash lamps as a primary light source. To be able to use laser ignition in a vehicle one must also ensure that the optics that guide the beam inside the combustion chamber is not affected by the heat and vibrations close to the engine.

Another issue is safety. Even scattered reflections of a powerful laser beam has enough energy to cause permanent damages to the eyes. A laser powerful enough to ignite fuel must be kept enclosed so that laser light can never escape the ignition system. This is especially important when dealing with infrared light since a leak may not be discovered before the damage is done [41].

It is also important to consider the expected lifetime of the ignition system. While the laser itself can be built to last for a long time (10 000 or more hours) the strong energy pulses will eventually cause significant wear on the optics, especially the protective window [32].

6.2 Design

During the design phase of the project the main concern of the laser ignition sub-project was to find suitable equipment to be used in the construction of a laser ignition system that could be used on the CVT engine. The desired specifications of this equipment was decided based on previous successful experiments on laser ignition and the running characteristics of the existing CVT engine.

Almost all lasers used in laser ignition experiments have been Q-switched Nd:YAG lasers. They can be made compact and energy efficient enough to be realistically installed in a vehicle, while also fulfilling the requirements regarding pulse energy and repetition rate.

6.2.1 Important considerations and design parameters

This section contains descriptions of some important properties of laser equipment as well as some recommended values for these properties. It can be seen as guidelines for the process of selecting suitable equipment for use in the CVT engine.

6.2.1.1 Pulse energy and pulse width

To properly ignite the fuel mixture in the combustion chamber each laser pulse must have enough energy. Pulse energy is usually measured in mJ per pulse.

Previous experiments report various requirements for minimum pulse energy, ranging from 3 mJ up to 18 mJ [29, 28]. These energies are reported as the lowest pulse energy to keep the engine running reliably, and were measured from the inside of the combustion chamber. Most lasers used in experiments have a pulse energy greater than the required minimum.

Pulse energies of 50 mJ has been the most common with some experiments using lasers delivering 200 mJ or more [29, 30, 31, 32, 28]. Since the pulse energy is measured on the inside of the engine during ignition tests it is important to also consider the quality of the optics. If less light is absorbed or reflected thanks to the use of high-quality optics the same pulse energy can be realised with a weaker laser.

The pulse width (duration of a single pulse) of the type of Q-switched Nd:YAG laser used in previous experiments is typically in the range of a few nanoseconds [31, 28]. This is much shorter than the average duration of a regular spark, which is about 1 millisecond [42]. In turn, the power delivered by the laser is much higher. The total amount of energy delivered per pulse from LI systems are similar to conventional SI [29]. An advantage of LI systems with active Q-switching is that ignition power can be adjusted during operation of the engine. This can be useful for certain condition such as cold starts.

6.2.1.2 Repetition rate

To be useful for engine ignition the laser has to be able to pulse at least once per ignition cycle. If the engine is designed to work at, for example, speeds up to 6000 rpm (the peak rpm of the current CVT engine is reported as 6500 rpm) this means that the laser must have a repetition rate of at least 50 Hz if the laser only fires at the ignition point of a four stroke engine.

This allows for greater engine speeds using slower pulsing lasers, but it is not optimal for other reasons. Many ignition systems are designed to fire once per revolution, similar to a two stroke engine. This allows for a simpler design of the ignition system. In a conventional ignition system the additional spark does nothing (the setup is commonly referred to as a "waste-spark" system) [43]. In a laser ignition system this redundant pulse serves a more important purpose as it can help clean the optical window in the cylinder head. This phenomenon is known as ablation and is described later in this chapter.

The main issue with using "wasted" laser pulses in LI is that the laser has to pulse once per revolution of the engine crank. Commercially available lasers that are capable of firing high-energy pulses with frequencies of 100 Hz or more do exist, but are often more expensive and physically larger than lower end systems. Finding a suitable balance between power, repetition rate, size and power consumption is crucial when designing an LI system.

6.2.1.3 Optics

Another challenge when designing a laser ignition system is to create the best possible path between the laser source and the combustion chamber of the engine. The most important function of the optics is to focus the beam so that the focal point of the beam (where ignition will happen) is optimally located inside the combustion chamber. Typically a lens made from borosilicate glass is mounted in close proximity to the chamber [31, 28].

To protect the lens from damage and pollution caused by the combustion process a window is usually installed at the end of the optical pathway, in contact with the inner wall of the combustion chamber. This window is typically made from sapphire or silica, and needs to be thick enough to withstand the pressure inside the combustion chamber [31, 28].

There are different ways to mount the optics on the engine. One way is to install the glass window permanently in the cylinder head [28]. This allows for more freedom in the choice of window dimensions since the cylinder head can be manufactured or modified to accommodate the window. The lens can then be mounted above the window. This setup also allows the user to move the point of ignition inside the combustion chamber in all three dimensions provided that the window has been made sufficiently large. This can be useful when trying to find the optimal point of ignition. One drawback of this setup is that the cylinder head has to be modified to accommodate the window. It is also harder to replace the window in case of failure.

Another solution to this problem is to mount the window and lens inside a container resembling a spark plug [29, 31]. This "optical spark plug" can then be installed in place of the conventional spark plug, without modifying the rest of the engine. One major benefit of this concept is that it allows for an easy comparison between spark and laser ignition without modifying the engine between tests. An optical spark plug can be designed so that the lens can be moved relative to the window, so that the point of ignition can be moved in one dimension. A simple design for an optical plug, was developed during the design phase.

If space is limited on the cylinder head the lens and window may have to be made small, with diameters of only a couple millimeters. In that case it is also important to consider the width of the laser beam. In some cases it might be necessary to narrow the beam using additional optics before it enters the optical path mounted on the engine.

6.2.1.4 Optics pollution and ablation

All internal combustion engines produce some amount of combustion byproducts. When running an engine fitted with an optical window such as is necessary for laser ignition, this window will eventually be coated in soot and other pollutants. Fortunately, experiments show that the laser is able to clean the glass through a process known as thermal ablation [31, 32].

This basically means that the heat from the laser beam vaporises the polluting particles on the window and the system becomes self-cleaning. Additional laser pulses such as the aforementioned “waste spark” can be used to enhance this feature.

For long-term use of laser ignition systems it is uncertain whether this process of ablation may cause significant damage to the glass itself, eventually resulting in failure of the protective window [32]. Since LI is still an experimental technology there is limited data on the long term wear on the ignition system.

6.2.1.5 Wavelength

Previous experiments with Nd:YAG lasers for engine ignition have used the lasers at their natural wavelength of 1064 nm. Due to the light being invisible, aligning an experimental setup becomes more difficult. If the laser is frequency doubled to 532 nm the light becomes visible. This makes it easier to see if the laser and optics are properly aligned and also get an idea of how much light is reflected by the optics. Visible light is also inherently safer than infrared since it activated the blink reflex of the eyes. While the blink reflex is no complete protection from eye damage it can help limit it. With infrared light potentially dangerous leaks may never be discovered and if the ignition system is built as a closed unit it is harder to verify that no light can escape the unit.

If frequency doubling equipment is to be used as part of the ignition system one must also consider the energy losses in the frequency doubler and adjust the power of the laser accordingly.

6.2.2 Laser selection

For the main component, the laser, it was decided that the best choice for this application would be a diode-pumped, Q-switched Nd:YAG laser with a pulse energy of at least 50 mJ and a repetition rate of 100 Hz. Ideally it would also be small enough to be easily mounted alongside the engine for testing.

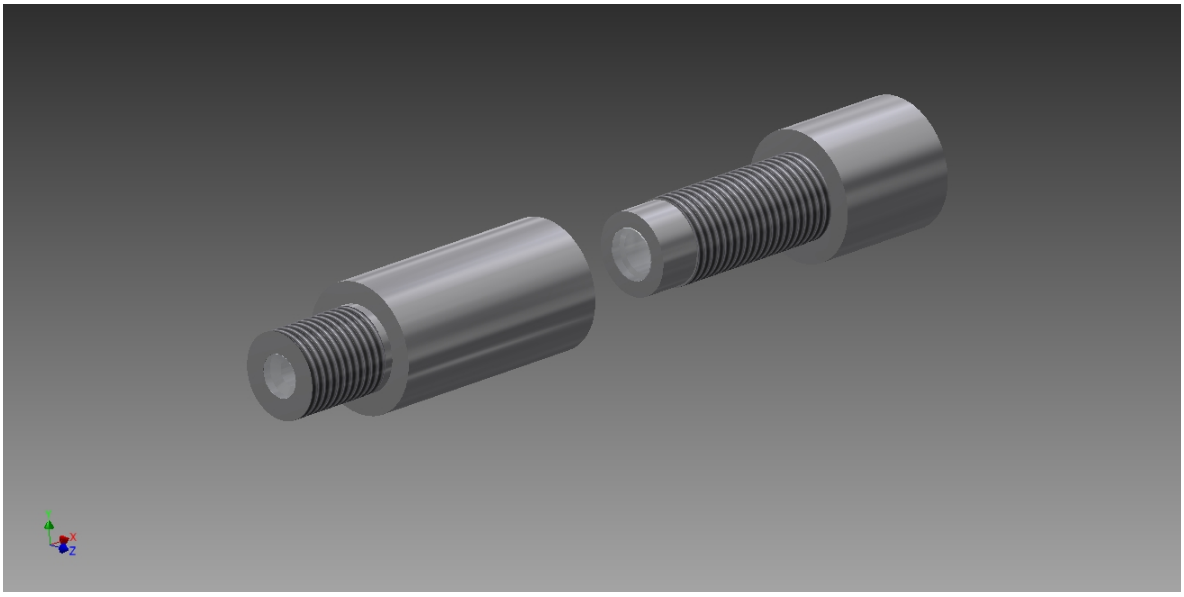


Figure 26: Optical plug consisting of lens holder (background) and window holder (foreground)

Based on these specifications several lasers from different manufacturers were selected as potential candidates, but very few of them fulfilled all requirements. The most common issue was that the repetition rate was too low to be able to perform any meaningful testing. Other lasers were simply too large and powerful for our application.

The best candidate turned out to be the Quantel Centurion+ from Quantel Laser. It is a diode-pumped, Q-switched Nd:YAG laser with a maximum pulse energy of 50 mJ and a repetition rate variable between 1-100 Hz. This laser also has a compact design, is entirely air cooled and has an integrated power supply. The total weight of the assembly is 6 kg.

6.2.3 Optics selection

It was decided early in the design phase to construct the optical pathway in the form of an optical spark plug for installation in the existing cylinder head. A design suggestion can be seen in figure 26. The primary reason for this was to enable side-by-side comparison between spark and laser ignition without dismantling the engine. Because of the small size of the original spark plug used in the CVT engine and the limited space available on top of the cylinder head the size of the lens and protective window had to be restricted. The proposed design of the optical plug uses a window 5 mm in diameter and a lens 6 mm in diameter with a focal point located 21 mm from the lens. This assembly can be mounted on the existing cylinder head without modifications. The design allows for some adjustment of the point of ignition.

Finding suitable optics turned out to be much easier than finding a laser supply. The suggested lens and window were available for order through Edmund Optics, an optics manufacturer located in New Jersey, USA. They also offered non-reflective coating on all optics to reduce energy losses in the optical path.

6.3 Manufacturing

Several manufacturers were contacted in order to assess the plausibility of ordering a suitable laser unit for use in the project. Resellers of laser equipment in Sweden and worldwide were also contacted to inquire about cost and delivery time. Of all the companies contacted, only one (Azpect Photonics) answered with an offer. Their offered system was however too big and too expensive for this application. It was later decided that purchasing a dedicated laser system for use with the technology demonstrator would not be reasonable within the limits of the project.

Another option was to use laser equipment already available in laboratories at Chalmers. The Department of Applied Mechanics owns several YAG lasers that could possibly be used for engine ignition. While these lasers certainly were powerful enough to ignite an engine, their maximum repetition rate of 10 Hz was too low to perform meaningful tests on the CVT engine under the engine's usual running conditions.

6.4 Results and Discussion

Due to the limited availability of suitable laser equipment and the many unsuccessful attempts to contact the manufacturers and resellers of said equipment, the laser ignition sub-project got severely delayed during the design phase. It was eventually decided that it could not be realistically completed within the time frame of this project and was subsequently cancelled in order to focus on other parts of the project.

As stated in the introduction to this chapter, laser ignition remains a highly experimental technology. While it has been proven to have several significant advantages over regular spark ignition, there is still a lot of development to do before it can compete with conventional ignition systems in terms of cost and physical size.

