IIT MMAE Dept. Research Project
The Homogeneous Charge Thermal Ignition (HCTI) engine

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1 Introduction:

Nowadays the main kinds of engines that are used in ground transportation are, gasoline Spark Ignition engines and diesel Compression Ignition engines. As every day more fuel is being used by a growing number of vehicles, fuel dependency growth and a growing concern for our environment health, it is a crucial point to gain in fuel efficiency for ground transportation engines.

Many approaches are being investigated, but we will focus in one kind that we call the HCTI, homogeneous charge thermal ignition engine. The goal with this new kind of ignition is to increase engine efficiency both by igniting the mixture using a much bigger surface that a spark plug kernel would be, thus lowering pumping losses, and also by burning leaner mixtures that are thermodynamically more efficient with a reticulated ceramic matrix, something that couldn’t be possible with SI because such lean mixtures wouldn’t burn with a spark plug.

In this paper an SI engine will be monitored and also the same version of the HCTI to try to determine if there are gains in efficiency according to the theory.

2 Experimental process:

2.1 Experimental plan:

The experimental plan was to first monitor torque, power, fuel consumption and fuel efficiency, emission gases, cylinder pressure and the thermodynamic cycle of the engine in the Spark Ignition mode, and then modify it to a HCTI engine and do the same monitoring (when possible) to be able to compare between the two kinds and then see if the HCTI improved or not with respect the SI engine and in which aspects there were gains or losses.

Each measurement process, the systems used for each measurement and also the results will be discussed further in this paper.
2.2 System description:
The experimental system used for this project is shown in the diagram below. Dashed lines indicate data wires. Blue lines are water lines. Black line is the throttle cable. Brown line is the gas analyzer prove tube. All the elements are listed below the diagram with their description.

1. PC laptop
2. QSB4 USB digital interface from US Digital, receives both pressure data signal after being amplified (dashed line 4) and shaft position data signal (dashed line 5). It buffers data at a high rate and send it via USB to the laptop.
3. PLC signal amplifier, used to convert the charge (in picocoulombs) from the piezoelectric pressure transducer to volts and then sent to the QSB4 interface.
4. Analog data shielded wire, transmits the charge generated by the pressure sensor.
5. Digital data shielded wire, transmits the position of the shaft captured by the digital encoder.
6. Prove tube, this prove captures gases from the exhaust muffler and circulates them into the gas analyzer.
7. Vetronix PXA-1100 gas analyzer, used to analyze the exhaust gases and calculate the AFR in different regimes.
8. Land and Sea DYNOMAX console, it can be used both as a direct data storage or a real time interface to the computer. In our case it was used a as a real time interface, it can measure
dozens of parameters of an engine, but in our case we only took RPM, torque and horsepower.

9. Mikuni 30mm carburetor, it was used as a replacement to the stock carburetor as it is very easy to tune at the desired AFR at almost all the regimes of the engine with this kind of carburetor.

10. Briggs & Stratton 1450 series engine, 305cc engine with OHV head design and 14.5 maximum raw torque (more information about the engine found in the appendix).

11. Muffler, the outlet port has been specially modified to allow EGR so that the combustion is more complete and the missions are lowered, more on that later.

12. Land and Sea water brake, with inbuilt RPM sensor and torque sensor, directly coupled to the 1” engine output shaft with keyway.

13. US Digital H6 digital encoder with 2 channels and index channel, resolution of 2500 cycles per revolution.


15. Water brake pressure regulator, used to increase or decrease the load on the engine trough the water brake.

16. Fresh water inlet, the circuit bifurcates to a second smaller line to cool the pressure transducer, the main stream is used to induce a load in the water brake.

17. Used water outlet to drain.

18. Kistler 6014A pressure transducer, piezoelectric type, induces a charge when a deformation in the measuring membrane occurs, translating it to pressure.

NOTE: other systems like exhaust gas ventilation system, water piping system details and electrical power outlets are not shown for diagram simplicity.

This system was entirely build from zero at the beginning of this project, many custom made parts had to be designed and built by us or machined by third parties. More pictures of the entire system can be seen in the appendix attached at the end of this paper, however it is not the aim of this paper to report the construction of the system rather than discussing the experimental data achieved with it.
3 Spark Ignition data gathering:

3.1 SI Torque and Power 3D mapping:
First the torque and power monitoring were measured in the following form; with the throttle open at different positions, from 10% opening to 100% opening in 10% intervals, the different power bands were obtained using the dynomax water brake and thanks to the dynomax console and dynomax software the curves were obtained. The way to do that was to induce a load in a given throttle position to keep the engine running as a low speed as possible, then the load was slowly decreased to allow the engine to gain speed to the rpm limit, all this while recording the points in real time, thus recording the torque through all the rpm range. Gathering all this curves a 3D mapping can be obtained. NOTE: this power curves were all obtained with the closest stoichiometric AFR of 14.7 as possible to keep emissions as low as possible, more on that later.

The results of the torque readings are as follows:

Isometric view:

![Torque vs RPM vs throttle opening position](image)
First of all we should note that each curve starts in a different regime, this is due to the fact that the engine can't work out of the given range, so if no torque is measured in a given rpm is because the engine could not stand that load at that regime, on the right of the graphics we can see that the end of the recording has been stopped at different regimes (randomly due to the recording method), however all the data was kept entirely for the sake of more information available. NOTE that this engine is not intended to work above 4200 rpm, as it could easily break, so not much of importance are the reading past the 4200 rpm line.

