CHALMERS



Optimizing Fuel Efficient Engine for ECO-marathon

Development of an energy efficient internal combustion engine, based on previously conducted bachelor thesis.

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Eco Marathon Project

Optimizing Fuel Efficient Engine for ECO-marathon
Project in applied mechanics
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Abstract

This report presents methods on how to increase the efficiency of a small internal combustion engine. The engine developed throughout this project is used in Chalmers Eco Marathon vehicle, Vera. The main errors of the previous year's engine are; excessive oil consumption, compression leakage, engine control signal disturbance, relative twisting of the crankshaft halves and flywheel oscillations. New implementations to the engine are; cooling- and heating system, ease the connection between the engine and the electric system, variable compression ratio and electromagnetic clutch.

The oil consumption problem is solved by investigating the possibility to use new piston rings and by increasing the crankhouse ventilation. However, new piston rings can not be manufactured, so the result is not yet investigated. The ventilation did not result in any significant change in oil consumption. Piston rings should be outsourced due to their complexity.
Compression leakage is solved by increasing the height of the cylinder lining, creating a higher pressure between the cylinder head and liner. Compression leakage around the valve seal has to be evaluated. Engine control signal disturbance originated from electromagnetic fields around the ignition cables. This is successfully solved by using cables with inductive resistors.
The crankshaft halves are remade, out of aluminum, with higher precision and higher degree of interference fit. However, during high engine load, these slipped. The conclusion is to make them out of steel and evaluate the result. The flywheel oscillation is substantially reduced by increasing the size of the crankshaft's support shoulder from 1.4 to 3.75 mm. The cooling system is implemented which reduced the cylinder's temperature from approximately 75 to 35 degrees Celsius. The heating system is never evaluated. No changes are made regarding the engine's electrical system.
To vary the compression ratio, a cambelt tensioner must be implemented. The ratio is therefore fixed to 10.76, due to the length of the available cambelt. The electromagnetic clutch is not implemented due to the high level of complexity.

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1 Introduction

This section aims to provide a deeper understanding of the purpose of this project. It presents the background, i.e. the reason as well as focus and boundaries for this project.

1.1 *Background*

An important discussion in the society today is the depletion of fossil fuel supplies. The depletion results in an increase in fossil fuel price and has a negative effect on global warming which is an ever-increasing topic of discussion.
Every year, Shell arranges a competition called “Shell Eco Marathon”. The purpose of the competition is to increase the awareness of vehicle fuel consumption and its effect on fossil fuel depletion. The competition also encourages engineering ingenuity in order to be on top of the podium. Winning the competition is solely based on driving as far as possible using one liter of gasoline or its equivalent energy in different form, i.e. electricity, hydrogen etc. Chalmers enters the competition using two different vehicles, namely Smarter and Vera. The base of the engine was developed during a project course during the year 2010.
This engine was further developed during spring 2011 as a bachelor thesis. The resulting engine proved good results but is in need of improvement.

1.2 *Scope*

The aim of this project is to further develop the existing engine for Vera, focusing on developing and manufacturing a more reliable and fuel efficient engine. The engine will be tested and optimized in order to verify its performance regarding reliability and fuel economy.

1.3 *Boundaries*

The project only considers improvements in the engine and the drive train, hence omits any research regarding improvements to the current chassis and body. Components which are too complex to manufacture in-house will be purchased from other companies. In order to keep a low budget, the project will strive to produce as much as possible in-house.

2 Methodology

This section aims to provide a deeper understanding of the different steps and practical methods used throughout this project. The section starts with an investigation of the previous engine in order to identify points of improvements. The method part is then divided into nine subsections, one for each point of improvement.

2.1 *Previous Engine*

The focus of this project is improving an already existing engine for the competition vehicle Vera. The old engine is module based and made up of different subsystems, which means changes in one system does not impact another. This eases the work of further developments. The old engine, which is going to be the foundation of this project, has the specifications presented in *table 1*.

Table 1, engine specifications from spring 2011

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	$degrees$

2.2 *Literature Study*

Based on literature study of the written report about the previous engine, different subsystems that are in need of improvement are brought to light in this section.

2.2.1 Problems regarding the previous engine

One of the main known problems is that the engine consumes a considerable amount of the lubricating oil. This problem was identified while comparing the amount of oil before start of engine and then after a while of running. The problem is identified to originate from leakage between the piston rings and the cylinder lining due to the oil film that is created on top of the piston head. Oil leakage has several associated problem:

- A major problem can arise if the engine consumes all of the oil, which would lead to poor lubrication of the bearings within the crank house as well as the cylinder lining. This will result in increased wear of engine components and lowered engine efficiency due to the increased friction.

- The second problem is the increased risk of engine knock due to lubrication oil within the combustion chamber. The lubrication oil contains long hydrocarbon chains, i.e. has a low octane number. When exposed to a combination of high pressure and temperature it is highly flammable (1). During the compression stroke the intake gas mixes with the lubrication oil and the pressure and temperature increases. This may result in both pre- and post-ignition knock.

The second issue is leakage during compression stroke, resulting in poor compression ratio. The origin of this problem is not yet identified. However, some possible areas of interest can be identified. These are the sealing between the cylinder head and the valves and/or between cylinder lining and piston rings. The compression leakage has the negative effect of lowering the overall efficiency due the lowered compression ratio and peak pressure of the engine (2). It might also lead to unpredictable behavior of the engine.

During bench-testing with the current Civinco control unit, signal disturbance can be identified, while running the engine, using a signal logging feature built in Civinco. The signal that showed disturbances came from the crankshaft trigger pulse generator. The disturbance entailed many other problems such as:

- Difficulties to control the ignition and fuel injection timing, due to false trigger pulses from the crankshaft pulse generator.
- Misleading output results.
- Cycle to cycle variations.

When the engine is subjected to higher loads, for example during competition, the two crankshaft halves tend to rotate relative to each other. This resulted in engine failure due to the increased friction force within the crankshaft bearings. One of the main contributing factors to the rotation motion is due to poor interference fit between the two halves and the wrist pin.

