CHALMERS

ECO-marathon engine

Development of an energy efficient internal combustion engine

Bachelor thesis project in applied mechanics

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Abstract

The purpose of this project was to continue the development of Chalmers new fuel efficient ECO-marathon engine and to build a working prototype. The new engine was to be used in the Shell ECO marathon competition. A few main development points were focused on during the project.

The first of these points was the development and testing of different cylinder linings for reduced friction losses. A lining with low friction against the piston and the piston rings was desirable. The cylinder lining should not affect the compression. To achieve even lower friction the friction coefficient of the piston and low-friction piston ring was reduced by using a surface treatment. The weight of the piston was also reduced to lower the reciprocating weight.

For the development of cam profiles a simulation model was required to accomplish low fuel consumption. The choice of engine fuel and ignition management system and optimization of fuel and spark parameters was of great importance for engine efficiency. This demanded sufficient knowledge of the engine management system.

The cylinder lining was manufactured in cast iron and then coated with a nano-composite surface coating containing tungsten disulfide (WS_2). The coating has extremely low friction (0.04) and thus it helps to reduce the friction losses between the piston ring and the cylinder lining. Piston rings in cast iron were Teflon treated. This surface treatment reduced the friction coefficient to about 0.04. It was not completely clear if the treatment would resist the high temperature and wear, so this was tested and evaluated.

The weight of the piston was reduced by removing a lot of material from the piston skirt. By removing this material, more than half of the contact surface between the piston skirt and the cylinder lining was eliminated which reduced friction. To further reduce the friction, the piston skirt was coated with a low friction surface, the same kind as was used on the cylinder lining.

Cam profiles were developed using the simulation program GT-power. These profiles were also manufactured and used for the first startup of the engine. The engine fuel and ignition management system was chosen based on a several aspects, such as performance and ability to get feedback from the manufacturer. To learn about the management system, parts of the group participated in a one day education at the supplier of the system. Besides the specified areas presented above, many other areas that were necessary to address to be able to implement tests and complete the engine was worked with and they are presented in this report.

Sammanfattning

Målet med detta projekt var att vidareutveckla Chalmers nya bränslesnåla ECO-marathonmotor och bygga en fungerande prototyp av den. Motorn ska användas i tävlingen Shell ECO marathon. Under projektet har fokus legat på ett antal punkter för att åstadkomma dessa mål.

Att utveckla och utvärdera olika cylinderfoder var ett delmål. Ett foder med låg friktion mot kolvringarna och kolven var önskvärt. Cylinderfodret fick inte försämra kompressionen.
Kolv och lågfriktionskolvring skulle utvecklas, delvis samma problem som för cylinderfodret med det var också önskvärt att minska kolvens vikt då all massa i rörelse kräver energi.

Framtagning av kamprofiler behövdes för att dels komplettera motorn för att överhuvudtaget kunna starta den, dels skulle kamprofilerna designas för bästa bränsleeffektivitet. En virtuell modell skapades för att underlätta detta arbete. Motorstyrsystem behövde också väljas samt optimeras med avseende på BSFC. Detta krävde att kunskap om styrsystemets mjukvara skaffades.

Cylinderfoder i gjutjärn tillverkades och dessa ytbehandlades därefter med en nano-komposit-behandling innehållande volframdisulfid (WS_2). Denna ytbeläggning har extremt låg friktionskoefficient (0.04) och den hjälper därmed till att minska friktionsförlusterna mellan kolvringen och cylinderfodret. Kolvringar i gjutjärn behandlades med en teflonyta som har en friktionskoefficient på 0.04. Från början var det något oklart om denna behandling skulle hålla med tanke på det stora slitage och värme den utsätts för, så detta testades och utvärderades.

Kolvens vikt minskades genom att eliminera en hel del material från kolvkjolen. Genom att ta bort material från kolvkjolen eliminerades även mer än halva kontaktytan mellan kolven och cylinderfodret och därmed minskades friktionsförluster. För att ytterligare minska friktionsförlusterna ytbehandlades kolvkjolen med samma typ av behandling som cylinderfodret.

Kamprofiler togs fram med hjälp av simuleringsprogram och dessa tillverkades sedan och användes vid motorns uppstart. Valet av styrsystem gjordes utifrån ett antal punkter och för att sedan kunna använda och förstå styrsystemet deltog delar av gruppen i en heldagsutbildning hos styrsystemets leverantör. Förutom de specifika förbättringsområden som är beskrivna ovan, behandlades också många andra problem som var nödvändiga att lösa för att kunna starta motorn och utföra tester på den. Dessa är också presenterade i denna rapport.

List of abbreviations

BSFC – Brake Specific Fuel Consumption
TDC – Top Dead Center
BTDC – Before Top Dead Center
ATDC – After Top Dead Center
BDC – Bottom Dead Center
CAD – Computer Aided Design
CAM – Computer Aided Manufacturing
ECU – Electronic Control Unit
RPM – Revolutions Per Minute
ANS – Applied Nano Surfaces
TPS- Throttle Position Sensor
MAP-Manifold Air Pressure

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1 Introduction

The petrol driven internal combustion engine has been around for more than a century. It has been developed during the years and is still used in many vehicles today. The use of petrol and other fossil fuels is heavily debated today. Even though many people agrees that a complete change of energy source would be the best way to lower our impact on the climate, alternative energy sources needs further development before they are ready to fully replace fossil fuels. This makes it important to continue the development of the petrol driven engine in order to minimize the impact on the environment.

Shell Eco Marathon is a competition between universities where the goal is to develop a vehicle with as low fuel consumption as possible. Chalmers competes with two vehicles, Smart and Eco Vera. The current propulsion design for Eco Vera consists of a conventional petrol driven four-stroke engine from Honda. Smarter uses the same engine but in series with an electric engine, making it a hybrid vehicle. The Honda engine is of model GX 35 and it has a specific fuel consumption of 333g/kWh in its best point. It is designed to power chainsaws and leaf blowers and therefore high reliability, not low fuel-consumption, has been a priority. With this engine as a base, attempts to increase the efficiency have previously been made, for instance by installing electronic fuel injection and electronic ignition. Since the efficiency of the Honda engine was close to its maximum potential, the interest in developing a new engine had increased.

During the summer of 2010 at Chalmers University of Technology at the department of Applied mechanics, division of combustion, Anders Johansson and Andreas Thulin carried out a project to construct a modular engine. The engine was primarily meant to be used in the Shell Eco Marathon vehicle Vera. However there was not enough time to fully complete the construction of the engine during the initial project. As a result there were some parts yet to be designed when this project started. The most critical issue was the construction of the cam lobes that are used to lift the valves in the engine. Although many of the already existing parts could be used in the engine, further development of these would favor the fuel efficiency of the engine.

The purpose of this project was to complete and further develop the already existing Shell Eco Marathon engine that was developed by Anders Johansson and Andreas Thulin. This development would be carried out with fuel efficiency in mind. Exhaust emissions won't be taken into consideration since this isn't a part of the goal, although portion of hydrocarbon will be minimized to achieve as complete combustion as possible.

To succeed with this extensive task of improving the fuel-efficiency a quick analysis was made to divide this problem into smaller tasks and a number of developments points have been chosen:

- Development and testing of different cylinder linings. A lining with low friction against the piston and the piston rings was desirable. The cylinder lining shall not affect the compression.
- Piston and low-friction piston ring, partly the same problems as with the cylinder linings but it is also important to reduce the weight of the piston since all mass in movement consumes energy. Reliability of the engine must not be affected.

- Development of cam profiles, no camshaft have been developed before which means the task is to design a cam profile to optimize the fuel efficiency. A model for simulation is required to accomplish this and the next task.
- Choice of control system and optimization of the control parameters regarding BSFC. This demands that sufficient knowledge of the control software is gathered.

2 Method

The project was divided into two main parts. First the already existing engine was further developed to reduce fuel consumption and missing parts were designed. New designs and new materials of selected engine components were considered in order to improve the engine. The development work was carried out by conducting researches of literature from websites and books and through contact with different companies. When adequate knowledge was achieved for the different components, new designs and materials could be chosen. Also a simulation model was created in the computer program GT power to ease the designs of missing parts.

When the developed components were designed and manufactured, they were assembled into the engine and evaluated through a number of experiments and tests. These tests and experiments were the second main part of the project and their purpose was to evaluate the development work carried out in the projects first part.

3 Design, development and manufacturing of selected engine parts and prototype

This chapter describes the process when different components were developed for fuel efficiency. It also describes the manufacturing process of the parts that were not bought. For every part there is a short background and some theory in order to make it clear how the part affects the fuel efficiency. The final prototype is presented at the end of the chapter.

3.1 *Piston ring and cylinder lining*

The main function of the piston rings is to provide a gas and oil seal between the piston and the cylinder wall. When the piston moves to compress the fuel/air mixture and during the combustion, a high pressure is created above the piston. To achieve the high pressure that is desirable, there has to be a tight fit between the piston ring and the cylinder wall. To fulfill this function, the piston ring is manufactured with a larger diameter than the cylinder bore, but with a gap.

When the piston is assembled the piston ring is located in a groove in the piston. It is then compressed so that its dimension matches the cylinder and therefore allows the piston to be inserted into the cylinder. As a result of the compression the gap in the piston ring is closed and a built in tension is created. This pressure between the piston ring and the cylinder is what makes sure no gases slips pass, thus making it possible to achieve a high pressure above of the piston. If some combustion chamber gases are allowed to pass by the piston efficiency is reduced and the oil may be contaminated.

For four stroke engines, it is most common to have three piston rings. The first or the top ring has the main function described above. The second, often called the scraper ring, combines the main function with the function to limit the oil film so that oil not can enter the combustion chamber. The third ring is an oil ring, which has the function to provide the piston with oil. Only one piston ring is used in this engine. The reason for this was mainly to keep the friction low in order to increase efficiency. Every ring causes friction losses and our engine is not going to run for an extended period of time and can thereby be sufficiently lubricated without oil rings.

One of the most important things when the engine originally was designed was to make it modular. The ability to have changeable cylinder linings was one important aspect of this. The cylinder lining is the layer of the cylinder wall that is in contact with the piston and the piston ring.

If friction can be reduced in an engine, efficiency is gained. 6-9 % of the energy that is supplied to an internal combustion engine is lost to friction between the piston ring and the cylinder liner (Applied Nano Surfaces 2011), therefore the internal friction was given extra attention.

3.1.1 Design and development

To not affect the main function described in the previous section, no reduction in the normal force was made, i.e. the stiffness of the ring would be the same as for factory made piston rings. The only parameter left to improve was the friction coefficients. To achieve the goals with a reduced friction between the piston ring and the cylinder liner, the first thought was to explore if piston rings in Teflon would work, due to their low friction coefficient (~0.04) (see Figure 1)(Engineering toolbox). In house manufactured piston rings were lathed and tested in the Honda GX35 engine, but despite some modifications, for instance with a thin steel wire that would increase the stiffness, it seemed as if piston rings in pure Teflon wouldn't work for our application.

Figure 1 Piston ring in pure Teflon with steel wire.