It is our recommendation that further research and development of laser ignition for the CVT engine is carried out as its own project. It would then be important to establish good contact with one or more manufacturers of lasers early on to be able to discuss with them the type of equipment needed and also what can be expected in terms of cost and delivery time. It is important that any laser purchased for this purpose has the necessary specifications to be used together with the CVT engine to produce meaningful test results. This chapter contains some guidelines and recommendations regarding the kind of equipment that could be used in such a project.

7 Variable compression

This section describes the theory behind variable compression and how the technology is chosen to be implemented in the technology demonstrator.

7.1 Theory

The compression ratio of an engine is calculated by measuring the volume of the combustion chamber when the piston is at its lowest position and dividing it by the volume of the combustion chamber when the piston is at its highest position. When a higher compression ratio is used in an engine, the fuel and air molecules come closer together due to the smaller volume in the combustion chamber. This is a large advantage because every fuel molecule gets a shorter distance to ignite the next molecule. The result is a faster combustion process where more energy is converted to mechanical work instead of heat losses [44].

According to equation 1 on page 25 the efficiency of an Otto cycle engine increases when a higher compression ratio is used. The equation states that the highest efficiency is acquired when the compression ratio goes to infinity. This description of an Otto engine gives a quick estimate of the change in pressure and volume over the four strokes but as some of the energy transforms into heat it will not be fully accurate [45].

A problem with increasing compression ratio and hence the temperature in the combustion chamber, is that it can result in a phenomenon called knock. Knock are auto-ignitions or detonations which occurs in the combustion chamber ahead of the propagating flame. This leads to high local pressures which lowers the total efficiency of the engine and causes the characteristic knocking sound in the engine. Low rpm and high loads will increase the chance of knock and the compression ratio therefore needs to be lower than with high rpm and low loads [45].

Exactly which compression ratio used is always a compromise between using the highest compression ratio possible but at the same time keeping the knock intensity to a minimum. On an engine with fixed compression ratio, the knock intensity is adjusted with the ignition angle. Because of the time it takes for the fuel to combust it is common to ignite the fuel before the piston reaches TDC. This is to get the pressure peak from the combustion right after the piston passed TDC and therefore produce as much mechanical work as possible. The number of degrees before TDC when the spark plug ignites the fuel mixture is called ignition angle. If the fuel mixture ignites later you get a lower cylinder pressure and a shorter time for knock to occur. The drawback is that the efficiency in the engine decreases as the ignition angle is moved away from its optimum [46, 47].

If a variable compression ratio is used in the engine the compression ratio can be changed to always be optimal for each rpm/load condition. It will also bring another possibility to change the knock intensity. The big challenge is to find the most efficient combination between the ignition angle and the compression for the different rpm:s and loads [47].

7.2 Current applications

Almost every car manufacturer has their own solution for a variable compression engine. During the literature survey the different solutions were evaluated and compared against each other. Figure 27 shows an overview of some of the concepts which have been investigated.

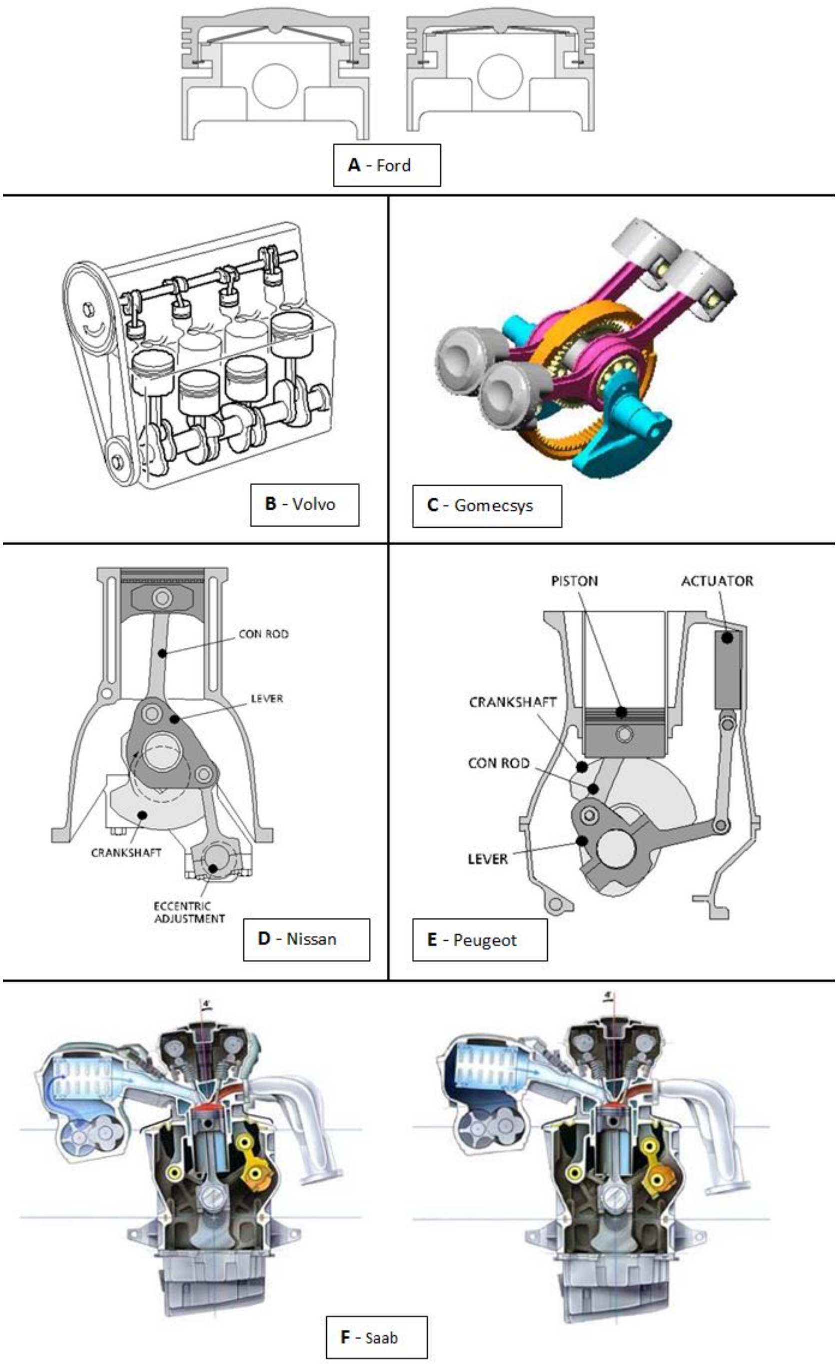


Figure 27: Current designs of variable compression engines

7.2.1 Moving head

7.2.1.1 Angling of the cylinder/cylinder head

This concept has been developed by SAAB and changes the volume in the combustion chamber by tilting the cylinder in relation to the crankcase (seen in figure 27 concept F). The different volumes makes it possible to have different compression ratios. The advantages with this solution is that the geometry of the combustion chamber is more or less the same and therefore have small implications on the gas exchange in the engine. It has a simple design which makes it possible to implement on the CVT engine and on the same time it can be realised with few and small components.

7.2.1.2 Linear movement of the cylinder

This is also an interesting concept which moves the cylinder in a linear motion compared with the crankcase. It is interesting because, as the concept above, it does have a small impact on the gas exchange in the engine. Compared to concept F it had a more complex design but it was still a good solution because it could be implemented on the CVT engine without taking up too much space.

7.2.2 Variable piston height

As seen in figure 27 concept A, Ford have a piston which consists of two main pieces that can move relative to each other. To raise the compression the top of the piston is moved upwards and thereby reducing the volume in the combustion chamber. The advantage of this concept was that it could be realised with small adjustments on the CVT engine. On the other hand the piston is complicated to manufacture and is often controlled with hydraulics which makes it hard to handle in a reliable way.

7.2.3 Variation of the combustion chamber volume

This solution changes the volume in the combustion chamber often by lowering a small cylinder into the chamber through the cylinder head. Concept B in figure 27 shows Volvo's concept of accomplishing that by connecting the small cylinder to an eccentric axle. This is a relatively easy concept to implement in the CVT engine but it will have a noticeable effect on the gas exchange in the engine. This because the small cylinder will interfere with the flow in and out of the engine as its lowered down in the combustion chamber.

7.2.4 Variable length of the conrod

Like the headline advices, this concept varies the length of the conrod to accomplish different compression ratios. This way it has a minimal effect on the gas exchange but on the other hand it will be hard to manufacture and it will even be complicated to design a concept like this.

7.2.5 Conrod linkages

During the survey there were many concept found which relates to this concept group (D and E in figure 27). The common ground is that every concept in this category has some type of link-system which cause the conrod bolt to move in comparison to the crank. The movement of the conrod bolt makes the volume in the combustion chamber change depending on if the conrod bolt is on a high place or a low place. Even this concept have a small impact on the gas exchange and would be accomplishable although its quite complicated design. But compared to the other concepts it would have a greater impact on the size of the engine which is a big drawback in the CVT engine where both size and weight is to be kept at a minimum.

7.2.6 Movement of the crankshaft

These concepts changes the compression ratio by moving the crank assembly in different ways. Gomecsys (C in figure 27) for example does this by a gear transmission inside the crankcase. Many of these concepts are complicated to both manufacture and implement in the CVT engine.

7.3 Design

During the design phase all the different concepts listed above were investigated and compared regarding complexity, impact on the gas exchange, manufacturing possibilities and size. With these four things in mind two main concepts were chosen to further look into. The concepts were the two moving heads and they were chosen mainly because of their low impact on the gas exchange and the implementation possibilities in the CVT engine.

7.3.1 Concepts

To further improve the two current concepts and make it possible to implement them in the CVT engine three new concepts were generated (Figure 28).

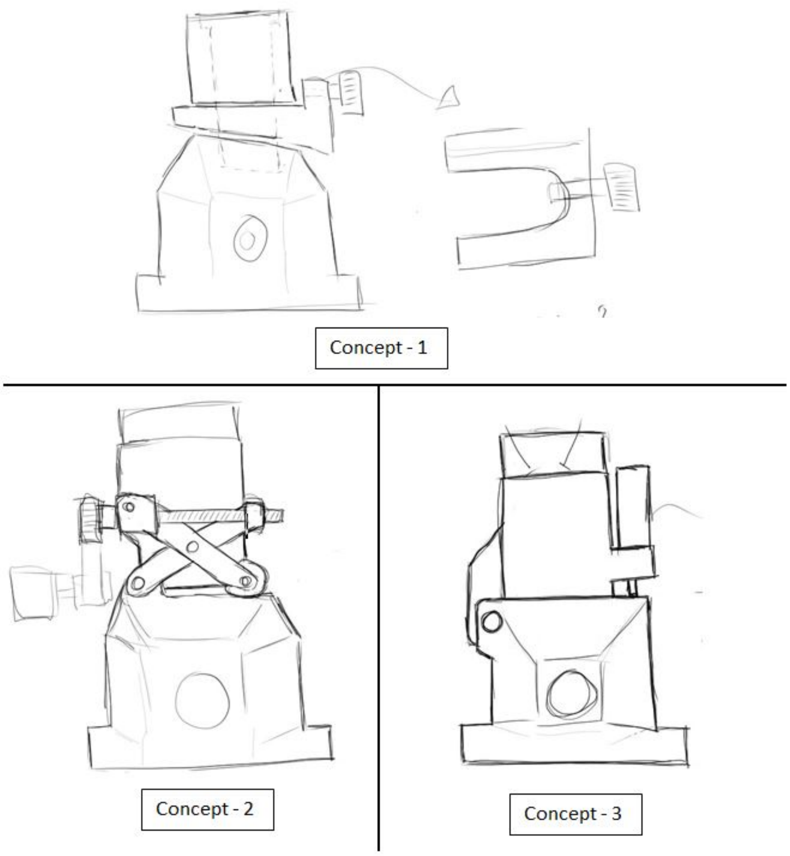


Figure 28: The three concepts that were generated during the design phase

7.3.1.1 Concept 1

This concept consists of a wedge between the crankcase and the cylinder which moves towards or away from the cylinder liner. This movement makes the cylinder and cylinder head either go up or down depending on which way the wedge moves. The advantage of this concept is that it is easy to get a high tolerance in the height adjustment because it is only depending on the angle of the wedge. On the other hand it was hard to find a solution to fasten the wedge in a way that it only moves the cylinder and cylinder head vertically. When it on the same time should have low friction against the crankcase and the cylinder.