As we can see, the torque in each throttle position has a peak at the beginning of each curve, growing at the beginning and rapidly getting to a maximum torque and then tend to decrease, this is a specific behavior of this engine, as load has to be decreased in order to gain regime there is no point in which the engine increases in torque. Anyhow the engine has a pretty flat torque delivery at throttle openings above 50%, this kind of torque delivery is desired for easy to drive engines as high torque is always available, making its behavior smooth and not picky. We must also note that the engine used in this project is intended for stationary use, so it's not designed for high rpm levels or a wide range of rpm.

The maximum torque recorded was 14.4 at 100% throttle and 3300 rpm, just around the rpm in which the engine was designed to deliver pick torque by the manufacturer, this is an expected result.
We must also note that when the engine was run in a rich mixture the behavior was smoother and a bit more powerful, but the emissions were much worse and the fuel efficiency was also lower, we don’t want to study the engine with rich mixture as we are trying to compare a clean efficient SI engine to a clean efficient HCTI engine, more on that later.

Let’s take a look to the horsepower mapping:

Isometric view:

Horsepower vs. RPM vs. throttle opening
As we can observe in this horsepower graphs, at the beginning from 10% to 50% throttle the power bands have a maximum later on the beginning on the curve, the torque loss in the middle ranges is boosted in power by the increasing rpm. From 50% to 100% throttle, because the torque delivery is so flat, the horsepower delivery boosts the horsepower in a increasing slope fashion, the more open the throttle is, the less volumetric loses are in the engine (less manifold restriction) and the higher the power curve is in respect to the one before. The maximum horsepower obtained is 11.41 hp at 4700 rpm, but however the engine cannot work in a such high speed, so we could conclude that the maximum useful horsepower is 11.01 hp at 4200 rpm.
3.2 Emissions, fuel consumption and efficiency:

First of all it must be exposed that some trial emission data was first gathered to get a general idea of the engine emission rates. It was found that because of the valve overlap, too much oxygen was always present in the exhaust pipe, so the gas analyzer machine calculations were giving readings of AFR higher than 25, which is extremely lean and was obviously not a real reading.

So in order to ensure complete Oxygen reaction, a restrictor was coupled to the muffler outlet pipe, creating backwards pressure into the exhaust port and then generating exhaust gas recirculation and dramatically lowering the O2 that was coming out of the exhaust pipe. An average of 0 to 0.5 [lb/ft] of torque loss was registered though out the torque bands, but it was found convenient for the sake of a good emission data capturing and also to get good non toxic emission products.

The process to gather all this data was as follows:

While maintaining the engine in a given rpm range and a given throttle position, it was given some time to exhaust gases to circulate through the probe tube and into the machine sensors and waited till the readings were stabilized, then the numbers were recorded.

To measure fuel consumption, a pipette was attached in the fuel line, that pipette was always full while the fuel was running from the main fuel tank, when the fuel from the tank was shut off, the level in the pipette started to drop as the engine was consuming the fuel trapped in it. By measuring how much took to the engine to consume a given amount of fuel volume, the fuel consumption in liters/hour could be calculated.

Also by knowing how much work does a liter of gasoline produce, it was very simple to compute the energy released by the engine output shaft by converting horsepower to watts, multiplying them by time to compute the total energy released during the fuel consumption time and finally dividing that energy by the total energy that the consumed volume of fuel was supposed to release with a 100% efficiency.

\[ \text{Efficiency \ [%]} = \left( \frac{\text{Torque} \times \text{rpm}}{5250} \right) \times \left( \frac{135962 \times \text{Fuel consumption} \times 9.68}{1.35962 \times \text{Fuel consumption} \times 9.68} \right) \times 100 \]
3.2.1 Emissions Data:

AFR chart:

In order to study the engine's performance in terms of emissions and fuel efficiency, it was considered that it would be reasonable to study such parameters within the most usable range of rpm, in other words when the engine had the best response and smoothness, without lugging or misfire, which is from around 3300 rpm to 4200 rpm and also from 20% to 100% throttle as the 10% throttle power band proved to be very inefficient and couldn't even reach high rpm.

Let's first talk about the AFR that we were able to achieve through tuning the carburetor. This process took some time and trial and error to tune the carburetor parameters to get as close to the stoichiometric point as possible in all different situations. Note that our aim was to have an engine with low toxic emission generation such as non-burned HC and CO and NOx.

NOTE: NOx emissions couldn't be obtained due to the failure of the gas analyzer NOx sensor.
Oxygen chart:

In the diagram below we can take a look at the evolution of the Oxygen that came out of the exhaust pipe. We will first review oxygen as its evolution will help us understand the evolution of the rest of products. Note how first the uncombined oxygen is very high at low throttle openings, this is due quite surely to poor turbulence and low reaction speed in the combustion chamber, as we will see HC and CO are also high at low throttle openings, demonstrating then that the low turbulence causes slow reaction speed and then causing products resulting from incomplete combustion to get out the exhaust pipe. Also note that in comparison, the slower the engine turns, the more uncombined oxygen there is, and the more un finished reaction products there are, demonstrating again that slow engine speed causes less turbulence in the intake port and the combustion chamber.