An even pressure between the piston ring and cylinder liner is desirable, and this was not easy to achieve with Teflon, despite the built in steel wire. In order to be able to use Teflon to the piston ring, the stiffness of the materials in some way had to be increased.

After a lot of searching, a material with a mix of 60 % brass and 40 % Teflon was found. This material had a friction coefficient of 0.13 (Nordbergs Tekniska AB 2011), to compare with pure Teflon's friction coefficient of 0.04 (Engineering toolbox 2011). If this material was to be used, the piston ring had to be manufactured in our workshop at Chalmers. This was believed to be hard, at least if the result was to meet the requirement that we had on the piston ring's function. With this difficulty in mind, and the not too impressive friction coefficient for the brass mixed Teflon material, this concept was abandoned.

A piston ring which could create the same surface pressure between the piston ring and cylinder lining as the factory made and at the same time reduced the friction losses was sought. To succeed with the main function, it was decided that factory made piston rings would be used. However a surface treatment with Teflon would be added. The critical point with this surface treatment was if it would resist the big wear and the high temperature caused by the combustion. After conversations with companies in the business and temperature estimates, it was decided that it was worth trying. Three factory made piston rings where sent to Ahlins AB for Teflon coating.

Figure 2 From left: Teflon treated and untreated piston ring.

The piston rings that were decided to be evaluated in the friction test (see chapter 4.1) were the Teflon coated and the untreated cast iron piston rings (see figure 2). Together with the piston rings, different cylinder linings were also tested.

To get a sense of which materials would be possible to use for the cylinder liner, research on which materials that are in commercial use today was performed. The plan was to find some materials that seemed to have potential and then test the different liners to see how they differ.

It was found out that cast iron and Nikasil coated aluminum was commonly used for this application. After some more research, a company with a quite new technology for surface treatments for this application was found. The treatment they offered would give a friction coefficient of 0.04 (Nordbergs tekniska 2011). This is roughly the same friction coefficient as Teflon, which has one of the lowest friction coefficients for solid materials.

The reason why Teflon was not an alternative for the liner was because of the heat in the combustion chamber that would melt the Teflon coating. The company ANS was contacted. ANS's surface treatment is a nano-composite, containing tungsten disulfide (WS_2), and can work in a wide temperature interval.

According to ANS the surface is durable and does not scratch or flake. A round bar of cast iron was ordered for turning cylinder linings. Three linings in cast iron were manufactured, two of them were sent for coating by ANS, and one was kept untreated for reference (see figure 3). The cylinder linings that were sent to ANS were tested and the result of the test can be found in section 4.1.

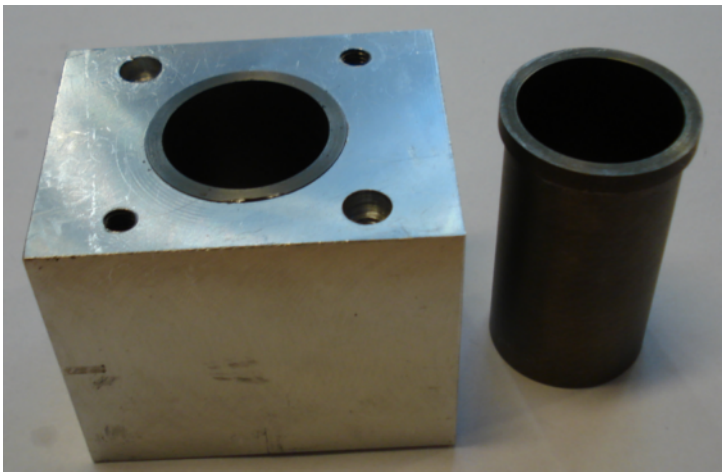

Figure 3 From left: Cylinder block with lining treated by ANS and untreated lining.

3.2 *Piston*

All moving parts in the engine are a source of friction and therefore contribute to fuel consumption. A heavier part produces a bigger friction force than a lighter one and thus it is important to reduce the weight of the moving parts if you want to improve fuel efficiency. A heavier part also demands more energy for the same acceleration according to Newton's second law, although in the ideal case this extra energy is given back when decelerating the engine. The engine will be running in short intervals where the engine will be revved from standstill to a desired rpm and then turned off. When the engine is turned off its clutch will disengage and as a result the energy built up in the engine's moving parts will be lost. Lighter engine parts means less energy wasted when the engine is turned off thus improving fuel efficiency.

The piston transports the energy from the combustion to the mechanical parts of the engine. Because the engine's optimum operating point is at a quite high rpm, the piston accelerates and decelerates fast, making the benefits of a light piston even bigger regarding fuel consumption.

Although the majority of friction loss against the cylinder lining comes from its contact with the piston rings, the contact between the skirt, the longer bottom part of the piston, and cylinder lining also contributes to the losses. It is therefore worth paying attention to which material that is chosen for the skirts surface.

The piston design will also affect the gas flow in the combustion chamber which has to be taken into consideration. A piston with a flat top crown design, the top part of the piston, will not induce the same flow as one with a dome on the top. A fast flow helps to spread the flame front, which ignites the air-fuel mix faster. The piston crown shape will also affect the compression of the engine by changing the volume of the combustion chamber.

3.2.1 Design and manufacturing

To reduce the weight of the piston, different designs and materials were taken into consideration and compared. The final design would have to minimize the amount of material used without risking mechanical failure of the piston. In order to know which material could be removed from the piston it was divided into two different parts, the crown and the skirt. They are separated by the piston ring groove. The crown must have enough material to withstand the high pressure and the high

temperature of the combustion chamber but the shape of the crown will also affect the flame front and the compression ratio.

The biggest contribution to friction from the piston is that from the contact between the piston skirt and the cylinder lining. This means that a minimal contact surface is desirable. The classical circular design of the piston skirt will by these principles lead to a lot of friction. This meant that this old design had to be abandoned to reduce friction. Inspiration for the new design came from studying pistons that were already in production. A very popular design was called the X-design. This basically means that more than half the skirt is removed on the side perpendicular to the piston pin axis (see figure 4). To further reduce the friction of the piston skirt the same surface treatment that was used on the cylinder lining was applied.

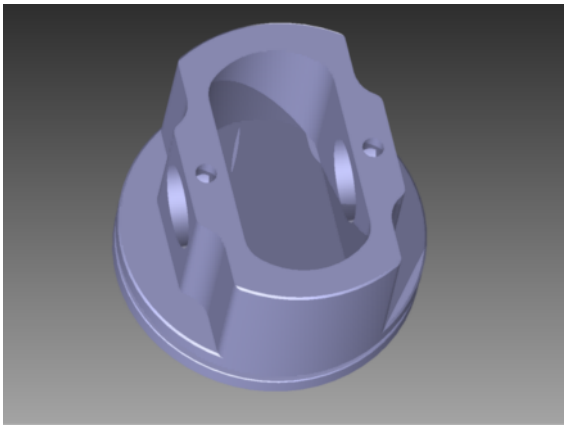

Figure 4 The piston skirt.

To avoid mechanical failure the material was made thicker where the hole for the piston pin is. Two small holes were drilled in the bottom of the piston skirt to maintain the lubrication from the oil mist. This idea was taken straight from the old piston design. The pocket in the middle of the piston is there so that the connecting rod can rotate freely around the piston pin axis. Since the piston pin can move freely from side to side it is important to make sure that the piston pin cannot move outside the piston and damage the cylinder lining.

The diameter of the piston skirt was set to 28.9 mm while the piston crown has a smaller diameter of 28.8mm. This is because of the thermal expansion of the piston crown due to high temperatures in the combustion chamber. The clearance between the piston and the cylinder lining is needed for the piston to run smoothly. With the cylinder lining having a diameter of 29mm a radial clearance of 0.05 mm was used on the piston skirt. These clearances were chosen because they were in the range recommended and because of the tolerances of the machines used to make the piston (Malcolm James Nunnery, 2007). Since it is impossible to make the piston exactly as on the drawing, a clearance in the middle of the span had to be chosen. The top part of the piston was made flat since the dome effect earlier mentioned already had been achieved due to the shape of the combustion chamber.

The material chosen for the piston would have to be light weight and easy to machine. Alumec is an aluminum based alloy with these desired properties. It’s also reasonably strong and have the strength required to mechanically hold together with a higher melting point and a higher tensile strength than Aluminum. This material was already available since it had been previously used by Ander Johansson and Anders Thulin when they were constructing the old version of the engine. These factors made it beneficial to make the piston in Alumec.

The piston was manufactured in a lathe out of an Alumec cylinder that had a diameter of 40 mm. By using a 4-jaw chuck the material was centered so that the throw in radius direction was tolerated. Since this piece needed further processing in both a lathe and a mill, 30mm extra material was added to the cylinder's/piston's length.

The piston's bottom had a more complex design with a lot of different radius milling and holes that had to be drilled with high precision. The workshop has a CNC-mill that was used to obtain the correct geometric design. The next step in the manufacturing was to drill the holes for the piston pin (see figure 5). For this operation one of the manual milling machines was used.

Figure 5 Piston after being machined in the CNC and drilled.

The piston crown needed a higher precision then the workshops smaller lathe could offer. The solution was to make a bush that the piston would fit in. The bush would center the piston material and was also needed for being able to put the piston in the lathe after the CNC-operation (see figure 6). Without the bush there would not be enough material to mount it into the lathe. It would then be possible to cut off the extra 30mm material.

Figure 6 Piston material before and after being machined in the lathe.

3.3 *Electronic Control Unit (ECU)*

The ECU is the brain of any modern engine. It receives signals from all of the engine's sensors and uses this information to control the fuel injection and the ignition. In a normal car it would also control a lot of other functions as the Electronic Climate Control. Examples of sensors are: Temperature (oil, water, exhaust, air temp in the intake), Pressure (oil, intake), Throttle position sensor, Lambda sensor (measures the air fuel ratio) and also any digital/analog signal that the engine manufacturer would like to send to the ECU. The ECU also controls the fuel pump, fuel injectors and the ignition.

In order to get the engine to run as desired it has to have the correct amount of fuel. When the engine is warm, the ECU should use lambda correction to accomplish as clean combustion as

possible. The Lambda value is a way of measuring the fraction between air/fuel in the exhaust. The lambda sensor is located in the exhaust system.

The ECU also need additional information to adjust the amount of fuel injected. One of these factors can be how much the throttle is open and how fast the throttle position is changed. The ECU needs these other signals in order to predict how much fuel that is needed in order to achieve a clean combustion. If you for instance step on the throttle, a lot more air will be sucked in to the engine than earlier, but the lambda sensor will tell the ECU that everything is fine. This would result in an incorrect air-fuel mixture that would reduce the efficiency of the engine.

An advanced aftermarket ECU lets you configure the fuel and ignition parameters depending on the value of the sensors. For example the system can handle almost any temperature-sensor after some calibration, and this sensor's lower values could for instance be defined as cold start and the higher as automatic emergency shutdown.

3.3.1 Choice of Electronic Control Unit (ECU)

The old ECO Marathon engine uses a NIRA I2 ECU. This ECU had a few problems that made it justified to look for an alternative system to use with the new engine. The main problem with NIRA was that it lacked support for small fuel injectors. This made it difficult to get the right amount of fuel injected. Another big problem was that NIRA did not support starter engines with high RPMs. This meant that the ECU thought that the engine was running whilst it was being dragged by the starter engine. This behavior led to the engine not getting the extra fuel needed to start it smoothly. The lambda regulation was also a problem since it took too long before it was activated.