7.3.1.2 Concept 2

Concept 2 moves the cylinder in an upward direction using a small scissor lift. The height of the scissor lift is changed with a stepper motor connected with a belt to a screw on the lift. One side of the scissor lift is mounted on the cylinder and the crankcase but the problem was to design a mount on the other side of the engine which was stable enough to withstand the forces produced by the engine.

7.3.1.3 Concept 3

This is a redesign of concept F in figure 27 only implemented in the CVT engine. The difference is that the angle adjustment is accomplished by a linear actuator instead of an eccentric axle as in the SAAB engine. The linear actuator is mounted with one side on the cylinder and the other side on the engine mount. This concept is easy to manufacture, it is stable and have potential to resist the forces produced by the CVT engine. Referring to these points, concept 3 were chosen as the final concept to implement in the CVT engine.

7.3.2 Calculations

For ordinary commercial Otto engines it is common to use a compression ratio around 10:1 but it is also interesting to investigate compression ratios down to 8:1, at least when high loads and low rpm is used [46]. The upper limit for the compression ratio were chosen to be 21:1, this to be able to investigate the Atkinson concept which will lower the compression in the engine due to its change of intake closing. Another advantage is the possibility to, in the future, investigate the engine running with compression ignition. Calculations, with respect to the angles between the engine and the ball screw, showed that a change in compression ratio from 8:1 to 21:1 will need the linear actuator to move approximately 10 mm vertically.

Calculations of the forces in the CVT engine were made based on cylinder pressure plots which had been simulated in an former project. Unfortunately the calculations showed forces up to 7600 N which made it hard to find a linear actuator, small enough for the CVT engine, that could withstand these forces. To solve this problem a custom linear actuator was designed based on a stepper motor and a ball screw.

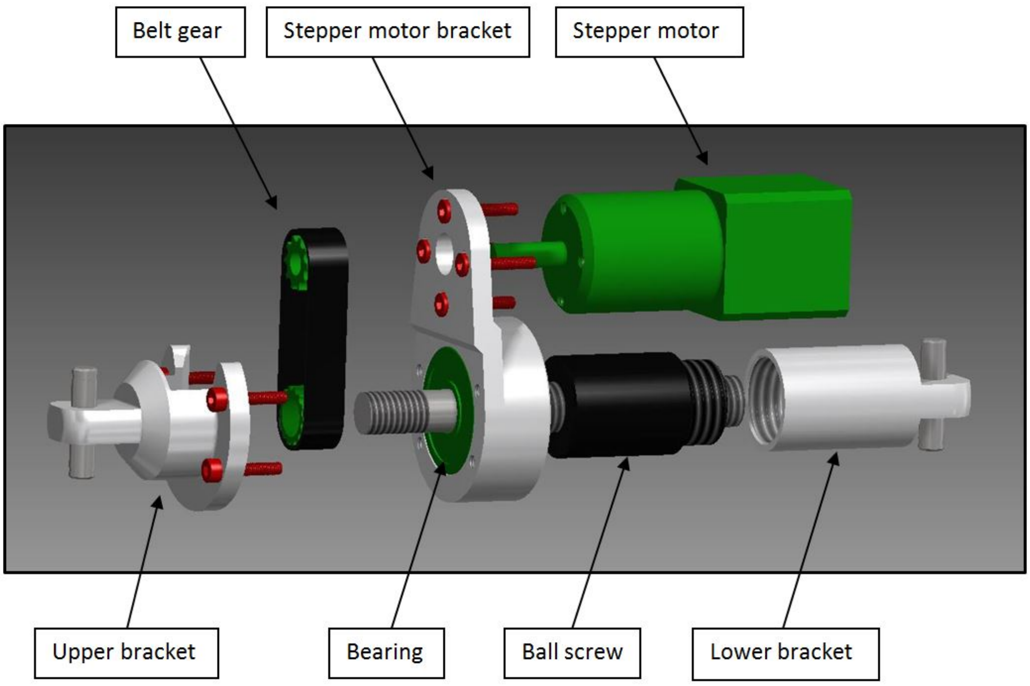


Figure 29: The components of the linear actuator

7.3.3 Linear actuator

The main components in the linear actuator seen in figure 29 are, as said earlier, a ball screw and a geared stepper motor. The stepper motor was chosen to be a Nema 11 motor and to defy the forces from the CVT engine it needed a gearbox with a ratio of 27:1. This will give the motor a torque of 1.3 Nm which is enough to turn the ball screw even with the highest simulated cylinder pressures. The ball screw and the motor are joined together with a belt and two pulleys. To resist the axial forces created by the engine a double row angular contact ball bearing was installed on the ball screw. The linear actuator is then mounted with one end on the cylinder and one on the engine mount with a steel axle holding it in place. Figure 30 shows the whole concept mounted together.

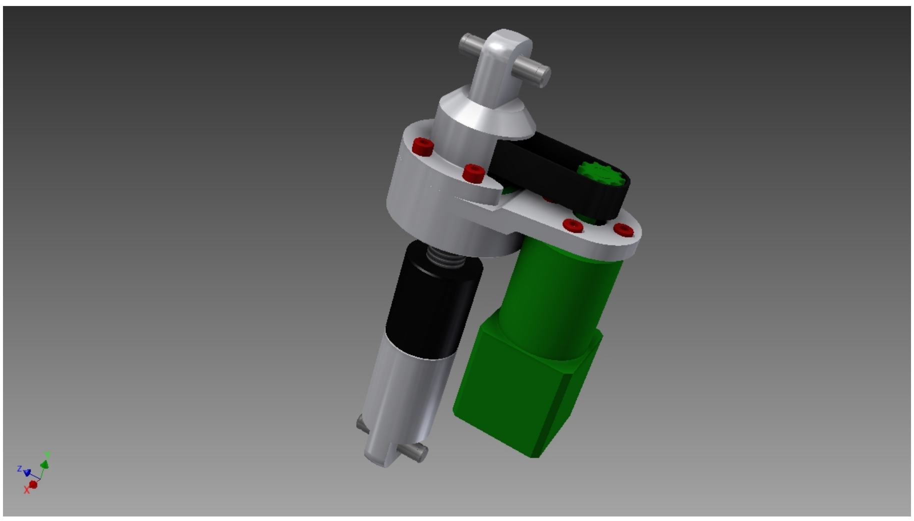


Figure 30: Assembly of the linear actuator

7.3.4 Engine

Figure 31 below show the concept as whole. On the opposite side of the linear actuator the cylinder is mounted with another steel axle going through both the cylinder and the crankcase. To lower the friction between these three components two bushings are mounted in the cylinder hole for the axle. The steel axle is kept in place with two stop screws which prevents the axle to move when the engine is running.

7.4 Manufacturing

All of the engines components were not manufactured during the time frame that was given. This because both the design phase and the manufacturing phase took more time than planned. Therefore a redesign of the concept were made to be able to test the other concepts included in this thesis. The drawback of the redesign was that the compression ratio only could be changed manually with the engine turned off. Another change from the main concept were that the cylinder head were taken from an old engine which limits the highest compression ratio to around 16:1.

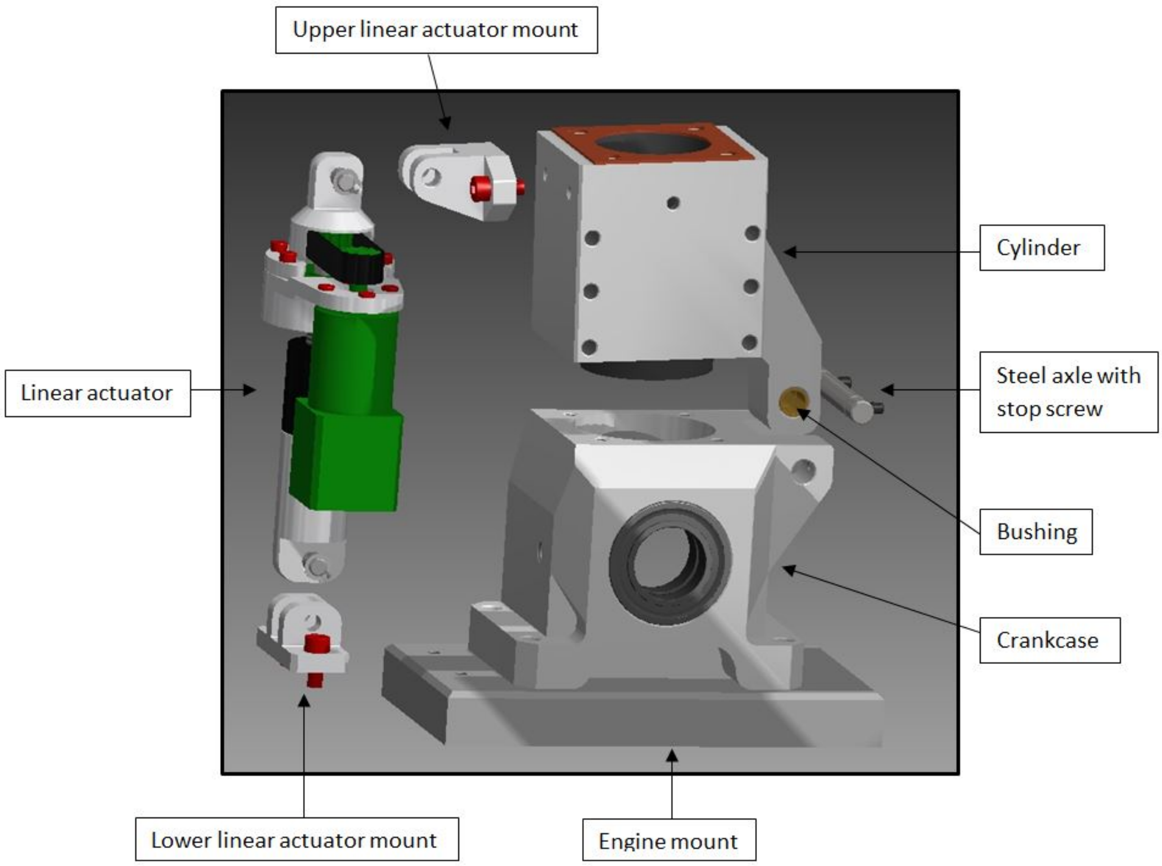


Figure 31: The components of the variable compression concept

7.5 Results and discussion

The variable compression engine is an interesting concept which have potential to further lower the fuel consumption in the CVT engine. But due to lack of time no tests of the concept developed in this thesis have been made. The next step of this project is to test the engine that was built with different compression ratios adjusted manually. This is to see how much knock that occurs and its impact on the fuel consumption.

To implement this concept in the CVT car some optimisations have to be made both to lower the weight and to make it smaller. Some considerations also have to be made concerning if the variable compression lowers the fuel consumption enough that it compensates for its extra weight.

To fully test the variable compression engine it is recommended to buy a ball screw which will make it possible to automatically change the compression ratios. Some time must be spent creating a reliable control system for the linear actuator to make sure that it moves in the right way. There are affordable controller units made for this purpose which can be fed signals from the CVT engine control unit. When this is done the testing of the variable compression can start. To find the most efficient compression ratios for the different loads and engine speeds the engine needs to be run with a cylinder pressure sensor mounted on the cylinder head. This to investigate when knock starts to occur and then calibrate the compression ratio according to the knock intensity.

A further development of this project is to investigate the possibility to run an engine with auto ignition. Auto ignition can be accomplished by raising the compression ratio and thereby making the fuel mixture ignite without a spark. Many reports have been made in this area and the main problem is to make a reliable system that runs good over the whole rpm spectra. But on the other hand it has a big potential to further increase the efficiency of the engine.

8 Direct injection

Direct injection is a method of fuel injection which in this section is evaluated with regards to fuel efficiency. The theory behind direct injection is described and the design chosen for the implementation in the technical demonstrator is described and motivated. The results are then presented and analysed.