At 40% throttle the mixture becomes most rich overall, so little oxygen is left to react, this doesn’t mean that the combustion is complete, this only indicates that oxygen is scarce in this regime as the rest of products indicate.

As throttle keeps opening from 60% to 100% the levels of oxygen rise a little bit, but as we will further see, the rest of products produced by incomplete combustion decrease, indicating that combustion is more complete at higher throttle opening for two main reasons, the AFR is closer to stoichiometric and also the turbulence in the combustion chamber is higher providing better air fuel kinetics and favoring the combustion reaction.
Unburned HC chart:

Very related to the oxygen evolution the unburned HC chart reflects how when incomplete reaction is happening, oxygen levels are high, HC levels are also high and CO levels are also high. Again due to a rich mixture and poor combustion reaction kinetics affected by low turbulence. As throttle is opened, turbulence grows and AFR is closest to stoichiometric the HC combine more effectively with oxygen lowering HC and CO levels and increasing CO2 levels (CO2 is a product of a complete combustion reaction, so a good indicator of good combustion process).
**CO chart:**

Exactly the same as unburned HC, CO indicates a low reaction efficiency at low throttle opening, that combined with the mixture being rich, it creates both a high level of oxygen and a high level of HC and CO. As throttle is opened and mixture becomes closer to stoichiometric and turbulence grows, the products related to unfinished combustion reaction drop dramatically.

Note again that how engine speed alone, regardless of throttle position overall in all products, helps turbulence, and then, overall (looking to all charts), obtaining cleaner emissions as combustion reaction is improved by the air fuel mixture kinetics.
CO2 chart:

Finally taking a look to the CO2 chart, we can see how richer mixtures at low throttle openings produce less CO2 due to the lower reaction efficiency. As throttle was opened and AFR was closest to stoichiometric, CO2 emissions increase as HC and CO decrease.

There is the a clear inverse evolution between CO2 vs. CO and HC.

**NOTE** that there are some irregularities in the 3300 rpm range, may be due to instability in the gas analyzer machine which was not noticed at the time of gathering the data.
3.2.2 Fuel consumption and efficiency

Fuel consumption chart:

Taking a look at this first chart a clear correlation between throttle opening and fuel consumption can be observed. The more open the throttle is, the more air/fuel mixture is let into the cylinder, so more fuel is consumed. Also a clear correlation between engine speed and consumption can be observed, this is due to the fact that the higher the engine speed is, the higher the air speed is in the carburetor's venturi, so more vacuum effect is produced by the venturi effect and more fuel is sucked from the carburetor's main jet. The maximum consumption is 3.1 liters/hour at 100% throttle opening and 4200 rpm.
This torque chart shows how torque is maximum at 3300 rpm as we already knew, and decreases when rpm increases as we also already knew. However, this more paused measurement of toque, has shown that at 100% throttle opening torque decreases. This might be due to the fact that in the previous torque measurements, the torque curves were measured at the same time as the load in the water brake was released and the rpm increased. This method may have caused inertia effects in the engine to produce torque picks that weren’t actually produced by the combustion process.

Anyhow, this measurement was done much more accurately and paused, so this data is considered to be more accurate than the 3D torque mapping.

It can be clearly observed that the maximum torque is achieved at 80% throttle opening, that means that the engine has maximum efficiency at this point, and volumetric efficiency at 80% is maximum probably because of the intake port length, intake port width and resonance effect in the intake and exhaust ports is in tune.
Efficiency chart:

Totally correlated to the torque chart, a clear relation can be observed between torque and efficiency overall, this correlation is a known fact and is no surprise. We can see how maximum efficiency is reached at 80% throttle opening but at 4200rpm. Due to the fact that fuel consumption at this point drops slightly, the lower torque produces anyway higher efficiency.

In any case, experimental data may have some variation, but, we can say for sure that in this engine in particular maximum efficiency of around 26% is reached at 80% throttle opening, regardless of the rpm range. Also a 26% efficiency might seem a bit high for a simple construction carburetor engine, this might be due to combined deviation of all the data gathering.
3.3 SI cylinder pressure graphs

To obtain the pressure values inside the cylinder, the cylinder head was modified in order to screw the pressure transducer in a conduct that connected to the combustion chamber, being then able to measure the pressure inside the cylinder as the sensor's membrane was exposed towards the cylinder through the conduct. The modification was as follows: one of the sides of the head was flat milled and a hole was drilled trough one side then threaded and the pressure transducer was screwed in. More detail pictures of this modification can be found in the appendix.

The data acquisition was controlled by a National Instruments 60009 series interface and programmed with LabView software. The pressure data was recorded at 3600, 3900 and 4200 rpm and 80% throttle opening as the efficiency was found to be maximum at those regimes.

NOTE that volts to bar conversion could NOT be performed accurately due to deviation once the system started measuring high pressures. The Kistler 6041 pressure transducer is intended for engine monitoring, it cannot measure actual pressure values as too much deviation occurs! This is a fact stated by the manufacturer.

However, the purpose of this measures is to compare the engine pressure between SI and HCTI, so as long as the same reference is taken, volts are still a good unit to compare pressures between the different kinds of engine due to the fact that both engines are measured with the same system.