When inspecting the flywheel during engine operation, it is obvious that there is an oscillating axial motion. This is a result of poor fixation of the part to the crankshaft. The oscillating motion can create large vibrations and also create a problem while engaging the starter motor. Another problem is the change in crankshaft pulse voltage due to the variation in distance to the flywheel.

2.2.2 New implementations

Earlier testing of the engine has proven that a cooling system is of great importance due to the risk of overheating the engine while running bench tests. In order to conduct successful bench tests, the engine's temperature has to be kept on a steady level. This needs to be controlled by some sort of cooling system.

During competition it is of great importance that the engine starts the competition with an appropriate operating temperature in order to perform and have the same characteristics as during the bench-testing.

In order to ease the work of connect and disconnect the engine from the chassis interface, improvements on the existing electric system needs to be made. The connection and disconnection of the engine is important in order to move the engine between the chassis interface and the optimization bench interface. If the engine easily can be disconnected from the frame structure, it also provides higher degree of accessibility when it comes to disassembly of the engine.

An important factor when it comes to the engine efficiency is the compression ratio. The compression ratio is dependent on the cylinder bore, stroke and clearance volume (2). The bore and stroke is at this moment already known. However, the clearance volume is not yet fully known. It is also interesting to investigate the engine behavior and efficiency in relation to the compression ratio. It is therefore of great importance to integrate a solution which provides a possibility to change the compression ratio of the engine without too much trouble.

There have also been suggestions that the efficiency of the current drivetrain can be improved. The current drivetrain basically consists of a centrifugal clutch, chain and chain sprockets mounted at the engine and at the rear driving wheel. The focus is mainly on the clutch system, where there is a need for higher level of controllability and increased efficiency.

2.3 *Requirements*

In order to proceed with the development of the engine a requirement specification has to be established. The purpose of the requirement specification is to give an overview of the functions and their performance on the engine and also how they are going to be validated. The functions or solution implementations is also graded, with either a demand or a wish, which is prioritized from 1-5. This highlights the magnitude of the different functions. *Table 2* displays the requirement specification with a short description and validation method.

Table 2, Displaying the requirement specification.

Description	Wish/ Demand	Priority (1-5)	Verification method
Reduce engines consumption of lubrication oil by 50% during operation.	D	-	Insert a certain amount of lubrication oil into the engine. Run the engine for 30min and measure the remaining lubrication oil. Compare the inserted to the extracted amount of oil.
Reduce compression leakage by 50%.	D	-	Test the compression before any modifications. Implement solution and test the compression again. Compare the result before and after implementing solution.
Reduce crankshaft trigger signal disturbances during operation to a point where the Civinco system don't get affected.	D	-	Measure the crank signal using an oscilloscope before modification. Implement the modification and measure the change in peak voltage on the disturbance.
Interference fit between crankshaft and wristpin should manage maximum available load from the engine. The angle between the two crankshaft halves shall not deviate more than +-2 degrees.	D	-	Run the engine with the modifications for 10 minutes with the maximum load which the engine i capable of. Disassemble the engine and measure the difference in angle between the two crankshaft halves.
Reduce the time needed to connect and disconnect the electricity system by 50%.	W	2	Measure the time needed to perform a connection between the engine and the chassis interface. Compare the time needed after implementing solution.
Implement a system which provides an adjustable compression ratio from 10-14.	W	4	See if the compression ratio is changed.
Increase the efficiency of the drivetrain by 25%.	W	3	Measure the force needed to rotate the drivetrain without rear wheel and engine. Implement the new solution and measure the force again.
Implement a combined cooling and heating system which are able to keep a steady temperature during constant bench-testing for 30min at 50% of max load. The engine should be capable to heat the engine to 70 degrees at 1h.	W	5	The cooling performance should be measured by comparing the difference in temperature on the cylinder after 30min of operation and compare the result with and without cooling package. The heating should be validated by comparing the cylinder temperature with and without the system while the engine is not operating.

2.4 *Points of improvement*

This section describes the nine points of improvement. It aims to help the reader get a deeper understanding of the problems as well as the corresponding countermeasure.

2.4.1 Oil Consumption

Like stated in previous section, the oil consumption originates from leakage between the piston rings and the cylinder lining. The reference engine, which is the base of this report, was operated using both one and two compression rings with rectangular cross sections.

When investigating commercial solutions, which has proven to be durable and provide low leakage, it is identified that they use another piston ring setup. The most common setup for piston rings in four-stroke IC engines is to use three types of piston rings, namely;

- 1st groove ring (closest to piston crown); main compression ring.
- 2nd groove ring; secondary compression ring/oil scraper ring.
- 3rd groove ring; oil ring.

The 1st and main compression ring has the purpose of sealing the gap between the cylinder linings and the piston in order to provide compression during compression stroke. The secondary purpose is to transfer the heat from the piston to the cylinder lining. The contact pressure between the piston ring and the cylinder lining needs to be sufficient in order to create the gas seal. This is maintained during compression by allowing a small amount to pass into the void between the piston and the piston ring. This pressure will force the piston ring out to the cylinder lining. The main compression ring usually has a rectangular cross section.

The 2nd piston ring, namely scraper ring, has the main purpose of controlling the oil film of the cylinder lining. This is controlled by scraping of the excessive oil during expansion and intake stroke. The scraping effect is achieved by utilizing a piston ring with a trapezoidal cross section. By reducing the excessive oil, a lower amount of oil reaches the combustion chamber, hence lowering the oil consumption. The secondary purpose is to act as a gas seal and aid the compression.

The 3rd piston ring, namely oil ring, acts as an oil control ring. When the excess oil is scraped off by the 2nd piston ring, the oil ring cooperates and provides a path for the oil to the crank house. This is achieved by letting the oil pass through small holes is the piston located behind the oil ring. The oil ring also has the purpose of distributing the oil evenly on the cylinder lining during compression and exhaust stroke. The oil ring is usually composed of three parts combined together in order to make it possible for the oil to pass through (3), (4).

The commercial solution has proven to provide low oil- and compression leakage and should therefore be implemented.