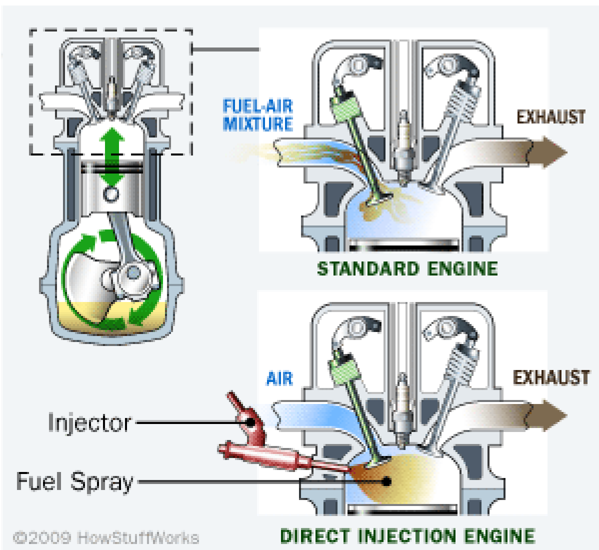
8.1 Theory

There are several different methods for getting fuel into an internal combustion engine, most based on pumping pressurized fuel through a small nozzle, thereby atomizing the fuel. When a carburetor is used, however, the fuel is transported with the air as the fuel-air mixture is sucked into the cylinder during the intake stroke. The suction occurs as the pressure in the cylinder is lower than the air pressure in the carburetor.

The method for injecting the fuel into the cylinder by the CVT is port injection where the fuel is injected in the intake duct and then flows into the combustion chamber with the air as the intake valve opens. Another method for injecting the fuel is to use direct injection, DI, which is when the fuel is injected directly into the combustion chamber during compression using higher pressure than the in-cylinder pressure. Figure 32 illustrates how DI works and how it differs from a standard port injected engine. The main theory of direct injection is that the higher fuel pressure used creates smaller fuel droplets and a more atomised spray which can be evaporated faster. This is because smaller droplets have a greater combined surface area than bigger droplets, and therefore more contact with the surrounding air.

8.1.1 Relation air and fuel

The way that the air and fuel is mixed can be described in different ways. If the air and fuel are mixed so that precisely all the fuel and oxygen in the air are combusted with no excess air, the mixture is called a stoichiometric mixture. For pure octane, the stoichiometric mixture ratio is 14.7:1, that is 14.7 mass parts air and 1 mass part fuel. Another way to measure the ratio between air and fuel is to use lambda, λ, which describes the relation



[48]

Figure 32: How Direct Injection engines work

between the combusted air and fuel residuals in the exhaust fumes. For a stoichiometric mixture λ equals 1.0, rich fuel mixtures has a $\lambda < 1.0$ and lean fuel mixtures $\lambda > 1.0$. After combustion rich fuel mixtures still contains uncombusted fuel. This occurs when there is not enough oxygen molecules to combust all the fuel molecules. Lean mixtures has on the other hand more air than needed and this mixture results in uncombusted oxygen. The air-fuel mixture which is to be ignited is also called the charge.

8.1.1.1 Lean Mixture

An engine is usually dimensioned to provide enough power for its maximum load condition. However many engines mostly operates during much less load and the power required is therefore much lower. This means that the power needs to be cut at this operating point which is done in most engines by using a throttle that is kept partially open to reduce the air flow. The air then needs to be pumped through the throttle which causes losses in efficiency.

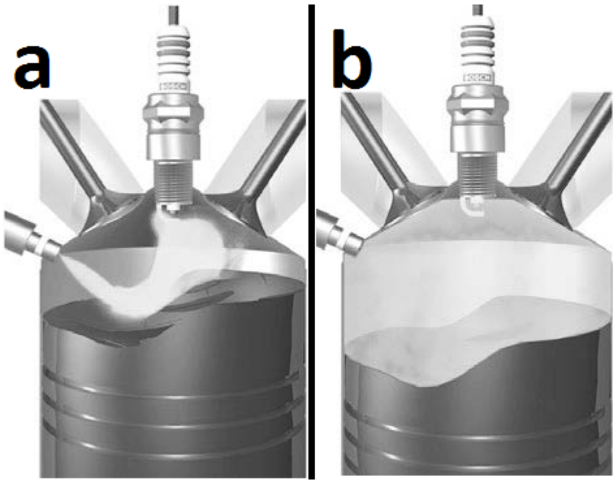
By employing a lower air-fuel ratio, AFR, i.e. a lean mixture, the throttle can be kept almost fully open and still achieve the required power, thereby reducing the throttle losses. This mixtures does however mean that the exhaust after treatment is more complicated and the lean mixture can therefore increase the amount of NO_x emissions [49].

8.1.1.2 Rich Mixture

By using a rich mixture, then it is possible to gain more torque from the engine. This is because in a rich mixture, the hydrocarbon in the fuel reacts with the oxygen, and as there is a deficit of oxygen, carbon monoxide is created. In a lean mixture, where there is more oxygen, carbon dioxide is created. When carbon monoxide is created more energy is released which heats the gases and more torque is achieved.

8.1.2 Advantages and disadvantages

A main advantage using DI is that it offers the opportunity to achieve higher torque and power than port injection. One of the reasons for this is that the potential for charge cooling is increased. Charge cooling is when the charge, intake air, is cooled and hence the density is increased. This increases the volumetric efficiency as more air molecules can fit in the same space. In a DI engine where the system uses highly pressurized fuel there is a higher probability that the fuel is vaporized before hitting any chamber surfaces. This means that the energy used to evaporate the fuel is transferred from the intake air instead of the chamber surfaces, and charge cooling occurs. This increases the volumetric efficiency by 2-3 %. By decreasing the charge temperature, the probability of knock, see section 7.1, is decreased due to lower compression temperatures [50]. When the fuel hits the chamber surfaces it is called walled wetting and it common in port injection. When the fuel hits the chamber surfaces, the energy needed to evaporate the fuel is transferred from the surface to the fuel, instead of from the surrounding gas. By transferring the energy from the surfaces it cools the surface and decreases the volumetric efficiency of the engine, as the potential for charge cooling is decreased.



[51]

**Figure 33: Different combustion processes,
a. Stratified-charge b. Homogeneous charge**

Another advantage of DI is that the fuel does not displace the air, as port injection does, which means that the air trapped in the cylinder is increased, thereby increasing the volumetric efficiency.

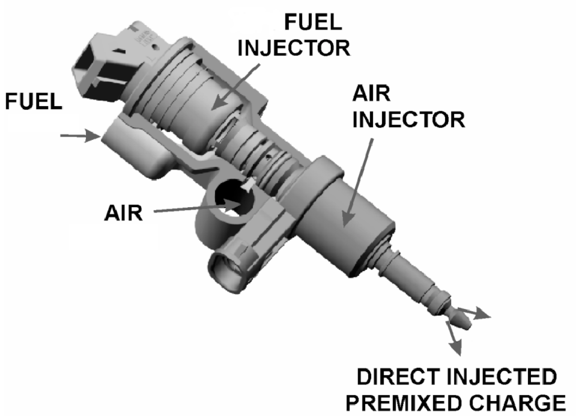
A drawback with DI is that it requires that the fuel injector nozzle is mounted directly into the cylinder, the materials need to be more durable than when the injector is placed in the intake manifold. This means that the injector needs to be made of a material with a high quality which is expensive. The pressure with which the fuel is injected into the cylinder also requires costly high pressure fuel pumps.

8.1.3 Combustion process

There are different types of DI all based on the principle that injecting pressurized fuel directly into the combustion chamber creates an atomized spray. In the following sections the different types of combustion processes are described. The combustion process is defined as the way in which the mixture is formation and energy conversion takes place. The combustion process can then switch between several operating modes of which the combustion process is made up of, see section 8.1.4

8.1.3.1 Stratified-charge

The stratified charge is when the fuel is injected at high pressure, and transported as a stratified-charge cloud to the spark plug. This means that an ignitable mixture is only present in a local cloud and not in the entire combustion chamber. The stratified-charge combustion process has a lean mixture, with the advantage of running the engine unthrottled in greater ranges. Figure 33a, shows the stratified-charge combustion process. The stratified-charge combustion process can be run at different operating modes, see section 8.1.4. It consists of two types of stratified-charge, depending on where the injector is placed in the combustion chamber, and how the mixtures is guided to the spark plug [51].



[52]

Figure 34: AADI component setup

Placing the injector centrally near the spark plug creates a spray-guided combustion process, which has the advantage of directly guiding the mixture to the spark plug. However since the fuel needs to be injected just before ignition, the air-fuel mixture has less time to be prepared and therefore need to be injected at higher pressure, approximately 200 bar [51].

The injector can also be placed close to the intake valves where the fuel then needs to be injected at a pressure of about 50 to 150 bar. The mixture then somehow needs to be guided to the spark plug and there are two ways of doing this. Both consists of utilising a recess in the piston head. The mixture of air and fuel can be guided to the spark plug via direct contact with the piston head recess, wall-guided, or by guiding the air flow in the combustion chamber so that the mixtures interacts with an air cushion on the piston recess instead, air-guided [51].

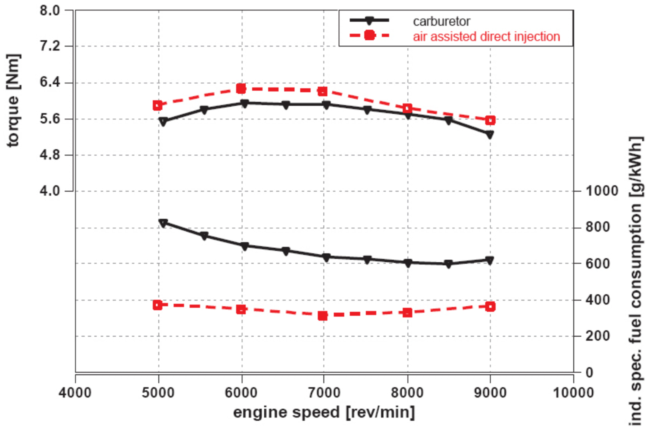
8.1.3.2 Homogeneous

The homogeneous combustion process is based on the fact that the fuel is to be homogenized, evenly spread, through the combustion chamber. Generally a stoichiometric mixture, λ=1 is injected and the process is considered an emission-reducing process as the NOx emissions reducing exhaust gas treatment is avoided. Figure 33b shows the stratified-charge combustion process. [51]

8.1.3.3 Air assisted

The Air Assisted Direct Injection, AADI, is a system commercially available for scooter engines. AADI, is based on a two injector assembly after each other, where one is connected to the fuel system, and the other to the air system. In the cycle, fuel is first injected to a space which hold compressed air, secondly the air-fuel mixture is injected to the combustion chamber. The setup can be seen in figure 34.

This sequence requires the fuel pressure to be higher than the air pressure to create a flow in the right direction. Since the fuel metering is separate from the injection into the combustion chamber, the duration of the injection into the combustion chamber only serves to alter the amount of air injected. The method creates even smaller droplets than regular DI as the air shears the fuel droplets into smaller droplets as the air accelerates and expands through the nozzle. The smaller droplets evaporate faster than larger ones as



[53]

Figure 35: Torque and indicated fuel consumption comparison between AADI and carburetor

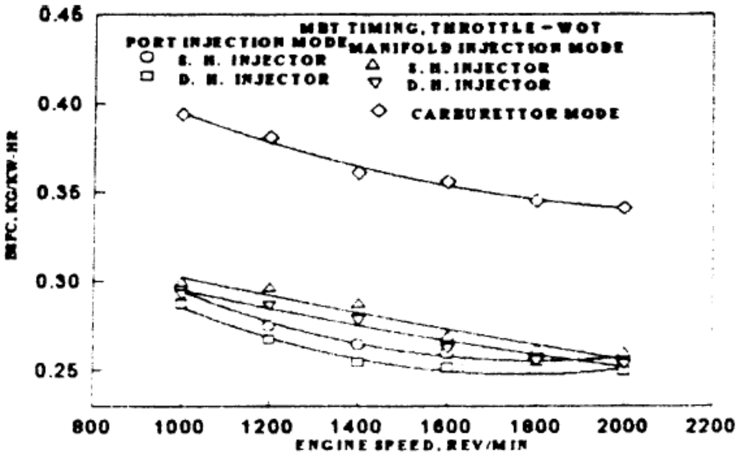
they have a larger surface area. The advantage with AADI compared to direct injection is that an atomised spray is still achieved even though the pressure with which the fuel is injected into the combustion chamber does not need to be as high as with conventional direct injection [52].

No study was found comparing the current CVT fuel injecting system, port injection, to AADI. However, in figure 35 it can be seen that compared to a system with a carburetor, the AADI system achieves slightly higher torque and significantly lower specific fuel consumption, as the mixture formation is enhanced because of the premixture with air [53]. The reason why there are few tests comparing AADI with other DI or port injection systems is probably because AADI is currently used in scooters where the carburetor system also is common.