Thanks to numeric simulation performed with software Engine analyzer pro v3.3, we could obtain many maximum pressures, and minimum pressures in the cylinder varying from 3600 to 4200 rpm, being respectively from 37 to 45 bars maximum, and minimum pressures due to the intake of -0.55 to -0.35 bars. This numbers give us an idea of the magnitudes of pressure we are dealing with in this project.

In the charts depicted in the next page one can first observe that pressure picks are not always de same height, thus not always the same peak pressure. This is due to the fact that mixture and engine variables change between cycles. note also that higher rpm does not mean necessarily higher pressures. High pressures mean higher torque, however this is difficult to compare between such random measurements. Higher rpm is equal to more cycles per time unit, thus more energy delivered per time unit, thus more horsepower, something that can be observed easily in the charts.

NOTE this are not the only measurement taken, hundreds of thousands of points have been recorded in each regime, having all a great similitude between all of them within the same regimes.
The following pressure data was gathered:

**Pressure (3600 rpm)**

![Graph of pressure data at 3600 rpm]

**Pressure (3900 rpm)**

![Graph of pressure data at 3900 rpm]

**Pressure (4200 rpm)**

![Graph of pressure data at 4200 rpm]
Detailed pressure peaks:

In the charts below detailed pressure peaks can be observed, differences in pressure between them are barely perceptible since torque values are almost the same, only cycle time is a major difference.

However it’s interesting to observe how pressure resonance occurs at some points during valve actuation and gases moving through the ports, note also how the higher regimes also increase turbulence, this perturbation has been identified (thanks to the encoder) as the moment when piston speed is maximum (between 80 and 100 degrees), causing then great turbulence in the intake gases because then gas aspiration is maximum and it’s velocity too, the pressure transducer records this turbulence as it is very close to the intake valve:
Pressure (3900 rpm)

- Volts
- Time [s]

Pressure (4200 rpm)

- Volts
- Time [s]
In the next graph below, as we are able to measure shaft position with the encoder, the valve lifts and spark timing are represented in order to help understand the evolution of pressure and the pressure behavior related to the engine mechanical cycle.

If we take a look to the spark advance, which is 22 degrees btdc, and the evolution of the pressure, we can conclude that combustion in this particular engine is somehow slow. First of because the same advance gives us information about how early the spark must jump to achieve complete combustion and maximum pressure, ideally at 14 degrees atdc. If faster combustion was to happen it would be observed a steeper and more sudden pressure increase after the spark, something that isn't much observed in this particular engine. In this particular engine the ignitions is relatively smooth, and the pressure curve has no perturbations.

NOTE! at 360 degrees there is a discontinuity due to the fact that the curve was composed from 2 readings and the cycles varied a little bit in pressure!

It’s interesting to see how cylinder pressure drops faster just after the exhaust valve opens (EVO) and the same happens when the intake valve opens (IVO), this might indicate that there is still positive pressure inside the cylinder when the intake valve opens, something that is not desirable and that lowers engine performance. Then as piston travels downwards negative pressure is built in the cylinder.
3.4 SI thermodynamic cycle

The thermodynamic cycles were obtained using the QSB4 digital interface, by programming a data acquisition program in LabView, data was buffered and then sent to the laptop computer. Note once again that the pressure transducer has deviation therefore it was not possible to scale pressure accordingly to reality, this 6041 transducer is only intended for monitoring use! It cannot measure actual pressure. However the data is sufficient to compare between the two different kinds of engine. It should be acknowledged that variability between cycles may occur, as pressure peaks are very random, as it is not possible to compare the hundreds of thousands of points gathered at first look the same variability will happen in the thermodynamic cycles.

Let's take a first look at the thermodynamic cycles individually. In this first 3600rpm cycle it can be perfectly observed how maximum pressure is quickly built into the cylinder and peak pressure is achieved shortly afterwards the tdc. This particular cycle is an image of a highly energetic peak of pressure, that was recorded randomly. Note also how the pumping effect of the piston creates a energy loss translating in the lower loop of the cycle which's area indicates negative work or lost energy.

![Thermodynamic cycle (3600 rpm)](image)
The next two graphics are the ones corresponding to 3900 and 4200 rpm, in this particular cases the pressure peaks measured were not particularly energetic, the peak of pressure is very round, indicating low combustion and low pressure achieved, as peak pressure is quite low as it can be observed. However it has to be acknowledged that torque delivery at 3600, 3900 and 4200 rpm was very similar, so thermodynamic cycles in average should be as similar between the tree regimes.
In this next chart the three thermodynamic cycles can be compared, the randomness of the pressure peaks is clearly observable in the different cycles. Note how much of a power increase can be between cycles. This could surely be avoided by running the engine excess rich, but however this is not the aim of the project as it is not desirable to produce any bad emissions and it is desired to run the engine stoichiometric. All in all, the most representative thermodynamic cycle of the engine is considered to be the 3600rpm 80% as it represents a good power delivery and a shape close to the theoretical thermodynamic cycle.
The step shape of the cycles is due to the digitalization of the signal performed by the interface. The bigger the array it was desired to capture the less resolution could be obtained due to the limited buffer size. In addition to the cycles presented above a higher resolution cycle was afterwards obtained at full throttle at 3900 rpm, and in which the pressure changes and perturbations can be observed easier.

**Full throttle 3900 rpm Thermodynamic cycle**
3.5 Conclusions

Thanks to the monitoring and measurements of the engine one can now have a picture of how the SI engine performs. There is also a solid base now on which any alteration on the engine can be compared in terms of power output, emissions, efficiency and so on.