When investigating the oil consumption, another interesting fact comes into play during work- and intake stroke. During these two strokes, the crankhouse gases are compressed and needs to be ventilated in order to control the pressure. If the pressure in the crankhouse reaches above that of the combustion chamber, the excess oil will not flow down into the crankhouse. Hence, the oil will be retained within the combustion chamber. In order to lower the pressure within the crankhouse there needs to be sufficient ventilation for the gases. The current engine has existing crankhouse ventilation. In order to minimize the risk of high crankhouse gas pressure the ventilation shall be enlarged.

2.4.2 Cylinder Lining

Thorough investigations of the cylinder lining from the previous year's engine revealed that it is conical. Due to the conical shape, there is an increased risk of compression leakage and even failure due to increased contact pressure and friction between the piston and the cylinder lining. The lining is therefore remade.

Cylinder lining is used to decrease friction between the piston rings and the cylinder. A good cylinder lining is supposed to; reduce wear on piston rings, reduce wear on the cylinder lining itself and reduce the consumption of lubricant. It should be made of a material with high anti-galling properties (5). Using the cylinder's aluminum as lining is not an option due to the aluminum's poor resistance against deformation during operation, thereby increasing the risk of engine failure (5).

Cast iron has good properties for a cylinder lining because of its ability to bind oil, creating an oil film which is available even directly after startup. Another reason to use cast iron is because of its graphite phase. The graphite acts as a dry lubricator, meaning the piston will run smooth, to some extent, even if there is a problem with the lubrication oil (6).

In this year's project it is decided that the cylinder lining should be made out of cast iron. In order to ensure a sufficient gas seal between the cylinder and the cylinder head, the lining will be designed to extend above the cylinder. This will result in a small contact surface between the cylinder head and lining, thereby increasing the contact pressure.

Finally, in order to create a smooth surface with low friction the cylinder lining will be honed. The honing process demands very high degree of precision, something which is impossible with the in-house machines, it will therefore be outsourced to a specialist.

Signal disturbances

The source of the disturbances, from the crankshaft pulse signal, is not yet fully identified. The first step to identify the source is to measure the signal with an oscilloscope. By plotting the signal in the oscilloscope, the amplitude and the frequency of the disturbance can be visualized.

The crankshaft trigger pulse signal is fetched while running the engine. The resulting signal is significantly affected by disturbances. The amplitude of the disturbance is approximately 15% of the crankshaft trigger pulse signal. The frequency of the disturbance is correlated to that of the trigger signal. Another conclusion from the signal analysis is that the timing of the disturbance coincide with the spark ignition. When the signal is analyzed without the electrical system turned on there are no disturbances.

The conclusion can be drawn that the source for the disturbance originates from the ignition. This is a likely source due to the fact that it generates an electromagnetic field around the ignition wire which can induce a current into a nearby wire. When current passes through a wire, a magnetic field is formed. The spark ignition produces a high peak voltage which passes through the engine which might interfere with other sensors (7).

In order to investigate if the electromagnetic field is the cause of the disturbance, the coil together with the cable leading to the spark plug is shielded using thin strips of aluminum foil. The shield is grounded in order to remove the induced current in the shield. The signal is analyzed while the engine is running and compared to the previous test. The result of shielding the coil and the wire has a small but not a significant effect on the amplitude of the disturbance.

In order to further reduce the magnetic field an ignition cable with inductive resistor is implemented. These wires are based on a non-conductive core such as Kevlar. Wound around this core there is a fine alloy wire in a helix. The helix wire will aid to suppress the generated magnetic field (8).

2.4.3 Interference fit between crankshaft and wrist pin

Previous section states that the two halves of the crankshaft twisted, in relation to each other, during high engine load. This is a result of poor tolerances for the interference fit between the crankshaft and the wrist pin. In order to solve this problem, a proper interference fit needs to be calculated based on material properties and wrist pin diameter. The change in interference fit also implies that the crank shaft halves need to be re-manufactured.

The wrist pin that is going to be used is manufactured from a steel alloy with high young's modulus. The crankshaft halves are going to be manufactured from the Uddeholm aluminum alloy called Alumec (9). The choice of using Alumec is due to the fact that it eases the manufacturing process in terms of machining (10). Based on an interference fit chart, together with the wrist pin diameter, a minimum and maximum interference fit can be obtained. The minimum fit is 0.008 mm and the maximum 0,040 mm. Due to the relatively low young's modulus of Alumec (71,5 GPa) (9) it is desired to have a relatively high interference fit in order to maintain a high contact pressure between the parts. The chosen interference fit is 0,035mm.

2.4.4 Flywheel oscillations

In order to minimize the risk of flywheel oscillations, countermeasures have to be carried out. The flywheel is manufactured from one piece of steel and lathed in one single operation without refastening. The resulting surfaces should therefore be in close parallel to each other. The parallelism is measured using a micrometer. The conclusion is that the oscillations are not due to the design of the flywheel.

Another possible source of oscillations would be from radial oscillations of the crankshaft. The rotational tolerance of the crankshaft is measured using a precision dial indicator while rotating the crankshaft positioned in the crankhouse. The point of measurement is at the flywheel's position on the crankshaft. The measuring results in variations in the scale of 0.01mm. The conclusion is that this does not have a significant contribution to the flywheel oscillations.

The only possible source for the problem would be the height of the crankshaft shoulder, which is supposed to provide stability for the flywheel. The only way to test this theory is to implement a shoulder which is larger. The implementation implies a new design of the crankshaft and also a redesign of the crankhouse due to the need of increased bearing dimensions. The resting shoulder for the flywheel is 1.4 mm in radial measurements. The new design of the shoulder is controlled by available bearing sizes and also the housing of the bearing in the crankhouse. In order to minimize the risk of oscillations, the decision is made that the shoulder should be controlled by the maximum bearing size that fit within the crankhouse. The bearing chosen for this application is 61804_2RZ from SKF. Using this bearing as a reference the maximum shoulder height is obtained. The resulting radial height is 3.75 mm which will provide stability for the flywheel.