As the current mode of fuel injection in the CVT engine is port injection, the difference between port injection and carburetor is of importance to evaluate the potential for AADI. In figure 36 it is shown that the brake specific fuel consumption, BSFC, is significantly lower for all types of port injection, as well as manifold injection. BSFC and ISFC, Indicated Specific Fuel Consumption both refer to the fuel consumption [54].

Comparing the two graphs, figure 35 and figure 36 indicates that the AADI system can lead to an improvement in fuel efficiency compared to the current port injected system. The tests are however done at different engines at different engine speed intervals which means that the conclusion is not entirely reliable. This is an uncertainty in assessing the potential of AADI, although from the results of these tests, it appears promising.

In a test performed on a one cylinder, 4-stroke petrol 500 cc engine, larger than the CVT engine, different injection pressures with AADI was tested. With a direct injection pressure into the combustion chamber of 6.5 bar the AADI system was shown to operate in a stratified manner with low fuel consumption. If the pressure was increased to up to 7.5 bar, there were slight advantages at a moderate load, that is 2000 rpm, 3.0 bar indicated mean effective pressure, IMEP. At lighter loads, increasing the pressure from 6.5 bar did not improve on fuel consumption. However pressure above 7.5 did not improve on fuel efficiency at all. The constant system pressure of 6.5 bar is considered to be the best compromise between fuel consumption, system costs and emissions [55].



[54]

Figure 36: Torque and indicated fuel consumption comparison between port injection and carburetor

8.1.4 Operating modes

The operating mode for direct injection is set by the engine management system, and depends on the engine operating point.

8.1.4.1 Stratified-charge and homogenous combustion process

In this mode the fuel is injected during the compression stroke, and therefore needs to be injected with high pressure to make the fuel flow in the right direction as the pressure in the cylinder increases. The fuel-air mixtures needs to be sufficiently spread through the stratified cloud, homogenized, in order for the mixture to be ignited. The stratified-charge operating mode is usually run at low load and low engine speed as the advantages made in fuel consumption is then achieved. This is because at faster engine speed there is not enough time to homogenize the mixture. [51]

For this mode the fuel is injected during the injection stroke and the fuel is then evenly distributed through the combustion chamber during the compression stroke. The air-fuel mixture is usually stoichiometric or rich, to increase the power at a full power load. This type of operating mode is similar to port injection [51].
There are several more operating modes, where the timing of the fuel injection and the lambda value varies. There are also operating modes where the fuel is injected twice, dual injection to form a rich mixture near the spark plug which can be easily ignited. These different variations have a variety of advantages and disadvantages including fuel emissions and fuel efficiency among other things. For further information see [51].

8.1.4.2 Air assisted

The influence of the mixture of air and fuel, lambda on the indicated specific fuel consumption for AADI is shown in figure 37, in the upper graph. In the figure it can be seen that as the AFR is increased, ISFC is decreased to a minimum to remain constant or increased in some cases. This occurs as the pumping work, using the throttle, is decreased as the AFR increases because the inlet manifold pressure is increased. The inlet manifold pressure can be seen in the lower graph in figure 37 [53].

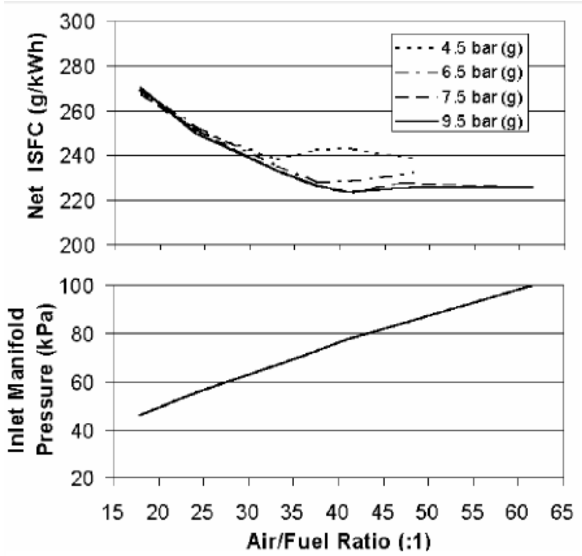


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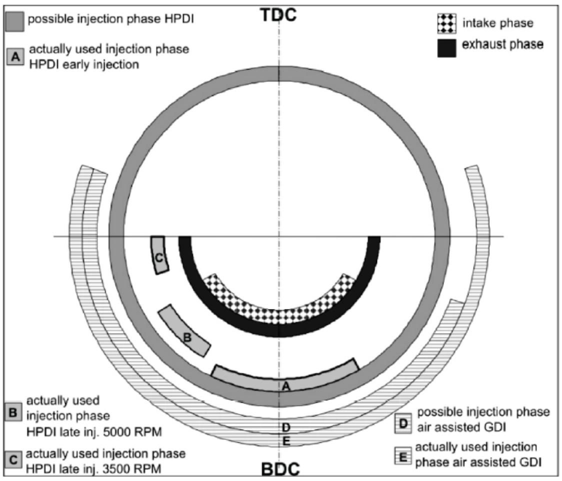
Figure 37: ISFC and inlet manifold pressure for different AFR



[53]

Figure 38: Possible injection timing

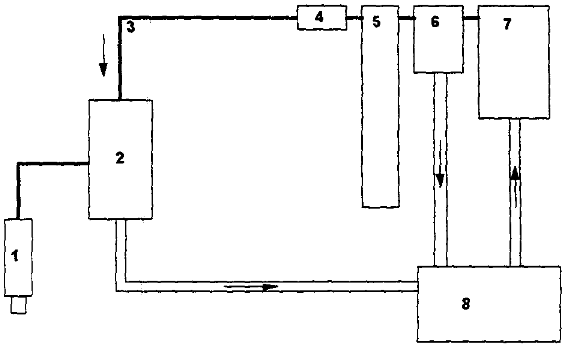
The possible and actual injection timing for AADI can be seen in figure 38, albeit for a two-stroke engine as that is the current application for the AADI system in an engine [53].

8.1.5 Pressurising systems

For direct injection the fuel needs to be pressurized to inject it into the combustion chamber. The method for doing this by the CVT is a pressure tank with pressurized air and a control unit to regulate the pressure. In larger engines, the pressurising system consists of a compressor that mechanically pressurises the air by reducing the volume. The pressure is then used to pressurize the fuel.

8.1.5.1 Ram tuned

Ram-tuned injection is a method for fuel pressurisation which utilises the phenomenon that when a fluid is halted the kinetic energy it contains is converted to a local pressure rise. The principle of this type of injection is shown in figure 39 where the components 6-8-7 in the figure is the primary circuit. When the solenoid valve (component 2) opens , the fuel



[56]

Figure 39: Components in ram tuning

is accelerated in the acceleration pipe (component 3). The fuel is accelerated to a desired velocity that can then be converted to a desired pressure, and when that desired velocity is reached, the solenoid valve closes. The fuel then impacts the solenoid valve which results in a local pressure rise 5 to 10 times higher than the initial fuel supply pressure. The pressure rise is transmitted as a sonic wave which has a high enough pressure to overcome the spring in the injection nozzle (component 1) and the fuel is sprayed through the injector [56].

8.1.6 Injector placement

As mentioned above, the injector can be placed in different locations within the combustion chamber. The favourable postion for AADI and spray-guided combustion process is to place the injector close to the spark plug to facilitate ignition. There are however different ways to organize the spark plug and injector even as they are positioned close to each other. Figure 40 shows the different positions on the cylinder head. Figure 41 shows how changing the position of the injector affects the specific fuel consumption at different injection pressures. In the figure, it can be seen that the most favourable position when fuel consumption is concerned, is the central position. Figure 42 shows the torque achieved at the different position and different injection pressures, and it is shown that the torque is not greatly affected by the position of the injector. The reason why the central position is favourable is that wall-wetting is less likely to occur when the injector is placed centrally on the cylinder head as it then is directed as far from all the walls of the combustion chamber as possible [53].

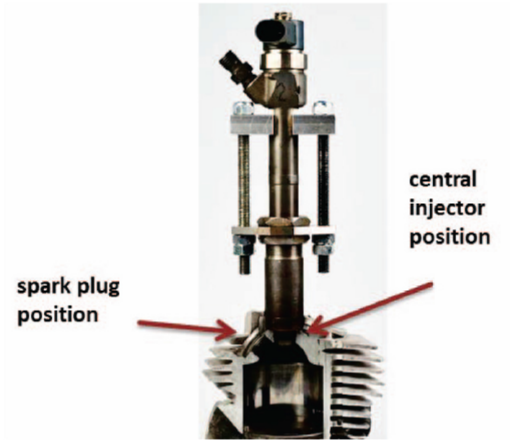


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Figure 40: Placement of the injector in the cylinder head

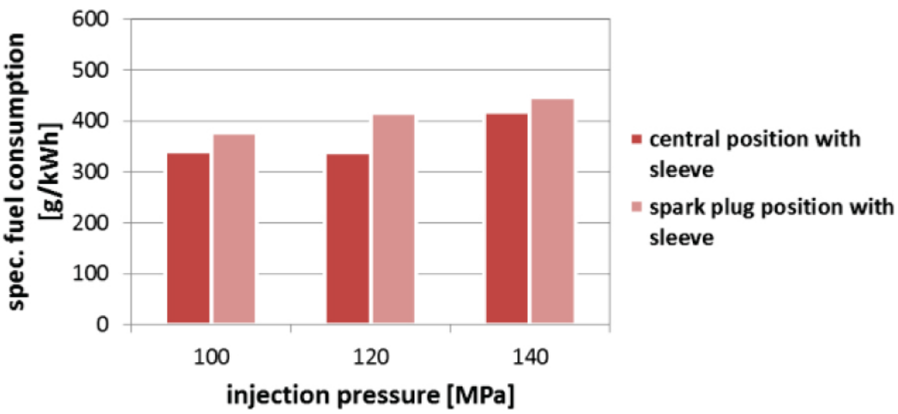

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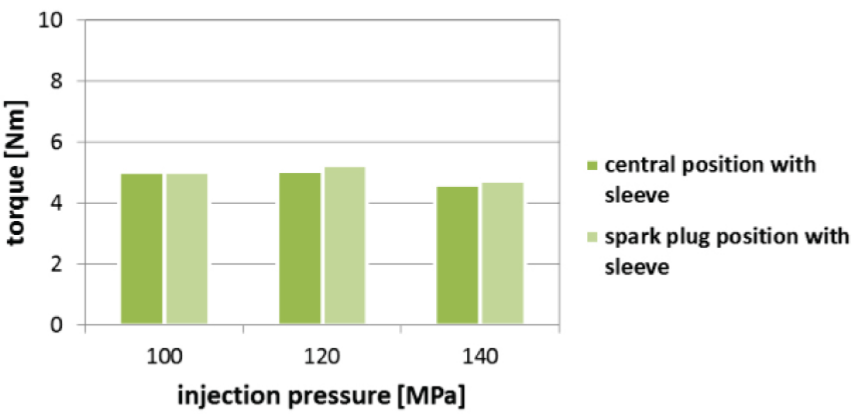
Figure 41: ISFC for different injector positions and injection pressures



[53]

Figure 42: Torque for different injector positions and injection

8.2 Design

To implement Direct injection in the CVT engine, the AADI system was chosen as the application method as it was shown to have potential to lower the fuel consumption, although it was previously mostly used in two-stroke engines. The reason for choosing AADI, was that the rules for the Eco Marathon limits CVT to 5 bar injection pressure into the combustion chamber, and the stratified and homogenous system uses too high injection pressure to stratify the mixture. A simplified compressor was chosen as the pressurising system, the same kind already used by CVT as the pressurizing system for the fuel injection and the air injection into the cylinder. The reason why ram-tuned was not chosen as the pressurising system was that it had never been used in a working engine, and that there therefore was less reliable information to be found on the benefits and methods of implementation. However if more time was given, ram-tuned could be a system worth investigating further. AADI is currently used in Aprilia scooters which means that the system is tested, reliable and shown to have a longer life expectancy than the CVT engine needs.