The problems mentioned above were solved by using Civinco's ECU instead of NIRA's. The amount of fuel injected is related to how long the fuel injectors are open. In Civinco you are able set the time they stay open. This means that you can use any fuel injectors as long as you know the relation between the amount of time open, manifold air pressure, the fuel pressure and the amount of fuel injected. In Civincos ECU you can control which RPM that is to be considered idle RPM. This means that the ECU will not consider the engine started whilst it is being dragged by the starter engine. The problem with the lambda regulation was not an issue with Civinco's ECU. Their ECU allows the user to set the time after the engine has started in which the lambda regulation is activated.

The fact that Civinco was able to solve all the old problems and that Civinco is located in Gothenburg made it beneficial to choose their system. They are a small company located closer to Chalmers than NIRA which meant that a better support was to be expected. Civinco was also willing to add extra functions to the ECU if required.

After choosing ECU for the engine, sensors were needed in order to give the ECU sufficient information about the engine. The sensors that was used in the new engine was a MAP (Manifold Air Pressure) sensor which also measures the manifold air temperature, a engine temperature sensor, a throttle position sensor, a crank shaft sensor which reads the angle of the crankshaft, a cam shaft sensor which knows in which phase the engine is and a lambda sensor with its control system.

The TPS sensor was chosen to be the main load sensor and would control how much fuel that was injected. The other sensors will be used to adjust the amount of fuel injected depending on their value. If for instance the engine got too much fuel, this would be noted by the lambda sensor that would give this information to the ECU. The ECU would then adjust the amount of fuel and the lambda sensor would give new feedback. This created a self adjusting system which uses the sensors as input and steers the fuel injectors and ignition.

3.4 *Gas exchange*

Piston engines inhales and emits gas during its cycles, this process is called gas exchange. To provide the desired characteristics for an engine, the gas exchange process is the most important. The timing of opening and closing both the intake and exhaust valve is crucial for engine performance. Camshaft design is all about designing a valve lift profile to satisfy the engines gas exchange process.

In 4-stroke engines poppet valves is used to control the gas flow. The motions of the poppet valves are controlled by the camshaft. The camshaft consists of a number of lobes mounted on a shaft. These lobes forces the valves to open simply by pushing directly on them or by a mechanism, the valve is forced to close by a coil spring.

If one look at the cam lobes separately there are two main design aspects that have to be determined:

- Valve lift, how much the valve lifts as maximum
- Cam lobe duration, how many degrees the valve is open (measured in crankshaft degrees)

When the variables mentioned above are determined, the next step is to connect the point where the valve starts to lift and where it reaches its maximum. This is done by a curvature which is divided into four stages. The first stage is the lash ramp, this is where the valve lash is removed and the valve starts to open. This ramp has to be designed to have low velocity in order to safely remove the valve clearance and to start to open the valve smoothly. When the contact and the opening point are reached the velocity and acceleration can be increased to make the valve lift quickly, this part is the opening ramp. This is also the part where the force rises to its maximum and could be critical to the design and the construction.

The next step is the nose, this is where the greatest lift occur. The final two stages is closing ramp and closing lash ramp. Usually the cam lobe is symmetric, meaning the closing action is the same as the opening only in reverse. See figure 7 for an illustration of the different sections of the cam lobe.

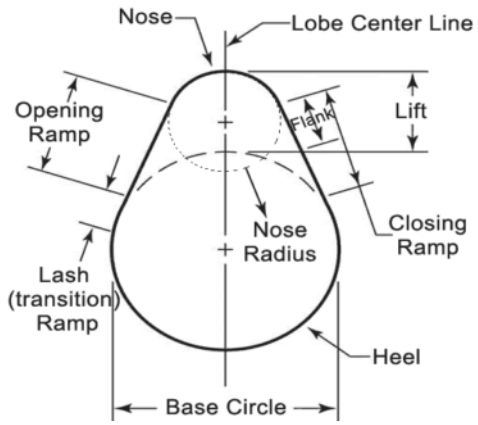

Figure 7 Cam lobe.

Valve lash is the mechanical clearance in the valve train between the camshaft and valve train mechanism in an internal combustion engine. Valve lash is usually about 0.10 mm depending on the engine specifications. Valve lash is intended to provide the greatest amount of valve opening on the high point of the camshaft lobe and assure that the valve is tightly closed on the low segment of the camshaft lobe.

In a commercial design software like GT-power four types of curves are used to describe the cam profile: lift (mm), velocity (mm/deg), acceleration (mm/deg^2) and jerk (mm/deg^3) (see figure8).

From these four curves it is possible to choose a number of variables to control the curvature and design the cam lobe. One feature to choose is called linear acceleration. This gives the user an opportunity to define the acceleration at each boundary which yields a linear acceleration profile. There are two constrains that are predefined by the software and they are: zero lift at boundary 0 and zero lift at boundary 12; this corresponds to start at opening ramp and the end of the closing ramp.

To minimize the wear of the cam lobe one seek to minimize the jerk (Tilden Technologies, LLC, 2009). To the left of graph four in figure 8 one can see that there is a maximum and a minimum jerk. These two peeks are irrelevant due to the lash ramp; it is just a transition when no contact occurs. To be able to get an efficient gas exchange one seek to get as much area under the lift curve as possible. The limitation is the acceleration that causes the force to rise upon higher acceleration. To save some time and minimize the complexity of the design a symmetrical cam lobe was made. Half a cam were designed and mirrored to get the symmetrical lobe.

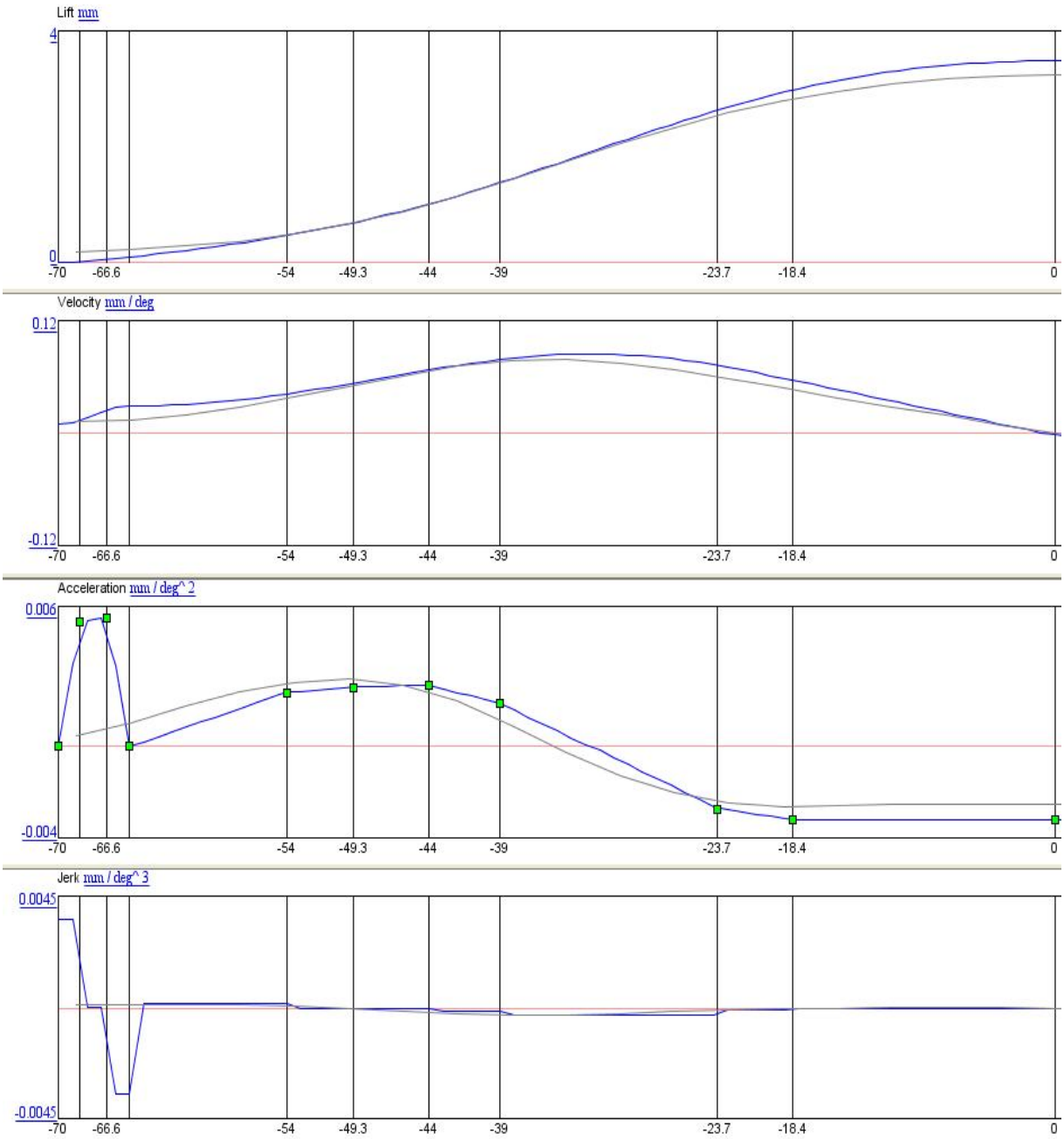

Figure 8 Different cam lobe parameters.

When a camshaft has been designed the next step is to create a follower mechanism which will provide the valve lift action. The follower mechanism was already designed by previous engine developers. A decision was made to keep this design and make a lift profile and camshaft for the engine. This mechanism could be used since the engine is operating at such a low speed. It is operating at an absolute maximum of 6000 rpm, compared to ~18000 rpm which is the maximum with a 40mm stroke, calculated by a maximum mean piston speed of 25 m/s.

3.4.1 Engine Computer simulation

To achieve the best possible result during the limited time offered, computer simulation tools were chosen to aid in the development of the engine. The main advantage is that the iteration-

optimization process can be made much faster. The software chosen is developed by Gamma Technologies, GT-Suite. This software offered all the features and variables necessary. The program features that were used were GT-ISE, VT-design and GT-post. GT-ISE gave the possibility to create a complete model of the engine which led to the opportunity to simulate and optimize different parts. In figure 9 you are able to see the complete structure of the engine model. VT-design made it possible to create and simulate the valve train mechanism. The last program GT-Suite delivers, GT-post, gave a complete view over the results in form of graphs and tables.