2.4.5 Cooling and Heating System

In order to run successful benchmark tests, the engine needs to be able to run for a long period of time. Running the engine excessively results in high heat production, due to the combustion process itself and also because of friction between engine parts. During combustion in the engine, the peak burn temperature can reach up to about 2200 degrees Celsius (2 s. 668). The cylinder head of the engine is made out of a special aluminum alloy which has a melting temperature of about 660 degrees Celsius (11). Creating a new cylinder head is not feasible due to the complexity of it in combination with sufficient performance at this stage. The surface of the combustion-side cylinder wall must also be kept below about 180 degrees Celsius in order to not disrupt the lubricating oil film (2 s. 668). The conclusion is that the heat needs to be transported away from the system in order to protect the components from failure during bench tests.

During the upcoming race, to run as fuel efficient as possible, the engine will be operated in short intervals and at high loads. For the engine to run smooth, i.e. taking advantage of the lubricating oil, it needs to be warm. During combustion, heat is transferred from the hot combustion gases to the cylinder walls. This will reduce the average gas temperature and cylinder pressure, thereby resulting in lower efficiency of the engine. In order to increase efficiency of the engine, the heat transfer from the combustion process to the cylinder walls must be kept as low as possible (2 s. 668).The engine will need some sort of heating system and also some sort of isolation in order to keep the engine warm during the race.

Engines used in commercial vehicles today use a mixture of water and glycol as heat transportation medium. There are two types of glycol used; ethylene- or propylene glycol. Ethylene glycol has better thermal properties (higher boiling point, lower freezing point, thermal conductivity etc.) than propylene glycol. Ethylene is more commonly used in automotive antifreeze. Water is one of the best liquid coolants due to its high heat capacity and thermal conductivity. Glycol is typically added to increase corrosion resistance and to avoid freezing during sub-zero degree weather, it does however decrease the cooling efficiency to some extent (12). Due to the previous mentioned points, water will be used, both for heating and cooling the engine, because of its good conductivity and availability.

In order to provide good heat conduction between the engine and the heat transportation system, as large surface as possible should be used. The solution for this project is to cover the walls of the cylinder with water “pipes”. The “pipes” will be created by milling 4 mm deep half-sphere splines outside three of the cylinder walls and connect them by drilling angled holes. Then, three plates with corresponding splines will be manufactured. The plates will be mounted on the cylinder and will thereby provide a piping system that will transport the water, shown in figure 1.

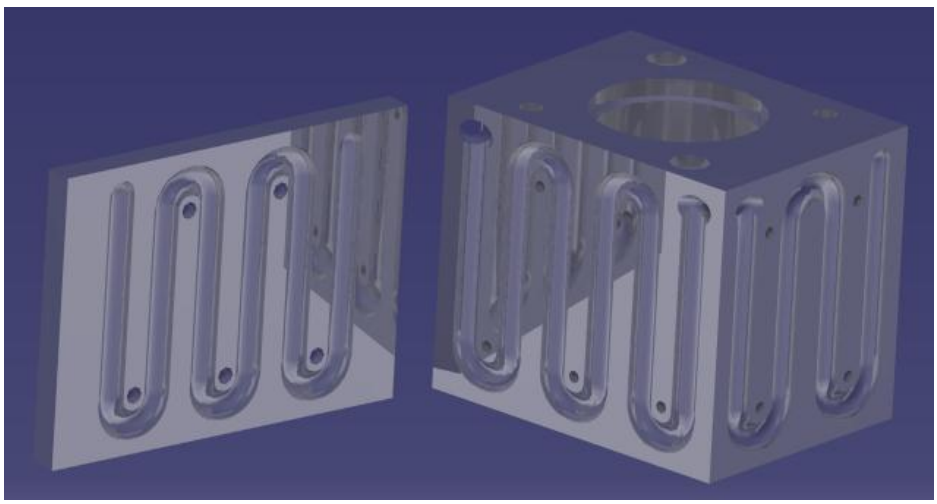


Figure 1, Displaying the CAD model of cylinder with cooling pipes

In order to cool the water it is possible to use a heat exchanger. However, due to the complex piping used in a heat exchanger, in combination with limited amount of time. A large water vessel will be used. The vessel will work both as a water source, i.e. a closed-loop cooling system, and as a heat exchanger. Due to the low volume of water inside the engines piping system compared to the volume of water inside the vessel, the water will have enough time to cool inside the large volume. The vessel will be made out of aluminum which is beneficial due to its good heat conductivity.

Regarding heating the engine prior to race, using the aluminum vessel and mounting a heating rod provides one simple solution for two problems. When the engine needs to be heated, the heating rod will be activated which will then warm the water. If the engine needs to be cooled, the heating rod will be deactivated.

In order to get the water flowing, the system can either be run by temperature difference, i.e. water would circulate from the warm area to the cooler area. This is however not feasible due to the slow motion and the low controllability. Instead, the flow will be controlled by an external pump. The pump will be driven externally by electricity, making the whole heat exchange system detachable from the engine.

To connect everything, i.e. creating a path for the water flow, special couplings with matching tubes will be ordered.

2.4.6 Electric system interface

The current electric system is tangled up and takes long time to connect and disconnect. It is difficult to distinguish the different connections and the connectors are not standardized. In order to ease the connection of the engine to the electronic control unit, the cords must be tangle free with distinct connections. The electric system must also be able to withstand harsh conditions to some extent, i.e. splashing water and vibrations.

One promising solution is to use a socket called HAN A from Harting. It supports 10-32 cable pairs, all packaged in one connector, and is IP65 classified (13).

2.4.7 Compression Ratio

Compression ratio has a great influence on an engine's performance. The aim of this part is to find a way to easily alter the compression ratio, to specific known values, and evaluate their performance in this particular engine. The highest performing compression ratio will then be used during the race. The focus will be put solely on geometrical compression. In order to calculate the compression ratio, eq.(1) is used.

$$r_c = \frac{\frac{b^2}{4} \cdot s \cdot \pi + V_c}{V_c}; \text{ Where} \begin{cases} \text{s} = \text{stroke} \\ \text{b} = \text{bore} \\ V_c = \text{clearance volume} \end{cases} \qquad eq.(1)$$

The stroke and bore of the engine is known to be 40 mm and 29 mm respectively. However, the clearance volume needs to be measured. In order to measure the volume of the cylinder head, which has a complex design, the following tools are used;

- High precision laboratory scale.
- See-through plastic plate.
- Sealant.
- Pipette.
- Graduated cylinder.
- Holding arm.
- Ethanol 95% T-röd OKQ8.