The engine has proved to have a very flat torque delivery, running stoichiometric. Also thanks to the stoichiometric carburetion and the gas recirculation it has proved to run very clean on the SI version.

A lot of randomness in the pressure peaks has been observed, therefore as well as in the thermodynamic cycles, this randomness could be quite possibly eliminated if the engine was run excess rich, however it would have altered the emissions, fuel efficiency and torque delivery. Something that wasn't wanted as the aim of the project was to compare the cleanest more efficient SI engine to the cleanest most efficient HCTI engine it could have been built.

Some more investigation is desired to be done for example on the unexpected high efficiency numbers that were achieved. It's also a drawback the fact that no actual pressure readings are possible with the 6041 pressure transducer, however it is enough for engine monitoring and cycle comparison.
4 Thermal Ignition Engine

Once the SI engine characteristics were measured, it is time to modify the engine and transform it into a HCTI engine. Many prototypes were designed and tested.

It must be acknowledged that in the end, none of the HCTI designs could improve the SI engine version. It is therefore desired to keep investigating the causes of that performance loss and also new possible designs will be discussed further in this paper.

4.1 Thermal Ignition approaches and design:

4.1.1 Ceramics in chamber construction:
The first and most simple approach was to literally place a reticulated ceramic matrix (which was a foam silicon carbide filter used for filtering molten metal as it is poured in the casting molds) in the combustion chamber. A hole was milled in the cylinder head and the ceramic piece was placed inside it. Part of the ceramic piece was held in a gutter shaped hole and the other side was held by a metal laminar spring. Then the spring was secured by the cylinder head gasket. The ceramic piece was hand cut to the desired shape. The location of the ceramic piece was chosen near the exhaust valve because it is the hottest zone in the combustion chamber and it would be easier for the ceramic piece to heat up. It was also vital to avoid the fresh air fuel mixture from the intake to get in contact with the ceramic piece to avoid pre-ignition and a possible failure of the engine. Some pictures of the modification are shown below:

Thanks to the gutter shaped hole, a good breathing of the mixture could be achieved through the ceramic foam matrix, thanks to the piston action and therefore facilitating the heat transfer between ceramic and air fuel mixture.
4.1.2 Ceramics in chamber results:
The ceramics in chamber proved to be able to ignite the mixture, but as the ceramic piece was so exposed to the fresh mixture (something natural due to the design of the OHV cylinder head) pre-ignition was inevitable the most part of the time once the engine and the ceramic foam matrix gained temperature. The engine could barely keep turning and pre-ignition was threatening the engine’s integrity.

One main problem of this design was the lack of control of the ceramic foam temperature, being the ignition very much random. It was so random that not even pressure measurements could be performed due to the short time in which the engine could keep running.

Anyhow, the performance at first glance was so poor that it was not worth it to keep investigating in this direction.

4.1.3 Glow plug in combustion sub-chamber construction
Next prototype was designed to gain control over pre-ignition. This was planned to be achieved with a combustion sub-chamber design that would prevent fresh mixture to get in contact with the hot glow plug while the intake stroke and then the piston action would push the mixture into this second chamber in which, thanks to the increasing pressure and temperature of the mixture and the high temperature and the wide surface of the glow plug the mixture would spontaneously ignite. This second combustion chamber is inspired in the swirl chambers used in indirect injection diesel engines. Also the temperature of the glow plug could be controlled by the voltage applied to it. A couple of autotransformers would be used to regulate the glow plug temperature. One to set maximum voltage to the second one and this second one to regulate this higher voltage in a more fine way.

This sub-chamber would be screwed directly to the original spark plug hole. Then a smaller 10mm thread spark would be screwed on the engine head to allow the engine to start up as an SI engine and then turned off to switch to HCTI mode.

Because of the additional sub-chamber volume that would lower the original compression ratio of 9.5:1, the necessary calculations were performed and the cylinder head was milled down (and also the piston to avoid interference between both) and the cylinder head gasket was removed. The sealing was achieved with high temperature cylinder head silicone. After the modifications were done the original 9.5 compression ratio was restored.

Some pictures of the device can be seen in the next page:
4.1.4 Glow plug in combustion sub-chamber results
This second prototype turned out to be a success in terms of ignition control. The engine was able to keep running for a long period of time. However late ignition and stalling were sometimes present. There was also difficulty in keeping up with high regimes.

The main problem with this system however was the fact that a high voltage had to be applied to the glow plug to keep it hot. As the power source was alternate current and the electric power needed relatively high (from 100 to 200 watts depending on the engine speed and throttle) the current circulated through the engine block and then to the electronic device's grounds, creating dangerous short circuits that could result in device damage or a fire as one of the ground wires burnt up. For this reason no measurement could ever be performed in with this HCTI prototype. This could be avoided using a closed loop continuous current power source but none was available to us. So a different prototype was then developed.

4.1.5 Tungsten wire spire in combustion sub-chamber construction
In order to totally isolate the electric current from the block, a new prototype was built. This new prototype would consist on a tungsten wire spire totally isolated from the engine block thanks to a non conductive extruded alumina cylinder with two holes through which the tungsten wire could go through to the exterior of the combustion chamber. A small bit of clay was used to seal the gap between wire and the alumina cylinder. Then using autotransformers it could be possible to heat the spire to a high temperature using any kind of current. Then the mixture would be ignited by this incandescent spire.