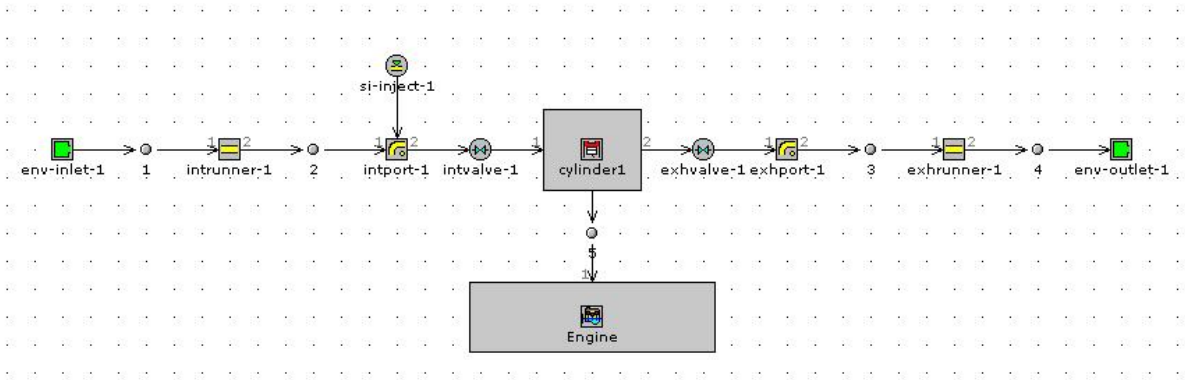

Figure 9 GT-ISE engine model structure.

The engine model is designed in the software by physical measurements and calculations. To create a realistic and accurate model of the engine, very precise data from the engine is required. Measurements were taken from both CAD models and the actual engine itself. To create an engine model, measurements from the engine while it was running was required. Because of this some predefined values was used to get the first model working.

The parameters that were considered to be of great importance to optimize with a computer simulation program were:

- Intake/exhaust runner length and diameter
- Camshaft design with lift, duration, timing and ramps

3.4.1.1 Cam lobe duration simulation

A critical aspect in the cam lobe design is the timing towards the crankshaft and the duration which corresponds to how many degrees the valve is open. The software GT-ISE was chosen to simulate and determine which duration and timing that corresponds to the lowest BSFC. Six different cam lobes were created in VT-design with different duration varying from 240 to 290 degrees with a step size of 10 degrees. This would give a wide span of different durations to be simulated. The next thing was to simulate intake and exhaust lobes with the same durations but with different timings toward the camshaft. The timings that were tested were:

- 450 Intake/270 Exhaust
- 450 Intake/255 Exhaust
- 450 Intake/240 Exhaust
- 465 Intake/270 Exhaust
- 465Intake/255 Exhaust
- 465Intake/240 Exhaust

From these simulations of each timing one cam lobe with specific duration and timing was chosen, this could be found in figure 10. The selection was based on the lowest BSFC. The cam lobe that was chosen was 260 degrees duration with 450 degrees Intake timing and 255 degrees exhaust timing. The timing was measured from TCD to the greatest lift.

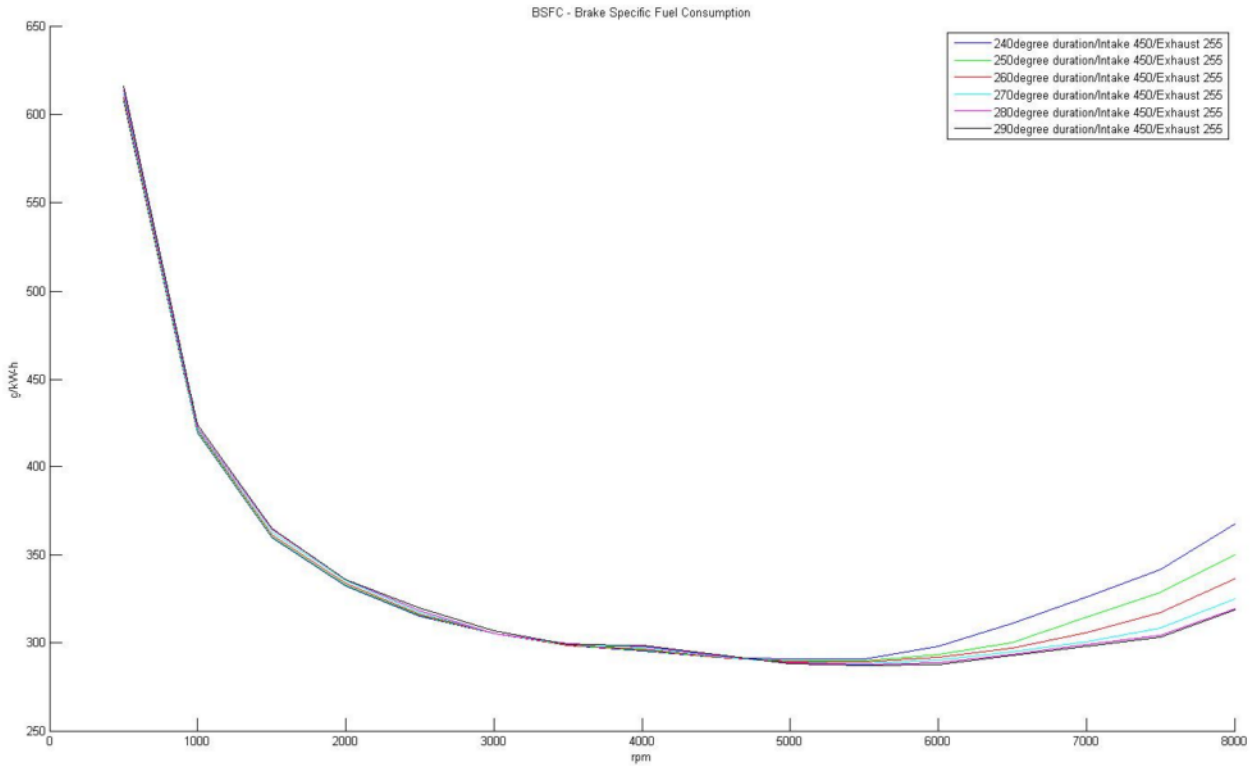

Figure 10 Graph of BSFC-RPM with different cam lobes.

3.4.1.2 Bench flow

The engine has two valves, one exhaust valve and one intake valve. To be able to accurately run simulations in GT-ISE the air flow of the intake and exhaust had to be determined. Therefore the complete head was sent to a professional engine tuner (Erland Cox Topplocksverkstan) to be flow tested and ported for optimal flow. Erland's results were presented as CFM (cubic feet per minute at 28" inches of water pressure difference). The results had to be converted to match our software which uses a flow coefficient Cd. This was made in a spreadsheet provided by Gamma Technologies. Our head was flowing about 0.4 Cd with 3.5mm lift on the valve. This was acceptable considering no initial testing was done during previous development. The calculated Cd values for each valve are presented in two different graphs in figure 11. The exhaust port flows better than the intake. Usually it is the other way round. The exhaust port should flow approximately 75% of the intake flow to have a good flow balance.

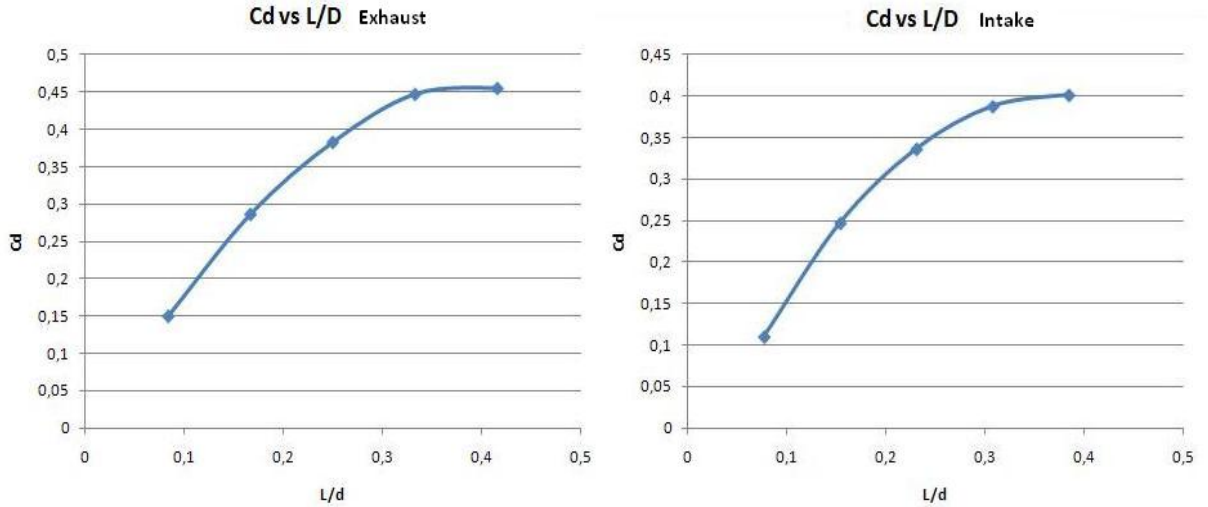

Figure 11 Graph of Cd values- L/d for intake and exhaust valves.

3.4.1.3 Final computer simulation results

When the symmetrical lift profile was determined with the specific parameters a cam profile had to be created, this was done in VT-design. Because of the valve train mechanism, the program calculated an asymmetric cam profile. This is seen in figure 12 where the lift profile corresponds to the red graph and the cam profile to the blue line. You can see that the cam curvature is slightly offset to the left. This is due to the gearing of the valve train mechanism.

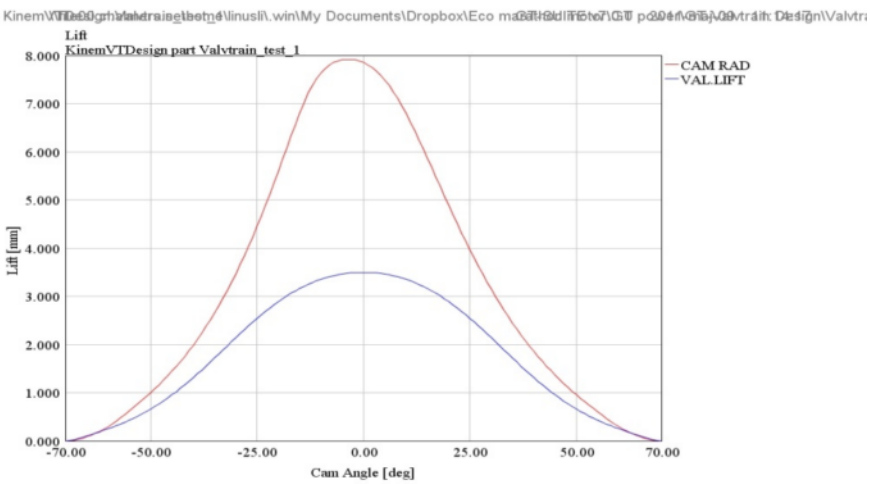

Figure 12 Graph of Lift [mm]-Cam angle [degree] with cam-/valve lift

The final simulation was done with GT-ISE. In order to make the manufacturing process easier and to save time, the same cam profile was chosen for intake and exhaust valve. The data that was of interest was BSFC, torque and BHP and is presented in figure 13-15. The BSFC reach a minimum of 286 g/kWh at 4500 rpm. Due to the form of the BSFC curve it is of great advantage to run the engine in the span of 3000-6000 rpm.

The torque curve (figure 14) is not very flat. It has a distinct peak around 3000-5500. This indicates that the engine is precisely tuned for this specific interval where it will operate well.
The power graph is very linear which is of great importance to the drivability of the car.