The idea is to measure the amount of alcohol that can fit inside the cylinder head and thereby depict the clearance volume. First off, the accuracy of the scale has to be established. The accuracy is evaluated by pouring a fixed amount of alcohol into the graduated cylinder. The cylinder is then weighted using the scale and the resulting density is calculated by dividing the weight with the volume. The resulting density is 0.790 g/cm3. This is verified against the depicted density on the bottle, which are 0.789g/cm3. The low error in density foretells the high accuracy of the scale.

The test to establish the clearance volume is conducted two times. This in order to avoid errors due to the low mass of the alcohol inside the cylinder head, i.e. one spilled drop would have great impact on the outcome. Test number 1 results in 1.620 g of alcohol which yields a volume of 2.053cm3 when using the given density. When the calculated density is used, the resulting volume is 2.051cm3. Test number 2 results in 1.610 g of alcohol which corresponds to a volume of 2.041cm3 when using the given density. The calculated density yields a volume of 2.038cm3. The resulting clearance volume, which is used for calculating compression ratio, is derived as the mean value of the two tests. The final clearance volume is calculated to 2.046cm3. Figure 2 displays the test setup.

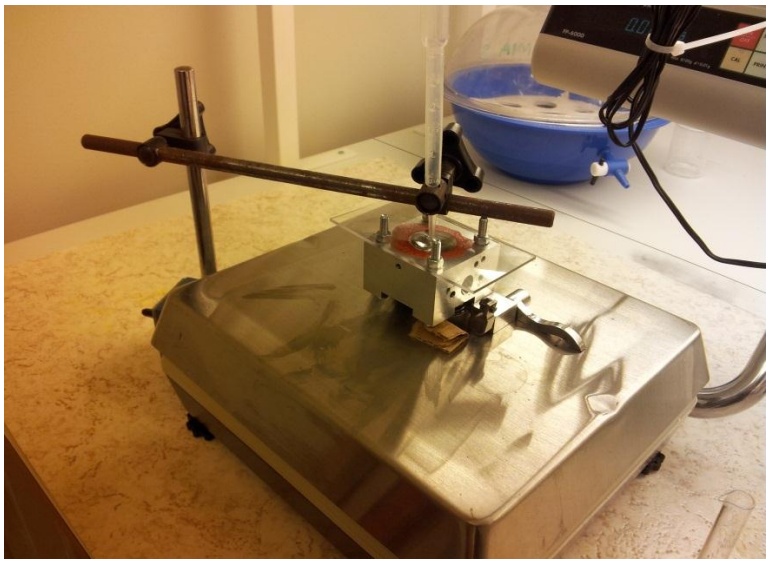


Figure 2, Displaying the test setup for measuring clearance volume.

When the clearance volume is known, in combination with known values of bore and stroke, the following equation can be used to determine the impact the height of the extension plates has on compression ratio.

$$r_c = \frac{\frac{b^2}{4} \cdot s \cdot \pi + V_c + \frac{\pi}{4} \cdot b^2 \cdot h_{extension}}{V_c + \frac{\pi}{4} \cdot b^2 \cdot h_{extension}}; \quad \text{Where} \begin{cases} \text{s} = \text{stroke} \\ \text{b} = \text{bore} \\ V_c = \text{clearance volume} \\ h_{extension} = extension\ plate\ height \end{cases} \qquad eq.(2)$$

Matlab is used to produce a diagram which depicts the height of the extension plate as a function of compression ratio.

2.4.8 Clutch and drivetrain

The current engine utilize a centrifugal clutch, this clutch is based on the centripetal acceleration, i.e. the engine speed. The current problem is the ability to control the clutch engagement and the amount of slip allowed within the clutch. The amount of slip is in direct relation to the drivetrain efficiency. It is of necessity that the clutch is controlled automatically due to the high amount of starts and stops of the engine during competition. The current engagement timing, in relation to engine speed, is controlled by varying the stiffness of three springs. In order to meet a satisfying ability to control the clutch, it is of high interest that it can be controlled by electric signals. This will provide the possibility to simulate the clutch engagement. A possible future clutch also needs to meet the necessary torque demand. In order to ease the installation of the engine to the chassis and drivetrain it is desirable that the clutch is positioned on the rear axle.

The current engine produces approximately 1.75 Nm in maximum torque and the current drivetrain is composed of a 12 tooth sprocket in front and a 150 sprocket in rear. Using these configurations with a rear axle mounted clutch results in a calculated torque demand of 21.875 Nm.

When investigating solutions available on the market, there is one particular type of clutch that has the possibility to meet these requirements. That is the electromagnetic clutch. This is due to the possibilities to control the clutch using a pwm signal (pulse wave modulated signal) during engagement and a steady signal while engaged.

3 Results

This section presents the results of the conducted methods. Results of various tests and manufacturing methods are presented for each part of the engine, structured according to area of interest.

3.1 *Oil consumption, piston rings*

Stated in the previous section, one of the main sources for the oil consumption is the lack of scraper ring and oil control ring. The development of the piston rings is provided in this section.

Due to difficulties in finding a suitable company to produce the piston rings, for a limited cost, the decision is made that it should be produced in-house. This results in several difficulties. The main issue is to manufacture the 3rd ring, oil control ring, due to the high complexity and tolerances. This ring can be simplified by creating a relatively thick ring with rectangular cross section and provide open slots around the ring. However, due to the limited amount of time this solution is neglected.

The main focus is creating the 2nd ring, i.e. the oil scraper ring. In order to minimize the production time of this part it is decided that the existing compression rings should be modified to have the same specifications as the oil scraper ring. The main difference between the compression ring and the oil scraper ring is the chamfered outer surface of the latter. The chamfered outside surface can be achieved by mounting the compression ring in a fixture and lathe it using angled cutting steel. However, due to poor installation tolerances and measuring procedures of the fixture, the outcome is not satisfying. The resulting chamfer is not evenly distributed around the outer surface, hence resulting in an elliptical piston ring. The scraper ring is never evaluated because of the poor quality.