The device is as follows:
The reason why the aluminum insert that holds the ceramic cylinder is so long is both to refrigerate as much as possible the hole device and also to make the ceramic cylinder easier to seal with clay as the longer the channel the harder is for the gases to come out the other side.

4.1.6 Tungsten wire spire in combustion sub-chamber results
This prototype was a success in terms of ignition control, both because of the retarding effect of the combustion sub-chamber and the voltage regulated temperature of the spire and also a success in achieving steady running of the engine.

However some pressure graphs were obtained and a very faulty combustion was diagnosed. Pre-ignition, detonation, late ignition and misfire were all observable randomly in only a 10 cycle period. Such faulty engine behavior resulted in a very poor engine performance and this kind of flaws in the combustion process were not admissible.

There was also an issue with tungsten oxidation. Although the voltage input was regulated not to achieve oxidation temperatures, it didn't matter when the engine ran for a while because the combustion itself ended up by heating the spire alone to limits way beyond the tungsten’s oxidation temperature, so the spire consumed after a short period of engine running. Some pictures can be seen below, note the yellow colored tungsten oxide deposits all over the sub-chamber:
4.1.7 Ceramics in combustion sub-chamber construction

After the tungsten was rejected, one last design was tested. It consisted in just filling up the combustion sub chamber all the way up with silicon carbide foam. The aim of this last design was to control ignition as it had been done so far with the combustion sub-chamber. While starting the engine conventionally with the secondary spark plug the ceramic would heat up and after the ignition would be turned off the ceramic would act as a regenerator, absorbing heat during combustion and returning it to ignite fresh mixture. The ceramics had to fit tight inside the chamber, because the blast of gases that were pushed inside the chamber when the engine ran at high speed proved to be enough fast and violent to reduce the ceramics to dust as they were hitting the chamber walls. A bolt was used to seal the other end of the chamber in this case. Some pictures on this designs are shown below:
4.1.8 Ceramics in combustion sub-chamber results

This final design proved to have the best performance compared to all the earlier ones.

Ignition timing was very reliable indeed, more even than in the tungsten version. Also the pressure graphs showed almost all the time correct ignition timing and complete combustion, no pre-ignition and no ignition delay or misfire were present at all, so this design is very robust in terms of ignition efficacy.

However, this design, although being the best and having a very good ignition performance it couldn't deliver as much power as the SI version (actually it could only deliver half of its SI equivalent), even running with the same carburetion. If the carburetion was done leaner the engine wouldn't run. Also at high throttle openings the engine wouldn't give any more power and it would very usually stall.

This behavior is thought to be produced either by volumetric losses due to the fact that now the engine would have to push a lot of air fuel mixture to a small hole into the combustion sub chamber, something that didn't have to do before, and also because of a possible pre-ignition at high regimes and high throttle openings that showered the engine down till it stalled.

More information on the data gathering section.
5 Thermal Ignition data gathering:

5.1 HCTI Torque and Power band (ceramic in sub-chamber only)

The main goal in this point was to obtain the same data with the HCTI engine as it had been obtained with the SI equivalent. However, the poor engine output and frequent stalling at throttle openings of 60% or more made only possible to obtain two good power curves at 20% and 40%.

Anyhow, the fact is that the HCTI engine proved to be much less powerful than its SI equivalent. So not much of an issue was not to be able to obtain a full 3D mapping.

In the next charts one can compare Torque and Horsepower of both engines at 20% and 40% throttle opening:
NOTE: it is very important to acknowledge that because of the testing period that was necessary to develop the HCTI engine prototypes, the engine itself, specially the cylinder skirt and the piston where damaged (worn) because of ceramic particles getting loose in the combustion chamber due to the natural degradation of the ceramic and also tungsten oxide deposits. Pre-ignition and detonation where also major contributors to engine damage due to the difficulty of getting a correct ignition timing.

That is why we should consider that some power might be lost, and also a decrease in cylinder pressure should be expected, but by any means the engine damage at this point is excuse for the poor performance of the HCTI engine as we will be able to demonstrate further in this paper.
5.2 HCTI cylinder pressure graphs

5.2.1 Tungsten wire spire pressure graphs

The graphs below show the pressure measurements of the HCTI tungsten wire version. As the pressure readings show, the combustion is pretty faulty if we carefully observe the pressure evolution. It can be seen how pre-ignition, late ignition, detonation and misfire occur randomly all at the same regimes and loads.

First of all, all of the different phenomena will be exposed in individual charts, to have a proper detail of how the pressure evolves in every case, finally some long arrays of data will be shown to give the reader an idea of how the engine actually works and to see the randomness of the HCTI tungsten wire combustion:

In the pre-ignition chart one can clearly observe how the combustion starts at a very early stage in which the piston hasn't been able to compress the mixture totally, note how the peaky shape of the curve and the serrated down slope that indicate very strong pressure pulses similar to the detonation ones that very commonly create damage to the engine due to the strong mechanical vibrations induced by this pressure peaks. Pre-ignition can also destroy the engine quickly at high throttle openings and high loads due to pressure over limits created inside the engine as the piston travels upwards which results in holes in the piston or broken connecting rods.
In the late ignition chart we can observe both late ignition and detonation phenomena. Late ignition is clear as the pressure of the cylinder drops as the piston travels downward and then a second peak of pressure develops as mixture is ignited. Because of the late ignition of the mixture when the piston is traveling downwards, the power output is very low, as the maximum pressure obtained in this case is very low compared to the SI engine pressures. Note also how a serrated form appears at the second peak of pressure. This serrated pressure evolution is a clear indicator of detonation.