The maximum power output of the engine is 1.6 horsepower. This is considered to be more than enough.

One must keep in mind that this software is a tool for quickly evaluating different concepts. The dyno equipment will provide data on the actual performance of the engine. It is possible to use the dyno data to create a more accurate computer model for future use.

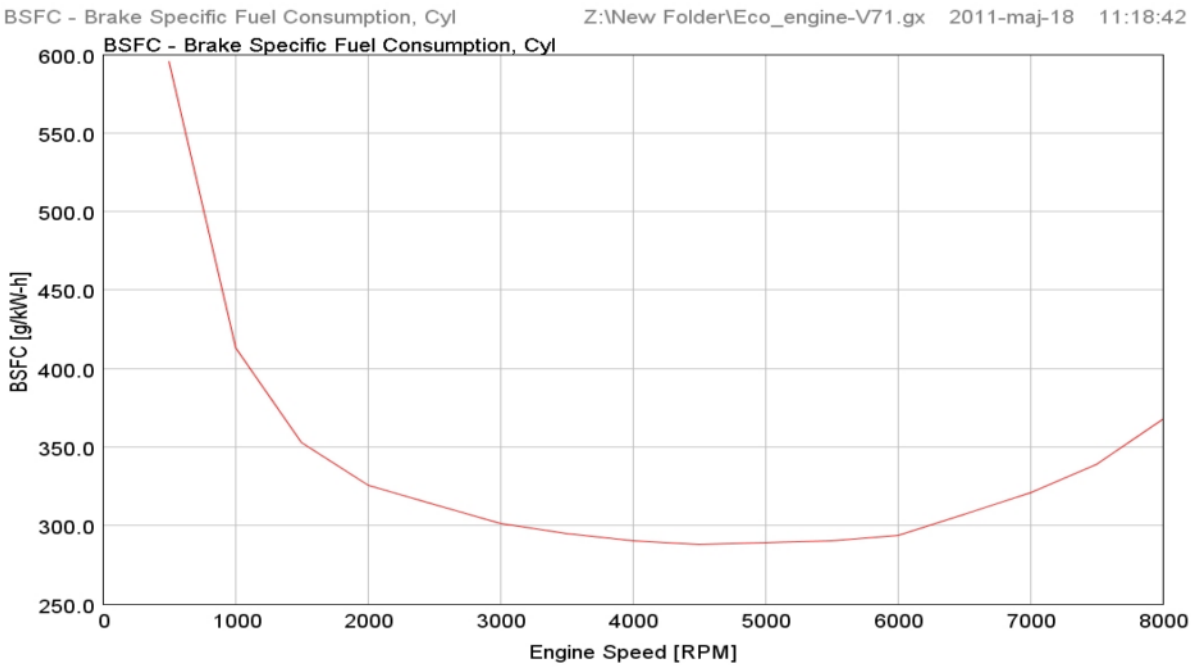

Figure 13 Graph of BSFC-rpm

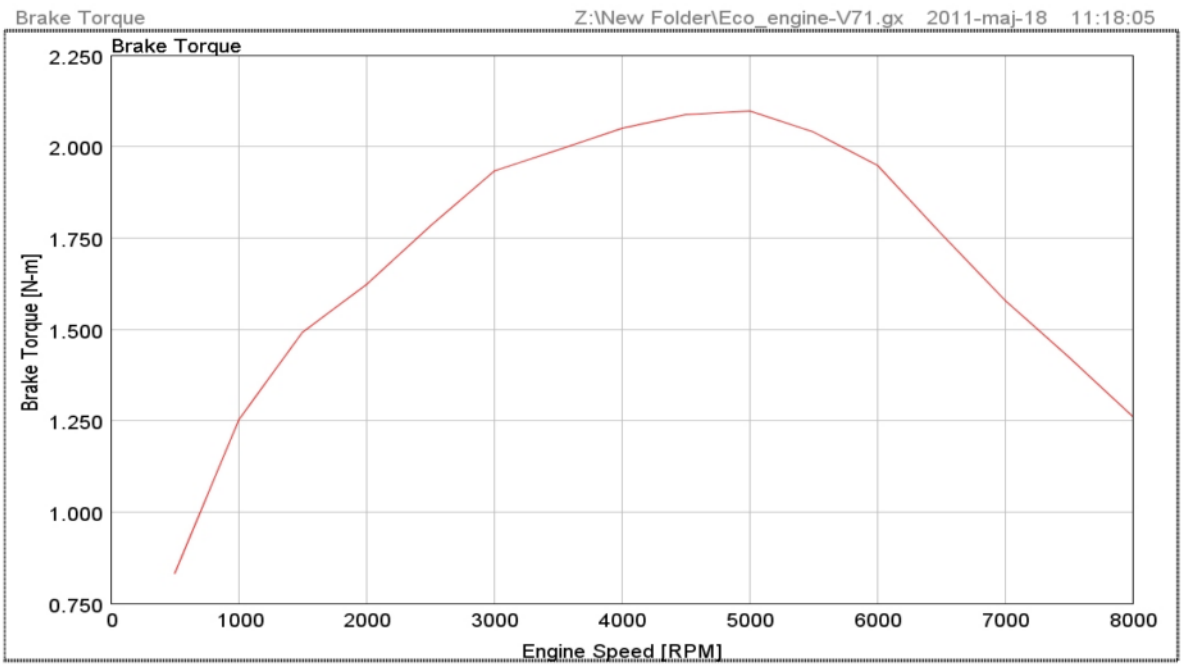

Figure 14 Graph of brake Torque-rpm

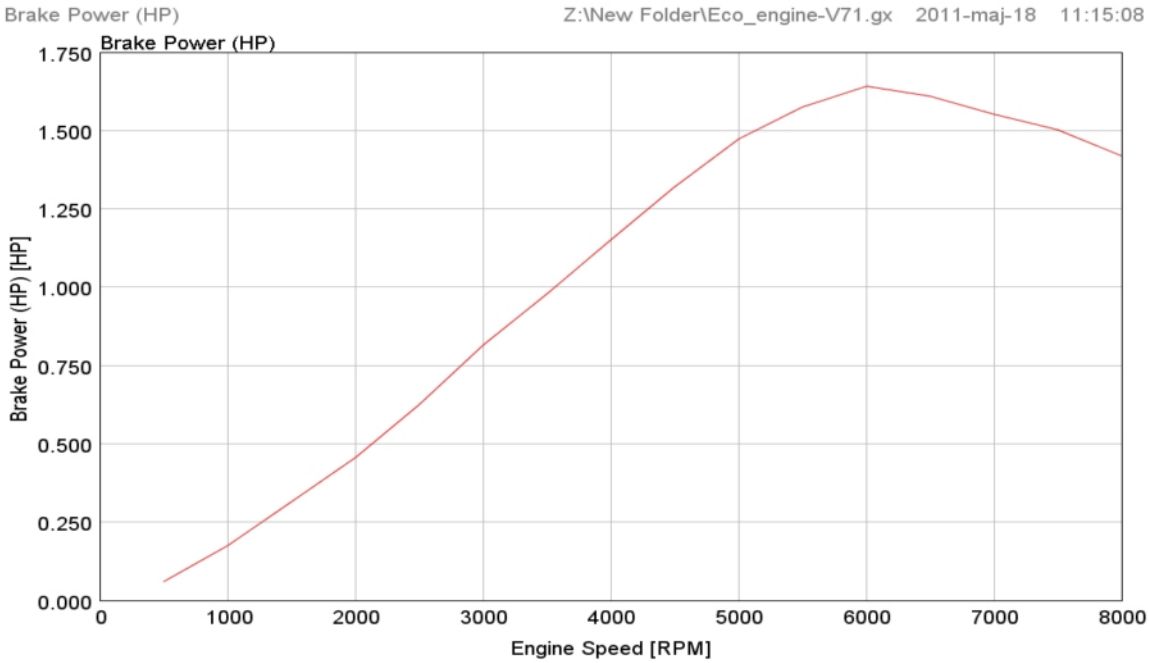

Figure 15 Graph of Brake horse power-rpm

3.4.2 Development and manufacturing of the camshaft

For the sake of simplicity a three piece camshaft consisting of the cam belt pulley and the two lobes was made. This allowed us to machine the lobes in a three axis CNC mill. The pieces could then be bolted together using three 12.9 M4 allen hex screws, spaced 120 degrees apart. By milling 4mm slots in the cam belt pulley and the intake lobe, the timing could be altered in a quick and stepless way during tuning. The slots were milled so that a 30 degree adjustment was possible, making it possible to correct faults in the simulation software. This design was compatible with the existing valve train mechanism. A complete exploded view of the camshaft is presented in figure 16.

To avoid deformation of the aluminum sprocket when the three pieces are bolted together, washers in steel were added to distribute the force to the sprocket more evenly. The cam is suspended in two Teflon bushings, one in the cam sprocket and the second one in the outer cam lobe. The choice of using Teflon bushings instead of ball-/needle bearings is due to the limited space. The middle lobe is suspended in a bush which rest upon the cam sprocket and the outer lobe. The material for the cam was chosen to be a low alloy tool steel (ss2258). This was available in large amounts in the workshop and easily hardened. The cam lobes were milled in the CNC and then polished on the contact surfaces. The first set of cams was not hardened.

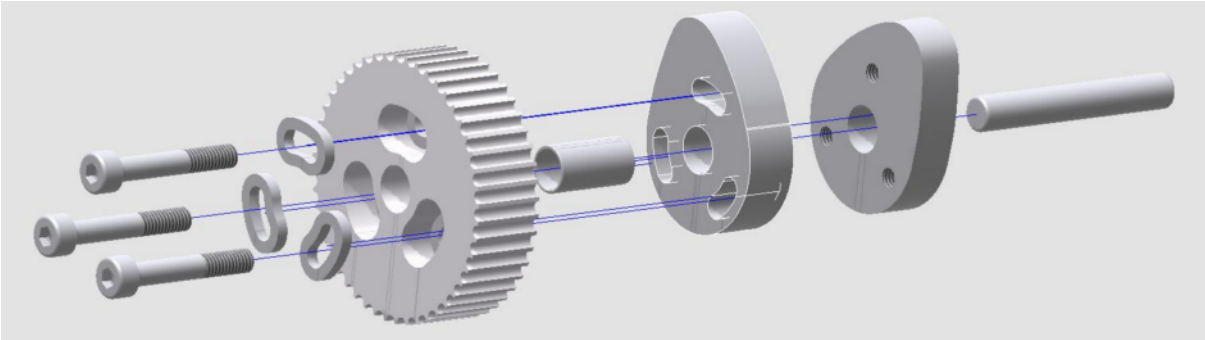

Figure 16 Exploded view of the camshaft.

A dummy camshaft was made to allow measurement of the torque at which the contact surfaces would start to slip, this dummy is presented in figure 17. This was measured in steps with a torque wrench. At 70Nm the force overcame the friction. This torque was sufficient, as the calculated torque in GT-power had a maximum of 5 Nm measured at 6000 rpm.

Figure 17 Dummy camshaft.