To design an AADI system, existing designs using AADI were studied and modified to be easier to manufacture and implement in the current CVT engine. The system designed is built with two parts with a cavity in between, and figure 43 shows the different components . The first part, Intake part 1 in figure 43 holds the air injector, also seen in figure 43 and also connects the air tube to the cavity via the brass nipple, to let the compressed air into the cavity. The second part, Intake part 2 in figure 43 connects the fuel injector to the cavity. Both Intake part 1 and Intake part 2, have M5 screws going through the parts to connect the intake assembly to the cylinder head. The function of the fuel heater is to create tension to hold the fuel injector in place, and to connect the fuel injector to the fuel tube using a hose clamp. The fuel connector is the component that the electrical signal to the fuel injector is submitted via. In figure 44 shows the assembly in half section view to see the cavity between the two intake parts. It also shows where the air and fuel travels to be injected into the cylinder.

This design is inspired by the current AADI system used by Aprilia, and calculations regarding weight optimisation or vibrations caused by the engine running has not been carried out. The size of the cavity was based on existing designs of the AADI system, and is an estimate of the size needed to mix the fuel but no calculations has been made regarding the actual size needed. This as the scope of the project only were to show how the technology can be implemented and optimisation and calculations is to be done in further development by the CVT. To control the system the current CVT ECU is to be modified to include the AADI system, that is, changing the injection timing and amount of fuel and air injected. The design of the fuel heater was based on the design of the current CVT MKVI fuel heater, with the added M5 screws to connect the fuel injector the the intake parts.

The placement of the injector into the cylinder was decided to be at the spark plug position, despite studies indicating that a central position is favorable. This since a previously constructed cylinder head could be modified to fit the injector assembly and thereby saving time in the manufacturing. It would also be difficult to compare the new AADI system to the old port injection system if the cylinder head had been modified. This as the injector nozzle is too wide in diameter to place centrally on the existing cylinder head without having to move the valves to fit the injector. Figure 45 shows this, as the red

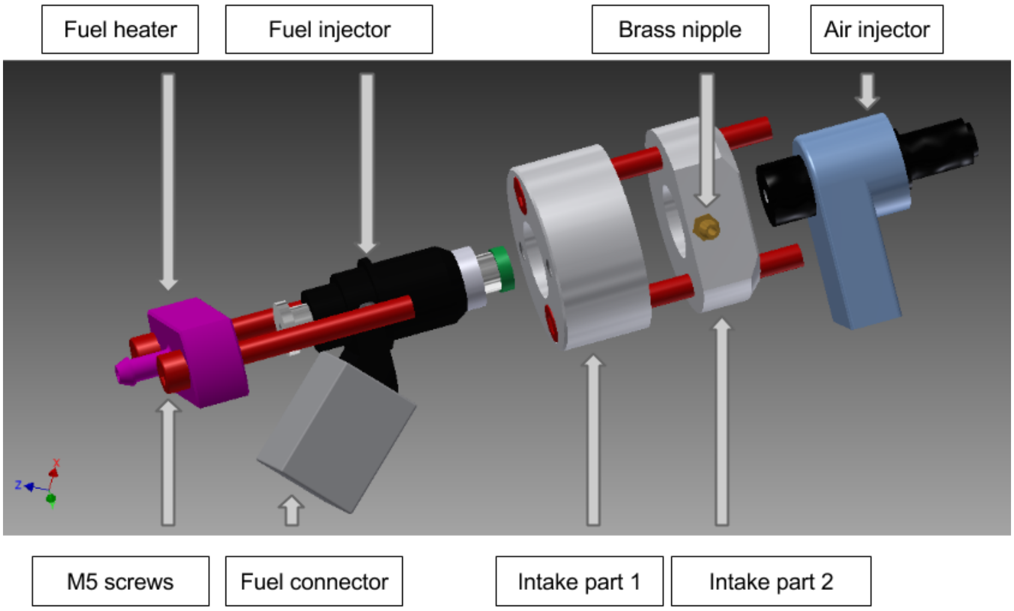


Figure 43: Intake assembly with components

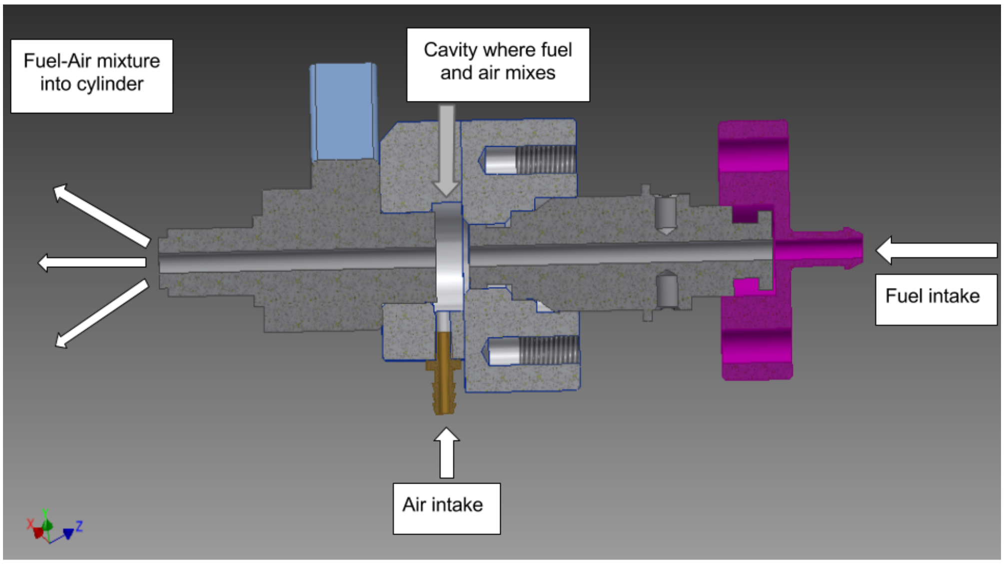


Figure 44: Intake assembly in half section view

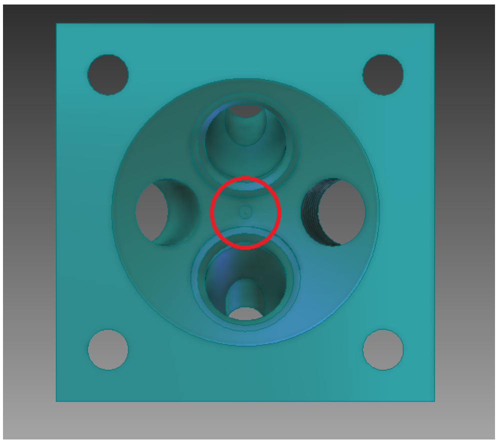


Figure 45: The bottom of the cylinder where the red circle is the diameter of the air injector nozzle

circle represents the diameter of the injector nozzle and it clearly shows that placing the injector centrally does not work for the cylinder head. The cylinder head would therefore need to be redesigned completely and redesigning the cylinder head would mean that more variables could impact the results compared to the port injection system which was used combined with the old cylinder head design. All components are designed to be made in Aluminum as it is a material that is easy to work with, which is desirable when an entire engine is to be made.It is also lighter than steel and weight optimising is desirable for CVT.

The components that could not be manufactured ourselves for the AADI system is the air injector, fuel injector, brass nipple, fuel tube, air tube. The air injector and fuel injector chosen are the same types as used in the AADI system in the Aprilia scooter, with the same specifications. The brass nipple was ordered to fit the current air tube used by the CVT team, with an inside diameter of 3 mm.

8.3 Manufacturing

The components that were manufactured for the intake assembly were Intake part and Intake part 2 and the Fuel heater, as well as components to hold the pressure regulating system. The cylinder head that was modified to fit the AADI was one that used to contain two spark plugs. As only one spark plug is currently used by CVT, the other spark plug hole was expanded to contain the air injector and holes were made in the cylinder head to attach the AADI assembly. All the parts were made in the Chalmers prototype workshop using aluminium.

8.4 Results and discussion

The finished design of the AADI system can be seen in figure 46 which shows the CAD model.

Due to complications the AADI system could not be tested before the finalisation of the project. Manufacturing the entire test engine was too time consuming within the time frame of the project and when the parts for the AADI system were manufactured, there were other parts of the engine that needed to be completed to test the system. The testing that needed to be completed for the AADI system include the durability of the intake assembly, and its ability to withstand the force and vibrations that occur when the engine is running. The specific fuel consumption is of utmost interest to be tested for the CVT as this is the main goal for the CVT. This can be done by measuring the fuel consumption at different operating points. Testing the timing of the injection is also of importance, as the optimal injection times may vary from the timings that is used in the current Aprilia system.

Direct Injection has very high potential to lower the fuel consumption for the CVT, and as AADI utilises lower injection pressure into the combustion chamber, it is considered a good system for the CVT. However, the effect of implementing AADI and direct injection in the CVT engine still has to be investigated. There are some uncertainties regarding the effects that AADI can have on the fuel consumption in the CVT engine, as most test that have been performed has been on larger engines, and engines that operate in a different way than the CVT engine. The AADI system is also currently used in Aprilia Scooters, which has a two-stroke engine, and the fuel efficiency needs to be evaluated for a four-stroke engine like the CVT engine as well.

The literature study showed that the best pressure for injecting the air-fuel mixture is 6.5 bar but as the CVT is limited to 5 bar this will have to be the maximal injecting pressure. Therefore the AADI system will not be optimal, and the effects on fuel efficiency are not guaranteed.

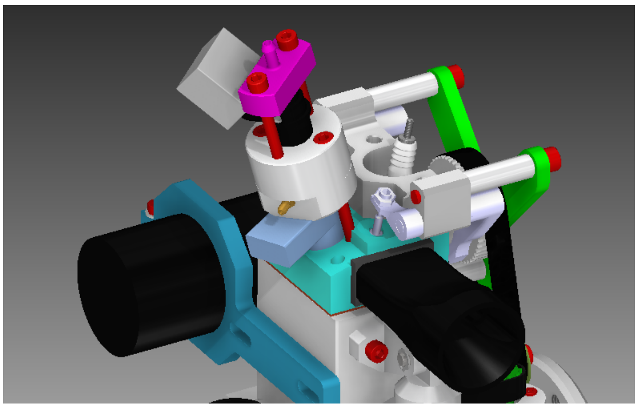


Figure 46: The manufactured AADI system on the technology demonstrator

9 Summary

The purpose of the project was to implement, evaluate and demonstrate different fuel efficiency improving ideas on an technology demonstrating engine for the CVT. In the finalized, manufactured engine all of the chosen technologies, except for the laser ignition, was implemented and thus demonstrated. Figure 47 shows the CAD model of the manufactured engine. However, due to complications and lack of time, final assembly, engine control and pressurizing systems together with testing of the engine could not be completed before the finalisation of this thesis.

In the ceramic chapter, two different techniques was investigated. One of the techniques investigated the possibility to use whole parts in ceramic material. This technique was realised as a ceramic cylinder liner where the selected material was zirconia KZY-8% which is a form of Y-PSZ. This manufactured cylinder liner is ready to mount in the CVT MKVI engine for testing and evaluating the affect on the heat losses and fuel consumption. The second technique that was investigated was ceramic coating and this was not realised because of the cost for coating the parts by plasma spraying were deemed too high within the budget of the project. A simpler and cheaper coating technique named SST was investigated and seemed to be realisable but for future implementations.
The mechanically easiest way of implementing the Atkinson cycle in a conventional Otto engine is to modify the cam lobes to reduce the amount of intake air. This project has chosen this method of implementation and developed a model for calculating the optimum volumetric efficiency and which cam lobe profile that could achieve this in the CVT MKVI engine. In addition, four different calibrating cam lobes have been manufactured for further testing and evaluation. To make fair comparisons of the different cam lobes effect on thermal efficiency the compression ratio needs to be held at a constant level and is therefore preferably tested together with the variable compression technology.

The concept for laser ignition of the CVT engine was developed based on previous experiments, and with regards to the specifications of commercially available equipment. Due to limited response from laser equipment manufacturers as well as the high cost of such equipment, laser ignition was never realised in the technology demonstrator. The recommended specifications for lasers and optics can instead be used as guidelines for future projects.

The variable compression concept was based on a currently existing concept developed by SAAB, just implemented on the CVT engine. The main complication with this concept was to find a linear actuator that was small enough to fit between the cylinder and the crankcase, but still be able to withstand the upcoming forces in the engine. Due to long shipping times of the components of the linear actuator the chosen concept was not completely manufactured. To still be able to test different compression raitos and its effect on the fuel efficiency a new design was developed and manufactured. This new design consists of a screw that would make it possible to manually change the compression ratio when the engine is turned off.