Misfire can be easily detected because it’s the same height as the first peak of the late ignition chart but no other pressure increment is present. No pressure perturbations are present as no combustion happens.
In this chart above regular ignition is achieved, may be a little late, but it is something to be expected as ignition can't be timed accurately in this kind of engine. Note the smoothness of the curve indicating that no detonation is present.

In the chart below ignition timing is better as the pressure increases more continuously, but however detonation can be observed as the serrated form of the pressure peak.
Pressure plots:
The conclusion is clearly that this engine performance is not acceptable at all. The combustion is too faulty and the peak pressures indicate that the output is very poor because of the low pressures reached while late ignition is present. Note how late ignition is mostly the most present phenomenon, that combined with misfire rests the engine with a very poor output.

One might think that the tungsten wire spire temperature could be increased to aid ignition, however oxidation temperature was already reached at this point, and spire destruction was pretty fast.

There is also a very important factor in this design to take into account: spire length.

The longest the spire the most heat can be accumulated in it and the more heat can be transferred to the mixture, knowing that the longest spires possible where used in this particular case. The sub-chamber dimensions however limited the total size of the spire and although being the spire of a notorious size, this were the best results obtained.

The spire mass could have been also increased by using a wider wire. In our case 0.025 inches wire was used. However, thicker wires of tungsten proved to be much less bendable, as tungsten easily breaks and splinters when bended.

We could basically conclude that heat transfer between tungsten spire and mixture is still an obstacle that must be solved. The faster the engine runs, the more heat generation must be produced in the spire by the current as a higher heat transfer rate is demanded by a larger amount of mixture circulation through the engine. However it was seen that electric heat generation in the spire had its limits. It obviously took too long for the spire to build this heat up in many cases, resulting in misfires and late ignition. Also the fact that oxidizing temperatures were achieved it was nonsense to try and increase even more the spire's temperature as when oxidation occurs spire destruction is imminent.
5.2.2 Ceramics in combustion sub-chamber pressure graphs

In this design we could observe a completely different ignition behavior. First of all we can observe that late ignition or misfire are never (or barely ever) present, this is a first success in terms of ignition reliability and consistency. Secondly it can be observed a good consistency in pressure peaks, being there a good and constant output. Note there are some higher peaks but they are attributed to mixture inconsistencies the same as in the SI equivalent.

In the next chart a higher pressure peak is amplified to see if it is produced by pre-ignition or by the other hand it’s related to mixture inconsistencies, as the curve is so smooth it can be concluded that no pre-ignition has been produced so far in the ceramic HCTI.
However as the more regular pressure peaks were amplified and analyzed, it was found that the ceramic HCTI ignited almost all the time producing detonation of the mixture. It could be said that this design relies only on detonation ignition something not desired as it is a very harmful way to run for the engine. It is also suspected that this ignition is a bit too late, as pressure is not built into the combustion chamber at levels comparable to SI equivalent. This detonation ignition is shown in the next chart, in which the classical serrated detonation wave can be observed:

Note that after amplifying a whole lot of this pressure peaks, it was discovered that almost all of them presented the same behavior, so it can be concluded that (at least) this particular design only runs on detonation.
Pressure plots:
5.3 HCTI thermodynamic cycle:
To have a better understanding of HCTI working cycle (the ceramic matrix in sub-chamber design in particular) the thermodynamic cycle was of course obtained too. Due to the fact that a huge array was needed to be collected in order to obtain valid cycles, the resolution of the cycle obtained couldn’t be improved.

As we can see in this chart the thermodynamic cycle of the HCTI is very different in nature to the SI equivalent, especially during ignition and the power stroke. The ignition of the HCTI is a little late, as peak pressure is achieved in the compression stroke no much pressure increment is built when the piston travels down in the power stroke. This bad timing of the HCTI produces less power each cycle than the SI equivalent and resulting in this pressure drop just after the tdc is achieved. Also it can be deduced that HCTI combustion is slower than SI because as piston travels downwards the combustion can’t build as much pressure as the SI equivalent does. There is also clear evidence of the already expected detonation as serrated shape is present during all the power stroke.

An easier comparison can be performed in the graph shown in the next page.
This graph is of vital importance at the time of comparing the performance of the different engines. The SI cycle is the one corresponding to the 3600 rpm shown before on the paper as it has a very good representation of a high output cycle. The HCTI cycle is the one shown just in the page before and was obtained trying to capture the cycle while working the engine at its highest output at 40% throttle opening.

Note how the SI engine can build a lot of pressure after the ignition starts, and on the other hand the poor combustion of the HCTI that can't even raise the pressure in the cylinder as the piston travels downwards. As we know the area of the P-V graph represents work, so as the HCTI can barely fill half of the SI cycle area it's no surprise that the HCTI engine can't even deliver half the output of the SI equivalent. Detonation is also a non-efficient way of combustion, and it's known for lowering any kind of engine's performance.