3.4.3 Design of the Intake and exhaust system

Both the intake and exhaust system were briefly optimized in GT-power as well as with classical formulas for estimating length and inner diameter of the pipes (A. Graham Bell 2001). The goal of this optimization was to take advantage of the pulses that arise when the valves are opening. By tuning the length and diameter the pulses will reflect at the end of the pipe and travel back to the valve to aid filling or scavenging of the cylinder. The inner diameter and the heat of the exhaust gas determine the speed at which the pulse will travel. The length of the pipe will then determine the time it takes for the pulse to travel back and forth. Therefore the pulses will benefit the filling and scavenging at a specific engine speed since the pulse returns to the engine at a specific phase in the gas exchange. In figure 18 the final result of the exhaust system can be seen. The connection hole on the top is for the lambda sensor.

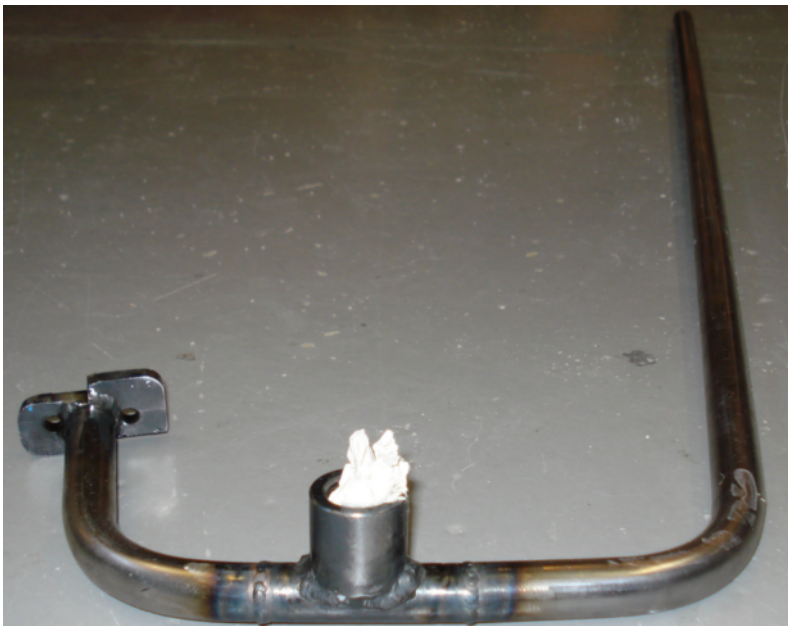

Figure 18 Exhaust pipe with connection for lambda sensor.

During racing, full throttle is used to allow for maximum air flow. However during initial testing and tuning a modified carburetor was fitted to allow for air restriction to protect the engine from over rev. A fuel injector and an air pressure sensor were fitted on the intake. In figure 19 the fuel injector and throttle are mounted and in the vertical pipe the air pressure sensor is meant to be fitted.

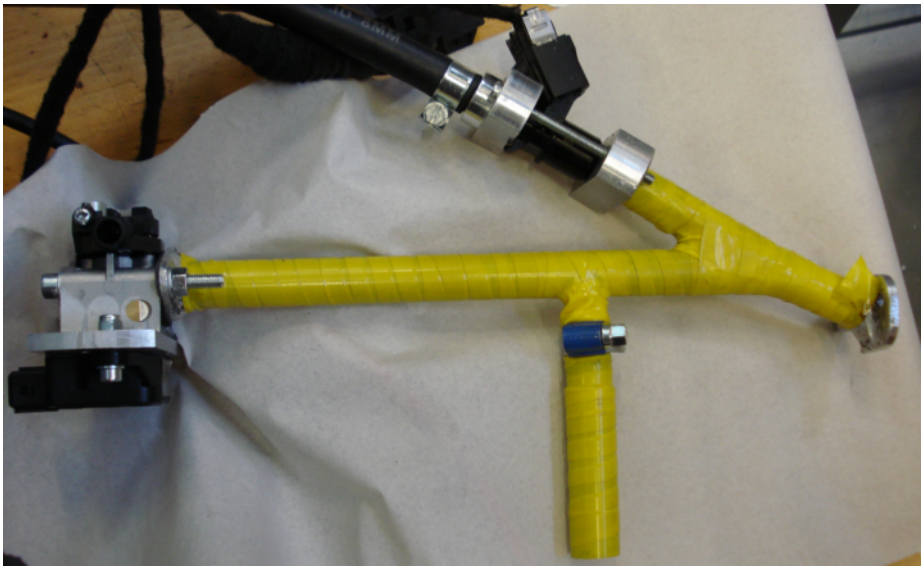

Figure 19 Intake pipe with throttle and fuel injector.

3.4.4 Development and manufacturing of the valve seats and guides

In the cylinder head provided by previous developers the valve was seated on the aluminum which the head was made of. This was not considered a durable and safe method since no known commercially available engine uses this method. To make the engine reliable, another design was developed. Two different designs were considered. The first was to make a new cylinder head out of steel and have the valves seat directly on the steel. This was the main practice before the 1980's and

has worked well. This design is heavier and wears out the valve seats more rapidly than an aluminum head with hard valve seat inserts.

A second cylinder head was made out of the same aluminum as the previous head; however this was made to have hard inserts put into it. The valve guides material needs to be a good heat conductor and wear resistant. Typically, cast iron or in a high power engine bearing bronze is used. The valve seat is almost always made of steel, sometimes of beryllium alloyed bronze in very high power engines such as four stroke motocross competition machines. Typically 80% of the heat of the valve is conducted into the valve seat and 20% into the guide.

For the simplicity of the manufacturing process it was decided to make the valve guide and valve seat in one piece and press it into the aluminum before finishing operations. This demanded the material to be suitable for both valve guides and seats. It was found that beryllium alloyed copper had both superior heat conduction and that it also has a very high hardness. This material was already available in the workshop. It is of great importance to keep the valves cool during the demanding dyno testing. Copper has great heat conduction. Beryllium may be poisonous, but after some research it was found that it is mostly the dust from grinding that is poisonous. However, the manufacturing and disposal of the chips was carefully performed. The final result can be seen in figure 20 and 21 below.

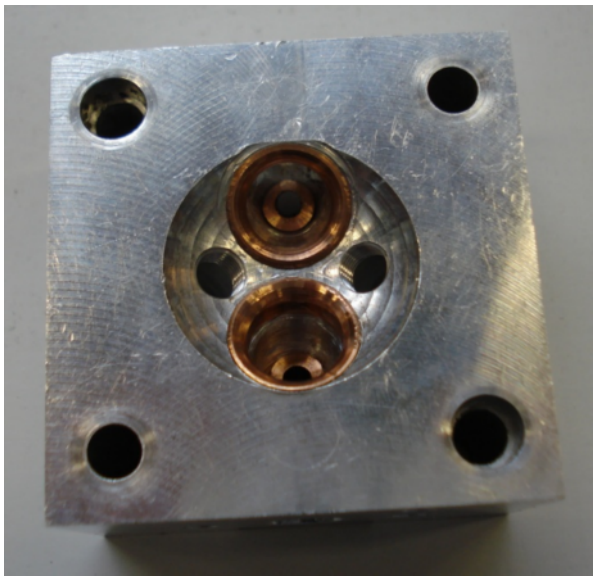

Figure 20 Valve seats.

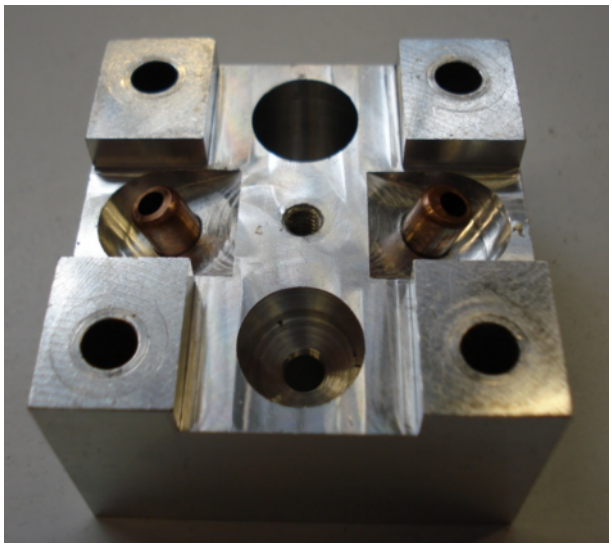

Figure 21 Valve guides.

3.5 *Balancing of the crankshaft*

To achieve low vibrations, and thereby low wear on the engine, it was desirable to balance the crankshaft of the engine. Vibrations in the engine are caused by several reasons but mainly because of unbalanced rotating masses and elastic deformations of components when rotating or under influence of the gas forces created from the combustion. For single cylinder engines, the unbalanced rotating masses are of relatively big importance and are quite hard to completely eliminate due to its unsymmetrical crankshaft with only one connecting rod.

The crankshaft was already designed when project started, but no regards had been taken to the balance of the shaft. The main formula for getting a balanced shaft is:

$$m_{rot}\omega^2 a = m_b \omega^2 R_b$$

Where m_{rot} is the rotating mass (crank pin mass+2/3 of the connecting rod), ω is the angular velocity of the crankshaft, a is the distance from shaft center to center of the crankpin, m_b is the balance weight and R_b is the distance from shaft center to mass center of the balance weight (Ingemar Denbratt, 1999).

The concept of balancing the crankshaft outlines in getting the mass center for the whole crankshaft, together with the rotating mass calculated as described above, in the shaft's center. To achieve this one method is to put a pin with the same mass as the rotating masses m_{rot} in the hole of the crank pin. The size of the balance weight can then be adjusted so that the mass center lies in the shaft's center. One easy way to balance the crankshaft by the method described above is to use a CAD program, for example Catia V5. For the final result from Catia V5 see figure 22.