The direct injection system which was chosen to be implemented in the technology demonstrator was AADI, which means that the fuel is pre-mixed with air before being injected into the combustion chamber. The reason why this was chosen is that the pre-mixture with air means that the injection pressure is lower than with conventional direct injection.

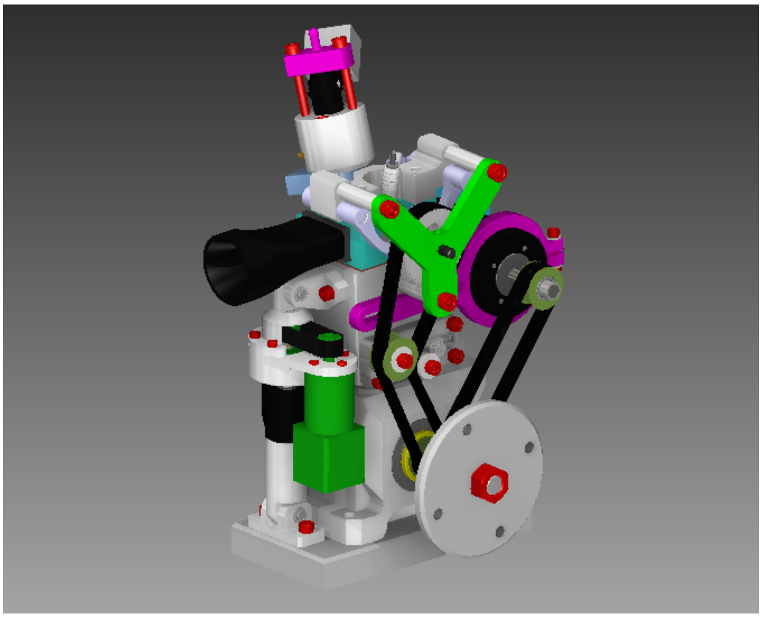


Figure 47: CAD model of the technology demonstrating engine

This is favourable for CVT as they are limited to 5 bar injection pressure by the rules of Shell Eco Marathon. The mechanical components for the direct injection technology have been obtained but since the engine control and pressurizing systems are yet to be manufactured the technique have not been tested and assessed. The fuel efficiency improving potential is however judged to be good.

The technology demonstrator was never tested as the complications mentioned above meant that the manufacturing process was delayed. The finished engine was assembled, the construction seemed promising and it had great potential for future testing.

Testing the engine would first consist of testing the individual systems on their own, in an old CVT engine, to assess their individual fuel efficiency potential and afterwards testing the complete engine. The variable compression components however would not be tested in an old engine since it would require a lot of modifications but could be tested separately in the new engine with the other technologies taken away or shut off.

There are uncertainties regarding the actual fuel efficiency gained from the combined systems as they are in some ways dependent on each other. For example the ceramic coating and Atkinson technologies may affect the gas exchange process, lowering the compression ratio, and should therefore be used with the variable compression technology to be compensated for this. Otherwise thermal efficiency losses due to a lower compression ratio would negatively influence the individual results.

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Appendices

A Matlab script: Atkinson Optimization modelling

```
clear variables
close all
clc

%Energy density, petrol [J/kg]
H = 45*10^6;

%Engine cylinder size [m,m]
bore = 0.039;
stroke = 0.039;

%Lambda value
lambda = 1.05;

%Cv @ 20  C
cv = 0.718;

%Kappa
kappa = 1.4;
%Temperature, atmospheric air [ C ]
T = 20;
%Pressure, atmospheric air [bar]
p_atm = 1.035;

%Volumetric efficiency VE
ve = 0.8;

%Spec. gas coefficient, air [J/kgK]
R = 287.058;

%% Measurements on the MKVI engine from Invntor CAD model
%mm, distance between crankshaft centre and conrod mount centre
L_vc_vs= 19,5
%mm, length conrod c/c
L_vs=80
%mm, length from piston bolt centre to piston head
L_vs_kt = 13.8
%mm, distance between crankshaft centre and cylinder top
L_vc_ct=115.8

%Calculated least distance between piston head and cylinder top
L_tdc=L_vc_ct-(L_vc_vs+L_vs+L_vs_kt)

%mm^3, volume in the piston head
V_kolvtopp= 19368.819-18509.019
%mm^3, volume in the cylinder head
V_topp=58927.238-55889,794
%mm^3, volume between piston head and cylinder top, at TDC
V_tdc=L_tdc*(bore*1000/2)^2*pi

%Total compression volume in m^3
```

```
V_comp=(V_kolvtopp+V_topp+V_tdc)/10^9
%Total cylinder volume in m^3
vol = ((bore/2)^2*stroke*pi+V_comp);
%Previously calculated Compression ratio with 80% VE
Komp_org=10.3615

 %% Friction losses during one cycle
    FMEP = 1 %bar, according to report "GT Power GX35 VERA 2009 -
    %David Willermark & Fredrik Dunert"

    Floss = FMEP*10^5*(bore/2)^2*pi*stroke*4 %[J] Total friktionloss
    %during one cycle

%% Different Atkinson levels
m=100 %number of steps (atkinson levels) to try
%percent of the orininal compression volume
vf=linspace(0,1,m)
for j=1:m

    %% First point, start of compression
    p1_at(j) = p_atm;
    t1_at(j) = T+273.15;
    v1_at(j) = vol*vf(j); %(cylinder + cylinder head) * current atkinson
    %level

    %% Second point, start of combustion
    v2_at(j) = v1_at(j)/Komp_org; %Compresses to the same volume as the
    %original MKVI engine, assumed to be able to be changed
    p2_at(j) = p1_at(j)*(v1_at(j)/v2_at(j))^kappa
    t2_at(j) = t1_at(j)/((p1_at(j)/p2_at(j))^((kappa-1)/kappa))

    %% Combustion process
    %air mass [kg] (calculated with the VE)
    m_luft = v2_at(j)*p2_at(j)*10^5/(R*(t2_at(j)));
    %Air-Fuel ratio
    afr = 14.7*lambda;
    %Energy, one cycle
    m_br = m_luft/afr;
    E_at(j) = m_br*H
    %pv-diagram:
    m_tot = m_luft + m_br;

    %% Third point, after combustion
    t3_at(j) = E_at(j)/(m_tot*cv*1000)+t2_at(j)
    p3_at(j) = p2_at(j)*(t3_at(j)/t2_at(j))
    v3_at(j) = v2_at(j)

    %% Fourth point, after expansion on the atkinson cycle but stopped
    % at the current atkinson level volume and not to atm. pressure
    v4_at(j) = vol
    p4_at(j) = p3_at(j)*(v3_at(j)/v4_at(j))^kappa
    t4_at(j) = t3_at(j)/((p3_at(j)/p4_at(j))^((kappa-1)/kappa))

    %% Fifth point, after the pressure in the exhaust have been let out
    p5_at(j) = p_atm
    v5_at(j) = vol
```

```
    t5_at(j) = t4_at(j)*p5_at(j)/p4_at(j)

    %% Thermodynamic losses
    %Loss from point 4 to 5
    loss4till5(j) = m_tot*cv*1000*(t4_at(j)-t5_at(j))
    %Loss from point 5 to 1
    loss5till1(j) = m_tot*cv*kappa*1000*(t5_at(j)-t1_at(j))

    %% Thermal efficiency at current atkinson level
    etaAtkinsonNonIdeal(j)=(E_at(j)-loss4till5(j)-loss5till1(j)-Floss)/...
    E_at(j)

end

%% Plot of the atkinson cycle
% Chose which of the atkinson level cycles to plot
m=100

P_1till2=@(v)(p2_at(m)*(v2_at(m)/v)^kappa)
P_3till4=@(v)(p3_at(m)*(v3_at(m)/v)^kappa)
figure(1)
hold on
fplot(@(v)P_1till2(v),[v2_at(m) v1_at(m)])
plot([v2_at(m) v3_at(m)],[p2_at(m) p3_at(m)])
fplot(@(v)P_3till4(v),[v3_at(m) v4_at(m)])
plot([v4_at(m) v5_at(m)],[p4_at(m) p5_at(m)])
plot([v1_at(m) v5_at(m)],[p1_at(m) p5_at(m)])
plot(v1_at(m),p1_at(m),'bo',v2_at(m),p2_at(m),'bo',v3_at(m),p3_at(m),...
    'bo',v4_at(m),p4_at(m),'bo',v5_at(m),p5_at(m),'bo')
xlim([0 v4_at(m)*1.2])
ylim([0 p3_at(m)*1.2])
Str= sprintf('Nonideal Thermal Efficiency at %2.0f%% VE is %2.0f%%',...
    vf(m)*100,etaAtkinsonNonIdeal(m)*100)
text(v3_at(m)+0.000005,p3_at(m)-5,Str)
xlabel('Volume [m^3]')
ylabel('Pressure [bar]')
title('P-V diagram')


%% The comparing Otto cycle
%% First point, start of compression
    p1 = p_atm;
    t1 = T+273.15;
    v1 = vol*ve;

    %% Second point, start of combustion
    v2 = v1/Komp_org;
    p2 = p1*(v1/v2)^kappa
    t2 = t1/((p1/p2)^((kappa-1)/kappa))

    %% Combustion process
    %air mass [kg] (calculated with the VE)
    m_luft = v2*p2*10^5/(R*(t2));
    %AFR
    afr = 14.7*lambda;
    %Energy one cycle
```

```
    m_br = m_luft/afr;
    E = m_br*H
    %pv-diagram:
    m_tot = m_luft + m_br;

    %% Third point, after combustion
    t3 = E/(m_tot*cv*1000)+t2
    p3 = p2*(t3/t2)
    v3 = v2

    %% Fourth point, after expansion back to v1
    v4 = vol
    p4 = p3*(v3/v4)^kappa;
    t4 = t3/((p3/p4)^((kappa-1)/kappa))

    %% Fifth point since VE makes it non-ideal
    p5 = p_atm
    v5 = vol
    t5 = t4*p5/p4

    %% Thermodynamic Losses Otto cycle
    loss4till5_otto = m_tot*cv*1000*(t4-t5)
    loss5till1_otto = m_tot*cv*kappa*1000*(t5-t1)

    %% Thermal Efficiency Otto
    %etaOttoIdeal=(E-loss4till5_otto-loss5till1_otto)/E
    etaOttoNonIdeal=(E-loss4till5_otto-loss5till1_otto-Floss)/E


%% Plots of Thermal efficiencys at differnet atkinson levels
figure(2)
hold on
plot(vf,etaAtkinsonNonIdeal)
[max, andel]=max(etaAtkinsonNonIdeal)
plot(andel/100,etaAtkinsonNonIdeal(andel),'o')
Str1= sprintf(...
'Maximum efficiency of %2.0f%% at %2.0f%% volumetric efficiency',...
max*100,andel)
text(0.2,max+0.1,Str1)
xlabel('Volumetric Efficiency')
ylabel('Non-Ideal Thermal Efficiency')
ylim([0 1])
legend('Thermal efficiency non-ideal modified Miller cycle')

%% Plot of the otto cykle
%Plot functions
P_1till2=@(v)(p2*(v2/v)^kappa)
P_3till4=@(v)(p3*(v3/v)^kappa)
figure(1)
hold on
fplot(@(v)P_1till2(v),[v2 v1],'r')
plot([v2 v3],[p2 p3],'r')
fplot(@(v)P_3till4(v),[v3 v4],'r')
plot([v1 v4],[p1 p4],'r')
plot(v1,p1,'ro',v2,p2,'ro',v3,p3,'ro',v4,p4,'ro')
ylim([0 p3*1.05])
```

```
xlim([0 v4*1.2])
Str= sprintf(...
'Nonideal Thermal Efficiency of %2.0f%% with conventional Otto cycle'...
,etaOttoNonIdeal*100)
text(v3_at(m)+0.000005,p3_at(m)-25,Str)
legend('Modified Miller Cycle','Otto Cycle')
```