If we take a look to the next chart it can be clearly seen how even a SI engine is giving a low output it can still manage to raise pressure in the cylinder thanks to a more effective and fast combustion.
Note the pressure increase always present in the SI engine due to a more effective and faster combustion compared to the poor slow HCTI combustion after TDC. It was thought also that there were pumping losses related to the fact that the mixture has to be pushed through the channel that connects the main chamber and the sub-chamber, however the thermodynamic cycle proves that poor combustion is a much critical factor at the time of explaining low power output rather than the pumping losses at the compression stroke.
5.4 Emissions and fuel efficiency

The HCTI engine proved to have a poor combustion, therefore a lower output than its SI equivalent, because the carburetion was the same in both engines, it can be concluded without any doubt that this HCTI engine in particular is not by any means as efficient as it's SI equivalent.

Another proof of its poor combustion is the emissions gathered at 40% throttle at 3300 and 3600 rpm. If we compare those to the SI equivalent, the following charts can be obtained and it can be clearly observed that HCTI has poor combustion as high O2, HCC and CO concentrations appear in the HCTI emissions at the same time that CO2 emissions are lower than its SI equivalent.
5.5 Conclusions and final thoughts

After all this data has been gathered and analyzed we can finally conclude that HCTI needs still a lot of improvement to become a successful machine. It has proven to be less powerful and less efficient then its SI equivalent.

The main problems that have been detected and need to be dealt with are that if ignition is not delayed to the an optimum point, pre-ignition won't let the engine run, but in the other hand if a sub-chamber is used to delay ignition, a very poor combustion is to happen. It is thought also that there are pumping losses related to the fact that the mixture has to be pushed through the channel that connects the main chamber and the sub-chamber. However the thermodynamic cycle has proven that poor combustion is a much critical factor at the time of explaining low power output rather than the pumping losses.

NOTE that it hasn’t been possible to measure combustion performance with the ceramics in cylinder head design due to the impossibility to run the engine properly.

Another problem is combustion smoothness, or in other words to avoid detonation. An engine cannot run all its life being ignited by detonation as early destruction will happen.

It would be desirable then to experiment with a hybrid design between placing the ceramic matrix straight into the cylinder head and using a sub-chamber, with an hybrid that exposed a little more the ceramic to the combustion chamber but at the same time managed to delay ignition at an optimal point would improve combustion at the same time that maintained a good ignition timing and also possible pumping loses would be reduced if that channel was widened. By widening the connecting channel between the main combustion chamber and the sub-combustion chamber would help in this endeavor, but the cylinder head would have to be further modified and a new sub-chamber should be built.

One other design which is however much more ambitious, is to create an hybrid between an HCCI and an HCTI by using a diesel engine with direct and indirect injection and a ceramic matrix in the cylinder head. It would consist on feeding the engine with a very much lean mixture during the intake stroke, and then at tdc, thanks to the hot ceramic and a jet of fuel provided by the direct injection the mixture would be ignited. The good thing about this design is that timing could be very much controlled by the fact that ignition would only occur when the fuel jet got into contact with the ceramic matrix and the flame front would then travel along the combustion chamber.
6  Appendix

6.1  Briggs & Stratton 1450 series technical data:

<table>
<thead>
<tr>
<th>Engine Specifications</th>
<th>Engine Technology</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
<td>OHV</td>
</tr>
<tr>
<td>Displacement</td>
<td>305</td>
</tr>
<tr>
<td>Bore</td>
<td>3.120 in (79.24 mm)</td>
</tr>
<tr>
<td>Stroke</td>
<td>2.438 in (61.93 mm)</td>
</tr>
<tr>
<td>Oil Capacity</td>
<td>25 – 28 oz (0.77 – 0.83 L)</td>
</tr>
<tr>
<td>Gear Reduction Oil</td>
<td>SAE 30</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Tune-up Specifications *</th>
<th>Model 200000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spark Plug Gap</td>
<td>0.030 in (0.76 mm)</td>
</tr>
<tr>
<td>Spark Plug Torque</td>
<td>180 lb-in (20 Nm)</td>
</tr>
<tr>
<td>Armature Air Gap</td>
<td>0.008 - 0.012 in (0.20 – 0.30 mm)</td>
</tr>
<tr>
<td>Intake Valve Clearance</td>
<td>0.004 - 0.006 in (0.10 – 0.15 mm)</td>
</tr>
<tr>
<td>Exhaust Valve Clearance</td>
<td>0.009 - 0.011 in (0.22 – 0.28 mm)</td>
</tr>
</tbody>
</table>
6.2  Detailed pictures of the system

6.2.1  System component's pictures

Main system overview all assembled on the Land and Sea bench

PLC amplifier
QSB4 US Digital interface, with analog input, encoder input, power inlet and usb data output.

National Instruments analog interface device
Vetronics PXA-1100 gas analyzer
Main fuel tank (black) and fuel consumption pipette meter
Land and Sea Dynomite console and interface

Bench control dash with accelerator, console/interface and load valve.
Close look to the engine and dynamometer assembly

Engine and dynamometer assembly
Mikuni 30mm carburetor
Close look to the dynamometer and encoder assembly

Dynamometer with encoder support and encoder
Vacuum ventilation tube and muffler outlet with restrictor
6.2.2 Muffler restrictor detail
6.2.3 Details of the pressure transducer mounting