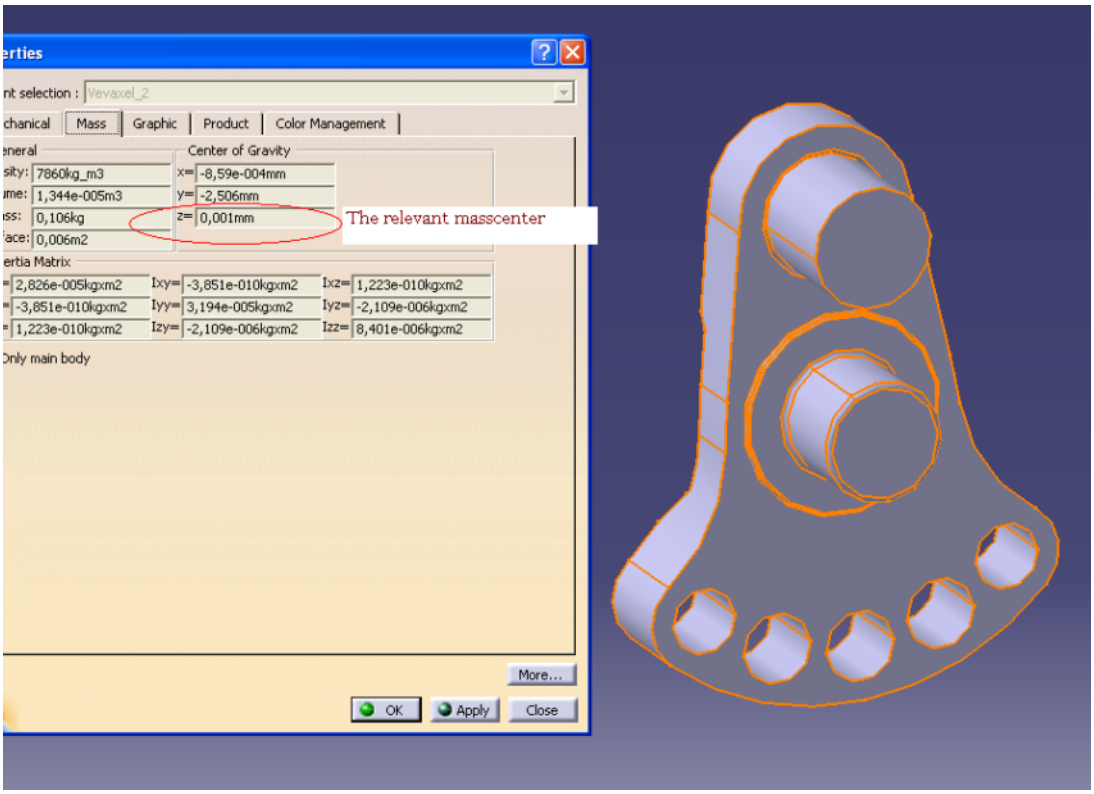

Figure 22 Balancing of the crankshaft in Catia V5

3.6 *Engine Prototype Specifications*

After constructing the different parts needed to complete the engine these were assembled and the first prototype was constructed (see figure 23). The first prototype was mainly built of parts from the old design that were left over. The reason for this was that there was an uncertainty whether the parts would hold together if the ECU was not correctly tuned. The final prototype will be quite different from the first one. For the engine's specifications see table 1.

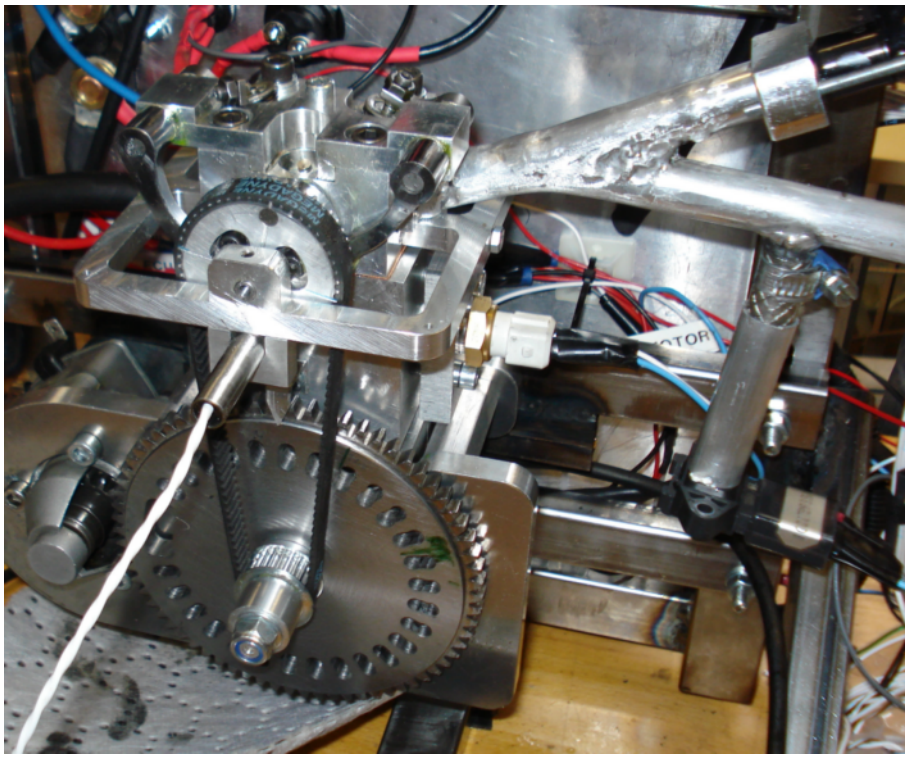

Figure 23 Assembled prototype ready for startup.

Table 1 The engine's specifications.

Parameter	Value	Unit	Parameter	Value	Unit
Displacement	26,2	cc	Sparkplugs angle	20	$degrees$
Cylinder	1	–	Fuel injection	Port injection	–
Stroke	40	mm	Operation cycle	Otto cycle	–
Bore	29	mm	Valve train	Single camshaft	–
Connecting rod length	70	mm	Fuel	Gasoline	–
Compression ratio	11:1	–	Fuel/ignition management system	Civinco	–
Number of valves	2	–	Fuel injector volume flow (at 3 bar)	46,0	cm^3/min
Intake valve diameter	13	mm	Fuel injector	Honda zoomer	–
Exhaust valve diameter	12	mm	Wastespark	Yamaha fazer	–
Port diameter	10	mm	Cam duration	260	$degrees$
Valve angle	20	$degrees$	Valve max lift	3,5	mm
Number of sparkplugs	2	–	Intake & exhaust timing	450 & 255	$degrees$

4 Prototype tests and tuning results

This chapter describes the different tests performed on the prototype in order to evaluate the results of the development work. It also describes the engine startup and tuning process.

4.1 Test of mechanical losses

The purpose of the test was to evaluate the power losses caused by mechanical factors. The main task with the test was to get data for the different cylinder linings and the piston rings with different friction coefficients. When that data for this was known, a decision of which components should be used to minimize the friction losses could be made.

The test was performed with an electric DC engine. The electric engine drove the engine but without the cylinder head mounted, and with different setups, ie. cylinder linings, piston ring. By measuring the current needed to run the engine at a specific speed (specified voltage), a comparison of how much power that was necessary for different setups could be made. The calculated torque losses from the test are mainly caused by the piston ring and the cylinder lining, but the bearings and the oil level also affects the losses. Since the ratios between the contributing losses are unknown, the exact torque losses of the cylinder lining and the piston ring cannot be calculated. Despite this a comparison of the torque losses between the setups can be made.

A similar test had been carried out before on Chalmers. The report from this test was read and the basic setup of the test was copied for the current test (Nebenfuehr, B. et al, 2009).

4.1.1 Electric formulas for calculate power losses

It was desirable to run the engine at different speeds to get an idea of which speed that was most advantageous for eliminating mechanical losses. To get a sense of how to control the speed, following formula was studied:

$$n = (U - RI)k$$

Here *n* is the engine speed in rpm, U is the voltage (in Volts) over the DC engine, *k* is a specific velocity constant for the engine (in rpm/V), R is the terminal resistance (in ohms), and I is the current (in Ampere) through the engine. Provided that enough current is available, it is possible to control the engine speed by setting a suitable voltage, calculated for the specific speed.

When the engine is running, the current is measured and the following formula is used to calculate the torque:

$$T = k_1 I$$

Here k_1 is the engines specific torque constant given in Nm/A.

Table 2 DC engine specifications.

Property	Value
Velocity constant *k* [rpm/V]	110
Terminal resistance R [Ω]	0.173
Torque constant k_1 [Nm/A]	0.087

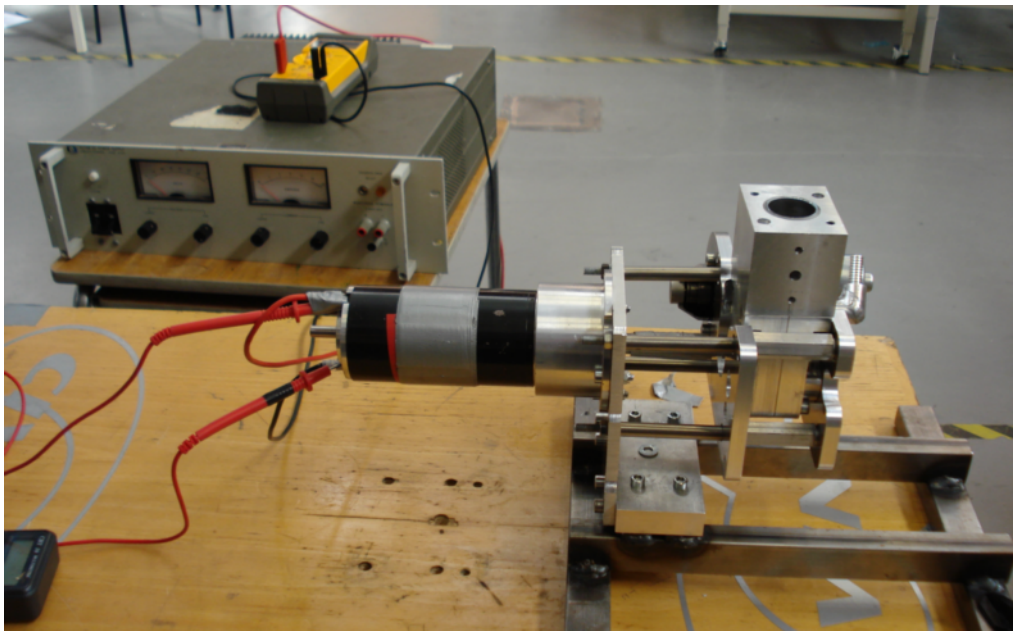

Figure 24 Test setup.

The electric actuators and the measurement equipments required to implement this test were borrowed from the electro division of Chalmers (see figure 24).

4.1.2 Test bench manufacturing

The method for this test was copied quite straight over from the Vera engine report (Nebenfuehr, B. et al, 2009) but the equipment needed to be modified. The electric engine was the same but a new connection between the engine shafts was manufactured. Also a rack for the engine needed to be manufactured. The rack was designed to fit to as many existing components as possible. A stand to mount the engine on a table already existed but the rack and the connection for the shafts was manufactured in Chalmers workshop. The overall test rig can be seen in figure 24 above.

4.1.3 Mechanical losses test results

As figure 25 shows, the best combination seems to be cylinder linings treated by ANS combined with untreated piston ring. However the result of the three last attempts, which at least had one surface treated component, is quite the same. The two setups containing cylinder linings treated by ANS were the two attempts with lowest torque losses which is a good grade this treatment.