B Matlab script: Air flow and cam lobe modelling

```
clear variables
close all
clc
%% Cam lobe and air flow modelling
%Calculates the air mass that is inhaled during the intake and
%compression stroke in the chalmers vera team MKVI engine with a defined cam
%lobe profile at a defined engine speed. Also calculates the VE.

%% General variables
%define variables
rpm=3000; %rpm on the crankshaft
ant_int=400; %number of calculating steps during the intake and compression
%stroke

%% Cam lobe variables
%define variables
xmax=0.0020125; %maximum valve opening in m, max 3,5 mm p  MKVI
Theta_vco=-20; %Crank angle degrees when valve opening shall occur before
%TDC shall be negative
Theta_vc=200; %Crank angle degrees when valve closing shall occur after TDC
%efter TDC
R_bas=0.014; %m base radius on the cam lobe
N_arm=3.5/8; %ratio of valve to cam lobe lift due to the rocker arm

%% Cam lobe calculations
L_kam=xmax/N_arm; %maximum cam lobe lift

Theta_vco=Theta_vco*pi/180; %valve opening in rad
Theta_vc=Theta_vc*pi/180; %valve closing in rad
dTheta_v=(Theta_vc-Theta_vco)/2; %total valve opening in rad at the cam
%profile

Theta_o=pi/2-dTheta_v/2; %opening angle from the positive x-axis on the cam
%profile in rad

Theta_c=pi/2+dTheta_v/2; %closing angle from the positive x-axis on the cam
%profile in rad

x_o=R_bas*cos(Theta_o); %x coordinate where opening starts
y_o=R_bas*sin(Theta_o); %y coordinate where opening starts

%% The cam profiles form function, 4th degree polynomial
%f(x)=c(1)*x^4+c(2)*x^3+c(3)*x^2+c(4)*x+c(5)
%calculates the coefficent c1-5 according to the conditions in matrices
%A and b below

%Conditions
%Row 1: f(0)=maximum lift  (L_kam)
%Row 2: f(x_o)=y_o
%Row 3: f'(x_o)=tangent to the base circle at x_o
%Row 4: f(-x_o)=y_o
%Row 5: f'(-x_o)=tangent to the base circle at -x_o
```

```
A=[0            0            0            0            1
   x_o^4        x_o^3        x_o^2        x_o          1
   4*x_o^3      3*x_o^2      2*x_o        1            0
   (-x_o)^4     (-x_o)^3     (-x_o)^2     (-x_o)       1
   4*(-x_o)^3   3*(-x_o)^2   2*(-x_o)     1            0];

b=[R_bas+L_kam
   y_o
   -x_o/(sqrt(R_bas^2-x_o^2))
   y_o
   -(-x_o)/(sqrt(R_bas^2-(-x_o)^2))];

c=A\b; %calculates the coefficents c1-5

%The cam profile form function
Fu=@(x)(c(1)*x^4+c(2)*x^3+c(3)*x^2+c(4)*x+c(5));
%%The cam profile base circle function
Fb=@(x)(sqrt(R_bas^2-x^2));

%% Engine design input
L=0.08; %m conrod length
SL=0.039; %m Stroke length
D_c=0.039; %m Cylinder diameter
D_i=0.013; %m diameter intake valve

%Compression volume, cylinder head volume + piston head volume + cylinder
%volume above TDC acc to CAD model in Inventor

V_0=7.4818e-06; %[m^3]

%One revolution of the crankshaft is one half revolution on the cam lobe
Theta_ka_start=Theta_o-Theta_vco/2; %Angle on the cam lobe where the intake
%stroke starts in rad

%loops from the start of the intake stroke and a half revolution forwards
Theta_ka=linspace(Theta_ka_start,pi+Theta_ka_start,ant_int);

%% Calculates the lift of the valve for each calculating step during the
%revolution
for i=1:length(Theta_ka)
    %checks wether the valve is open at the current step
    if Theta_ka(i) > Theta_o && Theta_ka(i) < Theta_c
        %If it is the cam lobe profile is further from the center than the
        %base circle radius.

        %x and y coordinate on the base circle at current calculating step
        xb(i)=R_bas*cos(Theta_ka(i)) ;
        yb(i)=R_bas*sin(Theta_ka(i)) ;

        %loops over the cam lobe profile in x-direction to find where a
        %straight line from origo to the base circle and the cam lobe
        %profile
        %intersects.
        xi=-R_bas ; %x-coordinate where the loop starts
        rest=1 ;  %predefine the reidual term
```

```
        %close to pi/2 the slope goes to infinity and the x-coordinate
        %is set to 0 automatically.
        if abs(Theta_ka(i)-pi/2) > 0.01
            while rest > 0.0001 && xi<R_bas
                %loops until the height-distance between the straight
                %line and the
                %cam lobe profile is less than 0.1 mm
                rest=abs(c(1)*xi^4+c(2)*xi^3+c(3)*xi^2+c(4)*xi+c(5)- ...
                    yb(i)/xb(i)*xi);
                %xu    x-koordinaten d r skillnaden berknas
                xi=xi+0.000001;
            end
        else
           xi=0 ;
        end
        %x-coordinate for the cam lobe profile at current calculation step
        xu(i)=xi ;
        %y-coordinate for the cam lobe profile at current calculation step
        yu(i)=c(1)*xi^4+c(2)*xi^3+c(3)*xi^2+c(4)*xi+c(5) ;
        %The distance between the cam lobe profile and the base cicrle
        %which equals the lift on the cam lobe
        diff(i)=sqrt(yu(i)^2+xu(i)^2)-sqrt(xb(i)^2+yb(i)^2) ;
        %The equal valve lift
        Vent_lyft(i)=diff(i)*N_arm ;
    else
        %if the valve is closed the lift is 0
        Vent_lyft(i)=0 ;
    end

end

%% Calculates the amount of inhaled air for each calculating step
Kappa=1.4; %c_p/c_v for air
R=287; %ideal gas coefficent
V=0; %air velocity before valve opening
P0=101325; %atmospheric pressure
T0=273+20; %atmospheric temperature
h=100; %Convection coefficient between the cylinder wall and the air
%[W/m^2*K]
cp=1009; %specific heat capacity air [J/kg*K]
T_cyl=273+200; %constant temp cyl wall [K]
P_efter(1)=P0; %Pa cyl pressure assumed to be atmospheric before intake
%stroke begins
V_c(1)=V_0; %cylinder volume at TDC
T_efter(1)=273+20; %Air temperature assumed to be atmospheric before
%intake stroke begins
m_c(1)=P_efter(1)*V_c(1)/(R*T_efter(1)); %air mass in the cylinder at TDC

dT=60/rpm/ant_int; % deltaT, time duration per calculating step,
%seconds per revolution / no of steps = s/step

%CAD to be calculated at, starts at the vertical axis and one revolution,
% intake and compression stroke = 2*Pi rad
Theta_a=linspace(0+0.00001,2*pi-0.00001,ant_int);

%Calculates the amount of inhaled air for each calculating step
```

```
for i=2:length(Theta_a)
%distance between crankshaft and piston bolt
Sy(i)=sin(pi-Theta_a(i)-asin(SL*sin(Theta_a(i))/(2*L)))/sin(Theta_a(i))*L;
%distance from TDC to piston head
y(i)=SL/2+L-Sy(i);
%Cylinder volume at current calculation step
V_c(i)=D_c^2/4*pi*y(i)+V_0;
%Cylinder pressure at current CAD, before inhaled air during the caluclated
%step
P_innan(i)=P_efter(i-1)*(V_c(i-1)/V_c(i))^(Kappa);
%Cylinder air temp at current CAD, before inhaled air during the caluclated
%step
T_innan(i)=T_efter(i-1)/((V_c(i)/V_c(i-1))^(Kappa-1));
%Intake valve open area at current calculation step
A_i(i)=Vent_lyft(i)*D_i*pi ;
%Mean valve open area during the calculating step
A_i_mean=(A_i(i)+A_i(i-1))/2 ;

%Cd discharge coefficient
%Corrects the ideal flow through the valve with flow resistance i percent
%of ideal flow. Values are obtained from the "Eco-marathon engine" Bachelor
%thesis project 2011:17.

%Ratio between valveopening and the valve diameter.
Ld=[0 0.08 0.16 0.24 0.3 0.38];
%The discharge coefficient that corresponds to that ratio
Cd=[0 0.11 0.25 0.34 0.38 0.4];

j=2; %initiate counter
%Loops to the correct discharge coefficient at the current calculation step
    if Vent_lyft(i)/D_i < max(Ld)
        while Vent_lyft(i)/D_i > Ld(j)
            j=j+1;
        end
        %Interpolates the discharge coefficient from the values above
        C_d(i)=Cd(j-1)+(Cd(j)-Cd(j-1))/(Ld(j)-Ld(j-1))*(Vent_lyft(i)/D_i-...
            Ld(j-1));
    else
    %if the valve opening is large enough the discharge coefficient is set
    %to its maximum value
    C_d(i)=max(Cd);
    end

%Calculates the air flow cuased by the pressure difference at the current
%calculation step acc. to eq. 9.46a and 9.47 in White, fluid mechanics book
    if P_innan(i)/P0 > 0.5283 %Checks if the flow is choked
        %if its not this formula is used
        m_dot(i)=C_d(i)*A_i_mean*P0/(sqrt(R*T0))*sqrt((2*Kappa/(Kappa-1))...
            *(P_innan(i)/P0)^(2/Kappa)*(1-(P_innan(i)/P0)^((Kappa-1)/Kappa)));
    else
        %if it is this formula is used instead
        m_dot(i)=C_d(i)*P0*A_i_mean/sqrt(R*T0)*Kappa^(1/2)*...
            (2/(Kappa+1))^((Kappa+1)/(2*Kappa-2));
    end
%Calculates the amount of air thats been inhaled during the time of the
%current calculation step, deltaT.
```

```
dm(i) = dT*m_dot(i);
%Adds it to the total air mass in the cylinder at the current calc. step.
m_c(i)= m_c(i-1)+dm(i);

%Heat energy transferred from the cylinder wall during deltaT
Q_heat(i)=(y(i)*D_c*pi+2*D_c^2/4*pi)*h*(T_cyl-T_innan(i))*dT; %[J]
%Updates the temperature of the air after the inhaled air and added heat
%energy
T_efter(i)=(m_c(i-1)*T_innan(i)+dm(i)*T0)/m_c(i)+(Q_heat(i)/(m_c(i)*cp));

%Updates the in cylinder pressure at the current calc. step after inhaled
%air and heat enegy added
P_efter(i)=m_c(i)*R*T_efter(i)/V_c(i);

end

%% Results
%Total cylinder displacment volume
Vc_tot=D_c^2/4*pi*SL+V_0

%Total normal air volume (at 1 atm pressure and T=293 K) that has been
%inhaled during one cycle
Vc_est=m_c(length(Theta_a))*R*T0/P0

%Calculates the volumetric efficiency of the cycle
VE=Vc_est/Vc_tot

%% Plots
% plot of the cam profile
figure(1)
hold on
fplot(Fb,[-R_bas R_bas])
fplot(Fu,[-x_o x_o],'black')
xlim([-R_bas R_bas])
xlabel('m')
ylabel('m')
axis equal

%adds the intake stroke interval on the cam profile plot
plot([0 R_bas*cos(Theta_ka_start)],[0 R_bas*sin(Theta_ka_start)],'r--')
plot([0 R_bas*cos(Theta_ka_start+pi/2)],...
    [0 R_bas*sin(Theta_ka_start+pi/2)],'r--')
title('Reduced lift design with timing -20 / 200 CAD')
legend('Base Circle','Opening and Closing Profile','Intake Stroke Interval')
hold off

%Shows the distance from the piston head to TDC
figure(2)
plot(Theta_a,y)
ylabel('Avstnd frn TDC')
xlabel('Vevaxelvinkel')

%In-Cylinder Pressure during intake and compression stroke
figure(3)
subplot(1,2,1)
plot(Theta_a*180/pi,P_efter)
```

```
title('In-Cylinder Pressure during intake and compression stroke ')
ylabel('Absolute Pressure in Pa')
xlabel('CAD from TDC')
xlim([0 360])
%In-Cylinder Pressure during intake stroke only
subplot(1,2,2)
plot(Theta_a(1:ant_int/2)*180/pi,P_efter(1:ant_int/2))
title('In-Cylinder Pressure during intake stroke')
ylabel('Absolute Pressure in Pa')
xlabel('CAD from TDC')
xlim([0 180])

%Valve opening
figure(4)
plot(Theta_a,Vent_lyft)
ylabel('Ventilppning')
xlabel('Vevaxelvinkel')

%Air mass in the cylinder
figure(5)
plot(Theta_a,m_c)
legend('Luftmassa i cyl')
xlabel('Vevaxelvinkel')

%In-Cylinder air Temperature
figure(6)
plot(Theta_a*180/pi,T_efter)
title('In-Cylinder air Temperature')
legend('Temperature in K')
xlabel('CAD from TDC')
xlim([0 360])
```