3.2 *Oil consumption, crankhouse*

One of the contributing factors to the oil consumption is the insufficient crankhouse ventilation. In order to achieve satisfying crankhouse ventilation, an increased diameter of the ventilation hole is needed. The crankhouse is therefore re-designed with a crankhouse ventilation diameter of 20mm in order to provide sufficient ventilation and minimize the risk of air flow restriction. The result of increasing the diameter has no significant effect on the oil consumption. However, this implementation is never evaluated in combination with the oil scraper ring, which might have increased its performance. Figure 2 displays the assembled crankhouse.

The crankhouse is also redesigned because of the change in bearing dimensions, as a result of increased crankshaft support shoulder for the flywheel.

3.3 *Flywheel oscillations*

In order to reduce the flywheel oscillations the crankshaft support shoulder is increased from 1.4 to 3.75mm. The oscillations of the flywheel are tested with an assembled crankhouse and crankshaft. The flywheel is then pressed against the support shoulder. Measurements are conducted on the outer axial surface of the flywheel using a precision dial indicator perpendicular to the surface. The increased shoulder resulted in a great reduction in the oscillations due to the increased contact surface. Figure 3 displays two crankshaft halves.

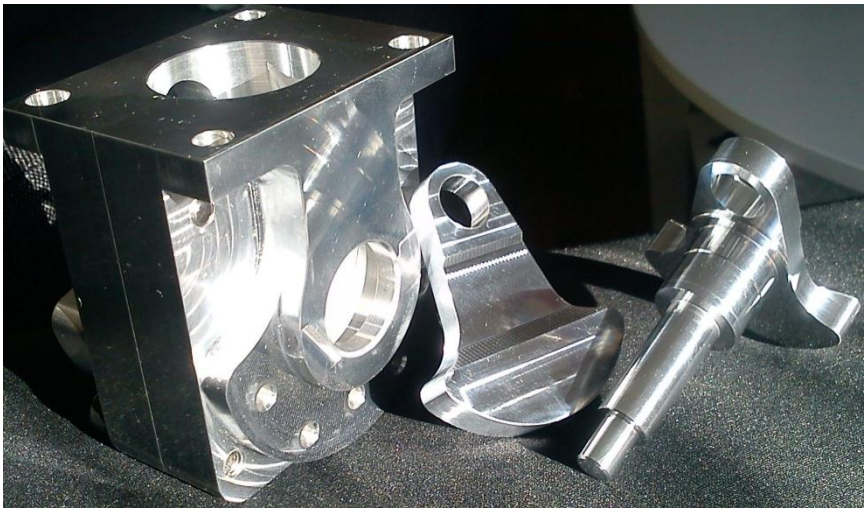


Figure 3, Displaying the assembled crankhouse and the two crankshaft halves.

3.4 *Cylinder Lining*

The inside of the new cylinder lining is measured using a three-point-micrometer. The micrometer has a precision of 0.01 mm and is calibrated using a fixed measurement piece. Its diameter is constant from top to bottom. The height of the lining is also measured and the height reached 0.05 mm above the cylinder. This yields a tight fit between the lining and the cylinder head which means close-to-zero compression leakage. The surface finish is visually inspected and the result is accepted.

3.5 *Signal disturbances*

The suppression of the crank pulse signal disturbances is evaluated using several different methods explained in the method section. The implementation of an ignition cable with inductive resistor is tested while running the engine. This clearly reduces the induced current and thereby the disturbances in the crankshaft trigger signal. The uneven and high amount of fuel pulses, due to the disturbances, is removed as a result. This leads to an increase in controllability due to the increased precision.

3.6 *Interference fit between crankshaft and wrist pin*

By using a relatively tight interference fit between the crankshaft and wrist pin, the engine is able to run for a relatively long period of time under low load. However, during competition and increased engine load the crankshaft halves rotate in relation to each other.

3.7 *Cooling and Heating System*

The base shape and mounting holes of the cylinder and side plates are manufactured in-house in the milling machine. The cooling pipes are created by implementing a CAD model in a CNC machine. The holes used to connect the three sides are drilled in the milling machine. In order to prevent any water from leaking between the cylinder and side plates, an adhesive sealing paste, in combination with M6 screws, is used. Pneumatic couplings and tubes are used to transport the water. The following pictures (figure 4-6) display the final cooling/heating system.

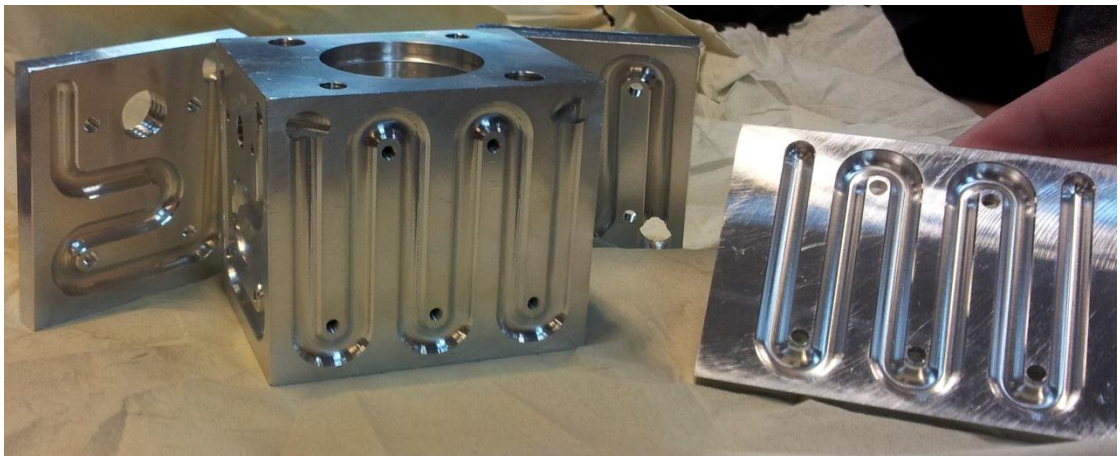


Figure 4, Displaying the manufactured cylinder with cooling pipes.