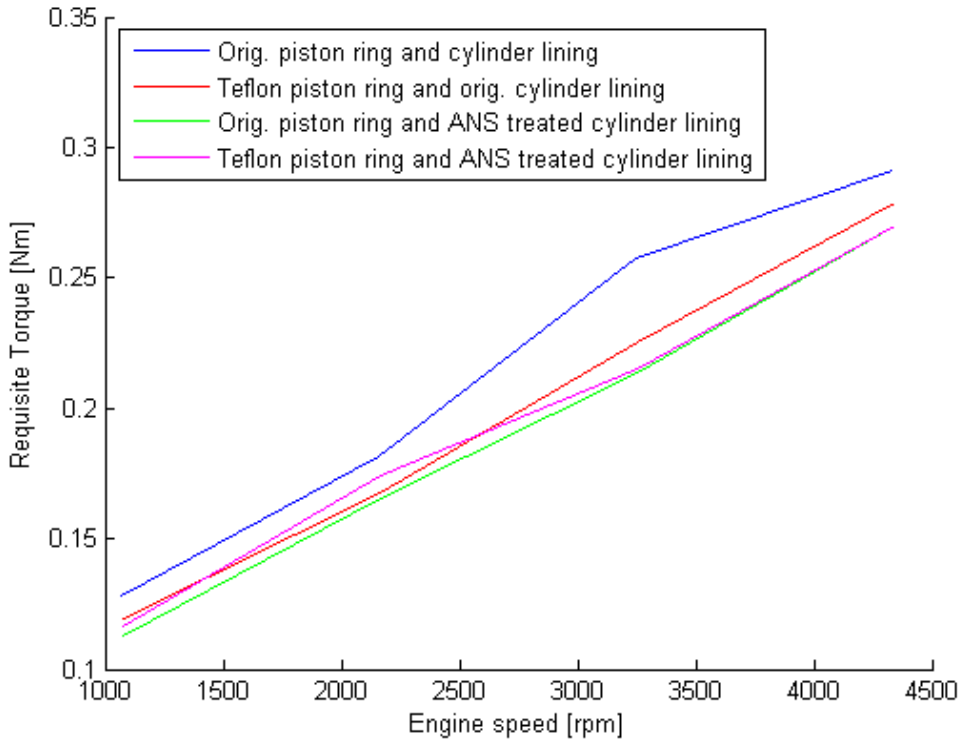

Figure 25 Test result of the different setups

4.2 Startup and engine tuning

To optimize the engine a dyno would be used to adjust the engine settings depending on different loads. These settings are controlled by the ECU. Before entering the dyno a basic setting for the engine was needed to make it easier to tune the engine. To get this basic setting a small rig was built and then mounted on a table in order to be able to run the engine without any loads (see figure 26). The ECU was connected to a computer so that all of the data from the start-up attempts could be logged.

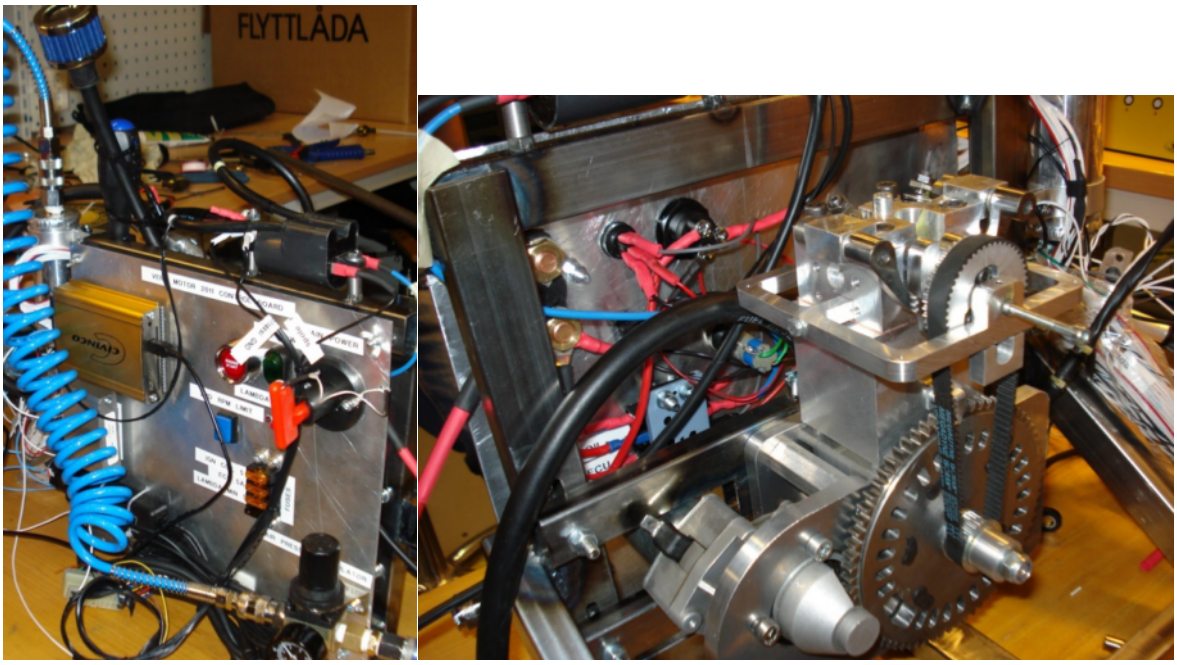

Figure 26 Engine rig.

The first start-up attempt was successful and the engine started several times after this. The problem was that the engine refused to run for a longer time and sometimes it would not even start. There was also a problem with the ECU not working properly. A few times the ECU did not give the system any ignition signal. This meant that several hours were wasted searching for other faults within the system before realizing that the ECU was the problem. At these occasions it was often enough to restart the computer and let the ECU rest for a couple of minutes.

Another reason for testing the engine without any loads before entering the dyno was to see if some parts would break. The first part to break was the rocker arm for the exhaust valve (see figure 27). The reason for this breakdown was believed to be that the rocker arm was fastened with too much torque. This created a crack in the rocker arm that continued to grow until failure. This theory was supported by the fact that the new rocker arm didn't break.

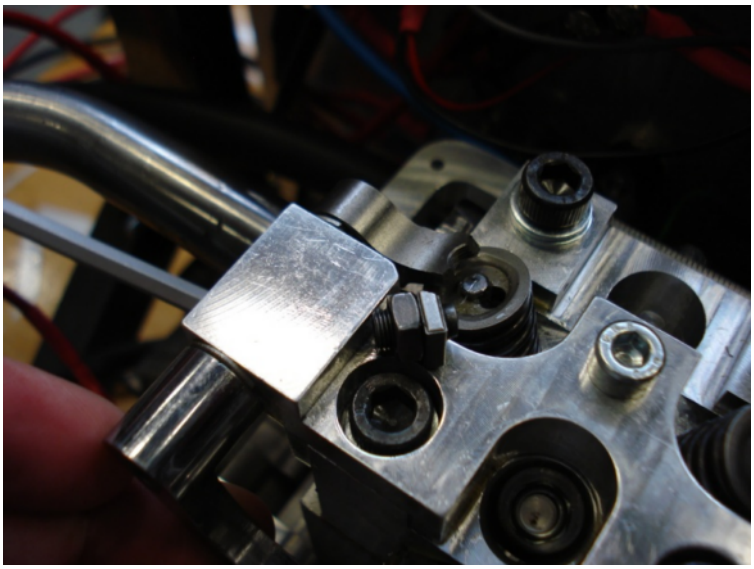

Figure 27 Broken rocker arm.

The gear wheel for the starter engine was the second part to fail (see picture 28). This happened after two hours of trying to get the engine to start. This meant that the starter engine had been used a lot during these hours. The gear wheel was taken from another application and had a few holes drilled into it. It was at one of these holes that it broke since a stress concentration was present. As a temporary fix the gears was welded back onto the flywheel. However the welding affected the hardening of the gears and a wear occurred rapidly. After this breakdown a new gear wheel without any holes was acquired in order to prevent this from happening again.

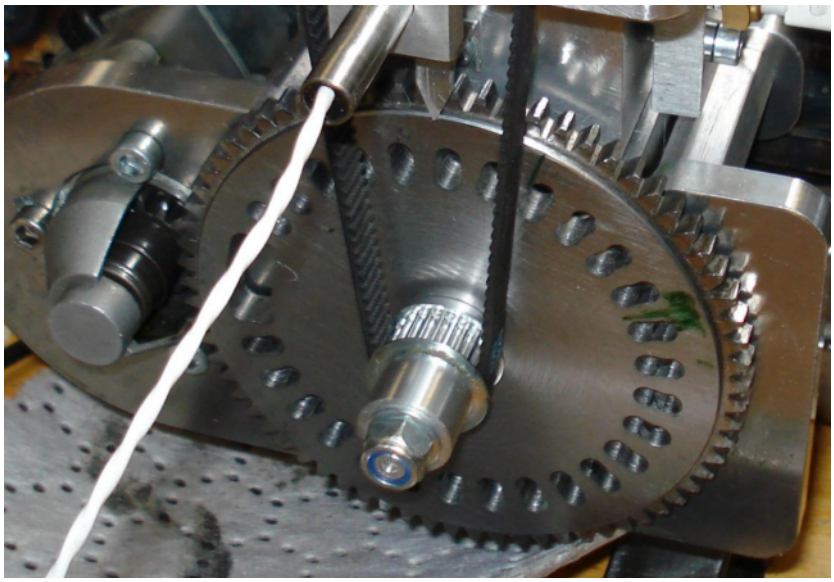

Figure 28 Gear wheel for starter engine.

5 Discussion and Conclusions

The goal of this project was to further develop and complete the engine, earlier constructed by Anders Johansson and Andreas Thulin. As described in this report, many areas have been improved in regards to minimize fuel consumption, but when the project started the engine was not complete and it is therefore not possible to compare any data for fuel consumption. Since it was hard to specify exactly how far the development work would come and that the areas that were decided to develop has been developed, it is justified to say that the first goal, to further develop the engine, is achieved.

The engine has been started and the first startups went quite well. Since the engine can be started, also the second part of the goal, to complete the engine, can be regarded as completed.

The expected result from the mechanical losses test was that ANS treated cylinder lining combined with Teflon treated piston ring would get the lowest torque losses. An explanation to the somewhat unexpected result can be that the Teflon layer was partly torn off after the first attempt together with original cylinder linings. It seems that the Teflon treatment from Ahlins AB, that never been tested for this application before, does not withstand the high wear. Maybe the varying level of oil, due to a leak, also affected the result.

Even though the two goals are reached, the engine still has several areas that can be improved. Since the engine was started for the first time later in the project than planned, less time for tests and tuning was available. These tests and tunings are very important for getting an engine that can be driven reliable and fuel efficient. The final tunings for the management system was planned to be done in a dyno, where we could put different loads on the engine and compare the engines power to the fuel consumption. This tunings could unfortunately not be done, since the project time ran out before we got a sufficiently reliable engine. Also mechanical reliability was planned to be evaluated in the dyno. Some mechanical problems occurred already when we ran the engine without any load, but for sure, more problems would have been found if the engine had been tested in the dyno.

The mechanical problem that occurred when running the engine for the first time was that some oil leaked from the crank house to the combustion chamber. Maybe an increase in number of piston rings would adjust this problem. With oil leaking to the combustion chamber, the tuning work was hampered. Since oil is flammable, the fuel rate became different than the management system thought because the engine combusted oil. Also the leak of oil lead to an uncertainty of the engines oil level which meant that the engine could not run for as long time as wanted.

However it was not time enough for us to work with these problems. For getting a competitive engine for the Shell eco marathon competition, more work has to be done. When we started the project, we divided the group into subgroups that would work with different components on the engine. Maybe more work for getting a complete engine should have been done earlier. The fuel and ignition management system is very important to get working well and without any serviceable settings on that, the BSFC will never be low enough for the competition. Maybe too much work for eliminating friction losses and weight reduction was done before the management system was paid any attention. Also much unscheduled work that took a lot of time had to be done for be able to start the engine, such as an engine stand, exhaust pipe, and intake pipe and so on. All these parameters made that the time for tests and tunings were much less than we had planned.

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