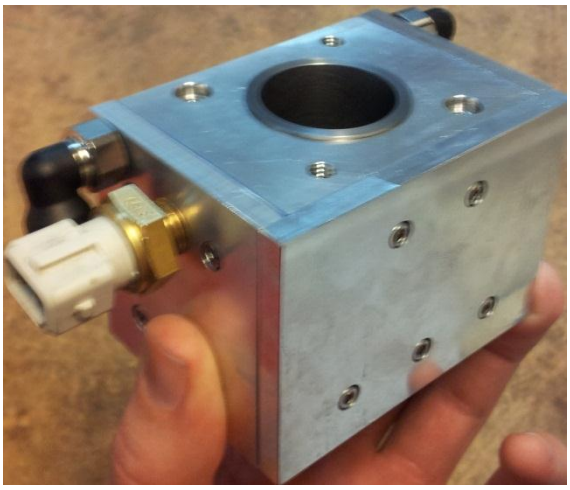


Figure 5, Displaying the assembled cylinder.

The cooling system showed great performance. The temperature, measured from the temperature sensor mounted in the cylinder side, showed a value of approximately 75 degrees C. When the water pump is engaged, the temperature decreases to 35 degrees C. The heat function is never evaluated or used due to the limited time available before race.

3.8 *Electric System Interface*

The extensive time spent in the workshop has led to a lack of time devoted to configuring the electric system. The system is therefore left as it is.

3.9 *Compression Ratio*

Using the Matlab script and assuming no extension plates, the resulting default compression ratio is 14. Figure 7 depicts the compression ratio as a function of extension plate height.

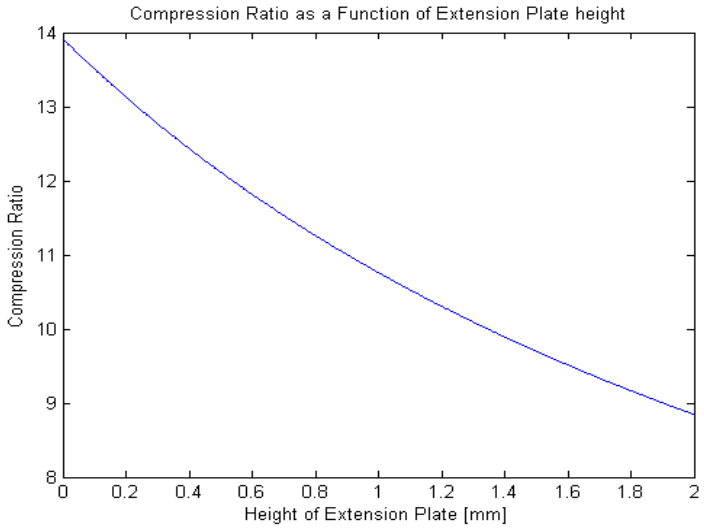


Figure 6, Displaying the Compression ratio as a function of extension plate height.

Looking into figure XX it shows that by adding a 2 mm extension plate, i.e. “elongating the cylinder” by 2 mm, the compression ratio drops from approximately 14 to 9. However, due to the lack of a camshaft belt tensioner, no suitable cam belt can be found to match any other extension than 1 mm. The different compression ratios have therefore not been compared or evaluated.

3.10 *Clutch and drivetrain*

Research about available electromagnetic clutches showed one promising solution. The clutch that meets the required torque is the INTORQ 14.105.10.1.1. It can transfer 30 Nm compared to the engines torque, with gearing ratio, which produces 21.875 Nm. However, if an electromagnetic clutch is to be installed, more components would be needed. A control module needs to be implemented to control the actuation of the clutch. Furthermore, two more bearings would be needed to provide stability for the clutch and the rear wheel. The last nail in the coffin is the weight of the clutch. It weighs 1.84 kg which corresponds to 5% of Vera’s 34 kg (14). The clutch specifications are presented under appendix figure 7, and size 10.
The lack of time and increase complexity of the drivetrain, in combination with increase weight due to bearings, control module and the clutch itself, yields the result that no modification of the drivetrain will be implemented during this project.

4 Conclusion

The problem regarding the oil consumption is never fully solved. However many suggestions is described but never implemented due to reasons described. The oil consumption is one of the main problems in the current situation and the implementations of piston rings should be further investigated. It is recommended that this development is outsourced because of its complexity.

By implementing the ignition cable with inductive resistance, the disturbance is greatly reduced to a level which did not disturb the engine control unit. This resulted in highly increased possibility to control and tune the engine more accurately.

In order to minimize the risk of the crankshaft halves rotating in relation to each other an increased interference fit is used. Even with increased fit the two halves rotated under high engine load. For further improvements other metals can be used to provide higher contact pressure between the wrist pin and the crankshaft. Other improvements can be to utilize splines on the wrist pin to increase resistance against rotation.

Due to time constraints the electromagnetic clutch is not fully investigated. Further development of this engine is to investigate in the possibility to utilize this kind of clutch.

The cooling system has proven to efficiently cool the engine during bench testing. However, the heating of the engine should be tested and evaluated with regard to heating time. Also a thermostat can be of interest in order to keep a steady temperature during bench testing.

Regarding the ability to change, and thereby test the effect of changing compression ratio, a cambelt tensioner must be implemented. The current problem is the course variations in length when changing cambelt. A tensioner would provide the ability to use the same belt for a wide range of cylinder extension plates. Thereby granting the ability to investigate the impact compression ratio has on performance.

5 Bibliography

1. Salter's Chemistry. *4college.* [Online] [Cited: 12 01, 2012.] http://www.4college.co.uk/as/df/octane.php.

2. Heywood, John B. *Internal Conbustion Engine Fundamentals.* New York : McGraw-Hill, 1988.

3. Coxon, John. Piston-rings. *ret-monitor.* [Online] 10 12, 2009. [Cited: 12 01, 2012.] http://www.ret-monitor.com/articles/630/the-oil-control-ring/.

4. Ring types and functions. *Samsun Segman Sanayi.* [Online] [Cited: 12 01, 2012.] http://www.samsunsegman.com/products.asp?id=53&cat=2.

5. Cylinder Liners. *TPR.* [Online] [Cited: 12 01, 2012.] http://www.tpr.co.jp/tp_e/products/cylinderliners/about_cylinderliners.htm.

6. P. Andersson, J. Tamminen, C.E. Sandström. *Piston ring tribology, A literature survey.* Esbo : VTT, 2002.

7. Ignition cables - principles. *NGK.* [Online] [Cited: 12 01, 2012.] http://www.ngk.de/en/technology-in-detail/ignition-cables/principles/.

8. Ignition cables with inductive resistor. *NGK.* [Online] [Cited: 12 01, 2012.] http://www.ngk.de/en/products-technologies/ignition-cables/ignition-cable-technologies/ignition-cables-with-inductive-resistor/.

9. Alumec 89. *Uddeholm.* [Online] 01 2011. [Cited: 12 01, 2012.] http://www.uddeholm.se/swedish/files/Alumec_89-swedish_p_R110126_e4.pdf.

10. Class V (Shaft-Hole) Tolerance Chart. *Engineers Edge.* [Online] [Cited: 12 01, 2012.] http://www.engineersedge.com/class_v.htm.

11. Aluminium. *Wikipedia.* [Online] 12 01, 2012. [Cited: 12 01, 2012.] http://en.wikipedia.org/wiki/Aluminium.

12. THE BEST HEAT TRANSFER FLUIDS FOR LIQUID COOLING. *LYTRON.* [Online] [Cited: 12 01, 2012.] http://www.lytron.com/Tools-and-Technical-Reference/Application-Notes/The-Best-Heat-Transfer-Fluids-for-Liquid-Cooling.

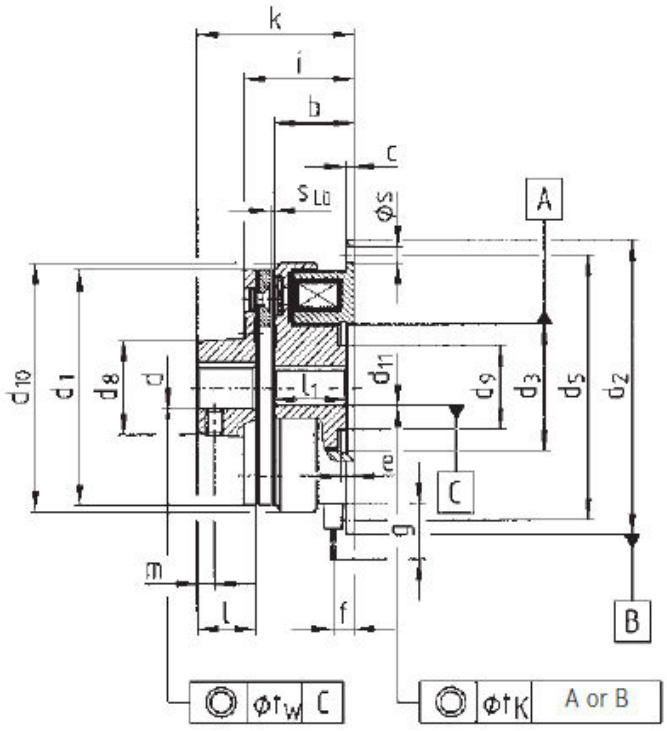
13. Harting Kontaktdon HAN A 10-32-poliga. *OEM Automatic.* [Online] [Cited: 12 01, 2012.] http://www.oemautomatic.se/Produkter/El-maskin/Kontaktdon_Harting/Insatser/Han_A_10-32_Pol_16A/Harting_Kontaktdon_HAN_A_10-32-poliga/562607-334320.html.

14. *Electromagnetic clutches and brakes.* INTORQ. 2009.

6 Appendix

Technical data

Flange-mounted clutches
INTORQ 14.105.□□.1.1



Size	M [Nm]	b	c	d H7 min.	d H7	max.	d_1 h8	d_2 h9	d_3 H8	d_5	d_8	d_9	d_{10}	d_{11} H7 min.	d_{11} H7	max.
06	7.5	24	2	10	10;12;14;15;17	17	63	80	35	72	27	23	68	10	10;12;14;15;17	17
08	15	26.5	2.5	10	12;14;15;17;19;20	20	80	100	42	90	32	28.5	85.5	12	12;14;15;17;19;20;25	25
10	30	30	3	14	15;19;20;24;25;28;30	30	100	125	52	112	42	40	107	15	15;19;20;24;25;28;30	30
12	60	33.5	3.5	14	20;24;25;28;30;35	35	125	150	62	137	49	45	134.3	20	20;24;25;28;30;35	40
16	120	37.5	4	20	25;28;30;35;38;40;45	45	160	190	80	175	65	62	170	25	25;28;30;35;38;40;45	50
20	240	44	5	25	35;38;40;42;45;50;55;60	60	200	230	100	215	83	77	214.3	25	35;38;40;42;45;50;55;60	65
25	480	51	6	25	40;45;50;55;60;65;70	80	250	290	125	270	105	100	266.5	30	40;45;50;55;60;65;70	80

Size	e	f	g	i	k	l	l1	m	s	$s_{Lü}$	t_k	t_w	m [kg]
06	3.5	5.5	400	31.5	43	15	22	5	4 x 4.5	0.2	0.2	0.1	0.53
08	4.3	6.5	400	35	51	20	24	6	4 x 5.5	0.2	0.3	0.1	0.96
10	5	6.5	400	40.9	60.9	25	27	6	4 x 6.6	0.2	0.3	0.1	1.84
12	5.5	7.1	400	46.5	70.5	30	30	10	4 x 6.6	0.3	0.3	0.1	3.24
16	6	8.6	400	53.5	84.5	38	34	10	4 x 9	0.3	0.4	0.2	5.79
20	7	12.4	400	64.4	103.4	48	40	15	4 x 9	0.5	0.4	0.2	11.4
25	8	14.9	400	74.9	118.9	55	47	20	4 x 11	0.5	0.5	0.2	20.4

- Dimensions in mm
- Keyway to DIN 6885/1-P9
- Recommended ISO tolerances for shafts: Up to Ø 50 mm: k_6
  Above Ø 50 mm: m_6

Figure 7, Displaying the data sheet of an electromagnetic clutch (14).